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IMPROVEMENT OF THE SERVICE LIFE AND STABILITY OF RAIL VEHICLES IN THE MINING AND INDUSTRIAL SECTOR BY MEANS OF FRICTION MODIFIERS

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Abstract. The scientific work includes the study of the impact of friction modifiers on reducing wear intensity, extending the service life, and improving the safety of technical equipment in mining and industrial railway transportation, particularly under challenging operational conditions with small curve radii. Mathematical modeling of dynamic interactions between the railway track and moving transport vehicles was conducted taking into consideration spatial vibrations of the transportation system in vertical, lateral, and longitudinal directions. Dependencies were established between the dynamics indicators of transport vehicles and railway tracks based on the speed of movement along connection joints of the jointed rail track with irregularities, as well as the relationship between the wear intensity of working surfaces and variations in friction coefficient values.

The purpose of the study is to extend the service life, stability, and safety of operation of technical equipment in mining and industrial railway transportation by reducing the friction coefficient. The methodology of the work includes methods of mathematical and computer modeling of dynamic interactions between mining rail transport vehicles and the railway track, which affect the side wear of rail heads and wheel treads. The obtained results of theoretical research on the influence of friction coefficient parameters between the side surfaces of rail heads and wheel treads on the dynamic indicators of interactions between transport vehicles and the railway track, as well as wear indicators of the "wheel-rail" pair, are presented. It is determined that reduce of the friction coefficient from 0.25 (dry friction) to 0.05 (use of friction modifiers) results in a 3.5–4.5 times decrease in wear indicators and a 1.3–1.5 times increase in stability against derailment of wheels. It is found that replace of the connection joints of rails with jointless track in curves with small radii leads to a reduction of horizontal loads by 15–56%, and the coefficient of safety margin against derailment of wheels increases by 19–26%. The obtained results are intended for the implementation of new friction modifiers and technologies in mining and industrial rail transport to enhance the economic efficiency and service life of moving transport vehicles and rail tracks, improve safety of their operation and work of specialists in railway transport.

Keywords: mining and industrial rail transport, service life, wear, friction coefficient, curved track sections.

1. Introduction

Curved sections of the rail track with particularly small radii of curvature, ranging from 80 to 400 meters, in mining and industrial rail transport pose the most complex and hazardous operational conditions. These conditions create extreme challenges for crucial performance indicators of the transport system, such as safety of movement, service life of components of rail track elements and flange tread surfaces of rolling stock's running gear, as well as operational expenses related to the rail track major and periodic repairs and maintenance.

In curved sections of the track, complex spatial oscillations of transport vehicles occur in the transverse, longitudinal, and vertical directions. This leads to significant loading on the elements of the rail track and the running gear of rolling stock, creating a high specific pressure in the contact zone between the side surfaces of the rail heads and the flange tread surfaces of the wheel rims in curved sections with small radii (80–400 meters) of mining and industrial rail transport.

The wear intensity of the side surfaces of rail heads amounts to 4–10 mm per 1 million tons of gross tonnage transported, and the service life of rail track elements is 0.5–2.0 years, which is several times less than the normative requirements. Further operation of the rail track leads to violations of traffic safety regulations and an increase in accidents [1].

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The research work [2] highlights the challenges of ensuring the efficiency and safety of mining enterprises due to the unsatisfactory condition of their essential production assets. It is noted that over the last 20 years in the coal industry, reconstruction and planned replacement of outdated equipment have been inadequately carried out. In the coal sector, the degree of wear of the main production assets reaches 60% (and up to 70% in some enterprises), and their further operation leads to equipment breakdowns and a significant increase in energy consumption. The recommendations provide proposals for a strategy of rational management for the functioning of mining enterprises. However, the study did not address the issue of extending the service life of mining and industrial rail transport equipment.

The scientific works [3–5] discuss the issues of extending the service life of the track of specialized industrial rail transport through the establishment of norms for the installation and maintenance of rail connection joints. Additionally, the studies involve strength calculations of rail connection joints subjected to the impact of quarry-specific rolling stock under various conditions of rail support on sleepers. However, the research does not address the topic of extending the service life of mining and industrial rail transport through the use of friction modifiers.

The known scientific studies [1; 6–9] are focused on improving the service life of the running gear components of wagons through the improvement of their design and wheel profiles. However, these studies did not address the issue of extending the service life of transport vehicles through the utilization of friction modifiers.

The scientific studies [10–13] address the issue of excessive wear on the contact side surfaces of rail heads and wheel treads in the railway industry. It is determined that on curved track sections with radii of 350–460 meters, the wear intensity of the rail head's side surface is 0.189–0.207 mm per 1 million tons of gross tonnage transported. For a curve radius of 325 meters, the wear intensity is 0.202–0.268 mm per 1 million gross tons, and for a radius of 268 meters, the wear intensity reaches 0.256–0.308 mm per 1 million gross tons. With such high values of rail head wear intensity, the service life of the track structure is reduced to 0.5–1 year, while the normative requirement is 6–8 years, resulting in a significant reduction in their operational lifespan.

The analysis of recent publications aimed at increasing the service life of various types of rail transport equipment revealed that insufficient attention was given in scientific works to the study of mechanics of dynamic interaction processes between transport system equipment accounting the properties and characteristics of friction modifiers.

An unsolved aspect of the overall problem is the underestimation of the influence of friction coefficient indicators on the wear intensity of interacting contact surfaces of working parts of vehicles under significant cyclic dynamic loads, with high specific pressures up to 600 MPa and relative slippage of up to 10% in areas of contact between the side surfaces of rail heads and wheel tread flanges.

The relevance of this issue is determined by significant operational costs for routine maintenance, capital repairs, and ensuring safety requirements for the operation of rail transport vehicles in mining and industrial enterprises.

The main direction of this scientific work is to study the influence of friction coefficient values on reducing wear intensity, increasing the service life, and improving the safety of mining and industrial rail transport vehicles under particularly challenging operating conditions with small curvature radii (80–650 m).

The purpose of the work is to increase the efficiency, service life, stability, and safety of mining and industrial rail transport vehicles through the reduction of the friction coefficient.

To achieve the set goal, the following research tasks are planned:

- to improve the mathematical model of the interaction processes between rolling stock and rail tracks in curved sections of mining and industrial rail transport, considering relative slippage of up to 10% between the side surfaces of rail heads and wheel flanges, with the influence of friction modifiers taken into account;
- to make a mathematical model of the interaction processes between rolling stock and rail tracks in curved sections of mining and industrial rail transport to investigate the loading, stability, and wear intensity of contact side surfaces of rail heads and wheel flanges, considering relative slippage of up to 10% between the side surfaces of rail heads and wheel flanges, with the use of friction modifiers;
- to evaluate the impact of friction coefficient values between the contact surfaces of the interacting side parts of rail heads and wheel flanges on wear intensity, service life, stability, and safety of mining and industrial rail transport vehicles.

2. Research Methods

In accordance with the outlined main research directions and objectives, the work assumes performing of mathematical modeling and theoretical investigations of dynamic processes of interaction between the rail tracks and rolling stocks to determine fundamental relationships governing the formation of extreme loads, wear, stability, and safety in curved sections with particularly small radii of curvature in mining and industrial rail transport under the influence of friction modifiers with varying friction coefficient values.

In contrast to the known computational schemes of dynamic interaction processes between rail tracks and rolling stock [1; 8; 9], this research takes into account the specific features of rail track construction and the character of influence of various specific factors, including friction modifiers, within the zone of contact between the side surfaces of rail heads and wheel flanges. The analysis includes the impact of spatial oscillations of vehicles and rail tracks with arbitrary volume outlines in both plan and trajectory profiles within curved sections of particularly small radii of curvature (80–400 m) for jointed and jointless track configurations in mining and industrial rail transport. Relative slippage of up to 10% between the contacting side surfaces of rail heads and wheel flanges is considered with taking into account the variation of friction coefficient values from 0.05 to 0.25 between the side surfaces of rail heads and wheel flanges.

To conduct research on wear processes and establish the influence of parameters and characteristics of friction coefficients between the side surfaces of rail heads and wheel flanges, considering slippage of up to 10% within these contact zones, a method of contact interaction investigations is presented. This method takes into account loads and stresses within the contact zone, as well as normal and tangential loads during longitudinal and circumferential mutual slippage with using friction modifiers and considering friction coefficients ranging from 0.05 to 0.25.

3. Theoretical study

To study the movement of rail transport vehicles along a track of arbitrary outline with small radii of curves in the plan, a differential equation for their spatial oscillations was formulated. This equation takes into account the equation of the curved axis of the track and the peculiarities of the friction process in the area of lateral contact between rail heads and wheel flanges by using friction modifiers with low friction coefficients.

The mathematical model is based on classical second-order Lagrange equations with using kinetic and potential energy, as well as energy dissipation functions within the transportation system. It also includes general forces without potential, along with generalized coordinates concerning rails, wheels, wheelsets, carriage, spring beams, side frames, and others. The methodology for researching dynamic processes is grounded on scientific principles outlined in publications by renowned scientists such as V.A. Lazarian, V.F. Ushkalov, M.O. Radchenko, T.F. Mokrii, E.P. Blokhin, V.D. Danovych, S.F. Redko, M.A. Frishman, V.V. Rybkin, V.V. Hovorukha, V.K. Garg, R.V. Dukkipati, and others [1; 14–19].

To research the dynamic processes of the transportation structure, a reference system was chosen following the established scientific works [1; 8; 9]. This system employs a single stationary base coordinate system $O^*\xi\eta\zeta$ and, for each rigid body of the rolling stock, two movable coordinate systems: the natural $OXYZ$ and the body-fixed $CX'Y'Z'$ (CX', CY', CZ' are principal central axes of inertia). All coordinate systems are assumed to be right-handed, with the $O^*\xi$, OX, and OX' axes directed from left to right, while the $O^*\zeta$, OZ, and OZ' axes are directed downward. The axes of the natural coordinate system are oriented along the tangent, normal, and binormal to the rail axis, respectively. The origin O for each rigid body is located at a distance of the traveled path S, in meters, from its initial position at the starting time. The position of the natural coordinate system relative to the fixed base system is characterized by the arc coordinate along the rail axis S, the angles χ in radians and φ_h in radians between the OX and $O^*\xi$ axes, respectively, in the planar and vertical longitudinal planes, as well as the angle θ_h in radians between the OY and $O^*\eta$ axes in the vertical transverse plane. The values of φ_h and θ_h are determined by the elevation of the outer rail h_R, in meters, the parameters of the track, χ, and h_R, which are specified functions of the coordinate S.

The position of each rigid body relative to the base natural coordinate system is described by translational displacements of lateral motion x, in meters, lateral sway y, in meters, and vertical bounce z, in meters, as well as angular wagging displacements ψ, in radians, galloping φ, in radians, and side roll θ, in radians.

A computational scheme of the general view for the rolling stock and rail track was taken from well-known scientific works [1; 8; 9]. The rolling stock is represented by a four-axle freight wagon with a mechanical system consisting of rigid bodies: the body, frames, bogies, wheelsets, etc., interconnected by elastic-dissipative, rigid, or hinge joints in vertical and horizontal planes.

The rail track under each wheel of the transportation vehicle is represented by a series of rigid bodies in the vertical and transverse directions, which model the main components of its parts (rails, fastenings with the sub-rail base and sub-ballast) interconnected by elastic-dissipative linkage nodes. These linkage nodes possess elastic characteristics with a stiffness coefficient and viscous friction.

When denoting the coordinates of the rigid bodies in the considered system, the following indices are introduced: for the spring beams – *be* ($e = 1, 2$ – carriage number in the wagon's motion), for the side frames – *fej* (for the left side of the carriage $j = 1$, for the right side $j = 2$), for the wheelsets – *sei* (I = 1, 2 – wheelset number in the carriage), for the wheels *weij*, and for the rails under the wheels – *reij*.

In the study of the interaction processes between rails, intermediate rail fastenings, and wheel flanges of the running gear of moving transportation vehicles in curved sections with small radii of curvature (80–650 m), significant importance lies in considering the mutual friction between the side surfaces of rail heads and the side surfaces of wheel flanges since transverse loads in small-radius curves are generated by extremely large centrifugal and inertial destructive forces. The friction coefficients for sliding and slipping in dry conditions fall within the parameter range of 0.25–0.4. These friction coefficient values are influenced by synthetic materials based on oil or friction modifiers.

To investigate these factors, external contact tangential forces are considered in the research method. Sliding forces ε_{nj} are determined based on the creep hypothesis from equation [1; 14–19]

$$F_{nj} = -f_{nj}\varepsilon_{nj}\left[\left(\frac{f_{nj}\varepsilon_{nj}}{k_f P_{nj}}\right)^2 + 1\right]^{-1/2}, \tag{1}$$

and the component of pseudo-sliding forces X_{nj}, Y_{nj} in the longitudinal and transverse directions as

$$X_{nj} = F_{nj}\frac{\varepsilon_{xnj}}{\varepsilon_{nj}};\; Y_{nj} = F_{nj}\frac{\varepsilon_{ynj}}{\varepsilon_{nj}}. \tag{2}$$

In these expressions:

$$\varepsilon_{xnj} = (-1)^{n+1}\frac{d_1}{\upsilon}(\dot{\chi}_n + \dot{\psi}_n) - \frac{\Delta r_{nj}}{r} + \frac{\dot{x}_n - \dot{\chi}_n y_n}{\upsilon};$$

$$\varepsilon_{ynj} = \frac{1}{\upsilon}(\dot{y}^{*}_{nj} - \psi_{nj}) + \frac{\dot{\chi}_n \dot{x}_n - (-1)^n l \dot{\chi}_n}{\upsilon}; \quad \varepsilon_{nj} = (\varepsilon^2_{xnj} + \varepsilon^2_{ynj})^{1/2}, \tag{3}$$

where $\dot{y}^{*}_{nj} = \dot{y}_{nj} - r\dot{\vartheta}^{*}_{nj} - \dot{y}_{njp}$ is the relative velocity of the wheel with respect to the rail at the contact point; f_{nj} is the coefficient of pseudo-sliding.

During the movement of transport vehicles along the rail track in curved sections with small radii, the radii of wheel rolling circles Δr_{nj} is increased due to lateral movements of the wheel's conical rims relative to the rail heads. Its value is determined by the formula:

$$\Delta r_{nj} = (-1)^n \mu \, y^{*}_{nj} + 0{,}007 \operatorname{abs}(r_{nj})^{n2} n_1 \sigma_{nj}, \quad n = 1, 2, \tag{4}$$

where μ is the conicity of the wheel rolling surface; n_1, n_2 are constants characterizing the degree of wear of wheel rims;

$$r_{nj} = 1000\left[y^{*}_{nj} - (-1)^j \delta \operatorname{sgn} y^{*}_{nj}\right]$$

$$\sigma_{n1} = \begin{cases} 1 & \text{ïðè} \quad r_{n_1} > 0; \\ 0 & \text{ïðè} \quad r_{n_1} \le 0, \end{cases} \qquad \delta_{n2} = \begin{cases} 1 & \text{ïðè} \quad r_{n_2} > 0; \\ 0 & \text{ïðè} \quad r_{n_2} \le 0, \end{cases} \qquad \Delta y_i = y_i - y_{oy}, \tag{5}$$

where δ is the gap in the inter-rail track within the linear portion of the wheel profile, mm.

The change in the wheel rolling surface radii on the rail head can also be found using the equation:

$$\Delta r_{nj} = \frac{(-1)^j}{2}(\mu_1 + \mu_2) y^{*}_{nj} + \frac{1}{2}(\mu_2 - \mu_1) \left\{ \sqrt{\left[(-1)^{j} y^{*}_{nj} - \delta_0\right]^2 + C_o} - \sqrt{\delta_0^2 + C_0} \right\}, \tag{6}$$

where $\mu_1, \mu_2, \delta_o, C_o$ are constants characterizing the wheel profile.

The lateral components of the forces acting on the wheels due to the curved profile of the wheel are determined by the formula:

$$W_{nj} = -P_{nj} \mu_{nj}, \tag{7}$$

where

$$\mu_{nj} = \frac{dr_{nj}}{dy_{nj}} = (-1)^j \frac{\mu_1 + \mu_2}{2} + \frac{\mu_2 - \mu_1}{2} \frac{\left[(-1)^j y_{nj} - \delta_o\right](-1)^j}{\sqrt{\left[(-1)^j y_{nj} - \delta_o\right]^2 + C_o}}.$$

In accordance with the scientific research [6; 7; 13–17], it is assumed that the tangential forces of interaction between the contact surfaces of wheel treads and rail heads are creep forces.

The creep factor ε is determined by the components ε_α and ε_ψ, whose direction depends on two angles: the inclination of the tangent at the contact point on the wheel to the horizon α (in radians) and the approach angle of the wheelset to the rail ψ_s, i.e.,

$$\varepsilon = \left(\varepsilon_\psi^2 + \varepsilon_\alpha^2\right)^{1/2} = \left[\left(\varepsilon_x \sec\psi_s\right)^2 + \left(\varepsilon_y \sec\alpha\right)^2\right]^{1/2}, \tag{8}$$

where $\varepsilon_\alpha, \varepsilon_\psi$ are dimensionless creep factors in the direction of the tangent at the contact point on the wheel to the horizon α and the direction of the wheelset approach to the rail ψ_s, respectively; $\varepsilon_x, \varepsilon_y$ are dimensionless creep factors in the longitudinal and transverse directions.

The components of creep forces *F*, kN, are determined by the formulas:

$$F_\alpha = \frac{\varepsilon_\alpha}{\varepsilon} F, \qquad F_\psi = \frac{\varepsilon_\psi}{\varepsilon} F. \tag{9}$$

N, kN, is the normal pressure of the wheel on the rail, which is given by:

$$N = S_z \sec\alpha, \tag{10}$$

where S_z is the vertical interaction force, kN, determined as the reaction of the rail:

$$S_z = m_{rz}\ddot{z}_r + k_{rz}\left(\gamma\dot{z}_r + z_r\right), \tag{11}$$

where z_r is the vertical displacement of the rail at the contact point with the wheel, m; m_{rz} is the mass of the rail, t, converted to one wheel in the vertical direction; k_{rz} is the corresponding vertical stiffness of the rail, kN/m; γ is the coefficient characterizing energy dissipation in the rail path.

In the phase of two-point contact between the wheel and the rail, the force S_z is distributed between two points, which is taken into account when solving the system of algebraic equations [6; 7; 16–18]. The geometric parameters of the contact depend on the relative transverse displacements of the bodies in the contact pair Δy, m, and the angle of wheelset lateral sway θ, rad. The output functions are those, which describe the profiles of the wheel rim $f_W(y)$ and rail head $f_R(y)$ surfaces.

The vector components of creep forces are represented as follows:

$$\vec{F}_\psi = \begin{cases} \vec{F}_\psi & \text{in the single-point contact phase,} \\ \vec{F}_\psi^{(1)} + \vec{F}_\psi^{(2)} & \text{in the two-point contact phase.} \end{cases}$$
$$\vec{F}_\alpha = \begin{cases} \vec{F}_\alpha & \text{in the single-point contact phase,} \\ \vec{F}_\alpha^{(1)} + \vec{F}_\alpha^{(2)} & \text{in the two-point contact phase.} \end{cases} \qquad (12)$$

The calculated disturbances in the vertical and horizontal directions are modeled in accordance with experimental studies under operational conditions [1; 10–13] and are represented by the formula:

$$\eta_i(x) = \frac{a_i}{2}\left(1 - \cos\frac{2 \cdot \pi \cdot x}{L_i}\right) \qquad (13)$$

where a_i is amplitude of the irregularity, m; L_i is total length of the irregularity, m; x is distance from the beginning of the irregularity to the ordinate $\eta_i(x)$, m; i is direction of the action of the contact disturbance (z, y).

The components of mechanical and geometric features of the railway track and the running gear of moving vehicles form methods that are considered in the study of the interaction processes of intermediate fastening, rails, and wheelsets in curved sections of railway transport with small radii of curvature.

Mathematical models of the movement of the transportation system are obtained using the second-order Lagrangian equation [1; 6; 7; 14–19]:

$$\frac{d}{dt}\frac{\partial \mathrm{T}}{\partial \dot{q}_v} - \frac{\partial \mathrm{T}}{\partial q_v} + \frac{\partial \Phi}{\partial \dot{q}_v} + \frac{\partial \Pi}{\partial q_v} = Q_v \ \ (v = 1,2,...,n) \qquad (14)$$

where q_v is generalized coordinates; Q_v is corresponding generalized forces; T and Π are kinetic and potential energies; Φ is energy dissipation function; n is number of freedom degrees of the system.

During the selection of generalized coordinates, the following constraints, imposed on the "wagon-truck" system due to structural features and assumptions, were taken into account:

– the side bearers move together with the body's pedestals during oscillations of rocking, lateral shifting, pitching, and galloping;

– the longitudinal movements of the side frames are determined by the movements of the side bearers, assuming the absence of longitudinal clearances between these bodies.

The kinetic and potential energies of the "wagon-truck" system are equal to the sum of the respective energies of the wagon and the track. The energy dissipation function in the system is calculated analogously.

According to the known Koenig theory, the kinetic energy of a vehicle is equal to the sum of the energies of all its solid components. The kinetic energy of each solid body is calculated as the sum of two components: the energy K_c of translational motion, which is determined by the motion of the center of mass,

$$K_c = \frac{1}{2}m\left(\dot{\xi}^2 + \dot{\eta}^2 + \dot{\zeta}^2\right) = \frac{1}{2}m\left[\left(\dot{S} + \dot{x} - y\dot{\chi}\right)^2 + \left(\dot{y} + x\dot{\chi}\right)^2 + \left(\dot{z} - \frac{1}{2}\dot{h}_R\right)^2\right], \quad (15)$$

and the energy $\hat{E}_\Omega$ of rotational motion relative to the center of mass

$$K_\Omega = \frac{1}{2}\left(I_x\omega_{x'}^2 + I_y\omega_{y'}^2 + I_z\omega_{z'}^2\right) =$$

$$= \frac{1}{2}I_x\left(\dot{\theta} + \dot{\theta}_h\right)^2 + \frac{1}{2}I_y\left(\dot{\varphi} + \dot{\varphi}_h\right)^2 + \frac{1}{2}I_z\left(\dot{\psi} + \dot{\chi}\right)^2, \quad (16)$$

where m is the mass of the considered body; I_x, I_y, I_z represent its principal moments of inertia; $\omega_{x'}, \omega_{y'}, \omega_{z'}$ are the angular velocities associated with the body-fixed coordinate system.

The potential energy of the wagon Π is equal to the sum of the energies of elastic deformations Π_1 and changes in energy Π_2 due to raising or lowering the centers of gravity of the bodies that make up the system. The potential energy Π_1 is determined using the Clapeyron theorem and is a quadratic function of the compressions of the elastic elements Δ:

$$\Pi_1 = \frac{1}{2}\sum_{\sigma=1}^{n1}\left(k_{\sigma z}\Delta_{\sigma z}^2 + k_{\sigma y}\Delta_{\sigma y}^2 + k_{\sigma x}\Delta_{\sigma x}^2 + k_{\sigma\psi}\Delta_{\sigma\psi}^2\right), \quad (17)$$

where $n1$ is the number of elastic elements in the system; $k_{\sigma z}, k_{\sigma y}, k_{\sigma x}$ are the stiffness of the σ-th elastic element in the vertical, horizontal transverse, and longitudinal directions, respectively; $k_{\sigma\psi}$ is the angular stiffness of the σ-th elastic element in the plane; $\Delta_{\sigma z}, \Delta_{\sigma y}, \Delta_{\sigma x}$ are the deformations of the σ-th elastic element in the vertical, horizontal transverse, and longitudinal directions, respectively; $\Delta_{\sigma\psi}$ is the angular deformation of the element in the plane.

The potential Π_2 is determined by taking into account the rise or fall of the centers of gravity of bodies during their lateral displacement. For each solid body in the system:

$$\Pi_2 = -mg(z + \theta_h y), \tag{18}$$

where g is the acceleration due to gravity.

The energy dissipation function of the transport system is expressed as:

$$\begin{aligned} \Phi = \frac{1}{2}\sum_{\sigma=1}^{n2}\left(\beta_{\sigma z}\dot{\Delta}_{\sigma z}^2 + \beta_{\sigma y}\dot{\Delta}_{\sigma y}^2 + \beta_{\sigma x}\dot{\Delta}_{\sigma x}^2 + \beta_{\sigma\psi}\dot{\Delta}_{\sigma\psi}^2\right) + \\ + \sum_{\sigma=1}^{n3}\left(F_{\sigma z}\left|\dot{\Delta}_{\sigma z}\right| + F_{\sigma y}\left|\dot{\Delta}_{\sigma y}\right| + F_{\sigma x}\left|\dot{\Delta}_{\sigma x}\right| + F_{\sigma\psi}\left|\dot{\Delta}_{\sigma\psi}\right|\right), \end{aligned} \tag{19}$$

where β is the coefficients of energy dissipation in viscous dampers; F is amplitude values of dry friction forces; $n2, n3$ are the number of viscous and dry friction dampers in the system, respectively.

The formula for calculating the kinetic energy of the railway track is given as:

$$K_r = \frac{1}{2}m_{rz}\sum_{e=1}^{2}\sum_{i=1}^{2}\sum_{j=1}^{2}\dot{z}_{reij}^2 + \frac{1}{2}m_{ry}\sum_{e=1}^{2}\sum_{i=1}^{2}\sum_{j=1}^{2}\dot{y}_{reij}^2. \tag{20}$$

The potential energy of the track is calculated as:

$$\Pi_r = \frac{1}{2}k_{rz}\sum_{e=1}^{2}\sum_{i=1}^{2}\sum_{j=1}^{2}z_{reij}^2 + \frac{1}{2}k_{ry}\sum_{e=1}^{2}\sum_{i=1}^{2}\sum_{j=1}^{2}y_{reij}^2. \tag{21}$$

The energy dissipation function in the track is represented as:

$$\Phi_r = \frac{1}{2}\beta_{rz}\sum_{e=1}^{2}\sum_{i=1}^{2}\sum_{j=1}^{2}\dot{z}_{reij}^2 + \frac{1}{2}\beta_{ry}\sum_{e=1}^{2}\sum_{i=1}^{2}\sum_{j=1}^{2}\dot{y}_{reij}^2, \tag{22}$$

where β_{rz}, β_{ry} are the coefficients of viscous resistance of the track, reduced to one wheel.

The generalized forces Q_v for the considered transport system can be expressed as the sum of generalized forces Q_v^t with no potential and generalized forces Q_v^{ni} that account non-inertial effects of the introduced reference systems.

The forces Q_v^t are equal to the components of the tangential interaction forces acting on the contact between wheels and rails, taking into account the possible phases of contact.

The forces Q_v^{ni} are determined by the formula:

$$Q_\text{v}^{ni} = \frac{d}{dt}\left(\frac{\partial K}{\partial \dot{q}_\text{v}}\right) - \frac{\partial K}{\partial q_\text{v}} = Q_\text{v}^{\hat{\imath}\grave{o}\acute{\imath}} + Q_\text{v}^{\hat{\imath}} + Q_\text{v}^{\omega} + Q_\text{v}^{\dot{\omega}} + Q_\text{v}^{\hat{e}op}, \tag{23}$$

where $Q_v^{\hat{\imath}\grave{o}\acute{\imath}} = f(\ddot{q}_v)$ is the generalized inertia force due to relative motion; $Q_v^{\hat{\imath}} = f(\ddot{S}, \dot{S}\dot{\chi}, \ddot{h}_R)$ is the generalized inertia force due to the translational motion of the bodies under consideration along with the reference systems *OXYZ*; $Q_v^{\omega} = f(q_v \dot{\chi}^2)$ is the generalized centrifugal force of inertia, which takes into account the translational rotation of the bodies together with the reference systems *OXYZ* around point *O*; $Q_v^{\dot{\omega}} = f(\ddot{\chi}, q_v \ddot{\chi}, \ddot{\theta}_h, \ddot{\varphi}_h)$ is the generalized inertia force associated with rotational motion, which accounts for the non-constant angular velocity of translational rotation of the bodies with the reference systems *OXYZ* around point *O* along a transitional; $Q_v^{\hat{e}op} = f(\dot{q}_v \dot{\chi})$ is the Coriolis generalized inertia force, which considers the mutual influence of relative motion and translational rotation of the bodies.

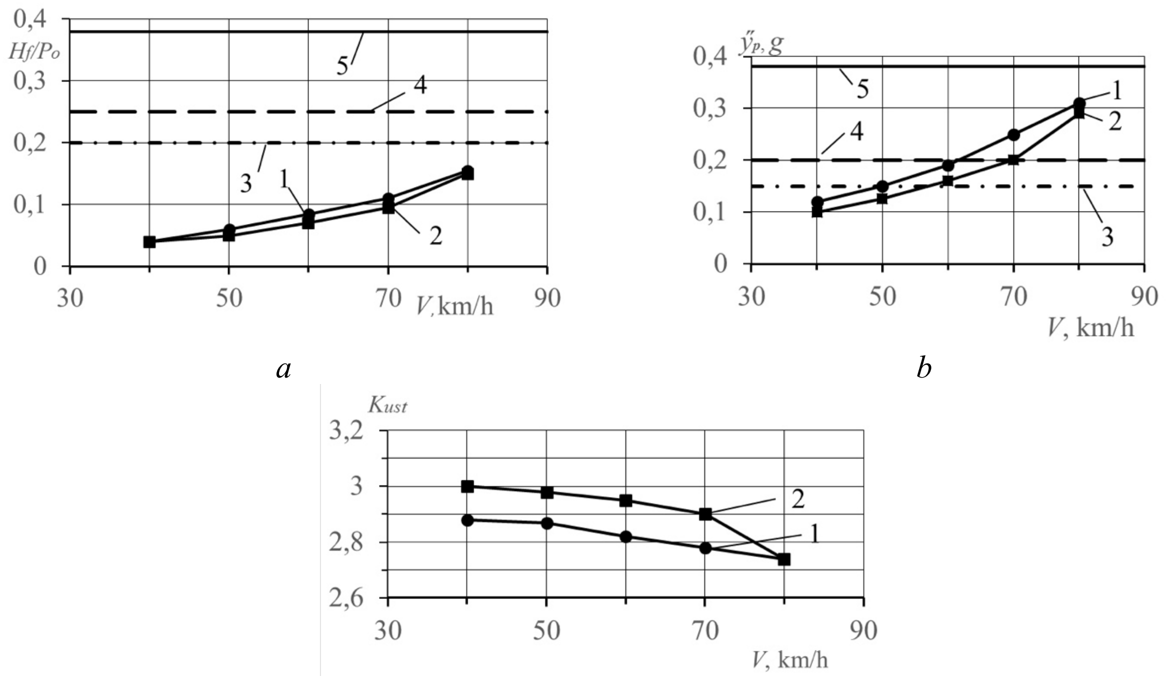
The presented equations (1–23) constitute an improved mathematical model for studying the interaction processes between railway tracks and the wheelsets of rolling stock. Based on this mathematical model, software was developed to perform mathematical simulations enabling the research for assessing dynamic performance indicators for the movement of the transport vehicle in curved sections with small radii of curvature. This research encompasses such factors as load-bearing capacity, oscillations, stability during motion, the magnitude of friction forces at points of contact, as well as evaluating the wear intensity of the contact pair between the rail head and the wheel flange during relative slippage of up to 10%. Additionally, the software allows considering friction modifiers with varying coefficient values between them [22].

4. Results and Discussion

The research findings include the fundamental relationships for maximum values of dynamic load indicators between the side surfaces of rail heads and the wheel flanges of moving transport vehicles and the rail track in both the vertical and horizontal directions within considered speed range of 40–80 km/h. These relationships take into account the influence of the magnitude of joint irregularities located in curved sections with small radii of curvature.

The parameters of joint irregularities were determined based on the results of experimental studies conducted on rail track sections with small radii of curvature [1; 11–13]. The length of the joint irregularities in the vertical direction was taken as L_z = 3 m. The length of the joint irregularities in the horizontal and transverse directions was set to L_y = 6 m. The depth of the joint irregularities varied depending on the different levels of tonnage, ranging from 1 to 8 mm in the vertical direction and 1 to 4 mm in the horizontal (transverse) direction. An increase in the outer rail relative to the inner rail by 80 mm was adopted in accordance with the regulatory materials [20–21].

Figures 1 and 2 depict the dependencies of the maximum dynamic performance indicators of freight vehicles on the speed of movement in the range of 30–80 km/h in curved sections of the rail track with small radii of curvature. Vertical irregularities are shown with magnitudes of 4 mm and 8 mm, and horizontal irregularities with magnitudes of 2 mm and 4 mm. The conditional notations on the figures are as follows: Line 1 represents the presence of rail joints, while Line 2 represents the absence of rail joints, Lines 3, 4, 5 represent the boundary normative values of the research indicators corresponding to "excellent", "good", and "acceptable" freight vehicle movement according to regulatory materials [20–21]. The permissible value of the safety margin coefficient for the vehicle's stability against derailing K_{ust} according to regulatory materials [20–21] should not exceed 1.4.



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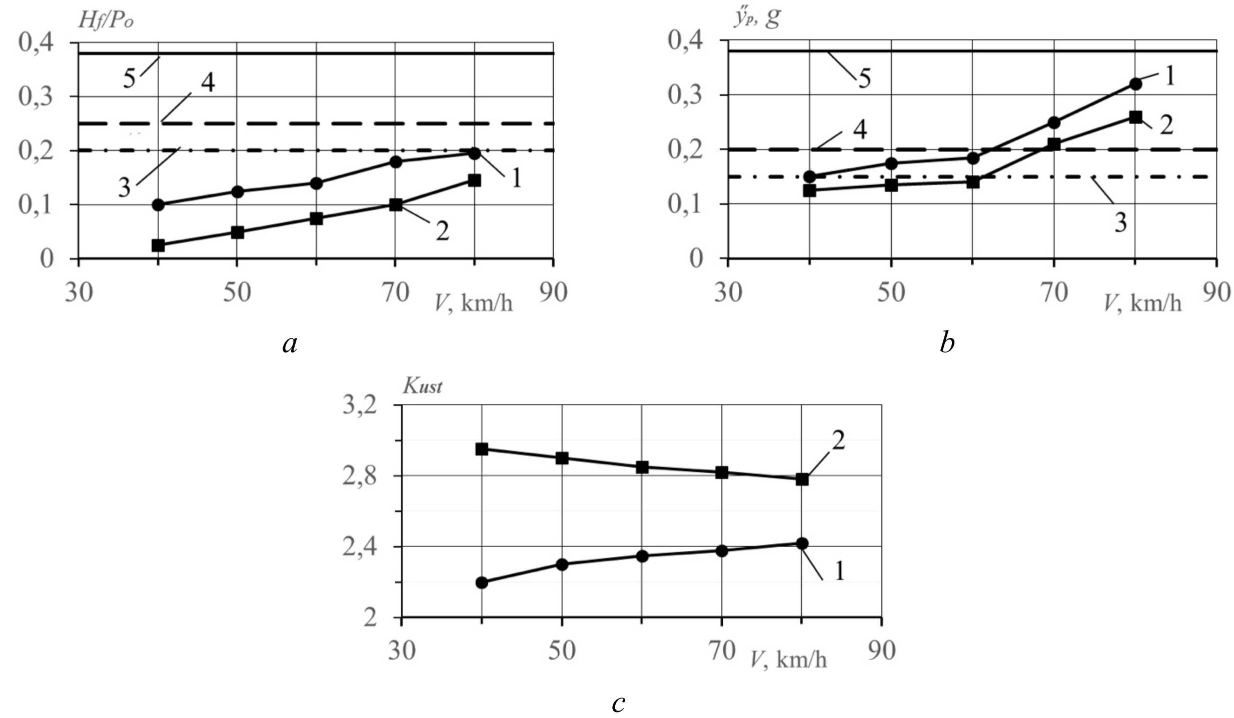
1 – with rail joints present; 2 – without rail joints; 3, 4, 5 – boundary normative values of indicators, respectively, for "excellent", "good" and "acceptable" movement of freight vehicles

a – frame forces as a fraction of static axial load; *b* – horizontal acceleration of the body heels in fractions of gravitational acceleration; *c* – stability margin coefficient against wheel derailment

Figure 1 – Dependencies of the maximum dynamic performance indicators of a freight transport vehicle on the speed of movement along a curve of small radius with vertical irregularities of 4 mm and horizontal irregularities of 2 mm

The analysis of dynamic characteristics of the moving vehicle was carried out for the horizontal (lateral) direction, which is the most critical for curved sections where wear of the side surfaces of the rail heads and wheel flanges is intensive. The analysis is based on frame forces H_f as a fraction of the static axial load P_o, horizontal accelerations of the body heels $\ddot{y}_p$ in fractions of gravitational acceleration g, as well as the safety factor against wheel derailment K_{ust}.



1 – with rail joints present; 2 – without rail joints; 3, 4, 5 – boundary normative values of indicators, respectively, for "excellent", "good" and "acceptable" movement of freight vehicles

a – frame forces as a fraction of static axial load; *b* – horizontal acceleration of the body heels in fractions of gravitational acceleration; *c* – stability margin coefficient against wheel derailment

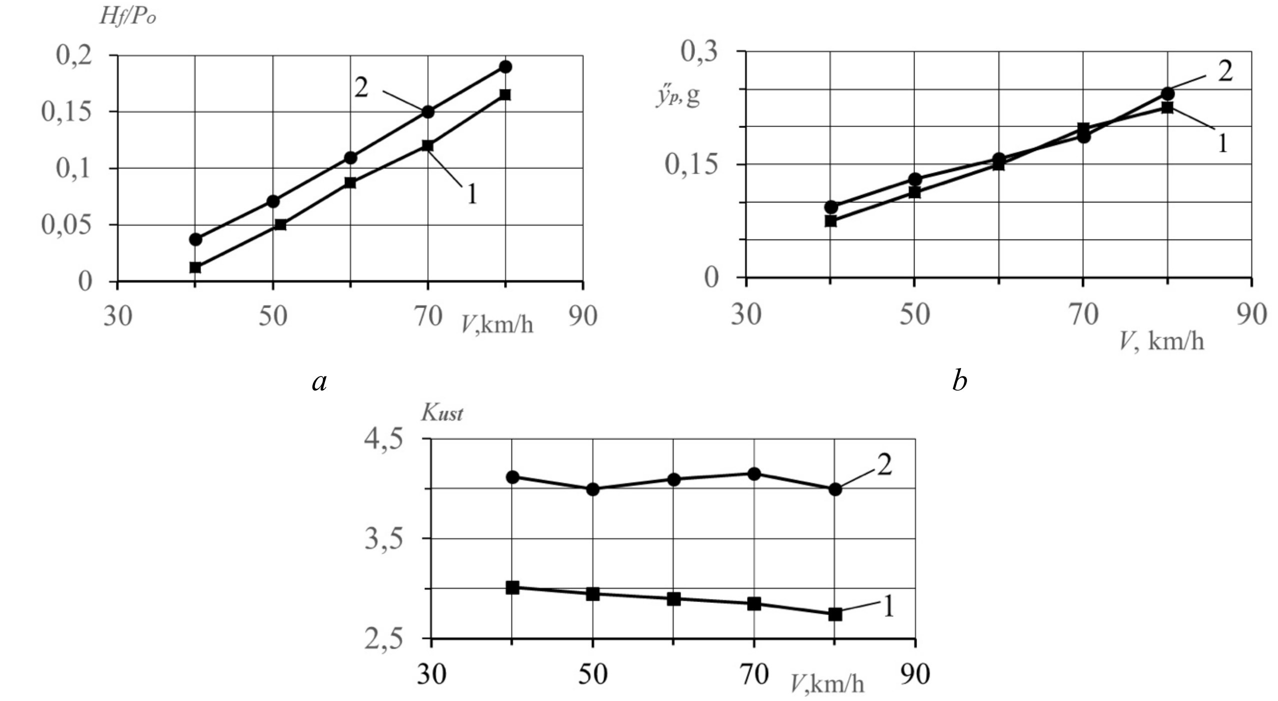
Figure 2 – Dependencies of the maximum dynamic performance indicators of a freight transport vehicle on the speed of movement along a curve of small radius with vertical irregularities of 8 mm and horizontal irregularities of 4 mm

The analysis of the dynamic performance indicators of the moving vehicle shown in Figures 1 and 2 reveals that the frame forces H_f and the horizontal accelerations of the body heels $\ddot{y}_p$ in curved sections with small radii decrease by 15–56% for the non-jointed track compared to the jointed track. Additionally, the safety factor against wheel derailment K_{ust} increases by 19–26%.

It is determined that the magnitude of side forces on the outer rails and the dynamics coefficient in curved sections with jointless rail track are 20–25% lower than that for the track with connection joints of the jointed rail.

Increasing the speed of vehicles from 30 to 80 km/h leads to an increase in the load on the outer rail in small radius curves by 45–50% for jointed track and by 40% for jointless track.

The research was conducted to investigate the influence of friction modifiers with different friction coefficient values on the dynamic load indicators and wear intensity indicators of the side surfaces of rail heads and wheel flanges in curved sections with small radii at a relative slip of 10%. Figure 3 shows the dependencies of the maximum dynamic load indicators of the freight transport vehicle on the speed of movement in the range of 30–80 km/h along a curve with a small radius for different values of the friction coefficient (μ) between the side surfaces of the rail heads and the wheel flanges in the range from 0.25 to 0.05. The figure presents the values of friction coefficients: 1 – μ=0.25; 2 – μ=0.05.



c

1 – friction coefficient μ=0.25; 2 – friction coefficient μ=0.05

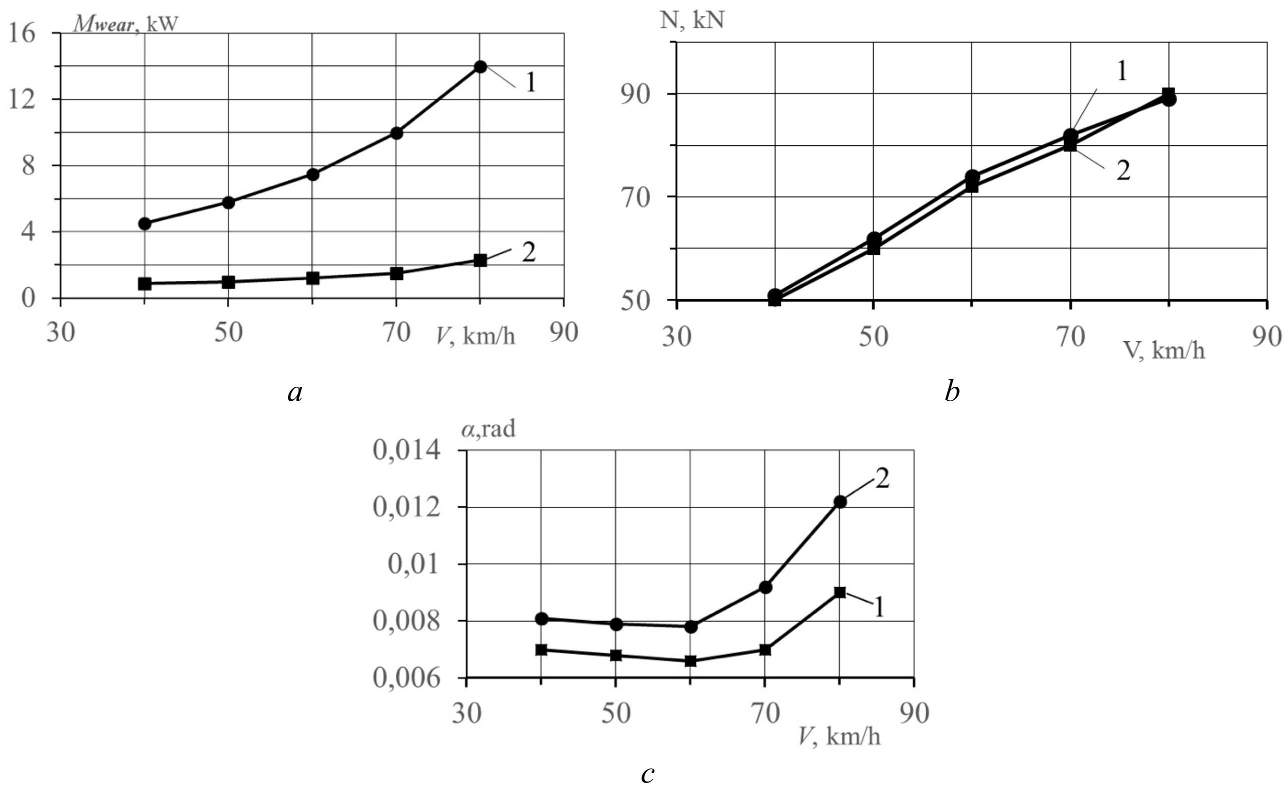
a – frame forces as a fraction of static axial load; *b* – horizontal acceleration of the body heels in fractions of gravitational acceleration; *c* – stability margin coefficient against wheel derailment

Figure 3 – Dependencies of the maximum dynamic load indicators of the vehicle on the speed of movement along a curve with a small radius for different values of the friction coefficient between the side surfaces of the rail head and the wheel flange

From the research results shown in Figure 3, it is established that in curves with small radii, as the friction coefficient decreases in the range from 0.25 to 0.05, the magnitude of dynamic loads on the outer rail decreases, and the stability margin coefficient against wheel derailment increases by 1.3–1.5 times.

As the speed of vehicles increases from 30 km/h to 80 km/h, the intensity of the decrease (deterioration) in the stability margin coefficient against wheel derailment is twice as high with a dry friction coefficient between the contact side surfaces μ=0.25 in comparison to the intensity of the decrease in the stability reserve coefficient against wheel derailment when using friction modifiers with the friction coefficient of 0.05. The use of friction modifiers improves by a factor of two the quality of dynamic processes of the moving transport unit in curves with small radii when the speed increases from 30 to 80 km/h.

Figure 4 depicts the dependencies of the maximum values of dynamic load indicators and wear intensity of the contact side surfaces of rail heads and wheel flanges as a function of speed and the friction coefficient in the range from 0.25 to 0.05 when moving in the curved section of track with a small radius. The wear intensity indicators include the wheel pair attack angle α (in radians) on the outer rail and the friction force power M_{wear} (B_m) (scalar multiplication of creep forces and wheel velocity changes relative to the rail during slip) at the point of contact between the side surface of the rail head and the wheel flange.



1 – friction coefficient μ=0.25; 2 – friction coefficient μ=0.05

a – friction force power at contact points; *b* – guiding force acting on the wheel from the rail; *c* – wheel pair attack angle on the outer rail

Figure 4 – Dependency of the maximum wear factor values for the side surfaces of rail heads and wheel flanges on the speed of movement of a freight vehicle on a small radius curve at different friction coefficients between the contact surfaces

The research results show a significant impact of reducing the friction coefficient on the formation of wear intensity indicators of the contact side surfaces of rail heads and wheel flanges based on the values of friction force power at the contact points M_{wear} (Figure 4, *a*) and the wheel pair attack angle on the outer rail α (Figure 4, *b*). Therefore, decreasing the friction coefficient μ from 0.25 (with dry friction) to 0.05

(when using friction modifiers) leads to a 3.5–4.5 times reduction in wear intensity indicators for the side surfaces of rail heads and wheel flanges.

The prospects for further development in this direction include advancements in creating new friction modifiers for the repair and restoration compound called "Ideal" [22, 23]. These modifiers are designed with a friction coefficient of μ=0.005 (close to zero) under specific conditions, such as a specific pressure range of 187–374 MPa, which is found in gearboxes, machinery, and mechanisms. Additionally, they are intended to maintain a similar friction coefficient of μ=0.04 under a specific pressure of 529 MPa corresponding to the interaction between the side surfaces of rail heads and wheel flanges with a relative slippage of 10% in curved sections with small radii.

The use of these new friction modifiers of the repair and restoration compound "Ideal" [22, 23] is expected to significantly improve the quality of dynamic processes in terms of normal, tangential, and longitudinal forces in the investigated transportation system. This improvement should also increase the service life, stability, and safety of transportation, while reducing the wear of the side surfaces of rail heads and wheel flanges in rail transport systems for mining and industrial enterprises.

5. Conclusions

1. An improved spatial mathematical model of the interaction between mining and industrial transport vehicles and rail tracks in curved sections with extremely small radii of curvature was developed. This model allows assessing the impact of friction modifiers on extending the service life, stability, and safety of movement by enhancing the quality of dynamic performance indicators, including normal, tangential, and longitudinal interaction forces of the transport system when slip reaches up to 10%. Additionally, it contributes to reducing the intensity of wear on the side surfaces of rail heads and wheel flanges.

2. It is found that dynamic loads, including lateral cross forces and horizontal accelerations of the transport vehicle in curved sections with small radii of curvature, are reduced by 15–56% when using continuous track without rail joints compared to the presence of rail joints. Additionally, the stability margin against wheel derailment increases by 19–26%.

3. The dependencies of the maximum values of the dynamics quality indicators of the transport vehicle (including lateral cross forces and horizontal accelerations) concerning small radius curves in relation to the speed of movement when using friction modifiers are established. When reducing the coefficient of friction from $\mu = 0{,}25$ (dry friction) to $\mu = 0{,}05$ (using friction modifiers), the magnitude of dynamic lateral loads on the rail decreases by 1.3–1.5 times.

4. It is established that the use of friction modifiers $\mu = 0{,}05$ in small radius curves, with an increase in the speed of transport vehicles from 30 to 80 km/h, leads to an improvement in the stability margin against wheel derailment by a factor of 2 compared to the presence of dry friction with a coefficient of friction of $\mu = 0{,}25$.

5. It is found that reducing the friction coefficient values in the range from $\mu = 0{,}25$ (dry friction) to $\mu = 0{,}05$ (using friction modifiers) leads to a decrease in

the wear rates of the side surfaces of rail heads and wheel flanges by 3.5–4.5 times and an increase in the stability margin against wheel derailment by 1.3–1.5 times.

6. The prospects for further development in this direction are associated with the use of friction modifiers in the repair and restoration compound "Ideal" with friction coefficient values $\mu = 0{,}005 - 0{,}04$ corresponding to specific pressures in the contact zone of 187–374 MPa and 529 MPa, as well as new technologies in mining and industrial rail transport [22, 23].

REFERENCES

1. Govorukha, V.V. (1992), *Fiziko-tekhnicheskie osnovy sozdaniya elementov relsovogo transporta shakht i karyerov: monograph* [Physical and technical basis for the creation of elements of rail transport of mines and quarries], Naukova dumka, Kyiv, Ukraine.

2. Chekhlatyy, N.A., Koval, A.N. and Myalkovskiy, V.I. (2012), “The main directions for improving the efficiency and safety of operating underground stationary installations in mines”, *Proc. of the International scientific conference “Forum of Mining Engineers”*, National Mining University, Dnepropetrovsk, pp. 101–107, https://ir.nmu.org.ua/handle/123456789/150426.

3. Darenskyi, O.M. (2011), *Teoretychni i eksperymentalni doslidzhennia roboty zaliznychnykh kolii promyslovoho transport: monograph* [Theoretical and experimental studies of the operation of railway tracks for industrial transport], UkrDAZT, Kharkiv, Ukraine.

4. Darenskyi, O.M. (2010), “Conditions of contact between wheels and rails in the horizontal plane”, *Sylovi i kinematychni zviazky ekipazhu i kolis*, vol. 113, pp. 171–177.

5. Darenskyi, A.N. and Klimenko, A.V. (2013), “Modelling of interaction between track and rolling stock at discrete under-rail base in the zone of rail joints”, *Informatsiino-keruiuchi systemy na zaliznychnomu transporti*, no. 4, pp. 15–22.

6. Ushkalov, V.F. and Bezrukavyi, N.V. (2015), “Increased wheel set service life due to the use of wear-resistant wheel profiles”, *Tekhnicheskaya mekhanika*, no. 1, pp. 97–103.

7. Ushkalov, V.F., Mokryi, T.F. and Malysheva, I.Yu. (2017), “Influence of changes in the shape of wheel profiles during operation on the dynamic qualities of freight wagons and indicators of their interaction with the rail track”, *Tekhnicheskaya mekhanika*, no. 1, pp. 65–71.

8. Hovorukha, V. (2020), “Movement stability of vehicles with a rigid and free wheels mounting on wheelset axles”, *International Conference Essays of Mining Science and Practice*, vol. 109, 00029, https://doi.org/10.1051/e3sconf/201910900029.

9. Hovorukha, V., Hovorukha, A. and Makaroz, Yu. (2022), “Research on the dynamic processes of vehicles and an arbitrary configuration rail track, influencing the side wear of the rail head and wheel flange contact surfaces at different values of friction coefficient between them”, *IOP Conf. Ser.: Earth Environ. Sci.*, vol. 970, 012029, https://doi.org/10.1088/1755-1315/970/1/012029.

10. Rybkin, V.V., Nastechik, M.P. and Arbuzov, M.A. (2013), “Investigation of the influence of track construction parameters on the intensity of side wear of rail heads in curved sections”, *Zbirnyk naukovykh prats Don IZT*, vol. 34, pp. 155–165.

11. Makarov, Yu.O. (2019), “Problems of excessive side wear of rails in sections with a difficult operating conditions”, *Zaliznychnyi transport Ukrainy*, no. 3, pp. 39–45.

12. Govorukha, V.V. and Makarov, Yu.O. (2017), “Research of side wear of rails at mechanical interaction of track and wheels of the running part of wagons on curvilinear sections of track”, *Geo-Technical Mechanics*, vol. 134, pp. 125–140. https://doi.org/10.15407/geotm2017.134.125

13. Hovorukha, V.V., Makarov, Yu.O. and Kizilov, V.K. (2019), “Study of rail head side wear in curved track sections with particularly small radii of curvature”, *Geo-Technical Mechanics*, vol. 149, pp. 111–121. https://doi.org/10.15407/geotm2019.149.111

14. Lazaryan, V.A. (1985), *Dinamika transportnykh sredstv* [Dynamics of vehicles], Naukova dumka, Kyiv, Ukraine.

15. Radchenko, N.A. (1988), *Krivolineynoye dvizheniye relsovykh transportnykh sredstv* [Curvilinear motion of rail vehicles], Naukova dumka, Kyiv, Ukraine.

16. Ushkalov, V.F., Mokryi, T.F. and Malysheva, I.Yu. (2015), “Predicting changes in the profile of railway carriage wheels due to wear during operation”, *Tekhnicheskaya mekhanika*, no. 4, pp. 148–154.

17. Govorukha, V.V. (2006), *Mehanika vzaimodeystviya relsovogo puti, podvizhnyih transportnyih sredstv i smezhnyih ustroystv* [Mechanics of interaction between railway track, rolling stock, and adjacent devices], Lira, Dnepropetrovsk, Ukraine.

18. Garg, V.K. and Dukkipati, R.V. (1988), *Dinamika podvizhnogo sostava* [Rolling stock dynamics], Transport, Moscow, Russia.

19. Hovorukha, V.V., Hovorukha, A.V., Sobko, T.P. and Semyditna, L.P. (2022), “Study method of spatial vibrations of a vehicle during its movement along the rail track on separate supports with elastic-dissipative and inertial properties”, *Geo-Technical mechanics*, vol. 162, pp. 13–28. https://doi.org/10.15407/geotm2022.162.013

20. Danilenko, E.I., Orlovskyi, A.M., Kurhan, V.O. and Yakovliev, V.O. (2012), *Instruktsiia z ulashtuvannia ta utrymannia kolii zaliznyts Ukrainy* [Instructions for the Construction and Maintenance of Railways in Ukraine], NVP Polihrafservis, Kyiv, Ukraine.

21. GosNIIV-VNIIZhT (1996), *Normy dlya rascheta i proyektirovaniya vagonov zheleznykh dorog MPS kolei 1520 mm (nesamokhodnykh)* [Norms for calculation and design of 1520 mm gauge (non-self-propelled) wagons of MPS railways], GosNIIV-VNIIZhT, Moscow, Russia.

22. Bulat, A.F., Voloshyn, O.I., Hovorukha, V.V. and Fedoseiev, S.V. (2015), *Kontseptsiia, retseptura, sklad, tekhnolohiia vyhotovlennia i zastosuvannia remontno-vidnovliuvalnoi sumishi "Ideal" na osnovi poiednannia trybotekhnichnoho skladu inhrediientiv z vuhletsevovmishchuiuchymy materialamy ta yikh vkraplenniam v bazovi poverkhni tertia v tryboziednanniakh mashyn, mekhanizmiv, ahrehativ i riznykh zasobiv transportu* [Concept, formulation, composition, manufacturing technology, and application of the repair and restorative compound "Ideal" based on the combination of tribotechnical ingredient composition with carbon-containing materials and their embedding into the base friction surfaces in tribological joints of machines, mechanisms, units, and various means of transportation], State Register of Patents of Ukraine, Kiev, UA, Pat. No. 60799.

23. Hovorukha, V.V. and Hovorukha, A.V. (2023), "Improvement of the service life of mining and industrial equipment by using friction modifiers", *Naukovyi Visnyk Natsionalnoho Hirnychoho Universytetu*, no. 4, pp. 74-81, https://doi.org/10.33271/nvngu/2023-4/074

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ПІДВИЩЕННЯ РЕСУРСУ ТА СТІЙКОСТІ РУХУ РЕЙКОВИХ ТРАНСПОРТНИХ ЗАСОБІВ ГІРНИЧОЇ ТА ПРОМИСЛОВОЇ ГАЛУЗІ ЗА ДОПОМОГОЮ МОДИФІКАТОРІВ ТЕРТЯ

Говоруха А.В.

Анотація Наукова робота включає дослідження впливу модифікаторів тертя на зменшення інтенсивності зносу , підвищення ресурсу роботи та безпеки технічних засобів гірничого та промислового рейкового транспорту в особливо складних умовах експлуатації з малими радіусами кривих. Проведено математичне моделювання динамічних процесів взаємодії рейкової колії та рухомих транспортних засобів при просторових коливаннях транспортної системи в вертикальному, поперечному та повздовжньому напрямках. Встановлені залежності показників динаміки транспортних засобів і рейкової колії від швидкості руху при стикових з'єднаннях ланкової колії з нерівностями, а також залежність інтенсивності зносу робочих поверхонь від зміни показників коефіцієнта тертя.

Мета роботи полягає в підвищенні ресурсу, стійкості і безпеки руху технічних засобів гірничого та промислового рейкового транспорту за рахунок зменшення величини коефіцієнта тертя. Методика роботи включає методи математичного та комп'ютерного моделювання динамічних процесів взаємодії гірничих рейкових транспортних засобів та рейкової колії , що впливають на бічний знос головок рейок та реборд бандажів коліс. Одержані результати теоретичних досліджень впливу параметрів коефіцієнтів тертя між боковими поверхнями головок рейок і реборд бандажів коліс на динамічні показники взаємодії транспортних засобів і рейкової колії , а також показники зносу пари «колесо-рейка». Встановлено, що зменшення коефіцієнта тертя від 0,25 (сухе тертя) до 0,05 (використання модифікаторів тертя) приводить до зменшення показників зносу в 3,5-4,5 рази, та збільшення стійкості від сходу коліс з рейок в 1,3-1,5 рази. Одержано, що заміна стикових з'єднань рейок на безстикову колію в кривих малих радіусів забезпечує зниження горизонтальних навантажень на 15-56 %, а коефіцієнт запасу стійкості проти сходу коліс з рейок збільшується на 19-26 %. Результати роботи передбачено використати при впровадженні нових модифікаторів тертя і технологій на гірничому та промисловому рейковому транспорті для підвищення економічної ефективності і ресурсу роботи рухомих транспортних засобів та рейкової колії, покращення безпеки руху та роботи фахівців в рейковому транспорті.

Ключові слова: Гірничий та промисловий рейковий транспорт, ресурс, знос, коефіцієнт тертя, криволінійні ділянки колії.