Applications of a Magnetic Bearing Acting as an Actuator In Conjunction with Conventional Support Bearings

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ABSTRACT

This paper describes three applications of Active Magnetic Bearings (AMBs) used as actuators in conjunction with conventional support bearings. These pilot studies demonstrate how AMB technology can be used to facilitate improvements in rotating machinery operation without concerns associated with full magnetic support of a rotor. These pilot studies include the use of an AMB actuator for the reduction of acoustic emissions due to gear noise, the reduction of subsynchronous rotor vibrations, and the reduction of synchronous vibrations in the lift fan driveshaft/engine assembly of the F35 STOVL Joint Strike Fighter.

AMB technology is often limited by the perception that it is just a support bearing option. In that scenario, limitations with full magnetic support of a rotor often eliminate consideration of the technology for many applications. The results of these studies demonstrate that the AMB is an enabling technology for facilitating improvements in rotating machinery design and operation in scenarios without full magnetic support of the rotor.

INTRODUCTION

The vast majority of research and commercial applications of Active Magnetic Bearings (AMBs) have utilized this technology in a support bearing capacity [1]. While there are many advantages to full magnetic support of a rotor in many applications, the use of the AMB as an actuator, in conjunction with conventional support bearings, for enabling performance gains in rotating machinery has been widely overlooked. In this manner, problems associated with full magnetic levitation of a rotor, such as power failure issues, high front-end costs, and concerns with low load capacity, are eliminated or significantly reduced, while many of the capabilities of AMBs can still be exploited. Researchers have published works examining the effectiveness of the AMB for use as a “third bearing” or Active Magnetic Damper (AMD) to add stiffness and damping at strategic locations to modify rotor dynamic characteristics. Schweitzer [2] demonstrated, analytically and experimentally, that a considerable reduction of the resonance amplitudes could be achieved using active damping in flexible beams and in flexible rotors supported in conventional bearings. Nikolajsen [3] demonstrated the use of AMDs for the suppression of vibrations in long transmission shafts. Kasarda [4] demonstrated experimentally that the location of the damper in a multi-mass rotor plays an important role in its effectiveness because of the rotor mode shapes, and reported that significant reductions of up to 88% in synchronous vibrations at the first and second critical speeds were attainable. These studies clearly demonstrate theoretically and experimentally the ability of a single AMB to significantly modify rotor dynamic performance of a rotor. The results of these studies imply that the manufacturer of rotating machinery can utilize the AMB to facilitate design changes such as smaller shaft diameters, resulting in lower weight and lower support bearing surface speeds, or higher operational speeds of existing and future designs.

To further demonstrate the capability of the AMB as actuators to successfully address other rotating machinery issues, the authors have conducted three pilot studies. This paper will present a few results from these pilot studies including examining the effectiveness of utilizing AMBs as actuators to attenuate gear noise, reduce subsynchronous vibrations, and reduce synchronous vibrations utilizing a feedforward control strategy, respectively.

STUDY #1: REDUCTION OF GEAR NOISE

Surprisingly little work has been done in using AMBs for the reduction of acoustic noise. The authors are aware of only one study where the filtered X-LMS feedforward algorithm has been used successfully with magnetic bearings by Piper and Calvert [5] in their application to actively control fluid borne noise from a centrifugal pump. They used a tachometer as a reference signal and a downstream hydrophone as an error signal. In all cases, the active vibration cancellation work in the literature has focused on systems supported entirely by magnetic levitation. The study shown here involves the novel use of AMB technology not for rotor support but rather as an actuator for reduction of acoustic noise emanating from a rotating system.

The authors are unaware of any work addressing the reduction of gear noise using AMB technology. Many rotating machines are coupled through gearing mechanisms that can cause large dynamic forces. These forces arise because there is often a change in load (dynamic tooth load) as the torque is passed from one tooth in the gear to the next [6,7]. The dynamic tooth loads can be affected by varying torque transmission levels, bearing pre-loads, rigidity of the casings and shafts and misalignment [8]. Dynamic tooth loading creates a forcing function that varies at the gear mesh frequency. Any dynamic forces caused by the rotor or gear can then transmit along the shaft, through support bearings, into the machinery casing and radiate as unwanted noise. This noise is often a health risk in the workplace and an annoyance to passengers in transport vehicles.

One proposed solution to this problem is to combine two emerging technologies, namely active magnetic bearings [1] and adaptive active control systems based on the filtered X-LMS algorithm [9], to create an effective, compact, and efficient solution to gear and rotor noise by counteracting the disturbance forces on the rotor. X-LMS feedforward active control systems have been used widely for the active control of noise and vibration [5,9,10]. The two main advantages of filtered X-LMS control systems are: (i) their flexibility of application and (ii) their adaptability to changing conditions. The controller has a system identification component such that the controller does not have to be fundamentally redesigned for each application. Presented here are experimental results from a pilot study to examine the reduction of unwanted acoustic emissions by using AMBs as actuators for the application of feedforward techniques to a rotating system. The AMBs are used in conjunction with conventional support bearings to eliminate issues associated with full magnetic support of rotating equipment.

The test rig developed to investigate the control of gear noise is shown in Figure 1. The test rig consists of two shafts and two gear sets intertwined in a *four-square* configuration and driven by an electric motor. This *four-square* gearing configuration allows the gears to be loaded at operating forces without requiring a large motor to drive the system. This is achieved by twisting the shafts before meshing the gears. Both gear sets are aluminum spur gears consisting of an 84 tooth gear with a pitch diameter of 13.3 cm and a 48 tooth pinion gear with a pitch diameter of 7.6 cm. Each shaft is approximately 9.52 mm in diameter and 61.0 cm long. Both shafts are mounted on conventional ball bearings for radial support and any axial loading was handled by the motor bearings. Each shaft is also equipped with an AMB and both AMBs are used in both a closed-loop configuration (acting as a "third bearing" for each shaft) and as an actuator for application of the feedforward control for noise reduction. The gear meshing tone is near 490 Hz, corresponding to approximately 613 rpm with a 48 tooth gear.

The AMBs used in the study are an eight-pole, 35 mm stator inner diameter heteropolar design with a digital PID controller manufactured by Revolve Magnetic Bearings, Inc. In order to directly measure a reference signal to drive the active control system, a proximity probe was placed near one of the gears with the gear as the target. In this manner, the proximity probe measured a series of on-off pulses proportional to the number of gear teeth and speed of the rotor. In this test both proximity probes and microphones were used as error sensors. Two microphones (#2 and #3) were placed radially, relative to the gear sets, and two microphones (#1 and #4) were placed axially, relative to the gear sets, as shown in Figure 2.

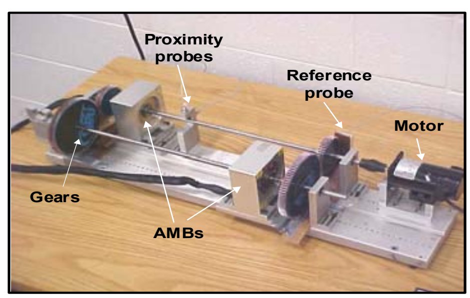


FIGURE 1: The four-square gear noise rig

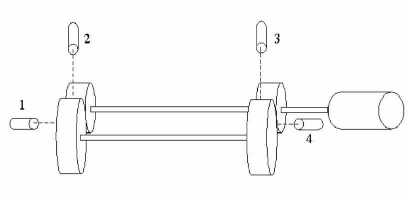


FIGURE 2: Microphone positions and orientation

Experimental Results. Experimental measurements were taken with the system operating with and without feedforward control. An AMB is inherently unstable and in all cases, the AMBs were operated in the same closed-loop configuration adding stiffness and damping to the system as a "third bearing" on each shaft. Figure 3 shows a spectrum of the sound at microphone #1 placed near to the gears with and without feedforward control. The gear meshing tone near 490Hz was effectively reduced by over 20dB using this system. The reductions at all of the microphones is not as good as for microphone #1 and results varied from a reduction of 0 to 5 dB for the other three microphones. The control system minimizes the sum of the squared error signals and therefore the largest reduction is at the microphone where the signal is largest An average pressure squared levels for all four microphones was determined and showed an overall reduction of 6dB for all microphones at the gear meshing frequency. The limitations of the control came mainly from variability in the drive system and four-square configuration. During the testing, the speed of the rotor varied making it difficult to make clear control on/control off comparisons. A more detailed discussion of this study can be found in Johnson, et. al [11]

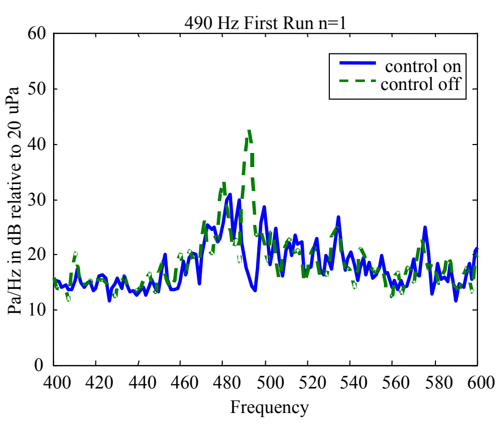


FIGURE 3: Experimental Results for Gear Mesh Frequency at 490 Hz at Microphone #1

STUDY #2: REDUCTION OF SUBSYNCHRONOUS VIBRATIONS

Rotor instabilities in turbomachinery often manifest themselves as a re-excitation of the first rotor critical speed resulting in lateral rotor vibrations at a frequency below the rotor operating frequency. Considerable work exists in the literature involving the analysis of destabilizing mechanisms and passive solutions for reducing subsynchronous vibrations. The authors propose here a novel active control solution utilizing AMB technology in conjunction with conventional support bearings. The AMB is utilized as an Active Magnetic Damper (AMD) at rotor locations inboard of conventional support bearings. Presented here are initial proof-of-concept experimental results using an AMD for vibration control of subsynchronous rotor vibrations in a high-speed laboratory rotor. The study shows that subsynchronous vibrations are reducible with an AMD and up to a 93% reduction in the amplitude of subsynchronous vibrations is demonstrated.

Test Apparatus. A multiple disk flexible rotor was used to evaluate the effectiveness of an AMD to reduce subsynchronous vibrations. The rotor is supported in conventional oil-lubricated bronze bushings and two independent AMDs are placed on the rotor, where one is used as an actuation source of subsynchronous excitation while the other provides active control of rotor vibrations. In this manner, the AMD used as a source is operated in open-loop mode with no feedback. A sine wave perturbation signal at or near the first rotor critical frequency is applied to the rotor through the electromagnets simulating excitation from a destabilizing mechanism (actual re-excitation of the first critical speed was not verified). Figure 4 shows the schematic and photograph of the experimental apparatus.

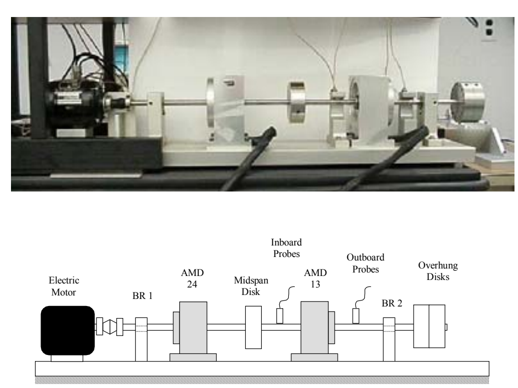


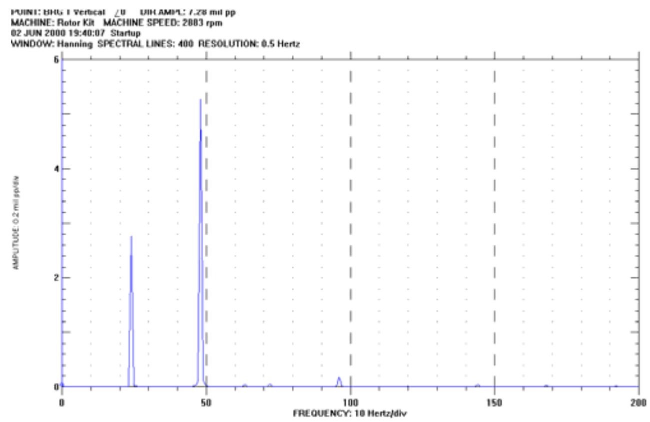
FIGURE 4: Schematic (not to scale) and photograph showing the single-disk rotor kit configuration

The three-disk rotor was assembled using a steel shaft 9.52 mm (0.375 inch) in diameter and 641.4 mm (25.25 in) long, one steel disk 72.2 mm (3 in) diameter, 25.4 mm (1 in) thick and a weight of 657 gr. (1.45 lbs) placed at the midspan, 285.8 mm (11.25 in) from the coupling, and two steel disks 72.2 mm (3 in) diameter, 19.1 mm (0.75 in) thick and a weight of 493 gr. (1.08 lb) each placed together close to the outboard end of the shaft, 600 mm (23.63 in) from the coupling. One actuator, AMD–24, was placed at approximately one quarter of the rotor span and its radial rotor was attached to the shaft at 155.5 mm (6.1 in) from the coupling. A second actuator, AMD-13 was located at two thirds of the rotor span and its radial rotor was attached to the shaft at 406.4 mm (16.0 in) from the coupling. The rotor was driven through a flexible coupling by a 0.1hp electric motor rated up to 10,000 RPM. Bently Nevada non-contact proximity probes were used in the rotor to monitor vibrations in two planes as shown in Figure 4. The actuators were manufactured by Revolve Magnetic Bearings Inc. and where the same design as used in the gear noise study.

Experimental Results

Kirk and Miller [12] state that compressors operating at a speed higher than twice the first critical speed would be likely to exhibit nonsynchronous vibrations. Therefore, scenarios where the subsynchronous excitation frequency is approximately one-half of the operating frequency of the rotor were examined here. For the results presented here, the stiffness and damping associated with the active AMD were determined to be 0 N/m (0 lb/in) and approximately 539 N-s/m (3.1 lbf-s/in), respectively, throughout the speed range of interest. The corresponding stiffness and damping associated with the bronze support bushings are determined to be approximately 312,991 N/m (1800 lb/in) and 1392 N-s/m (8.0 lbf-s/in), respectively.

To mimic an instability mechanism, a sine wave perturbation at approximately one-half of the operating frequency of the rotor was injected *open-loop with no feedback* through AMD-24, which acted as a quarter-span disturbance source, similar to a turbomachinery scenario where cross-coupling from oil seals at this location which may lead to instability. In this scenario, AMD-13 was used in an active control mode, adding damping in an attempt to reduce the magnitude of the resulting subsynchronous rotor vibrations. In this test, the rotor was taken up to 2880 RPM (48.0 Hz), approximately twice the first critical speed of the rotor configuration and a 0.4 V sine wave at 1440 RPM (24.0 Hz) was injected through the quarter-span AMD (AMD-24) resulting in a 2.8 mils pp subsynchronous vibration at 1440 RPM (24.0 Hz) at the vertical inboard probe. The two thirds rotor-span AMD (AMD-13) was subsequently turned on resulting in a reduction of this subsynchronous vibration component from 2.8 mils pp to 0.2 mils pp. In addition to the 93% reduction in subsynchronous vibration, a reduction of 17% of the uncompensated synchronous vibration was also achieved. Large shaft bow in the rotor is believed to have limited better performance of the AMD in reducing uncompensated synchronous vibrations. The spectra of the uncompensated vibration signal in mils (pk-pk) at the vertical inboard probe with the AMD-13 "off" and "on" are shown in Figure 5 (a) and (b), respectively.



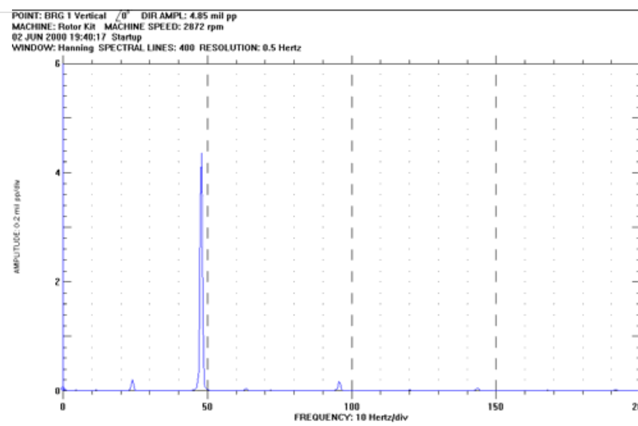


FIGURE 5: Spectrum of the vibration signal at the vertical inboard probe, three-disk rotor at 2880 RPM (48 Hz) and a perturbation at 24 Hz through AMD-24. (a) AMD-13 off. (b) AMD-13 on.

STUDY #3: F35 STOVL DRIVESHAFT VIBRATION REDUCTIONS

The Short Take Off Vertical Lift (STOVL) version of the F35 Joint Strike Fighter has experienced high vibrations due to unbalance in the driveshaft between the engine and lift fan. A theoretical study has been performed demonstrating the ability of a single magnetic actuator operating in a feedforward mode with the filtered X-LMS algorithm to significant reduce global and local vibrations in the engine-driveshaft assembly. This work involves the X-LMS algorithm used in the gear noise reduction work and the specific application is an extension of the theoretical and experimental work by Johnson, et. al[13] examining the effect of the number of sensors and actuators on local and global reduction of vibrations in a rotor supported in AMBs. The schematic of the rotor model used to determine mode characteristics is shown in Figure 6. The mode characteristics were used for input into a mode summation code in order to analyze various active control and unbalance scenarios. In addition to the modes, unbalance forces and control forces have been introduced at various locations to determine the optimal active control performance.

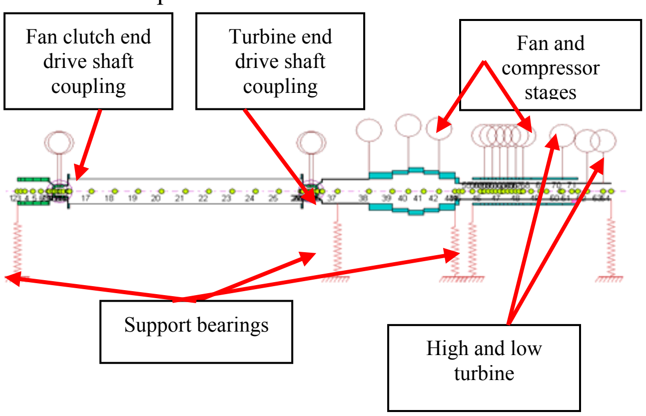


FIGURE 6: Schematic of rotor model for F135 and drive shaft

Multiple actuator and sensor scenarios were examined, and for brevity, the optimal design is presented. The operating range of interest is between 5000-9000 rpm. It has been determined that the optimum reduction for a general unbalance case occurs for a single AMB actuator when the actuator is placed at the engine-end of the driveshaft near the coupling in conjunction with three error sensor inputs from proximity sensors located at the driveshaft clutch-end, driveshaft mid-span, and driveshaft engine-end, respectively. A demonstration of global reductions for the length of rotor assembly including high and low turbine rotors as shown in Figure 6 is shown in Figure 7 for operation at a system critical speed at ~8000 RPM. The results in Figure 7 are for the case when unbalance is placed at four different locations to mimic expected worst case operating conditions. Corresponding Amplitude vs. Frequency plots are shown in Figure 8 for the driveshaft mid-span (Position 21) and the low-pressure compressor of the engine (Position 38), respectively.

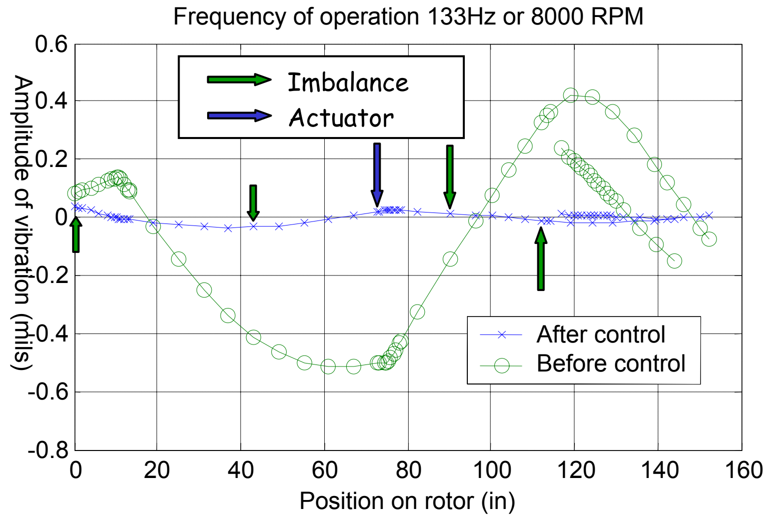


FIGURE 7: Amplitude of Vibration vs. Rotor Position With and Without Active Control

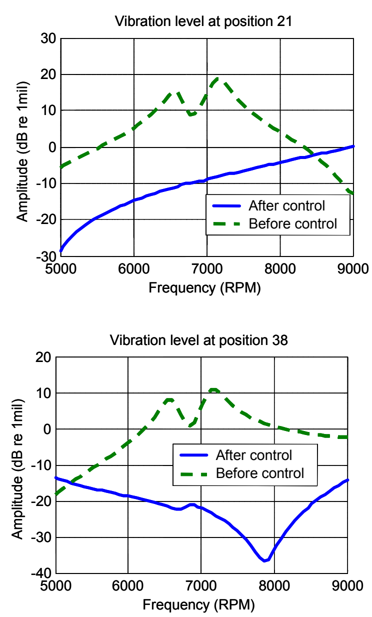


FIGURE 8: Vibration Level at Positions 21 and 38 versus Frequency with and without active control.

CONCLUSIONS

The purpose of this paper is to demonstrate three scenarios where AMB technology is used in conjunction with conventional support bearings to obtain specific performance objectives. In the first case, up to 20dB of reduction in acoustic levels of a gear meshing frequency using an AMB in a feedforward configuration was demonstrated experimentally. This pilot study shows the potential of this technology for reducing acoustic emissions which can be problematic in such applications as rotorcraft personnel compartments. The second study experimentally demonstrated the ability of a mid-span AMB to reduce subsynchronous vibrations in a rotor-bearing system by up to 93% in a laboratory test configuration. This pilot study demonstrates the potential of this technology to be used to enhance turbomachinery performance by allowing for higher speed and higher pressure operation since the AMB can be used to counter the instability mechanisms associated with fluid-structure interaction under high performance scenarios. In the final pilot study, a theoretical study was completed demonstrating the ability of a single AMB using a feedforward X-LMS algorithm with multiple input sensors to significantly reduce global vibrations in a gas turbine/fan drive shaft assembly for the vertical take-off version of the F35 Joint Strike Fighter. All of these pilot studies demonstrate the successful application of AMB technology as an actuator, in conjunction with conventional support bearings, for performance enhancement of rotating machinery.

ACKNOWLEDGEMENT

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