THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
Three Park Avenue, New York, N.Y. 10016-5990

99-GT-180

A 35,000 RPM TEST RIG FOR MAGNETIC, HYBRID AND BACK-UP BEARINGS

BREAK

Erik. E. Swanson
James F. Walton II
Hooshang. Heshmat

Mohawk Innovative Technology, Inc.
437 New Karner Road, Albany, NY 12205; USA
miti@albany,net

ABSTRACT

Magnetic bearings have long offered the potential for significant turbomachinery system improvements due to their oil-free, non-contact, low loss nature and their ability to actively control shaft dynamic motion. However, end-users and many designers are hesitant to apply this technology. There are two basic stumbling blocks: active magnetic bearings (AMBs) have little overload capacity, and failure of any portion of the AMB system could result in catastrophic damage to the machine. To cope with both of these problems, a secondary back-up bearing must be included in the system. This paper describes a new full scale, high speed test rig which has the capability to test a variety of back-up bearings at speeds of up to 35,000 RPM, and bearing loads of up to 6.7 kN. Preliminary data for two novel back-up bearings are presented as a demonstration of the test rig's capabilities.

INTRODUCTION

Active magnetic bearings provide an attractive alternative to conventional bearings and squeeze film dampers in advanced high speed turbomachinery where high energy efficiency and reliability are crucial. They offer a number of advantages, including:

- Electronic manipulation of rotor system dynamics for enhanced performance and stability
- Operation at high temperatures
- Lubricant free operation
- Potentially smaller shaft orbital motion, and hence smaller tip clearances
- Lower power loss

Offsetting the many advantages of the AMB however, are poor control of shaft motion during transient external shock events, and potentially catastrophic failure modes. The poor control during shock transients arises out of trade-off between over design with a large, heavy system which is far larger than required to support the steady-state load, and dynamic characteristics which are dynamically soft in the operating speed range to minimize the magnitude of transmitted bearing forces. Experience has shown that AMBs following this traditional design approach may easily be overloaded with a relatively light impact. While fault tolerant approaches are addressing the reliability issues, there are still valid concerns about potential failure modes. To address both of the areas of concern, backup or auxiliary bearings are essential.

It is evident that the backup bearing is a key determining factor in assessing total bearing system reliability, durability and size. The backup bearing must be capable of sharing loads and, providing full load support for continuous operation in the event of an AMB failure, and performing these functions reliably over the life of the machine with minimal maintenance. For low speed applications, many designs, including prelubricated bushings and rolling element bearings, are adequate. In higher speed applications, these designs may also be adequately in a short-term protective role,

Presented at the International Gas Turbine & Aeroengine Congress & Exhibition
Indianapolis, Indiana — June 7–June 10, 1999

since it is possible to shut-down the machine for bearing replacement following an overload or failure. However, for aerospace propulsion applications, lightweight auxiliary bearings capable of operation at high shaft speeds (10,000 to 50,000+ rpm), in the presence of significant shock loads (tens of G's or more), which are able to run for substantial periods of time in the event of total AMB failure, are required. Without this level of performance, it is unlikely a magnetie bearing system would be considered flight worthy.

Historically, rolling element bearings with clearances between the shaft and the bearing inner race were the primary choice for backup bearings due to their simplicity and familiarity of ball bearing designers. In most applications however, these loose clearance backup bearings do not have adequate life nor do they permit continued operation due to damage sustained during the transient events. This damage may be the result of insufficient bearing lubrication and/or excessively high bearing speed, as well as dynamic loads which can be far greater than the failure load for the bearing. There is also potential for damage from ball skidding and cage dynamics due to the high acceleration as the bearing is spun up to speed. Galling and non-uniform wear also occurs at the shaft/bearing race interface during operation, increasing the bearing internal clearance and producing irregular journal and bearing surfaces. In addition, impacts and rebounds can occur within the bearing clearance, which continually change the rotor support system stiffness, excite system natural frequencies and may induce backward whirl. Life of these backup bearings is inevitably very limited.

Although the back-up bearing issue is one of the major stumbling blocks to wider application of magnetic bearings, very little data from full scale machines for AMB overload or failure transients has been published. One of the few sources is a series of papers which describe the initial drop transients on ball bearings and bushing bearings for an approximately 200 kg rotor at up to 8000 RPM [1-4]. Other work which is available includes [5, 6]. There are also a variety of analytical models for the magnetic bearing drop transient such as [1,3, 7-12]. However, due to the scarcity of experimental data, most of these simulations have not been well validated. Thus, although there is a need for a design tool which will reliably predict the nature of transient shaft motion during a magnetic bearing failure or overload, there is not really a good tool available. Without such a tool, it is difficult for the designer to confidently develop a back-up bearing without resorting to expensive and potentially hazardous prototype testing.

With the eventual goal of the development of an adequate design tool, a test rig capable of exploring the rotor system response to a magnetic bearing failure transient with speeds, loads and shaft dynamics typical of a small gas turbine was developed. This test rig was also designed to be a safe test bed for development of advanced back-up bearing concepts which address the problems inherent in current bearing designs. This paper documents the test rig which was developed. As a demonstration of the test rig capabilities, preliminary test results for two advanced concept backup bearings will also be presented.

TEST RIG

The high speed test rig shown in Figs. 1 and 2 includes a high-speed motor drive, a ball bearing support, a magnetic bearing, a test back-up bearing, and a loading system. For simplicity, only the test end of the rotor is supported by the magnetic bearing/back-up bearing combination. A loader system to apply controlled steady loads is located just outboard of the test bearing. These systems will be described below. The drive end of the test rig is supported by a 20 mm bore, grease lubricated, deep-groove ball bearing installed in a damped, compliant mount to increase the shaft system damping. The test rig is driven from the ball bearing end with a 37 kW induction motor through a high speed flexible coupling. The motor speed is controlled with a variable frequency drive which allows bi-directional operation from 0 to 36,000 RPM, and has dynamic breaking capability. The motor bearings are oil-jet lubricated. A disk-pack type coupling with low angular stiffness is used to minimize the influence of the drive system on the test rig response. The rig is built-up on a heavy tee-slot base which allows considerable flexibility in mounting test components.

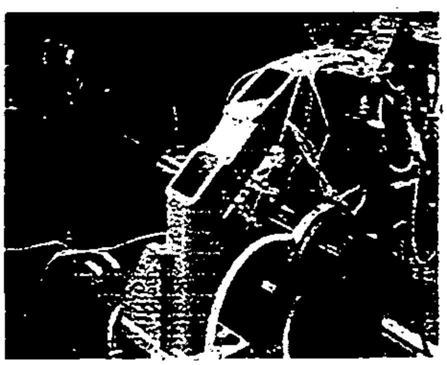

Figure 1 - Test Rig Photograph

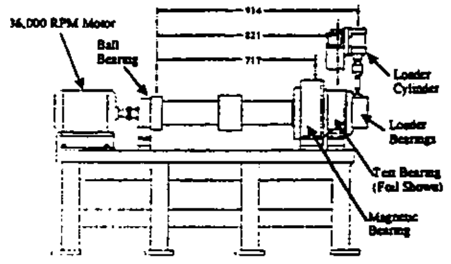

Figure 2 - Test Rig Side View

SHAFT SYSTEM

The test rig shaft, made of Inconel 718, is 985 mm long with a mass distribution designed to mimic the dynamics of a helicopter class gas turbine engine. The shaft has a mass of approximately 62 kg. Figures 3 and 4 present the rotor model and undamped critical speed map for the nominal compliantly supported ball bearing. The majority of the testing planned is above both rigid modes, but below the first bending mode, as would be typical of most magnetically supported machinery. The model presented here does not include the loader system dynamics. The shaft dead load at the test bearing location is approximately 400 N.

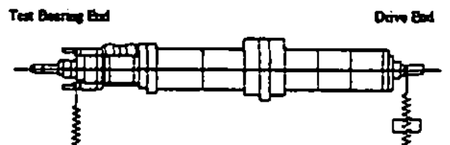

Figure 3 - Rotor Model

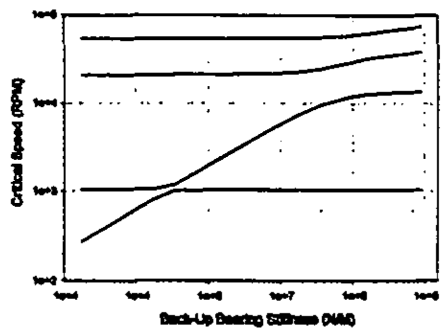

Figure 4 - Critical Speed Map for Nominal Support Bearing and Support Stiffness

MAGNETIC BEARING

The 4-pole homopolar magnetic bearing, located as shown in Fig. 2, was selected to minimize eddy-current losses during high speed operation. The bearing is a fully electromagnet design, with a separate coil to provide the bias flux. This approach was selected over a permanent magnet design to allow for separate evaluation of the system response to a loss of bias flux as well as response to loss of control flux. To reduce manufacturing cost, the bearing uses conventional silicon-iron laminations in both the stator and rotor. Table 1 presents the parameters for this bearing.

TABLE 1 - Magnetic Bearing Parameters

Load Capacity:	980 N in normal operation 1700 N at saturation
Shaft Diameter:	121 mm
Axial Length:	2x 21.8 mm
Control Current:	8 amps nominal/16 amps for max load
Control Windings:	50 Turns/pole
Bias Current:	8 amps nominal/16 amps for max load
Bias Winding:	200 Turns
Bias Design Flux:	6 kGauss nominal/10 kGauss for max load
Laminates:	0.178 mm Silicon-Iron
Pole Area:	1600 mm^2
Air Gap:	0.64 mm

The nominal currents indicated in Table 1 correspond to the nominal currents, as well as currents required to drive the bearing into saturation to obtain the maximum indicated load. The bearing is controlled by a PC based digital control system running a modified proportional-integral-derivative (PID) algorithm. This system includes a graphical user interface to allow the user to easily change the control parameters during operation. Two eddy-current sensors located inboard of the magnetic bearing are used for feedback. A total of five pulse width modulated (PWM) amplifiers are used to drive the bearing coils. One amplifier is used to provide the steady DC bias flux through the bias winding, two are used for the vertical control to drive the top and bottom control windings, two are used for horizontal control to drive the left and right control windings. This arrangement provides considerable flexibility in failure mode testing. The magnetic bearing stator is mounted on three load cells to directly measure the load carried by the bearing.

LOADING SYSTEM

To allow for characterization of the magnetic bearing and back-up bearings under steady-loads in excess of the shaft dead load, a hydraulic loading system can be attached to the shaft outboard of the test bearing as shown in Fig. 2. This system also gives the test rig the capability to evaluate back-up bearing response to simulated maneuver loads which overload the magnetic bearing and require that the back-up bearing support a portion of the load. The loader system consists of a duplex pair, angular contact bearing attached to the shaft outboard of the test bearing system, an oil-jet lubrication system, oil seals, load cells and hydraulic cylinders to apply load. This system is operated with a hand pump, and can apply unidirectional loads of over 6700 N to the shaft in the vertically and horizontally directions. The load cells used are piezo-electric force links attached to charge amplifiers operating in a long time constant mode. The system response has been experimentally verified as accurately reading a steady load to within five percent for slightly over 10 minutes, which was deemed sufficient for the goals of this test rig. An overtravel system in the load path allows for quick re-zeroing of the system during rig operation. Additional calibration to determine the scaling factors for equivalent loads at the test bearing and magnetic bearing were also performed.

TEST BEARING MOUNTING

The test bearings are mounted to the tee-slot baseplate outboard of the magnetic bearing as shown in Fig. 1. A 101 mm diameter, 100 mm long, tool-steel sleeve is mounted on the shaft as a running surface for the back-up bearings. This arrangement provides considerable flexibility for evaluating different back-up bearing configurations. The back-up bearing housings are essentially self contained, with provisions for displacement sensors, load sensors, cooling air, etc. In the current test rig configuration, the non-load carrying outer housing for the loader system also mounts to the test bearing housing. The results presented below illustrate the flexibility which has been built into the test rig. With only minor changes to the base rig, both a rolling element back-up bearing and a hydrodynamic air back-up bearing have been evaluated with this test rig.

SAFETY/PROTECTIVE FEATURES

A number of protective/safety features are designed into the test rig. Due to the high test speeds and the relatively heavy test rotor, the test rig is enclosed in a large double-layer steel enclosure. A heavy mid-shaft stator housing was also added to provide an additional level of restraint in the event of catastrophic failure. The drive is configured to provide dynamic braking during shut-down. The loader system is provided with a quick load release to dump load in the event of a problem. Since the loader system loads upward, operation of this release shifts the load from a possibly damaged bearing upper surface to an undamaged lower surface. Thermocouples are also provided to monitor key system temperatures.

DATA ACQUISITION AND CONTROL

A twenty-four channel, 16-bit, high-speed digital data acquisition system which samples all channels simultaneously is used in parallel with a fourteen channel analog data recorder to store test results. Each channel of the digital data acquisition system is currently configured for a 3 kHz data bandwidth, with a 12 kHz sample rate. A picture of this system is shown in Fig 5. The system user interface is a custom interface to MATLAB. A user interface developed in MATLAB allows for real-time as well as post-test data analysis. The data acquisition system can be readily expanded to 48 channels, and is capable of sampling all channels simultaneously at up to 100 kHz. The analog data recorder is used as a back-up to the digital data acquisition system, as well as to monitor rig performance between test events.

Depending on the goals of the test, a variety of data are recorded for evaluation. In the case of the back-up bearing testing which will be presented below, the measurements include the vertical and horizontal shaft displacements as recorded by the magnetic bearing control sensors, a once-per revolution signal as an angular position reference, an analog speed signal, magnetic bearing currents, signals from force transducers are located under the test bearing, and magnetic bearing load cells are recorded. For the foil bearing tests, a second set of displacement sensors adjacent to the foil bearing are also recorded. These measurements are sufficient to characterize the bearing loading performances, as well as the transient shaft motion for a drop transient.

The rig cooling air, buffer seal air, speed and the magnetic bearing disable controls are located adjacent to the data acquisition system as shown in the figure.

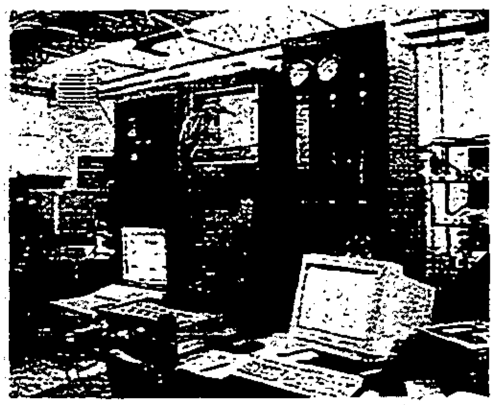

Figure 5 - Rig Control/Data Acquisition

INITIAL RIG/MAGNETIC BEARING CHECKOUT

Following magnetic bearing levitation and tuning, the first major milestone was to operate the test rig to 30,000 RPM with the magnetic bearing active. Figure 6 presents a peak-hold plot of the vertical displacement during a coast down from 30,000 RPM on the magnetic bearing. This figure shows well controlled motion, with adequately damped critical speeds around 50 and 70 Hz.

Figure 6 - Peak-Hold Frequency Response, Magnetic Bearing 30,000 RPM Coastdown

ZCAB Results

One of the first bearings which was evaluated with this test rig was a prototype 100 mm diameter, 8 Roller, Zero Clearance Auxiliary Bearing (ZCAB). This bearing has been described at length in [13, 14]. A conceptual schematic of the ZCAB is shown in Fig. 7. In essence, the ZCAB consists of a radial array of rollers positioned around the shaft such that under normal operation, there is a clearance space between the rollers and shaft as with a conventional rolling element back-up bearing. In the case of a magnetic bearing failure, the shaft drops onto several rollers which then move inward to eliminate the clearance space until all of the rollers contact the shaft. Damping and compliance are provided through the ZCAB mount. The elimination of the clearance space, lower component inertias, as well as the discrete roller contact will tend to eliminate the instabilities associated with a conventional clearance type rolling element back-up bearing. The design of this bearing also addresses many of the other concerns, such as ball skidding, cage instability, high rotation speed, etc. associated with a conventional rolling element back-up bearing. In the case of momentary magnetic bearing overload, the ZCAB can provide load sharing in either the open position, or a short term closure. Variations of the ZCAB which provide thrust load capability have also been developed.

Preliminary tests with the prototype of this novel bearing concept in a small demonstrator test rig were included in [13] and [14]. The loads and speeds available in this test rig were not in the range of the target small gas turbine applications. Therefore, during the initial debugging phases of the new test stand, further tests for loads more typical of a small gas turbine were conducted. Figure 8 presents a slow mag bearing turn-off transient during operation at 19,000 RPM. This transient corresponds to simulated failure of all PWM amplifier outputs. Despite being significantly overloaded at these operating conditions, a well controlled shaft coast down was observed.

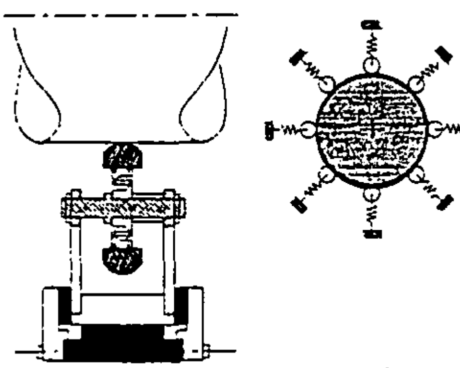

Figure 7 - Zero Clearance Auxiliary Bearing, Concept

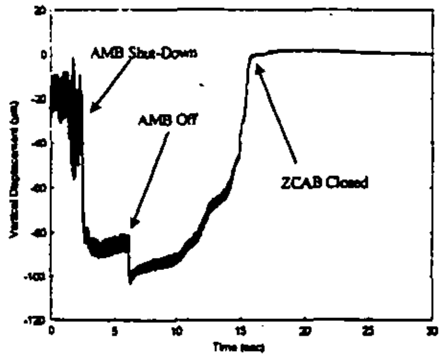

Figure 8 - ZCAB Transient at 19,000 RPM

Foil Bearing Results

A different approach to solving the back-up bearing problem associated with magnetic bearings is to develop a hybrid bearing system which augments the magnetic bearing with a self acting hydrodynamic bearing to provide a back-up as well as increase the total system load capacity through continuous load sharing. The compliant foil bearing is a natural candidate for this application. This bearing combines a low friction, air lubricated, hydrodynamic component with a structural component to increase the bearing damping and provide compliance to accommodate misalignment as well as thermal and centrifugal deformations as shown in Fig. 9. These bearings have been developed to a point where load capacity and shock tolerance capabilities are sufficient for the small gas turbine applications which are the focus of this test rig [15]. The only real drawback of these bearings are very low load capacity at low shaft speeds. However, as has been discussed in [16], the combination of foil and magnetic bearings reduces the impact of this drawback, thus synergistically combining the strengths of the two bearings.

For this hybrid bearing to be effective, the foil bearing must not only be able to share load under normal operation, but also support the entire shaft should the magnetic bearing fail or be overloaded. In addition, the system must exhibit a controlled response to magnetic bearing recovery. Therefore, it was crucial to demonstrate these capabilities as part of the development of this hybrid bearing concept. Figure 10 presents the measured shaft motion during a simulated magnetic bearing failure/recovery transient at 25,000 RPM with a 100 mm diameter, 75 mm long foil bearing acting as a back-up bearing. This bearing is similar to the smaller bearings described in reference [15]. For this test, the shaft dead load was supported almost entirely by the magnetic bearing prior to failure. Intuitively, this should be the most severe transient for the hybrid bearing system since the shaft must fall through the clearance space before the foil bearing develops useful load capacity. The transient shown in Fig. 10 shows a well controlled transition to foil bearing alone operation. Since the system must also transition back to the magnetic bearing for fault recovery, the magnetic bearing was re-enabled a few seconds after the simulated failure while the shaft was still operating at 25,000 RPM. To simulate a worst-case scenario, the digital controller was tuned to provide low bearing damping. In addition, no "soft-start" system was provided. The result, as seen in Fig. 8, is that the recovery transient is far more dramatic than the original failure transient.

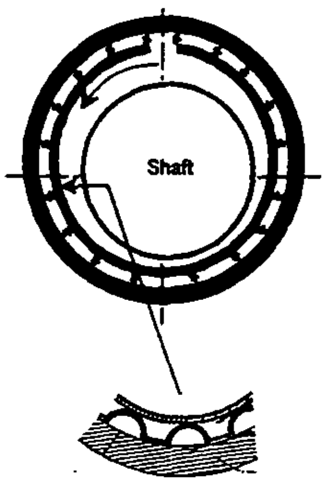

Figure 9 - Foil Bearing Concept

Figure 10 - 25,000 RPM Drop and Recovery on Foil Bearing

SUMMARY/COMMENTS

To fill the need for development of high speed back-up bearings for use in magnetic bearing applications, a test rig sized to simulate small gas turbines was constructed. This test rig is designed for operation at up to 36,000 RPM and can provide up to 6700 N of steady load to a candidate back-up bearing or hybrid bearing. It also has the capability to apply transient loads both through the magnetic bearing, as well as rapid changes in the supply pressure to the hydraulic loading system.

As demonstrations of the test rig capabilities, preliminary results for the "Zero-Clearance Auxiliary Bearing" (ZCAB) and a hybrid foil bearing configuration have been presented. For both bearings, successful operation through a loss of magnetic bearing support was achieved.

Future papers presenting the details of the ZCAB tests and the hybrid foil/magnetic bearing, such as load capacity measurements, failure mode behavior, hybrid operation and long duration tests are planned.

REFERENCES

1. Kirk, R.G., Raju, K.V.S., Swanson, E.E., "Evaluation of AMB Turbomachinery Auxiliary Bearings." *Proc Fifth Colloquium on Turbomachinery*, Turbo and Power Machinery Research Center, Seoul National University, Seoul. Korea, May 6-11, 1996.

2. Kirk, R.G., Swanson, E.E., Kavarana, F.H., Wang, X., "Rotor Drop Test Stand for AMB Rotating Machinery, Part I: Description of Test Stand and Initial Results." *Proc. 4th International Symposium on Magnetic Bearings*, Aug 1994, Zurich, pp. 207-212.

3. Swanson, E.E., Raju, K.V.S., Kirk, R.G.,"Test Results and Numerical Simulation of AMB Rotor Drop" *Proc. 6th International Conference on Vibrations in Rotating Mach.*, IMechE, 1996, pp. 119-131.

4. Swanson, E.E., Kirk, R.G., "AMB Rotor Drop Initial Transient on Ball and Solid Bearings." *Proc. MAG '95*,1995, pp. 227-235.

5. Schmied, J., Pradetto, J.C., "Behavior of a One-Ton Rotor Being Dropped into Auxiliary Bearings," *Proc. 3rd International Symposium on Magnetic Bearings*, Alexandria, VA, 1992,pp. 145-156.

6. Alves, P. and Lee, C., "Dynamic Test of Auxiliary Landing System for Magnetic Bearings (Drop Test)," *Proc. REVOLVE '92*, April, 1992.

7. Feeny, B.F., "Stability of Cylindrical and Conical Motions of Rigid Rotor in Retainer Bearings," *Proc. 4th International Symposium on Magnetic Bearings*, Zurich, 1994, pp. 219-224.

8. Fumagalli, M., Fenny, B., and Schweitzer, G., "Dynamics of Rigid Rotor in Retainer Bearings," *Proc. 3rd International Symposium on Magnetic Bearings*, Alexandria, VA, July 1992, pp. 157-166.

9. Fumagalli, M., Varadi, P., and Schweitzer, G., 1994. "Impact Dynamics of High Speed Rotors in Retainer Bearings and Measurement Concepts,"

Proc. 4th International Symposium on Magnetic Bearings, Aug 1994, Zurich, pp. 329-244.

10. Gelin, A., Pugnet, J.M., and Hagopian, J.D., "Dynamic Behavior of Flexible Rotors with Active Magnetic Bearing on Safety Auxiliary Bearings," *Proc. 3rd International Symposium on Magnetic Bearings*, Lyon, France, 1990, pp 503-508.

11. Ishii, T., and Kirk, R.G. "Transient Response Technique Applied to Active Magnetic Bearing Machinery During Rotor Drop," ASME Conference Proceedings, DE-Vol. 35, 1991, pp. 191-200.

12. Kirk, R.G., and Ishii, T., "Transient Rotor Drop Analysis of Rotors Following Magnetic Bearing Power Outage," *Proc. MAG '93*, 1993, pp. 53-61.

13. Chen, H.M., Walton, J., and Heshmat, H., "Zero Clearance Auxiliary bearings for Magnetic Bearing System," ASME paper 97-GT-112, 1997.

14. Chen, H.M., Walton, J., and Heshmat, H., "Test of a Zero Clearance Auxiliary Bearing," *Proc MAG '97*, 1997, pp. 111- 119.

15. Heshmat, H., "Advancements in the Performance of Aerodynamic Foil Journal Bearings: High Speed and Load Capability," *Journal af Tribology*, 116, 1994, pp. 287-295.

16. Heshmat, H., Chen, H.M., Walton, J.F., "On the Performance of Hybrid Foil-Magnetic Bearings," ASME paper 98-GT-376, 1998.