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EXPERIMENTAL VERIFICATION OF SEMI-ACTIVE MAGNETORHEOLOGICAL DAMPERS FOR EARTHQUAKE STRUCTURAL CONTROL

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ABSTRACT:

Magnetorheological (MR) dampers represent one of the most promising semi-active control devices for applications aiming at the seismic risk mitigation of new or existing civil structures. Two 30 kN MR dampers were experimentally tested subjecting them to assigned conventional displacement laws at their ends. The paper describes the main results of the above experimental activity and aims at improving the knowledge about mechanical and dynamical modeling of such devices. Three different phenomenological models are considered and compared one each other: a) an improved version of the Bingham model including the influence of the feeding current on the damping and friction force components; b) a widely adopted model, combination of viscous and elastic elements with an hysteretic part behaving as the Bouc-Wen law; c) a model superposing an hysteretic damper by a non linear viscous device.
The simplicity of using each model, the influence of the involved parameters and the capability in fitting the experimental results are discussed herein.

KEYWORDS: Energy dissipation, magnetorheological damper, smart materials

1. INTRODUCTION

Semi-active control systems represent one of the most promising innovative techniques for seismic protection of structures. Utilized as special devices whose dynamic properties can be real-time modified according to appropriated control algorithms, they lead to a substantial independence of the overall structural performance on the particular seismic input acting on the controlled structure. Magnetorheological (MR) dampers are time-varying properties devices able to achieve a wide range of physical behaviors using low-power electrical currents. Changing the current in the damper causes a very fast modification of the mechanical properties of the MR fluids, due to the particular magnetic field applied.
Modeling the strongly non-linear behavior of such devices represents a challenging issue. Many numerical models are available in literature, each one being based on a different philosophy and characterized by a different degree of complexity (for instance measured through the introduction of algebraic or differential equations, the use of few ore many parameters, and so on) and effectiveness.
Three different models are considered and compared herein, with reference to an experimental campaign conducted on two 30 kN MR dampers in the framework of the Italian *ReLUIS* research project, sponsored by the Italian Emergency Agency (*Dipartimento per la Protezione Civile*). The actual influence of the involved parameters and the capability of each model in fitting the experimental results are investigated.
The simplest model commonly used to describe the behavior of MR dampers is based on the properties of Bingham solids (Carlson et al., 2000). The force F_d in the device, derived through the analytical study shown in (Spizzuoco et al., 2001), can be expressed as the sum of two components, due to the fluid viscosity and to the magnetic field-induced yield stress, respectively:

$$F_d = C_d \cdot \dot{x} + F_{dy}(i) \cdot \mathrm{sgn}(\dot{x}) \tag{1}$$

In Equation (1), $\dot{x}$ is the relative velocity between the damper's ends, C_d the viscous damping constant, F_{dy} the variable plastic threshold controlled by the applied magnetic field which, in turn, depends on the current i in the coils inside the MR damper. By varying the current from zero to a maximum value, a wide range of plastic threshold values can be achieved. The Bingham model often does not satisfactorily fit the actual MR device's behavior. Occhiuzzi et al. (2003) proposed a modified version of the model obtained taking into account the variability of C_d with i. For such a model the best relationships $F_{dy}(i)$ and $C_d(i)$ are found herein with reference to the above-cited experimental tests.
The second considered model is a worldwide reference model for MR dampers. It has been derived from an experimental test campaign on two prototype MR dampers of different dimensions: a small-scale 0.3-ton (Spencer et al., 1997) and a full-scale 20-ton MR dampers (Yang et al., 2002). Several efforts of the researchers involved in such experimental activity have been addressed to the development of a numerical model able to describe the ″roll-off″ effect experimentally observed at small velocities in the force generated by the dampers. To better predict the damper's response in this region, a modified version of the Bouc-Wen model (Wen, 1976), through the addition of a dashpot and an elastic spring, has been proposed in (Spencer et al., 1997) and (Yang et al., 2002).
The third and last considered model for MR dampers is formulated by Weber et al., still not published but known to the authors thanks to private communications. Even if it deals with the behavior of MR dampers at zero current, it is effective for higher currents also. This model is obtained by superposing an hysteretic damper by a non linear viscous law, mainly trying to fit not only the force-displacement experimental trajectories, but also the force-velocity loops.
For the purpose of structural control design, the need of an accurate model for MR dampers has to be taken into account together with that of dealing with a limited number of parameters. A too much complex model, included in the more general numerical representation of the main structure hosting the devices, would result in inacceptable computational efforts.

2. PROTOTYPE MR DAMPERS

Two full-scale prototype semi-active MR dampers (Figure 1) have been designed and manufactured by the German company Maurer Söhne in the framework of the Italian *ReLUIS* research project, sponsored by the Italian Emergency Agency (*Dipartimento per la Protezione Civile*). Both devices have been experimentally tested in order to evaluate their mechanical behavior, subjecting their ends to assigned conventional displacements and measuring the corresponding forces value. The two prototypes shown practically the same behavior so that in the following everything will be referred to whatever one of them.

Figure 1. Full-scale prototype MR damper

The overall dimensions of the device are 675mm (length) × 100mm (external diameter) and its mass is about 16 kg. A maximum force of 30 kN can be developed along its longitudinal axis, whereas the presence of special spherical pin joints at both ends prevents the rise of bending, shear and torsional moment in the piston rod. The damper has a stroke of ± 25 mm, and the external diameters of the piston head and of the piston rod are 100 mm and 64 mm, respectively. A magnetic circuit composed by three coils, each of them with a resistance R = 1.11 Ω and an inductivity L = 92 mH, can generate the magnetic field in the device. The current in this circuit, in the range of i = 0÷3 A, is provided by a power supply commanded by a voltage input signal.

The MR dampers have been experimentally tested by using a 1200 kN actuator (Figure 2) at the laboratory of the Department of Structural Engineering of the Università degli Studi di Napoli Federico II (Naples, Italy), together with a 100 kN load cell, measuring the forces' value, and a LVDT transducer, measuring the displacement of the moving damper's end, during each test. The main electronic equipment used for the tests was an operational power supply from Kepco Inc. (New York, USA), model BOP 50-4 M, a real-time National Instruments CPU and a digital acquisition board (Figure 3).
Dynamic tests imposing harmonic displacement at the end of the device were performed (about five cycles per test), using each time a certain value of displacement amplitude (± 10 or ± 20 mm), a fixed frequency (0.5, 1.5, 3.0 Hz; the last one only for the ± 10 mm tests) and a preset current level (0, 0.9, 1.8, 2.7 A). For the sake of brevity, not all the tests results will be shown in the following.

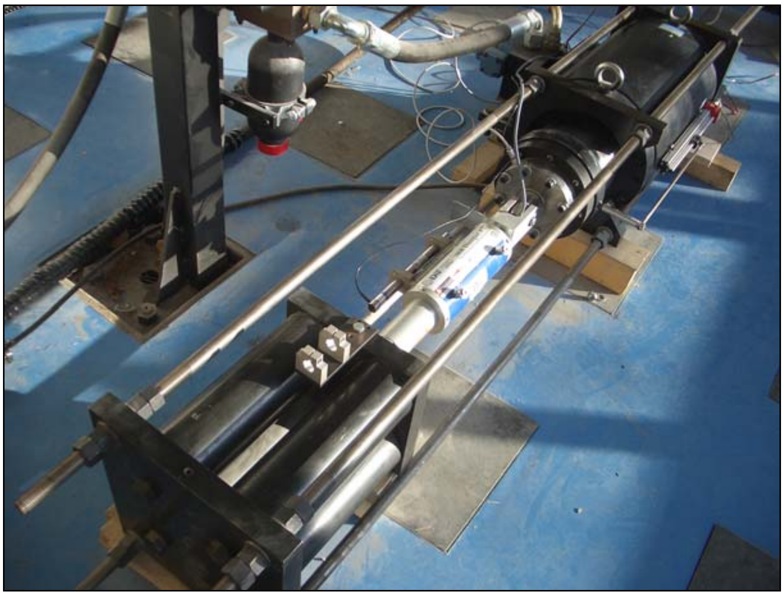

Figure 2. Experimental test set-up

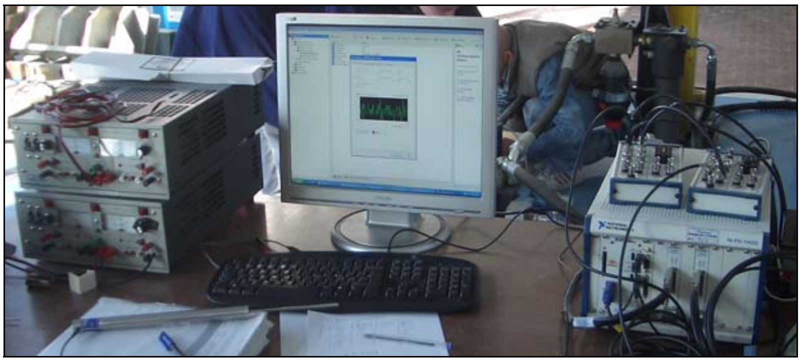

Figure 3. Electronic equipment for the tests

3. MODIFIED BINGHAM MODEL

This model, like the original one, considers the force in the damper as the sum of two components, due to the fluid viscosity and to the magnetic field-induced yield stress, respectively (Equation (1)). However, it takes into account the variability of the viscous damping constant C_d with i, besides that of the plastic threshold F_{dy}. The optimal value for C_d and F_{dy} was found for each test in order to achieve the better numerical fitting of the experimental data. For instance, with reference to harmonic displacement tests with amplitude ±20 mm, Figure 4 shows how the modified Bingham model works when the following parameters are set: $C_d(i\text{=0 A})$=5.5 kNs/m, $C_d(i\text{=2.7 A})$=27.0 kNs/m, $F_{dy}(i\text{=0 A})$=0.7 kN, $F_{dy}(i\text{=2.7 A})$=23.5 kN.

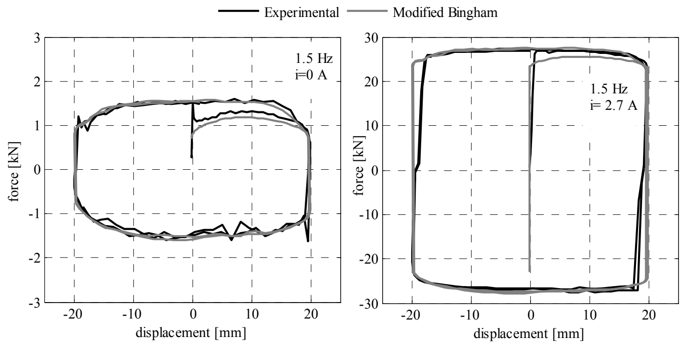

Figure 4. Experimental versus numerical (modified Bingham model)
cycles for harmonic displacement tests of amplitude ±20 mm

The optimal $C_d(i)$ and $F_{dy}(i)$ values resulted to be well predicted by the linear relationships (3) and (4), to be substituted in the (2) to calibrate the modified Bingham model ($\dot{x}$ is the relative velocity between the damper's ends):

$$F_d = C_d(i)\cdot \dot{x} + F_{dy}(i)\cdot \operatorname{sgn}(\dot{x}) \quad (2)$$

$$C_d(i) = 6.6 + 7.9i \quad (3)$$

$$F_{dy}(i) = 1.8 + 8.7i \quad (4)$$

Forces are expressed in kN, damping constant in kNs/m, current in Ampere, velocity in m/s.

4. SPENCER MODEL

The "Spencer" model indicates herein one of the most adopted worldwide model, formulated by Spencer et al. and described in Spencer et al. (1997) and Yang et al. (2002). This model gives special attention in modeling the ″roll-off″ effect experimentally observed at small velocities in the force generated by the dampers by the above research group. To better predict the damper's response in this region, a modified version of the Bouc-Wen model (Wen, 1976), through the addition of a dashpot and an elastic spring (Figure 5), has been proposed in Spencer et al. (1997) and Yang et al. (2002). The resulting model has ten parameters, or thirteen if a fluctuating magnetic field is considered.

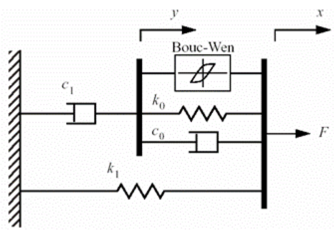

Figure 5. The Spencer model

The force generated by this reference model is the sum of the upper and lower sections shown in Figure 3, requiring an extra degree of freedom (DOF) y which actually does not exist:

$$F = c_1 \cdot \dot{y} + k_1 \cdot (x - x_0) \quad (5)$$

The artificial DOF y makes the damper c_1 work to simulate the roll-off effect at small velocities of the real DOF x. The elastic element k_1 accounts for a nitrogen accumulator inserted in the dampers tested in the USA (Spencer et al, 1997; Yang et al., 2002) to prevent cavitation and/or to compensate thermal expansion of the fluid, whereas x_0 represents an offset value of the real DOF x due to the accumulator. Imposing the internal equilibrium of the model leads to the following expression:

$$\dot{y} = \frac{1}{c_0 + c_1}\left[\alpha \cdot z + c_0 \cdot \dot{x} + k_0 \cdot (x - y)\right] \tag{6}$$

where c_0 models the viscous part of the behaviour of the MR fluid, k_0 accounts for the damper compliance and α is a scale factor needed to fit the experimental data by hysteretic loops governed by the evolutionary variable z, expressed as

$$\dot{z} = -\gamma \cdot |\dot{x} - \dot{y}| \cdot z \cdot |z|^{n-1} - \beta \cdot (\dot{x} - \dot{y}) \cdot |z|^n + A \cdot (\dot{x} - \dot{y}) \tag{7}$$

As in the original paper of Spencer et al. (1997), herein we set $n = 2$, $\beta=\gamma=200$ cm^{-2}, A = 207. The parameters k_1 , k_0 and x_0 has been set to 0 herein, because of their negligible contribution to the shape of the hysteretic loops. The three parameters α, c_0 and c_1 are depending on the current i. The constant c_1, able to reproduce the force roll-off effect, has been assumed simply much higher than c_0 in order to have the extra DOF "y" cancelled (c_0=10c_1 is assumed here). The optimal $\alpha(i)$ and $c_0(i)$ values, in terms of consequent agreement of the numerically evaluated results with the experimental ones (force-displacement loops), resulted to be well predicted by the relationships (8) and (9):

$$\alpha(i) = 2.41 + 12.61i \tag{8}$$

$$c_0(i) = 69.83 + 86.85i - 13.37i^2 \tag{9}$$

where α is expressed in kN/cm, c_0 in Ns/cm, current in Ampere.
Figure 6 shows, for instance, how the Spencer model fits the experimental data in terms of force-displacement loops with reference to two of the tests done.

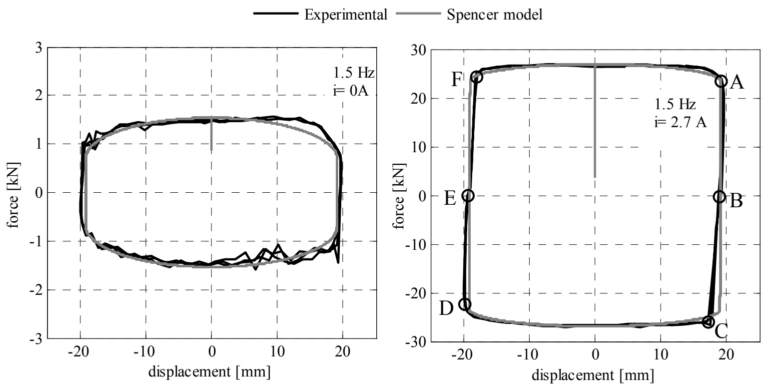

Figure 6. Experimental versus numerical (Spencer model) cycles for harmonic displacement tests of amplitude ±20 mm

5. WEBER MODEL

The third and last considered model is formulated by Weber et al., still not published but known to the authors thanks to private communications. The model tries to fit as closely as possible the behavior of MR dampers at zero current, but turned out to be effective for higher currents also. This model is obtained by superposing an hysteretic damper (f_h indicates the force in it) by a non-linear viscous law ($f_{v\text{-}nl}$ indicates the force in it), mainly trying to fit not only the force-displacement experimental trajectories, but also the force-velocity trajectories. By denoting with F_d the total force in the damper, the model becomes:

$$F_d = f_h + f_{v-nl} \tag{10}$$

Weber et al. define the f_h and $f_{v\text{-}nl}$ expressions focusing their attention on each single part of the force-displacement loops. For instance, they take into account the different damper behavior in the so called "pre-yield" region (corresponding to low velocities) and the "post-yield" region. In the first part of the pre-yield region (the offloading phase, where $x\dot{x}$>0, with x displacement, $\dot{x}$ velocity between the damper ends), the elastic energy stored during the last loading phase is recovered; non-dissipative active force acts in the damper leading to a practically vertical line in the force-displacement loop (see for example the A-B and D-E segments in Figure 6). In the second part of the same region, the system stores again energy until yield stress is reached and a finite slope k_h of the force-displacement trajectory is found (e.g. B-C and E-F segments in Figure 6). The cited Authors also modeled a so called "overshoot" effect neglected herein. Accounting for each of these considerations, Weber et al. define f_h and f_{v-nl} as follows:

$$f_h = \mathrm{sgn}(x)[f_{oh} - k_h(x_{\max} - |x|)] \tag{11}$$

when x is such that the point representing the damper state in the force-displacement loop lies in the second part (elastic) of the pre-yield region;

$$f_h = \mathrm{sgn}(\dot{x}) f_{oh} \tag{12}$$

otherwise;

$$f_{v-nl} = a_1 \ln(a_2 |\dot{x}| + 1)\,\mathrm{sgn}(\dot{x}) \tag{13}$$

As far symbols are concerned, f_{oh} indicates the hysteretic damper force in the plastic region, a_1 and a_2 are respectively scale and shape factors useful to modify the cyclic behavior of the non-linear viscous component.
The model is governed by four parameters: k_h, f_{oh}, a_1, a_2 that, carefully calibrated, allows to satisfactorily fit both force-displacement and force-velocity loops. By applying the model to the present experimental data, the optimal value for each of the above parameters was found in order to achieve the best curves fitting. They resulted to be well predicted by the following relationships with the i current value:

$$k_h(i) = 0.77 + 15.85i - 3.29i^2 \tag{14}$$

$$f_{oh}(i) = 1.46 + 7.75i \tag{15}$$

$$a_1(i) = 0.11 + 0.45i \tag{16}$$

$$a_2(i) = 0.5 + 10.9i \tag{17}$$

where f_{oh} and a_1 are in kN, k_h in kN/mm, a_2 in s/mm and the current i in Ampere.
Figure 7 shows some comparison between experimentally measured and numerically evaluated force-displacement (left) and force-velocity (right) loops. As said above, the model is able to fit very well both trajectories, unlike the previously examined ones that are substantially calibrated in order to satisfactorily reproduce the experimental force-displacement loops. Figure 8 shows an example of force-velocity loops derived from the modified Bingham (left) and the Spencer models (right), highlighting the anticipated substantial difference with the actual behavior.

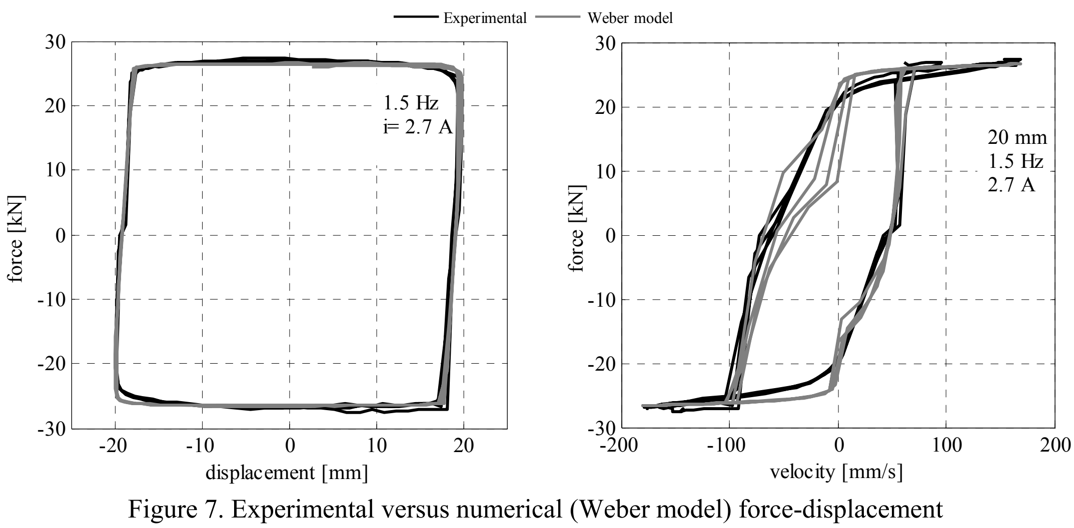

Figure 7. Experimental versus numerical (Weber model) force-displacement
and force-velocity cycles for an harmonic displacement tests of amplitude ±20 mm

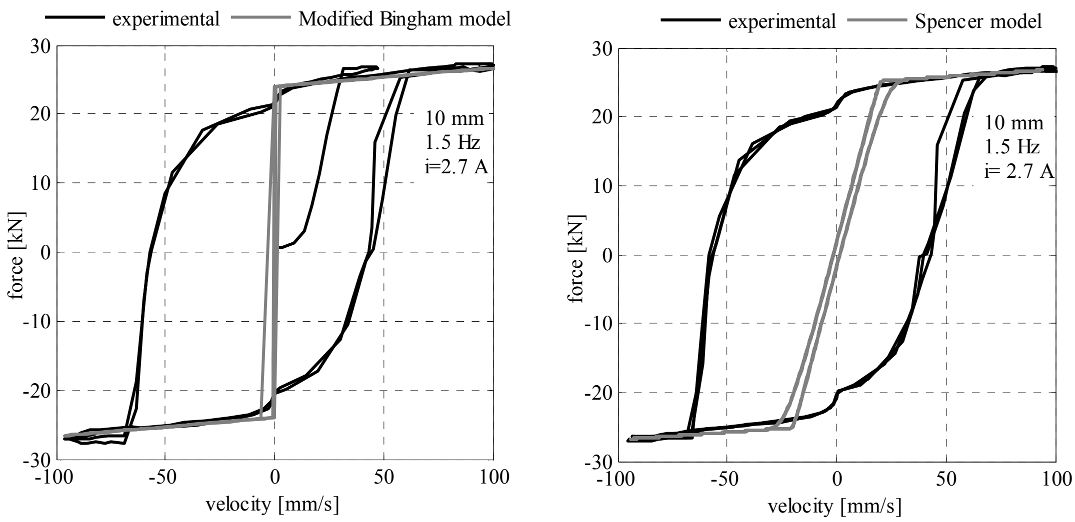

Figure 8. Example of force-velocity loops prediction by modified Bingham model and Spencer model (10 mm, 1.5 Hz, 2,7 A harmonic displacement test)

4. CONCLUSIONS

Modeling the strongly non-linear behavior of magnetorhelogical dampers represents a challenging issue. Many numerical models are available in literature, each one being based on a different philosophy and characterized by a different degree of complexity and effectiveness. Three different models have been considered and compared herein. They was synthetically indicated as “modified Bingham”, “Spencer” and “Weber” models respectively. They are applied with reference to an experimental campaign conducted on two 30 kN MR dampers in the framework of the Italian *ReLUIS* research project, evaluating the actual influence of the involved parameters and the capability of each model of fitting the experimental results. Table 1 summarizes for each model some informations useful for the comparison: *a*) the involved mechanical parameters, *b*) the corresponding amount (evaluated with reference to the experimental campaign under exam) of numerical values to be fixed in order to calibrate them, *c*) the amount of those actually needed to get satisfactorily results (referred to as “significant” in the table and indicated between brackets), *d*) the involvement or not of differential equations, *e*) the capability in simulating force-displacement (FD) and *f*) force-velocity (FV) loops.

Table 1. Comparison of the examined models

Model	*Involved parameters*	*(Significant) Calibrating values*	*Involved differential equations?*	*Fitting FD loops*	*Fitting FV loops*
Modified Bingham	C_d, F_{dy}	4 (4)	No	Good	Poor
Spencer	γ, n, β, A, k_0, k_1, x_0, α, c_0, c_1	12 (5)	Yes	Good	Poor
Weber	f_{oh}, k_h, a_1, a_2	9 (9)	No	Very good	Good

The modified Bingham model results to be completely defined once 2 numerical values for the $C_d(i)$ expression plus other 2 for $F_{dy}(i)$ are fixed. Conversely, the Spencer model requires to set twelve parameters, even if, as shown in Occhiuzzi et al. (2006), not all of them have the same influence. More precisely, only α and c_0 have to be carefully calibrated by properly fixing the 5 numerical values involved in the $\alpha(i)$ and $c_0(i)$ relationships. The Weber model involves 4 mechanical parameters, needing 9 numbers (each having a significant influence on the model) to express them.

When a MR damper has to be included in a control algorithm (needing to work in real-time) and/or in a non-linear structural model of an hosting structure in order to perform time-history analysis, it is important to use a model as simple as possible, able to work and give results at high velocity. For these reasons computational efforts may result inacceptable. To this aim, looking again at the Table 1, one can conclude that the Spencer model, besides being overparametrized, seems to be too much complex, also including differential equations. Conversely, the modified Bingham model offers the advantage of being very simple and fast to manage. The Weber model also results to be interesting for practical applications, even if it requires to set a larger number of parameters, often influencing one each other, not allowing an univocal determination of them. Advantages in using such a model could be found in its capability to fit in a very good manner also the sub-vertical segments of the force-displacement loops and the force-velocity cycles. But these aspects should not be considered as crucial for a MR damper model, the main purpose of which should be simulating in a realistic manner the actual dissipated energy by the device (proportional to the area enclosed by the force-displacement cycles), for each level of feeding current.

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REFERENCES

Carlson, J. D. and Jolly, M. R., MR fluid, foam and elastomer devices, *Mechatronics*, Vol. 10, pp. 555-569, 2000.

Spizzuoco, M. and Serino, G., Design of MR dampers' mechanical properties for the seismic response control of a steel frame structure, *7th Int. Seminar on Seismic Isolation, Passive Energy Dissipation and Active Control of Vibrations of Structures*, Assisi, Italy, GLIS, Vol. 2, pp. 353-362, 2001.

Occhiuzzi, A., Spizzuoco, M. and Serino, G., Experimental analysis of magnetorheological dampers for structural control, *Smart Mater. Struct.*, Vol. 12, pp. 703-711, 2003.

Spencer, B. F., Dyke, S. J., Sain, M. K. and Carlson, J. D., Phenomenological model for magnetorheological dampers, *J. Eng. Mech.* ASCE, Vol. 123, pp. 230-238, 1997.

Yang, G., Spencer, B. F., Carlson, J. D. and Sain, M. K., Large-scale MR fluid dampers: modeling and dynamic performance considerations, *Eng. Struct.*, Vol. 24, pp. 309-323, 2002.

Wen, Y. K., Methods for Random Vibrations of Hysteretic Systems, *J. Eng. Mech.*, ASCE, Vol. 123, pp. 249-263, 1976.

Weber, F., Feltrin, G. and Distl, H., Detailed analysis and modeling of MR damper behavior at zero current, to appear in *Structural Engineering and Mechanics*.

Occhiuzzi, A., Spizzuoco, M. and Caterino, N., Experimental tests and numerical modeling of magnetorheological dampers: the influence of model parameters, *Proceedings of the Italian National Conference "Sperimentazione su Materiali e Strutture"*, Venezia, Italy, 2006.