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DESIGN PROPOSAL OF A TESTING BENCH FOR A MAGNETORHEOLOGICAL DAMPER

Sebastián Jiménez
Oscar F. Avilés
Univesidad Militar Nueva Granada, Bogotá-Colombia
sebastian.jimenez@unimilitar.edu.com/oscar.aviles@unimilitar.edu.co

Mauricio F. Mauledoux
Univesidad Militar Nueva Granada, Bogotá-Colombia
mauricio.mauledoux@unimilitar.edu.co

Oscar I. Caldas
Univesidad Militar Nueva Granada, Bogotá-Colombia
oscar.caldas@gmail.com

Edilberto Mejía
Univesidad Militar Nueva Granada, Bogotá-Colombia
edilberto.mejia@unimilitar.edu.co

Juan C. Hernández
Univesidad Militar Nueva Granada, Bogotá-Colombia
juankmilo8405@unimilitar.edu.co

Abstract. *The magnetorheological dampers are based on no-Newtonian fluids, whereby the best approximation is the Bing-Ham model with a Bouc-Wen Hysteresis that generates a nonlinearity in the force vs speed function for the damping system model. This work presents a design proposal of an experimental assembly that allows not only the identification process, but also emulating different operation conditions in which the damper can be used, such as vehicles suspension systems, vibration control and prosthetics devices. This starting from physical models that permit to do the damper monitoring, characterizing and controlling, considering the input (LVDT and current sensor) and output (load cell) variables to measure the damper force according to the input speed. Therefore, the testing bench sizing, instrumentation and material selection are done considering the selected prototype (RD-8040-1 Lord®) operational maximum and minimum ranges for force, displacement and speed, looking for complete system dynamics identification, proposing a hardware and software architecture with a graphical user interface which can be used on future applications with control strategies for the identified model.*

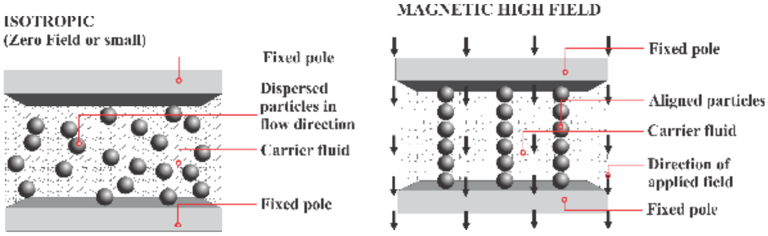
Keywords: *Magnetorheological dampers, Identification systems, Hysteresis, Parametric systems…*

1. INTRODUCTION

The magnetorheological dampers (MRD) base their behavior on the called ferrofluids: colloidal liquids made of nanoscale ferromagnetic particles (diameter 3-15 nm) suspended in a carrier fluid, usually mineral oil (Ghita and GIuclea (2004)), with the capability to show mechanical reactions through an electrical interface control quickly, simply and soundlessly. Magnetorheological fluids change their rheological properties when subjected to a magnetic field, due to the interaction between the induced dipoles and how this coerce the particles to form chained structures in the direction parallel to the magnetic flux, as shown in Fig.1. This behaviour exhibit a shear stress that increases with magnetic field.

These systems behave as semiactive actuators, because they preserve a part of the pasive behaviour, i.e. damping and energy absorption. It is important to clarify that this the energy absorption feature requires an external power source and a control signal to be changeable. For instance, the Maxwell elements, those with viscous damping and an elastic component (Butz and von Stryk (1998)), offer the posibility to change the model parameters according to an electrical current (for some work ranges), becoming dependant of the external power source. Details and considerations for modeling magnetorheological dampers will be explained later.

Semiactive dampers have several applications on controlling mechanical systems. There are applications in biomechatronics, mostly in controlling the braking system on intelligent protheses (Li and Xianzhuo (2009), Herr and Wilkenfeld (2003)) in accordance with the current phase of the human gait cycle, in order to achieve a more natural gait in terms of each person's cadence. They are also used for vibration control in structural engineering (Koo (2003), Jin *et al.* (2005)) and in MRD (Liao and Lai (2002)), by means of the parameters identification for a vibrations absorption system with 1 degree of freedom (DOF). Likewise, it is posible to control the absorption system for impact damping, which is useful in



(a) Without magnetic field (b) With magnetic

Figure 1: Principles of magnetorheological fluids with and without the presence of magnetic field (Ghita and GIuclea (2004))

the aircraft undercarriages (Da-wei *et al.* (2010)) and the blowback system of firearms (Hongsheng *et al.* (2009)), or even simulate these systems via rapid prototyping, using the Hardware in the Loop (HIL) technique to real time emulation of plants or control systems, in order to optimize the expenses, security and length of tests (Batterbee and Sims (2006)).

Regarding to the particular application of MRD, Tianjun and Changfu (2009) present a testing proposal for MRD identification via polynomial models, which experimentally obtains the coefficients of a fifth order polynomial, according with force and velocity data acquired by an universal testing machine. Sapinski and J. (2003) use an INSTRON machine to assess error performance of the phenomenologycal model proposed in terms of the experimental results, and then develop an appropriated control strategy. Likewise, Santos *et al.* (2009) assessed several type of controllers used in these systems, remarking advantages and disadvantages of implementing each one of them.

According to the above, a global architecture for MRD identification is presented in Fig. 2, which shows a magnetorheological damper, the position and force data acquisition and the input, set by either an electromechanical, pneumatical or hydraulical active actuator.

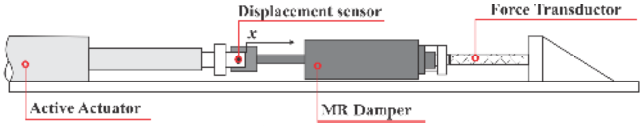


Figure 2: Assembly diagram for identification

Once observed the scope of this devices, a design proposal is presented to solve either two of the following procedures: on the one hand, it is performed an identification process with either black or gray box models, and on other one, control strategies are proposed for rough models simulated on mechanical assemlies that represents partially the real problem dynamisc (Chavez *et al.* (2009)).

This paper presents a design proposal of a testing machine configurable in such a way that both identification and control stages are able to be done, 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 characterized systems. Different control strategies can be performed with this variable plant and single testing bench, which considers the maximum operation conditions of the damper RD-8040-1 Lord®, in terms of forces and velocities.

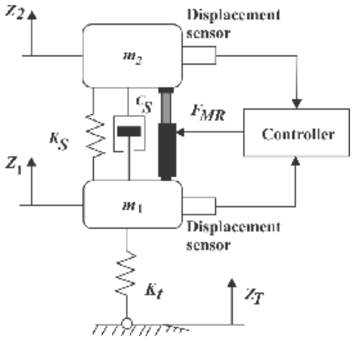


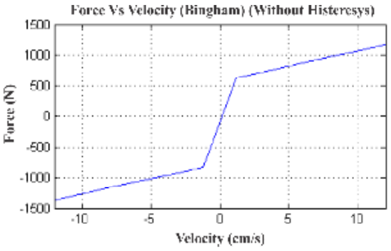
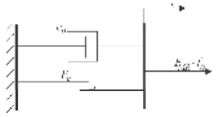
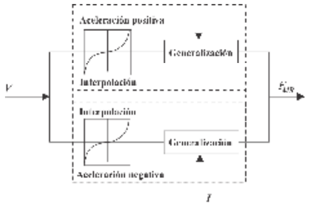
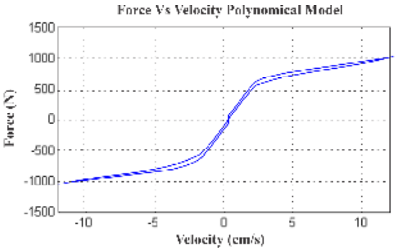
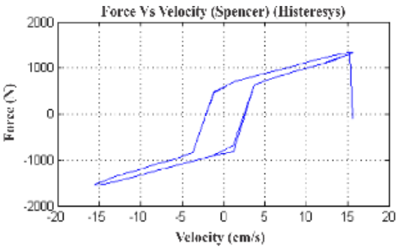
Figure 3: Schematic model of active suspension quarter of a Vehicle

2. MRD MODELS

There is a close relationship between the model and the selected identification strategy. In order to define a phenomenological characterization based on physical parameters, a representative model of the magnetorheological dampers dynamics must be obtained, which allows to assess what system state variables must be measured, as well as do the selection of appropiate instrumentation and the sizing of the testing bench general architecture. It is also necessary to decide the operation conditions in the the control set-up.

Explaining the system (MRD damper) and studying the effects of it's components, by means of mathematical models, is not a simple task due to its nonlinear behaviour. There are several theoretical models, but some of them are too complex or do not reproduce that behaviour good enough. This variety of models includes basic linear representations of damping force and velocity, complex approaches of the nonlinear fluid behaviour (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 as a disadvantage that the force not only depends on the input current I, but also on the system velocity, and assumes that the damping coefficient C is about linear with respect to current.

Table 1: Magnetorheological models adopted

Name	$\frac{Schematic}{Equation}$	Force-Deformation function	System Type
Linear Bingham (Ma *et al.* (2003))	$\tau = G^{*}\gamma, \tau < \tau_y$ $\tau = \tau_y(H) + \eta\dot{\gamma}, \tau > \tau_y$ $F_D = C(I)v$		Linear
Bingham simplified (Stanway *et al.* (1985))	$F_D = F_c sgn(\dot{x}) + C_0\dot{x} + f_0$ 		Linear
Nonparametric polynomial model (Tianjun and Changfu (2009))	$F_D = \sum_{i=0}^{n} a_i v^i$ $F_D = \sum_{i=0}^{n} (b_i + c_i I) v^i$ 		Nonlinear
Bouc-Wen model of MRD - Extended (Spencer *et al.* (1997))	$F_D = \begin{cases} c_0(\dot{x} - \dot{y}) + k_0(x - y) + k_1(x - x_0) + \alpha z \\ c_1\dot{y} + k_1(x - x_0) \end{cases}$ $\dot{z} = -\gamma \mid \dot{x} - \dot{y} \mid z \mid z \mid^{n-1} - \beta(\dot{x} - \dot{y}) \mid z \mid^{n} + \delta(\dot{x} - \dot{y})$ 		Nonlinear

The Bingham simplified model is based on the fluid rheological behaviour. It consist in a Coulomb friction element and a viscous damper connected in parallel. In this model, C_0 is the damping coefficient , f_0 is a constant force (to compensate the non displacement effect at the beginning of the force applying due to the endzone) and F_c is the fluid yield strength. The Bingham fluid requires a high level of strength before start flowing. The relation between shear strength and deformation is described by Eq. 1:

$$\tau = \tau_0 + \eta\dot{\gamma} \tag{1}$$

Butz and von Stryk (1998) present the extended Bingham model. This visco-plastic model consist in a Bingham model connected in series with elements that represent a linear solid, which includes the subactuated position states x_1 , x_2, with the corresponding elastic and damping constants k_1, k_2 and c_1, as shown in Fig.4 and described in Eq.2.

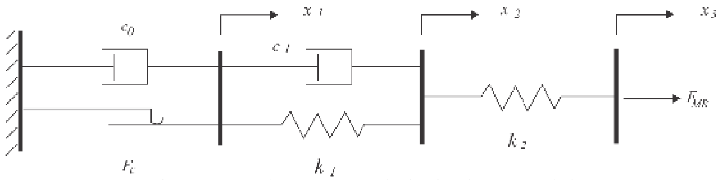


Figure 4: Scheme extended Bingham model

$$F_{MR} = \begin{cases} F_c sgn(\dot{x}_1) + C_0\dot{x}_1 & \\ k_1(x_2 - x_1) + c_1(\dot{x}_2 - \dot{x}_1) & , \mid F \mid > F_c \\ k_2(x_3 - x_2) & \\ & \\ k_1(x_2 - x_1) + c1\dot{x}_2 & , \mid F \mid \leq F_c \\ k_2(x_3 - x_2) & \end{cases} \tag{2}$$

Nevertheless, these equations do not represent the nonlinearities of a magnetorheological damper, that are produced by the existence of hysteresis.

On the other hand, there are complex models that represents MRD dynamics including nonlinearities, such as the called polynomial models, based on experimental processes, that perform a data acquisition of damping force and displacement, so that the chart force vs. velocity describes a hysteris cycle, subdivided in two regions: the Positive Acceleration Region (lower loop) and Negative Acceleration Region (upper loop), as shown in table1, both of them approached by a n-grade polynomial in terms of the piston velocity. The main advantage of this model is the ease for computational implementation, which could be useful for control purposes. Nonetheless, there are several disadvantages, because it is not physically parameterized and thus its coefficients do not describe any real parameter, being just a Black Box Model, i.e. an aproximation of the output can be obtained knowing the input value. Therefore, a linear relation is proposed between the current I and the coefficient a_i (Eq.3).

$$a_i = b_i + c_i, \quad i = 0, 1, \ldots.., n \tag{3}$$

2.1 Bouc-Wen model of MRD

According to Ikhouane and Rodellar (2007), this model is the reference point of the phenomenological models for magnetorheological dampers. As its name indicates, is based on the inclusion of the Bouc-Wen model of hysteresis, which effectively considers the system nonlinearity. It is also important to check if the sintonized parameters assure the two basic compatibility properties between the model and the physical laws (having input and output boundaries, and describing energy dissipation), according to Ikhouane and Rodellar (2007).

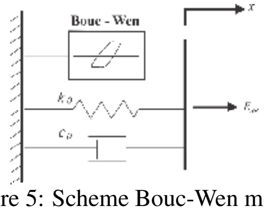


Figure 5: Scheme Bouc-Wen model

As presented in Fig.5, the output MRD force could be described as the sum of the following concepts:
1. The damping friction produced by the seals and the measurement bias.
2. The product of the mass, the inertial effects and the piston acceleration.
3. The product of the piston velocity and the plastic damping coefficient (post-yield), $c_0\dot{x}$.
4. The product of the piston position and the elasticity coefficient, $k_0(x - x_0)$.
5. The hysteresis term, αz.

Thus, the force generated by the damper is described as in Eq.4:

$$F_{MR} = c_0 x + k_0(x - x_0) + \alpha z \tag{4}$$

The time derivative of the hysteresis component is given in Eq.5:

$$\dot{z} = -\gamma \mid \dot{x} \mid \ z \ \mid z \mid^{n-1} - \beta \dot{x} \ \mid z \mid^n + A\dot{x} \tag{5}$$

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 measure in a MRD and hence allow to obtain the requirements for characterization and testing bench sizing.

3. GENERAL PROPOSAL ARCHITECTURE

The purpose of this device is being modular and configurable, allowing to identify and assess control strategies on the 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 to evaluate the MR damper dynamics. This is necessary for the appropiate electronical instrumentation and mechanical sizing in the experimental assembly, minimizing frictions in the actuator movement direction and assuring a good parameter estimation.

The magnetorheological damper RD-8040-1 Lord® has the following features:

- Stroke length = 55mm.
- Maximum damping force (peak-to-peak) = 2447N, at a linear velocidad = 5cm/s .
- Maximum damping force (peak-to-peak) < 667 N, at a linear velocidad 20cm/s .
- Maximum current =1A.

A correct identification requires a position variation in an ideally sine wave, which allows to study the damper complete stroke in both directions, in order to analyze the hysteresis.

There are two basic setups in the bench design process related with the operation modes: the identification setup and the control setup, which tests the identified system with a control strategy. Therefore, the testing bench must be modular and allow to adapt different attachable accesories (masses, springs, etc) and the electronical instrumentation.

It must be feedback of position, reaction force and current (control signal). Considering the above, the sensors listed in the table 2 were required, and were selected given the damper operation ranges and the state variables that should be monitored. For acquiring those sensor signals, it is used the National Instruments DAQ USB 6216 acquisition system, with a sampling rate of 25 Hz, which allows variations on the mechanical system dynamics,including the sampling theorem consideration by Nyquist-Shannon: the sampling time must be at least a half of the system response time, although it is common to use a tenth part in order to have integrity in the signal (Skoog (2008)). The complete system architecture is presented in Fig.6.

Table 2: Implemented Electronic Instrumentation

Sensor	Reference	Range	Sensitivity	Function
Current	ACS714	±5 A	185±5 mV/A	Sensing the control signal of the magnetorheological damper
LVDT	HC Metrolog	100 mm	0,01 ±0,005 mm/V	Feedback the mass or damper position
Acelerometer	MMA7361L	±1,5 g	800mV/g	Feedback the disturbance position to the control system. In function of the servomechanism angle
Load Cell	Lexus ¨SA Model¨	±100 kg	2±0,2mV/V	Feedback the reaction force generated by the MRD

3.1 Servomechanism system

An essential step in the testing bench design, for both operation setups, is the active actuator selection. The main required features are travel speed, force and type of control, being the minimum requirements to perform tests on magnetorheological dampers. It is necesary to have an independence between the applied force and the travel speed, considering that it is required an input with a sine waveform and a specific operation range. Besides, if there are speed and force requirements, for instance 5cm/s and 667N, the actuator should perform the test in such a way that it mantains a constant force, regardless the travel speed.

Regarding 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 travel speed is reached by means of both pressure and discharge servovalves. Also, this speed is restricted by the control system capabilities for driving the nonlinearities and the implicit delays of a pneumatic system (Jimenez *et al.* (2012)). This leads to use a rack and pinion system coupled to an Animatics SM34165DT servomotor, based on the functional features listed in table 3.

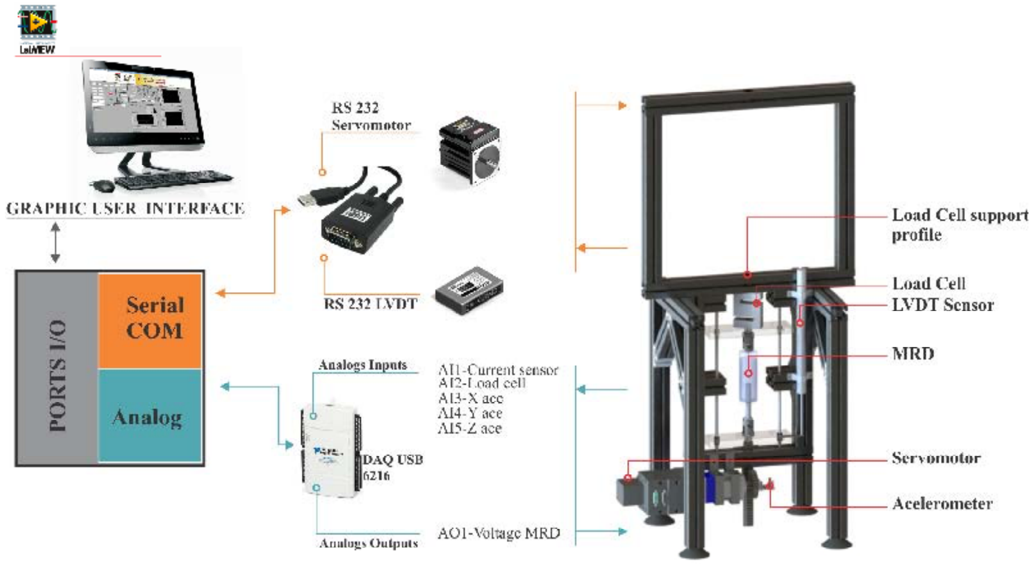


Figure 6: General Proposal Architecture of Testing Bench

Table 3: Animatics servomotor features

NEMA SM34165DT		
Continuous Torque	1,45	Nm
Peak Torque	3,39	Nm
No Load Speed	5100	RPM
Continuous Current	15,5	A
Encoder Resolution	8000	Count/Rev

Considering that the rack and pinion system has 1-DOF, it is necessary to generate a translational motion beneath the MR damper work range. It was selected a linear force of 1800N and a speed of 20cm/s as the input parameters, taking into account a safety factor of 2,25 in respect to the necessary load for the identification tests (up to 800N).

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

$$F_l = \frac{\tau_m}{(\frac{D_p}{2})}(\xi \cdot R) \tag{6}$$

$$P_c = \frac{D_p[mm]\pi}{N_t[teeth]} = 6,28\,\frac{mm}{teeth} \tag{7}$$

Thus, the rack displacement per pinion revolution is defined in Eq.8:

$$D_l = N_t[\frac{teeth}{rev}] \cdot P_c[\frac{mm}{teeth}] = 32 \cdot 6,28 = 200,96\,\frac{mm}{rev} \tag{8}$$

Finally, the rack maximum linear speed is as follows:

$$V_l = \frac{\omega_M[RPM]}{R} \cdot D_l[\frac{mm}{rev}] = \frac{2500}{40} \cdot 200,96 = 20,93\,\frac{cm}{s} \tag{9}$$

In addition, although the designed servomechanism achieves a constant power transmission and a translational motion, it is necessary to control its 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, as shown in Fig. 7,

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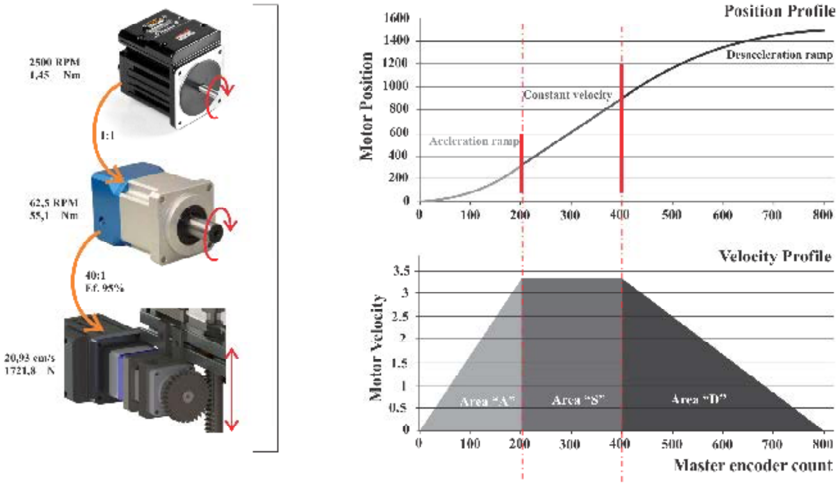
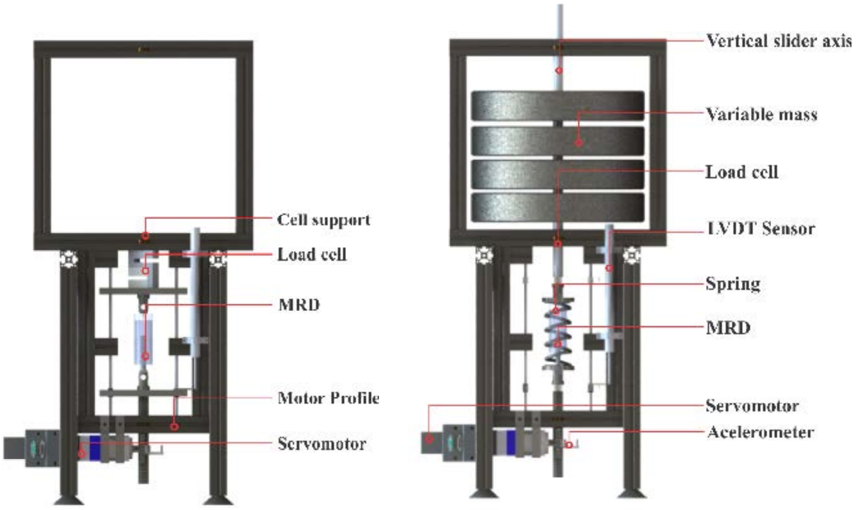


Figure 7: Servomechanism, positions and velocities ramps Rampa de posición y velocidad servomecanismo



(a) Identification Mode

(b) Control Mode

Figure 8: Mode setups

3.2 Proposal setups

The both setups are shown in Fig.8 and described below.

Identification Mode

This setup requires generate a reciprocating motion that describes a sine waveform on time, with a changeable frequency between 0,5 and 2 Hz, so as to concurrently acquire force, position and electrical current data, and then reconstruct the charts Force vs time, Velocity vs time and force vs velocity. For this, an experimental procedure must be established for obtaining the resulting force F_{MR} from the velocity and electrical current supplied to the MR actuador.

This design restricts motion for a 1 DOF system, minimizing friction and assuring the displacement ranges for the Magnetorheological damper (MRD) dynamics identification. As the maximum load would be 800N, the load cell is fixed to a testing bench end (reference point) and the other one to the actuador, so that the measured load is F_{MR}. The LVDT sensor measures the position variation produced by the servomechanism, meanwhile the current sensor measures the control signal for the MRD. All variables are centralized and monitored from the graphical user interface developed in Labview®, which further information is provided in subsection 4.1

A mechanical outline of the proposed setup is shown in Fig.8a.

Control Mode

As it was said before, the controller was designed for vibration attenuation in a quarter car active suspension system, using measurements of vertical positions of the car body and the tire, so that the damper performance can be assessed, even after parameter variations, such as spring stiffness K and mass M. Then, it is possible to emulate several operation conditions, by means of step, ramp, triangle, sine and random inputs, and even position disturbances that simulate land discontinuities. All that in order to obtain a tool for design and testing of various types of controllers, e.g. digital RST, clasical PID, adaptive, predictive, etc. (Santos *et al.* (2009)).

Fig.3 shows the suspension system outline, in accordance to the control setup in Fig.8b. In this case, the LVDT sensor measures the position Z_2 of mass M_2. Besides, the rack and pinion mechanism was restricted to a linear displacement of 5cm so that the pinion does not exceed the 90° of angular displacement, which allows the use of an accelorometer to measure the position disturbance Z_1 as a linear relationship with the Roll angle (same rack rotation axis).

The simplified dynamics of the active suspension system (Fig.3) was proposed by Chavez *et al.* (2009), ignoring the MRD internal friction and the implicit nonlinearities, by means of the following equations.

$$m_1\ddot{Z}_1 + k_s(Z_1 - Z_2) = F_{MR} \tag{10}$$

$$m_2\ddot{Z}_2 - k_s(Z_1 - Z_2) + k_u(Z_2 - Z_t) = -F_{MR} \tag{11}$$

4. RESULTS

The experiment was carried out on a configurable structure that allows to sense the system state variables and generate an input for dynamics evaluation in terms of force and displacement work ranges, in addition to sustain a sine waveform during the system identification process.

The testing bench at the identification mode is presented in the Fig.9 (the preliminar test for velocity variation are explained further in subsection 4.1). The entire mechanical design was made through the CAD (Computer Aided Design) software Solidworks®, validating the load-bearing elements with Finite Element Analysis (FEA), such as the cell support profile, which must assure sufficient stiffness to avoid higher deformations than $3e^{-4}$mm, as a consequence of applied loads.

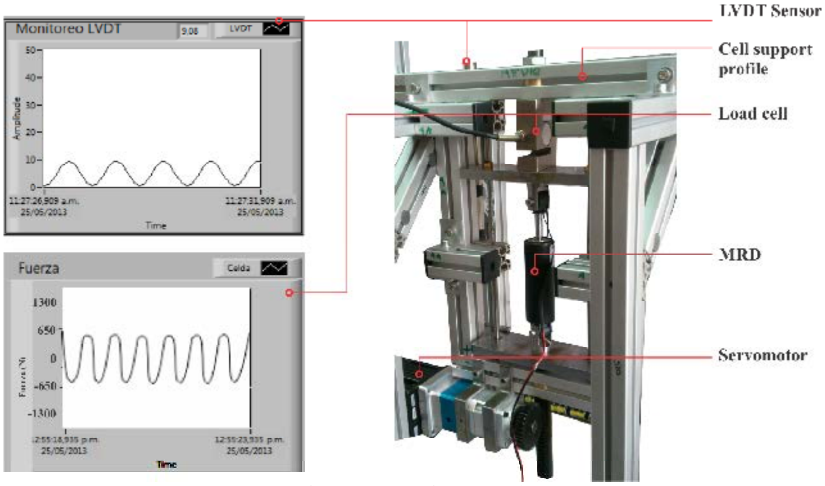


Figure 9: Testing Bench

4.1 Graphical User Interface

The application developed in Labview® works as a graphical user interface (GUI) from which the testing bench activity can be monitored and the operation modes (identification and control) can be enabled and disabled. Likewise, the GUI allows the data acquisition and the identification process needed for the plant control system.

The first block of the Fig.10 corresponds to the serial port configuration module, intended to set the communication settings between the LVDT sensor and the servomotor, including port name, baud rate, use of stop and parity bits, data number of bits and flow control.

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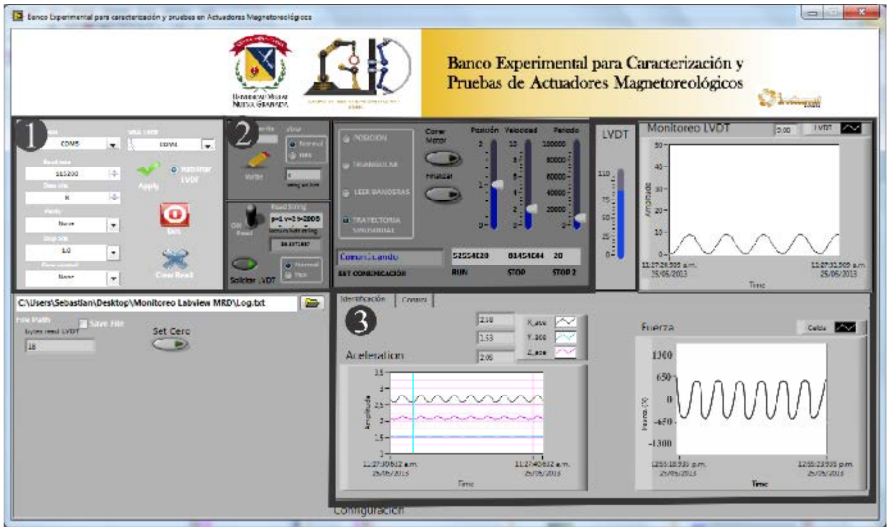

Figure 10: Graphical User Interface

Once the communication is established, the parameter variation for servomotor operation can be done at the block 2, i.e. setting the desired position and velocity of an applied step or the frequency and waveform (sine or triangle) of a periodical signal, as shown in Fig.11. The sine wave, in the range of is 0,5-2Hz, assures a maximum displacement of 25mm and the minimum required force of 800N which depends on the power supply (37V - 10A).

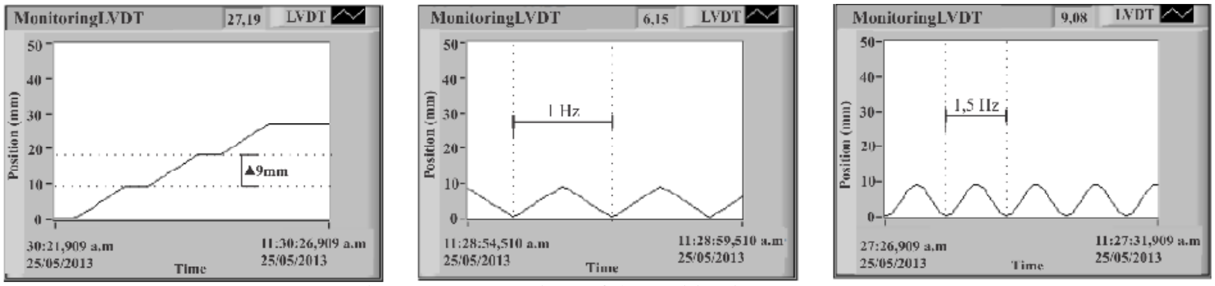

Figure 11: Comparison of the position input type

Finally, the block 3 allows the user to monitor the acquired signals by the LVDT sensor (linear displacement), the acelerometer, the current sensor and the load cell (force), according to the testing bench setup. The sensors response time and the sampling rate of the NI USB-6216 DAQ module allows to observe the system response. The chart force vs. time (block 3 in the Fig.10) shows the result of a test performed with a 10mm displacement and a 1,5Hz frequency sine wave, obtaining forces up to 650N and no current induction to the magnetorheological damper.

5. ACKNOWLEDGEMENTS

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