Development and Testing of a Passive Magnetic Support and Damping (PMSD) System

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ABSTRACT

This paper details the development and preliminary testing of a new type of passive magnetic bearing with integrated damping elements. The development of a "base excitation/resonant mass" test rig, which was developed to measure the stiffness and damping characteristics of the passive bearings, is also discussed. The results of parametric testing, which is being conducted to determine the effects of using different configurations of magnets and damping materials, is ongoing.

INTRODUCTION

One of the major limitations of passive magnetic bearings is their lack of damping. Various methods have been developed to incorporate damping into these bearings or to add damping to the system. For example, eddy current dampers have been extensively studied and many unique designs have been developed [e.g. 1, 2]. The drawback to these systems is that they are generally distinct from the passive bearing itself, thus adding additional components and expense to the overall system.

Other systems have been developed that mount the passive bearing in a damping mechanism, much like mounting a rolling element bearing in O-rings [e.g. 3, 4]. While significantly simpler than the eddy current approach, this also has disadvantages. The primary one being that, because the stiffness and damping elements are in series, the designer is limited in the ability to vary either stiffness or damping without altering the other factor.

The bearing described in this paper overcomes these limitations by directly incorporating damping elements into the passive bearing structure. The resulting bearing, referred to as a Passive Magnetic Support and Damping (PMSD) system, is essentially a composite structure with individual stator magnets mounted either conventionally (hard mount), or directly in a damping material (soft mount). By varying the number of hard mount and soft mount magnets, and the properties of the damping material, the stiffness and damping of the PMSD system can be independently adjusted to a greater degree than other systems. This paper discusses the design of this system more fully, and presents comparative test data between the PMSD system and conventional passive magnetic bearings.

DESIGN CONCEPT

Passive magnetic bearings are well described in the literature. Many configurations of these types of bearings are possible (e.g. US Patents 5,894,181; 5,619,083; 4,072,370; 3,958,842; and 3,614,18). Each of these configurations suffers from a lack of damping. Rotors supported on these types of bearings will, therefore, be poorly damped. This condition results in large vibrational amplitudes when the rotors traverse their critical speeds, increased sensitivity to imbalance forces, and decreased resistance to rotordynamic instabilities. This combination sometimes results in failure of the machines.

Many various techniques for introducing damping into passive magnetic systems have been developed (e.g. US Patents 5,910,695; 5,679,992; 5,521,448; and 5,386,166). Some of these methods utilize eddy current dampers, but these generally are very complicated and/or provide very low damping levels. An alternative method, utilized in some of the above patents, is to use a damping material, such as an elastomeric material or a woven material, to provide the damping. Many different configurations of this approach have also been disclosed. These configurations generally rely on introducing an intermediate housing between the rotor and the machine frame. In general, the stationary portion of the passive magnetic bearing is mounted in the intermediate housing. The damping material is then positioned between the intermediate housing and the machine frame. Rotor vibrations are transmitted from the rotor magnets to the stator magnets through the magnetic field. The transmitted vibrational forces cause movement of the stator magnets and the intermediate housing into which the magnets are mounted. This motion is resisted by the damping material, either in shear or in compression. The resistance of the damping material to the vibrations results in frictional forces, thus dissipating the vibrational energy.

This approach has several limitations. First, the intermediate housing represents an additional component that must be manufactured and assembled, adding to system cost and complexity. Secondly, the intermediate housing has a finite, and usually substantial, mass that is added to the bearing mass. This results in a loss of damping above the resonant frequency of the combined bearing stator and intermediate housing. Finally, in this configuration, all of the forces transmitted through the bearing must pass through the damping element. This limits the designers' ability to independently adjust the stiffness and damping of the bearing system to optimize rotordynamic performance.

In addition, several of these configurations rely on a single ring of magnetic material on each of the stator and rotor sections. Variations in the magnetic strength of the rotor and stator magnet materials results in variations of the magnetic forces as one ring rotates relative to the other. This results in "magnetic run-out," or a mechanical vibration of the rotor due to unbalanced magnetic forces. This sensitivity to variations in the magnetic field strength of the bearing magnets is undesirable.

It is therefore, desirable to develop a passive magnetic support and damping system without the above listed drawbacks. Such a system would: 1) be made of easily manufacturable components in a readily assemblable configuration; 2) provide a passive magnetic support and damping system that provides increased stiffness in response to large amplitude vibrations; 3) be minimally sensitive to variations in the magnetic properties of the permanent magnet materials used.

These objectives are achieved by providing a passive magnetic support and damping (PMSD) system, illustrated in Figures 2 and 4, in which the rotor portion of the system is comprised of a series of permanent magnet disks attached to the rotor of the machine. The stator portion is also comprised of a series of permanent magnet disks, which are positioned concentrically with the rotor magnets. To provide damping, at least one of the stator magnets is mounted directly in a damping material, which is in turn attached to the machine stator. This damping material may be an elastomeric material, a woven material, or any other type of material that exhibits frictional losses in response to shear or compressive strains.

These "soft mounted" stator magnet(s) provide damping to the system. The remaining stator magnets are rigidly attached to the machine stator and provide stiffness. These magnets are mounted in a conventional passive magnetic bearing configuration ("hard mounted"). By varying the number, size and magnetic strength of the stator magnets mounted in these two ways, the stiffness and damping of the bearing assembly can be varied independently.

In this type of configuration, only a single stator magnet is interposed between the rotor and any portion of the damping material. Because of this, the resonant frequency of the damping mechanism is very high (i.e. the effective mass of each damper element is minimized). This results in improved damping at higher frequencies than is available in previous designs.

An additional feature of this design is that the soft mounted stator magnets can be provided with a backing material that limits their displacement. When the soft mounted magnets come into contact with the backing material, they become effectively hard mounted, and contribute additional stiffness to the system. In this manner large excursions of the rotor, which cause large displacements of the soft mounted magnets, will result in increased bearing stiffness, tending to restore the rotor to the nominal position.

Finally, because multiple magnet rings are used on both the stator and rotor elements, the variations in magnetic strength inherent in the material causes minimal problems. The material variation are essentially "averaged out" over the multiple magnet rings.

TEST RIG DESIGN

To test the effectiveness of this design, and to conduct parametric studies of design variations, a test rig was developed (Figures 3 and 5). The test rig is of a "Base Excitation, Resonant Mass" (BERM) variety [5]. In this type of test rig, a shaker is used to introduce a sinusoidal vibration in the base of the rig. The test article is placed between the base and a variable mass. The frequency of the sine wave is varied until the system resonance is located. The stiffness and damping of the test article can then be determined from measurements of the amplification factor and the phase shift between the input vibration (base) and the output vibration (resonant mass) utilizing the following equations (5).

$$K = \frac{\omega^2 m\alpha^2 - \alpha\cos\phi}{\alpha^2 - 2\alpha\cos\phi + 1} \tag{1}$$

$$C = \frac{\omega \; m\alpha \;\; \sin\phi}{\alpha^2 - 2\alpha\cos\phi + 1} \tag{2}$$

where ω is the resonant frequency, αis the amplification factor, m is the mass of the resonant mass assembly, and ϕ is the phase angle between the input and output vibrations.

The resonance frequency, and therefore the measurement frequency, can be modified by adjusting the resonance mass, and/or by placing springs in series with the test article. If springs are used to increase the measurement frequency, then the stiffness and damping of the springs must be independently measured and subtracted from the results.

To eliminate the effects of static loading on the test bearings, the test rig is configured so that the bearing shaft is vertical. The shaker is mounted to provide horizontal motion. The base assembly, including the shaker, is mounted on low friction linear slides.

The resonant mass assembly is simply a plate, into which the "outer" portion of the test bearing is mounted. To minimize the effects of any support stiffness or damping, the plate is suspended in a set of linear active magnetic bearings. These bearings were constructed from "E-I" transformer laminations, with oversized "I" laminations to allow for lateral displacements. This configuration was tested by oscillating the base section with no test article or auxiliary springs in place. No vibrations were induced in the resonant mass section, thus verifying that no coupling through the AMBs existed.

TEST RESULTS

Early testing has revealed that this configuration is quite sensitive to many subtle design variations. It was difficult to achieve repeatable test data without strictly minimizing all sources of uncertainty in the test rig and the test procedures. Additionally, the signal to noise ratio was found to be very poor for the input vibration at resonance, especially in the low damping configurations (conventional PMB configurations).

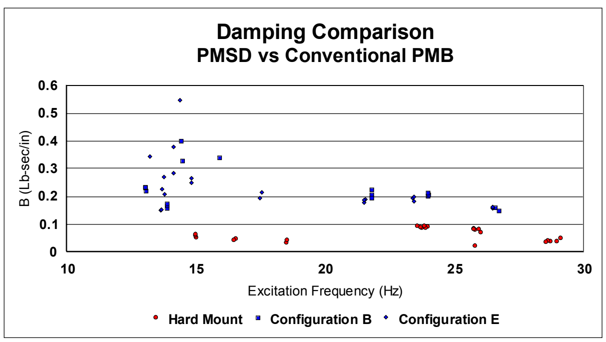

FIGURE1: Representative Data

Some representative data is presented in Figure 1. It is noted that the PMSD configurations have uniformly higher damping levels than the conventional AMB configurations. The improvement in damping levels ranges from 50% to almost 700%. The damping of the conventional PMB is actually overstated in this plot due to the signal to noise problem mentioned above. There is, however, a large spread in the PMSD data, especially at lower frequencies. Sources of these discrepancies are still being identified at this time. Additional testing is ongoing, and additional results will be presented at the conference.

CONCLUSIONS

A design concept to introduce damping into a passive magnetic bearing structure has been developed. Preliminary testing has revealed that the design is sensitive to many design parameters, such as the stiffness ratio between the magnetic fields and the damping material for the soft mount magnets, the relative axial and radial spacing between magnets, and the amount of shear that can be produced in the damping material. The effects of these, and other parameters, need to be better quantified through additional testing.

FUTURE RESEARCH

The nest step in the development of this technology is to demonstrate its use in actual applications. Plans are underway to incorporate the PMSD system in both a turbomolecular pump application and in a flywheel energy storage system. Both of these applications require high-speed rotation in a vacuum environment, and both require significant damping to ensure proper rotordynamic performance.

AKNOWLAGEMENTS

The material is based upon work supported by the National Science Foundation under Award Number: DMI-0078459. Any opinions, findings, and conclusions or recommendations expressed in this publication are those of the author and do not necessarily reflect the views of the National Science Foundation. The technology described in this work is further detailed in US Patent 6,448,679.

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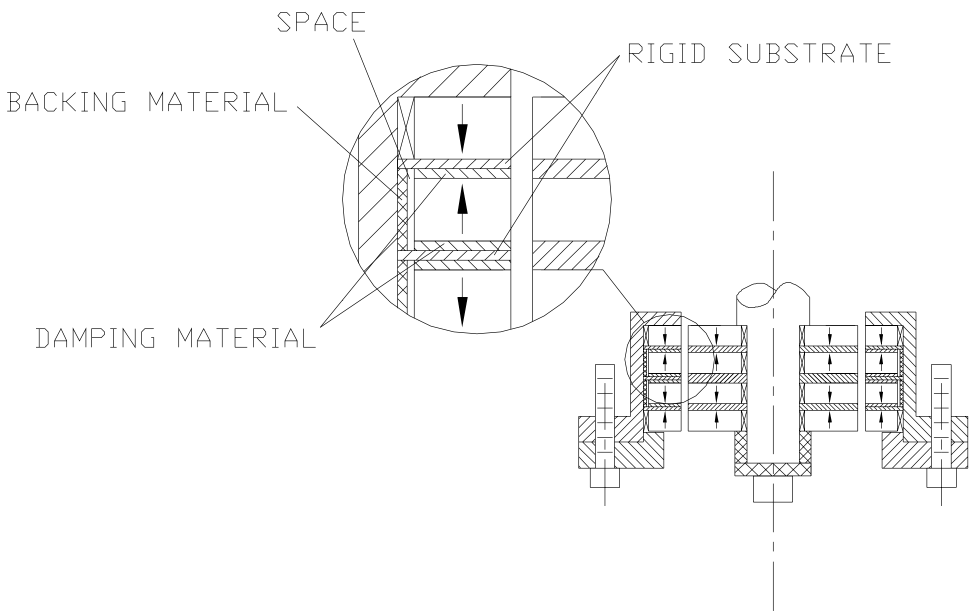

FIGURE 2: Schematic of PMSD Bearing

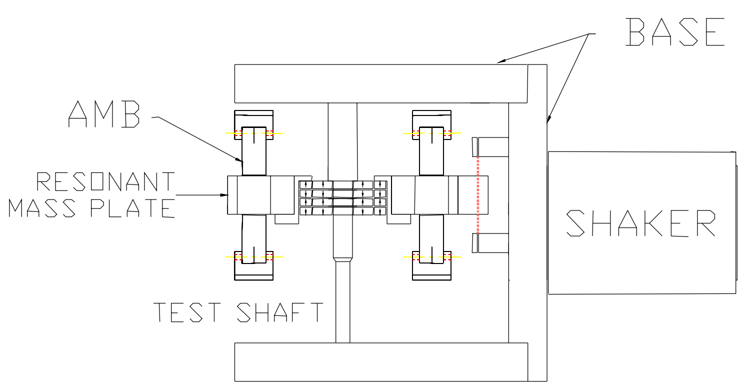

FIGURE 3: Schematic of BERM Test Rig

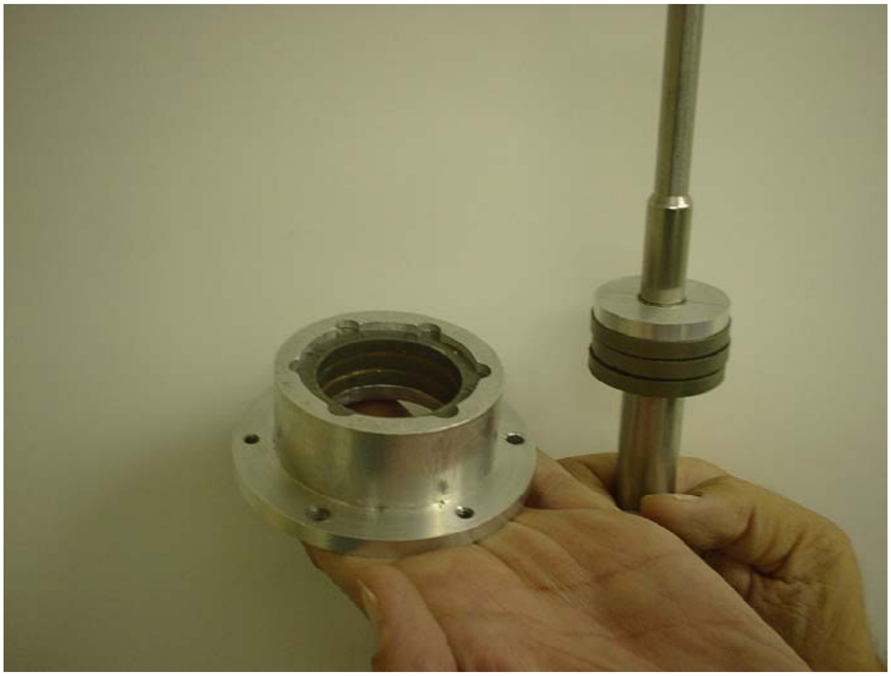

FIGURE 4: PMSD Test Bearing

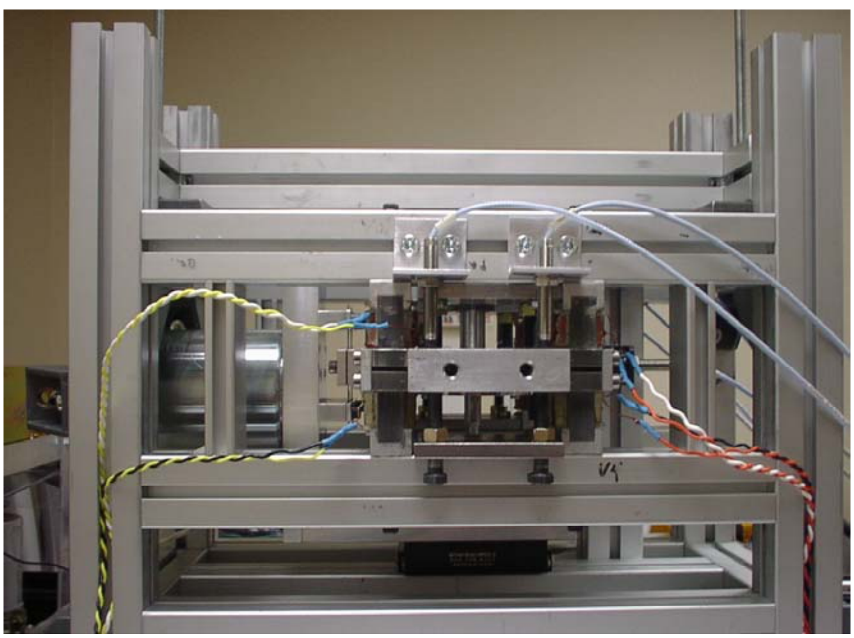

FIGURE 5: BERM Test Rig