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ON FORMATION OF KINEMATICAL AND DYNAMICAL PARAMETERS OF OUTPUT ELEMENTS OF THE MINE VEHICLES IN TRANSIENT MOTION

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ПРО ФОРМУВАННЯ КІНЕМАТИЧНИХ І ДИНАМІЧНИХ ПАРАМЕТРІВ ВИХІДНИХ ЛАНОК ШАХТНИХ ТРАНСПОРТНИХ ЗАСОБІВ У ПЕРЕХІДНОМУ РЕЖИМІ РУХУ

Purpose. To determine the parameters of elastic-dissipative connections in the running gear of the mine vehicle ensuring its safety motion through vertical and horizontal irregularities of the rail track under given angular kinematic movability of its output elements.

Methodology. We have formulated a differential equation system using Lagrange equations of the 2nd order, which have been solved in Wolfram Mathematica software after differentiation. The calculated data have been used in order to identify the values of dynamic loadings affecting the output elements of the mine vehicle and for further evaluation of elastic-dissipative connection parameters.

Findings. The mathematical simulation and analysis of dynamic processes of mechanical system "mine vehicle – railroad" has been carried out. The values of dynamic load of output elements of the running gear (on the example of mine cart ВГ-3.3-900 with movable and resilient mounting of wheel) have been determined. The received data became initial data for calculation of the elastic-dissipative connection "wheel centre - tread". We have determined that the additional kinematic wheel movability provided by the elastic-dissipative properties is one of the main factors that influence the safety factor at curvatures.

Originality. We have considered the elastic-dissipative connection of the output elements of the mine vehicle influence on the safety factor and exploitation safety of rolling stock in transient motion.

Practical value. The research results provide the possibility of calculation of the additional value of wheel movability when driving through curvilinear zones of the rail and to vary the movability degree depending on the exploitation environment. Mentioned factors affect the operating safety and reliability at the expense of dynamical load reduction.

Keywords: *mine vehicle, mine cart, Lagrange equation, safety factor*

Introduction. Exploitation conditions analysis of mining wagons at enterprises of Eastern Donbas and their breakage causes [1] allowed to develop a new design of axle-box, which reduces dynamic loading of chassis members and increases motion stability at curvatures [2].

The character of bogie's members load disturbance determines dynamically unstable motion regime of mining wagon. Therefore, to increase its dynamical features it is necessary to develop requirements to verisimilar relative displacements of bogie's members with additional kinematical movability.

The value of angular movability, as well as radial and axial displacements of wagon's output members must provide possibility to take rational wheel alignment angle within curved rail track, and while drive through horizontal track imperfections [1, 3].

The design of the bogie with kinematically movable output members provides wheel displacement in measure of resilient deformation of wheel center and bandage kinematic constraint, as well as free displacement of spherical bodies in conical bushing surface. The boundary value of possible displacement is determined by following geometrical parameters of interactive members of kinematical pair "seat-guiding bushing – sphere": angle of cone moving line, diameter of spherical bodies and curvature radius of sphere rolling on the cone surface. Power factors, which determine conditions and character of reciprocal relative motion, are: permissible value of axial component of full reaction, which is generated in bearing and depends on conicity of seat-guiding bushings, diameter of spherical bodies and on preliminary gripping torque of the axle-box that provides its axial rigidity. The gripping torque varies in tiny measures. This is explained by kinematical features of coupling – the necessary condition of its workability is the absence of clearances between contact pairs. Otherwise, redundant movability will arise, which results in accelerated wear of working surfaces of contact members, increase of dynamic loading on the wagon's bogie.

The conventional design of mining wagon ВГ-3,3-900 with rigid axle includes rubber-metal shock absorbers, which are connected with frame. However, as a practice shows, such elements work till the first repair, following which they do not recover (and in most cases are not being installed on the wagons). As a result, the increased dynamics, growth of loading onto wheel pairs and enlarging of destructive impact on the rail track, which is in poor condition as it is, are emerging. In the engineering solution, which is studied, there is proposed to substitute the axle shock absorbers by the elastic-dissipative element (EDE), which connects wheel center and bandage.

It allows reducing an unsprung mass of wagon and dynamic loading correspondingly.

Besides the advantages of such decision, that provides structural movability, additional wheel displacements in the range of EDE deformation are possible.

The work purpose – to develop qualitative and quantitative influence of elastic-dissipative coupling parameters of output members with additional kinematical movability, that provides reduction of dynamic loading and growth of motion stability in short radius curve.

Numerous theoretical and experimental researches of rail vehicle dynamics prove that most common cause of dynamically unstable motion of mining rollingstock is rail track, which has geometrical imperfections in horizontal and vertical surfaces. On the ground of authors' research, mathematical simulation of the dynamical system 'wagon-rail track' is provided [2]. The solution of the system of differential equations allowed determining reactive powers, which work on output members while drive through rail track imperfections. Such irregularities could be divided onto the following, which are typical for mine working: rail cut, gauge narrowing (widening), rail deflection, wheel eccentricity. The relation of rail reaction F_z while motion of wagon ВГ-3.3-900 on the rail track with vertical irregularities is depicted on Fig. 1. It takes into account rail cuts and other vertical irregularities (detritus, rail deflections, wheel eccentricity etc.). Vertical rail track geometry specifies the motion trajectory of a wheel center (Fig. 2). Movement describes both on exceeding (positive values z) and weak rail cut (negative values z).

There is necessary to point out, that the value of normal full rail reaction is determined as a sum of distributed wagon's mass and dynamical component while motion through rail irregularities. Movement on imperfect rail track, its cuts etc., generates the biggest local power oscillations while wheel-rail contacts (Fig. 1). When crossing of the rail with following cut occurres, value of vertical reaction decreases owing to acceleration vector of vertical displacement changes its sign onto opposite one.

Meanwhile, it can be denoted that dynamic power component, which affects the output member exceeds the reaction value of distributed wagon's mass significantly. It means that it is necessary to provide efficient damping of dynamic loads by EDE while high force interaction of mechanical system members. But the task is to save reduced design rigidity of the bogie.

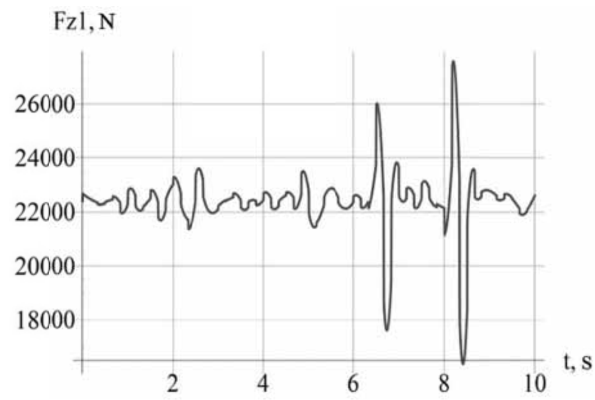

Fig. 1. The value of reactive loading, which takes wagon's wheel (of the left rail): F_{z1} – rail reaction under oncoming wheel, N; t – simulation time, s

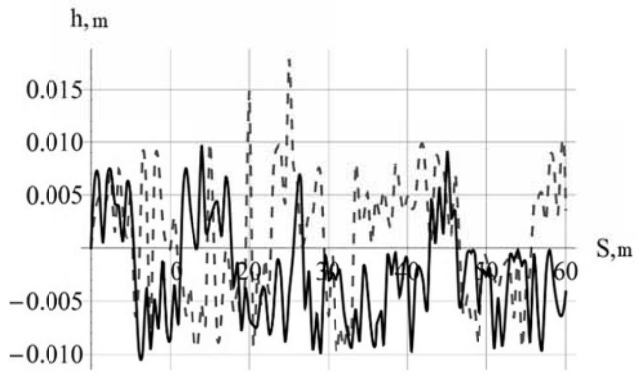

Fig. 2. Vertical trajectory of wheel center displacement: h – height of rail track irregularities, m; S – track length, m; ——— irregularities of the left rail; - - - - - - irregularities of the right rail

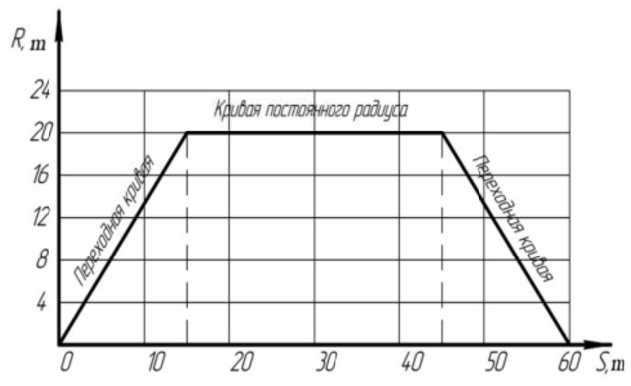

Fig. 3. Macrogeometry of rail track in horizontal surface: R – bending radius, m

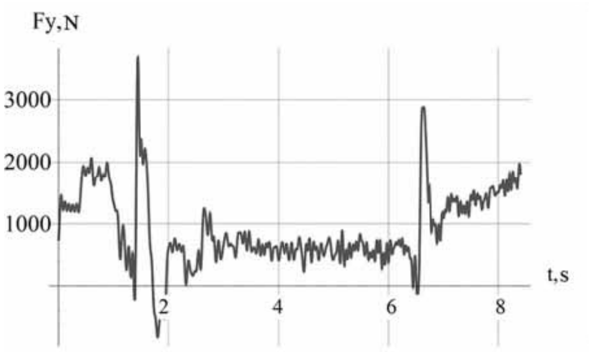

Fig. 4. Relation of the horizontal lateral force: F_y – horizontal lateral force, N

The technical decision, which is under the study, supposes the usage of cylindrical EDE as a coupling element. It is mounted in the wheel center and connects it with bandage. Due to difficult character of force interaction, caused by mining excavations, coupling element will be influenced by combined deformations of shift, coaxial spinning, axial and radial compression.

There are two design types of such elements: welded and assembled. The welded elements are produced by vulcanization of rubber element to metal armature and the assembled ones – by press-fitting a rubber element to a metal bushing.

The welded type is the most reliable for mine conditions. Its design provides durable coupling of metal frame and resilient element while high dynamic loads.

A calculation scheme of current EDE for modernized axle-box of mining wagon ВГ-3.3-900 is depicted on Fig. 5.

The following assumptions, which are determined by wagon's bogie design, are taken for EDE calculation: maximum vertical deformation of EDE does not exceed $f_\partial = 2{,}0$ mm; longitudinal displacement – $\Delta_p = 4{,}0$ mm; angular bandage displacement relative to vertical axis does not exceed $\xi = 0{,}036$ rad.

Calculation of main EDE parameters for rubberized wheels of railroad vehicles is provided by known methodic [4].

While motion within curve, influenced by horizontal lateral force EDE deforms and slides on the value

$$\Delta_p = \frac{F_Y}{C_y} - \frac{\sin\xi \cdot D}{2},$$

where Δ_p – longitudinal displacement of EDE, m; ξ – angular displacement of wheel bandage relative to vertical axis, grad; C_y – axle rigidity of EDE, N/m; F_Y – horizontal lateral guiding force, N.

EDE's axle compression rigidity is determined by equation

$$C_y = \frac{\pi \cdot (R_1^2 - R_2^2) \cdot E_p}{4 \cdot s},$$

where E_p – elastic modulus of EDE material, MPa; s – EDE length, m.

$$E_p = \frac{3 \cdot K \cdot F_Y{}^2}{3800},$$

where K – coefficient of EDE material hardening.

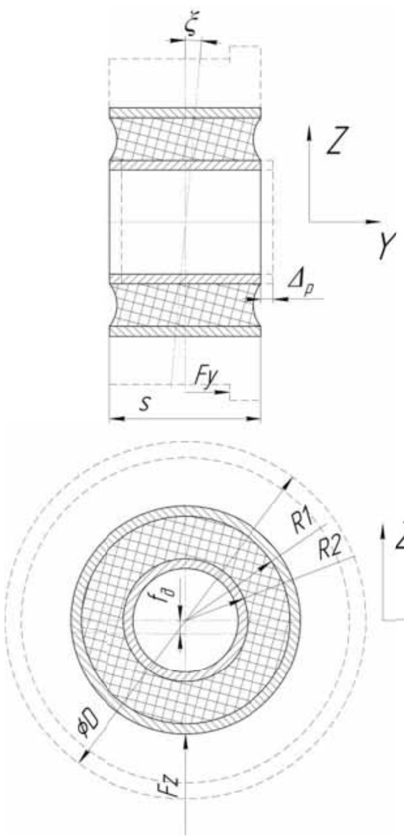

Fig. 5. Calculation scheme of elastic-dissipative element of modernized axle-box of wagon ВГ-3,3-900: D – wheel diameter, m; R_1 – external EDE radius, m; R_2 – internal EDE radius, m; s – EDE length, m; F_Y – horizontal lateral guiding force, N; F_z – full rail reaction, N; ξ – angular wheel bandage displacement relative to vertical axis, rad; Δ_p – longitudinal displacement, m; f_∂ – radial deformation, m; X, Y, Z – coordinate axis of calculation scheme

EDE's axle rigidity is limiting while determine axle frame displacement relative to wheel, which is dependent upon guiding force. It also influences on lateral dynamics while transient motion: angular frame oscillation caused by unstable motion, in easement, lateral irregularities et al. sources of dynamical force stress. In such motion regimes EDE damps arising dynamical lateral forces.

Relation of axial EDE's rigidity to numerous values of guiding force depicted on Fig. 6.

As shown above (Fig. 6), axial rigidity of dissipative element is in exponential relation to full rail reaction. Analysis of relation shows, that elastic-dissipative element could damp lateral oscillations, which are caused by rail track irregularities in wide range of loadings, efficiently.

Angular EDE rigidity C_ξ (N/rad), which specifies conditions of angular wheel displacement while motion in curvature, describes an equation

$$C_\xi = \frac{2F_Y}{s\lambda_r}\sin\xi,$$

where λ_r – wheel flange overshoot, m.

$$\lambda_r = (D/2 + c_w)\sin\varphi\, tg\alpha,$$

where c_w – flange height, m; φ – overshoot angle, rad; α – wheel conicity angle, rad.

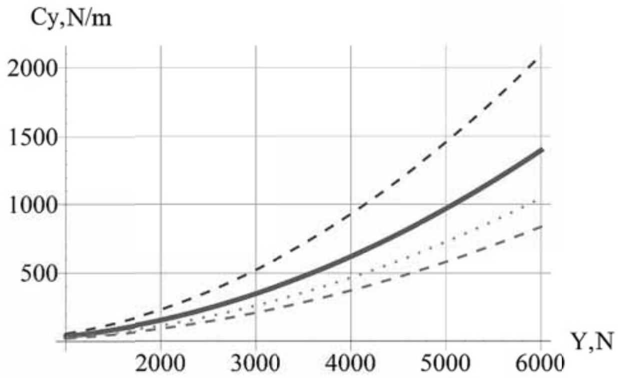

Fig. 6. Relation of EDE axle rigidity to guiding force: C_y – EDE axle rigidity, N/m; Y – value of guiding force, N

The relation of angular rigidity C_ξ to angular wheel displacement varies linearly (Fig. 7) and obeys the general exploitation principles of rubber damping elements. Maximal angular rigidity of EDE arises while high wheel displacement degree, that is typical for motion in short radius curve.

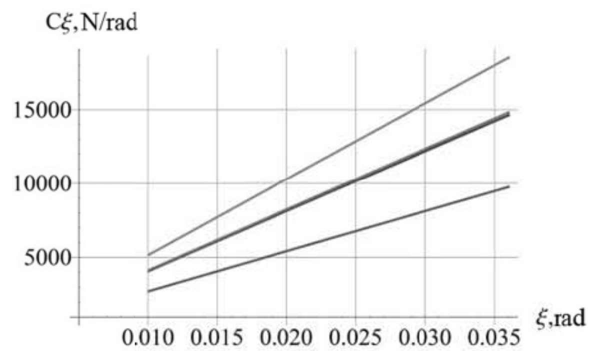

Fig. 7. Relation EDE angular rigidity to angular displacement: C_ξ – EDE angular rigidity, N/rad; ξ – value of output member angular displacement, rad

Dynamical characteristics of axial EDE displacement dependent upon guiding force is described by flexibility of resilient element, m/N, [4]

$$\Psi = \frac{1}{C_y} + \frac{D^2}{4C_y}.$$

As it is shown on Fig. 8, increase of axle rigidity causes decline of EDE flexibility. This parameter describes longitudinal shift under load 1 N. For further calculations it is necessary to choose such value C_y, which will guarantee damping of lateral oscillations, but does not exceed maximal shift $4\cdot10^{-3}$ m, that is defined by axle-box design.

The analysis of the relations, depicted on Fig. 6-8 showed, that value of EDE axial shift under high lateral loadings has small order $\Psi = 1\cdot10^{-7}$ and this parameter can be neglected in following calculations.

Maximum radial deflection under dynamic loading, that arises while overcome the rail irregularities, is defined as

$$f_\partial = \frac{F_z}{C_z},$$

where C_Z – radial wheel EDE rigidity, N/m.

Relation to define value of radial rigidity is [5]

$$C_z = \frac{3\pi}{2}Gs\frac{s^2 + 6(R_1 - R_2)^2(R_1 + R_2)^3}{s^2 + 3(R_1 + R_2)^2(R_1 - R_2)^3},$$

where G – rigidity modulus of a filler, MPa; s – EDE length in pressurized condition, m; R_1, R_2 – external and internal EDE armature diameters, correspondingly, m.

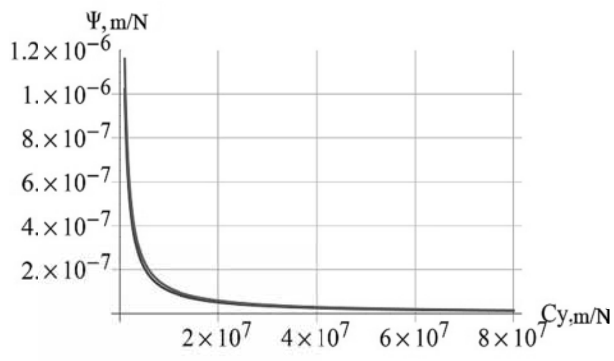

Fig. 8. Relation of output member axle flexibility to EDE rigidity: ψ – EDE flexibility, m/N

There is no necessity in calculation a temperature regime of EDE because the temperature of dissipative warming with temperature gradient of exploitation condition of mining-metallurgical enterprises is much lower than acceptable limits for technical rubbers are [5].

Foregoing relations define EDE characteristics of output members and allow determining the value of oncoming wheel angular displacement while motion on local and systematic rail track irregularities.

By usage of track parameters, which had been measured at mining surface complex of mine "Stepnaya" PAO "DTEK Pavlogradugol" the mathematical model of mechanical system 'wagon-rail track' have been composed. Thus the value of angular displacement of oncoming wheel (Fig. 8) and safety coefficient (Fig. 9) have been evaluated.

From Fig. 9 it is clearly understandable, that angular displacement of oncoming wheel arises while motion in circular curve and on easement. Maximal angular deviation of oncoming wheel is limited by axle-box design and its range lies in [-0,05; 0,05] rad.

Therefore, owing to a proper deformation, resilient element compensates lateral stress of the rail track and reduces dynamic interaction of the flange and rail.

Calculation results of safety coefficient (SC) for wagon with output members with additional kinematical movability (Fig. 9) show the momentary value of SC for each track point with refresh rate 0,5 m. The largest interest has diapason of results 1–5 unities (relation $F_z / F_y > 1$). The zones, where values are more then 5, we assume as safe. The task is to define safe exploitation zones of mining wagon provided for combined irregularities (vertical and horizontal). Thus in specified track points, where acceleration vector is directed to the negative zone, SC reduces due to dynamical unload of oncoming wheel. There are results for conventional, fixed oncoming wheel, and modernized, with additional movability and resilient coupling of oncoming wheel, bogie design on Fig. 9. Analysis of relation showed, that on track sections with slump changes of horizontal irregularities SC is 5–10% higher for new bogie design in comparison to conventional design.

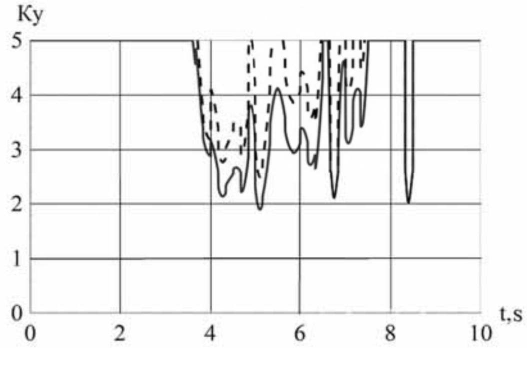

Fig. 9. Relation of momentary SC value to track features: K_y – safety coefficient: ——— fixed mounting of oncoming wheel; - - - - - - movable mounting of oncoming wheel with EDE

During mathematical study we have discovered the following: fixed oncoming wheel can cause local reduction of SC owing to difficult mining-technological conditions of mine excavations. Such motion regime is interpreted as partial flange climb on the rail without stability loose. Such regime arises in diapason 8–9 s (Fig. 9).

The forthcoming study of calculation algorithm of additional movability influence and parameters of EDE coupling allow defining rational displacement angles of output members and geometrical parameters of axle-box for ВГ-3.3-900.

Conclusions. There are certain conclusions, which could be done basing on the results.

1. Analysis of dynamical characteristics of mining wagon with output members' additional kinematical movability allowed us to determine EDE's parameters, that enables to reduce dynamic loading while motion though rail irregularities. The following parameters of EDE have been resulted: E_p=0,78 MPa, $C_y = 8{,}26 \cdot 10^3$ MPa, C_ξ=0,45 MPa, C_Z=3,8 MPa, R_1=40 mm, R_2=20 mm, G=1,5 MPa, s=0,06 m.

2. Elastic-dissipative element, mounted in the wheel center, which parameters allow to change wheel alignment angle, reduces unspung mass and rises safety coefficient up to 5–10% relative to velocity in curvature.

3. We have determined, that the fixed oncoming wheel design makes the occurrence of regime of partial flange climb on the rail more frequent, as indicated by local oscillations of SC while motion in sharp curvature.

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