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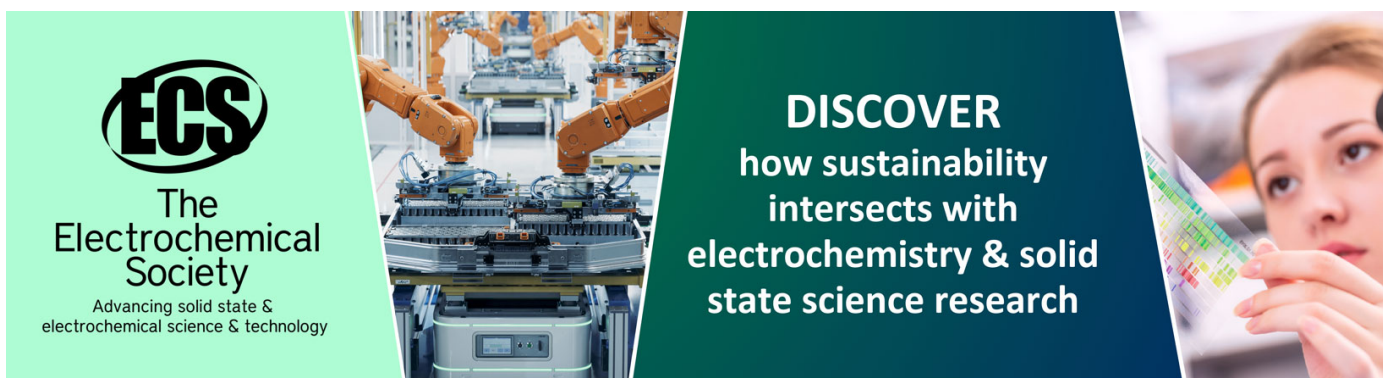
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To cite this article: Chengqiang Wang *et al* 2020 *IOP Conf. Ser.: Mater. Sci. Eng.* **780** 022027

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Study on line Vibration Characteristics of Equipment for Electric Multiple Units

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Abstract. The rigid-flexible vibration of high-speed train will cause the violent movement of equipment. This paper first introduced the high-speed EMU equipment and the theory of mass tuning vibration absorption. Then according to the test data, the vibration response of the bogie frame and the vibration response of the equipment, and the vibration response of the equipment under different lines are analyzed. The results show that the lateral vibration frequency of the frame is 4.5 Hz in normal section, and the lateral vibration frequency of the frame is 7.4 Hz in hunting excitation section. During the normal vibration line, the vibration of the equipment is significantly larger than that of the carbody, the vibration characteristics of the frequency domain of the equipment is mainly high-frequency magnetic vibration, the high-frequency vibration of the equipment above 15Hz is not transmitted to the carbody. When the running speed is increased, the vibration of the equipment is enhanced, and the line condition also has an important influence on the vibration of the equipment.

1. Introduction

There will be rigid-flexible coupling vibration in the multi-body system of railway vehicle, which is typically the structural vibration of vehicle body and its coupling with other connecting structures. The most prominent examples of elastic vibration of high-speed trains are the low-frequency elastic vibration and high-frequency structural noise of EMU, in which the low-frequency elastic vibration of the carbody is mainly the coupling vibration with the equipment under the vehicle. Many scholars have carried out research on this study. Wu Pingbo et al. considered the elasticity of the car body and calculated the dynamic response of the carbody [1]. Zhou JS calculated the influence of elastic carbody on passenger comfort by considering the flexibility of carbody [2]. Zeng jing established the vertical vibration model of the carbody and calculated the amplitude frequency characteristic curve of the vehicle system. At the same time, the random vibration analysis of the vehicle system with or without the dynamic absorber is carried out[3]. The results show that the dynamic absorber can suppress some frequency components of the carbody, and properly increase the mass, stiffness and damping of the absorber can achieve good damping effect. After considering the elastic response of the carbody, the carbody is considered as a constant cross-section Euler beam, and the secondary suspension is considered as a semi-active suspension. Considering the first-order and second-order elastic modes of the carbody, the first-order bending natural frequency of the carbody should avoid the hunting frequency of the frame. Due to the influence of the carbody elasticity, the vibration at the end of the carbody is greater than that in the middle of the carbody [4]. Ren zunsong used the rigid flexible



coupling model to calculate the vertical transmission law of EMU. Zhou Jinsong used the dynamic vibration absorber to restrain the elastic vibration of the carbody[5]. The optimal design of the dynamic vibration absorber can effectively control the elastic vibration of the carbody, and the larger the mass of the dynamic vibration absorber is, the better the vibration absorption performance is. When the mass of the dynamic vibration absorber is 1000 kg and the vertical first-order bending frequency of the car body is as low as 6.5 Hz, the high-speed bus with a speed of 250 km/h could have a good ride index [6]. Takigami has studied the elastic vibration of the carbody with piezoelectric elements [7-8]. Huang Caihong considered the elasticity of the car body, established the vertical model of vehicle track coupling, calculated the dynamic response of the elastic vibration of the carbody and the factors considered in the suspension design. Yang Guangwu et al. [10] calculated the vehicle body mode when considering the equipment under the vehicle, and compared it with the vehicle body mode without considering the equipment under the vehicle. Shi Huailong used the rigid flexible coupling model to calculate the influence of the elastic vibration of the carbody on the vibration of the equipment [11-12].

This paper first introduces the theoretical analysis of mass tuning and vibration absorption of the suspension equipment under the carbody, and then analyzed the vibration condition of the equipment under the train under normal line and hunting excitation line, as well as the impact of the operation speed and operation mileage on the suspension equipment under the train through the measured data of a high-speed EMU in China.

2. Mass tuning and vibration absorption theory of suspension equipment under vehicle

2.1. Introduction of suspension equipment under high speed EMU

According to the statistics of the types and functional characteristics of the equipment under the EMU, in order to design the connection parameters of the elastic elements, the equipment is classified according to the parameter design principles. At present, the common under vehicle equipment mainly includes high-frequency magnetic vibration equipment, rotating parts equipment, high-quality equipment and other passive equipment.

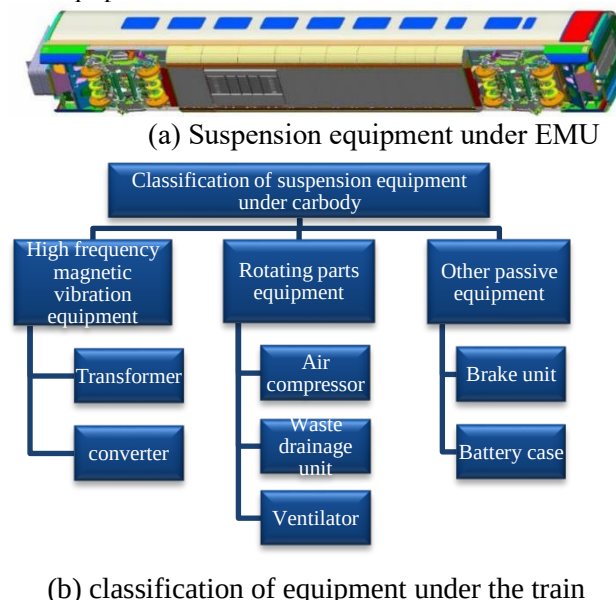


Figure 1 Equipment compartment and equipment type under EMU

2.2. Mass tuning vibration absorption theory

According to the vertical coupling vibration relationship between the elastic beam body model and the equipment, the model is shown in Figure 2. The mechanical model combined with the dynamic

vibration reduction of the elastic system is simplified as a hybrid dynamic system composed of the elastomer and discrete mass. The dynamic vibration absorption theory can be applied to match the modes of the equipment under the car and the elastic carbody, so as to restrain the elastic vibration of the carbody to the greatest extent and improve the elastic mode of the carbody.

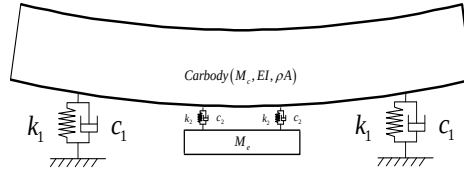


Figure 2 Coupling vibration model of carbody and equipment

R.g.jacquot used the modal analysis method, modal truncation and modal synthesis method to give the approximate solution of dynamic vibration absorption of general elastic components. For a uniformly straight elastic beam, if only its first main mode is considered, it is equivalent to solving the dynamic absorption problem of a single degree of freedom vibration system with a main mass of ρAl excited by a concentrated force $e^{j\omega}$. The amplitude frequency characteristic method of the discrete vibration system is adopted for the dynamic absorption of the uniform straight beam. The equation of motion of the whole system can be derived by using the mode truncation method and the mode synthesis method. Then the dynamic amplification coefficient of the generalized coordinate x_1 to the generalized force can be obtained. After the dynamic absorber is installed, the dynamic amplification coefficient is equal to the dynamic amplification coefficient of the dynamic absorption of the single degree of freedom system with the main mass of M . The main mass calculation formula is as follows:

$$M = \frac{\rho Al}{Y_1^2(a)} \quad (1)$$

Where: ρAl is the mass of elastic vehicle body, ρ is the density of unit length, A is the cross-sectional area of carbody, l is the length of carbody; $Y_1(a)$ is the first mode function of elastic beam.

The above formula shows that if the dynamic vibration absorber is installed at the position with the maximum vibration amplitude of the vehicle body, its equivalent mass M can be minimized. Compared with the determined mass m of the absorber, the mass ratio $\mu = m/M$ will get the maximum value, so the best vibration absorption effect can be obtained. The calculation of the optimal frequency ratio and damping ratio of the dynamic vibration absorber of the uniform beam is as follows:

$$f_{opt} = \frac{1}{1 + \mu Y_1^2(a)} \quad (2)$$

Where, μ is the mass ratio of the equipment to the elastic quantity; $Y_1(a)$ is the first mode function of the elastic beam; f_{opt} , ζ_{opt} are respectively the optimal suspension frequency ratio (the first bending frequency of the vehicle body) and damping ratio of the equipment.

According to the free vibration differential equation of the uniform beam, the vibration mode function can be derived. Combined with the boundary constraints of the free beam, the frequency equation and the natural frequency form can be obtained as follows:

Natural frequency equation of free beam:

$$1 - \cos \lambda_i \cosh \lambda_i = 0 \quad (3)$$

Free beam boundary conditions:

$$\ddot{Y}(0) = \ddot{Y}(l) = 0, \quad \ddot{Y}(0) = \ddot{Y}(l) = 0 \quad (4)$$

Substituting formula (4) into formula (3), the natural frequency is obtained:

$$\lambda_i = (2n+1)\pi/2 \quad (5)$$

According to the vibration differential equation of free beam, the vibration mode function of elastic uniform beam is obtained as follows:

$$Y_i(x) = \cosh \beta_i x + \cos \beta_i x - A(\sinh \beta_i x + \sin \beta_i x) \quad (6)$$

Where: $A = (\cosh p_i x - \cos p_i x) / (\sinh p_i x - \sin p_i x)$, $\beta_i = \lambda_i / l$ 。

2.3. Design criteria for equipment suspension stiffness

The equipment is mainly divided into three categories: A, B and C, Different design criteria are adopted for the design of the three categories of equipment, and the specific classification is as follows:

A. The vertical direction of the active equipment is about 7Hz, which is designed based on the vibration isolation theory. The frequency of lateral suspension is about 0.5 ~ 0.7 times of vertical suspension, and the frequency of longitudinal suspension is about 3 times of vertical suspension.

B. Passive high-quality equipment with a vertical direction of about 9Hz is designed based on the theory of mass tuning and vibration absorption, and optimized by dynamics and finite element method. The frequency of transverse suspension is about 0.7-1 times of vertical suspension, and the frequency of longitudinal suspension is about 3 times of vertical suspension.

C. The passive small mass vertical equipment is about 10 ~ 11Hz. Based on the vertical simplified beam model, the dynamic vibration absorption theory is designed, and the connection reliability is considered. The frequency of transverse suspension is about 0.7-1 times of vertical suspension, and the frequency of longitudinal suspension is about 3 times of vertical suspension.

3. Test process

Based on the long-term dynamic performance test of 350km/h EMU line, the vibration characteristics of equipment under the train are analyzed. The lateral and vertical acceleration of equipment and vehicle body under different bogie motion conditions are given, and the spectrum composition and vibration source are analyzed.

The maximum continuous operation speed of the train is 350km/h, and the wheel wear mileage is about 200000km. The sampling of the frame acceleration is 2kHz, and the band-pass filtering is 0.5-12Hz. The acceleration data of the equipment under the vehicle is sampled for 1kHz, with band-pass filtering (0.1~200Hz IIR4).

Select whether the equipment has significant elastic vibration time interval data for analysis, the lateral and vertical vibration acceleration of the vehicle body measuring points and equipment measuring points at the connection between the equipment and the vehicle body in local time interval are analyzed, compared the time-domain and frequency-domain signal characteristics, and summarize the dynamic response characteristics of the line vibration of the equipment.

This paper analyzes the vibration characteristics of typical under carbody equipment components, such as auxiliary converters, and gives the lateral stability analysis results of bogies in corresponding sections. This paper mainly analyzes the time-domain and frequency spectrum characteristics of equipment and carbody with and without hunting excitation vibration.

4. Analysis of test results

In order to explore the influence of the elastic vibration of the car body on the vibration of the equipment, the data of the normal road section and the elastic vibration significant section were intercepted respectively, and the vibration response of the frame, the vibration response of the equipment, and the vibration characteristics of the the line under the equipment was compared.

4.1. Frame vibration response

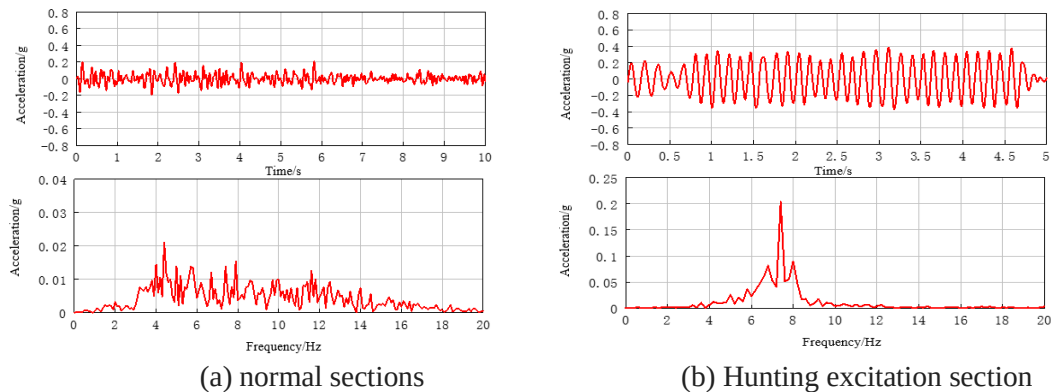


Figure 3 Lateral stability of the frame (0.5-12Hz filter)

The time-domain diagram and frequency domain diagram of the lateral acceleration of the end section of the normal section and the hunting excitation section are shown in Figure 3. In the time domain diagram, it can be seen that the lateral acceleration of the normal section frame is small, the maximum value is 0.2g, and the harmonic vibration of the hunting excitation section structure is obvious, and the maximum value is about 0.4g. The main frequency of the normal section is 4.5 Hz and the amplitude is 0.021 g. The lateral frequency of the elastic significant section is 7.4 Hz and the amplitude is 0.2 g.

4.2. Vibration frequency response analysis of equipment

The mass is 840kg, which is 1.84m away from the center of the carbody. The elastic component parameters (vertical design frequency 9.5Hz) are designed based on the quality tuning vibration absorption principle. The acceleration time domain waveform of the whole process is as follows: the vertical and lateral acceleration amplitudes of the equipment are 0.75g and 0.25g, respectively. The vertical vibration is significantly larger than the lateral direction; the acceleration amplitude at the carbody measurement point is about 0.1g.

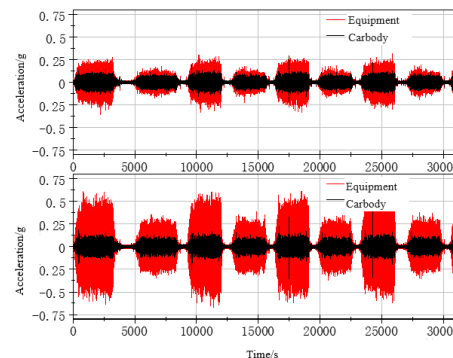
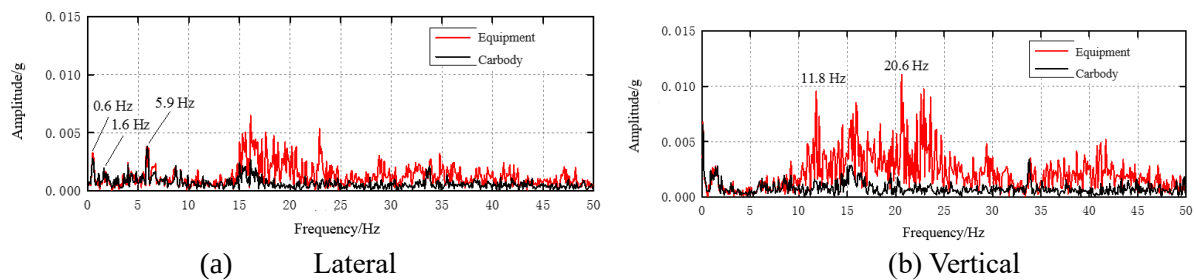
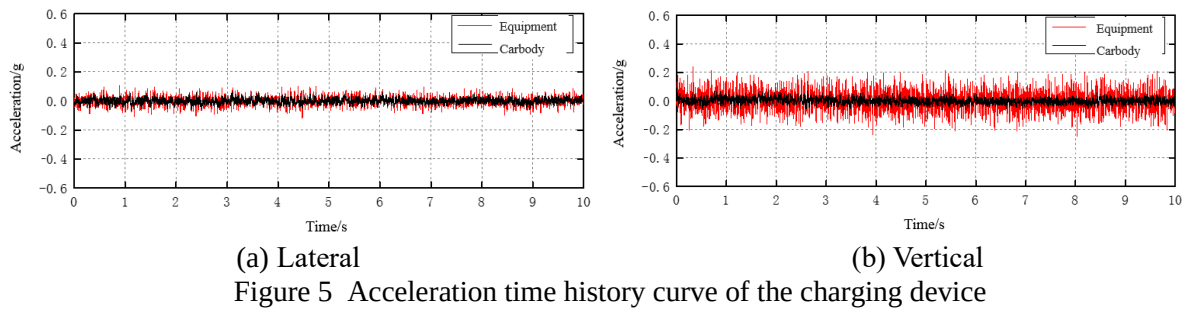
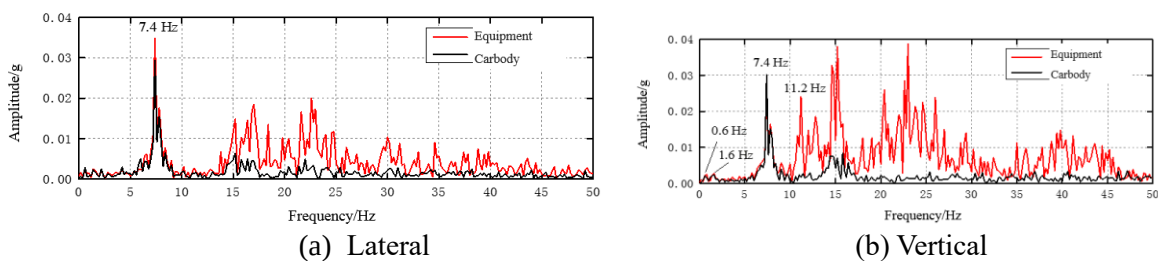
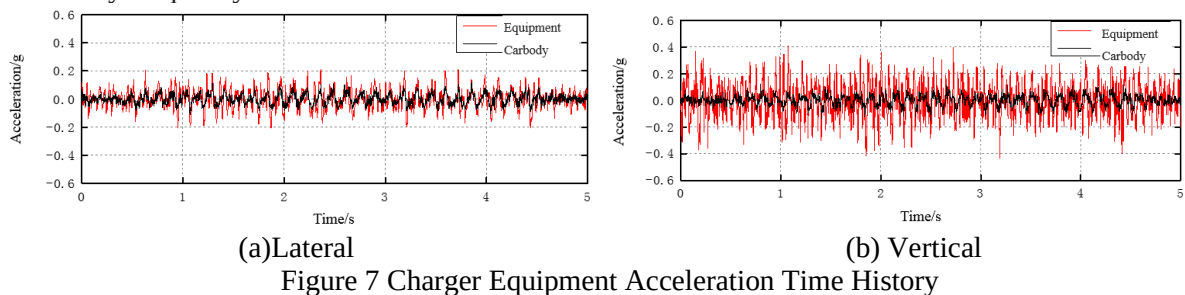


Figure 4 Charging machine equipment measuring point acceleration time history (lateral, vertical)

The data with low vibration level is selected for analysis, and the lateral and vertical vibration accelerations of the carbody measurement point and the equipment measurement point at the connection between the equipment and the carbody in the normal period are analyzed, and the time domain and frequency domain signal characteristics are compared, and the device is summarized. Line vibration characteristics. The horizontal and vertical amplitudes of the equipment are about 0.1g and 0.2g; the lateral and vertical amplitudes of the carbody are both 0.05g. It can be seen that the vibration of the equipment is significantly larger than that of the carbody, the high-frequency vibration of the equipment above 15Hz is not transmitted to the carbody, that is, effectively separated by elastic elements. The main frequency is 0.6Hz and 1.6Hz corresponding to the rigid body mode of the carbody.



The data of the bogie excitation period is selected for analysis: during the bogie excitation period of the bogie, the corresponding elastic vibration amplitude is significantly larger than the rigid body motion amplitude; the horizontal and vertical amplitudes of the equipment are 0.2g and the high-frequency vibration of the 0.4g device above 10Hz is not Passed to the car body, that is, effectively separated by the elastic element; the inherent suspension frequency of the device is not seen, and the main vibration characteristics are the same as the car body, which are all manifested as the lateral instability frequency of the frame.



4.3. Impact of operation speed

Figure 9 shows the influence of vehicle speed on the vibration of equipment under the vehicle. The vibration data of vehicle body and equipment under the vehicle (sleeper beam, underframe, apron, equipment under the vehicle) shows that when the vehicle speed is 380km/h or above, the vibration level is higher than 350km/h. When the speed is 350km/h, the lateral vibration acceleration is less than 300km/h and 380km/h. The vertical acceleration increases with the increase of velocity.

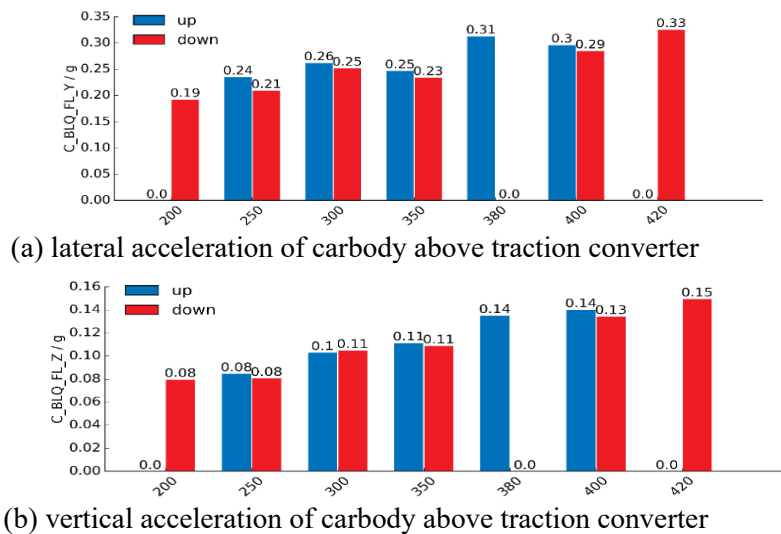


Figure 9 Effect of running speed on vibration of suspension equipment under the carbody

5. Conclusion

Based on the theoretical analysis of the coupling vibration of the high-speed EMU body and the equipment under the vehicle and the analysis of the test data, the following conclusions can be drawn:

(1) The main frequency and amplitude of the transverse vibration of the frame in the normal road section are 4.5Hz and 0.021g, while the main frequency and amplitude of the lateral vibration of the frame in the serpentine excitation road section are 7.4Hz and 0.2g respectively.

(2) During the normal period of vibration level, the vibration of equipment is significantly greater than that of carbody. The frequency-domain vibration characteristics of equipment are mainly high-frequency magnetic vibration, and the main frequency of time-domain waveform display is 100Hz, the high-frequency vibration of equipment above 15Hz is not transmitted to carbody. During the period of bogie hunting excitation, the coupling vibration frequency of auxiliary converter and vehicle body is 7.6Hz, 8.6Hz and 9.6Hz; the vibration phase of equipment and vehicle body is basically the same.

(3) With the increase of operation speed, the vibration of equipment under the vehicle is gradually enhanced.

Acknowledgment

The project was supported by the National Key R&D Program of China [grant no. 2018YFB1201702]. The Program of State Key Laboratory of Traction Power (Grant No.2019TPL_T20) and the Fundamental Research Funds for the Central Universities (Grant No.2682017CX010).

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