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TESTING TWO TYPES OF MAGNETO-RHEOLOGICAL (MR) DAMPER MODELS WITH QUARTER CAR SUSPENSION SYSTEM RESPONSE

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Abstract- Finding a sufficient and accurate mathematical model of Magneto-Rheological (MR) damper is not easy. Therefore, this paper will test the existing MR models to discover their behavior by involving MR models with a quarter car system under input circumference conditions such as impulse and step inputs. In this study, the comparison of simulation results by using the MATLAB/SIMULINK environment is conducted for passive and semi-active systems using both Bingham plastic and Bouc-Wen models. As a result, it was found that each model has strong and weak points, and one of them is found to be more suitable for the quarter-car suspension structure. The advanced Bouc-Wen model is used in conjunction with suspension control strategies. Comparing semi-active (non-controlled) damping along with the passive damping is achieved. The proposed controller, Skyhook with SFM damper control, is outfitted with a variable voltage for the MR damper according to the excitation inputs.

Keywords: Semi-Active Suspension, Simple Bingham Model (SBM), Modified Bouc-Wen Model (MBW), Signum Function Method (SFM), Modified Skyhook Control.

1. INTRODUCTION

The most pressing challenge in the automotive business today is obtaining the best vibration damping to ensure passenger safety, ride comfort, and road stability. Suspension systems are classified into three types based on their ability to control damping: passive [1], active [2], and semi-active [3]. The first one is considered one of the most widely used in suspension vehicles owing to their uncomplicated construction and low cost but constant damping properties, which is the main disadvantage of these systems.

As for the semi-active systems proposed by Karnopp in 1973, and since then, the semi-active suspension systems have gained popularity and wide demand in the applications of damping systems. These systems could control the required damping force relative to external disturbances. Furthermore, semi-active suspension systems rely on using Magneto-Rheological (MR) technology and other technologies instead of the oils used in passive suspension. MR fluid has grown in popularity and demand in numerous research investigations in the field of vibration reduction in recent years. MR fluids provide the capacity to respond quickly and simply through mechanical what is more electrical systems. Rabinow was the first to find liquids MR in the late 1940s.

The most notable attribute of MR fluids is their capability to switch from a free flow to a semi-solid form when an excited magnetic field is employed. Dynamic behavior of MR fluids (MRFs) is fluid and dramatic. These substances are mixtures of special oil and microscopic magnetic particles. As seen in Figure 1, when a current is applied, nanoparticles form straight chain up similar to the playing field; nevertheless, this process has restrictions Models for the design of MR fluids are essential in developing fluid MR devices. Phillips' work [4] started with the invention of nonlinear variants of Bingham's equations for fluid flow in parallel-walled channels, which were generally accepted because of the model's high level of accuracy. The Herschel-Bulkley model [5], the bi-viscous model [6], and the Bingham plastic model [7] are a few more models that have been used to define MR fluids (MRFs).

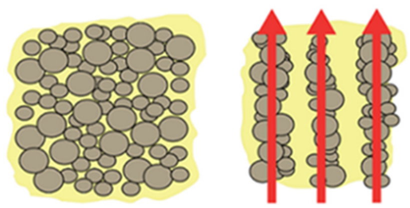

Figure 1. Behavior of MR with applying a magnetic

To use MR fluid technology in semi-active dampers, a mathematical model that explains the MR damper's (MRDs) nonlinear behaviors must be developed. Finding a suitable model that expresses a wide variety of hysterical MR dampers is one of the problematic issues. Using the Bingham fluid model as a foundation, Stanway et al. [9] suggested the mechanical model, also referred to as that of the Bingham incorporated the Dahl model's enhancement.

Other models for describing MR dampers exist. However, research is currently on to discover and formulate the best model for describing hysterical MR dampers. This work will be investigated the behavior of MRDs, as depicted in Figure 2, and compared it with the passive system. Therefore, they will be used through a simulation study using MATLAB/SIMULINK. plastic model.

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This work will be investigated the behavior of MRDs and compared them with the passive system. The most acceptable models are the Bingham and the Bouc-wen models. Therefore, they will be used through a simulation study using MATLAB/SIMULINK.

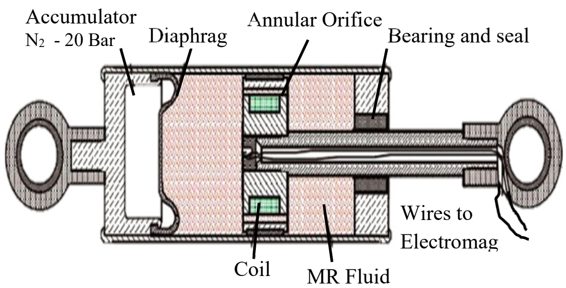


Figure 2. MR Damper [8]

2. FORMULATION OF A QUARTER-CAR MODEL

The suspension system must be formulated into a mathematical formula that describes the equation of motion to study vehicle system response.

In this paper, we will use a quarter-car model, as shown in Figure 3, that involves two degrees of freedom, which contains two masses, one representing the vehicle body Mb (sprung mass) and the other representing the mass of the tires Mw (un-sprung mass). On the other hand, it was assumed that the tires contained simple damping.

The Bingham model is considered simply formulated at constant excitation values; however, it cannot explain the behavior of semi-active dampers at speeds close to zero, which is the weak link in this model. The improved Bouc-Wen version captures a wide range of hysterical behavior of MR dampers, but it is considered a relatively solid model due to its complexity in its formulation. Until now, there is no accurate model for explaining the non-linear actions of the MRD. Studies are still in place to reach an ideal model for expressing the non-linear behaviors of the semi-active damper.

In our subsequent study, it will be searched more broadly between semi-active models and make a comparison between them. Also, it represents a passive damper in the experience of a malfunction in improved systems, as shown in Figure 3b. 1/4-car models can be expressed by Equations (1)-(4).

a) Passive suspension system:

$$M_b\,\ddot{Z}_b + C_b\left(\dot{Z}_b - \dot{Z}_w\right) + K_b\left(Z_b - Z_w\right) = 0 \tag{1}$$

$$M_w\ddot{Z}_w + C_b\left(\dot{Z}_w - \dot{Z}_b\right) + K_b\left(Z_w - Z_b\right) + \\ + C_w\left(\dot{Z}_w - \dot{Z}_r\right) + K_w\left(Z_w - Z_r\right) = 0 \tag{2}$$

b) Semi-active suspension system:

$$M_b\,\dot{Z}_b + K_b\left(Z_b - Z_w\right) - F_{mr} = 0 \tag{3}$$

$$M_w\ddot{Z}_w + C_w\left(\dot{Z}_w - \dot{Z}_r\right) + K_b\left(Z_w - Z_b\right) + \\ + K_w\left(Z_w - Z_r\right) + F_{mr} = 0 \tag{4}$$

where, C_b and K_b are damping coefficients and suspension stiffness; C_w and K_w damping coefficients and tire stiffness. The F_{mr} represents the force control produced by the MR actuators. On the other hand, Z_r represents the displacement resulting from the impact of the road, and from the analytical side, it represents the entry into the suspension system.

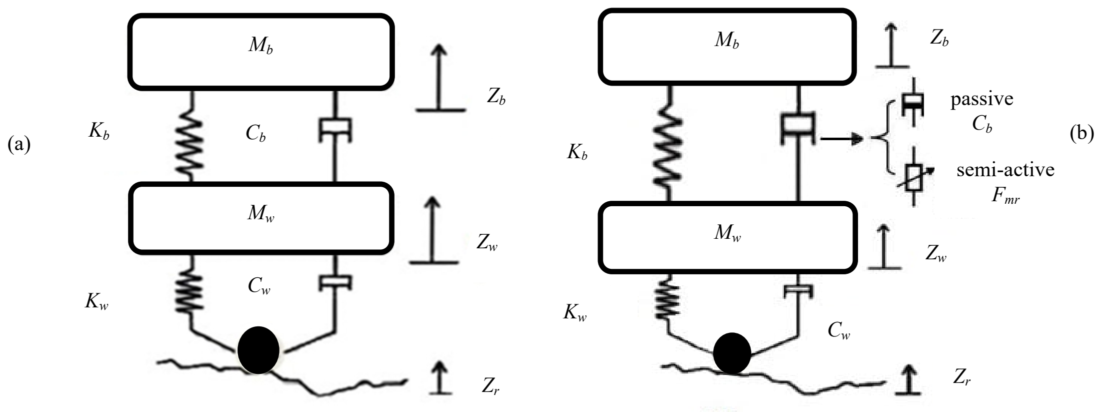


Figure 3. Suspension system of 1/4-car, (a) passive, (b) Semi-active [20]

3. MODELS FOR SIMULATION AND MATHEMATICAL FORMULATIONS

3.1. Simple Bingham Model (SBM)

Stanway, et al. [9] proposed the Bingham plastic model for describing Electrorheological dampers (ERDs). As display in Equation (5), the Bingham model has been formulated based on the Bingham fluid model. Therefore, MR can be described using the Bingham model and is one of the most basic and reliable mathematical models for describing MR damper hysteresis. As shown in Figure 4a mechanical model combines a viscous element parallel to the coulomb friction element [12].

$$\tau = \tau_y(H)\,\mathrm{sgn}(\dot{\gamma}) + \eta\gamma \quad (5)$$

where, τ_y denotes the yield stress caused by a magnetic field, and η indicates the fluid viscosity. The Bingham plastic model describes the damping force F_{mr} through the following equation:

$$F_{mr} = F_c\,\mathrm{sgn}(\dot{Z}) + C_0\dot{Z} + F_0 \quad (6)$$

where, $\dot{Z}$ is the derivative of the piston's velocity, and Z_r is the comparative movement of the piston. The damping constant is C_0, the offset force is F_0, and the frictional (control) force is F_c (constant force value). The signum function sgn(z) will take care of the vector of frictional force F_c according to the relative speed $\dot{Z}$ of the hysteresis (internal) variable Z.

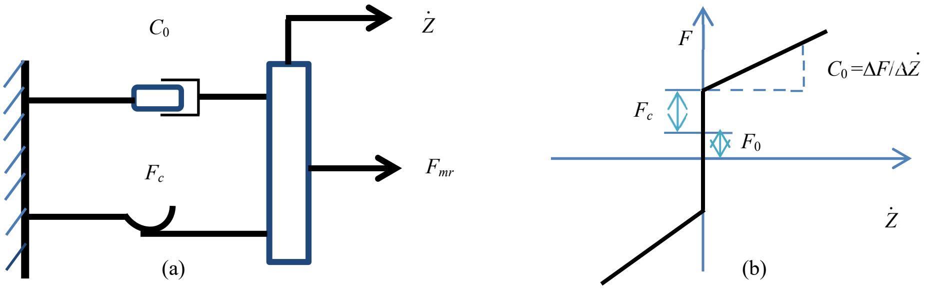


Figure 4. (a) Bingham model [13], (b) Response of Bingham model [13]

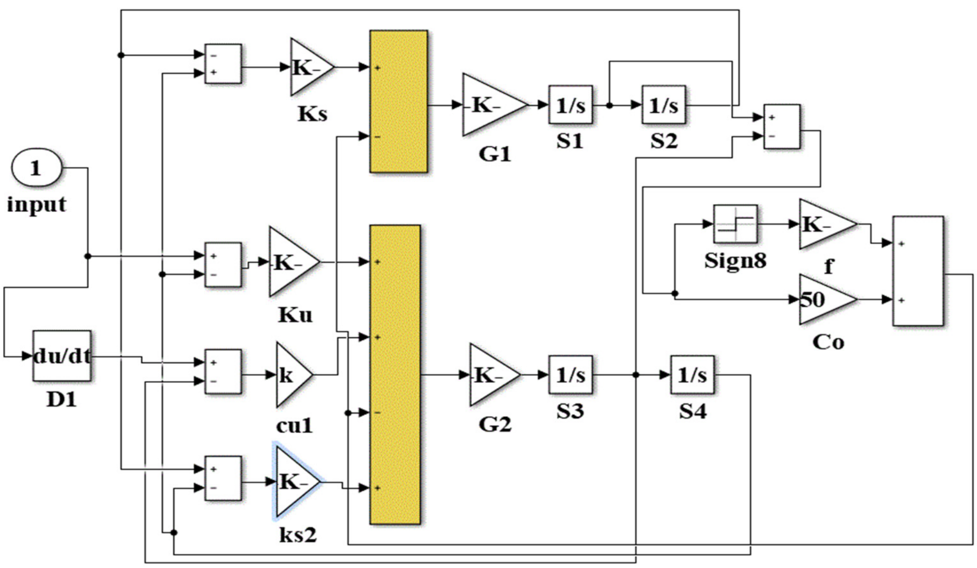


Figure 5. Bingham model in SIMULINK/MATLAB

In the Bingham model, the response between the damping force and the piston speed can be represented in Figure 4b, where we notice that the damping force is approximately equal to the coulomb. On the other hand, the Bingham model can be described as the Coulomb force elements paralleling the viscous damping elements, and the yield force is closely related to the Coulomb force. The damping force in the Bingham model expressed in Equation (6) represents the control force in a semi-active suspension model, as expressed in Equations (3), (4). Relying on semi-active suspension equations, it will be created a simulation model using Simulink, As is shown in next Figure 5, it was noted that the control force has been entered with a positive sign (+) on the spring-mass and with a negative sign (−) on the non-spring mass.

3.2. Modified Bouc-Wen Model (MBW)

Bouc, in the Bouc-wen model, is the first to characterize soft hysterics, and later on, Wen [15] generalizes an updated version of the Bouc model. This model describes a wide range of hysterical behavior of MR dampers on the stress-strain curve using an accurate and attractive mathematical formula. Figure 6 shows a schematic diagram illustration of the Modified Bouc-Wen version (MBW) by Spencer, et al. [10] which consists of representing the viscous damping, a stiffness element (spring), and Bouc-Wen hysteresis loop elements beside of reducing the accumulator stiffness. The hysterical performance that can alter depending on the coefficients is described in Equation (7). Moreover, the damping force is expressed by Equation (8), which considers the velocity-relative-displacement function [16].

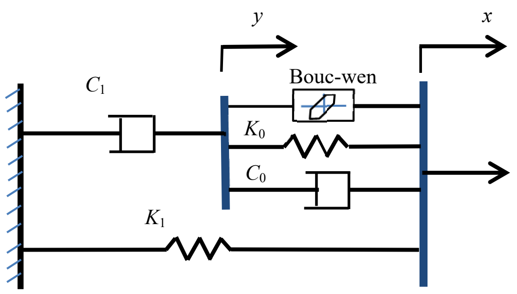


Figure 6. Modified Bouc-Wen model (MBW) [16]

$$\dot{z} = -\lambda\,|\,\dot{x}-\dot{y}\,|\,|\,z\,|^{(n-1)}\,z - \beta(\dot{x}-\dot{y})\,|\,z\,|^{n} + A(\dot{x}-\dot{y}) \quad (7)$$

$$F_{MR} = \alpha z + C_0\left(\dot{x}+\dot{y}\right) + K_0\left(x-y\right) + K_1\left(x-x_0\right) \quad (8)$$

where, y is MRDs internal motion and is determined by:

$$\dot{y} = \frac{1}{\left(C_0 + C_1\right)}\left[\alpha z + C_0\dot{x} + K_0\left(x-y\right)\right] \quad (9)$$

F_{MR} illustrates a linear connection to the control voltage u. The damper's pre-yield stress is taken into consideration by the force F. The Equations (7)-(9) were applied in SIMULINK/MATLAB to simulate the Bouc-Wen model after being inserted into the quarter-car system Figure 7. The parameters are viewed as being contingent on the affected current (I) that is defined by the voltage (v) given to the current driver. Considering the hypothesis of a model that functions for various magnetic field strengths, which depends on the linear connection between the factors and the supply voltage was established by Spencer, et al. [10].

$$\alpha = \alpha(u) = \alpha_a + \alpha_b\, u \quad (10)$$

$$C_1 = C_1(u) = C_{1a} + C_{1b}\, u \quad (11)$$

$$C_0 = C_0(u) = C_{0a} + C_{0b} u \quad (12)$$

where, C_{0a}, C_{1a} and α_a are the damping and Coulomb force parameters for MR damper at 0 V, respectively. u is the input relative to the supplied voltage v. The relationship between u and v is provided by the following equation:

$$\dot{u} = -\eta\left(u - v\right) \quad (13)$$

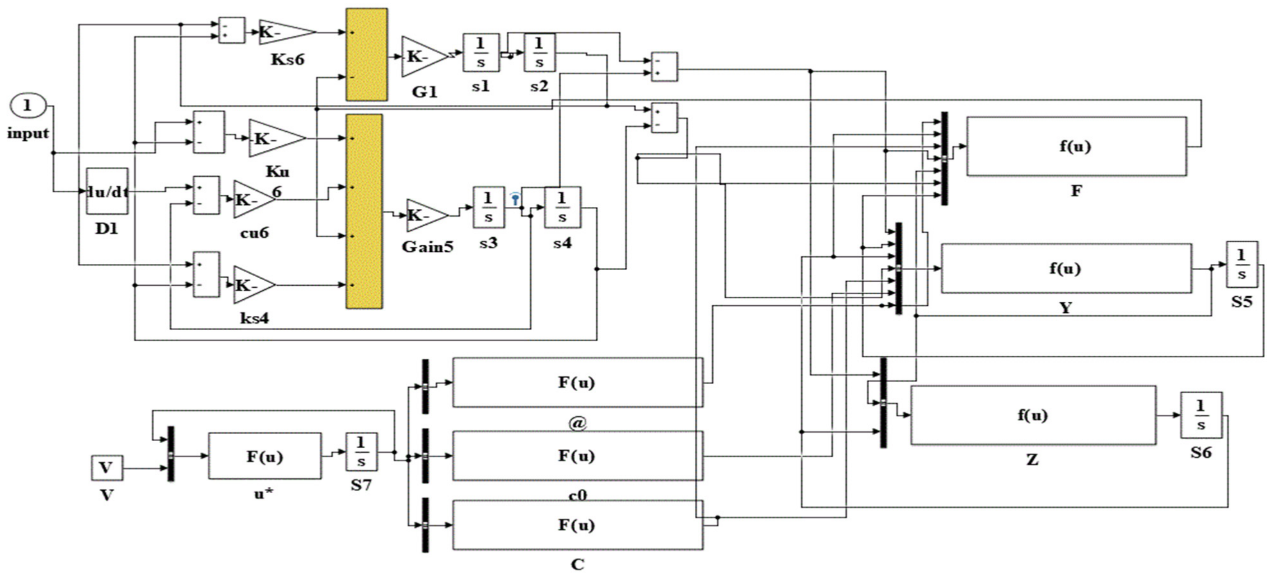


Figure 7. Modified Bouc-wen model in SIMULINK/MATLAB

4. SEMI-ACTIVE VEHICLE SUSPENSION CONTROL STRATEGIES

Two control units associated with the quarter-car model must be used when a semi-active suspension system of the car is engaged. [17]. The first control unit is the damper control, which provides the voltage supplied to the MR model. At the same time, the second control unit represents the System Controller, whose task is to give orders with the value of the desired force to dampen the excitement.

After reviewing the parametric models of the MR damper (Bingham and Bouc-Wen), it was discovered that only the Buck-Win model, which offers the ability to modify current-dependent parameters through Equations (10)-(13), can be used with control units.

Figure 8 shows a schematic diagram of the control system created on the MR damper. The system controller manipulating the needed force F_d in accordance with the system response, in order to produce a voltage appropriate for the MRD model.

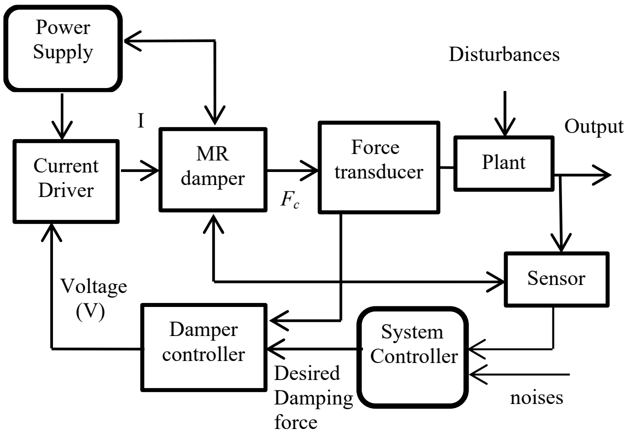


Figure 8. A semi-active control system [20]

4.1. MRD Controllers

The simplified Bingham model is unsuitable for magnetic resonance control because it neglects the input's nonlinearity and the force-velocity diagram's hysterical value, which depends on the variable P_{mr} value with the voltage value and is independent of the velocity value. Bouc-wen model is employed to apply control force to increase damping efficiency of the system response. The Signum Function Method (SFM) [18] will test the MR damper by projecting different and continuous voltages according to the dynamic excitation. The following equations represented Signum Function Method (SFM):

$$V_{sign1} = \frac{V_{max}}{2N} \left\{ \sum_{(0 \le j \le N-1)} \left\{ \text{sgn}\left\{ \left[F_c - \left(1+K_j\right) F \right] F \right\} + 1 \right\} \right\} \quad (14a)$$

$$V_{sign2} = 1 - \left[\frac{\text{sgn}\left(V_{sign1} - V_{max}\right) + 1}{2} \right] \cap \left[\frac{1 - \text{sgn}\left(F\dot{F}\right)}{2} \right] \quad (14b)$$

$$V_{sign} = V_{sign1} \times V_{sign2} \quad (15)$$

With respect to the magnetic field saturation in the system fluid damper, the high voltage to the control current has been V_{max}, and the needed and adjustable (measured) damping forces are F_c and F, respectively. The N is an integer with $0 \le j \le N-1$; sgn() is the signum function; M_w is the logical AND; K is constant. V_{sing} is the value of the voltage supplying the MR damper, which controls the damping value caused by the damper.

4.2. System Controller

To obtain the best possible environments for the system depicted in Figure 3b, there have been many system controller algorithms produced, and they can be generally categorized based on the control method employed to improve the system conditions:

- H∞ control [19]
- Linear-Quadratic-Gaussian (LQG) control [14, 20]
- Robust control [21]
- Skyhook control [22, 23]
- Sliding mode control [24]
- Neural Network (NN) system control [25]

In this paper, the Skyhook control system was selected due to the simplicity of its application and its acceptable results in improving the behavior of the MR damper. The modified of skyhook control strategy was described by Bessinger, et al. the identical approach was utilized in the investigation by Bakar, et al. [26]. The equation for the updated skyhook control method is given:

$$F_d = C_{sky} \left[\alpha \left(\dot{Z}_2 - \dot{Z}_1 \right) \right] + \left(1 - \alpha\right) \dot{Z}_2 \quad (16)$$

where, C_{sky} is the dampening factor for skyhook control and α is the passive to skyhook ratio. The value of is chosen to be 0.5, and an ideal value of C_{sky} is selected since it is determined that the required force derived from this controller would fall within the limit of damping forces of the proposed damper.

5. SIMULATION RESULTS AND DISCUSSIONS

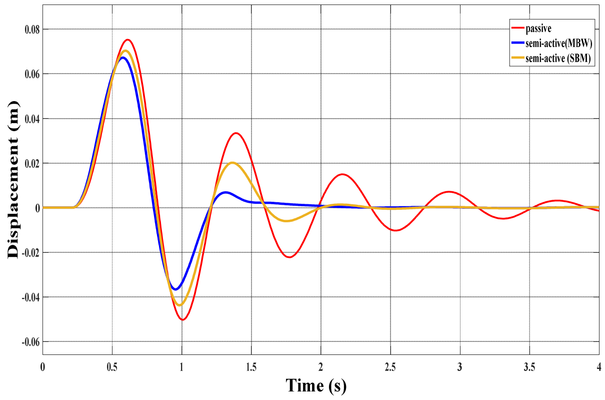
In this part, comparing of the analytical results of the semi-active and passive suspension were Its focus on the behavior and improvements of semi-active models (simple Bingham and Modified Bouc-Wen) after being introduced to the quarter-car model will be conducted. Through [1], the coefficient values for the 1/4-car are demonstrated in the Table 1.

Table 1. Quarter-Car coefficient values [1]

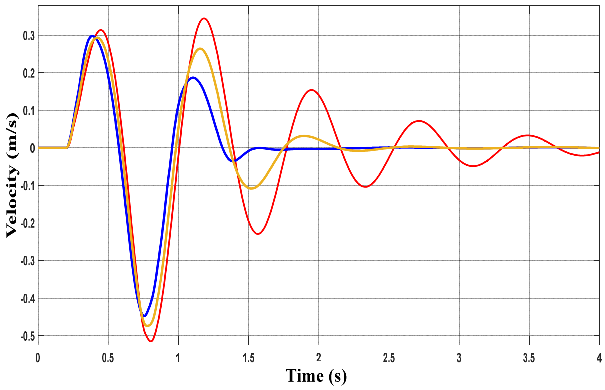
No.	coefficient name	coefficient notation	coefficient value
1	Sprung Mass	M_w	380 kg
2	Un-sprung Mass	M_w	31 kg
3	Stiffness of Suspension	K_b	29,000 N/m
4	Stiffness of Un-spring Mass (tire)	K_w	228,000 N/m
5	Coefficient of Damping of Sprung Mass	C_b	1500 N-s/m
6	Coefficient of Damping of Un-sprung Mass	C_w	110 N-s/m

It was found that the semi-active behavior significantly improved in comparison with the passive suspension system through employing two types of input: impulse and step inputs. Figure 9 shows the suspension systems considering the vehicle body was exposed to an impulse input that an amplitude of 0.05 m and a frequency of 6 rad/s.

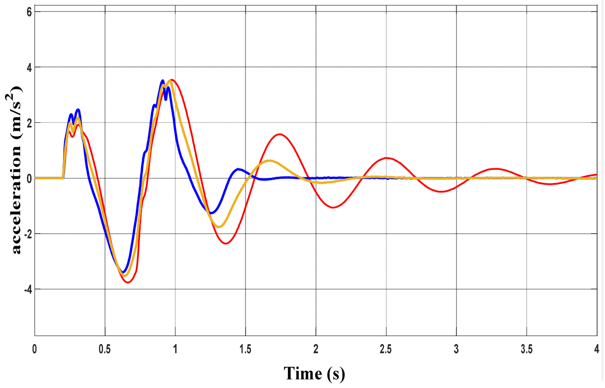
Figure 9 represents the response of both systems, including the Bingham model and the modified Buoc-wen (MBW) model. It is clearly seen that is an apparent decreased in the displacement in the semi-active system. The vehicle body reached stability after 4 seconds at the primary suspension, while the improved system reached the stability point within 1.5 to 2 seconds. However, the voltage that supplies the MR damper was set to a constant value (1.5 V). After using the semi-active system, the velocity and acceleration of the body mass were lowered, as shown in Figures 9b and 9c. It was discovered that the modified model provided the damper's behavior more widely and gave better damping than the Bingham model. The parameters values of both the simple Bingham model and the Modified Bouc-wen model are shown in Table 2 and Table 3. From the experimental results in [10], these values consider the damping behavior of MR at a constant voltage of 1.5.



(a)



(b)



(c)

Figure 9. Models of suspension with constant voltage (1.5 V) and an impulse input that are passive vs semi-active, (a) body displacement-time, (b) body velocity-time, (c) body acceleration-time

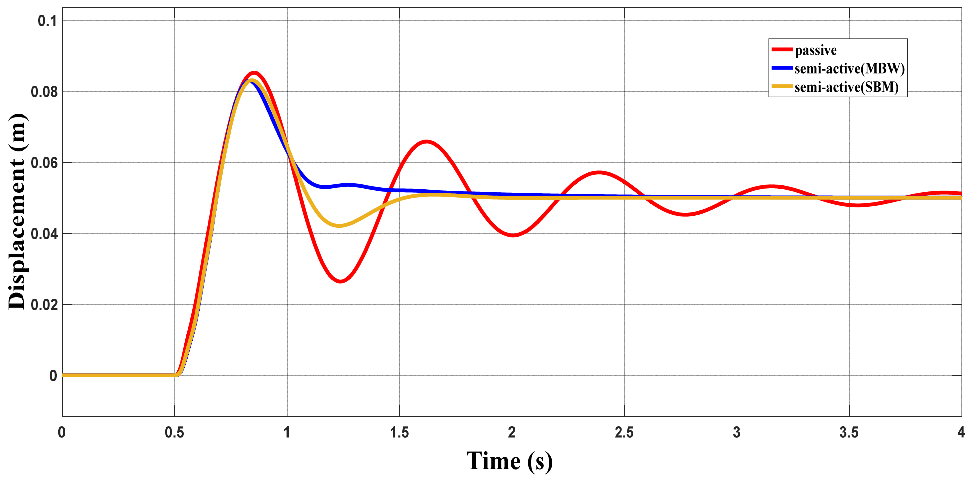
Table 2. The parameters for the Modified Bouc-wen model [10]

	α	C_0	K_0	C_1	K_1	β	γ	A
MBWM	963 N/cm	53 N.s/cm	14 N/cm	930 N.s/cm	5.4 N/cm	200 cm-2	200 cm-2	207

Table 3. The parameters for the simple Bingham model [10]

	F_c	C_0	F_0
SBM	670 N	50 N.s/cm	0

(a)



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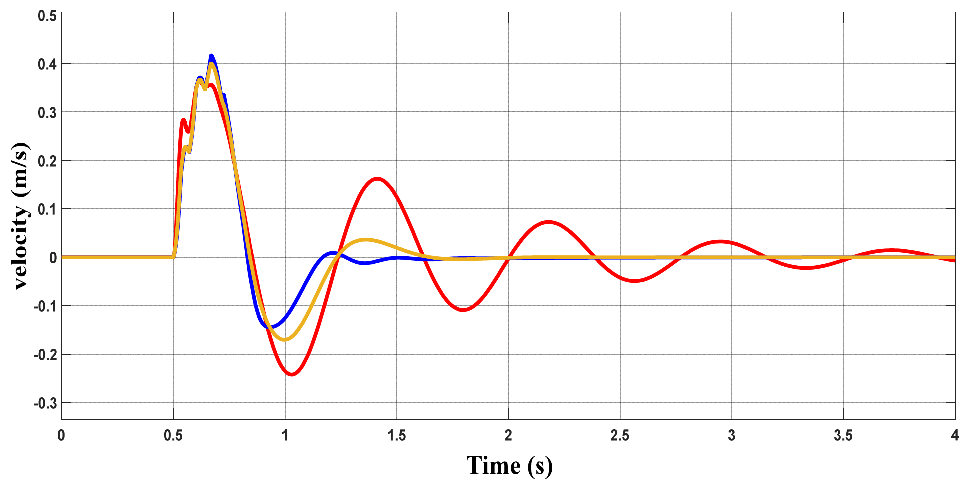

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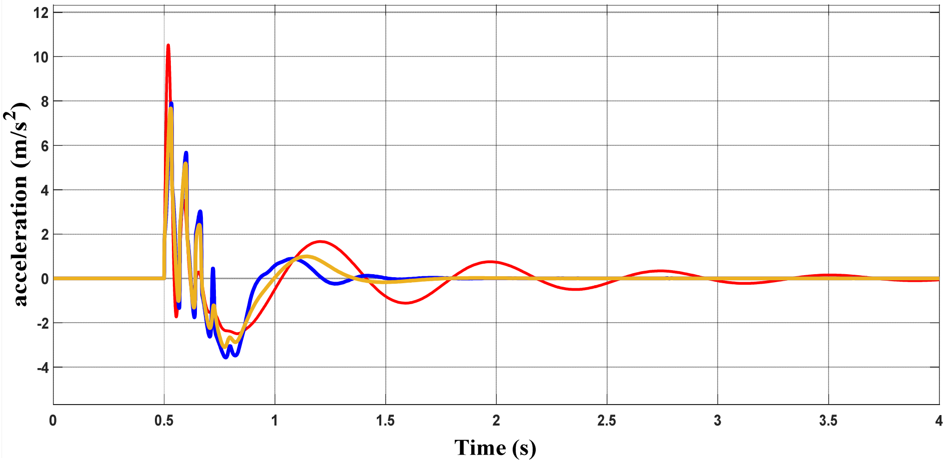

Figure 10. Models of suspension with constant voltage (1.5 V) and a step input that are passive vs semi-active, (a) body displacement-time, (b) body velocity-time, (c) body acceleration-time

(a)

(b)

(c)

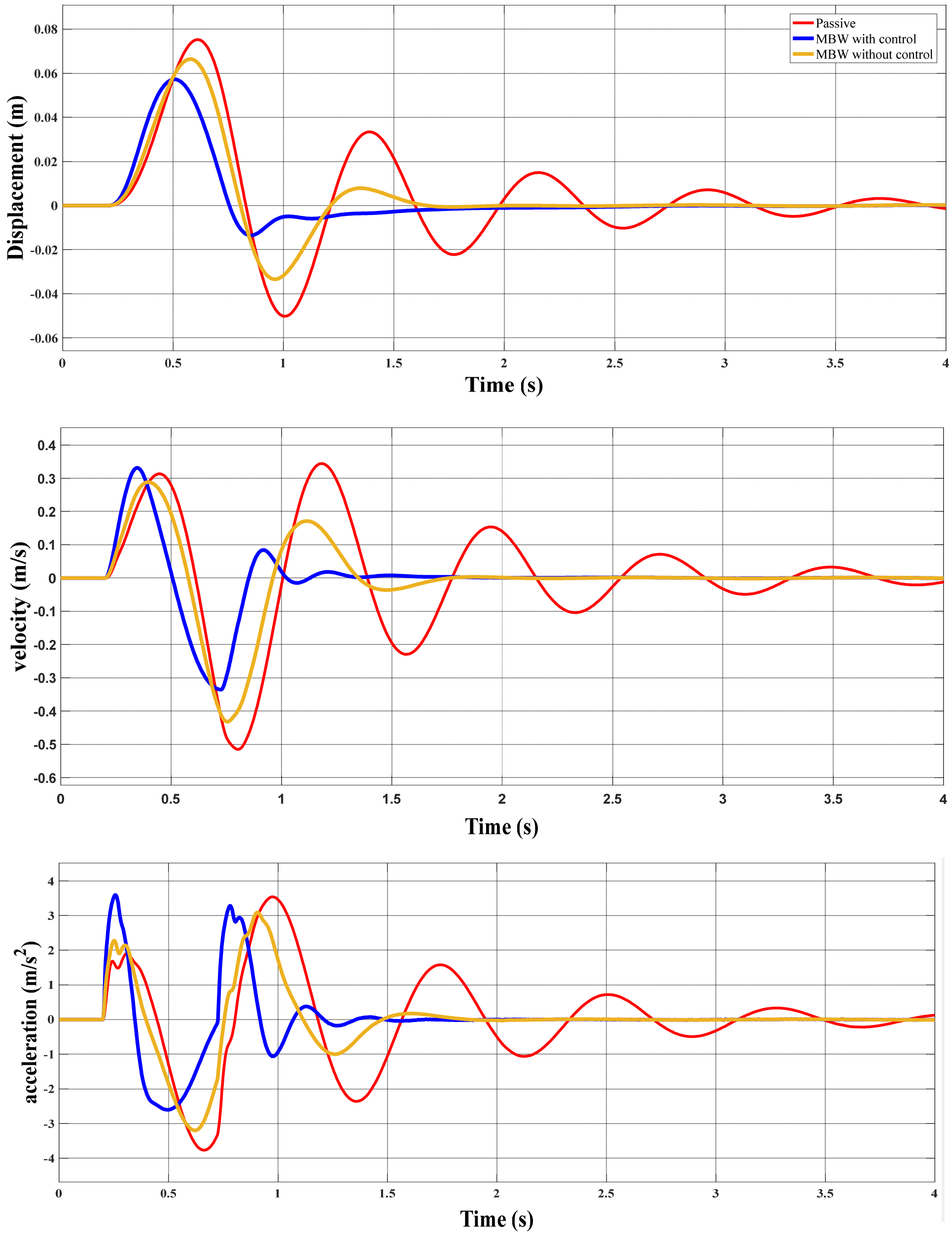

Figure 12. Passive vs. Semi-active controller and uncontrolled suspension (modified Bouc-Wen model) on step input excitation with variable voltage, (a) body displacement-time, (b) body velocity-time, (c) body acceleration-time

Figure 10 shows a noticeable superiority when using semi-active models in the quarter car system, which reduces road vibrations, were tested by step input excitation with a final amplitude of 0.05 (m). Despite applying a constant voltage to power the MR damper, the results show a significant improvement in the dampening of the acceleration value of the vehicle body when using the semi-active suspension system. Additionally, a stability period of 1.5 seconds from the beginning of the excitation of the vehicle body mass was established, with findings comparable to semi-active models.

After applying the control damper described by each of the Equations (14a), (14b), (15) to the semi-active suspension models, which led to the introduction of a variable voltage to the MR damper as needed, Figure 11 shows the value of the voltage required when using an impulse as an excitation of the road.

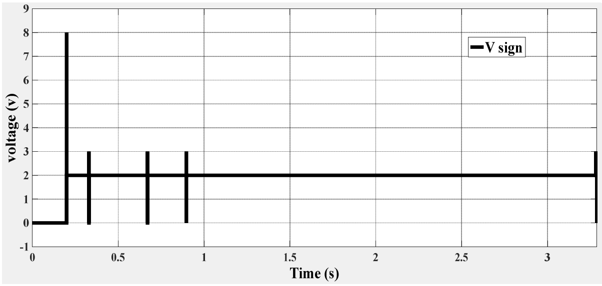

Figure 11. The supplied voltage is from the damper controller

The MR dampers were tested using the Bouc-wen model, which allows controlling the voltage values equipped with the damper, as the control strategies of the semi-active suspension mentioned earlier were used. Figure 12 compares both suspension systems using a constant voltage and the semi-active suspension using the damper and system control. The efficacy of damping is increased after employing the control strategies represented by Equations (15, 16), which gives satisfactory results in reducing the stability time and damping the overshoot value.

Figure 13 shows the nonlinear hysterical behavior of each semi-active model (MR dampers). The width of the hysteretic loop changes in the force-velocity diagram when the value of the current supplied to the MR dampers changes. However, hysterical testing of MR dampers was done using a sine wave where a frequency of 6 rad/s and an amplitude of 0.5 represents the road excitation with a constant value of 1.5 V. The force-velocity diagram of the Bingham plastic model is shown in Figure 13a. The results show that the model cannot accurately represent an MR damper's behavior at speeds near zero but provides excellent visualization of the damping force.

Figure 13c displays the force-velocity curve of the modified Bouc-Wen model. Also, each of Figure 13b and 12d represents the force diagram with the displacement of the damper model MR. As shown, there is a clear difference in the smoothness of the displacement-force curve between the Bingham and Bouc-wen model. The flowchart perfectly describes the nonlinear hysterical behavior of the MR damper and is a good representation of the damper's behavior.

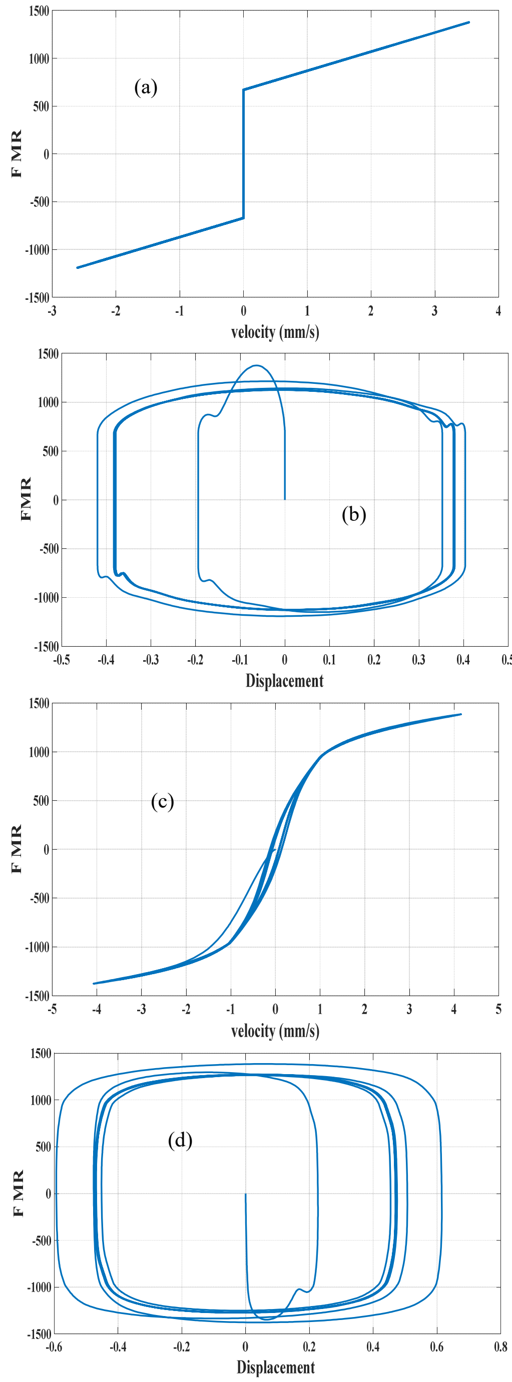

Figure 13. F_{MR}-Velocity and F_{MR}-Displacement in Semi-active suspension models with sine input excitation, (a, b) Simple Bingham Model, (c, d) Modified Bouc-Wen model

6. CONCLUSIONS

The MR fluid dampers have drawn much interest in the last ten years as a semi-active control device because they can rapidly change the damping qualities safely and fail-safely while using little power. However, understanding an MR damper's nonlinear hysteretic behavior under an applied magnetic field is essential for successful control. As a result, precise control methods that fully exploit the unique properties of MR dampers must be created using mathematical models that accurately capture the fundamental nonlinear behavior of these components.

In this paper, a comparison was made between the passive and semi-active suspension system employing MR damper, using both the Bingham and the modified Bouc-Wen models. Moreover, by imposing the road excitation as an impulse wave and a step input through simulation MATLAB /Simulink environment. The simulation results show that the semi-active models improve performance to absorb the vibrations generated by the road disturbances with high reliability.

On the other hand, the simulation results were compared between the Bingham model and the modified Bouc-Wen model. Although, it was noted that the Bouc-Wen model considers a wide range of damper behaviors and is considered one of the most accurate models since it is considered a complex model. Also, semi-active control strategies were used with the Bouc-wen model, and the results clearly showed an improvement in the damping of the excitement after using a variable voltage on the MR damper.

While the Bingham model is considered the simplest, formulated well in constant excitation values. However, the Bingham model cannot explain the behavior of semi-active dampers at speeds close to zero, which is the weak link in this model. The modified Bouc-Wen model captures a wide range of hysterical behavior of MR dampers, but it is considered a relatively solid model due to its complexity in its formulation. Until now, there is no accurate model for explaining the non-linear behavior of the MR damper. Studies are still in place to reach an ideal model for expressing the non-linear behaviors of the semi-active damper. In our subsequent study, we will search more broadly between semi-active models and make a comparison between them.

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