Journal of Engineering and Applied Sciences 13 (10): 3472-3478, 2018
ISSN: 1816-949X
© Medwell Journals, 2018

Characterization and Testing Bench Design for Magnetorheological Dampers

[2]Sebastian Jimenez, [1]Oscar F. Aviles, [1]Mauricio F. Mauledoux,
[1]Oscar I. Caldas F and [1]Hoffman F. Ramirez
[1]Department of Mechatronics Engineering Program, Faculty of Engineering,
Militar Nueva Granada University, Bogota, Colombia
[2]Escuela de Ingenieria de Antioquia University, Medellin, Colombia

Abstract: Magnetorheological dampers and actuators are based on the physical principle of non-Newtonian fluids and its behavior when under the presence of a magnetic field. A good mathematical approximation of this phenomenon is the Bing-Ham Model with the inclusion of the Bouc-Wen hysteresis which evaluates the non-linearity of the power ratio vs. velocity damping model. This study presents a proposed test for the identification process also experimenting with various operating conditions in which this type of actuator can be used such as in vehicle suspension systems, vibration control or prosthetic devices. The study began exploring various models representing the dynamic characteristics of these actuators, so that, to determine the state variables to be monitored. From the models studied it was determined that the test bench must consider the position obtained through a LVDT sensor and the current drawn by an electric motor as inputs as well as the damping force according to the speed variations as output, this is measured by a load cell. Dimensioning according to the ranks of the selected actuator operation was performed (RD-8040-1 Lord), so, minimum and maximum strength and speed at which the system will be subjected were defined in order to assess the full dynamics of the system. Therefore, a hardware and software based architecture is proposed along with a graphical user interface that allows applications to be used for identification and control of this actuators.

Key words: Control systems, identification, magnetorheological damper, parametric systems, interface

INTRODUCTION

MagnetoRheological Dampers (MRD) have had a big development during the last years and have been used in a variety of tasks, including semi-active suspension which allows controlling mechanical systems. There are developments on biomechatronics applied to smart prosthesis (Li and Xianzhuo, 2009; Herr and Wilkenfeld, 2003), mainly used as part of control systems for artificial knees braking, i.e., controlling the joint flexion and extension according to the each phase of the user gait cycle in order to obtain similar characteristics to those of a natural gait, based in the user specific cadence. There are also applications on civil structures (Koo, 2003; Jin *et al*., 2005), vibration control (Liao and Lai, 2002), damping for aircrafts landing gears and recoil systems for guns (Hongsheng *et al*., 2009) and even for rapid prototyping systems such as Hardware-In-the-Loop (HIL) (Betterbee and Sims, 2007), due to its importance in real-time emuling of control processes and real devices like sensors and actuators to optimize expenses, duration and safety during tests.

Tianjun and Changfu (2009) introduced an experimental test proposal for MR actuators identification based on polynomial models which let to obtain all the coefficients for up to fifth degree polynomials, using force vs. speed data as inputs in a universal testing machine and cyclic tests. The same methods were used by Sapinski and Filus (2003) by using an Instron machine to evaluate the error between the proposed phenomenological model and the experimental result with the purpose of selecting an appropriate control strategy according to specific applications. On the other side, Santos assessed different kind of controllers used for these systems in terms of “pros and cons” of implementation. Figure 1 shows the general architecture for MR actuators identification in a damping setup which consists in a MR damper with a force applied by an external active actuator (either electromechanical,

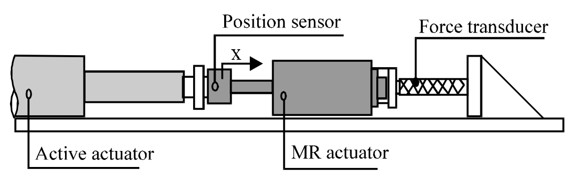

Fig. 1: MR identification assembly

Corresponding Author: Sebastian Jimenez, Escuela de Ingenieria de Antioquia University, Medellin, Colombia

pneumatic or hydraulic). Meanwhile, input data acquisition is performed by means of a displacement sensor and a force transductor.

Once observed the scope of these devices, a design proposal is presented to perform any of the two following procedures: on the one hand, an identification process with either black or gray box models and on the other hand, control strategies for rough models simulated on mechanical assemblies that represents partially real problems dynamics (Conde *et al*., 2009).

This study presents the above mentioned design as a configurable testing machine in such a way that both identification and control stages are able to be performed, so that, different operation conditions can be emulated by a 1-DOF mechanical system, e.g., a quarter car active suspension system with mass and spring stiffness as the variable parameters. This kind of tests allows to assess simulations and theoretical models and thus, the performance of the function approximation in order to apply control strategies to the characterization systems. Likewise, different control techniques can be applied with this testing bench and the variable dynamic system to be controlled which maximum operation conditions are considered in terms of forces and velocities (a RD-8040-1 Lord damper is used).

MRD Models: There is a close relationship between the model and the selected identification strategy, thus, it is important to define a phenomenological characterization based on physical parameters. Therefore, a representative model of the magnetorheological dampers dynamics must be obtained to assess what system state variables have to be measured as well as to do an appropriate instrumentation and sizing of the testing bench general architecture. It is also necessary to decide the operation conditions in the control set-up.

The first objective of explaining the system (MR damper) and the effects of its components, via. mathematical models, it is not a simple task due to its nonlinear behavior and can be easily seen when looking at the state of the art. There are several theoretical models, but some of them are too complex or do not reproduce the dynamics good enough. This variety of models includes basic linear representations of damping force and velocity, complex approaches of the nonlinear fluid behavior (friction and internal viscosity) and the hysteresis effect of the magnetic field applied (Choi *et al*., 2001). Some of the commonly adopted models are listed in Table 1, including models of the damping force F_D as a linear function of velocity v which shows a disadvantage due to the force not only depends on the input current I, but also on the system velocity and assumes that the damping coefficient C is relatively linear with respect to current.

Table 1: Models for MRD

Model	Equations	Type of system
Linear Bing-Ham Ma *et al*. (2003)	$\tau = G.\gamma, \tau < \tau y$ $\tau = \tau y(H) + \eta\gamma, \tau > \tau y$ $F_{MR} = C(I)v$	Linear
Bing-Ham simplified Stanway *et al*. (1985)	$F_{MR} = F_c.\text{sgn}(x) + C_0 x + f_0$	Linear
Non parametric polynomial model (Tianjun and Changfu, 2009)	$F_{MR} = \sum_{i=0}^{n} a_i v^i$ $F_{MR} = \sum_{i=0}^{n} (b_i + c_i I) v^i$	Nonlinear
Bouc-Wen Model-extended Spencer *et al*. (1997)	$F_{MR} = \begin{cases} C_o(\dot{x}-\dot{y}) + k_0(x-y) + k_1 \\ (z-zo) + \alpha z C_1\dot{y} + k_1(z-z_0) \end{cases}$ $z = -\gamma\lvert z\rvert\lvert z\rvert^{n-1} - \beta(\dot{x}-\dot{y})\lvert z\rvert^{n}$	Nonlinear

The Bingham simplified model is based on the rheological behavior of the internal fluid as the parallel of a Coulomb friction element and a viscous damper. This model considers a damping coefficient C_0, a constant force f_0 (to compensate the non-displacement effect at the beginning of the force applying by the end zone) and the fluid yield f_c. Nevertheless, Bingham fluids requires a high level of strength before start flowing. The shear strength and deformation ratio is described in Eq. 1:

$$\tau = \tau^0 + \eta\gamma \tag{1}$$

Bouc-Wen model of MRD: This model is a reference point when regarding to phenomenological models for magnetorheological dampers. As its name suggests, it is based on the on the use of the Bouc-Wen Model of hysteresis, due to the fact that it does consider the system nonlinearity. It is also important to check if the tuning parameters assure the two basic compatibility properties between the model and the physical laws: to have input and output boundaries and to describe energy dissipation (Ikhouane and Rodellar, 2007).

As shown in Fig. 2, output is the MRD force C_0 could be described as the sum of the following terms:

- The damping friction produced by the seals and the measurement bias
- The product of the mass, inertia effects and piston acceleration
- The product of the piston velocity and the plastic damping coefficient (post-yield) C_0x
- The product of the piston position and the elasticity coefficient $K_o(x-x_o)$
- The hysteresis term α z

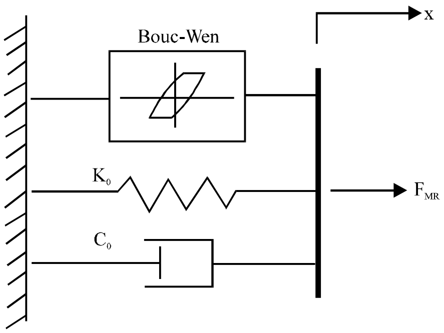

Fig. 2: Bouc-Wen Model schematics

Therefore the force generated by the damper is the one described in Eq. 2:

$$F_{MR} = c_0 x + k_0 (x - x_0) + \alpha z \quad (2)$$

The time derivative of the hysteresis component is given in Eq. 3:

$$z = \gamma |x| z |z|^{n-1} - \beta x |z|^n + Ax \quad (3)$$

These models and some variations of the Bouc-Wen Model mentioned by Spencer *et al*. (1997) give a clear outlook of the state variables needed to be measured ina MRD and hence, to obtain the requirements for characterization and testing bench sizing.

MATERIALS AND METHODS

Methods and experimental setup: General proposal architecture: The purpose of this device is to be modular and configurable, so, it allows identifying and assessing control techniques on the operating conditions for emulating 1-DOF (Degree of Freedom) mechanical systems.

The identification process seeks the force-vs-velocity function, thus it covers the whole design architecture according to velocity and work force ranges, allowing assessing the MRD dynamics. This is necessary for the appropriate electronical instrumentation and mechanical sizing in the experimental assembly in order to minimize frictions through the actuator movement direction and assure good parameter estimation. The MRD RD-8040-1 Lord® has the following features:

- Stroke = 55 mm
- Maximum damping force (peak-to-peak) = 2447 N, at a linear velocity of 5 cm/sec
- Maximum damping force (peak-to-peak) <7 N, at a linear velocity of 20 cm/sec
- Maximum curent = 1 A

Table 2: Electronic instrumentation

Sensor	Ref.	Range	Precision	Purpose
Current	ACS 714			Sensing the control signal of the MRD
LVDT	HC Metrolog	100 mm	0.01±0.005 mm/v	Feedback the mass or damper position
Accelero meter	MMA 736IL	±1.5 g	800 mV/g	Feedback the disturbance position
Load cell	Lexus SA	100 kg	2±0.2 mV/V	Feedback the reaction force

A correct identification requires a position variation in an ideally sine waveform which allows to study the damper complete stroke in both motion directions in order to analyze the hysteresis. On the other hand, the two basic setups must be considered for the testing bench design process, related with the operation modes: the identification setup and the control setup. Hence, it must be modular and let to adapt different attachable accessories (masses, springs, etc.,) as well as all the electronics.

Position reaction force and current (control signal) must be in feedback loop. Considering the above, the sensors listed in Table 2 were required and selected. via. the given operation ranges and the state variables of the MR that should be monitored. To acquire those signals, a NI DAQ USB 6216 acquisition system was used with a sample rate of 25 Hz to allow variations on the mechanical system dynamics, based on the Nyquist-Shannon sampling theorem: sampling time must be at least a half of the system response time, although it is common to use a tenth part in to have integrity in the signal (Skoog *et al*., 2008). The system architecture is presented in Fig. 3.

Servomechanism system: An essential step in the testing bench design for both operation setups is the active actuator selection. The main required features are speed, force and control technique as the minimum requirements to perform tests on MRDs. It is necessary to have independence between the applied force and speed, considering that it is required an input with a sine waveform and a specific operation range (Ikhouane and Rodellar, 2007). Besides, if there are speed and force requirements, e.g., 5 cm/sec and 667 N, the actuator should perform the test in such a way that it maintains a constant force, regardless the speed.

Regarding to the source of power, several options were considered. A pneumatic system requires a high initial pressure to assure the minimal desired force but it is not constant due to the fluid compressibility (nonlinearity), besides of the fact that the speed is reached by means of two servovalves: pressure and discharge ones. Also, this speed is constrained by the control system capabilities for driving the nonlinearities and the implicit delays of a pneumatic system

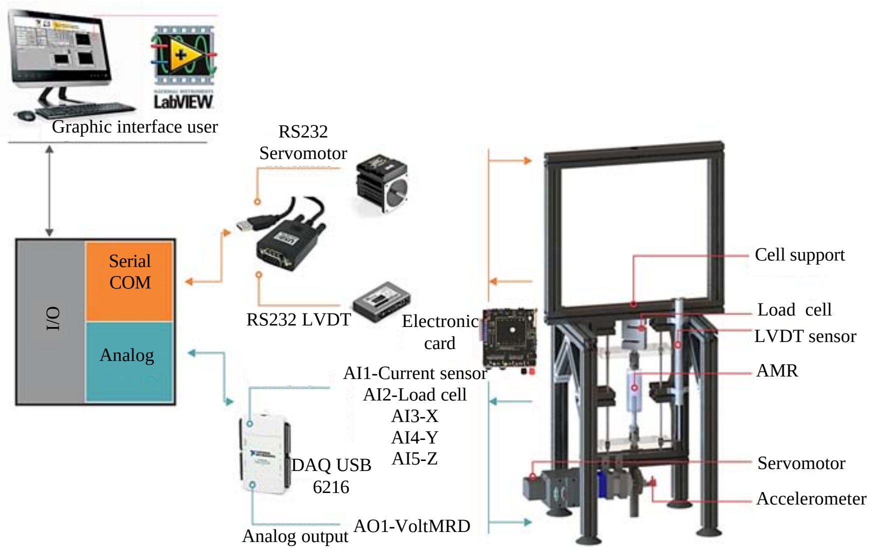

Fig. 3: General proposal architecture of testing bench

Table 3: Animatics SM34165DT servomotor features

Variables	Values
Continuous torque	1.45 Nm
Peak torque	3.39 Nm
No load speed	5100 rpm
Continuos current	15.5 A

(Jimenez *et al*., 2012). All this lead to use a rack and pinion coupled to an Animatics SM34165DT servomotor, based on the functional features listed in Table 3.

According to the above, it is necessary to generate a translational motion that assure the MRD stroke (the only one degree of freedom) via. the rack and pinion system. Therefore, it was selected a linear force of 1800 N and a speed of 20 cm/sec as the input parameters and a safety factor of 2.25 in respect of the necessary load for identification tests (up to 800 N).

If the servomotor torque is $\tau_m = 1.45$ Nm and R = 40 is the reduction ratio of a GBPH-0602-NP-040 gearbox (with an efficiency of $\xi = 95\%$) then the linear force F_l can be obtained using the standard primitive diameter $D^p = 64$mm and a modulus of Mod = 2, as described in Eq. 4-6:

$$F_l = \frac{\tau_m}{\left(\frac{D_p}{2}\right)}.\xi.R \tag{4}$$

$$N_t = \frac{D_p}{Mod} = 32 \text{ teeth} \tag{5}$$

$$P_c = \frac{D_p.\pi}{N^t} = 6.28\frac{mm}{Tooth} \tag{6}$$

Thus, the rack displacement per pinion revolution is described in Eq. 7:

$$D_l = N_t\left[\frac{teeth}{rev}\right].P_c\left[\frac{mm}{teeth}\right] = 200.96\frac{mm}{rev} \tag{7}$$

Finally, the rack máximum linear speed is shown in Eq. 8:

$$V_l = \frac{\omega_M\,[rpm]}{R}D_l\left[\frac{mm}{rev}\right] = 20.93\frac{cm}{s} \tag{8}$$

In addition, although the designed servomechanism achieves a constant power transmission and a translational motion, it is necessary to control itrs trajectories in order to assure a sine waveform position as well as velocity and acceleration ramps based on position data interpolation (for a smooth transition). Such curve fitting between desired initial and final positions could be linear or spline (polynomial), varying the transition period which in the end determines the control action for servomotor acceleration and deceleration.

Experimental design for gray box model identification: Gray box identification models (phenomenological models) are generally parameter optimizers of theoretical models that use acquired data from real physical systems (Oviedo, 2010).

In study 2 several theoretical models were described and could be used as the starting point for the parameter optimization along with optimization algorithms for square error minimization between the real signal and an estimated value. This research proposes the use of the multi-objective optimization algorithm NSGA-II

(Non-dominated Sorting Generic algorithm) which allows to do that offline with an experimental method that relates all variables with the designed machine, considering all testing inputs such as position, velocity and MRD controlling current as shown in Fig. 4.

Where U corresponds to stochastic perturbations of position and controlling current in the MRD, Y-real is output data measured by the load cell and Y-estim is data generated by the model by means of the input data themselves and the parameter optimization made with the algorithm mentioned above and described below.

Parameter optimization algorithm: The evolutionary algorithm NSGA-II is proposed to perform a multiobjective and multivariable optimization during the identification of the Mouc-Wen model physical parameters. To assess the algorithm and model performance, multiple simulation tests were applied replacing the experimental data with the theoretical model with known parameters but optimized by the algorithm. The theoretical model was enhanced by noises inclusion to emulate measuring perturbations. The objective functions are based on the error E as the difference between the measured signal and the one estimated by the parametrized model. Equation 9 correspond to both 2-norm and infinity-norm error measures:

$$\|E\|_2 = \frac{\sqrt{E(1)^2 + ... + E(k)^2}}{k}$$
$$\|E\|_\infty = \frac{\mathrm{Max}\,|E(1)| + ... + |E(k)|}{k} \quad (9)$$

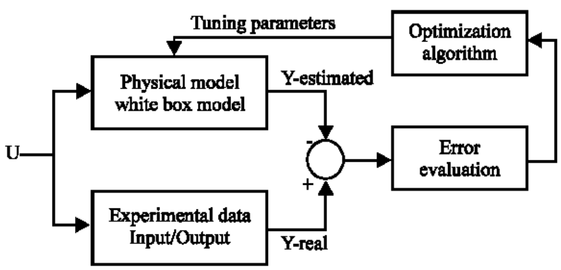

Fig. 4: Gray box model implemented

RESULTS AND DISCUSSION

A graphical user interface was developed in LabView® to supervise the testing bench activity and enable operation modes (identification and control). First, it is done a parameters variation related to the servomotor operation and signal configurations for position, velocity, acceleration and signal type. This way, it is configured to perform linear displacement with varying waveform due to perturbations (step, triangle and sine), as shown in Fig. 5a.

Sine waveform range is 0.5-2 Hz for up to 25 mm of stroke and up to 800 N of force; this ranges are reached by using a 37 V/10 A power supply. According to the testing bench setup, acquired signals supervising is made via. a LVDT sensor for position, an accelerometer, a current sensor and a load cell. The sensors response time and the sampling frequency set in the DAQ USB 6216 allow to perform monitoring on the system response.

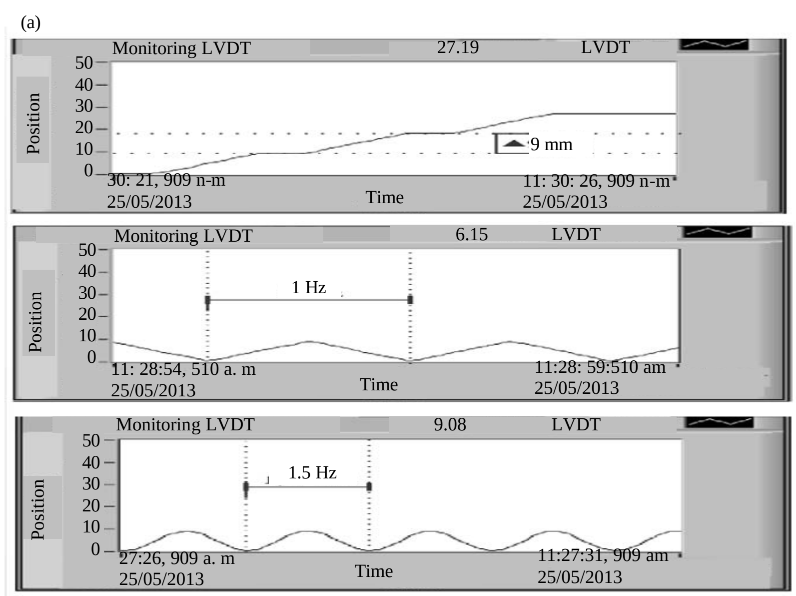

Fig. 5: Continue

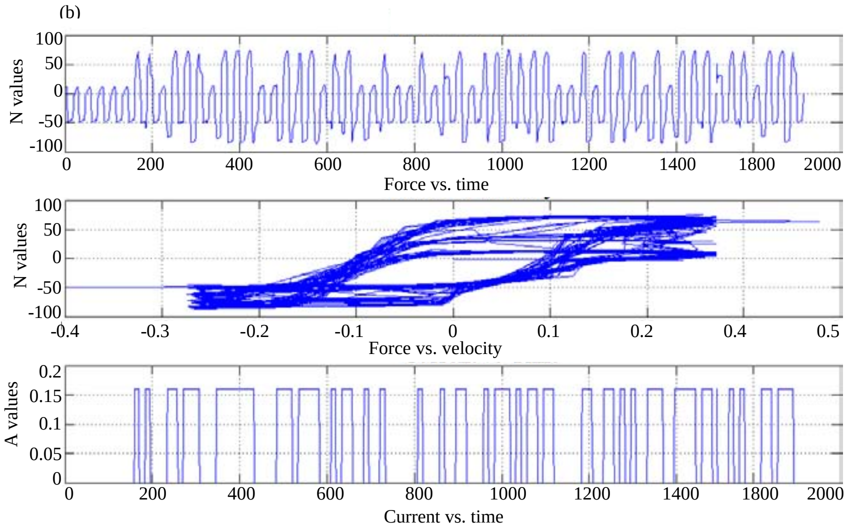

Fig. 5: Proposed general architecture testbed

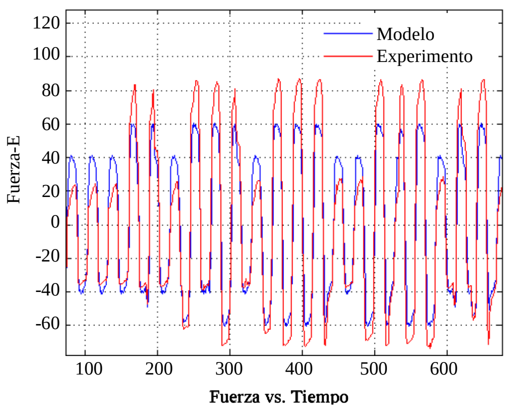

Fig. 6: Comparison of experimental data and Bouc-Wen parameterized model

Fig. 5b shows a force-vs-time chart with the result of a test performed with a 10 mm stroke at 1.5 HZ, reaching up to ±100 N; the controlling current was a 150 mA Pseudo-Random Binary Sequence (PRBs) in order to have a frequency rich signal to better asses the testing bench and sensor instrumentation performance.

Regarding to the Bouc-Wen parametrized model, it was obtained using the data generated by the experiment mentioned above and a scale factor F_s = 3.1878. This, along with the error tuning to get $\|E\|_2$ = 0.3770 and $\|E\|_\infty$ = 0.0245, result in a proper reproduction of the theoretical model with the experimental data, even following suitably the hysteresis non linearities (Fig. 6).

CONCLUSION

It was performed a study of models to represent Magneto Rheological Dampers (MRD) dynamics in order to generate specifications to design and instrument a testing bench. This testing bench allowed to do several actions that influence on the MRD performance, including the use of a variety of perturbations and control signals.

The pertinence of the gray box identification process lies in being able to use experimental data to obtain parameters for a mathematical model in such a way that they could be physically quantifiable as well as being available as a tool for failure identification and predictive maintenance.

During the parameter optimization with the NSGA-II algorithm, it was important to assess properly possible noises and trends of the test-acquired signals and then define boundaries from results to limit the feasible space. This also considered the stability conditions of the Bouc-Wen Model, mentioned in study 2.

The designed testing bench remarks for its modularity which let to perform identification processes as well as control tasks with 1-DOF dynamic systems, such as vehicle suspension systems, vibration tables, among others. Thus, it is possible to try the identified system out by means of the application of a control technique.

ACKNOWLEDGEMENT

The research for this study was supported by Universidad Militar Nueva Granada.

REFERENCES

Batterbee, D.C. and N.D. Sims, 2007. Hardware-in-the-loop simulation of magnetorheological dampers for vehicle suspension systems. Proc Inst. Mech. Eng. Part I. J. Syst. Control Eng., 221: 265-278.

Chen, D.W., H.B. Gu and H. Wu, 2010. Application of Magneto-Rheological (MR) damper in landing gear shimmy. Proceedings of the 2010 3rd International Symposium on Systems and Control in Aeronautics and Astronautics (ISSCAA), June 8-10, 2010, IEEE, Harbin, China, ISBN:978-1-4244-6043-4, pp: 1212-1216.

Choi, S.B., S.K. Lee and Y.P. Park, 2001. A hysteresis model for the field-dependent damping force of a magnetorheological damper. J. Sound Vibr., 245: 375-383.

Conde, E.C., F.B. Carbajal, A.B. Ortega and H.M. Azua, 2009. Sliding mode and generalized PI control of vehicle active suspensions. Proceedings of the IEEE Joint Conference on Control Applications (CCA) and Intelligent Control (ISIC), July 8-10, 2009, IEEE, St. Petersburg, Russia, ISBN:978-1-4244-4601-8, pp: 1726-1731.

Herr, H. and A. Wilkenfeld, 2003. User-adaptive control of a magnetorheological prosthetic knee. Ind. Robot: Int. J., 30: 42-55.

Hongsheng, H., W. Jiong, Q. Suxiang and J. Xuezheng, 2009. Test modeling and parameter identification of a gun magnetorheological recoil damper. Proceedings of the International Conference on Mechatronics and Automation ICMA, August 9-12, 2009, IEEE, Changchun, China, ISBN: 978-1-4244-2692-8, pp: 3431-3436.

Ikhouane, F. and J. Rodellar, 2007. Systems with Hysteresis: Analysis, Identification and Control using the Bouc-Wen Model. John Wiley & Sons, Hoboken, New Jersey, USA., ISBN:9780470513194, Pages: 222.

Jimenez, S., O. Caldas, E. Mejia, J.C. Hernandez and O. Aviles, 2012. Modeling and control of destructive test equipment for lower limb prosthesis. Proceedings of the 1st International Conference on Advanced Mechatronics, Design and Manufacturing Technology AMDM, September 5-7, 2012, Technological University of Pereira, Pereira, Colombia, pp: 463-468.

Jin, G., M.K. Sain and B.E. Spencer, 2005. Nonlinear blackbox modeling of MR-dampers for civil structural control. IEEE. Trans. Control Syst. Technol., 13: 345-355.

Koo, J.H., 2003. Using magneto-rheological dampers in semiactive tuned vibration absorbers to control structural vibrations. Ph.D Thesis, Virginia Tech University, Blacksburg, Virginia.

Li, F. and L. Xianzhuo, 2009. The modeling research of magnetorheological damper in advanced intelligent prosthesis. Proceedings of the International Conference on Control and Decision Chinese CCDC'09, June 17-19, 2009, IEEE, Guilin, China, ISBN:978-1-4244-2722-2, pp: 781-784.

Liao, W.H. and C.Y. Lai, 2002. Harmonic analysis of a magnetorheological damper for vibration control. Smart Mater. Struct., Vol. 11,

Ma, X.Q., E.R. Wang, S. Rakheja and C.Y. Su, 2003. Evaluation of modified hysteresis models for magneto-rheological fluid dampers. Proceedings of the 4th International Conference on Control and Automation ICCA'03, June 12, 2003, IEEE, Montreal, Quebec, Canada, ISBN:0-7803-7777-X, pp: 760-764.

Oviedo, D.D., 2010. [Optimization of the bouc-wen model of a magnetoreological buffer using genetic algorithms]. Master Thesis, Charles III University of Madrid, Getafe, Spain. (In Spanish)

Sapinski, B. and J. Filus, 2003. Analysis of parametric models of MR linear damper. J. Theor. Appl. Mecha., 41: 215-240.

Skoog, D.A., R.C. Stanley and F.H. James, 2008. [Principles of Instrumental Analysis]. Cengage Learning, Boston, Massachusetts, USA., ISBN:9789706868299, Pages: 1038 (In Spanish).

Spencer Jr., B.F., S.J. Dyke, M.K. Sain and J. Carlson, 1997. Phenomenological model for magnetorheological dampers. J. Eng. Mech., 123: 230-238.

Stanway, R., J.L. Sproston and N.G. Stevens, 1985. Non-linear identification of an electro-rheological vibration damper. IFAC. Proc. Vol., 18: 195-200.

Tianjun, Z. and Z. Changfu, 2009. Development of MRF damper modelling and validation of MRF damper test. Proceedings of the IITA International Conference on Control, Automation and Systems Engineering CASE, July 11-12, 2009, IEEE, Zhangjiajie, China, ISBN:978-0-7695-3728-3, pp: 238-241.