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Developments in efficiency and stability of fluid film bearings using new designs and test techniques

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ABSTRACT

Recently developed techniques in the measurement of fluid film bearing dynamic behaviour, including stiffness and damping, have highlighted some interesting observations on the performance of a range of bearings. These are outlined together with suggested strands of further research aimed at improving the efficiency, reliability and cost of operating plant and equipment.

1. INTRODUCTION AND BACKGROUND

The Open University Lubrication Research Group was inaugurated in 1997 to continue work previously completed at both the Open and Cranfield Universities. This work consisted of a variety of projects in forms such as PhD programmes, Post doctoral research contracts, consultancy projects and MSc projects, supervised mainly by Dr. D.W. Parkins. In addition to Cranfield the Group has links with the National Research Council of Canada, with which a formal collaborative research agreement has been signed, and with Xi’an Jiaotong University of China. A senior Professor from the latter is currently on a secondment visit with the Group funded by the Royal Society.

Work recently reported, or in the process of being reported, includes new test and measurement techniques of fluid film bearing performance, the use of such techniques in the testing of tilting pad journal bearings at high speed, the design, development and testing of a patented novel form of adjustable hydrodynamic bearing, the testing of an adjustable rotor bearing, the theoretical simulation of the adjustable bearing concept and observations on the behaviour of hydrostatic bearings not predicted by conventional empirical design models. The group is developing plans for future work building on these and other projects, some of which are outlined below.

2. RECENTLY DEVELOPED TEST TECHNIQUES

2.1 Velocity coefficients

Parkins (1) has described a novel experimental procedure for measuring the four velocity or damping coefficients of an oil film journal bearing from imposed dynamic orbits. All four damping coefficients are derived from one imposed journal centre dynamic orbit and thus effectively obtained at the same time. The method requires the production of a “figure of eight” shaped orbit and uses its “cross-over” point. Rowan (2) has carried out
an extensive study of a tilting pad bearing using the technique for journal speeds of up to 15 000 rev. min.\textsuperscript{1} An example orbit is shown in Figure 1. The key feature of the method is the use of a bespoke axis system, set at the orbit cross-over point, and aligned with the directions of the orbit at that point. The axes are thus not required to be perpendicular. Coefficients obtained with this axis set are then transposed to a suitably aligned and conventional orthogonal \(x, y\) axis set.

The main advantage is that the procedure is applied separately for each suitable experimental orbit obtained and recorded, so the axis system is adjusted to coincide with the particular test result. This is easier than the earlier approach whereby orbits had to be generated to suit a given set of axes.

2.2 Displacement coefficients

The method for velocity coefficients outlined in Section 2.1 was adaptable to the measurement of displacement coefficients in conjunction with an incremental load technique. This approach was used by Martin (3) for the determination of the displacement or stiffness coefficients of fluid film bearings supporting a rotor on a stationary shaft. Figure 2 shows the systems of axes used. The rotor was assumed to be rotating at a steady speed about the rotational centre \(O'\) under the influence of a known constant load. An incremental load, \(F_1\), was applied to the rotor causing its rotational centre to move a distance \(p\) to a new equilibrium position, No. 1. The incremental load \(F_1\) was removed and another one \(F_2\) applied in a different direction resulting in new equilibrium position No. 2, a distance \(q\) from \(O'\). The co-ordinates of points \(O', 1\) and \(2\) were determined from transducer measurements and represented as \((X_0, Y_0)\), \((X_1, Y_1)\) and \((X_2, Y_2)\) relative to a global axis system \(X, Y\). From these data the angles \(\alpha, \gamma\) and \(\psi\) could be determined, thus locating the non-orthogonal axes \(p\) and \(q\). From this point the process duplicated that of the velocity coefficients determination already mentioned.

3 ADJUSTABLE HYDRODYNAMIC BEARINGS

The group has carried out work on novel forms of adjustable bearing. The adjustable bearing concept has been patented in a number of industrialised countries by BTG International Ltd (4), and has been extensively studied in both rotor and conventional journal forms. Early work by Muhsin (5) was related to a segmented form of adjustment and clearly demonstrated greatly improved stability at zero loading conditions and the potential for reduced power losses. Figure 3 shows reductions in measured frictional torques available with the novel bearing.

Martin (3) was concerned initially with improving the precision of grinding machine bearings, in particular by considering a proposed design which combined hydrostatic and hydrodynamic segments, the latter with a number of pads. The design was unusual in that the bearing surface rotated on a stationary journal. After extensive theoretical modelling and studies, a system was developed in which the hydrodynamic segment pad tilt angles could be adjusted continuously (i.e. infinitely variably) in a proactive manner.
during operation. This bearing arrangement also exhibited high stability at zero loads and clearly demonstrated an ability to reposition the centre of rotation in any required direction irrespective of the load. Allied to this was an ability to maintain a zero eccentricity under load, and changes in load.

The concept was extended in the design of two large adjustable conventionally orientated hydrodynamic bearings for a Royal Navy Type 42 Destroyer main propulsion gearbox. The bearings were tested in a land based test rig at (then) DERA Pyestock, in a project funded by the Ships Support Agency. This work has been reported (6), but Figures 4 and 5 show two characteristics of potential benefit. Figure 4 shows typical traces of journal centre orbits before and after suppression by invoking bearing adjustments, during operation. This degree of suppression was demonstrated repeatably within a few seconds and could be maintained irrespective of loading conditions. Figure 5 shows the degree of reduced temperature rise of the lubricant oil available with the novel bearing, using the same inlet supply conditions and pressure as for the conventional bearing.

4   THEORETICAL MODELLING AND VALIDATION

The group has developed design guide computer programs for hydrostatic bearings (3) based on empirical models already in conventional use. These have been used for various design studies and in the design of the hydrostatic rotor bearing referred to in Section 3 above. Tests with this rotor bearing appeared to validate the design models. It was noted, however, that in further tests based on the procedure outlined in Section 2.1 above, the bearing exhibited displacement coefficients which differed significantly in value from stiffness as predicted by the design models. These tests also yielded displacement coefficient cross terms, a factor not addressed by the design models. It is hoped that these findings will be published and could form the basis for further investigation of hydrostatic bearings.

For the adjustable hydrodynamic bearings a comprehensive theoretical model and suite of computer analysis programs were written (7) and used in the design of the practical test bearings (8). Allowance was made for anti-elastic and synclastic deformations of the adjustable segments predicted by finite element modelling and analysis, interacting with the hydrodynamic pressure fields as they were formulated. Figure 6 shows a typical computed pressure field. The analysis was highly non-linear, not satisfied until the fields of profile thickness, temperature, viscosity and pressure had all simultaneously converged to a single solution set. This approach required an extended expansion of the governing Reynolds equation (9), and predicted parameters such as displacement and velocity coefficients, static loci, temperatures, adjustable eccentricities etc. which agreed closely with measured values. Figure 7 gives an example.
5 IMPLICATIONS OF RESEARCH

The recently developed analysis and test techniques facilitate a greater understanding of the behaviour of both hydrostatic and hydrodynamic fluid film bearings under a range of operating conditions. They provide useful tools with which to study bearings of a wide range of sizes and speeds.

Results and knowledge gained from this research into improving the performance of hydrodynamic bearings have been successfully integrated in the design, manufacture and testing of practical versions of an adjustable bearing system. Adjustments can be made whilst the bearing is in operation, under all conditions of loading, providing significant reductions in friction torques and temperature rises, leading to improved operating efficiencies.

Controlled lateral positioning of rotating centres, whilst in operation, have also been demonstrated under a range of loads leading to the possibility of proactive control and maintenance of given operating eccentricities, including zero, and hence greater accuracy of location of rotating components. The same feature could be of benefit in setting and maintaining equal load sharing in multiple in line bearing arrangements. It could also enable the fine adjustment of gear meshing alignment when the gears are in their hot, loaded and rotating condition.

High stiffness and damping can be achieved for all operating conditions including zero load and zero eccentricity, or both, leading to greater stability. The stiffness and damping can be adjusted in operation to tune rotor-dynamic performance in conditions of particular interest, for example the approach to and passing through critical speed regions.

It is intended to continue investigating the performance of fluid film bearings including the novel adjustable type, high speed tilting pad types and hydrostatic bearings. The procedures and facilities developed are ideally suited to such work and will be used for further studies and comparisons of bearings for differing operating conditions, lubricant types etc.

6 CONCLUSIONS

The design and operation of fluid film bearings is regarded as a key area of tribology today. Recent work and developments have been closely linked to newly active researchers at MSc and PhD level.

Results are manifest as developments in the design of test rigs, measurement techniques, theoretical analysis and in the design of bearings themselves. These have clearly shown the potential for gains and benefits in efficiency, reliability and cost in the operation of plant and equipment.
REFERENCES


FIG. 1 Generated “figure of 8” orbit

FIG. 2 Orbit local axes sets
FIG. 3 Measured friction torques

- \( \Delta \) Zero Tilt and Zero Radial Adj. -Case Study 1
- \( \circ \) -0.1° Tilt and 0.015 mm Radial Adj. -Case Study 3

- \( P_s = 2.07 \) bar (30 psi)
- Using VG32 Mineral Oil
- Tout = 43 °C
- Load = 598 Newton

Measured Torque, \( T_1 \) (Nm)

Journal Speed, \( N \) (rpm)
FIG. 4 Controlled journal orbit reduction
FIG. 5 Comparative oil temperature rises

FIG. 6 Computed pressure field
FIG. 7 Controlled adjustment of displacement coefficients