An analysis of vehicle - railway exchanger interaction

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Abstract. The fast variation of the section and contact characteristics of railway exchangers is a difficult matter for an analytical approach. At the same time, vehicle - railway exchanger system includes multiple contact patches, discontinuities in the contact points evolution and the necessity to compute stresses and deformations. As a consequence of the hars working conditions in the system there is a permanent concern to improve its performance. The analysis of the state of art in the research indicated the need of a complex theory for the study of the mechanical contact. In the paper, a tridimensional analysis of the vehicle - railway exchanger contact patch is carried out using “Contact” software, based on Kalker’s theory. The outcome of the study is the computation of tangential forces, creepage speeds and stresses distribution in the contact patches, in order to elucidate the vehicle evolution during this interaction and the estimation of the running safety.

1. Introduction

The contact with the track devices is analyzed in three hypostases: the contact with the tongue contact [1], the contact with the crossing heart and the contact with the rail which requires great efforts in support plates, which is why it cracks. The countermeasure plays an essential role in guiding latency over the track device.

Hiroyuki Sugiyama [2] presents a numerical analysis of the non-conformal wheel-decal contact. It parameterizes the contact surfaces and draws a parallel between the nodal analysis and the non-conforming contact, highlighting its efficiency from the perspective of an online analysis. The method is quite expensive.

According to the analyzed bibliography, many methods of analyzing the behavior of the vehicle were developed when passing over the track devices. Therefore, simulations performed using the GENSYS software have been experimentally validated by tests conducted on a convoy scale [3].

The finite element approach has been approached in more than 500 ways for analyzing the dynamic behavior of the vehicle when passing over the track devices [4]. The simulation of the contact using the SIMPACK package is described in the reference [5-6]. In reference [7-8], the authors used several Euler-Bernoulli beams in the path model to simulate the rails in the runoff zone over the track changer and in accordance with the path structure. Another model of analysis constitutes the model of real-time contact realized in NUCARS in which a bearing is used carriages on some lanes of passage.

Another method of analyzing the behavior of the vehicle when passing over the track devices is the VAMPIRE package [9]. To determine the contact forces with the help of this software there are four stages: - solid modeling of the conventional system of interaction is performed - the points of contact
between the axle and the running path are established, - the trajectory of the contact point is analyzed when crossing the crossing heart - dynamic behavior is analyzed (contact forces).

It is known that the track devices represent an unfavorable side of the tread and one of the most important factors limiting the speed when passing over them. For this reason it is necessary to analyze the dynamic characteristics at the contact level in order to improve the structure, to reduce the transverse and longitudinal shocks and mainly to increase the speed of movement in this area.

Zhai and Ren [10], Zhao [11] and Andersson and Dahlberg [12] studied the behavior of vertical direction of the interaction between the wheel and the track changer, analyzing the relationship between the speed of movement and the dynamic vertical forces. Wang [13] states in 1997 that a study of the dynamic characteristics of the wheel-changer system on a simplified model cannot be carried out precisely.

Drozdzie [14] analyzed the geometry of the tread deviations but does not include the dynamic effect of the vehicle. Baluch [15] analyzes the kinematic behavior of the vehicle when crossing the discontinuities of the track but does not include the dynamic effect, [16]

The pressure in the wheel-contact area is determined by the size of the contact force and the dimensions of the contact area. According to the specialized studies it is found that the pressure in the contact area does not significantly influence the speed of movement if the shape of the contact surfaces and the normal load on the wheel are taken into account, a fact stated by E. Kassa and G. Johansson. Thus, for the nominal profile of the wheel it is found that the pressure increases with the load and for a worn profile it reaches more maximums which contributes to reducing the wear [17].

Finally, when analyzing the specialized bibliography it is observed that most of the articles analyze the case of the vehicle's interaction with the tip of the crossing heart, so it is necessary the need for a three-dimensional analysis of the wheel contact with the tongue of the wheel in terms of traffic safety. This analysis is performed with the iterative software Contact - Kalker adapted for bicontact and the results are graphically visualized through the Matlab programming system.

2. The problem of wheel contact – tread
Since 1967 Kalker, and later other authors, have developed different theories specifically dedicated to wheel-rail contact analysis. These theories consider that the two bodies have identical elastic characteristics and respect Hertz's theory regarding the normal distribution of force. This allows the study of the contact to be divided into two independent problems: the problem related to the description of the normal force, the so-called normal problem and the problem related to the description of the tangential forces, the so-called tangential problem. According to Kalker's theory, the dependence of the tangential forces on the pseudo-slip in the contact area is given by the linear expressions:

\[
F_x = -a \cdot b \cdot G \cdot C_{11} \cdot v_x,
\]

\[
F_y = -a \cdot b \cdot G \left( C_{22} \cdot v_y + \sqrt{a \cdot b} \cdot C_{23} \cdot \phi \right)
\]

\[
M_{\phi} = -\left( \sqrt{a \cdot b} \cdot G \cdot C_{32} + (a \cdot b)^2 G \cdot C_{33} \cdot \phi \right)
\]

where: a, b - the semiaxes of the contact ellipse, \(v_x, v_y, \phi\) - the longitudinal, transverse and spin pseudo-slip at the contact points, \(C_{11}, C_{22}, C_{23}, C_{32}, C_{33}\) - Kalker's coefficients, which depend on the value of the coefficient Poisson being established based on the relationships below [18]:

\[
C_{11} = -\frac{\partial F_x}{\partial v_x} / (a \cdot b \cdot G) \quad C_{22} = -\frac{\partial F_y}{\partial v_y} / (a \cdot b \cdot G) \quad C_{23} = -\frac{\partial F_y}{\partial \phi} / \left( \sqrt{a \cdot b} \cdot G \right) \quad v_x = v_y = \phi = 0
\]
The values of the semiaxes of the contact ellipse are calculated with the relation, [19]:

\[ a = m \left( \frac{3}{2} \frac{1-\nu^2}{N \left( A + B \right)} \right)^{1/3} \]

\[ b = n \left( \frac{3}{2} \frac{1-\nu^2}{N \left( A + B \right)} \right)^{1/3} \]

(3)

where: \( m, n \) are sizes that depend \( \cos \beta = \frac{B - A}{B + A} \) on Hertz, \( N = \int_S \int p(x,y) dxdy \cdot \text{normal wheel load}. \)

Kalker also defines the pseudo-slip coefficients as follows:

\[ \kappa_x = \frac{a \cdot b \cdot G \cdot C_{11} \cdot v_x}{\mu \cdot N} \]

\[ \kappa_y = \frac{a \cdot b \cdot G \cdot C_{22} \cdot v_y}{\mu \cdot N} \]

\[ \kappa_\phi = \frac{2a \cdot b \cdot G \cdot C_{23} \cdot \phi}{\mu \cdot N} \]

(4)

According to [20], the modulus of transverse elasticity takes into account the nature of the material from which the rail and the wheel are made, so that it is given by the relation:

\[ G = \frac{2 \cdot G_r \cdot G_s}{G_r + G_s} \]

And Poisson's coefficient is dependent on the elastic characteristics of the material

\[ \nu = \frac{G \cdot \left( G_r \cdot \nu_x + G_s \cdot \nu_r \right)}{2 \cdot G_r \cdot G_s} \]

The longitudinal, transverse and spin pseudowords are established based on the relationships, [19]:

\[ v_x = \frac{w_x}{V}, \quad v_y = \frac{w_y}{V}, \quad \phi = \frac{r \cdot \omega_x}{V}, \]

(5)

where \( \omega_x \) – angular velocity of spin.
3. Numerical simulations
Within this section we analyze the behavior of an axle equipped with profile wheels S1002 (nominal rolling radius 460 mm) when passing over a simple track changer made from rail profile of type UIC 60. Axle speed over the track changer it is 100 km/h. The contact is analyzed from a three-dimensional point of view. Normal wheel load is considered to be 15-20 kN.

The profiles were drawn by the circle arcs method and the contact points were established by the minimum distance method. A gap of 6 mm was considered and two contact points were identified as follows: one on the lip of the wheel and one on the tread surface.

Starting from the idea that the tangential forces are transmitted by the adhesion between the two interacting bodies, which is dependent on the elastic deformations and the slips in the contact area.

At the same time, the pseudo-slip is a parameter that influences the transmission of the tangential forces in the wheel-rolling contact area, being defined by the ratio between the sliding speed and the axle advancement speed. The following shows the vectorial distribution of the tangential tensile force relative to the normal load on the wheel.

It can be observed that this increases with its vector field as the load increases, see figure 1-2.

![Figure 1. Vector distribution of the tangential tensile force for the normal load of 15kN](image-url)
At the contact point on the lip, according to the distribution of the tangential force field of traction, the wheel runs on the rail similar to the running situation without pivoting slips. In this case, the intensity of the traction field increases with the load, see figure 2. There are no major changes in the contact point on the tread surface.

![Figure 2. Vector distribution of the tangential tensile force for the normal load of 20 kN](image)

The distribution of the slip and adhesion zones is analyzed in the case of the bicontact that appeared at the interaction between the wheel and the hub tongue, see figures 3-4.

It is observed that at the point on the lip the distribution of the slip and adhesion zones is similar to the rolling in the conditions where the spin pseudo-slip is zero, is the wheel behaves like a puppet, see figures 3-4.

It is observed that the size of the normal load on the wheel significantly changes both the shape and the size of the grip area in the transverse direction, which is justified by an increase in the tangential force.
Figure 3. Distribution of the adhesion and slip zones for the normal load of 20 kN

Figure 4. Distribution of grip and slip zones for the normal load of 15 kN

Figures 5 - 6 show the distribution of the normal pressure in the wheel-axle contact area according to the load on the wheel. It is found that the pressure is higher when contacting the tongue tip, which is justified by the increased wear in this area.

The normal pressure in the contact area is directly influenced by the load on the wheel. It can be seen figures 5 - 7 that the wheel load significantly influences the maximum pressure at the wheel lip contact.
Figure 5. Normal pressure distribution for the normal load of 15 kN

Figure 6. First point of contact - Normal pressure distribution for the normal load of 20 kN
The justification of the necessity of the study carried out is taken into account that, for the study of the wheel-rail contact phenomena, essentially for the determination of the tangential forces, the knowledge of the sliding speeds and the distribution of the pressures in the contact area is defining for the behavior of the vehicle in the lane.

4. Conclusion
A method for the wheel-turnout interaction study is developed in the present work. The tridimensional Kalker theory is employed using the computer program CONTACT. The application is designed for frictional concentrated contact problems. The solution proposed is numerically efficient and it has been previously validated for the general case of the wheel-rail contact.

Multi-point contact is a common situation for the studied system. In the approach proposed, the bodies deformations and the contact areas can be determined from the contact loading. Further, the number and position of the contact points can be calculated either on-line or off-line. The scheme proposed also allows the computation of the loading and stress state starting from the local deformations.

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