A New Semi-Active Control Method of Yaw Damper in High-Speed Railway Vehicle and Its Experiment in Hardware-in-the-Loop System

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1 Introduction

With the rapid development of high-speed railway technology, many dynamic problems have been found and have been researched. In order to avoid hunting instability when the train is running at high speed on a straight line, greater longitudinal damping between the body and the frame is required in most cases, while the longitudinal damping should not be too large when the train passes through the curve to ensure the safety performance. Therefore, the contradiction between lateral stability and curving performance has always been difficult to solve in the process of increasing train speed. This is one of many problems. Engineers constantly optimize the suspension parameters and bogie structure to coordinate the relationship between them to finally achieve overall balance [1–5]. However, the process of parameter and structure optimization is not endless. When

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the optimization reaches a certain degree, the contradiction between lateral stability and curving performance will become a main bottleneck for further improving train speed.

Experts and scholars from all around the world have proposed a variety of solutions to the contradiction between lateral stability and curvilinear trafficability. A new type of radial bogie was proposed in [6]. According to the design principle of radial bogie, a breakthrough design of axle box positioning mode is carried out. The contradiction between curve negotiation performance and lateral stability of the bogie is solved with the least number of bars. The dynamic models of electric locomotive with radial bogies and conventional bogies are established by using multi-rigid body dynamics software in [7]. The two models are compared by simulation. The results show that the curving performance of the locomotive model with radial bogies is better than that of the other vehicle. At the same time, the wheel rail wear is reduced significantly. An independent wheelset structure with active control capability is proposed in [8]. The running ability on straight track and curving ability are simulated and analyzed. A curvature measuring method that can be implemented in a railway vehicle system is proposed to steer a railway vehicle in [9]. The field test is carried out by this method. The results show that the method can enhance the curving performance of trains and keep the lateral stability. The primary suspension stiffness is optimized to evaluate the curving performance and stability of trains to meet the operational requirements better in [10]. It can be concluded that the main ways to solve the problem are to use radial bogies and independent wheelsets, as well as parameter optimization, and so on. These solutions can solve the problem to a certain extent.

This chapter presents a semi-active control system of anti-hunting shock absorber, which can change the damping coefficient between straight track and curve track so as to solve the contradiction between lateral stability and curving performance of trains.

At present, semi-active control technology is a kind of control technology which scholars have studied extensively and deeply. Crosby and Karnopp [11] from the United States put forward the concept of semi-active suspension control in the 1970s. Its control mode was applied to actual vehicles in the early 1980s. Up to now, semi-active control technology has been applied in many fields [12–15] such as automobile, railway, and building. In 1983, Toyota developed an "on-off" semi-active suspension which can generate corresponding damping force by adjusting a switch and applied it to the 280GT car. In 1985, Dominy and Bulman designed a Formula One racing car with semi-active suspension. In 2014, Yutong Bus Company developed the semi-active suspension system with variable damping and variable stiffness, which greatly improved the ride comfort of the car. In order to verify the performance of semi-active vibration reduction and improve the ride comfort of high-speed trains, Japanese railway vehicle engineers carried out vehicle tests on 500 series EMUs at 300 km/h on Shanyang Shinkansen Line. The results showed a significant decrease in the energy spectrum of lateral acceleration and lateral acceleration of the vehicle body. Semi-active suspension has gradually become a standardized device on Shinkansen vehicles in Japan. Meanwhile, there are many types of semi-active control strategies for adjusting the damping coefficient [16–20].

Generally speaking, semi-active control is a mature and reliable control method. It is feasible to apply semi-active control technology to anti-hunting shock absorbers, which can solve the problems described in the chapter. This method does not need to change the structure of the system and to optimize some parameters to redesign the components. This method only needs to replace the traditional oil damper, which can reduce costs.

The content of this work discussed in this chapter is based on the railway field, and the research object is a railway vehicle. The first purpose is to build a simple single wheelset model to study the influence of large damping and small damping on curving performance.

The second goal is to establish the UM-MATLAB co-simulation model of railway vehicles, to propose a new semi-active control strategy, and to study the influence of the control strategy on the dynamic performance of trains, such as ride comfort, safety, and stability.

The third purpose is to carry out the hardware-in-the-loop test and apply the semi-active control strategy to the actual MR damper to verify the effectiveness of the control method.

2 Establishment and Simulation of Single Wheelset Model

This work first considers the problem from a simple model. A single wheelset model used in curve track is established. The longitudinal positioning stiffness *kx*, lateral positioning stiffness *ky*, and anti-yaw damping *cx* of single wheelset are considered in the model as shown in Fig. 1.

The model includes lateral and yaw degrees of freedom. The parameters denoted in Table 1 are derived from a type of high-speed EMUs.

The differential equations of the model are established, which are as follows [21, 22]:

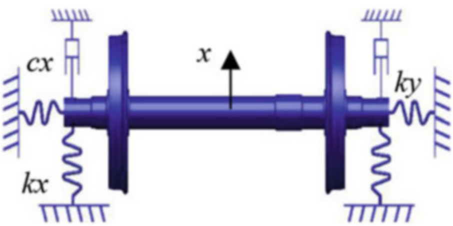


Fig. 1 Two freedom dynamics model of single wheelset

Table 1 The model parameters

Parameters	m/kg	J_w/kg	kx_l(N·m^{-1})	ky_l(N·m^{-1})	cx/(N·s·m^{-1})
Value	1901.8	685	9.8e06	9.8e05	2.45e06

$$\begin{aligned} & m\ddot{y}_w + 2f_{22}\left(\frac{\dot{y}_w}{v} - \varphi_w\right) + 2k_y y_w + k_g\,(y_w - y_a) - mg\phi + m\frac{v^2}{R} + mr_0\ddot{\varphi} = 0 \\ & J_w\left(\ddot{\varphi}_w + vd\left(\frac{1}{R}\right)\right) + 2f_{11}\left(\frac{\lambda b}{r_0}y_w + \frac{b^2}{v}\dot{\varphi}_w\right) + 2k_x l^2\varphi_w + 2c_x l^2\dot{\varphi}_w + k_{g\varphi}\varphi_w - 2f_{11}\frac{\lambda b}{r_0}y_a = 0 \end{aligned} \tag{1}$$

where f_{11} is the longitudinal creep coefficient, 1e07 N; f_{22} is the lateral creep coefficient, 1e07 N; R is the radius of the curve track,1500 m; λ is the equivalent conicity, 0.15; v is the running speed, 200 km/h; k_g is the gravity stiffness, 2e04 N/m; $k_{g\varphi}$ is the gravity angular stiffness, 1.1e04 N/m; r_0 is the rolling circle radius, 0.43 m; b is the half of the rolling circle distance, 0.7465 m; l is the half the length of axle, 1 m; and ϕ is the superelevation angle of track, 0.1 rad. y_{a} is input excitation, y_w is the lateral displacement of the wheelset, φ_w is the wheelset yaw angle.

According to the Eq. (1), the Simulink model of the single wheelset is established, and the parameters are input into the model for simulation. The radius of the curve track used in the model is 1500 m, and the random excitation is German lateral irregularity spectrum.

First, the lateral displacement curves of wheelset and longitudinal creep force under different yaw dampings are obtained, respectively, when the wheelset passes through the ideal curve track as shown in Fig. 2.

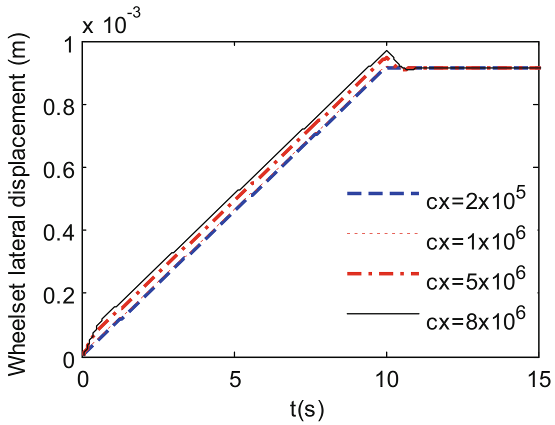
The lateral displacement curves of wheelset and longitudinal creep force under different yaw dampings are obtained under the random excitation condition as shown in Fig. 3.

The simulation results show that the larger the damping coefficient, the larger the wheel rail lateral displacement and longitudinal creep force. It is very obvious on the transition curve track. When the damping coefficient is too large, the safety of trains will be reduced, which is not conducive to passing through the curve track. Therefore, the damping coefficient should be smaller so that the train can easily pass through the curve track.

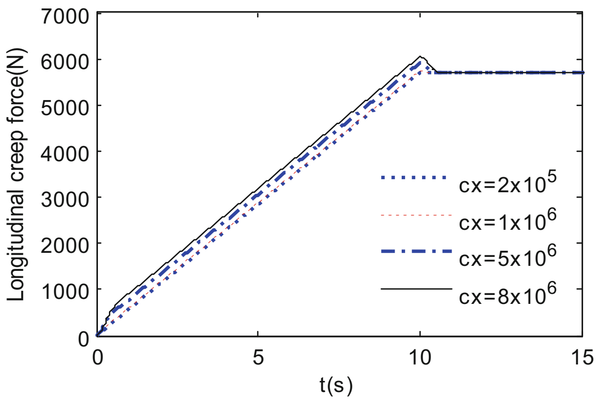
3 Research and Simulation of Semi-Active Control Based for a Whole Car Model

In the second section, the single wheelset is simulated under different damping coefficients. In this section, a vehicle model is built for more in-depth and accurate research. The model of a certain-type high-speed train is established as shown in Fig. 4 by using the UM dynamics simulation software.

The wheel tread is LMA type. The rail profile is T60. The train speed is 200 km/h. Dynamic simulation analysis is carried out using the UM software. The curve track is used. The radius is 1500 m.



(a) Lateral displacement of wheelset

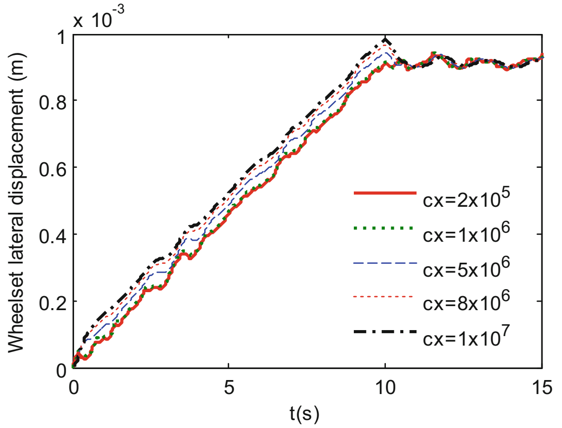


(b) Longitudinal creep force

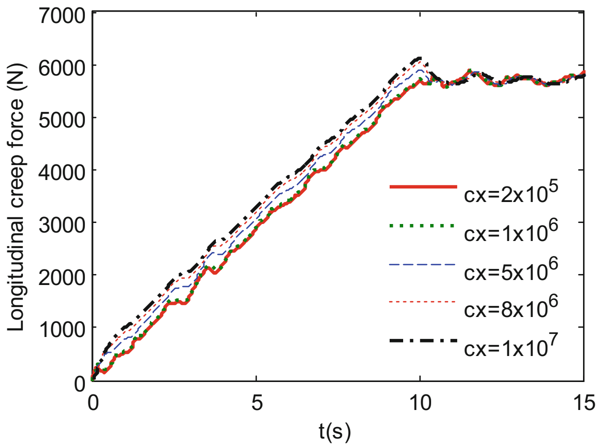
Fig. 2 Parameters using different damping coefficients under ideal track. (**a**) Lateral displacement of wheelset. (**b**) Longitudinal creep force

3.1 Control Strategy Design

Analyzing critical speed is a common way of evaluating train hunting stability. This chapter analyzes the stability of the train on the straight track and safety on the curve track. The critical speeds of the train are obtained under different damping coefficients of yaw dampers. The maximum derailment coefficients under different damping coefficients are obtained when the train runs on the curve track. Derailment coefficient which is used for evaluating train safety is defined as the vertical force on the wheel divided by the lateral force. In order to make the results clear and easy to understand, the simulation data are standardized to ensure them in the range of 0–1.The standardized formula is shown in (2). After normalization of simulation



(a) Lateral displacement of wheelset under the random excitation



(b) Longitudinal creep force under the random excitation

Fig. 3 Parameters using different damping coefficients under random excitation. (**a**) Lateral displacement of wheelset under random excitation. (**b**) Longitudinal creep force under random excitation

data, the two indexes are put on a picture for analysis as shown in Fig. 5.

$$f_{(0\sim1)} = \frac{x - X_{\min}}{X_{\max} - X_{\min}} \tag{2}$$

where $X_{\min}$ is the minimum value of sample data, $X_{\max}$ is the maximum value of sample data, and x and f are the values before and after standardization, respectively.

Fig. 4 Vehicle model built by UM software

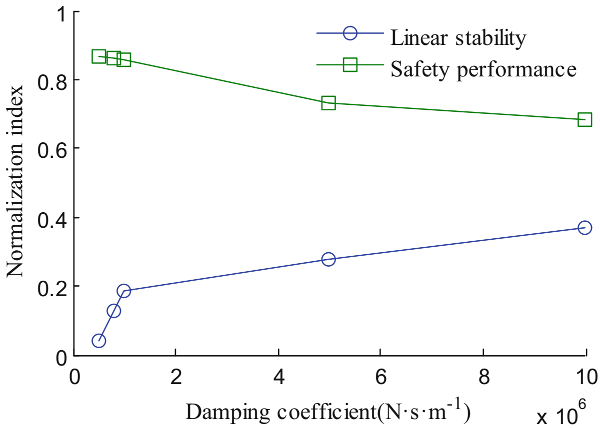


Fig. 5 Comparison chart of normalized indexes

It can be seen from Fig. 5 that the stability is getting higher and higher with the increase of damping coefficient, but the safety performance becomes worse and worse as the train passes through the curve track. Therefore, we need to control the damping coefficient. When a train passes through a straight track, in order to ensure the stability of the train, the damping coefficient needs to be increased, whereas when the train enters a curve track, the damping coefficient is reduced to improve the train running safety. Figure 6 shows the idea of damping control directly.

While, how to identify the train entering the curve track and control the damping has become a core problem of the semi-active control strategy. It is generally known that there will be a height difference between the two sides of a train when the train enters a curve track, while the height difference between the two sides of the train

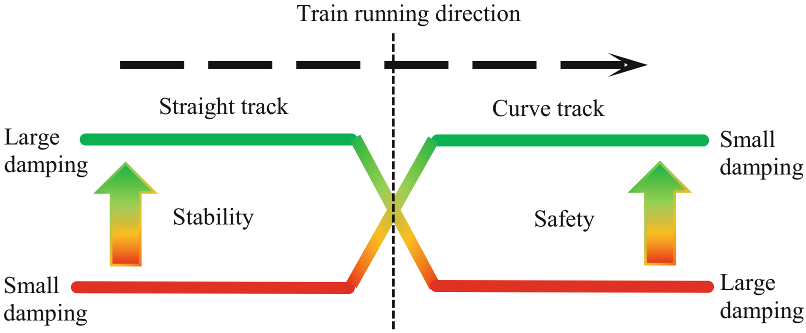


Fig. 6 Damping control diagram

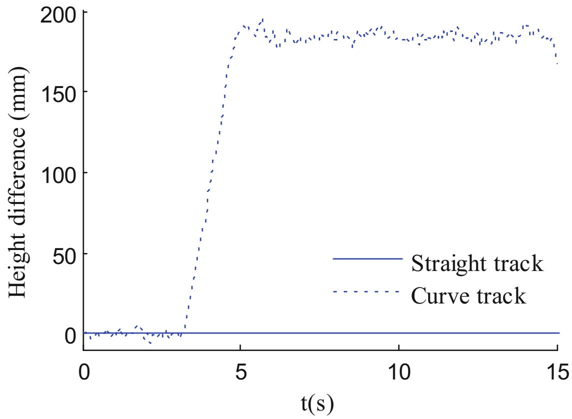


Fig. 7 Comparison of height difference between two sides of frame on straight track and curve track

on the straight track is close to zero. Therefore, the height difference is used as a control strategy to adjust the damping coefficient as shown in Fig. 7.

Therefore, the control strategy is as follows:

$$c = \begin{cases} c_{\max} & \Delta h \le 0 \\ c_{\min} & \Delta h > 0 \end{cases} \tag{3}$$

where Δh is the height difference between the two sides of the bogie.

When a train runs on a straight track with excitation, the height difference between the two sides will not be exactly equal to 0, and it will fluctuate within a threshold. After simulation, the threshold value obtained is 5 mm. Therefore, the control strategy is as follows:

$$c = \begin{cases} c_{\max} & \Delta h \le \delta \\ c_{\min} & \Delta h > \delta \end{cases} \tag{4}$$

where δ is the threshold.

This kind of control strategy is a switch-type control strategy. When the damping coefficient is switched, the shock absorber may be impacted. The continuous control strategy is discussed in this chapter. In other words, an intermediate damping coefficient is added between the maximum and minimum damping coefficients to mitigate the impact, which is formed by a first-order function, the independent variable of which is height difference between the maximum and minimum damping coefficients to mitigate the impact. The control strategy is as follows:

$$c = \begin{cases} c_{\max} & \Delta h \le \delta \\ a\Delta h + b & \Delta h > \delta \text{ and } \Delta h < \sigma \\ c_{\min} & \Delta h \ge \sigma \end{cases} \tag{5}$$

where a and b are the slope and intercept values of the first-order function respectively, and σ is a threshold which is bigger than δ, 10 mm in this paper.

In order to apply semi-active control strategy to the vehicle, it is imported into MATLAB/Simulink in the form of S-function. The co-simulation model built is shown in Fig. 8. The control strategy is applied to the yaw dampers in Simulink.

In this work, different damping coefficients of yaw dampers are used to simulate when the model is running on the curve track. For each damping value, a corresponding peak value of derailment coefficient is obtained. The simulation curve is shown in Fig. 9.

When the damping coefficient is 2×10^5 N s m^{-1}, the peak value of derailment coefficient is minimum and the train safety is best. The minimum damping coefficient for the control strategy is obtained based on the maximum derailment coefficient in the simulations according to Fig. 9; the maximum damping coefficient obtained is 5×10^6 N s m^{-1}. It can be specifically expressed as follows:

$$c = \begin{cases} c_{\max} & \Delta h \le 5 \\ -958000 & \Delta h + 9790000 \quad \Delta h > 5 \text{ and } \Delta h < 10 \\ c_{\min} & \Delta h \ge 10 \end{cases} \tag{6}$$

3.2 The Simulation Based on Semi-Active Control

The control strategy is applied to the UM model and the semi-active control simulation is carried out. The curving performance of the model is analyzed, and the comparison curves of derailment coefficient, wheel axle lateral force, and wear work are obtained as shown in Fig. 10.

It can be seen from Fig. 10 that the maximum derailment coefficient decreases by 41.75%, the maximum value of wheel axle lateral force by 17.65%, and the maximum wear power by 57.67%. The safety is enhanced and the curving performance is improved by using the semi-active control strategy. Meanwhile, the

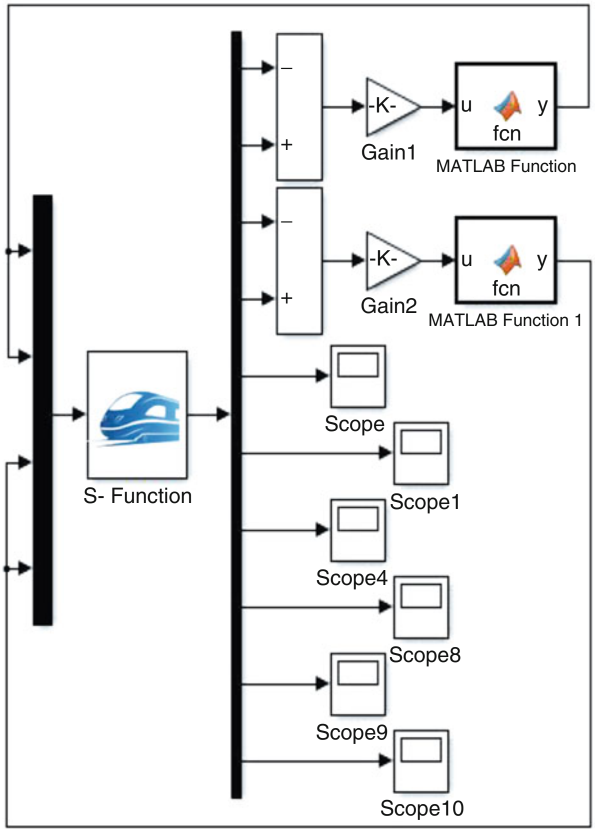


Fig. 8 The co-simulation model

wheel rail wear is reduced substantially. Therefore, the semi-active control strategy proposed in this chapter is effective.

The lateral acceleration curve of the train is obtained as shown in Fig. 11. Through comparison, it is found that the influence of semi-active control on the lateral acceleration is not obvious.

In general, the control strategy proposed in this chapter can improve the curving performance of the train. Meanwhile, it can ensure the stability of the train and the contradiction between the stability and curving performance is solved.

4 Hardware in the Loop Experiment

The semi-active control experiment is analyzed by the MR damper hardware in the loop test. Hardware in the loop (HIL) means that the research object is made into a

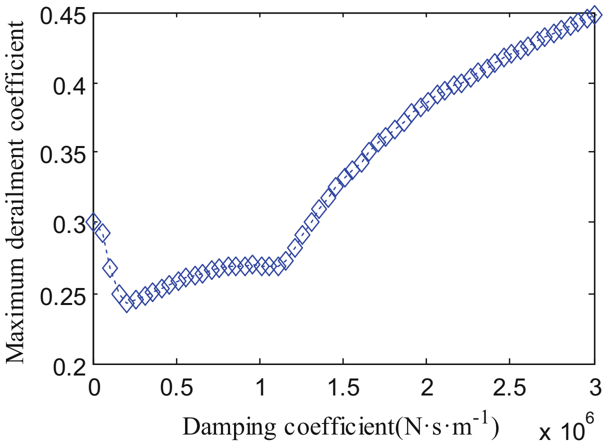


Fig. 9 Optimization of damping coefficient on curved track

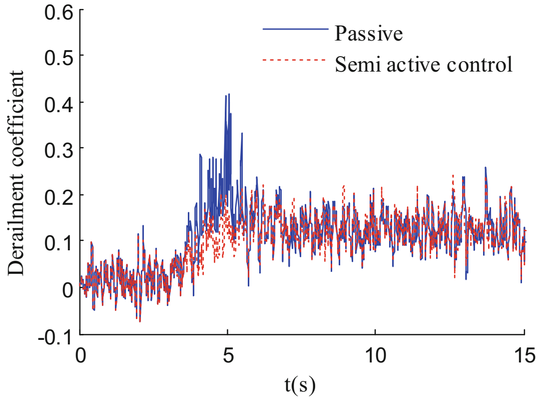
real object instead of a virtual model. The shock absorber is embedded in the two degree of freedom model in the work. The real shock absorbers are used to verify the effectiveness of the control strategy. The schematic and physical configuration of HIL test are shown in Fig. 12.

The experimental platform system built by the research team mainly includes upper computer/lower computer, servo electric cylinder, drive controller, force sensor, magnetorheological damper, input/output board, and so on. In the experiment, the RTX real-time simulation system of the ADI company is adopted. The drive controller receives the analog voltages from I/O boards and sends them to the electric cylinder making the MR damper move. The signal received by the electric cylinder comes from the relative velocity at both ends of the shock absorber in the model. The force generated by the real damper is returned to the model through the sensor. Thus, the whole system forms a loop. It can be close to the real operating conditions. In order to apply the control strategy to the test system, damping values in the control strategy need to be represented by the current values.

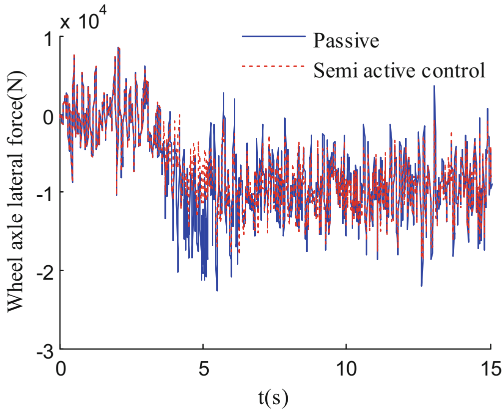
The lateral displacement and longitudinal creep force of wheelset are obtained by using the test to evaluate the effectiveness of the control strategy as shown in Figs. 13 and 14. In order to match the real shock absorber, the model needs to be scaled and random excitation is applied.

It can be found in the test from the Figs. 13 and 14 that the control effect of semi-active control is better than passive control under the condition of random excitation. This proves the effectiveness of the control strategy proposed in this chapter.

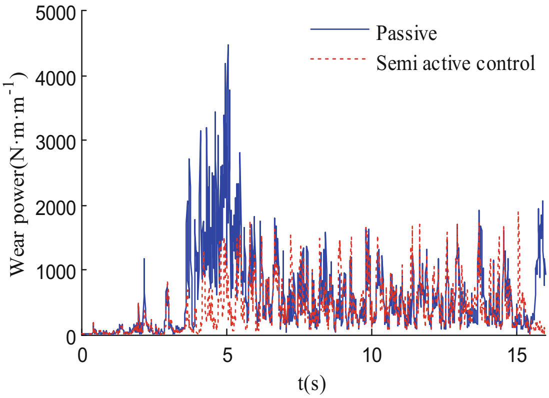
Fig. 10 Comparison of curving performance under different control. (**a**) Comparison chart of derailment coefficient. (**b**) Comparison chart of wheel axle lateral force. (**c**) Comparison chart of wear power



(a) Comparison chart of derailment coefficient

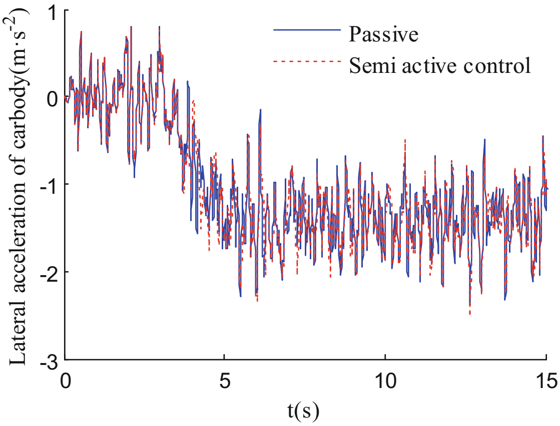


(b) Comparison chart of wheel axle lateral force



(c) Comparison chart of wear power

Fig. 11 Comparison of car body acceleration under different control



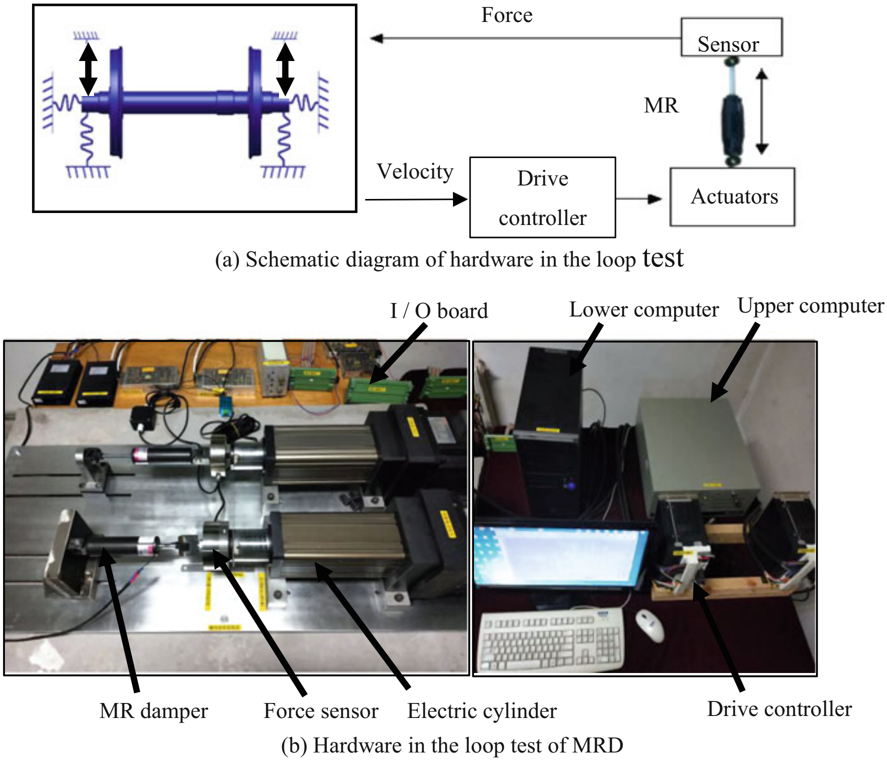


Fig. 12 Physical and schematic diagrams of hardware in the loop test. (**a**) Schematic diagram of hardware in the loop test. (**b**) Hardware in the loop test of MRD

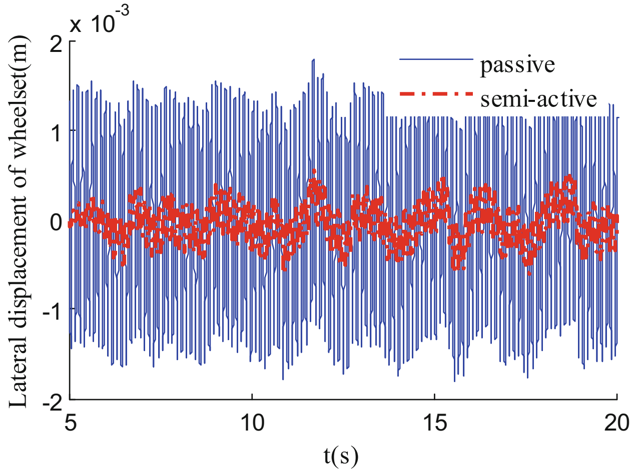


Fig. 13 Lateral displacement of wheelset obtained from the HIL test

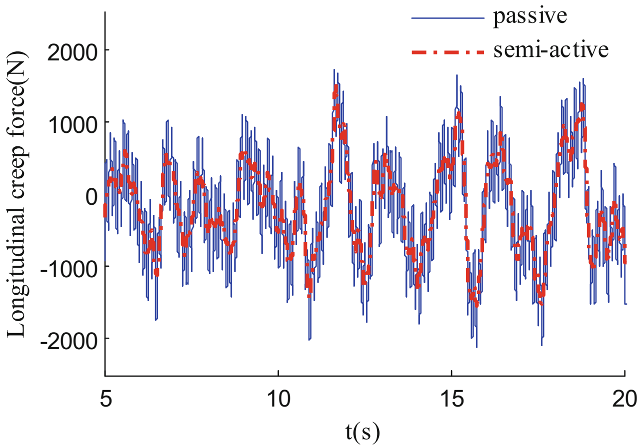


Fig. 14 Longitudinal creep force obtained from the HIL test

5 Conclusions

The results are here summarized.

1. The single wheelset model is established, and the curving performance of wheelsets is analyzed by changing the damping coefficient of yaw dampers. The results show that it is easy to pass the curve track by reducing the damping coefficient.

2. A new type of semi-active control is proposed through vehicle dynamics simulation. The UM-MATLAB simulation results show that the safety is enhanced and the curving performance is improved by using the semi-active control strategy. The contradiction between the stability and curving performance is solved.
3. In this chapter, the hardware in the loop method is used to analyze the new semi-active control strategy. The results show that the experimental results are consistent with the theoretical results, which prove that the control strategy proposed in this chapter is practical.
4. It is very interesting to study semi-active control strategies. The work discussed in this chapter is just the beginning. The most important thing is that the research results should be applied to the engineering practice. In the future, the control strategy will be materialized, the controller will be developed, and the real vehicle test will be carried out.

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