Determination of Kinematic and Force Parameters of the Special Bucket Shovel for the Development of Large-Block Soils

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Abstract— **Traditional designs of working bodies of earthmoving machines are not adapted for the development of large-block soils mudslides and other debris in emergency situations, which leads to a decrease in their productivity. Therefore, it is necessary to create special working bodies of single-bucket excavators for the development mudflows with large-block inclusions allowing to expand the functionality of these machines. The subject of the study is to determine the kinematic and power parameters mechanism extension of the special working body of the excavator on the basis analytical studies and computer modeling. The proposed method of kinematic and force analysis a special working body of the excavator: algorithms and methods of defining the position coordinates of the links; equation closure independent contours of extension mechanism special jaws the excavator bucket in a vector form to determine the angular velocities and accelerations of the links; programs for determination of numerical values of kinematic and power parameters, selection of the block diagram, calculation sizes links working body of the excavator and numerical and graphic values of parameters are received.**

Index Terms— **large-block soil, mudflow soil medium, excavator bucket, mathematical model, kinematic and power analysis**

I. Introduction

A significant part of the territory of the Republic of Kazakhstan is located in the mudflow and earthquake zones, where there are more than 70 engineering anti-settlement defenses most of which were commissioned in the period 1974-1985 y. During this same time, in the basins of the rivers debris flow was more than 40 mudslides [1], many of which were accompanied by removal large quantities of solid material to the defense. Analysis of the granulometric composition of large-block soils mudflow deposits in the dams, mud catchers shows that boulders with a diameter $d \geq 1000$ mm are 2,88 %; small fractions $D = 0,1\text{-}5$ mm 41 %; large fragments $d>500$ mm 32,38 %. As a result, there is an accumulation of mudslide soil environment [2], which is removed mainly by earthmoving machines (excavators, loaders, bulldozers) having traditional working equipment, the structures which are not adapted for work in such conditions [3]. At the same time, the productivity of earthmoving machines (ZM) with traditional working bodies (RO), as a rule, is reduced by 20-25 % or more and in many cases they lose their performance. Operation of anti-mudflow structures, cleaning mudflows and blockages is impossible without the choice of mechanization means with effective working bodies adapted to the specific mudflow soil environment [4]. The most difficult to work are large-block soils, which require the creation of special RO in order to increase the efficiency of traditional ZM, widely used in emergency situations [5]. Therefore, the task of substantiating the parameters and creating new working equipment for special purpose single-bucket excavators (OE) for the development of mudflows with large-block inclusions, allowing to expand the functionality of these machines is relevant [6]. The analysis of information materials shows that progress in the field of development of OE develops in the direction of complication of their designs. An example is the working bodies of intensifying action, manipulative working equipment, dragline buckets, etc. On the basis analysis of literary sources, patent information the main directions and tendencies development of working bodies of OE and their opportunities for development mudflows and blockages are defined [7]. The revealed main directions of development earthmoving equipment showed that

Manuscript received July 15, 2019; revised May 21, 2020.

doi: 10.18178/ijmerr.9.6.813-824

actually there are no RO adapted to development large-block soils of mudflows. A common disadvantage of the traditional used and proposed working bodies of the ZM is low reliability in operation, the inability to capture and move large boulders contained in the mudflow mass, loss of time when performing rescue and recovery work to replace the working body [8].

II. Materials and Methods

A. The Idea of Work

It consists in search ways to improve the design of the working equipment of single-bucket excavators in order to expand the functionality these machines for the development of large-block soils of mudflows.

B. Method of Research

Methods of engineering forecasting, positions of theoretical mechanics, theory of machines and mechanisms, computer methods of modeling were applied during the research.

C. Research Result

To eliminate these drawbacks is possible by use of the working equipment hydraulic excavator with a hydraulically controlled jaw for the development of mud offsets containing large boulders up to 1 m (Fig. 1). The working equipment comprises a rectangular handle 1 with a bucket 3 and a jaw 4 pivotally (through the hinge 2) attached to it. The movements of the bucket 3 and the jaw 4 are controlled by control mechanisms, each consisting of hydraulic cylinders 5 (6), levers 7 (8) and rods 9 (10). The positions parts of the jaw control mechanism 4 are shown in brackets [9]. The bucket lever 7 and the jaw lever 8 are attached to the handle 1 via a hinge 11 above the handle; the bucket 3 and jaw 4 are equipped with 14 cutting teeth, the front side 12 of the bucket and the back side 13 of the jaw 4.

D. Purpose of Research

Substantiation of the main parameters of the new working equipment of the excavator equipped with a bucket with a hydro-controlled jaw, providing expansion functional, technological capabilities in the development large-block soils of mudflows [10]. With the bucket rigidly connect the coordinate system OXY (Fig. 2). Let with according to the coordinate system OXY , the coordinates of the joints rigidly connected to the bucket are given: $A(A')$, $F(F')$; $G(G')$). Let also know the linear dimensions of the mechanism that extends the jaw: $AB(A'B')$. Let's define relative to the coordinate system OXY the positions of all links, as well as the coordinate’s points of interest to us links. Consider the movement links of the hydraulic jaw bucket new working body shovel excavator without taking into account the forces that generate these movements [11]. The main purpose of the mechanism is to perform the necessary movements, which are described by means of its kinematic characteristics. These include generalized coordinates, the coordinates of links a points, their velocities and accelerations. Among the kinematic characteristics are those that do not depend on the law of motion of the initial links [12], but are determined only by the structure of the mechanism, the size of its links and in General depend on the generalized coordinates. These are functions of position, analogs of speeds and accelerations links of the mechanism and their points.

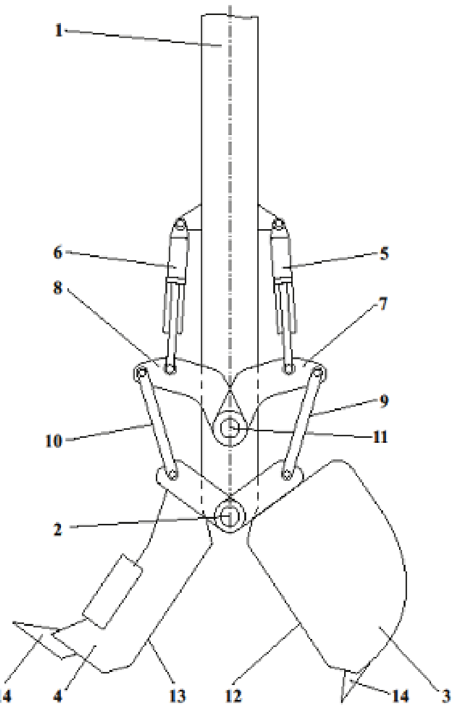

Figure 1. Working equipment of hydraulic excavator with hydraulic jaw.

As is known, the most common approaches to the kinematic analysis of plane lever mechanisms are based on the use method of independent vector contours [3]. General formulation of the positions flat lever mechanism is as follows: it is necessary to find the functions of the provisions output (slave) of links from the generalized coordinates input (leading) units with the given parameters and the positions of the outer joints on the basis of vector loops [4]. If there are several structural groups in the mechanism, the kinematic analysis is performed in the sequence of joining these groups. In this case, in addition to the coordinate systems associated with the individual links of the mechanism, for each structural group must be defined coordinate system, relative to which the links of the group forms a farm, i.e. has the number degrees of freedom equal to zero. To solve the problem about the provisions of the output (slave) units must be set kinematic scheme of the mechanism and position of the input (master) level with one degree of freedom, or position of an input (top), the links for mechanism with multiple degrees of freedom.

E. The Determination of the Positions of the Links

To determine the positions of all the bucket links, it is enough to set the generalized coordinate AB and constant parameters: the lengths of the links BF, BC, CF, CD, DG, GE, γ, rack coordinates $A(X_A, Y_A)$, $F(X_F, Y_F)$, $G(X_G, Y_G)$, stroke of the cylinder rod. To determine the coordinate of point B first find the distance AF:

$$AF = \sqrt{(X_F - X_A)^2 + (Y_F - Y_A)^2}$$

Find the angle $\theta_{AF} = arctg\left(\frac{Y_F - Y_A}{X_F - X_A}\right)$ and by the cosine theorem we find the angle $\varphi = \arccos\left(\frac{AB^2 + AF^2 - BF^2}{2 \cdot AB \cdot AF}\right)$, shown in Fig. 2. Then the angle:

$$\theta_{AB} = \theta_{AF} - \arccos\left(\frac{AB^2 + AF^2 - BF^2}{2 \cdot AB \cdot AF}\right)$$

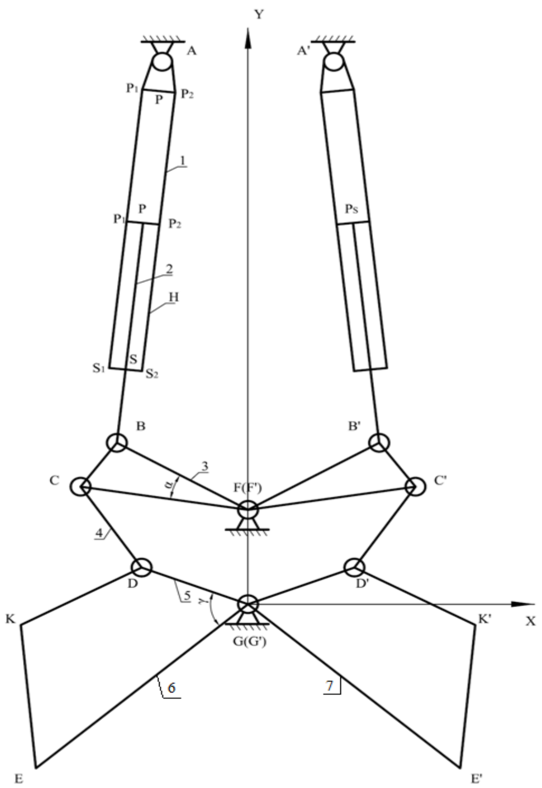

Figure 2. Kinematics of the working equipment hydraulic excavator with a hydraulically controlled jaw for the elimination consequences under emergency situations.

1-hydraulic cylinder, 2-rod, 3-upper lever, 4-rod, 5-lower lever, 6 - front jaw of the bucket, 7-rear jaw of the bucket.

Coordinates of a point B in a relatively fixed coordinate system GXY we define from the following relation:

$$\begin{Bmatrix} X_B \\ Y_B \end{Bmatrix} = \begin{Bmatrix} X_A \\ Y_A \end{Bmatrix} + \begin{Bmatrix} \cos(\theta_{AB}) & -\sin(\theta_{AB}) \\ \sin(\theta_{AB}) & \cos(\theta_{AB}) \end{Bmatrix} \begin{Bmatrix} AB \\ 0 \end{Bmatrix}$$

Where $AB = 1288 + 7{,}04 \cdot i$, $(i = 0,...,74)$.

To determine the coordinate's point C we first define from the cosine theorem a constant angle:

$$\alpha = \arccos\left(\frac{BF^2 + CF^2 - BC^2}{2 \cdot BF \cdot CF}\right)$$

and angle:

$$\theta_{FB} = arctg\left(\frac{Y_B - Y_F}{X_B - X_F}\right)$$

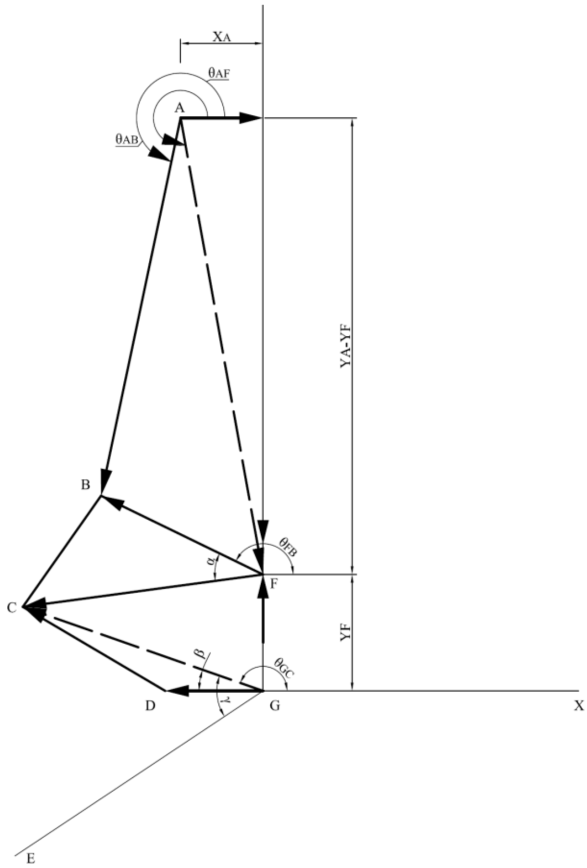

Figure 3. The design scheme of the link 1 hydro jaw to determine the coupling reaction in kinematic pairs.

Then we define the coordinates of point C from the following expression:

$$\begin{Bmatrix} X_C \\ Y_C \end{Bmatrix} = \begin{Bmatrix} X_F \\ Y_F \end{Bmatrix} + \begin{Bmatrix} \cos(\theta_{FB} + \alpha) & -\sin(\theta_{FB} + \alpha) \\ \sin(\theta_{FB} + \alpha) & \cos(\theta_{FB} + \alpha) \end{Bmatrix} \begin{Bmatrix} CF \\ 0 \end{Bmatrix}$$

In order to determine the coordinates of the point D, you must first determine the following values:

variable-length *CG:*

$$CG = \sqrt{(X_C - X_G)^2 + (Y_C - Y_G)^2}$$

Angle:

$$\theta_{GC} = arctg\left(\frac{Y_C - Y_G}{X_C - X_G}\right)$$

and constant angle β from the cosine theorem β:

$$\beta = \arccos\left(\frac{CG^2 + DG^2 - CD^2}{2 \cdot CG \cdot DG}\right)$$

Then, according coordinates of the points *D*, *E* are determined from the following relations:

$$\begin{Bmatrix} X_D \\ Y_D \end{Bmatrix} = \begin{Bmatrix} X_G \\ Y \end{Bmatrix} + \begin{Bmatrix} \cos(\theta_{GC}+\beta) & -\sin(\theta_{GC}+\beta) \\ \sin(\theta_{GC}+\beta) & \cos(\theta_{GC}+\beta) \end{Bmatrix} \begin{Bmatrix} DG \\ 0 \end{Bmatrix}$$

$$\begin{Bmatrix} X_E \\ Y_E \end{Bmatrix} = \begin{Bmatrix} X_G \\ Y_G \end{Bmatrix} + \begin{Bmatrix} \cos(\theta_{GC}+\beta+\gamma) & -\sin(\theta_{GC}+\beta+\gamma) \\ \sin(\theta_{GC}+\beta+\gamma) & \cos(\theta_{GC}+\beta+\gamma) \end{Bmatrix} \begin{Bmatrix} GF \\ 0 \end{Bmatrix}$$

Thus, we have determined the coordinates of points *D*, *E* relative to the local coordinate system XOY.

F. Determination of Speeds and Accelerations of Bucket Links

To determine velocities and accelerations of points in mechanisms, you must first define the analogues of the angular velocities and accelerations links [5], as well as analogues of angular velocities and accelerations points of the mechanism. Analogs of angular velocities and accelerations of mechanism links are found respectively by one-time and two-time differentiation of contour closure equations by generalized coordinate [6]. In this case, regardless class of the mechanism, we obtain a system of linear equations with respect to analogs of angular velocities or accelerations depending on the amount of differentiation on the generalized coordinate equation closure of independent circuits.

The leading link of the investigated mechanism (tong) is the piston on the connecting rod, the swinging cylinder *H* belonging to the link 1 (Fig. 2). In this mechanism, the generalized coordinate will be the variable distance AB.

Number of independent circuits (*k*) in the mechanism is determined by the topological Euler formula:

$$k = 7 - 6 + 1 = 2$$

where *p*, *n* – the number of kinematic pairs and links of the mechanism.

For the studying mechanism $p = 7$, $n = 6$, than $k = 7 - 6 + 1 = 2$, i. e. this mechanism has two independent circuits.

The first independent closed circuit (see Fig. 3):

$$\overline{AB} - \overline{BF} - (\overline{Y_A} - \overline{Y_F}) - \overline{X_A} = 0 \quad (1)$$

Projecting the equation of closure of vectors (1) on the *GXY* axis we obtain the following system of equation:

$$\begin{cases} AB\cos(\theta_{AB}) - BF\cos(\theta_{FB}) - \overline{X_A} = 0 \\ AB\sin(\theta_{AB}) - BF\sin(\theta_{FB}) - (\overline{Y_A} - \overline{Y_F}) = 0 \end{cases} \quad (2)$$

where $\theta_{FB} = arctg\left(\dfrac{Y_B - Y_F}{X_B - X_F}\right)$.

Differentiating the system (2) by the generalized coordinate AB we obtain:

$$\begin{cases} \cos(\theta_{AB}) - AB\sin(\theta_{AB})\dfrac{d\theta_{AB}}{dAB} + BF\sin(\theta_{FB})\dfrac{d\theta_{FB}}{dAB} = 0 \\ \sin(\theta_{AB}) + AB\cos(\theta_{AB})\dfrac{d\theta_{AB}}{dAB} - BF\cos(\theta_{FB})\dfrac{d\theta_{FB}}{dAB} = 0 \end{cases} \quad (3)$$

We give equations (3) in matrix form:

$$\begin{bmatrix} -AB\sin(\theta_{AB}) & BF\sin(\theta_{FB}) \\ AB\cos(\theta_{AB}) & -BF\cos(\theta_{FB}) \end{bmatrix} \begin{Bmatrix} \dfrac{d\theta_{AB}}{dAB} \\ \dfrac{d\theta_{FB}}{dAB} \end{Bmatrix} = \begin{Bmatrix} -\cos(\theta_{AB}) \\ -\sin(\theta_{AB}) \end{Bmatrix} \quad (4)$$

From the system of equations (4) we define analogs of angular velocities $\frac{d\theta_{AB}}{dAB}$ and $\frac{d\theta_{FB}}{dAB}$, angles θ_{AB} and θ_{FB}.

Differentiating the system of equation (2) by the generalized coordinate *AB* for the second time, we obtain the following system of equations:

$$\begin{cases} -\sin(\theta_{AB})\dfrac{d\theta_{AB}}{dAB} - \sin(\theta_{AB})\dfrac{d\theta_{AB}}{dAB} - AB\cos(\theta_{AB})\left(\dfrac{d\theta_{AB}}{dAB}\right)^2 - AB\sin(\theta_{AB})\dfrac{d^2\theta_{AB}}{dAB^2} + \\ BF\cos(\theta_{FB})\left(\dfrac{d\theta_{FB}}{dAB}\right)^2 + BF\sin(\theta_{FB})\dfrac{d^2\theta_{FB}}{dAB^2} = 0, \\ \cos(\theta_{AB})\dfrac{d\theta_{AB}}{dAB} + \cos(\theta_{AB})\dfrac{d\theta_{AB}}{dAB} - AB\sin(\theta_{AB})\left(\dfrac{d\theta_{AB}}{dAB}\right)^2 + AB\cos(\theta_{AB})\dfrac{d^2\theta_{AB}}{dAB^2} + \\ BF\sin(\theta_{FB})\left(\dfrac{d\theta_{FB}}{dFB}\right)^2 - BF\cos(\theta_{FB})\dfrac{d^2\theta_{FB}}{dAB^2} = 0; \end{cases}$$

or in matrix form:

$$\begin{bmatrix} -AB\sin(\theta_{AB}) & BF\sin(\theta_{FB}) \\ AB\cos(\theta_{AB}) & -BF\cos(\theta_{FB}) \end{bmatrix} \begin{Bmatrix} \dfrac{d^2\theta_{AB}}{dAB^2} \\ \dfrac{d^2\theta_{FB}}{dAB^2} \end{Bmatrix} =$$

$$=\begin{cases} 2\sin(\theta_{AB})\dfrac{d\theta_{AB}}{dAB}+AB\cos(\theta_{AB})\left(\dfrac{d\theta_{AB}}{dAB}\right)^2-BF\cos(\theta_{FB})\left(\dfrac{d\theta_{FB}}{dAB}\right)^2 \\ -2\cos(\theta_{AB})\dfrac{d\theta_{AB}}{dAB}+AB\sin(\theta_{AB})\left(\dfrac{d\theta_{AB}}{dAB}\right)^2-BF\sin(\theta_{FB})\left(\dfrac{d\theta_{FB}}{dAB}\right)^2 \end{cases} \quad (5)$$

From the systems of Eq. (5) we define analogs of angular accelerations $\dfrac{d^2\theta_{AB}}{dAB^2}$, $\dfrac{d^2\theta_{FB}}{dAB^2}$ angle adjustment θ_{AB} and θ_{FB}. The closure equation of the second independent circuit (see Fig. 2):

$$\overline{DG}+\overline{CD}-\overline{CF}-\overline{Y_F}=0$$

$$\begin{cases} DG\cos(\theta_{GD})+CD\cos(\theta_{DC})-CF\cos(\theta_{FC})=0 \\ DG\sin(\theta_{GD})+CD\sin(\theta_{DC})-CF\sin(\theta_{FC})-Y_F=0 \end{cases} \quad (6)$$

Where $\theta_{GD}=arctg\left(\dfrac{Y_D-Y_G}{X_D-X_G}\right)$, $\theta_{DC}=arctg\left(\dfrac{Y_C-Y_D}{X_C-X_D}\right)$, $\theta_{FC}=arctg\left(\dfrac{Y_C-Y_F}{X_C-X_F}\right)$.

We differentiate the system of equations (6) by the generalized coordinate *AB*, then we get:

$$\begin{cases} -DG\sin(\theta_{GD})\dfrac{d\theta_{GD}}{dAB}-CD\sin(\theta_{DC})\dfrac{d\theta_{DC}}{dAB}+CF\sin(\theta_{FC})\dfrac{d\theta_{FB}}{dAB}=0 \\ DG\cos(\theta_{GD})\dfrac{d\theta_{GD}}{dAB}+CD\cos(\theta_{DC})\dfrac{d\theta_{DC}}{dAB}-CF\cos(\theta_{FC})\dfrac{d\theta_{FB}}{dAB}=0 \end{cases}$$

$$\frac{d\theta_{FC}}{dAB}=\frac{d\theta_{FB}}{dAB}$$

These systems of equations are given in matrix form:

$$\begin{bmatrix} -DG\sin(\theta_{GD}) & -CD\sin(\theta_{DC}) \\ DG\cos(\theta_{GD}) & CD\cos(\theta_{DC}) \end{bmatrix}\begin{Bmatrix} \dfrac{d\theta_{GD}}{dAB} \\ \dfrac{d\theta_{DC}}{dAB} \end{Bmatrix}=\begin{Bmatrix} -CF\sin(\theta_{FC})\dfrac{d\theta_{FB}}{dAB} \\ CF\cos(\theta_{FC})\dfrac{d\theta_{FB}}{dAB} \end{Bmatrix} \quad (7)$$

From (7) we define analogs of angular velocities $\dfrac{d\theta_{GD}}{dAB}$ and $\dfrac{d\theta_{DC}}{dAB}$, angles θ_{GD} and θ_{DC}.

The second time differentiating equations (6) on the generalized coordinate AB, we get:

$$\begin{cases} -DG\cos(\theta_{GD})\left(\dfrac{d\theta_{GD}}{dAB}\right)^2-DG\sin(\theta_{GD})\dfrac{d^2\theta_{GD}}{dAB^2}-CD\cos(\theta_{DC})\left(\dfrac{d\theta_{DC}}{dAB}\right)^2-CD\sin(\theta_{DC})\dfrac{d^2\theta_{DC}}{dAB^2}+ \\ CF\cos(\theta_{FC})\left(\dfrac{d\theta_{FB}}{dAB}\right)^2+CF\sin(\theta_{FC})\dfrac{d^2\theta_{FB}}{dAB^2}=0, \\ -DG\sin(\theta_{GD})\left(\dfrac{d\theta_{GD}}{dAB}\right)^2+DG\cos(\theta_{GD})\dfrac{d^2\theta_{GD}}{dAB^2}-CD\sin(\theta_{DC})\left(\dfrac{d\theta_{DC}}{dAB}\right)^2+CD\cos(\theta_{DC})\dfrac{d^2\theta_{DC}}{dAB^2}+ \\ CF\sin(\theta_{FC})\left(\dfrac{d\theta_{FB}}{dAB}\right)^2-CF\cos(\theta_{FC})\dfrac{d^2\theta_{FB}}{dAB^2}=0; \end{cases}$$

or in matrix form:

$$\begin{bmatrix} -DG\sin(\theta_{GD}) & -CD\sin(\theta_{DC}) \\ DG\cos(\theta_{GD}) & CD\cos(\theta_{DC}) \end{bmatrix}\begin{Bmatrix} \dfrac{d^2\theta_{GD}}{dAB^2} \\ \dfrac{d^2\theta_{DC}}{dAB^2} \end{Bmatrix}=$$

$$=\begin{Bmatrix} DG\cos(\theta_{GD})\left(\dfrac{d\theta_{GD}}{dAB}\right)^2+CD\cos(\theta_{DC})\left(\dfrac{d\theta_{DC}}{dAB}\right)^2-CF\cos(\theta_{FC})\left(\dfrac{d\theta_{FB}}{dAB}\right)^2-CF\sin(\theta_{FC})\dfrac{d^2\theta_{FB}}{dAB^2} \\ DG\sin(\theta_{GD})\left(\dfrac{d\theta_{GD}}{dAB}\right)^2+CD\sin(\theta_{DC})\left(\dfrac{d\theta_{DC}}{dAB}\right)^2-CF\sin(\theta_{FC})\left(\dfrac{d\theta_{FB}}{dAB}\right)^2+CF\cos(\theta_{FC})\dfrac{d^2\theta_{FB}}{dAB^2} \end{Bmatrix} \quad (8)$$

Solving the system of Eq. (8) we define analogs of angular accelerations $\frac{d^2\theta_{GD}}{dAB^2}$, $\frac{d^2\theta_{DC}}{dAB^2}$ angle adjustment θ_{GD} and θ_{DC}. The coordinates of the point E are determined by the following expressions:

$$\begin{aligned} X_E &= GE\cos(\theta_{GE}) \\ Y_E &= GE\sin(\theta_{GE}) \end{aligned} \quad (9)$$

where $\theta_{GE} = arctg\left(\frac{Y_E - Y_G}{X_E - X_G}\right)$.

Differentiating (9) in time we find the velocity of the point E:

$$\begin{cases} \dfrac{dX_E}{dt} = -GE\sin(\theta_{GE})\dfrac{d\theta_{GD}}{dAB}\dfrac{dAB}{dt} \\ \dfrac{dY_E}{dt} = +GE\cos(\theta_{GE})\dfrac{d\theta_{GD}}{dAB}\dfrac{dAB}{dt} \end{cases} \quad (10)$$

The velocity modulus of point E will be equal to:

$$\vartheta_E = \sqrt{\vartheta_{x_E}^2 + \vartheta_{y_E}^2}$$

Differentiating (10) in time we find the acceleration of the point E:

$$\begin{cases} a_{x_E} = \dfrac{d^2X_E}{dt^2} = -GE\cos(\theta_{GE})\left(\dfrac{d\theta_{GD}}{dAB}\dfrac{dAB}{dt}\right)^2 - GE\sin(\theta_{GE})\dfrac{d^2\theta_{GD}}{dAB^2}\left(\dfrac{dAB}{dt}\right)^2 - \\ GE\sin(\theta_{GE})\dfrac{d\theta_{GD}}{dAB}\dfrac{d^2AB}{dt^2} \\ a_{y_E} = \dfrac{d^2Y_E}{dt^2} = -GE\sin(\theta_{GE})\left(\dfrac{d\theta_{GD}}{dAB}\dfrac{dAB}{dt}\right)^2 + GE\cos(\theta_{GE})\dfrac{d^2\theta_{GD}}{dAB^2}\left(\dfrac{dAB}{dt}\right)^2 + \\ GE\cos(\theta_{GE})\left(\dfrac{d\theta_{GD}}{dAB}\right)^2\dfrac{d^2AB}{dt^2} \end{cases}$$

The acceleration modulus of point E is equal to:

$$a_E = \sqrt{a_{x_E}^2 + a_{y_E}^2}$$

Computer modeling obtained dependencies according to the developed program was carried out in two modes:

- Operation of the hydraulic jaw extension mechanism to capture large-block boulders in the grapple mode;
- Joint operation of the hydraulic jaw extension mechanism in the mode of a conventional backhoe bucket.

III. Results and Discussion

Maple 18 software was used to obtain numerical results. The Essential Tool for Mathematics and Modeling [4]. The calculation program is developed and the kinematic and power characteristics mechanism of the hydraulic jaw of a special bucket hydraulic excavator are obtained in numerical and graphical form. This mechanism has the following parameters:

$$AB = 1288 + 7{,}04\cdot i,\ (i = 0,\ldots,74)$$

$BF = 500$ mm, $BC = 200$ mm, $CF = 538{,}51$ mm, $X_A = -427$ mm, $Y_A = 2150$ mm, $X_F = 0{,}0$ mm, $Y_F = 500$ mm, $X_G = 0{,}0$ mm, $Y_G = 0{,}0$ mm.

$$\gamma = \frac{2\cdot p_i \cdot 30}{360} = 0{,}5235987758 \text{ rad.}$$

As an example, Figs. 4, 5 and 6 present a graphical interpretation dependence of the projection angular acceleration of the point E on the X and Y axis and the acceleration modulus on the stroke of the cylinder rod. Analysis of the dependence presented in Figures 4-6 shows that the proposed design of hydraulically controlled jaws special bucket working equipment shovel excavator allows you to capture blocks of stone the size of 1 meter at the maximum opening of the jaw, which is provided when the stroke of the hydraulic cylinder 280 mm. Numerical values of kinematic parameters are obtained: coordinates of points, linear and angular velocities, accelerations links of the mechanism under study, which allowed to choose a block diagram and determine the size links mechanism of the hydro managed jaw of the bucket OE.

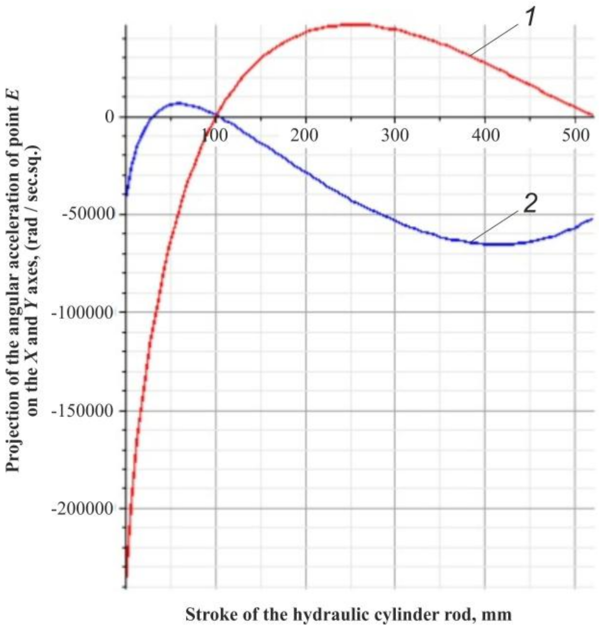

Figure 4. Dependence projection of the angular acceleration point E on the X(1) and Y(2) axis on the stroke of the cylinder rod.

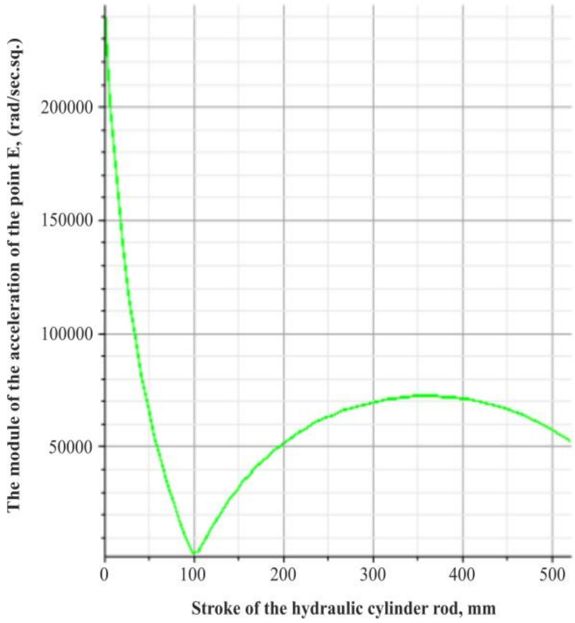

Figure 5. Dependence acceleration module point E on the stroke of the cylinder rod.

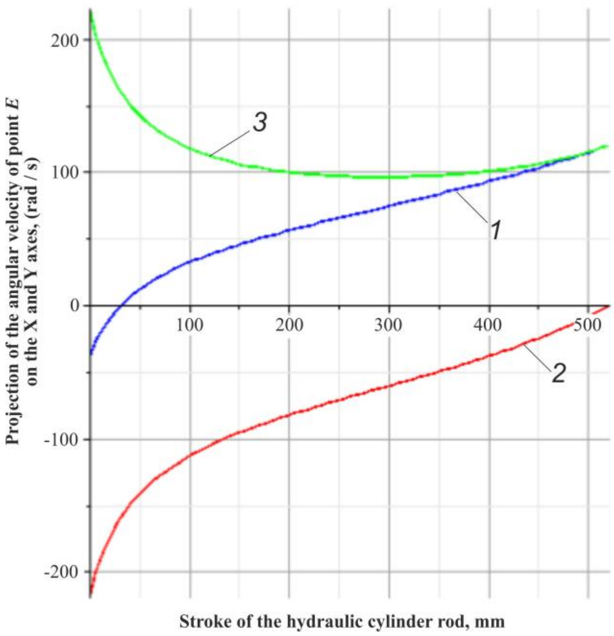

Figure 6. Dependence projection of the angular velocity of the point E on the axis X(1) and Y(2) and the velocity modulus (3) on the stroke of the cylinder rod.

Power analysis it is assumed that all the spatial elements of the working equipment can be represented as a set of absolutely solid bodies performing translational or rotational motion in the local or basic coordinate system associated with the axis rotation of the rotary platform and the support surface of the machine. In the analysis of the calculated positions, the digging force limits are taken into account for the stability OE equipment and the setting valves of the hydromechanisms equipment. To conduct a force analysis mechanism extension of the bucket jaw, consider it separately, applying external forces in accordance with Fig. 7. P_{EX}, P_{EY} - projections of the forces of useful resistance acting at the point E of the movable jaw. We form the design scheme of each link separately, applying all the external forces acting, we cut out the part shown in Fig. 7 from the tong.

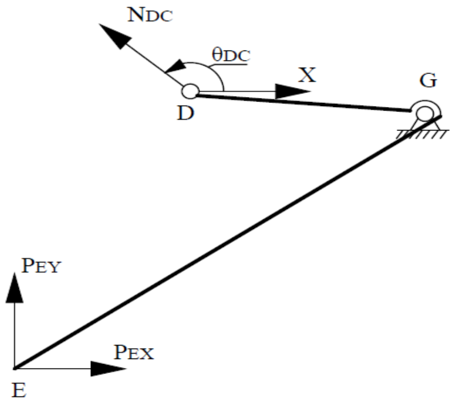

Figure 7. Scheme for determining the longitudinal force N_{DC} in the rod *DC*.

The action of the discarded part is replaced by a longitudinal force N_{DC}. Since the tong is a hinge rod system, so only the longitudinal force will appear in the DC rod.

Let the forces act at the point *E*, P_{E_X} and P_{E_Y}.

The sum of the moments all the forces acting on the system should be zero relative to the point *G*:

$$-N_{DC}\cos(\theta_{DC})(Y_D-Y_G)+N_{DC}\sin(\theta_{DC})(X_D-X_G)-$$
$$-P_{E_X}(Y_E-Y_G)+P_{E_Y}(X_E-X_G)=0$$

Where

$$N_{DC}=\frac{-P_{E_X}(Y_E-Y_G)+P_{E_Y}(X_E-X_G)}{\cos(\theta_{DC})(Y_D-Y_G)-\sin(\theta_{DC})(X_D-X_G)} \quad (11)$$

Cut out the part shown in Fig. 8. $N_{CD}=N_{DC}$ was found in the expression (11). The action of the upper discarded part of the tong is replaced by a longitudinal force N_{BP}.

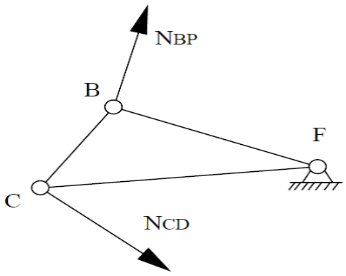

Figure 8. Scheme for determining the longitudinal force N_{BP} relative to the point F.

We write the equation of the sum moments of all forces acting on the system with respect to the point *F*:

$$-N_{BP}\cos(\theta_{BP})(Y_B-Y_F)+N_{BP}\sin(\theta_{BP})(X_B-X_F)-$$
$$-N_{CD}\cos(\theta_{CD})(Y_C-Y_F)+N_{CD}\sin(\theta_{CD})(X_C-X_F)=0 \quad (12)$$

where

$$\theta_{BP}=arctg\left(\frac{Y_P-Y_B}{X_P-X_B}\right),$$

$$\theta_{CD}=arctg\left(\frac{Y_D-Y_C}{X_D-X_C}\right).$$

From (12) we define:

$$N_{BP}=\frac{-N_{CD}\cos(\theta_{CD})(Y_C-Y_F)+N_{CD}\sin(\theta_{CD})(X_C-X_F)}{\cos(\theta_{BP})(Y_B-Y_F)-\sin(\theta_{BP})(X_B-X_F)}$$

Cut out the part shown in Fig. 9. The action of the upper discarded part is replaced by a longitudinal force N_{CB}.

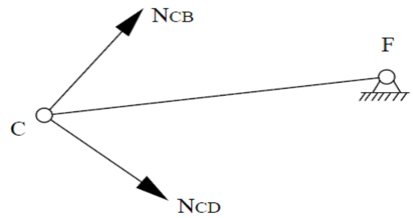

Figure 9. Scheme for determining the longitudinal force N_{CB} relative to the point F.

Write down the sum moments of all the forces acting on the system relative to the point *F*:

$$-N_{CD}\cos(\theta_{CD})(Y_C-Y_F)+N_{CD}\sin(\theta_{CD})(X_C-X_F)-$$
$$-N_{CB}\cos(\theta_{CB})(Y_C-Y_F)+N_{CB}\sin(\theta_{CB})(X_C-X_F)=0$$

from where

$$N_{CB}=\frac{-N_{CD}\cos(\theta_{CD})(Y_C-Y_F)+N_{CD}\sin(\theta_{CD})(X_C-X_F)}{\cos(\theta_{CB})(Y_C-Y_F)-\sin(\theta_{CB})(X_C-X_F)}$$

where $\theta_{CB}=arctg\left(\frac{Y_B-Y_C}{X_B-X_C}\right)$.

Let's project all the forces acting on the node C on the GX axis (Fig. 10).

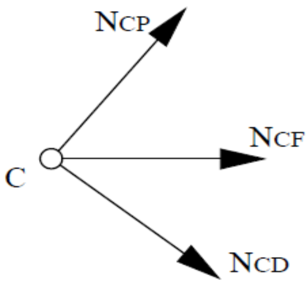

Figure 10. Scheme for determining the longitudinal force N_{CF} on the node C axes GX.

$$N_{CB}\cos(\theta_{CB})+N_{CF}\cos(\theta_{CF})+N_{CD}\cos(\theta_{CD})=0$$

Where

$$N_{CF}=\frac{-N_{CB}\cos(\theta_{CB})-N_{CD}\cos(\theta_{CD})}{\cos(\theta_{CF})}$$

where $\theta_{CF}=arctg\left(\frac{Y_F-Y_C}{X_F-X_C}\right)$.

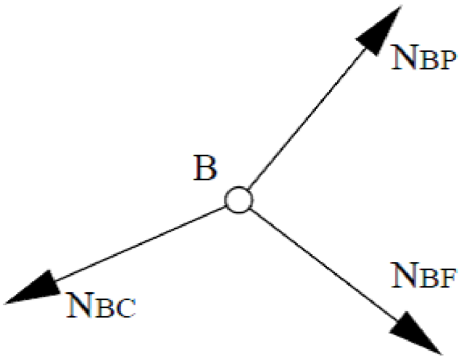

Figure 11. Scheme for determining the longitudinal force N_{BF} on the node B axes GX.

Projecting all the forces acting in node B (Fig. 11) on the GX axis, we obtain:

$$N_{BP}\cos(\theta_{BP})+N_{BF}\cos(\theta_{BF})+N_{BC}\cos(\theta_{BC})=0$$

where $\theta_{BF}=arctg\left(\frac{Y_F-Y_B}{X_F-X_B}\right)$, $\theta_{BC}=arctg\left(\frac{Y_C-Y_B}{X_C-X_B}\right)$

from where

$$N_{BF}=\frac{-N_{BP}\cos(\theta_{BP})-N_{BC}\cos(\theta_{BC})}{\cos(\theta_{BF})}$$

Let us project, (Fig. 12) on a movable coordinate system. Then have:

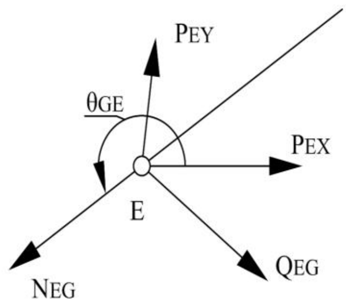

Figure 12. Scheme for determining the longitudinal N_{EG} and transverse Q_{EG} forces in the stem GE.

$$N_{EG}=P_{E_x}\cos(\theta_{GE})+P_{E_y}\sin(\theta_{GE}),$$
$$Q_{EG}=-P_{E_x}\sin(\theta_{GE})+P_{E_y}\cos(\theta_{GE})$$

I. e. we define longitudinal and transverse forces in the GE rod.

Similarly, we define the longitudinal and transverse forces in the rod DG (Fig. 13):

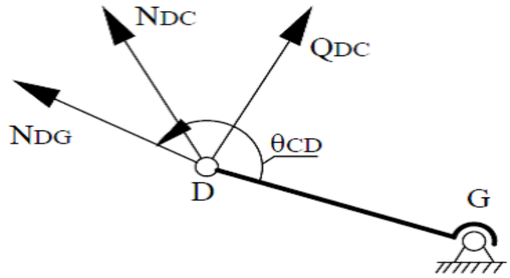

Figure 13. Scheme for determining the longitudinal N_{EG} and transverse Q_{EG} forces in the stem DG.

$$R_{DC_x}=N_{DC}\cos(\theta_{DC}),R_{DC_y}=N_{DC}\sin(\theta_{DC}),$$
$$\theta_{GD}=arctg\left(\frac{Y_D-Y_G}{X_D-X_G}\right)$$

Then have:

$$N_{DG}=-R_{DC_x}\cos(\theta_{GD})-R_{DC_y}\sin(\theta_{GD})$$

$$Q_{DG}=R_{DC\,x}\sin(\theta_{GD})-R_{DC_y}\cos(\theta_{GD})$$

The technique of kinetostatic analysis and determination coupling reaction in kinematic pairs mechanism of hydraulic jaw opening of the excavator bucket arising from external loads acting on the links of the mechanism, which allows to determine the coupling reaction in kinematic pairs, is proposed. Graphical interpretation obtained numerical values of the longitudinal and lateral forces, projections moment depending on the stroke of the hydraulic cylinder mechanism extension of hydraulically controlled jaws special bucket hydraulic excavator in the mode of operation grab shown in Figs. 14. The obtained dependences make it possible to determine the coupling reactions in kinematic pairs of ladle attachments with an opening hydro-controlled jaw arising from external loads acting on the links of the mechanism and to perform strength calculation links of the mechanism under study.

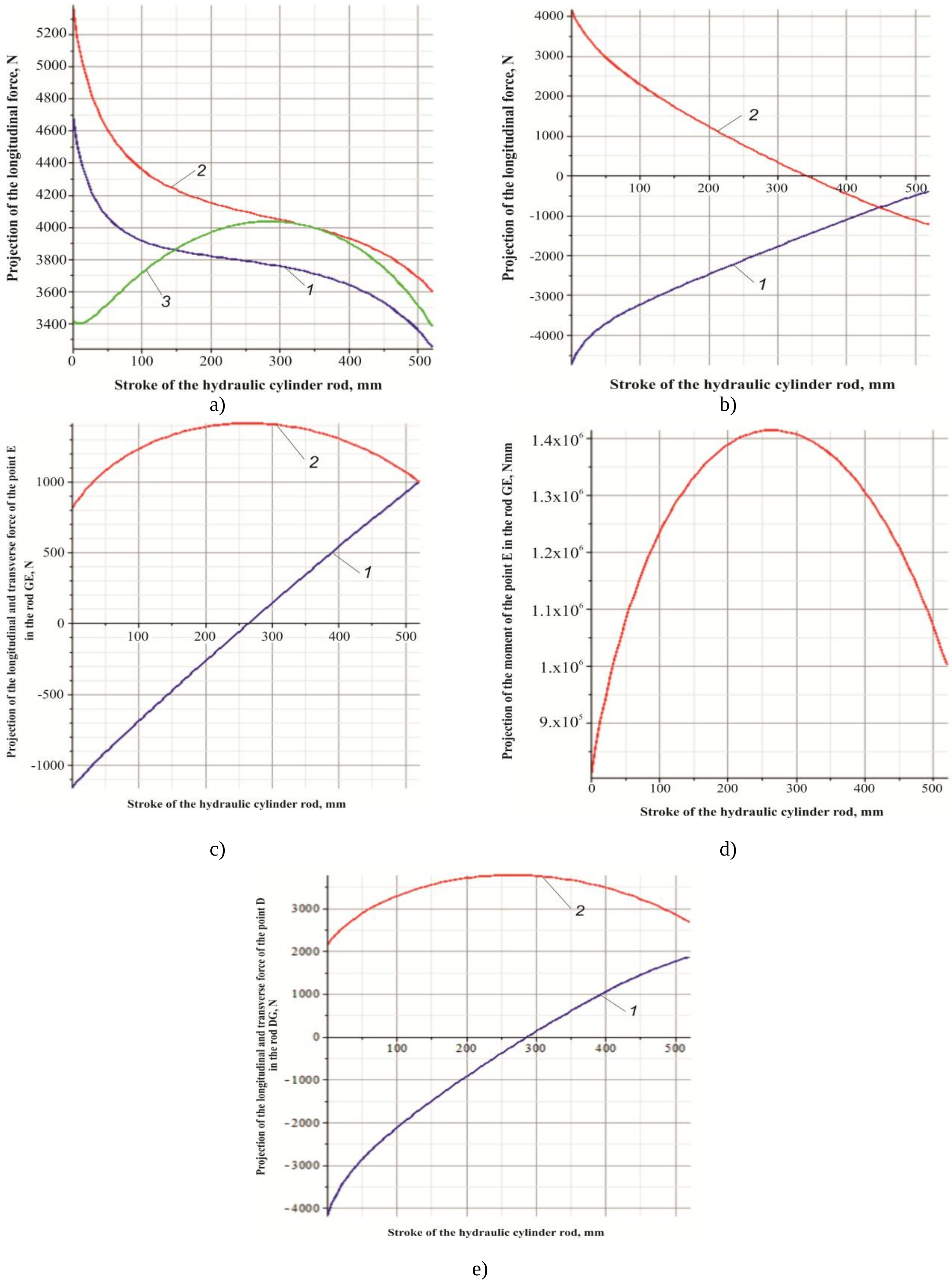

Figure 14. a) Dependence of the projection longitudinal force point G (1), point (2) and point (C) of the rod stroke of the hydraulic cylinder; b) Dependence projection of the longitudinal force point In (1) and point (C) on the axis GX of the rod stroke hydraulic cylinder; c) Dependence projection of the longitudinal force (1) and lateral forces (2) point E in the GE core from stroke hydraulic cylinder; d) Dependence projection of the point E in the GE core from stroke hydraulic cylinder; e) the Dependence projection of the longitudinal force (1) and lateral forces (2) point D in terminal DG from stroke hydraulic cylinder.

It is shown that the traditional working bodies of earthmoving machines are not adapted to the development of large-block soils of mudflows. A concrete example of the design special bucket of a single-bucket excavator and its kinematic schemes, providing a reliable capture of large-scale boulders up to 1 m. the Design of the proposed working body is simple, easily controlled, technological in manufacture, non-material and energy-intensive. The technical result from the use of the proposed technical solution is to increase the efficiency of the hydraulic excavator with a hydraulically controlled jaw due to the coordinated synchronous control of the moving parts of work equipment (bucket and jaw), a reliable fixation of the moment of transition from one operation to another and further coordinated-synchronous operation in each mode. Kinematic and power analysis of the proposed design of a special bucket is performed. Computer modeling of the obtained dependences according to the developed program was carried out in two modes: the operation of the hydraulic jaw extension mechanism to capture large-block boulders in the grapple mode; the joint operation of the hydraulic jaw extension mechanism in the mode of a conventional backhoe bucket. The direction of future research is the interaction working body of the earthmoving machine and a large stone object in static and in motion, taking into account the local and global representation of the surfaces of the captured object.

IV. Conclusion

The proposed method of kinematic and force analysis allows us to determine the geometric, kinematic and force parameters mechanism extension of the hydraulic jaw of a special working body of a single-bucket excavator for the development of large-block soils. The algorithm for the kinematic analysis of the working body hydraulically controlled jaw bucket OE is implemented in software for personal computers, the numerical values of the kinematic parameters of the mechanism, allowing to choose the block diagram and determine the dimensions of the links of the mechanism of hydraulically controlled jaws of the bucket.

Conflict of Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

Author Contributions

Co-authors brought the following contributions for the development of the paper. Prof. Kulgildinov M. developed the theoretical basics of hydraulic excavator. Ass. Prof. Kulgildinov B. developed the kinematic scheme for the hydraulic excavator. Dr. Zhauyt A. made mathematical model and numerical check of the system. Ass. Prof. Taran M. analyzed and calculated the parameters for the virtual model of the hydraulic excavator. Doctoral student Kaukarov A. developed the virtual model of the proposed hydraulic excavator within Maple 18 software. Simulation and numerical check of the virtual model have been done by Doctoral student Kamzanov N.

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