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The Potential Use of New Forms of Adjustable Hydrodynamic Bearings in the Intelligent Monitoring and Maintenance of Machine Accuracies

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Abstract

New forms of adjustable fluid film hydrodynamic bearings have shown great potential for suppressing vibrations and maintaining accuracy of location of rotational centres. First demonstrated on a simulated grinding wheel machine tool rotor and spindle system, the principal has also been applied to the support bearings in a large marine gearbox application. In both cases the bearing adjustment feature was used to maintain or displace in a pro-active manner the centre of rotation irrespective of loads and changes of load, and to suppress instabilities and vibrations provoked by running at low and zero loads. It is felt that these characteristics could be of significant benefit in a range of applications when coupled with appropriate intelligent systems for monitoring and control.
1.0 Introduction

Fluid film bearings are widely used in many areas of mechanical engineering where rotating parts are supported. In particular, hydrodynamic bearings by generating their own lubricant film pressures can carry large loads, provide significant stiffness and damping, while preventing physical contact between the bearing surfaces. They can be of radially and axially supporting forms. The radial type most commonly comprises a shaft or journal rotating within a stationary bearing housing. Less common is the inverse arrangement of a rotor on a stationary shaft, spindle or journal. The axial form is the thrust bearing where the rotating member bears on thrust faces, reacting axial loads. The clearance space between the two surfaces is small, typically 0.1% of the journal diameter, and this clearance space is occupied by the lubricant film which provides the hydrodynamic lubrication.

Hydrodynamic lubrication is a self-sustaining fluid film separation of the two bearing surfaces. Loads are carried by pressures within the lubricant film generated independently of the supply pressure, the latter is merely to ensure adequate supply of lubricant to the bearing. The lubricant, usually oil, must have significant viscosity, there must also be relative motion of the two surfaces, and the oil film shape must at some part be convergent. The magnitude of this convergence is very small, for example less than 0.1°, but its effect is most significant.

In a journal bearing arrangement the convergence is achieved by an eccentricity between the centres of the stationary and rotating members. This is produced by the loads acting, the more load - the greater eccentricity - the greater the convergence and higher film pressures to provide the reaction and hence carry the load. Figure 1 shows a cross section of such a pressure field, with the clearance and eccentricity both greatly exaggerated for illustration.

By implication, however, in the absence of loading there will be no eccentricity and no pressure field. This in turn leads to an unstable condition where the rotating member is only lightly supported and its centre of rotation can itself rotate or vibrate in the presence of any small disturbing forces such as an out of balance or nearby gear teeth meshing.

The phenomenon of hydrodynamic lubrication was analysed mathematically by Reynolds in 1886 and most hydrodynamic lubrication theory involves some form of Reynolds' pressure field equation. This relates velocity and pressure induced fluid flow based on assumptions of continuity of volume flow rate and equilibrium of fluid film forces. It predicts the substantial pressures generated which maintain full film lubrication under large loads, and indeed the absence of pressures under light loads. There are thus two main issues of concern in an operating hydrodynamic bearing: 1) that load carrying capacity involves the eccentricity required to produce the pressure fields, stiffness and damping, and 2) that in the absence of load there is no eccentricity, no pressure field and low stiffness and damping. The ideal arrangement would be to have the load capacity whilst maintaining a zero eccentricity for all loads, and significant stiffness and damping available at zero and light loads. This paper outlines recent developments which have led to this combination being achieved and demonstrated. It also anticipates the potential for a proactive monitoring and control system to both maintain such accuracy of location and quickly suppress any instability vibrations that may be provoked at light loads.

2.0 Previous developments in hydrodynamic bearings

There have been many developments in hydrodynamic bearings intended to improve performance in terms of stiffness, damping, load capacity etc. For example it is possible to
"build in" a converging oil film shape in the stationary member. This has the effect of reducing the eccentricity that would be required otherwise to provide a converging shape. This imposed convergence can be by special machining of the bearing surface or by machining and assembling the bearing in two halves to provide an offset. Another successful idea was the use of bearing pads which were pivoted and allowed to adopt, within limits, different angles of convergence to suit different loading conditions. These are generally known as tilting pad bearings, or pivoted shoe bearings. Figure 2 shows some of the options in sketch form, again with clearances and profiles greatly exaggerated (and one picture incorrectly orientated!).

In general required profiles have been difficult to make with conventional machine tools, but Albin et al have predicted the rise of precise profile generation in a production environment [1]. In China the Shanghai Machine Tool works introduced a contoured multiple wedge hybrid bearing with clearances about 40% of current designs, with consequent improvements in accuracy, stiffness and stability [2]. All of these developments have been to varying degrees successful in improving operating performance compared with the conventional forms. Like the conventional arrangement, however, they are still reactive to the loading conditions. The degree of eccentricity and hence pressure field forces are generated in response to the prevailing operating conditions.

3.0 Principles of the adjustable hydrodynamic bearing

An adjustable bearing concept was devised being concerned initially with improving the precision of grinding machine bearings, in particular by considering a proposed design which combined hydrostatic and hydrodynamic segments. This design was unusual in that the bearing surface rotated on a stationary journal, this rotor being a simulated grinding wheel assembly. The journal diameter was 70 mm and the hydrodynamic segments comprised two bearings in parallel. Each of these hydrodynamic bearings comprised 4 separate pads incorporating a means of adjustment. By such means the individual pad tilt angle could be varied from zero to a maximum, i.e. contact with the bearing surface. Each pad could develop its own pressure field and to a small degree was conformable and the overall bearing system was geometrically and materially highly non-linear. The analysis of such a system results in an expanded form of the original Reynolds' equation [3] from which a comprehensive computer model was developed [4]. After extensive theoretical modelling and studies [5], a practical system was designed in which the pad tilt angle could be adjusted continuously (i.e. infinitely variably) in a proactive manner during operation. The lubricant was a mineral oil of ISO VG 32 supplied to each pad separately.

The principle of adjustment involved the bearing pad being supported by a tapered pin. The pin in turn was located by a thread and when turned could translate along an axis parallel to the longitudinal axis of the journal. In so doing the pad tilt angle could be varied whilst still supporting a given load. A practical version and special test rig were manufactured in which the relative rotor radial position and thereby its centre of rotation could be determined by non-contacting inductive transducers. The bearing clearly demonstrated characteristics predicted by the theoretical model. This included the ability to move the position of the centre of rotation in any lateral direction. It was a simple matter to reset the centre of rotation to an initial position whenever the load magnitude, or direction, or both were altered. This included maintaining a zero eccentricity condition under radial load, and changes in load. Radial
stiffness was also high in all directions, with loads applied in any direction, and variable by means of the adjustment system. Figure 3 shows the test rotor. Figure 4 shows the adjusted radial positions of the centre of rotation of the rotor, adjusted during operation in 4 approximately orthogonal directions from an equilibrium position for a given load and speed combination. The movement was equivalent to a change in eccentricity ratio of about 0.6 to 0.7 in each case.

Further tests were conducted on a military ship gearbox test rig installation, [6]. This was a conventional orientation, i.e. rotating shaft within a stationary bearing housing. An adjustable bearing assembly was designed and manufactured to fit within the same packaging space and constraints as the hydrodynamic bearing normally used. The shaft contained the pinion of a double helical gear set and was supported by two bearings. One of these bearings was substituted for an adjustable bearing assembly. Figure 5 shows the installation of the adjustable bearing assembly prior to testing. The journal diameter was 190 mm and speed 1500 rev. min\(^{-1}\), and the oil was a mineral oil for steam turbine gear applications, supplied through the standard supply pipes and at the normal pressure. The adjustable test bearing comprised 4 separate bearing pads each supported by 2 tapered pins which on being turned could adjust the pad radial position and/or tilt angle. A schematic view is shown in Figure 6. The shaft was driven directly by an electric motor and the load on the test bearing could be set in response to changing the output resistance load of an AC alternator coupled to the main gear through a speed-increasing gearbox. The relative radial position of the journal and thereby its centre of rotation were again determined using inductive transducers.

All tests were conducted with the rig warmed up and running at its nominal pinion shaft speed of 1500 rev.min\(^{-1}\). Pad adjustments were carried out in situ during operation, with loading set constant. For test conditions giving low net radial loads, journal orbits were produced and maintained at shaft rotational speed. It was believed that they were driven by small variations in radial load due to the gear teeth meshing, possible unbalances in pinion and shaft masses and influences due to a shaft coupling. Once a journal orbit was established, investigations into orbit suppression were by manually imparting equal adjustments to each pad’s trailing pin (Nos. 2,4,6,8), leaving the leading pins (Nos. 1,3,5,7) set at zero. Equal pad adjustments of tilt angle 0.087\(^\circ\), then 0.174\(^\circ\), were set for all 4 pads. This had the effect of quickly suppressing the journal orbit in the time it took to make the adjustments (just a few seconds). The orbit could be equally quickly "reinstated" by reversing the adjustments. These experiments were repeated many times on different occasions, all with the same result. Figure 7 shows a typical orbit suppression. In this case the unsuppressed orbit was approximately 0.13 mm diameter, the same orbit but with the pad adjustments invoked was reduced nearly 80% to 0.03 mm diameter.

The effect of asymmetric adjustments was also tested. For given load conditions pad adjustments were all set first at zero. Then each pad was adjusted singly in turn to tilt angle 0.174\(^\circ\), with all other pads set at zero, and the effect on journal centre position noted. The effect was very similar to that observed with the rotor bearing. It was a simple matter to move the centre of rotation in any of 4 directions by single adjustments to the appropriate pad. Figure 6 illustrates the magnitudes and directions of tests given for one set of loading conditions, repeated a few days apart. The average journal centre adjusted movement was just over 0.1 mm, equivalent to a shift in eccentricity ratio of 40 % - in any direction and for the same constant load and speed. As with the adjustable rotor bearing it was also a simple matter to reinstate a given orbit centre position if the load was changed.
It should be emphasised that for all tests the adjustments were carried out whilst the bearing continued to operate at a constant speed, supporting a given load with no ill effects. Indeed in all cases the overall oil temperature rise was less than that observed with the conventional bearing for the same conditions. Individual pad oil exit temperatures also provided a useful secondary indication of the loading conditions on the pad. The temperatures increased with the severity of loading.

4.0 Potential use in machine tool context

Experiments and development so far have been conducted using prototype bearing designs and test rigs. In addition comprehensive theoretical models have enabled various studies to optimise design features and predict performance characteristics. Adjustments made to individual bearing pad tilt angles and radial position have been performed manually using hand tools. It has been shown that with 4 pad bearings, having symmetry of design and dimensions in 4 quadrants, that it is a simple matter to predict and implement particular adjustments to suit given conditions. For example to suppress a vibration orbit and/or to move the position of the centre of rotation in a desired direction is a straight forward procedure free of any complicated cross-coupling effects.

It is felt therefore that in terms of logic circuitry design, instrumentation and adjustment control systems the technology is already well developed and available. For example the basic instrumentation used for experimental work is shown in Figure 8. This comprises pairs of inductive proximity transducers and thermocouples with amplifiers, processing units and displays. This was all that was needed to monitor and measure the shaft relative position, the shape and size of journal orbits, the temperatures of oil inlet, outlet and exit from each pad. The oil temperatures at each pad exit provide a useful secondary monitoring of the loading and operating conditions on each pad. The inductive transducers themselves are reliable and not affected by the presence of any oil in the gap between their ends and the shaft and are already widely used in industrial applications. Thermocouples are also of course standard items.

For an automated control and monitoring function the output signals representing journal position, vibration and pad oil temperatures are all low voltages. These can be linearised and amplified in order to drive either electrical or hydraulic servo systems to apply individual adjustments. Figure 9 gives and overall view. Each adjustment pin could be connected via a threaded and worm drive arrangement providing 2 levels of gear reduction and non-reverse to balance and transmit any conceivable pad loading. All mechanical parts are relatively simple and can be made with conventional materials and machine tools. This system would provide means to maximise machine tool accuracies by virtue of a) precision control of rotating centres b) suppression of vibration instabilities generated in conditions of light loading and c) reduced temperature change effects.

Work is continuing in the feasibility of using the system in a number of areas where conditions of light load can generate instabilities in the fluid film bearing supports. The field of machine tools in particular is one where the maximum use can be made of the potential benefits of the adjustable bearing system so far demonstrated. It can be used to improve and maintain workplace accuracies using remotely connected condition monitoring and control activation software.
Conclusions

Two completely different forms of hydrodynamic bearing have been designed and manufactured to include a similar novel infinitely variable means of pad position and tilt adjustment. Both have been tested for a variety of operating conditions.

Both novel bearings demonstrated the ability to reposition the rotating member's centre of rotation whilst in operation, in any direction, for a range of loads with differing initial adjustment conditions (e.g. full adjustments or zero adjustments).

The size of journal orbits could be repeatedly reduced by at least 50% for any of the test load conditions, including that of zero radial load. Such reductions were made in seconds by manual means whilst the bearings continued in operation.

The novel bearings produced oil temperature rises which were consistently lower than those for comparable conventional bearings using the same oil supply system. Examination of all bearings subsequent to testing showed negligible wear or damage. The individual pad oil exit temperature provided a simple and reliable secondary means to monitor the loading conditions on the particular pad.

The characteristics demonstrated indicate a number of potential benefits and improvements available in the design and operation of fluid film bearings, using conventional materials and manufacturing techniques. With appropriate monitoring and control arrangements the bearing system could be adapted to provide automatic monitoring and maintenance of high precision accuracies in machine tools.

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References


Fig. 1 Conventional journal bearing scheme

Circumferential pressure distribution
Fig. 2 Variable profile journal bearings
(Reproduced by permission of the Council of the Institution of Mechanical Engineers)
Fig. 3 Bearing test rotor
Fig. 4  Rotor adjustment positions
Fig. 5 Adjustable bearing being installed
Fig. 6 Adjustable bearing pads and journal displacements
Fig. 7 Journal orbit suppression
Fig. 8 Basic instrumentation