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Advances in Applied Science Research, 2011, 2 (6):429-438

ISSN: 0976-8610
CODEN (USA): AASRFC

Vibration Control of Cantilever Beam Based on Eddy Current Damping

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ABSTRACT

A great difficulty is faced in the design of such structures due to the fact that the reduction in weight results in low rigidity and reduced vibration characteristics. Apart from the vibration is effectively controlled; it may destabilize the system and may, very often, result in complete failure of the system. Several advanced applications, such as those in jet fighters, automobiles and spacecrafts, require structures that are highly strong, lightweight and possess high structural damping property. Consequently, there is need to develop structures that are equipped with suitable vibration control. In modern vibration controller design for smart structures, much effort is devoted to the development of an efficient actuator along with high-performance control algorithms. Smart materials such as PZT, terfenol-D, ER, MR, and shape memory alloys have been exploited as candidates for actuators. In the present work, damping is obtained by the electromagnetic force which is generated by the movement of a conducting material through a stationary magnet or the movement of a magnet through a stationary conducting material. This causes ''eddy'' currents to flow in the conductor. These currents dissipate energy as they flow through the resistance of the conductor. The use of eddy currents for damping of dynamic systems has been known for decades and its application to magnetic braking systems and lateral vibration control of rotating machinery has been thoroughly investigated. Practically, it is unfeasible to prevent the vibration but it can possible to controlled and optimize the vibration up to certain limits.

Keywords: Vibration, Damping, Eddy current, Actuator, dissipate energy.

INTRODUCTION

In this dissertation, investigate the ability to use the eddy currents generated by magnetic fields to suppress the vibration of the ultra flexible devices intended for space. More often, vibration is undesirable, wasting energy and creating unwanted sound – noise. For example, the vibrational motions of engines, electric motors, or any mechanical device in operation are typically unwanted. Such vibrations can be caused by imbalances in the rotating parts, uneven friction or the meshing of gear teeth. Careful designs usually minimize unwanted vibrations. The study of sound and vibration are closely related. Sounds, or "pressure waves", are generated by vibrating structures (e.g. vocal cords); these pressure waves can also induce the vibration of structures (e.g. ear drum). when trying to reduce noise it is often a problem in trying to reduce vibration [1-2]. The eddy current phenomenon is caused when a conductive material experiences a time varying magnetic field. This time varying magnetic field can either can be induced either by movement of the conductor in the field or by changing the strength or position of the source of the magnetic field. The concept of using eddy currents for damping purposes has been known for a considerable time, with manuscripts dating to the late 1800's, the history of the eddy current damper will not be presented and only work from the past few decades will be reviewed [3-4]. By configuring the two oppositely poled magnets as shown in Figure1, the magnetic field is concentrated in the gap between the two magnet surfaces, therefore causing the conductive material passing through this region to experience the maximum change in magnetic flux and thus induce the greatest eddy currents and damping force.

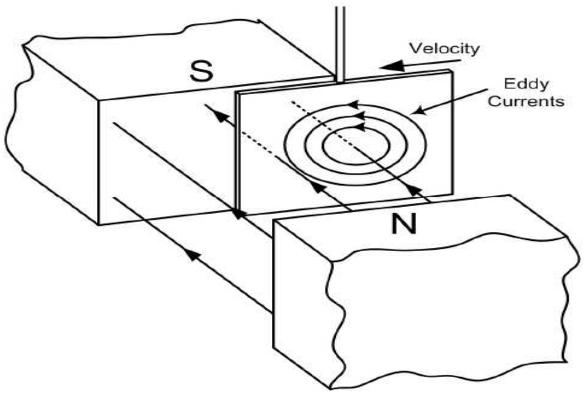

Figure1. Schematic the generation of eddy currents.

The magnets in the eddy current dampers used for the suppression of rotational vibrations are configured in the same manner as those used in magnetic braking applications. The damping concept was modeled using a rough finite element code and the damper's performance was estimated. The authors state that the damping generated by the system was sufficient to help suppress the rotor vibration; however results presented were hard to decipher. However, it is stated that the eddy current damper does form an effective vibration reduction mechanism for subcritical operation. To facilitate the identification of the damping ratio for each damper, the program AMPERES was used to generate the 3D magnetic flux of the magnet using boundary element techniques. For the structure in question, two tuned mass dampers were constructed for the two lowest modes of the structure. . The concept was constructed and tested to determine its effectiveness. It was shown through experiments that critical damping of the beam could be achieved using this system [5-6]. Using new model, the authors investigated the damping characteristics of the eddy current damper and simulated the vibration suppression capabilities of a cantilever beam with an attached eddy current damper numerically. The results showed the potential of this eddy current damper for suppressing the vibration of a cantilevered structure.

The studies that have investigated alternative methods using eddy currents to damp these vibrations have either been marginally successful or cumbersome and difficult to apply. For applications such as membranes the damping system must be applied such that it does not cause local surface imperfections, significant stiffness or mass loading [7- 8]. A need exists to develop an eddy current damping systems that can be easily applied to a structure while still providing significant damping. This dissertation will approach this problem and develop several non-contact methods utilizing eddy currents to applied significant damping to the vibrating structure. Furthermore, mathematical model of each method of eddy current damping will be developed to predict the amount (Morisue and Tsuboi, 1990) [9] of damping generated and dynamic response of the system. These models will differ from those previously developed because of the interaction of the magnet and conductive material utilizes only the radial magnetic flux rather than the flux in the direction of poling as done in magnetic braking applications. The models provided will also be shown to accurately predict the dynamic interaction of the system. The global mass and stiffness matrices used in these equations are derived from standard FEM techniques [10-11]. Modal Analysis technique has been used to convert the Lagrange's equations to eigenvalue problem. Different natural frequencies and mode shapes are determined from global mass and stiffness matrices [12-14].. Matrix deflation method is used to solve the eigenvalue problem, for pronouncement the natural frequencies and mode shapes. The bending moments produced by piezoelectric actuation are presented in mathematical form. Mathematical equations are presented for the conversion of the strain developed at the PZT patch, to the voltage generated [15-16].

MATERIALS AND METHODS

The goal of these experiments was to measure the damping of the beam as a function of the gap between the copper conducting plate and the surface of the permanent magnet. To do this, both the response to an initial displacement and the frequency response were measured. From these two tests the damping of the beam can be calculated by determining the log decrement of the initial condition response and applying the unified matrix polynomial approach (UMPA) to the frequency response. It was necessary to find the damping using both of these methods because

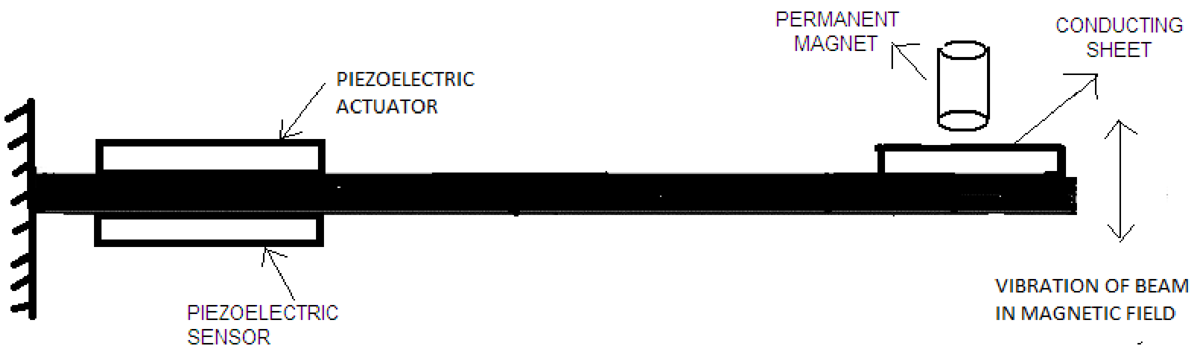

Figure3. Schematic experimental setup.

Significant damping is added when the magnet is placed in close proximity to the beam, making the damping measurement difficult. In order to accurately measure the damping of the aluminium beam using the log decrement, the initial condition must be consistent throughout all tests. This is necessitated further due to the need to measure the damping for numerous different gap distances between the magnet and conducting plate. To measure the frequency response of the aluminium beam, a piezoelectric patch was attached at the root of the beam as an excitation source, while the beam's response was measured using a PZT sensor. With the two excitation systems developed (initial condition and piezoelectric induced disturbance), the next step was to construct an accurate method of positioning the permanent magnet a fixed distance from the conducting plate. To allow the position of the magnet to be accurately varied, it was bonded to a wooden block that was fixed to a lead screw. Another figure4 shows a wooden block was used such that the magnetic field was not distorted due to high permeability materials in close proximity to the magnet. The combination of a lead screw for positioning, an electromagnet for consistent initial displacement, a permanently bonded piezoelectric patch and a non-contact sensing system, (laser vibrometer) allowed every test to be precisely repeated.

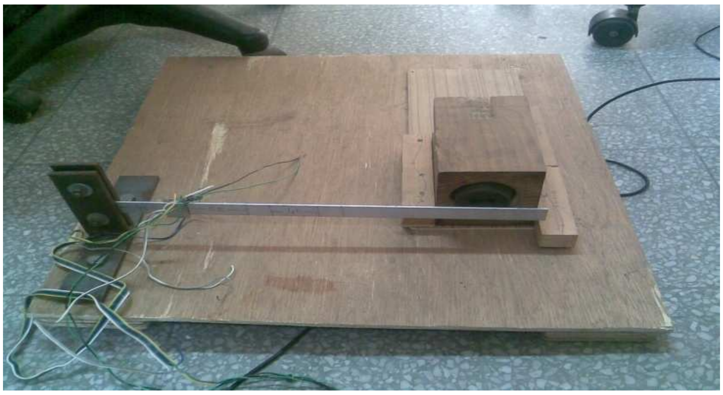

Figure4. Experimental Setup of cantilever beam.

2.1 Equipment characteristics
The following are the characteristics of the different equipments:

2.2 Software "Lab view RT"
The National Instruments 'Lab VIEW' is a highly productive graphical development environment for building data acquition, instrumentation and control systems. With 'Lab VIEW' it is easy to create interfaces that give interactive control of software system. The tight integration of 'Lab View' with measurement hardware facilities, rapid development of data acquisition and control is possible. This software contains powerful built – in measurement analysis and a graphical compiler for optimum performance. For applications that require real time performance, there is special system called 'Lab VIEW Real – Time'. 'Lab VIEW Real – Time' downloads standard 'Lab VIEW' codes to a dedicated hardware target running a real – time operating system independent of the operating system.

2.3 simultaneous I/O data acquit ion card (DAQ 6062 E)
National Instruments 'E-Series' technology is a complete data acquisition (DAQ) hardware architecture that takes advantage of the latest in electronics and technological innovations and advances the capabilities of PC – based DAQ solutions. 'E-Series' is a standard architecture in instrumentation class, for multichannel data acquisition.

This architecture includes the followings

- NI – PGIA gain independent, fast settling time amplifier.
- RTSI multi-board/multifunction synchronization bus.
- DAQ – STC counter/timer.
- MITE PCI bus master interface.
- Shielded, latching metal connectors.

Following are some of the important features of the E series data acquisition system

2.3.1 Analog input "FIFO"

The 'E series' devices perform both single and multiple Analog – to – Digital (A/D) conversion of a dynamic signal made of infinite number of samples. A large FIFO buffer holds the data during the A/D conversion so that no data is lost. Multiple A/D conversions can be handled with programmed input-output, interrupts, or direct memory access (DMA). Total 16 no. of channels can be handled. The maximum input voltage range is –10 Volts to +10 Volts. 12-bit resolution is there, which means that full-scale accuracy is 1.443 mV.

2.3.2 Analog output

The 'E series' devices also perform both single and multiple 'Digital-to-Analog (D/A) conversion' from a fixed number of data points to infinite number of samples. Multiple D/A conversions can be handled with programmed input-output, interrupts, or direct memory access (DMA). Total 2 numbers of channels can be handled. The maximum input voltage range is –10 Volts to +10 Volts. A 12 bit resolution is there, which means that full scale accuracy is 1.443 mV. Large FIFO DAC buffers for high-speed analog output updates are available to accommodate varying bus latencies and to ensure no data loss.

RESULTS AND DISCUSSION

3.1 Frequency response calculations

In order to obtain the frequency response function of the structural response with respect to the excitation w (t) at the PZT actuator, we describe the equation of motion by a state-space representation

$$\dot{z}(t) = A\,z(t) + B\,w(t)$$

Where **z** (t) is the state vector; **A** is the system matrix; **B** is the input matrix; and

$$z(t) = \begin{bmatrix} \dot{x}(t) \\ x(t) \end{bmatrix}$$

$$A = \begin{bmatrix} 0 & I \\ -M^{-1}K & -M^{-1}C \end{bmatrix}$$

$$B = \begin{bmatrix} 0 \\ -1 \end{bmatrix}$$

Where **I** be the identity matrix I and is a matrix of ones. In the case the displacement of the primary structure, x_S (t) if of interest, the output vector **y**(t) is given as

$$y(t) = C_y z(t) = [\,1\ 0\ .....0\,] \begin{bmatrix} x(t) \\ \dot{x}(t) \end{bmatrix}$$

For systems initially at rest, the transfer function of the state-space realization and derived by taking the Laplace transformation:

$$H(s) = C_y (sI - A)^{-1} B + Dy$$

where s is the complex argument, and L_Y (s) and L_W (s) are the Laplace transform of **y** and w(t), respectively. Since the Fourier transform is equivalent to evaluating the bilateral Laplace transform with complex argument s= i ω, the frequency response function is obtained as

$$h_s(\omega) = C_y (i\,\omega.I - A)^{-1} B + D_y$$

Where, ω is the circular frequency argument.

To demonstrate the effectiveness of this non-contacting magnetic damper for the suppression of the transverse vibrations of a beam, experiments were performed to determine the frequency response before and after placement of the magnet, the results of this test are shown in Figure 4. This figure shows that the first mode of vibration is reduced by 15 dB and the second and third mode are suppressed by 15 dB and 10 dB respectively when magnet is placed at a distance of 1mm. The response of the system in the transfer function form, from PZT actuator to PZT sensor is presented in this figure.

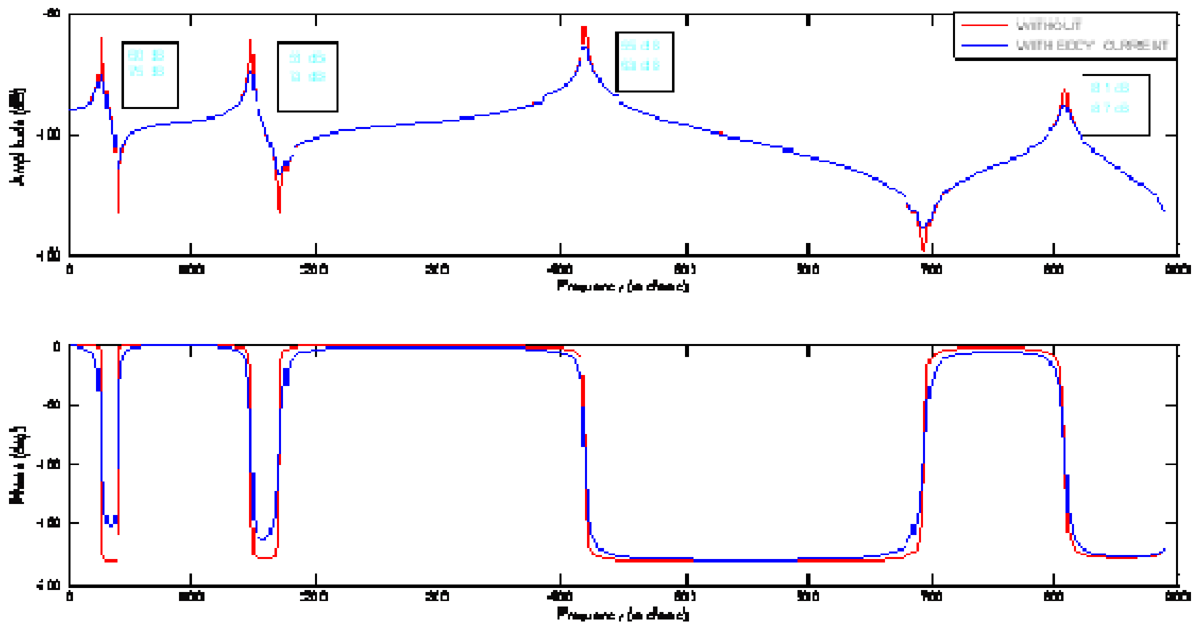

Figure5. Frequency response of the beam with the magnet at a distance, 1mm.

Figure 5 shows that the first mode of vibration is reduced by 10 dB and the second and third mode are suppressed by 4 dB and 4 dB respectively when magnet is placed at a distance of 2mm.

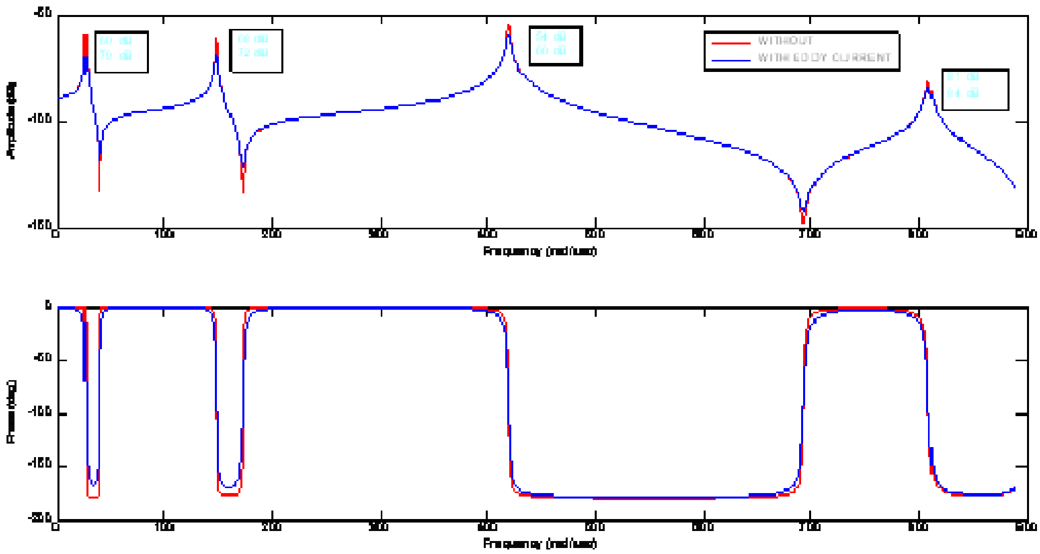

Figure6. Frequency response of the beam with the magnet at a distance, 2mm.

The response of the system in the transfer function form, from PZT actuator to PZT sensor is presented in this figure.

Figure 6 shows that the first mode of vibration is reduced by 6 dB and the second and third mode are suppressed by 4 dB and 4 dB respectively when magnet is placed at a distance of 3mm. It is very much clear from the above the three readings that the damping effectiveness get decreased marginally when the magnet is start moving away from the conductor

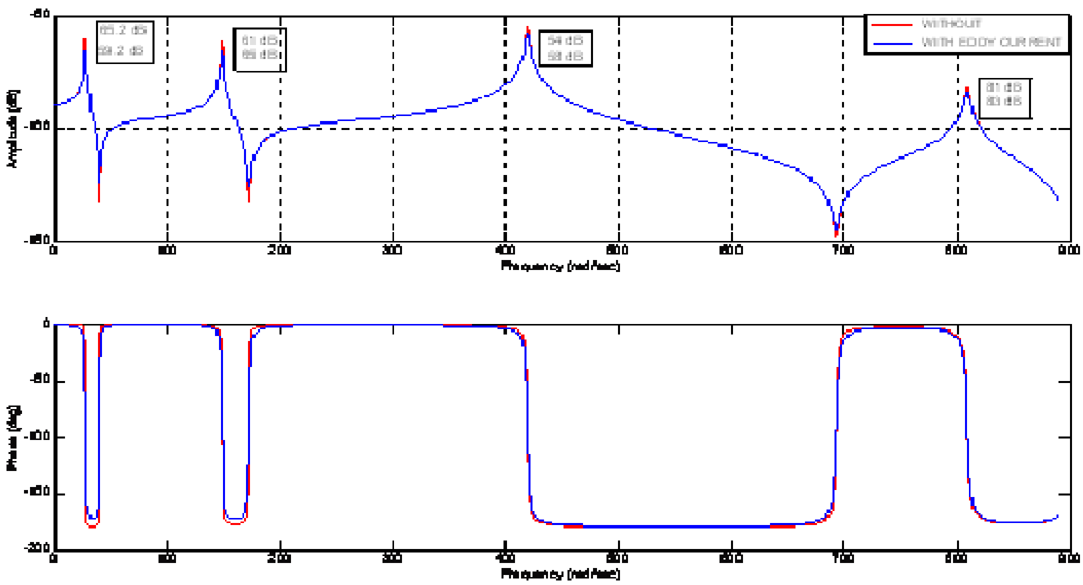

Figure7. Frequency response of the beam with the magnet at a distance, 3mm.

To view collectively the response of the system with and without magnet, tables (4-6) are made.

Table1. Damping magnitude at first mode, vibration.

Distances of magnet from the conductor	Damping magnitude at first mode
1.when magnet is at a distance of 1mm	15 dB
2.when magnet is at a distance of 2mm	10Db
3.when magnet is a distance of 3mm	6 Db

Table2. Damping magnitude at second mode, vibration.

Distances of magnet from the conductor conductor	Damping magnitude at second mode
1.when magnet is at a distance of 1mm	15 dB
2.when magnet is at a distance of 2mm	4dB
3.when magnet is at a distance of 3mm	4dB

Table3. Damping magnitude at third mode, vibration.

Distances of magnet from the conductor	Damping magnitude at third mode
1.when magnet is at a distance of 1mm	10 dB
2.when magnet is at a distance of 2mm	4dB
3.when magnet is at a distance of 3mm	4dB

4 EXPERIMENTAL TIME RESPONSE FUNCTION

To confirm the theoretical simulations, experimental time domain data is obtained with and without magnet. The magnet is placed at a distance of 1, 2 and 3 mm respectively. The following subsections discuss the performance of the system at various distances of the tip of the beam from the magnet. Figure4 shows the response of the system with and without magnet in time domain. The tip of the beam is displaced by 1mm from its equilibrium position. With the application of the permanent magnet the amplitude of vibration reduces to near zero in less than 0.15 seconds.

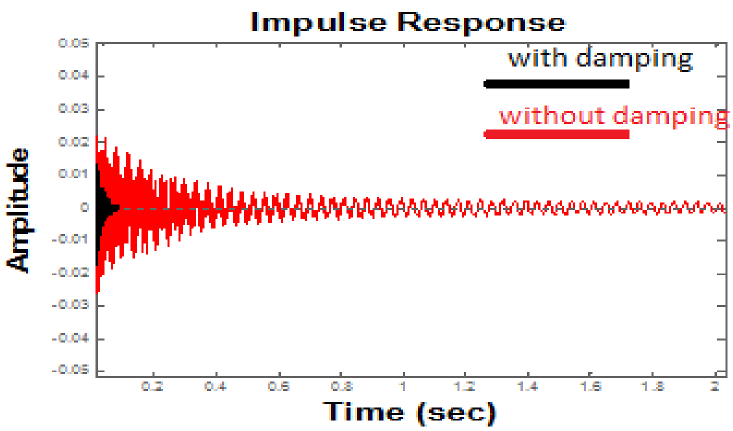

Figure8.Time response of beam as magnet sited at a distance, 1mm.

The response of the system with and without magnet in time domain. The tip of the beam is displaced by 1mm from its equilibrium position. However, the magnet is lying at a distance 2mm from the tip of the beam, figure8. With the application of the permanent magnet the amplitude of vibration reduces to near zero in less than 0.2 seconds.

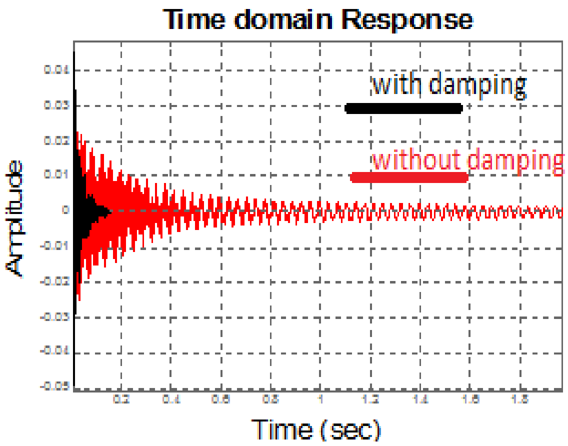

Figure9.Time response of beam as magnet sited at a distance, 2mm.

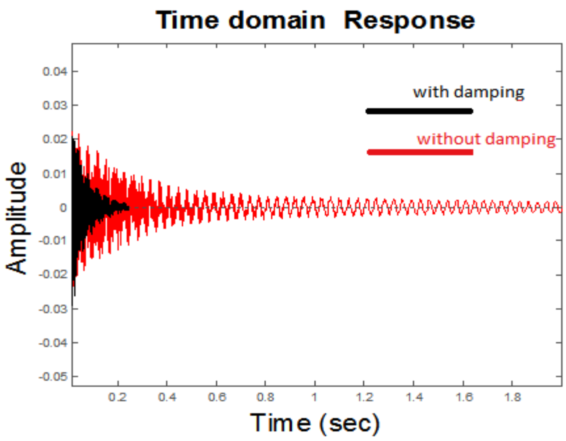

Figure10.Time response of beam as magnet sited at a distance, 3mm.

The response of the system with and without magnet in time domain. The tip of the beam is displaced by 1mm from its equilibrium position, figure9. However, the magnet is lying at a distance 3mm from the tip of the beam. With the application of the permanent magnet the amplitude of vibration reduces to near zero in less than 0.4 seconds.

CONCLUSION

1. The search for applicable actuation methods has lead to the development of systems that utilize a variety of active materials such as PZT, terfenol-D, electro-rheological, magneto-rheological, and shape memory alloys. However, one method of providing vibration suppression that has not seen significant research is eddy current damping. Dampers of this type function through the eddy currents that are generated in a conductive material experiencing a time changing magnetic field.
2. The density of these currents is directly related to the velocity of the conductor in the magnetic field. However, following the generation of these currents, the internal resistance of the conductor causes them to dissipate into heat. Because a portion of the moving conductor's kinetic energy is used to generate the eddy currents, which are then dissipated, a damping effect occurs. This damping force can be described as a viscous force due its dependence on the velocity of the conductor. While eddy currents form an effective method of applying damping, they have normally been used for magnetic braking applications. Furthermore, the dampers that have been designed for vibration suppression have typically been ineffective at suppressing structural vibration, incompatible with practical systems, and cumbersome to the structure resulting in significant mass loading and changes to the dynamic response.
3. To alleviate these issues, this dissertation has identified three previously unrealized damping mechanisms that function through eddy currents and has developed the necessary modeling techniques required to design and predict the performance of each. The dampers do not contact

the structure, thus, allowing them to add damping to the system without inducing the mass loading and added stiffness that are typically common with other forms of damping. The first damping concept is completely passive and functions solely due to the conductor's motion in a static magnetic field. The second damping system is semi-active and improves the passive damper by allowing the magnet's position to be actively controlled, thus, maximizing the magnet's velocity relative to the beam and enhancing the damping force.

4. The final system is completely active and uses an electromagnet, through which the current can be actively modified to induce a time changing magnetic flux on the beam and a controlled damping effect. Through theoretical and experimental results it has been observed that this type of passive damping technique is quite effective. The performance of the system improves dramatically if the distance between the tip of the beam and the permanent magnet is decreased. Experimental results validate the theoretical predictions.

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