Modeling of kinematics and kinetostatics of planetary-lever mechanism

R Karimov2*, N Begimov2, Sh Sadullaev2, B Urinov1, K Turajonov1 and Sh Ibodulloev3

1 Tashkent Institute of Irrigation and Agricultural Mechanization Engineers, Department of hydraulic construction, Tashkent, Uzbekistan
2 Tashkent State Technic University named after Islam Karimov, Tashkent, Uzbekistan
3 National University of Uzbekistan, Tashkent, Uzbekistan

q.turajonov@gmail.com

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 backstages 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 axes 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.](image)

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, \( \omega_0 = 0 \), 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

\[
x_{\text{s}_1} = x_A + AS_3 \cos \beta, \quad x_{\text{s}_2} = R_H \cos \varphi_H - O_2A \cos(\varphi_2 + AS_3 \cos \beta)
\]

(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 3, \( \beta \) is the angle of connecting rod, \( R_H \) is the length of the carrier, \( \varphi_H \) 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.

\[
y_{\text{s}_1} = BS_3 \cdot \sin \beta, \quad y_{\text{s}_2} = BS_3 \sin(\arcsin((R_H \sin \varphi_H - O_2A \sin \varphi_2)/L_3))
\]

(2)

\[
x_B = x_A + L_3x, \quad L_3 = L_2 \cdot \cos \beta
\]

\[
x_B = R_H \cos \varphi_H - O_2A \cos \varphi_2 + L_2 \cos(\arcsin(y_A/L_2))
\]

(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

\[
v_{x_{\text{s}_1}} = O_2A \omega_2 \sin \varphi_2 - AS_3 \beta' \sin \beta - R_H \omega_H \sin \varphi_H
\]

(4)

\[
v_{y_{\text{s}_1}} = \frac{BS_3(2A \omega_2 \cos \varphi_2 - R_H \omega_H \cos \varphi_H)}{L_3}
\]

(5)

\[
a_{x_{\text{s}_1}} = O_2A \omega_2 \sin \varphi_2 - R_H \omega_H \sin \varphi_H - AS_3 \beta' \sin \beta + O_2A \omega_2 \cos \varphi_2
\]

(6)

\[-R_H \omega_H^2 \cos \varphi_H - AS_3(\beta')^2 \cos \beta\]
\[ a_{y_A} = \frac{BS_3 \left( O_2 A_2 \cos \phi_2 - R_H \xi_H \cos \phi_H - O_2 A_2 \sin \phi_2 + R_H \xi_H \sin \phi_H \right)}{L_3} \]  

(7)

\[ v_B = O_2 A_2 \sin \phi_2 - R_H \omega_H \sin \phi_H - \frac{y_A \left( \frac{y_A}{L_3} \right)^2}{L_3 \sqrt{1 - \left( \frac{y_A}{L_3} \right)^2}} \]  

(8)

\[ a_B = O_2 A_2 \cos \phi_2 - R_H \omega_H \cos \phi_H - \frac{\left( \frac{y_A}{L_3} \right)^2 \left( \frac{y_A}{L_3} \right)^2}{L_3 \sqrt{1 - \left( \frac{y_A}{L_3} \right)^2}} \]  

(9)

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

\[ v_{S_3} = \sqrt{v_{S_{x3}}^2 + v_{S_{y3}}^2} \]  

(10)

\[ a_{S_3} = \sqrt{a_{x_{S_3}}^2 + a_{y_{S_3}}^2} \]  

(11)

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

\[ \sin \beta = \frac{y_A}{L_3} \beta = \arcsin \left( \frac{y_A}{L_3} \right) L_{x_A} = L_3 \cdot \cos \beta \ L_{x_A} = L_3 \cdot \cos \left( \arcsin \left( \frac{y_A}{L_3} \right) \right) \]  

(12)

\[ \omega_x = \frac{v_A}{L_3 \sqrt{1 - \left( \frac{y_A}{L_3} \right)^2}} \]  

\[ a_x = \frac{\left( \frac{y_A}{L_3} \right)^2 \left( \frac{y_A}{L_3} \right)^2}{L_3 \sqrt{1 - \left( \frac{y_A}{L_3} \right)^2}} \]  

(13)

Analytical formulas (1-13) were implemented in Math CAD 15 computer program. According to the results of calculations, patterns of changes were obtained \( x_{S_3}, y_{S_3}, v_{X_{S_3}}, v_{Y_{S_3}}, v_{x_{S_3}}, v_{y_{S_3}}, a_{X_{S_3}}, a_{Y_{S_3}}, a_{x_{S_3}}, a_{y_{S_3}}, \omega_3, \dot{\epsilon}_3, x_B, v_B, a_B \) 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 25 \( \text{s}^{-1} \). The step of variation was 5 \( \text{s}^{-1} \).

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’Alembert. When using the D’Alembert 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 \( G_1 \) is the force of gravity of carrier, \( G_2 \) is the force of gravity of pinion, \( G_3 \) is the force of gravity of connecting rod, \( G_4 \) is the force of gravity of slider, \( F_C \) is drag forces acting on the slider, \( M_y \) is balancing moment on the carrier shaft, \( F_{u3} \) is inertia force acting on the center of mass of the connecting rod, \( M_{u3} \) is moment of inertia acting on the connecting rod, \( F_{u4} \) 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.

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

Known forces act on the slider: the force inertia of the slider \( F_{u4} \), the technological resistance \( F_{c4} \), and the force of gravity of the slider \( G_i \). The inertia force \( F_{u4} = \frac{G_i}{g} \) and direction along the axis AX, the technological resistance force \( F_{c4} \) acting on the slider is also directed along the axis AX, the force of gravity of the slider \( G_i \) is directed vertically down. An unknown reaction in the hinge also acts on the slider \( R_B \), which we represent as two components \( R_{BX} \) and \( R_{BY} \) 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 \( N_s \), 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.

\[
\begin{aligned}
\sum X &= 0 \quad F_{a4} + F_{c4} + R_{BX} = 0, \\
\sum Y &= 0 \quad R_{BY} + N_s - F_4 = 0 \\
\end{aligned}
\]

(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 \( \bar{F}_{a3} \), which is presented in the form of two components \( \bar{F}_{ax} \), \( \bar{F}_{ay} \), gravity force of connecting rod \( \bar{F}_{g} \), which is directed vertically down, the moment from the forces of inertia \( \bar{M}_{a3} \). Components of the force of inertia \( \bar{F}_{a3} \), acting in the center of mass of the connecting rod \( S_3 \) were determined with following formulas \( \bar{F}_{ax} = -\frac{J_{ga}}{g} a_{ax} \), \( \bar{F}_{ay} = -\frac{J_{ga}}{g} a_{ay} \), the moment from the inertia forces of the connecting rod is \( M_{a3} = -J_{g3}\eta_3 \), where \( J_{g3} \) - connecting rod inertia moment relative to the centre of mass, \( \eta_3 \) - 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 \( \bar{R}_{AX} \), \( \bar{R}_{AY} \). 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 \( S_3 \).

\[
\begin{aligned}
\sum X &= 0 \quad R_{AX} + F_{u3x} - R_{BX} = 0, \\
\sum Y &= 0 \quad R_{AY} + F_{u3y} - G_3 - R_{BY} = 0, \\
\sum M_{s3} &= 0 \quad -R_{BY} \cdot l_{BS3} \cdot \cos \beta - R_{AX} \cdot l_{BS3} \cdot \sin \beta + M_{s3} - R_{AX} \cdot l_{AS3} \cdot \sin \beta - R_{AY} \cdot l_{AS3} \cdot \cos \beta = 0 \\
\end{aligned}
\]

(15)

Pinion equilibrium conditions: the following reaction forces act on the pinion from the connecting rod side: \( \bar{R}_{AX} \), \( \bar{R}_{AY} \), as well as the reaction force acting on pinion 2 from the side of the fixed central wheel \( \bar{R}_{21} \) and reaction force in the support \( O_3 \) of pinion, which are present below \( \bar{R}_{O3,X} \), \( \bar{R}_{O3,Y} \). The magnitude of the total reaction on the shaft of the pinion \( R_{O2} \) determined with the following formula

\( R_{O2} = \sqrt{(R_{O2X})^2 + (R_{O2Y})^2} \).

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.

\[
\begin{aligned}
\sum X &= 0 \quad R_{12} \cdot \cos((3\pi/2) + \varphi_H) + R_{12} \cdot \cos \varphi_H + R_{O3,X} + R_{AX} = 0, \\
\sum Y &= 0 \quad R_{12} \cdot \sin((3\pi/2) + \varphi_H) + R_{12} \cdot \sin \varphi_H + R_{O3,Y} - R_{AY} = 0, \\
\sum M_{O2} &= 0 \quad R_{12} \cdot r^2 - R_{AY} \cdot (O_2A) \cdot \cos \varphi_2 + R_{AY} \cdot (O_2A) \cdot \sin \varphi_2 = 0 \\
\end{aligned}
\]

(16)
The calculated scheme of the carrier with the forces of acting on is shown in Fig.4. (b). Known gravity acts on the carrier \( \overline{G} \), directed vertically down, the inertia force drove \( \overline{F} \) applied at its center of mass \( S \), which is equal \( \overline{F} = \frac{G}{g} \cdot \omega _n ^2 \cdot l_{0_1 S} \). 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 “\( O_1 \)”.

\[
\begin{align*}
\Sigma X &= 0 \quad R_{O_1 X} - R_{O_1 X} + F_{in} \cos \varphi_n = 0 \\
\Sigma Y &= 0 \quad R_{O_1 Y} - G_n - R_{O_1 Y} + F_{in} \sin \varphi_n = 0 \\
\Sigma M_{O_1} &= 0 \quad -G_n \cdot l_{0_1 S} \cos \varphi_n + M_y - R_{O_1 Y} (O_1 O_2) \cos \varphi_n + R_{O_1 X} (O_1 O_2) \sin \varphi_n = 0
\end{align*}
\]

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

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 \( \omega_H \) and central wheel \( \omega_1 \). The output parameters were: \( R_1 = 0.15 m \); \( R_2 = 0.05 m \); \( R_H = 0.2 m \); \( L_{O,A} = 0.07 m \); \( L_3 = 0.81 m \).

| Table 1. Numerical values of the main kinematic parameters of the planetary-lever mechanism with one degree of mobility |
|-------------------------------------------------------------------------------------------------|
| \( \omega_H \) (\( \text{c}^{-1} \)) | \( V_{B_{\text{max}}} \) (\( \text{m/s} \)) | \( V_{B_{\text{min}}} \) (\( \text{m/s} \)) | \( V_{S_{3_{\text{max}}}} \) (\( \text{m/s} \)) | \( V_{S_{3_{\text{min}}}} \) (\( \text{m/s} \)) | \( H_{S_{3}} \) (\( \text{m/s} \)) |
|---|---|---|---|---|---|
| 1 | 5 | 2.5 | -2.5 | 5.0 | 28 | -37 | 65 | 2.4 | 0.2 | 2.2 |
The angular velocity of the central wheel is negative if its direction does not coincide with the direction of the angular velocity of the pinion. In this case, the angular velocity of the pinion increases 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 affects the angular velocity of the pinion.

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

| $\omega_H$ | $a_3^{\text{max}}$ | $a_3^{\text{min}}$ | $H_a$ | $\omega_3^{\text{max}}$ | $\omega_3^{\text{min}}$ | $H$ | $\omega_3$ | $\epsilon_3^{\text{max}}$ | $\epsilon_3^{\text{min}}$ | $H\epsilon_3^{(c-2)}$ |
|----------|-----------------|-----------------|------|----------------|----------------|-----|--------|----------------|----------------|----------------|
| (c-1)    | (m/s$^2$)       | (m/s$^2$)       | (ω/2)| (c-1)          | (c-1)          |      | (c-2)  | (c-2)         | (c-2)         | (c-2)          |
| 1 5      | 34.3            | 11.7            | 22.6 | -2.9           | 5              | 42.4 | -42    | 84.4           |                 |                |
| 2 10     | 137.2           | 46.8            | 90.4 | -6             | 11.3           | 169.5| -169.5| 339            |                 |                |
| 3 15     | 308.7           | 105.2           | 205.6| 7.9            | -8.9           | 16.8 | 381.3  | -381.3         | 762            |                |
| 4 20     | 549             | 187             | 362  | -11.9          | 22             | 678  | -678   | 1356           |                |                |
| 5 25     | 858             | 293             | 565  | 13.2           | -14.8          | 28   | 1059   | -1059          | 2118           |                |

In Tables 1,2: $v_B^{\text{max}}, a_B^{\text{max}}$ are maximum values of linear velocity and acceleration of the slider, $v_B^{\text{min}}, a_B^{\text{min}}$ are minimum linear velocity and slider acceleration, $H_v, H_a$ are the swing range of linear velocity and acceleration, $v_{3\text{max}}, a_{3\text{max}}$ are maximum linear velocity and acceleration of the center of mass of the connecting rod, $v_{3\text{min}}, a_{3\text{min}}$ are minimum values of linear velocity and acceleration of the center of mass of the connecting rod, $H_v, H_a$ are linear velocity swings and acceleration, $\omega_3^{\text{max}}, \omega_3^{\text{min}}$ are the maximum and minimum value of the connecting rod angular velocity, $H\omega_3$ is swing range of angular velocity, $\epsilon_3^{\text{max}}, \epsilon_3^{\text{min}}$ are the maximum and minimum value of the angular acceleration of the connecting rod, $H\epsilon_3$ 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 $v_3, a_3, \omega_3, \epsilon_3, v_B, a_B$ 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 $\omega_2 = -(R_1/R_2) \cdot \omega_1 + (1 + R_1/R_2) \cdot \omega_H$

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 $(R_1/R_2) \cdot \omega_1$ 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 \((-R_1/R_2) \cdot \omega_h\) 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 \(\omega_H\), 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

| \(\omega_H\) (c-1) | \(R_{01\text{max}}\) (H) | \(R_{01\text{min}}\) (H) | \(H_{R_{01}}\) (H) | \(R_{02\text{max}}\) (H) | \(R_{02\text{min}}\) (H) | \(H_{R_{02}}\) (H) | \(R_{A\text{max}}\) (H) | \(R_{A\text{min}}\) (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            |

| \(\omega_H\) (c-1) | \(R_{B\text{max}}\) (H) | \(R_{B\text{min}}\) (H) | \(H_{R_{B}}\) (H) | \(N_{4\text{max}}\) (H) | \(N_{4\text{min}}\) (H) | \(H_{N_{4}}\) (H) | \(M_{y\text{max}}\) (Hm) | \(M_{y\text{min}}\) (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 \(O_A=0.04\), \(F_c=200H\) variation \(\omega_H\)

| \(O_A\) (m) | \(R_{01\text{max}}\) (H) | \(R_{01\text{min}}\) (H) | \(H_{R_{01}}\) (H) | \(R_{02\text{max}}\) (H) | \(R_{02\text{min}}\) (H) | \(H_{R_{02}}\) (H) | \(R_{A\text{max}}\) (H) | \(R_{A\text{min}}\) (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            |

The degree of mobility mechanism \(w=1\), wherein \(O_A=0.4\), \(\omega_H=10c^{-1}\) variation \(F_c\)

| \(F_c\) (H) | \(R_{01\text{max}}\) (H) | \(R_{01\text{min}}\) (H) | \(H_{R_{01}}\) (H) | \(R_{02\text{max}}\) (H) | \(R_{02\text{min}}\) (H) | \(H_{R_{02}}\) (H) | \(R_{A\text{max}}\) (H) | \(R_{A\text{min}}\) (H) | \(H_{R_{A}}\) (H) |
|-------------|------------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|-----------------|
| 1           | 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            |
It was found that with increasing distance from the axis of the pinion “O₂” 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 381 N to 2358 N; the magnitude of the swing between the slider and the guide increases from 1393 N to 33879 N; the magnitude of the swing range 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 MathCAD 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 “O₂" to the hinge "A".

References

[1] Semen A G Timofeev A M Loktionov A V 2008 Study of the mechanism of intermittent rotational motion with external gearing of gears. *Bulletin of the Belarusian-Russian University* 4 (21) pp 126-131

[2] Fedoseev G N Semin A G Korneyenko D V 2015 The study of the kinematic capabilities of the gear-lever mechanism with quasi-stops. *Bulletin of Herald State Technological University* 28 pp 111-115

[3] Shagiaxkhetov A I 2007 Dynamics of gear-link mechanisms based on elliptical wheels. *Bulletin of SUSU* 11 pp 42-47
[4] Abdukarimov A 2016 Graphographic analysis of the gear-lever differential gear with a parallelogram lever circuit Bulletin of the Kyrgyz State Technical University after the name I. Razzakova 1 pp 28-40

[5] Sinenko E G Konischeva O V 2013 Kinematics and geometry of misaligned epi-cyclic mechanisms Journal of the Irkutsk State University of Communication Modern Technologies. System analysis 4 (40) pp 39-43

[6] Komarov A V 2013 Kinematics of a rotary piston engine based on a gear-lever mechanism. Bulletin of the Pskov State University pp 159-162

[7] Grudinin V G 2016 Dynamic properties of planetary couplers with unbalanced satellites Bulletin of ISTU 3 pp 45-59

[8] Shagiakhmetov A I 2007 Structural synthesis and analysis of gear-link mecha-nisms of periodic movement with non-circular gears Bulletin of SUSU 25 pp 23-29

[9] Bat M I et al. 1990 Theoretical mechanics in examples and problems (Moscow Nauka)

[10] Bat M I et al. 1991 Theoretical mechanics in examples and problems (Moscow Nauka)

[11] Artobolevskiy I I 1988 Theory of mechanisms and machines (Moscow Nauka)

[12] Frolov K V Popov S A Musatov A K and et. 1987 Theory of mechanisms and machines (Moscow graduate School)

[13] Levitskiy N I 1990 Theory of mechanisms and machines (Moscow Nauka)

[14] Karimov R I Begimov N N 2015 Modeling of kinematics and kinetostatics of cycle mechanisms (Tashkent TashSTU)

[15] Karimov R I Karimov K A Begimov N N 2019 Theory of mechanisms and machines Training manual (Tashkent Sano-standart)

[16] Koltunov M A Mirsaidov M Troyanovskii I E 1978 Transient vibrations of axisymmetric viscoelastic shells Polym. Mech. https://doi.org/10.1007/BF00857468.

[17] Karpichev V A Bеспалко S V 2019 The method of approximate Esmiation of the Dynamic Load of the Train in the Braking Process https://doi.org/DOI:10.1109/FarEastCon.2019.8933873.

[18] Mirsaidov M M Abdikarimov R A Vatin N I Zhgutov V M Khodzhaev D A Normuminov B A 2018 Nonlinear parametric oscillations of viscoelastic plate of variable thickness. Mag. Civ. Eng https://doi.org/10.18720/MCE.82.11.

[19] Khodzhaev D A Abdikarimov R A Mirsaidov M M 2019 Dynamics of a physi-cally nonlinear viscoelastic cylindrical shell with a concentrated mass Magazine of Civil Engineering 91(7) pp 39-48 DOI:10.18720/MCE.91.4

[20] Teshaev M Kh Safarov I I Mirsaidov M M 2019 Oscillations of multilayer viscoelastic composite toroidal pipes J. of the Serbian Society for Computational Mechanics 13(2) pp 105-116. DOI:10.24874/jsscm.2019.13.02.084