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ACTIVE CONTROL OF GEAR NOISE USING MAGNETIC BEARINGS FOR ACTUATION

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ABSTRACT

This paper investigates experimentally the active control of gear noise and vibration using magnetic bearing actuators in a feedforward active control scheme. The dynamic forces caused by gear meshing can produce large noise and vibration signatures that can cause annoyance and also fatigue mechanical components. In this work active magnetic bearings were used as actuators to introduce control forces very close to the source of the disturbance i.e. directly onto the rotating shaft. The proximity of the actuators to the source ensures that substantial control can be achieved using a small number of actuators. A *four-square* gear rig was constructed in order to test the control methodology experimentally. A proximity sensor placed near the gear teeth was used as a reference sensor and used to drive the two magnetic bearing actuators through a time domain filtered X-LMS control system to minimize the outputs from both vibration and pressure error sensors. At one microphone over 20 dB of reduction in acoustic levels was achieved at the gear mesh frequency and an overall reduction of 6 dB was demonstrated at four microphones. It is also shown that gear mesh noise and sideband frequencies can be simultaneously controlled.

INTRODUCTION

Machines with rotating components are extremely common and are used in many industrial processes and in nearly all transport vehicles. 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 [1,2]. The dynamic tooth loads can be affected by varying torque transmission levels, bearing pre-loads, rigidity of the casings and shafts and misalignment [3]. 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 Bearing*s (AMB) [4] and Adaptive Active Control Systems based on the filtered X-LMS algorithm [5], to create an effective, compact, and efficient solution to gear and rotor noise by counteracting the disturbance forces on the rotor. Presented here are experimental results from a pilot study to examine the reduction of unwanted acoustic emissions by using Active Magnetic Bearings (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 Active Magnetic Bearing (AMB) is a feedback mechanism that is traditionally used to support a spinning shaft by levitating it in a magnetic field. Patents associated with passive, active, and hybrid magnetic bearings go back over 150 years and there are over 50,000 commercial applications of AMBs in the field today. Compared to conventional rolling element and hydrodynamic bearings, magnetic bearings have the capability for high surface speeds with low power losses. The AMB also has the added capability for active vibration control allowing for the reduction of rotor vibrations and better control of rotor positioning and

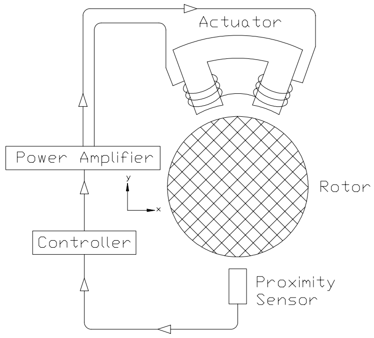

Figure 1: Active magnetic bearing principles

alignment. Shown in Figure 1 is one quadrant of a typical AMB system consisting of an input displacement sensor, a control circuit, a power amplifier and an electromagnetic actuator. For operation of the system shown in Figure 1, the displacement sensor monitors the position of the shaft and sends this information to the control system which determines the control signal necessary to keep the rotating shaft centered within the bearing actuator. This control signal is sent to a power amplifier to obtain the conversion and amplification from the small control signal voltage to the large current necessary for the electromagnetic actuator to support the required mechanical load.

Most radial magnetic bearings consist of electromagnetic actuators located radially around a ferromagnetic rotor attached to the shaft. Active magnetic thrust bearings also exist and operate under similar principles. They use planar air gaps and handle axial loads in an analogous manner. AMB support bearings have been used for various aspects of vibration cancellation. A considerable amount of literature exists on control algorithms used for reduction of rotor vibration. Knospe et al [8] discuss adaptive on-line balancing using digital control where algorithms in the frequency domain are used for the feedforward control scheme. Hope et al [9] discuss the use of two types of Active Vibration Control™ or AVC™ on a 6-stage hydrogen process compressor operating at 20.6 Mpa (3000 psi). One of the AVC™ modes is an open-loop rotating magnetic flux that is superimposed on top of the control fluxes. The rotating AVC™ flux is adaptive and effectively creates a force that is counter to the rotating unbalance force. The AVC™ flux is adaptive and can be applied based on the desired reduction of vibrations at a particular location, either along the machine shaft or on a baseplate to reduce transmitted forces. Surprisingly little work has been done in using AMBs for reduction of acoustic noise. The authors are aware of only one study where the filtered X LMS algorithm has been used successfully with magnetic bearings by Piper and Calvert [12] but in their application the control system was used 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 proposed work 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.

In terms of the reduction of acoustic noise, it has been shown that the most effective way to achieve good active control of a disturbance is to place the actuators close to the source of the disturbance [6]. AMBs provide an opportunity to apply dynamic control forces directly to a rotating shaft i.e. very close to the source of the disturbance. Take for example the noise created in helicopter cabins due to gear meshing [7]. The noise in this example is dominated by a gear meshing frequency between 800-1000Hz and if this disturbance is allowed to escape from the gearbox into the aircraft structure and into the interior acoustic space it becomes very difficult to actively control because the complexity of the system is too large (i.e. the number of actuators and sensors required to control the large number of degrees of freedom becomes prohibitive). However, if the disturbance is controlled at source, before it enters the structure, then the control system can remain relatively simple and compact. Figure 2 shows an example of how such a system could be used. AMBs placed close to the source can be used to isolate the disturbance and stop vibrational energy escaping into the supporting structure.

Although beyond the scope of this project, an additional benefit that may arise from this work is the reduction in the dynamic tooth loading on the gears and hence a reduction in tooth wear and fatigue. Active control systems often work to “unload” the primary source of the disturbance and hence the AMBs will in effect improve the smoothness with which the gears mesh with each other.

This paper will present results from an experimental study

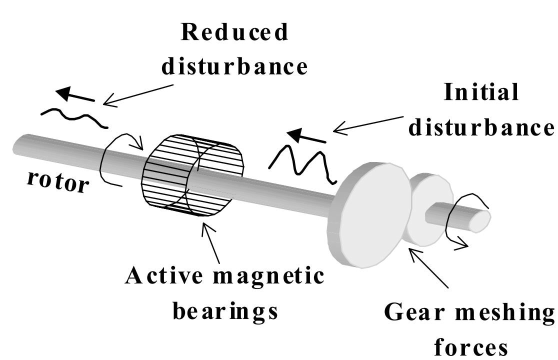

Figure 2: Using an AMB to reduce the disturbance due to gear meshing

demonstrating the viability of using AMBs as actuators for reduction of acoustic gear noise. In this study a small high-speed rotor arrangement is used to measure gear noise with and without the feed forward control applied through the two AMBs. At one microphone over 20 dB of reduction in acoustic levels were achieved at the gear mesh frequency and an overall reduction of 6 dB was demonstrated at four microphones.

EXPERIMENTAL SETUP

Test Apparatus

The test rig developed to investigate the control of gear noise is shown in Figure 3. 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. The pinion gear is mounted on the driving shaft. The gears are lubricated with Domade Red grease which is an all purpose grease with extreme pressure and anti-wear additives. 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 feed-forward control for noise reduction. The gear meshing tone near 490 Hz, corresponding to approximately 613 rpm with a 48 tooth gear. That is, the gear meshing is determined by the product of the number of gear teeth

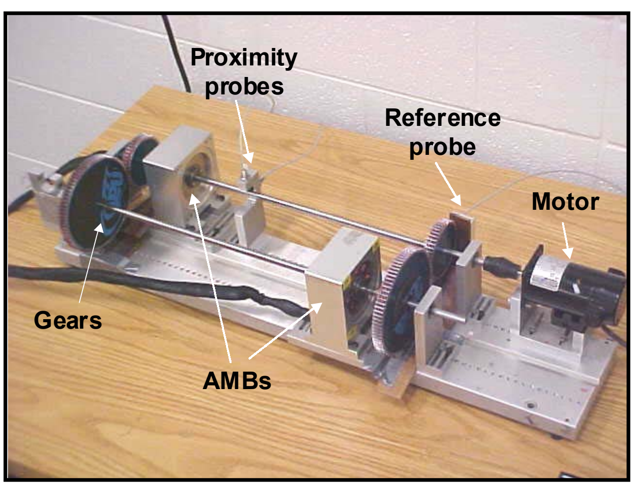

Figure 3: A picture of the four-square gear noise rig developed for the pilot study

and rotational speed of the rotor. In this configuration the vibration of the shaft was due to both rotor imbalance and gear meshing dynamic forces.

The AMBs used in the study are an eight-pole, heteropolar design with a digital PID controller manufactured by Revolve Magnetic Bearings, Inc. The bearings have a maximum rated load capacity of 53.4 N, and the saturation current is 3.0 A. The inner diameter of the stator is 35 mm, with a nominal diametric gap of 0.762 mm.

In order to directly measure a reference signal (x) 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 4.

Feedforward Control System: Filtered X-LMS

Filtered X-LMS feedforward active control systems have

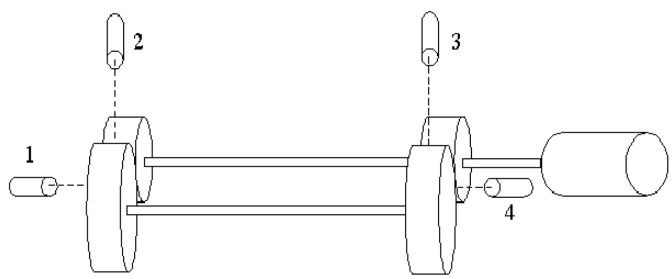

Figure 4: Microphone positions and orientation

been used widely for the active control of noise and vibration [5,6,13]. 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. Figure 5 shows a flow diagram for a filtered X-LMS time domain feedforward control system. This system requires a reference signal x, typically taken from a tachometer or upstream sensor, and uses it to drive the actuators through a digital finite impulse response (FIR) control filter W. The plant G is defined as the path between the input to the actuators y and the output from the error sensors e. For the algorithm to work it requires a model of the plant with which it filters the reference signal (also modeled with a digital FIR filter). This filtered reference or "filtered X" signal r is used

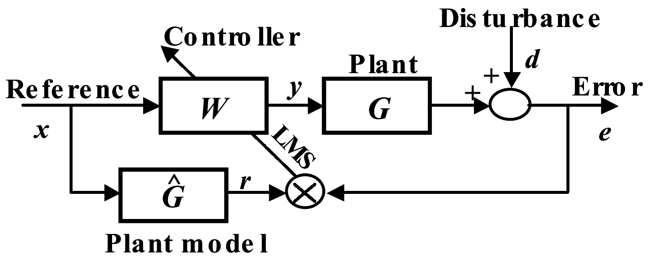

Figure 5: The flow diagram for a filtered X-LMS feedforward control system

with the error to automatically adjust the control filter *W*. The update equation is given by,

$$w_i(n+1) = w_i(n) - \alpha e(n) r(n-i)$$

where the new ith coefficient of the control filter $w_i(n+1)$ is changed from its old value $w_i(n)$ by a term proportional to the instantaneous error signal *e(n)* multiplied by filtered reference signal *r(n-i)* that is *i* samples old. The term α is a convergence parameter that determines how quickly the system adapts. If α is too large the system can become unstable but having α too small results in slow convergence. This adaptive component means that the system automatically converges to an optimal solution such that the signals from the error sensors (acoustic sensors in this application) are minimized. If the disturbance changes with time the controller can adapt itself to these changes and maintain performance.

In addition to performance advantages, advances in the speed of digital signal processors have reduced system costs and improved control system performance. Virginia Tech has developed high performance multi-input multi-output (MIMO) filtered X-LMS systems based on Texas Instrument's C40 DSP chips and is currently upgrading to the latest Texas Instruments C60 DSP platform. Virginia Tech's MIMO feedforward control system has been used successfully for the feedforward control of turbulent boundary layer noise [14], rotorcraft noise [7], car noise [15], piping noise [16] and used in the pilot work on controlling rotor imbalance [17]. It has proved to be a very flexible and robust control system and was easily adapted to work with AMBs. For this work a single proximity sensor, measuring the gear teeth, was used as a reference sensor (*x*). Four actuator channels were used (*y*), two for each of the two axes in each AMB, and used to minimize the outputs from four error microphones (*e*).

RESULTS

Experimental measurements were taken with the system operating with and without feed forward 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 6a shows a spectrum of the sound

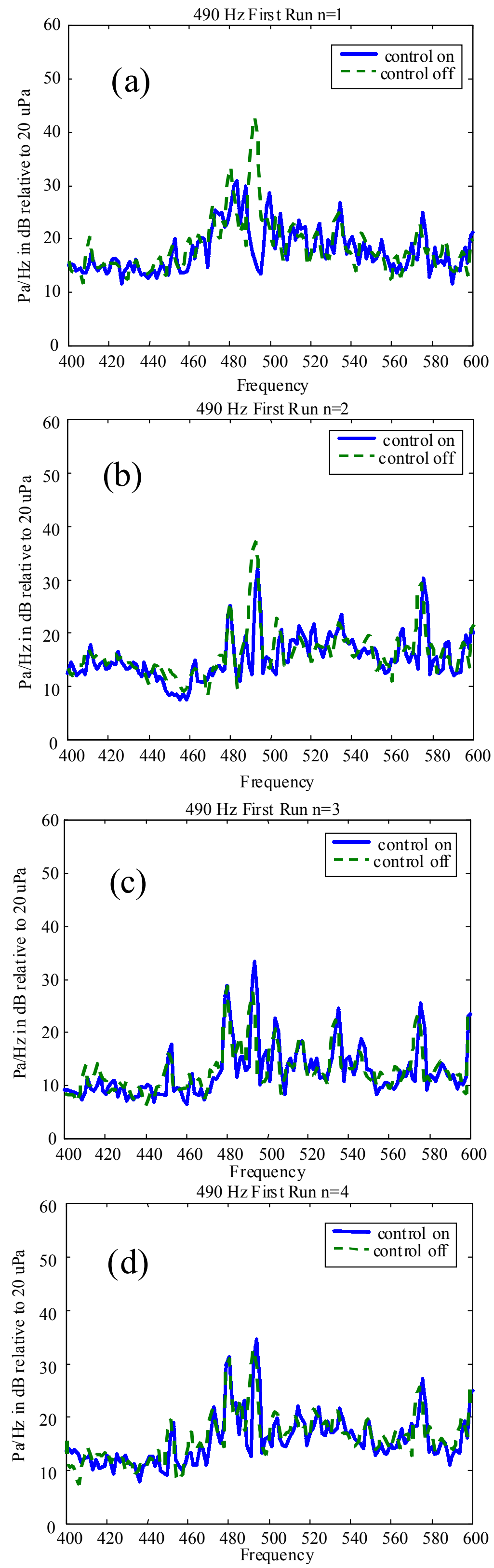

Figure 6: Experimental Results for Gear Mesh Frequency at 490 Hz at Microphones #1- 4 (a-d respectively).

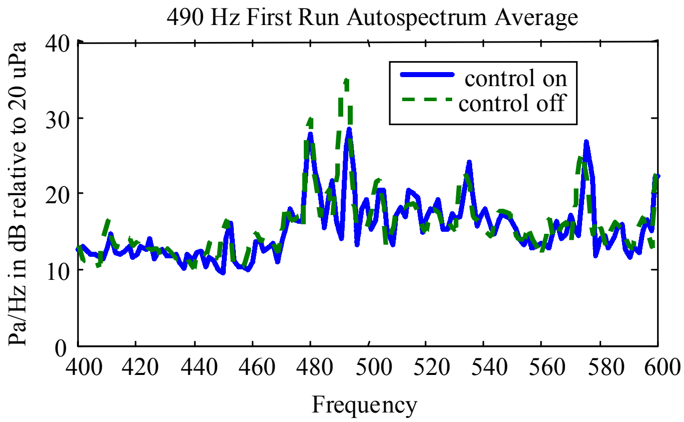

Figure 7: Average reductions in pressure at all four error microphones

at microphone #1 placed near to the gears with and without feed forward control. The gear meshing tone near 490Hz was effectively reduced by over 20dB using this system.

The reductions at all of the other microphones are not as good as for microphone #1 and results varied from a reduction of 0 to 5 dB for the other three microphones as shown in Figure 6 b, c and d, respectively. 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 which in this case is mic #1.

Shown in Figure 7 is the average pressure squared levels of all four microphones. This figure demonstrates 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. Slight variations in frequency between the two cases can be seen in Figure 7.

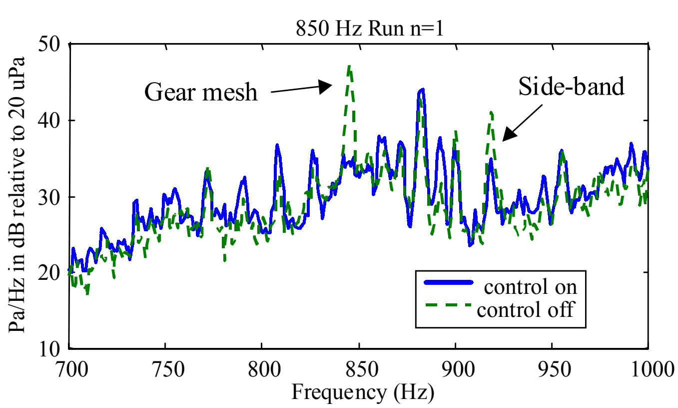

Figure 8: Simultaneous control of gear mesh tone and side-band

Side-band control

Figure 8 shows the control of the gear mesh noise at a single microphone for a higher rotor speed of 850Hz. This demonstrates that the control system is able to achieve control of side-band frequencies. Side-bands are cause by interactions between the gear mesh forces and the primary rotation frequency of the shaft. The feedforward control system used is able to achieve this because the reference sensor picks up the displacement of the gear directly and therefore can measured side band frequencies. These frequencies can then be used to drive the actuators to reduce the microphone levels.

CONCLUSIONS

Dynamic forces caused by gears can be transmitted along the shaft, through support bearings, into the machinery casing and radiated as unwanted noise. This noise is often a health risk in the workplace and an annoyance to passengers in transport vehicles such as helicopters. Active magnetic bearings can be used as an actuator to apply control forces to a rotating structure (shaft). In this study, a laboratory test rig was designed and built to test the viability of using AMBs as an actuator for the reduction of acoustic gear noise. 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. Both shafts are supported in conventional rolling element bearings and each shaft also has an AMB actuator for application of feed forward control.

An earlier pilot study successfully demonstrated that the filtered X-LMS system can be used in conjunction with AMB actuators to very effectively control the vibration of a rotor system [17]. The levels of attenuation achieved were large compared to the attenuations quoted in the literature validating the choice of reference signal and the control configurations. This paper extends the earlier work and showed that gear noise radiated from a rotor system can also be substantially reduced using this control methodology. A number of tests were conducted for a range of operating frequencies and demonstrated that the noise could be reduced under a variety of operating conditions using the same control setup. The first result presented in this paper shows control at the gear mesh frequency of 490 Hz and demonstrated large reductions (over 20 dB) in acoustic levels at the microphone with the highest noise level. An overall reduction of 6 dB in the average of the four microphones was also achieved.

A second test demonstrated that the system could also achieve control at a side band frequency while simultaneously controlling the gear mesh tone.

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