

INFLUENCE OF TRACK IRREGULARITY ON LONGITUDINAL VIBRATION OF WHEELSET AND CORRELATION PERFORMANCE

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Abstract: *A longitudinal vibration of wheelset with respect to bogie frame often exists with a high acceleration magnitude and relative high frequency. At first a simplified model with a single wheelset moving at a constant speed on a tangential track with irregularity is used to investigate the longitudinal vibration dynamics. Results of the longitudinal vibration study indicate that the longitudinal vibration frequency of the wheelset is most sensitive to the primary longitudinal stiffness and the mass of the wheelset. As to the locomotive model, the longitudinal vibration was concerned with cross-level irregularity and vertical profile irregularity. A method to estimate the resonance speed is presented. Finally, the paper shows a possible solution to extend wheel-rail service life by eliminating longitudinal vibration of the wheelset. The solution is simply arranging the primary vertical damper with a forward angle, so that its damping component can be applied to longitudinal direction.*

Keywords: *Track irregularity; Acceleration; Longitudinal vibration; Dynamics*

I. INTRODUCTION

Much attention has been focused on lateral performance of railway vehicles for a long time in order to achieve higher speed and better comfort. But the longitudinal dynamic performance was neglected except for the study on traction and braking performance. Since it has nothing to do with lateral performance, forward velocity was normally considered uniform. The results obtained in this paper show that it may be an important cause of wheel tread spalling and rail

corrugation with certain wavelength.

There are many puzzled enigma existing in rail vehicles, such as the rail vehicle may have some tremble in the course of speedup or at a not high speed; the ride index of rail vehicle in some low speed might be worse than that in the higher speed; the wheel tread spalls and the track wave wears with some wavelength; the vibration of rail vehicle was quite bigger on some tracks. These phenomena indicate that the performance of rail vehicle has some relation with the track irregularity and the longitudinal dynamic

behavior of the wheelset. In this paper, we investigate the relation between longitudinal vibration and the track irregularity from the longitudinal dynamic point of view.

Lots of researches have been carried out in the field of wheel-rail contact fatigue mechanism in last ten years [1, 2, 3, 4]. Wheel-rail wear is a necessary expense of railway transportation because of adhesion in contact patch between track and vehicle system. Total wear between wheel and rail is due to stress generated by static, dynamic and thermal loadings [6]. The best way to reduce wheel-rail wear is to reduce the unnecessary stress, which is caused by dynamic interaction between track and vehicle due to track irregularity and vehicle vibration. But it is difficult to determine which one is the main factor. According to the following analysis, wheelset longitudinal dynamic behavior may be a possible source of the severe wheel-rail dynamic interaction.

Since track irregularity is one of the main reasons of rail vehicles vibration [13], a large number of researches have been made on the response of rail vehicles to the track irregularity [10, 11]. They all aimed at the vertical or lateral vibration of rail vehicles, not referred to longitudinal vibration of the wheelset. In this paper, we analyse the influence of track irregularity on longitudinal vibrations.

II. THE TRACK IRREGULARITY

There always exists track irregularity in the realistic track from the track setup and the interaction wear between the railway vehicles and the track. Researches show that the track irregularity is a random process [11]. And it's difficult to decide what kind of the track

spectrum should be adopted in the vehicle dynamic evaluation in our country.

Usually the spectrum density used to describe track irregularity [5] can be expressed as:

$$S(\omega) = \frac{b_0 + b_1 \omega^2 + b_2 \omega^4 + b_3 \omega^6}{a_0 + a_1 \omega^2 + a_2 \omega^4 + a_3 \omega^6} \quad (1)$$

In which, $a_i, b_i (i = 0, 1, 2, 3)$ is the constant coefficient of the angular frequency of space ω .

The American Federal Railway Management Bureau (FRA) gets the track spectrum according to a great deal of measured data. It is divided into 6 Grades, among which Grade 1 is the worst and Grade 6 is the best. Germany has formulated the spectrum of the high-speed trunk with low interference and high-speed trunk with high interference. Our country has not formulated the standard track spectrum yet, but the department concerned has carried out a few research works, and offered some expressions for track spectrum according to the measured data. But the acquired data in our country's research is so few that it is unable to represent the statistical characteristic of track irregularity in our country [11], so we analyse the problem by using the Germany track irregularity.

According to the accumulative experience, aiming to the speedup track in our country, we consider the vertical profile irregularity of the medium track is corresponding to the vertical profile irregularity of Germany high-speed trunk with low interference; the alignment irregularity of the medium track is worse than the alignment irregularity of Germany high-speed trunk with

high interference; the cross-level irregularity of the medium track is corresponding to the cross-level irregularity of Germany high-speed trunk with low interference. To make a simple and proper calculation, we use the medium track irregularity in this paper.

Time-domain excitation is used both in the locomotive model and simplified model. It is calculated by a polynomial expression as following:

$$F(x) = \frac{b_0 + b_1 x}{a_0 + a_1 x + a_2 x^2} \quad (2)$$

The coefficients of excitation used in the model and the parallels with Germany high-speed trunk with high interference are listed in table 1

Table 1. Coefficients in polynomial expression

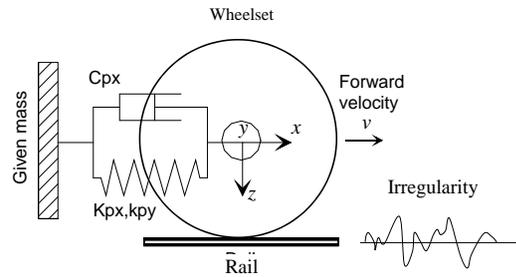
	a_0	a_1	a_2	b_0	b_1
Germany alignment	0.016987	0.8452	1	0.001144	0
Germany vertical	0.016987	0.8452	1	0.001519	0
Germany cross-level	0.00744	0.38718	1.283	0	0.00203
Model used alignment	0.034	1.6904	1	0.0057	0
Model used vertical	0.016987	0.8452	1	0.00093	0
Model used cross-level	0.00744	0.0774	1.283	0	0.0015

III. CALCULATION MODEL

3.1. Simplified model

The longitudinal vibration of wheelset was found in the locomotive model. It is necessary but much time-consuming to make more analysis for all kinds of railway vehicles to study the phenomenon. In order to prove the result obtained by the locomotive model and find the mechanism of longitudinal vibration, a simplified model with just one wheelset and minimum freedoms is set up

(see Fig.1).



K_{px} -primary longitudinal stiffness; K_{py} -the primary lateral stiffness; C_{px} -primary longitudinal damper;

Fig.1. The Simplified model with one wheelset

In the simplified model, mass of wheelset: 2500 kg;

Roll mass moment of inertia of wheelset: 500 kg/m²;

Pitch mass moment of inertia of wheelset: 100 kg/m²;

Yaw mass moment of inertia of wheelset: 500 kg/m²;

Mass of given mass: 19500 kg;

Degree of freedom of wheelset: 6;

Degree of freedom of given mass: 1;

K_{px} : 1.2×10⁷ N/m;

K_{py} : 6.0×10⁶ N/m;

C_{px} : 1000 N.s/m.

Rail: 60kg/m rail;

Track gauge: 1435mm;

Rail cant: 1/40;

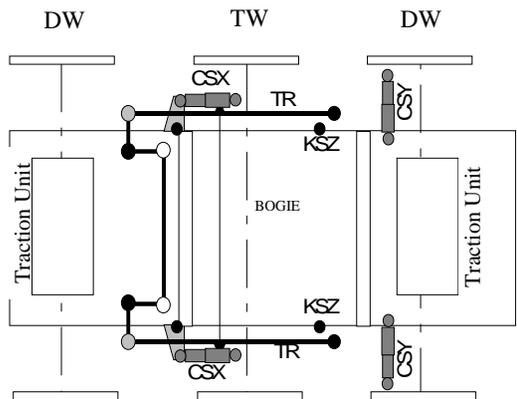
The lateral clearance between wheel and rail: 14 mm;

Tread profile: JM3 profile;

In the model, the friction coefficient μ between wheel and rail is supposed to be constant, and Kalker's simplification theory (FASTSIM) is adopted to calculate the wheel-rail creepage-creep force. The equivalent conicity of the wheel tread within the scope of 6 mm is about 0.12, and is about 0.2 within the scope of 6-8 mm, and will be greater than 0.25 beyond the scope of 8 mm.

3.2. Locomotive Model

The analysed locomotive is designed to meet the operation requirements of 200km/h. In order to reduce the axle load to 22 t, a trailer wheelset is considered besides two driven wheelsets in a bogie (Fig.2). Each driven wheelset has a set of driving unit that consists of one traction motor with a gearbox connected it rigidly. The driving unit can be considered mounted on bogie frame rigidly or mounted on it elastically in lateral direction in model.



DW—driven wheelset; TW—trailer wheelset; CSX—secondary suspension longitudinal damper; CSY—secondary suspension lateral damper; KSZ—secondary suspension vertical stiffness; TR—traction rod.

Fig.2. Model of A-1-A bogie

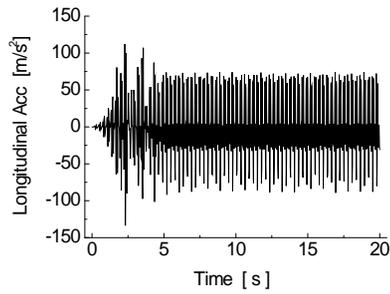
The model consists of car body, 2 bogies,

4 traction motors and 6 wheelsets. The longitudinal damper and the lateral damper are arranged in secondary suspension between each bogie and the car body. Four coil springs are arranged on each side of bogie as secondary spring suspension to keep good lateral running performance. The longitudinal force will be transferred from bogie frame to car body through a four-linking-rod mechanism. It causes a rigid connection between car body and front and rear bogies in longitudinal direction. The primary suspension is considered as a compact force element at each axle box with stiffness in three directions and vertical damping. To improve the speed of the calculation, the axle box itself will not be considered as a body element, and it doesn't influence the precision of the result.

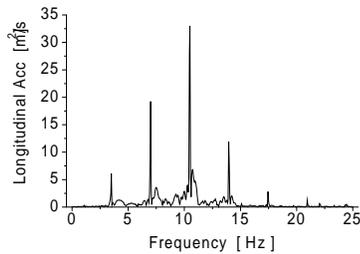
VI. ANALYSIS OF SIMPLIFIED MODEL

4.1. Longitudinal dynamic performance of wheelset

In order to study longitudinal dynamic performance in principal, an analysis is conducted for the simplified model shown in Fig.1. Longitudinal resonance vibration is revealed on the track with irregularity at the speed of 20km/h, and the longitudinal vibration acceleration is quite large at the frequency of 10.6Hz (see Fig.3). The longitudinal creepage and adhesion coefficient are shown in Fig.4. In Figs.3 and 4, we can see that the longitudinal vibration resonance is obvious and the adhesion coefficient always approaches the presumed saturation value 0.25. So we can say that longitudinal resonance vibration of the wheelset really exists in the simplified model.

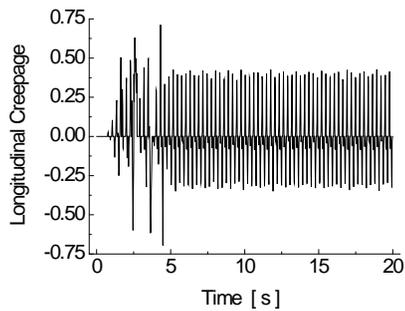


(a) In the time domain

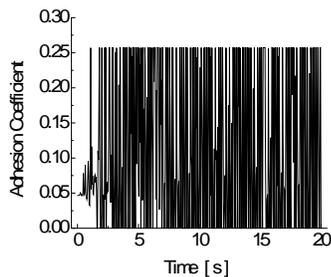


(b) In the frequency domain

Fig 3. Longitudinal acceleration in the simplified model at



(a) Longitudinal creepage



(b) adhesion coefficient

Fig 4. Longitudinal creepage and adhesion coefficient in the simplified model at 20km/h

4.2. Root loci analysis

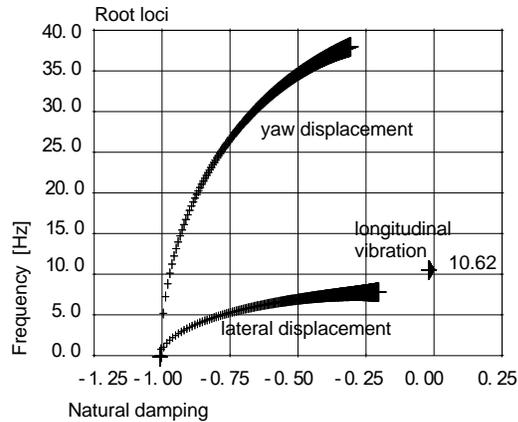


Fig 5. Root loci analysis of the simplified model

To find the inherent frequency of the simplified model, the linear analysis of the simplified model was carried out, and the results are shown in Fig.5. The speed is ranged from 1 km/h to 401 km/h and the increment is 5 km/h. We can clearly see that the longitudinal vibration frequency nearly unchanged with the speed, but the lateral displacement frequency and yaw displacement frequency of the wheelset changed with the speed. From the speed of 1km/h to 401km/h, the frequency of the wheelset longitudinal vibration is nearly 10.62Hz.

In the simplified model, the inherent frequency is related to the system, and not varied with the speed. But the frequency will change according to the mass of the wheelset and the primary longitudinal stiffness. The inherent frequency could be estimated by the following expression:

$$f_L = \frac{1}{2\pi} \sqrt{\frac{2k_x}{m}} \quad (3)$$

Where m is the mass of wheelset and k_x is the longitudinal stiffness of each axle box. Let $k_x = 1.2 \times 10^7 \text{N/m}$, $m = 5000 \text{kg}$, then

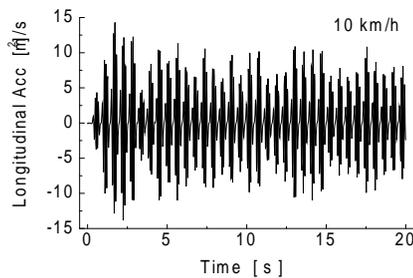
$$f_L = \frac{1}{2\pi} \sqrt{\frac{2 \times (1.2 \times 10^7)}{5000}} = 11.03 \text{ Hz}$$

If this vibration cannot be damped effectively, the rail corrugation with the wavelength λ is developed on rail, and the wavelength λ is:

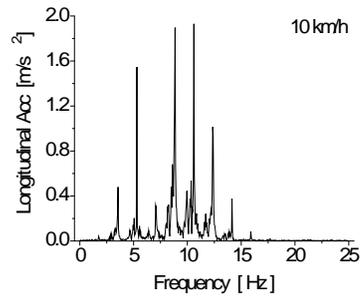
$$\lambda = \frac{v}{f_L} \approx \frac{20/3.6}{11.03} = 0.5\text{m}.$$

As the longitudinal resonance vibration only occurs at a given speed, so the wavelength λ doesn't increase with the increase of the speed. The possible cause of rail corrugation has been discussed in many papers [8]. Here a new possible solution is proposed on the formation of rail corrugation with certain wavelength from the longitudinal dynamic point of view.

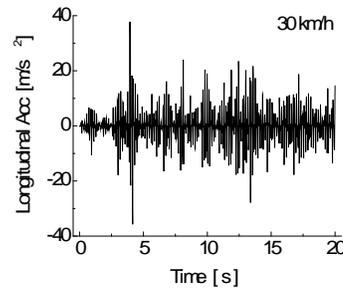
The longitudinal vibration resonance of wheelset doesn't occur at other speeds. For example, at the speed of 10km/h or 30km/h, the longitudinal vibration resonance of wheelset doesn't happen (see Fig.6). So we can draw the conclusion that the longitudinal vibration resonance is corresponding to the speed; higher or lower than 20km/h the longitudinal vibration resonance does not happen in the simplified model, although the longitudinal vibration is still very high.



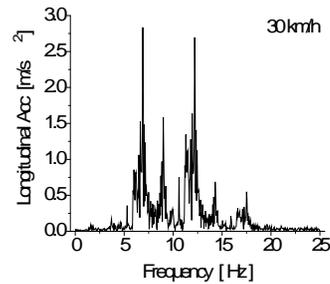
(a) In the time domain at 10 km/h



(b) In the frequency domain at 10 km/h



(c) In the time domain at 30 km/h



(d) In the frequency domain at 30 km/h

Fig 6. Longitudinal vibrations of wheelset under different speeds

4.3. The speed of longitudinal vibration resonance

Several frequencies can be recognized in Figs. 3 and 6. A fixed frequency of about 10.6Hz is corresponding to longitudinal vibration the wheelset with respect to bogie frame. A velocity depending on frequency is the rolling angular velocity of the wheelset.

With the forward speed increasing from 10km/h to 20km/h and 30km/h, this value varied from 5.56rad/s to 11.1rad/s and 16.66rad/s respectively. Suppose nominal radius of the wheelset is 500mm, then the value of 2.0rad/s equals 1cycle/s of wheel rolling. So above three angular velocities are corresponding to 2.78Hz, 5.55 Hz and 8.33 Hz of wheel rolling frequency. Dividing them by the frequency of 10.6Hz, we can get the results of 3.81, 1.91 and 1.29. Thus a ratio α is summarized as:

$$\alpha = \frac{f_L}{f_w}, \quad (4)$$

where f_L is longitudinal vibration frequency of wheelset according to bogie frame; f_w is rolling frequency of wheelset depending on forward speed and nominal radius, which is expressed as:

$$f_w = \frac{v/3.6}{2R_w} \quad \text{circle/s} \quad (5)$$

As the vibration resonance only occurs when the two vibration frequencies are close or when one is an integer multiple of the other, so a resonance vibration of wheelset in longitudinal direction tends to be induced when the vibration frequency is an integer multiple of wheel rolling speed, which means α is an integer.

From Eqs.3-5, we can get :

$$v = 3.6 \times f_w \times 2 \times R_w = 7.2 \times R_w \times \frac{f_L}{N} \quad (6)$$

According to Eq.6, the resonance forward speed for the simplified model is:

$$v = 7.2 \times R_w \times \frac{f_L}{N} = 7.2 \times 0.5 \times \frac{1062}{2} = 1912 \text{ km/h}$$

Suppose that the nominal radius of wheel is 500 mm. The resonance vibration occurs in 20km/h considering influence of other factors such as adhesion.

To further learn the influence of mass of the wheelset and the primary longitudinal stiffness on the longitudinal resonance vibration of the wheelset, we carried out the calculation of the speed at which resonance vibration occurs with different mass and primary longitudinal stiffness in the simplified model. The results with changing mass and constant primary stiffness are shown in table 2 and the results with changing primary longitudinal stiffness and constant mass are shown in table 3. In table 2-3, f_1 means the inherent frequency of the wheelset attained through the root loci analysis; f_2 means the inherent frequency calculated through Eq.3; v_1 means the speed of longitudinal resonance vibration calculated by incorporating f_1 into Eq.6; v_2 means resonance speed calculated by incorporating f_2 into Eq.6; V means the resonance speed gained by the simulation calculation of the simplified model.

Table 2. The resonance speed gained by different methods with different wheelset mass

Mass (kg)	f_1 (Hz)	f_2 (Hz)	v_1 (km/h)	v_2 (km/h)	V (km/h)
1000	20.86	24.66	37.54	44.38	45
2000	15.93	17.43	28.67	31.38	32
3000	13.38	14.24	24.08	25.62	26
4000	11.76	12.33	21.17	22.19	23
5000	10.62	11.03	19.11	19.85	20
6000	9.75	10.07	17.55	18.12	19
7000	9.07	9.32	16.32	16.77	17
8000	8.52	8.72	15.32	15.69	16

Table 3. The resonance speed gained by different methods with different primary longitudinal stiffness

K_{px} (N/m)	f_1 (Hz)	f_2 (Hz)	v_1 (km/h)	v_2 (km/h)	V (km/h)
1.0e7	9.69	10.07	17.44	18.12	18
1.1e7	10.16	10.56	18.30	19.00	19
1.2e7	10.62	11.03	19.11	19.85	20
1.3e7	11.04	11.48	19.89	20.66	21
1.4e7	11.47	11.91	20.64	21.44	22
1.5e7	11.87	12.33	21.36	22.20	23
1.6e7	12.26	12.73	22.07	22.92	23
1.7e7	12.64	13.12	22.74	23.62	24

Through table 2 and table 3, we can see that the softer the stiffness or the lighter the wheelset, the larger the difference between the approximated value (v_1, v_2) and the practical value V .

As the results of v_2 and v are very close, so we can use Eqs.3 and 6 to quickly calculate the resonance speed, and the result is quite close to that of the multi-body dynamic calculation. If we replace the track irregularity used in the model with the track irregularity of Germany high-speed trunk with high interference or Germany high-speed trunk with low interference, we can see that the longitudinal resonance vibration at the speed of 20km/h, and the difference between resonance vibrations on different trunk is only the amplitude of the vibration.

For the simplified model that just has one wheelset, the longitudinal resonance vibration will happen on the smooth, level and tangent track without irregularity as well. But the amplitude is smaller compared with the amplitude with track irregularity. It also

indicates that if the wheelset has occurred longitudinal vibration resonance, the longitudinal vibrations will not converge at a constant speed.

V. ANALYSIS OF LOCOMOTIVE MODEL

5.1. Root loci analysis

The root loci analysis of the locomotive model is shown in Fig.7. The movement of wheelset in front and rear bogies is out of phase at the frequency of 19.2Hz, and their interaction makes this vibration not shifted to the car body; whereas the movement of wheelsets in front and rear bogies is in phase at the frequency of 20.6Hz, and their interactions on car body is overlapping, which leads to a strong longitudinal vibration of car body. Moreover, as the gravity center of bogie frame doesn't coincide with geometrical center, the longitudinal forces from axle boxes to bogie frames will cause nodding movement on both bogie frames, and a vertical excitation due to the longitudinal force is thus produced on both ends of car body which leads to a strongly nodding vibration of car body. The frequencies of the discussed model will not change under an uniform running speed.

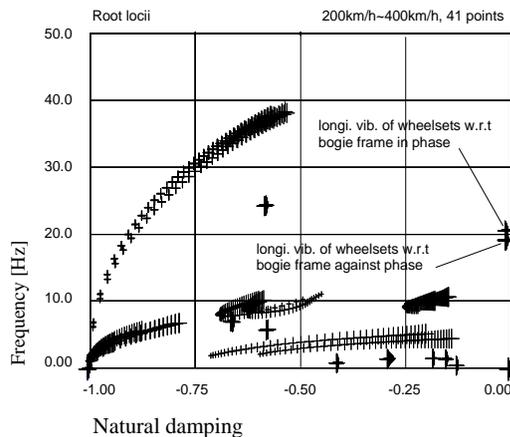


Fig 7. Root loci analysis of the locomotive model

Substitute $kx=2.0 \times 10^7 \text{N/m}$; $m=2500 \text{kg}$ into Eq.3

$$f_L = \frac{1}{2\pi} \sqrt{\frac{2 \times (2 \times 10^7)}{2500}} = 20.13 \quad \text{Hz}$$

Suppose the nominal radius of wheelset is 525mm. Then resonance speed for the locomotive model is:

$$v = 3.6 \times (2Rwfw) = 3.6 \times [2Rw \times (fL/N)] = 3.6 \times [2 \times 0.525 \times (20.6/1)] = 77.87 \text{km/h}$$

Further research showed that in the locomotive model the nominal mass used to calculate the inherent frequency should be slightly smaller than the wheelset mass; and the nominal stiffness used to calculate the inherent frequency should be slightly bigger than the primary longitudinal stiffness. In the locomotive model, the wheelset longitudinal inherent frequency is calculated just by the mass of wheelset and the primary suspension longitudinal stiffness, without consideration on the influence of the second suspension. So the actual frequency would be bigger than that calculated by Eq.3, and the actual speed which is 100km/h would be bigger than that calculated by Eq.6.

5.2. Wheel-Rail Longitudinal Interaction

Vibration corresponding to above eigenvalues is actually a small local rolling vibration of wheelset with respect to bogie frame, either in phase or out of phase (see Fig.8). Because of the track irregularity, a dynamic component is added to the nominal forward speed for wheelset.

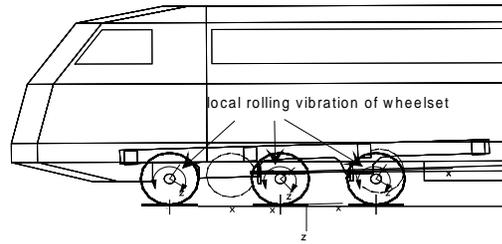
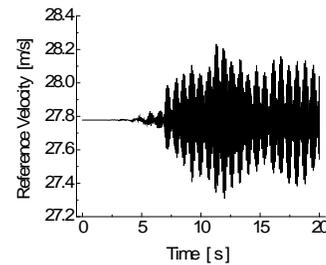


Fig 8. Local rolling vibration of wheelset within a bogie

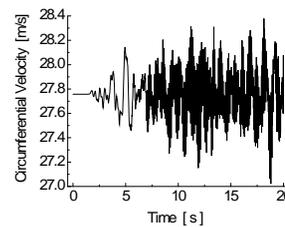
That means the reference velocity, which is often considered constant for creepage calculation is a variable. According to the definition of longitudinal creepage [6], it is expressed as:

$$\xi_x = (|v| - |c|) / v \quad (7)$$

where v is reference velocity and c is the circumferential velocity of wheel at contact point. There are different definitions of reference velocity [7], which lead to different calculation results of longitudinal creepage.



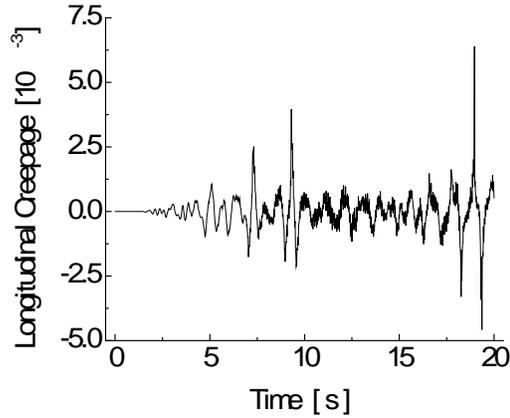
(a) Reference velocity



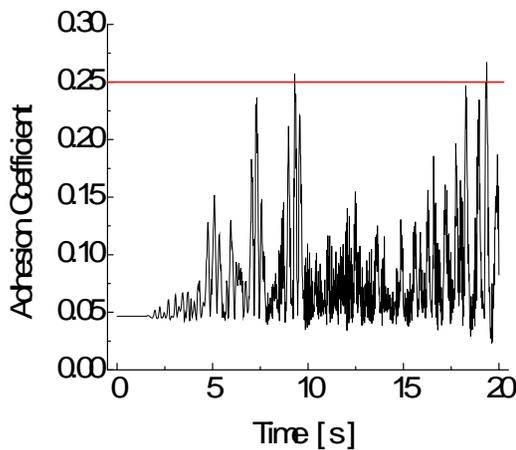
(b) Circumferential velocity

Fig 9. Reference velocity and circumferential velocity at contact point at 100km/h

For the analysed locomotive model, the reference velocity and circumferential velocity at contact point are given in Fig.9. The Longitudinal creepage and adhesion coefficient of wheelset are given in Fig.10.



(a) Longitudinal creepage



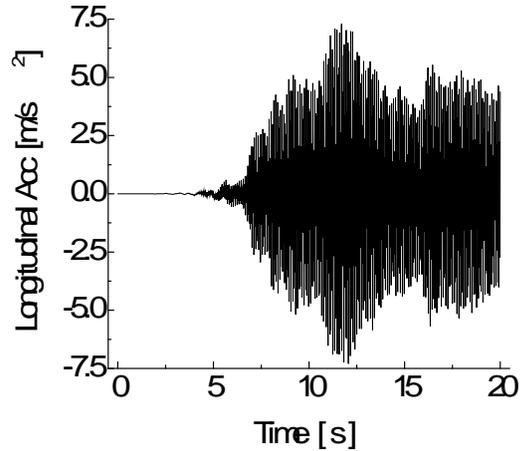
(b) Adhesion coefficient

Fig 10. Longitudinal creepage and adhesion coefficient of wheelset

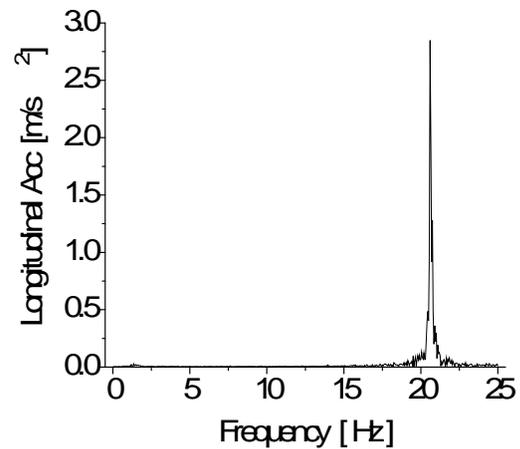
Longitudinal creepage of leading wheel at contact patch for example is thus obtained according to physical measurements in Figs.7 and 10, and at several points maximum adhesion coefficient reaches saturation value

0.25, see figure 10(b).

5.3. Effect of wheelset longitudinal vibration on carbody vertical vibration



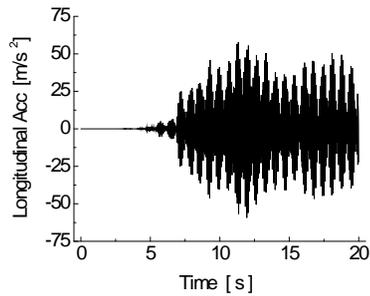
(a) In the time domain



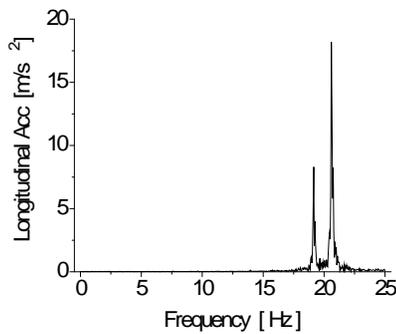
(b) In the frequency domain

Figure 11. Time history and frequency response of longitudinal vibration acceleration of car body

The time history and the frequency response of longitudinal vibration acceleration of the car body were shown in Fig.11, in which the longitudinal vibration acceleration of the locomotive is 7.5m/s^2 , and the main frequency is 20.6Hz .



(a) In the time domain



(b) In the frequency domain

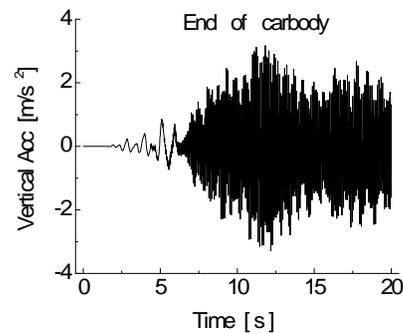
Fig 12. Time history and frequency response of longitudinal vibration acceleration of wheelset

The time history and the frequency response of longitudinal vibration acceleration of the first wheelset were shown in Fig.12, in which the amplitude of the longitudinal acceleration reached 70m/s^2 . It has two main frequencies that are very close and even intense-coupling. The vibration which frequency is 20.6Hz is transferred to car body.

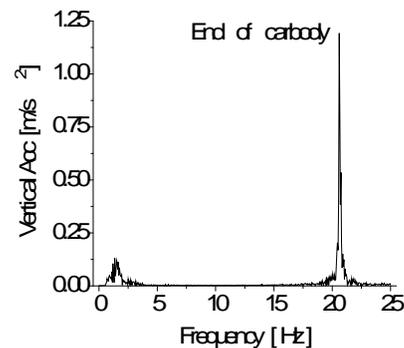
The maximum frequency of the longitudinal vibration of car body equals that of the wheelset. The vibration will not be transferred to the car body when the vibrations of two wheelsets in the same bogie are out of phase, as the vibrations of the two wheelsets within a bogie will be counteracted by the interaction. When the vibrations of the two wheelsets in the same bogie are in phase, it will cause high frequency vibration, which

would be transferred to car body, and produce the longitudinal oscillation and nod oscillation of the car body. From Figs.11 and 12, we can see that the effect of wheelset longitudinal vibration on locomotive longitudinal vibration is very large.

The time history and frequency response of vertical acceleration at the end of car body and the middle of car body were shown in Figs.13 and 14. The concerned result indicated that the vertical vibration of the locomotive was very intense, and clearly longitudinal oscillation occurred. Through the root loci analysis, we can see that longitudinal and nodding vibration occurred actually.

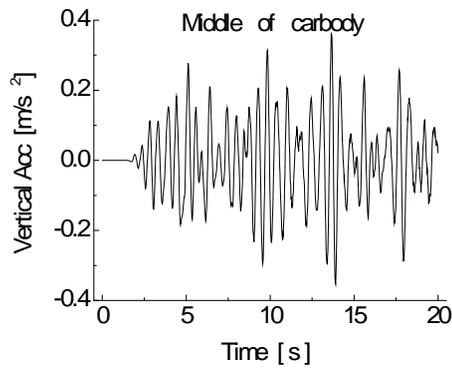


(a) In the time domain

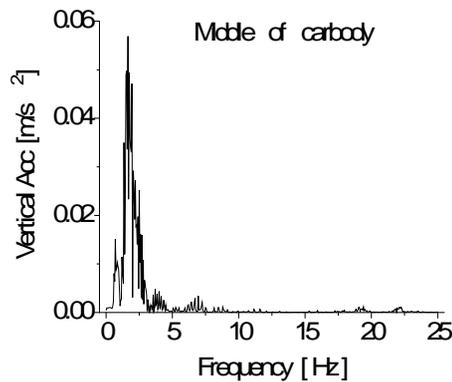


(b) In the frequency domain

Fig 13. Time history and frequency response of vertical vibration acceleration at the end of car body



(a) In the time domain

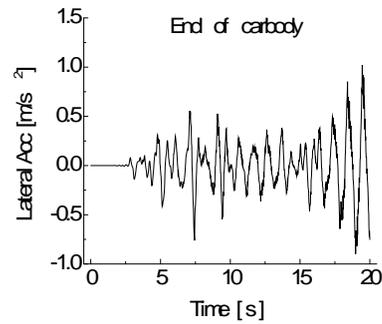


(b) In the frequency domain

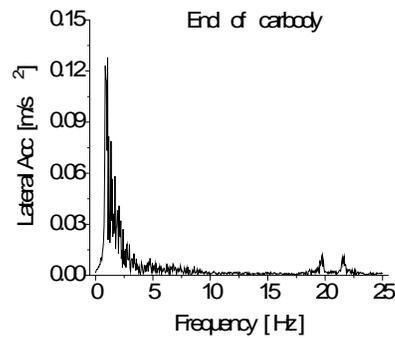
Figs 14. Time history and frequency response of vertical vibration acceleration at the middle of car body

The spectral analysis of the longitudinal vibration could find that a forced vibration with the frequency of 20.6Hz was applied at the both ends of the car body. The spectral analysis proved that the vibration frequencies leading to up and down and nodding of the car body are the values of 1.67 Hz and 1.37 Hz respectively.

Although the longitudinal vibration and vertical vibration of car body are extremely big, the amplitude of lateral acceleration is quite small (see Fig.15). It indicates that the calculation is stable and the longitudinal resonance vibration of the wheelset is not caused by the instability of the calculation.



(a) In the time domain



(b) In the frequency domain

Fig 15. Time history and frequency response of Lateral acceleration at the end of car body

5.4. The influence of wheelset longitudinal vibration on wheel treads spalling

Research results show that the strong wheel-rail dynamic force is the main reason that causes the wheel treads contact fatigue and spalling. Since the wheel-rail contact is a three-dimensional behavior on the contact point, the lateral wheel-rail dynamic force can be divided into three separate dimensional forces that are lateral force, vertical force and longitudinal force. On the condition of reasonable control on the lateral force and vertical force, reducing the longitudinal force can effectively reduce the wheel-rail dynamic force. Many factors have influences on

wheelset longitudinal vibration. A strong vibration, especially a resonance, may cause serious wheel-rail contact wear and shorten service life. If the vibration is suppressed, one of sources of wear on wheel-rail contact patch will be eliminated. It is hopeful to improve the wheelset operation condition through reducing the longitudinal force, and solve the wheel tread spalling to some degree. The relevant tests are presently underway.

5.5. Influence of the track irregularity on wheelset longitudinal vibration in the locomotive model

Through carrying out the linear vehicle system analysis adopting the Germany high-speed trunks with high interference and low interference and the medium irregularity as the track irregularity separately, we get the system response depending on the speed [14]. The speed corresponding to the vibration frequency that approximates to the inherent frequency is the very speed at which resonance vibration occurs.

The resonance frequencies with the three irregularities have slightly different amplitudes at the speed of 100km/h. So we can say that the calculated resonance speed is 100km/h. The difference of the locomotive model and the simplified model is that the longitudinal resonance vibration of the locomotive wheelset will not occur on the track without irregularity.

In this paper, vertical profile irregularity, alignment irregularity and cross-level irregularity are considered, and we find the phenomenon of longitudinal resonance vibration of the wheelset clearly. Now dispose one of the three irregularities respectively to find the source inducing longitudinal

vibrations.

First dispose vertical profile irregularity and reserve alignment irregularity and cross-level irregularity, the result is shown in Fig.16. It is shown that the longitudinal acceleration of the wheelset and vertical acceleration of the car body is quite small, longitudinal resonance vibration of the wheelset does not happen. It indicates that if there is not vertical profile irregularity, the longitudinal resonance vibration of the wheelset will not take place.

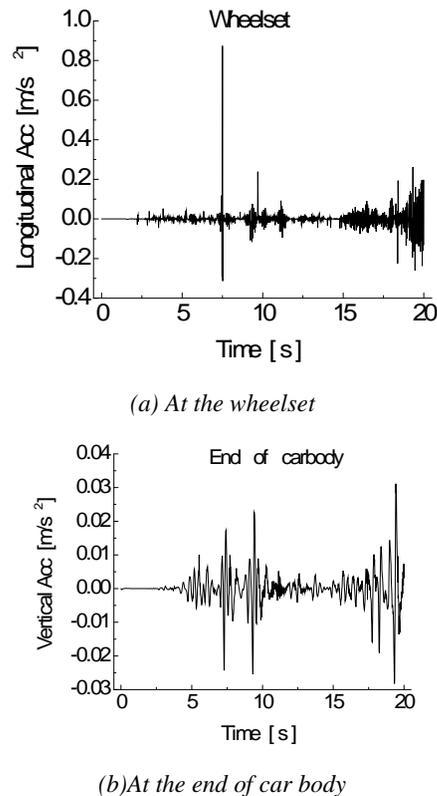
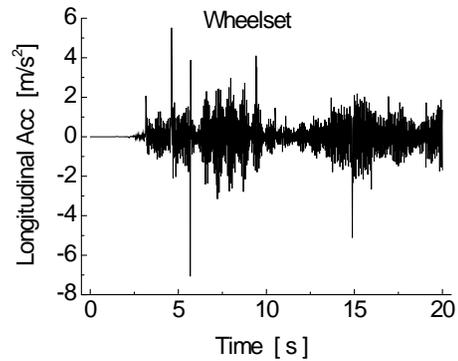


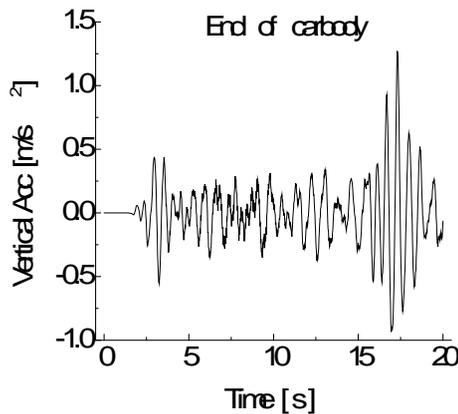
Fig 16. Longitudinal acceleration without vertical profile irregularity

Second dispose cross-level irregularity and reserve vertical profile irregularity and alignment irregularity, the result is shown in Fig.17. In this case, longitudinal acceleration

of the wheelset and vertical acceleration of the car body is much small, it doesn't cause longitudinal resonance vibration too. It proves that without the cross-level irregularity, the longitudinal resonance vibration of the wheelset will not take place, and the influence of cross-level irregularity is much bigger than that of the vertical profile irregularity.



(a) At the wheelset

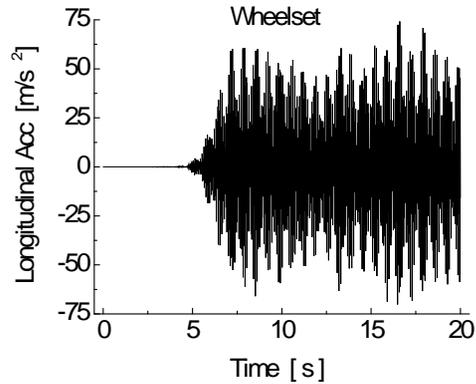


(b) At the end of the car body

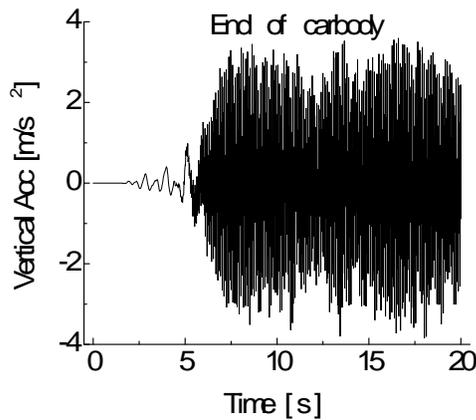
Fig 17. Longitudinal acceleration without cross - level irregularity

At last, dispose alignment irregularity and reserve vertical profile irregularity and cross-level irregularity, the result is shown in Fig.18. Obviously, with the vertical profile irregularity and cross-level irregularity, strong longitudinal resonance vibration takes place

on the wheelset.



(a) At the wheelset



(b) At the end of the car body

Fig 18. Longitudinal acceleration without the track alignment irregularity

The three above results show that the longitudinal resonance vibration is the product of interaction between vertical profile irregularity and the cross-level irregularity, and the alignment irregularity has little relation with the resonance. Single irregularity will not cause longitudinal resonance vibration of the wheelset at all.

If we replace vertical profile irregularity and cross-level irregularity with the corresponding Germany irregularity, the

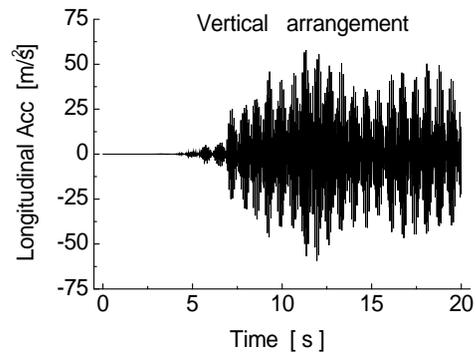
longitudinal resonance vibration will take place at the same speed. But on a smooth track under the same speed, the longitudinal vibration resonance will not taken place, which also validate the above conclusion. As the vertical profile irregularity and the cross-level irregularity always exists, the longitudinal vibration of wheelset always exists, and can even be developed into strong resonance within some ranges of the speed. If the vehicle runs under the condition of resonance vibration, wear of wheelset and rail will be very serious.

VI. A POSSIBLE SOLUTION FOR LONGITUDINAL VIBRATION

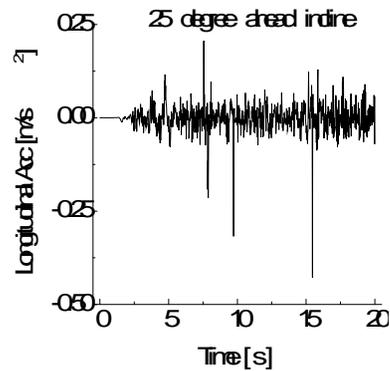
The control method is reducing the source causing the longitudinal resonance vibration. There are two kinds of methods: one is to improve the track quality, the condition of the track and reduce the track irregularity, but this method will cost a lot; the other is to improve the locomotive adaptability to the track irregularity, which can be easily achieved through changing the suspension parameter.

As the damper can reduce the vibrations, we can apply longitudinal damper to the primary suspension of the locomotive. Usually the locomotive do not have the primary longitudinal damper and a small slant angle is assigned to the damper in the vertical direction.

If the vertical damper has certain inclinations to longitudinal direction, it will produce a certain damping component in longitudinal direction. Generally the slant angle is 25° . Results for different damper arrangements are shown in Fig.19.



(a) With a vertical angle



(b) With a slant angle of 25°

Fig 19. Longitudinal accelerations with different damper arrangement

Result shows that the longitudinal damper is very effective for reducing longitudinal vibration. With the vertical arrangement, an extremely high resonance occurred, and its amplitude approached 70m/s^2 . With the slant angle of 25° , 28kNs/m damping is produced in longitudinal direction that eliminated resonance totally, and random vibration with much smaller amplitude become obvious. Comparing Fig.20 (right) with Fig.20 (left), we can conclude that a much longer service life of the wheel could then be expected.

VII. CONCLUSION

Longitudinal vibration of the wheelset with respect to bogie frame always exists while a vehicle operating on track. The amplitude of the vibration is influenced by many factors, such as vehicle structure, suspension and mass parameters, friction coefficient in wheel-rail contact patch and track irregularity, etc. In the worst case, a strong longitudinal resonance of the wheelset may be developed and cause severe wheel-rail contact fatigue, which lead to faster wheel spalling or rail corrugation with certain wavelength. In most cases, this vibration will not be transferred to car body but still exist within bogie with large amplitude and high frequency, which makes contribution on reducing wheelset service life. An approximate approach is presented to estimate the resonance speed according to the study on the simplified model.

The longitudinal resonance vibration of the wheelset will worsen the vertical dynamic performance and bring longitudinal shake of the locomotive. It relates to the cross-level irregularity and the vertical profile irregularity, and will not take place on the smooth track. Arranging the primary vertical damper with a forward angle is an effective solution to eliminate longitudinal vibration.

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