

Application of Nonparametric Magnetorheological Damper Model in Vehicle Semi-active Suspension System

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ABSTRACT

Nonparametric models do not require any assumptions on the underlying input/output relationship of the system being modeled so that they are highly useful for studying and modeling the nonlinear behaviour of Magnetorheological (MR) fluid dampers. However, the application of these models in semi-active suspension is very rare and most theoretical works available on this topic address the application of parametric models (e.g. Modified Bouc-Wen model).

In this paper, a nonparametric MR damper model based on the Restoring Force Surface technique is applied in vehicle semi-active suspension system. It consists of a three dimensional interpolation using Chebyshev orthogonal polynomial functions to simulate the MR damper force as a function of the displacement, velocity and input voltage. Also, a damper controller based on a Signum function method is proposed, for the first time, for use in conjunction with the system controller of a semi-active vehicle suspension. A mathematical model of a semi-active quarter-vehicle suspension using an MR damper is derived. Suspension performance criteria are evaluated in the time and frequency domains in order to quantify the suspension effectiveness under bump and random road disturbance.

The simulated results of the present study show that the applied nonparametric MR damper model is able to express the behavior of the damper precisely and the force tracking controller has the capability to track the desired damping force well. Compared with the passive suspension system, the proposed semi-active control strategy improves the suspension performance effectively.

CITATION: Metered, H., "Application of Nonparametric Magnetorheological Damper Model in Vehicle Semi-active Suspension System," *SAE Int. J. Passeng. Cars - Mech. Syst.* 5(1):2012, doi:10.4271/2012-01-0977.

INTRODUCTION

Nonparametric models, however, are more 'flexible' than the parametric models, by identifying the system's response with no assumption related to the system's physical characteristics [1]. The development and application of nonparametric MR damper models have been a great interest of many researchers because they do not involve any *priori* knowledge or memory of the systems characteristics [1, 2]. They are greatly valuable for predicting the nonlinear behaviour of MR dampers. However, the application of these models in vehicle suspension is still very rare and the majority of theoretical works available on this topic address the application of parametric models (e.g. Modified Bouc-Wen model [3]). Parametric models are based on mechanical idealization involving representation by an arrangement of springs and viscous dashpots [4]. The parameters of these elements are determined for a given damper through curve fitting of experimental results.

Nonparametric models can be used for modelling both the direct and the inverse dynamics of an MR damper. Proposed

nonparametric models can be classified as follows: (i) Neural-based methods; neural networks [4], this approach needs a memory into the model or a *priori* knowledge of the MR damper behaviour, which was applied in a vehicle semi-active suspension system to emulate the forward and inverse dynamics of MR damper [5], and adaptive neuro-fuzzy inference system (ANFIS) [6]. (ii) Interpolation techniques; simple polynomial model [7] which also was applied in a vehicle semi-active suspension system to emulate the forward and inverse dynamics of MR damper [8], nonparametric linearised data modelling [9] and Restoring Force Surface (RFS) techniques (e.g. Chebyshev polynomial interpolation) [2]. The advantage of the latter polynomial interpolation techniques is that they offer an explicit functional representation of the output variable in terms of the instantaneous values of the inputs. This is in contrast to the modified Bouc-Wen models and neural networks, which require time histories of the input variables to produce a prediction of the damper force. However, this means that these polynomial techniques are memory-less. This model is

applied in this paper to predict the MR damper force. It involves a three dimensional interpolation using Chebyshev orthogonal polynomial functions to produce the damping force as a function of the displacement, velocity and input voltage. The identification and validation of this model are done using data generated by solving the modified Bouc-Wen model.

The design of a better quality suspension system remains an important development objective for the automotive industry. An ideal vehicle suspension system should have the capability to reduce the displacement and acceleration of the vehicle body, maximizing ride comfort. It should also aim to minimize the dynamic deflection of the tyre to maintain tyre-terrain contact. Ride comfort and vehicle stability are two conflicting requirements which the suspension's vibration control strategy has to satisfy [10].

MR fluid dampers are becoming popular in semi-active vehicle suspension applications due to their mechanical simplicity, high dynamic range, low power requirements, large force capacity and robustness. Moreover, most vehicles have the facility to provide the voltage (or current) that is required to generate a controllable variable damping force. Semi-active suspension systems incorporating MR dampers combine the advantages of both active and passive suspensions. They can be nearly as efficient as a fully active suspension in improving ride comfort and is much more economical [11, 12]. They are also safer since if the control system fails, the semi-active suspension can still work as a passive suspension system. They have been shown to offer a compromise solution for ride comfort and vehicle stability requirements [13, 14, 15]. With passive damping these requirements are conflicting, since a "hard" damper results in better stability but reduced ride comfort, whereas a "soft" damper results in the converse effect [10].

An MR damper is a semi-active control device since it is only the voltage applied to its electromagnet that can be commanded directly. Hence, in addition to a system controller that calculates the desired damper force required for the system to achieve a good behaviour, a damper controller is also required. This latter controller determines the voltage v to be applied to the damper such that its actual force f_a tracks the desired force f_d . Different control strategies have been proposed and evaluated for the system controller: e.g. skyhook control [13, 16 and 17], H_∞ control [18, 19, 20], adaptive control based on neural network [21], robust control [22], Fuzzy Logic control [23], sliding mode control [5, 24, 25] and Linear Quadratic Gaussian control (LQG) [26]. The latter approach is used in this paper for the system controller.

With regard to the damper controller, its effectiveness depends on its ability to deal with the nonlinear and hysteretic nature of the device. The most basic MR damper controller algorithm is "on-off" control, also known as the Heaviside Step Function (HSF) method, where the applied voltage is either 0 or maximum [27]. It was first introduced in [27] to

research the MR damper control of structural responses due to seismic loads. It was subsequently used in [26] to research the MR damper control of an automobile suspension system. An improvement on this algorithm is the Signum Function method (SFM), which, under certain conditions, allows the applied damper voltage to switch between discrete voltage levels below the maximum [28]. The Continuous State control (CSC) allows for intervals of continuous voltage between periods of discontinuous pulses [29]. CSC was used in [24, 25] for an MR damper controller in vehicle suspension systems. An alternative method of commanding a voltage signal is through a damper controller that is a neural network model of the inverse dynamics of the MR damper. An inverse recurrent neural network damper controller was proposed firstly in [4] to predict the command voltage to the magnetic coil of the MR damper and then it was applied in [5] to research the MR damper control of an automobile suspension system.

The novel contributions of this paper are the application of nonparametric MR damper model based on Chebyshev polynomials and the implementation of an MR damper controller using Signum function approach, for the first time, in the investigation of MR damper control of a vehicle suspension system. Control performance criteria are evaluated in the time and frequency domains in order to quantify the suspension effectiveness for the two alternative system controllers (LQR and LQG), as well as MR passive damping (zero applied voltage) and conventional passive damping.

The rest of this article is organized as follows: an overview of the quarter vehicle model is introduced in the next section contained brief details of the MR damper model. This is then followed by a description of the proposed semi-active control algorithm. Finally, the results obtained for different road disturbance inputs are presented and discussed.

QUARTER VEHICLE MODEL

Figure 1 illustrates the two-degree-of-freedom (2DOF) system that represents the quarter vehicle suspension model. It consists of an upper mass, m_b , representing the body mass, as well as a lower mass, m_w , representing the wheel mass and its associated parts. The vertical motion of the system is described by the displacements x_b and x_w while the excitation due to road disturbance is x_r . The suspension spring constant is k_s and the tyre spring constant is k_t (tyre damping neglected). The data employed here for the quarter vehicle system is the same as that in [30] and listed in Table 1. By applying Newton's second law to the quarter vehicle model, the equations of motion of m_b and m_w are:

$$\begin{aligned} m_b \ddot{x}_b + k_s (x_b - x_w) + f &= 0 \\ m_w \ddot{x}_w - k_s (x_b - x_w) + k_t (x_w - x_r) - f &= 0 \end{aligned} \quad (1a,b)$$

where, the force from the damping device is given by

$$f = \begin{cases} c_s(\dot{x}_b - \dot{x}_w), & \text{for passive suspension} \\ f_a, & \text{for semi-active suspension} \end{cases} \quad (2a,b)$$

where, c_s being the coefficient for the passive suspension's damper.

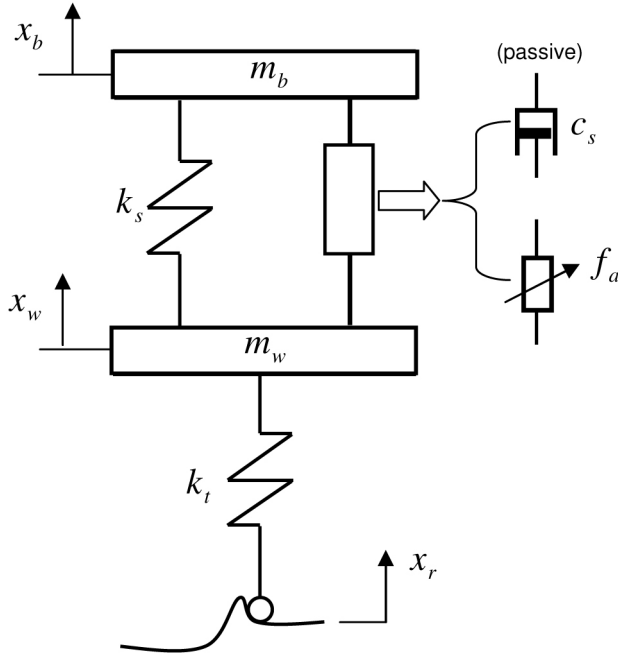


Figure 1. Quarter-vehicle suspension model

The proposed semi-active suspension system can be represented in the state space form as follows:

$$\dot{x} = Ax + Bf_a + Dx_r$$

Where, $x = [x_b \ x_w \ \dot{x}_b \ \dot{x}_w]^T$,

$$A = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ -\frac{k_s}{m_s} & \frac{k_s}{m_s} & 0 & 0 \\ \frac{k_s}{m_w} & -\frac{k_s + k_t}{m_w} & 0 & 0 \end{bmatrix}, \quad B = \begin{bmatrix} 0 \\ 0 \\ \frac{1}{m_s} \\ \frac{1}{m_w} \end{bmatrix},$$

$$D = \begin{bmatrix} 0 \\ 0 \\ 0 \\ \frac{k_t}{m_w} \end{bmatrix}$$

and

For the semi-active suspension, the MR damper force f_a depends on the time histories of the applied voltage to the magnetic coil v , the relative displacement x and relative velocity $\dot{x}$ across it:

$$\text{where } x_s = x_b - x_w \text{ and } \dot{x}_s = \dot{x}_b - \dot{x}_w \quad (3a,b)$$

x_s is also referred to as the suspension working space (SWS). The voltage signal v is calculated by the control system, which is described in the next section. For given signals v , x_s and $\dot{x}_s$, the force f_a is then determined by the Chebyshev polynomial methods in this research, as discussed in the Introduction. This method is presented in the next sub-section.

Table 1. Quarter vehicle suspension parameters [30]

Parameter	Symbol	Value (Unit)
Body Mass	m_b	240 (kg)
Wheel Mass	m_w	36 (kg)
Suspension stiffness	k_s	16 (kN/m)
Damping coefficient	c_s	980 (Ns/m)
Tyre stiffness	k_t	160 (kN/m)

OVERVIEW OF THE MR DAMPER MODEL

This sub-section gives a brief overview of the MR damper model, which was introduced recently by Metered et al. [2]. Using a three dimensional series expansion of the Chebyshev polynomials, the damping force was approximated as triple series involving variables x_s , $\dot{x}_s$ and v .

$$f_a(x_s, \dot{x}_s, v) \approx \hat{F}(x_s, \dot{x}_s, v) = \sum_{k,l,z=0}^{K,L,Z} C_{klz} T_k(\tilde{x}_s) T_l(\tilde{\dot{x}}_s) T_z(\tilde{v}) \quad (4)$$

where $F(x_s, \dot{x}_s, v)$ is the restoring force of the MR damper, the C_{klz} is the normalized Chebyshev coefficient, T_k , T_l , T_z constitute the polynomial basis and K , L , and Z are the highest polynomials' truncation orders of the Chebyshev polynomial of the normalized displacement ($\tilde{x}_s$), velocity ($\tilde{\dot{x}}_s$) and voltage ($\tilde{v}$) within the range of $[-1, 1]$ for x_s , $\dot{x}_s$ and v . The C_{klz} can be identified as:

$$C_{k,l,z} = \frac{8}{(1 + \delta_{k0})(1 + \delta_{l0})(1 + \delta_{z0}) Q_\eta Q_\omega Q_\psi} * \sum_{i=1}^{Q_\eta} \sum_{j=1}^{Q_\omega} \sum_{s=1}^{Q_\psi} \hat{F}(\eta_i, \omega_j, \psi_s) \cos\left(\frac{2i-1}{2Q_\eta} k\pi\right) * \cos\left(\frac{2j-1}{2Q_\omega} l\pi\right) * \cos\left(\frac{2s-1}{2Q_\psi} z\pi\right) \quad (5)$$

where δ_{ik} is the Kronecker delta and Q_η Number of quadrature points. The surfaces were constructed using data

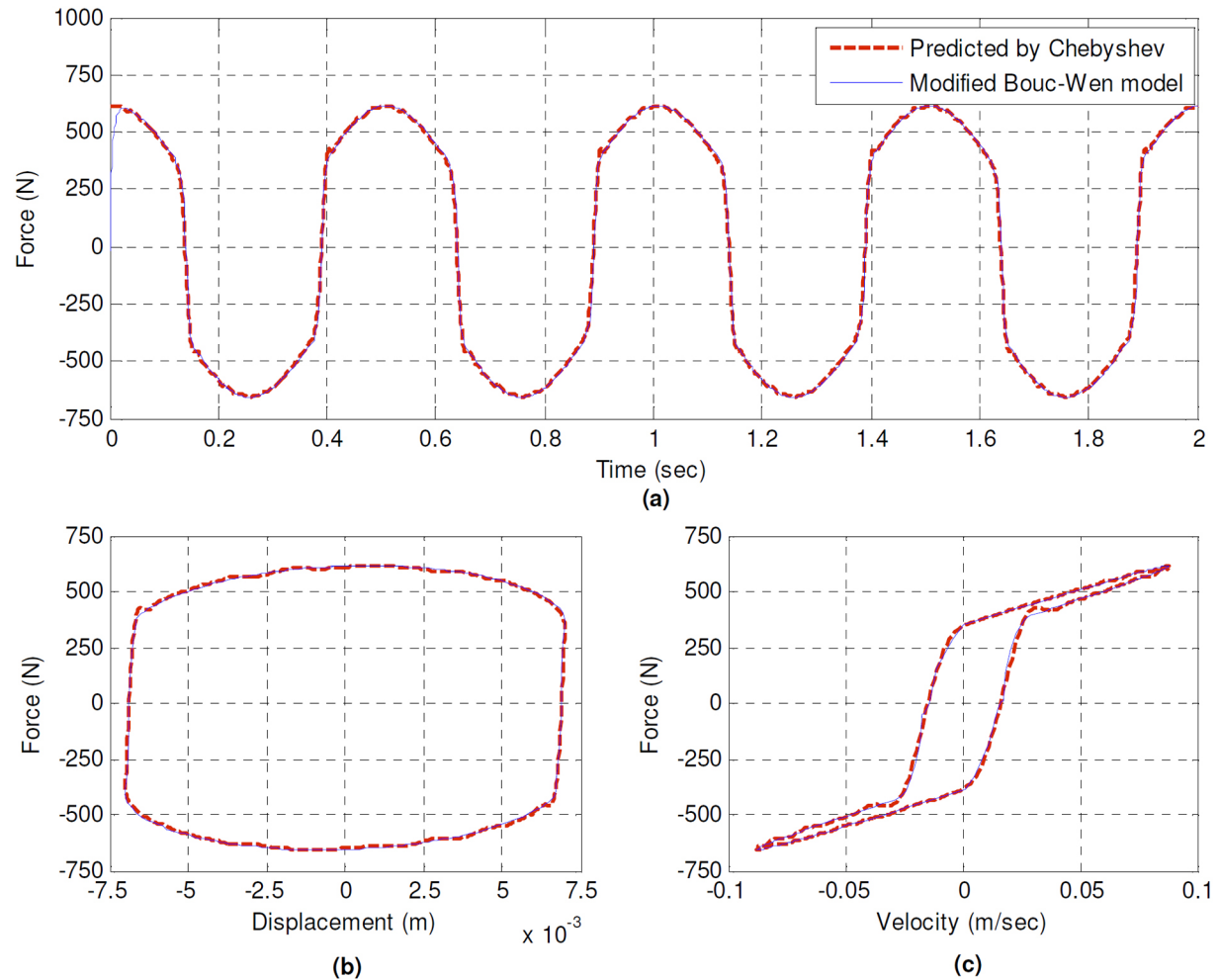


Figure 2. Validation of interpolation procedure using Chebyshev polynomials against modified Bouc-Wen [1]

generated by solving the modified Bouc-Wen model [3] after that the determination of the coefficients C_{klz} from the above expression requires the values of the damper force at the grid points i.e. $F^*(\eta_i, \omega_j, \psi_s)$. Once the C_{klz} is determined, the predicted damping force of the MR damper can be predicted using the following relationships [31]: or

$$T_0(x_s) = 1, T_1(x_s) = x_s, T_2(x_s) = 2x_s^2 - 1$$

or

$$T_{n+1}(x_s) = 2x_s T_n(x_s) - T_{n-1}(x_s) \quad (6)$$

Figure 2 demonstrates the validation of Chebyshev model against the modified Bouc-Wen, thus indicating that the applied model is able to predict the hysteresis force of the MR fluid damper accurately and efficiently. Moreover, the complete procedure, identification and validation of this model can be found in the text of Ref. [2].

SEMI-ACTIVE CONTROL STRATEGY

The complete semi-active vehicle suspension system using MR damper is illustrated in Figure 3. As stated in the Introduction, it consists of two nested controllers; a system controller and a damper controller. The system controller uses the dynamic responses of the plant to compute the desired damping force f_d according to some chosen algorithm. The damper controller adjusts the voltage v applied to the damper in order to track its actual force f_a to the desired force f_d . The following sub-sections provide a brief description of the damper controller and the system controller considered in this study.