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Simon Laflamme, Massachusetts Institute of Technology
Douglas Taylor
Mohamed Abdellaoui Maane, Massachusetts Institute of Technology
Jerome J. Connor, Massachusetts Institute of Technology

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Modified Friction Device for Control of Large-Scale Systems

S. Laflamme\textsuperscript{1*}, D. Taylor M.\textsuperscript{2}, Abdellaoui Maane\textsuperscript{1}, J. J. Connor\textsuperscript{1}

\textsuperscript{1}Department of Civil and Environmental Engineering, Massachusetts Institute of Technology, Cambridge, MA, 02139
\textsuperscript{2}Taylor Devices, North Tonawanda, NY, 14120

SUMMARY

Semi-active control of civil structures has shown to be promising at mitigating vibrations. Those systems typically perform significantly better than passive systems, and only necessitate small power to operate. Nevertheless, semi-active control devices are not yet accepted by the construction and structural engineering communities. Among impeding factors, semi-active devices are not perceived as sufficiently reliable or robust. In this paper, a new semi-active damping device based on existing reliable technology is proposed. It is composed of a stiffness element, a viscous damper, and a braking mechanism in parallel. The device, termed the Modified Friction Device (MFD), is essentially a variable friction damper based on vehicular braking technology, and has a fail-safe mechanism.

The MFD is investigated as a replacement to the actual viscous damping contained in an existing structure in Boston, Massachusetts, for mitigation of accelerations caused by wind excitations. Simulations have showed that the MFD 200 kN is capable of wind mitigation when compared to the performance of a passive viscous strategy of much larger capacity. Also, an MFD 1350 kN capacity can perform similarly by using only a third of the number of dampers currently contained in the existing structure. Finally, the proposed device shows to be promising at mitigating inter-story displacements in the occurrence of an earthquake.

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\textsuperscript{*}Correspondence to: Department of Civil and Environmental Engineering, Massachusetts Institute of Technology, Cambridge, MA, 02139

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1. Introduction

The field of control has thoroughly been developed since Norbert Wiener proposed to unify the field of feedback control under cybernetics [1], and has been widely extended to most engineering fields. Its application to civil engineering can be traced up to Nordell in 1969 [2] with the use of active tendons to protect structures from blast. Yao formally introduced the concept of structural control in 1971 [3]. Since then, several control strategies using active and semi-active control have been researched, but the physical implementation of those control strategies to large-scale civil structures has been limited. The very first application was achieved in 1989 in Tokyo by the Kajima Corporation, and consisted of an active mass driver installed at the top of the ten-story Kyobashi building [4]. In an extensive review of the structural control field, [5] reported that there were over 50 large-scale applications of feedback control systems in civil structures. In their paper, the authors noted that none of the applications were in the United States, and alleged that possible reasons include conservatism of the construction industry with respect to technologies, as well as lack of research and development expenditures. Nevertheless, the field of structural control has attracted a lot of attention over the last decades in search of robust and performing strategies for vibration mitigation.

However, many control schemes in the literature do not meet practical requirements of large force capability, mechanical reliability and robustness of actuators [6]. Power requirement is a specific issue arising from the large size nature of plants being controlled. During extraordinary events such as earthquakes, there is a high occurrence probability of a general power failure. It is therefore essential that actuators be capable of running on a secondary power system. Mechanical reliability and robustness of actuators are a primary concern for control of large-
scale systems. For instance, failure of an engine on an aircraft can be catastrophic. The same idea extends to civil structures, whereas actuators must be physically capable of exerting a force in the occurrence of a moderate-to-extraordinary load to ensure comfort and safety.

The autonomous dynamics of civil structures is inherently stable. This stability is quite advantageous for vibration mitigation, as passive and semi-active damping systems always enhance stability, in contrast to active control systems. Semi-active systems are characterized by a low energy demand, a control performance close to the active control schemes, and are thus ideal candidates to mitigate vibrations in large-scale systems. Despite their early emergence in the 1920s for vehicle shock absorbers, it is only in 1983 that they have been proposed for control of civil structures [7]. Symans and Constantinou [8] provided a state-of-the-art review of semi-active control systems. Several damping devices have been proposed, studied, and applied for applications to semi-active and hybrid control. Among these are controllable fluid [9, 10, 11, 12], variable friction [13, 14, 15], and variable orifice [16, 17, 18, 19] dampers. Magnetorheological (MR) dampers, a type of controllable fluid dampers, have received considerable attention over the last decades because of their significant low power requirement, large resistance force, and fail-safe nature, but their applications to civil structures is still in its infancy [20]. Some chemical issues specific to rheological fluids might impede the implementation of those devices in civil structures, such as sedimentation which occurs if the damper is not used for a long period of time [21]. Moreover, MR dampers may exhibit fluid leakage around the seal. The integrity of their seal is currently guaranteed on vehicle suspension systems for a life of 100,000 miles [22], but there is no indication in the literature on how well the seal can behave over the design period of a civil structure.

Variable friction devices are primarily composed of a friction pad and an actuator capable of generating a normal force. Kannan et al. [23] introduced a variable friction damper that uses an hydraulic actuator. The main disadvantage of hydraulic actuators is in their delay in reaching the actuation force [24]. For convenience, the field has introduced two other types of variable friction dampers capable of large frictional forces. The first one is the electromagnetic friction damper, proposed by [16]. The authors utilized the friction device for control of base-isolated
buildings. The semi-active scheme consists of a friction pad installed between two steel plates. The normal force $N$ is varied by changing the electric current in solenoids located at the outer surface. The device is capable of a force in the order of 20 kN. The electromagnetic friction damper has also been used for an engine mount in [25]. The second type is the piezoelectric friction device presented by [26]. It consists of two friction surfaces mounted on a piezoelectric stack actuator. The piezoelectric stack is activated to control the normal force on the friction surfaces. That type of variable friction device has been used as a friction joint for space truss structures [27]. Durmaz et al. [28] developed a high-capacity friction damper by extending the concept to larger contact areas. The semi-active damper has a force range of 0.890 kN to 11 kN. It is worth mentioning that [29] developed a variable friction device inspired by car braking systems, but the device also uses piezoelectric actuators and is limited to lightweight mechanical structures. Fig. 1a) shows the variable friction damper proposed by [16], and Fig. 1b) shows a typical piezoelectric friction damper.

![Diagram of variable friction devices](image)

**Figure 1:** variable friction devices: a) electromagnetic damper [14]; and b) piezoelectric damper [30]

The implementation of semi-active devices in civil structures is partly impeded by mechanical obstacles. Those include fluid and mechanical parts reliability, dependability on an external power source and electronics, performance degradation over long period of time, and damping...
capacity. This paper proposes a novel mechanically reliable damping device for large-scale structures that overcomes most of those practical obstacles to implementation, with the objective to enhance the applicability of semi-active damping systems to large-scale structures. The device, here termed the modified friction device (MFD), is inspired by the dynamic behavior of MR dampers and consists of a friction mechanism installed in parallel with a viscous and a stiffness element. The friction device is a rotating drum on which a variable friction can be smoothly applied. The MFD is novel by its capability of very large damping forces, on the order of 200 kN, while operating on 12-volts batteries. This is a major improvement compared to existing variable friction schemes proposed in the literature. For instance, piezoelectric variable friction dampers have recently been experimented [31, 32, 33], but to the best knowledge of the authors, the largest damping capacity for battery-operated variable friction was in the range of $10^4$ N, reported in [14]. The significant difference in the theoretical high operating range arises from the self-energizing capacity of the braking mechanism, which greatly amplifies the frictional force. The MFD is based on current reliable and robust mechanical technologies, and is thus a mechanically reliable and robust semi-active device.

The paper is organized as follows. Section 2 describes the proposed MFD. Its dynamics is described, along with its friction mechanism model and internal controller. Section 3 simulates the MFD as a replacement to the actual viscous damping strategy of an existing building and discusses the results. Section 4 concludes the paper.

2. Proposed Modified Friction Device

2.1. Device Dynamics

The proposed MFD consists of a stiffness, a viscous, and a controllable friction element installed in parallel, as schematized in Fig. 2a), where $x$ represents the device displacement and $F$ the reaction force. The variable spring in the variable friction element depicts a variable braking force. The controllable friction element differs from the other variable friction types by using a
reliable mechanical system analogous to the braking system of a vehicle. It is also novel by the incorporation of both a stiffness and a viscous element, which provides minimal damping when the current is switched off or in the unfortunate failure of the friction element, also termed fail-safe mechanism. The objective of the device is to develop a force capable of controlling large-scale systems. The proposed MFD has a maximum force capacity of 200 kN (45 kips), with a dynamic range of 10 (ratio of the maximum force over the minimum force). The design can be extended to 1350 kN (300 kips), which will be used in the simulations for comparison with an actual large-scale passive mitigation system. Fig. 2b) shows the dynamics of the MR damper based under the Bouc-Wen model representation. Since the MFD has a friction element in lieu of a Bouc-Wen element, the main difference in the dynamics of those devices is in their hysteresis. The MR damper hysteresis loop in the force-velocity plot is typically larger.

![Diagram](image)

Figure 2: schematic representation of the dynamics of a) the MFD; and b) the MR damper

As specified above, the MFD is at its conceptual phase. It is fundamental to utilize a model that accurately captures its dynamics. Several static and dynamic models have been developed to describe friction phenomena. Dynamic models are preferred over static models because of their higher precision and ease of simulation. Moreover, the hysteresis behavior of the friction
element must be modeled in order to avoid unnecessary discontinuities in the force when velocity changes sign. The hysteresis phenomena will be discussed later. Among the existing dynamic models, such as the Dahl Model, the Bristle Model, the Reset Integrator Model, the models from Bliman and Sorine, and the LuGre model [34], the LuGre model has been selected due to its capacity to accurately simulate the Striebeck effect and rate dependance of the friction phenomenon. It has also been widely studied and is simple to simulate. The analytical representation of the LuGre model is written as:

\[
\dot{z} = \dot{x} - \sigma_0 \frac{|\dot{x}|}{g(\dot{x})} z
\]

\[
F_{\text{friction}} = \sigma_0 z + \sigma_1 \dot{z} + f(\dot{x})
\]  

(1)

where \( z \) is an evolutionary variable, \( F_{\text{friction}} \) is the reaction force from the friction element, \( f(\dot{x}) \) is the viscous friction and is commonly taken as \( f(\dot{x}) = \sigma_2 \dot{x} \), \( \sigma_0, \sigma_1, \) and \( \sigma_2 \) are constants, and \( g(\dot{x}) \) represents the Striebeck effect and can be taken as:

\[
g(\dot{x}) = F_c + (F_s - F_c) e^{-\left(\dot{x}/\dot{x}_s\right)^\alpha}
\]  

(2)

where \( F_c \) is the Coulomb friction force, \( F_s \) is the magnitude of the Striebeck effect, \( \dot{x}_s \) and \( \alpha \) are constants. Note that at the steady-state (\( \dot{x} = 0 \)), the frictional force is uniquely described by \( F_c \) and the sign of the velocity: \( F_{\text{friction}} = F_c \text{sgn}(\dot{x}) \). Typical choices for \( \alpha \) are \( \alpha = 1 \) [35] and \( \alpha = 2 \) [34]. \( F_s \) is taken as proportional to \( F_c \), which allows the Striebeck effect to remain proportional to the applied voltage. The hysteresis phenomena is indirectly described in \( z \) by \( \sigma_0 \). The challenge in describing the hysteresis phenomenon for the MFD is in the limited information on the hysteresis found in large-scale friction devices. The hysteresis phenomena arises from slow processes, such as wear, accumulation of debris, and temperature variation, that cause a slow response to pressure changes [36]. Consequently, the hysteresis of a system is dependent on the friction material, load amplitude, and load frequency. Kim and Jeong [37] studied the hysteresis behavior of cast iron. The authors demonstrated that the hysteresis
increases with applied stress and temperature, and decreases with heat. They also found that cast iron showed an hysteresis between 12.20 N-mm and 193.8 N-mm for a temperature ranging between 27.3°C and 34.7°C under high stresses applied at 25 Hz. Nevertheless, the hysteresis behavior of the MFD is expected to reach higher values because of the low frequency nature of the excitations. Thus, $\sigma_0$ is selected such that the hysteresis of the MFD stays in the range of the values described in [37] on low voltage (0-3 V).

To obtain values for $\sigma_1, \sigma_2, F_s$, and $\dot{x}_s$, a model fit has been conducted on the experimental data of a 55 kN friction-type device. The fit is represented in Fig. 3. The resisting force $F$ of the MFD can be written:

$$F = F_{\text{friction}} + k_{\text{mfd}} x + c_{\text{mfd}} \dot{x}^\beta$$  \hspace{1cm} (3)

![Figure 3: model fitting of a 55 kN friction-type device](image)

where $k_{\text{mfd}}$ and $c_{\text{mfd}}$ are the stiffness and viscous coefficients of the MDF respectively, and $\beta$ is a constant and taken as $\beta = 1$ for a linear viscous damper. While $k_{\text{mfd}}$ can be designed based on the required stroke and dynamic range, $c_{\text{mfd}}$ can be selected based on fail-safe requirements or for enhanced performance of the MFD. Those two values have been selected based on the performance requirements of the simulated structure. $c_{\text{mfd}}$ is taken as 12 MN-s/m, which is a tenth of the large-scale viscous dampers. The force-displacement plot and the force-velocity
plot under an harmonic excitation of 0.5 Hz with an amplitude of 0.30 m for different levels of $F_c$ are shown in Fig. 4a) and Fig. 4b) respectively. Table I gives the parameter values used to model the MFD 200 kN dynamics. Note that in the case of the MFD 1350 kN, only $F_{c_{\text{max}}}$ changes, and takes the value $F_{c_{\text{max}}} = 1120$ kN. Fig. 5 illustrates the free body diagram of the MFD system. In an application, the drum would most likely be designed to roll on a flat surface.

Table I: MFD 200 kN parameter values

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$</td>
<td>2</td>
</tr>
<tr>
<td>$\beta$</td>
<td>1</td>
</tr>
<tr>
<td>$\sigma_0$</td>
<td>$5 \times 10^7$ kN/m</td>
</tr>
<tr>
<td>$\sigma_1$</td>
<td>$1 \times 10^5$ kN/s/m</td>
</tr>
<tr>
<td>$\sigma_2$</td>
<td>6.5 kN/s/m</td>
</tr>
<tr>
<td>$\dot{x}_s$</td>
<td>0.002 m/s</td>
</tr>
<tr>
<td>$F_c$</td>
<td>$1.17 F_c$</td>
</tr>
<tr>
<td>$F_{c_{\text{max}}}$</td>
<td>160 kN</td>
</tr>
<tr>
<td>$k_{\text{mfd}}$</td>
<td>$2 \times 10^4$ kN/m</td>
</tr>
<tr>
<td>$c_{\text{mfd}}$</td>
<td>$1.2 \times 10^4$ kN/s/m</td>
</tr>
</tbody>
</table>

Figure 4: dynamics of the MFD under a 7.62 mm amplitude sinusoidal excitation of 0.5 Hz: a) force-displacement; and b) force-velocity
2.2. Friction Mechanism

Large capacity friction devices have been used for decades by the railroad industry. Fig. 6 shows a 50 kN-m (36 kips-ft) draft gear capacity at 2200 kN (500 kips) reaction force, which is designed to absorb potential impacts of two adjacent cars. The challenge is to develop a device capable of large variable friction forces, while being mechanically reliable. The force variation must also be continuous in order to minimize accelerations in the controlled plant. The authors propose a variable friction mechanism based on existing mechanically reliable vehicle braking technologies: a duo-servo drum brake. The next subsection will derive the governing equations for duo-servo drum brakes, and will be followed by a description of the design of the braking mechanism.
2.2.1. Governing Equations for Duo-Servo Drum Brakes. The duo-servo drum brake is composed of two internal shoes anchored at a single pin, and attached with a rigid or floating link. A single hydraulic actuator exerts force on both shoes. Fig. 7(a) illustrates the concept, and Fig. 7(b) shows the force diagram for a negligible angle $\alpha$. For clarity of Fig. 7, the connection of the shoes to the pin is not shown. In Fig. 7(b), $N$ is the normal force to the friction force $F$, $r$ is the radius, $W$ is the actuation force, $R_{\text{link}}$ is the force reaction from the link, $R_x$ and $R_y$ are the reactions from the pin in the $x$ and $y$ directions respectively, and subscripts 1 and 2 indicate the shoes. For this design, the floating link scheme is selected. That type of braking system has been shown to have a higher force amplification factor than most of other types of drum brakes, because of the self-energizing nature of the mechanism in both spinning directions [39].

Figure 7: duo-servo drum brake: a) schematic; and b) force diagram

The amplification factor of the actuation force $W$ to the tangential forces $F_{1,2}$ is termed the force amplification factor $C$. The total frictional force is written $F_1 + F_2 = F_c = CW$, where $F_c$ is the coulomb force in (2). For the duo-servo drum brake, it can be shown that [39]:

$$C = \frac{\mu(a + b)}{a\sqrt{1 + \mu^2} - \mu r} \left[ 1 + \left( \frac{a + r \sqrt{1 + \mu^2}}{r(1 - \mu)} \right) - \frac{a + b}{r} \frac{\mu}{1 - \mu} \right] (4)$$

where $\mu$ is the friction coefficient between the drum and the shoes. Fig. 8 shows the pressure
distribution for the duo-servo brake drum. It is assumed that the distribution follows a triangular shape:

\[ p(\phi) = \frac{\phi p_{\text{max}}}{(\theta_1 + \theta_2)} \quad \text{for} \quad \phi \leq \theta_1 \]
\[ p(\phi) = \frac{(\phi - \theta_1)p_{\text{max}}}{(\theta_2 + \theta_1)} \quad \text{for} \quad \theta_1 + \theta_2 \leq \phi \leq \theta_1 + \theta_2 + \theta_3 \]  

(5)

where \( p_{\text{max}} \) is the maximum pressure and can be written as a function of the total pressure \( p_{\text{tot}} \):

\[ p_{\text{max}} = \frac{2p_{\text{tot}}}{\theta_1 + \theta_2} \]  

(6)

Recognizing that, in the case of civil structures, the braking pressure will be required for both rotating directions, it is useful to write (5) in term of a uniformly distributed average pressure \( p_{\text{avg}} \):

\[ p_{\text{avg}} = \frac{p_{\text{max}}}{2} \]  

(7)

The frictional torque \( \tau_f \) can be computed using the moment of the tangential frictional force \( F_{\text{friction}} \) about the center of the drum [40]:

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\[ \text{Prepared using stcauth.cls} \]
\[ d\tau_f = r\, dF \]  

with:

\[ dF = \mu b pr \, d\phi \]

where \( b \) is the shoe width. Integrating over the shoes leads to:

\[
\tau_f = \int_0^{\theta_1} \mu b_w r^2 \frac{\phi p_{max}}{(\theta_1 + \theta_2)} \, d\phi + \int_{\theta_1 + \theta_2}^{\theta_1 + \theta_2 + \theta_3} \mu b_w r^2 \frac{(\phi - \theta_3) p_{max}}{(\theta_1 + \theta_2)} \, d\phi
\]

\[
= \int_0^{\theta_1 + \theta_2} \mu b_w r^2 p_{max} \, \frac{2d\phi}{2}
\]

\[
= \frac{(\theta_1 + \theta_2)}{2} \mu b_w r^2 p_{max}
\]

The differential work done by the brake \( U_f \) is defined by:

\[
dU_f = \tau_f \, d\omega
\]

\[
= \frac{\tau_f}{r} \, dx
\]

Integrating (10) leads to:

\[
U_f = \frac{\tau_f}{r} x
\]

\[
U_f = F_{\text{friction}} x
\]

where \( x \) is the damper displacement, and \( \omega \) is the angular velocity. For an average velocity, the frictional power \( P_f \) (in kcal/sec) is given by:

\[
P_f = 238.8 \frac{\tau_f \dot{x}}{r}
\]
where 238.8 is a conversion constant for $\tau_f$ in N-m and $\dot{z}/r$ in s$^{-1}$. The energy absorption of the adjacent material to the braking shoes over a short period of time is related to the friction work and leads to the following relationship for temperature increase [40]:

$$\Delta T = \frac{1}{4187} \frac{U_f}{w_m c}$$

(13)

where 1/4187 is a conversion constant for $T$ in °C, $U_f$ in N-m, $w_m$ the mass of the material absorbing the energy in kg, and $c$ the average specific heat of the material in kcal/kg.°C. The allowable energy absorption by the braking mechanism is dependant on the lining material, the coefficient of friction $\mu$, and the rate of energy absorption. A greater shoe pad area would allow a greater energy dissipation, thus decreasing brake wear and fading. In the case of civil structure, the brake application is assumed to be of high intensity over infrequent short periods of time. For such, literature [40] suggests high allowable design values for energy dissipation.

2.2.2. Design of the Braking Mechanism  Asbestos was one of the most commonly used brake lining material due to its good mechanical properties and low cost availability, but health concerns have led to its replacement by asbestos-free materials, such as semi-metallic, organic, and ceramic fibers friction materials. Han and Yin [41] studied the performance of added ceramic fibers to 15% weight steel fiber lining. Their study showed that the coefficient of friction remained above $\mu = 0.46$ for temperatures ranging from 100°C to 350°C for a 10% weight added ceramic fiber, thus offering a good resistance to fade. The wear rate of that material is approximately 1.6x10$^{-7}$cm$^3$/(N-m)$^{-1}$.

Thus, cast iron using a steel fiber lining with added ceramic fibers is used for the drum and shoes. The design values are: $\mu = 0.46$, $c = 0.13$ kcal/kg.°C, and $w_m = 7500$ kg/m$^3$. The allowable maximum pressure is taken as $p_{\text{max,all}} = 130$ MPa.

The maximum required frictional force is $F_{\text{friction}} = 180$ kN for a 200 kN MFD, and $F_{\text{friction}} = 1200$ kN for a 1350 kN MFD. Selecting a shoe width $b_w = 0.1$ m and thickness $t_w = 0.025$ m for both shoes covering 0.65$\pi$ each, and a drum radius of $r = 0.2$ m, the maximum pressure at any time is $p_{\text{max}} = 15.1$ MPa for the 200 kN MFD, and $p_{\text{max}} = 118$
MPa for the 1350 kN MFD. Two 2 kN linear hydraulic actuator operating on 12 volts are installed at 0.086 m up from the center of the drum, and shoes terminate at 0.15 m down from the center of the drum, giving a force amplification factor (4) of 58.5, which translates into a theoretical frictional capacity of 234 kN. For the 1350 kN MFD, the actuation force demand would be of 20 kN, which would be equivalent to 10 times the number of linear actuator needed in the 200 kN MFD case. A design trade-off would be to use bigger power sources for larger capacity actuators, at the expense of the independence on an external power source.

2.3. Brake Actuator Control

Several controllers have been proposed for control of variable friction devices, such as the bang-bang controller presented by [42], along with some of its modifications [43, 44, 15, 45]. Those controllers are designed to include the variable friction mechanism in the feedback rule. In contrast, the brake controller presented in this subsection is an internal controller to facilitate the incorporation of the device in any closed-loop system. It is decoupled from the controller that computes the required force. The device is designed to receive a required force and the internal controller computes a voltage based on that force. In the simulations, an LQR controller is used to feed the required force to the brake controller.

A response delay of the linear actuator used in the braking mechanism is simulated by inducing a delay in the voltage response. The coulomb friction $F_c$ in (2) is written as:

$$F_c = F_{c,0} \cdot v_{act}$$

$$\dot{v}_{act} = -\eta(v_{act} - v_{req})$$

(14)

where $F_{c,0}$ is the nominal Coulomb friction and has a linear dependance on the actual voltage input $v_{act}$, $\eta$ is a positive constant representing the voltage delay, and $v_{req}$ is the required voltage for a desired force $F_{req}$. Here, the delay coefficient is taken as $\eta = 200$ to have a comparable dynamics with the 1000 N MR damper described in [9].
Two control rules are designed depending on the damper velocity state. In the region neighboring the hysteresis, the required voltage is applied following a saturation rule, where the voltage is maximum if the required force \( F_{\text{req}} \) is higher than the actual force \( F_{\text{act}} \) outputted by the damper, in absolute values, and of the same velocity sign. It is set to zero otherwise:

\[
v_{\text{req}} = v_{\max} \text{ if } |F_{\text{req}}| > |F_{\text{act}}| \text{ and } \text{sign}(F_{\text{req}}) = \text{sign}(\dot{x})
= 0 \text{ otherwise}
\] (15)

where \( v_{\max} \) is the maximum applicable voltage. This control rule is selected because of its simplicity in the region where the friction dynamics is complex to invert. The second control rule, outside the hysteresis region, is designed using a sliding controller. To select the required voltage \( v \), consider the following Lyapunov function specialized for a scalar control force [46]:

\[ V = \frac{1}{2} s^2 \] (16)

where \( s \) is the sliding surface \( s = F_{\text{act}} - F_{\text{req}} \). Taking the time derivative of (16), using (1), (2), (3) and (14), and recognizing that the system is at steady state \( (g(\dot{x}) = F_c \text{ and } \dot{x} \neq 0) \):

\[
\dot{V} = s \left[ \dot{F}_{\text{act}} - \dot{F}_{\text{req}} \right]
= s \left[ \dot{F}_{\text{friction}} + k_{\text{mfd}} \dot{x} + \beta c_{\text{mfd}} \dot{x}^\beta - \dot{F}_{\text{req}} \right]
= s \left[ \dot{F}_c \cdot \text{sgn}(\dot{x}) + \sigma_2 \ddot{x} + k_{\text{mfd}} \dot{x} + \beta c_{\text{mfd}} \dot{x}^\beta - \dot{F}_{\text{req}} \right]
= s \left[ F_{c,0} v_{\text{act}} \cdot \text{sgn}(\dot{x}) + F_{c,0} \nu_{\text{act}} \delta(\dot{x}) + \sigma_2 \ddot{x} + k_{\text{mfd}} \dot{x} + \beta c_{\text{mfd}} \dot{x}^\beta - \dot{F}_{\text{req}} \right]
= s \left[ -F_{c,0} \eta (v_{\text{act}} - v_{\text{req}}) \cdot \text{sgn}(\dot{x}) + \sigma_2 \ddot{x} + k_{\text{mfd}} \dot{x} + \beta c_{\text{mfd}} \dot{x}^\beta - \dot{F}_{\text{req}} \right]
= -\xi s^2
\] (17)

where \( \xi \) is a positive constant, the control voltage is selected to be:
\[
\dot{v}_{\text{req}} = \dot{v}_{\text{act}} + \frac{\left(-\sigma_2 \ddot{x} - k_{\text{mid}} \ddot{x} - \beta c_{\text{mid}} \dot{x}^{\beta-1} \ddot{x} + \dot{F}_{\text{req}} - \xi s\right)}{\eta F_{\text{c,0}}} \operatorname{sgn}(\ddot{x})
\]  
(18)

Note that \( \dot{F}_{\text{req}} \) cannot be directly measured, but can be approximated as \( \dot{F}_{\text{req}} \approx (F_{\text{req}}(t) - F_{\text{req}}(t-1)) / \Delta t \). The term \( \xi \) can incorporate system uncertainties or unmeasurable states.

Fig. 9 shows a 200 kN frictional brake controlled using (18) and (15) for a required damping force of 100 kN. The saturation control rule within the hysteresis region is clearly shown.

Figure 9: MFD behavior for a 100 kN required damping force under an harmonic excitation of 7.62 mm at 0.5 Hz: a) force-velocity; and b) voltage-velocity

3. Simulation and Discussion

3.1. Simulated Structure

A 39-story office tower located in downtown Boston, Massachusetts, is simulated for comparing and assessing the performance of the proposed semi-active device. The building was built in the 1990’s with fluid dampers in order to mitigate excessive acceleration levels produced by the proximity of an existing 52-story tower. The cost of the passive system was less than a million dollars [17]. Dampers are installed every other story from the 5th floor up to the 34th floor. The
structural system along with the dampers location is shown in Fig. 10.

![Diagram showing structural system with dampers](image)

Figure 10: elevation view of the simulated structure: a) X-direction; and b) Y-direction

The viscous dampers in the X-direction have a capacity of 1350 kN (300 kips) with a damping coefficient of 52550 kN·s/m (300 kips·s/in) below the 26th floor, and a capacity of 900 kN (200 kips) with a damping coefficient of 35000 kN·s/m (200 kips·s/in) from the 26th floor and above. The viscous dampers in the Y-direction have a capacity of 90 kN (20 kips) with a damping coefficient of 3500 kN·s/m (20 kips·s/in) below the 26th floor, and a capacity of 45 kN (10 kips) with a damping coefficient of 1750 kN·s/m (10 kips·s/in) from the 26th floor and above. Moreover, the viscous dampers in the Y-direction are installed using toggle braces [48] that amplify inter-story motion. A description of the actual damping strategy contained in the building can be found in [47], and details about the computer model are given in [49]. Table II summarizes the configuration of the viscous dampers.
Table II: Configuration of Viscous Dampers

<table>
<thead>
<tr>
<th></th>
<th>capacity (kN)</th>
<th>number of dampers</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>X-direction</td>
<td>Y-direction</td>
</tr>
<tr>
<td>below 26th floor</td>
<td>1350</td>
<td>90</td>
</tr>
<tr>
<td>above 26th floor</td>
<td>900</td>
<td>45</td>
</tr>
</tbody>
</table>

The MFD has been simulated as a replacement to the actual damping strategy for mitigation of accelerations caused by wind excitation. A random wind load has been generated, and scaled to match the acceleration of the 37th floor provided in [47]. A first simulation is ran using the MFD 200 kN at the actual viscous dampers location. A second simulation studies the performance of MFDs of equal capacity to the viscous dampers. A third simulation studies the possibility of a reduced number of dampers using the MFD 1350 kN in the X-direction to achieve a performance similar to the current viscous damping strategy. Lastly, despite that the current damping strategy is not designed for earthquake mitigation, the performance of the MFD at inter-story displacement mitigation, as well as its impact on the floor accelerations, are studied under the ElCentro 1940 North-South component earthquake scaled to a maximum of 0.12 g. Table III summarizes the configuration of the MFDs for each simulation.

Table III: configuration of MFDs for each simulations

<table>
<thead>
<tr>
<th>excitation</th>
<th>location</th>
<th>capacity (kN)</th>
<th>number of dampers</th>
</tr>
</thead>
<tbody>
<tr>
<td>simulation 1</td>
<td>wind below 26th floor</td>
<td>200</td>
<td>200</td>
</tr>
<tr>
<td></td>
<td>above 26th floor</td>
<td>200</td>
<td>200</td>
</tr>
<tr>
<td>simulation 2</td>
<td>wind below 26th floor</td>
<td>1350</td>
<td>90</td>
</tr>
<tr>
<td></td>
<td>above 26th floor</td>
<td>900</td>
<td>45</td>
</tr>
<tr>
<td>simulation 3</td>
<td>wind below 26th floor</td>
<td>1350</td>
<td>N/A</td>
</tr>
<tr>
<td></td>
<td>above 26th floor</td>
<td>1350</td>
<td>N/A</td>
</tr>
<tr>
<td>simulation 4</td>
<td>earthquake below 26th floor</td>
<td>1350</td>
<td>90</td>
</tr>
<tr>
<td></td>
<td>above 26th floor</td>
<td>900</td>
<td>45</td>
</tr>
</tbody>
</table>

3.2. Simulation Results

3.2.1. Simulation 1 A first simulation is conducted on the structure using the MFD 200 kN. The control objective is acceleration mitigation for serviceability. The maximum acceleration
profile is illustrated in Fig. 11 for both directions. The active control strategy refers to an ideal actuator saturating at the MFD capacity. The semi-active control case refers to an LQR controller computing the required force $F_{req}$ for the MFD. The passive-on case refers to a passive control using full voltage, and the passive-off case refers to passive control using no voltage. The passive-off case is essentially the fail-safe mechanism on its own (the spring and dashpot in parallel). The MFD without the fail-safe mechanism is also studies. Results show that the MFD 200 kN underperforms the 1350 kN viscous dampers in the X-direction, but is still capable of mitigating acceleration levels to the ranges of 50 mg. The good performance in the Y-direction is due to the larger device capacities, which is 15 times the 90 kN viscous capacity. For the control strategy, semi-active control gives similar performance compared to the passive-on case, in both directions. The fail-safe mechanism provides only a small mitigation. Its contribution is also found to be minimal when comparing results to the MFD without the fail-safe mechanisms. Certainly, the fail-safe mechanism could be designed with elements of higher capacity, depending on the design objectives. Finally, the internal force actuation of the $10^{th}$ MFD, as well as its voltage inputs in function of its velocity, are shown in Fig. 12 for the semi-active control case. The linear actuation force stays within the range of 4 kN, which is equivalent to two 2-kN linear actuators working on 12 V batteries.

![Figure 11: maximum acceleration profile, simulation 1: a) X-direction; and b) Y-direction](image-url)
3.2.2. Simulation 2 The second simulation compares MFDs of identical capacities to the viscous dampers currently installed in the building. Fig. 13 shows the maximum acceleration profile in both direction. The MFD strategy is clearly capable of outperforming the passive viscous strategy. In the X-direction, semi-active control reduces the maximum acceleration of the 37th floor by 59.2% compared to 46.7% for the viscous dampers. In the Y-direction, it is 27.4% versus 14.9% respectively.
3.2.3. Simulation 3  Inspired by the results from the second simulation, it is interesting to evaluate the number of MFDs that would be required to mitigate acceleration with similar performance to the viscous damping strategy. The X-direction is of interest because of the significant mitigation capacity of the MFDs demonstrated in the second simulation. Fig. 14 shows the maximum acceleration profile using an MFD every 6 floors starting at the 5th-floor. Results show that semi-active control and the passive-on strategies are capable of similar performance, using 10 dampers (2 dampers per floor, each 5 floors) instead of 30 for the viscous case. The semi-active control scheme could result in significant savings. Table IV summarizes the maximum acceleration of the 37th-floor under various control strategies. Remark: it was found that some devices under semi-active control do substantially more work than others, which signifies that a more optimal configuration could be established. This is out of the scope of this paper.

![Figure 14: maximum acceleration profile in X-direction, simulation 3](image)

Interestingly, both semi-active and passive-on control schemes give similar performance. The passive-on case is in essence a passive damper where the brake actuator is replaced by a spring. Thus, no semi-active control is necessary for the passive-on case, which could decrease the cost of the MFD. However, as it will be shown in the next simulation, it does not hold for earthquake mitigation. In addition, a passive MFD could not be used to mitigate small
Table IV: maximum acceleration at the 37th floor

<table>
<thead>
<tr>
<th>control strategy</th>
<th>X-direction</th>
<th></th>
<th>Y-direction</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$\ddot{x}_{37}$ (mg)</td>
<td>reduction (%)</td>
<td>$\ddot{y}_{37}$ (mg)</td>
<td>reduction (%)</td>
</tr>
<tr>
<td>simulation 1</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>uncontrolled</td>
<td>70.8</td>
<td>46.3</td>
<td></td>
<td></td>
</tr>
<tr>
<td>viscous dampers</td>
<td>46.7</td>
<td>34.0</td>
<td>39.4</td>
<td>14.9</td>
</tr>
<tr>
<td>active control</td>
<td>53.8</td>
<td>24.0</td>
<td>35.4</td>
<td>23.5</td>
</tr>
<tr>
<td>semi-active control</td>
<td>53.3</td>
<td>24.7</td>
<td>32.8</td>
<td>29.2</td>
</tr>
<tr>
<td>passive-on</td>
<td>50.7</td>
<td>28.4</td>
<td>34.0</td>
<td>26.6</td>
</tr>
<tr>
<td>passive-off</td>
<td>68.8</td>
<td>2.82</td>
<td>41.6</td>
<td>10.2</td>
</tr>
<tr>
<td>no fail-safe</td>
<td>53.9</td>
<td>23.9</td>
<td>33.7</td>
<td>27.2</td>
</tr>
<tr>
<td>simulation 2</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>semi-active control</td>
<td>28.9</td>
<td>59.2</td>
<td>32.7</td>
<td>27.4</td>
</tr>
<tr>
<td>passive-on</td>
<td>35.1</td>
<td>50.4</td>
<td>35.4</td>
<td>23.5</td>
</tr>
<tr>
<td>simulation 3</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>semi-active control</td>
<td>45.9</td>
<td>35.2</td>
<td>N/A</td>
<td></td>
</tr>
<tr>
<td>passive-on</td>
<td>40.8</td>
<td>42.4</td>
<td>N/A</td>
<td></td>
</tr>
</tbody>
</table>

magnitude excitations, as a substantial motion would be required to set the drum in motion. Also, using full voltage increases the work done on the damper, thus its heat. Table V shows the increase in temperature of the MFDs for several control strategies. The MFD increases temperature over the 50 seconds excitation by several degrees. Consequently, a release of the braking mechanism could be necessary if the device is to be utilized over a longer period of time. Results are also consistent with the simulations. For instance, larger MFDs (simulations 2 and 3) produce more heat, and the utilization of a third of the devices (simulation 3) results in a large demand on the dampers, thus higher heat. The passive-on strategy does more work than the control cases, as one would expect.

Table V: maximum increase in temperature ($^\circ$C/50 sec)

<table>
<thead>
<tr>
<th>control strategy</th>
<th>X-direction</th>
<th>Y-direction</th>
</tr>
</thead>
<tbody>
<tr>
<td>simulation 1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>semi-active control</td>
<td>12.8</td>
<td>1.35</td>
</tr>
<tr>
<td>passive-on</td>
<td>14.9</td>
<td>2.55</td>
</tr>
<tr>
<td>simulation 2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>semi-active control</td>
<td>41.5</td>
<td>4.37</td>
</tr>
<tr>
<td>passive-on</td>
<td>73.2</td>
<td>8.23</td>
</tr>
<tr>
<td>simulation 3</td>
<td></td>
<td></td>
</tr>
<tr>
<td>semi-active control</td>
<td>60.6</td>
<td>N/A</td>
</tr>
<tr>
<td>passive-on</td>
<td>74.8</td>
<td>N/A</td>
</tr>
</tbody>
</table>

Fig. 15 shows a comparison of hysteresis loops between the viscous damper and the MFD located between the 25th and the 26th floor. Fig. 15a compares results from the first simulation.
The hysteresis loops are in similar ranges for both systems. A slight increase in the MFD capacity would hypothetically result in similar mitigation performance than the viscous strategy. Fig. 15b compares results from the third simulation. Clearly, using MFDs of larger capacities allows the hysteresis loops to immediately reach the full damping capacity, resulting in a more efficient energy dissipation.

Figure 15: Comparison of hysteresis loops between the viscous damper and the MFD located between the 25th and the 26th floor: a) results from simulation 1; and b) results from simulation 3

3.2.4. Simulation 4  A last simulation is conducted on the structure subjected to the ElCentro 1940 earthquake. The control objective is inter-story displacement mitigation to minimize structural damage. For this simulation, the configuration of the MFDs from the second simulation is studied. Table VI summarizes the maximum inter-story displacement. In the X-direction, semi-active control performs significantly better than the viscous damping case as well as the passive-on case. In the Y-direction, the passive-on case performs better. This is explained by the low controllability range of the MFD 90 kN (20 kips). Table VII shows the maximum absolute acceleration of the structure under various control strategies, which includes the maximum acceleration of both the entire building $\ddot{x}_{1-39}, \ddot{y}_{1-39}$ and the maximum acceleration at the damper locations $\ddot{x}_{5-34}, \ddot{y}_{5-34}$. The control strategy does reduce the
acceleration for the entire structure, but performs the same as the passive viscous strategy at the damper locations due to the added stiffness. The passive-on control strategy gives the best performance for acceleration mitigation.

Table VI: maximum inter-story displacement

<table>
<thead>
<tr>
<th>control strategy</th>
<th>X-direction</th>
<th>Y-direction</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$</td>
<td>z</td>
</tr>
<tr>
<td>uncontrolled</td>
<td>27.9</td>
<td>36.0</td>
</tr>
<tr>
<td>viscous dampers</td>
<td>20.1</td>
<td>28.0</td>
</tr>
<tr>
<td>semi-active control</td>
<td>16.2</td>
<td>41.9</td>
</tr>
<tr>
<td>passive-on</td>
<td>19.6</td>
<td>29.8</td>
</tr>
</tbody>
</table>

Table VII: maximum absolute acceleration

<table>
<thead>
<tr>
<th>control strategy</th>
<th>X-direction</th>
<th>Y-direction</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$\ddot{x}_{1-39}$ (mg)</td>
<td>$\ddot{x}_{5-34}$ (mg)</td>
</tr>
<tr>
<td>uncontrolled</td>
<td>196</td>
<td>195</td>
</tr>
<tr>
<td>viscous dampers</td>
<td>186</td>
<td>171</td>
</tr>
<tr>
<td>semi-active control</td>
<td>180</td>
<td>172</td>
</tr>
<tr>
<td>passive-on</td>
<td>184</td>
<td>165</td>
</tr>
</tbody>
</table>

4. Conclusion

A new semi-active device for control of large-scale structure has been introduced: the MFD. The device has the advantage of using existing reliable technologies, which makes it an excellent candidate for implementation. Simulations have showed that the MFD 200 kN is capable of wind mitigation when compared to the performance of a passive viscous strategy of much larger capacity. Also, when a larger MFD is used, it has been shown that it can perform similarly with only a third of the number dampers currently contained in the existing structure. Finally, it was demonstrated that the proposed device was capable of mitigating inter-story displacements in the occurrence of an earthquake.
5. Acknowledgements

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