DOI: 10.3901/CJME.2013.03.498, available online at www.springerlink.com; www.cjmenet.com; www.cjmenet.com.cn

Skyhook-based Semi-active Control of Full-vehicle Suspension with Magneto-rheological Dampers

ZHANG Hailong[1], WANG Enrong[1, *], MIN Fuhong[1], SUBASH Rakheja[2], and SU Chunyi[2]

1 *School of Electric and Automation Engineering, Nanjing Normal University, Nanjing 210042, China*
2 *Department of Mechanical Engineering, Concordia University, Montreal H3G 1M8, Canada*

Received July 21, 2012; revised September 20, 2012; accepted February 25, 2013

Abstract: The control study of vehicle semi-active suspension with magneto-rheological (MR) dampers has been attracted much attention internationally. However, a simple, real time and easy implementing semi-active controller has not been proposed for the MR full-vehicle suspension system, and a systematic analysis method has not been established for evaluating the multi-objective suspension performances of MR full-vehicle vertical, pitch and roll motions. For this purpose, according to the 7-degree of freedom (DOF) full-vehicle dynamic system, a generalized 7-DOF MR and passive full-vehicle dynamic model is set up by employing the modified Bouc-wen hysteretic force-velocity (*F*-*v*) model of the MR damper. A semi-active controller is synthesized to realize independent control of the four MR quarter-vehicle sub-suspension systems in the full-vehicle, which is on the basis of the proposed modified skyhook damping scheme of MR quarter-vehicle sub-suspension system. The proposed controller can greatly simplify the controller design complexity of MR full-vehicle suspension and has merits of easy implementation in real application, wherein only absolute velocities of sprung and unsprung masses with reference to the road surface are required to measure in real time when the vehicle is moving. Furthermore, a systematic analysis method is established for evaluating the vertical, pitch and roll motion properties of both MR and passive full-vehicle suspensions in a more realistic road excitation manner, in which the harmonic, rounded pulse and real road measured random signals with delay time are employed as different road excitations inserted on the front and rear two wheels, by considering the distance between front and rear wheels in full-vehicle. The above excitations with different amplitudes are further employed as the road excitations inserted on left and right two wheels for evaluating the roll motion property. The multi-objective suspension performances of ride comfort and handling safety of the proposed MR full-vehicle suspension are thus thoroughly evaluated by comparing with those of the passive full-vehicle suspension. The results show that the proposed controller can ideally improve multi-objective suspension performances of the ride comfort and handling safety. The proposed harmonic, rounded pulse and real road measured random signals with delay time and asymmetric amplitudes are suitable for accurately analyzing the vertical, pitch and roll motion properties of MR full-vehicle suspension system in a more realistic road excitation manner. This research has important theoretical significance for improving application study on the intelligent MR semi-active suspension.

Key words: magneto-rheological damper, skyhook policy, semi-active control, multi-objective performances, full-vehicle suspension

1 Introduction

Suspension plays an important role in ensuring ride comfort and handling safety of the road vehicle and the semi-active controllable magneto-rheological (MR) damper has been widely explored to realize an intelligent semi-active vehicle suspension in the past decade. Lots of papers were published on modeling the hysteretic force-velocity (*F*-*v*) characteristic of MR damper and synthesizing semi-active controller aimed at the MR quarter-vehicle suspension[1–7]. Typical models include the Bouc-wen hysteretic *F*-*v* model proposed by SPENCER, et al[8] and the sigmoid hysteretic *F*-*v* model proposed by WANG, et al[9], the former has advantage of precisely describing transient process but lacks precision in describing steady state property, while the later has high precision in describing steady state property but lacks precision in describing transient process of the MR damper. The semi-active control studies, such as skyhook control, H_∞ control and neural fuzzy control, focus on improving the ride performance of MR quarter-vehicle suspension by suppressing vertical motion vibration. Whereas, the full-vehicle suspension has requirements of multi-objective suspension performances involving the vertical, pitch and roll motions, few studies have considered the semi-active control aimed at MR full-vehicle suspension. CHOI, et al[7], early investigated adaptive control of MR full-vehicle suspension with body mass variation by applying H_∞ control strategy. YANG, et al[10], proposed a semi-active controller employing fuzzy

* Corresponding author. E-mail: erwang@njnu.edu.cn
This project is supported by National Natural Science Foundation of China (Grant No. 51075215), and Research, Innovation Project for College Graduates of Jiangsu Province (Grant No. CXLX11_0889)

T-S neural network strategy for the MR full-vehicle suspension. LIAO, et al[11], proposed a coordinate semi-active controller combining skyhook, ground-hook and fuzzy control schemes. The above achievements have strongly put forward the semi-active control study of MR full-vehicle suspension. However, a proper analysis method has not been set up for evaluating the multi-objective suspension performances about the vertical, pitch and roll motions of full-vehicle, and the proposed semi-active controllers seem more complicate and difficult in real vehicle.

According to 7-DOF full-vehicle dynamic system[12], a generalized 7-DOF MR and passive full-vehicle dynamic model is set up by employing the proposed modified Boucwen hysteretic *F-v* model of MR damper, and an asynchronous semi-active controller with the proposed modified skyhook damping scheme is synthesized to realize independent control of the four MR quarter-vehicle sub-suspension systems in full-vehicle. Considering the distance between front and rear wheels in full-vehicle, the proposed harmonic, rounded pulse and real road measured random signals with delay time are employed as input excitations on the front and rear two wheels from road surface, so as to systematically analyze the vertical and pitch motion properties, and the above excitations with different amplitudes are further employed as the left and right two wheel road excitations for evaluating the roll motion property, in a more realistic road excitation manner. The multi-objective suspension performances of ride comfort and handling safety of the proposed MR full-vehicle suspension are thus thoroughly evaluated by comparing with those of the passive full-vehicle suspension.

2 7-DOF Full-vehicle Suspension Model

It is required to set up a proper MR full-vehicle suspension model, so as to evaluate the multi-objective performances of the vertical, pitch and roll motions. Fig. 1 shows a 7-DOF MR full-vehicle suspension model.

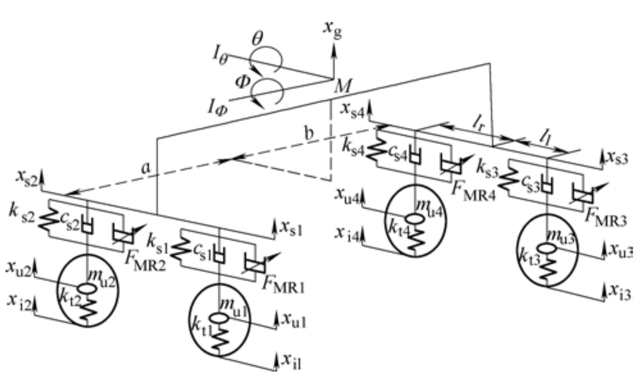


Fig. 1. 7-DOF dynamic model of MR and passive full-vehicle suspensions

In Fig. 1, M is sprung mass, I_θ and I_Φ are inertia moments of the pitch and roll motions. x_g, θ, Φ represent vertical displacement, pitch and roll angles, and a, b and l_l, l_r are distances from the front and rear, as well as the left and right wheels to the gravity center of sprung mass, respectively. m_{ui}, k_{si}, c_{si}, F_{MRi}, k_{ti} represent unsprung masses, suspension stiffness coefficients, suspension damping coefficients, MR damper forces, and tire stiffness coefficients, and x_{si}, x_{ui}, x_{ii} represent sprung and unsprung masses and road excitation displacements of the four quarter-vehicle sub-suspension systems, respectively. Herein, the subscript i=1, 2, 3, 4 denote the four front and rear sub-suspension systems at left and right sides of the full-vehicle, respectively. The proposed suspension model can be easily employed to the MR full-vehicle suspension by setting c_{si}=0 and to the passive full-vehicle suspension by setting F_{MRi}=0.

The generalized dynamic equations of MR and passive full-vehicle suspensions can be thus expressed as follows:

$$\begin{cases} M\ddot{x}_g = -\sum\limits_{i=1}^{4} c_{si}(\dot{x}_{si}-\dot{x}_{ui}) - \sum\limits_{i=1}^{4} k_{si}(x_{si}-x_{ui}) - \sum\limits_{i=1}^{4} F_{MRi}, \\ I_\theta\ddot{\theta} = a\sum\limits_{i=1}^{2} c_{si}(\dot{x}_{si}-\dot{x}_{ui}) + k_{si}(x_{si}-x_{ui}) + F_{MRi} - \\ \qquad b\sum\limits_{i=3}^{4} c_{si}(\dot{x}_{si}-\dot{x}_{ui}) + k_{si}(x_{si}-x_{ui}) + F_{MRi}, \\ I_\phi\ddot{\phi} = -l_l[c_{s1}(\dot{x}_{s1}-\dot{x}_{u1}) + k_{s1}(x_{s1}-x_{u1}) + F_{MR1} + \\ \qquad c_{s3}(\dot{x}_{s3}-\dot{x}_{u3}) + k_{s3}(x_{s3}-x_{u3}) + F_{MR3}] + \\ \qquad l_r[c_{s2}(\dot{x}_{s2}-\dot{x}_{u2}) + k_{s2}(x_{s2}-x_{u2}) + F_{MR2} + \\ \qquad c_{s4}(\dot{x}_{s4}-\dot{x}_{u4}) + k_{s4}(x_{s4}-x_{u4}) + F_{MR4}], \\ m_{ui}\ddot{x}_{ui} = c_{si}(\dot{x}_{si}-\dot{x}_{ui}) + k_{si}(x_{si}-x_{ui}) - k_{ti}(x_{ui}-x_{ii}) + F_{MRi}, \end{cases} \tag{1}$$

where $\dot{x}_{si}$ and $\dot{x}_{ui}$ represent displacement velocities of the sprung and unsprung masses, $\ddot{x}_g$, $\ddot{\theta}$ and $\ddot{\phi}$ represent vertical displacement, pitch angle and roll angle accelerations of the sprung mass, respectively.

F_{MRi} represent MR damper forces in the four MR quarter-vehicle sub-suspension systems and can be calculated through an appropriate hysteretic *F-v* model of the MR damper. In this study, the proposed generalized hysteretic *F-v* model decoupled with current control and hysteresis force is employed to modify the proposed original Bouc-wen hysteretic *F-v* model[13], consequently, the proposed modified Bouc-wen hysteretic *F-v* model in Eqs. (2)–(4) can accurately describe nonlinear hysteretic steady and transient characteristics of the MR damper:

$$F_{MRi}(i_{di}, v_{ri}) = C(i_{di})F_h(v_{ri}), \tag{2}$$

$$C(i_{di}) = 1 + \frac{k_2}{1+\exp(-a_2(i_{di}+I_0))} - \frac{k_2}{1+\exp(-a_2 I_0)}, \tag{3}$$

$$F_h(v_r) = c_1\dot{y} + k_1(x - x_0), \tag{4}$$

$$\dot{y} = \frac{1}{c_0+c_1}[\alpha z + c_0\dot{x} + k_0(x-y)],$$

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

where $C(i_{di})$ formulates the MR damper force in relationship with driving direct current (i_{di}), which can accurately describe the nonlinear variation and saturation properties of magnetic field depending on the controlled driving current. $F_h(v_{ri})$ formulates the MR damper force in relationship with relative motion velocity ($v_{ri} = \dot{x}_{si} - \dot{x}_{ui}$) of damper piston, which is expressed with the Bouc-wen hysteron.

A candidate MR damper with working source 12 V and limited maximum drive current 0.5 A, manufactured by CARRERA company[3–5], is employed in this study, and the tested data are used to identify parameters of the proposed modified Bouc-wen hysteretic F-v model, such as γ=8 816.9, β=233 849.1, α=20 373.7, c_0=1 368.7, c_1=6 222.7, k_0=184.1, k_1=1 528.1, k_2=10.092, n=2, A=20.6, x_0=−0.004, a_2=7.526, I_0=0.069. Fig. 2 shows comparison of the model results and measured data under varied drive current (0–0.4 A) and harmonic excitations with amplitude 12.5 mm at frequency 1.5 Hz and 2.5 Hz, respectively. The result illustrates that the proposed modified Bouc-wen hysteretic F-v model can precisely describe real operation characteristic of the MR damper[13].

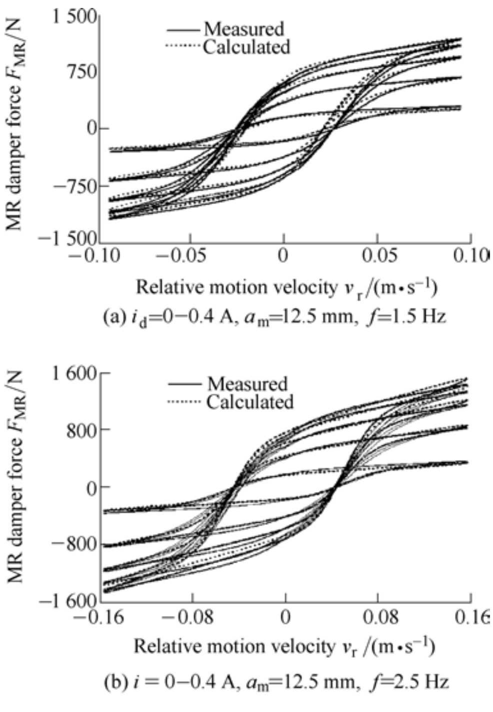


Fig. 2. Result comparison of model calculation and measured data

3 Skyhook-based Semi-active Controller of MR Full-vehicle Suspension

The original skyhook damping scheme, proposed by KARNOPP, et al[14], intends to suppress vibration acceleration of sprung mass by increasing and decreasing the suspension damping coefficients when the sprung mass velocity is in same and inverse phases with relative velocity of the damper piston, respectively, so as to ideally improve the ride comfort suspension performance. A modified skyhook semi-active control strategy in current form for the MR quarter-vehicle suspension, proposed by the authors[3–5], is directly applied to synthesize an asynchronous semi-active controller in Eq. (5), which is helpful to greatly simplify controller design complexity of the MR full-vehicle suspension, and can realize independent control of the four MR quarter-vehicle sub-suspension systems. The proposed modified skyhook-based asynchronous semi-active controller has further merits of easy implementation in real application, wherein only absolute velocities with reference to the road surface of sprung and unsprung masses are required to measure in real time when the vehicle is moving.

$$i_{di} = \begin{cases} k_d|\dot{x}_{si}|^m, & \dot{x}_{si}\ (\dot{x}_{si}-\dot{x}_{ui}) > 0, \\ 0, & \dot{x}_{si}\ (\dot{x}_{si}-\dot{x}_{ui}) \leqslant 0, \end{cases} \quad i = 1, 2, 3, 4, \tag{5}$$

where i_{di} are controlled drive currents of the four MR dampers, $\dot{x}_{si}$ and $\dot{x}_{ui}$ are displacement velocities of sprung and unsprung masses in the four MR quarter-vehicle sub-suspension systems, respectively. k_d denotes controller gain and m ($m \geqslant 2$) denotes controller order, which has potential to suppress the hysteresis effect and to improve the damping force bandwidth. In this study, the controller parameters are chosen as m=2 and k_d=3, and the parameters of 7-DOF full-vehicle model of the Ford Granada car are listed in Table 1[12].

Table 1. Parameters of the Ford Granada car

Parameter	Value
Sprung mass of vehicle M/kg	1 480
Inertia of pitching motion I_θ/(kg • m^2)	2 440
Inertia of rolling motion I_Φ/(kg • m^2)	380
Front unsprung mass m_{u1}, m_{u2}/kg	40.5
Rear unsprung mass m_{u3}, m_{u4}/kg	45.4
Tire stiffness coefficient k_{t1}–k_{t4}/(kN • m^{-1})	192
Front suspension stiffness coefficient k_{s1}, k_{s2}/(kN • m^{-1})	17
Rear suspension stiffness coefficient k_{s3}, k_{s4}/(kN • m^{-1})	22
Suspension damping coefficient c_{s1}–c_{s4}/(kN • s • m^{-1})	1 500
Distance from front suspension to center of gravity a/m	1.25
Distance from rear suspension to center of gravity b/m	1.51
Half of vehicle width l_l, l_r/m	0.74

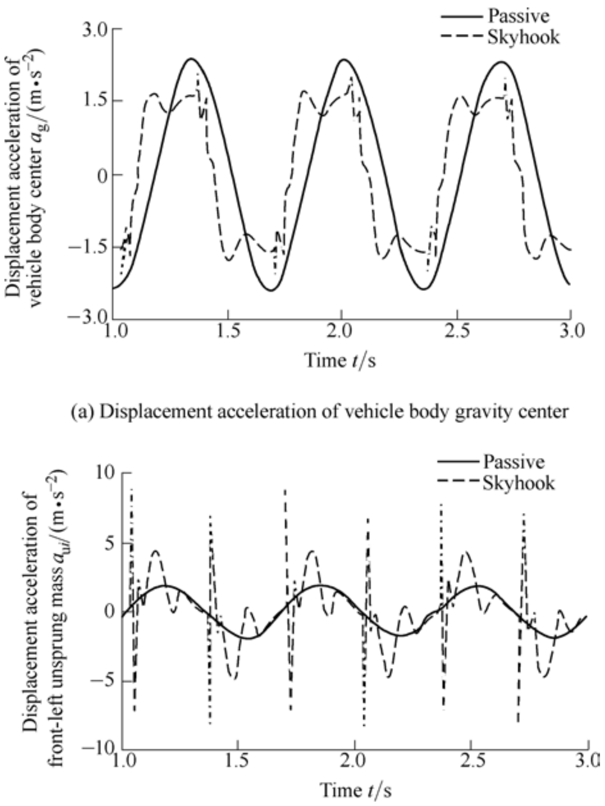
4 Analysis of Multi-objective Suspension Performances

The proposed skyhook-based asynchronous semi-active controllers are applied to the four MR dampers of 7-DoF Ford Granada dynamic system shown in Fig. 1, and a simulation platform with Matlab/Simulink is built up for analyzing both MR and passive full-vehicle suspension performances of the Granada car[15]. Considering delay time property between the front and rear two wheels when the car is moving, the proposed harmonic, rounded pulse and real road measured random signals[3–5] are modified with delay time (t_0) to imitate different road excitations, so as to analyze steady and transient state properties as well as multi-objective suspension performances of vertical, pitch and roll motions of the MR full-vehicle suspension in a systematic analysis manner. Herein, moving speed of the car is assumed as v=60 km/h, and delay time of the excitations inserted on the two rear wheels than those inserted on the two front wheels can be thus calculated as $t_0=(a+b)/v=0.17$ s.

4.1 Harmonic excitations

The harmonic excitation is usually applied to analyze the steady state response quality of a system. The ride comfort suspension performance can be evaluated by analyzing the steady state time responses of vehicle body gravity center displacement acceleration ($a_g = \ddot{x}_g$) and unsprung mass displacement accelerations ($a_{ui} = \ddot{x}_{ui}$) of the four quarter-vehicle sub-suspension systems under harmonic excitation at chosen frequency around natural frequency of the sprung mass, and by analyzing the frequency responses of displacement acceleration transmissibility (T_{ag}) of vehicle body gravity center and dynamic load coefficients of the four wheels (DLC_i) under varied amplitude harmonic excitations, in which T_{ag} and DLC_i are defined as ratios of Root Mean Square (RMS) of a_g to RMS of the excitation acceleration, and RMS of each wheel dynamic force ($F_{ti}=k_{ti}(x_{ui}-x_{li})$) to the wheel static force under a given harmonic excitation, respectively[3–5]. Moreover, the handling safety suspension performance can be reasonably evaluated through analyzing the steady state time responses of pitch and roll angular accelerations of vehicle body motion, under asymmetric amplitude harmonic excitations inserted on the left and right wheels.

The vertical motion property is analyzed by employing the harmonic excitations with delay time. Fig. 3 shows the steady state response comparison of vehicle body mass gravity center displacement acceleration a_g and unsprung mass displacement acceleration a_{u1} of front sub-suspension at left side in both MR and passive full-vehicle suspensions, under a harmonic excitation with amplitude 2.0 cm at frequency 1.5 Hz, in which the excitation inserted on the rear two wheels is delayed with 0.17 s than that inserted on the two front wheels. The results illustrate that the peak amplitude of a_g in the MR full-vehicle suspension decreases 30% than that in the passive full-vehicle suspension, which yields potential effectiveness for suppressing resonance of the sprung mass, whereas the peak amplitude of a_{u1} in the MR full-vehicle suspension increases 100% than that in the passive full-vehicle suspension, which sacrifices part of resonance suppression suspension performance of the unsprung mass. Moreover, a_{u1} yields obvious self-excited oscillations due to the strong hysteresis and force saturation nonlinear properties of the MR damper, and yields undesirable abrupt responses at about 1.05 s, 1.40 s, 1.75 s, due to discontinuous logic condition of the semi-active control manner as shown in Eq. (5).



(a) Displacement acceleration of vehicle body gravity center

(b) Displacement acceleration of unsprung mass

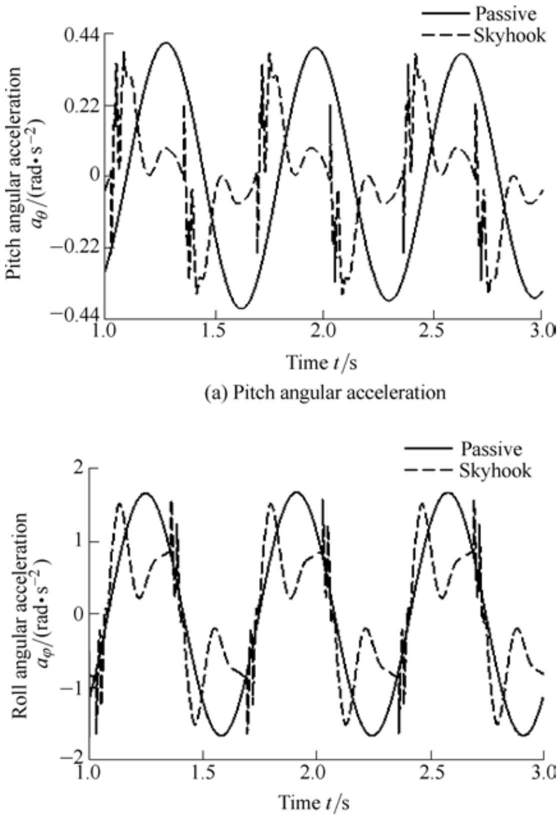
Fig. 3. Steady state response comparison of MR and passive full-vehicle suspensions under harmonic excitations with delay time

Furthermore, the pitch and roll motion properties are analyzed by employing the harmonic excitations with delay time and asymmetric amplitudes. Fig. 4 shows the time response comparison of pitch angular acceleration ($a_\theta = \ddot{\theta}$) and roll angular acceleration ($a_\phi = \ddot{\phi}$) of vehicle body motion, herein, in order to conveniently evaluate the roll motion suspension performance, asymmetric harmonic excitations with different amplitudes of 2.5 cm and 0.5 cm at same frequency 1.5 Hz are inserted on the left and right two wheels, respectively. Meanwhile, the excitations inserted on the two rear wheels are delayed with 0.17 s than those inserted on the two front wheels. The results show that amplitudes of a_θ and a_Φ in the MR full-vehicle suspension are obviously lower than those in the passive full-vehicle suspension, which illustrate that the proposed skyhook-based semi-active MR suspension has potential benefit of controlling the motion posture of vehicle body, and thus enhances the handling safety suspension performance. Furthermore, the results also show undesirable self-excited oscillation and abrupt responses, due to hysteresis of the MR damper and discontinuity of the semi-active control manner.

Fig. 5 shows the frequency response comparison of displacement acceleration transmissibility T_{ag} of vehicle body mass gravity center and dynamic load coefficient DLC_1 of front wheel at left side in both MR and passive full-vehicle suspensions, herein, the harmonic excitation with varied amplitudes as shown in Eq. (6) is employed to yield constant amplitudes at lower frequencies ($f \leqslant f_T$) and constant accelerations at higher frequencies ($f > f_T$), so as to avoid operation saturation of MR suspension dynamic system at higher frequencies due to nonlinear property of the MR damper[3–5].

$$x_{ii} = \begin{cases} a_m \sin(2\pi f t), & f \leqslant f_T, \\ a_m (f_T / f) \sin(2\pi f t), & f > f_T, \end{cases} \quad i = 1, 2, 3, 4. \tag{6}$$

The above function parameters are chosen as the amplitude a_m=1.0 cm, the cut-off frequency f_T=1.3 Hz, and the frequency f =0–25 Hz.

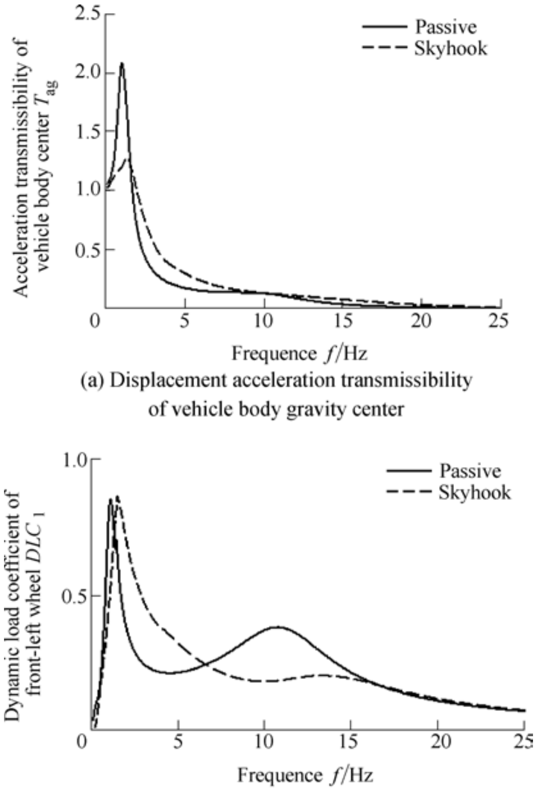


(a) Pitch angular acceleration

(b) Roll angular acceleration

Fig. 4. Steady state response comparison of MR and passive full-vehicle suspensions under harmonic excitations with delay time and asymmetric amplitudes

The results shown in Fig. 5 indicate that the peak value of T_{ag} around 1.5 Hz in the MR suspension decreases 40% than that in the passive suspension, while T_{ag} yields slight increases in the middle frequency range (2.0–8.0 Hz), which further illustrate that the proposed MR full-vehicle suspension has powerful role in suppressing the sprung mass resonance but sacrifices part of vibration isolation suspension performance. The results further show DLC_1 yields slight increases in the middle frequency range (2.0–6.5 Hz) and slight decreases in the higher frequency range (6.5–15.5 Hz), which is helpful to improve the road holding suspension performance of vehicle, and the sprung mass resonance frequency of MR suspension yields a little shift than that of the passive suspension.



(a) Displacement acceleration transmissibility of vehicle body gravity center

(b) Wheel dynamic load coefficient

Fig. 5. Frequency response comparison of MR and passive full-vehicle suspensions under harmonic excitations with varied amplitudes

4.2 Rounded pulse excitations

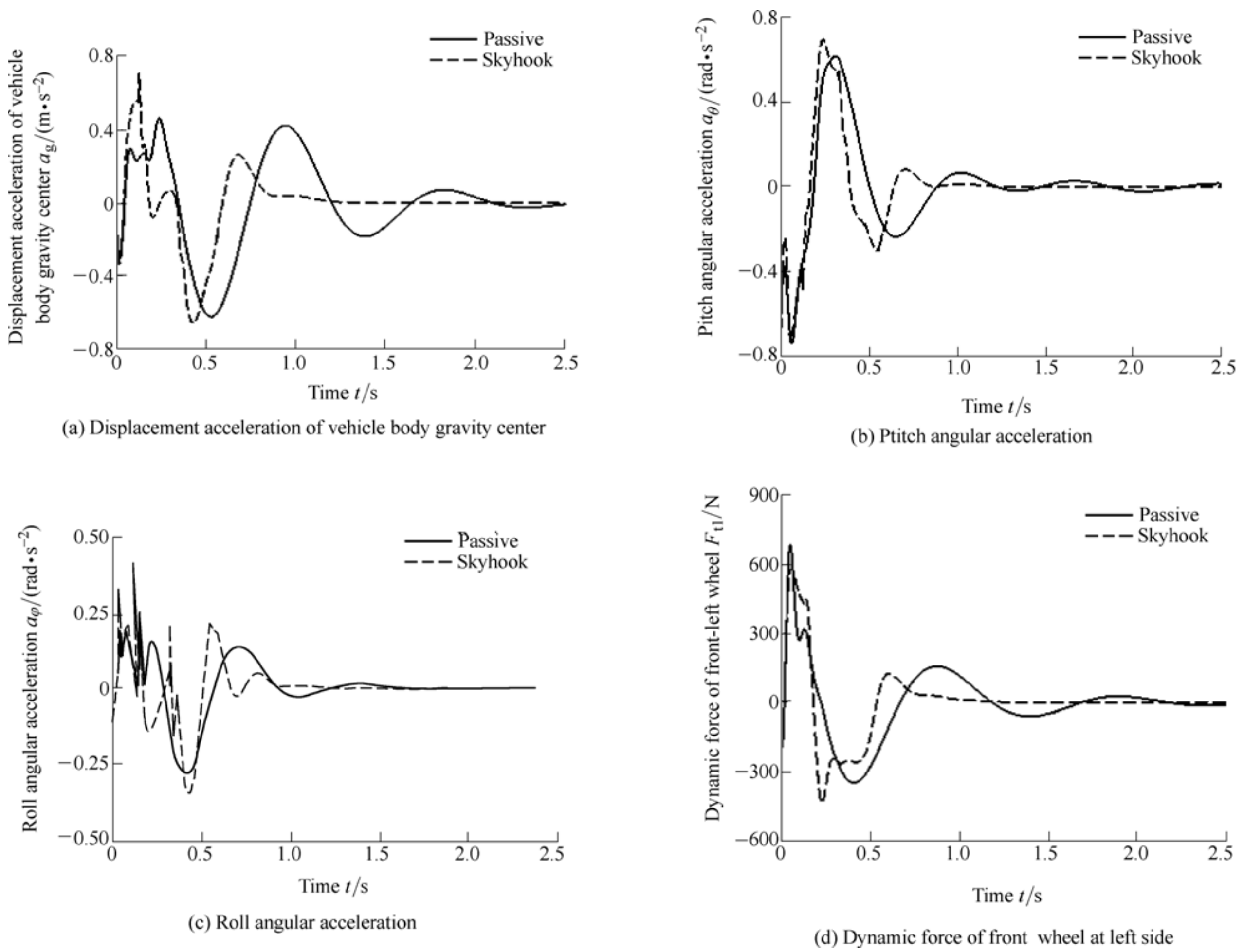
The moving vehicle often encounters harsh road surfaces such as hills and hollows, which can be formulated as the rounded pulse excitation to evaluate the shock attenuation suspension performance of vehicle suspension[3–5]. Herein, the rounded pulse excitations inserted on the two front wheels are generated using Eq. (7), and the excitations inserted on the two rear wheels are considered with a delay time in Eq. (8):

$$x_{i1,2} = 0.25 a_m \exp^2(\mu\omega_0 t)^2 \exp(-\mu\omega_0 t), \tag{7}$$

$$x_{i3,4} = 0.25 a_m \exp^2(\mu\omega_0 (t - t_0))^2 \exp(-\mu\omega_0 (t - t_0)) u(t - t_0), \tag{8}$$

where the function parameters are chosen as amplitude a_m=2.0 cm for the left two wheel excitations and a_m=1.0 cm for the right two wheel excitations, fundamental harmonic frequency ω_0=10.4 rad/s, pulse stiffness μ=3, and delay time t_0=0.17 s.

Fig. 6 shows the transient state response comparison of vehicle body gravity center displacement acceleration a_g, pitch angular acceleration a_θ, roll angular acceleration a_Φ and tire dynamic force F_{t1} of front wheel at left side, herein, the wheel dynamic force is defined as the dynamic load of each wheel, expressed by F_{ti}=$k_{ti}(x_{ui}$−$x_{ii})$. The results generally show that the transient processes of a_g, a_θ, a_Φ and F_{t1} in MR suspension are greatly shorten than those in the passive suspension, which benefits to improve the shock attenuation suspension performance. Whereas, the peak amplitudes of a_g, a_θ and a_Φ in the MR suspension yield a little increases than those in the passive suspension, which sacrifices part of ride comfort and handling safety suspension performances.



(a) Displacement acceleration of vehicle body gravity center

(b) Ptitch angular acceleration

(c) Roll angular acceleration

(d) Dynamic force of front wheel at left side

Fig. 6. Transient process response comparison of MR and passive full-vehicle suspensions under rounded pulse excitations with delay time and asymmetric amplitudes

4.3 Random excitations

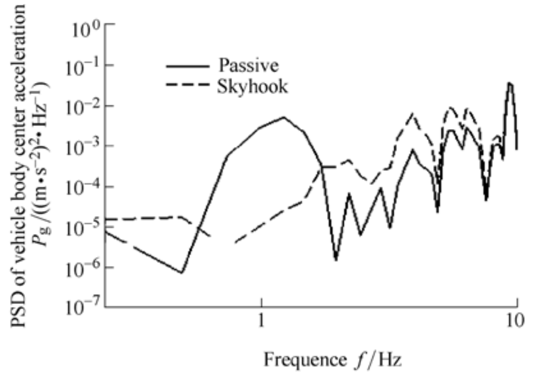
The real excitation inserted on wheels is usually random signal arising from road surface roughness when vehicle is moving on a road, and the real road measured roughness data reported in Canada[3–5, 16] are employed in this study, so as to actually evaluate the proposed skyhook-based asynchronous semi-active controller of MR full-vehicle suspension. The employed real road measured roughness data was processed by assuming that the moving speed of a car is v=50 km/h, such that the delay time of the excitations between the rear and front two wheels is $t_0=(a+b)/v=0.199$ s. Furthermore, amplitudes of the excitations inserted on the two right wheels are decreased 50% than those inserted on the two left wheels, so as to comprehensively evaluate the pitch, vertical and roll motion properties. Herein, the power spectral density(PSD) responses[16] of a_g, a_θ, a_Φ and F_{t1} are analyzed, and the results, as shown in Fig. 7, show that the magnitudes of PSD of a_g, a_θ and F_{t1} in MR suspension greatly decrease in the low frequency range(0.5–2.0 Hz) than those in the passive suspension, and thus illustrate that the proposed skyhook-based asynchronous semi-active controller has potential of the resonance attenuation of sprung mass, whereas a_Φ shows little variations. The results further show slight increases of the magnitudes of PSD of a_g, a_θ, a_Φ and F_{t1} in MR suspension than those in the passive suspension in the middle frequency range (2.0–8.0 Hz), which illustrate that the improvement of resonance attenuation suspension performance of sprung mass will sacrifice part of the resonance attenuation suspension performance of unsprung mass.

5 Conclusions

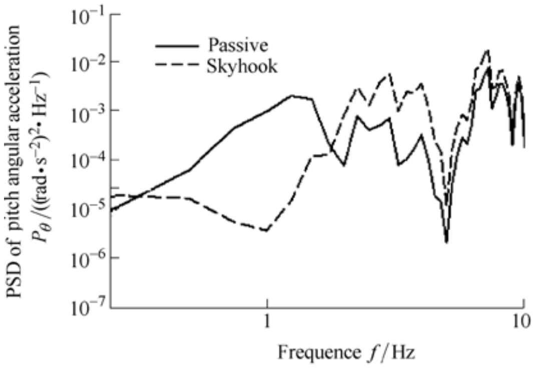
(1) According the modified Bouc-wen hysteretic F-v model of the MR damper proposed by authors, the established 7-DOF dynamic model comprising MR and passive full-vehicle suspensions puts forward an effective platform for the semi-active control study of MR full-vehicle suspension.

(2) Considering the distance between front and rear wheels in the real vehicle, the proposed harmonic, rounded pulse and

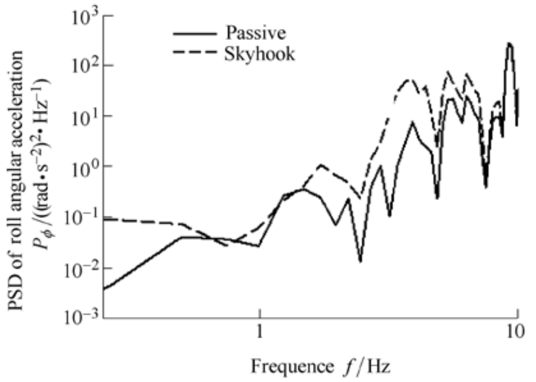
real road measured random excitations with delay time and asymmetric amplitudes set up a systematic analysis method for evaluating the multi-objective suspension performances of vertical, pitch and roll motions of MR full-vehicle suspension.



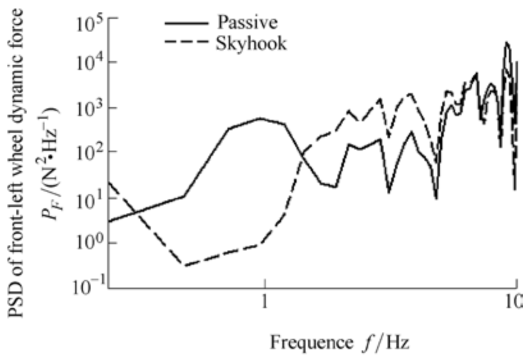
(a) Displacement acceleration of vehicle body gravity center



(b) Pitch angular acceleration



(c) Roll angular acceleration



(d) Dynamic force of front wheel at left side

Fig. 7. PSD response comparison of MR and passive full-vehicle suspensions under real road measured random excitations with delay time and asymmetric amplitudes

(3) Employing the proposed modified skyhook-based semi-active control strategy aimed at MR quarter-vehicle suspension, a skyhook-based asynchronous semi-active controller of MR full-vehicle suspension is synthesized for realizing independent control of the four MR quarter-vehicle sub-suspension systems, which has potential to comprehensively improve multi-objective suspension performances of the ride comfort and handling safety of full-vehicle.

(4) The above achievement establishes a solid fundamental for further study on decoupling semi-active control of the four quarter-vehicle suspensions in the MR full-vehicle suspension system.

References

[1] LEE H S, CHOI S B. Control and response characteristics of a magneto-rheological fluid damper for passenger vehicles[J]. *J. of Intelligent Material Systems and Structures*, 2000, 11(1): 80–87.

[2] YOKOYAMA M, HEDRICK J K, TOYAMA S. A model following sliding mode controller for semi-active suspension systems with MR dampers[C]//*Procs. of American Control Conference*, Arlington, USA, 2001, 4: 2652–2657.

[3] WANG Enrong, YING Liang, WANG Wanjun, et al. Semi-active control of vehicle suspension with MR-damper: Part I-Controller synthesis and evaluation[J]. *Chinese J. of Mechanical Engineering*, 2008, 21(1): 13–19.

[4] WANG Enrong, YING Liang, WANG Wanjun, et al. Semi-active control of vehicle suspension with MR-damper: Part II-Evaluation of suspension performance[J]. *Chinese J. of Mechanical Engineering*, 2008, 21(2): 52–59.

[5] WANG Enrong, YING Liang, WANG Wanjun, et al. Semi-active control of vehicle suspension with MR-damper: Part III-experimental validation[J]. *Chinese J. of Mechanical Engineering*, 2008, 21(4): 93–100.

[6] ZHU Maofei, CHEN Wuwei, ZHU Hui. Time-delay variable structure control for semi-active suspension based on magneto-rheological damper[J]. *J. of Mechanical Engineering*, 2010, 46(12): 113–120. (in Chinese)

[7] CHOI SB, LEE HS, PARK YP. H∞ control performance of a full-vehicle suspension featuring magneto-rheological dampers[J]. *Vehicle System Dynamics*, 2002, 38(5): 341–360.

[8] SPENCER B F, DYKE D J, SAIN K M, et al. Phenomenological model of a magneto-rheological damper[J]. *J. of Eng. Mech.*, 1997, 123(3): 230–238.

[9] WANG Enrong, MA Xiaoqing, RAKHEJA S, et al. Modeling hysteretic characteristics of an MR-fluid damper[J]. *J. of Automobile Engineering*, 2003, 217(D7): 537–550.

[10] YANG Moucun, YIN Chunbo. Semi-active suspension system design based on intelligent control strategy and stability analysis[J]. *J of Nanjing University of Aeronautics & Astronautics,* 2009, 41(3): 375–379.

[11] LI Rui, YU Miao, LIAO Changrong. Control of automotive suspensions vibration via magneto-rheological damper[J]. *Chinese J. of Mechanical Engineering*, 2005, 41(6): 128–132. (in Chinese)

[12] YU Fan, LIN Yi. *Dynamics of automotive system*[M]. Beijing: China Machine Press, 2005.

[13] WANG Wanjun, YING Liang, WANG Enrong. Comparison on hysteresis models of controllable magneto-rheological damper[J]. *J. of Mechanical Engineering*, 2009, 45(9): 100–108. (in Chinese)

[14] KARNOPP D C, CROSBY M J, HARWOOD R A. Vibration controlling semi-active generators[J]. *J. of Engg. for Industries*, 1974, 96, Sar. B(2): 619–626.

[15] ZHANG Zhiyong. *Proficient in Matlab*[M]. Beijing: Beijing University of Aeronautics and Astronautics Press, 2003.

[16] WANG Enrong. Syntheses and analyses of semi-active control algorithms for a magneto-rheological damper for vehicle suspensions[D]. Canada: Concordia University, 2005.

Biographical notes

ZHANG Hailong, born in 1988, is currently a master candidate at *School of Electric and Automation Engineering, Nanjing Normal University, China*. He received his bachelor degree from *Nanjing Normal University, China*, in 2010. His research interests include the semi-active control for implementing intelligent vehicle suspension with MR dampers.
E-mail: zhl_chosenone@163.com

WANG Enrong obtained his BS and MS degrees in 1985 and 1988, respectively, both from *Southeast University, China*, and obtained his PhD degree in 2006 from *Concordia University, Canada*. He joined *Nanjing Normal University, China* in 1988, where he is currently a professor and the dean of *School of Electric and Automation Engineering, Nanjing Normal University, China*. His research interests focus on the semi-active control for implementing intelligent vehicle suspension with MR dampers, and the related subjects in fields of electrical engineering and automation.
Tel: +86-25-85481043; E-mail: erwang@njnu.edu.cn

MIN Fuhong is currently a professor at *School of Electric and Automation Engineering, Nanjing Normal University, China*. Her research interests focus on the bifurcation and stability of discontinuous dynamic systems and its application in secure communication.
E-mail: minfuhong@njnu.edu.cn

SUBASH Rakheja is currently a professor at *Concordia University, Canada*. His research interests include advanced transportation systems and highway safety, human responses to workplace vibration, and driver-vehicle interactions.
E-mail: rakheja@vax2.concordia.ca

SU Chunyi is currently a professor at *Concordia University, Canada*. His research interests focus on automatic control theory and application about non-smooth dynamic system, hysteresis nonlinearities in smart actuators, robots.
E-mail: chunyi.su@gmail.com