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Semi-active control for vibration attenuation of vehicle suspension with symmetric MR damper

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Abstract: A semi-active force tracking PI controller is formulated and analyzed for a magnetorheological (MR) fluid-based damper in conjunction with a quarter-vehicle model. Two different models of the MR-damper are integrated into the closed-loop system model, which includes: a model based upon the mean force-velocity (f-v) behaviour; and a model synthesis comprising inherent nonsmooth hysteretic force and the force limiting properties of the MR damper. The vehicle models are analyzed to study the vibration attenuation performance of the MR-damper using the semi-active force tracking PI control algorithm. The simulation results are also presented to demonstrate the influence of the damper nonlinearity, specifically the hysteresis, on the suspension performance. The results show that the proposed control strategy can yield superior vibration attenuation performance of the vehicle suspension actuated by the controllable MR-damper not only in the sprung mass resonance and the ride zones, but also in the vicinity of the wheel-hop. The results further show that the presence of damper hystersis deteriorates the suspension performance.

Key words: magneto-rheological damper; vehicle suspension; hysteresis model; semi-active control

The automotive suspensions are designed to satisfy several conflicting requirements, such as adequate ride quality, road holding, and handling and control characteristics within the limited rattle space. A vast number of semi-active and active suspension systems have thus been explored to generate variable forces in accordance with the varying excitation and response variables in order to satisfy the various conflicting performance requirements. The studies on semi-active variable damping concepts, using conventional hydraulic dampers, have established that such dampers can effectively track the force generated by a fully active force generator when the force is of a dissipative nature, while the associated cost and hardware are considerably lower[1].

A new kind of semi-active controllable MR fluid-based damper has been commercially developed for vehicle suspension applications[2]. The MR-damper offers high viscous damping corresponding to low velocities in the pre-yield condition, while the post-yield saturation corresponding to high velocities can be characterized by a considerably lower viscous damping coefficient[3,4]. Such damping properties are considered to be well suited for vibration attenuation applications to achieve satisfactory compromise among different conflicting requirements. However, the MR-damper exhibits highly nonlinear variations in damping attributed to hysteresis and force-limiting as functions of the intensity of the magnetic field, displacement and velocity of the piston, and nature of excitation. Recent modeling work has been focused on these aspects. So far, a number of models have been evolved to characterize the hysteretic properties of the MR-damper[3-7]. Furthermore, the study of semi-active control using the MR-damper on vehicle suspension has also become a new challenging work since the MR-damper exhibits strong hysteresis and force-limiting nonlinearities. These nonlinearites have been proved to be harmful to system performance, and even to induce system instability[8]. Studies on the semi-active controller synthesis by employing the MR-damper have recently been reported[5,7].

In this study, on the basis of the proposed generalized hysteresis model of an MR-damper[3,4], an initial study on a semi-active control design and evaluation for vehicle suspension is conducted by employing a "quarter-car" suspension system. A semi-active force tracking PI control strategy is developed for the vehicle suspension vibration attenuation. The effects of nonlineaities, owing to the hysteresis and force saturation of the MR-damper, are thoroughly investigated.

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1 Hysteresis Model of an MR-Damper

Fig.1 illustrates a schematic of an MR-damper, comprising electro-magnetic coils and bleed orifices within the piston, and gas and hydraulic chambers within the cylinder that are separated by a diaphragm. The generalized force versus velocity (f-v) characteristics of an MR-damper are characterized by the generalized hysteretic loop shown in Fig.2. A hysteresis model synthesis was formulated to characterize the hysteretic and nonlinearly symmetric damping in the form of a nonlinear algebraic function upon consideration of the observed physical features, such as pre-yield behavior, post-yield force saturation, transition velocity and force, and control current dependence. The proposed model is expressed as a function of the relative velocity, the control current and the nature of vibration, and suits for a wide range of excitation frequency[3,4]. The model can be reviewed as

$$f_{\mathrm{d}}(i,v) = f_{\mathrm{t}}(i)\frac{1 - \mathrm{e}^{-\alpha(v+v_{\mathrm{h}})}}{1 + \mathrm{e}^{-\alpha(v+v_{\mathrm{h}})}}(1 + k_{\mathrm{v}}|v|) \quad (1)$$

where $f_{\mathrm{d}}(i,v)$ denotes the damping force in function of control current i and piston velocity v; v_{h} is the piston velocity corresponding to zero damping force,

$$v_{\mathrm{h}} = \mathrm{sgn}(\ddot{x})k_4 v_{\mathrm{m}}\left(1 + \frac{k_3}{1 + \mathrm{e}^{-a_3(i+I_1)}} - \frac{k_3}{1 + \mathrm{e}^{-a_3 I_1}}\right)$$

f_{t} is the transition force, which strongly depends upon the control current i and the peak velocity v_{m},

$$f_{\mathrm{t}}(i) = f_0(1 + \mathrm{e}^{a_1 v_{\mathrm{m}}})\left(1 + \frac{k_2}{1 + \mathrm{e}^{-a_2(i+I_0)}} - \frac{k_2}{1 + \mathrm{e}^{-a_2 I_0}}\right)$$

The parameters k_{v} and α, used to adjust the damping coefficients at high and low velocities, are expressed as functions of the peak velocity v_{m}, respectively.

$$k_{\mathrm{v}} = k_1 \mathrm{e}^{-a_4 v_{\mathrm{m}}}$$

$$\alpha = \frac{a_0}{1 + k_0 v_{\mathrm{m}}}$$

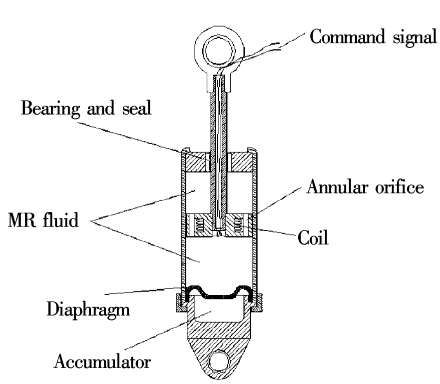

Fig.1 Configuration of an MR-damper

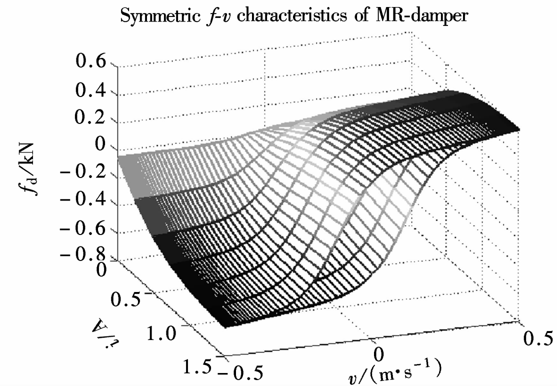

Fig.2 Symmetric hysteresis characteristics $f_{\mathrm{d}}(i,v)$

The model described in Eq.(1) requires identification of a total of 13 parameters from the measured data, and can be easily simplified to yield mean symmetric f-v characteristics by letting $k_4 = 0$.

The peak velocity parameter v_{m} can be estimated from the instantaneous position responses of displacement x, velocity $\dot{x}(v = \dot{x})$, and acceleration $\ddot{x}$. For harmonic excitation, parameter v_{m} is obtained from

$$v_{\mathrm{m}} = a_{\mathrm{m}}\omega = \sqrt{(\dot{x})^2 - \ddot{x}x} \quad (2)$$

where a_{m} and ω are amplitude and frequency of a sinusoidal excitation signal, respectively.

The model parameters for a particular damper are identified with the total measured data over a wide range of frequency 0 to 20 Hz and may refer to Refs. [3,4].

2 Semi-Active Force Tracking PI Controller

The controller synthesis for vehicle suspension, employing the MR-damper, is quite a challenging task owing to the hysteresis and force-limiting of the damper. The semi-active control strategy[9] indicates that if the suspension spring force takes the same direction as the damping force, then the damping force should be a minimum to reduce the sprung mass acceleration. Similarly, when the spring force and damping force are in opposite directions, the damping force should be adjusted to equal the spring force in magnitude to produce zero acceleration for the sprung mass. Using the above control strategy, in this paper a semi-active force tracking PI control law is developed for vehicle suspension vibration attenuation, where the PI control law serves for equaling the two forces yielded in the damper and suspension string when their directions are opposite.

For the purpose of the controller synthesis, a "quarter-car" vehicle suspension system actuated by an

MR-damper is employed as shown in Fig.3. Therein, the input x_i represents a displacement as a representative of a typical road profile; m_u is the unsprung mass representing the wheel, tire, and some suspension components; k_t and c_t denote the tire stiffness and damping properties, respectively; m_s represents the sprung mass; x_s and x_u denote the absolute displacement of the sprung and unsprung masses, respectively. The model parameters used are: m_s = 288.9 kg; m_u = 28.6 kg; k_s = 19.96 kN/m; k_t = 155.9 kN/m; and c_t = 100 N·s/m[10].

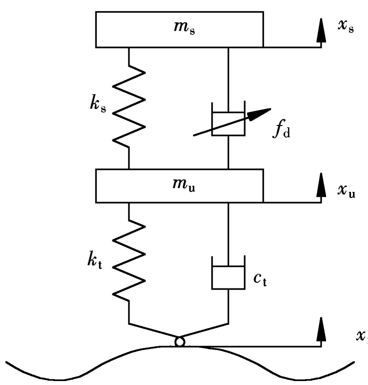

Fig.3 2-DOF "quarter-car" vehicle model

The controller synthesis, in corresponding to the "quarter-car" vehicle model, can then be described as

$$i(t) = \begin{cases} 0 & \text{if } (\dot{x}_s - \dot{x}_u)(x_s - x_u) > 0 \\ k_p(f_d(t) + f_k(t)) + k_i\int_0^t (f_d(\tau) + f_k(\tau))\mathrm{d}\tau & \text{if } (\dot{x}_s - \dot{x}_u)(x_s - x_u) < 0 \end{cases} \quad (3)$$

where $i(t)$ denotes the control current derived from the controller, $i(t) \in \{i(t) | 0 \leqslant i \leqslant 1.5\ \text{A}\}$ for the operation safety of the MR-damper, $f_d(t)$ is the damping force yielded by the MR-damper model of Eq.(1), $f_k(t) = k_s(x_s - x_u)$ is the suspension spring force in the "quarter-car" system, and k_p and k_i stand for the proportional and integral gains of the PI control law, respectively.

The force tracking error $e_f(t)$ is defined as

$$e_f(t) = f_d(t) + f_k(t) \quad (4)$$

It should be noted that the above error definition is different from the traditional error definition due to the semi-active control strategy[9]. The whole controller holds the advantage in real implementation because only relative position and velocity information of the MR-damper piston are needed.

3 Validation of the Controller

The above-described controller expressed as Eq.(3), based on the relative suspension signals, is simply referred to as the semi-active force tracking PI control. In this section, the performance of the control scheme is evaluated by simulation experiment, in which the control gains k_p and k_i are chosen as 5×10^{-4} and 3×10^{-2}, respectively.

Fig.4 shows the acceleration transmissibility responses of the sprung mass, in responding to the closed-loop system and the open-loop system with constant control current 0.2 A and 0.5 A. The system is uniformly excited by a single harmonic signal with amplitude 2.5 cm, and the transmissibility magnitudes are evaluated from the peak to peak response magnitudes over the interested spectrum (0 to 20 Hz) of the grounded-vehicle[3,4]. The results clearly show that the open-loop system yields poor sprung natural resonant attenuation and the proposed semi-active force tracking PI control can effectively suppress the sprung mass resonant response (around 1.2 Hz and 10.5 Hz), yielding superior vibration isolation. It thoroughly reveals that the MR-damper is an ideal semi-active controllable actuator for the vehicle suspension design. It should be mentioned that the control performance shown in Fig.4 achieves the requirement of adequate ride comfort[10].

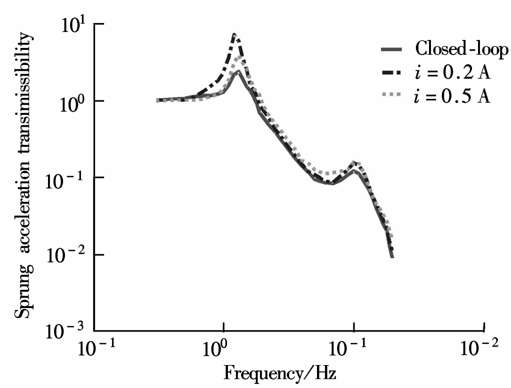

Fig.4 Transmissibility of sprung mass acceleration

To obtain the performance expressed in Fig.4, Fig.5 shows the steady responses of the closed-loop system with the proposed control scheme, wherein the excitation signal is fixed as a periodical sinusoidal signal with amplitude 2.5 cm and frequency 2.0 Hz. Amongst, Fig.5(a) shows the damping force yielded by the MR-damper, which is governed by the control current as shown in Fig.5(b). The current regulation responds to the control logic depicted in Fig.5(c), which is deduced from the signs of relative displacement and relative velocity as shown in Fig.5(d) and Fig.5(e). It is easy to check the correctness that the controller is governed by the control condition described in Eq.(3). Fig.5(f) shows the force tracking error of the controller as defined in Eq.(4).

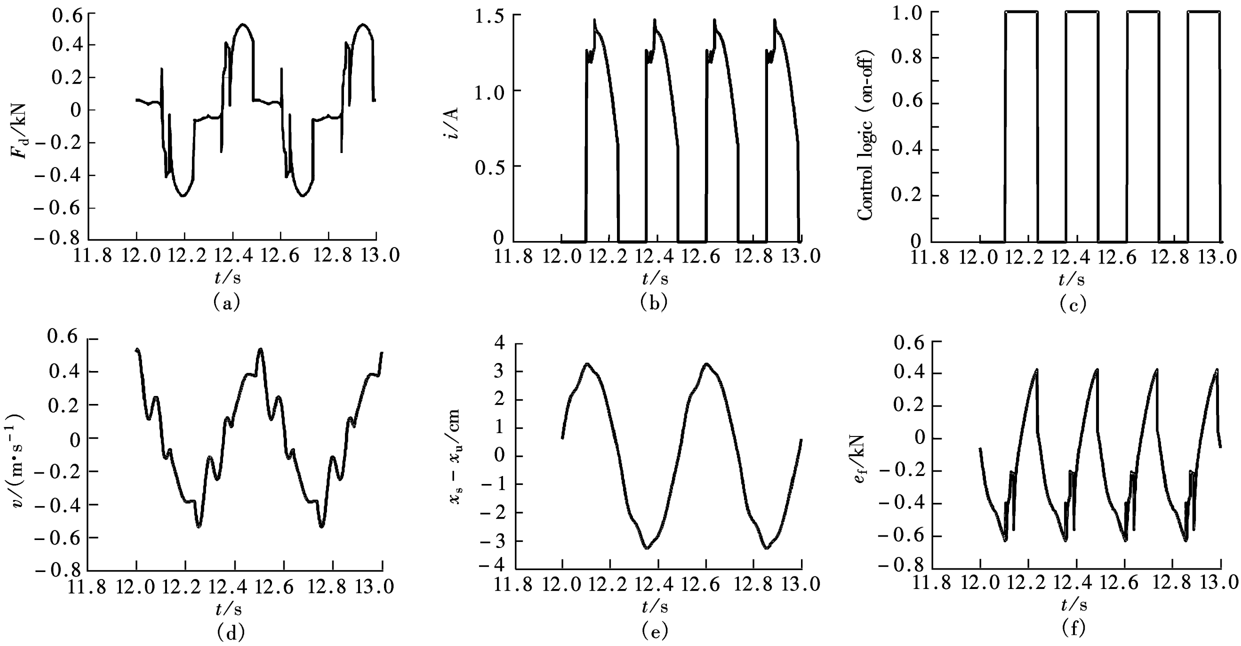

Fig.5 Steady state performance of the closed-loop system. (a) Time history of damping force; (b) Time history of control current; (c) Time history of control logic; (d) Time history of relative suspension velocity; (e) Time history of relative suspension displacement; (f) Time history of force tracking error

Fig.6 further shows the transient regulation performance of the dynamic system governed by the proposed control scheme. The vibration signal as shown in Fig.6(a) is emulated by a round step signal with final value of 0.25 m, which is also polluted by a random noise with peak value of 2.5 cm. Fig.6(b) shows the transient responses of the sprung mass of the open and closed-loop systems, a good transient regulation performance for the semi-active force tracking PI control can be concluded from the comparison of the results.

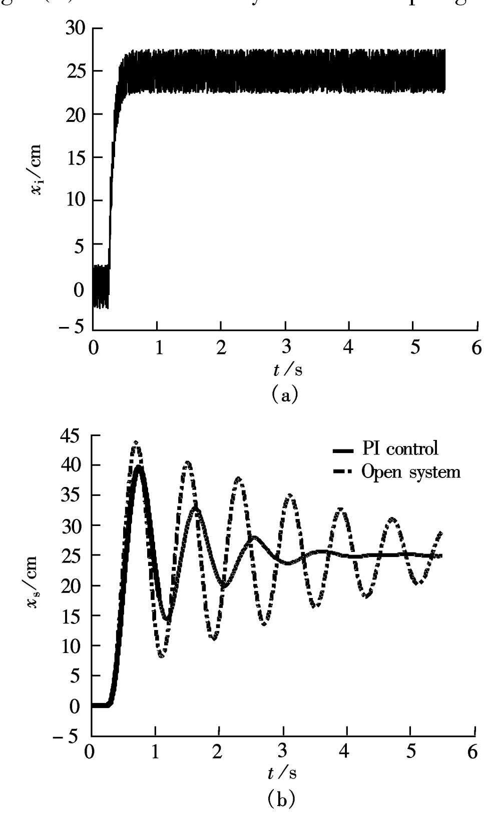

Fig.6 Transient performance of open and closed-loop system. (a) Vibration excitation signal; (b) Transient displacement response of sprung mass

4 Analysis of Hysteresis Effect

As is well established, the nonlinear property may cause serious damage to the system performance[8], such as self-excitation, instantaneous noise shoot, and may even lead to instability of the system. The controlled vehicle suspension system actuated by the MR-damper inevitably, exhibits strong nonlinearities due to the contributions of the hysteresis and force saturation properties of the MR-damper, as well as the semi-active control. Herein, the effects of the nonlinearities are thoroughly investigated with the help of the force tracking PI controller.

To take insight into the hysteresis effect on the system performance, the mean damping behavior (by letting the parameter $k_4 = 0$, i.e. $v_h = 0$ in the model expressed in Eq.(1)) is compared with that of the hysteresis damping. Fig.7 shows the simulation results of the closed-loop system with mean and hysteresis damping under sinusoidal excitation with an amplitude of 2.5 cm and frequency 1.5 Hz. Fig.7(a) describes the time history of damping force comparison, and Fig.

7(b) describes the time history of sprung mass acceleration, which is a key evaluation index of the vehicle suspension performance[10]. It shows that the responses of hysteresis damping features heavy instantaneous noise peak shoot, while the responses of meaning damping appears more smooth. The instantaneous noise peak shoot would surely cause bad effect on the dynamic performance of the system, and further measures have to be employed to overcome the shortage of hysteresis. Apart from the hysteresis effect, the nonliearities, attributed to the damping force saturation and "on-off" property of the semi-active control scheme, further cause discontinuities of the responses, which can be distinctly observed from the responses of mean damping.

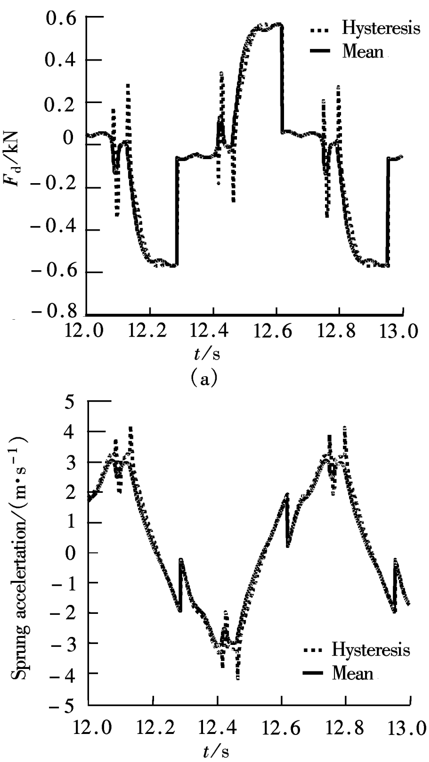

Fig.7 Comparison of the closed-loop system with mean and hysteresis damping. (a) Time history of damping force; (b) Time history of sprung mass acceleration

5 Conclusions

A semi-active force tracking PI control scheme is developed for the vehicle suspension vibration attenuation with controllable MR-damper. The simulation results verified that the synthesized controller could achieve superior attenuation performance of vehicle suspension. The proposed controller benefits in real implementation because only relative suspension signals are needed; these are facilely obtained by sensors. Moreover, the effects of nonlinearities, owing to the hysteresis and force saturation of the MR-damper, are thoroughly investigated. Since the semi-active control using the MR-damper on vehicle suspension is still an open problem, extensive studies are required to complete the controller design for practical application, such as compensation of hysteresis and nonlinearities, analysis of "full-car" dynamic system, adaptive control, etc.

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由 MR 阻尼器驱动的汽车悬架减振系统的半主动控制研究

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摘　要　设计了一种半主动力跟踪 PI 控制器，对由磁流液(MR)阻尼器作为悬架阻尼器的汽车模型进行控制分析.MR 阻尼器的 2 种不同模型在闭环的汽车悬架控制系统模型中得到了应用.2 种模型是基于均值阻尼力对速度(f-v)特性的均值 f-v 模型，和描述阻尼力非光滑滞环和饱和特性的滞环 f-v 模型.汽车模型用来研究力跟踪 PI 控制算法和 MR 阻尼器对车辆的振动抑制性能.仿真分析还指出了 MR 阻尼器的非线性，特别是滞环特性对汽车悬架系统的性能影响.结果表明所提出的控制方法对由 MR 阻尼器驱动的汽车悬架系统能产生很好的减振效果，不仅体现在对悬架弹簧支撑车厢的共振抑制和对驾乘人员舒适性敏感频域的振动抑制，还体现在对汽车轮胎共振频率周围的振动抑制.结果还进一步说明了 MR 阻尼器所存在的滞环特性对汽车悬架性能的不良影响.

关键词　磁流液阻尼器；汽车悬架；磁环模型；半主动控制

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