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A NOVEL VARIABLE FRICTION DEVICE FOR NATURAL HAZARD MITIGATION

Liang Cao[1], Austin Downey[1], Simon Laflamme[1, 2], Douglas Taylor[3] and James Ricles[4]

ABSTRACT

Implementation of high performance damping devices can ameliorate cost-effectiveness of structural systems for mitigation of natural hazards. However, the applications of these damping systems are limited due to a lack of 1) mechanical robustness; 2) electrical reliability; and 3) large resisting force capability. To broaden the implementation of modern damping systems, a novel semi-active damping device is proposed. The device, termed Modified Friction Device (MFD), has enhanced applicability compared to other proposed damping systems due to its cost-effectiveness, high damping performance, mechanical robustness, and technological simplicity. Its mechanical principle is based on a duo-servo drum brake, which results in a high amplification of the input force while enabling a variable control force. It is also possible to attach the MFD in parallel with a stiffness element and a viscous damper to provide a fail-safe mechanism, analogous to the dynamics of magnetorheological dampers. Here, we present the MFD, and experimentally demonstrate its principle. The hysteresis of the friction force is characterized at low displacements and velocities. Results show that the MFD is a promising semi-active device for mitigation of natural hazards.

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ABSTRACT

Implementation of high performance damping devices can ameliorate cost-effectiveness of structural systems for mitigation of natural hazards. However, the applications of these damping systems are limited due to a lack of 1) mechanical robustness; 2) electrical reliability; and 3) large resisting force capability. To broaden the implementation of modern damping systems, a novel semi-active damping device is proposed. The device, termed Modified Friction Device (MFD), has enhanced applicability compared to other proposed damping systems due to its cost-effectiveness, high damping performance, mechanical robustness, and technological simplicity. Its mechanical principle is based on a duo-servo drum brake, which results in a high amplification of the input force while enabling a variable control force. It is also possible to attach the MFD in parallel with a stiffness element and a viscous damper to provide a fail-safe mechanism, analogous to the dynamics of magnetorheological dampers. Here, we present the MFD, and experimentally demonstrate its principle. The hysteresis of the friction force is characterized at low displacements and velocities. Results show that the MFD is a promising semi-active device for mitigation of natural hazards.

Introduction

Designing structures for motion provides a desired level of performance for a given excitation (e.g., wind, earthquake), achieved by selecting appropriate structural stiffness and damping to limit structural response [1]. Supplemental damping has been shown to be cost-effective in reducing structural response via energy dissipation. Passive control systems are now widely accepted by the field of structural engineering, but are generally only applicable to limited bandwidths of excitation and do not perform well against near-field earthquakes due to the nature of the impact that comes in the form of a shock rather than an energy build-up [2-4]. Conversely, active systems require energy to operate, and they typically are capable of better mitigation performance. However, they are not widely used in structural engineering. Factors impeding their application include high power requirements, controller robustness, and possible actuator saturation [5, 6].

To cope with the numerous drawbacks of active control devices, while preserving a high level of mitigation efficiency, the field has introduced semi-active control devices [7-10]. It has been demonstrated that semi-active damping systems can have considerable economic benefits over passive energy dissipation systems, in addition to enhanced earthquake and wind mitigation. For instance, the authors have shown that the use of a semi-active damping system in lieu of an existing passive strategy currently installed in a high-rise building located in Boston, MA, would result in savings in the order of 20% to 30% in the cost of the damping system [11]. This

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substantial reduction in cost arose from a substantial reduction of the number of dampers required to attain the design performance. In other studies, the authors demonstrated that designs of mid-rise steel buildings with either passive or semi-active damper systems can readily lead to a reduced inter-story drift, while enabling the design to be lighter in weight compared to a building without dampers [12-13]. As a result, there is a reduction in the cost of construction while at the same time the building achieves a higher level of performance under the design earthquake. Despite that the economic and technical advantages have been discussed in some studies, the technology is not yet widely accepted by the field nor implemented. This is due to a lack of available devices that integrate: 1) mechanical robustness; 2) electrical reliability; 3) large resisting forces capability; 4) effective control; and 5) practical performance-based design procedures.

The authors have proposed a novel semi-active friction device capable of large damping forces on low power using mechanically reliable technologies. Its mechanical principle is based on a duo-servo drum brake, which provides a high amplification of the input force via its self-energizing mechanism. Figure 1 is the schematic of the device termed *Modified Friction Device* (MFD). The excitation force is dissipated by the friction of the braking shoes on the drum. The force from the brake shoes can be applied smoothly via a servo-controller and hydraulic or pneumatic actuator. In practical applications, the MFD can include a viscous and a stiffness element in parallel to provide a fail-safe minimal damping, analogous to magnetorheological dampers [11].

Others have proposed variable friction devices for control of civil infrastructure, typically utilizing a friction pad in combination with an actuator capable of generating a normal force. Types of actuators include hydraulic actuators [14-15], electro-magnetic [2][16], electro-mechanical [17] and piezoelectric technologies [18-20]. Large-scale studies and applications have been reported [21-26]. The proposed MFD differs from literature by providing a very large damping force using a mechanically reliable and robust technology.

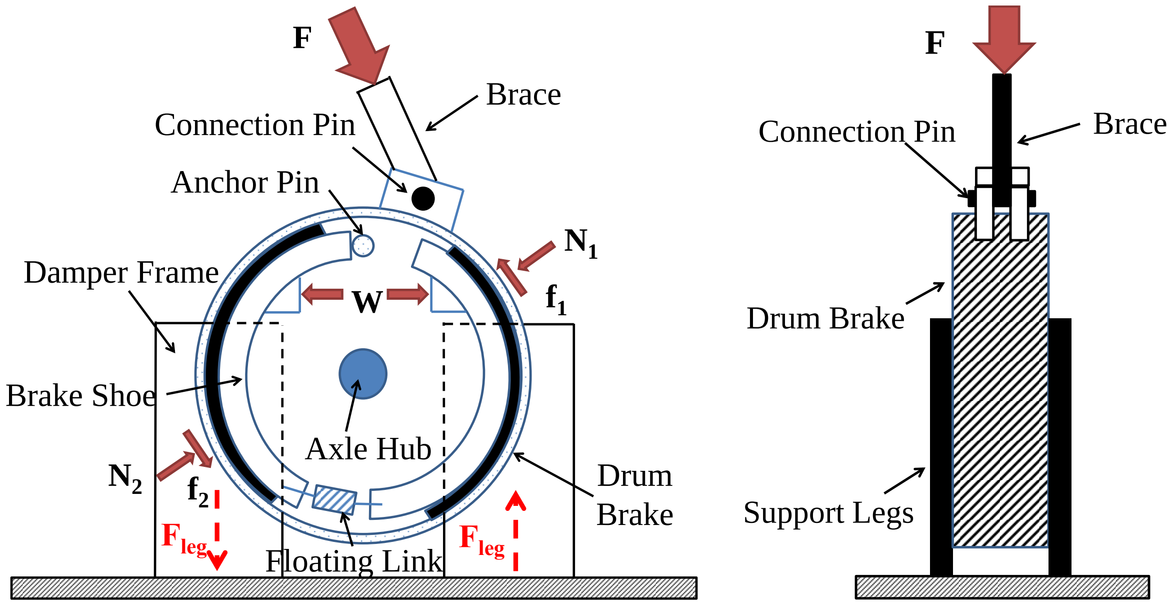
The promise of the MFD has been shown theoretically [11]. In this paper, we verify the technical viability of the device by constructing a first prototype by modifying an actual car brake. We characterize the friction behavior of the duo-servo drum brake at low displacements and velocities under large forces, which has never been reported for car brake systems. The objective is to further the understanding of the friction mechanism to guide future developments of the MFD. The paper first presents the construction of the prototype using a car brake. Then it derives the theoretical dynamic behavior of the MFD, and describes the governing equations used in characterizing the friction behavior. Finally, it characterizes the friction using experimental data and discusses the results.

Prototyping of MFD

The MFD was prototyped from the duo servo drum brake of a car, due to the readily availability of the components.

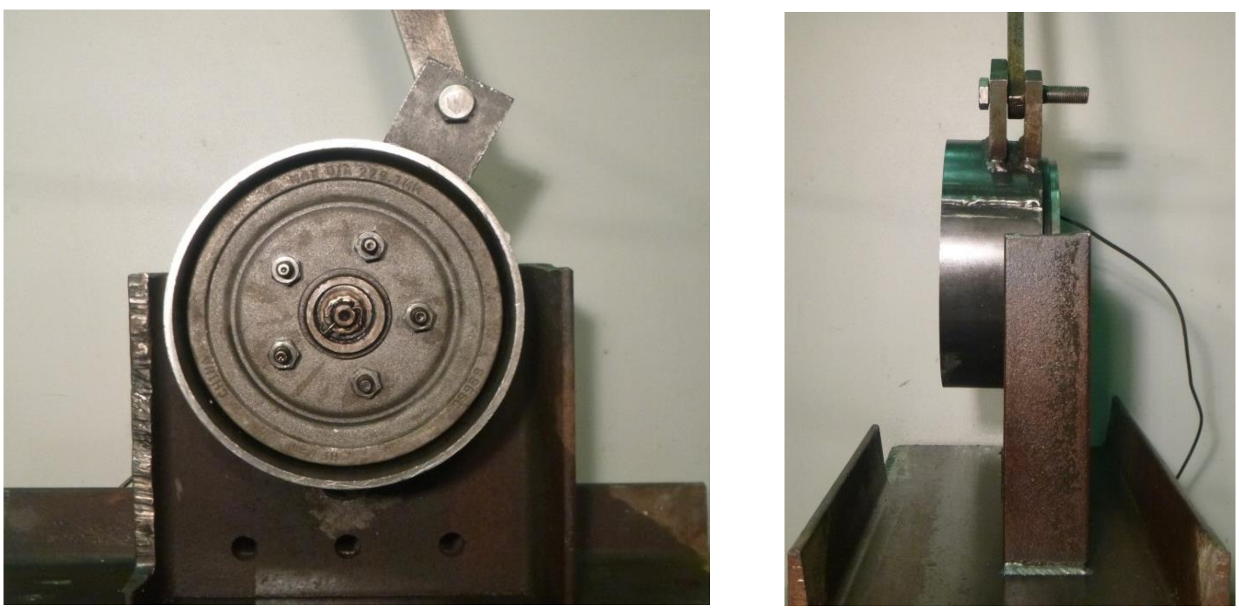
A schematic showing the principle of the MFD is shown in Fig. 1. The excitation force F is damped by the friction force f_1 and f_2 developed from the corresponding normal forces N_1 and N_2 generated by the actuator force W. The MFD is attached to the structure via two legs welded to the back plate to cancel the braking moment M. The setup shown in Fig. 1 dissipates forces that are transmitted axially; the MFD can be installed within a structural brace.

Fig. 2 is a picture of the frame used for the experiment. The frame differs from the schematic in two ways. The frame legs have been replaced by a welded C-shape plate for convenience of installation. Also, the frame is not symmetric on the side view; the resistance of the welded plate was enough to counteract the moment generated by the eccentricity of forces.



(a) (b)

Figure 1. Schematic of the braking mechanism in the MFD (a) front view; and (b) side view



(a) (b)

Figure 2. Prototype of the MFD (a) front view; and (b) side view

The braking mechanism has been modified to increase bi-directional performance. The localization of the braking shoes with respect to the anchor pin creates a static moment that may substantially amplify the actuator force, a phenomenon termed self-energizing mechanism. This self-energizing action is typically biased in one direction in a car brake application through the use of less friction material on the leading shoe. Consequently, the area of the friction on the leading shoe has been increased to provide equivalent area on both shoes. The contact area has also been increased to improve on the friction capacity [27] by rotating the drum brake under hydraulic pressure during several hours in both directions to wear the lining surface and increase the contact area. Fig. 3 shows the improvement on the frictional force for different levels of wear. The capacity increases with wear, but the initial friction decreases most likely due to the increase in smoothness of the frictional surface. Measurements of frictional force were obtained from static tests of the MFD under varying conditions of wear. Hydraulic pressure was applied to the MFD, while a displacement of 0.5 inch per minute was applied on the brace.

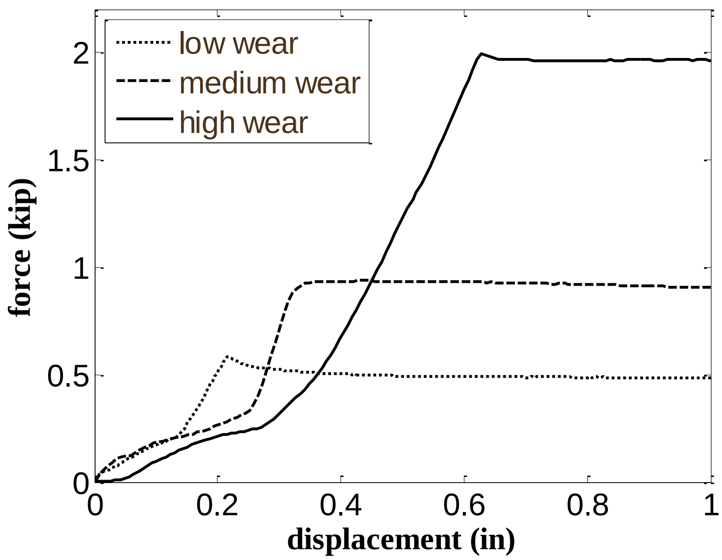


Figure 3. Increase of frictional force via wear

Dynamic Characterization of MFD

Friction Model

Several friction models have been proposed to characterize the friction mechanism, which can be divided into static and dynamic models. Given the significant hysteresis behavior of the device at low displacements and velocities, a dynamic model has been taken to characterize the MFD. In particular, the LuGre model [28]-[30] has been selected due to its capacity to accurately simulate the Stribeck effect and rate dependence of the friction phenomenon.

The LuGre Model describes the dynamic of the friction force $F_{friction}$ as:

$$F_{friction} = \sigma_0 z + \sigma_1 \dot{z} + \sigma_2 \dot{x}$$

$$\dot{z} = \dot{x} - \sigma_0 \frac{|\dot{x}|}{g(\dot{x})} z \tag{1}$$

where σ_0, σ_1, σ_2 are constants, z is an evolutionary variable, x and $\dot{x}$ are the displacement and velocity of the MFD, respectively, $g(\dot{x})$ is a function that describes the Stribeck effect:

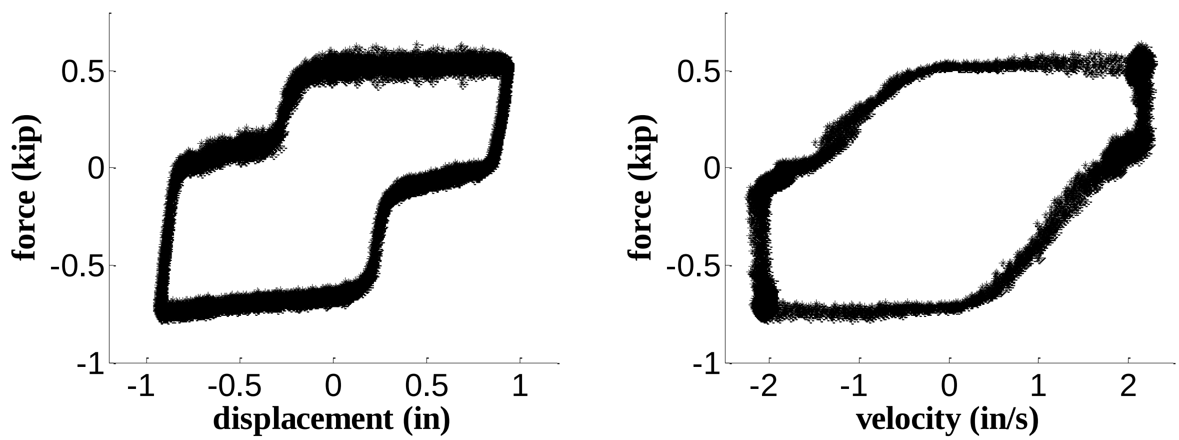
$$g(\dot{x}) = F_c + (F_s - F_c)e^{-\left(\frac{\dot{x}}{\dot{x}_s}\right)^2} \tag{2}$$

where $\dot{x}_s$ is a constant modeling the Stribeck velocity, F_c is the Coulomb friction force, F_s is the magnitude of the Stribeck effect.

Experimental Results

The characterization of the MFD presented in this paper is at an initial stage, conducted via a single and repeated harmonic test performed at a frequency of 0.5 Hz, amplitude of +/- 1 inch, 1500 psi hydraulic pressure corresponding to approximately 50% of the braking capacity, over 20 cycles per test. The objective is to establish a preliminary dynamic model capable of characterizing the dynamic behavior of the damping device.

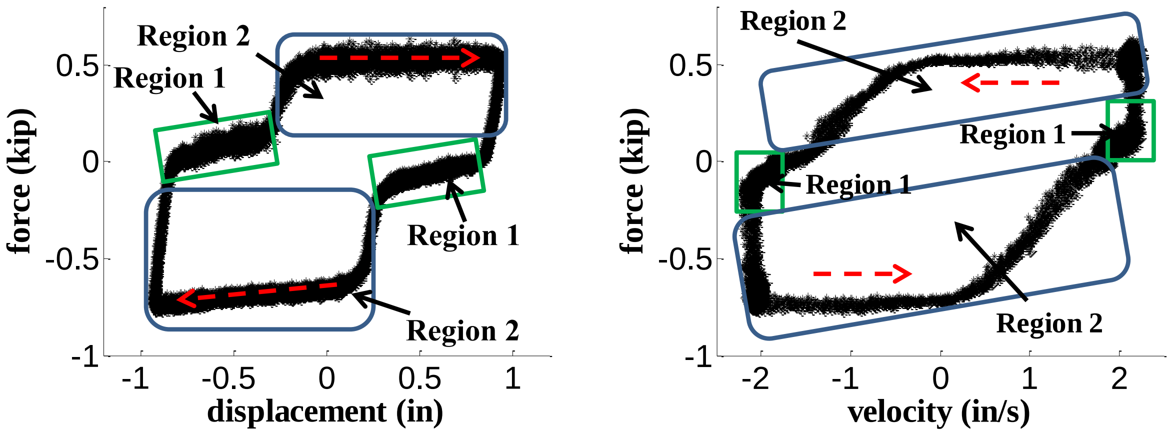
The force-displacement plot and the force-velocity plot of a typical result are shown in Figs. 4(a) and 4(b). Results show that the dynamic of the MFD is located within the hysteretic loop at this particular excitation. Another feature from the plots is the sudden drop of force that occurs when the velocity switches sign. This loss of force is explained by the braking shoes losing contacts with the drum when switching velocity, and a minimal stiffness is likely generated by the anchor pin.



(a) (b)

Figure 4. Dynamic response of the MFD under hydraulic pressure of 1500 psi: (a) force-displacement plot; and (b) force-velocity plot.

It follows that two different dynamics describe the behavior of the MFD: a pure stiffness zone and a friction zone. The LuGre model presented above is modified by dividing the hysteresis loop into two regions, shown in Fig. 5.



(a) (b)

Figure 5. Different regions of (a) force-displacement plot; and (b) force-velocity plot

For pure stiffness zone (region 1), as shown in Fig. 5, the governing equation of this behavior can be described by a stiffness element:

$$F_{stiffness} = kx + \alpha \tag{3}$$

where k is the stiffness of region 1 and α is a constant. Note that, due to a slight asymmetry in the directional braking force, this constant takes two values: α_{up} when the velocity is positive and α_{down} when the velocity is negative. The LuGre model (Eq. 1 and 2) is used to characterize the friction zone (region 2).

A C^{∞} transition function is used to provide a smooth transition between both dynamics, consisting of the following sigmoid function [31]:

$$m(x) = \frac{1}{1 + e^{-(x - x_0)/\gamma}} \tag{4}$$

where x_0 is the position of the transition region and γ is the bandwidth of the transition region. It follows that the MFD force F is described by

$$F = \begin{cases} (1 - m(x)) \cdot F_{stiffness} + m(x) \cdot F_{friction} & \text{stiffness zone} \rightarrow \text{friction zone} \\ (1 - m(x)) \cdot F_{friction} + m(x) \cdot F_{stiffness} & \text{friction zone} \rightarrow \text{stiffness zone} \end{cases} \tag{5}$$

Parameter Optimization

Model parameters are determined using MATLAB by minimizing the performance function $J = \left(\hat{f}_k - f_k\right)^2$:

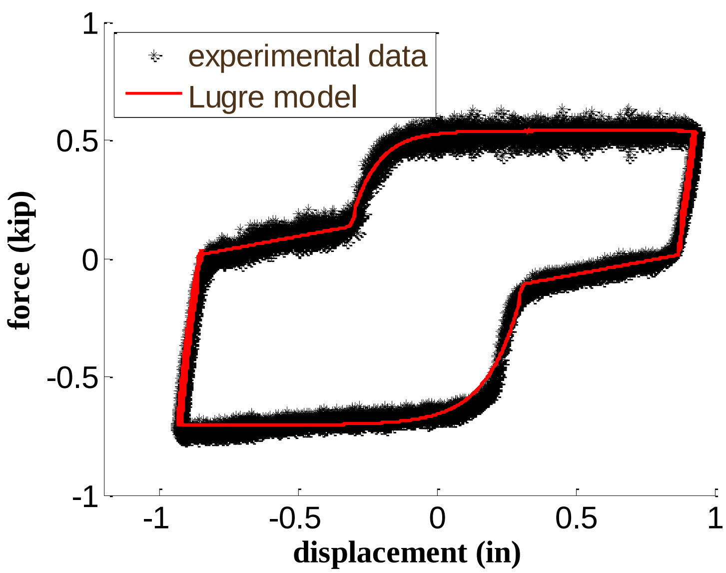
$$\min_{\mathbf{r}} \left\|\hat{f}_k - f_k\right\|_2 \tag{6}$$

where $\hat{f}_k$ is the estimated friction force from the friction model, f_k is the experimental friction force,. $\mathbf{r}$ is the parameter vector $[x_{0up}, x_{0down}, \gamma, k, \alpha_{up}, \alpha_{down}, \sigma_0, \sigma_1, \sigma_2]^T$, and $\|\cdot\|_2$ is the 2nd Euclidian norm. x_{0up} is the position of transition region when the velocity is positive and x_{0down} is the position when the velocity is negative.

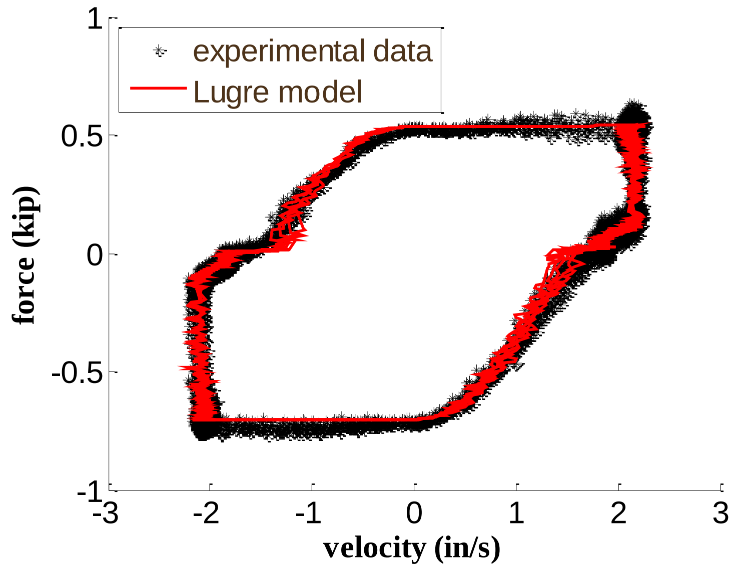
The optimal parameters r_* are shown in Table 1. Fig. 6 plots the experimental results versus the friction model. Results show that the proposed dynamic model can be used for characterizing the dynamic behavior of the MFD.

Table 1. Model Parameters

Parameter	Values
x_{0up}	{-0.9, -0.3} in
x_{0down}	{0.2, 0.8} in
γ	0.02 in
k	0.22 kip in^{-1}
α_{up}	0.22 kip
α_{down}	-0.16 kip
σ_0	5.754 kip in^{-1}
σ_1	0.002 kip s in^{-1}
σ_2	0.001 kip s in^{-1}



(a)



(b)

Figure 6. Model fitting of the MFD: (a) force-displacement loop; and (b) force-velocity loop

Discussion and conclusion

The dynamic behavior of a first prototype of a novel variable friction device, the MFD, has been characterized. The initial characterization of the prototype has been conducted other a single harmonic in a small displacements and velocities range.

Experimental results showed that the mechanism of the car brake led to a sudden drop of force when the velocity switches sign, because the rotation of the braking shoes is causing a temporary loss of contact with the drum. This drop of force is recovered once the MFD was beyond a given level of displacement. While it was possible to model this phenomenon using a stiffness element, future prototyping of the MFD will necessitate the minimization or elimination of this loss of contact. This will substantially increase the damping capability of the MFD.

The dynamics of the MFD outside the loss-of-contact zone has been modeled using the LuGre model due to the high hysteresis found in the device. Optimal model parameters have been determined by fitting experimental data. Results show that it was possible to use the proposed dynamic model to characterize the friction behavior.

Future research on the MFD includes the full characterization of the modified car brake at various levels of displacements, velocities, and actuation forces, as well as the fabrication of a second prototype to improve on the performance of the car brake.

Acknowledgments

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