Simulation and analysis of wheel wear prediction with rigid flexible couple wheelset

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Abstract: Wheel wear problem of high-speed train is one of the important factors that affect the dynamic performance of the train, from the angle of wear prediction, combined with the Hertz vehicle dynamics model, rolling contact model, wheel wear model, analyzed the main influencing factors of wheel wear. The rigid flexible coupling dynamic model is established to calculate the wheel rail contact relationship considering the wheel elastic deformation, and the Fastsim algorithm is used to calculate the creep force and contact parameters on the wheel rail contact spot. Combining the Fastsim algorithm and the Zobory wear model, the wear rate of the contact surface between the rigid wheel and the elastic wheel is obtained. The results show that in a straight section considering wheel elastic model to increase the width of the rigid model of wear, wear depth of the amplitude is also increasing. The maximum wear depth of rigid model is 9.23e-4mm, the maximum wear depth of the rigid flexible couple model is 1.019e-3mm, and the maximum depth of rigid flexible couple model increased 9.4%. The effects of the velocity and the longitudinal stiffness of the two models are analyzed. When the velocity and the longitudinal stiffness of the system are increased, the wear depth of the flexible wheelset model is more obvious than that of the rigid wheelset model.

1. Introduction
High speed train realizes traction and braking by means of the adhesion between wheels and rail. With the speed of high speed train, the running mileage increases and the wear of the wheel profile is becoming more and more serious. It shortens the cycle of the wheel rotation and increases the maintenance and operation cost of the vehicle. At present, the wear of the wheel surface is one of the most prominent problems in the operation of the high speed train [1]. Through a large number of tracking tests, it is found that the dynamic performance degradation of vehicle and the damage and loosening of parts are closely related to wheel wear.

Scholars at home and abroad have done a lot of research on this problem. On the basis of the theory of Achard wear, Jendel[2-3] based on the results of the three kinds of friction testing machines, the Jendel wheel wear model was put forward, and the calculation formula of the material wear volume in the sliding zone is given. The wheel rail contact parameters were calculated by dynamic simulation software GENSYS, and the contact wear depth was calculated by Fastsim and Jendal models. LI and other [4] have formed a set of wear calculation model by using the dynamic program of self woven vehicle track coupling dynamic program, using Contact to calculate wheel rail rolling contact and the friction and wear model of material. The model is used to analyze the wear of subway vehicles. Based on vehicle dynamics, non Hertz rolling contact mechanics and Archard wear model, the model of wheel wear prediction was established by Li Xia and other [5-7]. The effect of different axle load and...
wheel hardness on wheel wear and rolling contact fatigue of heavy load truck was analyzed by using the model and shakeout chart. The wheel wear is calculated by the numerical method adopted by BRAUHIN[8], and the influence of brake shoe on wheel wear is analyzed by ENBLOM R[9], and the simulation is carried out. [10-11] established the dynamic model of the rail coupling system and the material wear model, compared the wear model of Archard, the wear model based on the friction work, the wear model based on the wear index, and the wear distribution under different mileage, then compared the simulation results with the measured wheel wear. In predicting wheel wear life area, Tao Gongquan and other [12] combined the model of vehicle multibody system dynamics, wheel rail non-Hertz rolling contact model and material friction and wear model, calculated wheel wear conditions under different wear mileage, and analyzed the influence of wear on vehicle lateral stability. Huang Caihong and other [13] established the wear prediction model based on the vehicle track system dynamics model and the wear model of the Achard wear model, and simulated and analyzed the wheel tread shape of the CRH3 high speed train running on the Wuhan Guangzhou line, and compared the wear shape of the measured wheel on the wear shape to modify the wear prediction model.

In this paper, the rigid flexible coupling dynamic model of vehicle is established on the basis of wheelset elasticity, and a more accurate wheel rail contact parameter is used to better reflect the wear state of the wheel in the actual situation. Fastsim algorithm is used to calculate the wheel-rail force parameters in contact spot, and Jendel model is used to calculate the wear depth of wheel profile. The vehicle parameters are optimized locally, and the influence of the speed and the longitudinal orientation stiffness on the wheel wear is analyzed. Finally, through repeated calculation of the accumulative amount of wheel wear, the error and accuracy of the wheel wear prediction under the two models are analyzed by comparing the tread surface with the measured tread surface.

2. Wear depth calculation

2.1 Wheel rail contact algorithm

The wheel rail vertical force is calculated by Hertz contact theory, FASTSIMcontact theory is applied to calculate the tangential force, and the size of the elliptical contact spot is calculated by the Hertz contact theory, and the mesh unit is divided in the contact spot, \( \Delta i \) and \( \Delta j \) are the length of each unit. According to the Herz contact theory, the normal force on the contact element is:

\[
p_{ij}(i, j) = \frac{3N}{2\pi ab} \left( \frac{i}{a} \right)^{\frac{3}{2}} \left( \frac{j}{b} \right)^{\frac{3}{2}}
\]

(1)

\[
\Delta d = \Delta v \times \Delta t = \sqrt{\Delta v_l^2 + \Delta v_t^2} \frac{\Delta t}{v_j}
\]

(2)

Above them, \( p_{ij}(i, j) \) is the normal force distribution on the contact mesh; \( a, b \) are half of the long axis and the short axis of the contact spot; \( \Delta v, \Delta v_l, \Delta v_t \) is the relative sliding velocity, the longitudinal relative sliding velocity, the transverse relative sliding velocity, and \( v_j \) the velocity of the particle through the contact point.

It is assumed that the elastic displacement at any point \((i, j)\) in the contact spot is related to the normal force \( P \) and the compliance coefficient \( L \) in the same direction, that is, \( u(i, j) = L \cdot p(i, j) \) the sliding equation is described as follows:

\[
\begin{align*}
\frac{V_x}{V_y} & = \xi_x - \phi j - \frac{\partial u_x(i, j)}{\partial i} \\
\frac{V_y}{V_x} & = \xi_j - \phi i - \frac{\partial u_y(i, j)}{\partial i}
\end{align*}
\]

(3)

\( V_x \) and \( V_y \) are longitudinal and lateral sliding speed; for train running speed; for longitudinal and transverse creep rates; for spin creep rate; for longitudinal and transverse elastic displacements.

The elastic sliding velocity can be expressed as:
\begin{equation}
\begin{cases}
\Delta v_j = v_j(\xi_j - \phi j - \partial u_j(i,j)/\partial i) \\
\Delta v_i = v_i(\xi_i + \phi i - \partial u_i(i,j)/\partial i)
\end{cases}
\end{equation}

Figure 1 Diagram of elliptical contact spot

According to the existing wheel rail contact theory, the contact zone area is formed by the adhesion zone and the sliding zone, and the wear occurs in the sliding zone within the contact spot. The contact area is divided (Figure. 2(a)), assuming that the tangential force is linearly related to the tangential direction between the wheel and rail

\[ u(i,j) = L_s F(i,j) \]

Above them: \( u(i,j) \) is the elastic deformation, \( L \) is the flexibility coefficient, \( F(i,j) \) is the tangential force between the wheel and rail. The flexibility coefficient \( L \) can be obtained from the relationship between creep force and creep rate in Kalker theory.

\[ L = \frac{\left| \xi_i \right| L_1 + \left| \xi_j \right| L_2 + c \phi L_3}{\left( \xi_i^2 + \xi_j^2 + c^2 \phi^2 \right)^{1/2}} \]

\( L_1 = 8a/(3GC_{11}) \), \( L_2 = 8a/(3GC_{22}) \), \( L_3 = \pi a^2/(4GcC_{11}) \), \( c = \sqrt{a \beta} \) (\( C_0 \) is Creep coefficient)

The contact spot (long axis is 2a and short axis length is 2b) is divided into \( i \times j \) units. The length of the cell are respectively \( \Delta i \) and \( \Delta j \), as shown in Figure1. In any element, the sliding speed is related to the rigid sliding volume and tangential deformation.

\[ v(i,j) = \frac{u(i_{k+1},j)-u(i_k,j)}{\Delta i} + p(i,j) \]

\( v(i,j) \) is the sliding speed, \( p(i,j) \) is a rigid slip.

In the adhesive zone, the sliding speed is 0, and the tangential force is:

\[ F(i_{k+1},j) = L \times F(i_k,j) - \frac{p(i,j) \times \Delta i}{L} \]

In the sliding region, according to the Coulomb friction theory, when the tangential force on the contact spot unit is greater than its maximum friction force (the normal force is multiplied by the friction factor), if \( F(i,j) \leq F_{\text{max}}(i,j) \), then the cell is in the adhesive zone; if \( F(i,j) > F_{\text{max}}(i,j) \), then, the sliding zone appears in the tangential force of the cell.

\[ F(i,j) = F_{\text{max}}(i,j) \frac{F(i,j)}{F(i,j)} \]

2.2 Wear Model

In the Zobory wear model, the contact spot is divided into \( n_x \cdot n_y \) cells. From the above section, it is known that the wheel rail contact area is divided into adhesion zone \( A_a \) and sliding zone \( A_s \). According to the wheel rail creep theory, no wear occurs in the adhesion zone, only wear occurs in the sliding zone. The friction energy flow density in the contact spot
\[
\dot{E}_d(i, j) = \begin{cases} 
\sigma_x(i, j)V_x(i, j) + \sigma_y(i, j)V_y(i, j) & (i, j) \in A_t(t) \\
0 & (i, j) \in A_e(t)
\end{cases}
\]  

Above them: \( \dot{E}_d \) wear energy flow density, \( \sigma_x \) and \( \sigma_y \) are longitudinal and lateral shear stress, \( V_x \) and \( V_y \) are wheel rail longitudinal and lateral slip speed respectively.

\[
m_d(i, j) = k(i, j) \cdot \dot{E}_d(i, j)
\]

Above them, \( m_d \) is wear mass flow density, \( k \) is a wear coefficient related to energy density. The relationship between the wear coefficient and the energy flow density is shown in Figure 2.

The mass per unit area calculated by mass flow density is:

\[
\Delta m = \int_0^\infty m_d(i, j) \cdot \Delta A \cdot dt = \int_0^\infty m_d(i, j) \cdot \Delta i \cdot \Delta j \cdot dt
\]

Because the grid blocks are very small, it can be approximated as:

\[
\Delta m = m_d(i, j) \cdot \Delta A \cdot \Delta t = m_d(i, j) \cdot \Delta A / v
\]

The wear depth can be expressed as:

\[
\Delta z = \Delta m / (\rho \cdot \Delta A)
\]

Above them, \( \Delta z \) the depth of wear, \( \rho \) is the density of materials, formula (14) will be introduced into formula (15) to calculate the wear depth of contact classes.

3. Rigid flexible coupling model

A model of a type of high speed train in China is established. The train uses S1002CN tread, figure 3 is a vehicle system dynamic model. It selects 1 car bodies, 2 frames, 4 wheels and 8 revolving arms, and first suspension and second suspension system. The motor is suspended in the frame. The car body, the frame, the wheel pair and the motor take into account six degrees of freedom longitudinal, lateral, vertical, side rolling, nodding head and shaking head. The gear box includes large gear, small gear and box, one side of the box is hanged on the frame and the other end is on the wheel. A total of 90 degrees of freedom are considered. The vertical and roll motion of the wheel pair are coupled together; the rotor arm only takes into account the point head freedom; the direction of the vehicle is x axis, the Y axis is parallel to the track direction, and the Z axis is perpendicular to the orbit to establish the coordinate system. The vehicle model is shown in Figure 3. Considering the nonlinear contact relationship between wheel and rail and vehicle suspension system, the dynamic equation of vehicle system is expressed as

\[
M\ddot{x} + C\dot{x} + Kx = F(\ddot{x}, \dot{x}, t) + Re
\]

In formula (16), \( M \), \( C \) and \( K \) is the mass matrix, damping matrix and stiffness matrix of the system; \( x \) is the coordinate vector; \( F(\ddot{x}, \dot{x}, t) \) it is a nonlinear force element, including the nonlinear suspension force and wheel rail force, and the t is the distribution matrix of the time as the orbit input; \( e \) is the track irregularity.
4. simulation results and analysis

4.1 Calculation process

The wheel wear prediction mainly includes the following steps:

1. Vehicle dynamics simulation, the dynamic rigid flexible coupling model of vehicle dynamics is set up by SIMPACK to obtain the calculation parameters of wheel rail contact of each integral step, and the wheel rail contact parameters are output.

2. For local wheel-rail contact calculation, Fastsim was used to re-divide the contact patch mesh, distinguish the adhesion zone from the sliding zone, and calculate the parameters needed by the wear model.

3. Wheel wear calculation, calculation of contact spot wear depth and linear interpolation accumulation to the tread, and the accumulated wear depth smoothing;

4. Wheel tread is renewed by subtracting wheel wear depth from original longitudinal coordinates and smoothing the renewed tread.

The calculation process is shown in Figure 4.

Due to the lack of actual line data, this paper selects the line shown in Table 2. In order to reduce the amount of calculation, the following assumptions are made: (1) the distribution of the left and right curves on the line is symmetrical; (2) the train does not turn round and round, and the friction factor of each curve or line is 0.3; (4) the wear of the rail surface is not considered in the calculation. Based on the above assumptions, the wear of the left and right wheels of the first and fourth wheelsets are the same, and the wear of the left and right wheels of the second and third wheelsets are the same.

The wear of the wheel profile increases gradually when the train is running on the actual line. In the simulation calculation, the vehicle dynamics calculation is first carried out, and the wheel tread can not be updated in real time, but the wheel tread can not be updated in real time. Therefore, the vehicle dynamics is calculated first, and then the wear depth in the process is calculated. In order to make the simulation results closer to the real situation, each wear step must be limited. The vehicle running distance and the wheel wear depth are two commonly used methods to limit the wear step. In this paper, the maximum wear depth of 0.1mm is used to update the wheel profile. The smoothness of the wheel profile is essential. After each grinding step, the wear distribution is smoothed by three spline interpolation, then the wear distribution is superimposed to the last wheel profile, and the three spline interpolation is used to smooth the superimposed wheel profile.
4.2 Comparison of wear distribution of two kinds of model

Considering the elasticity of the wheelset, the wheel-rail contact parameters of the wheels have changed to a certain extent. From Figure 5, it can be seen that the wheel rail contact point position of the two models can be seen that the lateral displacement of wheel rail contact point considering the wheelset elastic model is larger than that of the rigid model, the overall transversal displacement of the wheel rail contact point is larger and the maximum transversal difference can reach 0.015mm, which is due to the bending mode of wheelset after considering the wheel elasticity. The contact will move out. Figures 6 and 7 are the wheel wear depth of the model using the rigid wheelset model and the wheelset elastic model under the simulation condition of Table 3. When the wheelset elasticity is taken into consideration, the range of the transverse wear is slightly increased and the amplitude is increased obviously. The maximum wear depth is 9.23e-4mm when the rigid model is adopted, and the maximum wear depth of the elastic model is obtained. The maximum wear depth increases by 9.4% when the wear degree is 10.19 E-4 mm. The left and right wheels show symmetrical distribution due to the linear working condition.

Figure 5 Lateral displacement of contact point
4.3 The influence of speed on wheel wear

As shown in Figure 8 and Figure 9, the wear depth of the rigid model and the elastic model on the 20000m line is calculated with the velocity variation. The figure shows that the rigid model increases with the speed, the abrasion depth increases, the maximum wear depth of the rigid model increases from 9.23e-4mm to 11.4e-4mm, and the maximum wear depth of the elastic model is from 10.14e-4mm to 12.6e-4mm, the wear loss of the model considering the elasticity of wheelset is greater with the change of speed. When the speed increases, the direction force, the tangent force and the relative sliding speed increase, the wheel rail wear is aggravated and the wear depth increases with the increase of speed.

4.4 The influence of positioning parameters on wheel wear

This section mainly analyzes the influence of the longitudinal orientation stiffness on the wear of the wheel. For example, when the wear depth of the elastic wheel is larger than that of the rigid wheel, when the longitudinal stiffness is increased from 60MN to 120MN, the maximum wear depth of the rigid wheel model increases from 9.26e-4mm to 9.62e-4mm and increases by 3.8%; The maximum wear depth of the elastic model of wheelset is increased from 9.63e-4mm to 10.16e-4mm and 5.5% respectively. Considering wheelset elastic model is more sensitive to a series of longitudinal positioning stiffness
Figure 10 Effect of Longitudinal Stiffness on rigid model and rigid flexible couple model wear

Figure 11 Effect of Lateral Stiffness on rigid model and rigid flexible couple model wear

5. Conclusion
(1) the wear prediction adopts the elastic wheel model, and the change range of the wheel rail contact point on the wheel is wider, and the model wear of the elastic wheel is increased by 9.4% more than that of the rigid wheel.

(2) When the speed changes, the wear loss of the model considering the elasticity of wheelset is greater with the change of speed.

(3) the influence of the longitudinal orientation stiffness on the wheel wear is larger than the lateral stiffness, while the wear depth of the elastic wheel is larger than that of the rigid wheel when the elastic wheel is considered.

Because the simulation conditions cannot be compared with the actual conditions of the line, ignoring the impact of some small curves, as well as the weather, temperature on the friction coefficient. Accurate prediction of wheel wear can provide reference for wheel spinning. This paper only provides a way of thinking in simulation.

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