Experimental Analysis of the Car Body Suspended Equipment Vibration for High-speed Railway Vehicles

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Abstract. Existing researches state that the rigid-flexible coupling vibration of the car body and car body suspended equipment deteriorates the ride comfort of a high-speed train, but a few works validate it through field tests. In this investigation, field measurements on the vibration of bogie, car body and car body suspended equipment of a high-speed train was carried out to reveal their coupling interaction characteristics. A design solution for the equipment suspension frequency is then summarized by its weight and excitations bears on. Tests show that bogie dominates a lateral vibration at 4.5 Hz in normal track segments but it rises to 7.4 Hz while hunting occurs. The equipment acceleration is significantly larger than that on car body since the high-frequency vibration bears on it is well isolated from the car body. When bogie hunting occurs, a structural resonance on the car body was noticed, and the lateral and vertical acceleration on equipment rise to 0.2 g and 0.4 g, respectively, with respect to 0.1 g and 0.2 g in a normal case. The vibration discrepancy on the equipment among track lines is quite obvious.

1. Introduction
Rigid-flexible coupling vibration occurs sometimes on modern railway vehicles, which typically involves the structural vibration of a car body and coupling vibration of other connected structures. The most prominent example of the flexible (or elastic) vibrations on the car body is the low-frequency elastic vibration and high-frequency structural noise. The low-frequency flexible vibration of the car body is always coupled with the vibration of the car body suspended equipment, which has been widely observed on-track and determines the ride comfort of a passenger coach.

Wu et al.[1] and Zhou et al.[2] considered the elasticity of the car body in the vehicle model, and its dynamic response was compared with that of a rigid vehicle model. The elastic vibration of the car body deteriorates the riding comfort of the high-speed train. Zeng[3] established a vertical dynamic model of passenger cars and calculated the frequency characteristics. The study[4] shows that the first bending natural frequency of the car body should be far away from the natural frequency of the bouncing of bogie frame. Due to the influence of the car body elasticity, the vibration at the end of car body is greater than that in its middle part of the car body. Ren and Liu[5] calculated the vertical transfer law of electric multiple units (EMUs) using a rigid-flexible coupling model. Zhou et al.[6] used a dynamic vibration absorber to suppress the elastic vibration of the car body. The greater the mass of the dynamic vibration absorber, the better the vibration damping performance. Takigami and Tomioka[7,8] studied the elastic vibration of the car body after using piezoelectric elements. Huang et al.[9] considered the car body elasticity, established a vehicle-track coupled vertical model and analyzed the dynamic vibration response of the car body and the factors considered in the suspension design. Shi et al.[10-11] studied the
influence of suspension parameters of the car body suspended equipment on the elastic vibration of the car body through a rigid-flexible coupling vehicle model. An active control actuator mounted on the chassis of car body contributes to improving bending stiffness which in turn improves the ride comfort[12]. Damping layers are placed to the car body structure to add more damping for the bending mode[13-14]. Also, a stiff rod can be added to stiff the car body against bending[15].

In this context, a field test on the vibration of bogie, car body and car body suspended equipment of a high-speed EMU in China was carried out to show their evolution with the running mileage and track irregularities. The vibration status of the car body suspended equipment in normal situation and hunting excitation situation is analyzed through analyzing the on-track measured data, and corresponding method for car body vibration reduction are discussed.

2. Categories of car body suspended equipment
An electric multiple unit (high-speed railway vehicle) is usually equipped with many functional devices or equipment under the car body chassis. For a generality, the car body suspended equipment is mainly divided into three categories, grouped as A, B and C according to their weight and excitation bears on them.

For the group A which has a small mass but bears severe vibrating excitations, its vertical suspension frequency is suggested to be around 7 Hz based on the vibration isolation theory to isolate the excitation bears on it to be transit to the car body can deteriorate the ride comfort. The lateral and longitudinal suspension frequencies are about 0.5~0.7 times and 3 times the vertical frequency, respectively, according to the rubber joint prototypes used by the high-speed train.

Regarding the group B which has a relatively large mass, its vertical suspension frequency prefers about 9 Hz based on the dynamic vibration absorption theory, in which the car body bending mode is assumed has a frequency 10~12 Hz. This suspension frequency is optimized to restrain the elastic vibration on car body bending[10-11]. The lateral and longitudinal suspension frequencies are about 0.7~1 time and 3 times the vertical frequency, respectively, according to the rubber joint prototypes used by the high-speed train.

For the group C which has a relatively small mass without any vibrating sources, its vertical suspension frequency is suggested to be selected based on the dynamic vibration absorption design regarding the car body bending or a rigid connection between the equipment and car body shall be employed considering the connection reliability. The lateral and longitudinal suspension frequencies are about 0.7~1 time and 3 times the vertical frequency or rigid connections shall be employed as the vertical suspension use.

3. On-track vibration test specifications
In order to understand the vibration transition between the two suspension systems and the coupled vibrating between the car body and its suspended equipment, an on-track vibration test was designed and carried out. Accelerations on the wheelset axle-box, bogie frame, connections between the car body and its suspended equipment were measured through accelerometers and displacement sensors, which were connected to the data acquisitions system in the equipment cabin through cables. The on-track vibration test was designed for long-term and continuously vehicle dynamics recording.

Based on the long-term dynamics performance test for EMUs, the vibration characteristics of the equipment are analyzed. The lateral and vertical accelerations of the equipment and the car body under different bogie running performances are measured, and then the spectrum composition and the vibration source are analyzed. The train has a continuous operating speed of 250~350 km/h and a wheel wear mileage of 200,000 km. The sampling frequency of the frame acceleration is 2 kHz and the passband frequency range are 0.5-12 Hz. The sampling frequency of the equipment acceleration is 1 kHz and the passband frequency range are 0.1~200 Hz.
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 in the normal section and the significant elastic vibration section were intercepted respectively.

4.1. Vibration analysis on the bogie
The time-domain diagram and frequency domain diagram of the lateral acceleration at the end of the bogie frame in the normal track segments and hunting occurs segments are shown in Fig. 1. In the time domain diagrams, it can be seen that the lateral acceleration in the normal section frame is much smaller than that in the hunting excitation section, the maximum value is 0.2 g and about 0.4 g respectively. The dominant frequency of the normal section is 4.5 Hz with the amplitude of 0.021 g, while the lateral frequency in the elastic significant section is 7.4 Hz with the amplitude of 0.2 g.

4.2. Vibration analysis of the car body and equipment
As presented in section 2, the auxiliary inverter has a mass of 840 kg without strong vibrating source and is grouped as C type equipment. Figure 2 illustrates the time history of acceleration exerted on the equipment and car body for one single trip with approximately nine hours. It is seen that the vertical and lateral acceleration of the equipment are 0.6 g and 0.25 g, respectively, which means that that vertical vibration is significantly larger than the lateral vibration. While the acceleration amplitudes of the car body along two directions both are about 0.13 g.

Figure 1. Lateral acceleration on the bogie frame (0.5-12Hz bandpass filtered) when the vehicle was operated at (a) normal track segments and (b) hunting occurs segments.

Figure 2. Time domain acceleration of the equipment and car body in (a) lateral and (b) vertical directions
The time and frequency domain characteristics are compared as shown in Figs. 3 and 4. The lateral and vertical amplitudes of the equipment are about 0.1 g and 0.2 g. The lateral and vertical amplitudes of the car body are both 0.05 g. It can be seen that the equipment vibration is significantly larger than that of the car body. The high-frequency vibration of the equipment above 15 Hz is not transmitted to the car body, which is effectively separated by the elastic suspension of equipment. The dominant frequency is 0.6 Hz and 1.6 Hz corresponding to the rigid mode of the car body.

The acceleration of the equipment and car body during the bogie hunting occurs are illustrated in Figs. 5 and 6. The corresponding elastic vibration is significantly larger than the rigid motion on the car body. The lateral and vertical acceleration of the equipment are 0.2 g and 0.4 g, respectively. The high-frequency vibration of the equipment above 10 Hz is not transmitted to the car body, which is effectively separated.
4.3. Vibration difference on various track lines

The vehicle was operated on different track lines with the similar operating speed (250~350km/h), and Figure 7 shows the acceleration of the equipment on these lines. The lateral vibration level of line 1, line 2 and line 3 are about 0.2~0.25 g, 0.25 g, and 0.18 g respectively. It can be seen that the vibration acceleration on line 3 is significantly smaller than that on line 1 and line 2. The lateral vibration data on line 1 and line 2 are highly discrete. The lateral vibration data of line 3 is relatively concentrated, but the vibration is relatively larger when the train operated on line 2.
5. Conclusions
The vibrating characteristics between the car body and car body-suspended equipment for a high-speed railway vehicle are examined through experimental study. Test results from a long-term field test on-track are used to demonstrate the coupled vibration among the vehicle components. Following conclusions can be drawn.

(1) In the normal track segments, the accelerations on the car body-suspended equipment vibration are about 0.1 g and 0.2 g in the lateral and vertical directions. For the car body, the lateral and vertical accelerations share same amplitude of 0.05 g. This implies that the vibration of the equipment is much larger than that of the car body, and the high-frequency vibration bears on it is not transmitted to the car body. It is contributed to the effective vibration isolation by the elastic element.

(2) The car body vibrates in the dominant frequency of 0.6 Hz and 1.6 Hz, corresponding to the rigid mode. In case of hunting occurs on the bogie when running on some track segments, an elastic structural vibration on the car body can be noticed which is significantly larger than its rigid motion. As a consequence, the lateral and vertical acceleration of the equipment rise to 0.2 g and 0.4 g, respectively. In this case, the vibrating modes are the same for the car body and equipment, which are both dominated by the frequency of lateral instability on the bogie frame.

(3) The difference in the track geometry and irregularities lead to a difference in the vibration level on the equipment. In a general, the acceleration locates between 0.18 g and 0.25 g, but the vibration of vehicle on line 3 is smaller than that on lines 1 and 2.

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References
[1] Wu P.B., Xue S.H., Yang C.H. (2005) Dynamic response of high speed passenger car based on flexible car body model. Journal of Traffic and Transportation Engineering, 5(2): 5-8.
[2] Zhou J.S., Goodall R.M., Ren L.H., et al. (2009) Influences of car body vertical flexibility on ride quality of passenger railway vehicles. Proceeding of the institution of mechanical engineers,
Part F: Journal of Rail and Rapid Transit, 223(5):461-471.

[3] Zeng J., Wu P.B., Hao J.H. (2006) Analysis of vertical vibration reduction for railway vehicle systems. China Railway Science, 27(3):62-67.

[4] Zeng J., Luo R. (2007) Vibration analysis of railway passenger car systems by considering flexible car body effect. Journal of the China Railway Society, 26(6):19-25.

[5] Ren Z.S., Liu Z.M. (2013) vibration and frequency domain characteristics of high speed EMU. Journal of Mechanical Engineering, 49(16):1-7.

[6] Zhou J.S., Zhang W., Song W.J. (2009) Vibration reduction analysis of the dynamic vibration absorber on the flexible car body of railway vehicles. China Railway Science, 30(3):86-90.

[7] Takigami T., and Tomioka T. (2005) Investigation to suppress bending vibration of railway vehicle car bodies using piezoelectric elements. Railway Technical Research Institute, Quarterly Reports, 46(4): 225-230.

[8] Takigami T., and Tomioka T. (2008) Bending vibration suppression of railway vehicle car body with piezoelectric elements (Experimental results of excitation tests with a commuter car). Journal of Mechanical Systems for Transportation and Logistics, 1(1):111-121.

[9] Huang C.H., et al. (2013) The vehicle system dynamics and suspension design with consideration of the flexible vibration of the car body. Railway ve51(10):1-7.

[10] Shi H.L., Wu P.B., Luo R., Zeng J. (2014) Coupled vibration characteristics of flexible car body and equipment of EMU. Journal of Southwest Jiaotong University, 49(4):693-699.

[11] Shi H.L., Wu P.B., Luo R. (2014) Suspension parameters designing of equipment for electric multiple units based on dynamic vibration absorber theory. Journal of Mechanical Engineering, 50(14):155-161.

[12] Schandl G., Lugner P., Benatzky C., Kozek M., & Stribersky A. (2007). Comfort enhancement by an active vibration reduction system for a flexible railway car body. Vehicle System Dynamics, 45(9), 835-847.

[13] Holst, C. (1998). Active damping of carbody vibrations. Royal Institute of Technology, Stockholm, 877-880.

[14] Takigami T., & Tomioka T. (2008). Bending vibration suppression of railway vehicle carbody with piezoelectric elements. Journal of Mechanical Systems for Transportation and Logistics, 1(1), 111-121.

[15] Dumitriu M. (2017). A new passive approach to reducing the carbody vertical bending vibration of railway vehicles. Vehicle System Dynamics, 55(11), 1787-1806.

[16] Zheng X., Zolotas A., Goodall R. (2019). Combined active suspension and structural damping control for suppression of flexible bodied railway vehicle vibration. Vehicle System Dynamics, doi.org/10.1080/00423114.2019.1572902, 1-31, online first.