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:

$$F_d = C_d \cdot \dot{x} + F_{dy}(i) \cdot \text{sgn}(\dot{x})$$  (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 unacceptable 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](image)

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 $R = 1.11 \ \Omega$ and an inductivity $L = 92 \ \text{mH}$, can generate the magnetic field in the device. The current in this circuit, in the range of $i = 0\rightarrow 3 \text{ A}$, 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: $C_d(i=0 A)=5.5$ kNs/m, $C_d(i=2.7 A)=27.0$ kNs/m, $F_{dy}(i=0 A)=0.7$ kN, $F_{dy}(i=2.7 A)=23.5$ kN.
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):

$$F_d = C_d(i) \cdot \dot{x} + F_{dy}(i) \cdot \text{sgn} (\dot{x})$$  \hspace{1cm} (2)$$

$$C_d(i) = 6.6 + 7.9i$$ \hspace{1cm} (3)$$

$$F_{dy}(i) = 1.8 + 8.7i$$ \hspace{1cm} (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.

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:

$$F = c_1 \cdot \dot{y} + k_1 \cdot (x - x_0)$$ \hspace{1cm} (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:
\[
\dot{y} = \frac{1}{c_0 + c_1} \left[ \alpha \cdot z + c_0 \cdot \dot{x} + k_0 \cdot (x - y) \right]
\]  \hspace{1cm} (6)

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

\[
\dot{z} = -\gamma \cdot |\dot{x} - \dot{y}| \cdot |\dot{z}|^{n-1} - \beta \cdot (\dot{x} - \dot{y}) \cdot |\dot{z}| + A \cdot (\dot{x} - \dot{y})
\]  \hspace{1cm} (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 \(\alpha\), \(c_0\) and \(c_1\) are depending on the current \(i\). The constant \(c_0\) 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):

\[
\alpha(i) = 2.41 + 12.61i
\]  \hspace{1cm} (8)

\[
c_0(i) = 69.83 + 86.85i - 13.37i^2
\]  \hspace{1cm} (9)

where \(\alpha\) 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.

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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 (\(f_{v-nl}\) 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:

\[
F_d = f_h + f_{v-nl}
\]  \hspace{1cm} (10)

Weber et al. define the \(f_h\) and \(f_{v-nl}\) 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:

$$f_h = \text{sgn}(x)[f_{oh} - k_h(x_{max} - |x|)]$$  \hfill (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;

$$f_h = \text{sgn}(x)f_{oh}$$  \hfill (12)

otherwise;

$$f_{v-nl} = a_1 \ln(a_2 |\ddot{x}| + 1) \text{sgn}(\dot{x})$$  \hfill (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:

$$k_h(i) = 0.77 + 15.85i - 3.29i^2$$  \hfill (14)

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

$$a_1(i) = 0.11 + 0.45i$$  \hfill (16)

$$a_2(i) = 0.5 + 10.9i$$  \hfill (17)

where $f_{oh}$ and $a_i$ 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](image-url)

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

Modeling the strongly non-linear behavior of magnetorheological 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 were 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

<table>
<thead>
<tr>
<th>Model</th>
<th>Involved parameters</th>
<th>(Significant) calibrated values</th>
<th>Involved differential equations?</th>
<th>Fitting FD loops</th>
<th>Fitting FV loops</th>
</tr>
</thead>
<tbody>
<tr>
<td>Modified Bingham</td>
<td>$C_d$, $F_{dy}$</td>
<td>4 (4)</td>
<td>No</td>
<td>Good</td>
<td>Poor</td>
</tr>
<tr>
<td>Spencer</td>
<td>$\gamma$, $n$, $\beta$, $A$, $k_0$, $k_1$, $x_0$, $\alpha$, $c_0$, $c_1$</td>
<td>12 (5)</td>
<td>Yes</td>
<td>Good</td>
<td>Poor</td>
</tr>
<tr>
<td>Weber</td>
<td>$f_{ohs}$, $k_h$, $a_1$, $a_2$</td>
<td>9 (9)</td>
<td>No</td>
<td>Very good</td>
<td>Good</td>
</tr>
</tbody>
</table>

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 $\alpha$ 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 unacceptable. 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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