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Mathematical modeling kinematics of double toggle jaw crusher

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Abstract. In this work, the problem of modeling the kinematic parameters of a jaw crusher with a simple movement of the jaw was consider. The dynamic model of the jaw crusher was consider as a flat articulated-lever closed mechanism. The crushing mechanism of the jaw crusher with a simple movement of the cheek modeled as a mechanism with five moving links and six rotary joints of the fifth mobility class, with the eccentric shaft modeled as a crank and the rotary jaw modeled as a rocker arm. The kinematic chain of the crank and the movable cheek was consider separately. Using vector equations, the interdependencies between the moving elements of the adopted kinematic scheme of the jaw crusher model were determined. Since this scheme has one degree of mobility, the functions of the position of all moving links of the given kinematic scheme were find, depending on the angle of rotation of the drive crank. Rotation angles was define in the Cartesian coordinate system relative to the horizontal plane.

In this work, using the obtained kinematic equations, the proposed simulation model of the jaw crusher was investigate and compared with a real SMD-117 machine using its typical dimensions.

The functions of the changes in the angles of rotation of the links of the kinematic scheme of the jaw crusher and their angular velocities obtained in the course of the research are important in the future for studies of the dynamics of such machine structures. Comparative data showed how the mechanism of changing kinematic parameters was implement in the design of the real SMD-117 machine.

Keywords: jaw crusher, mathematical modeling, kinematics, hinge-lever mechanism, SMD-117.

INTRODUCTION

Jaw crushers occupy a key place in the mining industry, particular such machines are use it in the first stages of crushing construction materials. Jaw crushers used for large and medium destruction of stone materials and minerals in order to obtain the required particle size. The principle of operation of a jaw crusher lies in the fact that material to be crushed is fed into a crushing chamber, which has a wedge-shaped form and is formed by two jaws, one of which is typically stationary, while the other is movable. Due to the wedge-shaped shape of the crushing chamber, the pieces of material arranged along the height of the crushing chamber depending on their size: larger - at the top, smaller - at the bottom. When the movable jaw approaches the stationary or movable (for models with two movable jaws), the material between them is crush into smaller parts [1].

Currently, two types of jaw crushers are widely used, its double toggle jaw crushers and compound jaw crushers [1-2].

According to research by the Australian Career Development Institute, jaw crushers have low energy efficiency when grinding solid materials due to an inefficient mechanism for transferring forces to the grinding elements [3].

In this work proposed to consider the kinematics of a double toggle jaw crusher of the jaw for the further development of a mathematical model of the crusher and analysis of the mechanisms of force transmission. The mathematical model of a jaw crusher is an important tool for optimizing its performance and efficiency [4].

LITERATURE REVIEW

Double toggle jaw crusher is one of the most common types of crushers. This type of crusher has a movable jaw that oscillates relative to a fixed axis of rotation [1].

Works of many authors [3-7], the issue of kinematics of jaw crushers was consider, but qualitative research in this direction was not carried out.

In the researches of Ham C. W., Crane E. J., Rogers W. L., Cao J., Rong X., Shehuan L. [4-5], the development of mathematical models of jaw crushers with complex movement of the cheek, which takes into account dynamic changes in the movement of the movable cheek, is considered. However, the results of such studies cannot be apply to crushers with a simple cheek movement.

Some authors it is Oduori M. F., Mutuli S. M., Munyasi D. M. [6-7] using the equations of motion, were able to describe the change in speed and acceleration of the moving jaw in time of a jaw crusher with simple motion. However, they did not perform a qualitative analysis of the obtained parameters on real systems, therefore it is do not known how the design of the crusher model affects its efficiency, and their obtained dependencies are quite simplify for its further use [8, 9].

PURPOSE OF THE ARTICLE

The goal is to determine the interdependence between the driving and driven links of the kinematic scheme of a jaw crusher with a simple movement of the cheek for qualitative further analysis of the kinematic characteristics of its motion transmission mechanisms and the development of a mathematical model of motion.

PRESENTING MAIN MATERIAL

In the research process, a jaw crusher with a simple movement of the jaw was consider, the main elements of which are (Fig. 1) [1]:

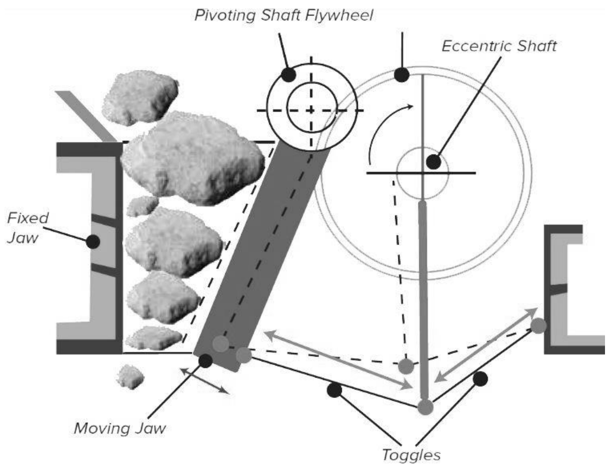

Fig. 1. Double toggle jaw crusher

- movable cheek (pendulum) is a part of the crusher that performs rotational oscillations around the upper suspension, thereby creating a crushing force;
- fixed jaw is a stationary part of the crusher, which is a support for the movable jaw and holds the material for crushing;
- adjusting device – allows you to change the distance between the cheeks, which affects the maximum size of the final product;
- crank or eccentric is a link that transmits movement from the drive to the swing mechanism of the movable cheek;
- connecting rod – an element that converts rotary motion into oscillations of spacer parts of the cheek.

The movable jaw of the crusher, the eccentric, the connecting rod, the spacer links and the frame make up the mechanism, which can modeled as a flat lever articulated mechanical system (Fig. 2). With the exception of the frame, class 5 hinges with one degree of mobility all links in such a mechanism can connected.

In the accepted on Fig. 2 kinematic model of a jaw crusher with a simple cheek movement, the eccentric shaft is modeled as a short crank length l_1, which rotates with a uniform angular velocity around its own fixed axis O_1. The connecting rod modeled as a stiff link O_2O_4 with in length l_2 is which performs a complex planar movement. The rear spacer plate is length l_3 rotates around a fixed axis at a point O_3.

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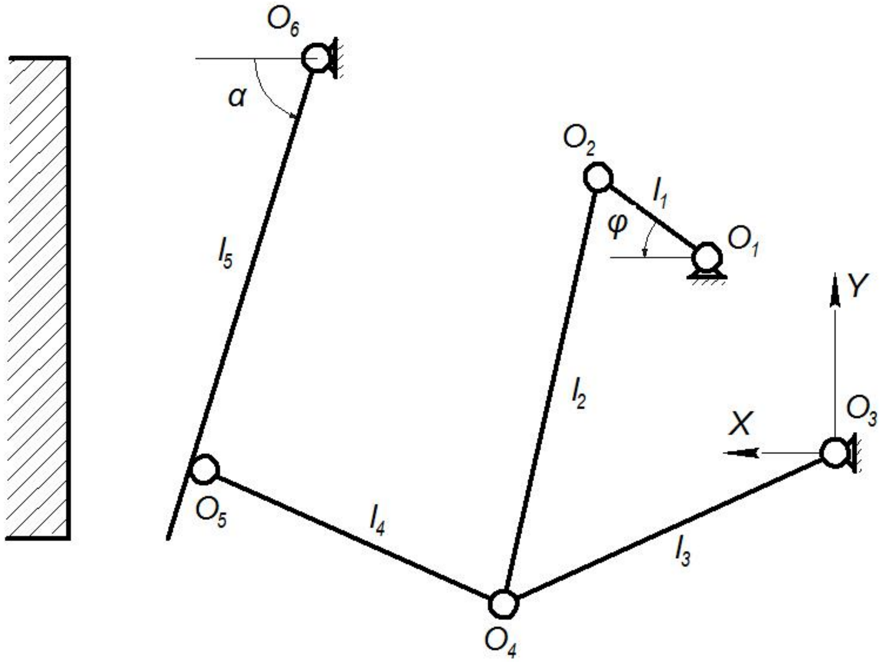

Fig. 2. Kinematical model

The front push rod length l_4 modeled as a link O_4O_5, which also performs a complex planar movement, i.e. it simultaneously performs rotation and rectilinear movement. The rotating cheek in the accepted model it modeled as a rocker O_5O_6, in length l_5, which oscillates about a fixed axis at a point O_6. In the adopted model of the jaw crusher, there is a rocker arm O_5O_6 is a part of the movable cheek and does not fully reproduce the surface of the crushing plate. The stationary jaw is consider of a part machine frame [10, 11].

This kinematic scheme of the machine has a total of one degree of mobility, which means that it is possible theoretical determine the functions of the connection between the drive link and all the driven parts of the machine.

Two closed kinematic chains are distinguished in the considered kinematic scheme is $O_1O_2O_4O_3$ (Рис. 3) and $O_3O_4O_5O_6$ (Рис. 4), which separately reflect the movement transmission mechanism from the eccentric to the oscillating spacer plate and from the oscillating spacer plate to the movable cheek.

Let's perform a separate kinematic analysis of chains $O_1O_2O_4O_3$ and $O_3O_4O_5O_6$.

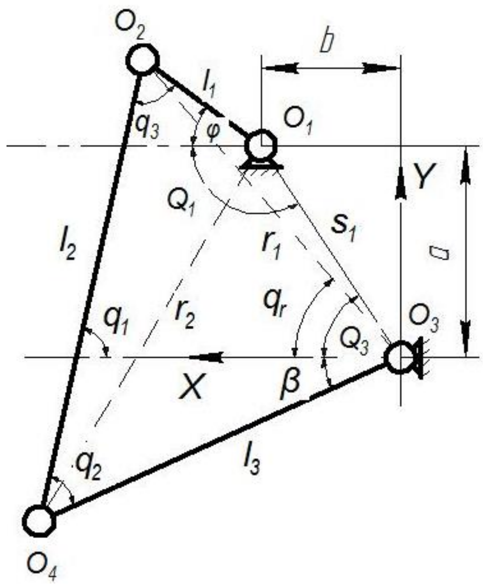

Fig. 3. Kinematic crank chain

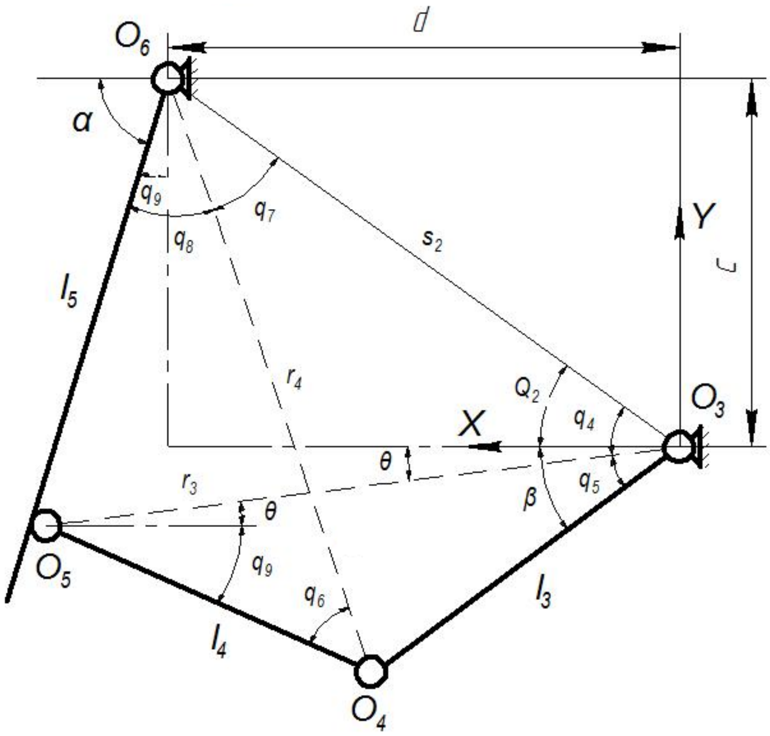

Fig. 4. Kinematic chain of the pendulum

Determine the angles of inclination of all moving links of the specified kinematic scheme of the jaw crusher to the horizontal direction.

According to Fig. 3 from $\Delta O_1O_2O_3$ we have the next vector equation [11]:

$$\vec{s}_1 + \vec{l}_1 = \vec{r}_1 . \qquad (1)$$

The projections of equation (1) on the vertical *Y* and horizontal *X* axes will be:

$$s_1 \sin Q_3 + l_1 \sin \varphi = r_1 \sin q_r , \qquad (2)$$

$$s_1 \cos Q_3 + l_1 \cos \varphi = r_1 \cos q_r , \qquad (3)$$

where: $s_1 = \sqrt{a^2 + b^2}$, $\sin Q_3 = \frac{a}{s_1}$, $\cos Q_3 = \frac{b}{s_1}$ are parameters of the kinematic scheme of the crusher; a, b – given dimensions of the model, m.

From equations (2) and (3), it was determinate:

$$\tan q_r = \frac{s_1 \sin Q_3 + l_1 \sin \varphi}{s_1 \cos Q_3 + l_1 \cos \varphi} . \qquad (4)$$

From $\Delta O_1O_2O_3$ and $\Delta O_2O_3O_4$ the distance of the radius vector r_1 was find, using the cosine theorem:

$$r_1^2 = l_1^2 + s_1^2 - 2l_1 s_1 \cos(\varphi + Q_1) , \qquad (5)$$

$$r_1^2 = l_2^2 + l_3^2 - 2l_2 l_3 \cos q_2 , \qquad (6)$$

where: $Q_1 = \pi - Q_3$.

Then from equations is (5) and (6) have been get:

$$\cos q_2 = \frac{l_2^2 + l_3^2 - l_1^2 - s_1^2 + 2l_1 s_1 \cos(\varphi + Q_1)}{2l_2 l_3} , \qquad (7)$$

$$r_1 = \sqrt{l_1^2 + s_1^2 - 2l_1 s_1 \cos(\varphi + Q_1)} , \qquad (8)$$

or

$$q_2 = \arccos\left(\frac{l_2^2 + l_3^2 - r_1^2}{2l_2 l_3}\right) . \qquad (9)$$

By the sinuses theorem with $\Delta O_2O_3O_4$ it have next question:

$$\sin(q_r + \beta) = \frac{l_2 \sin q_2}{r_1} , \qquad (10)$$

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then

$$\beta = \arcsin\left(\frac{l_2 \sin q_2}{r_1}\right) - q_r . \quad (11)$$

It is knows from the studied scheme that:

$$q_1 = q_2 + \beta . \quad (12)$$

According to Fig. 4 from $\Delta O_3O_4O_6$ defined the next vector equation:

$$\vec{s}_2 + \vec{l}_3 = \vec{r}_4 . \quad (13)$$

The projections of this equation on the vertical and horizontal axes will give the following expressions:

$$s_2 \sin Q_2 + l_3 \sin\beta = r_4 \sin(q_7 + Q_2) , \quad (14)$$

$$s_2 \cos Q_2 - r_4 \cos(q_7 + Q_2) = l_3 \cos\beta , \quad (15)$$

where: $s_2 = \sqrt{c^2 + d^2}$, $\sin Q_2 = \frac{c}{s_2}$,

$\cos Q_2 = \frac{d}{s_2}$ there are parameters of the kinematic scheme; c, d there are given dimensions of the model, m.

The angle of rotation is determined from equations (14) and (15) q_7

$$q_7 = \arctan\left(\frac{s_2 \sin Q_2 + l_3 \sin\beta}{s_2 \cos Q_2 - l_3 \cos\beta}\right) - Q_2 . \quad (16)$$

According to the theorem of cosines from $\Delta O_4O_5O_6$ have been define:

$$l_4^2 = l_5^2 + r_4^2 - 2 l_5 r_4 \cos q_8 , \quad (17)$$

where:

$$q_8 = \arccos\left(\frac{l_5^2 + r_4^2 - l_4^2}{2 l_5 r_4}\right) , \quad (18)$$

де $r_4 = \sqrt{s_2^2 + l_3^2 - 2 s_2 l_3 \cos(\beta + Q_2)}$.

Thus, the scheme in fig. 4, the angle of rotation of the movable cheek to the horizontal is determined:

$$\alpha = \pi - q_7 - q_8 - Q_2 . \quad (19)$$

From $\Delta O_3O_4O_6$ it have been established that:

$$s_2 \cos Q_2 + l_5 \cos\alpha = r_3 \cos\theta , \quad (20)$$

$$s_2 \sin Q_2 - l_5 \sin\alpha + r_3 \sin\theta = 0 , \quad (21)$$

then

$$\theta = \arctan\left(\frac{l_5 \sin\alpha - s_2 \sin Q_2}{l_5 \cos\alpha + s_2 \cos Q_2}\right) . \quad (22)$$

From $\Delta O_3O_4O_5$ it's obvious that

$$l_3 \sin\beta - r_3 \sin\theta = l_4 \sin q_9 , \quad (23)$$

$$r_3 \cos\theta - l_3 \cos\beta = l_4 \cos q_9 , \quad (24)$$

or

$$q_9 = \arctan\left(\frac{l_3 \sin\beta - r_3 \sin\theta}{r_3 \cos\theta - l_3 \cos\beta}\right) , \quad (25)$$

where: $r_3 = \sqrt{s_2^2 + l_5^2 - 2 s_2 l_5 \cos(q_7 + q_8)}$.

Thus, for this estimated kinematic scheme of the jaw crusher, geometric characteristics were determined that connect the drive link of the eccentric (crankshaft) with all the moving links of the crusher [12].

The resulting dependencies have a complex nature of relationships between the host and the given links. In order to check the obtained equations for correctness, a simulation model of a jaw crusher with a simple movement of the jaw it was define, on which the parameters of the angles of rotation of the moving parts of the machine were determined geometrically. The results of the comparison showed an almost complete coincidence of the obtained values of numerical and simulation modeling. In the table 1 shows the values of the parameters determined theoretically and on the simulation model (IM1) of the jaw crusher with the following dimensions of the model: l_1 = 100 mm; l_2 = 800 mm; l_3 = 700 mm; l_4 = 500 mm; l_5 = 500 mm; a = 400 mm; b = 300 mm; c = 650 mm; d = 900 mm.

Given the high convergence of model parameters, was also performed research of a real

Table 1. Results of comparison of theoretical data with data obtained on the simulation model IM1

№ п.п.	φ, degree	β, degree		q_1, degree		α, degree		q_9, degree	
		graph.	theor.	graph.	theor.	graph.	theor.	graph.	theor.
1	0	32,60	32,60	78,28	76,28	79,18	79,18	5,9	5,16
2	10	30,58	30,58	75,21	75,22	78,35	78,35	3,06	3,07
3	20	28,57	28,57	73,97	73,98	77,60	77,60	0,92	0,92
4	30	26,59	26,59	72,59	72,59	76,94	76,95	-1,24	-1,24
5	40	24,70	24,70	71,08	71,08	76,40	76,40	-3,37	-3,38
6	50	22,92	22,92	69,47	69,48	75,96	75,97	-5,45	-5,46
7	60	21,28	21,28	67,80	67,80	75,63	75,64	-7,42	-7,43
8	70	19,81	19,82	66,08	66,08	75,40	75,40	-9,25	-9,25
9	80	18,55	18,55	64,35	64,35	75,24	75,25	-10,87	-10,88
10	90	17,50	17,50	62,64	62,65	75,15	75,16	-12,24	-12,25
11	100	16,70	16,71	61,00	61,00	75,10	75,11	-13,31	-13,32
12	110	16,18	16,18	59,46	59,46	75,08	75,09	-14,03	-14,03
13	120	15,95	15,95	58,07	58,08	75,08	75,08	-14,34	-14,35
14	130	16,04	16,05	56,88	56,89	75,08	75,08	-14,21	-14,22
15	140	16,48	16,48	55,96	55,96	75,09	75,10	-13,62	-13,62
16	150	17,28	17,29	55,34	55,34	75,14	75,14	-12,53	-12,54
17	160	18,47	18,47	55,08	55,08	75,24	75,24	-10,97	-10,98
18	170	20,04	20,04	55,24	55,24	75,43	75,43	-8,97	-8,97
19	180	21,98	21,98	55,85	55,85	75,77	75,77	-6,57	-6,58
20	190	24,26	24,27	56,92	56,92	76,29	76,29	-3,87	-3,88
21	200	26,83	26,83	58,45	58,45	77,02	77,02	-0,97	-0,98
22	210	29,57	29,57	60,37	60,38	77,96	77,96	1,99	2,00
23	220	32,36	32,36	62,62	62,62	79,07	79,08	4,91	4,91
24	230	35,05	35,05	65,06	65,06	80,29	80,29	7,62	7,62
25	240	37,48	37,49	67,55	67,55	81,49	81,50	10,01	10,02
26	250	39,53	39,53	69,96	69,96	82,58	82,59	11,99	11,99
27	260	41,08	41,09	72,17	72,17	83,46	83,46	13,48	13,48
28	270	42,10	42,10	74,08	74,09	84,05	84,06	14,44	14,44
29	280	42,57	42,57	75,66	75,66	84,33	84,33	14,88	14,89
30	290	42,51	42,51	76,86	76,87	84,30	84,30	14,83	14,83
31	300	41,98	41,99	77,70	77,71	83,98	83,99	14,33	14,33
32	310	41,04	41,05	78,19	78,20	83,44	83,44	13,44	13,44
33	320	39,77	39,77	78,36	78,36	82,72	82,72	12,22	12,23
34	330	38,22	38,23	78,22	78,22	81,88	81,89	10,74	10,74
35	340	36,48	36,48	77,81	77,81	80,98	80,99	9,03	9,03
36	350	34,58	34,59	77,16	77,16	80,07	80,07	7,16	7,16

SMD-117 jaw crusher with simple jaw movement and standard size 1500×2100 mm. The model of such a crusher has the next dimensions: l_1 = 42 mm; l_2 = 2165 mm; l_3 = 1099 mm; l_4 = 1839 mm; l_5 = 3280 mm; a = 1800 mm; b = 1190 mm; c = 3410 mm; d = 2770 mm. The results of modeling listed in the Table. 2.

In the table 2 shows the ratio that estimates the transfer function between a given moving link and the angle of rotation of the drive crank, which it calculated from this ratio:

$$\frac{\Delta\delta_i}{\Delta\varphi_i} = \frac{\delta_i - \delta_{i-1}}{\varphi_i - \varphi_{i-1}} \;, \tag{26}$$

where: δ_i the angle of rotation of the given link (β, q_1, α, q_9).

According to the data from the Table 2, it can seen that the change in the rotation angles

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Table 2. Results of theoretical modeling of changes in the geometric parameters of the SMD-117 jaw crusher

№ п.п.	φ, degree	β, degree	Δβ/Δφ	q_1, degree	Δ q_1/Δφ	α, degree	Δα/Δφ	q_9, degree	Δq_9/Δφ
1	0	18,92	-0,039	95,10	-0,005	89,25	-0,008	15,35	-0,023
2	10	18,54	-0,039	95,02	-0,008	89,17	-0,007	15,12	-0,022
3	20	18,17	-0,037	94,91	-0,011	89,10	-0,007	14,91	-0,021
4	30	17,83	-0,034	94,77	-0,014	89,04	-0,006	14,71	-0,020
5	40	17,53	-0,030	94,62	-0,016	88,98	-0,006	14,53	-0,018
6	50	17,27	-0,025	94,44	-0,018	88,94	-0,005	14,39	-0,015
7	60	17,07	-0,020	94,25	-0,019	88,90	-0,004	14,27	-0,012
8	70	16,93	-0,014	94,05	-0,020	88,88	-0,002	14,19	-0,008
9	80	16,86	-0,008	93,85	-0,020	88,86	-0,001	14,14	-0,004
10	90	16,85	-0,001	93,66	-0,019	88,86	0,000	14,14	0,000
11	100	16,91	0,006	93,47	-0,018	88,87	0,001	14,17	0,003
12	110	17,03	0,012	93,31	-0,017	88,90	0,002	14,25	0,007
13	120	17,22	0,019	93,16	-0,015	88,93	0,003	14,35	0,011
14	130	17,46	0,024	93,04	-0,012	88,97	0,004	14,50	0,014
15	140	17,76	0,029	92,94	-0,009	89,03	0,005	14,67	0,017
16	150	18,09	0,033	92,88	-0,006	89,09	0,006	14,86	0,019
17	160	18,46	0,037	92,86	-0,002	89,16	0,007	15,07	0,021
18	170	18,84	0,039	92,87	0,001	89,23	0,007	15,30	0,023
19	180	19,24	0,040	92,92	0,005	89,31	0,008	15,53	0,023
20	190	19,64	0,039	93,01	0,008	89,39	0,008	15,76	0,023
21	200	20,02	0,038	93,12	0,012	89,47	0,008	15,98	0,022
22	210	20,37	0,035	93,27	0,014	89,54	0,007	16,18	0,020
23	220	20,68	0,031	93,43	0,017	89,60	0,007	16,37	0,018
24	230	20,94	0,026	93,62	0,019	89,66	0,006	16,52	0,015
25	240	21,15	0,021	93,81	0,020	89,70	0,004	16,64	0,012
26	250	21,30	0,014	94,02	0,020	89,73	0,003	16,72	0,008
27	260	21,37	0,008	94,22	0,020	89,75	0,002	16,77	0,004
28	270	21,38	0,000	94,41	0,019	89,75	0,000	16,77	0,000
29	280	21,31	-0,007	94,59	0,018	89,74	-0,001	16,73	-0,004
30	290	21,18	-0,013	94,76	0,016	89,71	-0,003	16,65	-0,008
31	300	20,98	-0,020	94,90	0,014	89,67	-0,004	16,54	-0,011
32	310	20,72	-0,025	95,01	0,011	89,61	-0,005	16,39	-0,015
33	320	20,42	-0,030	95,09	0,008	89,55	-0,006	16,22	-0,018
34	330	20,08	-0,034	95,15	0,005	89,48	-0,007	16,02	-0,020
35	340	19,71	-0,037	95,16	0,002	89,40	-0,008	15,80	-0,022
36	350	19,32	-0,039	95,15	-0,002	89,33	-0,008	15,58	-0,023

of the moving parts is real jaw crusher SMD-117 is carried out within small limits with one complete rotation of the drive crank. From this, we will assume that the developers of real systems of jaw crushers try to select the parameters of their schemes in such a way as to implement the operation of the machine with the least expenditure of kinetic energy. The implementation of the work of destruction by the moving cheek is perform by the power circuit due to the action of the massive moving masses of the machine

To study the dynamics of the jaw crusher model, it is necessary to know the angular velocities of the moving parts of the machine.

From Fig. 3, consider the following vector equation:

$$\vec{s}_1 + \vec{l}_1 = \vec{l}_2 + \vec{l}_3 . \qquad (27)$$

Let's determine the projections of equation (27) into the Cartesian coordinate system:

$$s_1 \sin Q_3 + l_1 \sin \varphi = l_2 \sin q_1 - l_3 \sin \beta ; \quad (28)$$

$$s_1 \cos Q_3 + l_1 \cos \varphi = -l_2 \cos q_1 + l_3 \cos \beta . \quad (29)$$

The left and right sides of the equations (28) and (29) can be squared, leading to the following:

$$\begin{aligned} & s_1^2 \sin^2 Q_3 + l_1^2 \sin^2 \varphi + 2 s_1 l_1 \sin Q_3 \sin \varphi = \\ & = l_2^2 \sin^2 q_1 + l_3^2 \sin^2 \beta - 2 l_2 l_3 \sin q_1 \sin \beta ; \end{aligned} \quad (30)$$

$$\begin{aligned} & s_1^2 \cos^2 Q_3 + l_1^2 \cos^2 \varphi + 2 s_1 l_1 \cos Q_3 \cos \varphi = \\ & = l_2^2 \cos^2 q_1 + l_3^2 \cos^2 \beta - 2 l_2 l_3 \cos q_1 \cos \beta . \end{aligned} \quad (31)$$

By adding equations (29), (30) and taking into account that $\sin^2 i + \cos^2 i = 1$, it will receive:

$$\begin{aligned} & s_1^2 + l_1^2 + 2 l_1 s_1 (\sin \varphi \sin Q_3 + \cos \varphi \cos Q_3) = \\ & = l_2^2 + l_3^2 - 2 l_2 l_3 (\sin q_1 \sin \beta + \cos q_1 \cos \beta). \end{aligned} \quad (32)$$

From trigonometry, the following formula is known:

$$\sin \varphi \sin Q_3 + \cos \varphi \cos Q_3 = \cos(\varphi - Q_3) . \quad (33)$$

Therefore, from equality (33), we will have the following expression:

$$\begin{aligned} & s_1^2 + l_1^2 + 2 l_1 s_1 \cos(\varphi - Q_3) = \\ & = l_2^2 + l_3^2 - 2 l_2 l_3 \cos(q_1 - \beta). \end{aligned} \quad (34)$$

From the expression, the formula for the angle of rotation of the spacer plate is determined:

$$q_1 = \beta + \arccos\left(K_1 - K_2 \cos(\varphi - Q_3)\right), \quad (35)$$

where: $K_1 = \dfrac{l_2^2 + l_3^2 - s_1^2 - l_1^2}{2 l_2 l_3}$, $K_2 = \dfrac{l_1 s_1}{l_2 l_3}$.

The resulting expression is similar to formula (12). Differentiating formula (35):

$$\begin{aligned} & \frac{dq_1}{dt} = \frac{d\beta}{dt} \frac{d\varphi}{dt} + \\ & + \frac{K_2 \sin\left(Q_3 - \varphi\right))}{\sqrt{1 - \left(K_1 - K_2 \cos(\varphi - Q_3)\right)^2}} \frac{d\varphi}{dt}, \end{aligned} \quad (36)$$

$$\frac{d\beta}{dt} = l_1 \frac{d\varphi}{dt} \left(\frac{1}{\dfrac{l_1^2 - s_1^2}{l_1 + s_1 \cos(Q_3 - \varphi)} - 2 l_1} - \varsigma_1 \right), \quad (37)$$

where:

$$\zeta_1 = \frac{s_1 (r_1^2 + l_2^2 - l_3^2) \sqrt{\dfrac{r_1^2 - l_2^2 + l_3^2}{l_3^2 r_1^2}} \sin(Q_3 + \varphi)}{(l_2 r_1 (r_1^2 - l_2^2 + l_3^2) \sqrt{-\dfrac{(r_1^2 - (l_2 - l_3)^2)(r_1^2 - (l_2 + l_3)^2)}{l_2^2 l_3^2}}} .$$

In a similar way, the derivatives for all other angles of rotation of the jaw crusher movable by the jaw crusher are determined, however, the analytical form of their recording does not allow to give these formulas in a compact form in this work, and therefore it is suggested to consider their numerical dependences on the graphs. Analyzing the function (36), we note that the angular velocities of the moving links will depend both on the position of the driving eccentric and on its speed. Since the main work of the jaw crusher is performed in a constant mode of movement, then we will assume that the angular speed of the crank will be constant. As a guide, we will use the parameters of the SMD-117 jaw crusher with a simple jaw movement, which contains an installed electric motor with a power of 280 kW and a rotation frequency of 490 rpm. Given the given drive parameters, the approximate constant angular speed of the eccentric of the crusher should be within 10.5...12.5 rad/s. In the future, the upper limit for calculations was adopted.

Fig. 5 shows the graphs of the speeds of the resonant links of the crusher. Fig. 6 shows the graphs of the angles of rotation of the crusher links. Fig. 7 and Figs. 8 shows the graphs of changes in the accelerations of the moving links of the SMD-117 crusher and the experimental simulation model IM1 in the mode of steady motion for one complete revolution of the crank.

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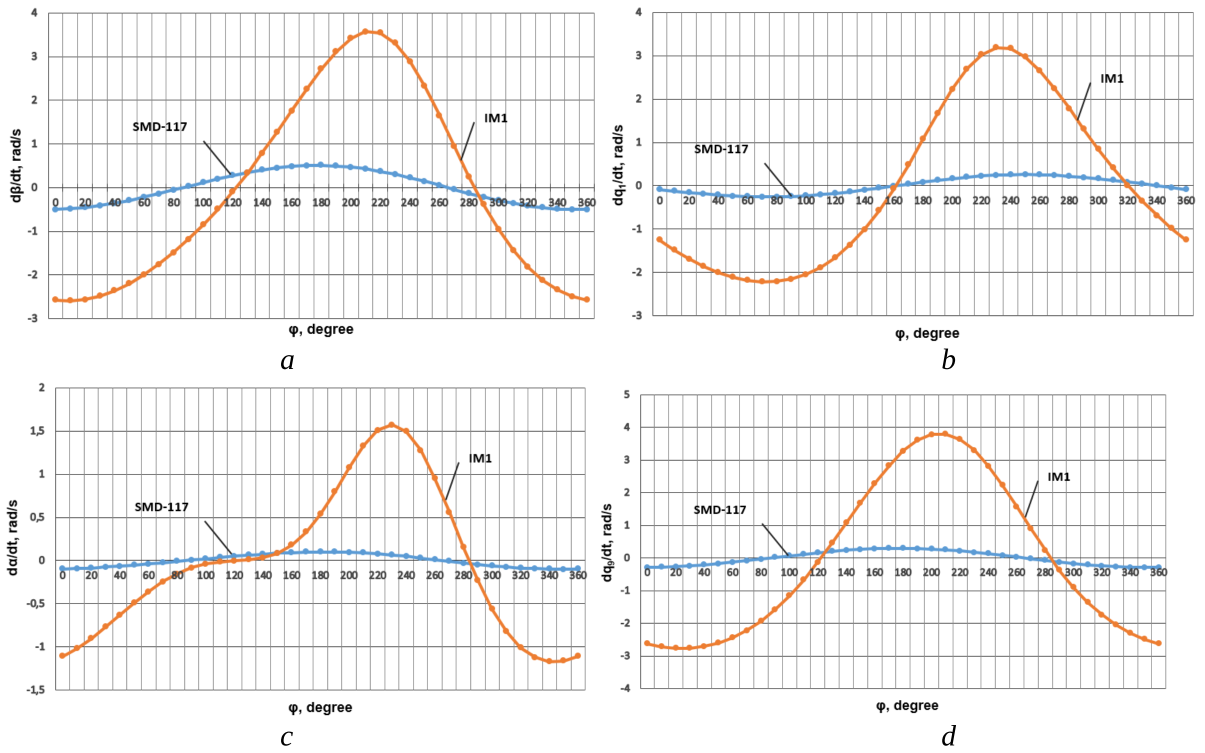

Fig. 5. Comparative graphs of changes in the angular velocities of the moving links of the kinematic scheme for jaw crushers of proposed the simulation model IM1 and the real SMD-117 machine in the mode of steady movement of the crank at a her velisity of 12,5 rad/s

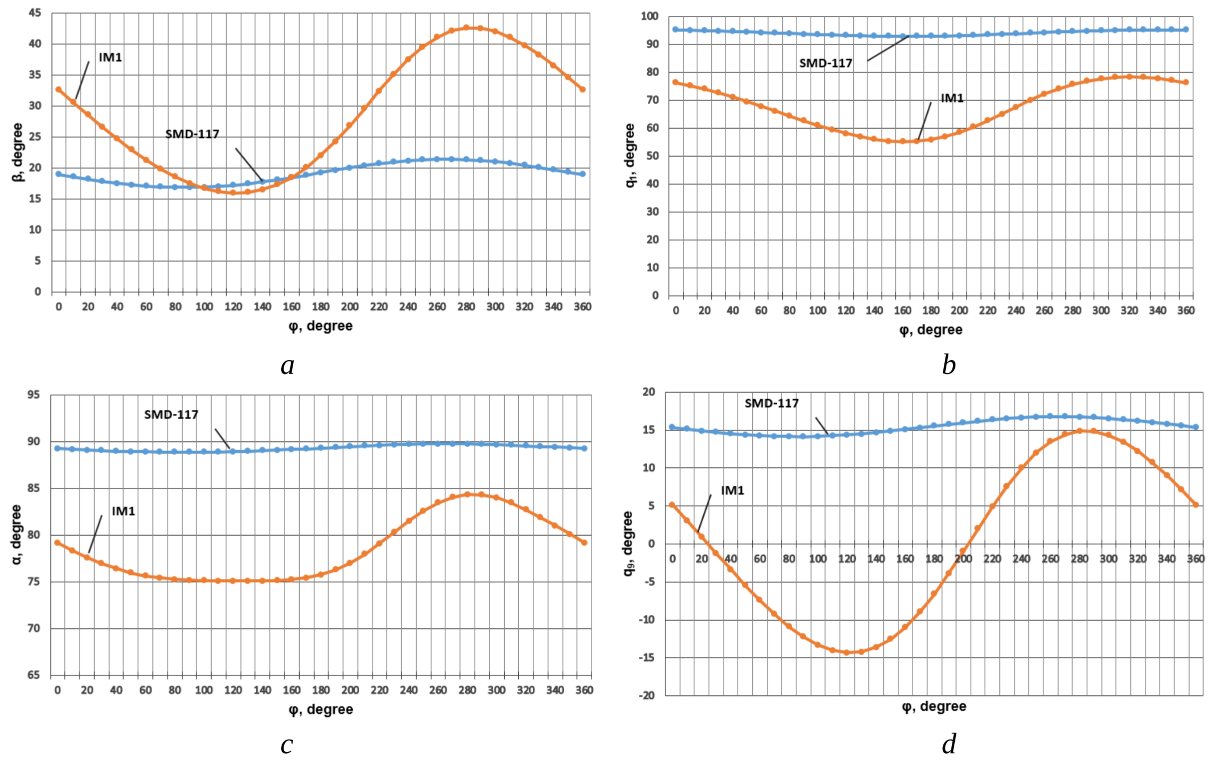

Fig. 6. Comparative graphs of changes in the angles of rotation of the links of the kinematic scheme for jaw crushers of proposed the simulation model IM1 and the real SMD-117 machine

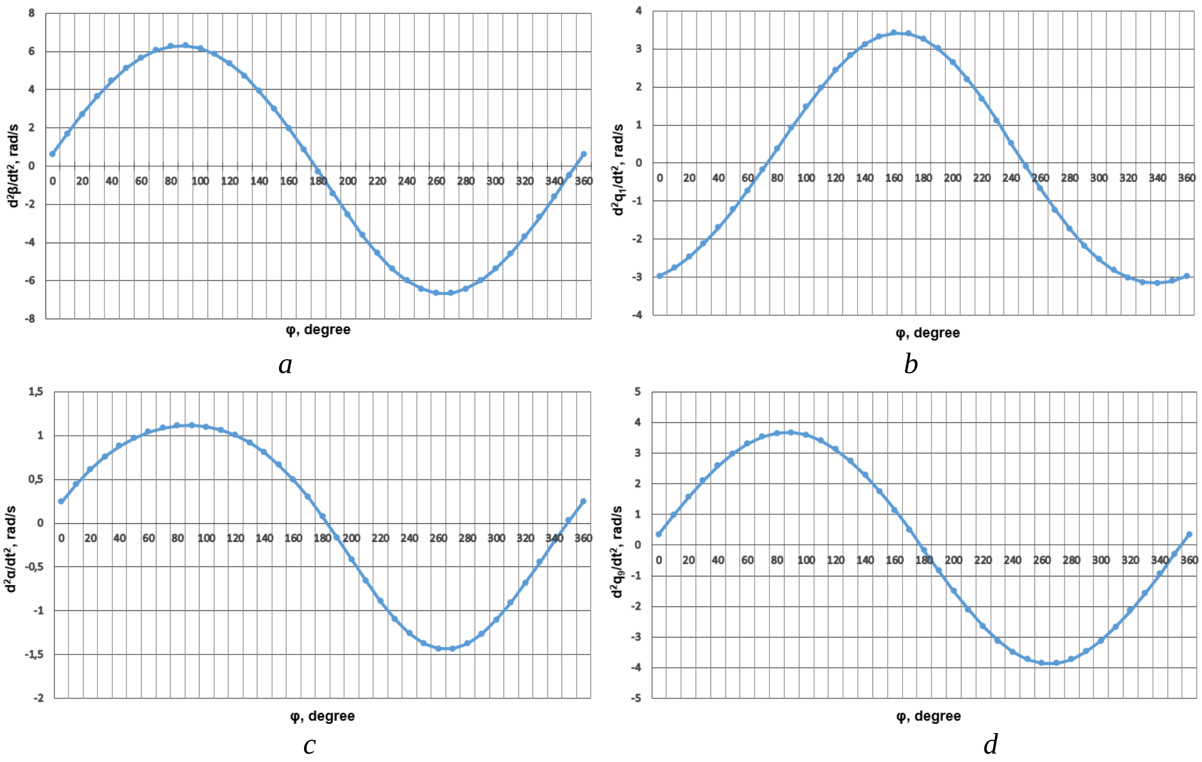

Fig. 7. Graphs of the accelerations of the moving parts of the SMD-117 jaw crusher in a steady mode of movement at a crank velisity of 12,5 rad/s

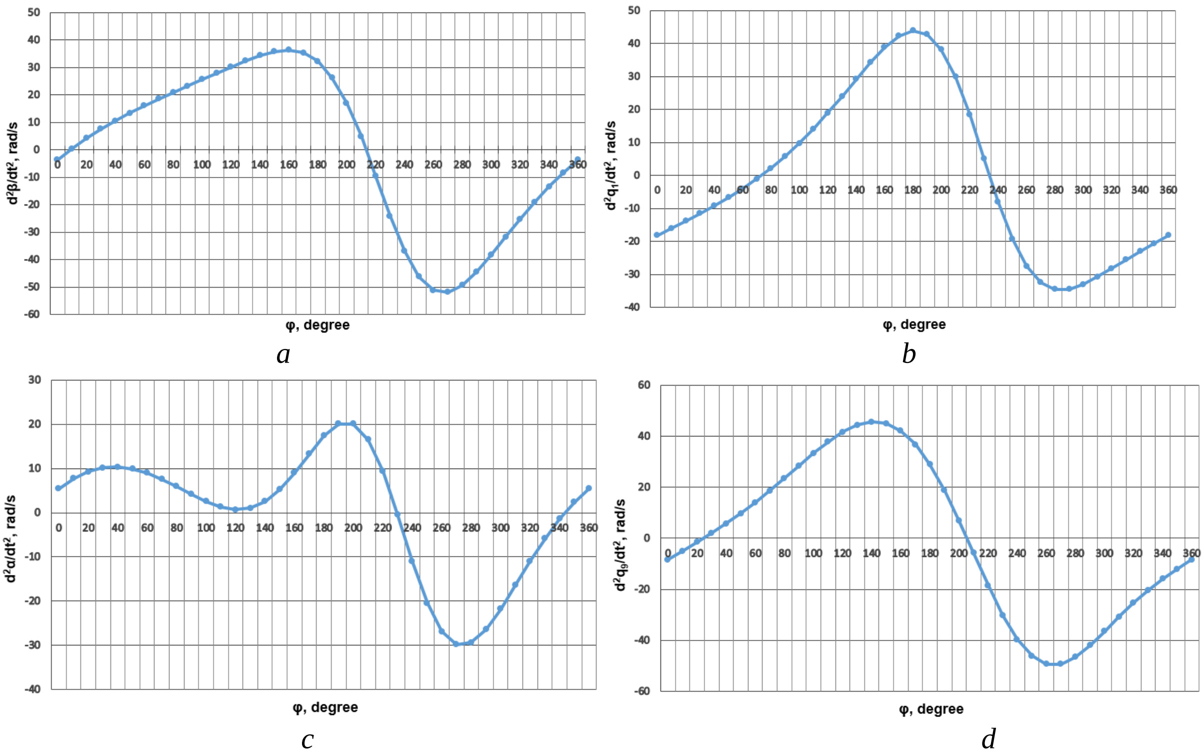

Fig. 8. Graphs of the accelerations of the moving parts of the IM1 jaw crusher in a steady mode of movement at a crank velisity of 12,5 rad/s

CONCLUSIONS

The obtained results of the study show a complex interdependence between the kinematic parameters of the jaw crusher's motion transmission mechanism with simple cheek movement. The speeds and accelerations of the moving parts of the real SMD-117 jaw crusher are significantly lower than in the proposed test simulation model. Since the real model of the SMD-117 has a much larger mass of moving parts. In part, this difference was explain by the fact that in the real system of the kinematic scheme of the jaw crusher with a simple movement of the jaw, static forces created by the drive implement the main mechanism of material destruction.

In the future, in order to determine the power of destruction, it is necessary to conduct a study of the dynamics and changes in forces in the jaw crusher mechanism.

In the future, a model of machine control based on IoT devices will be develop, as in the example of work [14] for efficient use of production resources.

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Математичне моделювання кінематики щокової дробарки з простим рухом щоки

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Анотація. В даній роботі розглянута проблематика моделювання кінематичних параметрів щокової дробарки з простим рухом щоки. Динамічна модель щокової дробарки розглядається, як плоский шарнірно-важільний замкнутий механізм. Механізм дроблення щокової дробарки з простим рухом щоки змодельований, як механізм із п'ятьма рухомими ланками та шістьма поворотними шарнірами п'ятого класу рухомості, причому ексцентриковий вал змодельований, як кривошип, а поворотна щелепа, – як коромисло. Було розглянуто окремо кінематичний ланцюг кривошипа та рухомої щоки. За допомогою векторних рівнянь визначено взаємозалежності між рухомими елементами прийнятої кінематичної схеми моделі щокової дробарки. Так як дана схема має одну ступінь рухомості, було знайдено функції положення всіх рухомих ланок заданої кінематичної схеми в залежності від кута повороту привідного кривошипа. Кути повороту визначено в Декартовій системі координат відносно горизонтальної площини.

В даній роботі із застосування отриманих кінематичних рівнянь було досліджено запропоновану імітаційну модель щокової дробарки та порівняння її з реальною машиною марки СМД-117 з використанням її типових розмірів.

Отримані в процесі дослідження функції зміни кутів повороту ланок кінематичної схеми щокової дробарки та їх кутових швидкостей важливі в подальшому для досліджень динаміки конструкцій таких машин.

З аналізу отриманих кінематичних залежностей для схеми щокової дробарки з простим рухом щоки зазначимо, що механізми руйнування матеріалу в камері дроблення є складним, а тому реальні конструкції машин виконують таким чином, щоб реалізувати режими їхньої роботи з малими динамічними навантаженнями на рухомі робочі органи. Для цього робочі органи створюють значної маси, а механізм руйнування реалізують статичними силами, що створюють приводом.

Ключові слова: щокова дробарка, математичне моделювання, кінематика, шарнірно-важільний механізм, СМД-117.