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Kinematic Parameters of the Biplanetary Mechanism (Intermittent Mixing Machines)

Erkin Nematov (0000-0002-4178-0168)[1], Amon Berdiev (0000-0002-5410-4601)[2], Peng Wang (0000-0003-4771-7984)[3]
[1]Faculty of Mechanichal engineering, Tashkent State Technical University named after Islam Karimov, University Str.2, 100174, Tashkent, Uzbekistan. E-mail: erkin_nemat@mail.ru.
[2]Faculty of technology of foodstuff products, Tashkent institute of chemical technology, Navoiy Str.32, 100011, Tashkent, Uzbekistan. E-mail: Amon84_1@mail.ru.
[3]School of Information Science and Electrical Engineering, Shandong Jiao Tong University, 5001 Haitang Road, Changqing District, Jinan, Shandong Province, China. E-mail: knightwp@126.com.

The subject of research is the kinematic parameters of a biplanetary mechanism of the intermitted mixing machines. The article substantiates analytical expressions for determining the kinematic parameters of the drives of the working body of the intermitted mixing machines with planetary ones with double satellites and biplanetary mechanisms; the laws of change of displacements, velocities and accelerations of the points of the working body for drives with planetary one with double satellites and biplanetary mechanisms are determined; the regularities of the influence of the velocity parameters of the driving links on the kinematic characteristics of these mechanisms are established.

Keywords: Drive, Biplanetary mechanism, Kinematic parameters, biplanetary mechanism, mixing machine.

1 Introduction

Biplanetary mechanisms of periodic action are widely used in various sectors of technology and industry. When designing such drives, an important role is played by the determination of the kinematic parameters of the drives and the laws of change in displacements, velocities and accelerations of the points of the working body.

Ways of operation and determination of kinematic, dynamic parameters of various mechanisms are widely highlighted in literary sources [1–10] . For example, a cam mechanism is a commonly used mechanism. An extensive review and systematization of the current state of kinematic and dynamic studies of the cam mechanism were conducted in [1] . Based on an in-depth analysis of existing mechanisms, a new coaxial eccentric indexing cam mechanism for high-speed automatic mechanisms was proposed and its advantages over other mechanisms were substantiated in [2] . Another type of mechanism is the friction mechanism. Based on the laws of the theory of mechanisms and machines, mathematical models for controlling the parameters and constraints of the friction mechanism were developed [3] . The theory and designs of friction mechanisms with controlled friction were developed in [4, 5] . Based on the results of the review of existing models, an analytical implementation of the mathematical model of the controlled motion of the positioning mechanism was developed and performed [6] . Based on the analytical expressions obtained according to the mathematical model developed, the patterns of changes in the angular velocity of the driven member in the steady-state operation of the crushing machine were determined. The results obtained were used in the design of a universal planetary mill [7] .

The article presents the results of the study of the kinematic parameters of the drives of the intermittent mixing machine working bodies with planetary with double satellites and biplanetary mechanisms.

2 Research methods

Research methods are based on: the method for determining the number of teeth of the reducer; the determination of the kinetic energy of the James reducer; the method for determining the dynamic characteristics of the electric motor.

3 Materials

Determination of the actual laws of motion of the working body of the mixing machine is a necessary factor in its design. Let us consider the working principle of the working body of a mixer. The working organ of the mixer receives motion from an asynchronous electric motor using a planetary reducer. To reduce the number of revolutions of the working disk of the working organ of the mixer, it is proposed to install a James planetary reducer. The kinematic diagram of the drive is shown in Fig. 1. Consider the method of determining the number of teeth of this given reducer [11] .

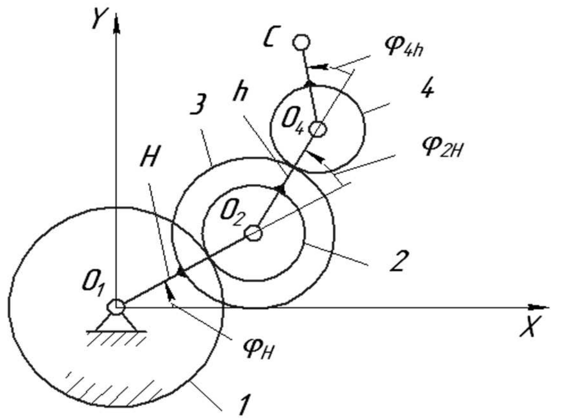

Fig. 1 *Scheme for deriving the equations of the kinematics of the planetary mechanism*

As can be seen in Fig. 1, the following relationship can be written for this mechanism

$$u_{4H} = (1 - u_{41}^{(H)}) \quad (1)$$

The angular velocity ω_H of all members of the mechanism is equal in magnitude and opposite in direction to the velocity of the main driver H. Then the carrier H becomes stationary, and wheel 1 receives an angular velocity ω_H. This reversal of motion leads to the formation of a conventional planetary mechanism with two simple pairs of wheels 1 and 2. Its gear ratio is:

$$u_{41}^{(H)} = u_{21}(1 - u_{43}^{(h)}) \text{ or } u_{41}^{(H)} = -\frac{z_1}{z_2}(1 - u_{43}^{(h)}) \quad (2)$$

Where $(1 - u_{43}^{(h)})$ is the gear ratio of the satellite planetary mechanism, which can be determined using the second reversal of motion, applying it only for this mechanism.

$$\text{Since } u_{43}^{(h)} = -\frac{z_3}{z_4} \text{ then } u_{41}^{(H)} = -\frac{z_1}{z_2}(1 + \frac{z_3}{z_4}) = -\frac{z_1}{z_2} - \frac{z_3 \cdot z_1}{z_4 \cdot z_2} \quad (3)$$

With this in mind, it is possible to write:

$$u_{4H} = (1 + \frac{z_1}{z_2} + \frac{z_1 \cdot z_3}{z_2 \cdot z_4}) \quad (4)$$

Thus, the absolute angle of rotation of the bisatellite is relatively equal to:

$$\phi_4 = \phi_H \left(1 + \frac{z_1}{z_2} + \frac{z_3}{z_4} \cdot \frac{z_1}{z_2}\right) \quad (5)$$

The absolute angular velocity of bisatellite is $\omega_4 = \omega_H \left(1 + \frac{z_1}{z_2} + \frac{z_1}{z_2} \cdot \frac{z_3}{z_4}\right)$ $[rad/s]$, the angular velocity of bisatellite 4 relative to the bicarrier h is $\omega_{4h} = \omega_H \left(\frac{z_1}{z_2} \cdot \frac{z_3}{z_4}\right)$ $[rad/s]$, the angular velocity of bisatellite 4 relative to carrier H is $\omega_{4H} = \omega_H \left(\frac{z_1}{z_2} + \frac{z_1}{z_2} \cdot \frac{z_3}{z_4}\right)$ $[rad/s]$, the absolute angular velocity of satellite 2 is $\omega_2 = \omega_H \left(1 + \frac{z_1}{z_2}\right) [rad/s]$.

In turn, the consideration of the kinematics of this biplanetary mechanism, considering the study given in [8], the kinematic equations for the point "C" of the bisatellite of the biplanetary mechanism can be written in the following form:

$$x_c(t) = A \cdot cos(\omega_H \cdot t) - B \cdot cos(k_1 \cdot \omega_H \cdot t) - D \cdot cos(k_2 \cdot \omega_H \cdot t) \ [m] \quad (6)$$

$$y_c(t) = A \cdot sin(\omega_H \cdot t) - B \cdot sin(k_1 \cdot \omega_H \cdot t) - D \cdot sin(k_2 \cdot \omega_H \cdot t) \ [m] \quad (7)$$

Where $k_1 = u_{2H} = \left(1 - u_{21}^{(H)}\right)$ or $k_1 = \left(1 + \frac{z_1}{z_2}\right)$, $k_2 = u_{4H} = \left(1 - u_{21}^{(H)} - u_{43}^{(h)}\right) = \left(1 + \frac{z_1}{z_2} + \frac{z_1}{z_2} \cdot \frac{z_3}{z_4}\right)$, $A = r_1 + r_2 \ [m]$; $B = r_{3.} + r_4 [m]$; $D = r_4$; r_1, r_2, r_3, r_4 $[m]$ are the radii of pitch circles of gears 1,2,3,4.

The velocities v_C of point *"C"* of bisatellite 4 were determined by the following equations:

$$v_{cx} = \frac{dx_c}{dt} = -A \cdot \omega_H \cdot sin(\omega_H \cdot t) + B \cdot k_1 \cdot \omega_H \cdot sin(k_1 \omega_H \cdot t) - D \cdot k_2 \cdot \omega_H \cdot sin(k_2 \cdot \omega_H \cdot t) \left[\frac{m}{s}\right], \quad (8)$$

$$v_{cy} = \frac{dy_c}{dt} = A \cdot \omega_H \cdot cos(\omega_H \cdot t) - B \cdot k_1 \cdot \omega_H \, cos(k_1 \cdot \omega_H \cdot t) - D \cdot \omega_H \cdot k_2 \cdot cos(k_2 \cdot \omega_H \cdot t) \left[\frac{m}{s}\right]. \quad (9)$$

Then the absolute speed of point "C" of bisatellite 4 is:

$$v_c = \sqrt{\left(\frac{dy_c}{dt}\right)^2 + \left(\frac{dx_c}{dt}\right)^2} = \sqrt{v_{cx}^2 + v_{cy}^2} \ \left[\frac{m}{s}\right]. \quad (10)$$

Likewise, the acceleration of point "C" of bisatellite 4 was determined by the following expressions:

$$a_{cx} = \frac{d^2x_c}{dt^2} = -A \cdot \omega_H^2 \cdot cos(\omega_H \cdot t) + B \cdot k_1^2 \cdot \omega_H^2 \cdot cos(k_1 \cdot \omega_H \cdot t) - D \cdot k_2^2 \cdot \omega_H^2 \cdot cos(k_2 \cdot \omega_H \cdot t), \left[\frac{m}{s^2}\right], \quad (11)$$

$$a_{cy} = \frac{d^2y_c}{dt^2} = -A \cdot \omega_H^2 \cdot sin(\omega_H \cdot t) + B \cdot k_1^2 \cdot \omega_H^2 \, sin(k_1 \cdot \omega_H \cdot t) + D \cdot \omega_H^2 k_2^2 \cdot sin(k_2 \cdot \omega_H \cdot t), \left[\frac{m}{s^2}\right], \quad (12)$$

The absolute acceleration of point "C" of bisatellite 4 was determined by the expression:

$$a_c = \sqrt{\left(\frac{d^2x_c}{dt^2}\right) + \left(\frac{d^2y_c}{dt^2}\right)^2} = \sqrt{a_{cx}^2 + a_{cy}^2} \ [m/s^2] \quad (13)$$

The kinematic parameters of the bisatellite point O_4were determined by the following formulas:

$$x_{O_4}(t) = A \cdot cos(\omega_H \cdot t) - B \cdot cos(k_1 \cdot \omega_H \cdot t) \ [m], \quad (14)$$

$$y_{O_4}(t) = A \cdot sin(\omega_H \cdot t) - B \cdot sin(k_1 \cdot \omega_H \cdot t) \ [m], \quad (15)$$

$$R_{O_4} = \sqrt{x_{O_4}^2(t) + y_{O_4}^2(t)} \ [m]. \quad (16)$$

Where $k_1 = u_{2H} = \left(1 - u_{21}^{(H)}\right)$, or $k_1 = \left(1 + \frac{z_1}{z_2}\right)$; $A = r_1 + r_2 \ [m]$; $B = r_{3.} + r_4 \ [m]$;

$$v_{O_4x} = \frac{dx_{O_4}}{dt} = -A \cdot \omega_H \cdot sin(\omega_H \cdot t) + B \cdot k_1 \cdot \omega_H \cdot sin(k_1 \omega_H \cdot t) \left[\frac{m}{s}\right], \quad (17)$$

$$v_{O_4y} = \frac{dy_{O_4}}{dt} = A \cdot \omega_H \cdot cos(\omega_H \cdot t) - B \cdot k_1 \cdot \omega_H \, cos(k_1 \cdot \omega_H \cdot t) \left[\frac{m}{s}\right], \quad (18)$$

$$v_{O_4} = \sqrt{\left(\frac{dy_{O_4}}{dt}\right)^2 + \left(\frac{dx_{O_4}}{dt}\right)^2} = \sqrt{v_{O_4x}^2 + v_{O_4y}^2} \left[\frac{m}{s}\right], \quad (19)$$

$$a_{O_4x} = \frac{d^2x_{O_4}}{dt^2} = -A \cdot \omega_H^2 \cdot cos(\omega_H \cdot t) + B \cdot k_1^2 \cdot \omega_H^2 \cdot cos(k_1 \cdot \omega_H \cdot t) \left[\frac{m}{s^2}\right], \quad (20)$$

$$a_{O_4y} = \frac{d^2y_{O_4}}{dt^2} = -A \cdot \omega_H^2 \cdot sin(\omega_H \cdot t) + B \cdot k_1^2 \cdot \omega_H^2 \, sin(k_1 \cdot \omega_H \cdot t) \left[\frac{m}{s^2}\right], \quad (21)$$

$$a_{O_4} = \sqrt{\left(\frac{d^2x_{O_4}}{dt^2}\right) + \left(\frac{d^2y_{O_4}}{dt^2}\right)^2} = \sqrt{a_{O_4x}^2 + a_{O_4y}^2} \ \left[\frac{m}{s^2}\right], \quad (22)$$

Kinematic parameters of the bicarrier O_h center of gravity is:

$$x_{O_h}(t) = A \cdot cos(\omega_H \cdot t) - \frac{B}{2} \cdot cos(k_1 \cdot \omega_H \cdot t) \ [m], \quad (23)$$

$$y_{O_h}(t) = A \cdot sin(\omega_H \cdot t) - \frac{B}{2} \cdot sin(k_1 \cdot \omega_H \cdot t) \ [m], \quad (24)$$

$$R_{O_h} = \sqrt{x_{O_h}^2(t) + y_{O_h}^2(t)} \ [m], \quad (25)$$

Where $k_1 = u_{2H} = \left(1 - u_{21}^{(H)}\right)$, or $k_1 = \left(1 + \frac{z_1}{z_2}\right)$; $A = r_1 + r_2 \ [m]$; $B = r_{3.} + r_4 \ [m]$;

$$v_{O_h x} = \frac{dx_{O_h}}{dt} = -A \cdot \omega_H \cdot sin(\omega_H \cdot t) + \frac{B}{2} \cdot k_1 \cdot \omega_H \cdot sin(k_1 \omega_H \cdot t) \left[\frac{m}{s}\right], \quad (26)$$

$$v_{O_h y} = \frac{dy_{O_h}}{dt} = A \cdot \omega_H \cdot cos(\omega_H \cdot t) - \frac{B}{2} \cdot k_1 \cdot \omega_H \, cos(k_1 \cdot \omega_H \cdot t) \left[\frac{m}{s}\right], \quad (27)$$

$$v_{O_h} = \sqrt{\left(\frac{dy_{O_h}}{dt}\right)^2 + \left(\frac{dx_{O_h}}{dt}\right)^2} = \sqrt{v_{O_h x}^2 + v_{O_h y}^2} \left[\frac{m}{s}\right], \quad (28)$$

$$a_{O_h x} = \frac{d^2 x_{O_h}}{dt^2} = -A \cdot \omega_H^2 \cdot cos(\omega_H \cdot t) + \frac{B}{2} \cdot k_1^2 \cdot \omega_H^2 \cdot cos(k_1 \cdot \omega_H \cdot t) \left[\frac{m}{s^2}\right], \quad (29)$$

$$a_{O_h y} = \frac{d^2 y_{O_h}}{dt^2} = -A \cdot \omega_H^2 \cdot sin(\omega_H \cdot t) + \frac{B}{2} \cdot k_1^2 \cdot \omega_H^2 \, sin(k_1 \cdot \omega_H \cdot t) \left[\frac{m}{s^2}\right], \quad (30)$$

$$a_{O_h} = \sqrt{\left(\frac{d^2 x_{O_h}}{dt^2}\right) + \left(\frac{d^2 y_{O_h}}{dt^2}\right)^2} = \sqrt{a_{O_{h4} x}^2 + a_{O_h y}^2} \left[\frac{m}{s^2}\right]. \quad (31)$$

4 Results and discussion

Calculations were carried out based on the regularities obtained. The graphs of the change in kinematic parameters of the point *"C"* belonging to the bisatellite for $z_1 = 72, z_2 = 18, z_3 = 36, z_4 = 18, m = 5$mm n_H =20 rpm are shown in Figs. 2-8; forn_H = 40 rpm in Figs. 2-8.

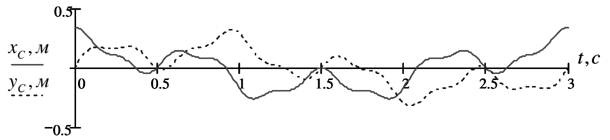

Fig. 2 *Patterns of changes in the projection of displacements of point "C" of the bisatellite for* $n_H = 20$ *rpm*

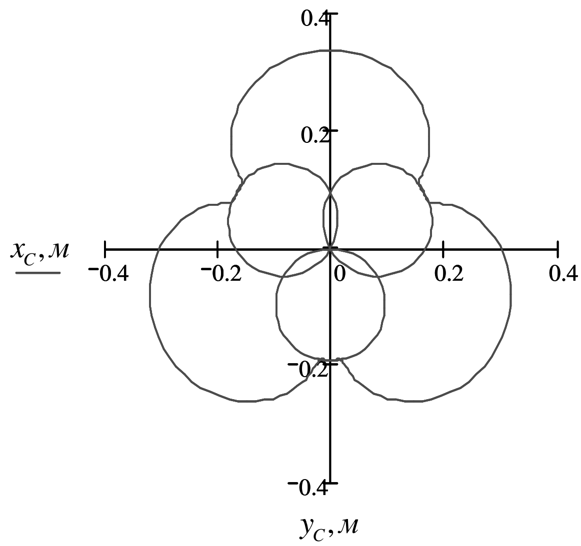

Fig. 3 *Trajectory of point "C" of the bisatellite for* $n_H = 20$*rpm*

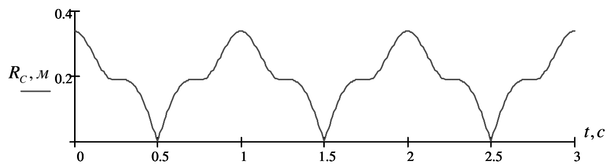

Fig. 4 *Pattern of changes in absolute displacements of point "C" of the bisatellite for* $n_H = 20$ *rpm*

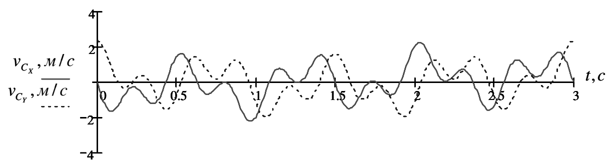

Fig. 5 *Pattern of changes in velocities of point "C" of the bisatellite for* $n_H = 20$ *rpm*

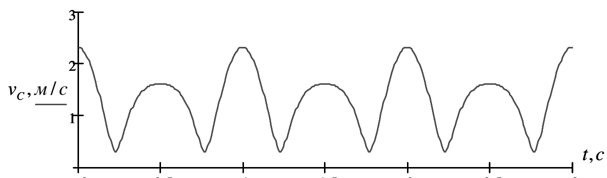

Fig. 6 *Pattern of changes in absolute velocities of point "C" of the bisatellite for* $n_H = 20$*rpm*

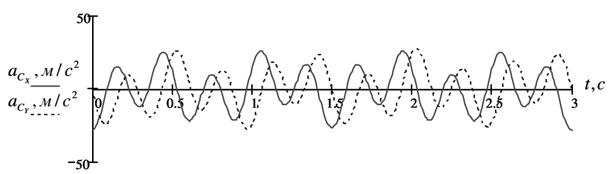

Fig. 7 *Pattern of changes in acceleration of point "C" of the bisatellite for* $n_H = 20$*rpm*

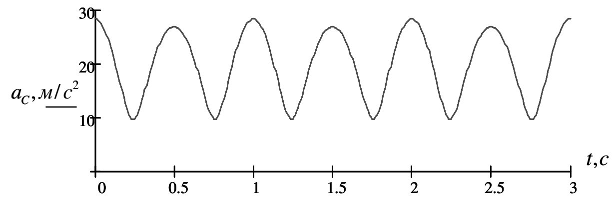

Fig. 8 *Pattern of changes in absolute acceleration of point "C" of the bisatellite for* $n_H = 20$*rpm*

In order to study the influence of the number of revolutions of the carrier on the kinematic characteristics of the bisatellite point, the center of gravity of the bisatellite and the bicarrier, we determined the extreme values of velocities and accelerations of these points v_{max}, v_{min}, a_{max}, a_{min}, a and the range of fluctuations in velocities and accelerations of points $H_v = v_{max} - v_{min}$, $H_a = a_{max} - a_{min}$, given in Tables 1, 2, 3.

Tab. 1 *Variation of the number of revolutions of the carrier* n_H *for point "C" of the bisatellite*

	n_H, rpm	$v_{max}\left[\frac{m}{s}\right]$.	$v_{min}\left[\frac{m}{s}\right]$.	$H_v\left[\frac{m}{s}\right]$.	$a_{max}\left[\frac{m}{s^2}\right]$.	$a_{min}\left[\frac{m}{s^2}\right]$.	$H_a\left[\frac{m}{s^2}\right]$.
1	20	3.11	0.34	2.77	49.15	18.49	30.66
2	30	4.67	0.51	4.16	110.6	41.6	69
3	40	6.22	0.68	5.54	196.6	73.6	123
4	50	7.78	0.85	6.93	307.2	115.5	191.7
5	60	9.33	1.02	8.31	442.4	166.4	276

Tab. 2 *Variation of the number of revolutions of the carrier* n_H *for the center of gravity of the bisatellite*

	n_H, rpm	$v_{max}\left[\frac{m}{s}\right]$.	$v_{min}\left[\frac{m}{s}\right]$.	$H_v\left[\frac{m}{s}\right]$.	$a_{max}\left[\frac{m}{s^2}\right]$.	$a_{min}\left[\frac{m}{s^2}\right]$.	$H_a\left[\frac{m}{s^2}\right]$.
1	20	1.88	0.94	0.94	15.79	13.82	1.97
2	30	2.83	1.41	1.42	35.53	31.09	4.44
3	40	3.77	1.88	1.89	63.16	55.27	7.89
4	50	4.71	2.36	2.35	98.7	86.4	12.3
5	60	5.66	2.83	2.83	142.12	124.4	17.72

Tab. 3 *Variation of the number of revolutions of the carrier* n_H *for the center of gravity of the bicarrier*

	n_H, rpm	$v_{max}\left[\frac{m}{s}\right]$.	$v_{min}\left[\frac{m}{s}\right]$.	$H_v\left[\frac{m}{s}\right]$.	$a_{max}\left[\frac{m}{s^2}\right]$.	$a_{min}\left[\frac{m}{s^2}\right]$.	$H_a\left[\frac{m}{s^2}\right]$.
1	20	1.18	0.24	0.94	8.39	6.42	1.97
2	30	1.77	0.35	1.42	18.88	14.43	4.49
3	40	2.36	0.47	1.89	33.56	25.66	7.90
4	50	2.94	0.59	2.35	52.43	40.1	12.33
5	60	3.53	0.7	2.83	75.5	57.74	17.76

5 Conclusions

Based on the research conducted, the following conclusions can be drawn:

- Analytical expressions are substantiated for determining the kinematic parameters of the drives of the working body of an intermittent mixing machine with planetary with double satellites and biplanetary mechanisms.
- A program has been compiled to implement them on a computer. Based on the results of computer calculations, the laws of change of displacements, velocities and accelerations of points of the working body for drives with planetary with double satellites and biplanetary mechanisms are determined. Regularities of the influence of the velocity parameters of the leading links on the kinematic characteristics of these mechanisms are established.

References

[1] CHEN, F.Y. (1977). A survey of the state of the art of cam system dynamics. In: *Mechanism and Machine Theory*. Vol. 12, No. 3, pp. 201–224. ISSN 0094-114X.

[2] YANG, Y., J. WANG, S. ZHOU, et al. (2019). Design of a novel coaxial eccentric indexing cam mechanism. In: *Mechanism and Machine Theory*. Vol. 132, pp. 1–12.

[3] KARIMOV, K., KARIMOV, R., AKHMEDOV, A. (2016). *Development of the theoretical foundations of mechanisms with controlled*

parameters and constraints. Monograph, p. 326. Tashkent State Technical University, Tashkent.

[4] KARIMOV, K., UMURZAKOV, A. (2015). *Theory and design of friction mechanisms with controlled friction*. Monograph, p. 147. Tashkent State Technical University, Tashkent.

[5] KARIMOV, K., NAGAEV R., USMANKHODJAEV, H. (1984). Analysis and synthesis of the planetary frictional mechanisms with unround wheels. In: *Proceedings of the Sixth World Congress on the Theory of Machines and Mechanism*, Vol. 1, p. 230–232. New Delhi, India.

[6] KARIMOV, K., ABDUVALIEV, U., AKHMEDOV, A. (2015). Development and analytical realization of the mathematical model of controlled motion of a positioning mechanism. Part 1. In: *European Applied Sciences*. Vol 3, pp. 61–63. Stuttgart, Germany. ISSN 2195 – 2183.

[7] NEMATOV, E., KHUDJAEV, M., KHASANOV, B. (2021). Development of a mathematical model of dynamic characteristics of a drive with a planetary mechanism. In: *E3S Web of Conferences*. Vol. 258, pp. 1–8. ISSN 22671242.

[8] MINÁRIK, M. BODNÁR, F. (2019). Dynamic Analysis of the Crank Mechanism through the Numerical Solution. In: *Manufacturing Technology*. Vol. 19, No. 6, pp. 1003–1009. ISSN 12132489.

[9] SEGLA, S. (2018). Kinematic analysis and optimization of a wheel loader mechanism. In: *Manufacturing Technology*. Vol. 18, No. 2, pp. 309–314. ISSN 12132489.

[10] VAVRO, J., J. VAVRO, P. KOVÁČIKOVÁ, et al. (2017). Kinematic and Dynamic Analysis and Distribution of Stress in Items of Planar Mechanisms by Means of the MSC ADAMS Software. In: *Manufacturing Technology*. Vol. 17, No. 3, p. 31. ISSN 12132489.

[11] ARTOBOLEVSKY, I. (1988). *Theory of mechanisms and machines*. Moscow: Nauka.