Influence of spin creepage on contact patch

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Abstract. The wheel-rail contact patch analysis methods are related to the evolution of modern computers. Contemporary computation systems allow the approach of tri-dimensional methods which take into account various contact parameters. With this background in mind, the wheel-rail contact problem is a study under development. The paper proposes a tri-dimensional approach, considering the spin creepage parameters. The spin effect is due to the increase of the wear around the wheel flange and the gage face. At the same time, the spin is influenced by the wheel rolling operating conditions. The paper gives an insight of wear mechanisms analysis, plastic deformation and fatigue phenomenon of the contact patch constituent material.

1. Introduction
An important application area for the Contact Kalker software is the detailed study of the wheel-rail interaction. The software features advanced characteristics and in each element of the contact area the dry friction Coulomb law may be applied. Contact Kalker is using an iterative algorithm which gives accurate results in the railway transportation domain [1-4]. By means of the Contact algorithm one can obtain elastic deformations, stick and slip areas, creepage values, contact forces and pressures as described in figure 1.
The preparing stages for the input data of the Contact Kalker software are the following:
1 - obtaining the coordinates of the first contact point using the method of the minimal level difference
2 - deformation calculus considering an isolated contact point
3 - testing the possibility of a new contact area
4 - computing deformations in the contact zones
5 - examination of the vertical components values; if the differences are not within assumed tolerances, step 4 is re-iterated using vertical differences computed at step 5, until the imposed accuracy is obtained.

In the following, a mounted axle with wear profiles rolling (nominal radius 460 mm) on rails with UIC 60 profiles is analyzed using Matlab and Contact Kalker software.

2. Aspects of wheel-rail contact
At the end of the XIXth century, Heinrich Hertz [5] accomplished sustained research regarding mechanic contact between two bodies and published his first paper in 1896.

His studies were based on optic interference produced by cylindrical lenses. He mentioned that both lenses were deformed under contact stress and that the contact area is elliptical. When the stress was removed, the contact patch disappeared and the lenses regained their initial shape. Starting from this observations, Hertz stated the theory presently used on a large scale to develop contact models due to its simplicity and the satisfactory accuracy. In spite of its advantages, the theory has certain limitations:
- it may be applied only to smooth surfaces with constant curvature in the contact area
- it allows to describe only normal forces on the contact surface

Since 1967, Kalker and subsequently other authors developed different theories suited especially to the wheel-rail contact study, [6]. These theories consider that the bodies in contact have identical elastic characteristics and that they observe Kalker theory regarding the normal distribution of the force. The main contribution of Kalker’s theory is that he considered the influence of the spin movement.

Kalker defines the creep coefficients as follows, [7,8]:

\[
\begin{align*}
\kappa_x &= \frac{a \cdot b \cdot G \cdot C_{11} \cdot v_x}{\mu \cdot N}, \\
\kappa_y &= \frac{a \cdot b \cdot G \cdot C_{22} \cdot v_y}{\mu \cdot N}, \\
\kappa_\phi &= \frac{3a \cdot b \cdot G \cdot C_{23} \cdot \phi}{\mu \cdot N}.
\end{align*}
\]

(1)

where: a, b – contact ellipsis dimensions, G – the modulus of transverse elasticity, C_{11,22,23} – Kalker’s coefficients, v_{x,y}, \phi - longitudinal transverse and spin creepage.

Longitudinal transverse and spin creepage are determined from equation (2):

\[
\begin{align*}
v_x &= \frac{w_x}{V}, \\
v_y &= \frac{w_y}{V}, \\
\phi &= \frac{r \cdot \omega_\phi}{V},
\end{align*}
\]

(2)

where: w_{x,y} – longitudinal and transverse relative velocities, r – wheel rolling radius, V – wheel rolling velocity, \omega_\phi – spin angular velocity.

The creep velocities are given by equations. (3, 4):

\[
0 = w_{x,y} = V \cdot (v_x - \phi \cdot y, \quad v_y + \phi \cdot x) - V \frac{\partial u(x,y)}{\partial x} \quad \text{- inside the contact ellipsis}
\]

(3)

\[
0 = p(x,y) \quad \text{- outside contact ellipsis}
\]

(4)
As stated in [9], the spin creepage is obtained decomposing angular velocity vector into normal and tangential components regarding the wheel-rail contact plane.

The spin effect is considerable when the contact takes place on the wheel flange. Besides that, spin creepage is significantly increased when the wheel is acted by a traction torque. This effect also increases wear of powered axles wheels.

Figure 2. Longitudinal transverse and spin creepage [3].

The analyses of the axle kinematics [10] is accomplished considering a symmetric loaded axle, rolling with velocity $V$ along a railway section either in traction or braking. The result is obtained using the torques equation (5):

$$M_{t-f} - r_e T_{ex} - r_i T_{ix} = 0$$  \hspace{1cm} (5)$$

where: $M_{t-f}$ – traction/breaking torque, $r_e, r_i$ – rolling radiuses in the contact point, $T_{ex}, T_{ix}$ – tangential forces in the contact point.

The torque exerted by the axle is positive for traction and negative for breaking. It is known that $M_{t-f}$ may not surpass $2r\mu N$ i.e., the value at the adherence limit.

In this situation the longitudinal creepage velocities are given by equation (6):

$$w_{ex} = v \left( 1 + e / R \right) - K \cdot r_e / r,$$

$$w_{ix} = v \left( 1 - e / R \right) - K \cdot r_i / r,$$  \hspace{1cm} (6)$$

where $v$ – axle rolling velocity, $e$ – half distance between rolling radiuses, $r$ – wheel nominal radius, $r_e, r_i$ – rolling radiuses in the contact points, $K$ – traction/breaking coefficient.

Tangential forces are given by the equation (7):

$$T_{ex} = -\kappa_{ex} \cdot \left( 1 + e / R - K \cdot r_e / r \right) Q_e$$

$$T_{ix} = -\kappa_{ix} \cdot \left( 1 - e / R - Kr_i / r \right) Q_t$$  \hspace{1cm} (7)$$
The torques equation (5) becomes:

\[ M + r_e \cdot \kappa_{ex} \cdot \left(1 + e / R - K \cdot p_e / r\right) \cdot Q_e - \ldots - \eta_i \cdot \kappa_{ix} \cdot \left(1 - e / R - K \cdot p_i / r\right) \cdot Q_i = 0 \]  

(8)

And hence:

\[ K = \frac{M + r_e \cdot \kappa_{ex} \cdot Q_e + \eta_i \cdot \kappa_{ix} \cdot Q_i + \left(\frac{e}{R}\right) \left(r_e \cdot \kappa_{ex} \cdot Q_e - \eta_i \cdot \kappa_{ix} \cdot Q_i\right)}{\left(1 / r\right) \left(r_e \cdot \kappa_{ex} \cdot Q_e + \eta_i \cdot \kappa_{ix} \cdot Q_i\right)} \]  

(9)

The following notations are used:

\[ r_e \cdot \kappa_{ex} \cdot Q_e = r_e \cdot C_{11} \cdot a_e \cdot b_e \cdot G, \quad \eta_i \cdot \kappa_{ix} \cdot Q_i = \eta_i \cdot C_{11} \cdot a_i \cdot b_i \cdot G \]  

(10)

\[ k_1 = r_e \cdot \kappa_{ex} \cdot Q_e + \eta_i \cdot \kappa_{ix} \cdot Q_i, \quad k_2 = r_e \cdot \kappa_{ex} \cdot Q_e - \eta_i \cdot \kappa_{ix} \cdot Q_i \]

where: \( \kappa_c, \kappa_a \) - longitudinal creep coefficients, \( Q_e,i \) – both wheels are equally loaded with load \( N \).

Considering the above notations the traction/braking coefficient becomes:

\[ K = \frac{M_{f-f} + k_1 + \left(\frac{e}{R}\right) \cdot k_2}{\left(1 / 2 \cdot r\right) \left(r_e \cdot \left(k_1 + k_2\right) + \eta_i \cdot \left(k_1 - k_2\right)\right)} \]  

(11)

The values of the traction/braking coefficient may be as follows:

- \( K = 1 \) – free axle
- \( 0 \leq K < 1 \) – braking axle
- \( 1 \leq K < \infty \) – powered axle

A null traction/braking coefficient is for a stocked wheel value tends to infinity when rolling at adherence limit.

3. Numerical simulations

In the present section the behaviour of an powered axle is analyzed. The axle wheels profile is S1002 (nominal rolling radius 460 mm) and the rail profile is UIC 60. The axle forward velocity is 160 km/h. The wheel normal load is between 15-25 kN.

After completing preparing stages, bi-contact is obtained. The contact points are located on the rolling surface and the wheel flange.

The distribution of the normal load on the wheel is based on equal elasticities. The distribution of vertical components of the normal load in the contact areas is given by the vertical components of the penetrations in these areas and the resultant load is equal with the wheel load.

Taking into account the influence of the spin upon wheel flange and gage face a 3D analysis of the frictional power density is presented in the following.

As the frictional power density is proportionate to tangential stress, forward velocity and creep in the contact area [1], in figures 3, 4 the values in the two contact points are given for different wheel loads, considering that spin coefficient is 0.000923 rad/mm.

The increase of the frictional power density when wheel load increases is considerable. As the slip area is larger on the rolling surface, the values are larger compared to those on the wheel flange.
It may be noticed that in figures 5-6 neglecting the spin effect, i.e., simulating a braking axle, significantly diminishes the values of the frictional power both for 25kN and 15kN wheel load, compared with the previous situations. This fact is explained by the axle tendency to slip forward.

Concurrently, on the wheel flange the shape of the stick area modifies in a similar way with the one predicted by the stripes theory. The increase in the spin rate makes the computation very difficult as in the case of the 3D theory or in the case of the simplified theory when taking spin into account makes impossible to obtain a solution. The stick area is very small compared to the slip one (bordered by the sharp edges on the plots).
In figures 7-8 the frictional power density is depicted for a powered axle with 25kN wheel load. Forward running velocities are 200 km/h and 30km/h, respectively. It may be seen that increased forward speed implies and increased frictional power loss in both contact points.
4. Conclusion
The paper presents a 3D analysis of wheel rail contact area considering the spin effect. The presence of the spin increases wear in wear of the wheel flange and the gage face areas. Moreover, it may be noticed that the spin effect is influenced by the axle torque when powered or when braking and further, the running safety is influenced by the axle torque.

The present study opens the perspective of the wear advance mechanism, plastic deformations and material fatigue in the wheel-rail contact area.

5. References
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