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ITJ

ISSN 1812-5638

INFORMATION TECHNOLOGY JOURNAL

ANSInet
Asian Network for Scientific Information
308 Lasani Town, Sargodha Road, Faisalabad - Pakistan

Information Technology Journal 12 (15): 3245-3248, 2013
ISSN 1812-5638 / DOI: 10.3923/itj.2013.3245.3248

Research on the Safety Threshold of High Speed Railway Wheel Ovalization Based on Numerical Simulation

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Abstract: By means of the vehicle-track coupled dynamics theory,a coupling dynamics model of railway vehicle with oval wheels and track is built and a new method is presented to describe wheel ovalization on high speed trains. The effect of wheel ovalization on the dynamic behavior of the vehicle system was investigated. The results show that wheel/rail impact force caused by the wheel ovalization depends greatly on the combinations of the wheel ovalization extent,the train speed,and the phase angle of the ovalization of the two side wheels of the wheel set. The safety threshold of wheel ovalization at vehicle speed of 250 km/h is determined according to the traditional criterion of wheelset loading reduction for safely running estimation.

Key words: High speed train, wheel ovalization, vehicle/track coupling dynamics, safety threshold

INTRODUCTION

The emergence of high-speed trains is one of the most significant technological advances in the transportation industry over the last part of the 20th century and the beginning of the 21st century. High-speed rail transportations quickly become a popular form of mass transit throughout the world. However, as the speed of the trains continues to increase, potential problems arise. Periodic non-roundness of railway wheel (OOR) is one of the most common concerns. It may contribute to passenger discomfort, or could potentially cause serious damage to the track and the train. Therefore, in order to guarantee safety and stability and to minimize costs for repair and maintenance and to meet noise legislation, it is desirable to determine a range for acceptable roundness values and to detect, remove and repair out-of-roundness wheels in time.

Much attention has been given to this subject including the dynamic influence and development mechanism of wheel out-of-round phenomena. The studies by Jenkins, et al. and Newton and Clark are contributions in experimental detection and numerical simulation of OOR that have played a significant role in the early understanding of OOR wheels. A literature survey written by Nielsen and Johansson (2000) discussed research on periodic OOR in North America and Germany and experiments on the Gotthard line. A series of numerical methods for predicting OOR are put forward by foreign scholars (Barke and Chiu, 2005). Meinke and Meinke (1999) adopted rigid body dynamics in combination with a wear model in predicting longer-wavelength non-roundness considering the rotor dynamics of high-speed wheel sets. Morys (1999) have investigated a rigid body model of an ICE carriage on an elastic track model. Johansson (2003), have carried out full-scale experiments, analyzed causes of different types of OOR wheels, simulated the influence of OOR on wheel/rail vertical contact force and track dynamics responses, put forward wavelength-fixing mechanisms of periodic OOR and proposed the development of periodic OOR wheel profiles. However, the mathematical models simulating periodic OOR wheels have not been generally known thus far.

In China, the development of high-speed railway is later than the developed countries. Many high-speed railways are being constructed since the Beijing-Tianjin intercity high-speed railway was put into operation in August of 2008. The problem of safety threshold of wheel OOR in high-speed operation are in need of further study. The primary intention of this paper is to put forth a new mathematical model to numerically simulate wheel ovalization, which is called the second order periodic non-roundness of the wheel. A coupling dynamics model of railway vehicle with oval wheels and track is built and a corresponding numerical method is developed. The effect of ovalization of the wheels on the dynamical behavior of the vehicle system is analyzed in details. It provides the foundation for determining the safety threshold of ovalization in high-speed operation. It also provides the theoretical foundation to conduct further research on the mechanisms that cause the development of periodic OOR.

DYNAMIC SIMULATION MODEL OF TRAIN-TRACK

Train-track system coupled model: By means of the vehicle-track coupled dynamics theory (Zhai, 2002) and

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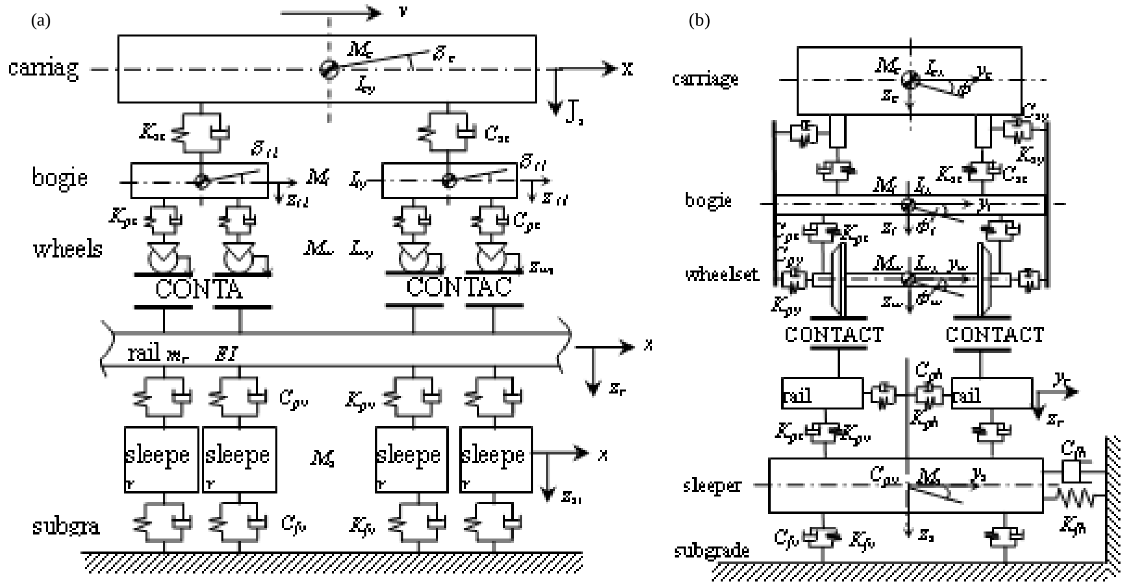

Fig. 1(a-b): Illustration of dynamic train/track interaction model

Table 1: Main technical parameters of vehicle model

Parameter	Value	Parameter	Value	Parameter	Value
M_c/kg	7831	I_{tx}/kg m^{-2}	1148	K_{px}/MN m^{-1}	0.40
M_t/kg	6.87×10^3	I_{ty}/kg m^{-2}	1236	K_{py}/MN m^{-1}	0.40
M_w/kg	$<2.0\times10^3$	I_{tz}/kg m^{-2}	2286	K_{pz}/MN m^{-1}	0.91
I_{cx}/kg m^{-2}	1.07×10^5	I_{wx}/kg m^{-2}	993	K_{sx}/MN m^{-1}	0.18
I_{cy}/kg m^{-2}	2.98×10^6	I_{wy}/kg m^{-2}	129	K_{sy}/MN m^{-1}	0.18
I_{cz}/kg m^{-2}	2.99×10^6	I_{wz}/kg m^{-2}	987	K_{sz}/MN m^{-1}	0.30
C_{pz}/kN s m^{-1}	4×19.6	C_{sz}/kN s m^{-1}	2×58.8		

the corresponding simulation software ADAMS,the vehicle-track system model is employed, in which elliptical railway wheels are considered in the calculation of the wheels and the rails in rolling contact. mode 1. In the vehicle sub-model, the car body is supported on two double axle bogies at each end. The bogie frames are linked with the wheelsets through the primary suspensions and linked with the car body through the secondary suspensions. Spring damper elements are used to represent the primary and secondary suspensions.The vehicle is assumed to move along the track at a constant traveling speed. In the track sub-model, both the left and right rails are treated as continuous Bernoulli-Euler beams which are discretely supported at fastener junctions.The calculation model of the whole-passenger car and the track is shown in Fig. 1. For the wheel rail contact, a coupling element including the elastic properties of the profiles, multiple contact patches and the simplified theory of contact by Kalker implemented in FASTSIM is used.

It is emphasized that the rolling circle is elliptical and not ideal circle. Vehicle-track coupling system dynamic responses are solved with new explicit integration methods. In this study, the Chinese high-speed passenger car CRH2 is considered and the main parameters are as follows: Lma wheel tread, wheel radius 430.0 mm, 60 kg m^{-1} rail. Values of other parameters of the vehicle are listed in Table 1.

Model of wheel ovalization: Wheel ovalization is the second order polygonalization. This type of OOR has a periodic radical irregularity around the wheel circumference. The traditional method naturally converted it to track geometric irregularity as system excitation. In this paper, a new method is put forth to numerically simulate wheel ovalization.

OOR shape are modeled by a variable radius around the tread as a function of the angle of wheel rotation φ. As depicted in Fig. 2, based on the "wavelength-fixing mechanism", the deference between the instant rolling radii and nominal contact radius of wheels ΔR can be expressed as:

$$\Delta R = a\sin\left(2\cdot\varphi(t)\right) \tag{1}$$

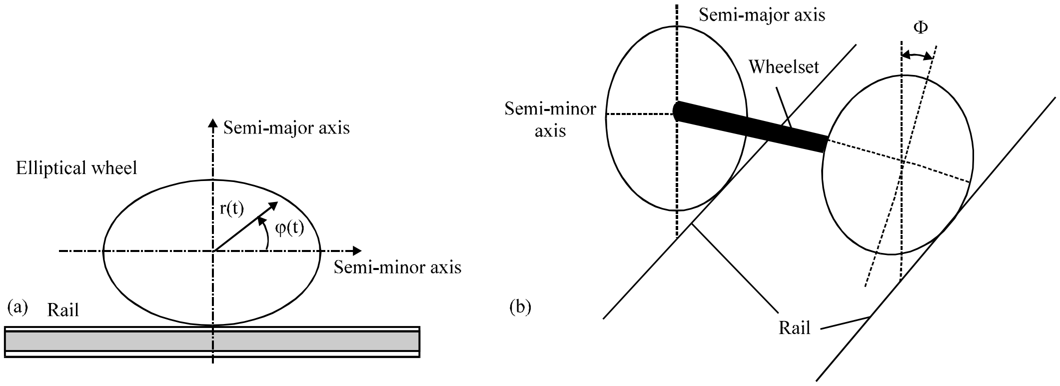

Fig. 2: Model of elliptical wheel

where, a is the difference between the length of the semi-major axis and the semi-minor axis, which is defined as ovalization extent and φ(t) is the angle between the initial contact radius of the elliptical railway wheel and the horizontal axis at t time, we have:

$$\varphi(t)=\varphi(t-1)+\omega\Delta t \tag{2}$$

where, ω is the rotational velocity of the wheel and Δt is the time step length of integration. The corresponding program has been compiled, which modifies the elliptical wheel initial nominal contact radius at each time step and the parameters of the contact geometry of the wheel/rail are obtained.

The two side wheels of the same wheelset are frequently asymmetrical. In this paper, assume that the two side wheels of the same wheelset are asymmetrical, there is phase angle difference Φ, as shown in Fig. 2.

NUMERICAL RESULTS AND DISCUSSIONS

The system dynamics are compared for different ovalization extent a and phases angle difference Φ in three different speeds 250, 300 and 350 km h^{-1} but with track irregularities being neglected. For brevity, only parts of the simulation results are plotted.

Case 1 corresponds to the condition that the two side wheels of the wheelset are in same phases and the ovalization extent a changes from 0.4-2.0 mm. Figure 3a illustrates the relationship of wheel-rail impact force versus train speed, when the wheel ovalization is 1.4 mm and the phase angle difference Φ of the same wheelset is 0. Figure 3b illustrates the relationship of wheel-rail impact force versus wheel ovalization at a train speed of 300 km h^{-1}.

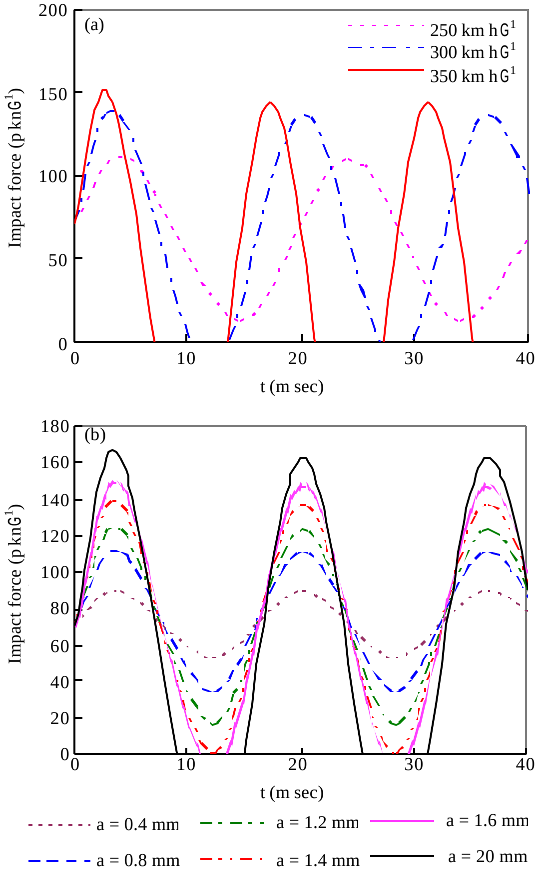

Fig. 3(a-b): Wheel-rail impact force under wheel ovaliation excitation. (a) a = 1.4 mm, (b) train speed 300 km h^{-1}

Simulations show that wheel-rail impact force periodical changes at a wavelength of 1.35 m, which is half of the rolling circumference of the railway wheel. The frequency increases with increasing train speed and the same trends as for the maximum wheel/rail contact forces are observed. The minimum of impact force is zero when train speed reaches at 300 km h^{-1}, that is the wheel tread detached from rail instantly. The detachment time

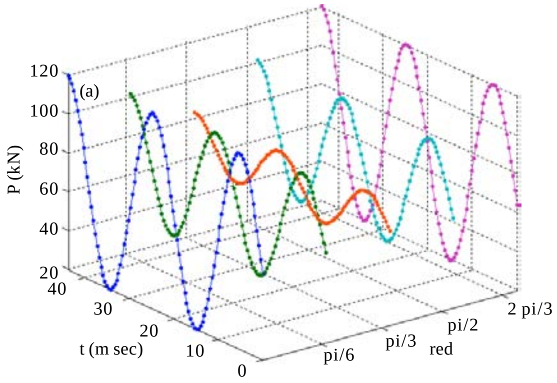

Fig. 4: Influence of phase angle on wheel rail impact force

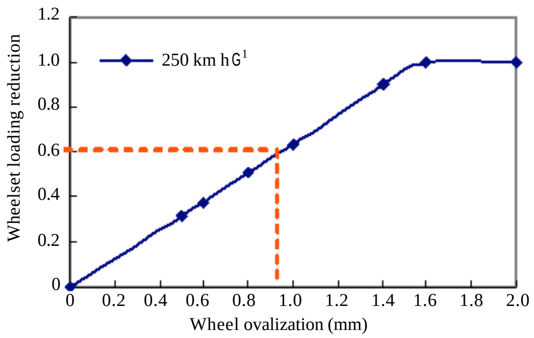

Fig. 5: Safety threshold of wheel ovalization

increases with increasing train speed. It can also be seen from Fig. 3b that the peak-to-peak wheel/rail force increases monotonically as the ovalization extent increases. Wheel/rail detachment occurs when a reaches at 1.4 mm and as ovalization extent continues to increase, the detachment time increases.

Case 2 corresponds to the condition that the two side wheels of the leading wheelset are in different phases. In Fig. 4, wheel rail impact forces versus phase angle difference Φ are shown, when the train speed is 300km/h and the ovalization extent is 1.2 mm. The results show that the peak-to-peak wheel/rail impact force increases with Φ changing from π/6 to π/2 and decreases with Φ changing from π/2 to 5π/6. It is concluded that the phase angle difference of the same wheelset often lead to large impact loads and the phenomena that one wheel of the bogie detachment from rail occurs in high-speed operations, which can have a detrimental influence on the train running safety.

It is viewed from simulation results that the limit of the wheel ovalization is determined according to the traditional criterion of wheelset loading reduction for safely running estimation. The relationship between reduction in wheelset loading and wheel ovalization status at the speeds of 250 km h^{-1} is illustrated in Fig. 5. It is shown that wheel ovalization extent is 0.94 mm when wheelset loading reduction $\Delta P/\bar{P}$ reaches at 0.6, which is the safety limit and the maximum of wheel/rail impact force is 117 kN.

CONCLUSIONS

The objective of the paper has been to determine the safety threshold of wheel ovalization in high speed operations, in order to detect and replace out-of-round wheel in times. High speed railway wheel ovalization causes various vehicle-track coupling system dynamic responses, which aggravates the running character and even decreases the service time of both the track and vehicle components. Therefore, further investigation need to be done to explore the formation and development mechanisms of OOR in order to instruct railway traffic on-site detecting and repairing.

ACKNOWLEDGMENT

This research was supported by the National Natural Science Foundation of China (No. 51208318, 11372199), the Hebei Province Natural Science Foundation (No. E2011210032) and also by the Educational Scientific Research Project of Hebei Province (No. Q2012025, ZD2010110).

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