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# MATHEMATICAL MODEL OF MECHANICAL PARTS INTERCONNECTED ELECTRIC DRIVES OF HEAD AND LIFT IN ROCK DIGGER

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*The need for development of new approach to modelling lift and head electric drives of quarry digger in the process of digging has been justified. The differences of the mathematical model suggested from the traditional approach are shown. The disadvantages of existing control systems in digger electric drives are revealed, the method of their removing is proposed.*

The important link in technological chain of open-pit mining for minerals is rock excavation. At the same time, recently there has been a decrease in the basic technical and economic parameters of quarry equipment utilization, including quarry diggers. It can be partially explained by insufficient reliability of operated machines. Search of ways to further increase technological level of career diggers demands mathematical model development adequately describing their basic working processes.

In order to describe dynamic processes taking place during work of quarry diggers, kinematic schemes of its drives are represented in the form of multimass mechanical systems. At the same time, the approach according to which full design schemes of drives are simplified up to two-mass [1, 2] is rather widespread. Parameters of such design schemes are thought to be constant. However, in actuality, design scheme parameters of digger drives is changed during the work. Thus, application of the standard approach leads to mistakes at loading calculation of specific joints in quarry diggers and complicates researches addressed to increase their reliability.

The most complex operation made by quarry digger is the process of digging, from the point of view of external loadings occurrence. In this connection, description of quarry digger work exactly in this mode is rather actual problem.

During digging two electric drives take part simultaneously: lift and head, therefore at creating of the described mathematical model it is necessary to consider both kinematic features of electric drives and their mutual connection. Thus the greatest role will be played by change in parameters of electric drives, caused by change in geometrical position of ladle and lever.

The simplified design scheme of interconnected electric drives of lift and head is shown in Figure 1. On this scheme the following designations are accepted:  $J_n$  is the total reduced inertia moment of the first mass of head drive, including inertia moment of engine rotor, reducer and head drum;  $c_n$  is the reduced rope rigidity of head mechanism;  $J_l$  is the total reduced inertia moment of the first mass of lift drive, including inertia moment of engine rotor, reducer lift drum;  $c_l$  is the total reduced rope rigidity of lift mechanism;  $m_p$ ,  $m_k$ ,  $m_n$  are the masses of lever, ladle and rock respectively;  $M_n$  is the electromagnetic moment of head engine, reduced to speed of head drum;  $M_l$  is the electromagnetic moment of lift engine, reduced to speed of lift drum;  $\omega_n$ ,  $\omega_l$  are the angular speeds of the first weight of head and lift drive respectively;  $F_{ch}$ ,  $M_{cn}$  is the force and resistance moment of head and lift drive respectively;  $v_k$  is the linear speed of ladle head;  $\omega_k$  is the angular speed of ladle and lever.

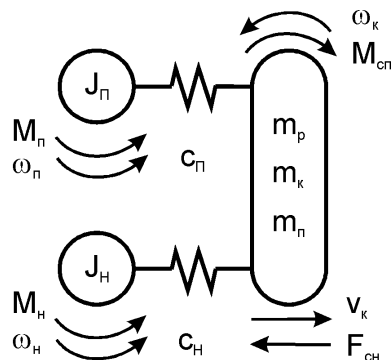
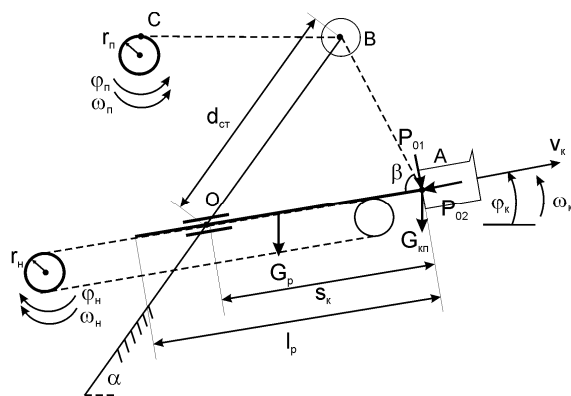


Fig. 1. Design scheme of lift and head electric drives in a quarry digger

It is necessary to consider the design of ladle suspension and lever mounting for definition of digger parameters de-

pending on their spatial position. Thus we shall believe that ladle is the material point, lever is the core with all the weight uniformly distributed on its axis. The constructive scheme used at model synthesis of interconnected electric drives of digger lift and head is shown in Figure 2. Here the following designations are taken:  $\alpha$  is the arrow inclination angle relative to horizon;  $\beta$  is the inclination angle of elevating rope relative to lever;  $r_1$  is the pressure of head drum;  $r_2$  is the radius of lift drum;  $l_p$  is the full length of lever;  $d_{cm}$  is the length of an arrow from the saddle bearing up to the head block;  $G_{kn}$  is the ladle weight with rock;  $G_p$  is the lever weight;  $P_{01}$  is the tangential force of resistance to cutting;  $P_{02}$  is the normal force of resistance to cutting;  $\varphi_n$ ,  $\varphi_p$  are the angular positions of head drum and lift drum respectively;  $s_k$  is the value of lever rundown;  $\varphi$  is the lever angular position.



**Fig. 2.** Schematic image of digger drives design

To obtain the equation of interconnected electric drives motion we shall take Lagrange's equation of the second sort:

$$\frac{d}{dt} \left( \frac{\partial L}{\partial \dot{q}_i} \right) - \frac{\partial L}{\partial q_i} = Q_i,$$

where  $L=W_k-W_n$  is the Lagrange's function;  $W_k$  is the kinetic energy;  $W_n$  is the potential energy;  $q_i$  is the generalized coordinate;  $Q_i$  is the generalized force. Solution of Lagrange's equation for mechanical subsystem of interconnected lift and head electric drives with selection of  $q_n, \varphi_n, s_n$  and  $\varphi_c$ , as generalized coordinates will be a system of equations:

$$\left. \begin{aligned} \frac{d\omega_n}{dt} &= \frac{M_n - r_n F_{12n}}{J_n}; \quad \frac{d\omega_n}{dt} = \frac{M_n - r_n F_{12n}}{J_n}; \\ dv_{\kappa} &= \frac{F_{12n} + m_p \omega_{\kappa}^2 \left( s_{\kappa} - \frac{l_p}{2} \right) + (m_{\kappa} + m_n) \omega_{\kappa}^2 s_{\kappa} - F_{cn} - F_{12n} \cos \beta - v_{\kappa} \frac{m_n}{dt}}{m_p + m_{\kappa} + m_n}; \\ d\omega_{\kappa} &= \frac{F_{12n} \sin \beta - M_{cn} - 2m_p \omega_{\kappa} v_{\kappa} \left( s_{\kappa} - \frac{l_p}{2} \right) - 2(m_{\kappa} + m_n) \omega_{\kappa} v_{\kappa} s_{\kappa} - \omega_{\kappa}^2 s_{\kappa} \frac{m_n}{dt}}{m_p \left( s_{\kappa} - \frac{l_p}{2} \right)^2 + m_p \frac{l_p^2}{12} + (m_{\kappa} + m_n) s_{\kappa}^2}; \\ \frac{d\varphi_n}{dt} &= \omega_n; \quad \frac{d\varphi_n}{dt} = \omega_n; \quad \frac{ds_{\kappa}}{dt} = v_{\kappa}; \quad \frac{d\varphi_{\kappa}}{dt} = \omega_{\kappa}, \end{aligned} \right\} \quad (1)$$

where designations are as follows:  $F_{12n}$  is the resilient force in a head rope;  $F_{12n}$  is the resilient force in lift rope. As it is possible to see, analyzing (1), besides resilient forces at work in head and lift electric drives of quarry digger there are forces of reaction in connections, and forces caused by change of drive parameters which create additional force, not considered at use of the traditional approach. Thus, the offered mathematical model re-

flects dynamic processes taking place in the mechanical part of head and lift drives more precisely.

Resilient force in a head rope  $F_{12H}$  is defined by the dependence:

$$F_{12_H} = c_H(r_H\varphi_H - (s_K - s_0)),$$

where  $s_0$  is the initial value of lever rundown. For definition  $F_{12n}$  it is necessary to consider in details possible conditions of a lift rope. With the change in ladle coordinates, leading to length reduction of the  $AB$  piece (Fig. 2), lift rope sagging occurs in a real digger. Thus, a lift rope can be only in stretched or in normal condition. In this connection the dependence of resilient force in a lift rope on coordinates of lift and head drives is nonlinear and can be described as:

$$\begin{cases} F_{12n} = c_n (r_n \varphi_n - (L_0 - L_{AB})), \text{ at } L_{AB} > L_0 - r_n \varphi_n; \\ F_{12n} = 0, \text{ at } L_{AB} \leq L_0 - r_n \varphi_n, \end{cases}$$

where  $L_0$  is the initial value of rope length in the piece  $AB$ . Current value of which is defined as:

$$L_{AB} = \sqrt{s_k^2 + d_{cm}^2 - 2s_k d_{cm} \cos(\alpha - \varphi_k)}.$$

It is necessary to consider when using this model (1), that weight of rock increases during the excavation process. Rock weight change  $m_n$  can be described by the dependence assuming, that ladle is fully filled at its raising to the maximal angle of elevation  $\varphi_{\max}$ . In this case:

$$\begin{cases} m_n = (\varphi_\kappa - \varphi_0) \frac{V_\kappa \rho_n}{\varphi_{\max} - \varphi_0}, & \text{at } \varphi_\kappa < \varphi_{\max}; \\ m_n = V_\kappa \rho_n, & \text{at } \varphi_\kappa \geq \varphi_{\max}, \end{cases}$$

where  $V_k$  is the ladle volume;  $\rho_n$  is the rock volumetric mass.

Besides rock mass, rigidity of resilient connections also changes during digging. This parameter depends on plenty of factors, most essential of which is rope lengthening. As the general length of head rope remains constant, the change of its rigidity will be insignificant whereas rigidity of lift rope changes in wider limits. Total rigidity of lift rope will be defined as [3]:

$$C_n = \frac{EF}{L_{AB} + L_{BC}},$$

where  $E$  is the rope resiliency module;  $F$  is the total area of all wires in a rope;  $L_{BC}$  is the length of a lift rope from the elevating drum up to the head block (Fig. 2).

During considered mathematical model operation it is necessary to consider changing character of electric drives loading. Magnitude of loading is influenced by spatial arrangement of ladle and lever, soil heterogeneity, coordination inconstancy of lift and head motions, casual changes in face height and turning angle of digger during work [4]. We shall believe that forces of resistance to cutting  $P_{01}$ ,  $P_{02}$  have casual character, ladle weight with rock  $G_{\kappa n}$  has determined character, and lever weight  $G_p$  is constant.

Head drive loading consists of composed resistance forces directed towards ladle lever. Considering direction of forces shown in Fig. 2 the equation of this loading has the view:

$$F_{cn} = P_{02} + (G_{kn} + G_p) \sin \varphi_k. \quad (2)$$

Lift drive loading is defined as the moment of force composed of resistance forces, directed perpendicular to ladle lever. The equation of this loading has the view:

$$M_{cn} = P_{01} s_k + G_p \cos \varphi_k \left( s_k - \frac{l_p}{2} \right) + G_{kn} \cos \varphi_k s_k. \quad (3)$$

For final description of interconnected lift and head electric drives loading we shall show likelihood character of resistance forces to cutting  $P_{01}$ ,  $P_{02}$ . As a rule, during calculations it is accepted that  $P_{02} = 0,1 P_{01}$  [5]. It is possible to write resistance to cutting for tangential force as:

$$P_{01}(t) = \bar{P}_{01}(t) + P_{01}''(t) + P_{01}^e(t), \quad (4)$$

where  $\bar{P}_{01}(t)$  is the mathematical expectation of resistance force to cutting;  $P_{01}''(t)$  is the low-frequency casual component characterizing variation of chip thickness;  $P_{01}^e(t)$  is the high-frequency casual component characterizing variation of rock resistance to excavation process. Each of these random variables is described by corresponding correlation function. Thus, the equations (2)–(4) completely define loading of lift and head electric drives in a quarry digger.

The analysis of interconnected lift and head drives work was made by the method of computer modeling by means of the program written in Delphi 7 environment. Parameters of quarry digger EKG8I with electric drive, built by the system TV-G-D, were used during modeling. Condition when the ladle lever is located vertically and the ladle stays on the ground was accepted as initial position. Modeling was conducted in two modes: in a mode of soil excavation at presence of sharp-changing component loading and increasing rock mass, and in a mode of loaded ladle lowering with constant rock mass. The results of modeling are presented in Fig. 3–6.

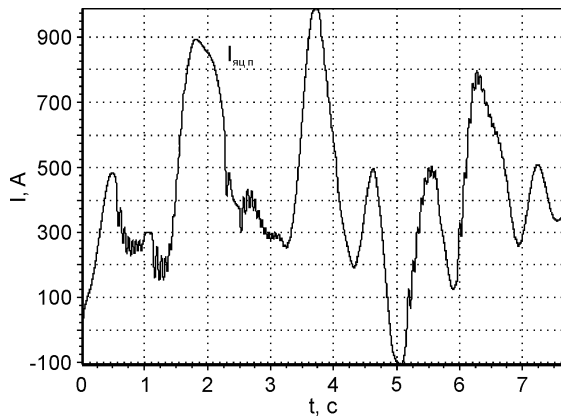


Fig. 3. Current of anchor circuit in the lift drive

Comparing transitional processes of anchor circuit current in the lift drive, obtained by means of modeling (Fig. 3) with similar transitional processes of a real digger presented in [1] it is possible to reveal their similarity which confirms adequacy of the developed mathematical model.

According to [2], regulators of control systems in digger drives are adjusted to technical optimum proceeding from parameters of electric part and engine moment of inertia. However presence of resilient connec-

tion or parameters change in drives are not considered. Hence, in digger real operating conditions proceeding in it processes will differ from technical optimum. Analyzing schedules of ladle position, shown in Fig. 4, it is possible to draw a conclusion that fluctuations caused by rope resilience occur at excavation operation and at loaded ladle movements in space. In Fig. 5 and 6 it is visible that elastic fluctuations, both in the head and in the lift rope have frequency nearby 2...3 Hz and amplitude of about tens of kN, which represents double excess of nominal loads. Presence of such fluctuations essentially increases fatigue wear process and, hence, reduces functioning reliability of rope and digger itself.

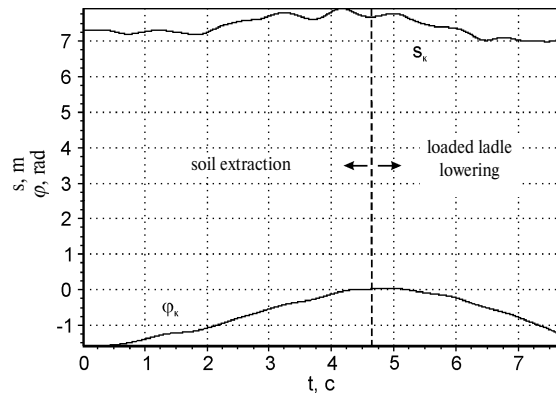


Fig. 4. Ladle position in space

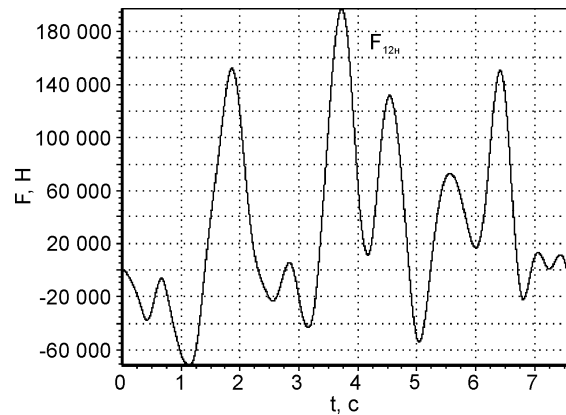


Fig. 5. Resilience force in the head drive

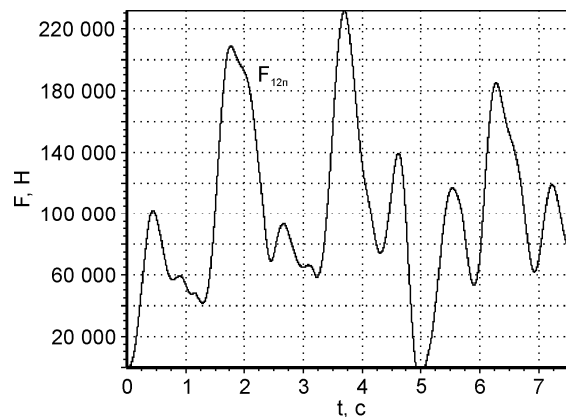


Fig. 6. Resilience force in the lift drive

From the aforesaid it is possible to draw a conclusion that application of subordinate regulation systems for digger electric drives can not provide demanded non-failure operation of its work. Replacement of subordinate regulation systems by more perfect systems

that would take into account dynamic processes is necessary in order to increase reliability of quarry diggers. It is necessary to use mathematical model of interconnected head and lift drives in diggers to synthesize such control systems.

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## ON IMPROVEMENT OF MATHEMATICAL MODEL OF LOOSE MATERIAL PRODUCTION

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*On the bases of studying the history of formation of stress fields and densities in fixed layer of cohesive compressed loose material, formation of secondary stress field, appearing in the substance layer under the external action, and determining the conditions of massive destruction the mathematical description of loose material production has been proposed.*

According to the modern ideas about the process, loose material (LM) production occurs in the form of stochastic shifts of substance blocks of arbitrary, constantly changing shape with low substance concentration on the boundaries between them [1–3]. Arcs formed at periodic formation and crushing of block structure may take any profiles, however, their average shape should be rather smooth. The process may be presented more simplified in the form of the system of periodically appearing and destructing arcs situated in the whole material volume. In this case lower arcs crushing precedes the crushing of upper ones and so the parameters of LM production are determined in general by the conditions of arc formation over the discharge outlet [1–5]. In connection with the examined mechanism of outflow process the supposition about the fact that substance motion in flared section of outlet zone occurs under the influence of stress field appearing due to discharge outlet opening is seemed to be physically based [2, 3, 6, 7].

The subject of the given investigation is a hydraulic or mass (in foreign terminology «*mass flow*») flow shape of loose material. Such shape is typical for well loose substance (as a rule for those having low humidity and content of fines) and at its presence the area of still material occurs only in the low part of the container (zones *CEM* and *NFD* in Fig. 1). According to the proposed design diagram it is accepted that in the upper cylindrical area *ABCD* the rate and volume density of the mate-

rial are constant and undergo changes only in flared section *CMND* of outflow zone. The validity of such supposition is confirmed by numerous experimental data of various authors.

To close the combined equation of motion and continuity of steady axis-symmetric outflow of compressible LM in spherical coordinates the stress tensor constituents are presented according to the hypothesis of P.I. Lukyanov [2] about stress redistribution in LM layer at discharge outlet opening, widely used at present by various authors. Expression (1) [7] describing the forces influence on substance layer having the shape of a spatial cone the special case of which is the ratio of Bussinesk-Frelich is used in the given paper [2]:

$$\nabla \sigma_r = \frac{\nu q \cos^{\nu-2} \theta}{2(1 - \cos^{\nu} \beta)}. \quad (1)$$

Here  $\Delta \sigma_r$  is the radial stress in material array;  $\nu$  is the coefficient of distribution ability;  $q$  is the vertical stress functioning on the level of discharge outlet plane;  $\beta$  is the slop angle of container walls to the vertical.

Taking into consideration the results of numerous investigations both theoretical and experimental having showed that loose material motion near the discharge outlet is close by the shape to the radial one [2–4, 6], the system may be considerably simplified. Along the lines with constant value of angle  $\theta$  for the case of radial motion we obtain the following combined equation: