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FUZZY HYBRID CONTROL OF OFF-ROAD VEHICLE SUSPENSION FITTED WITH MAGNETORHEOLOGICAL DAMPERS

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KEYWORDS

Fuzzy control, magnetorheological damper

ABSTRACT

The vibration caused by severe road excitation influences off-road vehicle suspension performance. The vibration control of the suspension system is a crucial factor for modern vehicles. Smart control devices (magnetorheological dampers) are proposed as a first step to handle a multiple suspension system of off-road vehicles. The magnetorheological (MR) dampers can be employed as smart dampers for vibration suppression of the suspension system; this is done by varying the produced damping force. In this paper an investigation is presented on the effectiveness of such smart dampers in attenuating the vibration of a multiple suspension system. This goal is accomplished by designing a new fuzzy hybrid controller and studying its effectiveness on the suspension performance. The multiple suspension system considered here comprises a chassis, five wheels and three MR dampers. The chassis is suspended over the five wheels through five compression springs. Three MR dampers are attached to the first, second and the fifth wheel. The stiffness of the wheels is represented by five compression springs. Only the bounce and the pitch of the chassis are considered. The assessment of the proposed model is carried out through a simulation scheme under bump and sinusoidal excitations in the time domain. The excitation of the five wheels is done independently. The simulation is accomplished using the MATLAB/SIMULINK software. The simulation results show the effectiveness and robustness of the new controller in conjunction with MR dampers in vibration suppression. Compared to the passive suspension, the body bounce, body displacement, angular acceleration and pitch angle can be well controlled

NOMENCLATURE

$\ddot{z}_{wi}$	Wheel acceleration
$\ddot{z}_b$	Body acceleration
G_{gi}	Groundhook gain
G_{si}	Skyhook gain
V_{max}	Maximum value of input voltage
c_{bi}	Damping coefficient
f_{ctrli}	Control damping force
f_{di}	MR damping force
f_i	Damping force for ith damper
k_{bi}	Suspension stiffness
l_i	Distance between wheel center and c.g
m_b	Body mass
m_{wi}	Wheel mass for ith wheel
t_i	Time shift for ith wheel
v_i	Applied input voltage
z_b	Body displacement
z_{ri}	Vertical excitation for ith wheel
z_{wi}	Wheel displacement for ith wheel
τ_i	Hybrid control weighting factor
Iy	Pitch moment of inertia
V	Vehicle velocity
θ	Pitch angle

INTRODUCTION

Recently numerous studies on vibration control of off-road vehicle suspension systems have been undertaken by incorporating electrorheological (ER) dampers [1-2]. However, ER dampers are not as effective as magnetorheological (MR) dampers and ER dampers are not commercially available but MR dampers are available. MR dampers are semi-active control devices that exploit MR fluids to create controllable damping forces. They can be used as smart dampers for suppressing vibration of suspension systems. The importance and fascinating characteristics of these devices come from their ability to dynamically vary their properties and go from liquid to solid state by varying the magnetic field. Moreover they require a minimal amount of power for activation of the fluid, they have instantaneous response times (usually milliseconds) and they lack cumbersome hydraulic valves. Due to the stated advantages, MR dampers are classified as the most prominent devices that can be used in semi-active suspension systems to enhance and improve the suspension performance.

The challenge in using these smart devices is in how to develop a suitable and adaptable control algorithm that can deal, in an accurate manner, with the nonlinear hysteresis inherent in those dampers. Also, producing a robust mathematical model that reflects all characteristics of the dampers is an issue. Much research uses a classical control policy (like skyhook, groundhook and hybrid control) to direct the MR damper in wheeled vehicles such as passenger vehicles [3-5]. In the case of skyhook control, the controller provides the desired damping force depending on the direction of motion of the vibrating body and relative velocities. In the case of positive direction i.e. body and relative velocities are in the same direction, maximum damping force is delivered; otherwise, the damping force is a minimum. For the groundhook control, the damping force depends on the direction of the wheel and relative velocities. Unlike these two controllers, the hybrid control is a mix between skyhook and groundhook control. There is very little research study on the effect of MR dampers on multiple suspension performance. Thus more investigations in this field are highly desirable.

In this study, a new controller featuring fuzzy hybrid control is adopted. This proposed controller is used to improve the ride comfort and stability of a multiple suspension system of an off-road vehicle by taking advantages of the hybrid and fuzzy controllers. The fuzzy controller is used to balance the adjusting weighting factors of the hybrid control to obtain optimum performance. The robustness and effectiveness of the proposed controller are derived by testing a seven degrees-of-freedom (DOF) suspension model through computer simulation under bump and sinusoidal excitations. The body acceleration (BA), body displacement (BD), angular acceleration (AA) and pitch angle (PA) are used as judgment criteria for the suspension system.

DYNAMIC MODEL OF 7 DOF SUSPENSION SYSTEMS

The schematic configuration of a typical off-road vehicle is illustrated in Figure 1. In this study, a semi-active half model of an off-road vehicle suspension is investigated. The model is a 7 DOF suspension system fitted with three MR dampers that replace the conventional hydraulic shock absorber in passive suspension as shown in Figure 2. The model incorporates a vehicle hull (sprung mass) suspended over five road wheels (unsprung masses) The vehicle body is connected to the wheels through five springs and three MR dampers. It is noted that the vehicle body has two DOF (bounce and pitch) while the wheels have five DOF (independent bounce).To derive the equations of motion for the model, all forces acting on the vehicle body and the five wheels are calculated and the moments about the center of gravity of the vehicle body are determined. Referring to notations in Table 1, the equations of motion for body bounce and pitch and for wheels bounce can be expressed respectively as follows:

$$m_b \ddot{z}_b = -\sum_{i=1}^{5} k_{bi}(z_b + l_i\theta - z_{wi}) - \sum_{i=1,2,5} f_i \qquad (1)$$

$$Iy\ddot{\theta} = -l_i \sum_{i=1}^{5} k_{bi}(z_b + l_i\theta - z_{wi}) - l_i \sum_{i=1,2,5} f_i \qquad (2)$$

$$m_{wi}\ddot{z}_{wi} = k_{bi}(z_b + l_i\theta - z_{wi}) + f_i - k_{wi}(z_{wi} - z_{ri}) \qquad (3)$$

where the force generated through the damper can be written as follows for $i=1,2,5$:

$$f_i = \begin{cases} c_{bi}(\dot{z}_b + l_i\dot{\theta} - \dot{z}_{wi}), & \text{for hydraulic damper} \\ f_{di}, & \text{for MR damper} \end{cases} \qquad (4)$$

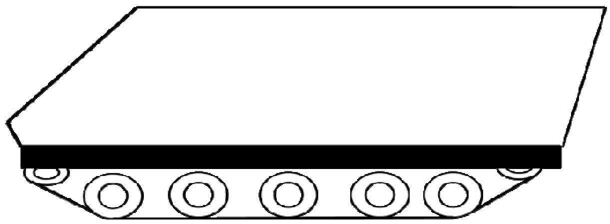

Figure 1Schematic configuration of an off-road vehicle

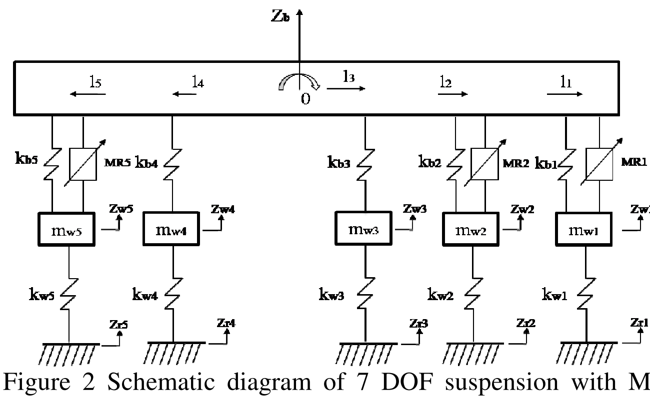

Figure 2 Schematic diagram of 7 DOF suspension with MR dampers

Table 1 Model suspension parameters [6-7]

Symbol	Value	Symbol	Value
m_b (kg)	5109	l_1(m)	1.35
m_{wi} (kg)	30	l_2(m)	0.69
Iy kg.m^2	12856	l_3(m)	0.02
k_{bi} (N/m)	120710	l_4(m)	-0.66
c_{bi} (N.s/m)	16500	l_5(m)	-1.32
k_{wi} (N/m)	10^6		

MR DAMPER MODEL

There are various models of MR fluids and MR dampers [8-11]. A dynamic damper model proposed in [11], is used in this research to obtain the controllable damping force of the MR damper. It is noted that the equations of motion were modified to suit the 7 DOF suspension with three MR dampers. Figure 3 illustrates a schematic diagram of the Modified Bouc-Wen model used. In the proposed model the accumulator stiffness is represented by a spring with constant, k_1, the damping at high velocities is denoted by, c_{oi}. A damper of damping c_{1i}, is introduced in the model to account for the hysteresis loop at low frequencies. However, a spring, with constant, k_o is introduced into the model for controlling stiffness at large velocities. It is supposed that the initial displacement due to the accumulator stiffness is, x_o. The internal displacement d_i for the ith damper ($i = 1,2,5$) is governed by the following equation [11].

$$\dot{d}_i = \frac{1}{c_{oi}+c_{1i}}\left[\alpha_i\psi_i + c_{oi}(\dot{z}_b + l_i\dot{\theta}) + k_o(z_b + l_i\theta - d_i) + c_{1i}\dot{z}_{wi}\right] \quad (5)$$

where, ψ_i is the evolutionary variable that represents the hysteresis loop for the i^{th} damper ($i = 1,2,5$) and can be expressed as follows [11]:

$$\dot{\psi}_i = -\gamma\,\psi_i\left|(\dot{z}_b + l_i\dot{\theta} - \dot{d}_i)\right|\left|\psi_i^{\,n-1}\right| - \beta(\dot{z}_b + l_i\dot{\theta} - \dot{d}_i)|\psi_i|^n + A(\dot{z}_b + l_i\dot{\theta} - \dot{d}) \quad (6)$$

The vertical bounce and pitch of the vehicle body are z_b and θ,respectively, and the wheel bounce is z_{wi}. The shape of the hysteresis loop can be adjusted by the constants,γ, β, n, and A. All parameters for the MR damper model are listed in Table 1. From the representation of the MR damper shown in Figure 3, the MR damper force can be written as:

$$f_{di} = c_1\dot{d}_i + k_1(z_b + l_i\theta - z_{wi} - x_o) \quad (7)$$

The applied voltage, v_i to the current driver is related to the output voltage, u_i for $i = 1,2,5$ according to the following equations:

$$\dot{u}_i = -\eta(u_i - v_i) \quad (8)$$

$$\alpha_i = \alpha_a + \alpha_b u_i \quad (9)$$

$$c_{oi} = c_{oa} + c_{ob}u_i \quad (10)$$

$$c_{1i} = c_{1a} + c_{1b}u_i \quad (11)$$

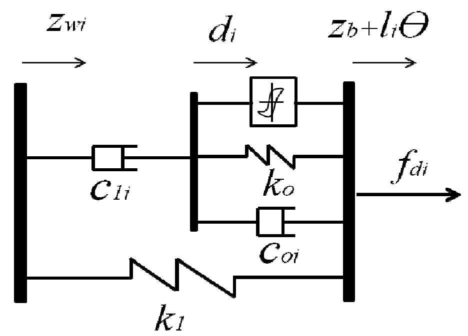

Figure 3 Modified Bouc-Wen model adjusted for 7 DOF suspension

SEMI-ACTIVE CONTROL VIA MR DAMPERS

In fact, any control system consists of a mechanical system to be controlled (usually referred to as the plant); sensors and a controller. It is known that a vehicle's passive suspension has two conflict parameters: ride comfort and vehicle handling. These parameters are preselected by the designer according to the objective and proposed application of the suspension. Passive suspension for all time is unable to eliminate the trade-off between ride comfort and stability due to its fixed parameters (i.e. spring and damping constant). If the designer intends to suppress the vehicle's vibration, suspension parameters are chosen to be soft (low damping), however vehicle stability will not be ideal. Alternatively, if the suspension is selected to enhance stability of the vehicle (high damping constant) the user is subjected to large amount of vibration and the ride comfort will be uneven. Superior design of passive suspension should eliminate this trade-off between ride comfort and stability or at least try to optimize the conflicting targets. As a result of the inability of a passive suspension to secure the compromise between ride comfort and stability, there is a growing demand to use a semi-active suspension. In this type of suspension, a controllable damper is used to alter the damping force consistent with control guidelines which is selected in advance to meet the applications needs of the suspension.

Table 2 MR damper model [12]

parameters	value	parameters	value
c_{oa}	2100 Ns/m	α_b	69500 N/Vm
c_{ob}	3500 Ns/Vm	γ	36300 m^{-2}
c_{1a}	28300Ns/m	β	36300 m^{-2}
c_{1b}	295/Vm	A	301
k_1	5 N/m	n	2
x_o	0.143 m	k_o	4690 N/m
α_a	14000 N/m	η	190

The semi-active control devices that employ a MR damper cannot input energy into the mechanical system being controlled[12-13]. The challenge of controlling these semi-active devices comes from the nonlinear dynamic behavior of such dampers. It is the command voltage applied to the current driver that is connected to the MR damper and that energizes the MR damper to produce the damping force, which can be directly controlled [14]. The robustness of the control method depends on how strictly it can deal with the nonlinear relationship between the damping force and the applied voltage/current. In multiple suspensions, semi-active control via a MR damper is achieved by two levels of control, namely, the damper controller and the system controller. The damper controller is exercised to create and amend the command voltage and to track the desired damping force that is specified by the system controller on the basis of desired and actual damping force [14]. Damper and system controllers will be explored in some details.

MR Damper Controller

The damper controller is used to allocate the damping force and to track the desired force produced by the system controller [14]. It is becoming increasingly difficult to ignore two factors that have great importance when designing the damper controller: (1) tracking ability of damping force, and (2) energy requirement for MR damper. The term tracking ability can be defined as the ability to make the controlled damping force to follow the desired force specified by the system controller. The error between these two forces is used as an evaluation for this factor. However the term energy requirement is generally understood to mean that less energy expenditure will extend the effective time of the power supply and hence enlarge the life time of the damper.

There are different types of MR damper controllers. This study will focus on Heaviside Step Function (HSF). In the HSF the input voltage is varied between two values i.e. zero or maximum to alter the damper force in order to closely track the desired force produced by the system controller. The governing equation for input voltage and damping force can be written as follows [12, 15]:

$$v_i = V_{max} H[(f_{di} - f_i) f_i] \quad (12)$$

System Controller

In contrast to the damper controller, the system controller is responsible for creation of the desired damping force according to the dynamic behavior of the plant [16]. It is necessary to use appropriate and simple control policy to realize reasonable isolation performance under various road excitation. The suggested control scheme used in this study is a fuzzy hybrid controller. This controller has many advantages as it combines the skyhook and groundhook effect with fuzzy logic. The construction of skyhook and groundhook is simple and realistic. Moreover, skyhook accounts for the ride comfort of a vehicle while groundhook affects vehicle stability and handling. The control scheme is illustrated in Figure 4 . Only the system controller integrated with the suspension model is shown. The fuzzy logic controller takes the body velocity (BV) and the wheel velocity (WV) of each damper and produces the appropriate weighting factor (τ_i) which is sent to the hybrid controller. Based on the value of the weighting factor, the hybrid controller generates the desired damping force ($f_{ctrl,i}$).

Afterward the desired damping force is compared with the actual damping force produced by the MR damper. Then, the necessary applied voltage is generated through the damper controller to steer the actual force to track the control force.

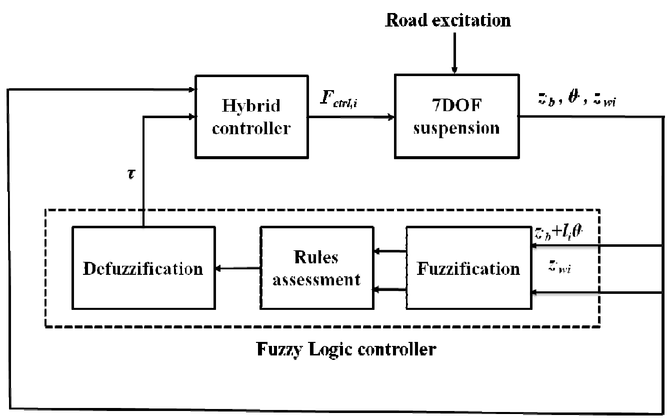

Figure 4 Block diagram of 7 DOF model with system controller

The governing equation used for the creation of variable damping force ($f_{ctrl,i}$) via the hybrid controller is expressed below for the i^{th} damper, *i=1,2,5*

$$f_{ctrl,i} = \left[\tau_i G_{si}(\dot{z}_b + l_i\dot{\theta}) - (1-\tau_i) G_{gi}\dot{z}_{wi}\right] \quad (13)$$

where,G_{si} and G_{gi} are skyhook and groundhook gain and they take values of 90000 and 9000 respectively. As can be seen from the above equation the weighting factor τ_i plays a crucial role in the controller output behavior. To explain this important role, let $\tau_i = 1$ .Then according to the above equation the controller will be pure skyhook. But if $\tau_i = 0$, then the controller will be pure groundhook. Thus it is possible through altering τ_i value, to switch from skyhook to groundhook controller. Consequently, enhancement of the ride comfort and stability can be achieved. Then, it is important to determine the values of the weighting factor based on the dynamic behavior of the suspension. For that reason, a fuzzy logic controller is employed in this study. The fuzzy input variables that are used in this controller are the vertical velocities across the damper ends, $\dot{z}_b + l_i\dot{\theta}$ and $\dot{z}_{wi}$ respectively while the fuzzy output variable is τ_i that is provided to the hybrid controller.

The input and output variables that are involved in the fuzzy logic controller are described below:

$$\begin{cases} \dot{z}_b + l_i\dot{\theta} = [VS, SM, ME, LA, VL] \\ \dot{z}_{wi} = [VS, SM, ME, LA, VL] \\ \tau_i = [ZE, Q_1, Q_2, Q_3, Q_4, Q_5, Q_6] \end{cases} \qquad (14)$$

where, VS= very small, SM =small, ME =medium, LA =large and VL = very large for the fuzzy input. For the output variable, ZE= zero,$Q_1 = 0.1$, Q_2=0.3, Q_3 =0.5, Q_4=0.7, Q_5=0.9, Q_6=1. The ranges for the three fuzzy variables take these values respectively [-0.5, 0.5], [-1, 1] and [0, 1]. The possible membership functions for the three fuzzy variables are illustrated below:

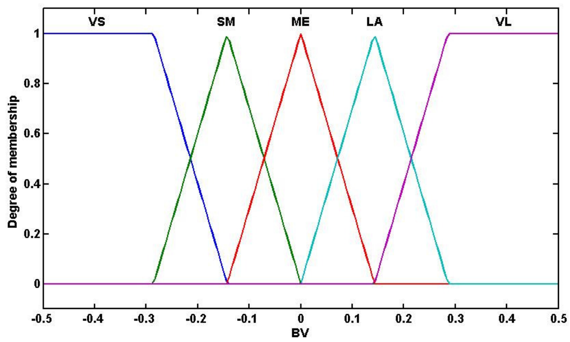

Figure 5 Membership function for i^{th} vertical body velocity

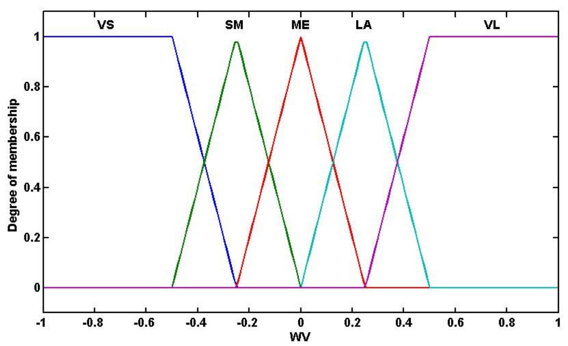

Figure 6 Membership function for i^{th} vertical wheel velocity

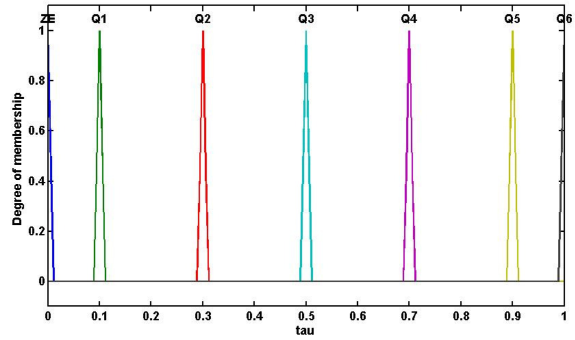

Figure 7 Membership function for i^{th} desired output

The fuzzy controller rules are listed in Table 3. There are 25 rules adopted for the proposed controller. Each rule is executed based on IF-THEN statement. For instance, rule number 5 (R5) and rule number 21 (R21) are executed as follows:

R5: IF ($\dot{z}_b + l_i\dot{\theta}$ = VL) and ($\dot{z}_{wi}$ =VS) THEN (τ_5=1)

R21: IF ($\dot{z}_b + l_i\dot{\theta}$ = VS) and ($\dot{z}_{wi}$ =VL) THEN (τ_5=0)

Table 3 Fuzzy control rules

$\dot{z}_{wi}$	$\dot{z}_b + l_i\dot{\theta}$				
	VS	***SM***	***ME***	***LA***	***VL***
VS	Q_3	Q_5	Q_5	Q_6	Q_6
SM	Q_1	Q_3	Q_4	Q_5	Q_5
ME	Q_1	Q_2	Q_3	Q_4	Q_5
LA	Q_1	Q_1	Q_2	Q_3	Q_4
VL	ZE	Q_1	Q_2	Q_2	Q_3

SIMULATION RESULTS AND DISCUSSIONS

Assessment of the proposed controller is evaluated through a computer simulation. SIMULINK models for both passive and semi-active suspension system are created in Matlab in conjunction with the MR damper model and fuzzy hybrid controller model. Consequently, the passive and semi-active suspension system with fuzzy hybrid control are compared for two different excitations (bump and sinusoidal). The vertical body acceleration and displacement (BA and BD), angular acceleration (AA) and the pitch angle (PA) are used as performance indicators for evaluation of the proposed controller. For better control performance, the magnitude of all these indicators should be reduced.

It is noted that the simulations were done for two different configurations of numbers and/or locations of MR dampers. The first configuration incorporates two MR dampers mounted at the first and the last wheels however the second one includes three dampers at the first, the second and the fifth wheel. It is revealed from these simulations that using three MR dampers offer potential improvement over using only two MR dampers at first and last wheel.

System Responses Under Bump Road Input

The displacement of the road excitation and the time histories of the passive and controlled suspension under this excitation are shown in Figures 8 to 12. The bump has amplitude of 0.1 m, wave length of the road is 10 m as in [17] and the vehicle crosses the bump with a constant speed of 20 km/h. It is noted that all wheels are subjected to the same excitation but with different phase i.e. $t_i = \frac{l_1 - l_i}{V}$, $i = 2,3,4,5.$ As can be seen from the figures, the suggested controller offers a percentage reduction in peak-to-peak (PTP) response of BA, BD, AA and PA less than the passive system by about 5.9%, 2%, 9.24% and 7.61% respectively.

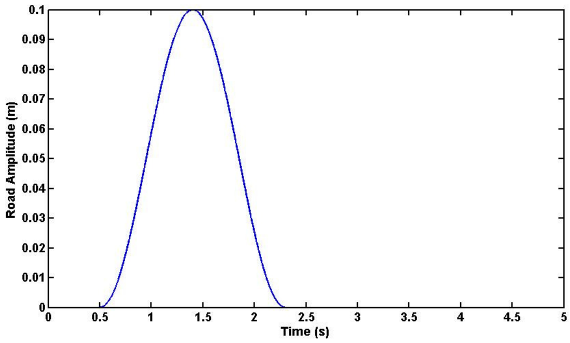

Figure 8 Typical road bump input

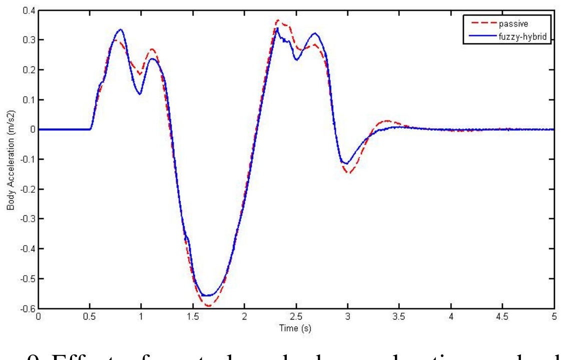

Figure 9 Effect of control on body acceleration under bump excitation

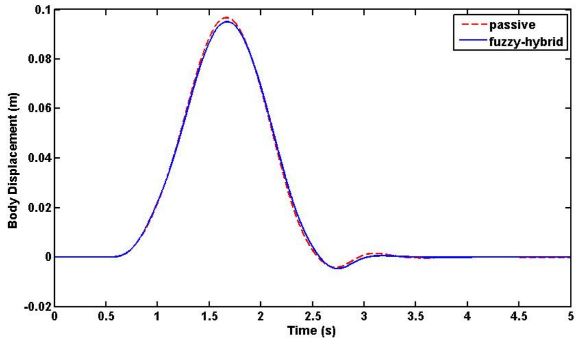

Figure 10 Effect of control on body displacement under bump excitation

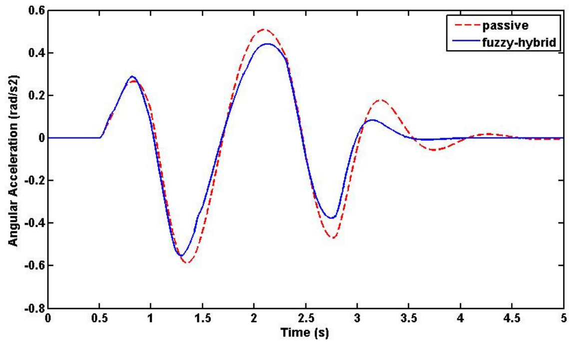

Figure 11 Effect of control on angular acceleration under bump excitation

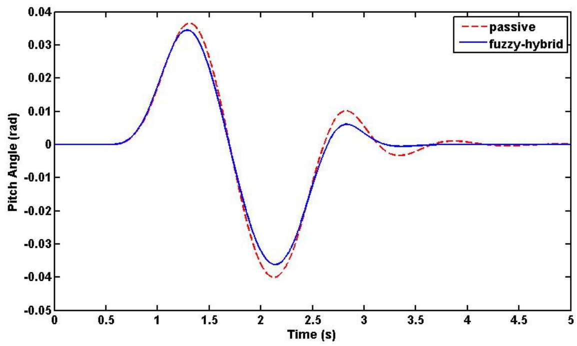

Figure 12 Effect of control on pitch angle under bump excitation

System Responses Under Sinusoidal Road Input

The displacement of the road excitation and the responses of the controlled system compared with passive system for BA, BD, AA and PA under sinusoidal excitation are shown in Figures 13 to 17. The road has amplitude equal 0.1 m with wave length 8 m as in [18] and the vehicle crosses the road with velocity 60km/h. It can be noticed that the fuzzy hybrid control provides slightly better performance than the passive suspension. The root mean square (RMS) values of BA, BD, AA and PA are reduced by about 14.12 %, 13.12 %, 32.89 % and 35.6 % respectively.

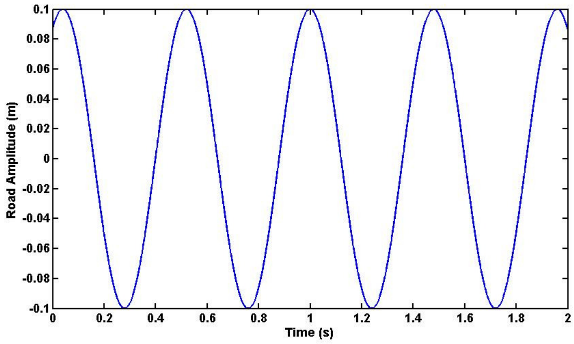

Figure 13 Typical road sinusoidal input

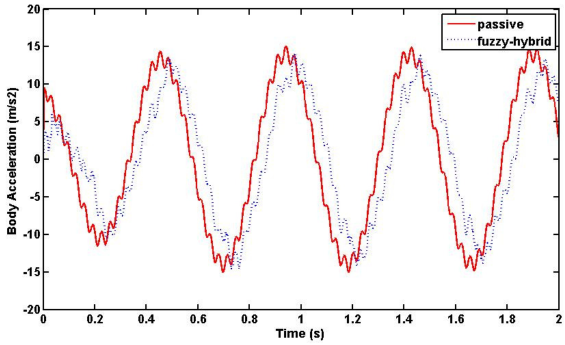

Figure 14 Effect of control on body acceleration under sinusoidal excitation

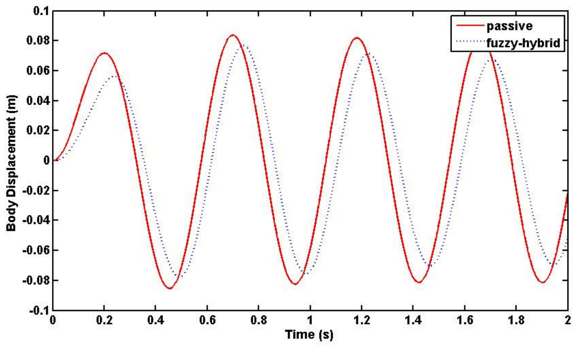

Figure 15 Effect of control on body displacement under sinusoidal excitation

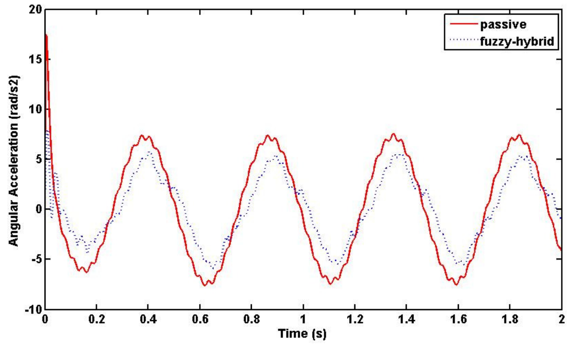

Figure 16 Effect of control on angular acceleration under sinusoidal excitation

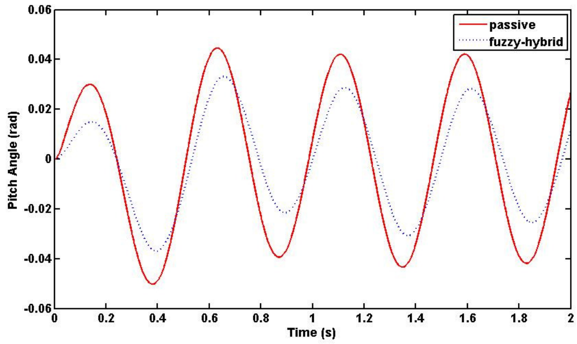

Figure 17 Effect of control on pitch angle under sinusoidal excitation

CONCLUSIONS

A 7 DOF suspension system integrated with three MR dampers have been presented. A new fuzzy hybrid control scheme is adopted to improve the suspension performance under various operating conditions. The proposed controller achieves good improvement in the suspension performance by increasing the ride comfort and stability.

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