Variable damping control strategy of a semi-active suspension based on the actuator motion state

Mingde Gong and Hao Chen

Abstract
A semi-active suspension variable damping control strategy for heavy vehicles is proposed in this work. First, a nine-degree-of-freedom model of a semi-active suspension of heavy vehicles and a stochastic road input mathematical model are established. Second, using a 1/6 vehicle as an example, a semi-active suspension system with damping that can be adjusted actively is designed using proportional relief and throttle valves. The damping dynamic characteristics of the semi-active suspension system and the time to establish the damping force are studied through a simulation. Finally, a variable damping control strategy based on an actuator motion state is proposed to adjust the damping force of the semi-active suspension system actively and therefore satisfy the vibration reduction requirements of different roads. Results show that the variable damping control suspension can substantially improve vehicle ride comfort and handling stability in comparison with a passive suspension.

Keywords
Heavy vehicles, semi-active suspension, damping dynamic characteristics, variable damping control

Introduction
Suspension system is the general term for all force transfer devices between the frame and the axle of a vehicle. This system transfers force and torque between the wheel and the frame and buffers impact loads transmitted from an uneven road surface to the frame or body, thereby attenuating the vertical vibration of the body and ensuring ride comfort, driving harshness and handling stability.

Three types of suspension, namely, passive, semi-active and active, are commonly used for heavy vehicles. The stiffness and damping of the passive suspension cannot be adjusted automatically with the change in driving speed and road conditions and cannot achieve the desired performance under various working conditions. Although the active suspension has strong adaptability and evident improvement of ride comfort and stability, it requires external energy supply which is more complex and costly than the semi-active suspension. A semi-active suspension is extensively used in heavy vehicles given its low cost, simple manufacturing process and favourable damping effect.

The stiffness of a spring is difficult to adjust. Thus, most semi-active suspensions are realised by changing the damping; thus, a variable-damping damper is the most important actuator in a semi-active suspension system. Variable damping dampers are classified into magnetorheological, electrorheological and solenoid valve. Among these damper types, the electromagnetic valve-type variable damping shock absorber has the most compact structure, quick response and reliable performance.
Guy, Ivers and Miller and Rajamani and Hedrick confirmed experimentally that a continuously controlled damping suspension can improve the ride comfort and road adhesion performance of a vehicle better than a traditional suspension. Besinger et al. conducted a hardware-in-the-loop test of vehicles equipped with continuously adjustable dampers. The test verified that the vibration acceleration of a vehicle body and the dynamic load of a tire are reduced by 28% and 21%, respectively. Many scholars have conducted performance analyses and experimental studies of an adjustable damping shock absorber. A suspension with controllable damping is used in extant research for light vehicles, but research on the suspension with adjustable damping for heavy vehicles is rare.

Many control methods have been used for a semi-active suspension of heavy vehicles. Sulaiman et al. and Valašek et al. studied the semi-active suspension groundhook control of heavy vehicles. Their findings showed that a groundhook control can effectively reduce the dynamic load of tires and improve driving comfort. Yarmohamadi and Berbyuk tested the control strategies of the semi-active suspension of a heavy vehicle; these strategies include groundhook, skyhook and groundhook–skyhook control strategies; the effect of a semi-active damper on vehicle dynamic performance was quantitatively displayed. Nicolas et al. expounded the application of a fuzzy logic method in a vehicle’s semi-active suspension and analysed its advantages over conventional control method. Salah designed a neuro-fuzzy controller to improve the damping and ride comfort of a semi-active vehicle suspension system. The simulation results showed that the proposed controller has reduced suspension travel and has improved ride quality. Nguyenf et al., Zheng et al. and Song et al. studied the fuzzy sliding mode controller of a semi-active suspension to provide improved vibration control capability with low power consumption. The whole vehicle control effect is disregarded despite adopting various control methods in the semi-active suspension system of heavy vehicles.

First, a nine-degree-of-freedom (9-DOF) model of the semi-active suspension system of heavy vehicles is established in the present work. Second, a variable damping semi-active suspension is designed for heavy vehicles. The damping dynamic characteristics of the semi-active suspension system and the time to establish damping force are investigated through a simulation. Variable damping control (VDC) strategy is used to control a heavy vehicle, and the control effect is analysed. Results show that, in comparison with the passive suspension, the VDC suspension can considerably reduce the vibration of body acceleration, pitch angle acceleration, roll angle acceleration, suspension deflection and tire deflection. Moreover, the VDC suspension has an improved control effect on vehicle ride comfort and handling stability.

**Semi-active suspension system dynamics**

The 9-DOF semi-active suspension system model of a heavy vehicle is illustrated in Figure 1. This model consists of six vertical unsprung masses and DOFs due to pitch, roll and vertical motion of the mass centre. The variables of the semi-active suspension model are listed in Table 1.
According to Newton’s second law, we can obtain the vertical motion of the body centroid, body pitching and roll rotation equations as follows:

\[
\begin{align*}
\ddot{m}z &= -F_i - F_2 - F_3 - F_4 - F_5 - F_6 \\
J_y \dot{\theta} &= \frac{l}{2} (F_1 + F_2 + F_3) - \frac{l}{2} (F_4 + F_5 + F_6) \\
J_x \ddot{\phi} &= -F_1 c + F_2 a + F_3 b - F_4 c + F_5 a + F_6 b
\end{align*}
\]

(1)

where \( F_i = K_i (z_i - z_{ui}) - U_i \), \( i = 1, 2, 3, 4, 5, 6 \); \( U_i \) is the output force of variable damping; the relationship between \( U_i \) and \( c_{mi} \) is discussed in the 'Mathematical model of the system'.

The dynamic equation of the vertical motion of the unsprung mass is

\[
m_{ui} \ddot{z}_{ui} = K_i (z_i - z_{ui}) - U_i - K_{ti} (z_{ui} - q_i)
\]

(2)

According to the spatial motion law of a rigid body, the dynamic relationship among the four suspension systems, the body connection points, body centroid vertical movement, pitching rotation and roll rotation can be demonstrated as

\[
\begin{align*}
z_1 &= z - \frac{l}{2} \sin \theta + c \sin \phi \\
z_2 &= z - \frac{l}{2} \sin \theta - a \sin \phi \\
z_3 &= z - \frac{l}{2} \sin \theta - b \sin \phi \\
z_4 &= z + \frac{l}{2} \sin \theta + c \sin \phi \\
z_5 &= z + \frac{l}{2} \sin \theta - a \sin \phi \\
z_6 &= z + \frac{l}{2} \sin \theta - b \sin \phi
\end{align*}
\]

(3)

Table 1. Variables of the semi-active suspension model (\( i = 1, 2, 3, 4, 5, 6 \)).

| Symbol | Quantity |
|--------|----------|
| M      | Mass of the vehicle body |
| \( m_{ui} \) | Unsprung masses |
| Z      | Vertical displacement of the vehicle body |
| \( z_i \) | Displacements between the vehicle body and unsprung masses |
| \( z_{ui} \) | Unsprung mass displacements |
| \( q_i \) | Road displacements |
| L      | Wheelbase |
| A      | Distance between the centre of mass and the first rear axle |
| B      | Distance between the centre of mass and the second rear axle |
| C      | Distance between the centre of mass and the front axle |
| \( \Theta \) | Pitch angle of a vehicle body |
| \( \psi \) | Roll angle of a vehicle body |
| \( J_x \) | Body moment of inertia of the X-axis |
| \( J_y \) | Body moment of the inertia of the Y-axis |
| \( K_i \) | Stiffness coefficients of springs |
| \( K_{ti} \) | Stiffness coefficients of tires |
| \( c_{mi} \) | Variable damping coefficients |
| \( v \) | Vehicle speed |
As a rigid body structure, the pitch and roll angles of the vertical body assume a change in a small angle range. Furthermore, the following equations are set

\[ \theta \approx \sin \theta, \quad \varphi \approx \sin \varphi \] (4)

The state equation is established, and the system state variable \( X \) is defined as

\[ X = \begin{bmatrix} \ddot{z} & \dot{\theta} & \dot{\varphi} & z_{u1} & z_{u2} & z_{u3} & z_{u4} & z_{u5} & z_{u6} & \dot{z}_{u1} & \dot{z}_{u2} & \dot{z}_{u3} & \dot{z}_{u4} & \dot{z}_{u5} & \dot{z}_{u6} \end{bmatrix} \] (5)

The semi-active suspension system can be expressed in the following state equation

\[ \dot{X} = AX + BU + HQ \] (6)

where \( U \) is the control vector that represents the output force of the variable damping of each suspension subsystem; \( Q \) is the road input vector of six tires and \( A, B \) and \( H \) are divided into coefficient matrices.

System output \( Y \) is defined as

\[ Y = \begin{bmatrix} \ddot{z} & \dot{\theta} & \dot{\varphi} & z_1 - z_{u1} & z_2 - z_{u2} & z_3 - z_{u3} & z_4 - z_{u4} & z_5 - z_{u5} & z_6 - z_{u6} \end{bmatrix} \] (7)

The output equation of the system is

\[ Y = CX + DQ \] (8)

where \( C \) and \( D \) are the coefficient matrices.

**Random road output model**

Road roughness is the most important factor that affects vehicle ride comfort which is typically used to describe the roughness degree of pavements. This factor makes the vehicle produce driving resistance and vibration and affects the ride comfort, handling stability, fatigue life of components and other aspects. The statistical characteristics of pavement roughness are frequently expressed by power spectral density function \( G_q(n) \)

\[ G_q(n) = G_q(n_0) \left( \frac{n}{n_0} \right)^{-\omega} \] (9)

where \( n \) is the spatial frequency, \( n_0 \) indicates the standard spatial frequency, \( G_q(n_0) \) is the road roughness coefficient and \( \omega \) is the frequency index which determines the frequency structure of road power spectral density.

When \( \omega = 2 \)

\[ G_q(f) = G_q(n_0) \left( \frac{n_0}{f} \right)^2 v \] (10)

The velocity power spectral density function is

\[ \dot{G}_q(f) = 4\pi^2 G_q(n_0)^2 v \] (11)

The time-domain mathematical model of road excitation on a single wheel is described as follows

\[ q(t) + 2\pi f_0 q(t) = 2\pi n_0 \sqrt{G_q(n_0)} \omega(t) \] (12)

where \( q(t) \) indicates the random excitation of the road on the wheel, \( \omega(t) \) indicates the unit Gaussian white noise and \( f_0 \) is the road spatial cut-off frequency.
Structure and characteristics

Structure and working principle of the semi-active suspension system

Using the right front suspension as an example, a diagram of the variable damping semi-active suspension structure for heavy vehicles is depicted in Figure 2.

A vehicle is motivated by the road surface whilst running, and a relative motion between the wheel and the vehicle body occurs. With the piston moving downwards, the suspension is in a compression stroke. The oil in the cylinder can be pressed into the accumulator through the proportional relief and throttle valves. These valves work under different piston speeds. The system pressure becomes low alongside the piston speed. Thus, the proportional relief valve is closed, and the oil enters the accumulator only through the throttle valve. When the piston accelerates, the system pressure increases until the proportional relief valve can open. Therefore, the oil enters the accumulator through both valves. The suspension is in the stretching stroke whilst the piston moves upwards. The oil in the accumulator can be sucked into the cylinder through the proportional relief and throttle valves. The working principle is the same as the compression stroke.

The opening pressure of the proportional relief valve is controlled by the voltage signal from the controller, and the voltage range is 0–10 V. The opening pressure of the proportional relief valve is directly proportional to voltage and can be regulated continuously by the voltage signal.

Mathematical model of the system

Hypotheses: (1) oil is incompressible; (2) the effect of temperature on oil characteristics is ignored; (3) the system has no oil leakage.

The flow formula for throttle valve is

\[ Q_1 = C_d A_1 \sqrt{\frac{2}{\rho} \Delta p} \]  

(13)

where \( Q_1 \) is the flow rate through throttle valve; \( C_d \) is the flow coefficient; \( A_1 \) is the area of the throttle orifice, m\(^2\); \( \rho \) is the oil density; \( \Delta p \) is the pressure difference between the two ends of the valve.

The valve core of the proportional relief valve can adopt the mathematical model of a poppet valve. The opening pressure is proportional to the control voltage. The input voltage on the electromagnet can change from 0 to 10 V, and the force produced will change accordingly. Thus, the continuous change in the relief pressure can be obtained.

Figure 2. 1/6 Vehicle structure diagram of an adjustable damped semi-active suspension.
The flow formula for the proportional relief valve is as follows

$$Q_2 = \begin{cases} 
0, & \Delta p < \frac{V}{10} \Delta p_{\text{max}} \\
C_d \pi d y_{\text{max}} \sqrt{\frac{2}{\rho} (\Delta p)}, & \Delta p \geq \frac{V}{10} \Delta p_{\text{max}} 
\end{cases}$$

(14)

where $Q_2$ is the flow rate through the proportional relief valve; $d$ is the diameter of the proportional relief valve port; $y_{\text{max}}$ is the maximum opening of the proportional relief valve; $V$ is the voltage signal.

When the piston speed is low, the proportional relief valve remains closed. In addition, the oil only flows through the throttle valve. The flow rate that flows out of the hydraulic cylinder is equal to the flow rate at the throttle valve. The mathematical model of the system in this state is as follows

$$\begin{cases} 
Q = A_p v_p \\
Q = A_1 v_1 \\
Q = C_d A_1 \sqrt{\frac{2}{\rho} (\Delta p)} 
\end{cases}$$

(15)

Thus

$$C_{m1} = \frac{\rho A_p^2 v_p}{2 C_d A_1}$$

(17)

$$U_1 = \frac{\rho A_p^2 v_p^2}{2 C_d A_1^2}$$

(18)

where $U_1$ is the variable damping output force for the right front suspension; $C_{m1}$ is the variable damping coefficient for the right front suspension; $v_p$ is the speed of the cylinder piston; $A_p$ is the area of the cylinder piston; $v_1$ is the speed that flows through the throttle valve.

The pressure difference between the two ends of the proportional relief valve increases alongside the piston speed. The valve opens when the pressure difference reaches the opening pressure of the valve. The mathematical model of the oil that flows through the throttle and proportional relief valves is presented as follows

$$\begin{cases} 
U_1 = \Delta p A_p \\
U_1 = C_{m1} v_1 
\end{cases}$$

(16)

$$\begin{cases} 
Q = Q_1 + Q_2 \\
Q_1 = C_d A_1 \sqrt{\frac{2}{\rho} (\Delta p)} \\
Q_2 = C_d \pi d y_{\text{max}} \sqrt{\frac{2}{\rho} (\Delta p)} \\
Q = A_p v_p \\
Q = (A_1 + \pi d y_{\text{max}}) v_e 
\end{cases}$$

(20)
Thus

\[
C_{m1} = \frac{\rho A_p^2 v_p}{2C_d^2(A_1 + \pi dy_{\text{max}})}
\]  
(21)

\[
U_1 = \frac{\rho A_p^2 v_p^2}{2C_d^2(A_1 + \pi dy_{\text{max}})^2}
\]  
(22)

where \( v_e \) is the equivalent speed.

In summary, the expression of the equivalent damping coefficient is

\[
C_{m1} = \begin{cases} 
\frac{\rho A_p^2 v_p}{2C_d^2A_1}, & \Delta p < \frac{V}{10}\Delta p_{\text{max}} \\
\frac{\rho A_p^2 v_p}{2C_d^2(A_1 + \pi dy_{\text{max}})}, & \Delta p \geq \frac{V}{10}\Delta p_{\text{max}}
\end{cases}
\]  
(23)

The expression of the damping force is

\[
U_1 = \begin{cases} 
\frac{\rho A_p^2 v_p^2}{2C_d^2A_1}, & \Delta p < \frac{V}{10}\Delta p_{\text{max}} \\
\frac{\rho A_p^2 v_p^2}{2C_d^2(A_1 + \pi dy_{\text{max}})^2}, & \Delta p \geq \frac{V}{10}\Delta p_{\text{max}}
\end{cases}
\]  
(24)

where

\[
\Delta p_{\text{max}} = \frac{\rho}{2} \left( \frac{A_p v_{\text{ymax}}}{A_1 C_d} \right)^2
\]  
(25)

**Simulation of the system characteristics**

The system is modelled and simulated using MATLAB/SIMULINK. The simulation and complete vehicle parameters are listed in Tables 2 and 3. The damping coefficient control characteristic, damping force control characteristic and damping force time curves are demonstrated in Figures 3 to 5.

The damping characteristic control curve is exhibited in Figure 3. The black curve is the damping coefficient control characteristic curve of the suspension system without adding voltage to the proportional relief valve. The curves from left to right are the control characteristics of the damping coefficient of the suspension system with the conditions that the proportional relief valve is added with 1–10 V. When the piston speed is low, the oil

| Parameter | Value |
|-----------|-------|
| Oil density (kg/m³) | 870 |
| Flow coefficient | 0.6 |
| Throttle valve orifice area (m²) | 0.00003 |
| Diameter of the proportional relief valve orifice (m) | 0.01 |
| Max opening of the proportional relief valve orifice (m) | 0.003 |
| Cylinder piston area (m²) | 0.0057 |
| Maximum speed of the piston rod (m/s) | 0.3 |
only flows through the throttle valve. In addition, the damping force is generated by the throttle valve because the slope of the curve enlarges accordingly. When the piston speed increases, the oil flows through the throttle and proportional relief valves. Thus, the damping force is generated by the two valves together, and the slope of the curve becomes small.

The damping force characteristic control curve is displayed in Figure 4. The black curve represents the damping force control characteristic curve of the suspension system without adding voltage to the proportional relief valve. The curves from left to right are the control characteristics of the damping force of the suspension system when the proportional relief valve is added with 1–10 V.

The time curve for establishing damping force is presented in Figure 5. In the curve, the damping force increases from time 0 to a certain time and surges to the peak value of the damping force at this certain time. Therefore, this time quantum is required to establish the damping force completely under any voltage. In addition, a large control voltage indicates a long time required to establish the damping force.

The simulation curve, we can control the damping characteristics of the semi-active suspension system by controlling the input voltage of the proportional relief valve.

The control signals of each suspension can be obtained by simulating the 9-DOF vehicle model. To present the verification results of the model, the control signal of the front axle is illustrated in Figure 6.

| Parameter | Value |
|-----------|-------|
| $m$       | 10,000 kg |
| $m_u$     | 100 kg |
| $l$       | 1.692 m |
| $a$       | 1.675 m |
| $b$       | 3.325 m |
| $c$       | 3.325 m |
| $J_x$     | 700 kg m² |
| $J_y$     | 3000 kg m² |
| $K_0$     | 15,000 N/m |
| $K_{ni}$  | 250,000 N/m |
| $v$       | 10 m/s |
| $f_0$     | 0.0628 Hz |
| $n_0$     | 0.1/m |
| $G_{eq}(n_0)$ | $256 \times 10^{-6}$ m² |

**Figure 3.** Damping coefficient control characteristic curve.
Figure 4. Damping force control characteristic curve.

Figure 5. Damping force time curve.

Figure 6. Control signal of the front axle.
Experiment and analysis of the VDC strategy

We propose a VDC strategy on the basis of an analysis of the damping characteristics of a semi-active suspension with variable damping in the fourth part. The displacement sensor is used to identify the motion state of the hydraulic cylinder. When the hydraulic cylinder is in the state of stretching, the control voltage is increased to improve the damping switching threshold of the system. Thus, the suspension is in a high damping state, and the vibration can be reduced quickly. When the hydraulic cylinder is in a compression state, the control voltage is reduced to not only decrease the damping switching threshold of the system but also maximise the elastic elements for moderating the impact. If the speed of the cylinder reaches the upper limit, then the control voltage must be reduced to increase the flow rate of the shock absorber. Thus, the damping force is constantly kept within a certain limit to avoid excessive impact and load. The VDC flowchart is demonstrated in Figure 7.

Using the road of Class C as an example, the control strategy is used to control the whole vehicle. The heavy vehicle used in this work is depicted in Figure 8. The 1/6 variable damping semi-active suspension is exhibited in Figure 9. To verify the effect of the VDC suspension proposed in this work, the heavy vehicle was driven to the test site displayed in Figure 10. In contrast to the traditional passive suspension, the validity of the control strategy is verified.

The curves show that the semi-active suspension with VDC has better control effect than the traditional passive suspension on vehicle ride comfort and handling stability. In comparison with passive suspension, the VDC
suspension can significantly reduce the vibration of body acceleration, pitch angle acceleration, roll angle acceleration, suspension deflection and tire deflection (Figures 11 to 17).

In Figure 11, the acceleration vibration of the vehicle body with a VDC suspension is considerably reduced in comparison with the vehicle body with a passive suspension system. Moreover, the results show that the peak acceleration of the vehicle body with the passive and VDC suspension systems are 2.91 and 1.38 m/s², correspondingly. In Figures 12 and 13, the pitch and the roll accelerations of the vehicle body are better with the VDC
suspension than with the passive suspension. In Figure 11, the peak pitch accelerations of the vehicle body with the passive and VDC suspension systems are 5.22 and 2.49 rad/s$^2$, respectively. In Figure 12, the peaks of the roll angle acceleration of the vehicle body with the passive and VDC suspension systems are 2.94 and 1.55 rad/s$^2$, respectively. The results show that the VDC suspension can effectively reduce body vibration and improve vehicle ride comfort.

Figure 12. Pitch acceleration.

Figure 13. Roll acceleration.

Figure 14. Deflection of the left suspension.
Figure 15. Deflection of the right suspension.

Figure 16. Deflection of the left tire.

Figure 17. Deflection of the right tire.
**Figure 18.** Vertical acceleration power spectral density.

**Figure 19.** Pitch acceleration power spectral density.

**Figure 20.** Roll acceleration power spectral density.
In Figures 14 and 15, in the front axle, the suspension deflections on the left and right sides are clearly lower in the VDC suspensions than in the passive suspension system. In Figures 16 and 17, in the front axle, the tire deflections on the left and right sides are evidently lower in the VDC suspension than in the passive suspension system. The results show that the VDC suspension can effectively reduce the deflection of suspension and tire, thereby improving handling stability.

To verify the control effects in the frequency domain, the energy spectral density figures of the vertical, pitch and roll accelerations of the two suspension systems are presented in Figures 18 to 20. In these figures, the amplitudes of the VDC suspension have been reduced considerably in comparison with those of the passive suspension.

Conclusions
1. In this work, a 9-DOF model of a semi-active suspension system and a random input model of a road surface are established. These models provide a basis for vehicle control.
2. A semi-active suspension with variable damping is proposed for heavy vehicles. The dynamic characteristics of damping and the time for establishing the damping force are presented on the basis of its working principle and configuration mechanism.
3. A VDC strategy is proposed on the basis of the dynamic characteristics of a semi-active suspension with variable damping to control heavy vehicles. The results show that, in comparison with passive suspension, the VDC suspension can significantly reduce body, pitch angle and roll angle accelerations. Therefore, VDC suspension can considerably improve vehicle ride comfort and handling stability in comparison with passive suspension.

Declaration of conflicting interests
The author(s) declared no potential conflict of interest with respect to the research, authorship and/or publication of this article.

Funding
The author(s) disclosed receipt of the following financial support for the research, authorship, and/ or publication of this article: This work was supported by the National Key Research and Development Program of China (Grant No. 2016YFC0802900).

ORCID iD
Hao Chen http://orcid.org/0000-0003-3951-9347

References
1. Evers WJ, et al. 2011. The electromechanical low-power active suspension: modeling control and prototype testing. J Dyn Syst Meas Control 2011; 133: 133–141.
2. Anakwa WKN, et al. Development and control of a prototype pneumatic active suspension system. IEEE Trans Educ 2002; 45: 43–49.
3. Dankan VG and Sadishiva C. Comparative analysis of passive and semi-active suspension system for quarter car model using PID controller. In: International conference on recent trends in signal processing, image processing and VLSI, ICrtSIV, Bangalore, 21–22 February 2014, pp.510–517.
4. Kawabe T, Isobe O, Watanabe Y, et al. New semi-active suspension controller design using quasi-linearization and frequency shaping. Control Eng Pract 1998; 6: 1183–1191.
5. Mardani A. Energy harvesting, handling and ride comfort trade-off between passive and active suspension systems of half vehicle model using PID controller for off-road vehicles. J Adv Veh Eng 2017; 3: 150–160.
6. Flores JL, Drivet A, Ramirez-Mendoza RA, et al. Hybrid optimal control for semi-active suspension systems. In: Mini conference on vehicle system dynamics, identification and anomalies, VSDIA 2006, Budapest, Hungary, 5–7 November 2006, pp.6–13.
7. Poussot VC, Spelta C, Senname O, et al. Survey and performance evaluation on some automotive semi-active suspension control methods: a comparative study on a single-corner model. Annu Rev Control. 2012; 36: 148–160.
8. Pavel P and Michal H. Design of characteristics of air-pressure-controlled hydraulic shock absorbers in an intercity bus. Multibody Syst Dyn 2008; 19: 73–90.
9. Hiromichi N. Technology for measuring the damping force of shock absorbers and the constant of coil springs mounted on a motorcycle by the un-sprung mass vibration method. SAE Technical paper 2004-01-2068, 2004.
10. Guy Y, et al. A solenoid-actuated pilot valve in a semi-active damping system. SAE Technical Paper 881139, 1988.
11. Ivers D and Miller L. Experimental comparison of passive, semi-active on/off, and semi-active continuous suspensions. SAE Technical Paper 892484, 1989.
12. Rajamani R and Hedrick JK. Semi-active suspension – a comparison between theory and experiments. Veh Syst Dyn 1992; 20: 505–518.
13. Besinger FH, Cebon D and Cole DJ. Force control of a semi-active damper. Veh Syst Dyn 1995; 24: 695–723.
14. Cebon D, Besinger FH and Cole DJ. Control strategies for semi-active lorry suspensions. Proc Inst Mech Eng, Part D 1996; 19: 161–178.
15. Wu JD, Lin CJ, and Kuo KY. A study of semi-active vibration control for vehicle suspension system using an adjustable shock absorber. J Low Freq Noise Vib Active Control 2008; 27: 219–235.
16. Ning XB, Shen J, Meng B, et al. Digital control damp valve in semi-active suspension. Adv Mater Res 2012; 433: 2534–2540.
17. Fan J, Sun X, Chu Y, et al. Optimization of two-state adjustable damping shock absorber. Key Eng Mater. 2011; 464: 332–335.
18. Sulaiman S, Mohd. Samin P, Jamaludd H, et al. Groundhook control of semi-active suspension for heavy vehicle. J Sound Vib 2012; 172: 391–411.
19. Valášek M, Novák M, Šika Z, et al. Extended ground-hook – new concept of semi-active control of Truck’s suspension. Vib Active Control 1997; 27: 289–303.
20. Yarmohamadi H and Berbyuk V. Effect of semi-active front axle suspension design on vehicle comfort and road holding for a heavy truck. SAE Technical Paper 2012-01-1931, 2012.
21. Nicolas C F, Landaluze J, Castrillo E, et al. Application of fuzzy logic control to the design of semi-active suspension systems. In: IEEE international conference on fuzzy systems, Barcelona, Spain, 1–5 July 1997, pp.987–993.
22. Salah GF. Neuro-fuzzy control of a semi-active car suspension system. In: IEEE Pacific Rim conference on communications, computers and signal processing, Victoria, BC, Canada, Canada, 26–28 August 2001, pp.686–689.
23. Nguyenf SD, Nguyen QH and Truong NT. Designing optimal fuzzy-compensator-enhanced sliding controller for train-car semi-active suspensions. In: IEEE international conference on fuzzy systems, Naples, Italy, 9–12 July 2017, pp.1–6.
24. Zheng MJ, Zhang XL and Zhao J. Fuzzy sliding mode variable structure control of semi-active air suspension. In: The 3rd annual international conference on design, manufacturing and mechatronics, China, 13–15 May 2016, pp.618–626.
25. Song BK, An JH and Choi SB. A new fuzzy sliding mode controller with a disturbance estimator for robust vibration control of a semi-active vehicle suspension system. Appl Sci 2017; 7: 1053.
26. Dong X, Zhao D, Yang B, et al. Fractional-order control of active suspension actuator based on parallel adaptive clonal selection algorithm. J Mech Sci Technol 2016; 30: 2769–2781.