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COIL BASED ELECTROMAGNETIC SEMI-ACTIVE VIBRATION CONTROL FOR FLEXIBLE STRUCTURES

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ABSTRACT: *In this study, the semi-active coil based electromagnetic vibration control was newly employed for vibration suppression of a cantilever beam integrated with a copper coil and a permanent magnet. The control system for the vibration suppression of the beam consists of a permanent magnet attached to an aluminum beam and a coil installed below the magnet. The switching system operates when the output of sensor is larger than the reference set value and a Proportional feedback control was implemented for faster settling time. The frequency response function of a beam with the theoretical model of the electromagnetic switching system was predicted, and its accuracy was compared with the experimentally measured data. Also, the time responses with and without switching control were investigated. The tested model is suggested to implement in vibration control of flexible structures.*
KEYWORDS: *Semi- Active Vibration Control (SAVC), PID control, Electromagnetic Actuator*

INTRODUCTION

Vibrations accompany us everywhere and in most cases these vibrations greatly affects the nature of engineering design. The vibration properties of engineering devices are often limiting factors in their performance. Vibration can be harmful and should be avoided, or it can be extremely useful and desired. In either case, knowledge about vibration analysis, measure and control is desired. Vibration control strategies have been used in many applications such as space structures, helicopters, aircraft structures, wind tunnel stings, civil engineering structures and machine tools. In all these cases an object has to be isolated from the source of vibrations. Automobiles and trucks have shock absorbers to damp out the vibration experienced due to roughness of the roads. However, energy in conventional shock absorbers gets dissipated as heat and was not used in any way. Regenerative electromagnetic shock absorbers provide a means for recovering the energy dissipated in shock absorbers, A. Gupta (2006). Electromagnetic devices have been widely used in various electronic and industrial applications including power generators, electric motors, loud speakers and propulsion trains. A microwave coil was employed to generate a magnetic field in the magnetic resonance force microscope (MRFM) [1]. Stepanek et al [2] studied the dynamic behavior of the movement of a magnetic micro-actuator. The electromagnetic micro-actuator with three diaphragms can be developed through fabrication, test results, or finite element analysis [3]. Chen et al [4] proposed a new type of non-conventional electro-magnetic actuator for micro-positioning. The electromagnetic damper also was used to actively reduce the vibration of 1-D.O.F suspension systems [5]. There are many applications of the electromagnet, such as actuators for aircraft flight control surface, long-stroke linear motors for high-speed packaging and manufacturing, reciprocating actuators for resonant electro-mechanical systems, magnetic bearings for flywheels, solenoid actuators for diesel fuel injector control, and actuators with multiple degrees-of-freedom [6]. The electromagnetic actuator was developed for the vibration control of a cantilever beam with a tip mass [7]. Some researchers [8-9] studied the design, modeling, and analysis of a novel magnetic spring–damper and also reported the modeling, simulation and testing of a novel eddy current damper to be used in vehicle suspension systems.

Piezoelectric shunt damping with voltage feedback can be classified into a semi-active control method with the power amplifier and the signal processing unit, is the most popular technique for minimizing the vibration of a smart flexible structures. The possibility of dissipating mechanical energy with a piezoelectric material shunted with passive electrical circuits is investigated by Hagood et al [13]. Applied switching R shunts, where a single resistor is either switched to, or disconnected from the piezoelectric transducer resulting in changes in patch stiffness [11]. With a heuristic switching law, they were able to damp vibration of a beam structure. Later, switching of R_L shunts have been proposed [12].

Figure 1 shows the concept of electromagnetic switching semi-active vibration control with single degree of freedom system in this study. It is called semi-active because of the shunt damping acts as passive damping given from the coil itself, and there was an external power supply for driving of the switching circuit and feedback control make it active control system.

The analog switch is connected with electromagnetic actuator which was placed on a mass, damper and spring system. The switch operating condition depends on the disturbance to the system. Switch "on" control when the displacement is more than reference displacement value, else "off". Here the displacement value means the vibration amplitude which increases with increasing of disturbance of system. In this study, the semi-active electromagnetic switching shunt damper was newly employed for vibration suppression of cantilever beam. The electromagnetic switching shunt damper consists of a magnet attached on a cantilever and a conducting coil placed under the bottom of the stack of permanent magnets, and both ends of the coil were connected to the switching circuits and amplifier.

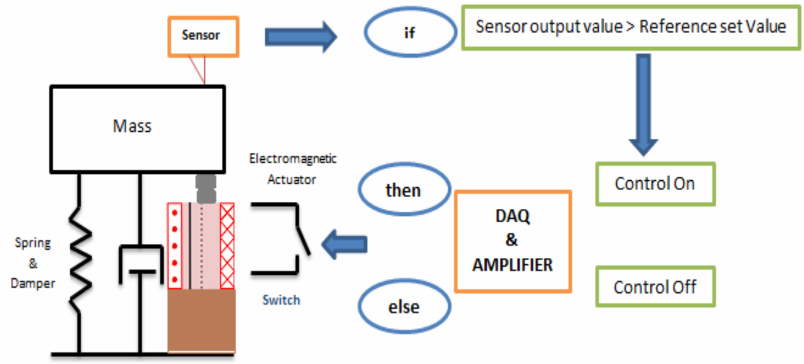

Figure1. Schematics on electromagnetic switching semi-active control for 1DOF system

The results showed good vibration damping effect with electromagnetic switching semi-active control. There has been 6.5 mm amplitude reduction at first mode of flexible beam structure in frequency domain.

ELECTROMECHANICAL COUPLING MATHEMATICAL FORMULATION

When a conducting coil moves through a non-uniform magnetic field or is in a region where there is a change in the magnetic flux, an electric current can be induced within the conducting media. Although the eddy current can be induced in any electrical conductor, the eddy current is most pronounced in solid metallic conductors. Eddy currents are utilized to induce heat and to damp oscillations in various devices.

In order to analytically model the dynamic governing equations of the cantilever beam with the copper coil, an unorthodox approach was used [10]. Using Hamilton's principle, the kinetic and potential energy of the vibrating flexible beams can be written by considering the tip mass effect of the copper coil as:

$$T=\frac{1}{2}\int_0^L\left(\rho A\left(\frac{\partial w}{\partial t}\right)^2+M_c\left(\frac{\partial w}{\partial t}\delta(x-x_c)\right)^2\right)dx$$

$$U=\tfrac{1}{2}\int_0^L EI\left(\left(\frac{\partial^2 w}{\partial x^2}\right)\right)^2 dx \qquad (1)$$

Here, M_C and x_C are the mass and location of the coil, respectively, and w denotes the transverse displacement of the cantilever beam. ρ, A, E, and I are the density, cross-sectional area, Young's modulus, and mass moment of inertia of the cantilever beam, respectively.

The virtual work consists of two terms: the first term is for work done by the external force, the second is due to the inherent damping force of a base:

$$\delta W=\int_0^L\left(f_{ext}(x,t)-c\frac{\partial w}{\partial x}\right)\delta w dx \qquad (2)$$

The equations of motion and all the natural and geometric boundary conditions can be obtained by applying Hamilton's principle:

$$\delta H=\int_{t1}^{t2}(T-U+W)dt=0 \qquad (3)$$

where t_1 and t_2 are the end points in the time domain and δ is the virtual work parameter. Substituting the strain energy and kinetic energy into Hamilton's principle yields the following equations of motion:

$$(\rho A+M_c\delta(x-x_c))\frac{\partial^2 y}{\partial t^2}+c\frac{\partial w}{\partial t}+EI\frac{\partial^4 w}{\partial x^4}=f_{ext}(x,t) \qquad (4)$$

The single mode approach can be used to create a discrete partial differential equation (4) into an ordinary differential equation. The flexural motion for a cantilever beam is simply approximated with a shape function, ψ (x)

$$w(x,t)=\psi(x)W(t) \qquad (5)$$

where

$$\psi(x)=\cosh\left(\alpha\frac{x}{L}\right)-\cos\left(\alpha\frac{x}{L}\right)-\beta\left(\sinh\left(\alpha\frac{x}{L}\right)-\sin\left(\alpha\frac{x}{L}\right)\right) \tag{6}$$

Here, the frequency coefficient, α, and the mode shape coefficient, β. By applying the assumed solution (5) to the equation of motion (4) and integrating the equation (4) multiplied with the shape function ψ(x), within the domain a single mode ordinary differential equation can be obtained.

$$M\ddot{W}(t)+C\dot{W}(t)+KW(t)=F(t) \tag{7}$$

where ,

$$M=\int_0^L \psi(x)\left(\rho A+M_c\delta(x-x_c)\psi(x)dx\right)$$

$$C=\int_0^L\left(\frac{d\psi(x)}{dx}C\frac{d\psi(x)}{dx}\right)dx \tag{8}$$

$$K=\int_0^L\left(\frac{d^2\psi(x)}{dx^2}EI\frac{d^2\psi(x)}{dx^2}\right)dx$$

$$F(t))=\psi\times f_{ext}(x,t)dx$$

The electro-magnetic force due to the induced eddy current, which is generated in the mechanism of the electro-magneto- mechanical coupling in the conductive coil that is located at x = x_C *, can be formulated in the following form through a point-load of the input force that is located at x =* x_S *.*

$$F(t)=\int_0^L f_{ext}(x,t)\psi dx=\left(-\phi i(t)\delta(x-x_c)+f^s(t)\delta(x-x_s)\right)\psi dx$$

$$=-\left(\phi\psi(x_c)\right)i(t)+\psi(x_c)f^s(t)=-\phi i(t)+F^sf^s(t) \tag{9}$$

where ϕ *=NBL is electromechanical coupling coefficient and i(t) is the induced current in the coil,* $f^S(t)$ *and* x_S *are the exciting force and the location, respectively. The single mode governing equation can be written as:*

$$M\ddot{W}(t)+C\dot{W}(t)+KW(t)+\Phi i(t)=F_Sf^S(t) \tag{10}$$

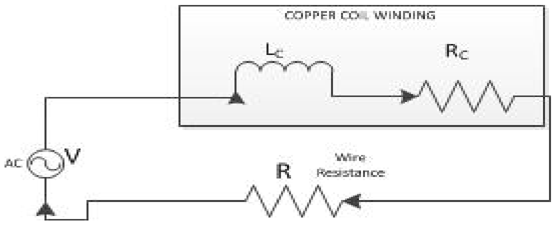

Figure 2. Circuit diagram of the switching shunt damper with voltage input.

Figure 2 shows the circuit diagram of the switching shunt damper. The copper coil has the electrical properties of inductance and resistance. The measured inductance and the electrical resistance of the coil are L_C *= 0.096 H and* R_C *= 12Ω respectively. Therefore the differential equation of the circuit can be written as:*

$$L_C\frac{di(t)}{dt}+R_Ci(t)=\phi\dot{w}\delta(x-x_c)=\Phi\dot{w}(t) \tag{11}$$

The electro-magneto-mechanical coupled equations can be expressed with 2x2 matrices.

$$\begin{bmatrix}M & 0\\ 0 & 0\end{bmatrix}\begin{bmatrix}\ddot{W}\\ \ddot{i}\end{bmatrix}+\begin{bmatrix}C & 0\\ -\Phi & L_c\end{bmatrix}\begin{bmatrix}\dot{W}\\ \dot{i}\end{bmatrix}+\begin{bmatrix}K & \Phi\\ 0 & R_c\end{bmatrix}\begin{bmatrix}W\\ i\end{bmatrix}=\begin{bmatrix}F^s\\ 0\end{bmatrix}\sin(\omega t) \tag{12}$$

Therefore, the magnitude of the frequency response function of the flexible beam integrated with the closed coil obtained by solving above equation. For free Vibration substitute f^S *in above matrices equal to zero.*

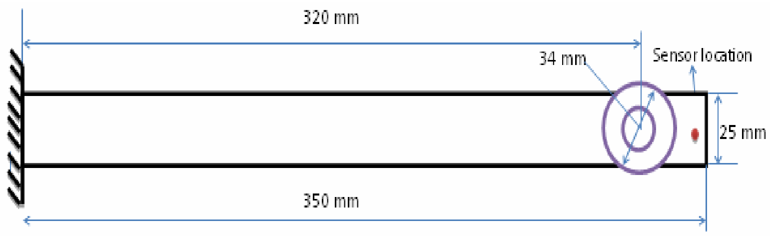

Figure 3. Cantilever geometry and sensing/actuator coil point's location.

Figure 4. Experiment setup with Laser Pickup (optoNCDT).

EXPERIMENTAL VERIFICATION OF ELECTROMAGNETIC SEMI-ACTIVE DAMPING CONCEPT

Electro-mechanical coupling mathematical model was validated by conducting experiment. The experimental tests were performed with an aluminum beam of density (ρ) =2700 kg/m^3, Young's modulus of Elasticity (E) = 70 x 10^9 N/m^2 and thickness (t) = 1.5 mm as shown in Figure. 3. The structural dimensions and locations of the laser displacement sensor (Micro-Epsilon optoNCDT 1400), and the conductive coil are depicted. The electromagnetic transducer consisted of a copper coil and a permanent magnet attached to flexible beam. The permanent magnet with a cylindrical neodymium was used to induce the magnet field in the coil. Considering the mass effect due to magnet attachment, the position of the copper coil in the cantilever beam was determined as 320 mm. As big mass is located at the end of the cantilever, the second mode will be similar, in terms of the shape, to the first mode of the Free -clamped beam. Figure 4 shows the experimental setup for the measurement of the frequency-response function of the integrated cantilever beam with an electromagnetic shunt damper. An external resistor was used for passive shunt circuit for tuning the resonant vibration of the cantilever beam. The National Instruments -PXI 1031 a 4 slot PXI chassis with Analog input module PXI-4496 at slot 2 and the LabVIEW program were used to obtain the time and frequency domain response by giving an initial displacement of 8.5 mm with the electromagnet switching mode. Active vibration control is achieved by providing a voltage to the coil proportional to displacement. A coil draws more current, TECHRON model 5507 was used as dual channel power amplifier designed for use in medium power systems.

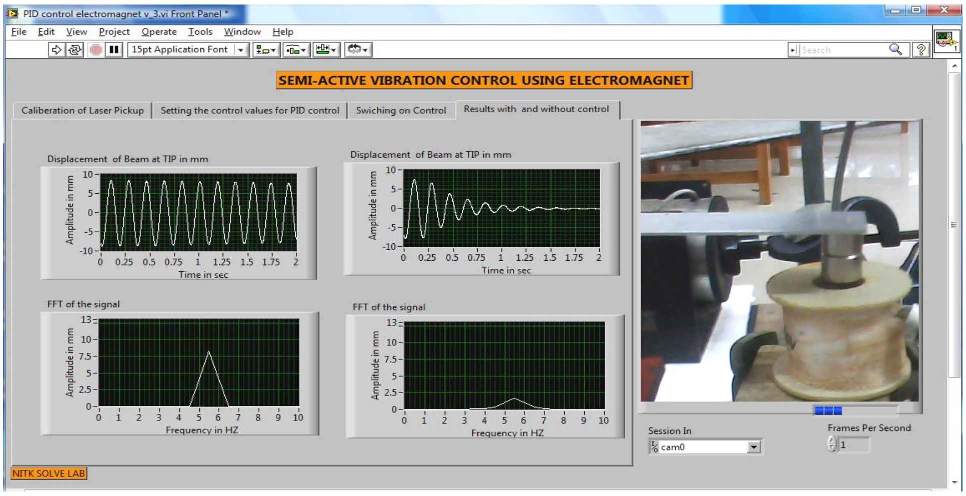

Figure 5. GUI with real time video streaming to visualize vibration control

RESULTS AND DISCUSSION

To predict the vibration characteristics of the cantilever beams with the electromagnetic shunt damper, the electro-magneto- mechanically coupled single mode equations were reformulated in this study. Figure 6 shows the time response of cantilever beam with and without switching semi-active control. A clear reduction in amplitude of vibration was observed. Also, time responses of the cantilever beam under an initial condition of a tip displacement of 8.5 mm at the sensing point were examined under both controlled and uncontrolled conditions as shows in Figure 7. The slow decay of the time response of the uncontrolled beam accelerates through the incorporation of the electromagnetic switching circuit.

Figure 9 shows the experimental frequency domain graph of the controlled and uncontrolled cantilever beams for first mode of beam. The electromechanical coupling matrices were solved using MATLAB and the resulting graphs are shown in Figure 10.

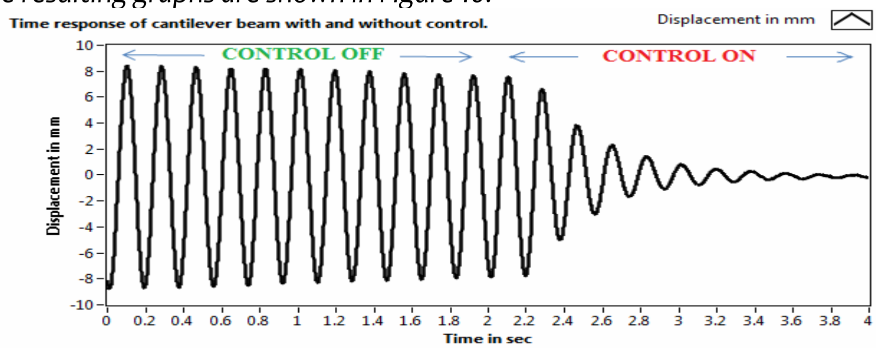

Figure 6. Time response of cantilever beam with and without control

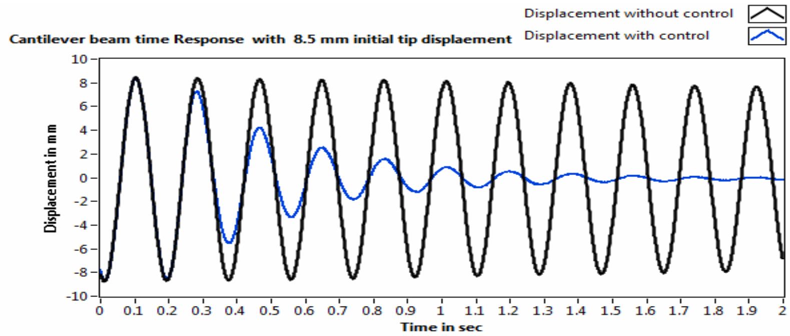

Figure 7. Time response of cantilever beam with initial displacement

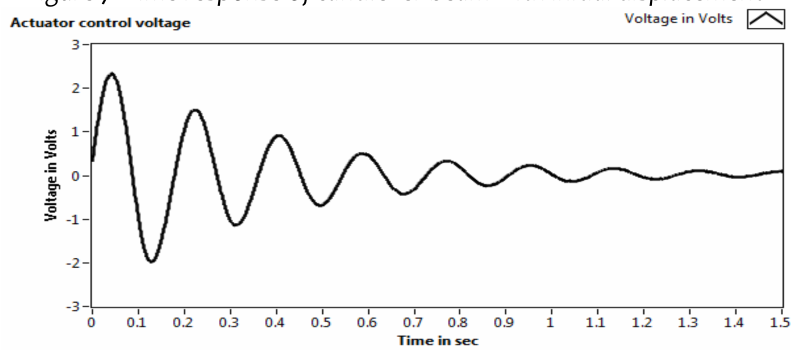

Figure 8.Control voltage with amplification gain equal to one of TECHRON amplifier

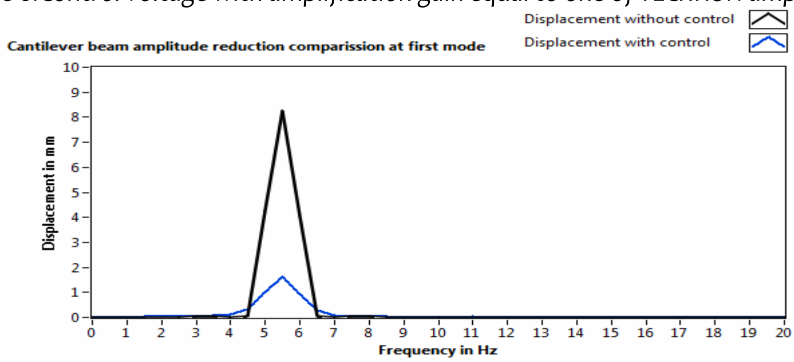

Figure 9. Experimental amplitude reduction with/without electromagnetic semi-active vibration control

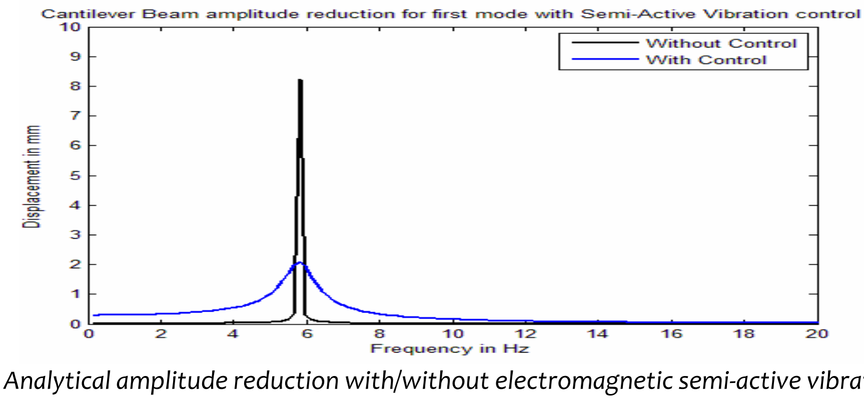

Figure 10. Analytical amplitude reduction with/without electromagnetic semi-active vibration control

The predicted results are consistent with the experimental data. This means that the used analytical model can provide a good solution for the design of the multi-mode electromagnetic shunt damper. From the fine tuning of the PID control, a dramatic reduction of 6.5 mm was obtained for the first mode with peak system disturbance amplitude of 8.5 mm.

CONCLUSIONS

In this study, semi-active vibration control has been achieved using coil based electromagnetic circuit. The advantage of this design is that if active voltage feedback control fails, passive shunt damping will control the vibration. The design was newly constructed for vibration suppression of flexible structures. The used analytical formulation of the electromagnetic switching system provides a good solution for the design of the shunt circuits and the dynamic analysis of the integrated cantilever beam with experimental results. This can be manufactured with large electromagnets on both sides and permanent magnets at the center for semi-active vibration suppression in automobiles. As the results *show that the theoretical data are in good agreement with measured data. One possible method to enhance the system's broadband performance is to use series of coils and permanent magnets with individual switching control with respect to vibration amplitude.*

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