ASPECTS OF MULTIPOINT CONTACT AT SWITCH TRAVERSAL

Mihaela Cristina TUDORACHE[1], Ionuț NEAGOE[2], Răzvan Andrei OPREA[3]

The operating mode of the switch traversal is incomparably more severe than other sections, influencing, on one hand, the grip-pseudo-slip-wear characteristics, and on the other hand, the dynamic behavior of the vehicle, because the pseudo-slip forces are significantly influenced by the contact area and pressures from within this area. In this context the paper presents aspects of the multipoint wheel - rail-way switch contact using the Contact software. Considering the influence of sliding speeds on the contact phenomena at the intersection of the wheel with the running path, the paper presents an analysis of the kinematic behavior of the axle while passing over a switch traversal.

Keywords: switch, adhesion zone, slip zone, pseudo-sliding speeds.

1. Introduction

In most cases, the rail-road has deviations from the theoretical geometric dimensions, which must not affect traffic safety and the demands of the running gear, and the track must be within the permissible limits.

As a consequence, just as important for the safety of the guide is the wheel's contact with the switch traversal. Thus, the track switches, in this case the switch traversal, ensure discontinuities on the railway causing high dynamic loads. In operation, the track devices must ensure three basic requirements: safety against derailment, stability while on the move and passenger comfort.

The problem of the influence of wheel-rail contact phenomena on steering safety has been done using simplifying hypotheses until recently and the approximate methods led to illustrative results, on one hand. On the other hand, the analysis of contact phenomena produced over switch traversal crosses is a field of expertise almost untouched in our country, being limited only to a visual inspection through the use of template checks.

A focus on the interaction of the vehicle with the tip of the crossing heart is present in the scientific literature in this scope, thus imposing the need for a

[1] Prof., Dept. of Railway Rolling Stock, University POLITEHNICA of Bucharest, Romania, e-mail: cristi_tudorache2003@yahoo.com
[2] PhD student, Doctoral School of Transports, University Politehnica of Bucharest, Bucharest, Romania, e-mail: ionutneagoe88@yahoo.ro
[3] Prof., Dept. of Railway Rolling Stock, University POLITEHNICA of Bucharest, Romania, e-mail: razvan.oprea@gmail.com

three-dimensional analysis of the wheel's contact with the switch traversal in terms of traffic safety, [1-12]. This analysis is based on the applicability of the CONTACT software, [13, 14], this being considered an advanced simulation program applied to contact between the wheel and the rail, where in each element of the contact ellipse the law of dry friction of Coulomb is applied. The originality of the work consists in the application of the software in the case of the bi-contact made by the wheel when passing over the switch traversal.

2. Basics of Kalker's theory

In France, Levi, a railway engineer, publishes two papers on the rolling contact, based on experimental investigations. His purely empirical representation can be found in Rocard's monograph on "Stability of locomotive running". The relation established by Levi between the coefficient of friction τ and pseudo-slips is of parabolic form:

$$\frac{1}{\tau}=\frac{1}{\mu}+\frac{1}{\chi\cdot\nu} \tag{1}$$

where $\mu = F_{max} / N$ is the maximum coefficient of friction, and $\kappa = tg\ \varphi = (\tau / \nu)$, $\nu \to 0$ is called the pseudo-slip coefficient.

An approach at superior levels regarding the contact is represented by the Kalker's linear theory, which tries to establish the delimitation of the adhesion / sliding zones and the unitary efforts within them according to the tangential forces. For this, the longitudinal, transverse and spin pseudo-slips and the local deformations of the two bodies are calculated at each point in the contact area and the normal force resulting from the pressure distribution ellipsoid is considered. It is considered that in the sliding zone the tangential forces in the two directions satisfy Coulomb's law, and in the adhesion zone they are smaller. The size of the areas results from local stresses and deformations, [15-19]. The geometry of the two surfaces in the first stage in determining the sliding speeds in the study of the contact and, based on that, the contact points are determined according to the position of the axle on the path. The contact surface, S, is divided into a S_{stick} adhesion zone and a slip zone, S_{slip}, for which the conditions must be met:

$$\begin{aligned}(x,y)\in S_{stick} &\Rightarrow \begin{cases} s(x,y,t)=0 \\ \left|q(x,y,t)\right|<\mu\cdot p(x,y,t)\end{cases}\\ (x,y)\in S_{slip} &\Rightarrow \begin{cases} s(x,y,t)\neq 0 \\ \left|q(x,y,t)\right|=\mu\cdot p(x,y,t)\end{cases}\end{aligned} \tag{2}$$

where: $q\ (x, y, t)$ - tangential stress in the contact area, μ - Coulomb's coefficient of friction, $p\ (x, y, t)$ - normal pressure in the contact area, $s\ (x, y, t)$ - sliding near the point of contact.

The tangential force at the point of contact has two components, one longitudinal, F_x, and one transverse, F_y, which results from the integration of the tangential force:

$$F(t)=\iint_S q(x,y,t),\quad F(t)=\{F_x,F_y\} \tag{3}$$

In kinematic regime the slip is defined by the following relations:

$$\begin{aligned} s_x(x,y,t)&=\nu_x(t)-\phi(t)\,y+\frac{\partial}{\partial x}u_x(x,y,t)-\frac{1}{V_m}\frac{\partial}{\partial t}u_x(x,y,t)\\ s_y(x,y,t)&=\nu_y(t)-\phi(t)\,x+\frac{\partial}{\partial x}u_y(x,y,t)-\frac{1}{V_m}\frac{\partial}{\partial t}u_y(x,y,t)\end{aligned} \tag{4}$$

where: ϕ - spin pseudo-slips, V_m – average speed of the surfaces in contact, $u_x(x,y,t)$, $u_y(x,y,t)$ – longitudinal and transverse displacements.

For the stationary regime the term $\partial/\partial t\ u(x,y,t)$ becomes null situation in which the relations (4) become:

$$\begin{aligned} s_x(x,y,t)&=\nu_x-\phi\cdot y+\frac{\partial}{\partial x}u_x(x,y)\\ s_y(x,y,t)&=\nu_y-\phi\cdot x+\frac{\partial}{\partial x}u_y(x,y)\end{aligned} \tag{5}$$

Pseudo-slips that occur in relation 1 together with the coefficient of friction and the normal force are essential parameters for determining the wheel-railway contact forces. The longitudinal, transverse, and spin pseudo-slips are established based on the relationships, [20]:

$$\nu_x=\frac{s_x}{V},\nu_y=\frac{s_y}{V},\phi=\frac{r\cdot\omega_s}{V}$$

ω_s – angular velocity of spin, V – forward speed.

Rolling without longitudinal and lateral slips is defined as pure rolling in [20-22] and is also called conical rolling, and the longitudinal or transverse pseudo-slip is defined as the ratio between the relative speed of the wheel in the longitudinal or lateral direction and its forward speed. This rolling can be performed only rarely and for short periods of time; all cases of abnormal rolling of the vehicle on the rails, such as acceleration or braking, lead to the occurrence of pseudo-slips at the wheel-railway interaction. In the case of pure slip, the tangential force between the two bodies reaches the friction limit. There is no slip below this limit. However, specialized studies have found an intermediate state in which the elasticity of the two bodies in contact allows the division of the contact area into an adhesion zone and a sliding zone. Thus, below the friction limit value, there is a transition from the predominance of the adhesion zone (1) to the sliding zone (4), according to fig. 1.

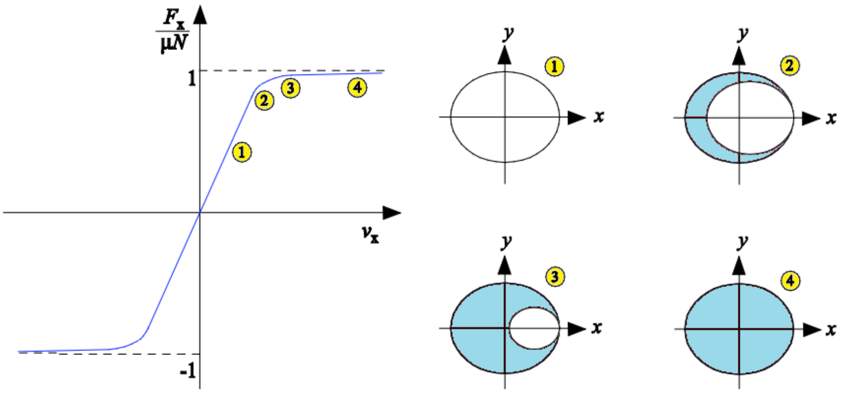


Fig. 1. Elliptical contact adhesion area [23]

The presentation will be clear and concise, and the symbols used therein will be specified in a symbol list (if necessary). In the paper the International System measurement units will be used. There will be no apparatus or equipment descriptions.

3. Numerical simulation

Within this section the analysis of the kinematic regime of the axle is performed considering a symmetrically loaded axle. For the presented case, the axle is equipped with S78 profile wheels and passes over a switch traversal with at a speed of 158 km/h in direct and 29 km/h in deviation. The load on the wheel is considered 20kN and 40kN respectively. The attacking wheel comes into contact with the switch traversal on the lip by making bi-contact between the switch traversal-railway-wheel, so that later, as the wheels advance, the contact moves to the outer side of the wheel's rolling surface, see fig. 2. The coordinates of the two contact points are established by the method of minimum elevation distances. At the same time, the distribution of normal loads on the two contact areas is made based on the fact that the rigidities are equal and the deformation is proportional to the ratio 3/2.

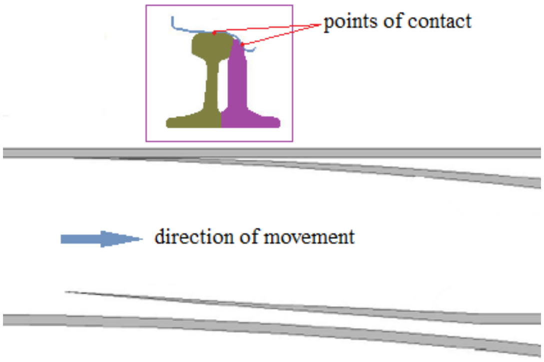


Fig. 2. The contact points at passing over switch

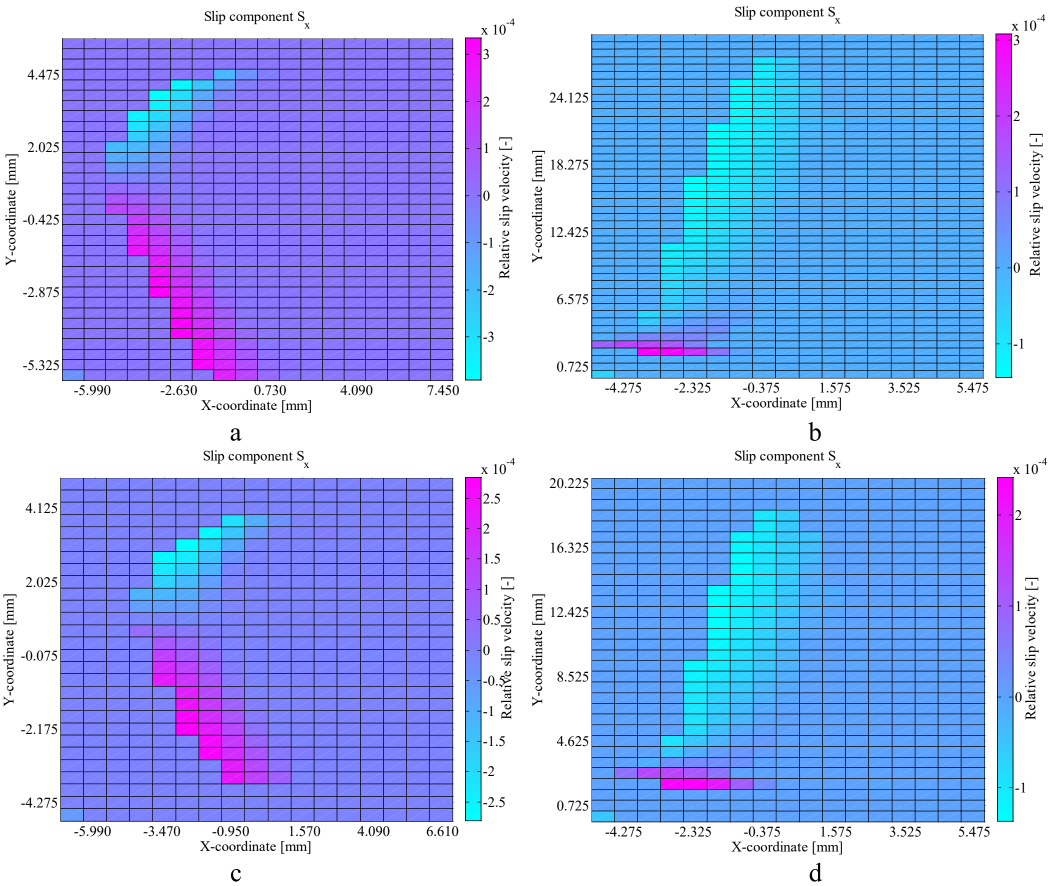


Fig. 3. Longitudinal pseudo-sliding speeds, load 40 kN-20 kN, speed 158 km/h

Figure 3 shows the distribution of the relative pseudo-slip velocity in the longitudinal direction considering the zero spin coefficient and the wheel load 40 kN (a,b) - 20 kN (c,d). An increase of the pseudo-sliding speeds is observed at the point of contact on the wheel flange (Fig. 3 a, c), where the sliding area of the contact spot is present. Pseudo-sliding speeds are reduced as the load on the wheel decreases. At the point of contact on the running surface, point of support, where the sliding area predominates, the sliding speeds show very small intensifications towards the inside of the running surface, fact justified by the takeover of the guide by the lip, see fig. 3 b, d.

Keeping the initial conditions of the simulation, fig. 4 shows the distribution of the relative pseudo-sliding speeds in transverse direction. An intensification of the transversal speeds can be observed, compared to the longitudinal ones. The maximum values are oriented towards the wheel flange at both points of contact and for both cases considered. There is a decrease in values as the load decreases.

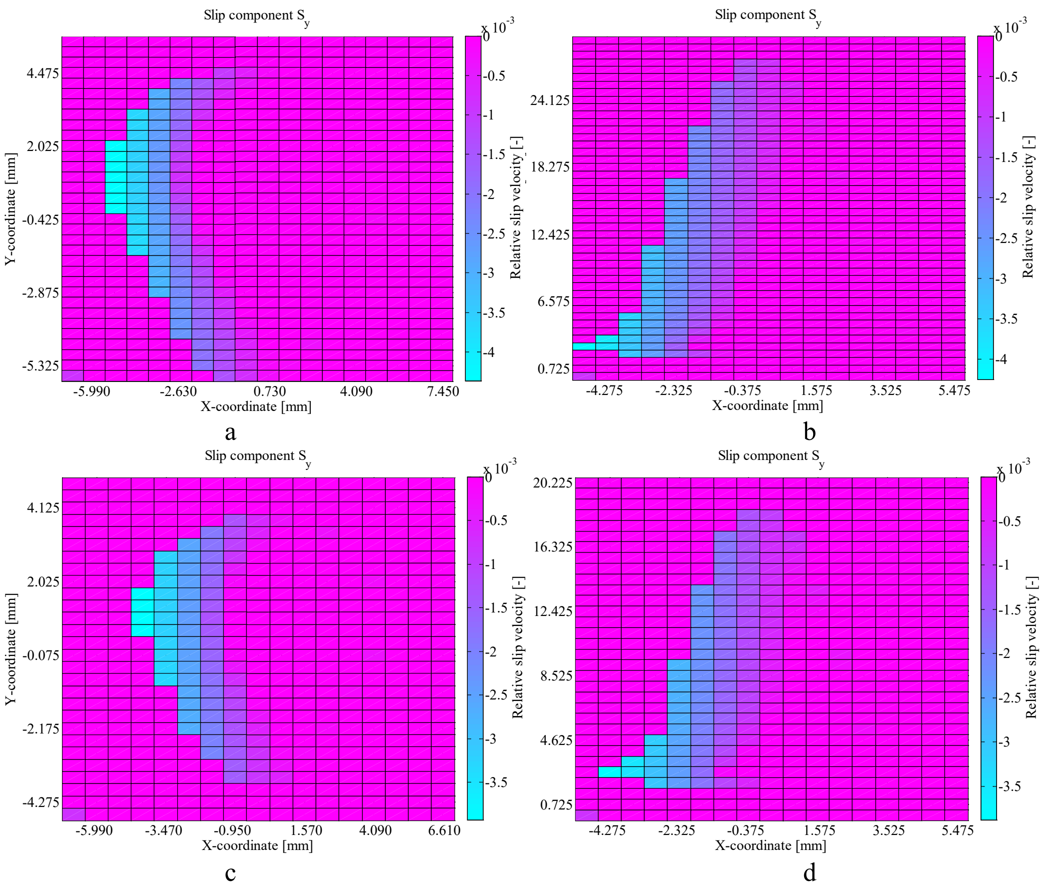
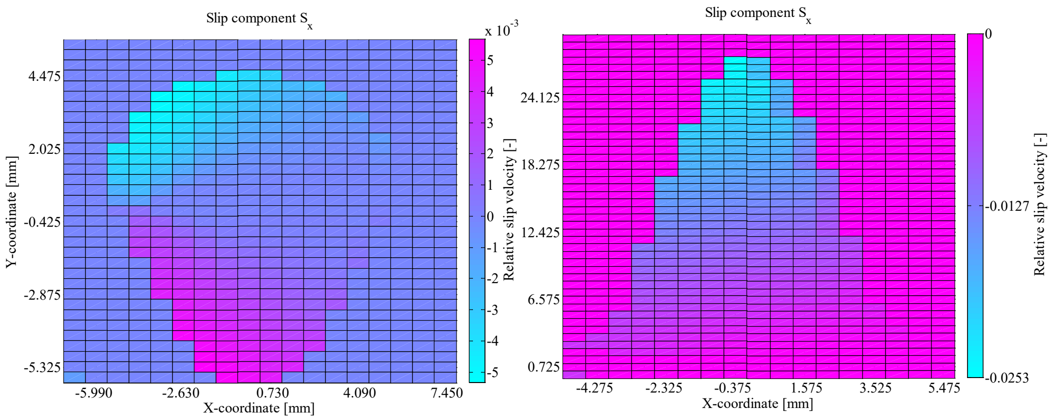


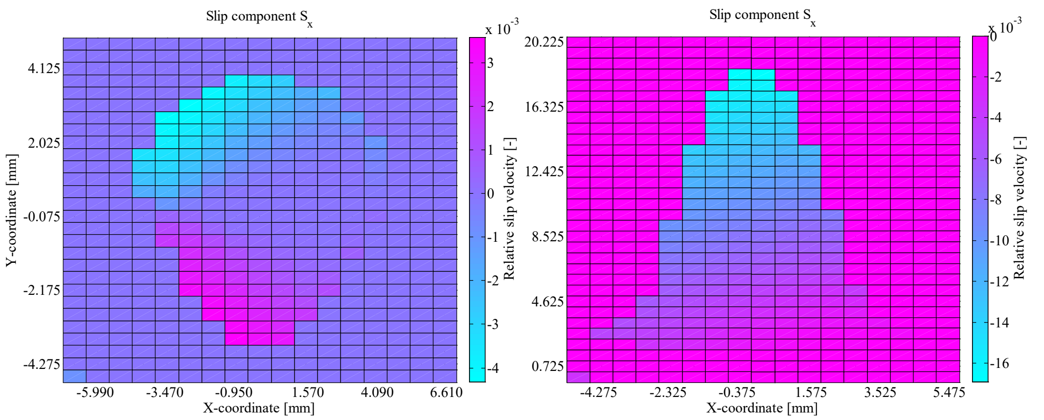
Fig. 4. Transversal pseudo-sliding speeds, load 40 kN-20 kN, speed 158 km/h



a b

Fig. 5. Longitudinal pseudo-sliding speeds, load 40 kN, speed 29 km/h, spin 0.000923 rad/s

If the traffic speed on deviation is reduced to 29 km/h, there is no significant change in the distribution of the pseudo-slip speed field.



a b

Fig. 6. Longitudinal pseudo-sliding speeds, load 20 kN, speed 29 km/h, spin 0.000923 rad/s

In this particular situation, also taking into account a non-zero value of the spin coefficient, 0.000923 rad/s, corresponding to the situation in which the axle runs at the adhesion limit, the distribution of pseudo-sliding speeds in longitudinal and transverse direction can be seen in fig. 5 to 8.

Pseudo-sliding speeds in the longitudinal direction are significantly increased for the situation where the load on the wheel is 40 kN and at the point of contact on the lip, see fig. 5a – 6a. This is justified by the guiding role of the wheel's lip. In the transverse direction, see fig. 7-8, compared to the situation in which the spin coefficient was considered zero, see fig. 4, there is a slight increase in values for the two cases considered, which is explained by the pivoting effect.

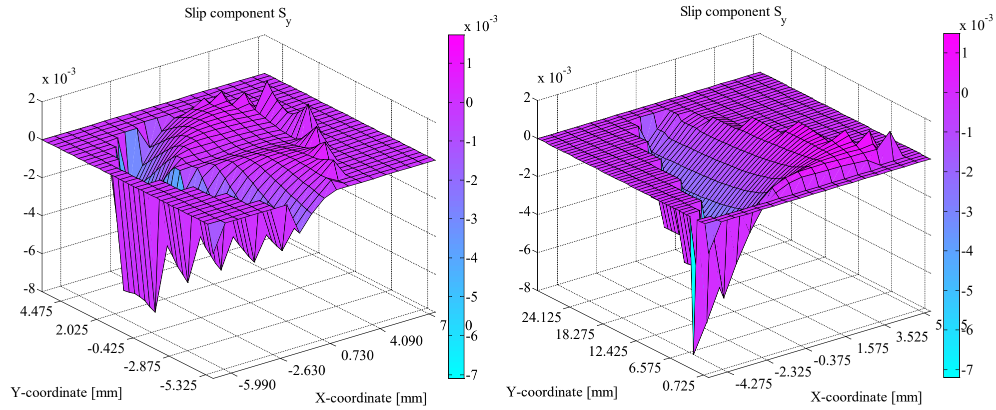


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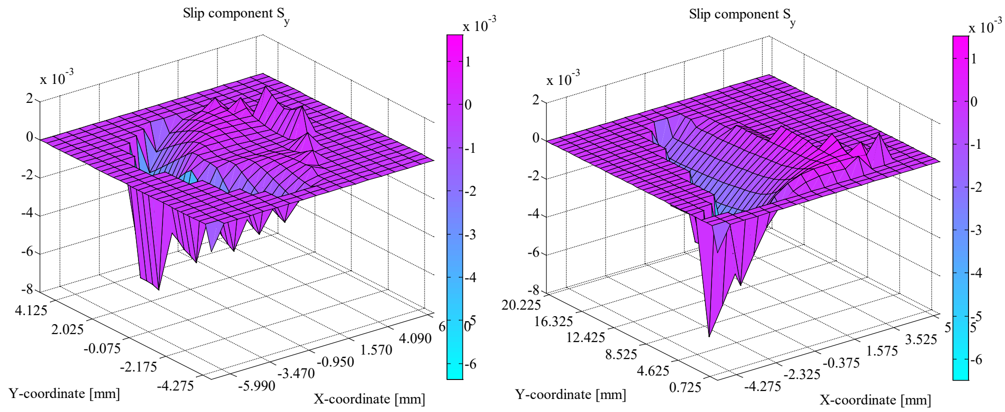
Fig. 7. Transversal pseudo-sliding speeds, load 40 kN.



b c

Fig. 8. Transversal pseudo-sliding speeds, load 20 kN.

4. Conclusions

In this paper a less studied side was approached from an analytical point of view in our country, namely the analysis of the contact between wheel and switch traversal and its influence on guidance.

Also, a three-dimensional approach to the kinematic behavior of an axle equipped with S78 profile wheels, by adapting the iterative Contact software to bi-contact.

The originality of the study consists in the application of the Contact software in case of wheel's interaction when crossing the switch traversal.

Friction forces and the distribution of stresses in the contact area are determined considering the pseudo-sliding speeds. The problem of tangential contact can also be solved, including the case of the secant contact. The values of the multipoint sliding speeds are of major importance when the second point of contact is on the wheel flange, which is crucial for safety against derailment.

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R E F E R E N C E S

[1]. *H. Sugiyama, T. Sekiguchi*, Wheel/rail contact dynamics in turnout negotiations with combined nodal and non-conformal contact approach, Multibody System Dynamics, 2012, 27:55–74, DOI 10.1007/s11044-011-9252-0.

[2]. *E. Kassa, J. C. O. Nielsen*, Dynamic interaction between train and railway turnout: full-scale field test and validation of simulation models, Vehicle System Dynamics , **46(5)**, 2008, pp. 521–534.

[3]. *E. Kassa, C. Anderson, J. Nielsen*, Simulation of dynamic interaction between train and railway turnout, Vehicle System Dynamics **44**, 2006, pp 247–258.

[4]. *H. Kono, Y. Suda, M. Yamaguchi, H. Yamashita, Y. Yanobu, K. Tsuda*, Dynamic analysis of the vehicle running on turnout at high speed considering longitudinal variation of rail profiles, Proceedings of ASME International Design Engineering Technical Conferences and Computer and Information in Engineering Conference, USA, 2005, pp. 2149-2154.

[5]. *S. Alfi, S. Bruni*, Mathematical modeling of train-turnout interaction, Vehicle System Dynamics **47**, 2009, pp. 551–574.

[6]. *Z. Ren, S. Sun, W. Zhai*, Study on lateral dynamic characteristics of vehicle/turnout system, Vehicle System Dynamics, **43(4)**, 2005, pp. 285–303.

[7]. *E. Kassa, G. Johansson* , Simulation of train–turnout interaction and plastic deformation of rail profiles, Vehicle System Dynamics, International Journal of Vehicle Mechanics and Mobility, **44:sup1**, pp. 349-359, http://dx.doi.org/10.1080/00423110600871566.

[8]. *X. Shu, N. Wilson, C. Sasaoka, J. Elkins*, Development of a real-time wheel/rail contact model in NUCARS and application to diamond crossing and turnout design simulations, Vehicle System Dynamics, **44(Suppl)**, 2006, pp. 251–260.

[9]. *Y. Q. Sun, C. Cole, P. Boyd*, A numerical method using VAMPIRE modelling for prediction of turnout curve wheel–rail wear, Wear **271**, 2011, pp. 482–491, doi:10.1016/j.wear.2010.10.010

[10]. *Z. Ren, W. Sun, S. Zhai*, Study on lateral dynamic characteristics of vehicle/turnout system, Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility, **43:4**, 2005, pp. 285-303, DOI: 10.1080/00423110500083262

[11]. *P. Wang*, Research on wheel/rail system dynamics on turnout, Phd thesis, Southwest Jiaontong University of China, 1997.

[12]. *J. Droździel, B. Sowiński, W. Groll*, The Effect of Railway Vehicle-Track System Geometric Deviations on its Dynamics in the Turnout Zone, Vehicle System Dynamics, **33:sup1**, 1999, pp. 641-652, DOI: 10.1080/00423114.1999.12063118.

[13]. ***https://www.cmcc.nl/User guide for CONTACT, Rolling and sliding contact with friction.

[14]. *K. E. Zaazaa , A. L. Schwab*, Review of joost Kalker's wheel-rail contact theories and their implementation in multibody codes, Proceedings of the ASME 2009 International Design Engineering Technical Conferences & Computers and Information in Engineering Conference IDETC/CIE, San Diego, California, USA, 2009.

[15]. *J. J. Kalker*, On the rolling contact of two elastic bodies in the presence of dry friction, Faculty Electrical Engineering, Mathematics and Computer Science, doctoral thesis, 1967.

[16]. *J.J. Kalker*, A fast algorithm for the simplified theory of rolling contact, Vehicle System Dynamics, Vol. **11**, 1983, pp 1-13.

[17]. *J.J. Kalker*, Survey of Wheel-rail rolling contact theory, Vehicle System Dynamics, Vol. **8**, 1979, pp 317-358.

[18]. *J.J. Kalker*, The computation of three dimensional contact with dry friction, International Journal Numerical Methods Engineering, Vol. **14**, 1979, pp. 1293-1307.

[19]. *J. Haines, E. Ollerton*, Contact stress distribution on elliptical contact surfaces subjected to radial and tangential forces. Proc. Inst. Mech. Eng. Vol. **177**, 1963, pp 95- 144.

[20]. *I. Sebesan*, The Dynamics of railway vehicles (in Romanian), Technical Publisher, 1995. pp. 87-85.

[21]. *C. Matei (Tudorache), M. Dumitriu, M. St. Sebesan*, The kinematic of the wheelsets at Y32 bogie (in Romanian), National Symposium of Railway Rolling Stock – Bucharest 2006, MatrixRom Publisher, ISSN 1843-9888, pp. 149-156.

[22]. *I. Sebesan, C. Tudorache, M. Dumitriu*, Aspects concerning kinematic aspects of the wheelset in curve traffic a railway vehicles (in Romanian), 3rd Symposium Durability and Reability of Mechanical Systems, Târgu-Jiu, 20-21 may 2010, „Constantin Brâncuşi" University of Târgu-Jiu, Engineering Faculty, Industrial Engineering Departament, in Proceedings Durability and Reliability of Mechanical Systems, Brâncuşi Academica Publisher, ISBN 978-973-144-350-8, pp. 53-58.

[23]. *A. Zougar*, Estudio del comportamiento vibratorio de vias ferroviarias mediante simulation numerica, tesis doctorale, Universitat Politecnica de Catalunyia, Barcelona, 2014.