A Magnetic Bearing Actuator for Detection of Shaft Cracks in Rotating Machinery Supported in Conventional Bearings *

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***Abstract* – This paper demonstrates experimentally the capability of an AMB actuator to detect shaft cracks during operation of a rotor that would otherwise go undetected by conventional health monitoring methods. The AMB actuator is used in a mid-span location on a rotor supported in conventional rolling element bearings, although the technique is expected to be applicable to AMB-supported rotors.**

***Index Terms* – Shaft Crack, Magnetic Bearings, Health Monitoring**

I. INTRODUCTION

This paper presents results from a study looking at the novel application of Active Magnetic Bearings (AMBs) as *actuators* for applying known force inputs to a rotating shaft supported in conventional bearings in order to monitor and evaluate response signals resulting from the AMB force inputs. Similar to modal analysis and other NDE techniques which apply input signals to static structures in order to monitor responses to those inputs, this approach allows for the measurement of both input and output response in a rotating system for evaluation. However, unlike these techniques, *the procedure developed here allows for multiple forms of force input signals to be applied to a rotating structure*. This procedure will facilitate the development of new improved techniques for diagnosing subtle changes in machinery health and the development of accurate prognosis models.

Although it is expected that this approach can be used in rotors supported in AMBs, an important aspect of the technique developed here is that the AMB can be used as an actuator with the rotor system supported in conventional support bearings. Therefore, this approach has the potential to be used on any rotating machine that can be designed or retrofitted with a single AMB actuator.

The condition monitoring of rotating machinery has been actively practiced for several decades in the aerospace, power generation and petrochemical process industries (Pusey [1], Eisenmann [2], Mitchell [3]). Most of the diagnostic methods used involve vibration monitoring simply because machinery distress very often manifests itself in vibration or a change in vibration pattern. Aside from static modal analysis testing, these conventional techniques rely on monitoring vibrations which are the result of operational machine conditions and not a result of applying additional forces for evaluation.

In addition to the general class of machinery health monitoring, there is a significant body of literature involving the analysis and diagnostics of shaft cracks. Papadopoulos [4] discusses unambiguous coupling due to the presence of a shaft crack between longitudinal, torsional, and bending vibrations obtained with a harmonic sweeping excitation throughout a large frequency range. The experimental portion of this work was completed on a stationary beam. Iwatsubo [5] presented work detailing the excitation of a rotor by external forces at specific frequencies for crack detection. In this study, a laboratory rotor was rotated at slow speeds and forces were applied by an exciter mounted to the shaft through a rolling element bearing. The new magnetic actuator technique presented here allows for advances to these earlier studies. In particular, Quinn [6] presents a technique using a magnetic actuator to apply a specified time-dependent forcing on the rotor system for identification of breathing cracks in the shaft of a rotating machine. The application of the correct forces by the AMB actuator can induce a combination resonance that can be used to identify the magnitude of the time-dependent stiffness arising form the breathing mode of the shaft crack. This technique was verified experimentally on the test rig described in the current paper.

The procedure presented in the current work is an extension of work performed by Humphris [7]. In his work, Humphris utilized the AMBs that were supporting a rotor as a source for applying various perturbations to the shaft and monitoring the response for health diagnosis. In addition to expanding the work by Humphris, the work here focuses on applying the approach through a separate

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actuator for machines not supported magnetically. This allows for a much broader application of the technique to the general class of rotating machinery.

Specifically, this paper presents results showing the system identification capability of the procedure as compared to traditional modal analysis techniques, and a demonstration of the ability of the technique to detect the presence of shaft cracks while the rotor is spinning at a constant speed. In this experimental investigation, a shaft with a machined notch near the mid-shaft location was used to represent a cracked shaft. Two notch depths, one at 10% of shaft diameter and one at 25% of shaft diameter, were examined for comparison to the response of a healthy (un-notched) shaft. Experimental results showed no change in the 1st and 2nd modes but a distinct and progressive shift in the 3rd mode for the two notched cases in the Frequency Response Function (FRF) output resulting from the AMB excitation input measured at a proximity probe. The ability to interrogate the shaft at select frequencies through a large range demonstrates the capability of the technique to detect shaft cracks during rotor operation that might otherwise go undetected by conventional steady-state vibration monitoring techniques.

While the addition of AMB actuator(s) to a rotating machine for improved health monitoring is not without cost, the actuators can potentially be economically justified by utilizing them to improve rotor dynamic performance. For example, Kasarda [8] has demonstrated that the midspan use of an AMB as a "third bearing," or magnetic damper, for the reduction for two modes of rotor synchronous vibration. Additional work has also demonstrated the reduction of subsynchronous vibrations such as those resulting from aerodynamic rotor instability using the third AMB bearing approach (Mendoza [9]). In jet engines, the addition of one or two AMBs as additional bearings may add enough stiffness to the system to allow for the reduction of shaft diameters resulting in weight reduction and lower surface speeds for the rolling-element support bearings (resulting in longer bearing life In addition to using the actuators as rotor dynamic design tools, they can also be used for active vibration cancellation. Hope [10] demonstrates the use of a feedforward approach for reducing unbalance forces in a commercial centrifugal compressor supported in AMBs. There are enough potential design advantages for the addition of magnetic actuators to a rotating system, in addition to the health monitoring potential, that will facilitate the broad use of the proposed health monitoring approach in aerospace, power generation, and petrochemical applications.

II. Test Set Up

A single disk test rotor shown in Fig. 1 was used in the experiments. The rotor is supported by rolling element bearings and has a bearing span of 610mm (24 inches) and a shaft diameter of 15.9mm (0.625 inches). Attached to the rotor is a 1.25kg (2.75 lb) disk with threaded holes located around the periphery which are used for adding balance or unbalance to the system. The test rotor is driven by a variable speed electric motor through a flexible coupling with a maximum operating speed of 10,000 rpm. The test rotor has been designed so that one or more critical speeds can be achieved in the operating speed range to mimic the dynamics of industrial rotating machinery. The magnetic bearing, which is used to excite the rotor, consists of a ferromagnetic 48 mm (1.9 inch) diameter disk attached to the shaft as a "target" and an 8-pole heteropolar stator with 2-axis of control.

Figure 1: Test Apparatus

III. Comparison to Modal Testing (Non-Rotating)

Testing procedures began with the system identification of a healthy system, which is a test rotor with no faults as described above. The first tests involved examining non-rotating rotor response due to different input forces from the AMB actuator as compared to classic modal test results from a hammer input.

The goal of this initial work was to examine relationships between various AMB excitation inputs and associated outputs, and to compare these to results from established modal test methods. For this non-rotating case, the accelerometer is placed on the stationary shaft to the left of the balance disk and the non-contact proximity probe is located in the same angular location as the accelerometer in a plane near the AMB actuator as shown in Fig. 2. An instrumented hammer was used near the AMB actuator for the modal tests as also shown in Fig. 2.

A function generator was used in conjunction with a single axis of the AMB actuator to achieve the various excitation forces to the system for comparison to the modal test approach.

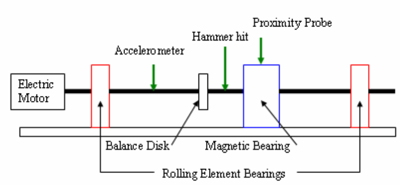


Figure 2: Diagram of Test Rig Including Modal Test Equipment

Shown in Figure 4 is the response of the rotor at the accelerometer and proximity probe due to a mechanical modal hammer impact near the AMB actuator towards the center of the rotor. As shown in Fig. 3, the main excitation frequencies identified at both sensors from this test are at approximately 50 Hz, 221 Hz, 509 Hz, and 788 Hz, respectively.

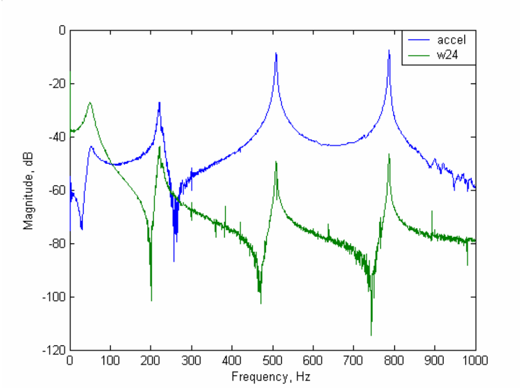


Figure 3. Mechanical Hammer Excitation to the Rotor, Output at Accelerometer and Proximity Sensor.

After the modal test with the mechanical hammer, a function generator was used to achieve a chirp excitation force applied to the rotor through the AMB actuator. As shown in Fig. 4, the response at the accelerometer and proximity probe for this AMB excitation scenario also shows response frequencies at approximately 50, 222, 509, and 786 Hz, which are almost identical to the frequencies identified with the mechanical hammer hit.

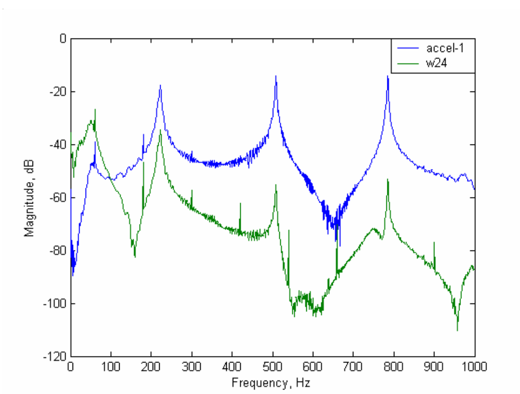


Figure 4. AMB Chirp Excitation to the Rotor, Output at Accelerometer and Proximity Sensor.

Results from both the traditional modal test and AMB excitation approach were essentially the same (within $\pm$ 1 Hz) with the main excitation frequencies identified from both tests at approximately 50 Hz, 221 Hz, 509 Hz, and 788 Hz, respectively.

As shown here, the frequencies identified using non-contact AMB excitation matched with the classic modal technique results using a mechanical hammer. However, unlike the modal hammer approach, the AMB actuator can be used to excite the system while the rotor is spinning. This will allow for identifying actual operating system frequencies which are a function of speed-dependent properties such as bearing stiffness and damping, gyroscopic effects, and fluid-structure interactions. In this manner, a more accurate picture of system resonances and their changes due to damage can be tracked.

IV. Shaft Crack Identification

After tests to demonstrate the general system identification ability of the approach were completed, a specific fault in the form of a shaft crack was chosen for identification. In an effort to approximate a shaft crack, a 0.635 mm (0.025 inch) wide notch was cut into the midspan of the shaft in two increments of 10% and 25% of the shaft diameter, respectively. Testing began with a "healthy" system, which means no damage. Then, the 10% shaft diameter notch was added and the system retested. This procedure was the repeated for the 25% of shaft diameter notch depth.

Dynamic testing was taken while the rotor was operating at 2400 rpm and a burst chirp frequency spectrum of force excitation was applied to the rotor through one axis of the AMB at frequencies up to 1000 Hz. Resulting FRF response was recorded at a proximity probe located near the AMB. An example of the FRF for the full frequency range for the healthy and 10% depth notch tests is shown in Fig. 5.

An examination for the FRF response for the healthy, 10% depth, and 25% depth notch showed no change in the frequency or magnitude of the 1^{st} or 2^{nd} natural frequency. However, there was a distinct and progressive shift in the third natural frequency for the damaged shaft cases. Specifically, the third natural frequency was recorded at 520 Hz for a healthy shaft, 517 Hz for the shaft with the 10% notch, and 511 Hz for the 25% notch as shown in detail in Figs. 6a and b, respectively. The absence of a frequency shift in the 1^{st} and 2^{nd} modes, and the presence of a frequency shift in the 3^{rd} mode for the notched rotors are also predicted from a standard undamped rotor dynamic analysis that was performed for each of the three cases. In addition, similar experimental results were obtained for output at different probe locations, and for different rotational speeds, but these results are not presented here for brevity.

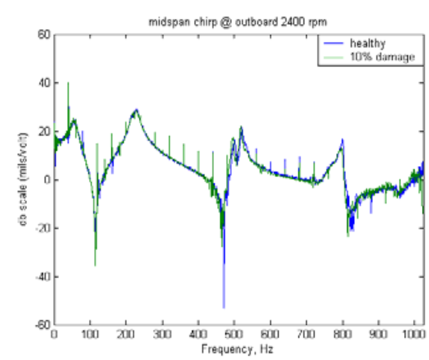
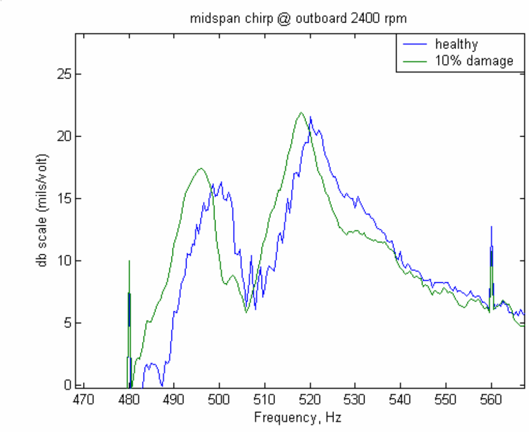
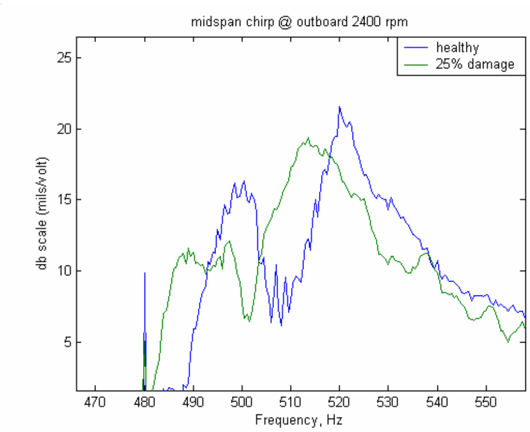


Figure 5. FRF Response to AMB Excitation for Healthy and 10% Depth Notch While Rotor is Operating at 2400 RPM.



a) healthy vs 10% notch



b) healthy vs 25% notch

Figure 6a and b: Test Results

V. CONCLUSIONS

A new method for improved health monitoring of rotating equipment has been introduced and verified on a laboratory rotor. The new method consists of using an AMB actuator to interrogate a rotating system by applying forces at specified frequencies in order to monitor the Frequency Response Function (FRF) for discrepancies indicating damage. The rotor in these tests is supported in conventional support bearings and the AMB used for actuation was placed in a mid-shaft location. The method was first compared to conventional modal analysis on a non-rotating shaft in the tests and demonstrated comparable results in identifying the natural frequencies of the system. A notched shaft was then chosen to represent a shaft crack for identification purposes while the rotor was operating at a speed of 2400 rpm. Three cases were examined including a healthy (un-notched) shaft, a shaft with a mid-span notch extending to a depth of 10% of shaft diameter, and a shaft with the same mid-span notch extending to a depth of 25% of shaft diameter. Excitation up to 1000 Hz was applied via one axis of the AMB actuator to the three rotor cases and corresponding responses were recorded. No changes in the 1st or 2nd natural frequencies were detected but distinct shifts in the 3rd natural frequency were detected from the FRF data. This is consistent with predictions from a rotor dynamic analysis of undamped critical speeds for progressively increasing notches. These results demonstrate the viability of the technique for detecting shaft cracks that might otherwise go undetected in typical steady-state vibration monitoring approaches.

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