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MATHEMATICAL MODELING OF GROUP DRIVES PUSH AND LIFT ELECTRIC EXCAVATOR MECHANISMS

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Purpose. An important feature of the excavator in the course of excavation is a change in a wide range of its kinematics scheme parameters and the impact on the formation of elastic connections of mechanical loads. To improve the research reliability in the reliability area of excavating equipment it is necessary to clarify the mathematical model of the mechanical excavator. **Methodology**. In the development of the excavator mathematical model it has been used classical mechanics laws: the law of conservation of angular momentum and the Huygens-Steiner theorem. It was performed the validation of the results. **Results**. On the basis of the excavator kinematics scheme and the relationship of its elements it was developed a mathematical model of the excavator mechanical part, which takes into account the change in the mutual arrangement of parts of the mechanical design and the weight of the bucket during the excavation. It takes into account the nonlinearity of mechanical parts, resulting slack rope hoist, and dependence on the height of the cutting force of the bucket lift. In the course of solving the developed mathematical model it was obtained numerical values of effort in the ropes, refined characteristics of the starting process of the excavator head drive. **Originality**. For the first time it was developed a mathematical model of the electric lifting mechanisms and dredge head, taking into account the effect of change in the location of the space elements mechanical design and elasticity of the links between the individual design elements. **Practical value**. Adjusted calculation of dynamic loads in excavator design elements allows one to increase the reliability of the equipment operation and will enhance the technical and economical parameters of excavating equipment operation. References 8, figures 6.

Key words: excavator, group electric drive, elasticity of ropes, mutual influence, mathematic model.

МАТЕМАТИЧНЕ МОДЕЛЮВАННЯ ГРУПОВОГО ЕЛЕКТРОПРИВОДА МЕХАНІЗМІВ НАПОРУ І ПІДЙОМУ ЕКСКАВАТОРА

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Зниження основних техніко-економічних показників використання обладнання кар'єрів, у тому числі й кар'єрних екскаваторів, частково пояснюється недостатньою надійністю експлуатованих машин. Покращення експлуатаційних показників екскаваторів неможливе без детального вивчення процесів формування навантажень у механічному обладнанні екскаватора, яке формує навантаження електроприводів. На основі аналізу кінематичної схеми роботи механізмів напору й підйому при копанні складено систему диференціальних рівнянь руху елементів механічної частини екскаватора. Розроблена система рівнянь ураховує пружність канатів, основні нелінійності при формуванні моментів та зусиль навантаження. Показано, що робота механізмів напору взаємоповя'зана, встановлено шляхи взаємного впливу. Обґрунтовано необхідність розглядання електроприводів напору й підйому як групового електроприводу.

Ключові слова: екскаватор, груповий електропривод, пружність канатів, взаємний вплив.

PROBLEM STATEMENT. An important link in the technology circle of the minerals in opened mining production is rock excavation. Recently, there is a decrease in basic technical and economic indicators of the quarries equipment, including mining shovels. This is partly explained by the lack of reliability of the exploited machine. Finding of ways for further improve of the technical level of mining shovels requires the development of mathematical model that adequately describes key working processes.

To describe the dynamic processes occurring during mining shovels operation, its kinematics of drive circuit are in the form of mechanical multi-mass systems. This is quite common approach, according to which full settlement schemes used to simplify the two-mass drive. Parameters of such calculation schemes are taken as constant. However, in reality, the parameters of the calculation schemes of excavator drives during operation vary. Thus, application of the standard approach leads to errors in determining the specific nodes of mining shovels loads and reduces the reliability in improving the reliability of their research.

The most complex operation undertaken by mining shovels, from the point of view of the emergence of external loads, is the digging process, and therefore very urgent task is to describe the operation of mining shovels in this mode.

The work target is development of mathematical model of the mechanical interlocking head mechanisms and lifting the excavator and analysis of dynamic modes of electric drives of mechanisms with taking into account the most important features of a mechanical part of excavator.

EXPERIMENTAL PART AND RESULTS OBTAINED. The digging process is ensured with coordinated work of two drives: pressure and lifting movement and, therefore, in the preparation of a mathematical model there must be taken into account both kinematics features of the electric-wire and their interrelation.

In this case the most important role will be played by a change of electric parameters caused by changes in the geometrical position of the bucket and lever handle in space. In the digging process the excavator's shovel completes complex translational-rotational motion in the process of digging, not only the position and the geometrical arrangement of the shovel excavator, also changes the weight of the bucket during digging ore.

Simplified design scheme of related electric drive of pressure and lifting movement is shown in Fig. 1.

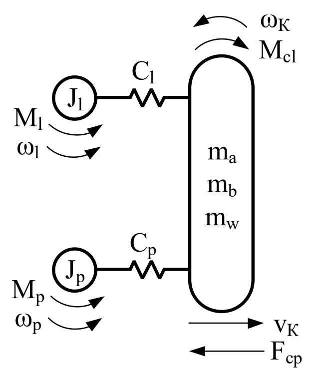


Figure 1 – Design diagram of the interaction of electric drives of pressure and lifting of the excavator

In this diagram, there are the following notations: J_p is the total reduced moment of inertia of the pressure drive, including the moment of inertia of the rotor, gearbox and the pressure drum; G_p are the ropes rigidity of pressure mechanism; J_l is the total reduced moment of inertia lifting actuator includes including moment inertia of the motor, the gearbox and the lifting drum; C_l are the ropes rigidity of lifting mechanism; m_a , m_b , m_w are the weight of lever handle, bucket and rock, respectively; M_p is the electromagnetic torque of pressure motor, reduced to the velocity of the pressure drum; M_l is the electromagnetic torque of lifting motor, reduced to the velocity of lifting drum; ω_p , ω_l are the angular velocity of the pressure drum and the lifting drum, respectively; F_{cl} , M_{cl} are the force and moment resistance about the head and lifting, respectively; v_k is the linear velocity of movement of the bucket; ω_k is the angular velocity of the bucket and lever handle.

To determine the parameters of an excavator, according to their attitude, you need to look at design and mounting of bucket and lever handle. We assume that the bucket is the material point, the lever handle is the core, and the entire mass is uniformly distributed along its axis. The design concept is shown in Fig. 2.

The following notations are: α is the boom angle relative to the horizon; β is the tilt angle of the rope with respect to the handle; r_p is the radius of the discharge drum; r_l is the radius lifting drum; l_p is the total length of the handle; d_m is the length of the arrows from the saddle to the main bearing unit; G_{bw} is the bucket weight with the ore; G_p is the weight of the handle; P_{01} is tangential force cutting resistance; P_{02} is the normal force of cutting resistance; φ_p , φ_l are the angular position of the drum and the pressure of the lifting drum, respectively; S_k is the value of run-handle; φ_k is the angular position of the handle.

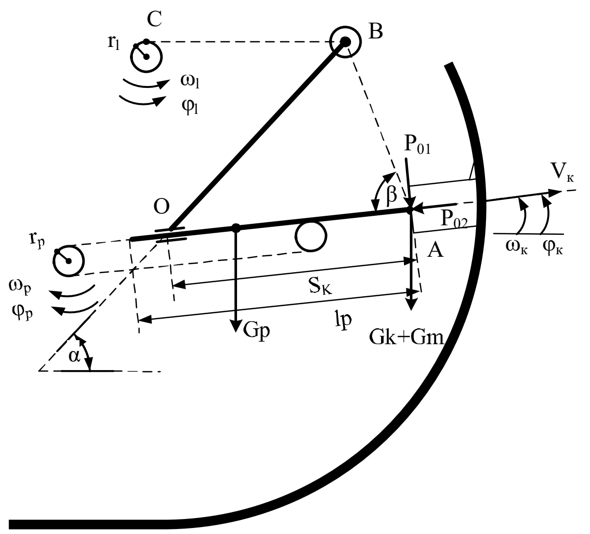


Figure 2 – Diagrammatic view of a schematic diagram of drives kinematics excavator

As the coordinates describing the state of excavator shovels will consider the following variables: ω_k is the handle angular velocity around axis, passing through the point O mount saddle bearing; φ_k is the handle angle relative to the same axis; υ_k is the linear velocity of the handle, S_k is the path traversed handle radius handle.

The equation of rotational motion handle dredge the axis passing through the point O from the general equation of dynamics of rotational motion will look like

$$M = \frac{d(J\omega)}{dt}. \qquad (1)$$

Applying differentiation general dynamic equations of rotational motion, we get the following:

$$\frac{d\omega}{dt} = \frac{M - \omega \frac{dJ}{dt}}{J}. \qquad (2)$$

Assuming that handle excavator homogeneous core, draw up the equation for the moment of inertia about the axis of the handle that passes through the point O. Using Huygens-Steiner theorem, we get the expression for the moment of inertia of the handle as the following:

$$J = m_a \frac{l_p}{12} + m_a \left(S_k - \frac{l_p}{2} \right)^2 + (m_a + m_b) S_k^2, \quad (3)$$

where the first term is the moment of inertia about the axis of the handle that passes through the center of gravity handle, the second term takes into account the actual position of the axis of rotation of the handle, the third term is the moment of inertia of the rock bucket, presented in the form of a point.

In calculating the first derivative of the moment of inertia of the handle should be noted that the weight of the excavator shovel digging in the process is not sustainable. It increases by weight of rocks collected in buckets excavator.

The first derivative of the handle moment of inertia is

$$\frac{dJ}{dt} = 2m_a \upsilon_k \left(S_k - \frac{l_p}{2} \right) + 2(m_a + m_b)\upsilon_k S_k + S_k^2 \frac{dm_w}{dt}. \quad (4)$$

Finally, the equation of rotational motion of the handle about the axis O is

$$\frac{d\omega_k}{dt} = \left[F_{12n} \sin(\beta) S_k - M_{cl} - 2m_a \omega_k \upsilon_k \left(S_k - \frac{l_p}{2} \right) - \right.$$
$$\left. - 2(m_a + m_w)\omega_k \upsilon_k S_k - \omega_k S_k^2 \frac{dm_w}{dt} \right] \times \quad (5)$$
$$\times \left(m_a \left(S_k - \frac{l_p}{2} \right)^2 + m_a \frac{l_p}{12} + (m_a + m_0) S_k^2 \right)^{-1}.$$

The equations of translational motion of the handle dredge the axis passing through the point O by universal dynamic equations of translational motion will look like

$$F = \frac{d(m\upsilon)}{dt}. \quad (6)$$

Applying differentiation, we get the following equation:

$$\frac{d\upsilon}{dt} = \frac{F - \upsilon \frac{dm}{dt}}{m}. \quad (7)$$

Given the effect of centrifugal force to the equation of a point as

$$F_{cf} = m_p \omega_k^2 \left(S_k - \frac{l_p}{2} \right) + (m_b + m_w)\omega_k^2 S_k. \quad (8)$$

Finally, the equations of translational motion of the handle about the axis OA will look like

$$\frac{d\upsilon_k}{dt} = \left[F_{12p} + m_a \omega_k^2 \left(S_k - \frac{l_p}{2} \right) + (m_b + m_w)\omega_k^2 S_k - \right.$$
$$\left. - F_{cp} - F_{12l} \cos(\beta) - \upsilon_k \frac{dm_w}{dt} \right] (m_b + m_w + m_a)^{-1}, \quad (9)$$

where F_{12p} is the resilient force in the pressure rope; F_{12n} is the resilient force in lifting rope.

Elastic force in the pressure rope F_{12p} determines by the following dependence:

$$F_{12p} = c_p \left(r_p \varphi_p - (S_k - S_0) \right), \quad (10)$$

where S_0 is the initial value coincidence handle. To determine F_{12n} need a closer look, you may condition the lifting rope.

With this change of position of the bucket, which reduces the length of the segment AB (Fig. 2), a real excavator is slack lifting rope. Thus, the lifting rope can only be stretched or in good condition.

In this regard, the dependence of elastic force on coordinate of lifting and pressure drives is non-linear and can be described as

$$\begin{cases} F_{12l} = c_l \left(r_l \varphi_l - (L_0 - L_{AB}) \right);\ L_{AB} > L_0 - r_l \varphi_l; \\ F_{12l} = 0;\ L_{AB} \le L_0 - r_l \varphi_l, \end{cases}$$

where L_0 is the initial value of the length of the rope on the segment AB, the current value is determined as

$$L_{AB} = \sqrt{S_k^2 + d_{cm}^2 - 2S_k d_{cm} \cos(\alpha - \varphi_k)}. \quad (11)$$

The angle β can be determined by the law of sines of triangle ABO (Fig. 2):

$$\frac{L_{AB}}{\sin(\alpha - \varphi_k)} = \frac{d_{cm}}{\sin(\beta)}.$$

And we have

$$\beta = \arcsin\left(\frac{d_{cm} \sin(\alpha - \varphi_k)}{L_{AB}} \right). \quad (12)$$

In above described model equations of mechanical excavator must be considered that ore mass during digging increases. The change in ore mass can be described addiction, which provides that final bucket filled with his ascent to the maximum lifting angle $\varphi_{\max}$. In this case

$$\begin{cases} m_w = (\varphi_k - \varphi_0) \dfrac{V_k \rho_n}{\varphi_{max} - \varphi_0},\ \varphi_k < \varphi_{max}; \\ m_w = V_k \rho_n,\ \varphi_k \ge \varphi_{max}, \end{cases}$$

where V_k is the bucket volume; ρ_n is the volume mass of rock.

The first derivative of the ore mass over time can be calculated as follows from the above equations or M_r,

$$\begin{cases} \dfrac{dm_w}{dt} = \omega_k \dfrac{V_k \rho_n}{\varphi_{max} - \varphi_0},\ \varphi_k < \varphi_{max}; \\ \dfrac{dm_w}{dt} = 0;\ \varphi_k \ge \varphi_{max}. \end{cases}$$

When using ehe mathematical model there must be take into account the changing nature of the electric load. The magnitude of the load affect the spatial arrangement and bucket handle heterogeneity of soil instability co-ordination and pressure lifting movements; random changes slaughter height and angle of rotation of excavator in its work. We assume that the resistances of cutting P_{01}, P_{02} are random, the weight of the bucket with the ore G_{bw} is deterministic in nature, and handle G_p constant weight.

For definitive description of interconnected electric load lifting and recovery will take into account the probabilistic nature of the resistance cutting P_{01}, P_{02}. Typically, the calculations take that $P_{02} = 0.1\,P_{01}$ [5]. For the tangential cutting resistance forces can write:

$$P_{01}(t) = \overline{P_{01}}(t) + P_{01}^{l}(t) + P_{01}^{h}(t), \qquad (13)$$

where $P_{01}(t)$ is the expectation of the resistance cutting; $P_{01}^{l}(t)$ is the random low-frequency component, which describes the variation of the thickness of the chip; $P_{01}^{h}(t)$ is the random high-frequency component that characterizes the excavation process variation resistance. Each of these random variables is described by the corresponding correlation function.

Assembly average of P_{01} cutting the resistance can be calculated by the following equation:

$$P_{01} = k_F bt10^6\ , H,$$

where b is the bucket width, m ; t is the thickness of the chips, m ; k_F is the resistivity of digging rocks. For rock, heavy ores with few cracks k_F is (0.38 – 0.5).

For excavator EKG–8I value of P_{01}, approximately equal to (180–200) кН.

In determining the resistance of cutting P_{01}, P_{02} there must be taking into account the current geometric position of the excavator handle because these efforts occur only in contact with the rock bucket. If the current value of the run-length handle is less than digging radius, these efforts take zero. Also please note that cutting force occurs only when bucket lifting. When lowering the bucket cutting efforts also take zero. Mathematically, these provisions can be recorded using the following irregularities:

$$P_{01}(t) = \begin{cases} \overline{P_{01}}(t) + P_{01}^{n}(t) + P_{01}^{в}(t), \neg\left(\left(S_k \le S_{k0}\right) \wedge \left(\omega_k \le 0\right)\right); \\ 0,\ \left(S_k \le S_{k0}\right) \wedge \left(\omega_k \le 0\right). \end{cases}$$

The load on the drive head consists of the components of the resistance directed along the handle of the bucket. Given the direction of forces shown in Fig. 2, the equation of the load will look like

$$F_{cn} = P_{02} + g\left(m_k + m_n + m_p\right)\sin(\varphi_k). \qquad (14)$$

The load on the drive recovery is defined as the moment of force, consisting of the components of the resistance directed perpendicular to the handle of the bucket. The equation of the load is

$$\begin{aligned} M_{cn} = {} & P_{01}S_k + gm_p\cos(\varphi_k)\left(S_k - \frac{l_p}{2}\right) + \\ & + g\left(m_k + m_n\right)\cos(\varphi_k)S_k. \end{aligned} \qquad (15)$$

Thus, equations (13)–(15) is completely determined by load pressure and electric lifting career excavator.

Finally, we get the following system of equations for a mechanical interlocking of electric pressure and lift (16).

Thus, the mechanical mechanisms of pressure and recovery seem excavator system of ordinary differential equations of eighths order. This system of equations is essentially nonlinear, with discontinuous right-hand sides, which requires for its solution specialized numerical methods, for example a Rozenbrok method.

$$\begin{cases} \dfrac{d\omega_p}{dt} = \dfrac{M_p - r_pF_{12p}}{J_p}; \\ \dfrac{d\omega_l}{dt} = \dfrac{M_l - r_lF_{12l}}{J_l}; \\ \dfrac{d\upsilon_k}{dt} = \dfrac{F_{12p} + m_p\omega_k^2\left(S_k - \dfrac{l_p}{2}\right) + \left(m_b + m_w\right)\omega_k^2S_k - F_{cn} - F_{12l}\cos(\beta) - \upsilon_k\dfrac{dm_n}{dt}}{m_b + m_w + m_a}; \\ \dfrac{d\omega_k}{dt} = \dfrac{F_{12n}\sin(\beta)S_k - M_{cl} - 2m_p\omega_k\upsilon_k\left(S_k - \dfrac{l_p}{2}\right) - 2\left(m_b + m_w\right)\omega_k\upsilon_kS_k - \omega_kS_k^2\dfrac{dm_w}{dt}}{m_p\left(S_k - \dfrac{l_p}{2}\right)^2 + m_p\dfrac{l_p}{12} + \left(m_p + m_0\right)S_k^2}; \\ \dfrac{d\varphi_l}{dt} = \omega_l; \\ \dfrac{d\varphi_p}{dt} = \omega_p; \\ \dfrac{d\varphi_k}{dt} = \omega_k; \\ \dfrac{dS_k}{dt} = \upsilon_k. \end{cases} \qquad (16)$$

As can be seen by analyzing above-stated system of equations, in addition to elastic forces at work of electric lifting and lifting career dredge the forces of reaction relations and forces caused by the change of parameters of drives that create extra efforts that are not accounted for using the traditional approach . Thus, the mathematical model more accurately describes the dynamic processes occurring in mechanical drives of lifting and pressure.

According to the initial position it was made a condition where the bucket handle is vertically bucket standing on the ground (Fig. 3).

In this case, the initial values of linear and angular velocity is zero, the initial angle φ_k is $-\pi/2$ rad. Initial freewheel handle S_k take equal radius digging RK.

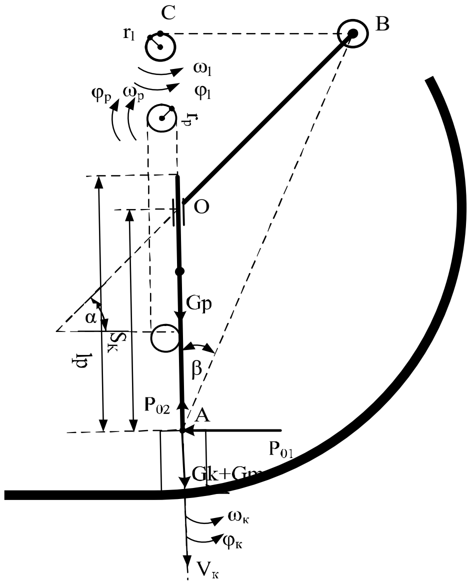


Figure 3 – Location of the mechanical system elements to dredge original position

Due to the high order system of equations, the presence of a significant number of non-linear dependencies implementation in a Simulink model block is inappropriate. For practical implementation of the above model in Simulink apparatus was used so-called S-functions of the second level. As a programming language was chosen m-language, which is a natural programming language for MATLAB.

The first two equations of system (16), which realize the connection of mechanical dredge with the drive motor has been implemented as usual block model.

The paper does not address the issue of differential equations assembly drive motor current position as previously known and carefully considered in the scientific and technical publications [9, 10].

When configuring the model specifications of electric motor of drive pressure type DE 812–U1 drive and lift-type DE 816–U2 were used.

In the simulation parameters used for career excavators EKG–8I (Tab. 1) with the DC drive.

Testing the mathematical model of the mechanical excavator performed excluding the electric drive at constant values of moments (Fig. 4).

Table 1 – Parameters of EKG–8I excavators

Parameter	Value
Bucket volume, m^3	8
The density of the ore, kg/m^3	2100
Pressure drum radius, m	0.7
Lifting drum radius, m	0.7
Pressure drum weight, kg	1460
Lifting drum weight, kg	1350
Excavator handle length, m	13.35
The angle to the horizontal boom, degrees	45
The length of the handle of the excavator, m	11.425
Lever handle weight, kg	10300
Bucket weight, kg	15480

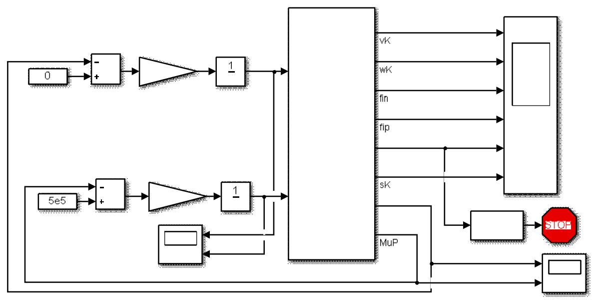
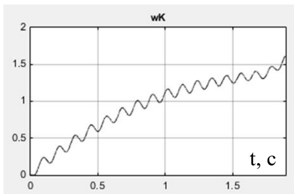
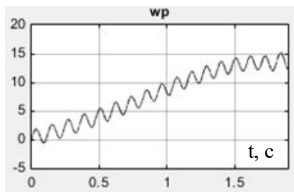


Figure 4 –Model of mechanical part related mechanisms of pressure and lifting of the excavator

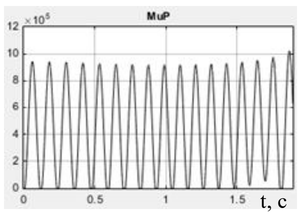
The results are presented in Fig. 5.



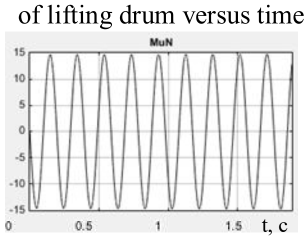
The angular velocity of the handle versus time



The angular velocity of lifting drum versus time



Elastic rope lifting moment versus time



Elastic rope moment in the pressure versus time

Figure 5 – Simulation of the mechanical excavator during the digging

The graph shows that the elastic ropes efforts to significantly affect the fluidity handle excavator. Elastic efforts to rope in 2–2.5 times higher than applications of the drive force.

Complement the mechanical model of excavator models drive motors. Ignore the fact that the recovery drive is implemented as double motor, use the equivalent model of one engine. It is permissible, because the connection shaft of the engine hard enough.

The model of the mechanical excavator considering electric is represented in Fig. 6.

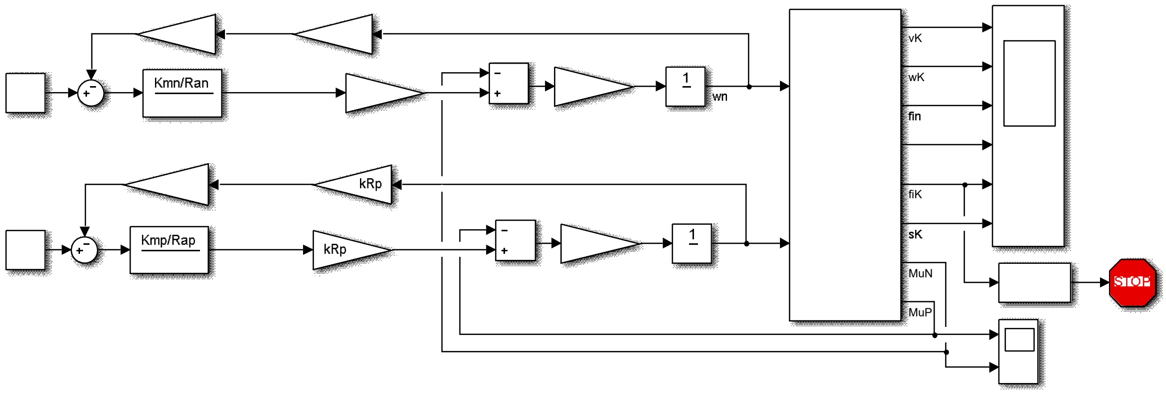


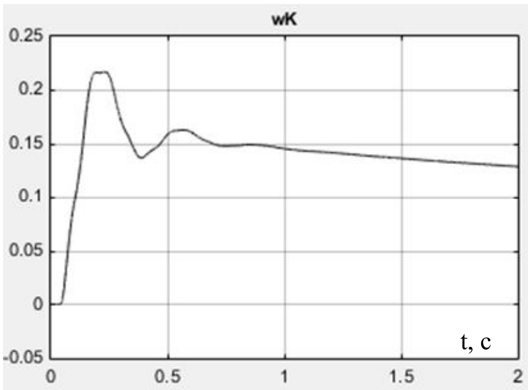
Figure 6 – Model of mechanical part related mechanisms of pressure and lifting of the excavator with electric drives

Results obtained using this model are presented in Fig. 7.

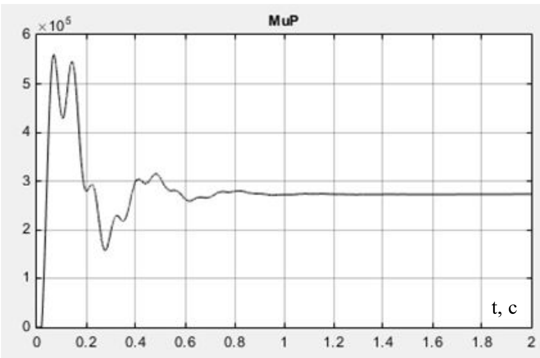
When you run the electric drive of lifting mechanism, elastic forces in ropes significantly reduces. Current anchor contains high fluctuations dictated by the influence of the resilient efforts in lifting rope.

CONCLUSIONS. Basing on the analysis of kinematics scheme of lift mechanisms and pressure excavator EKG–8I it was developed a system of differential equations of motion of mechanical elements, taking into account efforts in elastic ropes. The equations of derived mathematical model show that the mechanisms of pressure rise and affect each other. To improve the accuracy of analysis of effort and motion parameters of the mechanical part of excavator these mechanisms must be considered jointly, and the two electric drive mechanism should be seen as integrally-electric group.

Based on the proposed equations using MATLAB/Simulink software the model of interconnected electric pressure rise and excavators on the basis of the functioning of the mechanical excavator was developed. Adequacy test of mathematical model is done.



a)



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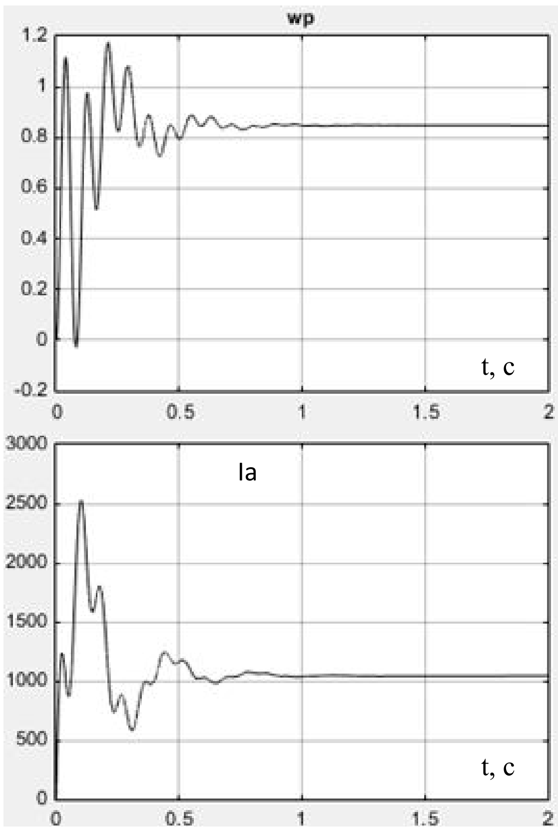

c)

Figure 7 – The results of modeling mechanical excavator with a glance of drives during digging:
a) the angular velocity of the handle V_k versus time; b) elastic rope in the lifting moment versus time;
c) the angular velocity of drum lifting and drive motor armature current versus time

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МАТЕМАТИЧЕСКОЕ МОДЕЛИРОВАНИЕ ГРУППОВОГО ЭЛЕКТРОПРИВОДА МЕХАНИЗМОВ НАПОРА И ПОДЪЕМА ЭКСКАВАТОРА

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Снижение основных технико-экономических показателей использования оборудования карьеров, в том числе и карьерных экскаваторов, частично объясняется недостаточной надежностью эксплуатируемых машин. Улучшение эксплуатационных показателей экскаваторов невозможно без детального изучения процессов формирования нагрузок в механическом оборудовании экскаватора, которое формирует нагрузки электроприводов. На основе анализа кинематической схемы работы механизмов напора и подъема при копании составлена система дифференциальных уравнений движения элементов механической части экскаватора. Разработанная система уравнений учитывает упругость канатов, основные нелинейности при формировании моментов и усилий нагрузки. Показано, что работа механизмов напора взаимосвязана, установлены пути взаимного воздействия. Обоснована необходимость рассмотрения электроприводов напора и подъема как группового электропривода.

Ключевые слова: экскаватор, групповой электропривод, упругость канатов, взаимное влияние.

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