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SYNTHESIS OF ADJUSTABLE MECHANISM OF SEWING MACHINE AND ITS KINEMATIC ANALYSIS USING SOLIDWORKS

The paper is dedicated to the task of optimal design of eight-linked linkage mechanism of the chain stitch sewing machine, for which it is necessary to solve the problem of the adjustable stroke of the needle. The algorithm and corresponding formulas for conducting a kinematic study of the specified mechanism are presented in the article that allows conducting the optimization design using different criteria. The results of calculations using kinematic research formulas are already given, that made it possible to determine the optimal limits of the needle stroke. The conducted research also took into account the need for optimal values of pressure angles in the links of structural groups of the mechanism, which ensures that they do not jam during the operation of the mechanism, with optimal values of the efficiency, without the loss of its functionality.

To verify the correctness of the results obtained by the analytical method, a computer model of the sewing machine mechanism was created in the computer modeling software SOLIDWORKS and its main kinematic characteristics were determined using the Computer-Aided Engineering system SOLIDWORKS Motion. It enabled to determine the values of displacements of the links of the mechanism, its velocities and accelerations, besides it proves the working capacity of the mechanism.

Key words: kinematic synthesis, analysis, computer modeling, stroke adjustment, pressure angles

ХАРЖЕВСЬКИЙ В'ЯЧЕСЛАВ, МАРЧЕНКО МАКСИМ, ТКАЧУК ВІТАЛІЙ
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СИНТЕЗ РЕГУЛЬОВАНОГО МЕХАНІЗМУ ШВЕЙНОЇ МАШИНИ ТА ЙОГО КІНЕМАТИЧНИЙ АНАЛІЗ ЗА ДОПОМОГОЮ SOLIDWORKS

В роботі розглядаються питання оптимального проектування важільного восьмиланкового шарнірно-важільного механізму швейної машини ланцюгового стібка, для якого ставиться задача забезпечення регульованого ходу голки. Наведено алгоритм та відповідні формули для проведення кінематичного дослідження зазначеного механізму, що дозволяє провести його оптимізаційне проектування. Результати проведеного кінематичного дослідження дозволили визначити оптимальні межі ходу голки. В проведених дослідженнях враховано також необхідність оптимальних значень кутів тиску у ланках структурних груп механізму, що забезпечить їх незаклинювання. Проведено комп'ютерне моделювання механізму швейної машини у системі SOLIDWORKS та визначення його основних кінематичних характеристик за допомогою системи комп'ютерного моделювання SOLIDWORKS Motion.

Ключові слова: кінематичний синтез, аналіз, комп'ютерне моделювання, регулювання ходу, кути тиску

Problem statement

The optimal design of mechanisms of sewing machines is an important practical task, because by means of kinematic synthesis methods it is possible to obtain various kinematic characteristics of their points and links, in particular, the laws of motion of the output link – the needle guide, including the values of displacements, velocities and accelerations. As it is known [1], the needle mechanism of a sewing machine performs the following functions: piercing and passing threads through the material, forming a loop and tightening the stitch. Moreover, depending on the purpose of the sewing machine, the needle can perform a simple movement (straight-line or curvilinear), as well as a complex planar or spatial movement. For example, the 876 class sewing machine uses an eight-link linkage mechanism, the structural scheme of which is shown in the Fig. 2. According to the Assur classification, this linkage mechanism consists of a ground link to which crank 1 (mechanism of class I) is connected in series, besides – two groups of the 2nd class of the 1st type (links 2-3 and 4-5, respectively) and one group of 2nd class of the 2nd type (links 6-7).

The needle installed on the needle guide 7 makes a reciprocating movement, the maximum stroke of which is 36 mm. Practically all types of mechanisms of such machines have an unregulated movement of the needle, including the 876 class machine. This mechanism can be upgraded if the stroke of the needle could be adjustable, which can be done by changing the position of the fixed hinge *F* (see Fig. 2). For normal stitch formation conditions, the lower position of the needle should remain unchanged (Fig. 2, b).

Aim of the work: designing of the mechanism of the needle guide of the sewing machine in order to regulate the movement of the needle within the specified limits, as well as its research in order to determine its main kinematic characteristics and working capacity.

Analysis of the literature

The problem of optimal kinematic synthesis of linkage mechanisms is one of the most difficult problems in the theory of mechanisms and machines. There are a number of tasks for the kinematic synthesis that require designing such mechanisms according to various criteria, in particular, according to the given law of motion of the output link or the by the maximum value of the output link displacement [1, 6]. After conducting the kinematic synthesis, it is also necessary to carry out kinematic analysis of the designed mechanisms and perform an optimization procedure, using all the possible variants, according to various criteria, in order to choose the mechanism that meets all the requirements of the designer and the technological requirements of the machine. For this purpose, the synthesis and analysis methods that are described in the papers [2] can be used.

As it is known, there are several methods to study the kinematics parameters of linkage mechanisms. As shown in the paper[1], the method of closed vector contours is the most convenient for calculating the kinematics of planar linkage mechanisms. Based on the method of closed vector contours, a method of groups for studying the kinematics of mechanisms has been developed, which consists of the following: the mechanism is divided into Assur groups, and each structural group is represented in the form of a closed vector contour. For each contour, vector equations of closure are drawn up separately, projected onto the coordinate axes and projection equations are obtained, by which the positions of the links are found, then analogs of velocities and accelerations are determined by differentiating the projection equations. During the kinematic study of mechanisms, the analogues of the kinematic parameters are determined, instead of their actual values, because it allows comparing the laws of motion of different mechanisms. The basics of analytical research of linkage mechanisms are outlined in the fundamental course of the Theory of Mechanisms and Machines [1], as well as in specialized literature [1-7].

The usage of the Computer-Aided Design software SOLIDWORKS for modeling of linkage mechanisms, as well as the usage of the Computer-Aided Engineering system SOLIDWORKS Motion for kinematic study of such mechanisms, is considered in the work [4].

The kinematic study of the adjustable mechanism

Let's consider the sewing machine mechanism, the structural scheme of which is shown in the Fig. 2. The dimensions of the links of the mechanism are as follows: $l_{OA}=15$ mm, $l_{AB}=175$ mm, $l_{BC}=25$ mm, $l_{CD}=17$ mm, $l_{DE}=30$ mm, $l_{EF}=25$ mm, $l_{EG}=30$ mm, $x_{OF}=18$ mm, $b=20$ mm. Let's determine the value of the interaxial distance l_{CF} from the condition that in the lowest position of the needle guide 7, the rocker arm takes a horizontal position $(\varphi_5\approx 0)$. To do this, let's consider a four-bar linkage (Fig. 1). After carrying out simple transformations, we get the following: $l_{CC'}=l_{CD}\cos\varphi'_{30}; l_{DD'}=l_{DE}\cos\varphi_{40}$,

where $\varphi'_{30}=\varphi_{30}-\dfrac{\pi}{2};\varphi_{40}=\arcsin\left(\dfrac{l_{EF}-l_{CD}\sin\varphi'_{30}}{l_{DE}}\right),\varphi_{30}-$ angle that determines the position of the link 3 in the lowest position of the needle guide *7*. From the Fig. 1, it is clear that the length of *CF* can be determined as follows: $l_{CF}=l_{CC'}+l_{DD'}=37{,}945$ mm.

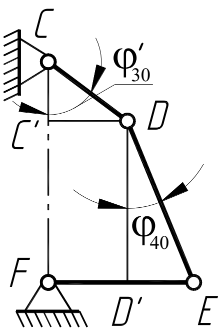
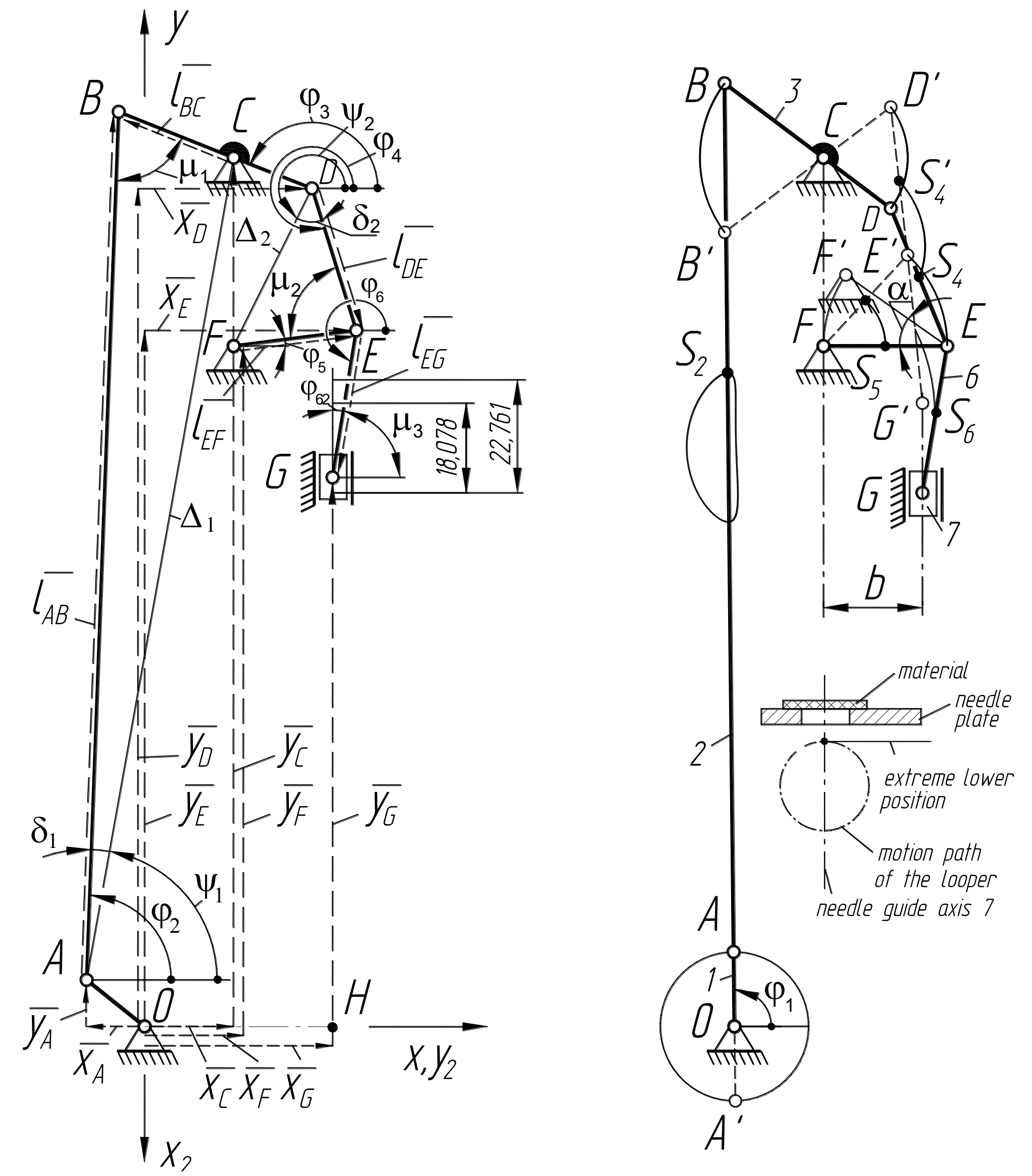


Fig. 1. – To determine the value l_{EF}

The mechanism that is described in the Fig. 2 is a 2nd class II mechanism, which consists of the mechanism of the 1st class and three attached structural groups. To carry out a kinematic study of the mechanism, it is necessary to make up the vector equations, separately for each structural group (Fig. 2, a):

$$\begin{aligned}&\text{for the group 2-3:}\quad \vec{x}_A+\vec{y}_A+\vec{l}_{AB}=\vec{x}_C+\vec{y}_C+\vec{l}_{BC};\\&\text{for the group 4-5:}\quad \vec{x}_D+\vec{y}_D+\vec{l}_{DE}=\vec{x}_F+\vec{y}_F+\vec{l}_{EF};\\&\text{for the group 6-7:}\quad \vec{x}_{E_2}+\vec{y}_{E_2}+\vec{l}_{EG}=\vec{x}_{G_2}+\vec{y}_{G_2}.\end{aligned}\qquad(1)$$

The kinematic study of the mechanism starts with the analysis of the 1st class mechanism. The mechanism of the 1st class is formed by the initial link (crank) and the ground link, that form a rotating kinematic pair; during rotational movement, the position of the crank is determined by the angle φ_1, which is called the generalized coordinate.



a *b*

Fig. 2 – Kinematic scheme of the mechanism: a) general calculation scheme; b) mechanism in extreme positions with trajectories of movement of individual points

Let's take the center of the fixed hinge *O* as the origin of the coordinate system, then the coordinates of the point *A* of the crank *OA* in the *xOy* coordinate system can be calculated as follows:

$$x_A = l_{OA}\cos\varphi_1;\ y_A = l_{OA}\sin\varphi_1. \tag{2}$$

The value of the angle of crank rotation φ_1 varies within the limits $\varphi_1 = \varphi_0,...\varphi_0 + 2\pi,$ where φ_0 – the value of the angle φ_1 in the lowest position of the output link 7 ($\varphi_0 = 90{,}6^\circ$).

Let's consider structural group 2-3 of the mechanism (2nd class, 1st type). The structural group is formed by the links 2 and 3, that are connected to the main mechanism by rotating kinematic pairs *A* and *C*. To determine the position of the links of the structural group in the *xOy* coordinate system, it is necessary to determine the angles φ_2 and φ_3 :

$$\varphi_2 = \psi_1 + \delta_1;\ \varphi_3 = \psi_1 + \delta_1 + \mu_1. \tag{3}$$

Let's determine the value and position of the auxiliary line Δ_1 and the angle $\psi_1,$ that determines its position, and other values included in (3):

$$\Delta_1 = \sqrt{\left(x_C - x_A\right)^2 + \left(y_C - y_A\right)^2};\ \psi_1 = \mathrm{arctg}\left(\left[y_C - y_A\right]/\left[x_C - x_A\right]\right);$$
$$\mu_1 = \arccos\left(\frac{l_{AB}^2 + l_{BC}^2 - \Delta_1^2}{2 l_{AB} l_{BC}}\right);\ \delta_1 = \arccos\left(\frac{l_{AB}^2 + \Delta_1^2 - l_{BC}^2}{2 l_{AB}\Delta_1}\right). \tag{4}$$

It should be noted that the angle ψ_1 must be considered as such that it can take values from 0 to 2π, which means that when performing calculations, it should be used not the usual function $\mathrm{arctg}(x)$, but the function $\arctan 2(x, y)$, that determines the position of the angle in any quarter. The function $\arctan 2(x, y)$ is presented in almost all modern programming languages.

The points B, S_2 and D can be considered as ones that belong to the links 2 and 3 respectively, and the position of those points can be determined by the angles φ_2 and φ_3, so the coordinates of mentioned points can be found as follows:

$$x_B = x_A + l_{AB}\cos\varphi_2; x_D = x_C + l_{CD}\cos\varphi_3; x_{S_2} = x_A + l_{AS_2}\cos\varphi_2;$$
$$y_B = y_A + l_{AB}\sin\varphi_2; y_D = y_C + l_{CD}\sin\varphi_3; y_{S_2} = y_A + l_{AS_2}\sin\varphi_2. \quad (5)$$

Consider the next structural group that is connected to the mechanism. The position of the links DE and EF of the structural group 4-5 that is connected to the main mechanism by the rotating kinematic pairs D and F, is determined by the following angles, respectively:

$$\varphi_4 = \psi_2 + \delta_2; \; \varphi_5 = \psi_2 + \delta_2 + \mu_2. \quad (6)$$

Similarly to the previous structural group, let's determine the values that are necessary to calculate the angular displacements of links 4 and 5:

$$\Delta_2 = \sqrt{\left(x_F - x_D\right)^2 + \left(y_F - y_D\right)^2}; \; \psi_2 = \mathrm{arctg}\left(\frac{y_F - y_D}{x_F - x_D}\right);$$
$$\mu_2 = \arccos\left(\frac{l_{DE}^2 + l_{EF}^2 - \Delta_2^2}{2 l_{DE} l_{EF}}\right); \; \delta_2 = \arccos\left(\frac{l_{DE}^2 + \Delta_2^2 - l_{EF}^2}{2 l_{DE} \Delta_2}\right), \quad (7)$$

and the coordinates of the fixed hinge F can be determined as follows:

$$x_F = x_C + l_{EF} - l_{EF}\cos\alpha; y_F = y_C - y_{CF} + l_{EF}\sin\alpha. \quad (8)$$

To determine the coordinates of the points S_4, S_5 and E, the following equations can be used:

$$x_E = x_F + l_{EF}\cos\varphi_5; x_{S_4} = x_E + \cos\varphi_5 \cdot l_{DE}/2; x_{S_5} = x_F + \cos\varphi_5 \cdot l_{EF}/2;$$
$$y_E = y_F + l_{EF}\sin\varphi_5; y_{S_4} = y_E + \sin\varphi_5 \cdot l_{DE}/2; y_{S_5} = y_F + \sin\varphi_5 \cdot l_{EF}/2. \quad (9)$$

To calculate the structural group 6-7 that is connected to the main mechanism by the kinematic pair E, it is convenient to rotate the main coordinate system xOy by an angle $\xi = 270^\circ$ and calculate the kinematic parameters in the coordinate system $x_2 O y_2$. Here and further, the index "2" means the corresponding value in the rotated coordinate system. To determine the position of the guide of the slider G, we add a fixed point H, the position of which in the xOy coordinate system can be determined as follows (Fig. 2a):

$$x_H = x_C + b; y_H = 0. \quad (10)$$

The position of points E and H in the rotated coordinate system, the value of the angle φ_6, that determines the position of the connecting rod 6 and the transmission angle μ_3:

$$x_{E_2} = -y_E; y_{E_2} = x_E; x_{H_2} = 0; y_{H_2} = x_H; \varphi_{62} = \arcsin\left(\frac{y_{H_2} - y_{E_2}}{l_{EG}}\right), \; \varphi_6 = \varphi_{62} + \xi; \; \mu_3 = \frac{\pi}{2} - \left|\varphi_{62}\right|. \quad (11)$$

Coordinates of the center of gravity S_6, of the link 6 can be also calculated:

$$x_{S_6} = x_E + l_{EG}\cos\varphi_6/2; y_{S_6} = y_E + l_{EG}\sin\varphi_6/2; \quad (12)$$

The coordinates of the slider 7, in the rotated and basic coordinate systems respectively, are determined by the following dependencies:

$$x_{G_2} = x_{E_2} + l_{EG}\cos\varphi_{62}; y_{G_2} = y_{E_2} + l_{EG}\sin\varphi_{62};$$
$$x_G = x_{G_2}\cos\left(2\pi - \xi\right) + y_{G_2}\sin\left(2\pi - \xi\right);$$
$$y_G = y_{G_2}\cos\left(2\pi - \xi\right) - y_{G_2}\sin\left(2\pi - \xi\right). \quad (13)$$

During the research process, it was established that the mechanism has two extreme positions that occupies in the positions of the crank that are defined by the angles $\varphi_1 = \varphi_0$ and $\varphi_1 = \varphi_0'$. Then, it is obvious that the stroke of the slider 7 can be determined as follows:

$$S = y_G\left(\varphi_0'\right) - y_G\left(\varphi_0\right). \quad (14)$$

After projecting equation (1) on the coordinate's axes, and differentiating the obtained expressions by the generalized coordinate, and performing some transformations, we can obtain formulas for calculating analogues of velocities and accelerations of the points and links of the mechanism. These transformations were carried out in the Mathcad system and are not presented here.

The results of the performed calculations are presented below. As can be seen from the diagrams in the Fig. 3, with an increase of the value of the angle α, that determines the position of the fixed hinge F, the stroke of the slider 7 increases, and the lower position of the slider remains unchanged. Another dependence is observed – with the increase in the stroke SG of the output link, the analogs of velocities and accelerations also increase.

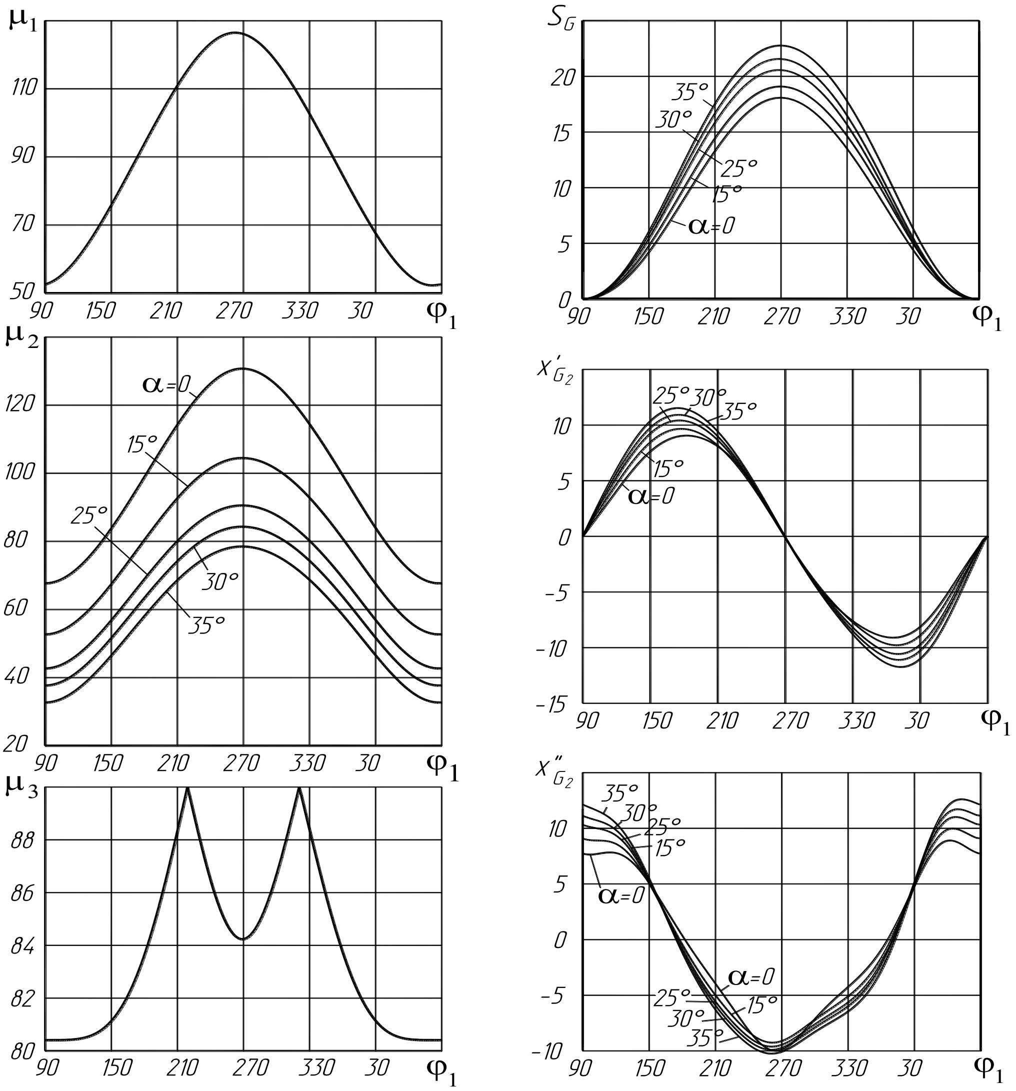


Fig. 3 – Results of the kinematic analysis

One of the conditions the efficiency of the mechanism are the values of the transmission angles μ_1, μ_2, μ_3, the values of which must be within the following limits:

$$30^\circ < \mu_i < 150^\circ. \quad (15)$$

As the conducted kinematic studies have shown, this condition for all the transmission angles is fulfilled for all designed linkage mechanisms.

In order to check the correctness of the obtained analytical formulas and the operability of the proposed mechanisms with adjustable stroke of the needle, a computer simulation of the sewing machine needle guide mechanism was carried out in the Computer-Aided Design system SOLIDWORKS, and the kinematic characteristics of the mechanisms were determined using Computer-Aided Engineering system SOLIDWORKS Motion. In particular, it was determined the law of motion of the output link, as well as the movement and acceleration of all points and links of the mechanism. The developed model and examples of the obtained results of the kinematic study are shown in the Fig. 4.

Conclusion

As it can be seen from the obtained calculation results and provided diagrams, the increase of the stroke of the output link of the mechanism leads to a decrease of the minimum value of the transmission angle μ_2. So, it can be stated about an expediency of adjusting of the stroke of the slider 7 only within limits such limits as: $0 < \alpha < 35^\circ$. Further increase of the angle α will lead to jamming of the mechanism links. As a result of the study, it was established that the stroke of the needle guide 7 can be adjusted in the range from 18 to 23 mm. The conducted computer modeling and corresponding kinematic analysis in SOLIDWORKS Motion confirmed the correctness of the analytical calculations and the operability of the proposed mechanisms.

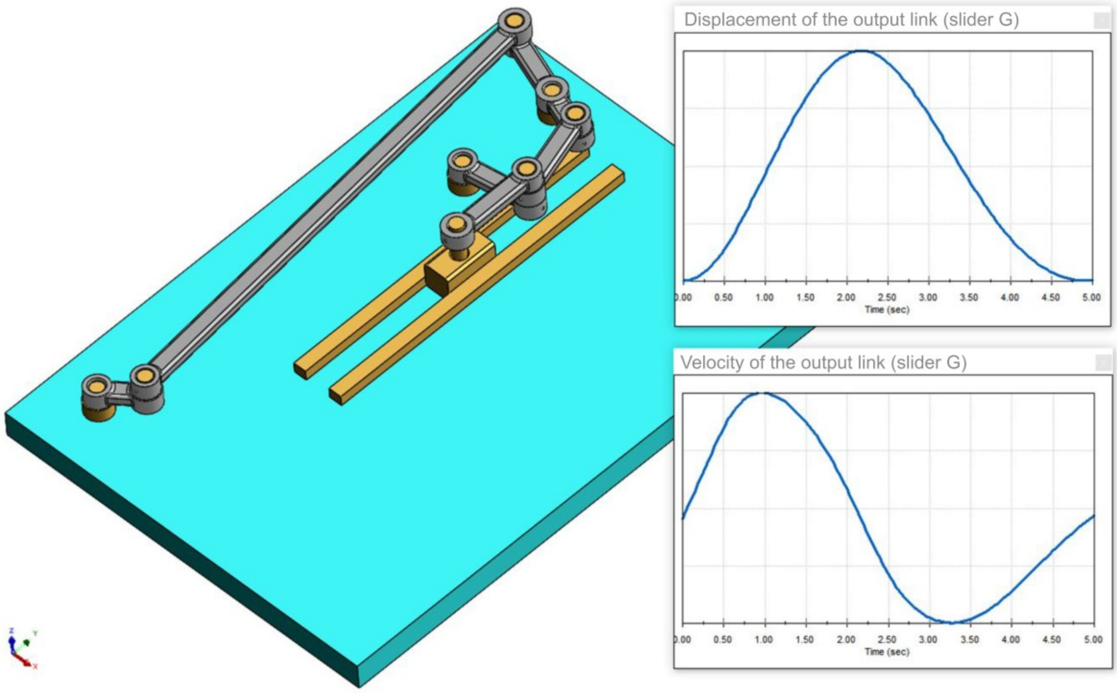


Fig. 4 – Mechanism and results of calculation in SOLIDWORKS Motion: displacement and velocity of the output link

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