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NONLINEAR DAMPING FOR VIBRATION ISOLATION AND CONTROL USING SEMI ACTIVE METHODS

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ABSTRACT

This paper describes the semi active control of a single as well as two degree-of-freedom (sdof and 2dof) system with a magnetorheological (MR) damper. The investigation was done using a MATLAB/SIMULINK model. The objective of this paper is to present a comprehensive analysis of a SIMULINK model for a control strategy (using a Proportional-Integral Controller) to study the effect of a cubic viscous damping characteristic parameter on the transmissibility of sdof and 2dof vibration isolation systems. This research has substantial deductions for the analysis and design of viscously damped vibration isolators for a broad scope of engineering applications.

Keywords: Vibration control, magnetorheological damper (MR damper), degree-of-freedom (dof), mass-spring-damper system, force transmissibility.

INTRODUCTION

Vibration is the motion of a particle, a body or a system of connected bodies displaced from a state of equilibrium. A large number of vibrations are unwanted in machines and structures because they generate increased stresses, energy losses, cause added wear and absorb energy from the system. The source of the unwanted vibration signals could be natural for example, strong wind or artificial like vibrations from an electric generator (Thureau, 1981). The destructive effect of these signals needs to be suppressed hence the need for a vibration isolation system. Vibrations developing from machineries or some other sources are transferred to a supporting structure such as a facility floor, inducing a damaging environ and undesirable levels of vibration. The aim of vibration isolation is to curb undesirable vibration to keep its harmful effects maintained within tolerable levels.

Vibration suppression/ isolation could be implemented employing one of many methods that have been proposed, which can be classified into several families: passive, semi-passive, semi-active and active control methods (Genta, 2010).

Transmissibility is a concept broadly used as a performance measure for vibration isolation. It is the ratio of the vibration transmitted after isolation to the disturbing vibration or harmonic excitation. If the disturbing vibration is applied at a frequency which coincides with the natural frequency, a condition known as 'resonance' occurs (Lang et al., 2009; Piersol, 2002). Resonance is the tendency of a system to oscillate at greater amplitude, at some frequencies, than at others. These are known as the system's resonant frequencies (Rao, 2010). At these frequencies, even small periodic driving forces can produce large amplitude oscillations, because the system stores vibration energy. In a vibration isolation system, viscous damping is employed in the suppression of vibration amplitude at resonance. However, if the viscous damping has a linear effect, increasing the damping level results to the decrease in the transmissibility of the system at the resonant region but it also causes an increase in the transmissibility above the resonant region. Nonlinear viscous damping which is implemented using semi-active methods is proposed to solve this challenge and an MR damper is used as the controllable damper.

BACKGROUND AND RELATED LITERATURE

Vibratory systems are made up of a part that stores energy in its potential form (spring), a part that stores energy in its kinetic form (mass or inertia) and a part that dissipates energy gradually (damper) and are often referred to as a mass-spring-damper (MSD) system.

Mathematical Model of an MSD system

For a mass-spring-damper system depicted in figure 1, the dynamics of the system can be approximately described using ordinary differential equations as in equation 1 (Inman, 1994).

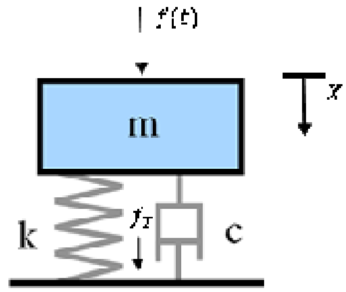

Figure 1. Mass-spring-damper (MSD) system

$$m\ddot{x} + c\dot{x} + kx = f(t)$$

$$f_T = c\dot{x} + kx \qquad (1)$$

Where

m = Mass, c = Damping coefficient, k = Spring stiffness, x = Displacement of mass

$\dot{x}$ = velocity of the mass, $\ddot{x}$ = acceleration of the mass

f(t) = Disturbance or Excitation force, f_T = Force transmited to the base or foundation

Transmissibility Ratios

Absolute transmissibility evaluates how much the isolator has suppressed the transmitted force to the foundation. It is the ratio of the force transmitted at the base of the structure to the force of excitation. The foundation is assumed to be immovable in this case.

Isolation Methods

Karnopp reviewed the historical growth of vibration isolation and also showed how various isolation methods started to mark their presence in the 1970's (Korenev *et al.*, 1993). Some established methods of vibration isolation are briefly reviewed in this section.

Passive Isolation

Passive elements were the major components of previous isolators. This implies no external power source was needed to store or dissipate energy.

Active Vibration Control

Active vibration control is the active application of force in an equal and opposite fashion to the forces imposed by external vibration

Semi-Active Vibration Isolation

The active isolation system is very efficient however the resulting system requires a significant amount of power, like many other active control systems, costs more and is more complex to design (Hrovat *et al.*, 1998; Karnopp, 1974). Another way in attaining same performance is to implement a semi-active isolation system discussed in this paper.

Semi-active vibration isolation systems can be realized by mass control, stiffness control or damping control. Various damper-types applied in engineering applications include: friction dampers; viscoelastic dampers; viscous dampers; tuned liquid dampers; magnetorheological fluid dampers; tuned mass dampers; electrorheological fluid dampers (ER damper); and shape memory alloy dampers. In this study, a magnetorheological fluid damper was used as the controllable damper device.

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Magnetorheological (MR) damper

The MR damper, just like the ER damper, contains a type of smart fluid and comprises a hydraulic cylinder which contains ferromagnetic particles, of micron size, suspended in a fluid (often oil) (Meirovitch, 1986). When exposed to an electric or magnetic field, through a solenoid embedded inside the damper, the MR material (or fluid) has the ability to change from a free flowing viscous fluid to a semi-solid state in a few milli-seconds (Sims *et al.,* 1999; Sims *et al.,* 1996). These devices do not contain moving parts hence are mechanically reliable. Due to the polarizing magnetic field, the particles form chains thereby modifying the value of the oil yield stress resulting to the changes in state of the fluid. Hence, by adjusting the solenoid current, continuously variable damping can be produced without requiring moving parts like variable orifices and valves. Detailed analysis of MR fluids and their properties can be found in Carlson *et al* in (Tanaka, 1992). A schematic diagram of the MR damper is shown in figure 2.

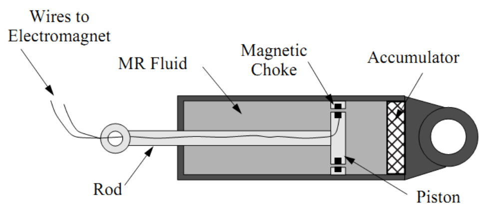

Figure 2. Schematic of an MR damper (Sims, *et al.,* 1999)

SDOF Vibration Isolators with A Linear Damping Characteristic

Consider the sdof vibration isolation system shown in figure 4, where $\mathrm{F_{in}(t) = F_o \sin wt}$ is the harmonic disturbance acting on the system with frequency w and magnitude $\mathrm{F_o}$. $\mathrm{F_{out}(t)}$ is the force transmitted to the foundation assumed to be static and X(t) is the mass displacement (Inman, 1994).

k = Spring stiffness and c = linear damper coefficient.

Assuming the vibration isolation system has a linear spring and damper as that shown in figure 1, then the equations of motion are thus given by;

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = F_{in}(t) = F_o \sin wt = f(t) \tag{2}$$

$$F_{out}(t) = c\dot{x}(t) + kx(t) \tag{3}$$

From basic vibration analysis, the following definitions are given:

$$w_n^2 = \frac{k}{m}, c_c = 2\sqrt{km}, \ \zeta = \frac{c}{c_c} = \frac{c}{2\sqrt{km}} \text{and } 2\zeta w_n = \frac{c}{m} \tag{4}$$

Where w_n = Undamped natural frequency, c_c = Critical damping and ζ = damping ratio.

$$\ddot{x} + 2\zeta w_n \dot{x} + w_n^2 x = \frac{F_o \sin wt}{m} = \frac{f(t)}{m} \tag{5}$$

Force Transmissibility Analysis for SDOF System with Linear Damper

As defined in 2.2, force transmissibility is the ratio of the magnitude of the maximum force transmitted to the base to the magnitude of the maximum input force. It can be expressed as a percentage. A good vibration isolation system should have a low transmissibility ratio, far less than 1 (Fuller *et al.,* 1996).

Considering the *m-c-k* system of figure 4 with C2 = 0 and modifying equations 2 and 3 from its time domain to its frequency domain,

The force transmissibility, *TR* becomes,

$$TR = \frac{F_{out}}{F_{in}} = \frac{c\dot{x} + kx}{m\ddot{x} + c\dot{x} + kx} \tag{6}$$

Transforming into frequency response and taking the magnitude of equation (6);

$$\Rightarrow TR = \left|\frac{F_{out}}{F_{in}}\right| = \frac{\sqrt{1+\left(2\zeta\frac{w}{w_n}\right)^2}}{\sqrt{\left[1-\left(\frac{w}{w_n}\right)^2\right]^2+\left[2\zeta\frac{w}{w_n}\right]^2}} \tag{7}$$

The term $\frac{w}{w_n} = r$, is the non-dimensional excitation frequency ratio. Based on equation (7), a MATLAB simulation plot of transmissibility, *TR* versus frequency ratio, $r = \frac{w}{w_n}$ for damping parameter values of $\zeta = 0.1, 0.2, 0.4$ and 0.7 is shown in figure 3 and it is seen that the frequency ratio has a value of one, $(r = 1)$ where the disturbance or excitation frequency equals the natural frequency of the system (Walsh *et al.,* 1996; Fuller *et al.,* 1996). This region, known as the resonant region, experiences maximum vibration amplitude and as the damping characteristic parameter is increased, the transmissibility in this region is reduced. However, it is also observed that the transmissibility over the region where isolation is desired $(r \gg 1)$, increases. Nonlinear viscous damping, discussed in section 4, was employed to solve this problem.

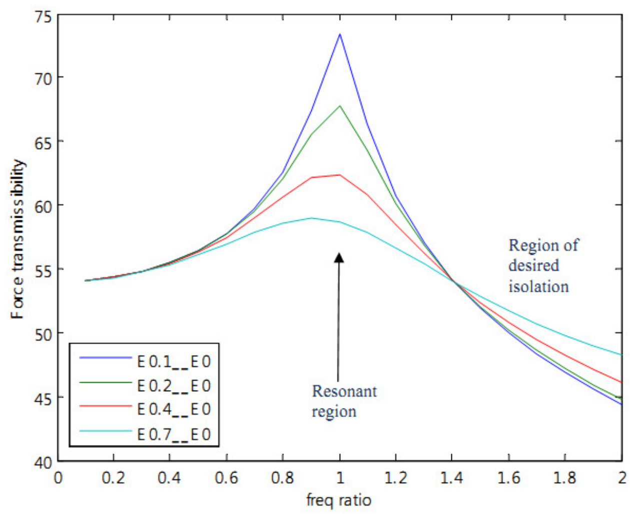

Figure 3. Transmissibility curve for vibration isolation system with linear viscous damping characteristics where $\zeta_1 = 0.1, 0.2, 0.4, 0.7$ and $\zeta_2 = 0$ for all.

SDOF Vibration Isolator with Nonlinear Viscous Damping Characteristics

Consider the sdof vibration isolation system shown in figure 4, where $F_{in}(t) = F_o \sin wt$ is the harmonic disturbance acting on the system with frequency w and magnitude F_o. $F_{out}(t)$ is the force transmitted to the foundation (assumed to be static) and X*(t)* is the mass displacement.

Here k = Spring stiffness, c_1 = linear damper coefficient and c_2 = nonlinear damper coefficient.

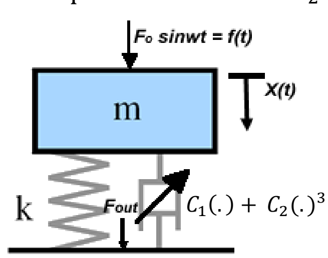

Figure 4. SDOF vibration isolator system with cubic nonlinear viscous damping characteristics

Assuming the vibration isolation system has a linear spring and a cubic nonlinear damper as indicated in figure 4, then the equations of motion of the SDOF vibration isolation system are thus given by (Lang *et al.,* 2009; Fuller *et al.,* 1996);

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$$\left.\begin{aligned} &m\ddot{x}(t) + c_1\dot{x}(t) + c_2[\dot{x}(t)]^3 + kx(t) = F_{in}(t) = F_o \sin wt = f(t) \\ &F_{out}(t) = c_1\dot{x}(t) + c_2[\dot{x}(t)]^3 + kx(t) \end{aligned}\right\} \quad (8)$$

Where k, c_1 and c_2 are the spring and viscous damping characteristic parameters of the *m-c-k* sdof system respectively.

$$\frac{F_{out}(t)}{F_{in}(t)} = \frac{kx(t)+c_1\dot{x}(t)+c_2[\dot{x}(t)]^3}{F_o} = z_1(\tau) + \zeta_1\dot{z}_1(\tau) + \zeta_2[\dot{z}_1(\tau)]^3 = z_2(\tau) \quad (9)$$

Hence, the force transmissibility $TR(\overline{w})$ of the sdof isolation system of equation (9) in terms of the normalized frequency $\overline{w}$ is given as

$$\boldsymbol{TR(\overline{w}) = |Z_2(j\overline{w})|} \quad (10)$$

Where $\boldsymbol{Z_2(j\overline{w})}$ is the spectrum $\boldsymbol{Z_2(j\omega)}$ of the second output of the system (9) computed at frequency $\omega = \overline{w}$. Hence, the transmissibility of the sdof isolation system can be obtained by investigating the spectrum of the second output of system (9). This complex evaluation, which is beyond the scope of this project, has been investigated by Z.Q. Lang et al. (2009).

Z.Q. Lang et al. (2009), by using the concept of the output frequency response function (OFRF) of nonlinear Volterra systems, derived an analytical relationship between $\boldsymbol{Z_2(j\omega)}$ and the system nonlinear viscous damping characteristic parameter, ζ_2. In this work, simulation studies were carried out using a SIMULINK model of the system, to determine the effect of the nonlinear viscous damping characteristic parameter. The simulation result is shown in figure 5.

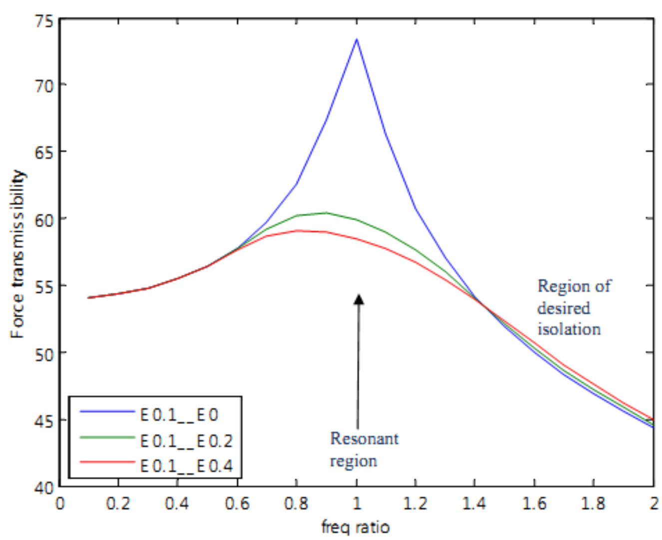

Figure 5. Transmissibility curve for vibration isolation system with nonlinear viscous damping characteristics

where $\zeta_1 = 0.1, \zeta_2 = 0, 0.2$ and 0.4.

Figure 5 shows the transmissibility in cubic nonlinear viscous damping cases for $\zeta_2 = 0, 0.2$ and 0.4 while the linear viscous damping characteristic parameter $\zeta_1 = 0.1$ in all cases. The simulation plot shows that an increase in the nonlinear viscous damping characteristic parameter ζ_2 cannot only suppress the transmissibility over the resonant region but also keep the transmissibility almost unchanged over the region where isolation is desired (Lang *et al.*, 2009; Fuller *et al.*, 1996).

2DOF Vibration Isolator with Nonlinear Viscous Damping Characteristics

Consider the 2dof vibration isolation system shown in figure 9. Just like in the previous sections, where$F_{in}(t) = F_o \sin wt$ is the harmonic disturbance acting on the system with frequency w and magnitude, F_o while $F_{out}(t)$ is the force transmitted to the foundation which is assumed to be static. $X_1(t)$ and $X_2(t)$ are the respective mass displacements for masses, M_1 and M_2.

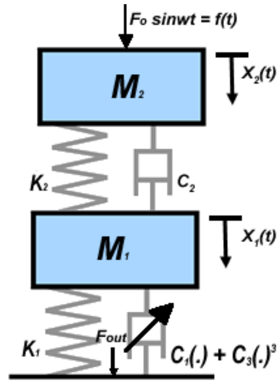

Figure 6. 2DOF vibration isolation system with a cubic nonlinear viscous damping characteristic

Where K_1, K_2 and C_1, C_2, C_3 are the spring stiffness's and viscous damping characteristic parameters of the system, respectively. The equations of motion are derived by resolving the free body diagram of the system.

$$M_2\ddot{X}_2 + C_2(\dot{X}_2 - \dot{X}_1) + K_2(X_2 - X_1) = F_{in}(t) \quad (17)$$

$$M_1\ddot{X}_1 + K_1X_1 + C_1\dot{X}_1 - C_2(\dot{X}_2 - \dot{X}_1) - K_2(X_2 - X_1) = -U(t) \quad (11)$$

Where $U(t) = C_3[\dot{X}_1]^3$

Hence the matrix equation implies;

$$\begin{bmatrix} M_1 & 0 \\ 0 & M_2 \end{bmatrix}\begin{bmatrix} \ddot{X}_1 \\ \ddot{X}_2 \end{bmatrix} + \begin{bmatrix} C_1 + C_2 & -C_2 \\ -C_2 & C_2 \end{bmatrix}\begin{bmatrix} \dot{X}_1 \\ \dot{X}_2 \end{bmatrix} + \begin{bmatrix} K_1 + K_2 & -K_2 \\ -K_2 & K_2 \end{bmatrix}\begin{bmatrix} X_1 \\ X_2 \end{bmatrix} = \begin{bmatrix} -U(t) & 0 \\ 0 & F_{in}(t) \end{bmatrix} \quad (12)$$

Where

$$M = \begin{bmatrix} M_1 & 0 \\ 0 & M_2 \end{bmatrix}, C = \begin{bmatrix} C_1 + C_2 & -C_2 \\ -C_2 & C_2 \end{bmatrix}, K = \begin{bmatrix} K_1 + K_2 & -K_2 \\ -K_2 & K_2 \end{bmatrix}, X = \begin{bmatrix} X_1 \\ X_2 \end{bmatrix}$$

and

$$f(t) = \begin{bmatrix} -U(t) & 0 \\ 0 & F_{in}(t) \end{bmatrix}$$

Obtaining the state-space parameters; **A = [-M\C -M\K; eye(2,2) zeros(2,2)]; B = [inv(M); zeros(2,2)]; Cs = [zeros(2,2) eye(2,2)]; D = zeros(2,2)**

The SIMULINK model is shown in figure 7.

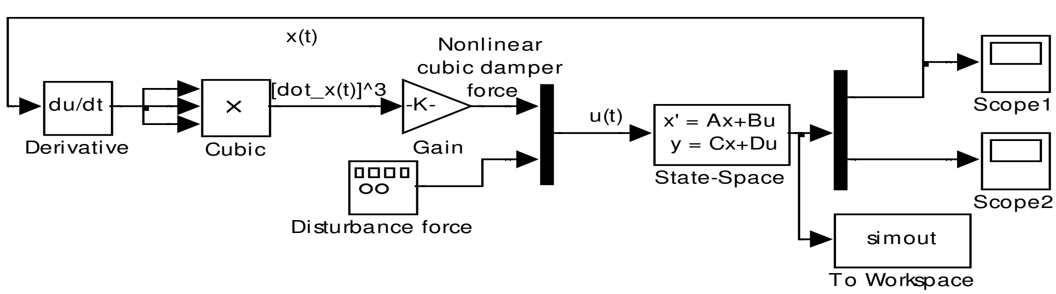

Figure 7. SIMULINK model of 2dof vibration isolation system with a nonlinear cubic damper

Considering figure 6, the force transmissibility required is given as the ratio of the force transmitted to the foundation of the system, to the force transmitted into mass, M_1. Therefore

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$$TR = \frac{F_{out}}{F_{in}2} = \frac{K_1X_1 + C_1\dot{X}_1 + U(t)}{K_2(X_2 - X_1) + C_2(\dot{X}_2 - \dot{X}_1)} \tag{13}$$

Where $F_{in}2 = K_2(X_2 - X_1) + C_2(\dot{X}_2 - \dot{X}_1)$ is the force transmitted to the mass M_1 and

$F_{out} = K_1X_1 + C_1\dot{X}_1 + U(t)$ is the force transmitted to the foundation/ base.

For $U(t) = 0$, a linear damper emerges and the force transmissibility plot is as indicated in figure 3. When $U(t)$ exists, a cubic nonlinear damper exists and figure 5 illustrates the simulation plot obtained (Lang *et al.,* 2009).

Semi-Active Implementation using MR Damper

The cubic nonlinear viscous damping characteristic was implemented using the MR damper whose current input was controlled with a PI (Proportional-Integral) controller. A nonparametric model of the MR damper was simulated and tested for a range of current inputs. Simulation plots were generated for damper force time trace, force-velocity and force-displacement data. These plots were used to validate the MR damper model designed (Liu *et al.,* 2005; Song *et al.,* 2005).

Nonparametric Model Development of an MR Damper

Four mathematical functions were suggested by X. Song et al. to obtain the predominant parts of MR dampers. The MR damper model is made up of the following functions (Liu *et al.,* 2005; Song *et al.,* 2005):

- A polynomial function such as
  $$A_{mr}(I) = \sum_{i=0}^{n} a_i I^i \tag{14}$$

which describe the maximum damping force as a function of the current applied, where A_{mr} = Maximum damping force, a_i = Polynomial coefficients with appropriate units, n = Order of the polynomial, I = Current applied to the MR damper

- A shape function used to maintain the ensued wave-shape correlation between the damping force and relative velocity of the damper. This function likewise represents the bilinear behaviour of the force-velocity curve (Song *et al.,* 2005).
  $$S_b(V) = \frac{(b_0 + b_1|V - V_0|)^{b_2(V - V_0)} - (b_0 + b_1|V - V_0|)^{-b_2(V - V_0)}}{{b_0}^{b_2(V - V_0)} + {b_0}^{-b_2(V - V_0)}} \tag{15}$$

Where b_0, b_1, b_2 are constants with $b_0 > 1, b_1 > 0$ and $b_2 > 0$.

V = Velocity across MR damper and V_0 = Constant. Combining equations (14) and (15) gives the damper force as a function of MR damper current and relative velocity. Hence,

$$F_s = A_{mr}(I)S_b(V) \tag{16}$$

- A delay function (a first-order filter) is used to generate the hysteresis loop. The filter is developed in its state space form as
  $$\dot{x} = -(h_0 + h_1I + h_2I^2)x + h_3F_s$$
  $$F_h = (h_0 + h_1I + h_2I^2)x + h_4F_s \tag{17}$$

Where x = State variable of the filter, h_i ($for\ i$ = 0-2) = Constants, h_i ($for\ i$ = 3-4) = constants

or functions of current, I = Current applied to the MR damper as defined in equation (17), F_h combines the damping force F_s shown in equation (14) and the hysteresis function (Song *et al.,* 2005).

- An offset function could be required in some cases if the damping force misses the zero centre mark due to the effect of the gas-charged accumulator in the damper. A force bias is hence included in the model such as
  $$F_{mr} = F_h + F_{bias} \tag{18}$$

Where F_{bias} = Nonzero centered damping force due to the accumulator, F_{mr} = MR damping force.

Combining the four functions mentioned above provides the nonparametric model as given in equation (18). A selection of the model parameters and evaluation of the model accuracy was investigated by X. Song et al. and the optimal parameter values obtained are given in table 1.0. Simulation plots for damper force-time trace, force-velocity and force-displacement curves are given in figures 8, 9 and 10 respectively. The damper model simulation curves were obtained for constant current inputs, $i(A)$ = 0, 0.25, 0.5, 1, 1.5 and 2.

Table 1. Optimal values of the magnetorheological (MR) damper model (Song *et al.*, 2005)

Parameter	Value	Parameter	Value	Parameter	Value
A_0	164.8	b_0	5.8646	h_2	566.0
A_1	1316.5	b_1	0.0060	h_3	1
A_2	1407.8	b_2	0.2536	h_4	0
A_3	- 1562.8	h_0	299.7733	V_0	0.6248
A_4	388.8	h_1	-210.320	F_{bias}	0

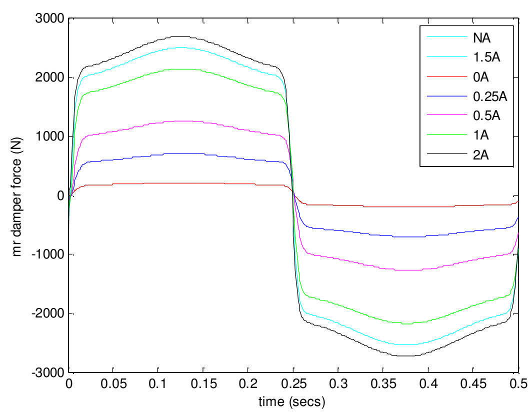

Figure 8. MR damper force-time trace

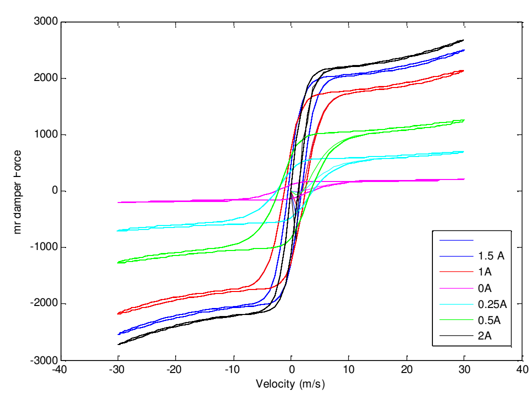

Figure 9. MR damper force-velocity curve

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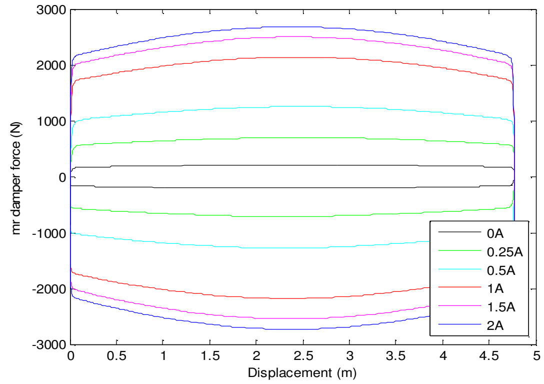

Figure 10. MR damper force-displacement curve

Nonlinear Damping Characteristics Implemented with an MR Damper

The cubic nonlinear damper characteristic was implemented using the MR damper in a closed loop system as illustrated in figure 11 (Spencer *et al.*, 1996; Song *et al.*, 2005). The displacement X_1 of mass M_1 was constantly measured using a sensor and its derivative taken to obtain its velocity, $\dot{X}_1$. The desired damper force was generated using velocity, $\dot{X}_1$. Hence,

Desired damper force, $F_d = C_d * [\dot{X}_1]^3$ (19)

Where C_d = Desired Damping coefficient and $\dot{X}_1$ = Velocity of mass M_1

The actual MR damper force was measured using a force sensor and compared with the desired force, F_d. A PI controller was used to minimize the error (difference) of comparison to zero (Song *et al.*, 2005). The output of the controller supplies the MR damper current input, I amps. In the actual experimental setup, the current was applied to the MR damper using a solenoid.

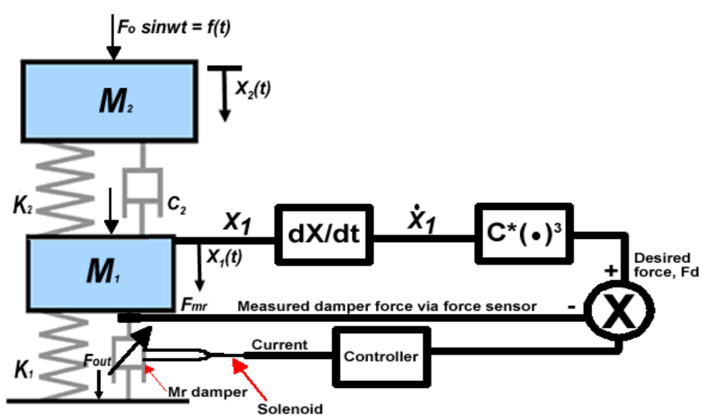

Figure 11. Schematic of a nonlinear damping characteristic implemented using an MR damper

The cubic nonlinear damping force (desired force) was tracked by adjusting the current input to the MR damper. The current became zero as soon as the desired force was achieved by the MR damper. The MR damper velocity is same as the velocity of mass M_1 and was used in the design of the SIMULINK model for the MR damper.

Force Transmissibility with Cubic Nonlinear Viscous Damper Model

The force transmissibility simulation curve of the vibration isolation system shown in figure 12 demonstrates the effects of nonlinear viscous damping on vibration isolation. This indicates that an increase in the nonlinear viscous damping characteristic parameter $\zeta_{3rms} = 0, 0.2, 0.4$ *and* 0.7, does not only reduce the transmissibility as well as suppress the vibration at the resonant frequency but also maintain the transmissibility at frequency ranges greater than and less than the resonant frequency (Lang *et al.*, 2009).

Force Transmissibility with MR Damper Model of Cubic Nonlinear Viscous Damping Characteristic

Same cubic nonlinear viscous damping characteristic was modelled but using an MR damper. The MR damper force was made to track the desired cubic nonlinear viscous damping characteristic and was controlled using a PI controller which fed the MR damper through a solenoid. The MR damper force can be adjusted to track any desired viscous damping characteristic. It is shown in figure 13 how much, in terms of the transmissibility of the vibration isolation system, the MR damper modelled the cubic nonlinear viscous damping force. Just like in section 4, an increase in the nonlinear viscous damping characteristic parameter $\zeta_{3rms} = 0, 0.2, 0.4$ and 0.7, does not only reduce the transmissibility as well as suppress the vibration at the resonant frequency but also maintain the transmissibility at frequency ranges greater than and less than the resonant frequency (Lang et al., 2009).

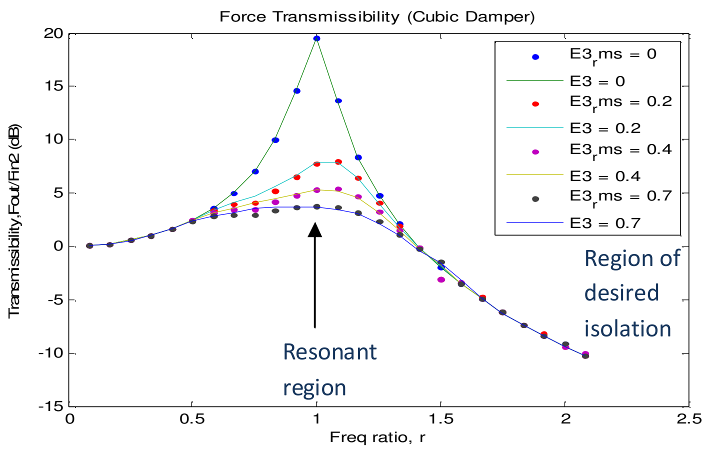

Figure 12. RMS Force transmissibility plot using cubic nonlinear viscous damper model

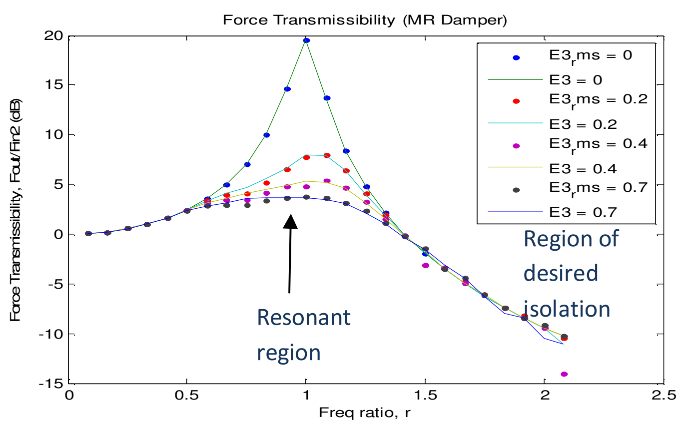

Figure 13. RMS Force transmissibility plot using an MR damper

RESULTS

In this section, the results of the simulation studies for the force transmissibility curve when an MR damper model was used to implement the desired cubic nonlinear viscous damping force characteristics, is analyzed and discussed. It was compared with the force transmissibility curve obtained when the desired cubic damping force in its mathematical model form, was used. The simulation plots are shown in figures 12 and 13. It is observed that both simulation plots are almost identical which was as a result of the close tracking of the cubic nonlinear viscous damping force by the MR damper force. As illustrated in figure 11, the error difference between the desired cubic nonlinear damping force and the MR damper force was continuously minimized to zero by the PI controller as the MR damper force approaches the value of the desired cubic nonlinear viscous damping force (Spencer *et al.,* 1996). At that instance, the MR damper force emulates the

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characteristics of the desired cubic nonlinear viscous damper force. This result is favourable since a semi-active implementation (using the MR damper) of the cubic nonlinear viscous damping force was achieved. The benefits of nonlinear viscous damping characteristics were highlighted and illustrated with simulation studies. Hence, with the results of the simulation studies, it was deduced that implementation of nonlinear damping for vibration isolation by semi-active methods were feasible using a controllable viscous damper like the MR damper (Liu *et al.,* 2005).

CONCLUSION AND RECOMMENDATION

Vibration isolation is a crucial problem for a broad scope of engineering applications. Conventionally, the design of a vibration isolation system involves the determination of the spring stiffness as well as the damping coefficient of the linear viscous damper. These characteristic parameters basically make up a passive vibration isolation system. A commonly known problem associated with passive vibration isolators is that although the presence (and increase) of the linear viscous damping effect can greatly suppress the transmissibility over the resonant region of the system however, this has got detrimental effects on the range of normal working frequencies of the system where isolation is desired. This problem has been offered solutions using active vibration control methods but it also comes with increased complexities, cost and high power demand.

The problems of active vibration isolation methods have been addressed by a series of studies conducted by Z.Q. Lang et al. (1996). The use of nonlinear viscous damping, investigated by Z.Q. Lang et al. (1996) and nonlinear approach to resonance suppression investigated by S.A Billings et al. (1996) provides an efficient passive solution. This present study concerns the implementation of the nonlinear viscous damping characteristics using semi-active vibration control methods using an MR damper. The MR damper's viscous damping characteristic was controlled using a PI controller and regulated to meet desired nonlinear viscous damping characteristics. This implies that a vibration isolation system with a nonlinear viscous damping characteristic implemented with an MR damper (Semi-active means) can also achieve the desired amount of vibration isolation as an active control system. Simulation studies were used to show the validity and engineering benefits of the proposition. The conclusions reached by this study have important implications for the engineering design of semi-active vibration isolation systems in a broad range of applications.

More investigations on the effect of more complex nonlinear viscous damping characteristics on vibration isolation could be carried out to improve on the results achieved in this study (Lang *et al.,* 1996). The MR damper can be generally modelled for any desired nonlinear viscous damping characteristics hence making semi-active methods a better performing, more beneficial and effective method.

REFERENCES

Beards, C.F. (1996). *Structural Vibration: Analysis and Damping,* 3rd Edition, New York: John Wiley & Sons.

Franchek, M.A., Ryan, M.W. and Bernhard, R.J. (1995). Adaptive-passive Vibration Control, *Journal of Sound and Vibration*, vol. 189, p.565-585.

Fuller, C.R., Elliot, S.J. and Nelson, P.A. (19966). Active Control of Vibration, Academic Press.

Genta, G. (2010). *Vibration Dynamics and Control* 2nd edition, Springer, Croatia.

Hrovat, D., Margolis, D.L. and Hubbard, M. (1998). An approach toward the optimal semi-active suspension, *Journal of Dynamic Systems, Measurement and Control* vol. 110, p.288-296.

Inman, D.J. (1994). *Engineering Vibration,* 2nd Edition, Prentice Hall.

Karnopp, D. (1990). Design principles for vibration control systems using semi-active dampers, *ASME Journal of Dynamic Systems, Measurement and Control*, vol. 112, p.448-455.

Karnopp, D.C. and Crodby, M.J. (1974). Vibration control using semi-active force generators, *ASME Journal of Engineering for Industry*, vol. 96, p.619-626.

Korenev, B.G. and Reznikov, L.M. (1993). *Dynamic Vibration Absorbers: Theory and Technical Applications,* John Wiley & Sons.

Lang, Z.Q. Jing, X.J. Billings, S.A. Tomlinson, G.R. and Peng, Z.K. (2009). Theoretical study of the effects of nonlinear viscous damping on vibration isolation of sdof systems, *Journal of Sound and Vibration,* vol. 323 p.352-365.

Liu, Y., Waters, T.P. and Brennan, M.J. (2005). A comparison of semi-active damping control strategies for vibration isolation of harmonic disturbances, *Journal of Sound and Vibration,* vol. 280 , p.21-39.

Meirovitch, L. (1986). *Elements of Vibration Analysis,* 2nd edition, McGraw-Hill.

Olgac, N. and Holm-Hansen, B. (1994). A novel active vibration absorption technique: Delayed resonator, *Journal of Sound and Vibration*, vol. 176, p.93-104.

Piersol, A. and Paez, T. (2002). *Harris' Shock and Vibration Handbook,* 6th Edition, New York: McGraw-Hill, Inc.

Rao, S.S. (2010). *Mechanical Vibrations* 5th edition*,* Prentice Hall.

Richard, C., Guyomar, D., Audigier, D. and Bassaler, H. (2000). Enhanced semi-passive damping using continuous switching of a piezoelectric device on an inductor, *Smart Structures and Materials: Damping and Isolation,* Proc. SPIE Vol. 3989, 288-299.

Sims, N.D., Stanway, R. and Johnson, A.R. (1999). Vibration control using smart fluids: a state-of-the-art review, *The Shock and Vibration Digest*, vol. 31, p.195-204.

Song, X., Ahmadian, M. and Southward, S.C. (2005). Modeling Magnetorheological Dampers with Application of Nonparametric Approach, *Journal of Intelligent Material Systems and Structures*, vol. 16, p.421-432.

Spencer, B.F. Dyke, S.J. Sain, M.K. and Carlson, J.D. (1996). Modeling and Control of Magnetorheological Dampers for Seismic Response Reduction, *Smart Materials and Structures*, vol. 5, p.565-575.

Spencer, B.F., Dyke, S.J., Sain, M.K. and Carlson, J.D. (1996). Phenomenological Model of a Magnetorheological Damper, *ASCE Journal of Engineering Mechanics*, vol. 123, vol. 230-238.

Tanaka, N. and Kikushima, Y. (1992). Impact Vibration Control Using A Semi-active Damper, *Journal of Sound and Vibration,* vol. 158, p.277-292.

Thureau, P. and Lecter, D. (1981). *An Introduction to the Principles of Vibrations of Linear Systems,* UK: Nelson Thornes Ltd.

Walsh, P.L. and Lamnacusa, J.S. (1992). A variable stiffness vibration absorber for minimization of transient vibrations, *Journal of Sound and Vibration*, vol.158 p.195-211.

Zhang, B., Billings, S.A., Lang, Z.Q. and Tomlinson, G.R. (1996). A novel nonlinear approach to suppress resonant vibrations, *Journal of Sound and Vibration,* vol.317, p.918-931.