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# THE CONTACT BETWEEN A WHEEL AND A RAILWAY TRACK AND ITS INFLUENCE ON TRANSPORTATION SAFETY

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*The wheel-railway track contact phenomena determine the dynamic performances required for a safe transportation system. The present paper will present an actual analysis of the pressures and the sliding speeds that occur in the contact area between the wheel and the railway track underlining at the same time its importance in assuring the safety of transportation. The interaction between an axle-tree equipped with S78 type wheels and a railway track model UIC60 in relation to the contact area will be analyzed.*

**Keywords:** wheel-railway track contact, interaction forces, contact ellipse, guiding safety.

## 1. Introduction

The geometry and the interacting forces located in the contact area between the wheel and the railway track assure the safety of vehicle movement. The analysis of the way in which this type of contact can influence the safety of the transportation starts with establishing the coordinates of the contact points through a mathematical description of the surfaces, using high level polynomials or level 2 curves (circular curves, parabola curves, etc) [1]. Studies regarding the contact between the wheel and the railway-track, more specifically for determining the wheel's contact profile, have been done by Boedeker, Ihn, Heumann, Borgeaud [2, 3, 4]. Thus, in order to determine the coordinates of the wheel's elusory profile, Borgeaud, [2], proved two properties which are further described. First, he used two measures, $\alpha$ and $\beta$, and showed that the contact point from the apparent profile is not depending on the y, r and $\tan\gamma$ coordinates (abscissa, rolling radius and the tangent of the flanking angle in front of the contact point) of the real profile. For a specific tilting angle of the axle-tree, compared with the horizontal level, the position of the point depends on the shape of the wheel's profile, instead. Second, the contact point may be determined using an equidistant line for the profile. If the distance equals the curvature radius, the point will be given by the coincidence with the center of the railway's arch profile [2]. The development of railway transportation has always had at its core a

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detailed knowledge of the contact phenomena between the wheel and the railway track. The analysis of the contact area between two elastically deforming bodies was done by Hertz for establishing the shape of the contact surface and of the distribution of contact forces. This theory is valid only in the following cases [5]:

- the surfaces which are in direct contact are smooth and unequal;
- friction exists between the surfaces found in direct contact;
- the contact surface is very small in comparison with the size of the bodies to which they belong;
- at the contact level there are only compression stresses;
- the proportionality limit for elastic deformation is not exceeded.

Within the wheel-railway system, with the exception of the wheel's tor edge and of railway's connection, the third condition is generally fulfilled. Regarding the fourth condition, the tangential force has a major influence over tracing the lines of the adhesion and sliding areas without affecting the shape of the contact ellipse, which remains exclusively dependent on the size of the normal force [1].

The first theory of rolling with friction was developed by Carter [6] and uses a bi-dimensional model where the wheel is depicted as a cylinder while the railway tracks as a plate. The bodies respect the hypothesis of the elastic half-space and only the longitudinal glide is taken into consideration.

Lévi [7], establishes a mathematical expression which expresses the dependence of the friction coefficient on pseudo-slide. Thus, the shape of the curve is hyperbolic (first degree) while for small pseudo-sliding it can be considered linear. Regrettably, the isotropy principle is not properly and correctly expressed.

Müller [8], after some systematic measurements taken at Minden, during the ORE C9 committee, shows that Lévi's law is not entirely correct: the shape of the curve of the friction coefficient is indeed hyperbolic but of first degree as he considered it; its shape depends on the burden.

Johnson and Vermeulen [9] have extended Carter's bi-dimensional theory for random halves of spaces. The contact surfaces located between moving bodies that are transmitting tangential forces have been divided into two unequal areas, a sliding area and a sticky (adhesion) area. The latter was assumed to be an ellipse which is tangential toward inside with the contact ellipse. Also Halling, Haines and Ollerton [10, 11] have come with mostly the same theory regarding the elliptic contact with longitudinal pseudo-sliding. The contact area has been divided into a series of strips which are parallel with the rolling direction and each strip was then studied by extending Carter's bi-dimensional theory.

In the end, taking into consideration the theories presented above and the fact that the contact area is elliptic and divided into two zones (adhesion / sliding), one of the most exact models of analysis is developed by Kalker [12,13] for calculating the stress in the contact area. He takes into consideration each dot from the contact area as crucial in determining the local sliding forces as well as the local deformations of the two bodies found in direct contact. After that, Frederich [14] made some tests for determining the loads resulted from the interaction of the forces in the wheel of a vehicle which is rolling in the attack position and ends with confirming Kalker's theory. On the other hand, several other studies were made for averting Kalker's theory.

In the present paper which focuses on the interaction between the wheel and the railway track we use Kalker's tri-dimensional theory. The pressures, the sliding speeds within the contact area between the wheel and the railway track as well as the elements which influence the safety of guiding are analyzed.

This analysis is achieved using the CONTACT software, which is currently considered an advanced simulating program, applied to the contact point between the wheel and the railway track where within each element of the contact ellipse the law of dry friction is applied. CONTACT assimilates the contact area through half-spaces and divides itthrough the method of the finite element.

The present paper will present, in the second part, the issues encountered by the contact between the wheel and the railway track while in the third part, an example of interaction between an axle equipped with wheels that have the S78 profile and a railway track with UIC 60 1:20 profile ia presented.

## 2. The issue of the rolling contact.

The problem of the tangential contact between the wheel and the railway track consists in determining the distribution of the surface where the traction is exerted, specifically the contact pressure and the relative sliding speed in relation to the dimensions of the contact area [12, 15, 16].

If Hertz's hypotheses are fulfilled, then the distribution of pressure in the contact is done following an elliptic paraboloid described by the following equation [1, 5, 17]:

$$p(x, y) = p_o\sqrt{1-\left(\frac{x}{a}\right)^2-\left(\frac{y}{b}\right)^2} \tag{1}$$

where $p_o$ represents the maximum pressure located in the very center of the contact ellipsis while $a$ and $b$ represent the semi-axes of the contact ellipsis. The normal load, $N$, applied to the wheel depends of the normal pressure as shown in the following relation:

$$N = \int\int pdxdy \tag{2}$$

If "$p$" from the two equations above is replaced then the integration is established and the following result is obtained:

$$N = \frac{2}{3} p_o \pi a b \tag{3}$$

Thus, relation (2) becomes:

$$p(x, y) = \frac{3}{2 \cdot \pi \cdot a \cdot b} N \sqrt{1 - \left(\frac{x}{a}\right)^2 - \left(\frac{y}{b}\right)^2} \tag{4}$$

The semi-axes listed as *a* and *b* as well as their orientation along the railway or transversally on it are determined through the following relationship:

$$a = m\left(3\pi N \frac{k_1 + k_2}{4(A+B)}\right)^{1/3}, \quad b = n\left(3\pi N \frac{k_1 + k_2}{4(A+B)}\right)^{1/3} \tag{5}$$

where *m* and *n* – are coefficients established by Hertz in relationship to $\cos\tau = (A - B)/(A + B)$, $A = 1/r$, $B = 1/\rho_r + 1/\rho_s$, *r* being the rolling radius of the wheel, $\rho_r$, $\rho_s$ being the curvature radius of the wheel/railway at the contact point and $k_{1,2}$ being the constant which depends of the properties of the material out of which the two bodies are being made $k_{1,2} = \left(1 - \vartheta_{1,2}^2\right)/\pi E_{1,2}$, $\upsilon_{1,2}$ – Poisson's coefficient and $E_{1,2}$ – the longitudinal elasticity modulus.

The analysis of the axles' kinetic regime is done by considering an axle symmetrically loaded that is rolling with a given *v*-speed along a sector of railway developing a motor or a friction moment, [17, 20] . The result of this equation is obtained in conditions of static and dynamic equilibrium related to the system in which the axle is found.

Thus the equation for fig.1 can be written like this:

$$M_{t-f} - r_e T_{ex} - r_i T_{ix} = 0 \tag{6}$$

where: $M_{t\text{-}f}$ is the traction and braking moment, $r_e$ and $r_i$ are the wheel's rolling radii at the contact point and $T_{ex}$ and $T_{ix}$ are the tangential forces developed at the contact point.

The influence of the torque developed by the axle is positive for the motion and negative for braking. It is acknowledged the fact that $M_{t\text{-}f}$ cannot have a greater value than $2r\mu\ N$, meaning the value corresponding to the limit of adherence. Through this situation the sliding speeds along the longitudinal axis are given by the following relations, [18] :

$$\begin{aligned} w_{ex} &= v\left(1 + e/R\right) - K r_e / r \\ w_{ix} &= v\left(1 - e/R\right) - K r_i / r \end{aligned} \tag{7}$$

where *v* is the rolling speed of the axle, *e* is the semi distance between the nominal rolling circles, *r* is the wheel's nominal radius, $r_e$ and $r_i$ are the wheel's rolling radii at the contact level, *K* is the axle regime coefficient and *R* is the radius curve.

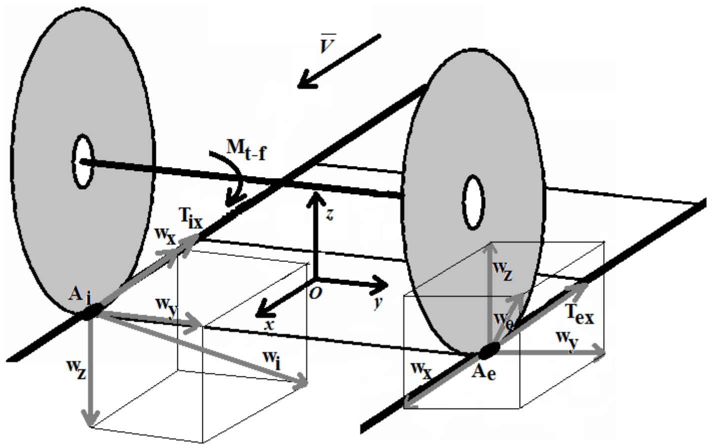


**Fig. 1.** The kinematics of the conventional wheel-railway track system.

The tangential forces are given by the following relationships:

$$\begin{aligned} T_{ex} &= -\kappa_{ex}\left(1 + e/R - Kr_e/r\right)Q_e \\ T_{ix} &= -\kappa_{ix}\left(1 - e/R - Kr_i/r\right)Q_i \end{aligned} \tag{8}$$

The equation of motion (6) becomes:

$$M + r_e\kappa_{ex}\left(1 + e/R - Kr_e/r\right)Q_e - r_i\kappa_{ix}\left(1 - e/R - Kr_i/r\right)Q_i = 0 \tag{9}$$

and

$$K = \frac{M + r_e\kappa_{ex}Q_e + r_i\kappa_{ix}Q_i + (e/R)\left(r_e\kappa_{ex}Q_e - r_i\kappa_{ix}Q_i\right)}{(1/r)\left(r_e^2\kappa_{ex}Q_e + r_i^2\kappa_{ix}Q_i\right)} \tag{10}$$

The following notations are thus made:

$$r_e\kappa_{ex}Q_e = r_eC_{11}a_eb_eG, \qquad r_i\kappa_{ix}Q_i = r_iC_{11}a_ib_iG$$

$$k_1 = r_e\kappa_{ex}Q_e + r_i\kappa_{ix}Q_i, \qquad k_2 = r_e\kappa_{ex}Q_e - r_i\kappa_{ix}Q_i$$

where $K$ is the system regime ratio which can range depending on the mode of movement of the axle: $K = 1$ – for a free axle; $0 \le K < 1$ - for an axle that is braking; $1 \le K < \infty$ for an axle that is driving.

The locked axle has a zero ratio value, corresponding to a large braking force (emergency braking). In the situation where $K$ tends to infinite the axle is running at the limit of adhesion.

$$K = \frac{M_{t-f} + k_1 + (e/R)k_2}{(1/2r)\left(r_e\left(k_1 + k_2\right) + r_i\left(k_1 - k_2\right)\right)} \tag{11}$$

## 3. Numerical application

Using the Matlab and the CONTACT algorithm in the following we analyze the kinematic behavior of a wheelset equipped with profile model S78 that is running on a railway track profile UIC 60 1:20. Considering a nominal radius of 460 mm for the wheel's travel, wheel load capacity of 10 kN, about 6 mm gap, running speed 35 m/s and traction/braking couple the wheelset is loaded at 2915 Nm. For a correct analysis of the contact area it is necessary to draw the tread profile of the rolling surfaces and to establish the points of contact between them, [19]. The profiles are drawn using circle arcs and the contact points are determined by the minimum distance method, see Fig. 2.

The analysis of the distribution of contact pressure determines the contact and sliding area, which in turn influences the wear of the contact surfaces, as wear occurs only in the sliding area.

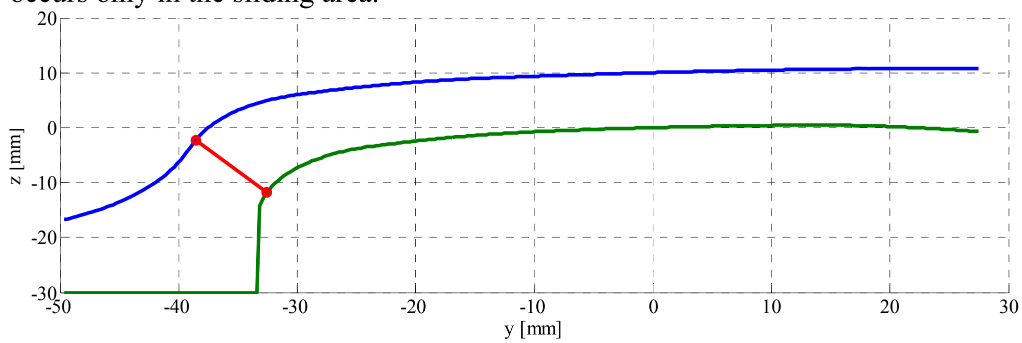


Fig. 2. The contact between a wheel having the S78 profile and a railway track model UIC60 for a 6 mm gap

Thus, in Fig. 3 the distribution of pressure in transversal direction corresponding to the bonding area can be seen. One can notice that the loading is at maximum at the extremity of the contact and then decreases toward inside, near the sliding area.

Along the railway the aspect is different, Fig. 4, the pressures are greater and their distribution towards the sliding area is smoother. This is caused by the orientation of the ellipse with large semi-axis along the rail. For the S78 profile analyzed the contact ellipse of major axis is oriented perpendicular to the rail.

The tangential contact problem can be solved in terms of sliding velocity based on which the friction forces and the tangential pressure distribution can be determined. Also, the sliding velocities and their distribution in the contact area of the wheel-rail system determine the wheels reliability.

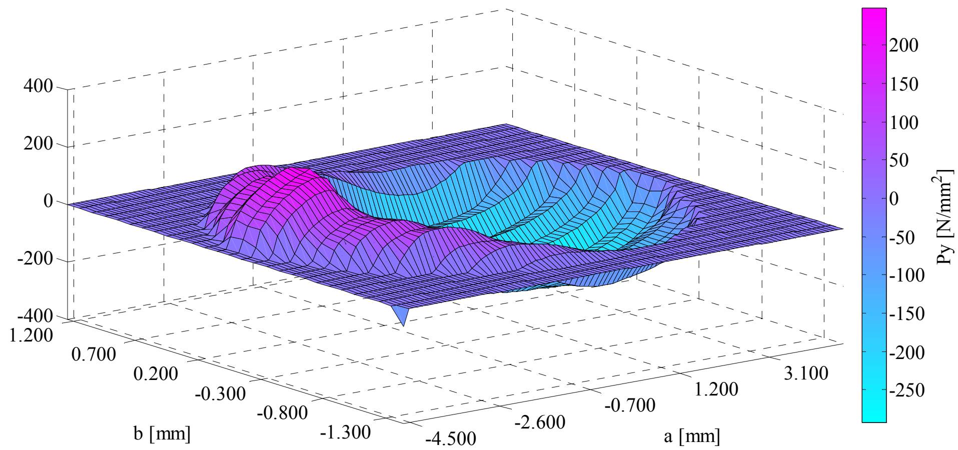


Fig. 3. Transversal pressure distribution

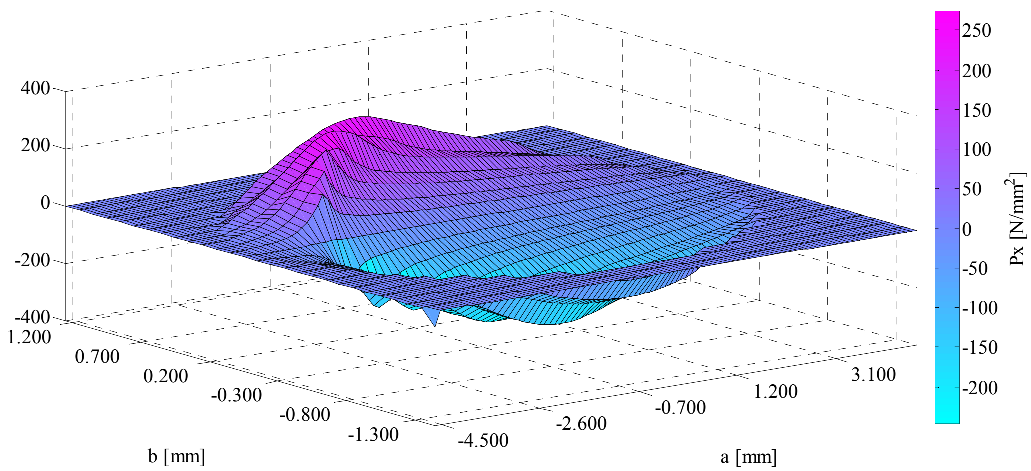


Fig. 4. Pressure distribution in longitudinal direction

According to the graphical representation, it can be seen that the sliding speed in the transverse direction is at a maximum towards the wheel flange and decreases towards the opposite direction, see Fig. 5 and 6.

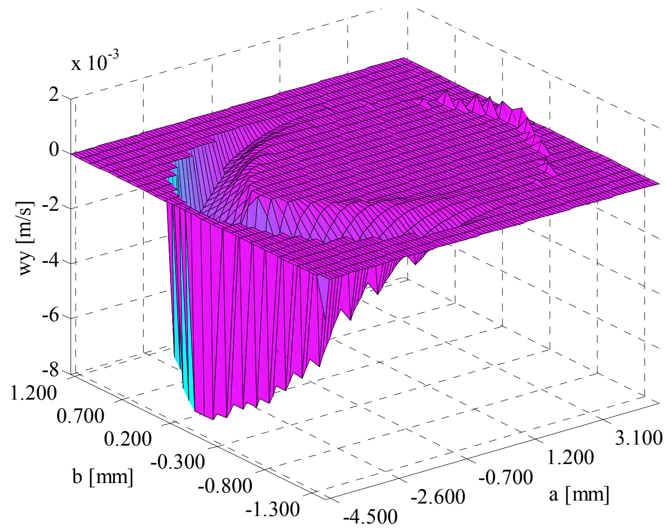

Fig. 5. Sliding speed distribution in the transverse direction

Also, given the directly proportional dependency of the sliding cross angle velocity, one can say that its reduction flattens the distribution from Fig. 5 almost to zero, the ideal and suitable radial position of the wheel set. Its spatial orientation cancels the transversal sliding movements which is a good thing as it encourages the increase of the longitudinal dynamic performances. If the axle develops a friction moment the longitudinal sliding speeds have different behavior than for the case when the axle is free (see Fig. 6).

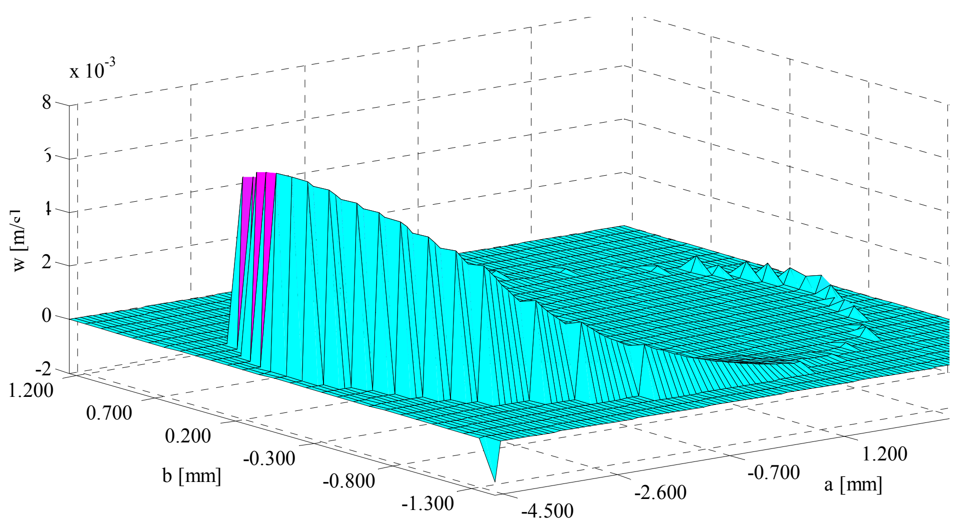

Fig. 6. Distribution of the relative sliding velocity at the point of contact

Thus, it can be seen that when the driving axle develops a traction moment the maximum values of the sliding forces developed on the longitudinal directions appear in the middle of the sliding area. This area corresponds to the longitudinal acting component of the friction force, value which added to the one from the inner part or the wheel gives the traction force (see Fig. 7). In this case, the wheel has a tendency to slide backwards.

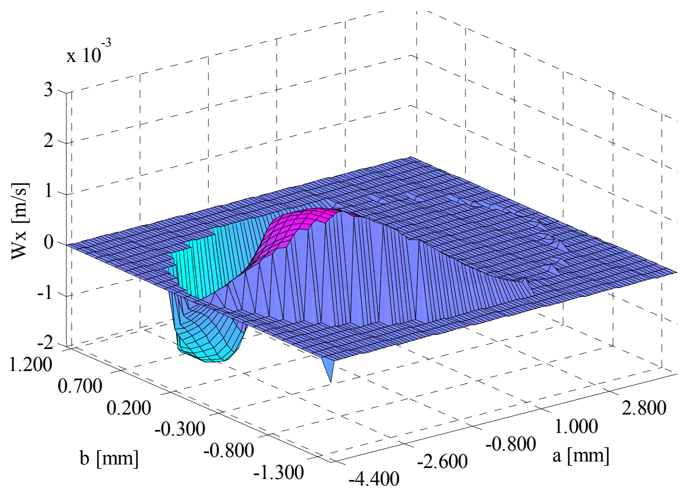


Fig. 7. Sliding speed in the longitudinal direction on the driving axle

If the axle moves under braking, the sliding area increases compared to the situation when the driving axle runs, while tending to slide forward.

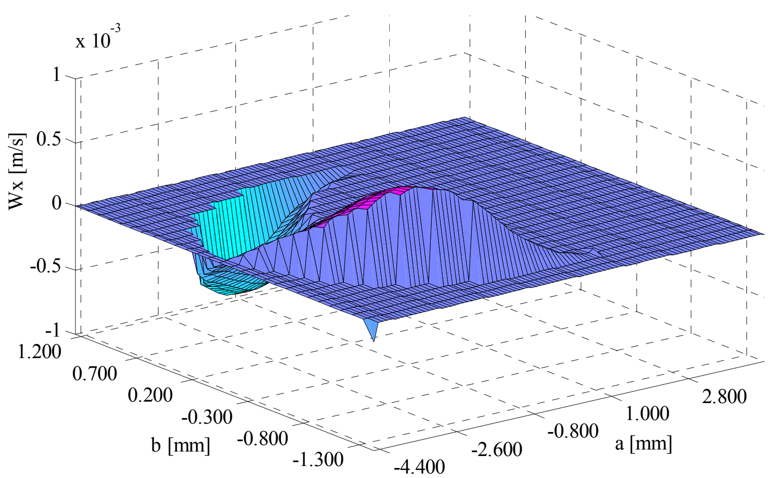


Fig.8. Sliding speed in the longitudinal direction for the axle while braking

In Fig. 8 it can be seen that the maximum speed of sliding in the longitudinal direction is located in the middle of the contact area namely on the major axis of the ellipse of contact. The value of the sliding speed is correlated with longitudinal component of the aggregate friction which gives the traction force.

Also, if the wheel develops a traction force, the shape of the adherence area tends to be similar to the one of traveling without longitudinal sliding movements. If, at a given moment, a braking movement occurs, the shape of the adherence area tends toward the situation in which a pure sliding movement takes place, according to the straps law.

The wheel loads an influence on the shape of the slip zone. If the wheel's guiding tends to be made by the edge of the wheel traction increases to the extremity contact ellipse with decreasing load, see Fig. 9. This occurs even if the point of the contact is located on the rolling surface.

The analysis of the interaction between the wheel and the railway track through the dimensions of the contact area and the distribution of the sliding speeds offers information about the position of the wheel, a crucial aspect in safety of guiding.

As shown (see Fig. 7, 8, 9), modification of the sliding area offers information about the rolling system, distribution of weight and also about the orientation of the forces within the contact area, which is a crucial aspect in the guiding technique.

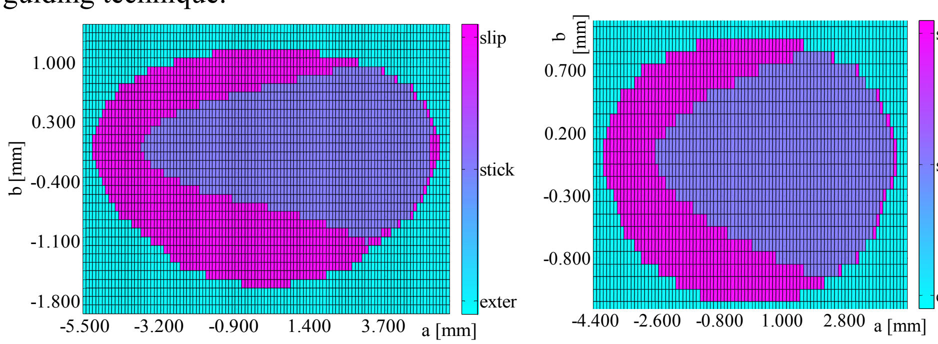


Fig. 9. The distribution of traction at the contact point

This three-dimensional approach of the contact issue simplifies the way of analyzing the tangential contact between the wheel and the railway track as well as its influence over the safety of transportation because it can be used to analyze the ellipse's sizes, orientation, and then the allure of the contact parameter. In essence, the parameters' allure of wheel-railway contact is evident in terms of road safety.

## 6. Conclusions

The present paper presents a tri-dimensional method of analysis of the contact point between the wheel and the railway track through the use of the CONTACT software. The numerical simulation software is applied to the contact between the wheel and the railway track with dry friction. For the wear and tear profile taken into consideration the results thus obtained are notable for evaluating the safety on guidance.

The evaluation of the safety on guidance was done from the point of view of the size and distribution of the sliding and adherence areas from within the contact point as well as of the distribution of the sliding speeds on the longitudinal and transversal directions.

The proposed numerical solution is efficient for the evaluation of the contact between the wheel with the S78 profile and the railway track with the UIC 60 1:20 profile, being validated only for S1002 and USC 60 1:40. Also, the solution brings forth a new parameter which takes into account the wheel's rolling movement. For the dedicated literature the proposed method offers an analytic evaluation of the pressures and sliding movement from within the contact area for tearing profiles used in our country.

It should also be noted that the decrease in wheel load reduces the ability of the guide shaft. Therefore, a reduced load was taken into account for the wheel, while the speed has been reduced to a maximum of 60 km / h to simulate the real behavior of the wheel passing over the curves at low speed and high torque maximum super elevation .

In justifying the necessity of the study that was done we must consider that the study of wheel-railway contact phenomena, essentially for determining the tangential forces and the sliding speeds as well as the distribution of pressures in the contact area, is crucial for knowing the behavior of the vehicle.

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