Experimental investigation on vibration characteristics and frequency domain of heavy haul locomotives

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Abstract

This paper aims to investigate the vibration characteristics and frequency domain of heavy haul locomotives at operation speed of 70 km/h based on the field test. A 10000t freight trains that included double HX-type heavy haul locomotives and 105 C80-type freight wagons, was used to conduct an on-track field tests. Test results indicate that suspension systems of locomotives can effectively attenuate high-frequency vibration induced by wheel-rail interface excited by the track irregularity. Axle box vibration energy distributes in a wide frequency range which includes both low-frequency and high-frequency vibrations, while the vibrations of bogie frame and car body mainly concentrated on low-frequency ranges. Forced vibrations induced by sleepers (32.1 Hz) can be transmitted from axle box to bogie frame and car body, and contribute to significant vibration energy, especially for bogie frame. Besides, although vibration energy induced by wheel perimeter (5.0 Hz) is a small proportion of both axle box and bogie frame vibration, it has a significant effect on car body vibration. According to UIC-518, test locomotives have an excellent hunting stability at operation speed of 70 km/h.

Keywords: Filed test; Locomotive; Vibration characteristic; Frequency domain; Running stability

1. Introduction

As increases in running speed and axle load, dynamic behaviors of heavy haul trains become more complex and have a significant effect on structural stability, running safety and riding comfort of railway vehicles. Therefore, it is necessary to study the vibration and transmission characteristics of heavy haul locomotives, which is benefit to understand service status of locomotives and ensure its long-term operation safety.

Scholars around the world have conducted a lot of theoretical and experimental researches to investigate dynamic performance and vibration characteristics of locomotives and obtained abundant results. Sugahara et al. [1-2] built a vibration analysis simulation model and conducted running tests using Shinkansen vehicle to improve ride comfort in railways vehicles, which indicates that the controllable suspension damping can effectively restrain the rigid modal and first-order elastic mode of vehicle body during high-speed operation. Ribeiro et al. [3-4] investigated vibration modals of railway vehicles, including car body and bogie frames, and built the finite element modals that were proven by test results. Zhai et al. [5-6] conducted a field test to analyze high-speed train–track–bridge dynamic interaction relationship and investigated vibration of a CRH train at speed of 350km/h. Guo et al. [7] conducted a series of low-speed derailment test to investigate dynamic performance of derailed vehicles under different test condition, and proposed effective safety measures to reduce post-derailement consequences. For the theoretical analysis of railway vehicle vibrations, Cole et al. [8-9] built a train model considering coupler behavior to study wagon instability behavior in long trains. Ma and Xu [10-12] built a train dynamic simulation model to analyze the dynamic performance and rotation behavior of heavy haul couplers and its effect on heavy haul locomotives on curve and straight lines. Wu et al. [13-14] analyzed the stability mechanism of three typical couplers and investigated some derailments of heavy haul trains caused by coupler jackknifing during braking. Ren and Wang [15-16] investigated the vibration transmission and frequency domain of high-speed railway vehicles based on simulation model and experimental data. However, most of the aforementioned experimental studies focus on vibration behavior and transmission characteristics of high-speed railway vehicles. Conversely, vibration characteristics of freight trains are mainly studied based on dynamic simulation model while experimental researches are rather scarce. As the power units of freight trains, dynamic performance of long-term service locomotives can have an important impact on running stability and safety of freight trains. Therefore, it is very important to conduct experimental studies of vibration characteristics of long-term service locomotives.

In this paper, vibration behavior and transmission characteristics of heavy haul locomotives were investigated based on a running test using a 10000t heavy haul freight train. According to the test results, acceleration responses of heavy haul locomotives were analyzed to study the vibration characteristics and frequency domain of heavy haul locomotives opera-
2. Structure running test of heavy haul trains

2.1. Test conditions

In China, heavy haul locomotives are mainly HX-type and SS-type locomotives, especially for 10000t or 20000t class long heavy haul freight trains, and ballasted tracks are commonly used in heavy haul railway. As shown in Fig. 1, test heavy haul trains was 10000t class long heavy haul freight trains which consisted of double HX-type heavy haul locomotives and 105 C80-type freight wagons. Fig. 2 shows the configuration of the test train. Note that a set of HX-type heavy haul locomotives includes two identical locomotives. Locomotive No.2A was the interested target in this paper. The main parameters of test locomotive are listed in Table 1.

Test line is Baotou-Dongshengxi line with ballasted track which is the section of Bao-Xi railway. Rail mass per unit length is 60kg, and track gauge is 1435mm. The minimum radius of curve track is about 3000m. The maximum gradient is 13‰. The maximum operation speed is 85km/h.

2.2. Test methods

In order to acquire the dynamic behavior of test locomotives, a series of acceleration sensors were installed on car body, bogie frame and axle box of test locomotives. Note that acceleration sensors can record acceleration data in the lateral and vertical direction, respectively. Fig. 3 shows the arrangement of the acceleration sensors. Car body acceleration sensors were installed on the car body floor (Fig. 3a), and the bogie frame acceleration sensors were fixed on the top surface of bogie frame near the secondary suspension system (Fig. 3b), the axle box acceleration sensors were installed on the upper side of the axle box end cover (Fig. 3c). Acceleration data was gathered by integrated measurement and control devices. Considering the vibration characteristics of railway vehicles in a wide frequency range, the sampling frequency of acceleration sensors was set to be 2000 Hz.

3. Analysis of vibration characteristics

The vibration statuses of the main components of heavy haul locomotives were obtained by the running test on the actual line. The dynamic performance and vibration characteristic of locomotives can be analyzed according to the test results. In this paper, a series of typical results of vibration accelerations of the car body, the bogie frame and the axle box were measured at running speed of 70 km/h on tangent track.

Fig. 4 and Fig. 5 show the test results of axle box vertical and lateral acceleration responses and their power spectral density, respectively. As shown in Fig. 4a, the peak value of axle box vertical acceleration at 70km/h is approximately 5.95g (g=9.81 m/s^2). It can be seen from Fig. 4b that there are two distinct dominant frequencies in the vertical vibration frequency of the axle box. The first main frequency is 32.1 Hz, which is caused by sleepers. The vibration frequency \( f_d \) due to the spacing of sleepers can be calculated:

\[
\frac{v}{3.6d} \\
\]

where \( v \) is the running speed of heavy haul locomotives, \( v=70 \) km/h, and \( d \) is the spacing of sleepers, \( d=0.6 \) m.

Accordingly, \( f_d=32.4 \) Hz, which is close to the first main frequency of axlebox vertical acceleration \( 32.1H \). It should be pointed that the second main frequency is 60.9 Hz that is the double frequency of the \( f_d \). It is clear that the axle box vertical acceleration distributes in a wide frequency rang below 500 Hz. The reason is that the wheel-rail contact vibrations excited by the track irregularity are tend to be transmitted to the axle box. Note that the contact vibration of wheel-rail interface is high-frequency vibration because of the large contact stiffness between rails and wheels. Clearly, there are two distinct dominant frequency ranges in axle box vertical acceleration, as shown in Fig. 4b. The first one is a wide frequency range of 30-90 Hz, which is mainly related to the vibrations caused by sleepers and the elastic vibration of the bogie frame. The second one is high frequency 160-260 Hz, which is related to the elastic vibration of wheelsets and the high-frequency contact vibration of wheel-rail interfaces.

As shown in Fig. 5a, axle box lateral acceleration is smaller than its vertical acceleration. The peak of axle box lateral acceleration is approximately 2.36g. Similar to the vertical vibration, lateral acceleration also distributes in a wide frequency rang below 500 Hz, as shown in Fig. 5b. Axle box lateral acceleration also reflects the vibration behavior induced by sleepers.

The measured results of vertical and lateral accelerations of bogie frame are shown in Fig. 6 and Fig. 7, respectively. Comparing with axle box vertical acceleration, bogie frame vertical vibration is weaker, and the maximum value of bogie frame vertical acceleration is about 1.79g which is only one third of the peak value of axle box vertical acceleration. Its vibration energy distributes mainly in the range of 30-80 Hz, which mainly reflects the low-order elastic modal frequencies of the bogie frame. This phenomenon indicates that primary suspension systems can effectively attenuate the high-frequency wheel-rail contact vibration. In particular, there is a significant frequency component at 32.1Hz, which indicates that the forced vibration caused by sleepers can be transmitted to the bogie frame and cannot be attenuated by primary suspension systems.

Similar to the vibration of axle box, bogie frame lateral acceleration is smaller than its vertical acceleration, as shown in Fig. 7. Its peak value is about 1.14g. In the frequency domain, the vibration energy distributes mainly in the range of 30-80 Hz, which includes the low-order elastic modal frequencies of bogie frame and forced vibration induced by sleepers. Note that the vibration energy of low-order elastic modal frequencies of bogie frame is larger than that of forced vibration, which is different from its vertical acceleration.
Fig. 8 shows the car body vertical acceleration and its power spectral density. As shown in Fig. 8a, its peak value is about 0.049g. It can be seen from Fig. 8b that there are three distinct dominant frequencies. The first one is the range of 0.5-2.5 Hz which reflects the natural vibration frequency of the vertical suspensions of car body. There are significant frequency components at 5 Hz which is induced by wheel perimeter. The vibration frequency $f_2$ due to wheel perimeter can be calculated:

$$f_2 = \frac{v}{3.6\pi D}$$  \hfill (2)

where $D$ is the rolling diameter of wheel, $D=1.25$.

Thus, the vibration frequency due to wheel perimeter is 4.95 Hz that is close to the second main frequency 5.0 Hz. It should be pointed that the forced vibration induced by wheel perimeter are also transmitted to both the axle box and bogie frame, but its vibration energy is rather smaller proportion of both axle box and bogie frame vibration. For the railway vehicle, the high-frequency vibrations have been effectively attenuated by suspension systems of locomotives, while low-frequency vibrations can be transmitted from axle box to car body. The third main frequency is 32.1 Hz that is the forced vibration induced by sleepers. Comparing with bogie frame vertical acceleration, the vibration energy due to sleepers is attenuated by secondary suspension systems.

The measured lateral acceleration of the car body is shown in Fig. 9a. Similar to axle box and bogie frame, the car body lateral acceleration is smaller than its vertical acceleration. Its peak value is 0.037g. In the frequency domain, its vibration energy also distributes mainly in three distinct dominant frequencies ranges, as shown in Fig. 9b. The first one is 0.5-2.5 Hz which represents the natural vibration frequency of the suspension systems. The second and third ones are respectively 5.0 Hz and 32.1 Hz, which are the force vibration induced by wheel perimeter and sleepers. Note that the natural vibration energy is larger than forced vibration energy.

According to the above analysis of test results, the vibrations of heavy haul locomotives are mainly induced by wheel-rail contact vibration excited by the track irregularity and are transmitted to the axle box, bogie frame and car body successively. The vibration amplitudes of axle box, bogie frame and car body decrease successively. Although axle box vibration contains a lot of high-frequency components, the suspension systems can effectively attenuate high-frequency vibration so both bogie frame and car body vibration accelerations mainly distribute below 100 Hz. Besides, both lateral and vertical accelerations have similar frequency domain characteristics and the lateral vibration is generally weaker than vertical vibration.

Figure 1. Test train and test track.

Figure 2. Configuration of the test train
Table 1. Main parameters of the test locomotive

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<td>Mass of wheelset</td>
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Figure 3. Arrangement of acceleration sensors

Figure 4. Test results of axlebox vertical acceleration: (a) time history and (b) power spectral density
Figure 5. Test results of axlebox lateral acceleration: (a) time history and (b) power spectral density

Figure 6. Test results of bogie frame vertical acceleration: (a) time history and (b) power spectral density

Figure 7. Test results of bogie frame lateral acceleration: (a) time history and (b) power spectral density

Figure 8. Test results of car body vertical acceleration: (a) time history and (b) power spectral density
4. Evaluation of locomotive running stability

The running stability is a very important index to analyze the operation performance of heavy haul locomotives. According to UIC-518, the running stability of test locomotives can be evaluated by lateral acceleration of bogie frame. A safety value of bogie frame lateral acceleration can be obtained:

\[ \text{Safe}_{\text{lateral}} = 12 - \frac{M_b}{5} \]  \hspace{1cm} (3)

where \( M_b \) is the mass of the complete bogie (including wheelsets), \( M_b = 19 \) tonnes. Therefore, \( \text{Safe}_{\text{lateral}} = 8.2 \) m/s\(^2\). Once the peak value of bogie frame lateral acceleration after 10 Hz low-pass filtering is more than the \( \text{Safe}_{\text{lateral}} \), the locomotive is regarded as losing its stability.

As shown in Fig. 10, the maximum value of bogie frame lateral acceleration after 10 Hz low-pass filtering is much smaller than the \( \text{Safe}_{\text{lateral}} \). The phenomenon indicates that test locomotives have the perfect running stability at operation speed 70 km/h according to the UIC-518.

5. Conclusions

The dynamic performance and vibration characteristics of heavy haul locomotives at running speed of 70 km/h are investigated based on a running test on Bao-Xi line. The following conclusions are drawn from the present results.

** Axle box vibration energy distribute in a wide frequency ranges below 500 Hz which contains both low-frequency and high-frequency vibrations. However, both bogie frame and car body vibration energy mainly concentrate on the low-frequency range below 100 Hz which includes natural vibration of bogie frame and car body and forced vibration induced by sleepers and wheel perimeter.

** Vibration amplitudes of heavy haul locomotives gradually decrease along with the transmission path. Vertical vibration acceleration has a similar frequency domain to lateral vibration acceleration, but lateral vibration amplitude is smaller than that of the vertical vibration.

** Suspension systems of locomotives can effectively at-
tenuate the high-frequency vibration but not low-frequency vibration. Thus, low-frequency vibrations induced by sleepers and wheel perimeter can be transmitted from axle box to bogie frame and car body, and contribute to a great proportion of both bogie frame and car body vibration acceleration.

** Running stability of test locomotives fully satisfies the evaluation criterion of UIC-518.

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**References


