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DEVELOPMENT OF AN OPTIMAL AND FUZZY SEMI-ACTIVE CONTROL SYSTEM FOR VEHICLE SUSPENSION

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Abstract. *This paper aims to evaluate the performance of a semi-active controlled suspension system using a magneto-rheological (MR) damper to provide better ride comfort and safety to vehicle passengers than an uncontrolled or passive suspension system. Passive systems represent a conventional solution for vibration control of suspension systems. Although this system is a proven, reliable and economic technology, their parameters cannot be modified according to the road conditions. On the other hand, active systems allow a continuous control of the suspension motion, but require a complex and energy demanding actuator. The proposed suspension system has the adaptability of active systems with lower energy consumption, which constitute an economic and efficient option for vibration control in vehicle suspensions. The analysis was carried out with a set of numerical simulations in Matlab/Simulink using a 1/4 vehicle suspension model with two degrees of freedom for a passive system and two semi-active control modes based on fuzzy and optimal controllers.*

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1 INTRODUCTION

As is well known, the purpose of automotive suspension systems is to minimize the effect of the road roughness in the vehicle to allow a comfort, safety and smooth drive. A large selection of vibration control system has been proposed to reduce unwanted vehicle vibrations. These systems are designed to reduce and dissipate quickly large oscillations and essentially they can be classified according with the control approach as passive, active and semi-active systems. Passive control systems use absorbing or dissipating energy devices operating in a passive mode, i.e., using the suspension motion to command the actuator. Thus, no external power source is required. Although they are simple and inexpensive when compared with other control approaches, they operate in a narrow frequency bandwidth since they cannot adjust the control mode for different road conditions. On the opposite side, active control systems are able to deal with modifications in the system properties including to road disturbances. However, complex actuators are often needed usually with high power requirements making them more complex and expensive than passive systems. In this context, semi-active control systems combining simple controllable or adjustable semi-active devices with low power requirements have been seen as a promising technology for automotive suspension systems with substantial advantages over both active and passive control systems.

Like active control systems, semi-active control approaches also require sensors, actuators and controllers to compute the control action, but the latter use simpler and therefore low-cost actuators which can operate with reduced power sources such as small batteries. The main issue is related with the selection of an effective controller to command the semi-active actuator. So far, a wide range of controllers have been proposed for semi-active vehicle suspension systems. While optimal controllers represent a well-known and simple classical vibration control approach, the hard dependence on the system model and lack of robustness can hinder their applicability to uncertain systems. On the other hand, fuzzy logic controllers seem to be a good approach to design advanced control systems due to its robustness to deal with the inherent uncertainty of automotive suspension systems [1-3].

This paper aims to evaluate the performance of a semi-active controlled suspension system using a magneto-rheological (MR) fluid based actuator. The analysis was carried out with a set of numerical simulations in Matlab/Simulink using a 1/4 vehicle suspension model with two degrees of freedom for a passive system and two semi-active control modes based on fuzzy and optimal controllers. The objective is to compare the effectiveness of each controller in increasing the ride comfort and road handling of a vehicle. The results obtained with the MR damper operating in a passive mode are compared with those obtained with semi-active control modes to assess the performance of the suspension control systems.

2 VEHICLE SUSPENSION

Typically, basic vehicle suspension systems are described by two masses: the body or vehicle mass and the suspension mass. Besides, suspension systems have a symmetric distribution over the car body that allows a simplified vibration response analysis using only one wheel or the so-called ¼ vehicle model shown in Figure 1 [4].

According with the two degrees-of-freedom (DOFs) mass system of the simplified model, the equations of motion of the system can be written as

$$\begin{aligned} m_1\ddot{x}_1 &= F_s \\ m_2\ddot{x}_2 &= F_t - F_s \end{aligned} \tag{1}$$

in which m_1 is the sprung mass, m_2 is the unsprung mass, F_s and F_t are the suspension and tire forces, respectively, described by

$$F_s = k_1(x_2 - x_1) + c_1(\dot{x}_2 - \dot{x}_1)$$
$$F_t = k_2(x_0 - x_2) \tag{2}$$

where x_1 is the motion of the vehicle body and x_2 is the wheel displacement, k_1 is the suspension stiffness and k_2 the spring constant of the tire, and finally c_1 is the damping coefficient of the suspension system. In this case, x_0 represents the road disturbance.

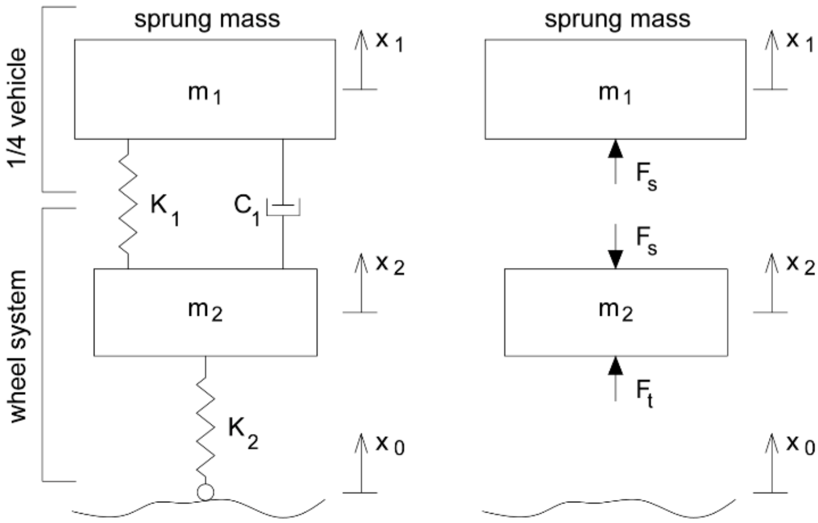

Figure 1 - ¼ vehicle suspension system (Two DOFs model).

The equations of motion of the suspension system can be rewritten as

$$\ddot{x}_1 = \frac{1}{m_1}[k_1(x_2 - x_1) + c_1(\dot{x}_2 - \dot{x}_1)]$$
$$\ddot{x}_2 = \frac{1}{m_2}[k_2(x_0 - x_2) - k_1(x_2 - x_1) - c_1(\dot{x}_2 - \dot{x}_1)] \tag{3}$$

which can be used to describe a state space model of the system as

$$\dot{X} = AX + Bx_0 \tag{4}$$

where X is the state vector, x_0 is the input vector, matrix A given by

$$A = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ -\frac{k_1}{m_1} & \frac{k_1}{m_1} & -\frac{c_1}{m_1} & \frac{c_1}{m_1} \\ \frac{k_1}{m_2} & -\frac{(k_1 + k_2)}{m_2} & \frac{c_1}{m_2} & -\frac{c_1}{m_2} \end{bmatrix} \tag{5}$$

is the state or system matrix and B is the input vector given by

$$B = \left\{0, 0, 0, \frac{k_2}{m_2}\right\}^T \tag{6}$$

3 SEMI-ACTIVE MR SUSPENSION

Vehicle suspension system usually relies on passive dampers with constant damping characteristics to absorb and/or dissipate the kinetic energy generated by the wheel motion due to the road disturbance. These conventional shock absorbers can be replaced by active or semi-active actuators with real-time controllable damping and/or stiffness properties. Among these devices, MR fluid based dampers represent a novel technology that combines the simplicity of passive dampers with the adaptability of active actuators that have been widely used in advanced vehicle suspension systems. Basically, the MR fluid flow within the damper can be controlled using a magnetic field generated by an electromagnet located in the piston rod. This allows for a real-time modification of the rheological behavior of the MR fluid, which results in a modification of the damping force that can be adjusted in accordance with the vehicle motion and road condition. Besides, MR dampers allow a passive or semi-active operation of the vehicle suspension system by using a constant magnetic field or changing the power of the electromagnet, respectively.

The simplified model of a MR based semi-active suspension system is shown in Figure 2. The passive suspension stiffness k_1 and damping c_1 remain as constant values while the damping force represented by the variable damping c(*t*) can be controlled in accordance with prescribed criteria or required operating settings.

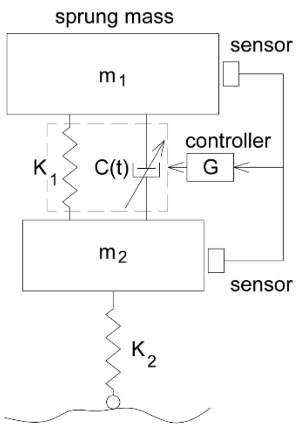

Figure 2 - Semi-active MR based vehicle suspension system.

In this case, the equations of motion are given by

$$\ddot{x}_1 = \frac{1}{m_1}[k_1(x_2 - x_1) + c_1(\dot{x}_2 - \dot{x}_1) + F]$$
$$\ddot{x}_2 = \frac{1}{m_2}[k_2(x_0 - x_2) - k_1(x_2 - x_1) - c_1(\dot{x}_2 - \dot{x}_1) - F] \tag{7}$$

where *F* the control force of the suspension system generated by a MR damper located between the sprung and unsprung masses of vehicle suspension system. The corresponding state space model can be written as

$$\dot{X} = AX + Bx_0 + B_1F \tag{8}$$

where B_1 is the vector of the control force representing the effect of the actuator in the suspension system is given by

$$B_1 = \left\{0, 0, \frac{1}{m_1}, -\frac{1}{m_2}\right\}^T \tag{9}$$

and F is the damping force produced by the actuator. If only the MR damper is responsible by the damping in the suspension system, then c_1=0.

The damping force can be numerically simulated using a numerical model such as the modified Bouc-Wen model shown in Figure 3 [5].

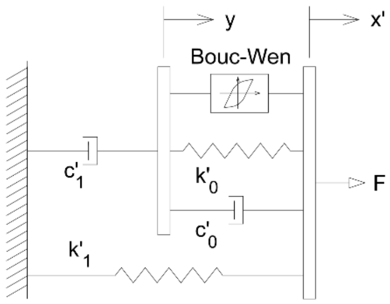

Figure 3 - Phenomenological model of the MR damper (modified Bouc-Wen model).

This model is able to represent the highly non-linear hysteretic behavior of MR dampers. According with the model depicted in Figure 4, the damper force F is obtained by

$$F = c_1'\dot{y} + k_1'(x' - x'_0) \tag{10}$$

where

$$\dot{y} = \frac{1}{c_0'+c_1'}\left[\alpha z + c_0'\dot{x}' + k_0'(x' - y)\right] \tag{11}$$

is dependent on the hysteresis effect given by the evolutionary variable

$$\dot{z}(t) = -\beta\left|\dot{x}'\right|z(t)|z|^{n-1} - \gamma\dot{x}'|z|^n + \delta\dot{x}' \tag{12}$$

Some of the modified Bouc-Wen parameters are current independent parameters (constant values) and frequently, α, c_0' and c_1' are current dependent parameter described by

$$\begin{aligned} \alpha(u) &= \alpha_a + \alpha_b u \\ c_0'(u) &= c_{0a}' + c_{0b}' u \\ c_1'(u) &= c_{1a}' + c_{1b}' u \end{aligned} \tag{13}$$

Besides, the dynamics involved in the MR fluid reaching equilibrium state is represented through first order filter given by

$$\dot{u} = -\eta(u - i) \tag{14}$$

in which the applied current u is described with a time delay relative to the desired current i and η is a first-order filter time constant. The model parameters are determined based on experimental data.

In this study, a numerical model of a commercial MR damper (model RD-1005-3 by Lord Co., USA) is used to represent the damping force. A model parameter identification procedure was carried out and the following current independent parameters were found: δ=10.013, β=3.044 mm, γ = 0.103 mm, k_0=1.121 N/mm, f = 40 N and n =2 [6].

It was also found that the current dependent parameters can be described by polynomial functions as follows

$$\alpha(u) = -826u^3 + 905u^2 + 412u + 38$$
$$c_0'(u) = -11.7u^3 + 10.5u^2 + 11.0u + 0.6 \tag{15}$$
$$c_1'(u) = -54.4u^3 + 57.0u^2 + 64.6u + 4.7$$

Finally, a first order filter with η= 130 sec^{-1} was used to represent the first-order time lag involved in the current driver/electromagnet during a step command signal.

4 OPTIMAL CONTROLLER

The damping properties of MR dampers are frequently controlled using classical control algorithms in combination with a secondary or clipping unit that generates a bang-bang control operation. This approach represents a well-known semi-active control strategy for these devices known as Clipped-Optimal (CO) control algorithm. Besides, this controller has been shown to be very efficient in estimating the control signal and exploring the variable damping force of these semi-active actuators.

The basic idea is to append a force feedback loop to induce the device to produce approximately a desired control force. As has been said, this control strategy combines two controllers:

- A primary controller that includes an optimal control unit which is responsible for determining the optimal or desired control forces of an ideal active control system that should be applied to the structure to reduce the system response;

- A secondary controller. Since only the current/voltage applied to the current driver of the MR damper can be directly controlled, this controller has the function to generate the corresponding control signal in the form of a bi-state control output by clipping the optimal control forces. This accounts for the non-linear nature of MR dampers ensuring that they only produce dissipative forces (i.e., by adapting the ideal control to the semi-active nature of the actuator).

The Linear Quadratic Regulator (LQR) is a simple and well-known optimal controller that provides practical feedback gains. In these vibration control problems, the goal is to minimize the following quadratic cost function

$$\mathrm{J} = \int_0^\infty (X^T Q X + F^T R F)\, dt \tag{16}$$

in which the weighting parameter Q is a diagonal semi-positive matrix that evaluates the state vector X and the weighting parameter R is a positive matrix that evaluates the control force F. The controller law is defined by a state feedback gain G given by

$$G = R^{-1} B_2^T P \tag{17}$$

where P represents the solution for the reduced Riccati equation

$$P A_1^T + \overline{A_1^T} P - P B_2 R^{-1} B_2^T P + Q = 0 \tag{18}$$

Thus, the optimal control problem is to find the feedback gain G to obtain the corresponding optimal control force, which is given by

$$F = -GX \tag{19}$$

In this study, the weighting matrices Q and R were found by keeping Q with a constant value and changing R until the optimal control force is within the force range of the MR damper. The constant weighting matrix Q is related with displacements and velocities of each mass and R is a value that weights the importance of the damper force (there is only one actuator), which are defined as

$$Q = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0.5 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0.5 \end{bmatrix}, \quad R = 10^{-8} \tag{20}$$

Then, the optimal state feedback gain G can be found by solving the Riccati equation, which can be carried out using standard numerical tools in linear algebra. In this case, it follows that

$$G = [0.010 \quad -1.752 \quad 0.318 \quad -0.169] \tag{21}$$

5 FUZZY LOGIC CONTROLLER

Recently, soft computing and intelligent algorithms such as fuzzy logic based controllers (FLCs) have been proposed for semi-active vehicle suspension systems. This advanced modeling technique represents a powerful tool to deal with non-linear systems. FLC controllers use human-like reasoning through the use of fuzzy sets and linguistic variables related by a set of IF-THEN fuzzy rules. The design of a FLC requires the definition of membership functions (MFs) and the selection of appropriate fuzzy inference rules that relate the controller inputs and outputs. In this case, triangular MFs are used to represent both input and output variables as shown in Figures 4 and 5. The FLC inputs are the relative displacement and velocity across the MR damper, which represent state variables obtained via sensor measurements. The controller output is the control signal used to operate the MR damper.

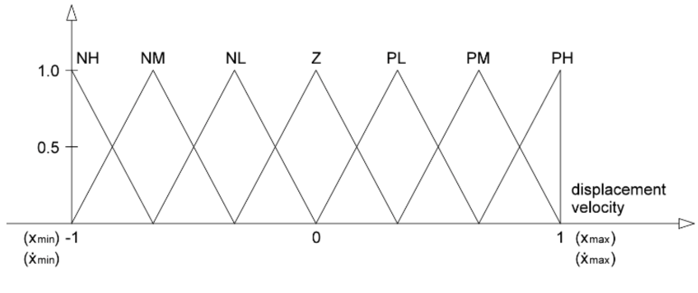

Figure 4 - MFs for displacement and velocity variables.

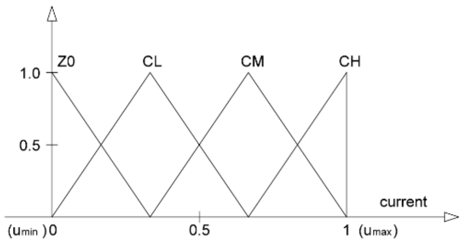

Figure 2 - MFs for the output variable (operating current).

As can be seen in Figure 4, seven linguistic variables were defined for each input as follows: NH- negative high, NM- negative medium, NL, negative low, Z- zero, PL- positive low, PM- positive medium, PH- positive high. The linguistic variables for the fuzzy controller output shown in Figure 5 are de-fined as: ZO- zero, CL- current low, CM- current medium and CH- current high.

		Velocity						
		NH	**NM**	**NL**	**Z**	**PL**	**PM**	**PH**
Displacement	**NH**	CH	CH	CM	PM	ZO	ZO	ZO
	NM	CH	CM	CM	PM	ZO	ZO	ZO
	NL	CM	CM	CL	ZO	ZO	ZO	ZO
	Z	CM	CM	CL	ZO	CL	CM	CM
	PL	ZO	ZO	ZO	ZO	CL	CM	CM
	PM	ZO	ZO	ZO	CM	CM	CM	CH
	PH	ZO	ZO	ZO	CM	CM	CH	CH

Table 1 – Fuzzy rules of the proposed FLC.

The fuzzy inference system (FIS) must be defined assuming some knowledge about the system behavior. In this case, the following concept is used: if the displacement and velocity across the damper are very large, then the output is also large. The resultant fuzzy inference rule is shown in Table 1 [4].

6 NUMERICAL SIMULATION

A Matlab/Simulink model with the following parameters was used: m_1 = 350 kg; m_2 = 40 kg; k_1 = 50000 N/m; k_2 = 190000 N/m and c_1 = 500 N.s/m. The road disturbance is modeled as impulse signals as shown in Figure 6. In this case, two positive impulses with 5.0 mm alternated with a negative impulse with 2.5 mm are used.

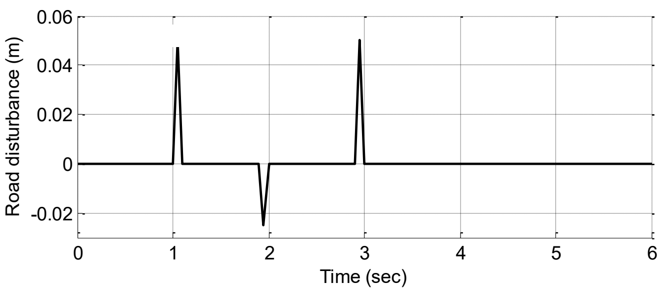

Figure 6 - Road disturbance - impulsive signals.

The clipped-optimal algorithm uses a clipping unit (or secondary controller) to adapt the control forces estimated by the primary optimal controller into a bi-state control signal (bang-bang/on-off controller) to command the MR damper. The damping forces generated by the actuator are dependent on the local responses of the structural system and therefore the device cannot always produce the desired optimal control force. Thus, the following command signal algorithm is applied

$$f_{MR} = \begin{cases} f_c, & f_c \cdot \dot{x} < 0 \\ 0, & \text{otherwise} \end{cases} \tag{22}$$

When the MR damper is delivering the desired optimal force, the operating current should remain the same. If the magnitude of the damper force is smaller than the magnitude of the desired optimal force and the two forces have the same sign, the voltage applied to the current driver is increased to the maximum level. Otherwise, the voltage is set to zero. The algorithm for selecting the command signal for the MR damper can be stated as

$$u = V_{max} H(f_c - f) f \tag{23}$$

where V_{max} is the saturation voltage/current of the MR damper, fc is the desired optimal control force, f is the measured damper force and H is the Heaviside step function.

The results obtained with the two control approaches are shown in Figure 7 for the displacements of each mass and relative displacement between the two masses and Figure 8 for the acceleration of each mass.

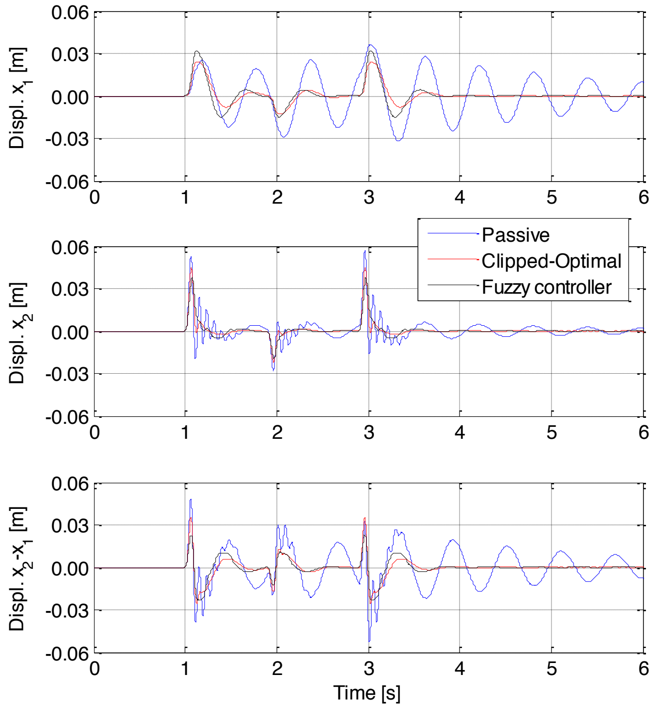

Figure 7 - Displacements (passive and semi-active modes).

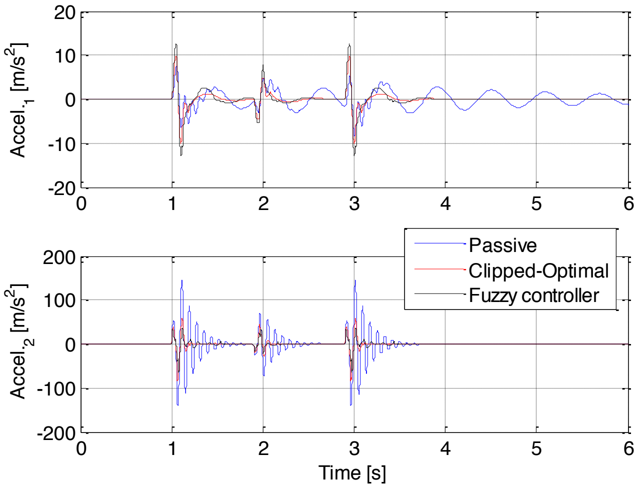

Figure 8 - Accelerations (passive and semi-active modes).

The semi-active control approach is more effective than the passive mode in eliminating the oscillation of the individual components. The effectiveness of the semi-active suspension system can be assessed measuring the root mean square (RMS) value of the body vertical motion (m_1) in both passive and semi-active control modes. The results are summarized in Table 2. Essentially, the results show that the proposed semi-active control system presents a higher performance in reducing the vehicle body vertical displacements and accelerations (clipped-optimal) than that obtained with the passive system.

Control system	RMS x_1	RMS $\ddot{x}_1$
Passive	0.0148	1.9369
SA – optimal	0.0062	1.5724
SA – Fuzzy	0.0073	2.1138

Table 2 – Controller evaluation (RMS criteria).

The semi-active fuzzy controller is less effective in reducing the peak vertical acceleration than the passive and clipped-optimal control mode, which is related with the fuzzy parameters used in this case. Searching procedures could be used to optimize the MFs and inference system in order to enhance the performance of the fuzzy controller.

7 CONCLUSION

In this study, a simplified model with two degree of freedom under impulsive road excitation was used to assess the performance of passive and semi-active suspension systems. The proposed semi-active suspension integrates a clipped-optimal and a fuzzy logic based controller in combination with a MR damper to reduce the motion of the vehicle body. Although the peak values of the body vertical acceleration are increased, the proposed clipped-optimal and fuzzy based control modes provide better ride comfort and safety reducing or attenuating the vibration level of the vehicle body when compared with the passive suspension system.

REFERENCES

[1] L. Cheng, Z. Qiang, *Fuzzy Control of Vehicle Semi-active Suspension with MR Damper*, WASE International Conference on Information Engineering, Vol. 3, pp. 426-429, 2010.

[2] A. Cherry, R. Jones, *Fuzzy logic control of an automotive suspension system*, IEE, Vol. 142, No.2, pp. 149-160, 1995.

[3] B. Sapinski, *Fuzzy Control for MR dampers in a Driver's Seat Suspension*, Journal of Theoretical and Applied Mechanics, Vol. 43-1, pp. 179-201, 2005

[4] K. Oliveira, M. Braz-César, J. Gonçalves, *Fuzzy based Control of a Vehicle Suspension System using a MR Damper*, Series Lecture Notes in Electrical Engineering Vol. 402, pp. 571-582, 2016.

[5] B. Spencer Jr., S. Dyke, M. Sain, J. Carlson, *Phenomenological model of a magnetorheological damper*, Journal of Engineering Mechanics, Vol. 123, pp. 230.238, 1997.

[6] M. Braz-César, R. Barros, *Experimental behaviour and numerical analysis of MR dampers*, 15WCEE - 15th World Conference on Earthquake Engineering, 2012.