Journal of Sound and Vibration 409 (2017) 94–111

Contents lists available at ScienceDirect

Journal of Sound and Vibration

journal homepage: www.elsevier.com/locate/jsvi

Method of multi-mode vibration control for the carbody of high-speed electric multiple unit trains

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ARTICLE INFO

Article history:
Received 19 December 2016
Received in revised form
26 April 2017
Accepted 6 May 2017
Handling Editor: L.G. Tham

Keywords:
High-speed EMU trains
Multi-mode vibration control
DVA
Equipment
Belleville spring

ABSTRACT

A method of multi-mode vibration control for the carbody of high-speed electric multiple unit (EMU) trains by using the onboard and suspended equipments as dynamic vibration absorbers (DVAs) is proposed. The effect of the multi-mode vibration on the ride quality of a high-speed EMU train was studied, and the target modes of vibration control were determined. An equivalent mass identification method was used to determine the equivalent mass for the target modes at the device installation positions. To optimize the vibration acceleration response of the carbody, the natural frequencies and damping ratios of the lateral and vertical vibration were designed based on the theory of dynamic vibration absorption. In order to realize the optimized design values of the natural frequencies for the lateral and vertical vibrations simultaneously, a new type of vibration absorber was designed in which a belleville spring and conventional rubber parts are connected in parallel. This design utilizes the negative stiffness of the belleville spring. Results show that, as compared to rigid equipment connections, the proposed method effectively reduces the multi-mode vibration of a carbody in a high-speed EMU train, thereby achieving the control objectives. The ride quality in terms of the lateral and vertical vibration of the carbody is considerably improved. Moreover, the optimal value of the damping ratio is effective in dissipating the vibration energy, which reduces the vibration of both the carbody and the equipment.

1. Introduction

The frequencies of the low-order elastic modes of high-speed electric multiple unit (EMU) train manufactured using lightweight technology are usually low, which includes the frequencies of the diamond-shape deformation mode, first-order vertical bending mode, breathing mode, and torsional modes, etc. Occasionally, these frequencies may be below 10Hz, which is close to the vibration frequencies to which the human body is sensitive [1]. Additionally, the ride quality of an EMU train will further deteriorate at a high speed [2]. Therefore, it is important to control the elastic vibration of the carbody of a high-speed EMU train in order to improve passengers' comfort.

The conventional method for reducing elastic vibration is to increase the stiffness of the carbody; however, this method also causes the carbody mass to increase, which does not comport with the general principle of achieving a lightweight design [3]. Therefore, as an alternative, primary and secondary active suspension systems and novel types of actuators are

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http://dx.doi.org/10.1016/j.jsv.2017.05.010

used. Sugahara et al. [4] controlled carbody elastic vibration by using damper control system for axle dampers based on LQG control and sky-hook damper control, respectively, and the vibration reduction was compared. The results showed that, as compared to the sky-hook damper control, the LQG control allows for the design of some state vectors of the system. Therefore, the vibration energy of the basic frequency can be selectively suppressed based on the distribution of carbody vibration frequencies. Foo and Goodall [5] installed a PI control-based hydraulic servo actuator below the middle part of the carbody in order to control the carbody flexural vibration, and they achieved a good control effect. Gong et al. [6] used secondary active suspensions which designed based on optimal control method that considering track irregularity spectrum in order to reduce the vibration transfer from the track to the middle part of the carbody. Their method controlled the rigid vibration of the carbody as well as reduced the carbody first-order vertical bending vibration. Schandl et al. [7] divided 12 piezo-stack actuators into two groups, which were respectively installed in the carbody side beam. Then, the elastic flexural vibration of the carbody was suppressed through the design of the controller based on the optimal control theory. However, EMUs usually have a large mass, and it may be difficult to harvest the necessary energy for active control. Moreover, the intervals between the suspensions in the bogie are limited. For these reasons, the active control systems for reducing carbody vibration are more structurally complex and cost more than the simple structural vibration control systems for bridges and buildings, and the maintenance is also difficult. As a result, active vibration control systems are not yet widely applied in the railway industry.

Dynamic vibration absorbers (DVAs) provide another solution for reducing carbody elastic vibration. Gong et al. [8] installed a DVA below the middle part of the carbody and conducted calculations on the DVA control of the first-order vertical bending vibration of the carbody. In this study, the existing onboard and suspended equipment are treated as DVAs, and a multi-mode vibration control method is proposed. Then, using the theory of dynamic vibration absorption, the natural frequencies and damping ratios of lateral and vertical vibrations are optimally designed for each equipment based on the multiple target modes of vibration control. Finally, simulation is conducted on carbody vibration control for verification.

2. Determination of the target modes for carbody vibration control

In order to determine the target modes for carbody vibration control, a rigid-flexible coupling dynamic model of an EMU train considering the carbody flexibility was first constructed. This process was done using ANSYS software and the multibody dynamic software SIMAPCK. The carbody under investigation primarily consists of the underframe, side walls, end walls, roof, and interiors. The cylindrical bearing structure of the carbody is manufactured by the assembly welding of large-size full-length hollow aluminum alloy profiles. In the finite element analysis (FEA), the entire carbody is discretized using 4-node shell elements. Fig. 1 shows the elastic modal calculation for the first 5 orders. It can be seen that both the eigenfrequencies of the diamond-shaped mode and first-order vertical bending mode are below 10 Hz, which is not conducive to the ride quality of the vehicle [2]. Moreover, both the lateral and vertical deformations are produced by the diamond-shaped mode, first-order torsional mode, and first-order lateral bending mode. In other words, these modes have an impact on both the lateral and vertical vibrations.

The degree of freedom (DOF) of the carbody FE model nodes is carefully selected as the master DOF in order to construct the rigid-flexible coupling dynamic model. The Craig-Bampton condensation method [9] is applied to the FE model of the carbody, and the precision of the calculation is about 3%. The standard documents containing the carbody structure and the modal information needed for multibody dynamics modeling are obtained and are input into the software for the modeling process. Fig. 2 shows the resulting rigid-flexible coupling dynamic model of the vehicle, and its dynamic parameters are presented in Appendix Table A1. The model consists of 1 flexible carbody, 2 bogies, 3 under-chassis-equipment, 1 air conditioner, 4 wheelsets, and 8 axle boxes. The elastic deformations of the bogies, wheelsets, and axle boxes are very small as compared to the carbody, therefore, they are considered to be rigid. Based on the carbody flexibility's contribution to the vibration energy [10], only the elastic modes given in Fig. 1 are considered for the carbody. Moreover, the model also takes into consideration the increasing stiffness feature of the secondary lateral elastic bump stops, the non-linear creep force and creep moment, the non-linear geometry of the wheel-rail contact, and the non-linear features of hydraulic dampers. This study uses the track irregularities of high speed railway as system inputs [11]. Samples of a length of 2000 m are plotted in Fig. 3 for lateral and vertical irregularities.

For the multi-mode vibration control, the target modes are the modes that make the greatest contribution to the vibration energy and ride quality of the carbody. On this basis, the vibration is reduced through the existing onboard and suspended equipment as the absorbers. The model shown in Fig. 2 is used firstly to study the vibration spectra of the carbody without the equipment and the degree to which the elastic modes of the carbody contribute to the vibration is assessed qualitatively. Figs. 4 and 5 show the calculated results of the vibration acceleration in frequency domain at the ride quality testing points on the floors of the middle part of the carbody and above the front and rear bogies. From the calculation results, the following conclusions are determined:

(1) Except for the rigid vibration of the carbody below 2 Hz, the first-order vertical bending is the main contributor to the vertical vibration in the middle part of the carbody, followed by the diamond-shaped deformation mode. The first-order lateral bending mode is the main contributor to the lateral vibration in the middle part of the carbody, followed by the diamond-shaped deformation mode.