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A TECHNIQUE TO MEASURE ROTORDYNAMIC COEFFICIENTS IN HYDROSTATIC BEARINGS

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ABSTRACT

An experimental technique is described for measuring the rotordynamic coefficients of fluid film journal bearings. The bearing tester incorporates a double-spool shaft assembly that permits independent control over the journal spin speed and the frequency of an adjustable-magnitude circular orbit. This configuration yields data that enables determination of the full linear anisotropic rotordynamic coefficient matrices. The dynamic force measurements were made simultaneously with two independent systems, one with piezoelectric load cells and the other with strain gage load cells. Some results are presented for a four-recess, oil-fed hydrostatic journal bearing.

INTRODUCTION

New-generation rocket engine turbopumps will make greater use of fluid film journal bearings. This type of bearing has potential for long life and can use the engine's propellants as lubricant. Part of the design process for a turbopump involves rotordynamic analysis of the rotor-bearing system. As part of the rotor dynamic analysis, bearing coefficients, in the form of stiffness, damping and inertia, are necessary inputs. While rotor mechanics is well-characterized and relatively straightforward, determination of accurate bearing coefficients is not so well-established. Lund's review of the concept of fluid film bearing coefficients has a comprehensive reference list that includes experimental and analytical work (1). Taken as a whole, these references reveal the general difficulties in measuring bearing coefficients and the lack of uniform agreement between measured and analytical coefficients. The data base for rocket engine-type fluid film bearings is extremely small. The turbopump requires a bearing that operates at high speed, with small clearance and with low viscosity fluid. These parameters make experimental coefficient determination especially challenging. Butner and Murphy's report, as an example, reveals the added research and operational complexities of measuring coefficients for such a bearing (2).

As part of NASA's effort to develop fluid film bearings for rocket engine turbopumps, it has an ongoing program with Case Western Reserve University (CWRU) to develop a reliable and accurate measurement technique for bearing coefficients that can be ultimately applied to high-speed, low-viscosity bearings. The current experimental setup tests a hydrostatic oil journal bearing, but can accommodate other types of fluid film bearings. Coefficients have been extracted at low static eccentricity ratios and compare well with theory. In addition, the tester's unique load measuring system gives a high level of confidence in the data.

EXPERIMENTAL FACILITY

The linear anisotropic bearing model is shown in Figure 1, where K,C and D designate respectively stiffness, damping and inertia. In addition, the two fundamental methods to extract the coefficients from a dynamic system (i.e. fluid film bearing) are shown. The coefficient extraction method used at CWRU, and fully explained in reference 3., is the linear impedance model.

The test facility is located at CWRU and is fitted with independent air, water and oil systems. The bearing tester, shown in Figure 2, consists of a double-spool shaft assembly supported by rolling element bearings, and overhung test section. The double-spool shaft assembly is configured so that the orbital eccentricity of the inner shaft (relative to the outer shaft) can be accurately set from zero to 0.060 inch. This is achieved by having the outer spindle shaft comprised of two closely fitted cylindrical portions, with their respective mating surfaces eccentric to their respective centering surfaces. Each spindle shaft is independently driven by a variable speed drive and can be driven in either forward or backward rotation. The net result is a controlled circular orbit of the journal, with an orbit frequency independent of the journal spin speed. The resulting dynamic force signals exerted upon the test bearing are then measured by the load support system.

Figure 3 shows a lobe bearing installed in the tester. In this earlier version of the tester, the load support system consisted of only four piezoelectric load cells. Figure 4 shows the current load support system. Each load path consists of a strain-gaged load link in series with a piezoelectric load cell. This arrangement provides simultaneous measurement of the dynamic forces by two independent systems. The strain-gaged system also permits measurement of static radial forces. Both systems are calibrated in place. Journal displacements are measured by eddy current proximity probes.

The outer spindle, which produces the orbit frequency, is equipped with a timing disk at its drive end. The timing disk contains 360 equally space slots that interrupt a light beam, triggering an A-to-D converter as an external clock to the data acquisition PC. Eight channels of data are taken A-to-D, which include four force signals and four displacement signals. Thus, 45 digitized data points are taken from each channel per cycle of orbit. Typically, data is taken for 50 consecutive cycles and time-averaged. Data is taken at sufficiently off-synchronous frequencies so that extraneous signals not coherent with the orbit frequency are essentially filtered out (i.e. mechanical and electrical runout and 60 Hz). The eight time-averaged signals are then Fourier decomposed to extract the fundamental orbit frequency signal components, which provide the inputs to extract the bearing coefficients.

SAMPLE OF RESULTS

The test bearing details are given in Figure 5. A typical example of comparison between the dynamic load signals of the two independent systems is shown in Figure 6. The two signals shown are time-averaged over 50 consecutive cycles, and show very few harmonics. Numerical comparison between the fundamental frequency components shows only a 1.8 % difference in amplitude and 0.3 degree difference in phase.

Figure 7 is a typical example of comparison between experimental and analytical stiffness and damping coefficients. This test condition was run at zero static eccentricity. As expected, the measured bearing coefficients are close to being isotropic. Direct coefficients are approximately equal in x- and y- directions. Cross-coupled stiffness in y- direction is equal in magnitude and opposite in sign to that in x-direction. Cross-coupled damping is close to zero. Predictions agree quite well with measured values. The biggest discrepancy is in over-prediction of direct stiffness. This is probably due to not including the effects of oil compressibility in the relatively deep (0.1875 inch) bearing recesses.

CONCLUDING REMARKS

The experimental results obtained so far with the CWRU tester have agreed quite well with analytical predictions. The self-checking nature of the dual load measurement system provides for greater accuracy in measuring fluid film forces, which has always been a difficult and uncertain aspect of coefficient determination.

A new test bearing holder is currently being fabricated so that tests can be performed at static eccentricity ratios up to 0.9, and the eccentricity can be quickly and accurately adjusted. This will allow for tests over a wide range of operating conditions and also for quicker setup and testing of a given bearing.

Testing to higher eccentricities will also verify over what range of eccentricities the linear impedance model can be expected to hold, as fluid film bearing coefficients should theoretically become nonlinear when the bearing becomes highly eccentric.

REFERENCES

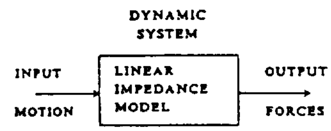
1.) Lund, J.W., "Review of the Concepts of Dynamic Coefficients for Fluid Film Journal Bearings," ASME Journal of Tribology, Vol. 109, 1987, pp. 37 - 41.

2.) Butner, M., Murphy, B.T., "SSME Long-Life Bearings," NASA CR179455, 1986.

3.) Adams, M.L., Sawicki, J.T., Capaldi, R.J., "Experimental Determination of Hydrostatic Journal Bearing Rotordynamic Coefficients," IMechE Proceedings of 5th Rotating Machinery Symposium, C432/145, 1992.

Anisotropic Model

$$\begin{Bmatrix} f_x \\ f_y \end{Bmatrix} = -\begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix}\begin{Bmatrix} x \\ y \end{Bmatrix} - \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix}\begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} - \begin{bmatrix} D_{xx} & D_{xy} \\ D_{yx} & D_{yy} \end{bmatrix}\begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix}$$



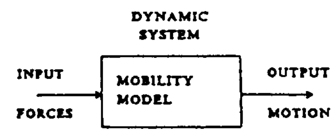
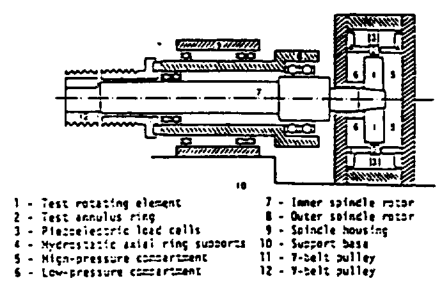
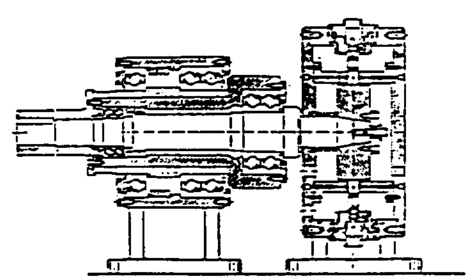


Figure 1. The Linear Anisotropic Bearing Model



Conceptual Sketch of Rotor Support Component Test Apparatus



Assembly Layout of Rotor Support Component Test Apparatus

Figure 2. CWRU Bearing Tester

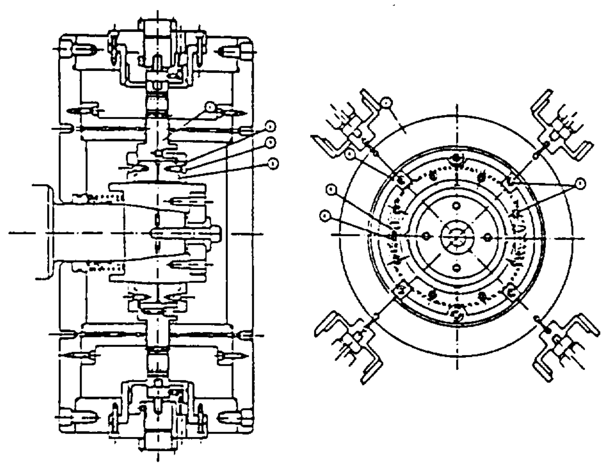


Figure 3. Arrangement for Bearing Test

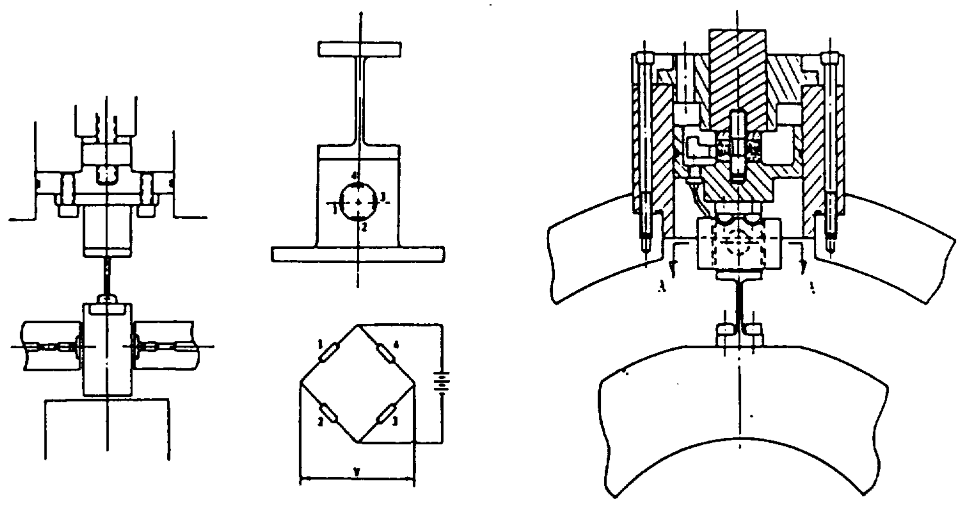
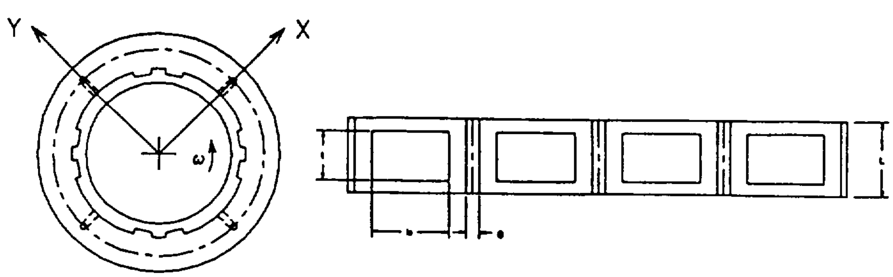


Figure 4. Dual Piezoelectric-Strain Gage Load Measurement System



LUBRICANT:

Motor Oil SAE 30
Viscosity @ 100°F:
14.212 micro-reyns
(97987 micro-Pa*s)
Operating Temperature 100-110°F

GEOMETRY:

Diameter 4.516 in. (114.71 mm)
Length 2.125 in. (53.98 mm)
Recess (l) 1.40 in. (35.56 mm)
Recess (b) 2.21 in. (56.13 mm)
Recess Depth 0.1875 in. (4.763 mm)
Groove (g) 0.375 in. (9.525 mm)
Radial Clearance 0.0083 in. (0.211 mm)

Figure 5. Test Bearing and Parameters

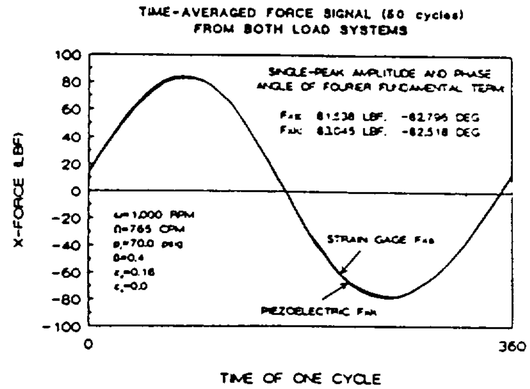


Figure 6. Total Time-Averaged Force Signal from Both Load Measuring Systems

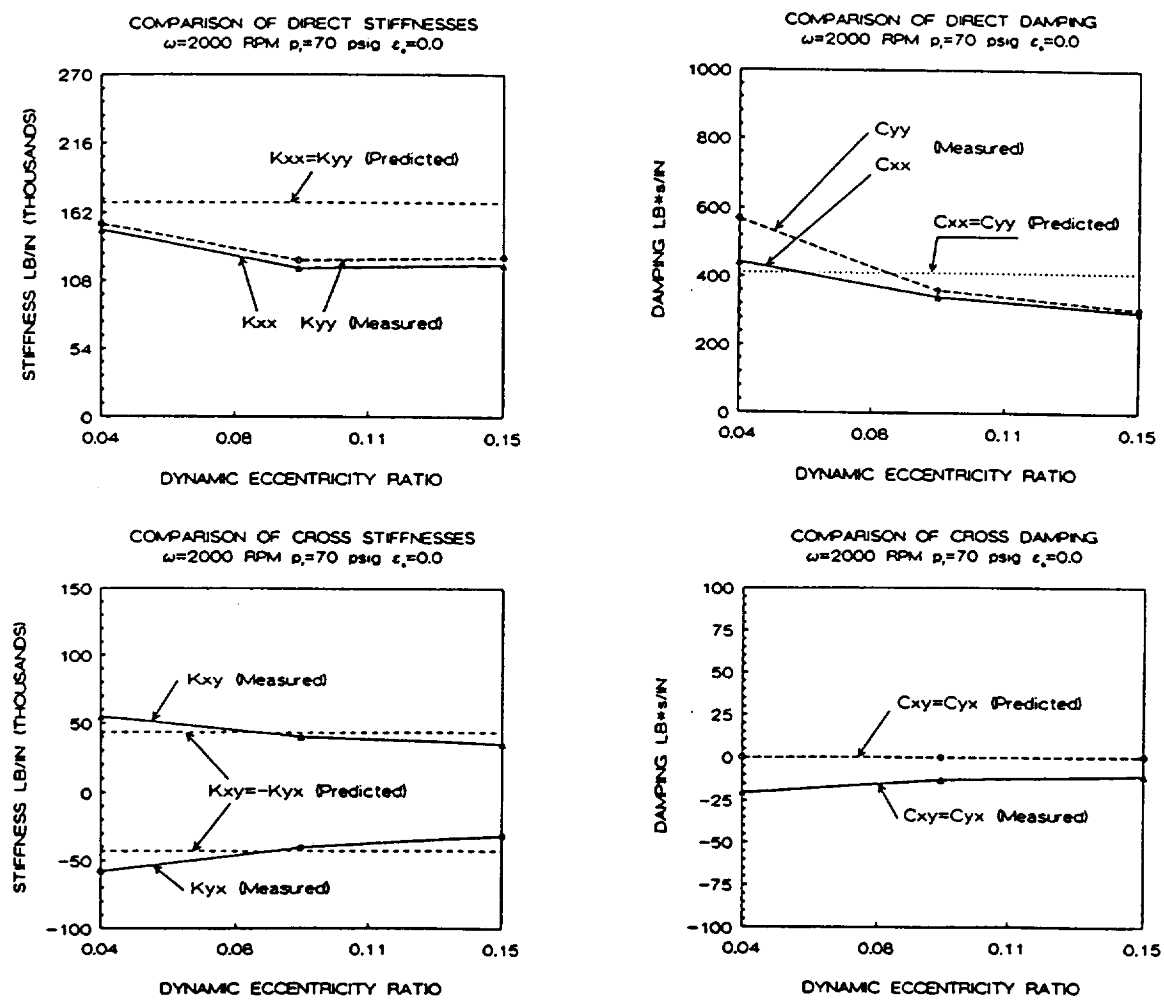


Figure 7. Comparisons of Test Results with Computations for Hydrostatic Oil Journal Bearing