Semi-active Control of a Banded Rotary Friction Device

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ABSTRACT
Friction-damping devices are robust, low-cost solutions for structural control. Semi-active friction dampers in particular are able to dissipate sizable amounts of energy with comparatively little input power. However, modeling friction is a difficult problem, and the development of control algorithms for a semi-active friction damper proves challenging. Previously introduced by the authors is a novel semi-active friction damper termed the Banded Rotary Friction Device (BRFD). During operation, the BRFD develops friction between an internal steel drum and friction bands; this is achieved by transducing input displacement into rotation of the drum. Given its unique geometry, the BRFD is capable of providing sizable damping force, especially compared to that afforded using a traditional, planar friction surface. In this preliminary work, a semi-active model for the BRFD is introduced whereby damping is controlled via displacements to electric actuators. The model is validated using devised semi-active displacement profiles, and results show that the BRFD is capable of achieving large levels of force amplification, lending itself to applications in structural control and multihazard mitigation.

Keywords: Variable friction damper (VFD), Semi-active control, Structural control, LuGre model

INTRODUCTION
The effective reduction of structural vibrations has been the subject of much research. Damping devices are a common solution for structural control as they are able to attenuate vibration-induced displacements at targeted frequencies [1]. In particular, the tuned mass damper (TMD) has shown success at dissipating energy associated with wind loads [2]. Yet, the efficacy of TMDs for seismic excitation is comparatively poor, and in general passive TMDs suffer from limited functional bandwidth [3]. Research has been conducted to address this shortcoming, and both semi-active and active classes of TMDs have been explored [4]. In the interest of providing damping over a larger range of excitation frequencies, developments have led to a variety of damping techniques and devices, including the friction damper.

Installed within a building, friction dampers dissipate structural kinetic energy as heat by developing friction between two surfaces. Compared to other passive devices, friction dampers generally provide greater levels of energy dissipation over a larger range of excitation frequencies [5]. However, due to an inability to self-center, passive friction dampers can introduce residual deformation in structures. Moreover, the stick-slip operation of friction dampers is such that jerky responses are induced under large accelerations [6]. For enhanced performance, the slip force of a friction damper can be controlled by electrically altering a parameter of the device; such semi-active friction dampers are termed variable friction dampers (VFDs). A large advantage of VFDs compared to active control strategies is that VFDs require the input of little energy, a scarce resource during seismic or sustained wind events [7]. VFDs are mechanically robust devices capable of providing competitive, reliable damping [8].

Friction is notoriously difficult to model, and a large barrier to VFD implementation has been the design of control algorithms consistent with the dynamic behavior of friction. Introduced for control systems involving dry friction [9], the LuGre model is able to describe both hysteretic and velocity-dependent phenomina associated with friction [10]. Using the LuGre model, general semi-active, bang-bang control laws were proposed to maximize instantaneous energy dissipation for VFDs [11]. The development of internal control laws for a VFD, however, requires much knowledge of device geometry, dynamics, and operating characteristics [12].

In this work, a semi-active model for a novel rotary friction damper is introduced that controls damping with displacement inputs to electric actuators. Using data collected from damper operation in passive mode, developed static and kinetic friction are written as functions of electric actuator positions, and a modified, semi-active LuGre model is proposed. The model is validated using devised semi-active displacement profiles, and conclusions are drawn from model performance and damper applicability. The contributions of this work are twofold: (1) select relationships necessary for semi-active control of a rotary friction damper are characterized; and (2) the viability of said device for semi-active structural control is analyzed.

BACKGROUND

The Banded Rotary Friction Device (BRFD) is a semi-active rotary friction damper based on band-brake technology [13]. The BRFD consists of an internal steel drum wrapped by three elastic bands lined with friction material. During excitation, the BRFD transduces lateral displacement into angular motion, and friction develops as the drum rotates against the anchored elastic bands. For testing, the BRFD is bolted to a steel foundation beam. A hydraulic actuator provides lateral displacement profiles to the drum, and electric actuators attached to either end of the elastic bands allow for control inputs to the BRFD. By retracting or extending the electric actuators, the force with which the bands clamp onto the drum may be varied, altering normal force and thus damping. Load cells are connected to the hydraulic and electric actuators to measure output friction (damping force) and band tension (actuator force), respectively. The BRFD and testbed used for control and data acquisition are displayed in Figure 1.

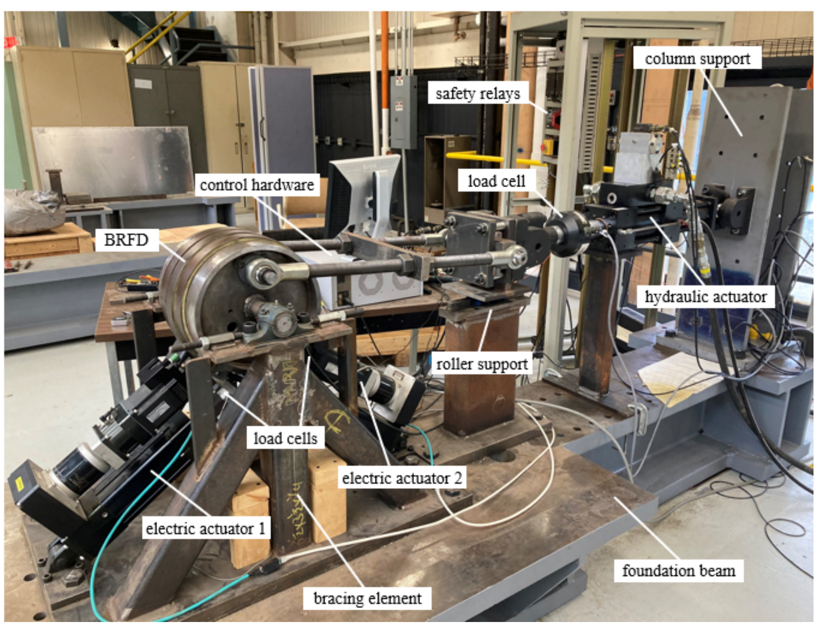


Figure 1: BRFD and testbed with key components labeled.

Although equipped with electric actuators and proposed as a semi-active damper, the BRFD has only been characterized while operating in passive mode. That is, tests to date have simply commanded the electric actuators to maintain position during excitation. As the drum rotates and draws the elastic bands taut, the electric actuators deflect between a slack and taut condition, and contact pressure between the drum and bands increases with respect to the point of applied force. Illustrated in Figure 2, this phenomenon contributes to a self-energizing effect that stores energy in the system as the drum displaces from equilibrium. During passive operation, the level of damping afforded by the BRFD varies with applied force. Figure 3 shows a typical set of hysteresis loops resulting from BRFD excitation with harmonic input. Despite all depicted tests having the same displacement amplitude and frequency, notice that damping increases with the pretension forces in the friction bands. This relationship is further elucidated in Figure 4, a plot of BRFD force amplification factors against pretension forces applied by the electric actuators. Defined to be the ratio of damping force to slack-actuator force, the BRFD is able to provide large levels of force amplification, a parameter that may be controlled by the position of the electric actuators.

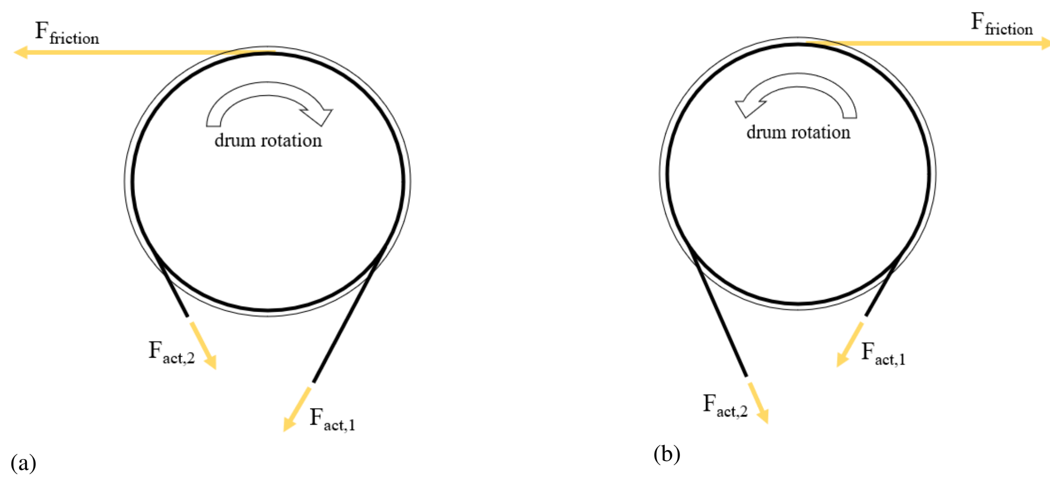


Figure 2: Diagram of forces acting on the BRFD: (a) forward rotations, $F_{act,2} \gg F_{act,1}$; (b) backward rotations, $F_{act,1} \gg F_{act,2}$.

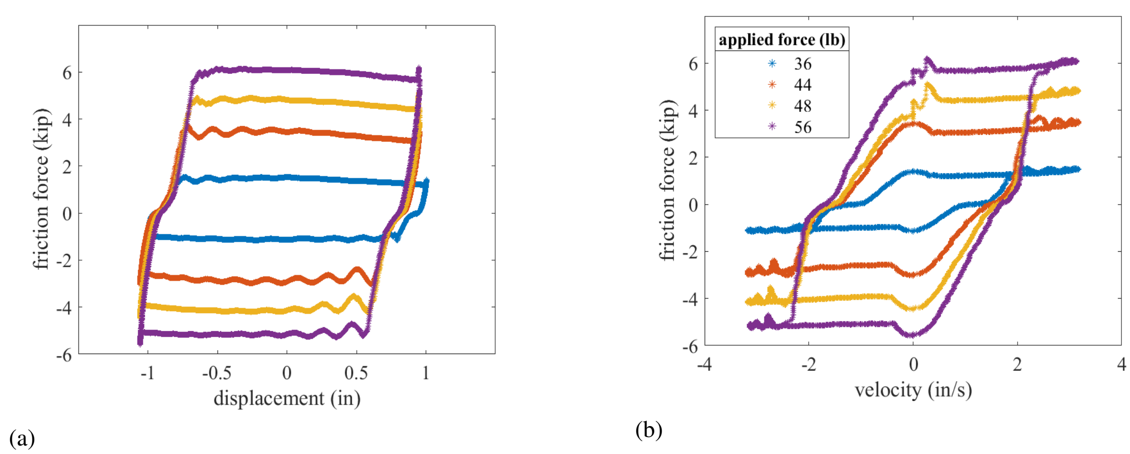


Figure 3: Passive mode hysteresis loops for various pretension forces: (a) force-displacement plots; (b) force-velocity plots.

Friction modeling of the BRFD has been the subject of much analysis. The LuGre model has historically served as a baseline for friction characterization of the BRFD as it is able to describe stick-slip motion and the Stribeck effect [13, 14]. Presented in equations 1-3, the LuGre model consists of a first-order, nonlinear differential equation with state variable z. The model describes a system of microscopic bristles at the interface of two surfaces which – upon relative motion – deflect and give rise to friction. The model parameters σ_0, σ_1, and σ_2 are aggregate bristle stiffness, microdamping, and the coefficient of viscous friction, respectively. To imitate the Stribeck effect, the function $g(v)$ evolves friction from a static value for small velocities, F_s, to a steady-state, kinetic value for larger velocities, F_c. Notably, F_s and F_c are constant parameters. While they can be written as a function of drum velocity to capture asymmetries associated with displacement profiles, to model changes in damping

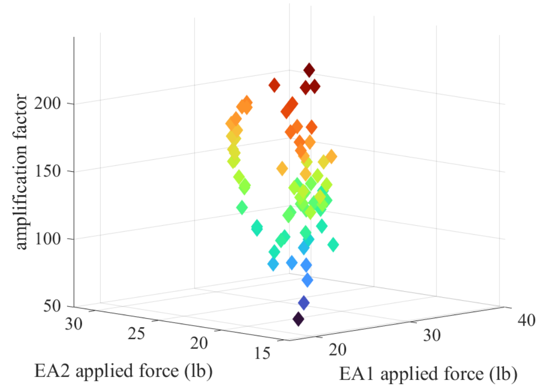


Figure 4: Plot of BRFD force amplification factors against applied electric actuator forces generated from passive test data.

induced by electric actuator displacements, it is necessary to characterize how F_s and F_c depend on the positions of the electric actuators [15]. Such an insight is a key first step towards achieving semi-active control of the BRFD.

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

$$g(v) = F_c + (F_s - F_c)e^{-(v/v_s)^2} \tag{2}$$

$$F = \sigma_0 z + \sigma_1 \dot{z} + \sigma_2 v \tag{3}$$

ANALYSIS

For device characterization, a single sinusoidal displacement profile with an amplitude and frequency of 1 in and 0.5 Hz, respectively, is selected for use in testing. Initial positions for the electric actuators are identified to ensure the following: (1) developed static and kinetic friction for forward and backward rotations of the drum are nearly equal; and (2) accrued actuator forces are only a fraction of their maximum capacity. Beginning with the electric actuators at these initial positions throughout their stroke, sets of passive tests are conducted on three separate dates. Between tests, the positions of the electric actuators are incrementally retracted. Tests are conducted until the force on either electric actuator exceeds a set safety limit.

Using the collected passive characterization data, developed static and kinetic friction forces are plotted against electric actuator initial positions. To capture the rates at which these forces change with displacements of the electric actuators, multiple linear regression (MLR) is performed, and models of the form $y = \beta_0 + \beta_1 x_1 + \beta_2 x_2 + \varepsilon$ are fit to the data. Figure 5 depicts corresponding MLR models for an example set of tests. By using linear models, the potential for a coupling effect to exist between the electric actuators is ignored. However, obtained adjusted R^2 values are all at and above 0.88, suggesting that strong linear relationships exist between developed friction and actuator positions.

With the force-position models in hand, developed static and kinetic friction are written as functions of electric actuator positions and drum velocity as

$$F_s(x_1, x_2, v) = \begin{cases} s + S_1(x_1 - x_1') + S_2(x_2 - x_2'), & v \geq 0 \\ s + S_3(x_1 - x_1') + S_4(x_2 - x_2'), & v < 0 \end{cases} \tag{4}$$

and

$$F_c(x_1, x_2, v) = \begin{cases} c + C_1(x_1 - x_1') + C_2(x_2 - x_2'), & v \geq 0 \\ c + C_3(x_1 - x_1') + C_4(x_2 - x_2'), & v < 0, \end{cases} \tag{5}$$

respectively. For both functions, x_1' and x_2' correspond to the initial positions of electric actuators one and two, whereas x_1 and x_2 correspond to the time-dependent positions of electric actuators one and two. The S_i and C_i slopes capture, respectively, the rates at which static and kinetic friction vary with displacements of the electric actuators; they are identified by averaging parameters obtained from force-position plots like that in Figure 5. s and c are those static and kinetic friction values which

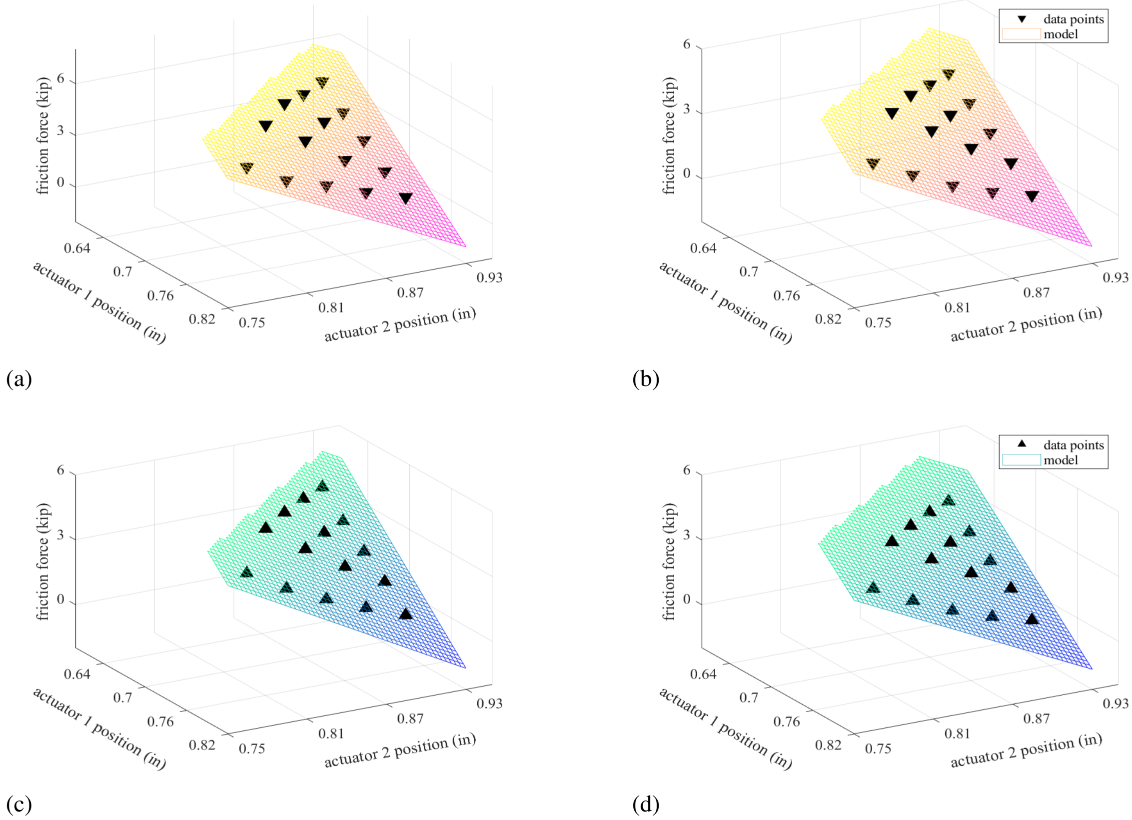


Figure 5: Static and kinetic friction forces against electric actuator initial positions: (a) static, forward rotation; (b) static, backward rotation; (c) kinetic, forward rotation; (d) kinetic, backward rotation.

develop with no displacements of the electric actuators from their initial positions. Substitution of the derived functions into equation 2 gives

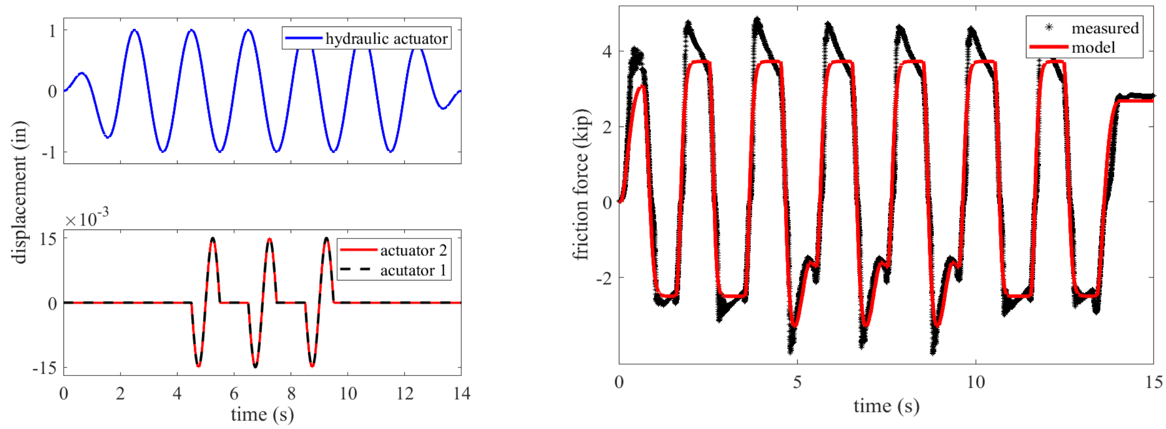
$$g(v) = F_c(x_1, x_2, v) + (F_s(x_1, x_2, v) - F_c(x_1, x_2, v))e^{-(v/v_s)^2}, \tag{6}$$

a small modification to the standard LuGre model allowing changes in developed static and kinetic friction to be predicted from electric actuator displacements.

To validate the proposed model, semi-active tests are devised that simultaneously displace the drum and electric actuators. 12 validation tests are conducted in total, six with harmonic inputs to the electric actuators and six with step inputs to the electric actuators. All tests excite the drum with the same sinusoidal input of amplitude 1 in and frequency 0.5 Hz. Table 1 details electric actuator displacement parameters for considered validation profiles. Semi-active model predictions are made using measured responses of the electric actuators. Figures 6 and 7 show actuator displacement commands along with damper response and model predictions for a harmonic- and step-type validation test, respectively. Notice that – for those cycles of drum rotation with inputs to the electric actuators – non-constant static and kinetic friction values emerge.

Table 1: Electric actuator displacement parameters for harmonic and step validation tests.

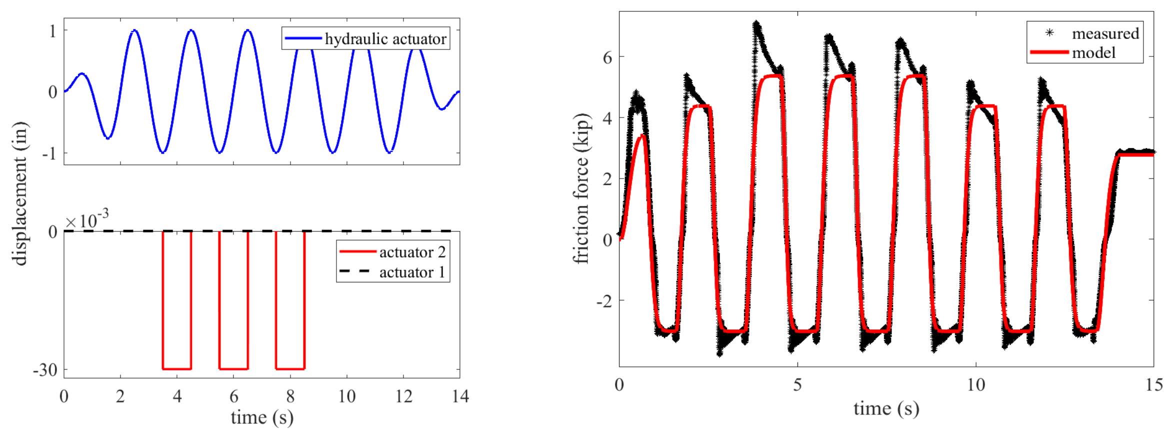
Test #	**Controlled Actuator(s)**	**Displacement Amplitude (in)**	**Drum Rotation**
1	one	0.03	forward
2	one	0.03	backward
3	two	0.03	forward
4	two	0.03	backward
5	one & two	0.015	forward
6	one & two	0.015	backward



(a)

(b)

Figure 6: Harmonic validation test 6 data showing (a) actuator displacement commands and (b) damper response with model predictions.



(a)

(b)

Figure 7: Step validation test 3 data showing (a) actuator displacement commands and (b) damper response with model predictions.

Having dynamic static and kinetic friction parameters, the semi-active, modified LuGre model successfully captures changes in damping induced by electric actuator displacements. Yet, the model does exhibit sizable error on the validation dataset. Importantly, however, much of model error stems not from the introduced F_s and F_c functions, but rather from the following identified error modes: (1) initial discrepancies between measured friction and model predictions due to residual static force; (2) difficulties in fitting LuGre model parameters to semi-active data; and (3) an inability to capture spikes in damping observed upon reversal of drum rotation.

CONCLUSION

This worked developed a semi-active model for a novel rotary friction damper using data collected from damper operation in passive mode. Adopting the LuGre dry friction model as a baseline, static and kinetic friction parameters were modified to be functions of the control inputs of the analyzed damper – electric actuator positions. Semi-active tests were developed to validate the proposed model, and results demonstrate that the semi-active model can successfully predict changes in damping induced by displacements of the electric actuators. Altogether, the analyzed damper shows much promise for semi-active structural control. Displacements of the electric actuators by just 0.03 in were able to realize 1 kip increases in damping. With the relationship between electric actuator displacements and developed friction now understood, future work may focus on the development of internal control algorithms for the BRFD.

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