DAMP-BY-WIRE : MAGNETORHEOLOGICAL VS. FRICTION DAMPERS

Emanuele Guglielmino*[1], Charles W. Stammers*[2], Kevin A. Edge*,
Tudor Sireteanu#, Danut Stancioiu#

* *University of Bath, Dept. of Mechanical Engineering, UK*
# *Romanian Academy, Institute of Solid Mechanics, Bucharest, Romania*

Abstract: A comparison is made of a magnetorheological damper (MRD) and a friction damper (FD) in semi-active suspension design; both devices are closed-loop controlled. Experimentally validated models of both dampers have been developed and a hybrid variable structure-fuzzy control algorithm designed. It is shown that the two devices, albeit based on very different physical principles, can be described by a general common mathematical model, with the FD model being regarded as a particular case of the model for a MRD. The control strategy is targeted to improve occupant comfort. Performance has been investigated numerical simulation in a quarter car vehicle model and compared to that obtained with a classical passive damper. *Copyright © 2005 IFAC*

Keywords: vibration, dampers, friction, fuzzy control.

1. NOTATION

B	Hydraulic oil bulk modulus
c_0	MRD equivalent viscous damping
C	Road roughness coefficient
C_q	Valve discharge coefficient
D	Valve bore diameter
F	Frequency
f_c	Road roughness roll-off frequency
f_0	MRD equivalent constant force
F	Control force
G	Road roughness spectral density
i	MRD solenoid current input
k_0	MRD equivalent stiffness
k_s	Suspension stiffness
k_t	Tyre stiffness
m_1	Sprung mass
m_2	Unsprung mass
P_s , P_A	Supply and actuator pressures
Q_c	Compressibility flow in line
Q_p	Pump flow
Q_{rv}	Flow past the relief valve
Q_1, Q_3	Flow past the control valve
Q_2	Compressibility flow in actuator
s	Valve spool position
u	Valve lap
x	Suspension relative displacement
x_1	Sprung mass displacement
x_2	Unsprung mass displacement
V_t	Actuator chamber volume
z	Bouc-Wen evolutionary variable
z_0	Road profile displacement
A, α, β, γ, n	Bouc-Wen model coefficients
δ	Control gain
λ	MRD damping force gain
μ	Friction coefficient
ρ	Hydraulic oil density

2. INTRODUCTION

Semi-active suspension systems have been shown to perform comparably well to active suspensions. This performance is achieved with a smaller power consumption, simpler design and overall reduced costs. A review can be found in Crolla (1995).

The present authors have explored the use of semi-active devices in terrain vehicle suspension systems. Initially they investigated the feasibility of using friction dampers (FDs) in automotive applications (Stammers and Sireteanu, 1997; Guglielmino *et al.*, 2000; Guglielmino and Edge, 2001) and most recently they have focussed their attention on magnetorheological dampers (MRDs) (Sireteanu *et al.,* 2002).

A MRD damper uses an oil which contains micron-embedded solenoid creates a magnetic field which induces particles to align and form chains which

[1] Now with GE Energy (Florence, Italy)
[2] Corresponding author. Phone +44 1225 385962, fax +44 1225 386928, email: enscws@bath.ac.uk

results in a modification of the fluid yield stress. In such a state, rheological properties of the oil change and the fluid passes from liquid state to semi-solid state. A detailed analysis of MR fluid properties can be found in Agrawal *et al.* (2001). By controlling solenoid current a continuously variable damping is produced without employing moving parts, such as variable orifices, and with low energy requirements: for control purposes, it is only necessary to supply the solenoid from a conventional battery. As far as reliability is concerned, MRDs are fail-safe systems: in case of sudden loss of control, they behave broadly as conventional viscous dampers.

The main domains of application of MRDs are in the automotive and structural fields. In the latter, they are employed for earthquake protection and for damping wind-induced oscillations of bridges and flexible structures (Dyke *et al.*, 1996). In the automotive field they are employed in semi-active suspensions. MRD-based semi-active suspensions are used on some high-segment market cars.

A FD is essentially composed of an actuator with a friction pad and a metallic plate, connected to a mass with a motion relative to the actuator. An external normal force is applied by the actuator to the mass *via* the pad and, consequently, because of the relative motion, a frictional damping force is produced. FDs can be found in structural applications (Nishitani *et al.*, 1999), in freight train suspensions and in some rotating machinery. Generally these are non-controlled (i.e., there is no external control of the friction force).

With a controlled FD, a broader range of application areas becomes possible, including rotating machinery, seat suspensions and suspensions for agricultural vehicles. Moreover, existing friction damper-based systems could be retrofitted with a closed-loop control system, with a net improvement in the performance. For the specific case where hydraulic actuation is employed, the actuator is pressure-controlled. This means that the friction force can be scheduled as a function of feedback sensor signals, and hence it is possible to obtain any desired generalised damping characteristics as a function of any combination of displacement, velocity and acceleration.

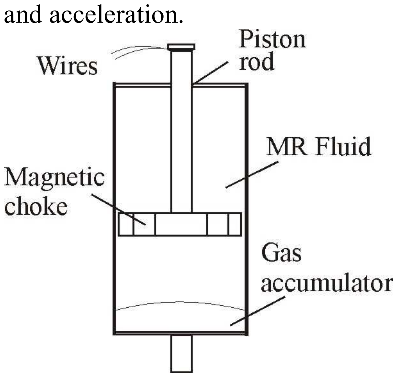

Fig. 1. Schematic of the MRD.

In this work a comparative study of a MRD and a FD in a semi-active vehicle suspension is presented. The system performance is assessed in a quarter car model. The controller has been designed to reduce vertical acceleration in order to improve occupant comfort.

3. MRD AND FD MODELLING

The MRD and the FD considered for this work are now briefly described. The MRD is a commercial off-the-shelf damper, sketched in Figure 1. It is essentially a monotube damper filled with MR oil and with the solenoid mounted on to the piston. A gas accumulator provides a nominally constant offset force. The accumulator allows for the change in volume available for the fluid and also for thermal expansion of the fluid.

The FD considered here has been designed and developed as a prototype by modifying an original viscous damper of a vehicle (Figure 2). The frictional coupling has been obtained with a piston in a cylindrical housing which contains two diametrically-opposed pistons upon which are bonded the friction pads. The damper is controlled by an electrohydraulic control drive. The pistons are supplied with hydraulic oil through the centre of the piston rod, with the control valve mounted externally.

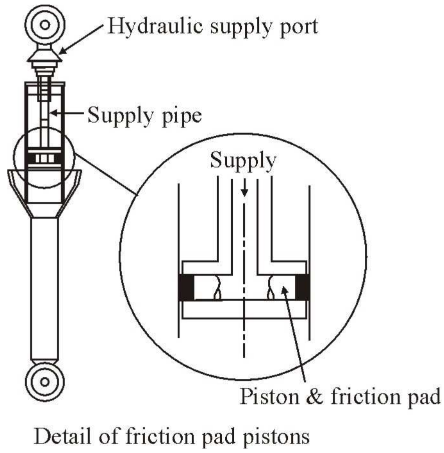

Fig. 2. Schematic of the FD.

In order to identify a model to support controller design, a bench test assessment of the static characteristics of both devices was undertaken on a hydraulically-powered shaker. A series of tests was conducted to measure their responses.

Figures 3 and 4 depict the force *vs.* velocity static characteristics of the MRD and FD respectively. Both dampers contain some degree of hysteresis. In the MRD this is due to fluid internal dissipation, whereas in the FD is is due to the "frictional memory" effect (Armstrong *et al.*, 1994). Both characteristics also exhibit some residual nonlinear damping of the mounting rubber bushes. Hence MRD and FD static characteristics are relatively similar. Therefore, for purposes of generality, it is reasonable to develop a model which incorporates the common features of both MRD and FD characteristics. A Bouc-Wen model has been used which can capture a wide variety of hysteretic behaviour (Wen, 1976).

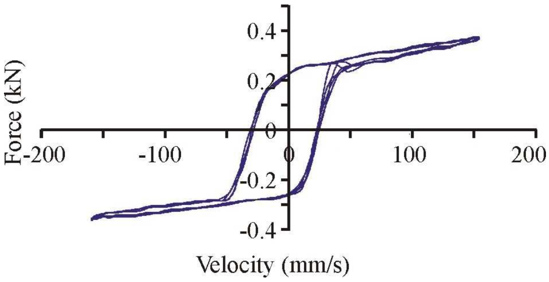

Fig. 3. MRD static characteristics.

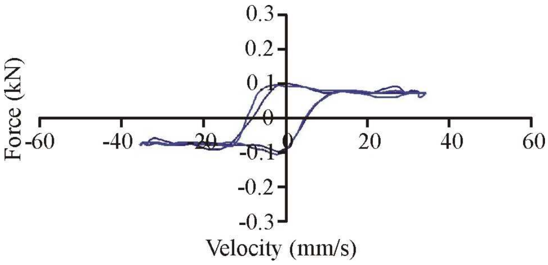

Fig. 4. FD static characteristics.

The MRD (Figure 5) is modelled as a hysteresis block in parallel with a linear elastic element and a dashpot, while the FD only requires a hysteresis block, because at higher velocities damping is constant and dependent only on the sign of velocity. Hence from a mathematical viewpoint the FD model can be considered as a particular case of that for an MRD.

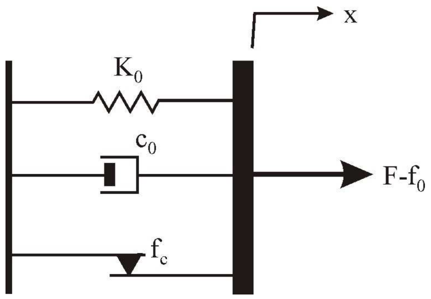

Fig. 5. Schematic of MRD model.

The MRD model that captures the experimental response is:

$$F(x,\dot{x},i) = a(i)z + c_0(i)\dot{x} + k_0(i)x + f_0 \quad (1)$$

$$\dot{z} = -\gamma|\dot{x}|\cdot z\cdot|z|^{n-1} - \beta\dot{x}\cdot|z|^{n} + A\dot{x} \quad (2)$$

where x is the damper displacement, i the solenoid current, α a yield stress dependent gain, c_0 a viscous damping coefficient, k_0 an elastic coefficient and f_0 a constant force term.

The FD model that captures the experimental response is:

$$F(x,\dot{x},P_A) = \alpha(P_A)z \quad (3)$$

$$\dot{z} = -\gamma|\dot{x}-\dot{y}|z|z|^{n-1} - \beta(\dot{x}-\dot{y})|z|^{n} + A(\dot{x}-\dot{y}) \quad (4)$$

where the variable P_A is the actuator pressure.

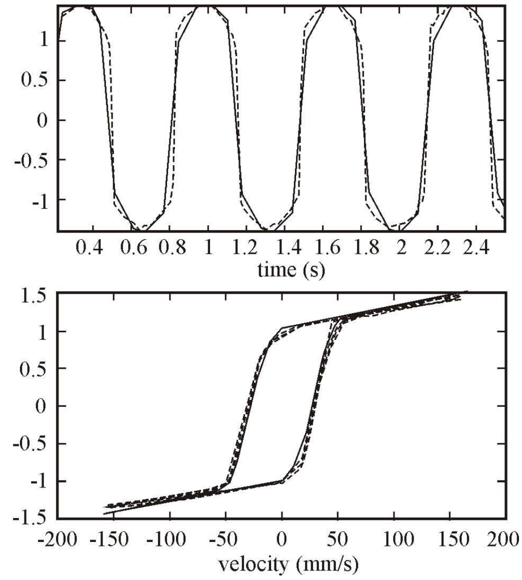

Fig. 6. MRD damping force [kN] response; experimental (dotted) and simulated (solid).

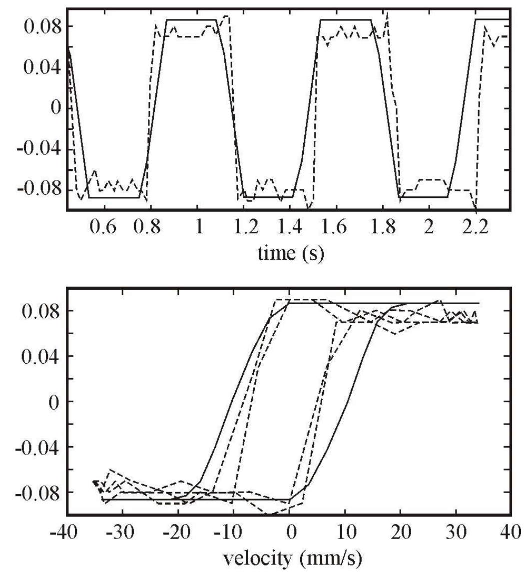

Fig. 7. FD damping force [kN] response; experimental (dotted) and simulated (solid).

Model parameters have been found using a genetic algorithm inverse method for the MRD and a least squares approximation for the FD. In Figures 6 and 7 experimental and simulated trends are superimposed. MRD stimulus is sinusoidal at a frequency of 1.5 Hz, with the MRD solenoid supplying a constant current of 0.6 A, whereas the FD input is sinusoidal of the same frequency with a supply pressure of 10 bar. Both models provide a good prediction of the experimentally measured responses. Less sophisticated models exist which do not take into account the low-speed hysteric behaviour, such as the Bingham plastic model (Sireteanu *et al.*, 2002) which, for the FD model, would degenerate into the simple Coulomb friction model. The disadvantage of such a model is the poor representation of the hysteretic behaviour of the force-velocity loops at low velocities.

Under feedback control, damper responses rely upon drive dynamics. Therefore an accurate model of the drive is of paramount importance. The control variables are respectively current for the MRD, and actuator pressure for the FD. The MRD drive is constituted by a simple solenoid. If magnetic saturation is not reached, the solenoid electromagnetic dynamics are well approximated by a linear first-order model with time constant of 10 ms (Carlson *et al.*, 1996).

In the prototype FD, the hydraulic servo-actuator system is comprised of an electrohydraulic proportional underlapped valve which is supplied by a pump working at nominally constant pressure. The valve drives a single-chamber actuator under pressure control. The hydraulic valve essentially behaves in a manner analogous to a potential divider in an electrical circuit.

The FD hydraulic model introduces stronger nonlinearity due to the nonlinear flow-pressure characteristics of the valve. Figure 8 shows the equivalent hydraulic circuit.

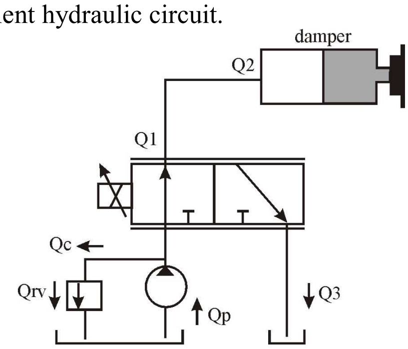

Fig. 8. FD hydraulic drive.

The pressure-valve opening characteristics can be obtained by consideration of flow continuity at the second node of the circuit:

$$Q_1 = Q_2 + Q_3 \tag{5}$$

$$Q_1 = C_q \pi D(u+s)\sqrt{2(P_s - P_A/\rho)} \tag{6}$$

$$Q_2 = \frac{V_t}{B}\frac{dP_a}{dt} \quad Q_3 = C_q \pi D(u-s)\sqrt{\frac{2P_A}{\rho}} \tag{7}$$

The dynamic model of the hydraulic valve is completed by the flow continuity at the valve supply port node:

$$Q_P = Q_1 + Q_c + Q_{rv} \tag{8}$$

Spool-solenoid electromechanical dynamics are closely approximated by a linear second-order linear differential equation.

By manipulation of equations (6)-(7), the non-linear static characteristic pressure-valve opening is obtained: it is saturation-shaped with the gradient dependent upon valve underlap and expressed by Merrit (1967):

$$P_A(s) = P_s \frac{(s+u)^2}{2(s^2+u^2)} \tag{9}$$

with $-u \le s \le u$.

A more detailed hydraulic model including leakage flows, valve spool and solenoid dynamics is presented in Guglielmino and Edge (2001).

If hydraulic dynamics are linearised, the system hydromechanical response can be expressed by a first order model with a hydraulic time constant of 40 ms and spool dynamics by a second order model with a natural frequency of 105 Hz and damping ratio of 0.60.

In terms of power consumption the required power to drive the MRD is only 24 W, provided by a 12 V-2A power supply, whereas in the case of a FD, the power required is higher. Calculations show that the average power required is higher. Calculations show that the average power required is about 500 W per damper for FD, but still far smaller than the power required by a fully active system.

4. CONTROLLER DESIGN

The starting point in the MRD controller design was a previous study on FD control (Stammers and Sireteanu, 1997). A variable structure controller was designed for the FD, targeted to reduce sprung mass acceleration. The aim is pursued by reducing the forces transmitted to the chassis by generation of a spring-like control force, at predfined time intervals (computed using Lyapunov stability theory) which is proportional to the suspension vertical deflection and with opposite sign to spring force:

$$F(x,\dot{x}) = \begin{cases} \delta k_s |x| \operatorname{sign}(\dot{x}) & \text{if} \quad x\dot{x} \le 0 \\ 0 & \text{if} \quad x\dot{x} > 0 \end{cases} \tag{10}$$

where δ is a gain term proportional to friction coefficient μ. Such a control logic can be classified as variable structure control (VSC) and it is robust, in a control sense (Itkis, 1976). However, it is a switching logic and this may cause chattering problems when the controller switches from one state to the other which can worsen ride quality. This is even more critical for the MRD which has faster dynamics than the FD (where the hydraulic dynamics help smooth the switching action). The problem can be tackled at control level by the introduction of fuzzy logic which softens the fast switching action without the need for low-pass filters which would reduce system bandwidth.

The VSC algorithm has been fuzzified by choosing as fuzzy variables normalised relative displacement and velocity; the linguistic variables are: negative (neg) and positive (pos). The membership functions are depicted in Figure 9.

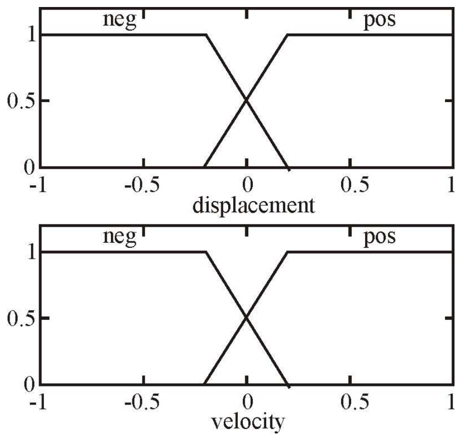

Fig. 9. The fuzzy logic membership functions.

The fuzzy controller function is a 2-value function $i(\dot{x}(t), x(t)) \in [0, i_{\max}]$ (Small (0) and Big (2)). The fuzzy set rules (Table 1) have been obtained by fuzzifying controller (11). In this way, the transition between the two structures defined by (11) is continuous.

Table 1. The fuzzy logic rules

Velocity / Displacement	Neg	Pos
Neg	SMALL	BIG
Pos	BIG	SMALL

5. VEHICLE AND ROAD MODEL

Damper performance has been assessed for a quarter car (Figure 10).

The quarter car model of a typical saloon car moving with constant speed has been considered. The equations of motion are:

$$\begin{aligned} m_1\ddot{x}_1 + F(x_1 - x_2, \dot{x}_1 - \dot{x}_2, i) + k_s(x_1 - x_2) &= 0 \\ m_2\ddot{x}_2 + m_1\ddot{x}_1 + c_2(\dot{x}_2 - \dot{z}_0) + k_t(x_2 - z_0) &= 0 \end{aligned} \quad (11)$$

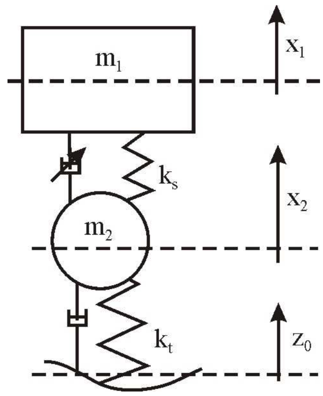

Fig. 10. Quarter car model.

The output variables of interest are the sprung mass absolute acceleration response expressed as a comfort index $\sigma_{\ddot{x}_1}$ (sprung mass r.m.s.) and the unsprung mass relative displacement response expressed as a road holding index $\sigma_{x_2 - z_0}$ (unsprung mass displacement r.m.s., relative to the road).

Two road roughness descriptions were used to simulate the quarter car model input excitation (Sireteanu *et al.*, 2000) corresponding to a "good" road profile and a "poor" road profile. The first has a spectral density described by the following equation:

$$G(f) = \frac{C}{v} \cdot \left(\frac{f_0}{f}\right)^{n_j} \; n_j = n_1 \; f \le f_0, \; n_j = n_2 f > f_0$$

where v is the vehicle speed and $2 \ge n_1 > n_2 > 1$.

The proposed road excitation model with this spectral density is:

$$z_0(t) = \sum_{k=1}^{n} \alpha_k \sqrt{2G(f_k) \cdot \Delta f_k} \cos(2\pi \cdot f_k t + \theta_k)$$

where f_k frequencies are initially imposed, and the weighting coefficients α_k as well as the random phases θ_k were generated using computed spectral density (14) and the theoretical one (13). The case considered here is for $C = 2.10^{-5}$ and with a r.m.s. surface roughness of 0.012 m.

The second road excitation corresponds to a spectral density expressed by:

$$G(f) = 4\sigma^2 \left[\frac{a \cdot (4\pi^2 f^2 + a^2 + b^2)}{(4\pi^2 f^2 - a^2 - b^2)^2 + 16\pi^2 b^2 f^2} \right]$$

where parameters *a, b* are imposed by the simulated road profile quality and depend on vehicle speed. The proposed roughness employs the spectral density given by (15) and it is a transfer function of a system with white noise input. The following parameters have been used: a = 0.22 m^{-1}, b = 0.44 m^{-1} and road r.m.s profile of 0.012 m.

6. NUMERICAL RESULTS

The values of the control parameters that minimise the performance indices were obtained for both the controlled system and for an equivalent classical passively-damped system. The simulation results are summarised in Table 2 and show that a reduction in the comfort index $\sigma_{\ddot{x}_1}$ has been obtained with both controlled dampers in the case of a poor road profile: MRD and FD achieve equivalent performance. Road holding index $\sigma_{x_2-z_0}$ values are fairly similar in both passive and semi-active cases. However, with the MRD a slight improvement is obtained whereas the FD has produced slightly worse results. This is mainly related to the different dynamic response of the two dampers. The slight increase in this index is a consequence of the control logic which is aimed to improve comfort.

Table 2. Comparison of passive and semi-active systems

Vehicle speed [km/h]	$\sigma_{\ddot{x}_1}$ [m/s^2]	$\sigma_{x_2-z_0}$ [mm]	$\sigma_{\ddot{x}_1}$ [m/s^2]	$\sigma_{x_2-z_0}$ [mm]
	Poor road profile		Good road profile	
Passive damper				
50	1.40	2.8	0.49	0.9
90	1.82	3.5	0.60	1.1
Semi-active MRD				
50	0.69	2.4	0.36	0.9
90	0.73	3.4	0.48	1.1
Semi-active FD				
50	0.65	3.0	0.38	0.9
90	0.65	4.1	0.48	1.1

7. CONCLUSIONS

Experimentally-validated models of a MRD and a FD have been obtained based on the Bouc-Wen equation. The merits of the controlled systems over the passive case are most evident with a poor road profile. Overall the MRD performs better than the FD; this is due to its higher bandwidth response. MRDs offer undoubted benefits in suspension design, both in terms of ease of design and control logic implementation as well as power consumption and reliability. Controlled FDs require a more involved hydraulic drive for their control. However, they could offer benefits in some applications, particularly in those applications where a passive FD is already present to which a semi-active control scheme can be retrofitted.

Future work will address the investigation of a hybrid system based on a magnetorheologically-controlled friction damper, which should merge the benefits of the two systems.

8. REFERENCES

Agrawal A., Kulkarni P., Vieira S.L. and Naganathan N.G. (2001), "An overview of magneto- and electro-rheological fluids and their applications in fluid power systems", *International Journal of Fluid Power,* **2**, pp. 5-36.

Armstrong-Helouvry B., Dupont P. and Canudas de Wit C. (1994), "A survey of models, analysis tools and compensation methods for the control of machines with friction", Automatica, **30 (7)**, pp. 1083-1138.

Carlson J.D., Catanzarite, D.N. and St Clair K.A. (1996), "Commercial Magneto-Rheological Fluid Devices", *Proc. 5th Int. Conf. on ER Fluids, MR Suspensions and Associated Technology*, pp. 20-28.

Crolla D A (1995) "Vehicle dynamics-theory into practice". *Proc ImechE Journal of Automobile Engineering,***209(4)**, 671-684

Dyke S.J., Spencer B.F., Sain M.K. and Carlson J.D. (1996), "Modelling and control of magnetorheological dampers for seismic response reduction", *Smart Materials and Structures*, **5**, pp. 565-575.

Guglielmino E., Stammers C.W. and Edge K.A. (2000), "Robust force control in electrohydraulic friction damper systems using a variable structure scheme with non-linear state feedback", *2nd Internationales Fluidtechnisches Kolloquium,* Dresden, Germany, pp. 163-176.

Guglielmino, E., Edge, K.A. (2001), "Modelling of an electrohydraulically-activated friction damper in a vehicle application", *ASME IMECE 2001*, New York, USA, November 2001.

Itkis Y. (1976), "*Control systems of variable structure*", John Wiley & Sons, New York.

Merritt, H E. (1967), "*Hydraulic Control Systems*", John Wiley & Sons, New York.

Nishitani A., Nitta Y., Ishibashi Y. and Itoh A. (1999), "Semi-active structural control with variable friction dampers", Proc. of the American Control Conference, San Diego, USA, pp. 1017-1021.

Sireteanu T., Stancioiu D. and Stammers C.W. (2002), "Use of Magnetorheological fluid dampers in semi-active seat vibration control", *Active 2002*, Southampton, UK.

Sireteanu T., Stancioiu D. and Stammers C.W. (2000), "Semi-active vibration control by use of magneto-rheological dampers", *Proc. Rom. Acad.*, Series A, **1**, No. 3/2000, pp.195-199.

Stammers C.W. and Sireteanu T. (1997), "Vibration control of machines by using semi-active dry friction damping", *Journal of Sound and Vibration*, **209** (4), pp. 671-684.

Wen Y.K. (1976), "Method of Random Vibration on Hysteretic Systems", *Journal of Engineering Mechanics, ASCE*, **102**, pp. 249-263.