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Research on Control of Semi-Active Suspension System Using Hydraulic Single-Tube Shock Absorber

***Ho Huu Hai**, Do Ngoc Sang**
Hanoi University of Science and Technology, Hanoi, Vietnam
**Corresponding author email:hai.hohuu@hust.edu.vn*

Abstract

There are some factors that influence a running vehicle. The dynamic forces acting at the contact between tires and rough road surfaces can have a detrimental impact on passenger health and vehicle safety. The purpose of the automotive suspension system is to reduce the impact of these forces and vibrations on passengers and also improve mobility, safety and the vehicle's longevity itself. The stiffness of the springs and the damping characteristic of the shock absorbers should be sufficiently non-linear for system's substantial performance. Many studies on the control of vehicle suspension system have lately been conducted in order to increase ride comfort and maneuverability but shock absorber's model has not been described detailly. This paper proposed a PID controller for a semi-active suspension system with a hydraulic single-tube shock absorber. A quarter-car model with a sub-model of the single-tube shock absorber was used to perform the simulation. In comparison with the non-controlled system, the damping performance of the controlled system increased significantly.

Keywords: Hydraulic single-tube shock absorber, semi-active suspension system, PID controller.

1. Introduction

Suspension is critical for ensuring vehicle ride comfort and a relaxing experience for passengers. In the conventional suspension system (non-controlled system or passive system), the spring and shock absorber are not controlled, so the ride comfort is not constantly good for various riding conditions. Active suspension systems have the best performance for cars thanks to actuators that produce the force acting between sprung and unsprung parts. However, since these systems need a significant amount of additional energy for actuator operation, they are not widely used in automobiles. Semi-active suspension systems considerably enhance ride comfort by changing the cross-sectional area of orifice valves or changing the viscosity of the working fluid of the shock absorber. and require little energy for operation. Because of this advantage, semi-active systems are widely used on not only luxury but also popular cars nowadays.

Articles related to suspension system control have been frequently published in recent times. Some of them could be mentioned as follows:

In [1], Abramov *et al.* described in detail a full-car model, road disturbance, and applied Skyhook control law to improve the vehicle's oscillating characteristics. However, the model of vibration damper - the element that generates the control force - was not mentioned in the article. That absence of the actuator model somewhat reduces the practical significance of the study.

In [2], a comparison between the dynamic characteristics of passive and semi-active suspension systems was presented. The semi-active suspension system with a PID controller was proposed for fine damped vibration of the vehicle. Nonetheless, the actuator producing the controlling force was not mentioned in the article.

Jamil *et al*. [3] investigated the functioning of a quarter car semi-active suspension model using the designed PID controller to adjust its damping parameters. However, this work has not partly achieved the utmost accuracy yet due to the undefined damper model properties.

Ali and Hameed in [4] focused on modeling an active-Nishimura quarter car model system, applying the rules of Fuzzy controller. The coil spring was replaced by an air spring and hydraulic damper with the use of an air actuator to generate the contact force between sprung and unsprung mass. Nevertheless, this force could not fulfill scientific accuracy to some degree since the air actuator model was not represented.

For non-linear model development, Yadav *et al.* considered quadratic non-linearity for suspension stiffness and cubic non-linearty for tyre stiffness [5]. Simulink model of semi active suspension system consists of a controllable damper - a form of MR (Magnetorheological) damper to produce the damping force. However, the damping coefficient determination was not mentioned in the paper.

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In [6] a modified PID controller was used to control the suspension system in a quarter car model. The controller's output (the force acting between sprung and unsprung parts) improved the system's dynamic response, whereas the actuator responsible for this force was not shown. That may partially reduce the practically significant of the paper.

In order to to achieve quality ride comfort, Ghoniem *et al.* [7] proposed a new semi-active suspension system including a hydraulic cylinder with a proportional valve. The change in the opening of the proportional valve has a great effect on the performance of the suspension system, meanwhile, the equation that expresses the proportional valve opening was not shown in the paper.

In [8], Ma *et al.* constructed a novel compensation system aimed at modeling the regulating mechanism of the nonlinear hydraulic adjustable damper (HAD) in a semi-active suspension system instead of building the model of a specific HAD directly to realize the desired damping force, which somewhat reduces the accuracy of the damping force.

In general, previous researches focused solely on optimizing control methods for suspension systems without actuator of the controller (shock absorbers or hydraulic cylinders). Meanwhile, these elements contribute significantly to the scientific accuracy and the practical applicability of the studies.

This paper proposed mathematical model of a quarter-car semi-active suspension system including sub-model of the hydraulic single-tube shock absorber as an actuator. A PID controller with two variations of feedback signal (that were vehicle body velocity and vehicle body acceleration) was proposed for controlling the cross-sectional area of damping orifices to get better-damped characteristics in comparison with the conventional passive approach.

2. Model of Semi-Active Suspension System Using PID Controller

2.1. Hydraulic Single-Tube Shock Absorber Model

Besides the damping force, hydraulic single-tube shock absorbers are renowned for generating the non-linear elastic force that is consistent with the ideal characteristics of automotive suspension system. Therefore, this type of shock absorber is widely employed nowadays in automobiles, particularly in passenger car.

The operation principles of this type of shock absorber can be briefly described as follows: when the damper is functioning, hydraulic fluid is pumped by moving up and down of the piston (compression and extension strokes) from one chamber to the other through small orifice holes (2) and (3) respectively (Fig. 1), causing a damping force to quench the car vibration rapidly. The damping coefficient is determined as a function of piston velocity, fluid viscosity, size and geometry shape of the holes (orifices) through which fluid flows. As a result, for a certain vibration velocity, the damping coefficient almost remains unchanged when the cross-sectional area of orifices is constant. The damping coefficient, on the other hand, can be changed by regulating the size of the orifices. That is principle of a controlled shock absorber in semi-active suspension system [11].

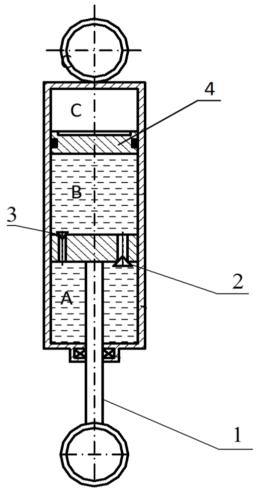

Fig. 1. Scheme of the hydraulic single-tube shock absorber: (1) piston rod, (2) compression stroke, (3) orifice hole for extension stroke, (4) floating piston, (A) and (B) hydraulic chamber, (C) compressed gas chamber.

The hydraulic single-tube shock absorber model described in this paper was based on the one that had been previously published in [9] and [10].

The damping force is induced by the pressure difference between the extension chamber (A) (with the cross-sectional area A_1) and the compression chamber (B) (with the cross-sectional area A_2) as the following equation:

$$F_{gc} = (p_A - p_0)A_1 - (p_B - p_0)A_2 \tag{1}$$

where:

p_A, p_B are the hydraulic pressure in chamber (A) and chamber (B) respectively;

p_0 is the initial pressure of compressed gas in chamber (C);

The hydraulic pressure in chambers (A) and (B) can be determined by the following equations:

$$p_A = \frac{K}{V_A}\int (Q_A + A_1\dot{x})dt + p_0 \tag{2}$$

$$p_B = \frac{K}{V_B}\int \left(Q_B - A_2(\dot{x} - \dot{y})\right)dt + p_0 \tag{3}$$

where:

V_A, V_B are the volume of chambers (A) and (B) respectively;

K is the bulk modulus of fluid;

x is the displacement of piston rod (1);

y is the displacement of floating piston (4). It can be determined from the equation of motion:

$$m\ddot{y} = (p_B - p_C)A_2 \tag{4}$$

where:

m is the mass of floating piston (4);

p_C is pressure of compressed gas in chamber (C) and it can be determined from the equation:

$$p_C = p_O \frac{V_O^n}{V_C^n} \tag{5}$$

where:

V_O is the initial volume of chamber (C);

V_C is the volume of chamber (C);

n is polytropic coefficient of gas expansion.

Q_A, Q_B are the fluid flow rates into chambers (A) and into chamber (B), which are given by the following equation:

$$Q_A = -Q_B = Q_{BA} - Q_{AB} \tag{6}$$

Q_{AB} and Q_{BA} are the fluid flow rates from the chamber (A) to the chamber (B) and vice versa. With the attention to the direction of the flow from the higher pressure chamber to the lower pressure chamber, these flow rates can be written as below:

$$Q_{AB} = \beta A_{AB}\sqrt{\frac{2|p_A - p_B|}{\rho}}\, sign(p_A - p_B) \tag{7}$$

$$Q_{BA} = \beta A_{BA}\sqrt{\frac{2|p_B - p_A|}{\rho}}\, sign(p_B - p_A) \tag{8}$$

where:

β is the flow rate coefficient;

ρ is the density of hydraulic fluid;

A_{AB} and A_{BA} are respectively the cross-sectional area of compression orifices and extension orifices. Their values depend on the pressure difference between damping chambers and the constant pressure p_k referring to as "critical pressure", at which the relief valves begin to open. The cross-sectional area of these orifices can be described as [9]:

$$A_{AB} = A_{AB}^{const} + A_{AB}^{max}[\delta_1 v(p_A - p_B, 0) + \\ +\delta_2 v(p_A - p_B, p_k)] \tag{9}$$

$$A_{BA} = A_{BA}^{const} + A_{BA}^{max}[\delta_1 v(p_B - p_A, 0) + \\ +\delta_2 v(p_B - p_A, p_k)] \tag{10}$$

where:

A_{AB}^{const} is the cross-sectional area of the permanently open orifice holes;

A_{AB}^{max} is the cross-sectional area of the variable opening orifice holes (relief valve);

δ_1, δ_2 are the relief valve' design coefficients;

v is a variable that represents the opening of valves.

2.2. Quarter-car model using the single-tube shock absorber

As mentioned earlier, a model of quarter-car suspension including the single-tube shock absorber model was proposed to carry out the simulation. The scheme of the system is illustrated in Fig. 2.

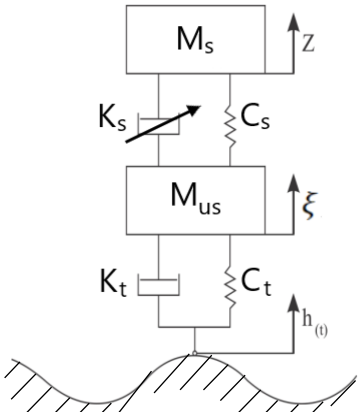

Fig. 2. Model of quarter car semi-active suspension system

The equations of motion for sprung and unsprung parts are:

$$\begin{cases} M_s\ddot{z} = C_s(\xi - z) + F_{gc} \\ M_{us}\ddot{\xi} = -C_s(\xi - z) - F_{gc} + C_t(h - \xi) \\ \quad +K_t(\dot{h} - \dot{\xi}) \end{cases} \tag{11}$$

where:

M_s and M_{us} are the mass of sprung part and unsprung part respectively;

C_s and C_t are the stiffness of the suspension spring and the tire respectively;

F_{gc} is the damping force generated by the shock-absorber;

K_t is the tire damping coefficient;

h is the road surface profile (disturbance);

z is the displacement of sprung part;

ξ is the displacement of unsprung part.

Regarding the shock absorber in the system, its orifice's cross-sectional area can be changed to vary the damping coefficient. It was assumed that the cross-sectional of damping orifices is modified by an amount of change by PID controller - Δ_A and the two equations (9) and (10) could be rewritten as:

$$A_{AB} = A_{AB}^{const} + A_{AB}^{max}[\delta_1 v(p_A - p_B, 0) + \delta_2 v(p_A - p_B, p_k)] + \Delta_A \quad (12)$$

$$A_{BA} = A_{BA}^{const} + A_{BA}^{max}[\delta_1 v(p_B - p_A, 0) + \delta_2 v(p_B - p_A, p_k)] + \Delta_A \quad (13)$$

Damping force F_{gc} was determined by the equations (1) to (8), (12) and (13).

2.3. PID controller

PID controller consists of three components: proportional (P), integral (I), and derivative (D) component (Fig. 3).

Semi-active suspension system has a feedback mechanism to control the damping force (by changing the damping coefficient). The error signal was fed to PID controller to adjust the size of orifices of the shock absorber so that the output reaches the reference value (setpoint).

For study purposes, there were two cases of the feedback signal to the controller: velocity and acceleration of the vehicle body. Block diagrams and Simulink models for these feedback signals are shown in Fig. 4 to Fig. 6 below:

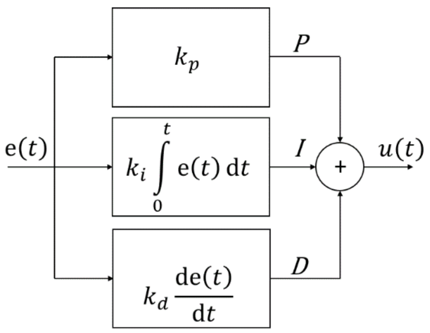

Fig. 3. The structure of PID controller

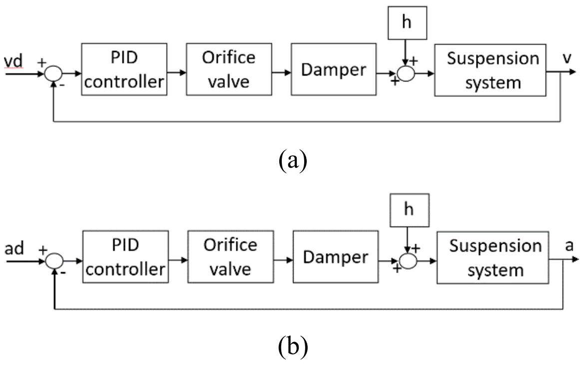

Fig. 4. Block diagram of semi-active suspension system with body velocity control (a) and body acceleration control (b)

The notation parameters for the model are:

h: Road surface profile;

v: Vehicle body velocity;

v_d: Desired vehicle body velocity;

a: Vehicle body acceleration;

a_d: Desired vehicle body acceleration.

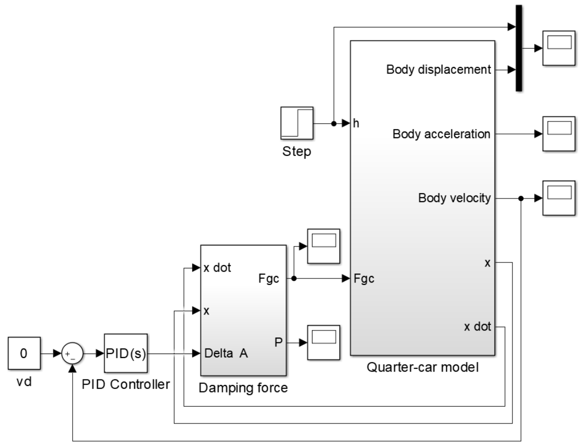

Fig. 5. Simulink model of semi-active suspension system with vehicle body velocity control

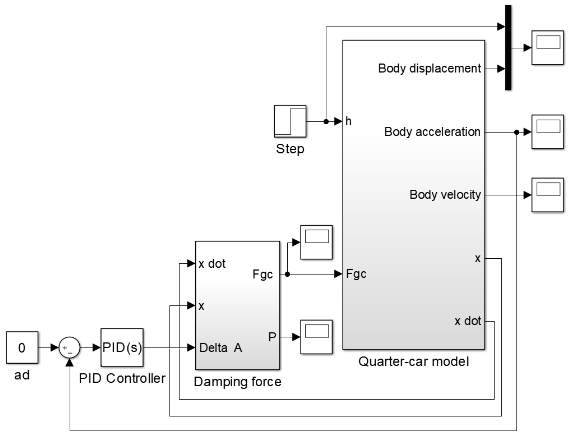

Fig. 6. Simulink model of semi-active suspension system with vehicle body acceleration control

Simulink model contains two main sub-systems:

- The sub-system "Damping force" (Fig.7) simulating the single-tube shock absorber to generate the damping force F_{gc}.
- The sub-system "Quarter-car model" (Fig. 8) simulating the motion of sprung and un-sprung masses to calculate vehicle body's displacement, velocity and acceleration.

As mentioned above, the PID controller regulated value of Δ_A to change the cross-sectional area of the damping orifices according to the value of the feedback signals, that are sprung mass' velocity $\dot{z}$ (vehicle body velocity control) and acceleration $\ddot{z}$ (vehicle body acceleration control).

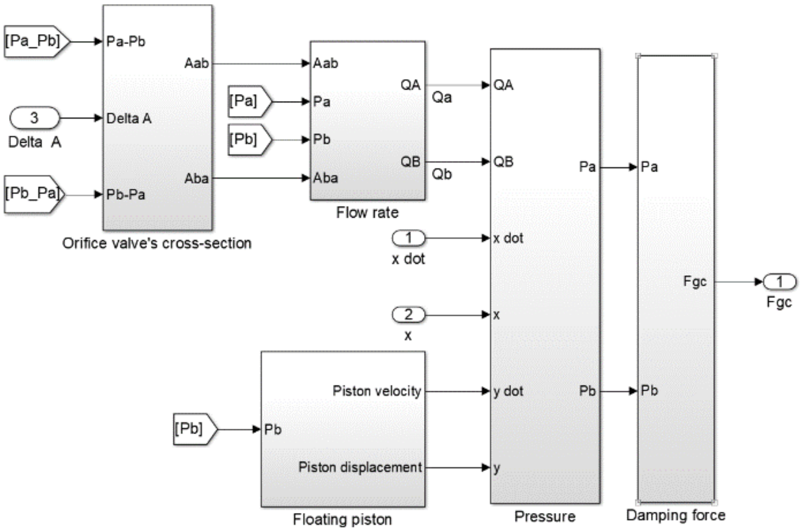

Fig. 7. “Damping force” sub-system

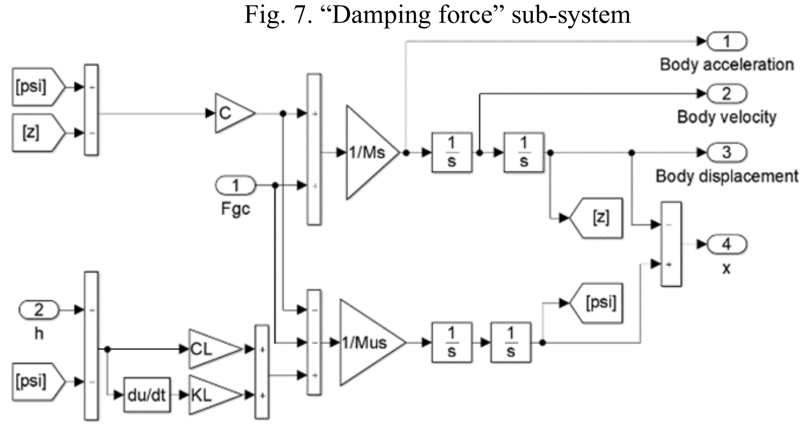

Fig. 8. “Quarter-car model” sub-system

The simulation was carried out with the parameters of a normal passenger car, which are listed in Table. 1.

From Section 3 below, the controlled suspension system is considered as semi-active suspension system while the non-controlled one is considered as passive suspension system.

Table 1. The parameters of quarter-car suspension model

Parameters	Value
M_s	430 (kg)
M_{us}	21 (kg)
C_s	21595 (N/m)
C_t	40000 (N/m)
K_t	2000 (Ns/m)

3. Simulation Results

The simulation was carried out with an external disturbance, which was the road bump as a step function of 0.05 (m) at time 1 (s) (Fig. 9). The simulation for the two cases is shown as follows.

3.1. Vehicle Body Velocity Control

The comparison of the hydraulic fluid pressure variation and damping force variation (as a consequence of pressure change) in passive and semi-active system are shown in Fig. 10 and Fig. 11. It could be seen from these figures that when the vehicle hit the road bump, the pressure difference between the damping chambers in the controlled shock absorber was much higher, causing the greater damping force to quench oscillation more efficiently. Moreover, the controlled damping force was reduced to zero quickly in the free-oscillation periods, leading to the damped performance improvement of the semi-active system.

Fig. 9 and Fig. 12 illustrate vehicle body displacement and suspension system’s working space. It was generally considered a significant enhancement in system performance in terms of vehicle riding comfort because the curves showed a decreasing trend in vibration amplitude of the semi-active system.

A similar trend could be seen in Fig. 13 of vehicle body velocity and Fig. 14 of vehicle body acceleration. The sprung mass in the semi-active system stabilized faster in comparison with the passive system.

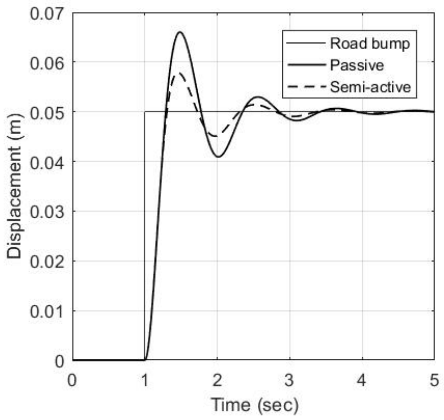

Fig. 9. Road bump and vehicle body displacement

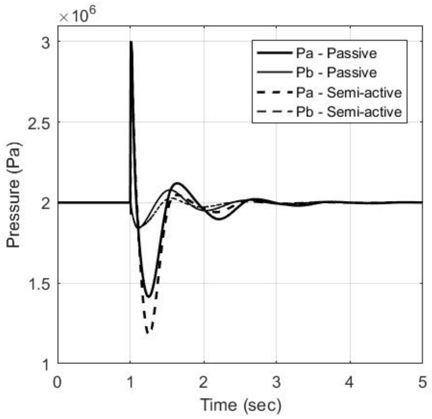

Fig. 10. Pressure variation

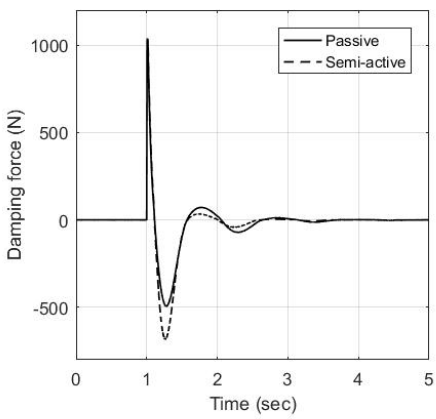

Fig. 11. Damping force

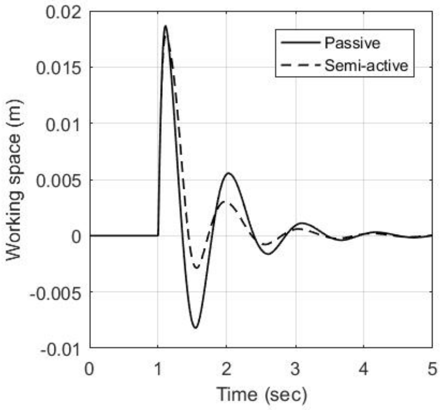

Fig. 12. Working space of suspension system

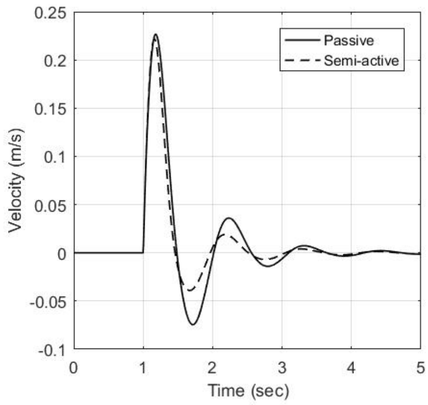

Fig. 13. Vehicle body velocity

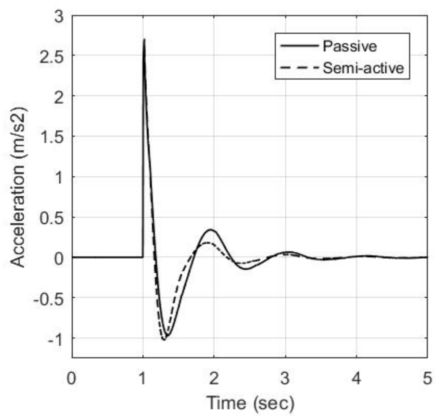

Fig. 14. Vehicle body acceleration

3.2. Vehicle Body Acceleration Control

Fig. 15 to Fig. 20 show the similar response of system in case of body acceleration control as in case of the velocity control.

Fig. 15 shows the variation of hydraulic pressure in absorber's chambers, while Fig. 16 showed the damping force curve. The figures demonstrated the higher damping efficiency of the absorber regulated by the PID controller in comparison with the non-controlled one, which was similar to the comments in Fig. 9 and Fig. 10.

As we can see from the vehicle body displacement curve in Fig. 17, and from suspension system's working space in Fig 18, the amplitude reduction of all the curves also contributed greatly to vehicle ride comfort and maneuverability.

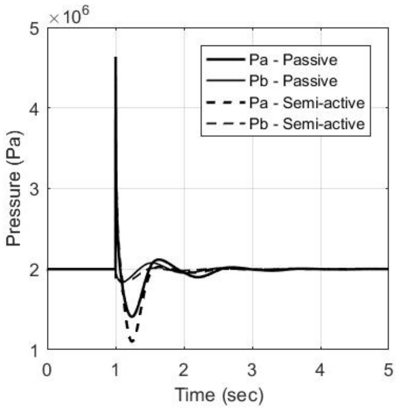

Fig. 15. Pressure variation

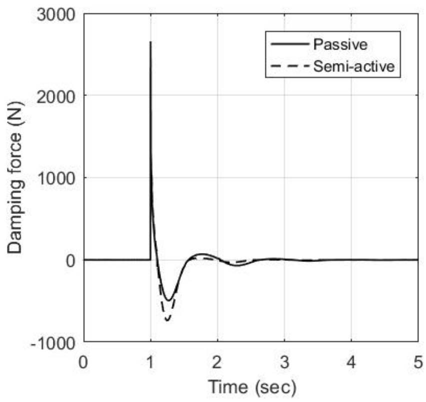

Fig. 16. Damping force

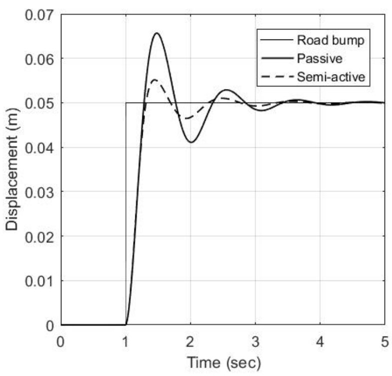

Fig. 17. Vehicle body displacement

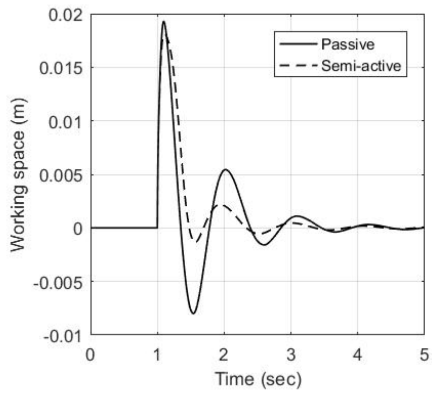

Fig. 18. Working space of suspension system

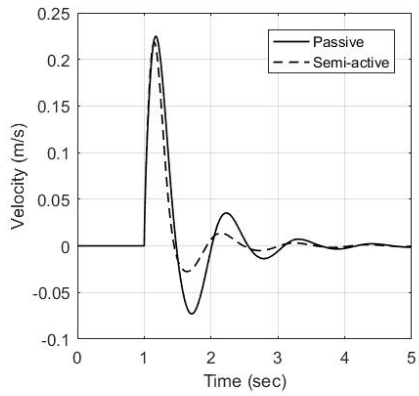

Fig. 19. Vehicle body velocity

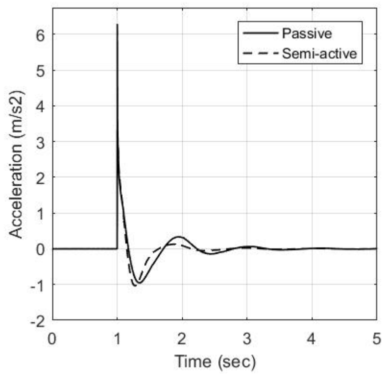

Fig. 20. Vehicle body acceleration

The controlled shock absorber's damping force produced comfortable velocity and acceleration of sprung mass for the passengers, which is depicted in Fig. 19 and Fig. 20. It was clear that velocity and acceleration had been reduced by the semi-active system, particularly in free-oscillation periods.

4. Conclusion

In this paper, a quarter-car suspension system with a hydraulic single-tube shock absorber regulated by a traditional PID controller has been modeled and simulated.

A comparison between simulation results of the passive and semi-active suspension systems has been done for two cases of velocity control and acceleration control.

The performance improvement of systems in two cases has been shown: velocity and acceleration control are relatively comparable. With semi-active system, the vehicle body displacement, velocity, and acceleration all decreased approximately 50% in comparison with passive system for the given operating condition. Moreover, the working space of suspension system was also reduced considerably, allowing the vehicle to lower the center of gravity to enhance stability.

Because of the similar operation of the velocity controlled system and the acceleration controlled system, it could be proposed a comment for practical application of acceleration control: the measurement of vehicle body acceleration as the feedback signal for PID controller would be more convenient in practice. Acceleration sensors are today more reasonably priced and have high accuracy for suspension system control applications.

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