Measured displacement coefficients of an adjustable hydrodynamic journal rotor bearing
Coefficients de raideur mesurés d'un palier hydrodynamique ajustable

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In simulating a proposed machine tool grinding wheel design, a bearing system comprised a stationary spindle or shaft with fluid film bearings supporting a belt driven rotor. Two hydrodynamic bearings acted in parallel, and built into the shaft were the means to effect continuous pro-active adjustments to their performance characteristics during operation, irrespective of load, speed and other running conditions. The design is outlined, as is a specially constructed test rig for evaluating it. A method for determining the combined displacement coefficients is given which used an incremental load technique with dedicated axis system selection. Observations are given on the bearings' performance characteristics. The effects of the adjustments on rotor eccentricity are shown, along with the ability to maintain a given rotor eccentricity, including zero, irrespective of load and changes in load. Results of measured rotor displacements and displacement coefficients are also given showing that the bearing exhibited high stiffness at zero load and eccentricity, and that stiffness could be changed by adjustment if required, thereby "tuning" the system rotordynamics behaviour. Comparisons are made with results of others' work on the measured displacement coefficients of a conventional type of tilting pad bearing. An approach to uncertainty estimation of measured data is included.

Dans la simulation de l'axe d'une machine de rectification, le système comporte une broche ou un rotor stationnaire avec des paliers fluides qui supportent un rotor entraîné par une courroie. Deux paliers hydrodynamiques construits dans l'arbre et fonctionnant en parallèle ont permis la modification continue de leur performance durant le fonctionnement, sans dépendre de la charge, de la vitesse de rotation ni des autres conditions de fonctionnement. Cette conception est présentée dans l'article tout comme un banc d'essais spécialement construit. Une méthode est ensuite proposée pour déterminer les coefficients de raideur en utilisant une méthode de charge incrémentale. Les caractéristiques des paliers sont ensuite données. Sont donnés les effets des modifications sur l'excentricité ainsi que la capacité de maintenir une excentricité donnée, y compris nulle, pour toute charge ou variation de la charge. Les déplacements et les raideurs mesurés sont aussi donnés et montrent que le palier a une raideur élevée pour une charge et une excentricité nulle. Ils montrent aussi que la raideur peut être modifiée à la demande, réalisant ainsi un réglage du comportement dynamique du rotor. Des comparaisons avec les raideurs mesurées par d'autres auteurs pour des paliers à patins oscillants conventionnels sont aussi données. Une approche pour estimer l'incertitude des données mesurées est également incluse.

1 INTRODUCTION

There is an impetus for improving performance of fluid film bearings whether indicated by stiffness, damping, stability, noise signatures, temperature rises, energy loss etc., aspects previously reported by the authors and others, [1,2,3,4]. A novel form of adjustable fluid film bearing has hydrodynamic characteristics that can be changed in a continuously controlled manner during operation, irrespective of the loading conditions on the bearing, a *pro-active* characteristic. The principle can be applied to conventionally orientated journal bearings, i.e. shaft rotating; to inverse orientations, i.e. a rotor on a stationary shaft, and to thrust bearings. Investigations have been carried out on theoretical and practical models for journal bearings of both orientations [5,6,7]. This paper concerns potential applications in high-speed machine tools where stability and accuracy are vital, for example in grinding operations. Some aspects of performance as measured in tests are presented.

A large rotor was supported on a stationary shaft within which were a pair of hydrodynamic oil film bearings each comprising 4 adjustable surface segments. Each segment was formed from the shaft material rigidly attached at one end and supported by an adjuster pin such that by turning the adjuster pin the free end of the segment could be displaced radially into the clearance space. Figure 1 shows the shaft, Fig. 2 shows a cross section of one bearing with adjuster segments, G1 – G4 (with displacements greatly exaggerated). The diameter of the shaft was 70.0 mm with adjustments set at zero, notional radial clearance was 0.035 mm, being reduced in the warmed up condition as noted below. Each adjustable segment had a width of 25.0 mm with the free end having a radial displacement of 0.0254 mm (0.001 inches) for one full turn of the adjuster pin. Forces were applied to the rotor by large external non-contacting electro-magnets. These could be changed to provide a net steady load, W , in any direction, acting in conjunction with the rotor weight. Full details of the load application method have been published, [8]. Pairs of orthogonal inductive transducers sensed the radial displacements of the rotor assembly.

Fig 1 – Central shaft with one set of adjuster pins

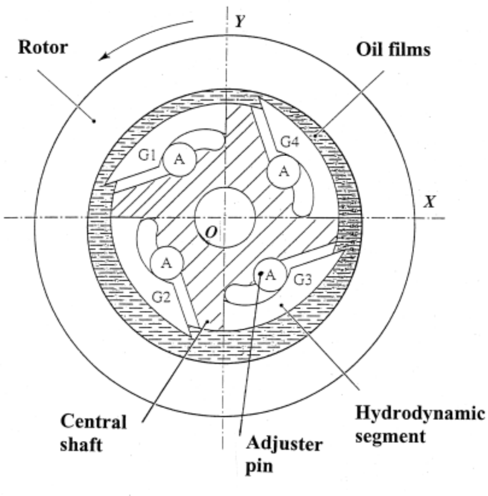

Fig 2 – Cross section scheme of adjustable hydrodynamic segments

2 OUTLINE OF TESTS

In determining bearing running data, adjustments to the hydrodynamic bearings were made equally to both and results divided by 2 when referring to a single bearing. Advantage was also taken of the 4 planes of symmetry of each bearing cross section to avoid the need to determine the shaft centre position during the test warmed up running condition. (In warmed up conditions determining the bearing system absolute stationary centre can be problematical owing to thermally induced distortions and movements in test rig structures). Figure 3 shows the reference parameters used.

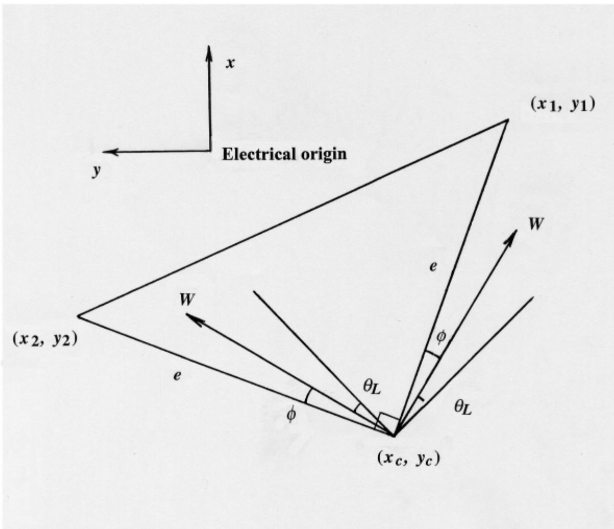

Fig 3 – Shaft centre and rotor eccentricity

Point 1 at x_1, y_1 represents the rotor equilibrium position for a given net load in direction θ_L. Point 2 at x_2, y_2 represents the rotor equilibrium position with the same load but at a direction 90° to the first direction. Both points are for the same warmed up test conditions. The following expressions were derived and used:

$$\text{Eccentricity, } e, = \frac{\sqrt{(y_1 - y_2)^2 + (x_1 - x_2)^2}}{2\sin \pi/4} \qquad (1)$$

$$\text{Attitude angle, } \phi_A = \frac{\pi}{4} - \tan^{-1}\frac{y_2 - y_1}{x_1 - x_2} - \theta_L \text{ when } \begin{matrix}|x_2| > |x_1| \\ |y_2| < |y_1|\end{matrix} \text{ and } \begin{matrix}|x_2| < |x_1| \\ |y_2| > |y_1|\end{matrix}$$

$$\phi_A = \tan^{-1}\frac{y_2 - y_1}{x_2 - x_1} - \theta_L + \frac{\pi}{4} \text{ when } \begin{matrix}|x_2| > |x_1| \\ |y_2| > |y_1|\end{matrix}$$

$$\phi_A = \tan^{-1}\frac{y_2 - y_1}{x_2 - x_1} - \theta_L - \frac{3\pi}{4} \text{ when } \begin{matrix}|x_2| < |x_1| \\ |y_2| < |y_1|\end{matrix} \tag{2}$$

For θ_L referred to displacement transducer set local x axis.

Thus to determine eccentricities and attitude angles it was not necessary to know the shaft centre position in absolute terms nor the transducer electrical origin in the warmed up test condition, provided symmetry was maintained in equalising all segment adjustments, for both hydrodynamic bearings. This avoided the need to determine such absolute data in testing. This technique had the added advantage in allowing for thermal drift of inductive transducer electrical origins. It had been noted that during testing and over time the transducers were prone to drift errors but that their calibrated characteristic gradients remained constant. Thus data derived from *changes* in position measurement were more reliable. Tests were carried out for chosen rotor load directions, usually "on-segment" or "off-segment" and the load value increased in steps, with both hydrodynamic bearings acting. Rotor lateral/radial positions were noted and a specially written computer program processed the transducer voltage readings to produce the net displacements (i.e. changes in position) and hence rotor attitude angles and the eccentricities.

Each test was repeated for equal segment adjustments of zero, small, and medium designation, equivalent respectively to radial clearance, C_{32}, in the warmed up condition of 0.023 mm, 0.014 mm, and 0.010 mm. A large adjustment, equivalent to C_{32} = 0.006 mm, was invoked sparingly to avoid the risk of contact damage. It was also known from the theoretical studies that small adjustments were sufficient to confer all the noted characteristics of the bearing system. For static locus plots the rotor centre eccentricities were non-dimensionalised with the zero adjustment radial clearance, C_{31}, determined as 0.0226 mm for an average warmed up condition. Static locus results were plotted in this case for both hydrodynamic bearings acting summatively in parallel.

In terms of coefficients a rotor mass system of mass M supported by 2 fluid film bearings in parallel and when subject to a repeating dynamic force or load of magnitude F is described by the equations of motion:

$$\begin{aligned} F\cos\Omega t &= a_{xx}x + a_{xy}y + b_{xx}\dot{x} + b_{xy}\dot{y} + M\ddot{x} \\ F\sin\Omega t &= a_{yx}x + a_{yy}y + b_{yx}\dot{x} + b_{yy}\dot{y} + M\ddot{y} \end{aligned} \tag{3}$$

Where the coefficients also in this case apply to the bearings acting summatively. For a steady state situation with the bearings supporting the rotor at an equilibrium position the velocity and acceleration terms vanish leaving the displacement coefficients as indications of stiffness thus:

$$\begin{aligned} F_x &= a_{xx}x + a_{xy}y \\ F_y &= a_{yx}x + a_{yy}y \end{aligned}$$

For small changes in forces and small displacements the coefficients can be assumed to be linear and so can be determined from the net measured displacements in response to known incremental forces, ΔF_x and ΔF_y . The process for doing this has been published in detail [8] and also used in evaluating hydrostatic bearing stiffnesses [9]. The method itself was adapted from an extended selected orbit technique reported by Parkins for determining velocity coefficients [10]. The rotor was assumed to be rotating at a steady speed about a rotational centre O' under the influence of a known constant load, shown in concept in Fig. 4.

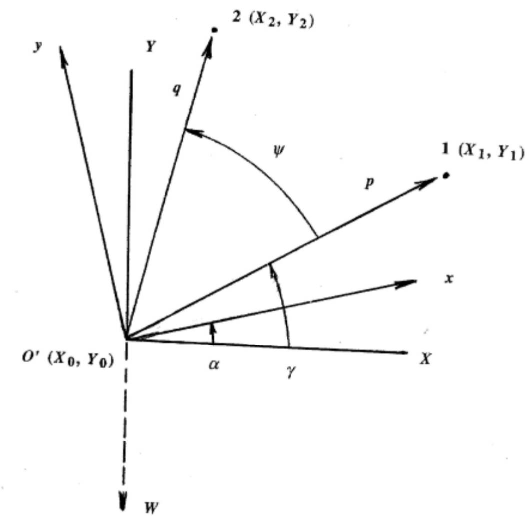

Fig 4 – Reference axis sets and angles used

An incremental load, F_1, was applied to the rotor causing it to move a distance p to a new equilibrium position, No. 1. The 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'. F_1 and F_2 were chosen so as to make ψ a reasonable size, and the whole test could be conducted quickly for the same warmed up running condition.

The co-ordinates of points O', 1 and 2 were noted as (x_0, y_0), (x_1, y_1) and (x_2, y_2) and the angles α , γ and ψ evaluated, thus locating axes p and q. These were then used in defining forces F_p and F_q, and for transposing to x, y axes to determine the required displacement coefficients. Displacement coefficients thus obtained for both hydrodynamic bearings acting in parallel were then divided by 2 to relate to each hydrodynamic bearing. A range of tests was carried out with the bearing operating in zero segment adjustment plain mode, and with segment adjustment settings small, medium and large, as defined above. The tests were carried out for 2 rotor speeds, 2500 and 3500 revolutions per minute. Equilibrium positions, O', were established for loads of 3 magnitudes, 0, 300 and 450 N, each in 2 separate directions, on-segment and off-segment. Each test was conducted for two different pairs of incremental load. For the large adjustment settings the equilibrium load was limited to the zero case to avoid the risk of possible segment contact with the rotor bearing surface. This set of tests yielded a map of coefficient values from which general trends were observed.

The need to consider uncertainty of data in fluid film bearing measurements was emphasised by Kostrzewsky and Flack [11], particularly for deriving stiffness and damping coefficients from induced orbits. Uncertainty in measurements was reduced as noted above by relying on changes of position measurement for a given test rather than absolute values. Also for each single transducer channel the characteristic gradients of signal voltage output to known displacements were plotted before and after testing and the worst case variation in gradient applied as a plus-minus tolerance to all transducer channels. This tolerance was compounded with resolution and error tolerances for other instrument readings such as voltages from the digital voltmeters and each data point expanded to a circle to embrace the likely area of uncertainty. Finally conclusions and observations were based on trends in data plots rather than reliance of absolute values.

3 DISCUSSION OF RESULTS

The position of the rotor centre of rotation could be displaced on demand by significant amounts in a predictable and repeatable manner irrespective of loads acting. Such a degree of adjustment could be of great benefit in maintaining the location of the rotational centre when loading conditions change, for example in the machine tool application originally conceived. For static locus plots for a given range of loads directed on-segment, the rotor eccentricity ratio decreased markedly as adjustments were invoked. Figure 5 shows an example. The maximum eccentricity with the large adjustment (C_{32} = 0.006 mm) was less than 50 % of that for zero adjustment, an indication of stiffness increasing with adjustments applied. The attitude angle also increased with that for the larger adjustment, backing away from the load line. In this condition the space available for side leakage of the oil would be less (for the main loaded hydrodynamic segment) and the locus tends towards the Sommerfeld prediction for a conventional plain journal bearing ($\phi_A = 90\ ^o$) which neglects side leakage.

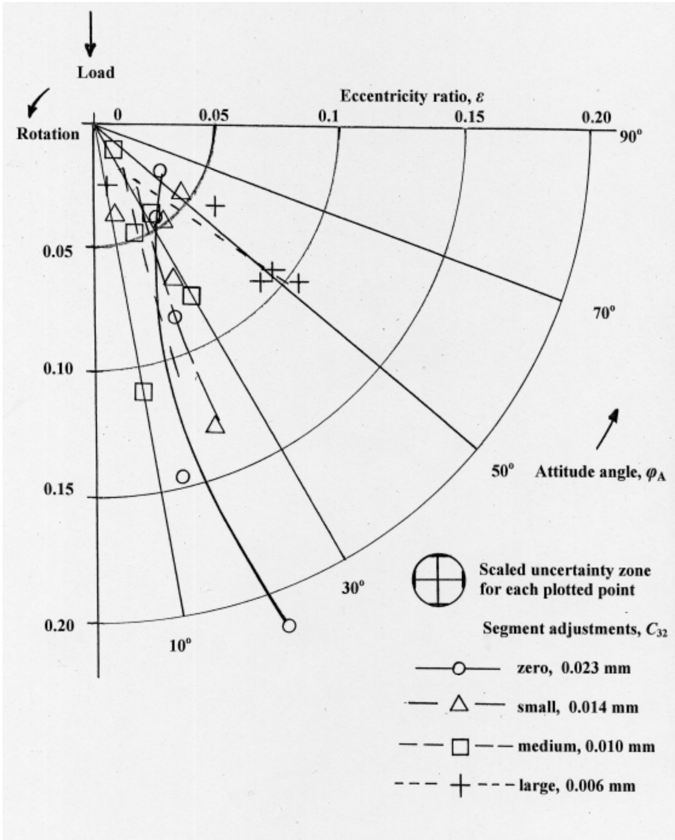

Fig 5 Static locus plots for different adjustments, 3500 rpm

For the off-segment direction, however, the reverse was the case with the attitude angle veering towards the load line, but the relative magnitudes of eccentricity were again reduced markedly with increasing adjustments. Such a capability in reducing eccentricities could be useful in for example suppressing an unstable vibration orbit.

The direct coefficients a_{XX} and a_{YY} increased with increasing adjustment in an approximately linear manner, for both on and off-segment equilibrium load directions. Values were also in close agreement with predictions from the theoretical model, [5]. Figure 6 shows a sample result for a_{YY}.

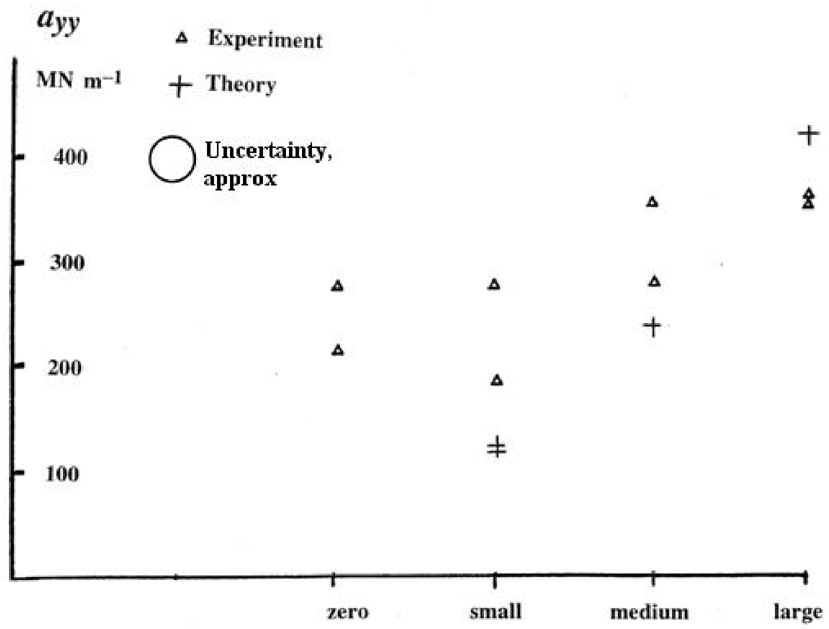

Fig 6 – Sample result of displacement coefficients

Indirect coefficients a_{xy} increased (i.e. more negative) non-linearly with adjustment, for the on-segment load case, but appeared to peak at around the medium adjustment for the off-segment case. The latter case is also closely reflected by the theoretical model results. The indirect coefficient a_{yx} also appeared to peak for both load orientations. Magnitudes of cross coefficients were consistently as high as direct coefficient values, again as predicted by the theoretical model.

In general the direct coefficient a_{yy} was measured higher than a_{xx} for each adjustment setting when the load direction was off-segment. For the on-segment load direction the converse was observed. Similar types of variations to these and to those of the cross coefficients were reported by Parkins and Horner [12] using a 5 tilting pad bearing of 80 mm shaft diameter, at various speeds including 3000 rpm. Santos reported a control of direct coefficients by hydraulic means in tilting pad bearings which showed an increase with pad support pressures [13]. Overall the rotor hydrodynamic bearing discussed in this paper exhibited high stiffness in all directions, with loads applied in any direction, and that stiffness could be changed by pro-active means on demand. These are characteristics which could be of significant benefit in a number of applications, including that of a machine tool grinding wheel as originally envisaged.

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6 NOMENCLATURE

a_{xx}	Direct displacement coefficient, force *x*, displacement *x*
a_{xy}	Cross displacement coefficient, force *x*, displacement *y*
a_{yx}	Cross displacement coefficient, force *y*, displacement *x*
a_{yy}	Direct displacement coefficient, force *y*, displacement *y*
C_{31}	Concentric radial clearance = $(D_3 - D_1)/2$
C_{32}	Concentric radial clearance with adjustments applied = $(D_3 - D_2)/2$
D_1	Nominal diameter of shaft, no adjustments
D_2	Diameter of shaft, including adjustments
D_3	Rotor bore diameter
e	Eccentricity of shaft and rotor axes
F	Exciting force amplitude
$F_{1,2}$	Loads applied to rotor, rotor positions 1,2
$F_{p,q}$	Loads on rotor, *p,q* axes components
O'	Rotor centre equilibrium position for a known load
p, q	Local axes, variable orientation, not necessarily perpendicular
t	Time
W	Rotor load
x, y	Local axes, variable orientation, perpendicular
X, Y	Fixed global axes, perpendicular
$x_{1,2}, y_{1,2}$	Rotor translatory displacement components, positions 1,2
α	Angle between *x, y* and *X, Y* axes sets
γ	Angle between *X* and *p* axes
ΔF_x , ΔF_y	Incremental changes in load, *x*, *y* components
θ_L	Angle to load direction from *X* or *x* axes
ϕ_A	Attitude angle
ϕ	Angle between *x* and *q* axes = $\psi + \gamma - \alpha$
ψ	Angle between *p* and *q* axes
Ω	Frequency of exciting force