Theoretical studies of horizontal dynamics parameters of the «wheel-rail» kinematic pair

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Summary. The article made a deep analysis of scientific publications researching horizontal dynamics performance of "wheel-rail" kinematic pairs. There has been worked out the scheme of the cohesive forces in the main crest contacts and "wheel-rail" kinematic pair by applying the theory of mechanisms and machines. There has been given calculation data pf transverse displacements of wheelsets and bogie frames, yaw angles wheelsets , angular velocities of wheel pairs at maximum speed and normal loads in the main contacts. There has been worked out the scheme for the passage of wheel wags turnout and shown the calculated dependence of rudder crest contact with racking wheel on the rail. There has been made a conclusion about the reliability of the theoretical research results.
Key words: "wheel-rail" kinematic pair, cohesive forces, horizontal dynamics, crest contact, sliding speed.

INTRODUCTION

Lack of reliable scientific data on frictional contacting of wheels and rails is an indirect cause of many problems in the "wheel-rail" kinematic pair. Among them - excessive wear of the wheels and rolling surfaces of the rails, particularly the undercut crests, increased resistance to movement of the kinematic, dynamic elevated horizontal load on the track, and others.

The main problems of horizontal dynamics and surface wear are associated with contacting the crest. In this regard, accurate simulation of the two-point contact with the rail wheels with the description of the crews kinematics direction by rail track, including the clash of the crest on the rail and wheel input turnout is crucial in the study of horizontal dynamics indicators in "wheel-rail" kinematic pair.

PUBLICATION ANALYSIS

The most important research of horizontal dynamics indicators in the "wheel-rail" kinematic pairs, forming the basis of the dynamics of railway rolling stock as a section of mechanics, are the works of the following scientists: MF Verigo [22], A.L.Golubenko [3], V.D.Danovich [2], A.I.Karmazin [4] V.G.Masliev [6], and others.

In studies conducted by scientists of the Dnepropetrovsk scientific school – E.P.Blohin, G.I.Bogomaz, Yu.V.Demin, M.L.Korotenko, V.F.Ushkalo, there was made a significant contribution to the science of the interaction of track and rolling stock composition. There have been developed fundamental methods of statistic dynamics and stability theory of railway vehicles.

In [9, 20, 1, 18, 21, 8] there have been suggested mathematical models, with varying degrees of completeness, showing the main features of the "kolesos" and allows you to examine both steady states and stability of the unperturbed motion of railway vehicles, taking into account the contact forces.

Design diagrams, mathematical models of the crew and track to a great extent reflect the physical processes that take into account the characteristics of the relationships between elements of systems, disturbing factors and mode of traffic.

The resulting system of nonlinear differential equations with right-hand sides is examined by using a computer [12, 23].

According to the nature of oscillatory processes elements crews, there are made conclusions about the stability of their motion, the amplitudes of forces and accelerations acting on them.

This method allows us to study the dynamics of the vehicles movement and get the motion characteristics of the system elements power interaction, wear characteristics.

In [13, 14] there has been suggested a fundamentally new approach to the classical problem of the wheels contact with the rails and has been used in studies of the "carriage-way."

Questions of coupling dynamics, crews rail gauge direction, kinematic frictional resistance to movement, were first investigated at a precise description of the wheels in contact with the rail, including a two-point [15].

In particular, in [16], the problem of redistribution of normal stresses in the wheel contact with the rails, was solved with contact deformations.

The considered works, depending on the level of detail modeling of contact wheels with rails, can be divided into three groups.

To first refer operation in which the wheel profile is seen in a simplified form as a conical surface.

This, above all, is a study where differential equations linearization systems are performed to study the stability of motion.

Naturally, in this case contact is considered as a single point. Same type of modeling is most often used in studies of locomotives traction qualities.

The second group consists of works in which the profiles of wheels and rails are considered to approximate the actual profiles, but the contact is considered as a single point.

It uses both linear and nonlinear friction characteristics.

It should be noted that while introducing of external friction characteristics as a depending of coefficient on the relative coupling coefficient of sliding, a precise description of the profile of the wheel and the rail is not essential, since the restriction of singleton contacting key pair - meters kinematics interaction - the radius of the rolling circle and taper wheel - changes slightly [5, 7].

The third group includes a few studies in which the contact of wheel and rail is considered as a two-point.

However, there are no works where at the same time there would be considered the following features of two-point contacting:

- the redistribution of loads between normal contacts,
- the velocity distribution of the sliding contact,
- taking into account the difference in the laws of friction on the contact surface of rolling contact and raised bed,
- describing kinematics and dynamics of the two-point contact system "wheel-rail" in a turnout with regard to the shape profile of the wheel and rail [22].

PURPOSE OF RESEARCH

Therefore, the aim of this work is to develop methods of calculating horizontal transverse crews vibrations by improving the modeling of two-point wheel contact with the rail.

The task of theoretical research is to obtain the comparative dynamics horizontal engine with extremely worn crests of wheels bandages. To check the validity of the results of theoretical research, the model debugging was performed by checking derivatives and intermediates for the parameters of robust criteria:

- transverse displacement of wheelsets and bogies frames,
- periods of wavelengths and wiggle angles in wheelsets and bogies frames,
- angular speed of wheelset, velocity slip in the main crest and contacts,
- normal stress in the core and the crest contacts of wheels with rails and turnout elements.

RESEARCH RESULTS

To calculate the contacts coupling force there has been used the method of professor V.Tkachenko [17, 18], which is based on the dependence of the longitudinal and transverse forces coupling on longitudinal and transverse velocities in sliding contacts.

Shows a diagram of adhesion forces in the main contacts K_{1jk}.

In the main contacts K_{1jk} relative movement of the wheel and the rail has rolling nature of slip or grip.

Longitudinal S_{x1jk} and transverse S_{y1jk} components of adhesion forces in the main contacts (Fig. 1) [10, 11]:

$$S_{x1jk} = -N_{1jk} \cdot \Psi_0 \cdot k_{x1jk};$$
$$S_{y1jk} = -N_{1jk} \cdot \Psi_0 \cdot k_{y1jk}, \quad (1)$$

k_{x1jk}, k_{y1jk} – regarding the longitudinal and transverse coefficients using the clutch, which are defined by the formulas [19]:

$$k_{xjk} = \frac{\varepsilon_{xjk}}{a_x \cdot \varepsilon_{xjk}^2 + b_x \cdot \left|\varepsilon_{xjk}\right| + c_x} \cdot \frac{d_x}{e_x + \varepsilon_{yjk}^{g_x}}, \quad (2)$$

$$k_{yjk} = \frac{\varepsilon_{yjk}}{a_y \cdot \varepsilon_{yjk}^2 + b_y \cdot \left|\varepsilon_{yjk}\right| + c_y} \cdot \frac{d_y}{e_y + \varepsilon_{xjk}^{g_y}}, \quad (3)$$

where:

$a_x, b_x, c_x, d_x, e_x, g_x, a_y, b_y, c_y, d_y, e_y, g_y$ – empirical correlation coefficients grip [13],

ε_{xjk}, ε_{yjk} – respectively, the longitudinal and transverse sliding core in the main (first) contacts in coordinate systems for wheel profiles $O_{jk} X_{jk} Y_{jk}$:

$$\varepsilon_{xjk} = \frac{v_{x1jk}}{\dot{\varphi}_k \cdot R_{1jk}}, \quad \varepsilon_{yjk} = \frac{v_{y1jk}}{\dot{\varphi}_k \cdot R_{1jk}}, \quad (4)$$

or, given the fact that $\bar{v}_{1jk} = \bar{V} + \bar{V}_{\varphi 1jk} + \bar{V}_{\psi 1jk} + \bar{V}_{y1k} + \bar{V}_{p1jk}$:

$$\varepsilon_{x1k} = \frac{V \cdot cos\psi_k + \dot{\psi}_k \cdot A}{\dot{\varphi}_k \cdot R_{1jk}} - 1, \quad (5)$$

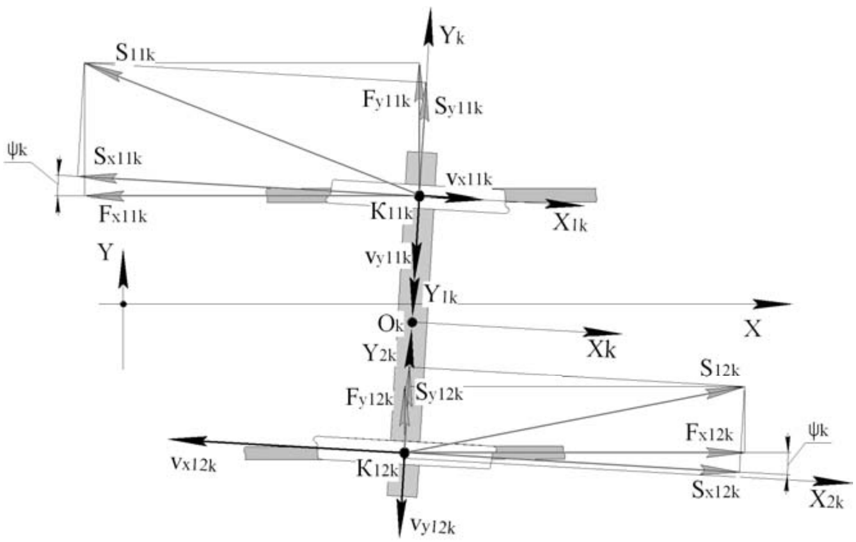

Fig. 1. Scheme of adhesion forces in the main contacts

$$\varepsilon_{x2k} = \frac{V \cdot cos\psi_k - \dot{\psi}_k \cdot A}{\dot{\varphi}_k \cdot R_{12k}} - 1, \qquad (6)$$

$$\varepsilon_{yjk} = \frac{\dot{y}_k - V \cdot sin\psi_k - \dot{y}_{pjk}}{\dot{\varphi}_k \cdot R_{1jk}}. \qquad (7)$$

Traction in key contacts in the absolute coordinate system: OXY :

$$S_{1jk} = \sqrt{S_{x1jk}^2 + S_{y1jk}^2}. \qquad (8)$$

The force of adhesion F_{x1jk} i F_{y1jk} in key kontacts of the absolute coordinate system OXY :

$$F_{x1jk} = S_{x1jk} \cdot cos\psi_k + S_{y1jk} \cdot sin\psi_k, \qquad (9)$$

$$F_{y1jk} = S_{y1jk} \cdot cos\psi_k - S_{x1jk} \cdot sin\psi_k. \qquad (10)$$

In Fig. 2 there is shown a diagram of adhesion forces in the main and crest contacts for two-point contacting of the wheel with a rail.

In the crest contacts K_{2jk} relative movement of the wheel and the rail has the character of sliding friction, so the strength of adhesion in crest contact S_{22k} can be determined by Culon's law:

$$S_{22k} = -N_{22k} \cdot f_0 \cdot sign(v_{22k}), \qquad (11)$$

where: f_0 − friction coefficient in crest contact.

Longitudinal S_{x2jk} and vertical S_{z2jk} components in the adhesion force of crest contacts systems of wheels coordinates $O_{jk}X_{jk}Y_{jk}$:

$$S_{x22k} = S_{22k} \cdot sin\zeta_{2k}, \qquad (12)$$

$$S_{z22k} = S_{22k} \cdot cos\zeta_{2k}. \qquad (13)$$

Longitudinal F_{x2jk} and transverse F_{y2jk} components of adhesion forces in crest contacts in the absolute coordinate system OXY :

$$F_{x2jk} = S_{2jk} \cdot cos\psi_k, \qquad (14)$$

$$F_{y2jk} = S_{2jk} \cdot sin\psi_k. \qquad (15)$$

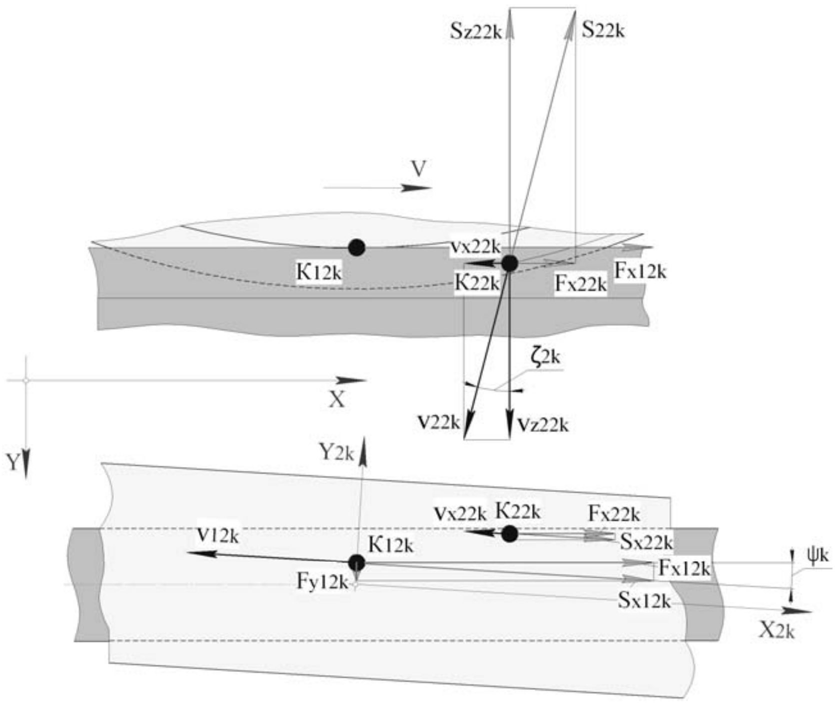

Fig. 2. Scheme of adhesion forces in the crest contacts

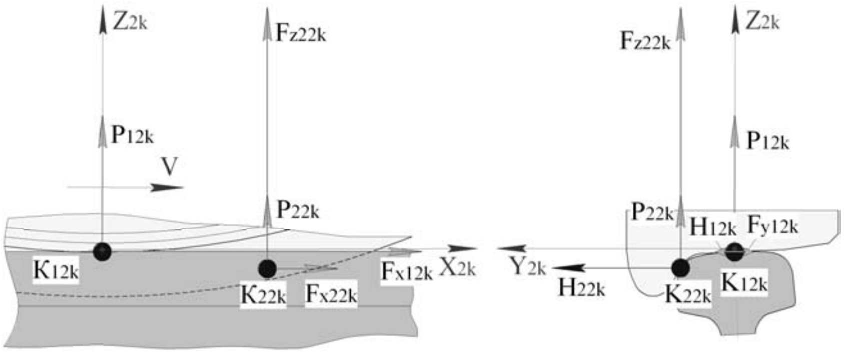

Fig. 3. The contact forces structure of two-point contact with the rail wheels

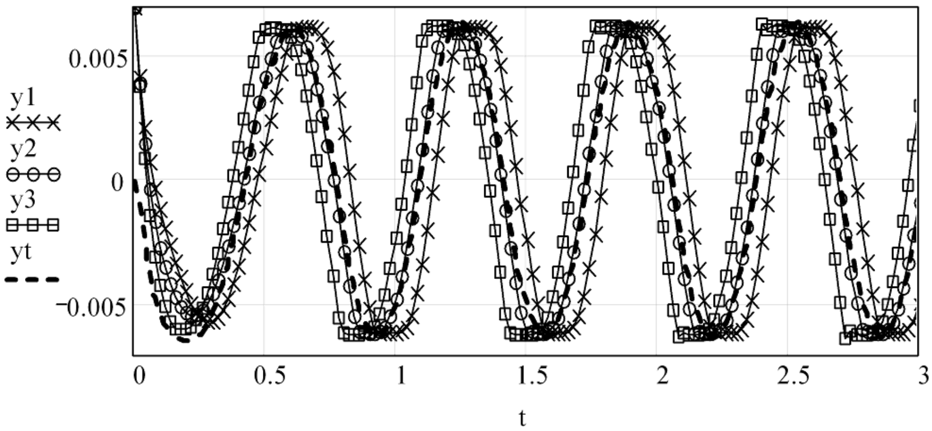

Fig. 4. Transverse displacement of wheelsets and bogies frames for velocity of 30 m/s: y1, y2, y3 – lateral movement (side assignment) of wheelsets, yt – lateral movement (side assignment) of the frame cart, t – time, s

Fig. 3 shows the structure of the contact force to the wheels that have a two-point contact with the rail.

Vector equation of contact forces acting on the wheel bogies from the rails along the axes of the absolute coordinate system, are as follows:

$$\overline{X}_{jk} = \sum_{i=1}^{2} \overline{F}_{xijk} , \qquad (16)$$

$$\overline{Y}_{jk} = \overline{F}_{y1jk} + \sum_{i=1}^{2} \overline{H}_{ijk} , \qquad (17)$$

$$\overline{Z}_{jk} = \overline{F}_{z2jk} + \sum_{i=1}^{2} \overline{P}_{ijk} . \qquad (18)$$

Calculation results of deterministic dependency of kinematic and dynamic crew motion parameters in a straight section lines are shown in Fig. 4-8.

In Fig. 4 there is shown the calculation of the transverse displacements (lateral assignment) of wheelsets and bogies frames at maximum speed of movement – 30 m/s. The period of oscillation for speeds is 0,72 s, and the wavelength wiggle – 21,6 m.

The calculation of wheelset wiggle angles shown is shown in Fig. 5. The maximum wiggle angles for the speed of 30 m/s were 0,0028 rad.

In Fig. 6 shows the calculation of the angular rotation velocity of the wheelset around their rotation axes. Fluctuations in the angular velocity is associated with the wheelsets wiggle noticeable at speeds over 20 m/s.

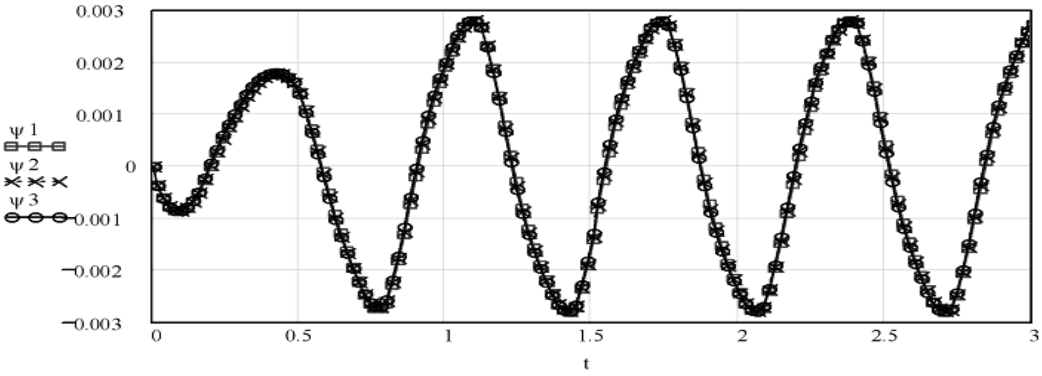

Fig. 5. Angles of wheelsets wiggle for velocity of 30 m/s, ψ1, ψ2, ψ3 – wiggle angles, respectively, 1, 2 and 3 first trolley wheelsets, t – time, s

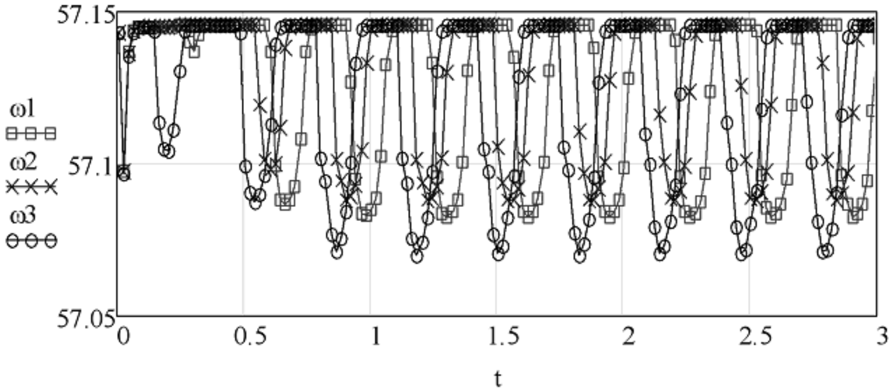

Fig. 6. Angular wheelset velocity for velocity of 30 m/s, ω1, ω2, ω3 – angular velocity of the 1, 2 and 3 wheelset, rad/s

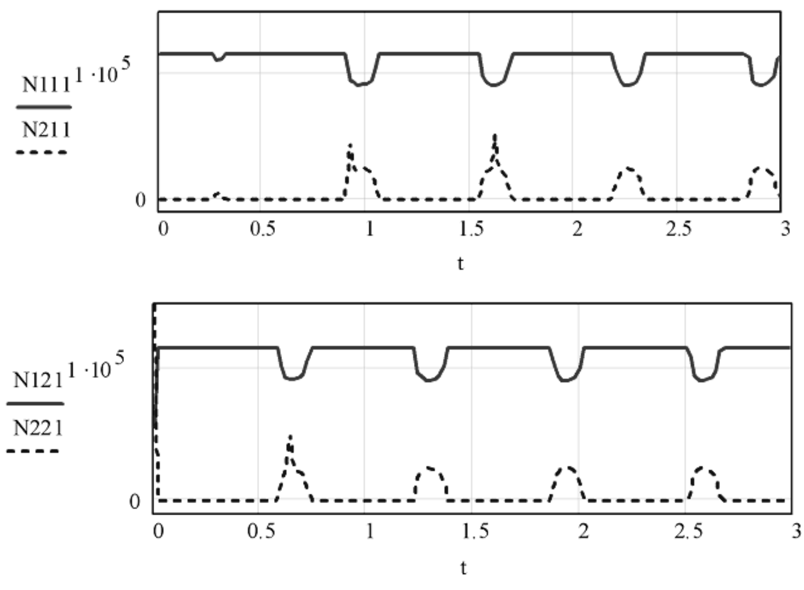

Fig. 7. Normal load on the main N1jk and crest N2jk contact wheels with the rails when driving in a straight section of track at a speed of 30 m/s

Fig. 7 shows the calculation of normal reactions in the core - N111 ... N121 and N211 ... N221 crest contacts of wheelset wheels for velocity of 30 m/s: N111, N121 - basic, N211, N221 - the crest contact of first wheelset wheels.

Sliding velocity in the main and crest contacts are considered as the velocity of the wheels crest surface under the surface of the rails, and, regardless of whether there is direct contact surfaces or not. Therefore, the rate of slip is conditional and is called the conventional sliding speed.

Fig. 8 shows an example of changes in conventional velocity sliding in crest contacts in the third wheelset wheels locomotive at a speed of 30 m/s.

Fig. 9. scheme of wheel studs passage in protysherstnomu movement in turnouts is shown in.

In Fig. 10 shows calculations of relaying the crest contact at vkochuvanni of wheel on the studs of a turnout at protysherstnomu motion for speeds of 10 m/s and 20 m/s.

Depending on the crest contact pattern there can be identified three phases hitting the wheel on studs (Fig. 11).

The first phase - hitting the rail of frame – exists on the section from point I to point II. At this phase the crest contacts with the groove of frame rail at point K_2, and between the studs and the edge of the crest there is a gap τ_1, which decreases as we approach point II. In this case, the vertical and horizontal components of the normal crest reactions depend on the angle of contact:

$$P_2 = N_2 \cdot sin\,\mu\,, \qquad H_2 = N_2 \cdot cos\,\mu\,, \quad (16)$$

where: contact angle $\mu = 90 - \gamma_2$.

The second phase – hitting on the tip studs - takes place in the area from point II to point III (Fig. 12). In this phase, the crest contacts with elements of turnout at two points: the point K_2 - with frame rail and at the point of K_3 – with studs. Changing the values of the reaction N_2, N_3 in section II-III is shifting nature: at the point of II – N_3=0, and in point III – N_2=0.

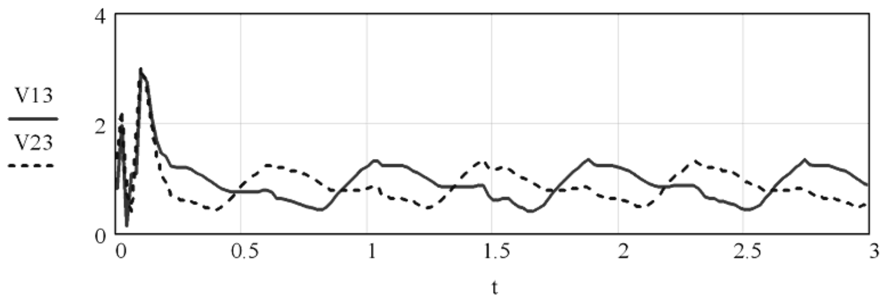

Fig. 8. Changing the conventional speed of sliding in crest contacts of wheels with tracks for velocity of 30 m/s: V13, V23 - the first and second wheel of the third wheelset

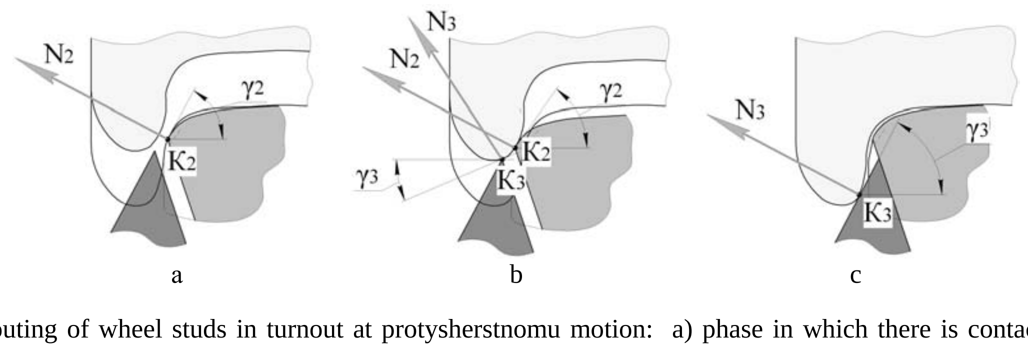

Fig. 9. Routing of wheel studs in turnout at protysherstnomu motion: a) phase in which there is contact with the crest of frame rail - point K2, b) phase of crest contact shifting from frame rails on the studs - the point K_2 and K_3, c) phase of full crest contact with the studs at the point K_3.

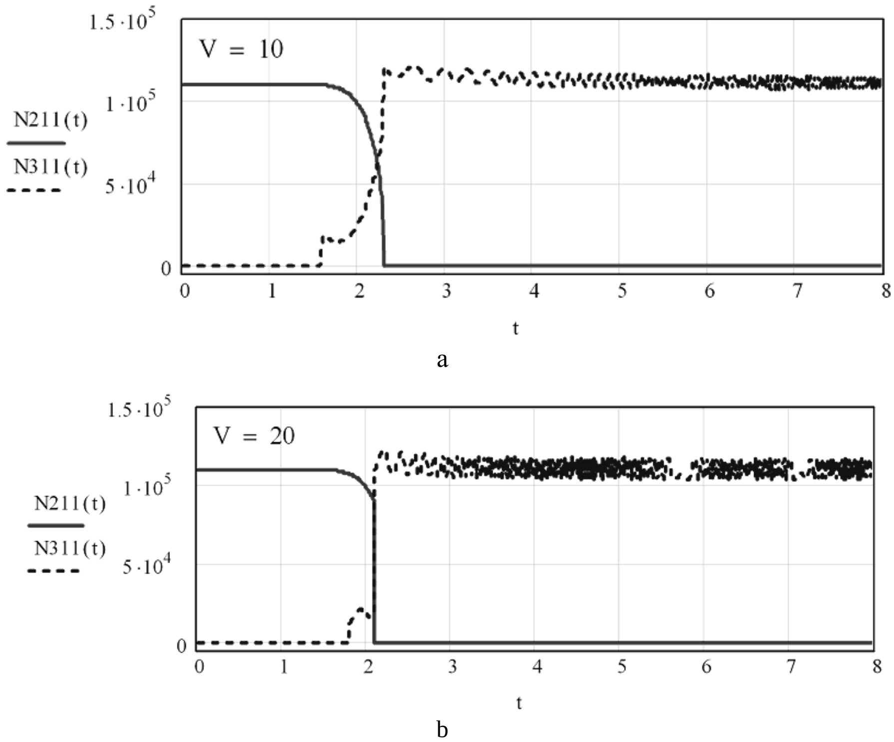

Fig. 10. Relaying of crest contact at wheel vkochuvanni on the studs on the turnout at protysherstnomu motion: a) speed of 10 m / s, b) 20 m/s

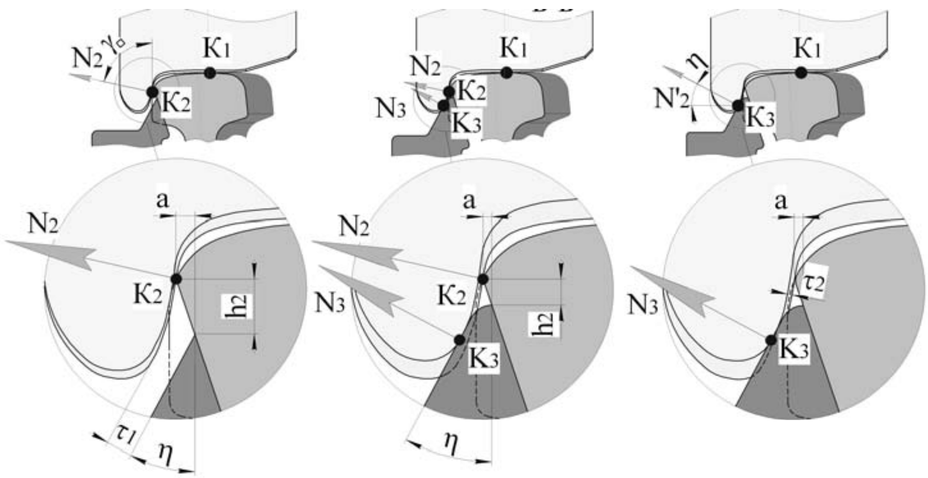

Fig. 11. Scheme of the crest hitting the maximum tie worn profile on turnout studs

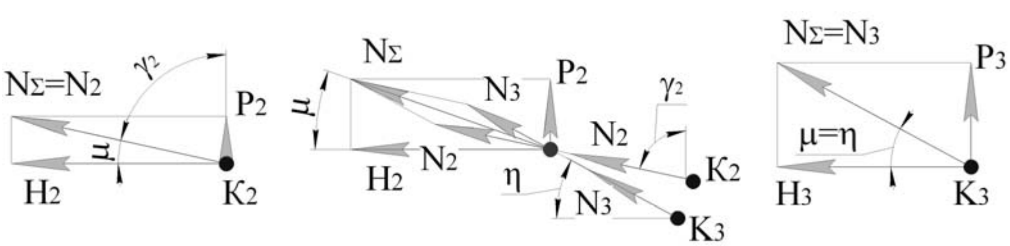

Fig. 12. Scheme of crest reactions on turnouts

Overall reaction of the crest on the turnout consists of reactions at points K$_2$ i K_3 ($\overline{N}_\Sigma = \overline{N}_2 + \overline{N}_3$) and is defined by the formula:

$$N_\Sigma = \sqrt{N_2^2 + N_3^2 + 2 \cdot N_2 \cdot N_3 \sin(\gamma_2 + \eta)} \,. \quad (17)$$

The third phase – after the third point – is the movement of the wheel studs. In this phase, between the frame rail and the crest, there is a gap τ_2 and steering effort is passed through a point on the crest K_3.

$$P_3 = N_3 \cdot \sin\eta \,, \qquad H_3 = N_3 \cdot \cos\eta \,. \quad (18)$$

In the absence of excess wear and defects studs its geometry does not create conditions for vkochuvannya of wheels on the rail. There are manual stair cases that are usually associated with defective studs.

CONCLUSIONS

1. The results of theoretical studies suggest an adequate level of research results and the use of models for the study of the horizontal parameters dynamics of locomotive wheels with worn profiles.

2.The results of the research model are significantly affected by the redistribution of contact stress between contacts of the wheel crest and turnout elements: frame rails and studs.

REFERENCES

1. **Blokhyn E.P., Manashkyn L.A., 1982.:** Train dynamics (transient longitudinal oscillations). Transport. 222. (in Russian).
2. **Danovych V.D., Reidemeister A.G., 1999.:** Mathematical model of wheel-rail interaction. Transport: Collection of scientific works. 2, 17-22. (in Russian).
3. **Golubenko A.L., 1993.:** Clutch of wheel and rail. VIPOL. 448. (in Russian).
4. **Karmazyn A.Y., 1955.:** About wheel slip at tangential force. Railways technologies. 8,12-14. (in Russian).
5. **Kolodyazny P., 2010.:** Some problems of improvement of ultrasonic control of wheel pair axles while their diagnostics. TEKA. Commission of motorization and power industry in agriculture. XC, 132–139.
6. **Maslyev V.H., 1996.:** Study of dynamic processes in the movement of railway vehicles. East Ukraine Univ. Herald. 4, 176-178. (in Russian).
7. **Myamlin S., Neduza L., Ten A., Shvets A., 2013.:** Research of friction indices influens on the freight car dynamics. TEKA. Commission of motorization and power industry in agriculture. XIII, 4, 159-166.
8. **Mytrokhyn A.N., 1998.:** "Wheel-rail": there is required more advanced teoriya. Railway transport. 7, 41-44. (in Russian).
9. **Pevzner A.Z., 1996.:** On the problem of interaction between wheel and rail. locomotive. 9, 19-20. (in Russian).
10. **Sapronova S., Tkachenko V., 2013.:** Justification of the choice of admissible parameters wear paddlewheel locomotives. TEKA. Commission of motorization and power industry in agriculture. XIII, 4, 270-278.
11. **Sapronova S.U., 1997.:** Distribution of power and geometrical parameters for the two-point contact of the wheels of railway vehicles with rails. Herald of East Ukraine Volodymyr Dahl National University. №2(6), 205-207. (in Russian).
12. **Sapronova Svetlana., 2010.:** Modeling of Locomotive wheel profile form. TEKA. Commission of motorization and power industry in agriculture. XC, 270–278.
13. **Tkachenko V., Sapronova S., 2007.:** Steerability of railway vehicles. Transport Problems. International Scientific Journal. T.2, Z.4, 9–16.
14. Tkachenko V.P., 1996.: Kinematic resistance to movement of rail vehicles. Monograph. 200. (in Russian).
15. **Tkachenko V.P., 1996.:** Kinematic resistance to movement of rail vehicles.Herald of East Ukraine Volodymyr Dahl National University. 5(14), 200–208.
16. **Tkachenko V.P., 1997.:** Control of railway vehicles. SUDU. 92. (in Russian).
17. **Tkachenko V.P., 1999.:** Calculating and predicting road frictional resistance to movement of railway carriages. Dis. Dr. ... Engineering. sciences. 05.22.07. 371. (in Russian).
18. **Tkachenko V.P., Sapronova S.U. 1997.:** Kinematics direction crews rail track. 22. (in Russian).
19. **Tkachenko V.P., Sapronova S.U., 2009.:** Studies on the characteristics clutch wheel and rail on a special stand. Herald All-scientific.-research. and design Konstr. Inst electric locomotive. 89-99. (in Russian).
20. **Ushkalov V.F., Malyshev Y.U, 1989.:** Influence of creep model choice for joint spatial variation rail vehicle and uprugodissipativnogo inertial way. Vibrations and dynamic quality of railway rolling stock. Intercollege. Sat Nauchn. tr. Dnepropetrovsk. 266/36, 10-21. (in Russian).

21. **Ushkalov V.F., Reznykov L.M., Red'ko S.F., 1982.:** Statistical dynamics of railway vehicles. Naukova dumka. 360. (in Russian).
22. **Verigo M.F., Kogan, A.Y., 1986.:** Interaction of the way and rolling stock. Transport. 559. (in Russian).
23. **Wickens A.H., 1996.:** Fundamentals of rail vehicle dynamics: guidance and stability. International Heavy Haul Association. − Virginia Beach: 2808 Forest Hills Court (USA). 481.

ТЕОРЕТИЧЕСКИЕ ИССЛЕДОВАНИЯ ПОКАЗАТЕЛЕЙ ГОРИЗОНТАЛЬНОЙ ДИНАМИКИ КИНЕМАТИЧЕСКОЙ ПАРЫ «КОЛЕСО-РЕЛЬС»

Сапронова Светлана

Аннотация. В статье выполнен глубокий анализ научных публикаций по исследованию показателей горизонтальной динамики кинематической пары «колесо-рельс». Разработана схема сил сцепления в основных и гребневых контактах кинематической пары «колесо-рельс» с применением теории механизмов и машин. Представлены расчетные данные поперечных перемещений колесных пар и рам тележек, углов виляния колесных пар, угловых скоростей вращения колесных пар при максимальной скорости движения, а также нормальных нагрузок в основных контактах. Разработана схема прохождения колесом остряка стрелочного перевода и представлена расчетная зависимость перекладки гребневого контакта при вкатывании колеса на рельс. Сделан вывод о достоверности результатов теоретических исследований.

Ключевые слова: кинематическая пара «колесо-рельс», силы сцепления, горизонтальная динамика, гребневой контакт, скорости скольжения.