Magnetorheological Dampers

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***Abstract*—This paper reviews and summarizes magnetorheological (MR) dampers. MR dampers are hydraulic dampers filled with an MR fluid, and while there are a variety of applications, they are most-widely used in motor vehicle suspensions. By adjusting the magnetic field that surrounds the MR fluid, the damping coefficient can be adjusted to provide better control of the system dynamics. Positional sensors placed in strategic locations send positional information as an input to a system controller which then outputs an electrical current to each damper, creating an electromagnetic field that alters the viscosity of the fluid. This paper details the operating mechanisms of MR damper systems including the physical hardware, the electrical components, and the different ways that they can be controlled.**

***Keywords*—Magnetorheological damper, MR damper, MR fluid, semi-active suspension, Bouc-Wen model, non-linear model, numerical simulation**

I. Introduction

Dampers have been incorporated into vehicle suspensions since the early twentieth century. In fact, one of the earliest hydraulic dampers invented was produced and commercialized in 1912. Automotive suspension is intended to provide a smooth and comfortable ride for passengers by mitigating effects of irregular terrain, but it must also control the dynamic tire load to improve stability and safety of the vehicle. Conventional dampers, also referred to as shock absorbers, involve a spring (to store the kinetic energy of the load) and a damper (to dissipate that energy, usually as heat). As technology advanced, semi-active suspensions were invented to incorporate control into the damper system. This was usually done mechanically by changing the damper orifice by means of hydraulic valves. Unfortunately, these systems are often noisy, and neither cost- nor time-effective. This is where magnetorheological dampers (MR) came onto the scene.

Rheology is the branch of physics that deals with the deformation and flow of matter, especially the non-Newtonian flow of liquids and the plastic flow of solids, and the prefix *magneto-* implies a relation to magnets or magnetism. There is such thing as an *electro-rheological* damper which uses an electric current, instead of a magnetic field, in a fluid, but these systems require a much larger power supply voltage making them less advantageous than an MR damper. In the last 20 years or so, MR dampers have revolutionized the world of automobile suspensions, providing a control solution using smart fluids with adjustable viscosity to control the damping force. The key component to this phenomenon is the MR fluid, which in the presence of a magnetic field, can change instantaneously from a free-flowing liquid into a semi-solid with infinite controllability and precision in terms of viscosity and yield strength. Tireless research is being conducted to optimize the controllability of MR dampers, but they are already used in seat suspensions as well as main suspensions of passenger vehicles by large corporations like General Motors, BMW, and others.

II. Technology & Hardware

A. Mechanical System

The mechanical system of magnetorheological dampers contains only a single moving part. A damper consists of a cylindrical housing within which a piston on a shaft moves through a fluid. The viscosity of the fluid causes it to resist the motion of the piston through the fluid. In the case of a magnetorheological damper, the piston is filled with a magnetorheological fluid. A magnetorheological fluid is a fluid which contains magnetic particles that, when exposed to a magnetic field, line up and cause the viscosity of the fluid to change. This change in particle alignment is shown in Figure 1.

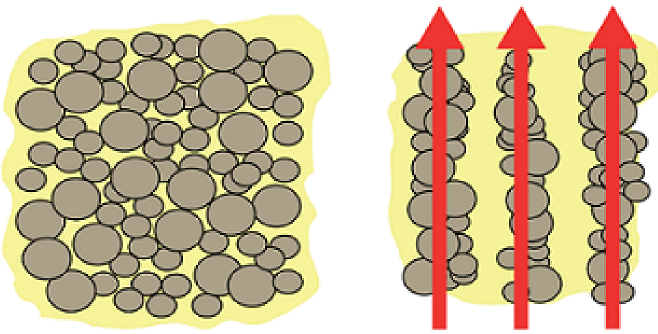

Figure 1. MR Fluid Particles aligning in a magnetic field [1].

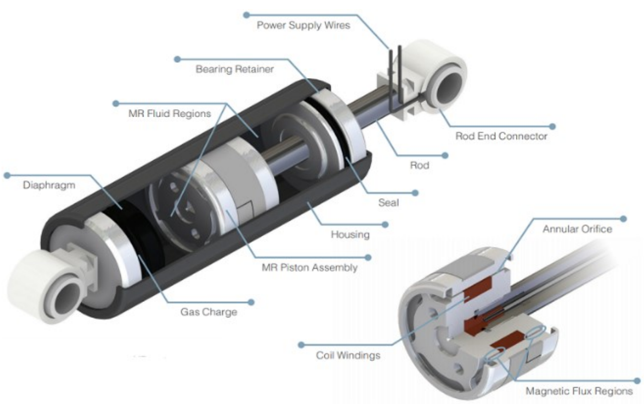

Figure 2. Typical MR damper assembly [1].

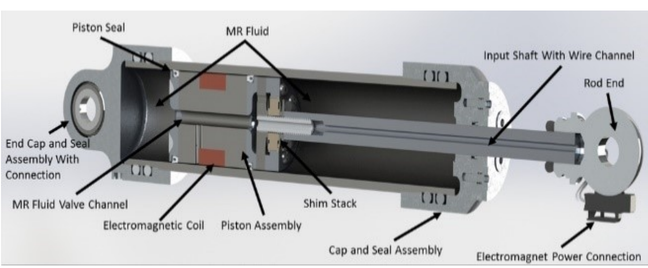

Figure 3. Cross-sectional anatomy of MR damper.

By placing coils of wire within the piston, the magnetorheological fluid within the damper can be exposed to a quickly and easily adjustable magnetic field. The layout of an MR damper can be seen in Figures 2 and 3. The changes in viscosity made possible by this arrangement allow for damping force to be rapidly adjusted and infinitely variable. This is particularly valuable in instances such as suspension on automobiles, where the usage of traditional dampers with fixed damping coefficients results in an unavoidable compromise between performance and response in one condition versus another.

B. Electric Circuit

The electrical system of magnetorheological dampers requires several elements. A voltage source is needed to provide the electric current through the piston to generate the magnetic field which the entire system depends on. Load and position sensors are placed in strategic locations on the car and there is a controller which is able to vary the electrical current being sent to each damper independently. These sensors can monitor the road surface up to 1000 times per second and the controller can make variations within a millisecond [2]. Typically, displacement sensors on the car are divided into five categories based on the aspect of vehicle dynamics that they are monitoring: body control, handling control, wheel hop control, end stop control, and landing control. The data from all of these sensors then gets sent to the closed loop current controller [1]. The controller reads the data and then according to its programming adjusts the current according to its programming to the relevant dampers to provide the desired system response characteristics. For example, if a vehicle is turning to the right and it is experiencing body roll as vehicle weight shifts to the left, the load and positional sensors located in the left side of the car would detect this leftwards weight shift and send that information to the controller. The controller could then send more current to the two dampers on the left side suspension to increase the viscosity, and therefore the damping force, in the left dampers to resist the vehicle's body roll. This can improve the vehicles handling and therefore make it safer in emergency situations. The damper itself does not require usage of must electricity, with a typical electrical usage of 5-25W per damper. Current draw typically stabilizes around 2.5 amps, with momentary spikes of up to 5 amps for shocks [2].

III. MATHEMATICAL MODELS

As a semi-active MR-damper is a non-linear dynamic system, this non-linear nature can be used as an advantage to allow more efficient and accurate control of the system [3]. In the case of MR dampers, a hysteresis loop is observed in magnetic or magnetized materials and magneto-rheological (MR) liquids, and it is found to be one of the most suitable material for use in vibrational dampers and shock absorbers. Hysteresis is a memory dependent non-linear behavior in which the system output is not only dependent on the instantaneous input but also on the history of the input. MR-dampers are a non-linear control system where the inputs are the stroke (displacement), and the command current; the current is the control input which modulates at high-bandwidth the damping characteristic through the variation of a magnetic field. The output is the force delivered by the damper.

A. Bouc-Wen Model

The Bouc–Wen model is a set of differential equations describing the hysteretic characteristic of the damper force/velocity response [4, 5]. They are given as:

$$m\ddot{u}(t) + c\dot{u}(t) + F(t) = f(t) \tag{1}$$

$$F(t) = ak_i u(t) + (1-a)k_i z(t) \quad (2)$$

Here, m represents the mass, $u(t)$is the displacement, c is the linear viscous damping coefficient, $F(t)$ is the restoring force and $f(t)$ is the excitation force while the over dot denotes the derivative with respect to time, a is the ratio of post-yield k_f to pre-yield (elastic), k_i is stiffness, F_y is the yield force, u_y is the yield displacement and $z(t)$ is a non-observable hysteretic parameter (usually called the *hysteretic displacement*) that follows a non-linear differential equation.

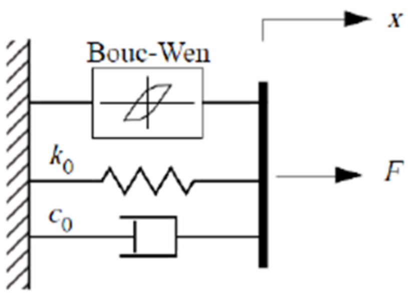

Figure 4. Schematic diagram of Simple Bouc-Wen model for MR damper.

The model parameters for the Bouc-Wen model can also be identified using various methods such as computationally efficient genetic algorithm and adaptive charged system search optimization. By adjusting twelve parameters of the simple Bouc-Wen model or fourteen parameters of the modified model through solving an optimization problem with the objective of fitting model response to the experimental data for a prototype MR damper, it is possible to predict the response of damper to any random inputs (displacement and applied voltage) before and after the yield areas.

B. *Dahl Model*

The Dahl model is simultaneously one of the simplest and most useful models. This model considers quasi-static bonds of friction and treats the relationship between frictional force and position to be analogous to the stress-strain curve [6]. The model is represented by the following equations:

$$F_{mr} = k\dot{z} + (k_{wa} + k_{wb}v)w \quad (3)$$

$$w = \rho(\dot{z} - |\dot{z}|w) \quad (4)$$

Where F_{mr} is the exerted force from the MR-damper, v is the control voltage, w is the dynamic hysteresis coefficient, and k , k_{wa}, k_{wb} and ρ are the parameters that control the shape of the loop. The above equations are used to design the control system in Simulink where there is a feedback control input velocity $\dot{z}$(dZ s/dt) and an output signal to the un-sprung mass and sprung mass, represented by F_{mr}.

C. *Bingham Model*

Bingham Model is based on the Bingham plastic [7], which is a viscoplastic materials that behaves as a rigid body at low stress but flows as viscous fluid at high stresses and was proposed in 1985 by Eugene C. Bingham who proposed its mathematical form. It follows the following equation:

$$F_{mr} = F_c sgn(\dot{y}) + c_o\dot{y} + F_o \quad (5)$$

Where y is a piston's relative displacement and $\dot{y}$ is its derivative that is the velocity if a piston; F_c is frictional force; c_o is damping coefficient; F_o is offset force (constant force value). The signum function $sgn(\dot{y})$ takes care of the direction of the frictional force depending on the relative velocity of the hysteresis variable. Substituting the variables from our simulation system the equation becomes the following:

$$F_{mr} = F_c sgn(\dot{z}) + c_o\dot{z} + F_o \quad (6)$$

D. *LuGre Model*

For the hysteresis loops, the LuGre model was developed and applied for damping simulation studies [8]. This model accounts for three different type of frictions which are observed in dry friction and fluid flows, viz., Coulomb, stick-slip and Stribeck effects that follows the following function:

$$F_{mr}(t) = \sigma_o y(t) + \sigma_1\dot{y}(t) + \sigma_2\dot{z}(t) \quad (7)$$

$$\dot{y}(t) = \dot{z}(t) - \frac{|z(t)|}{y_{ss}(\dot{z}(t))}y(t) \quad (8)$$

$$y_{ss}(\dot{z}(t)) = \frac{1}{\sigma_o}(F_c + (F_s - F_c)e^{\left(-\frac{z(t)}{v_s}\right)^2}) \quad (9)$$

Where $\sigma_o, \sigma_1, \sigma_2$ are stiffness, damping and viscous friction coefficients, respectively; y(t) is the friction state (average deflection of the bristles), $\dot{y}(t)$ is the velocity of the friction state, $\dot{z}(t)$ is the relative velocity of the sprung mass, F_c is the Coulomb friction force, F_s is the sticktion force, and v_s is the Stribeck velocity.

Based on the research of the above mathematical models, the Bingham model follows a linear pattern and doesn't follow a hysteresis loop. Although Dahl and LuGre models are able to produce a hysteresis loop, their controllability is fixed and therefore there are less parameters that can be tweaked. Hence, Bouc-Wen model has been identified as the best mathematical model for its simplicity and also its ability to control the hysteresis loop because of availability of 12 controllable parameters.

IV. CONTROL ABILITIES

A real-time closed-loop feedback control of the MR damper is achieved by changing the damping force under a direct current generated by a controller according to the signal of a force sensor and response of the displacement transducer sensor. A piezoelectric force sensor could be a possible choice for the force sensor. A schematic diagram of a MR semi-active control systems structure is shown in Figure 6.

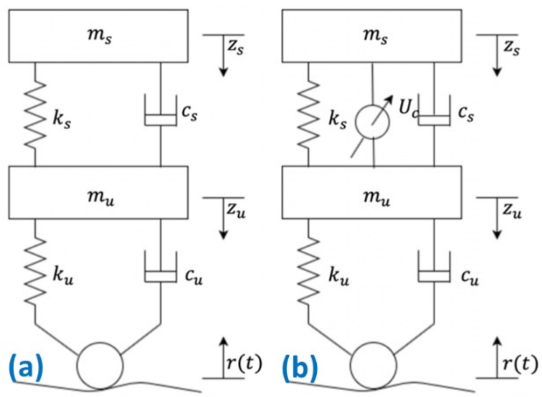

Figure 5. Passive (a) and semi-active (b) suspension models.

A. Quarter Car Model with MR damper

The use of MR dampers for vehicle suspension is compared with a conventional suspension system using a quarter-car model for a single scenario, i.e. for a constant current value. Figure 5 shows the two degree of freedom suspension systems representing quarter car models with different suspension systems. Equations of motion can be derived for the two mass bodies which have an un-sprung mass (half of axle mass) m_u and sprung mass (quarter car body mass) m_s as follows:

(a) For passive suspension system

$$m_s\ddot{z_s} + c_s(\dot{z_s} - \dot{z_u}) + k_s(z_s - z_u) = 0 \quad (10)$$

$$m_u\ddot{z_u} + c_s(\dot{z_u} - \dot{z_s}) + k_s(z_u - z_s) + c_u\dot{z_u} + k_u z_u = k_u r + c_u\dot{r}$$

(b) For semi-active suspension system

$$m_s\ddot{z_s} + c_s(\dot{z_s} - \dot{z_u}) + k_s(z_s - z_u) = U_c \quad (11)$$

$$m_s\ddot{z_u} + c_s(\dot{z_u} - \dot{z_s}) + k_s(z_u - z_s) + c_u\dot{z_u} + k_u z_u = -U_c + k_u r + c_u\dot{r}$$

Where $z_s, \dot{z_s}$ and $\ddot{z_s}$ are displacement, velocity and acceleration of the sprung mass (quarter car body mass), respectively; $z_u, \dot{z_u}$ and $\ddot{z_u}$ are displacement, velocity and acceleration of the un-sprung mass (half of axle mass and one wheel), respectively; c_s and c_u are damping coefficients of suspension and tire; k_s and $k_u$are stiffness of suspension and tire; $r(t)$ and $\dot{r}$ are terrain roughness (disturbance) displacement and velocity with respect to longitudinal speed of the vehicle, U_c is the force generated by the controller that takes into account terrain roughness $r(t)$, and vertical displacement and velocity of the vehicle. In this model, the Bouc-Wen model can be applied to account for hysteresis effect of a MR damper.

A simulation model for the suspension systems is shown in Figure 7 is built in Simulink by using the equations expressed in equations (10) and (11). The simulation model has two input sources, viz. $z(t)$ displacement and $dz(t)$ velocity of the sprung mass (m_s) of the system, and two output signals for control force F_{mr} going to the sprung mass (m_s) with (-) minus sign and to the un-sprung (m_u) mass with (+) plus sign. Note that $dz(t)$ is equal to $\dot{z_s}$ and F_{mr} is equal to U_c in the equation (2). Also note that in the MR model, there are two input signals and one output signal. The input signals are $z(t)$ and $\dot{z}(t)$ displacement and velocity of the sprung mass and the output signal is the control force F_{mr} generated by the MR damper.

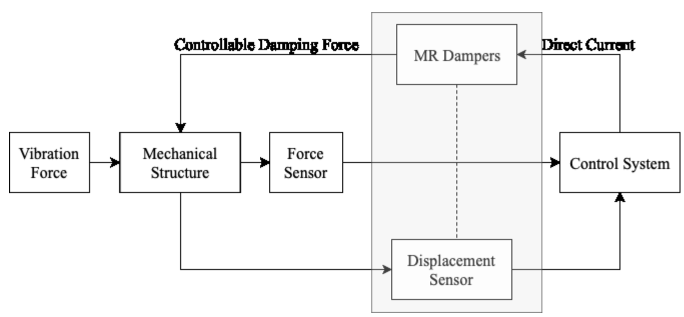

Figure 6. Schematic diagram of MR semi-active control systems structure.

V. SIMULATION & RESULTS

The mathematical formulation is implemented in Simulink models to compare the performances of each type of suspension system. In the simulations, the control force from Equation 11 U_c is set to be equal to F_{mr} and vibration damping is evaluated in the sprung mass. Displacement values of the sprung mass with MR damper and conventional suspension system are compared.

The parameters of the quarter car model, hydraulic actuator and Bouc-Wen model were obtained from Spencer et al. [3] and listed as follows: k_s= 80000 N/m, k_u = 500000 N/m, m_u=320 kg, m_s = 2500 kg, c_s = 320 Ns/m, c_u=15050 Ns/m, $\gamma = 1,\ \beta = 0,\ A = 1.5,\ n = 2$, k_0=300 N/m, $\nu = 5\,V, c_{0a} = 4400,\ c_{0b} = 442, \alpha_{0a} = 10872, \alpha_{0b} = 49616$, and f_0= 0 N.

For numerical simulations four different road excitations are considered: viz. step function, bump, sine wave, and random white noise. These excitations are shown in Figure 8.

From the numerical simulations it is clear that MR damper suspension system outperforms conventional suspension system for all the four excitation signals. Figures 9 through 12 show the system responses (displacement of the car body) of the passively and semi-actively controlled models for the four excitation signals.

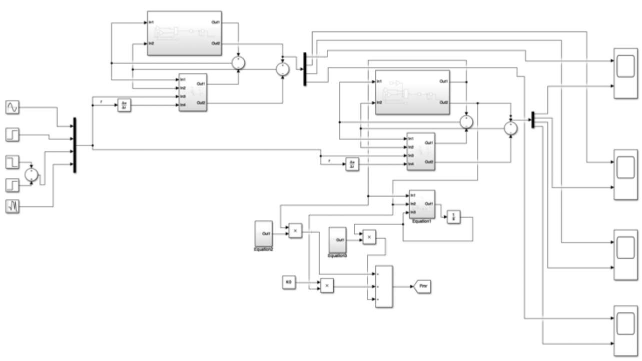

Figure 7. Simulink Model for Passive and MR suspension systems.

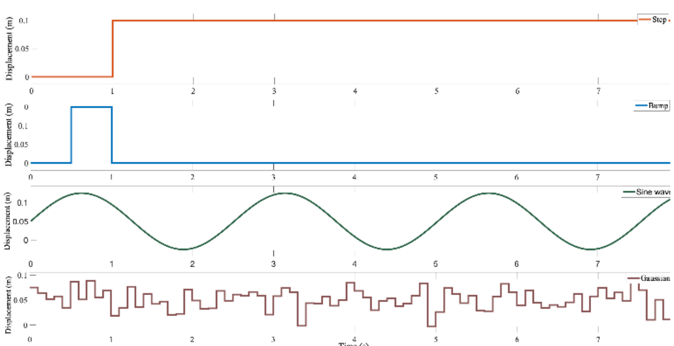

Figure 8. Road excitations: step, bump, sinusoidal and Gaussian road profile inputs.

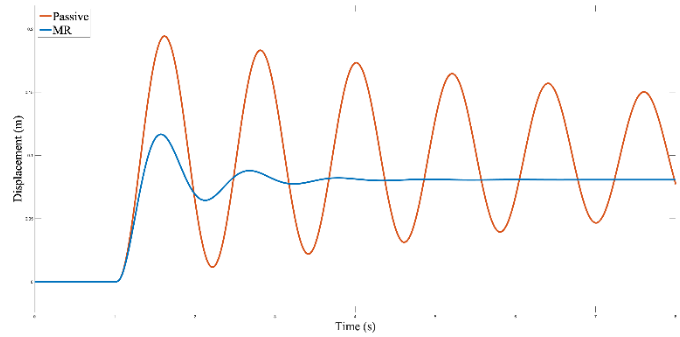

Figure 9. Model responses under step function road model.

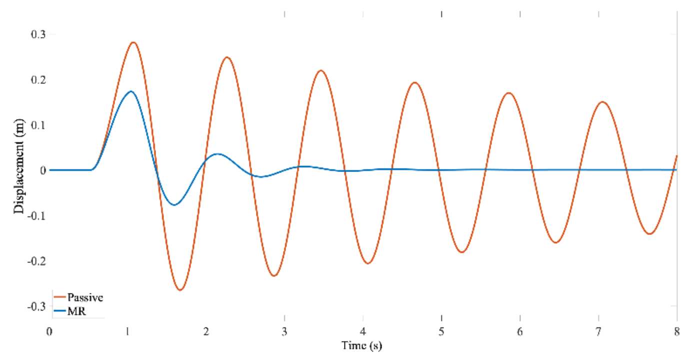

Figure 10. Model responses under bump road model.

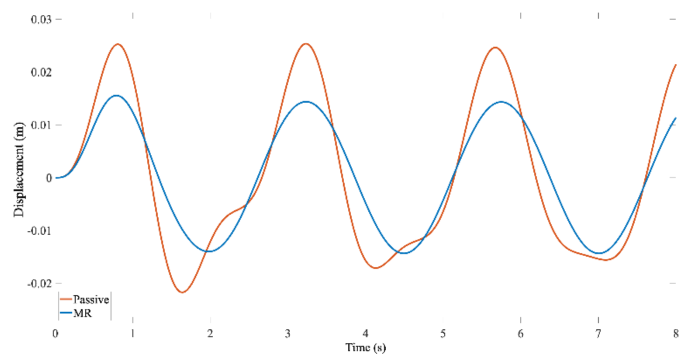

Figure 11. Model responses under sine wave function road model.

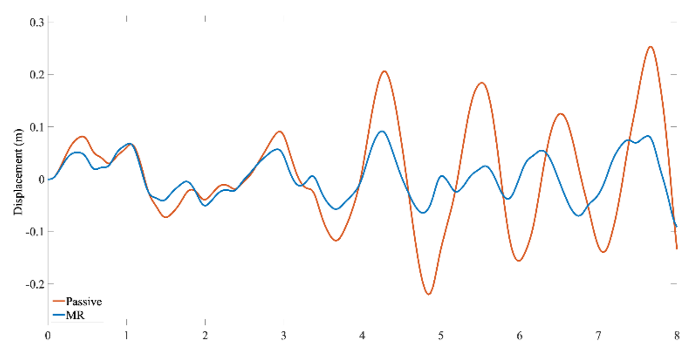

Figure 12. Model responses under random (Gaussian white noise) excitation from

VI. Conclusions

Based on the research and simulation results obtained using the Bouc-Wen model, MR dampers are found to be a more advantageous damping system for vehicle suspension and vibration absorption than traditional shock absorbers. The major benefits of an MR damper include: minimal energy consumption, continuous adjustability to varying terrain, enhanced passenger comfortability, and better handling abilities for the vehicle. That being said, there are certain challenges associated with MR damper system controls due to its non-linearity. Inaccuracies in derived parameters can be minimized by training neural networks with large experimental datasets. Other limitations include the high cost and finite life-span of MR fluids, thus further research is necessary to make them more cost-effective.

VII. Future Work

A. Modified Bouc-Wen Model

The Bouc-Wen model can be modified to make it more efficient, as proposed by Spencer [3,5]. An additional dashpot c_1 is added along with a spring k_1 to make the system more accurate. The model is governed by the following equation:

$$F = \alpha z + c_o(\dot{x} - \dot{y}) + k_o(x - y) + k_1(x - y) \quad (10)$$

In this modified model, the stiffness is represented by k_1, the viscous damping coefficient observed at large velocities is represented by c_o, and the dashpot is represented by c_1. This is introduced to cover the nonlinear force–velocity response of the Bouc–Wen model in roll-off in the region where the acceleration and velocity have opposite signs and the magnitudes of the velocities are small. In order to better asses the property of MR damper in vibration control application and make full use of the apparatus, a mechanical model was developed by Spencer et al. [3] .

By comparing the modified model to the simple Bouc-Wen model, it can be observed that adding the internal displacement factor was able to capture a better behavior of the damper along with cases when velocities with a small absolute value and there is an operational sign opposite to the acceleration. Although the modification does add more factors to consider, it provides a better control and feedback system for the controller.

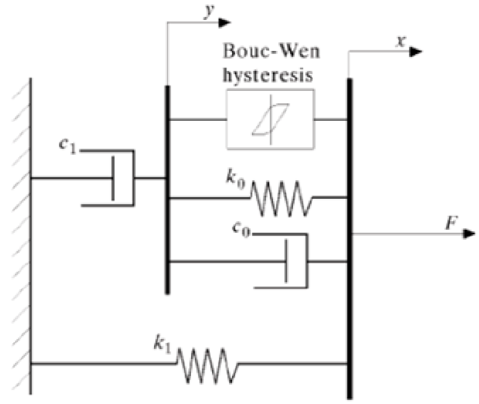

Figure 13. Schematic diagram of modified Bouc-Wen model for MR damper.

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