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Effect of Wear on Wheel/rail Normal Contact Relationship

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Abstract. According to the non-Hertz contact theory, a three dimensional spatial model was established, which can accurately reflect the wheel/rail contact, the program can be used to calculate normal contact for the arbitrary wheel/rail profile, study the effect of wear on wheel/rail normal contact relationship, the non-Hertz contact theory is adopted to calculate the contact span and normal force. The results indicated that the influence of wear on the wheel / rail contact stress and the contact spot distribution is negligible.

Introduction

E.Meli, S.Magheri[1] argue that wheel-rail contact is the basis of vehicle system dynamics analysis, and the wheel-rail contact model should provide accurate and clear intercourse between global and local contact, especially contact points, contact stresses and contact Spot shape. They used Matlab / simulink to combine with the multi - body system dynamics model to provide a flexible wheel - rail contact model. The linear elasticity theory and Navier equation are used to calculate the wheel - rail contact stress.2002 Braghin[2] based on Kalker's CONTACT numerical algorithm to solve the non-Hertzian contact problem, the calculation of wheel wear is based on the elastic half-space contact theory, using the wear model assumes that the contact spot material wear and friction work density Proportional, calculated by the influence function, the computing time is too long, only update two wheels while the vehicle is running in 10500km. Compared with the experimental data, the simulation results show that the two results are different in terms of flange wear, but in terms of tread wear, the results are more consistent, and his study clarifies the wear coefficient of tread wear area.In 2005, A Alonso, JG Gimenez [3] proposed a new method to avoid the Hertz theory requires that the contact object gap must be expressed by quadratic polynomial expression, and the contact stiffness is used to simplify the wheel-rail normal contact The problem is given a detailed method to deduce the normal contact of the wheel and rail, and the calculated results are compared with the Kalker method. K.Knothe[4] gave a detailed introduction to the development history and research methods of wheel-rail contact theory of railway vehicles around the world, and made clear that the theoretical branches of the current research are based on Heinrich Hertz, Hans Fromm, Frederick William Cater , And summarizes the research theories and achievements of Joost Kalker, Ken Johnson et al., Who have made great contributions in the field of wheel and rail contact in the last 50 years. Põdra and Andersson[5] proposed a model called winker to simulate the wheel-rail normal contact, which is faster and more accurate than the previous model. The Winker model is equivalent to a mass consisting of many springs, Regardless of the lateral deformation between objects, the spring stiffness is used as the parameter of the study condition, and the normal stress on the contact spot is calculated according to the deformation of the spring. Massimiliano Pau, Francesco Aymerich[6] In paper[7 ~ 10], the abrasive evolution of the tread surface shape, the tread shape optimization and the influence on the dynamic performance. The ultrasonic method was used to study the contact problem, give the distribution of the contact patch and the size and distribution of contact stress and the results were compared with the Hertz method. The wheel rail contact parameters are determined, and the test equipment is calibrated by the finite element method.

Model and Calculation Method

Non - Hertzian Contact Theory

Using the non Hertz theory to solve the wheel rail contact problem, the assumption of Hertz contact can be avoided, and more accurate position and results can be obtained. For the non-Hertz problem, the lack of analytic form, for the form of contact form is particularly prominent. Generally speaking, can ignore the tangential force to the normal force, because the two are coupled, ignoring the tangential force, accuracy loss is very small, assuming two elastic surface friction free, you can use this method to solve the simplified method to the normal force.

In the contact of wheel and rail materials isotropic, ignoring friction between the effect of wheel rail contact, the contact surface is discretized into a rectangular grid, and then according to the deformation of an elastic half space on the surface of concentrated force, in each unit central are the following relationship:

$$\bar{u}_{zi} = \sum_{j=1}^{n} \varphi_{ij} p_j \qquad (i = 1, 2, 3, \cdots, n)$$

Where: are the influence factor, are elastic displacement, are stress.

For rectangular elements, the mathematical expression of the influence coefficient:

$$\varphi_{ij} = \frac{1-\nu^2}{\pi E} \int_{x_j - a_j}^{x_j + a_j} \int_{y_{j-b_j}}^{y_j + b_j} \frac{dxdy}{\sqrt{(x - x_i)^2 + (y - y_i)^2}}$$

According to the displacement equation of each element, the equations of the whole contact area can be obtained :

$$[\Phi][P] = [G]$$

Non-Hertz Calculation

In order to obtain accurate and reliable simulation results, we need to establish a model that can reflect the real situation of wheel and rail. In the vehicle operation process, the wheel shape and rail shape can not always meet the ideal design shape, with the increase in operating mileage, wheel and rail will wear, according to statistical wear is divided into uniform wear and non-uniform wear, will inevitably lead to the wheel / rail profile and wheel rail contact changes. Thus, a new wheel-rail contact model was established, as shown in Fig.1.Calculation parameters: CN60 rail, S1002CN wheel, wheel weight 105kN, rolling circle from the inside of the 1353mm, 1435mm gauge, rail bottom slope 1/40, nominal rolling radius, rail and wheel for homogeneous materials, elastic modulus, Poisson's ratio, the calculation object is right right wheel rail contact, yaw angle 0, lateral displacement is 0. The calculation flow is shown in Fig. 2.

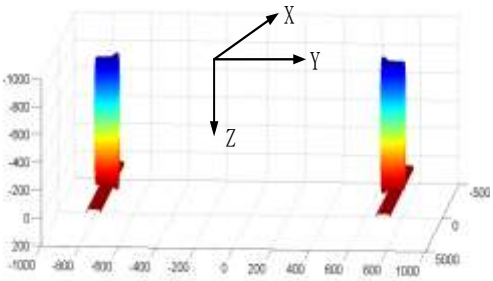


Figure 1. wheel and rail space model and coordinate system direction

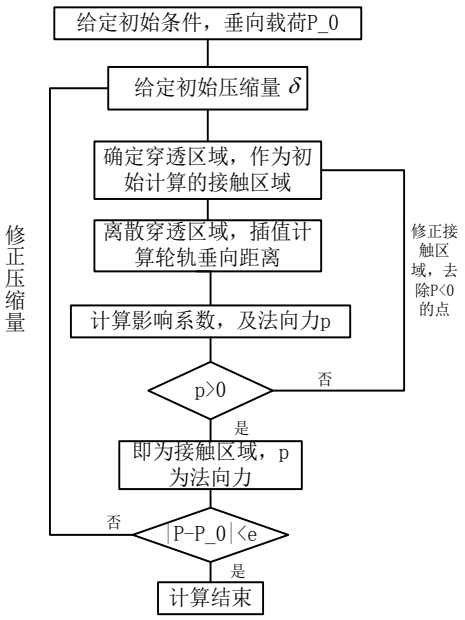


Figure 2. calculation flow of non normal force

The effect of Wheel Wear on Normal Contact

The Effect of Uniform Vertical Wear on the Normal Contact

The effect of the wheel continuously on the rail, running for a period of mileage, it is bound to happen between wheel and rail wear, wear with the development of the wheel profile and rail profiles are in constant change, which leads to the contact between wheel and rail is also changing. Using the non-Hertz method to study the relationship between the wheel wear and the normal contact of the wheel and rail using the 15-car No. 1 tread profile data measured in the wear tracking test, it is assumed that the rail does not wear, the wheel wear mileage and profile is shown in Fig. 3.

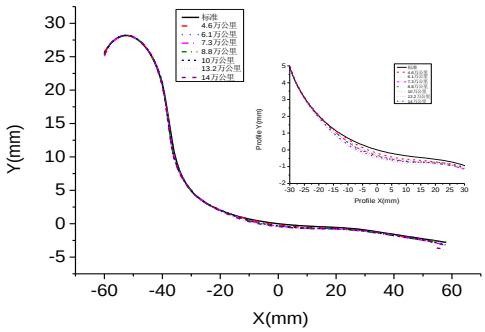
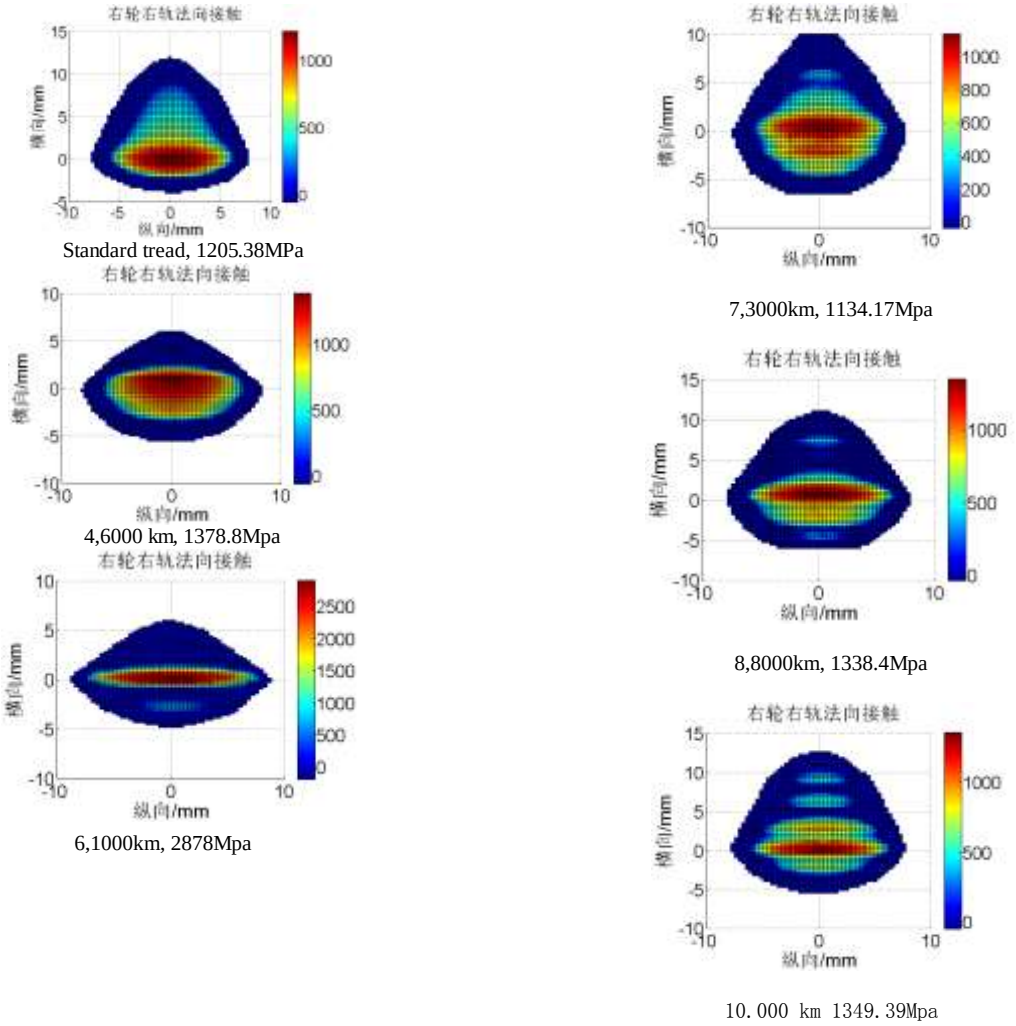


Figure 3. wheel wear tread profile

Calculate the relationship between the normal contact patch with the grinding mileage of the 15 wheel 1 , and it can be seen from the distribution of the contact patch: When the wear mileage is 4.6000 and 6,1000 km, the contact patch shape changes from ellipse to jujube nucleus, and the normal stress value changes obviously and the stress concentration occurs; In the wear mileage of 7,3000 and 8,8000 km, the contact patch shape close to the normal stress distribution close to; With the increase of wear mileage, the shape of the contact patch is consistent, which is similar to the ellipse.



Standard tread, 1205.38MPa

4,6000 km, 1378.8Mpa

6,1000km, 2878Mpa

7,3000km, 1134.17Mpa

8,8000km, 1338.4Mpa

10.000 km 1349.39Mpa

Figure 4

Fig.4 shows the relationship of maximum normal contact stress and wear with the operating mileage , we can see from the figure 4, in the case of a certain mileage, 15-1 wheel maximum normal contact stress and wear trends Basically no coincidence, 0 ~ 61,000 km wheel wear and changes in contact stress trends consistent, was positively correlated; 73,000 km, 132,000 km between the two trends in the opposite. With the increase of wear mileage, the contact stress of wheel and rail increases first and then decreases, and the change of normal stress is about 1250MPa. When the wear mileage is 61,000 km, the normal stress is 2878MPa , The analysis may the possible reason is that the error of abrasion measurement, can also be used to the contact stress is only related to tread profile, and wear track in the test is just about the amount of wear section data, can not reflect the actual profile of the tread.

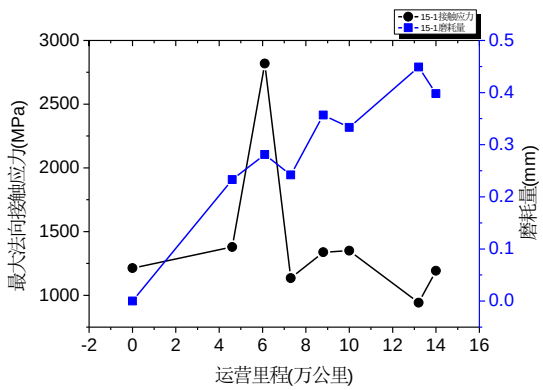


Figure 5. The relationship between contact stress and wear with operating mileage

The Effect of Nonuniform Wear on the Normal Contact

According to the results of the tracking test of wear that is wheel OOR(out of roundness) on the Beijing-Shanghai line, we can see that the wheel OOR on the Beijing-Shanghai line are mainly 1, 3, 6 and 11 order periodic polygons. Therefore, the fourth-order periodic polygon the influence of the normal contact law and the regularity of the polygon on the normal stress distribution in the rolling process is calculated. Calculation parameters: CN60 rail, S1002CN wheel, wheel weight 105kN, rolling circle from the inside of the 1353mm, 1435mm gauge, rail base slope 1/40, nominal rolling radius, rail and wheel for homogeneous materials, elastic modulus, Poisson's ratio, yaw angle is 0, lateral displacement is 0, the amplitude is 0.1mm polygon. The wheel is calculated by rolling 5O every time, and the normal contact of the polygonal wheels in a symmetrical period is calculated.

Calculate the normal force of 1 order, 3 order, 6 order and 11 order periodic polygon in the rolling process of the wheel, The results are shown in Figure 5, and the maximum contact normal stress is shown in Figure 7. The maximum normal contact stress of the 1and 3 order polygon is not changed during the rolling process of the wheel rolling 1/3 circle. The 6 order polygon produces the maximum contact stress of the wheel and rail, and the wheel-rail method caused by the 11 order polygon The contact stress is small.3 order polygon appears 2 times the peak, 6 order polygon appears 4 times the larger peak, 11order polygon appears 8 times the larger peak, we can see that the high order will increase the peak stress frequency, the wheel line normal contact stress The size of the polygon is not directly related to the order.

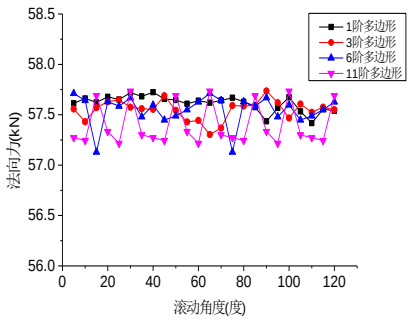


Figure 6. normal force of wheel polygon

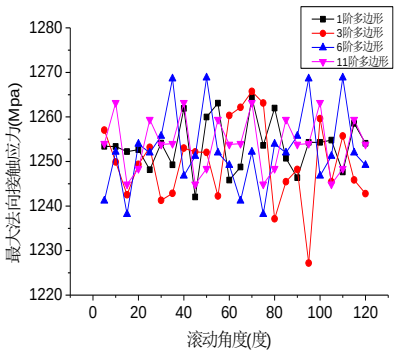
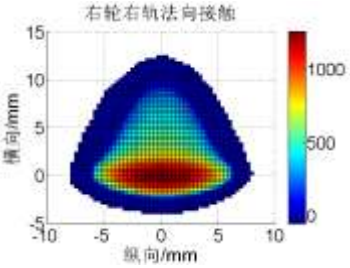
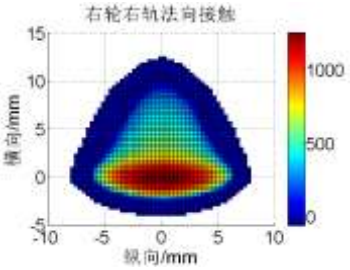


Figure 7. normal contact stress of wheel polygon

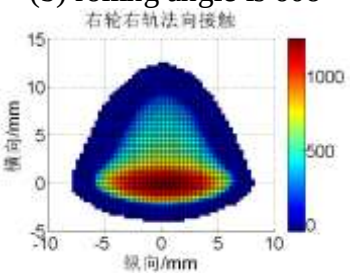
The calculation results of the contact patch shape, as shown in figure 8 to figure 11, show that the normal contact contour of the right wheel of the right polygon is almost the same, and the interior is elliptical, and the whole is fan-shaped.



(a) rolling angle is 30o

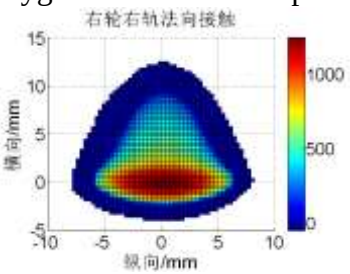


(b) rolling angle is 60o

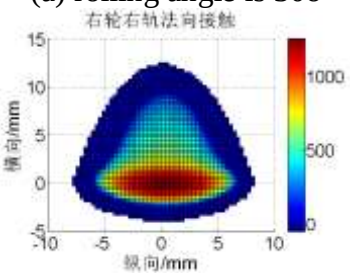


(c) rolling angle is 120o

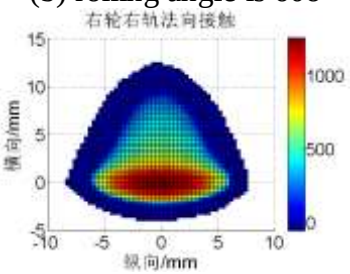
Figure 8. 1 order polygon normal contact patch of right wheel-rail



(a) rolling angle is 30o

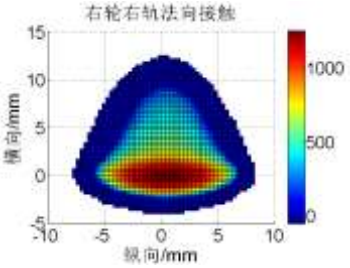


(b) rolling angle is 60o

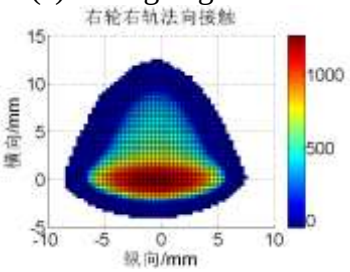


(c) rolling angle is 120o

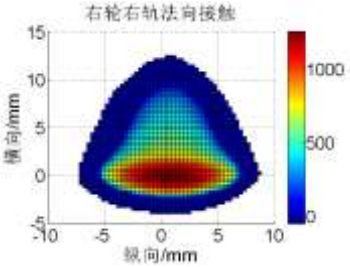
Figure 9. 3 order polygon normal contact patch of right wheel-rail



(a) rolling angle is 10o

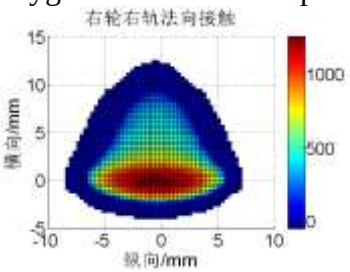


(b) rolling angle is 30o

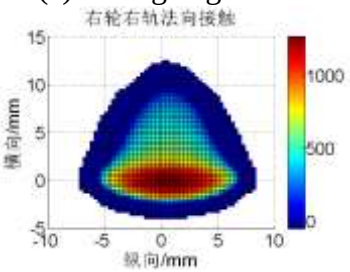


(c) rolling angle is 60o

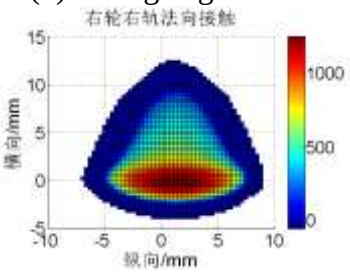
Figure 10. 6 order polygon normal contact patch of right wheel-rail



(a) rolling angle is 5o



(b) rolling angle is 10o



(c) rolling angle is 15o

Figure 11. 11order polygon normal contact patch of right wheel-rail

Conclusion

The influence of abrasion on the normal contact behavior of wheel and rail is analyzed, and an accurate three - dimensional wheel - rail space model which can consider any wear profile of wheel is established. Based on the non - Hertzian contact theory, the wheel - rail normal contact program can be prepared to solve arbitrary wheel profile. The vertical uniform wear tread and nonuniform wear (wheel polygon tread) under different wear mileage are taken as input, and the following conclusions are drawn:

(1)With the increase of the wheel wear mileage, the normal force change is small, always around 57.5kN; the normal contact stress change is large and there is no regularity, the contact stress is about 1200MPa, when the operating mileage is 6,1000 km when the normal contact stress up to 2878MPa; contact patch shape in the 100,000 km after the convergence, the shape of the approximate oval.

(2) There is no obvious relationship between the wear and the normal contact stress. The polygon order has no effect on the distribution of the contact patch and the normal contact stress, but it will affect the frequency of the maximum contact stress.

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