23rd ABCM International Congress of Mechanical Engineering
December 6-11, 2015, Rio de Janeiro, RJ, Brazil

VIBRATION CONTROL OF A LIFT SYSTEM USING MAGNETORHEOLOGICAL DAMPER

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Abstract. *In this work, the horizontal response of a vertical transportation with nonlinearities under excitation by guide rail deformations is analyzed. In order to improve the comfort of passengers, a control strategy based on optimal linear feedback control using MR damper is used. The control force of the damper is a function of the voltage applied in the coil of the MR damper that is based on the force given by the controller. Numerical simulations are performed in order to study the nonlinear behavior of the adopted mathematical model. In addition the robustness of the proposed control technique is tested considering parametric errors and noise measurement. The results confirm that the control strategy using MR damper can be effective in reduce the vibration of the vertical transport.*

Keywords: *Optimal feedback control, Vertical transportation, MR damper*

1. INTRODUTION

High-speed elevators became a necessity, however, high speeds can result in decreased quality of ride (Mitsui and Nara, 1971), this being one of the main problems in the high-speed elevators systems (Nai *et al*., 1994). Thus, it is necessary improve the travel quality, without losing the efficiency of elevators and there are limits of the levels of horizontal and vertical vibrations, lateral and longitudinal acceleration and jerk, necessary to ensure good ride quality to passengers (Fortune, 1997).

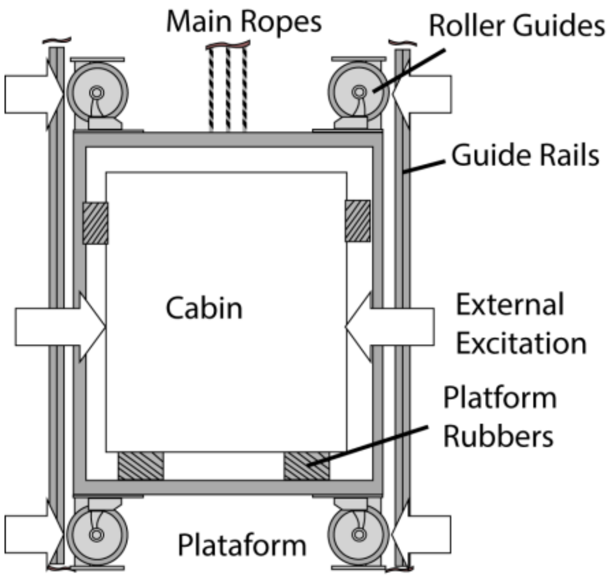
Motivated by necessity to improve passenger's comfort level, this work presents a form of the controlling the dynamic horizontal behavior of a three degrees of freedom model of a vertical transportation system excited through guide rail deformations by a controlled magnetorheological damper.

For the use of MR damper, in the suppression of unwanted oscillations variable, which it can be controlled is the electrical current or voltage, which changes the viscosity of the fluid's internal damper. The damping force will depend on the speed of the piston of the damper and the density of internal fluid.

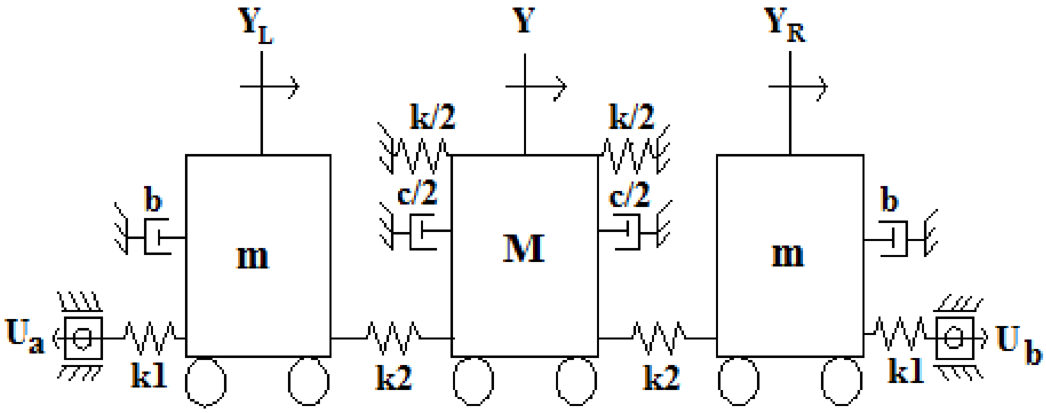
The feedback of state variables by means of sensors can provide information about the behavior of the controlled system, over time and thereby on the optimal intensity of force being applied by the MR damper. Here, we will utilize a mathematical model to transform the values of force control in electric current signal proposed by (Tusset *et al*., 2012).

2. MATHEMATICAL MODELING

Figure 1a shows the schematic diagram of the cabin elevator, where the roller guides support the platform with springs to avoid the transmission of the external excitation caused by misalignment and deformation of guide rails. And figure 1b shows an equivalent physical model to represent the horizontal motions of an elevator system.



(a)



(b)

Figure 1. (a) Schematic diagram of cabin structure (adapted from Funai, 2004). (b) Equivalent model for the horizontal motion of the elevator (adapted from Lopez et *al.*, 2010).

In this model, M is the mass of the cabin [kg], m is the mass of the suspension system [kg], b is the damping coefficient of the suspension [N.s/m], c is the damping coefficient of the cabin [N.s/m], k_1 is the stiffness coefficient of the guide rollers [N/m], k_2 is the stiffness coefficient of the suspension [N/m], k is the stiffness coefficient of the spring equivalent to tilting motion of the cabin [N/m], Y is the displacement of the cabin [m], Y_L is the displacement of the left suspension system [m], Y_R is the displacement of the right suspension system [m]. U_a and U_b are external excitations caused by guide rails deformations, defined in Eq. (1).

$$U_a = U_b = a\sin(\omega t) \tag{1}$$

where: a is the amplitude of external excitation [m], ω is the external excitation frequency [rad/s].

The equations of motion that represent the vertical transportation system can be represented by:

$$
\begin{aligned}
&m\ddot{Y}_L + b\dot{Y}_L + (k_1 + k_2)Y_L - k_2 Y = k_1 U_a \\
&M\ddot{Y} + c\dot{Y} + 2k_2 Y + kY^3 - k_2 Y_L - k_2 Y_R = 0 \\
&m\ddot{Y}_R + b\dot{Y}_R + (k_1 + k_2)Y_R - k_2 Y = k_1 U_b
\end{aligned}
\tag{2}
$$

Or in space of estate:

$$
\begin{aligned}
&\dot{x}_1 = x_2 \\
&\dot{x}_2 = -\alpha_2 x_1 - \alpha_1 x_2 + \alpha_3 x_3 + \alpha_4 U_a \\
&\dot{x}_3 = x_4 \\
&\dot{x}_4 = \beta_3 x_1 - \beta_2 x_3 - \beta_5 x_3^3 - \beta_1 x_4 + \beta_4 x_5 \\
&\dot{x}_5 = x_6 \\
&\dot{x}_6 = \alpha_3 x_3 - \alpha_2 x_5 - \alpha_1 x_6 + \alpha_4 U_b
\end{aligned}
\tag{3}
$$

where: $x_1 = Y_L$, $x_2 = \dot{Y}_L$, $x_3 = Y$, $x_4 = \dot{Y}$, $x_5 = Y_R$, $x_6 = \dot{Y}_R$, $\alpha_1 = \frac{b}{m}$, $\alpha_2 = \frac{(k_1 + k_2)}{m}$, $\alpha_3 = \frac{k_2}{m}$, $\alpha_4 = \frac{k_1}{m}$, $\beta_1 = \frac{c}{M}$, $\beta_2 = \frac{2k_2}{M}$, $\beta_3 = \frac{k_2}{M}$ $\beta_4 = \frac{k_2}{M}$ and $\beta_5 = \frac{k}{M}$.

Consider, now, the introduction of the MR damper controlled in parallel with the spring roller guides as shown in figure 2.

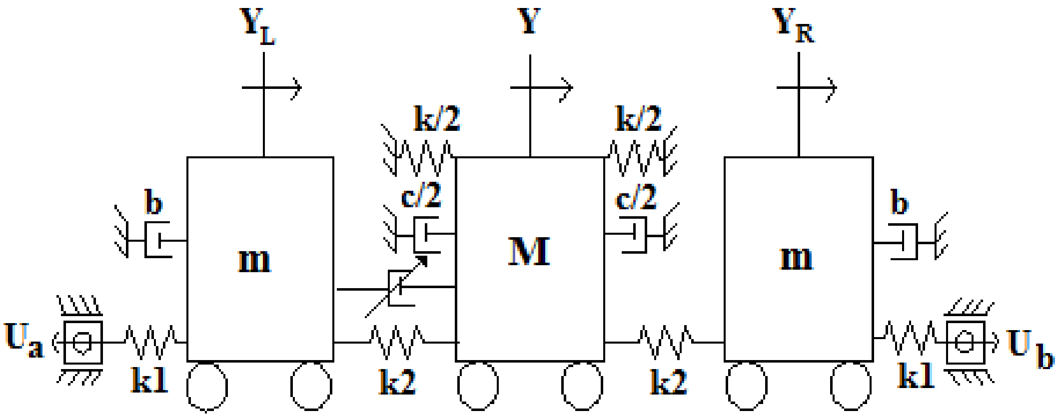


Figure 2. Equivalent model for the horizontal motion of the elevator whit MR damper.

The system (3) with the introduction of active control can be represented by the following system:

$$
\begin{aligned}
&\dot{x}_1 = x_2 \\
&\dot{x}_2 = -\alpha_2 x_1 - \alpha_1 x_2 + \alpha_3 x_3 + \alpha_4 U_a + \delta_1 u \\
&\dot{x}_3 = x_4 \\
&\dot{x}_4 = \beta_3 x_1 - \beta_2 x_3 - \beta_5 x_3^3 - \beta_1 x_4 + \beta_4 x_5 - \delta_2 u \\
&\dot{x}_5 = x_6 \\
&\dot{x}_6 = \alpha_3 x_3 - \alpha_2 x_5 - \alpha_1 x_6 + \alpha_4 U_b
\end{aligned}
\tag{4}
$$

where: u represent the MR damper force, $\delta_1 = \frac{1}{m}$ and $\delta_2 = \frac{1}{M}$.

Figure 6 it shows the main characteristics of the variation of the force of the damper in function of both velocity and electric current applied to the coil of the damper, noting that is the force [N], (i) is the electric current [A] and v is the velocity of the displacement of the piston of the damper [m/s] (McManus *et al*., 2002; Tusset *et al*., 2012).

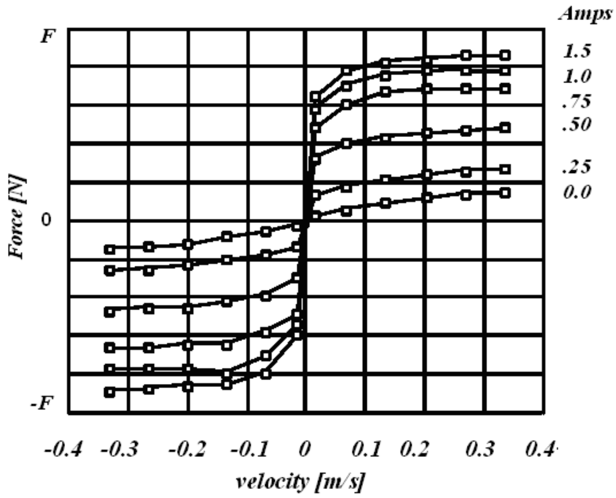


Figure 3. Characteristics of the normalized force-velocity of a MR damper in function of the current (Tusset *et al*., 2012)

Considering of the force normalized and that the velocity will be bounded on $v \le 0.4$ [m/s], the electrical current to be applied, can be determined by solving numerically the following function:

$$C = \begin{cases} f(i)|v|^{-g(i)} \tanh(0.43v) - F, & if \quad v \neq 0 \\ 0, \; if \quad v = 0 \end{cases} \tag{5}$$

Where: C is the electrical current applied voltage in the damper coil, F is the estimate force applied in MR damper, $f(i)$ and $g(i)$ are opted by equations:

$$f(i) = \frac{2.6}{(3.7058e^{-3.2934i}) + 1} \tag{6}$$

$$g(i) = \frac{0.9}{(1.1548e^{-6.8239i}) + 1} \tag{7}$$

3. DETERMINATION OF FORCE OF THE MR DAMPER BY OPTIMAL CONTROL

Considering the system (4) in the following way:

$$\dot{\mathbf{x}} = \mathbf{Ax} + \mathbf{Bu} + \mathbf{GU} \tag{8}$$

where:

$$\mathbf{A} = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ -\alpha_2 & -\alpha_1 & \alpha_3 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ \beta_3 & 0 & -\beta_2 & -\beta_1 & \beta_4 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & \alpha_3 & 0 & -\alpha_2 & -\alpha_1 \end{bmatrix}, \; \mathbf{x} = \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \\ x_5 \\ x_6 \end{bmatrix}, \mathbf{B} = \begin{bmatrix} 0 \\ \delta_1 \\ 0 \\ -\delta_2 \\ 0 \\ 0 \end{bmatrix} \text{ and } \mathbf{G} = \begin{bmatrix} 0 \\ \alpha_4 U_a \\ 0 \\ -\beta_5 x_3^2 \\ 0 \\ \alpha_4 U_b \end{bmatrix}.$$

The control of the damping force can be obtained through the LQR control, given by:

$$\mathbf{u} = -\mathbf{R}^{-1}\mathbf{B}^{\mathrm{T}}\mathbf{Px} \tag{9}$$

where: $\mathbf{Q}$ and $\mathbf{R}$ are positive definite matrices, the matrix $\mathbf{P}$ is obtained of the solution of the Riccati equation:

$$\mathbf{PA} + \mathbf{A}^T\mathbf{P} - \mathbf{PBR}^{-1}\mathbf{B}^T\mathbf{P} + \mathbf{Q} = \mathbf{0} \tag{10}$$

The using of the control (9) minimizes the functional:

$$J = \frac{1}{2}\int_{t_0}^{\infty}\left[\mathbf{x}^T\mathbf{Q}\mathbf{x} + \mathbf{u}^T\mathbf{R}\mathbf{u}\right]dt \tag{11}$$

The minimization of the functional (11) implies the minimization of the states $\mathbf{x}$, and of the force ($\mathbf{u}$) applied by MR damper.

4. NUMERICAL SIMULATION

Table 1 show the parameters experimentally obtained, used for numerical simulations.

Table 1 show the parameters used for numerical simulations (Rivas and Perondi, 2010; Santo *et al.*, 2014).

Parameters	Unit	Description	Value
M	kg	Mass of the cabin	1120
m	kg	Mass of the suspension system	17.5
b	N.s/m	Damping coefficient of the suspension	668.21
c	N.s/m	Damping coefficient of the cabin	2058.2
k_1	N/m	Stiffness coefficient of the guide rollers	250000
k_2	N/m	Stiffness coefficient of the suspension	19027
k	N/m	Stiffness coefficient of the spring equivalent to tilting motion of the cabin	19027
a	m	External excitation amplitude	0.01
ω	rad/s	External excitation frequency	31.4159

The control signal $\mathbf{u}$ is determined using the matrices $\mathbf{A}$ and $\mathbf{B}$, and defining the positives definite matrices $\mathbf{Q}$ and $\mathbf{R}$:

$$\mathbf{Q} = 10^5\begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1000000 & 1 & 0 & 0 \\ 0 & 0 & 1 & 1000000 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \text{ and } \mathbf{R} = 10^{-1}.$$

Substituting the matrices $\mathbf{A}$, $\mathbf{B}$, $\mathbf{Q}$ and $\mathbf{R}$ in equation (10), we obtain the matrix $\mathbf{P}$:

$$\mathbf{p} = \begin{bmatrix} 1.9048(10^7) & 784.5849 & 1.9345(10^6) & 8.6144(10^5) & 8.0549(10^4) & -1.5830(10^4) \\ 784.5849 & 1.3293(10^3) & 7.6146(10^4) & 8.2427(10^4) & 1.9614(10^4) & 50.5085 \\ 1.9345(10^6) & 7.6146(10^4) & 1.0018(10^{11}) & 1.1260(10^8) & 1.6852(10^6) & 640.4774 \\ 8.6144(10^5) & 8.2427(10^4) & 1.1260(10^8) & 1.1716(10^8) & 3.3662(10^6) & 6.7063(10^3) \\ 8.0549(10^4) & 1.9614(10^4) & 1.6852(10^6) & 3.3662(10^6) & 2.1187(10^7) & 2.5678(10^3) \\ -1.5830(10^4) & 50.5085 & 640.4774 & 6.7063(10^3) & 2.5678(10^3) & 1.3755(10^3) \end{bmatrix}$$

Substituting the matrices $\mathbf{R}$, $\mathbf{B}$ and $\mathbf{P}$ in equation (9), we obtain the control u:

$$u = 7243.1294x_1 - 23.6365x_2 + 961849.1073x_3 + 998972.4480x_4 + 18847.0373x_5 + 31.0160x_6 \tag{12}$$

Figures in (4) and (5) can observe the behavior of the elevator car, considering the speed of the piston MR damper obtained from ($v = x_4 - x_2$) and the force of variation MR damper obtained through the control (12), considering the variation the current applied to the coil of MR damper obtained from equation (5).

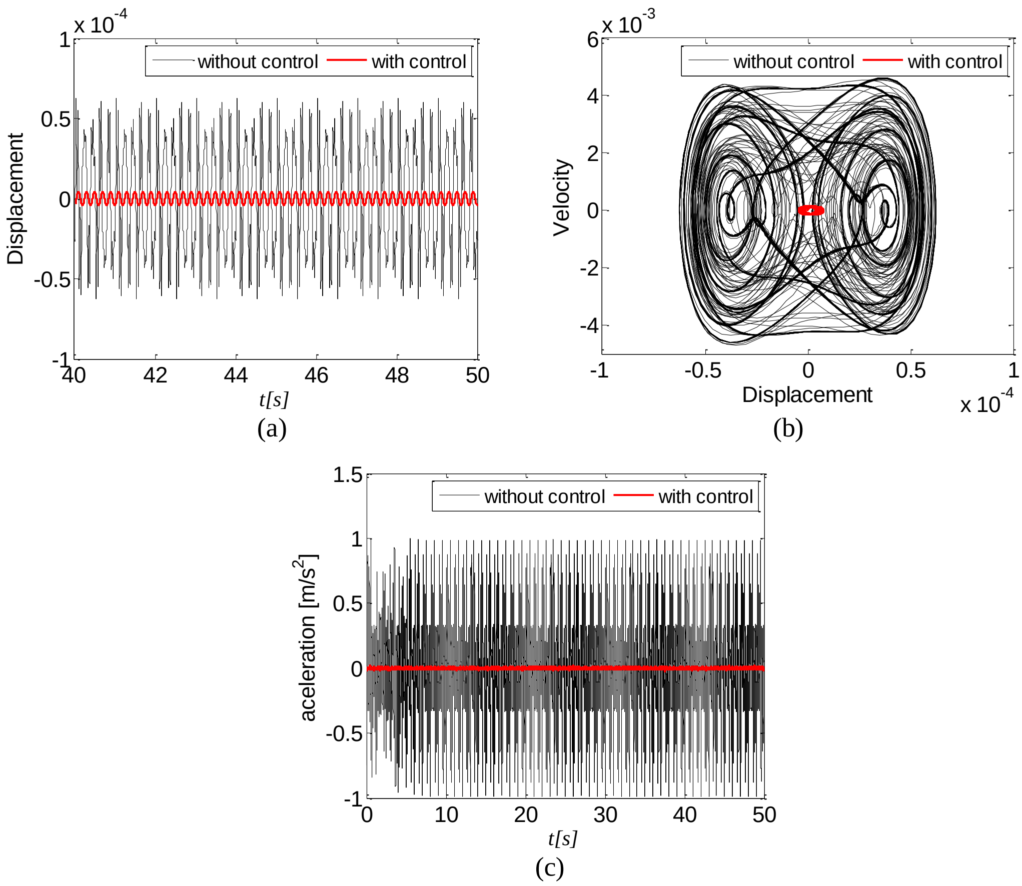
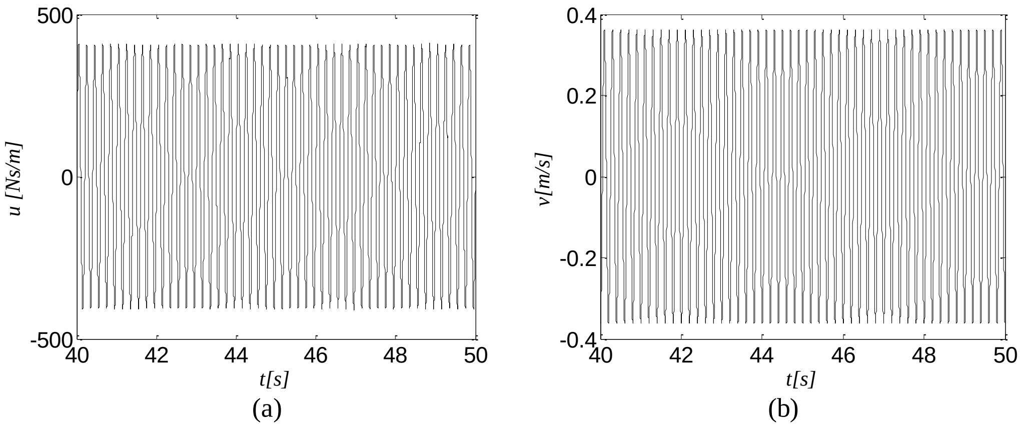
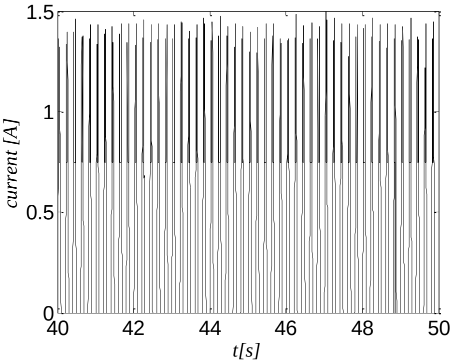


Figure 4. Comparison between uncontrolled (black) and controlled using MR damper (red) cabin response (x_3, x_4 and $\dot{x}_4$). (a) History of the displacement (x_3). (b) Phase plane (x_3 and x_4). (c) History of acceleration ($\dot{x}_4$)

It is possible to observe that the proposed control by MR damper reduced the system oscillation amplitude, figures 4a and 4c. The rms of the magnitude acceleration peak has been reduced in about 98.6367% figure 4c, reducing uncontrolled system acceleration ($\ddot{x}_{3_{rms}} = 0.2714$) for acceleration with control ($\ddot{x}_{3_{rms}} = 0.0037$).





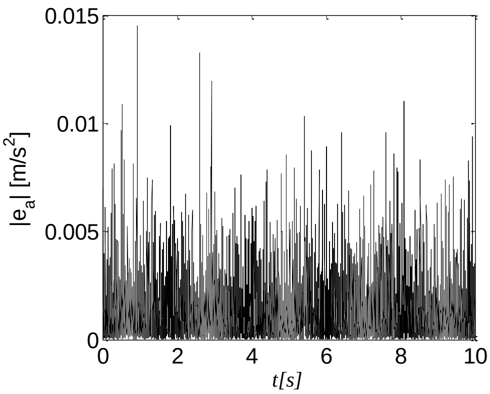
(c)

Figure 5. (a) Force control *u*. (b) Velocity *v*. (c) The current applied in the damper.

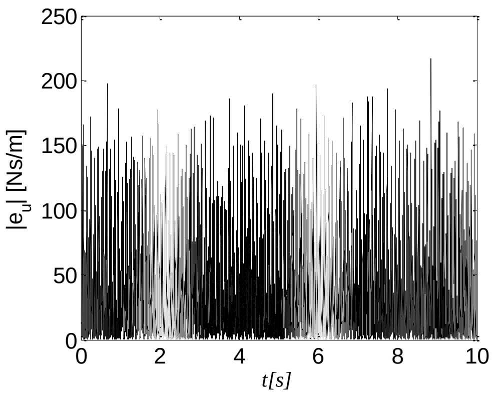
As shown in figure 5a the strength to control is less than 420 Ns/m, common force into (MR dampers) available. Also we can see in figure 5c the electric current required to control the MR damping force.

To consider uncertainties parameter effects on the control performance, the parameters used in the matrix of state variables **A**, will be considered as a random error of $\pm 20\%$, strategy similar is used in (Nozaki *et al*., 2013; Tusset *et al*., 2015; Balthazar *et al*., 2014).

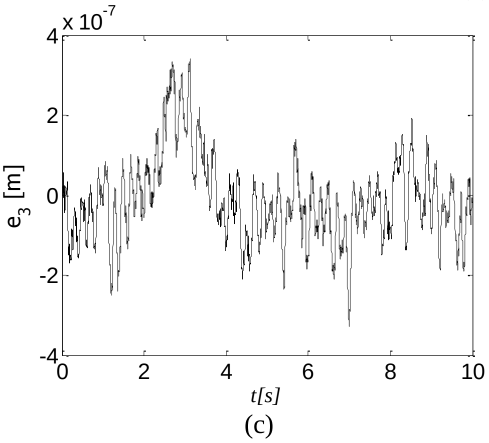
In figure 6 we observe the robustness of the control where $|e_a| = |\dot{x}_4 - \dot{\hat{x}}_4|$ represents the acceleration deviation obtained by control without uncertainties ($\dot{x}_4$) and acceleration obtained by control with parametric uncertainties ($\dot{\hat{x}}_4$), and $|e_u| = |u - \hat{u}|$ represents the signal control obtained by control without uncertainties (u) and signal control with parametric uncertainties ($\hat{u}$), and where $e_3 = x_3 - \hat{x}_3$ that represents the trajectory deviation obtained by the control without uncertainties (x_3) and trajectory obtained by control with parametric uncertainties ($\hat{x}_3$).



(a)



(b)



(c)

Figure 6. (a) Acceleration deviation ($|e_a| = |\dot{x}_4 - \dot{\hat{x}}_4|$). (b) Signal control deviation ($|e_u| = |u - \hat{u}|$). (c) Displacement deviation.

As can be seen in figure 6 the proposed control isn't sensitive to errors in the parameters of the matrix **A**, maintaining the level of acceleration and displacement below the uncontrolled system.

5. CONCLUSION

This work presented the horizontal behavior of a vertical transportation system harmonically excited by guide rails, as well as a proposal for a control by MR damping.

The control technique used proved to be effective for case without parametric error in state matrix **A**, reducing cabin's vibration and contributing on prevention of elevator's components integrity. Also, the amplitude of the acceleration has been reduced, increase passengers' comfort level during the movement of the elevator.

Therefore the proposed control has proved to be efficient both for comfort and for safety. Using equation (5) it was possible to control the coil voltage of the MR damper (figure 5c). We can conclude that in general the proposed active control is appropriate and efficient.

6. ACKNOWLEDGEMENTS

The authors thank CAPES, FAPESP and CNPQ. The first author thanks CNPQ (proc: 447539/2014-0 and 303903/2014-7).

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