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Modeling of kinematics and kinetostatics of planetary-lever mechanism

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Abstract. The article provides analytical formulas for determining the kinematic parameters of a planetary-lever mechanism with one and two degrees of mobility. The obtained formulas were realized on a computer, in the program Math CAD 15. According to the results of calculations on a computer, regularities were established for the change in the kinematic parameters of the links of the mechanisms under consideration. The influence of the direction and magnitude of the angular velocity of the central wheel on the kinematic characteristics of the planetary-lever mechanism with two degrees of mobility has been determined. Formulas are created to determine the reaction in kinematic pairs of a planetary-lever mechanism with one degree of motility. Based on the calculation results and analysis provided, regularities of the reaction change in the kinematic pairs, as well as balancing moments in the carrier, were established during the basic geometric parameters of this mechanism were varied.

1. Introduction

Modern technological machines are widely used mechanisms whose slave links make complex movements. Further improvement of these mechanisms is inextricably linked with the creation of new combined mechanisms with a wider range of changes in kinematic parameters. The object of the study is a combined planetary-lever mechanism, which is formed by connecting a second-lead group of the second type to the planetary gear pinion.

In this mechanism, the offset of the change in the gear ratio between the central wheel and the pinion, as well as the change in the point of connection of the two-wire group of the second type to the pinion, allows changing the laws of motion of the driven link, in this case, the slider in a wide range of diapason in the kinematic parameters of the driven link. Despite a large number of studies and analyses on combined mechanisms in the issue of kinematics and kinetostatics of combined planetary-lever mechanisms, they were not adequately directed in literary sources. Therefore, research directed at creating mathematical models describing the kinematics and kinetostatics of planetary-lever mechanisms with a two-lead group of the second type is actual.

The object of the study the state of the issue, a review of scientific papers on combined gear-lever mechanisms, which are given below.

The article [1] presents the results of a kinematic analysis of the gear-lever mechanism of intermittent rotational movement with external gearing of the wheels. It has been established that the backstage has a stop equal to 23% of the cycle time corresponding to half a turn of the carrier. The regularities of the change in the angular velocity and acceleration of the output link of the mechanism are determined.

The work [2].presents a calculation of the geometric characteristics of the loop of the epicyclic trajectory of the motion of the link point of the gear lever mechanism, which allows for an approximate stop to the actuator. A comparative analysis of various mathematical methods for determining the transverse and longitudinal sizes of the epicycloid loop, which serve as the basis for calculating the duration and quality of the quasi-stop of the actuator, is given.

The [3] presents a mathematical model of a gear-lever mechanism based on elliptical gear wheels that implements movements with the output link stopping without breaking the kinematic chain. A comparison of the dynamic characteristics various mechanisms are made and suggested some recommendations for choosing mechanisms with the best dynamic properties.

In the article [4], the accessibility of the study of kinematic analysis of a gear-lever differential gear with a parallelogram lever circuit by the graph-analytical method is given. This method has clearness, is convenient for monitoring results and allows you to quickly solve applied design problems, and also develops engineering intuition to evaluate the capabilities of a mechanism by its kinematic scheme.

In the article [5], the features of an epicyclic misaligned mechanism are described, the axles of the central wheels of which are shifted by a given eccentricity. Four schemes of mechanisms with different kinematic capabilities and with a complex plane rotational movement of their output links are obtained.

In the article [6], the kinematics and synthesis elements of a rotary piston engine based on planetary-gear and rocker linkage mechanisms are investigated. The proposed design allows you to create a rotary piston engine with four working chambers. The formulas of the kinematic characteristics of the links of the mechanism are presented.

In the article [7], an analysis of a connecting device based on a planetary mechanism with an unbalanced pinion was carried out. The use of pinions with a fixed unbalanced additional mass in the connecting device is considered. An option is proposed for using an unbalanced load with the ability to mix relative to the pinion. Differential formulas of system motion are compiled. The method of bringing the system to the operating mode due to the purposeful change in dynamic properties is studied.

In the article [8], a generalized model of the gear-lever mechanism of periodic movement based on non-circular wheels is presented, which implements movements with the output link stopping without breaking the kinematic chain. The main types of gear-lever mechanisms with elliptical gears are presented, their ultimate kinematic capabilities are determined. Formulas are obtained for determining the conjugate curves of non-circular gears for gear-link mechanisms that implement a different number of stops of the driven link.

An analysis of studies on gear-lever mechanisms showed that practically unexplored issues related to the determination of the kinematic parameters of planetary-lever mechanisms formed by attaching a two-lead group of the second type to the pinion of the basic planetary mechanism. Analytical formulas for determining the load of kinematic pairs of these mechanisms are not presented.

The main aim of the research is to create the theoretical foundations for the design of combined planetary-lever mechanisms in particular:

1. Preparation of analytical formulas to determine the kinematic parameters of links and points of the planetary-lever mechanism with one and two mobility;
2. Justification of analytical formulas for determining the reaction in the kinematic pairs of the planetary-lever mechanism;
3. Development of programs for the implementation of the obtained mathematical models on a computer.

2. Methods

Research methods of this research are the general methods of theoretical mechanics and the theory of mechanisms and machines. One of the ways to expand the kinematic capabilities of the driven links of mechanisms is to develop and study combined mechanisms [9-11]. In Fig. 1. the kinematic diagram of the combined planetary linkage mechanism is given. As it is seen, this mechanism is formed by connecting to the pinion of the planetary mechanism a two-lead group of the second type. The mechanism which is under consideration consists of the following links; 1 is the central wheel, *N* is the carrier, 2 is the pinion, 3 is the connecting rod, 4 is the slider.

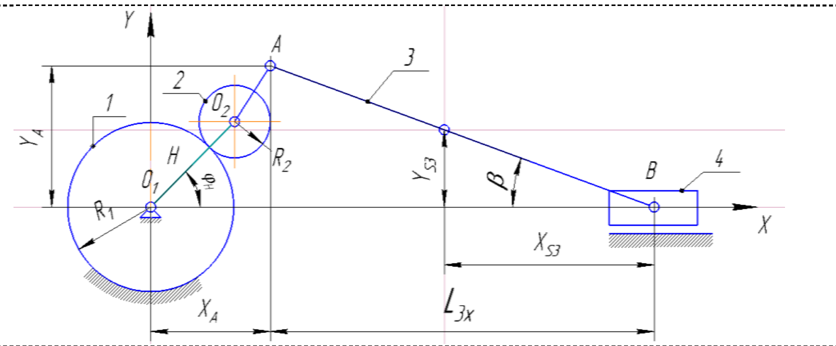


Figure 1. Kinematic diagram of the planetary link mechanism with a two-wire group of the second type.

Hinge *A* to which the connecting rod joins is rigidly connected to the pinion. This mechanism may have one or two degrees of mobility. If the central wheel is stationary, then the degree of mobility of the mechanism will be equal to one. If the central wheel is movable, then the mechanism will have two degrees of mobility. Analytical formulas are compiled to determine the kinematic parameters of the planetary-lever mechanisms with one and two degrees of mobility.

When, $\omega_1 = 0$, i.e. the central wheel is stationary, the following analytical formulas are obtained for determining the displacements, velocities, and accelerations of the center of mass of the connecting rod and slider, which has the following form

Projections of the center of mass of the connecting rod, the slider on the coordinate axis

$$x_{S_3} = x_A + AS_3 \cos\beta \,,\ x_{S_3} = R_H \cos\varphi_H - O_2A\cos(\varphi_2 + AS_3\cos\beta) \qquad (1)$$

Where $x_A = R_H \cdot \cos\varphi_H - O_2A \cdot \cos(\varphi_2)$, $AS_3 = BS_3 = L_3/2$ distance from point A to the center of mass of the connecting rod, L_3 is the length of connecting rod 3, β is the angle of connecting rod, R_H is the length of the carrier, φ_H is the angle of the carrier, $\varphi_2 = (1 + R_1/R_2)\varphi_H$ is the rotation angle of pinion 2; where R_1 is the radius of a pitch circle of a motionless gear wheel 1, R_2 is the pinion pitch radius 2.

$$y_{S_3} = BS_3 \cdot \sin\beta \ \ y_{S_3} = BS_3 \sin\left(\arcsin\left((R_H \sin\varphi_H - O_2A\sin\varphi_2)/L_3\right)\right) \qquad (2)$$

$$x_B = x_A + L_3x \ \ L_{3x} = L_3 \cdot \cos\beta$$

$$x_B = R_H \cos\varphi_H - O_2A\cos\varphi_2 + L_3\cos\left(\arcsin\left(y_A/L_3\right)\right) \qquad (3)$$

The analytical expression for the projection of the center of mass of the connecting rod and slider on the coordinate axis has the following formula

$$v_{x_{S_3}} = O_2A\omega_2 \sin\varphi_2 - AS_3\beta' \sin\beta - R_H\omega_H \sin\varphi_H \qquad (4)$$

$$v_{y_{S_3}} = \frac{BS_3(O_2A\omega_2\cos\varphi_2 - R_H\omega_H\cos\varphi_H)}{L_3} \qquad (5)$$

$$a_{x_{S_3}} = O_2A\varepsilon_2\sin\varphi_2 - R_H\varepsilon_H\sin\varphi_H - AS_3\beta''\sin\beta + O_2A\omega_2^2\cos\varphi_2 \qquad (6)$$
$$-R_H\omega_H^2\cos\varphi_H - AS_3(\beta')^2\cos\beta$$

$$a_{y_{S_3}} = \frac{BS_3\left(O_2A\varepsilon_2\cos\varphi_2 - R_H\varepsilon_H\cos\varphi_H - O_2A\omega_2^2\sin\varphi_2 + R_H\omega_H^2\sin\varphi_H\right)}{L_3} \quad (7)$$

$$v_B = O_2A\omega_2\sin\varphi_2 - R_H\omega_H\sin\varphi_H - \frac{y_A\left(y_A\right)'}{L_3\sqrt{1-\left(\frac{y_A}{L_3}\right)^2}} \quad (8)$$

$$a_B = O_2A\omega_2^2\cos\varphi_2 - R_H\omega_H^2\cos\varphi_H - \frac{\left(y_A'\right)^2}{L_3\sqrt{1-\left(\frac{y_A}{L_3}\right)^2}} - \frac{\left(y_A'\right)^2(y_A)^2}{L_3^3\sqrt[3]{1-\left(\frac{y_A}{L_3}\right)^2}} - \frac{\left(y_A''\right)^2}{L_3\sqrt{1-\left(\frac{y_A}{L_3}\right)^2}} \quad (9)$$

The full values of the velocities and accelerations of the center of mass of the connecting rod were determined by the following formulas

$$v_{S_3} = \sqrt{\left(v_{S_3x}\right)^2 + \left(v_{S_3y}\right)^2} \quad (10)$$

$$a_{S_3} = \sqrt{\left(a_{y_{S_3}}\right)^2 + \left(a_{x_{S_3}}\right)^2} \quad (11)$$

The angle of rotation of the connecting rod, its angular velocity and acceleration was determined by the following formulas

$$\sin\beta = \frac{Y_A}{L_3} \;\; \beta = \arcsin\left(\frac{Y_A}{L_3}\right) \;\; L_{3x} = L_3\cdot\cos\beta \;\; L_{3x} = L_3\cdot\cos\left(\arcsin\left(\frac{Y_A}{L_3}\right)\right) \quad (12),$$

$$\omega_3 = \frac{v_{A_y}}{L_3\sqrt{1-\left(\frac{y_A}{L_3}\right)^2}}$$

$$\varepsilon_3 = \frac{a_{A_y}L_3\sqrt{1-\left(\frac{y_A}{L_3}\right)^2} + \frac{y_A\left(v_{A_y}\right)^2}{L_3\sqrt{1-\left(\frac{y_A}{L_3}\right)^2}}}{L_3^2\left[1-\left(\frac{y_A}{L_3}\right)^2\right]} \quad (13)$$

Analytical formulas (1-13) were implemented in Math CAD 15 computer program. According to the results of calculations, patterns of changes were obtained x_{S3}, y_{S3}, v_{XS3}, v_{YS3}, v_{S3}, a_{XS3}, a_{YS3}, a_{S3}, ω_3, ε_3, x_B, v_B, a_B in graphic and numerical forms. To study the influence of the angular velocity of the carrier on the kinematic parameters of the planetary-lever mechanism with one degree of mobility, the angular velocity of the carrier was changed from 5 to 25s^{-1} the step of variation was 5s^{-1}.

It is important that during the designing of mechanisms have the determination of the reaction in kinematic pairs, as well as the forces acting on the links of mechanisms. Calculation of the forces acting on various parts of the mechanism during its movement can be done if the laws of motion of all parts of the mechanism and external forces applied to the mechanism are known.

Force calculations of mechanisms can be made by a variety of methods. In the theory of mechanisms and machines, the method of force calculation of mechanisms based on the ordinary formulas of equilibrium of solids has become widely used [12-14] . This method is reduced to use in solving problems of the dynamics of equilibrium formulas in the method of D'Alember. When using the D'Alember principle, in addition to external forces acting on a single link, the mechanisms of inertia are applied to the calculation of mechanisms [15–20].

In figure 2 below the design diagram of a planetary-lever mechanism with one degree of mobility with forces to moments acting on its links is presented.

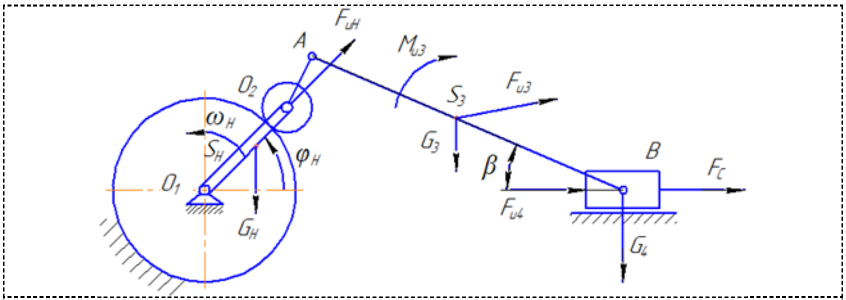
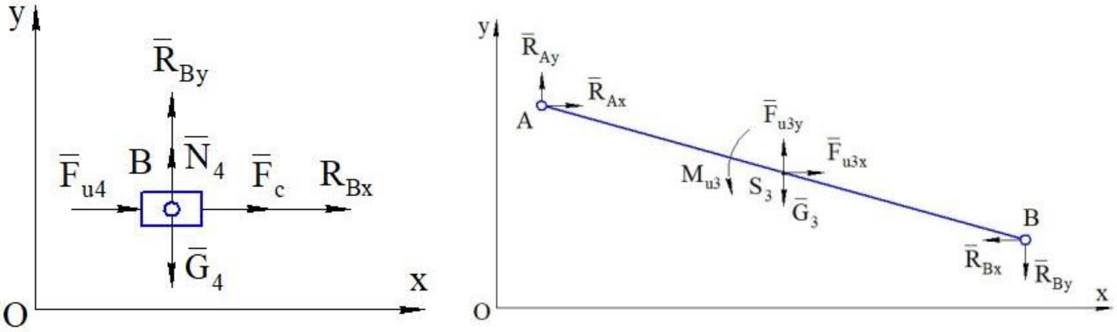


Figure 2. The design diagram of a planetary-lever mechanism with one degree of mobility with forces to moments acting on its links is presented.

Where $\overline{G}_H$ is the force of gravity of carrier, $\overline{G}_2$ is the force of gravity of pinion, $\overline{G}_3$ is the force of gravity of connecting rod, $\overline{G}_4$ is the force of gravity of slider, $\overline{F}_C$ is drag forces acting on the slider, M_Y is balancing moment on the carrier shaft, $\overline{F}_{u3}$ is inertia force acting on the center of mass of the connecting rod, M_{u3} is moment of inertia acting on the connecting rod, $\overline{F}_{u4}$ is forces of inertia acting on the slider.

If the angular velocity of the carrier is constant, then the force of inertia acting on the carrier and the pinion will be directed along the carrier. As it is seen, this mechanism is formed by attaching a two-lead group of the second type to an arbitrary pinion point, and this mechanism has one degree of mobility. Consider the equilibrium conditions of the individual links of this mechanism. Figure 3. shows the design diagram of the slider (a) and connecting rod (b) with the forces and moments acting on them.



(a) (b)

Figure 3. Calculated schemes of slider (a) and connecting rod (b)

Known forces act on the slider: the force inertia of the slider $\overline{F}_{u4}$, the technological resistance $\overline{F}_{C4}$, and the force of gravity of the slider $\overline{G}_4$. The inertia force $\overline{F}_{u4} = -\frac{\overline{G}_4}{g}\cdot\overline{a}_B$ and direction along the axis AX, the technological resistance force $\overline{F}_{c4}$ acting on the slider is also directed along the axis AX, the force of gravity of the slider $\overline{G}_4$ is directed vertically down. An unknown reaction in the hinge also acts on the slider $\overline{R}_B$, which we represent as two components $\overline{R}_{BX}$ and $\overline{R}_{BY}$ directed along the coordinate axes.

The magnitude of the total reaction in the hinge «B» equal $R_B=\sqrt{(R_{BX})^2+(R_{BY})^2}$. The slider also has a reaction from the side of the guide $\overline{N}_4$, which in this case passes through the hinge axis "B" and directed perpendicular to the guide. The equilibrium conditions of the slider can be written as follows.

$$\begin{cases}\Sigma X=0\\ \Sigma Y=0\end{cases}\quad\begin{cases}F_{u4}+F_{C4}+R_{BX}=0_{и}=0\\ R_{BY}+N_4-G_4=0\end{cases}\qquad(14)$$

Now proceed to consider the balance of the connecting rod. The design diagram of the connecting rod with the forces and moments are shown in Fig. 3 (b).

Known forces act on the connecting rod: the force of inertia of connecting rod $\overline{F}_{u3}$, which is presented in the form of two components $\overline{F}_{u3X}$, $\overline{F}_{u3Y}$, gravity force of connecting rod $\overline{G}_3$, which is directed vertically down, the moment from the forces of inertia $\overline{M}_{u3}$. Components of the force of inertia $\overline{F}_{u3}$, acting in the center of mass of the connecting rod S_3 were determined with following formulas $\overline{F}_{u3X}=-\frac{\overline{G}_3}{g}\cdot\bar{a}_{S3X}$, $\overline{F}_{u3Y}=-\frac{\overline{G}_3}{g}\cdot\bar{a}_{S3Y}$, the moment from the inertia forces of the connecting rod is $M_{u3}=-J_{S3}\cdot\varepsilon_3$ here J_{S3} - connecting rod inertia moment relative to the centre of mass, ε_3 - angular acceleration of connecting rod. Unknown reaction force components in the hinge "B" joint act on the connecting rod; their directions are opposite to the directions of the reaction components in the "B" joint acting on the slider. The unknown reaction in hinge "A" is presented in two components $\overline{R}_{AX}$, $\overline{R}_{AY}$. The magnitude of the total reaction in the hinge "A" was determined with the following formula $R_A=\sqrt{(R_{AX})^2+(R_{AY})^2}$.

For a connecting rod, three formulas of equilibrium can be written: two in projections on the coordinate axis X, Y, and one formula of moments relative to the center of mass of the connecting rod S_3.

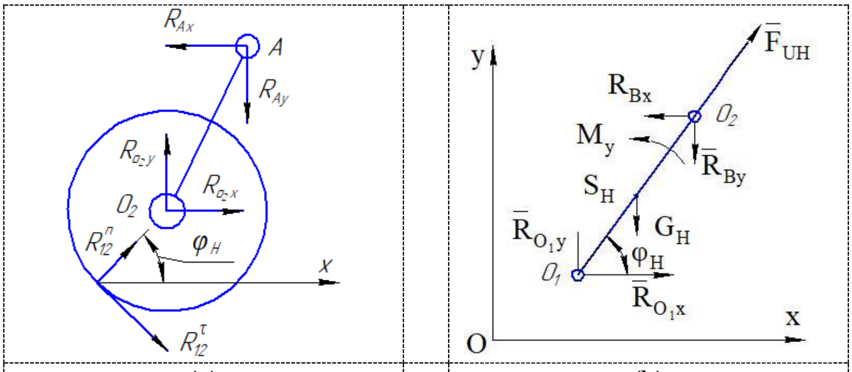
$$\begin{cases}\Sigma X=0 & R_{AX}+F_{u3X}-R_{BX}=0\\ \Sigma Y=0 & R_{AY}+F_{u3Y}-G_3-R_{BY}=0\\ \Sigma M_{S3}=0 & -R_{BY}\cdot l_{BS3}\cdot\cos\beta-R_{BX}\cdot l_{BS3}\cdot\sin\beta+M_{u3}-R_{AX}\cdot l_{AS3}\cdot\sin\beta-R_{AY}\cdot l_{AS3}\cdot\cos\beta=0\end{cases}\qquad 15)$$

Pinion equilibrium conditions: the following reaction forces act on the pinion from the connecting rod side; $\overline{R}_{AX}$, $\overline{R}_{AY}$, as well as the reaction force acting on pinion 2 from the side of the fixed central wheel 1 $\overline{R}_{21}$ and reaction force in the support O_2 of pinion, which are present below $\overline{R}_{O_2X}$, $\overline{R}_{O_2Y}$. The magnitude of the total reaction on the shaft of the pinion $\overline{R}_{02}$ determined with the following formula $R_{02}=\sqrt{(R_{02x})^2+(R_{02y})^2}$.

Fig. 4. shows the calculated scheme of the pinion (a) and carrier (b)

For a pinion may write three formulas of equilibrium: two in projections on the coordinate axis X, Y, and one formula of moments relative to the axis of rotation of the pinion.

$$\begin{cases}\Sigma X=0 & R_{12}^{\tau}(\cos((3\pi/2)+\varphi_H))+R_{12}^{n}(\cos\varphi_H)+R_{O_2X}+R_{AX}=0\\ \Sigma Y=0 & R_{12}^{\tau}(\sin((3\pi/2)+\varphi_H))+R_{12}^{n}(\sin\varphi_H)+R_{O_2}y-R_{AY}=0\\ \Sigma M_{O2}=0 & R_{12}^{\tau}\cdot r^2-R_{AY}(O_2A)\cdot\cos\varphi_2+R_{AY}(O_2A)\cdot\sin\varphi_2=0\end{cases}\qquad(16)$$



(a) (b)

Figure.4. Calculated scheme of the pinion (a) and carrier (b)

The calculated scheme of the carrier with the forces of acting on is shown in Fig.4. (b). Known gravity acts on the carrier $\bar{G}_H$, directed vertically down, the inertia force drove $\bar{F}_H$ applied at its center of mass S_H, which is equal $\bar{F}_H = \frac{\bar{G}_H}{g} \cdot \omega_H^2 \cdot l_{0_1 S_H}$. Unknown reaction components in the joints act on the carrier "O_1" and "O_2", as well as a balancing moment M_y.

For a carrier may write three formulas of equilibrium: two in projections on the coordinate axis X, Y and one formula of moments relative to the hinge"O_1".

$$\begin{cases} \Sigma X = 0 & R_{O_1X} - R_{O_2X} + F_{uH} \cos\varphi_H = 0 \\ \Sigma Y = 0 & R_{O_1Y} - G_H - R_{O_2Y} + F_{uH} \sin\varphi_H = 0 \\ \Sigma M_{O_1} = 0 & -G_H \cdot l_{O_1S_H} \cdot \cos\varphi_H + M_y - R_{O_2Y}(O_1O_2)\cos\varphi_H + R_{O_2X}(O_1O_2)\sin\varphi_H = 0 \end{cases} \tag{17}$$

The complete reaction in the carrier $\bar{R}_{01}$ determined with the formula $R_{01} = \sqrt{(R_{01x})^2 + (R_{01y})^2}$

The system of formulas (14-17) was solved on a computer program Math CAD 15. According to the results of calculations on a computer, patterns of changes in the reactions in the kinematic pairs of the mechanism as well as the balancing moment on the carrier shaft were obtained.

Calculations on a computer were performed with variations in the angular velocity of the carrier from 5 s^{-1} to 25 s^{-1} with steps 5 s^{-1}; the distance from the center of the pinion to the hinge "A" from 0 to 0.1 m in increments of 0.02 m; the resistance force acting on the slider from 0 to 1000 N in increments of 200 N. The calculation results are shown in Table 3

3. Results and Discussion

Tables 1 and 2 show the numerical values of the main kinematic parameters of the planetary-lever mechanism with one and two degrees of mobility with a variation in the angular velocity of the carrier ω_H and central wheel ω_1. The output parameters were: $R_1 = 0.15м$; $R_2 = 0.05м$; $R_H = 0.2м$; $L_{O_2A} = 0.07м$; $L_3 = 0.81м$.

Table 1. Numerical values of the main kinematic parameters of the planetary-lever mechanism with one degree of mobility

The degree of mobility mechanism $w=1$, wherein $\omega_1 = 0$, variation ω_H										
	ω_H (c^{-1})	$v_{B\max}$ (м/с)	$v_{B\min}$ (м/с)	Hv_B (м/с)	$a_{B\max}$ (м/с^2)	$a_{B\min}$ (м/с^2)	Ha_B (м/с^2)	$v_{S3\max}$ (м/с)	$v_{S3\min}$ (м/с)	Hv_{S3} (м/с)
1	5	2.5	-2.5	5.0	28	-37	65	2.4	0.2	2.2

2	10	5	-5	10	111.8	-148.8	260.6	4.8	0.4	4.4
3	15	7.5	-7.5	15	251.5	-334.9	586	7.2	0.6	6
4	20	10	-10	20	447.2	-595.5	1042	9.7	0.8	8
5	25	12.5	-12.5	25	698.8	-930	1628	12.1	1	11.1

	ω_H (c-1)	$a_{S3\max}$ (м/c2)	$a_{S3\min}$ (м/c2)	Ha_{S3} (м/c2)	$\omega_{3\max}$ (c-1)	$\omega_{3\min}$ (c-1)	$H\omega_3$ (c-1)	$\varepsilon_{3\max}$ (c-2)	$\varepsilon_{3\min}$ (c-2)	$H\varepsilon_3$ (c-2)
1	5	34.3	11.7	22.6	2.6	-2.9	5	42.4	-42	84.4
2	10	137.2	46.8	90.4	5.3	-6	11.3	169.5	-169.5	339
3	15	308.7	105.2	203.5	7.9	-8.9	16.8	381.3	-381.3	762
4	20	549	187	362	10.6	-11.9	22	678	-678	1356
5	25	858	293	565	13.2	-14.8	28	1059	-1059	2118

Table 2. Numerical values of the main kinematic parameters of the planetary-lever mechanism with two degrees of mobility

The degree of mobility mechanism $w = 2$, wherein $\omega_H = 5c^{-1}$, variation ω_1

	ω_1 (c-1)	$v_{B\max}$ (м/c)	$v_{B\min}$ (м/c)	Hv_B (м/c)	$a_{B\max}$ (м/c^2)	$a_{B\min}$ (м/c^2)	Ha_B (м/c^2)	$v_{S3\max}$ (м/c)	$v_{S3\min}$ (м/c)	Hv_{S3} (м/c)
1	10	1.38	-1.38	2.76	11.9	-6.4	18.3	1.5	0.15	1
2	15	2.65	-2.65	5.3	47.8	-48.8	96.6	2.6	0.4	2.2
3	20	3.8	-3.8	7.6	113	-124	237	3.8	0.9	2.9
4	25	4.9	-4.9	9.8	207.9	-236.7	444.6	4.8	1.4	3.4
5	30	5.9	-5.9	11.8	332.4	-383.4	715.8	5.9	1.9	4

	ω_1 (c-1)	$a_{S3\max}$ (м/c^2)	$a_{S3\min}$ (м/c^2)	Ha_{S3} (м/c^2)	$\omega_{3\max}$ (c-1)	$\omega_{3\min}$ (c-1)	$H\omega_3$ (c-1)	$\varepsilon_{3\max}$ (c-2)	$\varepsilon_{3\min}$ (c-2)	$H\varepsilon_3$ (c-2)
1	10	11.9	0.2	11.7	2.1	-1.1	3.2	8.7	-8.7	17.4
2	15	48.2	19.6	28.6	3.4	-3.4	6.8	57.3	-57.3	114.6
3	20	118.8	53.3	65.5	4.7	-4.6	9.3	146	-146	292
4	25	222.7	102.9	119.8	5.9	-5.9	11.8	276	-276	552
5	30	361	168	193	7.2	-7.2	14.4	448.8	-448.8	896.4

In Tables 1,2: $v_{B\max}$, $a_{B\max}$ are maximum values of linear velocity and acceleration of the slider, $v_{B\min}$, $a_{B\min}$ are minimum linear velocity and slider acceleration, Hv_B, Ha_B are the swing range of linear velocity and acceleration, $v_{S3\max}$, $a_{S3\max}$ are maximum linear velocity and acceleration of the center of mass of the connecting rod, $v_{S3\min}$, $a_{S3\min}$ are minimum values of linear velocity and acceleration of the center of mass of the connecting rod, Hv_{S3}, Ha_{S3} are linear velocity swings and acceleration, $\omega_{3\max}$, $\omega_{3\min}$ are the maximum and minimum value of the connecting rod angular velocity, $H\omega_3$ is swing range of angular velocity, $\varepsilon_{3\max}$, $\varepsilon_{3\min}$ are the maximum and minimum value of the angular acceleration of the connecting rod, $H\varepsilon_3$ is the amplitude of the angular acceleration of the connecting rod.

Analysis of the calculations showed that, with an increase in the angular velocity of the carrier in the above range, in the diapason the swing range v_{S3}, a_{S3}, ω_3, ε_3, V_B, a_B increases. Now consider the case when the planetary-lever mechanism has two degrees of mobility. In this case, the angular velocity of the central wheel affects the angular velocity of the pinion. According to the formula of Villis [11], the angular velocity of the pinion is equal to $\omega_2 = -(R_1/R_2)\cdot\omega_1 + \left(1 + R_1/R_2\right)\cdot\omega_H$

The angular velocity of the central wheel is negative if its direction does not coincide with the direction of the angular velocity of the carrier. In this case, the angular velocity of the pinion increases by an amount equal to $\left(R_1/R_2\right)\cdot\omega_1$ in comparison with the angular velocity of the pinion with a degree of mobility of the mechanism equal to unity. If the direction of the angular velocity of the central wheel coincides with the direction of the angular velocity of the carrier, then the angular velocity of the pinion decreases by an amount equal to $\left(-R_1/R_2\right)\cdot\omega_1$ in comparison with the angular velocity of the pinion with a degree of mobility of the mechanism equal to unity. The kinematic parameters of the planetary linkage mechanism can be determined by analytical formulas compiled for the planetary linkage mechanism with one degree of mobility. In this case, it is necessary to take into account the direction of rotation and the value of the angular velocity of the central wheel.

Analysis of the calculation results showed that, with variation ω_H , the swing range: the reaction in the carrier increases from 1408 N to 33879 N; on the shaft of the pinion increases from 1393 N to 33879 N; the magnitude of the swing range between the slider and the guide increases from 764 N to 19,100 N, the balancing moment on the carrier shaft increases from 464 Nm to 11,000 Nm

Table 3. The results of calculations of the reaction in kinematic pairs of a planetary-lever mechanism with one degree of mobility

The degree of mobility mechanism $w=1$, wherein $O_2A=0.04м$, $F_c=200H$ variation ω_H

	ω_H (c-1)	$R_{01\max}$ (H)	$R_{01\min}$ (H)	$H_{R_{01}}$ (H)	$R_{02\max}$ (H)	$R_{02\min}$ (H)	$H_{R_{02}}$ (H)	$R_{A\max}$ (H)	$R_{A\min}$ (H)	H_{R_A} (H)
1	5	1440	32	1408	1400	7.1	1393	634	394	240
2	10	5500	39.5	5460.5	5500	23	5477	2500	1600	900
3	15	12000	22.3	11977	12000	42	11958	5600	3600	2000
4	20	22000	58	21942	22000	89	21911	10000	6400	3600
5	25	34000	121	33879	34000	153	33847	16000	10000	6000

	ω_H (c-1)	$R_{B\max}$ (H)	$R_{B\min}$ (H)	H_{R_B} (H)	$N_{4\max}$ (H)	$N_{4\min}$ (H)	H_{N_4} (H)	$M_{y\max}$ (Hm)	$M_{y\min}$ (Hm)	H_{M_Y} (Hm)
1	5	422	142	280	472	-292	764	229	-235	464
2	10	1570	570	1000	1600	-1400	3000	895	-897	1782
3	15	3500	1300	2200	3500	-3300	6800	2000	-2000	4000
4	20	6200	2300	3900	6200	-6000	12200	3500	-3500	7000
5	25	9600	3600	6000	9600	-9500	19100	5500	-5500	11000

The degree of mobility mechanism $w=1$, wherein $\omega_H=10c^{-1}$, $F_c=200H$ variation O_2A

	O_2A (м)	$R_{01\max}$ (H)	$R_{01\min}$ (H)	$H_{R_{01}}$ (H)	$R_{02\max}$ (H)	$R_{02\min}$ (H)	$H_{R_{02}}$ (H)	$R_{A\max}$ (H)	$R_{A\min}$ (H)	H_{R_A} (H)
1	0	426	262	164	407	282	125	407	282	125
2	0.02	1021	76	945	1006	65	941	878	69	809
3	0.04	1963	227	1736	1948	228	1720	1374	415	959
4	0.06	3225	100	3125	3210	112	3098	1892	742	1150
5	0.08	4815	69	4746	4799	68	4731	2432	1049	1383
6	0.1	6814	68	6746	6820	38	6782	2992	1336	1656

	O_2A (м)	$R_{B\max}$ (H)	$R_{B\min}$ (H)	H_{R_B} (H)	$N_{4\max}$ (H)	$N_{4\min}$ (H)	H_{N_4} (H)	$M_{y\max}$ (Hm)	$M_{y\min}$ (Hm)	H_{M_Y} (Hm)
1	0	434	142	292	156	24	132	79	69	148
2	0.02	588	38	550	478	298	776	149	162	311
3	0.04	832	11	821	820	641	1461	325	342	667
4	0.06	1163	73	1090	1178	998	2176	559	580	1139
5	0.08	1522	25	1497	1552	1372	2924	851	875	1726
6	0.1	1905	113	1792	1945	1765	3710	1202	1230	2432

The degree of mobility mechanism $w=1$, wherein $O_2A=0.4м$, $\omega_H=10c^{-1}$ variation F_c

	F_c (H)	$R_{01\max}$ (H)	$R_{01\min}$ (H)	$H_{R_{01}}$ (H)	$R_{02\max}$ (H)	$R_{02\min}$ (H)	$H_{R_{02}}$ (H)	$R_{A\max}$ (H)	$R_{A\min}$ (H)	H_{R_A} (H)

1	0	470	64	406	454	73	381	302	133	169
2	200	1844	247	1597	1829	248	1581	1257	573	684
3	400	1923	237	1686	1908	238	1670	1335	515	820
4	600	2003	216	1787	1987	217	1770	1414	315	1099
5	800	2176	102	2074	2181	97	2084	1496	116	1380
6	1000	2429	65	2364	2436	78	2358	1613	85	1528

	F_c (H)	$R_{B\max}$ (H)	$R_{B\min}$ (H)	H_{R_B} (H)	$N_{4\max}$ (H)	$N_{4\min}$ (H)	H_{N_4} (H)	$M_{y\max}$ (Hm)	$M_{y\min}$ (Hm)	H_{M_Y} (Hm)
1	0	211	52	159	260	-81	341	69	-68	137
2	200	723	118	605	759	579	1148	265	283	548
3	400	791	80	711	800	620	1420	305	322	627
4	600	912	52	860	841	661	1502	346	362	708
5	800	1084	195	889	882	702	1584	387	403	790
6	1000	1266	206	1060	924	744	1668	429	445	874

It was found that with increasing distance from the axis of the pinion “O_2” to the hinge "A", the swing range: the reaction in the support of the carrier increases from 164 N to 6746 N; on the pinion shaft increases from 1393 N to 33879 N; the magnitude of the swing between the slider and the guide increases from 764 N to 19100 N.

The variation F_c revealed that changes in the technological resistance forces F_c from 0 to 1000 N lead to an increase in the range of oscillations in the support carrier from 406 N to 2364 N, on the pinion shaft from 381 N to 2358 N; the magnitude of the swing range between the slider and the guide increases from 341 N to 1668 N

4. Conclusions

1. Analytical formulas have been made to determine the kinematic parameters of a planetary-lever mechanism with one and two degrees of mobility;

2. It is established that for a mechanism with two degrees of mobility, the angular velocity of the central wheel influences the analytical formulas of only the pinion, the analytical formulas for the connecting rod and the slider are identical for the analytical formulas of the planetary-lever mechanism with the same degree of mobility;

3. The realization of the obtained formulas on a computer made it possible to determine the kinematic parameters of planetary-lever mechanisms with one and two degrees of mobility. According to the results of calculations on the computer, regularities of changes of movements, velocities, and accelerations of the considered mechanisms were obtained.

4. Formulas for determination of reaction in kinematic pairs of planetary-lever mechanism with one degree of mobility are made. The formulas obtained were implemented on a computer program Math CAD 15.

5. The regularities of the change in the reaction in kinematic pairs, as well as the balancing moments in the carrier support, were obtained when the basic geometric parameters of the mechanism under consideration were varied. It is established that the greatest influence on the reactions in kinematic pairs has a distance from the axis of the pinion“O_2” to the hinge "A".

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