A RELIABLE MAGNETIC BEARING SYSTEM FOR TURBOMACHINERY

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ABSTRACT

A reliable magnetic bearing system for turbomachinery was fabricated and tested. The bearing system features high performance, high temperature actuators with integrated sensors, and an advanced rotor control system. Fault tolerance was achieved by using redundant sensors and actuator coils. The control system, which uses a triple modular redundant architecture with four processors, was integrated with the sensor electronics, amplifiers, and power supplies. The control software takes advantage of the redundant sensors and coils to provide continuous operation even in the event of a sensor, coil, or amplifier failure. The control system also includes an automatic inertial balancing system, which minimizes forces transmitted to the stationary structure by shaft unbalance. An advanced monitoring system was developed, which interfaces to the control system. It monitors system performance such as rotation rate, rotor position, and unbalance. The monitoring system also analyzes the unbalance, calculating its location and magnitude. The bearing was tested at temperatures up to 800 °F (427 °C).

INTRODUCTION

Magnetic bearings are under development for gas turbine engines for industrial, marine, and aircraft applications. Lack of friction and higher permissible operating temperatures minimize the cooling requirements. The result is reduced system weight, higher thermodynamic efficiencies, higher reliability, and reduced maintenance (Penfield, 1995). Synchrony Inc. and Allison Engine Co. are together developing magnetic bearing systems for gas turbine engines and other applications. Also participating in this effort is the University of Virginia, which has developed a fault-tolerant, multi-tasking digital controller suitable for critical applications (Fedigan, 1993).

This paper discusses some of the recent results in the development and testing of a prototype bearing system for a gas turbine engine of around 4000 HP. As such, it was designed for reliability and high temperature operation. The control system was designed for fault-tolerant operation of the five axis bearing system (two radial bearings and one axial bearing).

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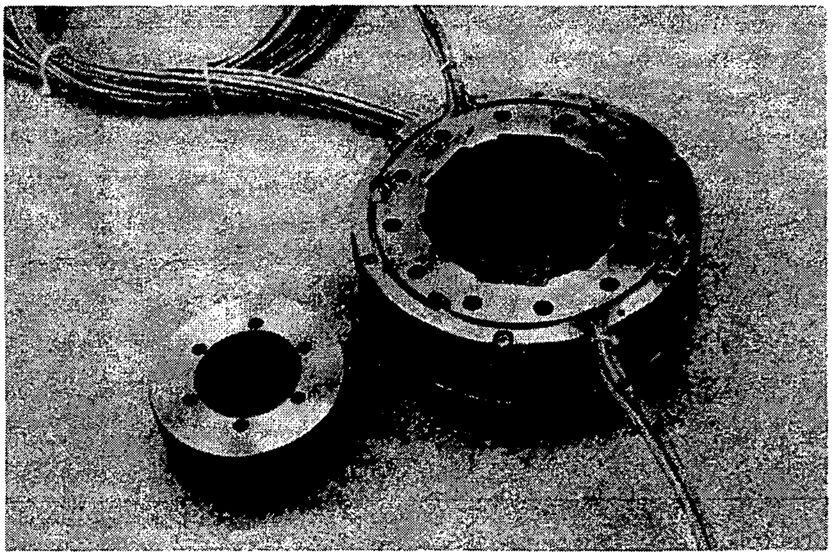

Figure 1. Photograph of fault-tolerant radial bearing and rotor.

ACTUATORS

Reliability is a primary design consideration for magnetic bearings for critical applications. A radial magnetic bearing can operate with just two sensors and three or four coils, but if one sensor, coil or amplifier fails, the entire bearing may fail because the actuator cannot effect a force in an arbitrary direction. Reliability dictates the use of multiple redundant sensors and coils. We have used six coils and six sensors for our radial bearings.

The radial bearing and rotor are shown in Figure 1. In order to reduce the weight, an optimized E-shaped pole design was selected. This design splits the total flux of each magnet into two paths. As a result, the rotor laminations and the outer region of the magnet laminations only carry half the magnetic flux for each magnet, and can be half the width that might otherwise be required. The same design was used for both low temperature and high temperature versions of the bearing, the only difference being that high temperature materials and magnet wire were used in the high temperature version.

The design also incorporated airflow channels next to the coils to prevent excessive heat buildup in the coil area. This was expected to be critical for high temperature operation, since the increase in electrical resistance at high temperature would dramatically increase the power required to operate the coils. (At 1100 °F the coil resistance increases by a factor of 3.4). Although the "cooling air" to flow through these passages would itself be hot, this air would remove energy generated by resistive heating of the coils, thereby lowering the peak temperature experienced by the coils.

High temperature compatibility, magnetic properties, strength, and thermal expansion coefficients determined the materials selected for the high temperature bearings. Solid ceramic-coated nickel-clad copper wire was selected for the coil wire. This wire is rated for continuous operation to 1000 °F, and its insulation is rated at 200 volts DC. Stranded nickel-coated copper wire insulated with a thick composite of mica and fiberglass was chosen for the lead wire. It is rated up to 1000 °F. It was found experimentally that the lead wire insulation could stiffen and become brittle after exposure to high temperature, so a stainless steel overbraid was added to improve its strength. Hiperco (iron-cobalt) was selected for the lamination material based on its high temperature strength and magnetic properties. The laminations were 0.005" thick for optimum performance over a wide frequency range. Various specialty steels (primarily Inconel 625 and 416 stainless steel) were used for the stator and rotor housings and fittings. The low temperature radial bearing housing was constructed of brass.

The high temperature bearings were heat-tested in an oven at 1100 °F. Figure 2 shows the coil resistance as a function of temperature. This information was also used to calculate the coil temperature during bearing operation, based on the coil resistance.

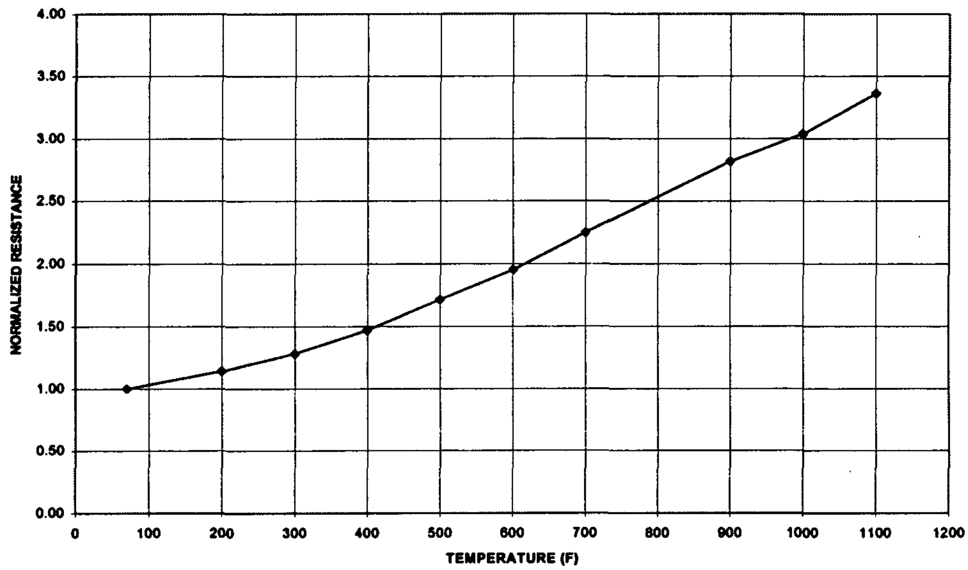

Figure 2. Coil resistance versus temperature.

SENSORS

Sensors are integrated into our radial bearings, using a configuration that does not increase the size of the bearing. Two of the six sensors can be seen on the inside diameter of the stator in Figure 1. This close "co-location" of sensing and actuation avoids some of the control problems that can occur with flexible shafts. Another concern for high temperature, high speed applications is growth due to centrifugal forces and/or thermal expansion. These effects can change the rotor diameter and lead to position errors under normal operating conditions. For this reason, we have developed a differential sensing method which uses two sensors per axis, which accurately preserves the zero position calibration independent of temperature. By itself, this requires four sensors (two opposed sensors on two axes). For increased reliability, however, redundant sensors are required, and so we have also used a total of six sensors per radial bearing. An added benefit is the elimination of harmonics at integer multiples of 2 and 3 of the rotational frequency (x2,3,4,6,8,9,10,12,14,15...) due to sensing surface imperfections.

Capacitance sensors were selected because they are relatively easy to construct of high temperature materials, and their performance is relatively independent of temperature. The commercially available high-temperature capacitance sensors we tested were not suitably linear. Instead, we developed proprietary high-temperature capacitance sensors and special detector circuitry that exhibited good linearity (Pat. Pending). The voltage output is linear within 1% for displacements of ±12 mils between a pair of sensors, which is the full range of radial motion allowed by the radial bearing.

The capacitance sensors were successfully operated from 70 °F to 1050 °F. The radial bearing was placed in an oven with the sensors connected, and the rotor position was continuously monitored. Any two sensor axes can be used to resolve the x and y position of the rotor. Since there are three sensor axes, the sensor axes can be paired so that three different (x,y) coordinates are calculated. When the sensors are in calibration, the three (x,y) coordinate pairs agree. After calibration of the sensors at 70 °F, the (x,y) coordinates were compared as a function of temperature up to 1050 °F. The maximum position disagreement over the temperature range was less than 1.0 mils, which is acceptable. Because the capacitance between the sensor and the shaft is almost independent of temperature, we believe the 1-mil discrepancy is due to thermal distortion of either the sensor housing or rotor.

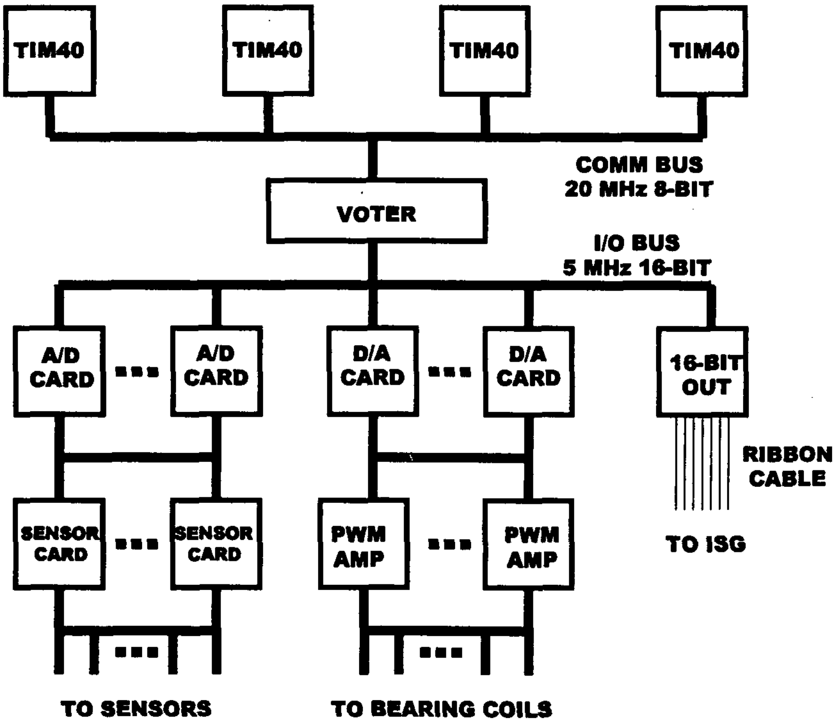

Figure 3. Block Diagram of Digital Control System.

CONTROLLER

The digital processing engine of the controller, which includes redundant digital signal processors (DSPs), was developed by the University of Virginia (Fedigan, 1993). A block diagram of the integrated digital controller is shown in Figure 3. The signal processing is performed by four TI TMS320C40. Each DSP is located on its own TIM40 module, and each TIM40 includes dedicated EEPROM and RAM. The C40 DSP has 6 asynchronous, 8-bit wide ports for communicating to other processors or other peripheral devices.

The digital controller is a triple modular redundant (TMR) system. The processors are synchronized such that each processor begins its calculation "simultaneously." When the controller is configured so that the DSPs are in the fault-tolerant mode of operation, each processor receives identical data from a "voter module," which is the link between the processor communications bus and the I/O bus. Each processor is executing an identical program so that at the end of the calculation cycle, each should output an identical result. The voter module compares the outputs from three of these processors. The fourth processor is in hot standby. If the calculation results are identical for the three processors, the result is passed along to the I/O bus. However, if one of the processors differs from the others, this processor is deemed "failed," and the fourth processor is swapped in to replace. Because of the fourth spare processor, the system continues to function normally through the failure of two processors.

The I/O bus allows communication between the voter and the I/O devices. For a 5-axis controller, two A/D cards, each comprising 8 16-bit analog input channels, and two D/A cards, each comprising 8 16-bit analog output channels, are included. The analog inputs to the A/D cards are position signals from the capacitance sensor cards and coil currents from current monitors. The analog outputs from the D/A cards go to the PWM converters and then to the amplifiers, which are connected to the coils in the bearings. Sufficient capacity is included for the redundant sensors and coils of the 5-axis system.

The fault-tolerant digital controller, sensor cards, and amplifiers for a complete 5-axis magnetic bearing system are integrated into a single enclosure. Synchrony has developed sensor cards, signal conditioning, power amplifiers, and software, to complete the magnetic bearing control system. Amplifiers were configured in a proprietary dual amplifier package for reduced weight and power supply requirements. Amplifiers, shown in Figure 4, plug into a proprietary amplifier backplane that includes current transducers and high-voltage power bus.

Fault tolerant 5-axis control requires approximately 14 capacitance sensor inputs and 14 power amplifier outputs, depending on the exact system configuration. The digital controller, I/O boards, capacitance sensor cards, amplifiers, current detectors, cooling fans, and power supplies are all integrated into an enclosure weighing 67 lb.

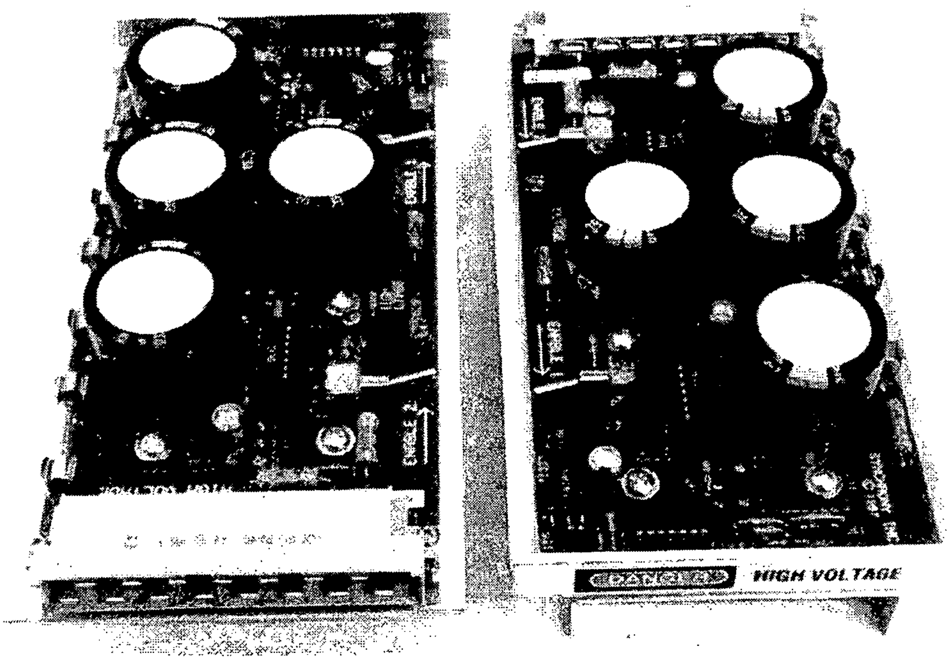

Figure 4. Photograph of Two Dual Amplifier Modules.

CONTROL ALGORITHM

The magnetic bearing control software was written in C language for the C40 processors. Several unique features have been incorporated in this software, including full 5-axis control (3 bearings), support of the fault tolerant features of the actuators, inertial balance control, and flux command operation of the actuators.

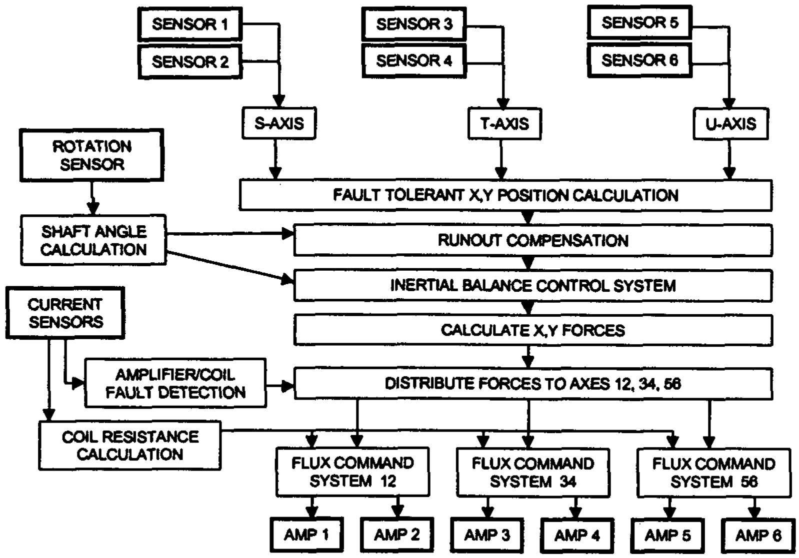

Figure 5. Control Algorithm for One Radial Bearing.

The 3 bearings are controlled simultaneously using similar algorithms. The general algorithm for fault-tolerant control of one radial bearing is seen in Figure 5. The six sensor values are converted into three axial positions (S, T, and U) with true center position accuracy. This conversion is done in hardware using the capacitance sensor cards, and sent to the analog input channels of the controller. The three positions are converted to a raw (x,y) position using the fault-tolerant scheme of Figure 6. This algorithm uses all three axial position inputs to calculate (x,y) unless one input goes out of range, in which case it uses only the two remaining inputs.

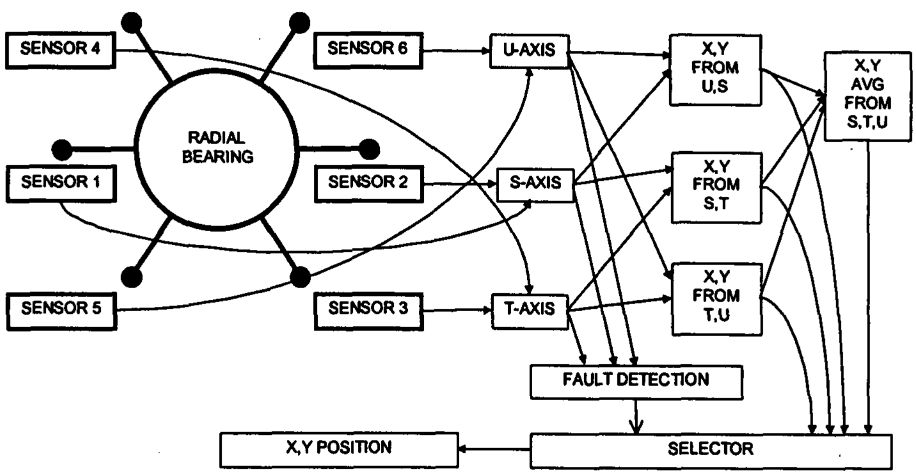

Figure 6. Fault-Tolerant Calculation of Radial Position.

Two types of open-loop unbalance compensation schemes were devised and tested:

- *Forced Balancing.* Minimize the synchronous vibration (orbit of the geometric center) of the shaft by injecting a synchronous, rotating bearing force to counteract the unbalance. The phase and magnitude of the synchronous force is adaptively adjusted until the objective is met.
- *Inertial Balancing.* Minimize the synchronous force by injecting a synchronous, rotating position correction term in the algorithm. The phase and magnitude of the correction term is adaptively adjusted until the objective is met.

Either type can greatly reduce the peak vibration experienced by the shaft as it crosses critical speeds (Beale, 1992; Knospe, 1993; Ku, 1993). *Forced Balancing* results in smaller shaft orbits at the expense of higher bearing forces transmitted to the static structure. In extreme cases, the high bearing force may saturate either the actuator (maximum force) or amplifier (maximum force slew rate). *Inertial Balancing* results in almost no synchronous force transmitted to the static structure, at the expense of allowing unrestrained orbit of the shaft about its inertial center. If the unbalance is extreme, the orbit may be large enough to cause contact in the auxiliary bearing. The choice between the two is therefore dictated by available clearance, force, and force slew rate. We have primarily used *Inertial Balancing.*

In *Inertial Balancing* there are two correction terms that are applied to the fault-corrected position. Consider a single radial bearing with shaft geometric center vector X, lumped mass M, and controller transfer function G. We assume the mass is rotating at angular frequency ω and that its center of mass is displaced from its geometric center by a displacement vector U. We also assume that the sensing surface of the shaft is displaced (non-concentric) relative to the center of the shaft by a displacement vector X_s, and the shaft is unbalanced by vector X_b. With Z as the control position, the equation of motion for this system is

$$M\ddot{X} + G(Z) = MU\omega^2 \tag{1}$$

$$Z = X_m + X_s + X_b \tag{2}$$

X_m is measured position of the shaft and X_b and X_s are the corrections. X_s is applied to compensate for non-concentricity of the sensing surface with the geometric center of the shaft. Note that without including this term, the non-concentricity of the bearing surface would cause a force to be produced by the magnetic bearing even if the shaft were centered (X=0). The geometric center is defined as the center of a reference surface of the shaft, such as the journal for the auxiliary bearing. We refer to X_s as the sensor runout correction. This is a geometric correction that would be calibrated once for a given shaft.

The second correction term to the measured position is the inertial balance correction X_b, which is dynamically changed by the controller. The inertial balance algorithm adaptively calculates the phase angle and magnitude of X_b to zero the synchronous portion of the sum (X_m+X_s+X_b). When this occurs, Eqs. (1) and (2) reduce to

$$X = -U \tag{3}$$

Eq. (3) shows that when the inertial balance term is applied, the shaft will orbit with a radius equal to the eccentricity of the center of mass and 180 degrees out of phase with the eccentricity. Said otherwise, the shaft will orbit about its inertial center. Under these conditions, there will be no synchronous force transmitted to the stationary structure by the bearing because the synchronous force has been eliminated. Please note that although the synchronous force applied by the bearing is eliminated, the shaft (geometric center) orbit will in general be non-zero.

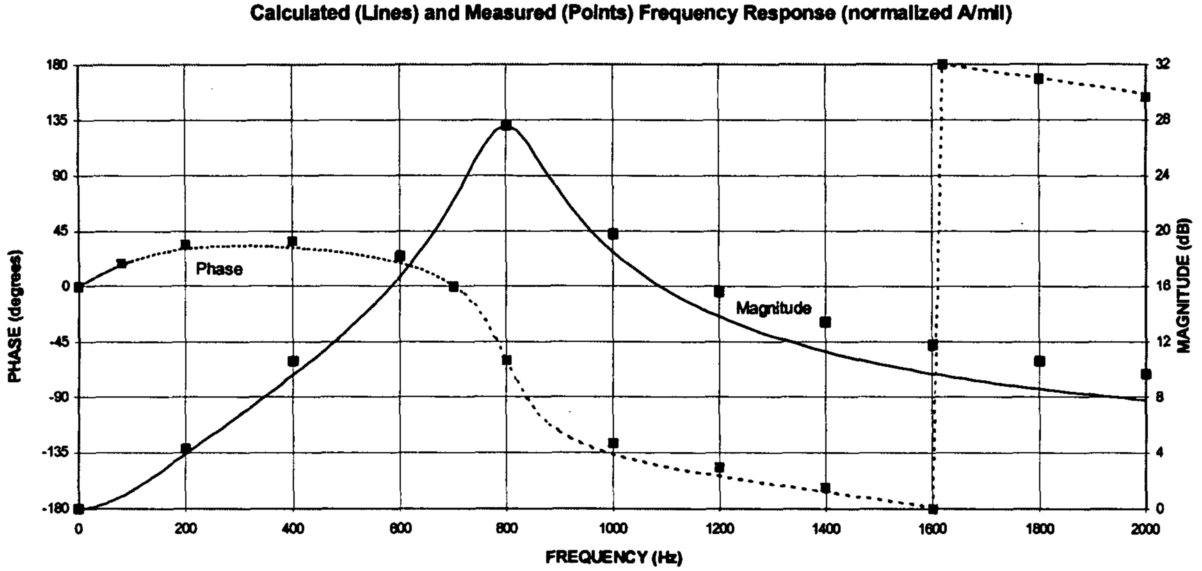

Figure 7. Measured and calculated frequency response of controller.

The forces to be applied to the shaft are calculated using a 4th order infinite impulse response (IIR) filter with separate integral term. The control parameters are determined by using proprietary software in which a state space model for the flexible rotating structure is coupled with a state space model for the magnetic bearing. The general solution procedure is to choose control parameters that provide sufficient damping of lower order modes and sufficiently low gain at higher frequencies so that higher order modes are not excited. These two criteria tend to be conflicting so that a compromise is typically achieved between stability of higher and lower order modes.

A typical frequency response spectrum of the controller with typical control parameters is shown in Figure 7. The experimental values are displayed as square points; the calculated values are displayed as lines. For these experiments, the gap was held constant using plastic shim material, and the actual applied coil current was measured in response to a perturbation of the position signal. A positive phase response is found up to 700 Hz for this particular set of control parameters, which means that movement in this frequency range is damped. It is easily possible to extend the positive phase response well beyond 700 Hz, but only at the expense of increased high-frequency gain, which can excite higher order modes in the system where the phase response is negative.

The x and y forces are split among the six coils of the bearing. This system is inherently fault-tolerant because if one coil or amplifier ceases to function, the integral term will reduce the force applied to the opposing coils and build up the force on adjacent coils, until the desired (centered) position is restored. Alternatively, a fault detection system has been developed which detects axis failure and immediately switches the applied force to the remaining coils.

In a typical magnetic bearing system, current amplifiers are used which convert the output voltage of the controller to a current with a predetermined amps-per-volt gain factor. The flux in the bearing is usually assumed to be proportional to the current. The problems with this technique are:

- Due to eddy currents in the magnetic materials, the actual flux will lag the amplifier current in phase and magnitude at higher frequencies.
- Because the flux is dependent on gap of the bearing for a given current, there is an undesirable "negative stiffness" effect that reduces the overall stiffness of the bearing for a given electronic gain.
- Due to saturation effects in the magnetic materials, the flux will be less than desired at higher field levels.
- The amplifier, in converting its control signal to amperes, has its own control loop with finite bandwidth and stability considerations.

In our system the coils are operated in "flux command mode." The flux command algorithm takes advantage of a particular feature of electromagnets, that is, the flux (and hence the force) remains constant during changes in the air gap, even though the inductance and current change, as long as no voltage is applied. A voltage is applied to the coil only as necessary to change the flux (and force), and to account for the resistive voltage drop associated with the coil current. Furthermore, the amount of voltage that must be applied to achieve a given change in force is independent of the inductance at any moment (which is a function of gap and magnetic properties of the materials). This is particularly helpful in high temperature bearings, in which the inductance of the coil may change significantly with temperature. The flux command method is also very fast since it applies whatever voltage is necessary to achieve the target force in one sample period (up to the maximum amplifier voltage). In contrast, a commercially available amplifier with integrated current controller took about 10 times as long to settle to the commanded current.

ROTORDYNAMIC MONITORING SYSTEM

In order to minimize the size and weight for aircraft applications, the control system does not have external displays. Typically, extra output channels may be configured to display, analyze, and archive position signals. A standard monitoring and communications for the controller was developed which runs on MS-Windows95 or NT. The monitoring program is written in Visual Basic. It provides a user interface to the control system for the purpose of monitoring bearing performance, calibration, control parameters, inertial balance, and unbalance analysis. Because the communication from the digital controller is via an RS232 port, the interface is sufficiently generalized so that the monitoring parameters can be exchanged between the controller and most other processing systems.

The primary functions monitored during operation of the magnetic bearing system are position, force, current, temperature, rotation speed, unbalance, and control system status. The temperature of a bearing is monitored by comparing the calculated coil resistance with a pre-calibrated temperature/resistance curve. The rotational speed and shaft unbalance are displayed in the Inertial Balance window of the monitoring program, which also shows the status of the inertial balance system. For example, the screen reproduced in Figure 8 shows a rotation speed of 62 Hz (or 3715 RPM). The inertial balance system is ON and is applying an offset of 0.38 mils on the first radial bearing and 0.73 mils on the second radial bearing. Based on this result, and knowing the mass of the rotating structure and the position of the center-of-mass relative to the position sensors, the inertial balance system calculates the magnitude, shaft angle, and position of the unbalance. In this case, an unbalance of approximately 1.5 ounce-inches was intentionally added to the system.

Two test rigs were used to demonstrate the performance of the magnetic bearings. One is located at Synchrony, and is capable of rotational speeds of about 6,000 rpm at room temperature. The other rig is located at Allison Engine Co. in Indianapolis, IN. This is a complete rotordynamic test facility, complete with shielding, instrumentation, heated air, and an integral turbine drive capable of speeds up to 12,000 rpm (derated at high temperatures). The magnetic bearings were designed for integration into either test rig. Stable operation was achieved to full speed at room temperature, and to derated speeds at high temperatures. The maximum operating temperature of the magnetic bearing compartment was non-uniform, ranging from 650 °F to 800 °F. Fault tolerance of the radial bearing was tested while the system was operating, by disabling sensors and amplifiers and observing and recording the response of the system. We found that the system was immune to failures of any single sensor, coil, amplifier, or processor.

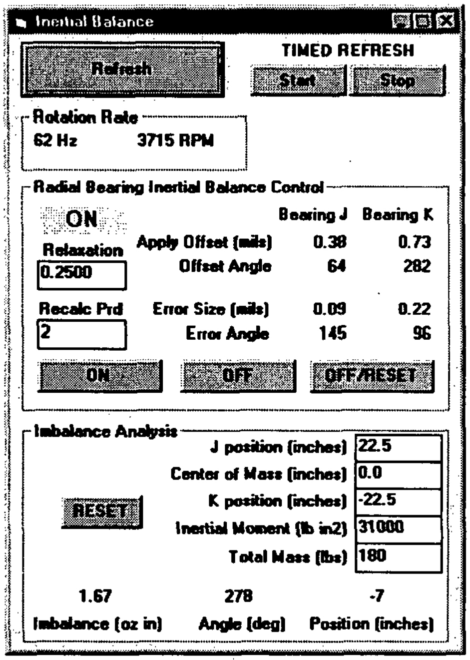

Figure 9. Inertial Balance Display.

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REFERENCES

Beale, S. et al. 1992. "Adaptive Forced Balancing for Magnetic Bearing Systems," *Proc. Third Internat. Symposium on Magnetic Bearings '92*, Technomic Publishing, Lancaster, PA.

Fedigan, S. J. and R. D. Williams. 1993. "An Operating System for a Magnetic Bearings Digital Controller," *Proceedings of MAG '93*, Technomic Publishing, Lancaster, PA.

Knospe, C. R. et al. 1993. "Adaptive On-Line Rotor Balancing Using Digital Control" *Proceedings of MAG '93*, Technomic Publishing, Lancaster, PA.

Ku, C.-P. R. and H. M. Chen. 1993. "Optimum Shaft Balancing at a Rotor Bending Critical Speed with Active Magnetic Bearings" *Proceedings of MAG '93*, Technomic Publishing, Lancaster, PA.

Penfield, S.R. et al. 1995. "How to Avoid the Pitfalls in Using Magnetic Bearings in Turbomachinery", *Proceedings of MAG '95*, Technomic Publishing, Lancaster, PA.