Research of Wheel-Rail Coupled Vibration Excited by Wheel Polygon for Heavy-Haul Locomotive

Tianlong Wang, Pengfei Liu, Kailong Zhang, YunqiangCao

School of Mechanical Engineering, Shijiazhuang Tiedao University, Shijiazhuang 050043,China

E-mail:1249591174@qq.com

Abstract. In order to study the wheel-rail impact vibration of the heavy-haul locomotive with 25t axle-load, the typical wheel polygonal data for different running distance were field measured. The radial circular run-out, radial geometric deviation and harmonic composition of wheel polygon were analyzed. The vehicle-track vertical coupling dynamics model was established and verified by comparing the vibration test results and numerical simulation results. The research results show that, with the increase of service mileage, the wheel polygon wear deteriorates, leading to the violent wheel-rail dynamic interaction. The impact force caused by the wheel polygon has a great influence on the primary suspension force, and has almost no influence on the secondary suspension force.The maximum acceleration of the car body, frame and wheelset caused by the wheel polygon is 3.148g, 0.177g and 0.00149g, so the vibration attenuation phenomenon of the locomotive is very obvious. The maximum acceleration of the rail, sleeper, ballast is 3.38g, 2.81g, 2.1g, therefore, the track system vibration attenuation is weaker. The main frequency of the vibration is 30Hz, which corresponds to the 7th-order harmonics, the result is consistent with the analysis of the wheel polygonal harmonic components .

1. Introduction

After railway vehicle runs for a period of time, wheel tread produces uneven wear. When the wear reaches a certain value, the wheel polygon causes additional vibration, impact and noise to the vehicle, affecting the stability of the vehicle and causing serious threat to the safety of the vehicle [1]. With the increase of axle load and speed, the problem of wheel-rail wear is becoming more and more serious. The research on the impact vibration between wheel and rail cannot be ignored.

Many experts and scholars have done a lot of research on the wheel-rail relationship. Literature [2] established a vehicle-track coupling dynamics model, taking the motor drive system and traction force into account. As the traction force varied between 150 and 450kN, the dynamic interaction of the wheel and rail when the locomotive passing through a curved track was analyzed. Literature [3] based on the dynamic test data, the vibration level and vibration transmission of the components of the bogie with wheel polygonization were analyzed. Literature [4] established a vehicle dynamics simulation model.The effects of wave depth and harmonic order of wheel polygonization on vehicle dynamic performance under high-speed train operation were studied by numerical simulation.

In this paper, a vehicle-track coupling dynamics model was established for heavy-load locomotives with 25t axle-load. The wheel-rail vibration response caused by wheel polygons was analyzed,the transmission relationship of vibration in locomotive and track system was studied, and the effect of different orders' wheel polygon on vehicle was explored.

2. The field test and geometric feature analysis of wheel polygon

2.1.Field test

In order to study the wear evolution of heavy load locomotives for different mileage, our research team went to field three times to measure wheel polygon. Fig. 1 shows the measuring instrument and field. The position of the measuring point selected during the test is the nominal rolling circle, that is, 70 mm from the rim. In the test ,wheel roughness measuring instrument has a sampling length of 1 mm, and the circumference of the measured wheel is about 3850 mm. Fig. 2 shows the spalling phenomenon of the measured wheel. Since the wavelength of the wheel radial deviation should be greater than the length of the wheel-rail contact patch longitudinal axis, it is necessary to filter the high-frequency wheel roughness component. After that, the shortest wavelength that can be considered is 100mm, which corresponds to the harmonic order of wheel exceeds 20th order,and it can meet the research requirements.

Figure 1. Field measuring.   Figure 2. Wheel spalling

2.2. Analysis of wheel radial runout and geometric characteristics

During a Wheel reprofiling cycle, the No. 2 locomotive with the mileage of 43, 000, and 170,000 kilometers were measured respectively. The measured results of the wheels radial run-out are shown in Fig. 3. As the running mileage increases, the wheel wear value increases due to the further increase of the wheel wear, and the maximum value reaches 0.867 mm. Since China has not yet formed a unified standard for wheel maintenance and reprofiling, the standard for the radial runout of the wheels specified by the German Railways should not exceed 0.6 mm. Obviously, this value exceeds the standard. Overall, half wheels of the vehicle exceeded the standard, indicating that the wheel polygon wear of the No. 2 locomotive was very serious when the mileage reaches 170,000 km.

Next, the radial circular run-out of the No. 8 locomotive before and after wheel reprofiling is compared in Fig. 4. The mileage of the locomotive before wheel reprofiling was 88,000 kilometers. Only one wheel appeared abnormal wear, and the value of radial circular run-out reaches 1.31mm. After wheel reprofiling, the value of wheel radial circular run-out is below 0.1mm.
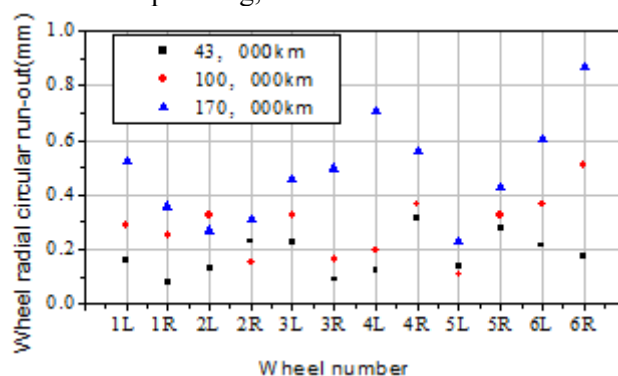

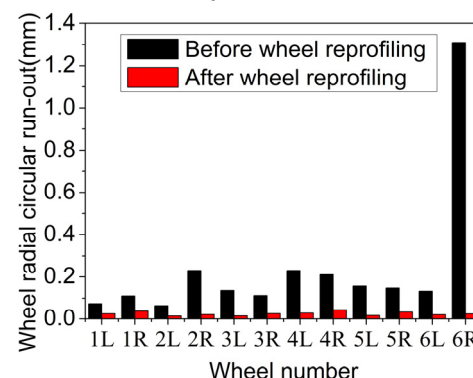Figure 3.The radial circular run-out of No. 2 locomotive with different mileage.

Figure 4. Comparison of the radial circular run-out before and after wheel reprofiling.

The geometric characteristics of the wheel polygon for the No. 2 locomotive's 6R wheel with 170,000 km is analyzed, as shown in Fig. 5 and Fig. 6, the wheel has a long-wavelength local defect, its wavelength is about 1298 mm and the depth of the wave is about 0.5 mm. The main harmonic components are 1st, 2nd, 3rd, and 7th order. Fig. 7 and Fig. 8 show that the radial deviation and harmonic components of wheel polygon for the No. 8 locomotive's 6R wheel, which radial circular run-out is 1.3mm before wheel reprofiling.The wheel has obvious short-wavelength local depression, the wavelength is about 850mm, and its wave depth is 1mm, the main harmonic components are 1st, 2nd and 4th order.

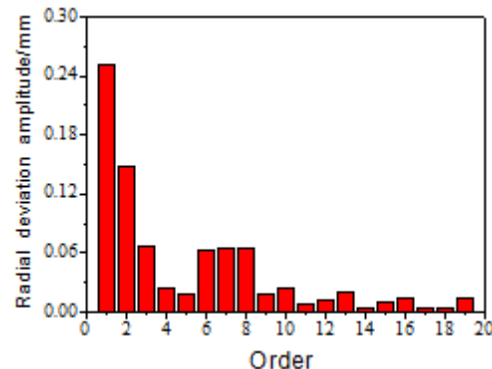Figure 5. 6R wheel polygon for No.2 locomotive with 170,000 km

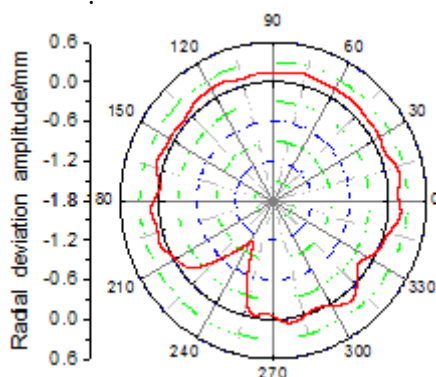Figure 6.Harmonic order distribution for No. 2 locomotive's 6R wheel with 170,000 km.

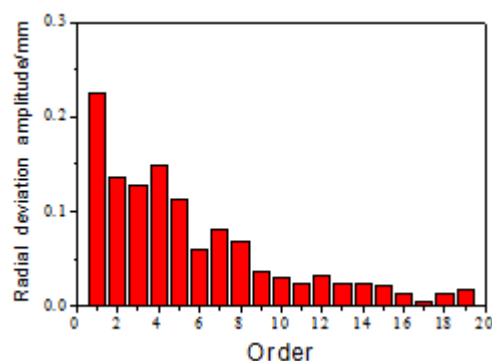Figure 7. Polar diagram of the wheel polygon for No. 8 locomotive's 6R wheel.

Figure 8. Harmonic order of the No. 8 locomotive's 6R wheel.

In general, through statistical analysis, the main components of the measured wheel polygons are mostly in the 1st to 10th orders, and the dominant components are ranked in the order of 1st, 2nd, 3rd, 5th, and 7th. The main wavelength concentrates in the range of 556 to 3890mm. When the locomotive operates normally within the speeds of 60~80km/h, the vertical vibration with the frequency from 4 to 40Hz can be excited.

3. The establishment and verification of vehicle-track coupling vertical model
The locomotive and track system dynamics model reference [5], no longer repeat, Z_{0i} (t) can input wheel polygon data.

After measuring wheel polygon, in order to verify the model, a dynamic test was designed for No.2 locomotive. Vertical, horizontal and vertical acceleration sensors were arranged on the axle box and the motor gearbox respectively, and a vertical acceleration sensor was arranged on one side of the bogie. According to the measured data of the sensor, the vertical acceleration data of the axle box 1 is

intercepted.The polygon of No. 2 locomotive's 1R wheel is selected to input into the numerical simulation model, the resulting comparison is shown in Fig.9 and Fig. 10.
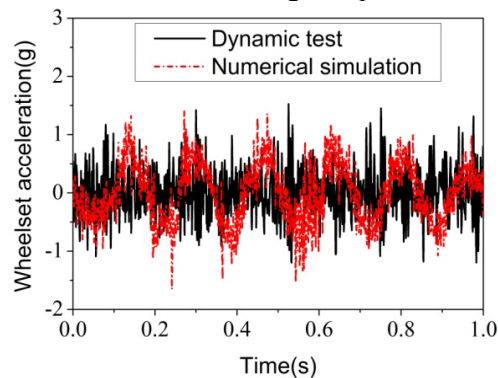

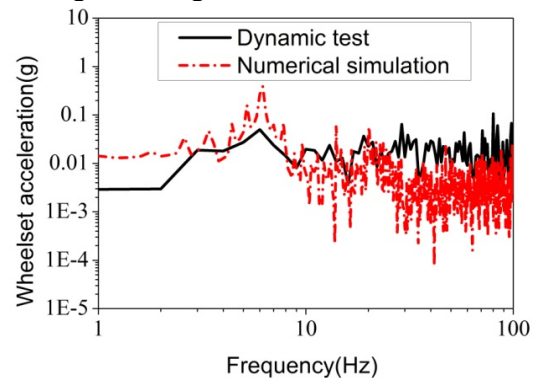Figure 9. Time-history graph of wheelset acceleration.

Figure 10. Spectrogram of wheelset acceleration.

As can be seen from the time history diagram, both acceleration images fluctuate between -15 and 15g with very close amplitudes. Because the measured data is affected by factors such as the irregularity of rail surface and the high-order wheel roughness, there are certain errors in the time-history graph and the spectrogram, but the basic trend of the reaction process can be reflected, and it has certain significance for the research of locomotives vibration.

4. Analysis of the transmission process for vibration caused by typical wheel polygons

4.1. The analysis of wheel- rail force caused by different wheel polygons
The typical wheel polygon data is selected as the excitation input into the model. In order to highlight the influence of the wheel polygon on the wheel-rail impact vibration, the track irregularity is not considered in the model. The simulated working conditions in the numerical simulation model are: the locomotive runs linearly at a speed of 60 km/h, and the stationary wheel load is 123 kN. Fig. 11 and Fig. 12 compare the wheel-rail vertical force of the No. 2 locomotive 6R wheel with different mileage. The results show that the wheel-rail force fluctuation increases significantly with the increase of the mileage. In the spectrogram, when the frequency is 30hz, the amplitude reaches the maximum, and 30Hz corresponds to the 7th-order harmonic, the result is in good agreement with the previous order analysis.
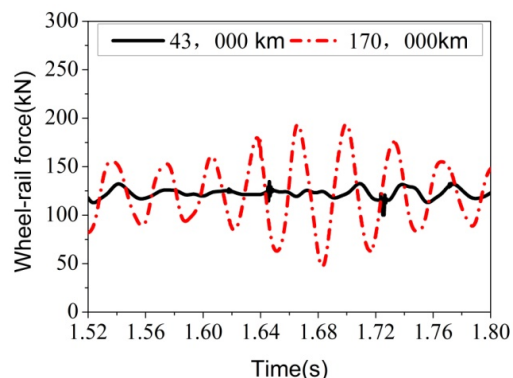

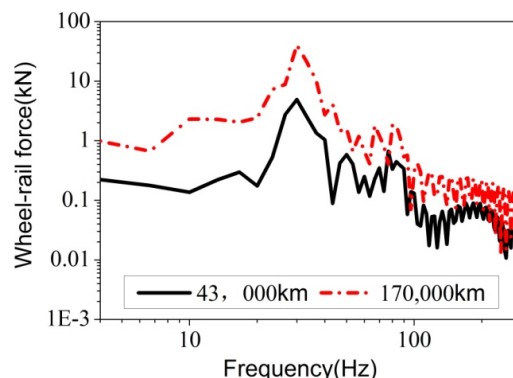Figure 11. Time-history graph of wheel-rail force.

Figure 12. Spectrogram of wheel-rail force.

4.2. Analysis of transmission relationship for vibration caused by wheel polygons
Taking the 6R wheel for No. 2 locomotive with 170,000 km as an example to illustrate its vibration transmission characteristics, wheel-rail force transmission process caused by wheel polygon is shown

in Fig.13. The wheel-rail force fluctuates between 52 and 189.6kN, and the fluctuation of the primary suspension force and the second suspension force is significantly reduced. The primary suspension attenuates most of the wheel-rail impact force, for the reason that it is soft. Furthermore, the secondary suspension force is hardly affected by the wheel-rail force. The spectrogram Fig.14 shows that the wheel-rail force and the primary suspension force reach the peak at 30 Hz, while the second suspension force peak appears at around 15 Hz, indicating that the wheel-rail force has little effect on the second suspension force again.
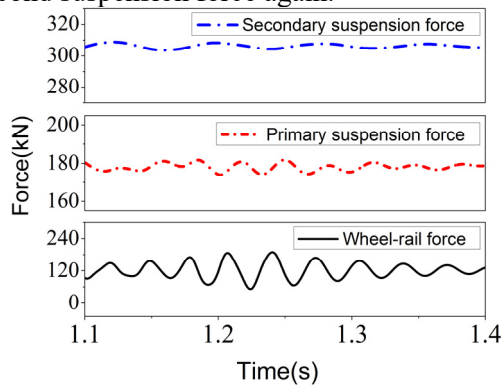

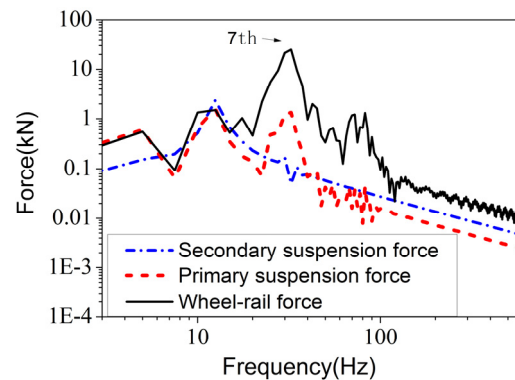Figure 13. Force transmission process

Figure 14. Spectrogram of force transmission

The time-history diagram and spectrogram of the vibration transmission for the locomotive is shown in Fig. 15 and Fig. 16 , and the nodding acceleration caused by the vibration is considered when calculating the acceleration of frame and car body. The maximum vertical acceleration of car body, bogie and wheelset in the time history diagram is 3.148g, 0.177g and 0.00149g, the vibration attenuation of primary suspension is about 17 times, and the second suspension attenuation is about 119 times. Different harmonic components have different effects on the vertical acceleration. At 30 Hz, the amplitude is the largest, corresponding to the 7th harmonic, at the same time, peaks appear at 10, 13, 43, 60, 70, and 83 Hz, corresponding to the 3rd, 10th, 14th, 16th, and 19th harmonics, and the amplitudes of the wheelset, bogie, and body are decreased.
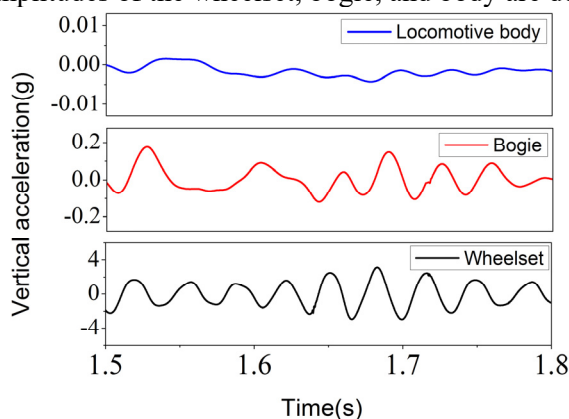

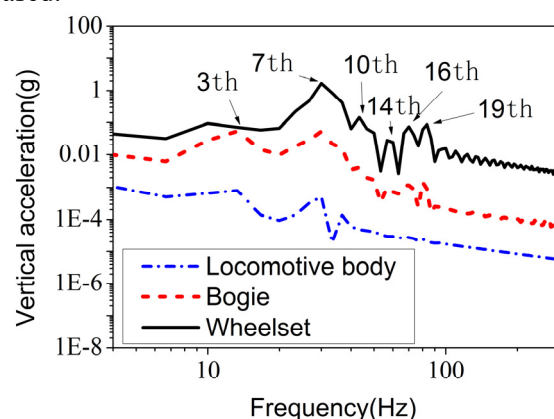Figure 15. Time-history graph of locomotive vertical acceleration.

Figure 16. Spectrogram of locomotive vertical acceleration.

The vibration transmission relationship of the track system is shown in Fig. 17 and Fig. 18. The acceleration at the contact point of the wheel-rail is selected, and the sleeper block and the ballast directly under the wheel can better reflect the process of vibration transmission. The time history diagram shows that the amplitudes of the rail, sleeper and ballast are decreased, and the maximum values are 3.38g, 2.81g, 2.1g, the attenuation is weaker than locomotive. The main frequency of the

rail, sleeper and ballast all appear at 33Hz, and there are four side frequencies: 47Hz, 57Hz, 73Hz and 83Hz, corresponding to the 8, 11th, 16th and 19th order harmonics respectively.
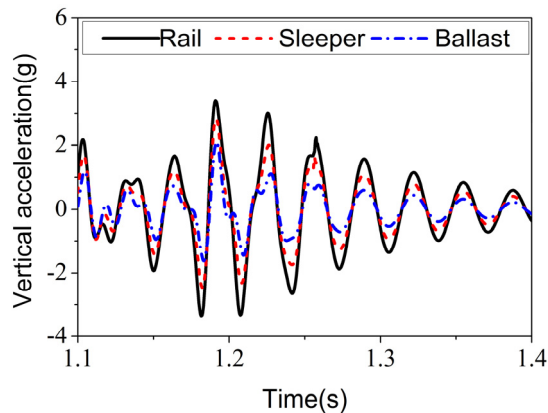

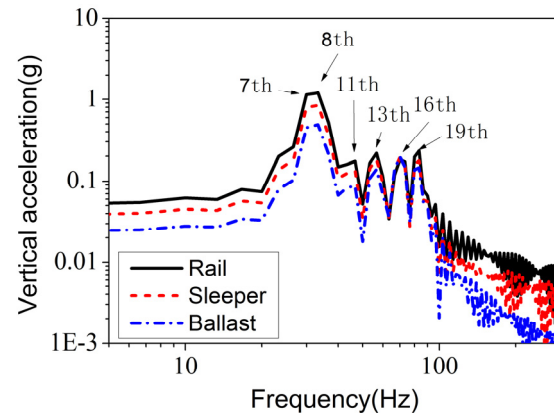Figure 17. Time-history graph of track system vertical acceleration.

Figure 18. Spectrogram of track system vertical acceleration.

After establishing and verifying the model, this section inputs the wheel polygon data of 6R wheel for No. 2 locomotive with 170,000 km into the model, and analyzes the vibration transmission process by comparing secondary, primary suspension force and wheel-rail force, and by comparing the vertical acceleration of the locomotive and rail systems. The results of 7th-order harmonic corresponding to the dominant frequency are consistent with the results of section 2.

5. Conclusion

In this paper, for the heavy-haul locomotive with 25t axle-load, through the on-site investigation, the tracking test of the No. 2 locomotive in a wheel reprofiling cycle was carried out three times, and the No. 8 locomotive before and after wheel reprofiling was tested separately. The vehicle-track vertical coupling dynamics model was established and verified by comparing the vibration test results and numerical simulation results. The research results show that, the main harmonic components of the wheel polygon input in the model are 1st, 2nd, 3rd, 5th,7th. With the increase of service mileage, the wheel polygon wear deteriorates, leading to the violent wheel-rail dynamic interaction. The impact force caused by the wheel polygon has a great influence on the primary suspension force, and has almost no influence on the secondary suspension force. The maximum acceleration of the car body, frame and wheelset caused by the wheel polygon is 3.148g, 0.177g and 0.00149g, so the vibration attenuation phenomenon of the locomotive is very obvious. The maximum acceleration of the rail, sleeper, ballast is 3.38g, 2.81g, 2.1g, therefore, the track system vibration attenuation is weaker. The main frequency of the vibration is 30Hz, which corresponds to the 7th-order harmonics,the result is consistent with the analysis of the wheel polygonal harmonic components .

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