руху елемента опорядження, гідропривід маніпулятора, методи вирішення оптимізаційних задач

Purpose. Development of the mathematical model for justification of rational parameters of translational motion of tunnel lining machine manipulator.

Methods. Solving of optimization problems by defining the class of test functions; calculation of some of the features of this class; and confirmation of the uniqueness of the solution. Application of mathematical modeling methods for calculation of the motion of the fluid in the hydraulic systems in laminar and turbulent flows.

Findings. We have developed the mathematical model of the hydraulic drive arm allowing us to find the hydraulic control valve spool motion control law under any law of motion of the block. The results of the simulation show that to decrease the control signal anticipation the area of the rod end should be increased under given line pressure.

Originality. It is proved that: when acceleration rate value is limited, the U-shaped motion law provides the fastest lifting of the block to certain height; if necessary, to decrease the control signal anticipation over the time of movement speed change the area of the rod end should be increased under given line pressure.

Practical value. The mathematical model of the hydraulic drive lift has been developed. Its application allows determining of the rational parameters of the tunnel lining machine manipulator on order to create the tunnel lining machine manipulators of reduced weight, increased strength and durability, competitive at the world market.

Keywords: *tunnel lining machine manipulator, hydraulic control valve spool motion control law, lining element motion law, manipulator hydraulic drive, optimization problem solving methods*

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K.M. Bass, Cand. Sci. (Tech.), Associate Professor,
S.M. Kuvayev, Cand. Sci. (Tech.), Associate Professor,
V.V. Plakhotnik, Cand. Sci. (Tech.), Associate Professor,
V.V. Krivda

State Higher Educational Institution “National Mining University”, Dnipropetrovsk, Ukraine, e-mail: KMBass@yandex.ru

PLANAR AND SPATIAL MATHEMATICAL MOTION SIMULATION OF OPEN PIT MINING VEHICLES

К.М. Бас, канд. техн. наук, доц.,
С.М. Куваєв, канд. техн. наук, доц.,
В.В. Плахотнік, канд. техн. наук, доц.,
В.В. Крівда

Державний вищий навчальний заклад „Національний гірничий університет“, м.Дніпропетровськ, Україна,
e-mail: AlekseevM@nmu.org.ua

ПЛОЩИННЕ ТА ПРОСТОРОВЕ МАТЕМАТИЧНЕ МОДЕЛЮВАННЯ РУХУ КАР’ЄРНОГО АВТОТРАНСПОРТУ

The article presents the mathematical simulations of dump truck linear motion along a straight road segment as a motion of multimass system with eight degrees of freedom.

Purpose. To analyze the results of the study of dump truck dynamic and traction-speed performance in the course of motion.

Methodology. General scientific and special study methods, including scientific generalization, methods of integrated assessment of technical level, mathematical modeling and linear programming were used. To solve the problem we have formulated differential equations of motion, using the Lagrange equations of the second kind and the corresponding expressions of kinetic and potential energy as well as dissipative function.

Findings. By mathematical calculations in the Wolfram Mathematica software, the traction and dynamic characteristics calculation method for the dump truck moving along the road with longitude inclination was received.

Originality. A planar and spatial design diagrams and motion equations of the vehicle in the course of straight-line motion were made, considering elastic and dissipative characteristics of elastic constraints, longitudinal slope and road profile, changes of design characteristics which can provide a near real picture of motion dynamics.

Practical value. The method of calculation of the dump truck dynamic characteristics in motion has been developed. Based on the analysis of the dump truck design parameters presented, one may provide recommendations on reduction in capital expenditure for development of a quarry.

Keywords: *design diagram, dump truck*

Introduction. Study of the vehicle dynamic as a complex multimass system requires considering different level of influence of mechanical characteristics of individual vehicle elements in the different motion modes.

A significant contribution to the development of the theory and practice of vehicle use in the open pit mining was made by M.V. Vasylyev, A.A. Kuleshov, A.N. Kazarez, N.V. Melnikov, M.G. Potapov, V.V. Rzhevsky, V.P. Smirnov, I.M. Tsyperfin, V.D. Shtein, B.Ya. Yakovenko, I.V. Zyryanov, A.V. Bunyakin, G.A. Smirnov [1, 2, 3], etc. They set up mathematical models to study the dynamic processes of open pit mining vehicles that take into account operational characteristics (in particular road irregularities and impact from excavator loading). Compiled motion equations are presented in this case as a system of nonlinear differential equations, which are solved with given initial conditions in a given integration interval with the use of specially developed iteratively-differential type algorithm with determination of the point of discontinuity for the right members of equations [1–3].

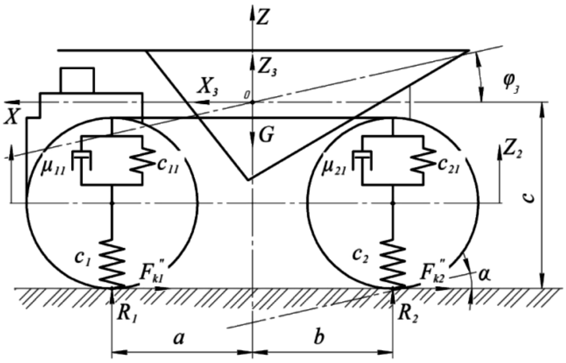
Analysis of the earlier studies. Further design improvement of the dump truck, operating in particularly difficult environment, and assessment of their performance requires involvement of more structural and technological factors in compiling the design scheme, which will allow determining their effect on the vehicle performance and its reliability. Therefore, the study of the vehicle dynamics and obtaining the amplitude-frequency characteristics depending on the physical and mechanical characteristics of elastic constraints and masses of the vehicle parts may be considered as a relevant objective.

The investigation of dynamic of the linear vehicle motion without side slope of the road is presented below.

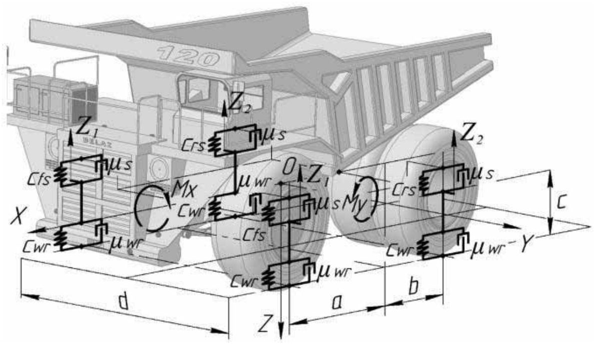
The purpose of the study is to draw the design diagrams and formulate the motion equations describing the straight vehicle motion, considering the characteristics of its elastic constraints and cross-section of road.

The planar design diagram and motion equations drawing up. In drawing up the planar vehicle design diagram assume that the mechanical characteristics of the wheels, located at one axis of the respective double-reduction axles, are equal. In this case, the dump truck can be represented as a system of bodies (Fig. 1), connected by elastic and inelastic constraints; provided that the functions describing the interaction of the vehicle wheels with the road simulate various motion conditions.

At Fig. 1: *a, b, c* are dimensions determining the position of the vehicle center of mass relative to the wheel axes and the road surface, axle spacing (dump truck base) – $L = (a + b)$; d is dump truck wheel track; c_{wr}, μ_{wr} are stiffness and damping factors of the front and rear wheels; c_s, $c_{wr,}$ μ_s, μ_{wr} are stiffness and damping factors relative to the vehicle suspension and wheels; Z is relative movement of the sprung body mass perpendicular to the vehicle direction; Z_1, Z_2 are moving axes relative to the front and rear wheels and perpendicular to the vehicle direction; X is the vehicle center of mass moving at the vehicle direction; φ is body rotation angle relative to the axis passing through the center of mass and perpendicular to $X0Z$-plane; φ_1, φ_2 are rotation angle of the wheels relative to the axis; ψ is rotation angle of the body around X-axis; α is road slope angle by the vehicle direction; m_1, m_2, m_B, $G1$, $G2$, G are mass and weight relative to the front and rear wheels and the dump truck body. Interaction with the road surface is expressed through R_F, R_R normal reactions, and F_{k1} and F_{k2} rolling resistance.



a



b

Fig. 1. Design diagrams of the open pit dump truck motion: a is planar design diagram of the open pit dump truck; b is spatial vehicle design diagram

Consider the vehicle motion in the $X0Z$ subspace, passing through the longitudinal axis of the vehicle. To compile the motion equations we use the Lagrange equation of the second kind. In the preparation of kinetic T and potential P energy expressions of D dissipative function we used m_1, m_2, J_1, J_2 values of mass and moment of inertia of the front and rear wheels, m_3, J_3 body mass and moment of inertia, (Z_1, Z_2, Z_3, x, φ_3 taken as generalized coordinates) for the planar design diagram, as well as m_1, m_2, m_B, J_{YB}, J_{XB}, J_{Y1}, J_{Y2} used as values of mass and moments of inertia. X, Z_1, Z_2, Z, φ_1, φ_2, φ, ψ were taken as Qi generalized coordinates for spatial design diagram.

The kinetic energy of the system (planar design diagram)

$$T=\frac{1}{2}m3\cdot\left(\frac{a\cdot\dot{Z}_2^{2}+b\cdot\dot{Z}_1^{2}}{a+b}+\dot{Z}_3^{2}\right)+\frac{1}{2}\cdot m_1\cdot\dot{Z}_1^{2}+ +\frac{1}{2}\cdot\left(\frac{(J_1+J_2)\cdot\dot{X}_3^{2}}{2r^2}\right)+\frac{\dot{X}_3^{2}}{2}(m_1+m_2+m_3)+ +\frac{1}{2}J_3\cdot\dot{\varphi}_3^{2}+\frac{1}{2}m_1\cdot\dot{Z}_1^{2}+\frac{1}{2}m_2\cdot\dot{Z}_2^{2}. \quad (1)$$

The potential energy of the system (planar design diagram)

$$
\begin{aligned}
\ddot{I} &= \frac{1}{2}\cdot\tilde{n}_{11}(-a\cdot\varphi_3 - Z_1 + Z_3)^2 + \frac{1}{2}\cdot\tilde{n}_{21}(b\cdot\varphi_3 - \\
&- Z_2 + Z_3)^2 + +\frac{1}{2}\cdot\tilde{n}_1\cdot Z_1{}^2 + \frac{1}{2}\cdot\tilde{n}_2\cdot Z_2{}^2 - \\
&- G1\cdot\cos\alpha\cdot Z1 - G2\cdot\cos\alpha\cdot Z2 - G3\cdot\cos\alpha\cdot Z3.
\end{aligned} \tag{2}
$$

The kinetic energy of the system (spatial design diagram)

$$
\begin{aligned}
T &= \frac{1}{2}\cdot(J_{XB}\cdot\dot{\psi}^2 + J_{YB}\cdot\dot{\varphi}^2 + \dot{X}^2(\frac{2(J_1+J_2)}{r^2} + \\
&+ 2m_1 + 2m_2 + m_B) + +J_{Y1}\cdot\dot{\varphi}_1^2 + J_{Y2}\cdot\dot{\varphi}_2^2 + \\
&+ m_B\cdot\dot{Z}^2 + m_1\cdot\dot{Z}_1^2 + m_2\cdot\dot{Z}_2^2).
\end{aligned} \tag{3}
$$

The potential energy of the system (spatial design diagram)

$$
\begin{aligned}
\ddot{I} &= \frac{1}{2}\cdot(\tilde{n}_s(\Delta_1^2 + \Delta_2^2 + \Delta_3^2 + \Delta_4^2) + \\
&+ \tilde{n}_{wr}(\delta_1^2 + \delta_2^2 + \delta_3^2 + \delta_4^2)).
\end{aligned} \tag{4}
$$

Dissipative function (spatial design diagram)

$$
\begin{aligned}
Д &= \frac{1}{2}\cdot(\mu_s(\dot{\Delta}_1^2 + \dot{\Delta}_2^2 + \dot{\Delta}_3^2 + \dot{\Delta}_4^2) + \\
&+ \mu_{wr}(\dot{\delta}_1^2 + \dot{\delta}_2^2 + \dot{\delta}_3^2 + \dot{\delta}_4^2)),
\end{aligned} \tag{5}
$$

where Δi, $\dot{\Delta} i$ are motion and velocity of the dump truck body mass center respectively in the motion at a given truck profile; δi, $\dot{\delta} i$ – respectively motion and velocity of mass center of the dump truck wheel at the front and rear suspension in the course of motion along given road profile.

After substitution and solution of the kinetic and potential energy expressions, and dissipative function, considering constraints (1–5), as well as related transformations in the *Wolfram Mathematica* software, we obtain the system of six differential equations of second order for planar design diagram and the system of seven differential equations of second order for spatial design diagram.

The solutions of equations describe the changes of generalized coordinates which allow us to estimate the vehicle dynamics, taking into account the nature of the road.

$$
\begin{aligned}
&m\ddot{X}_3 = \frac{M_g}{r} - f_{k1}R_1 - f_{k2}R_2 + mg\sin\alpha; \\
&(m_1 + m_3 k_0^2)\ddot{Z}_1 - m_3 k_0^2(\ddot{Z}_2 + \ddot{Z}_3) + (c_1 + c_{11})Z_1 - \\
&- c_{11}(Z_3 + \varphi a) + c_{11}\mu_{11}(\dot{Z}_1 - \dot{Z}_3 - \dot{\varphi}a) = \\
&= g\cos\alpha(m_1 - m_3 k_0); \\
&(m_2 + m_3 k_0^2)\ddot{Z}_1 - m_3 k_0^2(\ddot{Z}_1 + \ddot{Z}_3) + (c_2 + c_{21})Z_2 - \\
&- c_{21}(Z_3 - \varphi b) + c_{21}\mu_{21}(\dot{Z}_2 - \dot{Z}_3 + \dot{\varphi}b) = \\
&= g\cos\alpha(m_2 + m_3 k_0); \\
&m_3\ddot{Z}_0 - c_{11}\mu_{11}(\dot{Z}_1 - \dot{Z}_3 - \dot{\varphi}a) - c_{21}\mu_{21}(\dot{Z}_2 - \dot{Z}_3 + \dot{\varphi}b) - \\
&- c_{11}(Z_1 - Z_3 - \varphi a) - c_{21}(Z_2 - Z_3 + \varphi b) = m_3 g\cos\alpha; \\
&J_3\ddot{\varphi}_3 - c_{11}\mu_{11}a(\dot{Z}_1 - \dot{Z}_3 - \dot{\varphi}a) + c_{21}\mu_{21}b(\dot{Z}_2 - \dot{Z}_3 + \dot{\varphi}b) - \\
&- c_{11}a(Z_1 - Z_3 - \varphi a) + c_{21}b(Z_2 - Z_3 + \varphi b) = 0;
\end{aligned} \tag{6}
$$

$$
\begin{aligned}
&(\frac{2(J_1+J_2)}{r^2} + 2m_1 + 2m_2 + m_B)\cdot\ddot{X} = 0; \\
&\frac{1}{2}\tilde{n}_s(2(\cos(\frac{L}{2}\varphi_1) + Z - Z_1 + a\varphi - \frac{L}{2}\psi) + 2(\cos(\frac{L}{2}\varphi_2) + \\
&+ Z - Z_2 - b\varphi - \frac{L}{2}\psi) + 2(-\cos(\frac{L}{2}\varphi_1) + Z - Z_1 + a\varphi + \frac{L}{2}\psi) + \\
&+2(-\cos(\frac{L}{2}\phi_2) + Z - Z_2 - b\phi + \frac{L}{2}\psi)) + \frac{1}{2}\mu_s(2(-\sin(\frac{L}{2}\dot{\phi}_1) + \\
&+\dot{Z} - \dot{Z}_1 + a\dot{\phi} - \frac{L}{2}\dot{\psi}) + 2(-\sin(\frac{L}{2}\dot{\phi}_2) + \dot{Z} - \dot{Z}_2 - b\dot{\phi} - \frac{L}{2}\dot{\psi}) + \\
&+2(\sin(\frac{L}{2}\dot{\phi}_1) + \dot{Z} - \dot{Z}_1 + a\dot{\phi} + \frac{L}{2}\dot{\psi}) + 2(\sin(\frac{L}{2}\dot{\phi}_2) + \\
&+\dot{Z} - \dot{Z}_2 - b\dot{\phi} + \frac{L}{2}\dot{\psi}))\cdot m_B\ddot{Z} = 0;
\end{aligned}
$$

$$
\begin{aligned}
&\frac{1}{2}\left(4c_{wr}(-P + Z_1 - \varphi_1\right) + c_s(-2(\cos(\frac{L}{2}\varphi_1) + Z - Z_1 + \\
&+ a\varphi - \frac{L}{2}\psi) - 2(-\cos(\frac{L}{2}\varphi_1) + Z - Z_1 + a\varphi + \frac{L}{2}\psi))) + \\
&+ \frac{1}{2}\left(4\mu_{wr}(-\dot{P} + \dot{Z}_1 - \dot{\varphi}_1\right) + \mu_s(-2(-\sin(\frac{L}{2}\dot{\varphi}_1) + \dot{Z} - \\
&- \dot{Z}_1 - a\dot{\varphi} - \frac{L}{2}\dot{\psi}) - 2(\sin(\frac{L}{2}\dot{\varphi}_1) + \dot{Z} - \dot{Z}_1 - a\dot{\varphi} + \frac{L}{2}\dot{\psi}))) + \\
&+ m_1\cdot\ddot{Z}_1 = 0;
\end{aligned}
$$

$$
\begin{aligned}
&\frac{1}{2}\left(4c_{wr}(-P + Z_2 - \varphi_2\right) + c_s(-2(\cos(\frac{L}{2}\varphi_2) + \\
&+ Z - Z_2 + b\varphi - \frac{L}{2}\psi) - 2(-\cos(\frac{L}{2}\varphi_2) + \\
&+ Z - Z_2 + b\varphi + \frac{L}{2}\psi))) + +\frac{1}{2}\left(4\mu_{wr}(-\dot{P} + \dot{Z}_2 - \dot{\varphi}_2\right) + \\
&+ \mu_s(-2(-\sin(\frac{L}{2}\dot{\varphi}_2) + \dot{Z} - \dot{Z}_2 - b\dot{\varphi} - \frac{L}{2}\dot{\psi}) - \\
&- 2(\sin(\frac{L}{2}\dot{\varphi}_2) + \dot{Z} - \dot{Z}_2 - b\dot{\varphi} + \frac{L}{2}\dot{\psi}))) + m_2\cdot\ddot{Z}_2 = 0;
\end{aligned}
$$

$$
\begin{aligned}
&\frac{1}{2}\mathrm{c_s}(2\mathrm{a}(\cos(\frac{\mathrm{L}}{2}\phi_1) + \mathrm{Z} - \mathrm{Z}_1 + \mathrm{a}\phi - \frac{\mathrm{L}}{2}\psi) - \\
&-2\mathrm{b}(\cos(\frac{\mathrm{L}}{2}\phi_2) + \mathrm{Z} - \mathrm{Z}_2 - \mathrm{b}\phi - \frac{\mathrm{L}}{2}\psi) + \\
&+2\mathrm{a}(-\cos(\frac{\mathrm{L}}{2}\phi_1) + \mathrm{Z} - \mathrm{Z}_1 + \mathrm{a}\phi + \frac{\mathrm{L}}{2}\psi) - \\
&-2\mathrm{b}(-\cos(\frac{\mathrm{L}}{2}\phi_2) + \mathrm{Z} - \mathrm{Z}_2 - \mathrm{b}\phi + \frac{\mathrm{L}}{2}\psi)) + \\
&+\frac{1}{2}\mu_\mathrm{s}(2\mathrm{a}(-\sin(\frac{\mathrm{L}}{2}\dot{\phi}_1) + \dot{\mathrm{Z}} - \dot{\mathrm{Z}}_1 + \mathrm{a}\dot{\phi} - \frac{\mathrm{L}}{2}\dot{\psi}) - \\
&-2\mathrm{b}(-\sin(\frac{\mathrm{L}}{2}\dot{\phi}_2) + \dot{\mathrm{Z}} - \dot{\mathrm{Z}}_2 - \mathrm{b}\dot{\phi} - \frac{\mathrm{L}}{2}\dot{\psi}) + \\
&+2\mathrm{a}(\sin(\frac{\mathrm{L}}{2}\dot{\phi}_1) + \dot{\mathrm{Z}} - \dot{\mathrm{Z}}_1 + \mathrm{a}\dot{\phi} + \frac{\mathrm{L}}{2}\dot{\psi}) - \\
&-2\mathrm{b}(\sin(\frac{\mathrm{L}}{2}\dot{\phi}_2) + \dot{\mathrm{Z}} - \dot{\mathrm{Z}}_2 - \mathrm{b}\dot{\phi} + \frac{\mathrm{L}}{2}\dot{\psi})) + \mathrm{J_{YB}}\ddot{\phi} = 0;
\end{aligned} \tag{7}
$$

$$\frac{1}{2}\left(-4c_{wr}(-P+Z_1-\varphi_1)\right)+c_s(-L(\sin(\frac{L}{2}\varphi_1)\times$$
$$\times(\cos(\frac{L}{2}\varphi_1)+Z-Z_1+a\varphi-\frac{L}{2}\psi)+$$
$$+L\sin(\frac{L}{2}\varphi_1)(\cos(\frac{L}{2}\varphi_1)+Z-Z_1+a\varphi+\frac{L}{2}\psi)))+$$
$$+\frac{1}{2}\left(-4\mu_{wr}(-\dot{P}+\dot{Z}_1-\dot{\varphi}_1)\right)+\mu_s(-L(\cos(\frac{L}{2}\varphi_1)\times$$
$$\times(-\sin(\frac{L}{2}\varphi_1)+\dot{Z}-\dot{Z}_1+a\dot{\varphi}-\frac{L}{2}\dot{\psi}+L\cos(\frac{L}{2}\varphi_1)\times$$
$$\times(\sin((\frac{L}{2}\varphi_1)+\dot{Z}-\dot{Z}_1+a\dot{\varphi}+\frac{L}{2}\dot{\psi})))J_{Y1}\ddot{\varphi}_1=0;$$

$$\frac{1}{2}c_s(-L(\cos(\frac{L}{2}\varphi_1)+Z-Z_1+a\varphi-\frac{L}{2}\psi)-$$
$$-L(\cos(\frac{L}{2}\varphi_2)+Z-Z_2-b\varphi-\frac{L}{2}\psi)+$$
$$+L(-\cos(\frac{L}{2}\varphi_1)+Z-Z_1+a\varphi+\frac{L}{2}\psi)+$$
$$+L(-\cos(\frac{L}{2}\varphi_2)+Z-Z_2-b\varphi+\frac{L}{2}\psi))+$$
$$+\frac{1}{2}\mu_s(-L(-\sin(\frac{L}{2}\varphi_1)+\dot{Z}-\dot{Z}_1+a\dot{\varphi}-\frac{L}{2}\dot{\psi})-$$
$$-L(-\sin(\frac{L}{2}\varphi_2)+\dot{Z}-\dot{Z}_2-b\dot{\varphi}-\frac{L}{2}\dot{\psi})+$$
$$+L(\sin(\frac{L}{2}\varphi_1)+\dot{Z}-\dot{Z}_1+a\dot{\varphi}+\frac{L}{2}\dot{\psi})+$$
$$+L(\sin(\frac{L}{2}\varphi_2)+\dot{Z}-\dot{Z}_2-b\dot{\varphi}+\frac{L}{2}\dot{\psi}))+J_{XB}\ddot{\psi}=0.$$

Fig. 2–3 shows graphs of the main dynamic parameters variance for the dump truck moving along the road segment with i = 7% longitudinal gradient of the road, given small irregularities for 10 and 20 seconds for planar and spatial diagrams respectively.

The obtained systems of five and of eight equations (6) and (7) respectively, describing straight movement of vehicle, taking into account the influence of the transverse and torsion vibrations (Fig. 2, 3) that may occur during destabilization or in transient regimes, we can recommend for research of the dynamics of quarry vehicles for natural and disturbing vibrations source. Value of natural frequencies and amplitudes of the oscillations are the parameters of evaluation of movement smoothness, and vehicles reliability.

Conclusions. A mathematical model of the dump truck motion has been built, as well as the dependences for assessing the impact of the road profile on major dynamic and operational characteristics of the trucks have been received. The calculation results of characteristic curves shown at Fig. 2–3, allow us to estimate the offset value of the front and rear wheel suspensions as well as of the center of the body mass. Furthermore, we have received corresponding acceleration values, allowing us to estimate the inertial load on the vehicle structure parts when passing the obstacles, movement smoothness, and vehicles reliability. The obtained characteristic curves can be used in the process of design of operational and technical parameters of the individual mechanisms of dump truck. This allows us to give recommendations for design institutes concerning methods determining power dissipation (Fig. 2, *a, c, e, g*, Fig. 3, *a, c*) and the maximum speed in the process of moving along the road with the longitudinal inclination and irregularities of 0.2 m.

The received mathematical dump truck motion simulations will allow estimating the impact of isolated road irregularities, as well as structural dump truck parameters on major dynamic and operational tractive characteristics. The results presented at Fig. 2–3 show that natural vibrations affect the value of dissipation essentially.

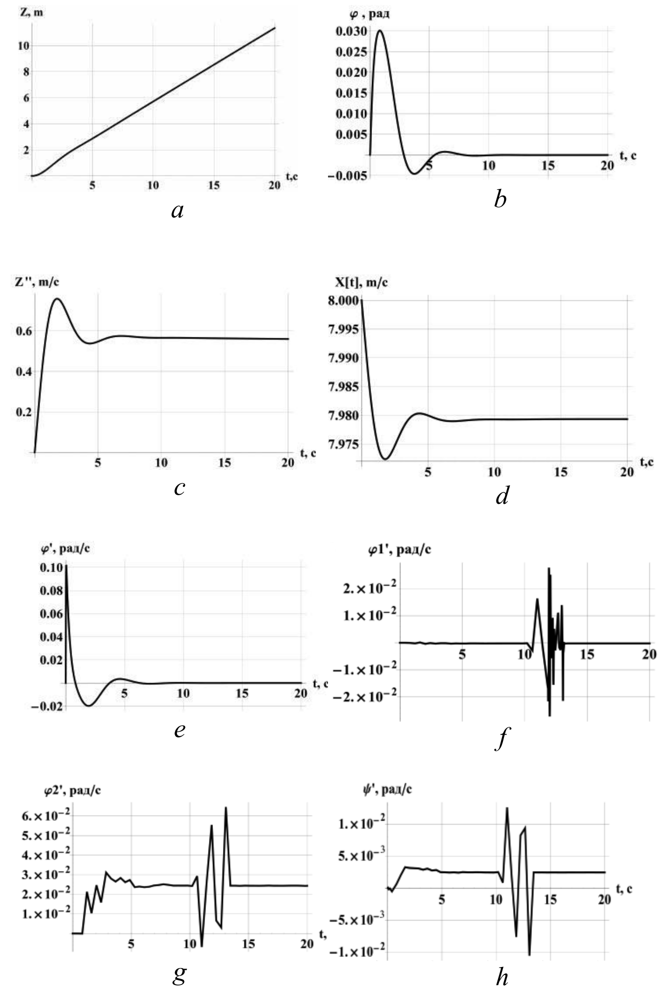


Fig. 2. Graphs of motion parameters on a road segment with i=7% longitudinal gradient of the road, given small irregularities for 20 seconds: a is vertical movement of the center of dump truck mass considering function P(x); b is angular movement relative to the axis center of the dump truck mass (longitudinal pitch); c is acceleration; d is dump truck acceleration by motion direction with i=7% slope; e is dump truck angular acceleration relative to Y transverse axis, passing through the center of dump truck mass; f is angular acceleration of the front axle relative to the center of mass; g is angular acceleration of the rear axle relative to the center of mass; h is dump truck angular acceleration relative to X longitudinal axis, passing through the center of mass

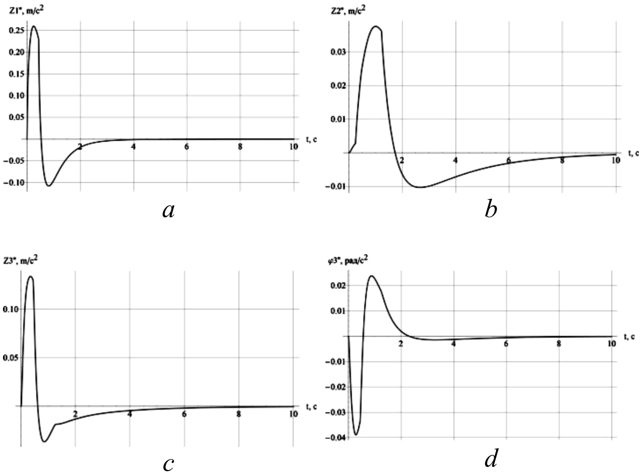


Fig. 3. Graphs of the motion parameters on a road segment with i=7% longitudinal gradient of the road, given small irregularities for 10 seconds: a is vertical acceleration of the front dump truck axle; b is vertical acceleration of the rear dump truck axle; c is vertical acceleration of the center of the dump truck mass; d is angular acceleration relative to the axis of the center of the dump truck mass

Using the obtained values of displacement of front and rear suspensions, and the center of mass of the vehicle moving along the road with the longitudinal inclination, we can define external disturbing vibrations of body parts and the vehicle in general. By choosing the values of the arguments influencing the movement smoothness and speed we can adjust the value of dissipation using the mathematical models (6–7), and thus correct the stroke of suspension and body of the dump truck.

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Мокін Б.І. Математичні моделі багатомасових розподілених динамічних систем для задач оптимізації (частина 1) / Б.І. Мокін, О.Б. Мокін // Вісник Вінницького політехнічного інституту. – 2008. – № 6. – С. 55–58.

У статті наведені математичні моделі прямолінійного руху кар'єрного автосамоскида як багатомасової системи з вісьмома ступенями свободи на прямолінійній ділянці дороги з урахуванням стану дорожнього покриття.

Мета. Аналіз дослідження динаміки та тягово-швидкісних характеристик у процесі руху автосамоскида.

Методика. При виконанні роботи використовувались як загальнонаукові, так і спеціальні методи досліджень, включаючи наукове узагальнення, методи комплексної оцінки технічного рівня, математичне моделювання та апарат лінійного програмування. Рішення даної задачі базується на складанні диференціального рівняння руху, для чого були використані рівняння Лагранжа другого роду, а також відповідні вирази кінетичної, потенційної енергії та дисипативної функції.

Результати. За допомогою математичних розрахунків у програмному продукті математичного аналізу та розрахунку – “Wolfram Mathematica”, отримана методика розрахунку тягових і динамічних характеристик автосамоскида у процесі руху по дорозі з поздовжнім ухилом.

Наукова новизна. Складені розрахункові схеми та рівняння руху при прямолінійному русі машини з урахуванням пружних і розсіювальних характеристик пружних зв'язків, поздовжнього ухилу та профілю дороги, зміни конструктивних характеристик, що дозволить дати близьку до реальної картину динаміки руху.

Практична значимість. Розроблені методики розрахунку динамічних характеристик автосамоскида у процесі руху, а також виконаний аналіз параметрів конструкції автосамоскидів, на підставі чого можна дати рекомендації щодо скорочення капітальних витрат на розробку кар’єрів.

Ключові слова: *розрахункова схема, автосамоскид*

В статье приведены математические модели прямолинейного движения карьерного автосамосвала как многомассовой системы с восемью степенями свободы на прямолинейном участке дороги с учетом состояния дорожного покрытия.

Цель. Анализ исследования динамики и тягово-скоростных характеристик в процессе движения автосамосвала.

Методика. При выполнении работы использовались как общенаучные, так и специальные методы исследований, включая научное обобщение, методы комплексной оценки технического уровня, математическое моделирование и аппарат линейного программирования. Решение данной задачи базируется на составлении дифференциального уравнения движения, для чего были использованы уравнения Лагранжа второго рода, а также соответствующие выражения кинетической, потенциальной энергии и диссипативной функции.

Результаты. С помощью математических расчетов в программном продукте математического анализа и расчета – "Wolfram Mathematica", получена методика расчета тяговых и динамических характеристик автосамосвала в процессе движения по дороге с продольным уклоном.

Научная новизна. Составлены расчетные схемы и уравнения движения при прямолинейном движении машины с учетом упругих и рассеивающих характеристик упругих связей, продольного уклона и профиля дороги, изменения конструктивных характеристик, что позволит дать близкую к реальной картину динамики движения.

Практическая значимость. Разработаны методики расчета динамических характеристик автосамосвала в процессе движения, а также выполнен анализ параметров конструкции автосамосвалов, на основании чего можно дать рекомендации по сокращению капитальных затрат на разработку карьеров.

Ключевые слова: *расчетная схема, автосамосвал*

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V.P. Franchuk[1], Dr. Sci. (Tech.), Prof.,
O.Yu. Zhyvotov[2]

1 – State Higher Educational Institution "National Mining University", Dnepropetrovsk, Ukraine, e-mail: franchuk@nmu.org.ua
2 – Yuzhnoye SDO, Dnepropetrovsk, Ukraine,
e-mail: a-zhivotov@ukr.net

ROTATING MOMENT FOR STATICALLY UNBALANCED ROTOR WITH ELASTIC SHAFT

В.П. Франчук[1], д-р. техн. наук, проф.,
О.Ю. Животов[2]

1 – Державний вищий навчальний заклад „Національний гірничій університет“, м.Дніпропетровськ, Україна,
e-mail: franchuk@nmu.org.ua
2 – ДП „КБ „Південне“, м.Дніпропетровськ, Україна,
e-mail: a-zhivotov@ukr.net

ПОВНИЙ ОБЕРТАЮЧИЙ МОМЕНТ ДЛЯ СТАТИЧНО НЕВРІВНОВАЖЕНОГО РОТОРА З ПРУЖНИМ ВАЛОМ

Purpose. Study of total rotational moment components for statically unbalanced system of the "rotor – supports" type with a bent elastic shaft.

Methodology. In this paper, theoretical approaches of classical mechanics were used to study a process of unbalanced rigid body rotation. These approaches are based on a methodology to determine axial moments of inertia of statically unbalanced system of the "rotor – supports" type with a rigid (non-deformable) shaft and an elastic (deformable) shaft.

Findings. The conducted researches show that a statically unbalanced rotating system of the "rotor – supports" type, with an elastic shaft, is a system with variable moments of inertia that depend on the rotor's rotation velocity. Therefore, moments that impede the rotor's rotation occur in the system. The obtained general equation of moments makes it possible to find total rotational moment of the drive in each particular case. Total rotational moment is defined as the sum of rotational moments needed to ensure the "rotor –supports" system's rotation and the production process.

Originality. The equation of rotation for the "rotor – supports" deformed system, which is given in the paper, reflects the behavioral features of a statically unbalanced rotor with a bent elastic shaft at different rotation velocities, such as the bent shaft's middle line position with respect to the rotational axis, the rotor's center of mass position with respect to the bent shaft's middle line and the rotational axis, and the rotor's angle of turn around the shaft's middle line. The equation of total rotational moment, which is obtained in this paper, confirms and explains why the rotational moment, with rotor's shaft bend, exceeds the rotational moment sufficient for rotor rotation with no shaft's bend.

Practical value. The equation of total rotational moment can be used as a master equation in computer software to calculate required rotational moment of the drive or to process experimental results.

Key words: *moment of inertia; "rotor – supports" system; damping forces; deformed system; total rotational moment*

Introduction. Various rotating assemblies and parts are widely used in the up-to-date mining machine designs. Even a small unbalance of the rotating assemblies and parts (rotors) may lead to the occurrence of undesirable vibrations, which are sometimes hazardous to the whole machine integrity. Hazardous vibrations cause in-