Performance comparison between semi-active suspension and passive suspension based on ceiling algorithm

Dongqi He, Pingyang Yang, Xiaodong Zhang*

College of Mechanical Engineering and Automation, Huaqiao University, Xiamen, China

* Corresponding Author Email: zhangxd@hqu.edu.cn

Abstract. This research explores continuous damping for vehicle suspension systems. We devise two models of suspension systems: one being a traditional passive suspension system and the other a semi-active variant. The semi-active suspension system is highlighted for its capability to enhance vibration reduction via adjustable damping, offering cost-effective and practical benefits over fully active counterparts. To further augment the performance of the semi-active suspension, we introduce a ceiling algorithm, which is instrumental in diminishing the vibrations affecting the rear sprung mass, thereby enhancing suspension system performance overall. Subsequently, we conduct a comparative analysis of the sprung acceleration responses of both types of suspension systems under various excitations, including step inputs, sine waves, random waves, and B-class road profiles. Our results demonstrate that the semi-active suspension system, when equipped with the enhanced ceiling control algorithm, surpasses both the traditional passive and semi-active cases utilizing the standard ceiling control algorithm. Specifically, the enhanced algorithm-equipped semi-active suspension exhibits lower root-mean-square acceleration and peak acceleration values, affirming its superior damping performance.

Keywords: Semi-Active Suspension System, Passive Suspension System, B-class Road Profile, Superior Damping Performance.

1. Introduction

As technology progresses, there is a growing emphasis on pursuing convenience, comfortable and fast travel experience. The car with good shock absorption performance has become a necessity of people's life. The vehicle suspension system, acting as a conduit for transferring forces and moments between the wheel and the chassis, plays a crucial role in dampening vibrations caused by rough road surfaces and ensuring vehicle stability during sharp turns. This makes the study of suspension systems highly significant.

Automobile suspension systems are primarily categorized into passive and semi-active types. Passive suspension is a traditional system that cannot adjust the stiffness of the suspension according to the irregularity of the road and the load of the vehicle. It typically comprises springs and shock absorbers that provide support and cushioning based on the impact forces from the road. As the most fundamental type of suspension, it continues to be refined through various optimization techniques and is extensively utilized in everyday life. [1] A new type of ideal mechanical single-port network unit is introduced [2]. Several simple passive suspension elements are compared and the potential performance advantages of the unit are discussed. To enhance the passive suspension performance of rail vehicles, an interferometer mechanical device was proposed [3]. With the increasing maturity of genetic algorithm and the wide application of artificial neural network, the combination of the two has become a new trend [4]. Genetic algorithms are employed to optimize the transmission passive suspension by minimizing acceleration root mean square (RMS) and enhancing driving comfort. Subsequently, artificial neural networks are utilized to forecast the optimal suspension parameters [5]. By integrating advanced computational techniques with traditional mechanical designs, the pursuit of improved suspension systems continues to push the boundaries of automotive technology, aiming to deliver a smoother and safer ride for all.

Comparatively, the primary benefit of the semi-active suspension over passive is its capability to maintain optimal wheel contact with the road, thus enhancing handling and adhesion. Additionally, it ensures the car maintains stable body posture under acceleration, braking, and cornering to prevent loss of control. Genetic algorithm-based multi-objective optimization design method of Mr Damper was employed to minimize vibration, enhance passenger comfort, and optimize road handling of a quarter vehicle suspension [6]. The application of deep reinforcement learning for semi-active suspension control further underscores the promising future of semi-active suspension systems in improving ride comfort [7].

Regarding semi-active suspension systems, to mitigate the impact of road-induced excitations on the sprung mass, a control algorithm employing ceiling damping is utilized to attenuate sprung mass vibrations, thereby further reducing overall vehicle oscillations. Subsequent advancements in this area are detailed in [8] [9] [10].

In this paper, the ceiling algorithm of passive suspension and active suspension based on 2-DOF model is introduced to test the damping performance of different road surfaces.

2. Pavement model

2.1. Random pavement

Turbulence is a ubiquitous phenomenon when a vehicle is in motion, with road surface conditions typically represented by random excitation signals. These signals are characterized statistically through their power spectral density. Generally, the smoothness of pavement is quantified using displacement as the statistical characteristic. In this study, we employ a pavement roughness input model that leverages the rational function of power spectral density to depict the conditions of B-class roads. The formula for the power spectral density is as follows:

$$G_q(n_0) = G_q(n)\left(\frac{n}{n_0}\right)^{-w} \tag{1}$$

In the formula, n spatial frequency (m^{-2}), n is the reference spatial frequency (m^{-1}), generally 0.1, $G_q(n)$ is the road power spectral density (m^3), $G_q(n_0)$ n is the road power spectral density value (m^3) under the reference spatial frequency, w is the frequency index, w=2.

According to the power spectral density of the road surface and the road classification method, the road surface roughness is divided into 8 grades A to F.

Table 1 shows the geometric mean and upper and lower limits of the road roughness coefficient $G_q(n_0)$ of the first four grades.

Table 1. random road roughness classific

Grade of pavement	$G_q(n_0)\ (10^{-6}m^3)$		
	floor	Mean value	Upper limit
A	8	16	32
B	32	64	128
C	128	256	512
D	512	1024	2048

Random pavement excitation signals in the time domain are often synthesized from white noise, with various generation techniques at our disposal. These include the random harmonic superposition method [11], filtered white noise method [12], integrated white noise method [13] and inverse Fourier transform method, because the pavement velocity spectral density G_q (n_0) has a definite amplitude in the frequency range. In this paper, we opt for the forming filter method to generate the time-domain signal for random pavement excitation. The associated transfer function is detailed below:

$$H_1(s) = \frac{2\pi\sqrt{G_q(n_0)v}}{s + 2\pi f_0} \tag{2}$$

Time domain signal is

$$\dot{q}(t) = 2\pi n_0 \sqrt{G_q(n_0) v} w(t) - 2\pi v f_0 q(t) \tag{3}$$

where v is the speed (m/s), f_0 is the lower limit of the cutoff frequency (HZ), $q(t)$ is the road surface, $w(t)$ is the gaussian white noise with the mean value of 0 and the intensity of 1.

2.2. Pulse pavement

In actual driving, the vehicle will also cause severe tremors in a short time because of road bulges, pits, etc., affecting the driving and riding comfort, which is a test of the shock absorber function. Therefore, this paper introduces a unit pulse signal with an amplitude of 5mm to simulate this kind of pavement.

2.3. Frequency sweep pavement

With a road input signal bandwidth of 1~5Hz or 10~20Hz, the vehicle's unsprung mass and sprung mass resonate with the road's oscillatory signal. Research indicates that 4~8Hz is the frequency range where humans are most sensitive to vibration [14]. Therefore, it's crucial to scrutinize the frequency amplitude characteristics of system vibration within these three frequency ranges during automotive suspension's vertical dynamic investigation. In this paper, a swept-frequency signal spanning 0.01Hz~50Hz is adopted for analyzing the vehicle suspension, highlighting the efficacy of vibration mitigation achieved by the proposed model.

2.4. Step pavement

In addition to the random vibration of pavement, different frequency amplitude of pavement and sinusoidal excitation, the periodic fluctuation of pavement caused by various reasons is also quite common. This paper uses step signal to simulate this kind of pavement.

3. Control model

3.1. Passive suspension control

Reconstructing from the vehicle single-axle two-degree-of-freedom suspension model illustrated in the figure below, we have converted the stiffness and damping coefficients of the shock absorber into their equivalent vertical stiffness and damping values, respectively [15] [16]. The equation of motion is then formulated as follows:

$$\begin{gathered} M\ddot{z} = -c(\dot{z} - \dot{z}_t) - k(z - z_t) \\ m\ddot{z}_t = c(\dot{z} - \dot{z}_t) + k(z - z_t) - k_t(z_t - z_r) \end{gathered} \tag{4}$$

M represents the mass of the sprung mass in the suspension system (kg), m denotes the mass of the unsprung mass (kg), z_r is the variation in road height (m), z_t signifies the vertical displacement of the unsprung mass (m), z is the vertical displacement of the sprung mass (m), k is the equivalent vertical stiffness of the suspension (N/m). c is the equivalent vertical damping coefficient of the suspension (N/(m/s)), and k_t is the tire stiffness (N/m).

This paper employs the Simulink module within MATLAB to analyze the frequency response characteristics of a passive suspension system across various damping coefficients. The system model, as constructed, is illustrated in Figure 1.

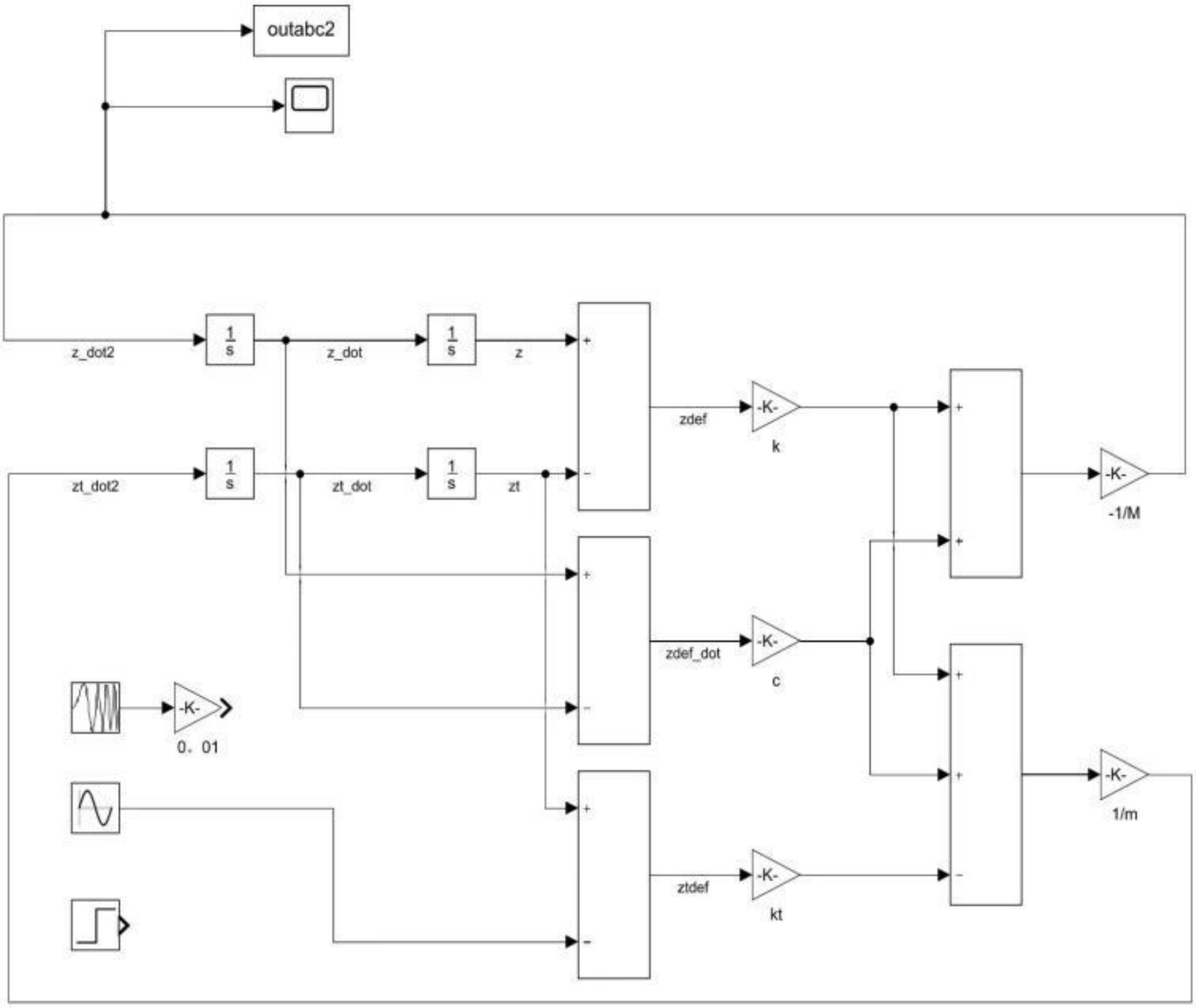

Fig 1. Simulink model of single-axis two-degree-of-freedom passive suspension

Applying the target vehicle's total mass and passenger mass to front and rear axles yields the front and rear unsprung masses. Regarding the target vehicle's front suspension, unsprung acceleration serves as the stability indicator. First, the relevant parameters of the single-axle suspension model are list in TABLE II.

Table 2. Front suspension model parameter

Argument	Numerical value
M	1280.1 kg
m	89.9 kg
k	6120N/m
c_{max}	11400N/(m/s)
c_{min}	1910N/(m/s)
k_t	268000N/m

3.2. Semi-active suspension optimized by ceiling algorithm

Semi-active suspension systems are a class of adjustable suspension mechanisms that leverage sensor technology to monitor road surface conditions and the vehicle's posture [16]. These systems fine-tune damping characteristics to bolster both the ride comfort and the stability of the vehicle. In this context, the canopy damping algorithm is utilized to develop an optimal model for the semi-active suspension. This model introduces a virtual damper that links the sprung mass to an inertial reference frame above the vehicle, ensuring that the distance between the sprung mass and the road remains constant regardless of the vehicle's motion. By eliminating the physical damper between the sprung mass and the unsprung mass, results in the sprung mass being insulated from road excitations. Consequently, the influence of road excitations on the sprung mass is reduced, resulting in less sprung mass vibration.

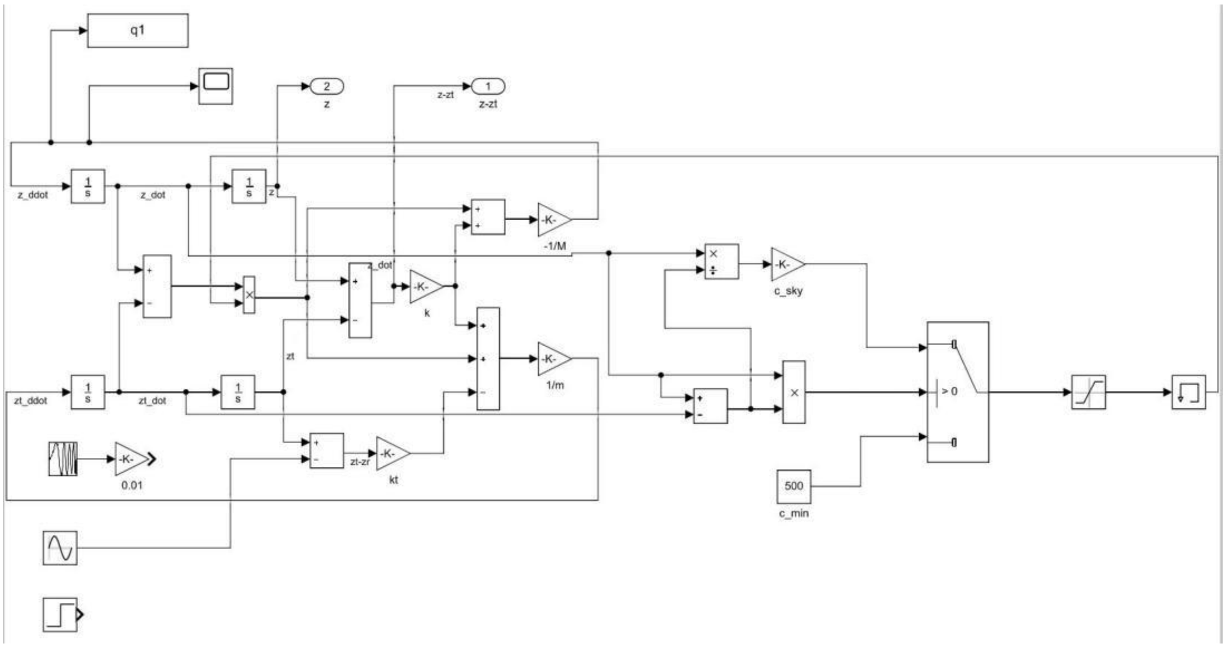

Fig 2. Simulink model of single-axis two-degree-of-freedom semi-active suspension

Figure 2 illustrates the aforementioned model. Specifically, the amplification of the virtual damper's damping coefficient and its stiffness significantly impact the sprung mass vibration suppression. As depicted in Figure 3 [17], the real suspension system lacks an inertial reference frame and a damper between the frame and the sprung mass. Therefore, estimating the virtual damper's damping force on the sprung mass in the ideal model controls the variable-damping shock absorber's damping force, yielding an equivalent damping effect, permitting the sprung mass [18] to achieve a vibration suppression performance similar to that of the ideal model.

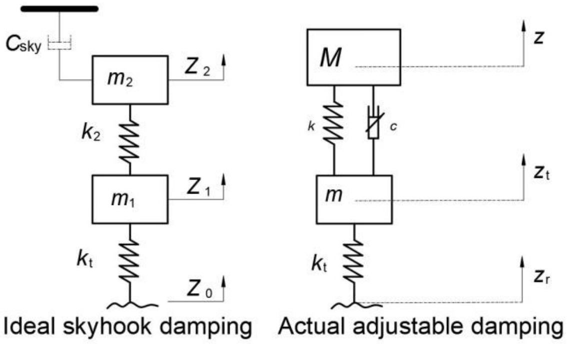

Fig 3. Ideal Skyhook Damping and Actual Adjustable Damping in a 1/4 Vehicle Suspension Model

Compared to the actual adjustable damping shown in (4) the semi-active suspension under ideal ceiling optimization is shown as follows

$$M\ddot{z} = -C_{sky}\dot{z} - k\left(z - z_t\right)$$

$$m\ddot{z}_t = k\left(z - z_t\right) - k_t\left(z_t - z_r\right) \qquad (5)$$

Based on the above, formula (4)(5)is equivalent, so it can be obtained

$$c = \frac{\dot{z}}{\dot{z} - \dot{z}_t} c_{sky} \qquad (6)$$

where c_{sky} is the damping coefficient of the virtual shock absorber (N/(m/s)), $\dot{z}$ is the vertical velocity of the sprung mass (m/s), and $\dot{z}_t$ is the vertical velocity of the unsprung mass (m/s).

c_{sky} is the calibration parameter and C is the continuously adjustable damping coefficient, both of which are real numbers greater than zero in the physical sense. Since both may be negative in formula (4) and (5), the following logical summary of the switching ceiling damping [19] is given and shown in Figure 4.

(1) When the body and the wheel move in opposite directions, the control system should apply high damping to effectively dissipate vibration energy and quickly reduce the vibration.

(2) When the body and the wheel move in the same direction, if the body speed exceeds the wheel speed, the control system should also apply high damping to reduce body vibration.

(3) When the body and the wheel move in the same direction, if the body speed is lower than the wheel speed, the control system should reduce damping, allowing more of the wheel's vibration energy to be absorbed and stored by the spring, and less of it to be transferred to the body.

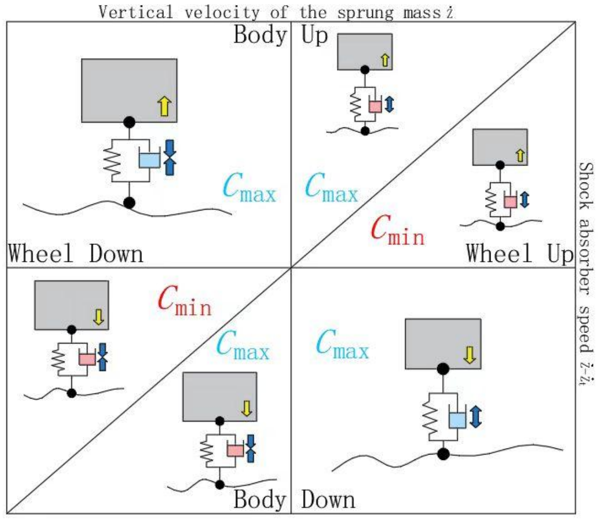

Fig 4. Damping switching logic of ceiling algorithm

Ceiling damping control strategies are divided into SkyhookOn-off and SkyhookLinear, both of which have the same control idea. The linear ceiling damping control strategy of the continuously adjustable damping shock absorber studied in this paper is expressed as

$$c = \begin{cases} c_{\max} & \dot{z} \cdot \dot{z}_{def} > 0 \\ c_{\min} & \dot{z} \cdot \dot{z}_{def} \le 0 \end{cases} \tag{7}$$

4. Simulation

MATLAB's Simulink model employs the root-mean-square of sprung acceleration for vehicle ride comfort evaluation. The superiority of passive and semi-active suspension models is reflected by lower vehicle vibration control.

Table 3. The maximum and rms of sprung acceleration of passive suspension and semi-active suspension under four kinds of excitation

Statistical index	Stimulation	Controller Suspension	
		Passive	Semi-active
RMS	Step input	0.5487	0.3221
	Sinusoidal input	1.2556	0.1710
	Random fluctuation	1.0217	0.3184
	Class B pavement	6.5600	2.5918
Maximum value	Step input	7.6297	2.3611
	Sinusoidal input	1.8291	0.2882
	Random fluctuation	2.1721	2.1660
	Class B pavement	19.6050	7.4071

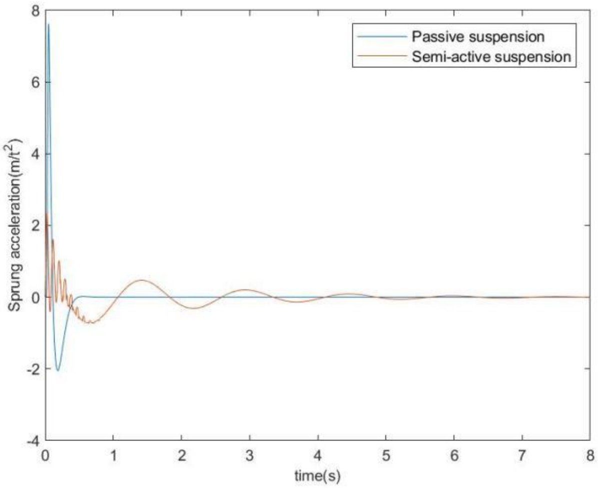

Fig 5. Response values of active suspension and semi-active suspension under step input

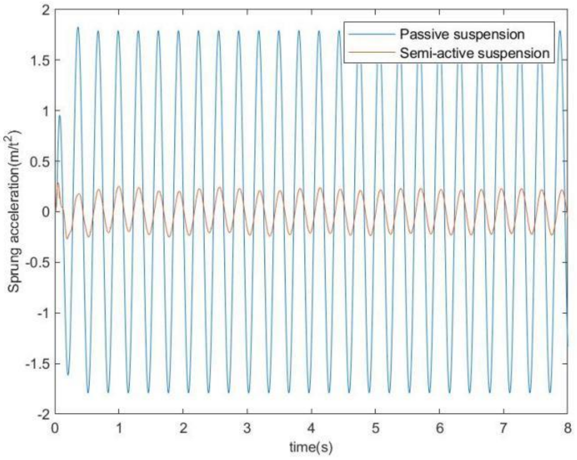

Fig 6. Response values of active suspension and semi-active suspension under sinusoidal input

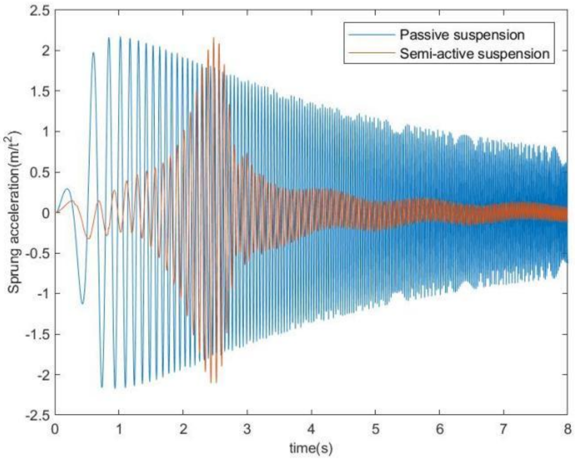

Fig 7. Response values of active suspension and semi-active suspension under random fluctuation input

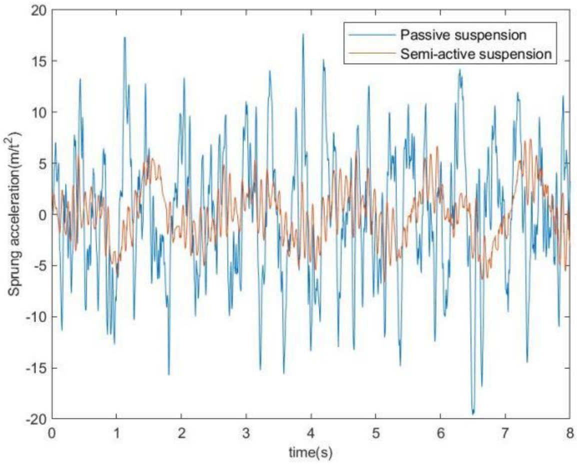

Fig 8. Response values of active suspension and semi-active suspension under B-class road input

TABLE III presents the maximum unsprung acceleration and the root-mean-square values for both passive and semi-active suspensions under step input, sine input, random input, and B-class road excitation. It is visualized in Figure 5-8. During the initial 8seconds of simulation, the improved ceiling algorithm-assisted semi-active suspension response amplitude noticeably reduces under sine input and B-class random road excitations. Although the convergence is gradual, the overshoot is minimal, leading to enhanced vehicle ride comfort. Under random input road excitation, the response amplitude of the semi-active suspension exceeds that of the passive suspension in the initial 2-3 seconds, but the overall response is minimal. This demonstrates that the performance of the semi-active suspension, enhanced by the ceiling algorithm, is superior to that of the passive suspension.

5. Conclusion

Based on the research presented in the article, the optimization of passive suspension systems using ceiling algorithm has been successfully demonstrated to reduce the RMS of acceleration, thereby enhancing driving comfort. In this study, a comparative analysis between semi-active suspension and passive suspension systems, utilizing the improved ceiling algorithm. The core of this approach lies in the calibration and adjustment of the damping coefficient, denoted as csky, which is pivotal in controlling vibrations. The study emphasizes two primary scenarios: (1) when the body and wheel move in reverse, necessitating large damping to rapidly attenuate vibration energy; and (2) when the body and wheel move in the same direction, with the body speed exceeding the wheel speed, requiring substantial damping to mitigate body vibrations. The integration of the improved ceiling algorithm further refines the control strategy, ensuring that the damping coefficients are optimally adjusted to handle varying road conditions and driving scenarios. By refining the control algorithms, semi-active suspension systems are poised to deliver enhanced performance and comfort, setting the stage for the next generation of vehicle dynamics and control.

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