A TEST RIG FOR AIR BEARINGS DYNAMIC CHARACTERIZATION

Bellabarba, E., Ruiz, R., Díaz, S., Rastelli, V.
Universidad Simón Bolívar
Departamento de Mecánica
Valle de Sartenejas, Baruta 1080-A, Miranda
VENEZUELA

ABSTRACT
This paper describes the design and operation of an experimental facility for measurement of equivalent stiffness and damping of air bearings. The rig uses two magnetic bearings to impose any given orbit to the journal, including displacement in two perpendicular directions on the rotation plane and tilting on the conical mode. Dynamic forces are measured directly on the test bearing housing. Data is gathered and processed using PC based data acquisition boards and software. Only the stiffness and damping coefficients of the fluid film are calculated as a function of the excitation frequency, being it synchronous or not. The present design allows testing air bearings up to 44 mm in external diameter and at frequencies up to 1 KHz. Preliminary testing was performed on this research that demonstrates the capability of the apparatus to measure the dynamic properties with ease and accuracy.

INTRODUCTION
Dynamic behavior of fluid film bearings has been recognized as a key factor in turbomachinery design for a long time. Both analytical and experimental approaches to determine dynamic coefficients have been tried. Tiwari et al. [1] presents an extensive review about experimental identification techniques. Most these studies verse on hydrodynamic bearings and seals, and address synchronous coefficient determination [2]. Some studies use time domain techniques [1], while most use frequency domain algorithms since they provide a more flexible and reliable measurement [3]. Air bearings pose additional complexity by the introduction of strong non-linearities, frequency dependency and, in many practical installations, the need to know pitch and yaw dynamic coefficients. Experimental identification of rotodynamic coefficients on air bearings has been addressed by quasi-static measurement of stiffness and hysteresis [4,5], and more recently by dynamic measurements using unbalance excitation [6]. The following presents a test rig which uses magnetic bearing actuators to identify coefficients in the frequency domain, allowing for asynchronous identification, and with the potential to measure the full 16-coefficient rotodynamic matrices.

TEST RIG DESCRIPTION
The test bearing is supported by three load cells that allow for centering and record all dynamic forces on the bearing. A shaft, with a built-in journal, is supported by a radial magnetic bearing on each end. Positioning precisions down to $10^{-4}$ mm can be attained at the magnetic bearing journals, which can be controlled independently. Common orbit shapes, white noise, multi-harmonic motion, one dimensional trajectories, cylindrical or conical modes, are just a few of the possible excitation functions that can be produced with this arrangement. Whirling of the rotor is independent of shaft rotating speed, thus allowing asynchronous excitation. Fig. 1. shows a schematic of the test rig. The magnetic suspension imposes an orbit or known motion, the load cells record reaction forces, and the data acquisition system computes the bearing transfer function. Since this transfer function relates only to the test bearing, the shaft inertia is irrelevant to the analysis, and only stiffness and damping coefficients are computed (equivalent film added mass could also be easily computed, but is neglected herein).

Each Magnetic Bearing has a load capacity of 76N up to 550Hz. Above this frequency, the amplifier slew limit produces a decrease down to 50% of full capacity at 1KHz. The factory supplied PID controller has a built-in antialias filter and the controller itself works at a scanning rate high enough to prevent leakage (above 100,000 Hz). The rotor is driven by a DC motor (max. 10,000 RPM). Magnetic levitation controller and amplifier are able to reproduce fed control signals up to 1 KHz. Swiveling tip bolts are used to support the load cells, minimizing the effect of misalignment and other production defects that could set an oblique force on the sensor. All parts of the aluminum plate, steel chassis and rotor, and bronze bearing are optimized as to have the lightest mass while assuring natural frequencies above the rig’s operating range. FEM analysis and experimental modal testing of the rig guarantee the elastic deformations are kept two orders of magnitude lower than the displacements being measured. This allows for the interpolation of the journal motion from the measurement of the rotor motions at the magnetic bearing locations (which are used as control feed-back signals).
PRELIMINARY TESTING AND RESULTS

To fully prove test rig functionality, a series of preliminary studies were performed on a fixed geometry, three lobe, load between pads air bearing, fabricated on bronze by wire EDM and similar to one previously studied in [6]. The test bearing is 30mm long, 28.526mm nominal diameter, 0.070mm radial clearance, 0.020mm preload, and has 6 between pads air bearing, fabricated on bronze by wire EDM studies were performed on a fixed geometry, three lobe, load between pads air bearing, fabricated on bronze by wire EDM. A PC based, data logging and processing system is implemented to control the eleven signal streams present in the test rig: four feed control signals and four position signals to and from magnetic bearings as well as three force measurement signals. The designed software displays all seven logged data channels in frequency domain and, upon request, calculates and displays the transfer functions.

CONCLUSIONS

A test rig capable of easily determining fluid film bearing rotordynamic coefficients was developed and successfully tested. The test rig can operate up to 1kHz and can produce almost any orbit desired by the user. Proper modal response was achieved for the all parts, assuring a precise analysis. The rig is capable of producing asynchronous excitations, allowing an extended test range. It can also produce angular motion, thus, with a proper load cell arrangement, angular coefficients could also be determined. This capability might be highly desirable when testing air bearings for small turbomachinery that might have large L/D ratios and have a shaft supported by a single bearing.

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REFERENCES


Figure 1. Test rig schematic

Figure 2. Test rig mounted on magnetic bearing system
1) Chassis, 2) Shaft, 3) Distribution Ring, 4) Test Bearing 5) Force sensor

Three piezoelectric load cells measure bearing loads, four variable reluctance probes measure shaft motion at the magnetic bearings, and a hall effect sensor records shaft speed. A PC based, data logging and processing system is implemented to control the eleven signal streams present in the test rig: four feed control signals and four position signals to and from magnetic bearings as well as three force measurement signals. The designed software displays all seven logged data channels in frequency domain and, upon request, calculates and displays the transfer functions.

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