Study on Numerical Analysis Method to Wheel/Rail Wear of Heavy Haul Train

Chongyi CHANG, Chengguo WANG, Bo CHEN, Lan LI
Railway Science and Technology Research & Development Center,
China Academy of Railway Sciences, 100081 Beijing
E-mail: changchy@rails.com.cn, wangchengguo@rails.com.cn

Summary: A wheel/rail rolling contact finite element model based on Arbitrary Lagrangian Eulerian finite element method is developed and applied to the wheel/rail steady state rolling contact analysis. Based on the wheel/rail steady state rolling contact analysis, a wheel profile wear prediction methodology is developed and applied to the wheel wear of flange and gauge when the heavy haul train passes in curved track with the small radius. The vehicle dynamics model is built in the ADAMS software and the forces of the bearing boxes from the vehicle system dynamics model are loaded on the model of three-dimensional nonlinear finite element steady state analysis for wheelset rolling contact process. Three-dimensional nonlinear finite element steady state analysis is solved by the ABAQUS software. In the FEM simulation, a material model with bilinear kinematic hardening property was used. The wear modeling is based on Archard's wear model. Laplacian smoothing with area weights is used to smooth the meshes of wheel and rail. At last the simulation results of the wheel of heavy haul train are analyzed.

Index Terms: arbitrary lagrangian eulerian, finite element, lagrange multiplier formulation, rolling contact, steady-state

1 INTRODUCTION

The life of railway wheels is usually limited by wear. The wheel/rail surfaces are subjected to high sticking, sliding and contact stress in rolling contact. The removal of material from the surfaces by wear is relative to sliding, contact stress and material property. On curved track, the wear rate of wheel flanges and rail gauge faces are extremely large since the wheel flange and gauge corner contact are subject to a large contact stress and creepage. They are one major factor in vehicle/trace maintenance cost, so the study on wheel/rail wear attracts many researchers. Simulating wear of railway wheels was done in [1-6]. In their models multi-body code was mainly used for the dynamic railway vehicle simulation and the local contact analysis was solved by applying Hertz theory and FASTSIM. The Hertz method [7] assumes that the contact surfaces are smooth and can be described by a second-degree polynomial. The material model is linear-elastic and there is no friction between the contacting surface. Furthermore, the contact is assumed to be half-space. The half-space assumption puts geometrical limitations on the contact. This means that the significant dimensions of the contact area must be small compared with the relative radii of curvature of each body. Especially in the gauge corner of the rail profile, the half-space assumption is questionable since the contact radius here can be of the same size as the contact zone. With the contact software [8] the contact zone is divided into cells and a boundary element analysis with a half-space assumption is used to solve the contact problem. Furthermore, a linear material model is used in the analysis. But the finite element method is not limited by the two assumptions. T. Telliskivi and U. Olofsson [9] found that there were significant differences between the FEM and the Hertz method and the contact software in all of output data in a sharp curve case. The finite element parametric quadratic programming method is used to compute elastic and elasto-plastic rolling contact problems between wheel and rail in [10]. U. Nackenhorst [11] [12] apply ALE-formulation to analyze wheel/rail steady state rolling contact problems, and the method makes the
problems really become into a dynamic problem. Weak formulation of the equation of motion for the rolling contact problem is based on virtual displacement. Virtual work of the contact forces is described by relative slip velocities and force impact.

The objective in the present work is that the numerical method to predict wheel profile evolution due to wear is developed. The method is based on ALE steady state rolling contact FEM, and is used to analysis wheel wear of flange and gauge when the heavy haul train passes in curved track with the small radius. Three-dimensional nonlinear finite element analysis code ABAQUS is used in the simulation of wheel/rail disc steady state rolling contact process. For the wear prediction, Archards wear model was applied. Laplacian smoothing with area weights is used to smooth the meshes of wheel. At last, the simulation results are analyzed.

2 ROLLING CONTACT FEM MODEL BASED ON ALE

To solve the rolling contact problem of wheel/rail by classical Lagrangian FEM, numeral precision lies on the fine size of FEM grid in contact zone of wheel/rail. If one would apply this approach for rolling contact a couple of difficulties would arise. First, the wheel and rail would have to be discretized on their complete circumference with a fine mesh. Secondly, even for stationary rolling the time integration of the rolling process until a stable solution is reached required a very long part of the rail to be discretized as well. Third, an extremely small time step has to be chosen for a reasonable integration taking into account the fine discretization of the contact zone. The large free degrees and small time step lead up to waste much computer time.

This contact problem is not solved only by Lagrangian FEM, but it can be done by ALE FEM.

2.1 The Arbitrary Lagrangian-Eulerian Approach for Rolling Contact

For the ALE approach, total deformation of the rolling wheel is decomposed into a rigid body motion and material deformation. The rigid body rotation is described in a spatial or Eulerian and the deformation in a material or Lagrangian manner. The velocity of the center of the wheel is \( v_o \), and the wheel is rotating with a constant angular rolling velocity \( \omega_o \) around a rigid axis of wheel.

In an ALE method, \( x \) is the Eulerian coordination, \( X \) is Lagrangian coordination, and \( \chi \) is ALE coordination. The function \( x = \phi(X, t) \) maps the body from the initial configuration \( \Omega_0 \) to the current or spatial configuration \( \Omega \), and The function \( \chi = \chi(X, t) \) maps the current or spatial configuration \( \Omega \) to the reference configuration domain \( \hat{\Omega} \), and \( x = \phi(X, t) \) maps the reference configuration domain \( \hat{\Omega} \) to the current or spatial configuration \( \Omega \). The map of Lagrangian, Eulerian and ALE domains is shown in Figure 1.

In an ALE method, when the wheel is moved to the position \( y \) at the velocity \( v_o \) along the rail, then \( y \) could be described as

\[
y = x + v_o t
\]

(1)

The velocity of material particle of wheel in the relative kinematical description reads as

\[
v = \dot{y} = \frac{DX}{Dt} + v_o
\]

(2)

Where,

\[
\frac{DX}{Dt} = \frac{\partial \hat{\phi}}{\partial t} + \frac{\partial \hat{\phi}}{\partial x} \frac{\partial x}{\partial t}
\]

(3)

The circumferential direction \( S \) is defined by below equation

\[
S = \frac{T \times (X - x_o)}{R}
\]

(4)

Where \( T \) is the rigid axle of the wheel at \( X_o \); \( R = |X - X_o| \) is the radius of a point on the reference body.

Since the material particle of the wheel rotates with the axle of the wheel in grid and the constant angular velocity is \( \omega_o \), we now define the referential particle velocity as

\[
w = \frac{\partial \chi(X, t)}{\partial t} = \frac{\partial \chi}{\partial t} |_{\omega_o RS}
\]

(5)
Substituting equation (5) into equation (3), the material velocity is written as

\[
\frac{D\mathbf{v}}{Dt} = \frac{\partial \mathbf{u}}{\partial t} + \mathbf{u} \cdot \nabla \mathbf{u} = -\nabla \mathbf{p} + \frac{1}{\rho} \mathbf{f} = \frac{\partial \mathbf{u}}{\partial t} + \mathbf{u} \cdot \nabla \mathbf{u} = \nabla \times \mathbf{t} - \frac{1}{\rho} \mathbf{f}.
\]

Substituting equation (6) into equation (2), we can obtain

\[
v = \frac{\partial \mathbf{u}}{\partial t} + \mathbf{u} \cdot \nabla \mathbf{u} = \nabla \times \mathbf{t}
\]

At the same time, the material acceleration is obtained by a second differentiation and some manipulation:

\[
a = \frac{\partial^2 \mathbf{u}}{\partial t^2} + \mathbf{u} \cdot \nabla \mathbf{u} = \nabla \times \mathbf{t}.
\]

For steady-state condition these expressions reduce to

\[
v = \omega \mathbf{RS} \cdot \frac{\partial \mathbf{u}}{\partial \mathbf{x}} + v_0
\]

And

\[
a = \omega^2 \mathbf{RS} \cdot \frac{\partial^2 \mathbf{u}}{\partial \mathbf{x}^2}
\]

2.2 Principle of Virtual Power for ALE

The Arbitrary domain \( \Omega \) is introduced, of which boundary is \( \Gamma \), and the body force \( \rho b \) and surface force \( t \) are loaded on the domain. The momentum equation is described as

\[
\rho b - \rho b - \Delta \cdot \mathbf{u} = 0
\]

The principle of virtual power is the weak form of the momentum equation, the traction boundary conditions and the interior traction continuity conditions.

The space of test function is defined by:

\[
\delta \mathbf{u}(\mathbf{x}) \in \mathbf{u}_0, \mathbf{u}_s = \{ \delta \mathbf{u} | \delta \mathbf{u} \in C^0(\mathbf{X}) \} \text{ on } \Gamma,
\]

Where \( C^0 \) denotes that the 0th derivative of the function is a continuous function.

A boundary is called a traction boundary and denoted by \( \Gamma_t \), so \( \Gamma_t \) is described by equation (13). It is called a displacement boundary and denoted by \( \Gamma_s \), so \( \Gamma_s \) is described by equation (14)

\[
\mathbf{n}_t = \mathbf{t} \text{ on } \Gamma_t
\]

\[
\mathbf{n}_s = \mathbf{t} \text{ on } \Gamma_s
\]

\[
\mathbf{n}_t = \mathbf{t} \text{ on } \Gamma_t,
\]

Applying the differentiation theorem, the weak form about the momentum equation, the traction boundary conditions and the interior traction continuity conditions, principle of virtual power is described as:

\[
\int_{\Gamma_t} \delta \mathbf{u} \cdot \mathbf{t} d\Gamma + \int_{\Gamma_s} \delta \mathbf{u} \cdot N d\Gamma - \int_{\Gamma_t} \delta \mathbf{u} \cdot \mathbf{f} d\Gamma = 0
\]

Where \( D \) is the rate of deformation. For each of the terms in the weak form is a virtual power.

2.3 Contact Constitutive Models

Contact constitutive model includes the definition of the contact pressure and contact tangential traction. The contact pressure between two surfaces at a point, \( p \), as a function of the "overclosure" \( h \), of the surfaces.

\[
\begin{align*}
 p &= 0 \quad \text{for } h < 0 \\
 p &= Kh \quad \text{for } h \geq 0
\end{align*}
\]

Where \( K \) is contact stiffness.

Contact shear stress is defined by Coulomb friction model. In the adaptation of Coulomb friction models to continua, they are applied at each point of contact interface. If the two surface are in contact at \( x \), then

\[
\begin{align*}
 &a) \text{ if } \tau_t(x, t) \leq \mu p(x, t) \text{, } \lambda_t(x, t) = 0 \\
 &b) \text{ if } \tau_t(x, t) = \mu p(x, t) \text{, } \lambda_t(x, t) = -k(\tau_t(x, t)) \text{, } k > 0
\end{align*}
\]

Where \( k \) is a variable which is determined from the solution of the momentum equation; \( \mu \) is friction coefficient; \( \tau_t \) is the contact tangential traction; \( \lambda_t \) is the relative tangential velocity.

In equation (18), condition a) is known as the stick condition when the tangential traction at point is less than the critical value, no relative tangential motion is permitted. Condition b) corresponds to sliding and the second part of that equation expresses the fact that the tangential friction must be in the direction opposite to the relative tangential velocity.

2.4 The Total ALE Contact Weak Form

A common approach to imposing the contact constrains is by means of Lagrange multipliers. Let the Lagrange multipliers trail function be \( \lambda(\xi, t) \), and the corresponding test functions be \( \delta \lambda(\xi, t) \). These function reside in the following spaces:

\[
\lambda(\xi, t) \in j^*, j^* = \{ \lambda(\xi, t) | \lambda \in C^{-1} \} \text{ on } \Gamma^c
\]

\[
\lambda(\xi, t) = \bar{u}_c(\xi, t) \text{ on } \Gamma_v
\]
\[ \delta \lambda (\xi, t) \in \mathcal{D} \left\{ \delta \lambda (\xi, t) \right\} \quad (20) \]

Since the weak form is equivalent to the momentum equation, the traction boundary conditions, the interior traction continuity conditions and the contact interface conditions, the total ALE contact weak form is described as:

\[
\int_{\Gamma} \delta v \cdot \rho \ddot{\omega} d\Omega + \int_{\Gamma} \delta \sigma : \sigma d\Omega - \int_{\Gamma} \delta v \cdot \rho \dot{\omega} d\Omega - \int_{\Gamma} \delta v \cdot \tau_{\text{f}} d\Gamma + \int_{\gamma_{\text{c}}} \delta (\gamma x_{\text{c}} + \gamma y_{\text{c}} \cdot \tau_{\gamma}) d\gamma + \int_{\gamma_{\text{c}}} \delta (\lambda y_{\text{c}}) d\gamma + \int_{\gamma_{\text{c}}} \delta (\gamma_{\text{c}} \cdot \lambda_{\gamma}) d\gamma \geq 0 \quad (21)\]

Where \( \gamma_{\text{c}} \) is the rate of interpenetration of the two body, \( \int_{\gamma_{\text{c}}} (\delta \gamma x_{\text{c}} + \delta \gamma y_{\text{c}} \cdot \tau_{\gamma}) d\gamma \) is the virtual power of the contact traction, \( \int_{\gamma_{\text{c}}} \delta (\lambda y_{\text{c}}) d\gamma \) is impenetrability constraint by means of Lagrange multipliers, and \( \int_{\gamma_{\text{c}}} \delta (\gamma_{\text{c}} \cdot \lambda_{\gamma}) d\gamma \) is the constraint of no tangential slip imposed by Lagrange multipliers.

### 3 WEAR MODEL

Historically, the wear depth on the profile has been related to the energy dissipation (work done by friction forces or the product of creep forces and creepages) in the contact surface [1-5]. In this study, Archard’s wear model will be used since it is more commonly used in the tribology community for modelling sliding wear. For instance, Archard’s wear model was applied to predict railway wheel profile evolution due to wear in reference [6].

According to Archard’s wear model [13, 16] the volume of material worn away is proportional to the sliding distance and the normal force, and is inversely proportional to the hardness of the worn material according to Eq. (22)

\[
V_v = k \frac{3N}{3H} \quad (22) \]

Where \( V_v \) is the volume of material removed by wear from the surface (m³), the sliding distance (m), \( N \) the normal force (N), \( H \) the hardness of the worn material (Hv) and \( k \) the wear coefficient.

Since the contact relation of wheel/rail is computed in ABAQUS software, the interface normal pressure and the interface slip rate in nodes are obtained from the output of ABAQUS analysis. According to Archard’s wear equation there will be no wear in the adhesive zone of the contact surface since the slip rate is zero for all elements inside that zone (Figure 2).

In Eq. (22),

\[
V_v = \int_{\gamma_{\text{c}}} \dot{\gamma}_{\text{eq}} \ddot{A}_{\text{eq}} dt \quad (23) \]

\[
s = \int_{\gamma_{\text{c}}} \gamma_{\text{eq}} \gamma_{\text{eq}} dt \quad (24) \]

\[
N = p d \quad (25) \]

Where \( \dot{h} \) is a nodal ablation velocity (m/s), the nodal contact area (m²), \( \gamma_{\text{eq}} \) is the equivalent slip velocity (m/s), and \( p \) the interface normal pressure (N/m²).

Then, the nodal ablation velocity in an element can be calculated according to Eq. (22)-(25)

\[
\dot{h} = k \frac{\gamma_{\text{eq}} p}{3H} \quad (26) \]

Finally, the time integration enables the following expression for the wear depth in an element:

\[
\Delta h = \int_{\gamma_{\text{c}}} \frac{k \gamma_{\text{eq}} p}{3H} dt \quad (27) \]

Since the FEM models of wheel and rail are created by spinning the 2D mesh of axial symmetry, the wear depths of the nodes in the boundary of the base planes are computed by averaging the wear depths of the surface nodes about axial symmetry. This work is done by dealing with the results of three-dimensional nonlinear transient dynamic finite element model.

Wear coefficient of hardened tool steel on hardened tool steel is \( k = 10^{-4} \) in article [17], which is used in this wear simulation. H, hardness distribution is from labora-
tery experiments (Table 1). The hardness of nodes is obtained by linear interpolation and it is changed by linear method following rolling cycles between the double wear test data.

4 ADAPTIVE MESH ADJUSTMENT TECHNIQUE

The updating strategy of the wheel and rail profiles is a key point of the profile wear prediction model. Its purpose is to determine the mileage after which wheel and rail profiles should be updated and new mesh should be adjusted after the revolutions, and new calculation of wheel/rail contact should be performed. Different wheel profile updating strategies were compared [13] and it was found that the most efficient one is based on the maximum wear depth, i.e. the profile is updated when a given threshold of the maximum value of cumulative wear depth is reached. A sensitivity analysis showed that a threshold of 0.1 mm is low enough to guarantee a good accuracy and at the same time does not lead to excessive computational effort. The worn wheel profile is then smoothed in order to avoid short wavelength concavities along the wheel profiles. Therefore, the smoothing process is necessary and allows for improving approximating the continuous wear process with a discrete sequence of profile updates. A cubic spline interpolation algorithm of smoothing is applied on wear distribution before starting a new iteration of the wear prediction.

Mesh adjustment technique is the key in the wear simulation based on FEM, but mesh adjusting of common 3D mesh is very difficult. Since the FEM model of wheel is axial symmetry, mesh adjusting of the 3D mesh is transform into the problem of 2D mesh which is the mesh in the base plane of the FEM model. After adjusting 2D mesh, the 3D mesh is adjusted by spinning the 2D mesh about symmetry axis. The adaptive mesh adjustment technique about 2D is introduced in the following text.

After the wear depth of a node in the boundary of the base plane has been obtained, the site of the node is adjusted along the wear direction at the node. It is obvious that wear direction at a node and normal vector of the node are identical, so unit normal vectors should be computed for all the nodes on the boundary of the base plane. For example, the unit normal vector \( n \) is computed by averaging the unit normal vectors \( N_{12} \) and \( N_{23} \) of segments 1-2 and 2-3, which is showed in Figure 4. The adjusting vector is obtained by

\[
v_i = \Delta h_n n
\]  

(28)

After the sites of boundary nodes of 2D elements in the base plane have been adjusted by \( v_i \), elements close to boundary will became narrow or negative shape if the sites of interior nodes are not adjusted. It is necessary that sites of interior nodes are adjusted. Laplacian smoothing is often used in mesh modification and it may improve quality of 2D meshes for FE computation. In the paper, Laplacian smoothing with area weights is used, which keep boundary nodes fixed and adjust the sites of interior nodes. The adjusting vector of interior \( i \) node is obtained by

\[
v_i = \sum_{j=1}^{n} w_j (x_j - x_i)
\]  

(29)

Where \( x_i \) is the site coordinate of \( i \) node, \( x_j \) and \( w_j \) are respectively the site coordinate of the centers of gravity, and area rate of \( j \) element around \( i \) node. An example of Laplacian smoothing with area weights is showed in Figure 3.

After the updated 2D meshes in the base plane are done, the 3D meshes of rail head and wheel tread are created by spinning the 2D meshes about symmetry axis. By above adaptive mesh adjustment technique, 3D Meshes of wheel tread are obtained, which is showed in Figure 5.

![Figure 3 wear directions at nodes](image3)

![Figure 4 Laplacian smoothing with area weights](image4)

![Figure 5 3D Meshes of wheel tread](image5)
5 THE MODEL OF WHEEL/RAIL ROLLING CONTACT WEAR

Da Qin railway is main heavy line rail network in China, and the present wear prediction tool is applied to a vehicle operating the rail network. This network is mainly operated by C80 vehicle which is shown in Figure 6. The vehicle is equipped with K5 bogey which is shown in Figure 7. The technical parameters of the vehicle include: axle load of 25t; the LM wheel tread; the tape-circle radius of 460 mm; flange back separation of 1353 mm. The curve radius discretization of the rail network is a key factor for the shape of the wear distribution. The total track length of the curve radius of 800m is 44.1% of total track length sum. It is found that there is severity side wear on the track of the curve radius of 800m, so the curve track is studied in this paper. The velocity of the vehicle is 80km/h on the curve track. The technical parameters of the track include; CHN75 rail profile; track gauge of 1435mm above a plane that rest across the two rails; Rail cant of 1/40; track superelevation of 60mm. The wear profiles of the track in k403 are investigated at intervals. The initial designed rail profile, the profile passed 20 million gross tons and the profile passed 420 million gross tons have been measured. The predicted wear profiles of the track in k403 are shown in Figure 8. When the vehicle passes the small curve track, its wheel wear is measured rail profiles are shown in Figure 8. When the vehicle passes the small curve track, its wheel wear is measured. For the simulation results are prevalent in intervals. The initial designed rail profile, the profile passed 20 million gross tons and the profile passed 420 million gross tons have been measured. The predicted wear profiles of the track in k403 are investigated at intervals. The initial designed rail profile, the profile passed 20 million gross tons and the profile passed 420 million gross tons have been measured. The predicted wear profiles of the track in k403 are shown in Figure 8. When the vehicle passes the small curve track, its wheel wear is predicted. For the simulation results are prevalent in fact, the CHN75-wear1 form Figure 8. is used as the rail profile in the wear simulation.

![Figure 6 C80 tram](image)

![Figure 7 K5 bogy](image)

The dynamic model which couples C80 vehicle model and track model is developed in the commercial MBS (multi-body system) software Adams Rail and consisted of a full dynamic rigid multi-body model with a car body, two K5 bogies and track model. In Adams Rail the contact between wheel and rail is calculated by Kalker's simplified theory (FASTSIM). Depending on the track irregularities, the dynamic responses of coupling vehicle model and track model are solved by Adams Rail. The averages of these responses are loaded on the model of FEM based on ALE. The numerical method of steady state rolling contact of wheel/rail is used to analyze the shape of contact pitch, the contact pressure, shear friction stress, relative slip velocity and stick-slip state. In FEM model the forward velocity of wheelsets is 80km/h, and angular rolling velocity around a rigid axis of wheel is 3031.4°/s.

A classical bilinear elastic-plastic constitutive material model is used in numerical analysis, and Poisson's ratio for wheel and rail is 0.28. The material properties of wheel and rail are shown in Figure 9. In the wheel/rail contact property, static and kinetic friction and stick-slip motion are described by friction model which friction coefficient is related with the interface slip rate. According to the article [18] in which the brake test was done in Lyons when wheels of vehicle were slipping on rails, for dry wheel/rail contact surface, relation between Friction coefficient and slip velocity is showed in Figure 10, which is used in this simulation.

The wheel/rail rolling globe coordinate system is right handed Cartesian, with 'x' parallel to the tracks, in the direction of rolling, 'y' horizontally to the right and 'z' vertically down. The coordinate system is shown in Figure 11a. The right wheel and right track are selected to be analysis object and they are meshed by three-dimensional finite element meshes. The minimum size of the finite element meshes is 1mm nearby the wheel/rail contact point and the model has 267649 elements and 314077 nodes, as shown in Figure 11b. The parts close
6 NUMERICAL RESULTS

The wheel/rail rolling contact finite element model based on ALE finite element method is developed and applied to the wheel/rail steady state rolling contact analysis. The Von Mises stresses for static state contact of the wheel/rail are shown in Figure 12, and those for steady state contact of the wheel/rail are shown in Figure 13. On no tangential traction the maximum of the Von Mises stresses is 639 MPa, which appears inside the bodies at a location 3.2 mm below the surface of wheel. Those are seen in Figure 12. On the steady state the maximum of the Von Mises stress is 806 MPa, which appears on the surface of wheel. Those are seen in Figure 13. The tangential friction forces have an effect on the maximum of the Von Mises stresses and its location. The distribution of stresses has a large effect on the fatigue damage of the wheel/rail surfaces.

The contact traction and relative slip velocity distribution...
on the contact patch of the wheel is shown in Figure 14. The nonlinear FEM takes into account all geometrically nonlinear and material nonlinear effects, so the shape of the contact patch is large different with the result of Hertz theory. From Figure 14b the “main contact center” is seen, and friction vector distribution is rotating field along counter-clockwise, the rotating center of which is the main contact center. The relative slip velocities are very small near the main contact center, and the domain is close to adhesive state. The relative slip velocity increases with the distance from the main contact center increasing. And its vector rotating distribution is shown in Figure 14c.

The contact pressure and friction distributions are shown Figure 14d, Figure 14e and Figure 14f. The contact pressure distribution takes on the wave shape, which is relative to the angle of attack when the vehicle passes the curve track. The vector distributions of friction and relative slip velocity take on the self-spin character, which is take a effect on the wheel/rail rolling contact wear. The FEM model based on ALE takes into account the rolling velocity of the wheel and elastic-plastic deformation of wheel and rail geometry, so the results are very close to reality.

Based on the wheel/rail steady state rolling contact analysis, a wheel profile wear prediction methodology is developed and applied to the wheel wear of flange and gauge when the heavy haul train passes in curved track with the small radius. For a time period of about 2 years extensive consecutive wheel profile measurements on C80 vehicles are averaged according to different wear phases, and the three wear profile of the wheels are got, which are shown in Figure 15.

**Figure 14** Surface traction and relative slip velocity distribution

**Figure 15** Consecutive wheel profile measurements on C80 vehicles

7 **CONCLUSIONS**

1) The wheel/rail rolling contact finite element model based on ALE finite element method is developed and applied to the wheel/rail steady state rolling contact analysis. It is not limited by the half-space assumption and linear-elastic assumption. The method makes the prob-
lems of wheel/rail contact really become into a dynamic problem.

(2) The presented numerical results for the rolling contact situation underline that the contact friction and the relative slip velocity take on the self-spin character, which have a large effect on the wear of the wheel/rail rolling contact.

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