Obrabotka metallov (tekhnologiya, oborudovanie, instrumenty) = Metal Working and Material Science. 2024 vol. 26 no. 1 pp. 80–98
ISSN: 1994-6309 (print) / 2541-819X (online)
DOI: 10.17212/1994-6309-2024-26.1-80-98

NSTU NETI

Obrabotka metallov - Metal Working and Material Science

Journal homepage: http://journals.nstu.ru/obrabotka_metallov

Synthesis of the heddle drive mechanism

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ARTICLE INFO

Article history:
Received: 12 December 2023
Revised: 08 January 2024
Accepted: 17 February 2024
Available online: 15 March 2024

Keywords:
Loom
Heddle motion mechanism
Warp threads
Kinematic chain
Structural synthesis of mechanism
Assur groups
Kinematic scheme
Cam mechanism
Cam radius
Roller
Speeds
Accelerations
Motions
Axes

Funding
This study was supported by a NSTU grant (project No. TP-PTM-1_24).

Acknowledgements
Research were conducted at core facility "Structure, mechanical and physical properties of materials".

ABSTRACT

Introduction. Domestic enterprises in various industries use a variety of process equipment, including weaving machines. Modern weaving machines have several unique features, including a close relationship between technical condition, productivity, and product quality. Weaving machines are widely used in the textile industry in Russia and other countries. To produce cotton, silk, wool, linen, and other types of fabrics, appropriate machines are designed, including shuttle, shuttleless, pneumatic, and hydraulic machines. One of the most crucial parts of the machine is the heddle lifting mechanism, which determines the weave pattern and the quality of the fabric produced. **The purpose of the work** is to reduce the dimensions of the loom by changing the design parameters of the heddle lifting mechanism. **The research methods** are based on the theory of machines and mechanisms. They enable the development of a method for synthesizing the heddle lifting mechanism and designing a device with reduced dimensions. The paper presents the synthesis and analysis of the Assur group algorithm, which can determine the kinematic characteristics of the mechanism. **Results and discussion.** Following the proposed methodology, the mechanism design was modified by removing the fixing device from the lever mechanism operating area. This allowed for a reduction in interaxial distances and a change in the kinematic scheme. As a result of the new position of the fixed axes, some levers, the connecting rod, and the angle of the double-arm lever were also altered. The synthesis of the mechanism is proposed to begin with the last Assur group, setting it a specific value for the G-point motion equal to 75 mm. (motion of the fourth heddle shaft). As a limitation, the equality of arcs (chords) $E'E = F'F$ was accepted. By assigning these values to the input element for the second-class first-type Assur group and bearing in mind the accepted conditions, the motions for point D were obtained. Thus, the value of the swing angle β of the roller shaft equal to 22.46° was obtained, which is 27.44 mm along the chord. Applying the interpolation principle, we found the initial motion value of 28 mm. Since the loom is planned to produce interlacing fabric patterns using 10 heddles, the design provides for a variable parameter that allows changing the motion of the heddles depending on their location in the depth of the machine. This role was assigned to the lever $B0_3D$. A cam pair synthesis was performed after determining the maximum and minimum values of the center of the roller motion. In total, 5 types of laws of motion were considered: straight-line, harmonic, double harmonic, power-law, cycloidal ones. For the center of the roller, the cycloidal law of motion was selected since it better corresponds to the specified conditions. The synthesis's accuracy was confirmed by the constructed cam profile and conducted kinematic studies for the Assur groups.

For citation: Podgornyj Y.I., Skeeba V.Y., Martynova T.G., Lobanov D.V., Martyushev N.V., Papko S.S., Rozhnov E.E., Yulusov I.S. Synthesis of the heddle drive mechanism. *Obrabotka metallov (tekhnologiya, oborudovanie, instrumenty) = Metal Working and Material Science*, 2024, vol. 26, no. 1, pp. 80–98. DOI: 10.17212/1994-6309-2024-26.1-80-98. (In Russian).

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Introduction

Domestic enterprises in various industries use a variety of process equipment, including weaving machines. Modern weaving machines have several unique features, including a close relationship between technical condition, productivity, and product quality. An essential feature of the process equipment is also the high kinematic complexity of the main mechanisms' movement and the dynamic intensity of the machines' operating modes [1–5]. One of the trends in the development of modern mechanical engineering is focused on improving existing and creating new high-performance equipment for weaving production. The increase in dynamic tension combined with that of operating speeds places higher requirements on the design of individual elements and assemblies, including drives that ensure intermittent movements of the machine's working bodies [1, 6–9]. Currently, the production of a mass assortment of fabrics for consumer needs, including strong ones, is carried out mainly on shuttleless looms [2–18]. Shuttleless looms offer several advantages, including small dimensions, high performance, and automated fabric production processes. They are used to manufacture cotton, silk, wool, linen, technical, and other types of fabrics [3, 4, 9, 19]. One of the most important requirements for modern machines is that the followers are required to perform movements that accurately correspond to a specific law. This requirement is sometimes not feasible if simple part connections, such as levers, are being used. Therefore, shuttleless looms use cam links with various contour surfaces obtained using mathematical dependencies in their mechanisms. Compared with other transmission mechanisms, they have a number of advantages. The cam can be shaped to meet the kinematic and dynamic requirements of the developer. This allows for easy adaptation. The design of a cam is simple, allowing for precise execution of the required follower motion [1, 4, 10, 12, 19–25]. However, fabric formation on such machines can present several challenges, including increased vibrations and accelerated wear of mechanisms. These factors reduce the performance and quality of the fabric. In this regard, when designing machine mechanisms, it is important to consider dynamic characteristics, which depend on the smoothness and continuity in the graphs of the followers' kinematic characteristics [10–12, 19–43]. The industrial use of shuttleless looms indicates that it is not possible to increase performance without considerable changes in definite mechanisms. First of all, it is necessary to modernize the mechanisms directly involved in the formation of fabrics. These include a mechanism designed to move the warp threads, i.e., a heddle lifting mechanism. The process of fabric formation on shuttleless looms is similar to that on shuttle looms: shed opening, picking of the weft thread, shed closing, battening of the weft thread to the cloth fell, and then the cycle repeats [40]. In the process of weaving, the warp threads bend around the weft threads and move from one side of the fabric to the other. Each main overlap on one side of the fabric corresponds to a weft overlap on the other. The pattern is created by various interlacing. This function is performed by a heddle lifting mechanism [40, 43]. Significantly, there are a large number of shuttleless looms in the factories of the Russian Federation. Even a small reduction in the size of a machine can allow for more equipment to be placed in the factory, resulting in a significant increase in performance per unit of production area. Consequently, reducing the dimensions of the shuttleless loom by reducing the size of the heddle lifting mechanism is an urgent and practical task.

The purpose of the work is to reduce the dimensions of the loom by changing the design parameters of the heddle lifting mechanism.

To achieve this goal, the following tasks were solved:

– to analyze the possibility of changing the size of the kinematic scheme of the mechanism;

– to develop a synthesis technique for the lever mechanism;

– to select the necessary parameters for the synthesis of the cam pair and perform the synthesis;

– to present the methodology of kinematic analysis and establish criteria for objectively evaluating the proposed solution.

Research Methodology

Consider the constructive scheme of the mechanism of the remission motion as shown in Fig. 1. It includes drive cams (*7*), a shaft with rollers (*6*), a connecting link (*10*), an eccentric mechanism (*11*), a double-arm lever (*1*), and a horizontal rod (*9*). As can be seen from the diagram, an eccentric

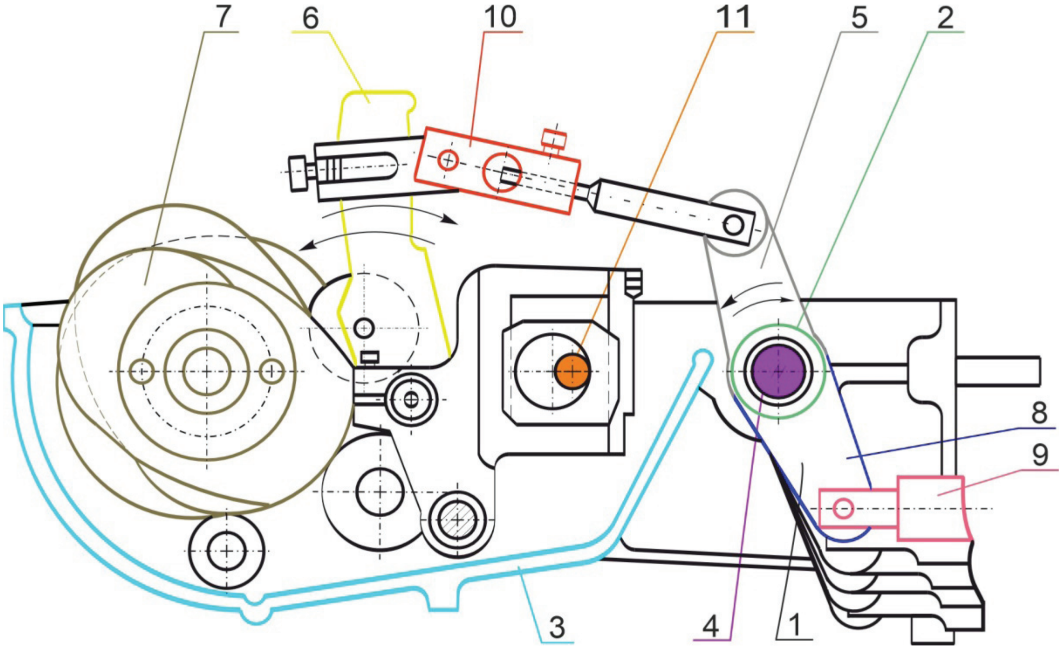


Fig. 1. The design scheme of the heddle motion consists of several components, including a two-arm lever (*1*), a hub (*2*), a body (*3*), a shaft (*4*), a top arm (*5*), a roller lever (*6*), eccentric drive (*7*), a bottom arm (*8*), a horizontal rod (*9*), a connecting rod (*10*), eccentric mechanism (*11*)

mechanism (*11*) is located inside the lever system. The purpose of this mechanism is to bring the system of levers and rods to a certain position, which contributes to their setting parameters, when installing a set of cams and heddle frames [44].

Relocating the eccentric mechanism (*11*) from the lever system to the body side is suggested. In this case, it will be possible to change the positions of the fixed axes and reduce the distances between the axes of the double-arm levers and the roller shaft. Due to the change in these positions, the dimensions of the levers and rods will change, which will require a new synthesis of the lever system.

The reduction in the dimensions of the mechanism is due to the removal of the mechanism for fixing the position of the heddle (eccentric mechanism) from the area of the lever system. This solution allowed for the reduction of the distance between the O_2 and O_3 axes. As a result of the change in these parameters, the synthesis of the attached structural groups was necessary. Some of the elements, such as the double-arm lever FO_4E and the slider G, which is a heddle shaft, will not change their geometric parameters. The fixed axes of the mechanism's kinematic scheme are marked in Fig. 2. The O_2 axis is at a distance of 151 mm from the origin, the O_3 is at a distance of 311 mm from the O_1 axis, the dimension of the O_2B lever is 192.5 mm. Due to the new position of the axes, the levers O_2B, O_3C, and O_3D, as well as the rod BC, should be changed, and their values should be obtained as a result of synthesis. In addition, the angle of the double-arm lever AO_2B should be reduced by 35 degrees so that it does not take up much space when deflected.

The lever mechanism is synthesized assuming that it starts from the last link, which is responsible for the stroke provided by the amount of heddle lifting. For instance, the stroke of the fourth heddle shaft should be 75 mm [1, 19]. The symmetry of the heddle stroke relative to the horizontal axis was chosen as the main criterion for synthesis. Thus, for the fourth heddle shaft, it was 37.5 mm. According to the technical documentation, the lever has a size of O_4E = 138.5 mm. Then, for the fourth heddle shaft, the value of the angle is μ_1 (see Fig. 2).

$$\mu_1 = \arctan\left(\frac{EE'}{2 \cdot O_4E}\right), \tag{1}$$

where EE' = 75 mm.

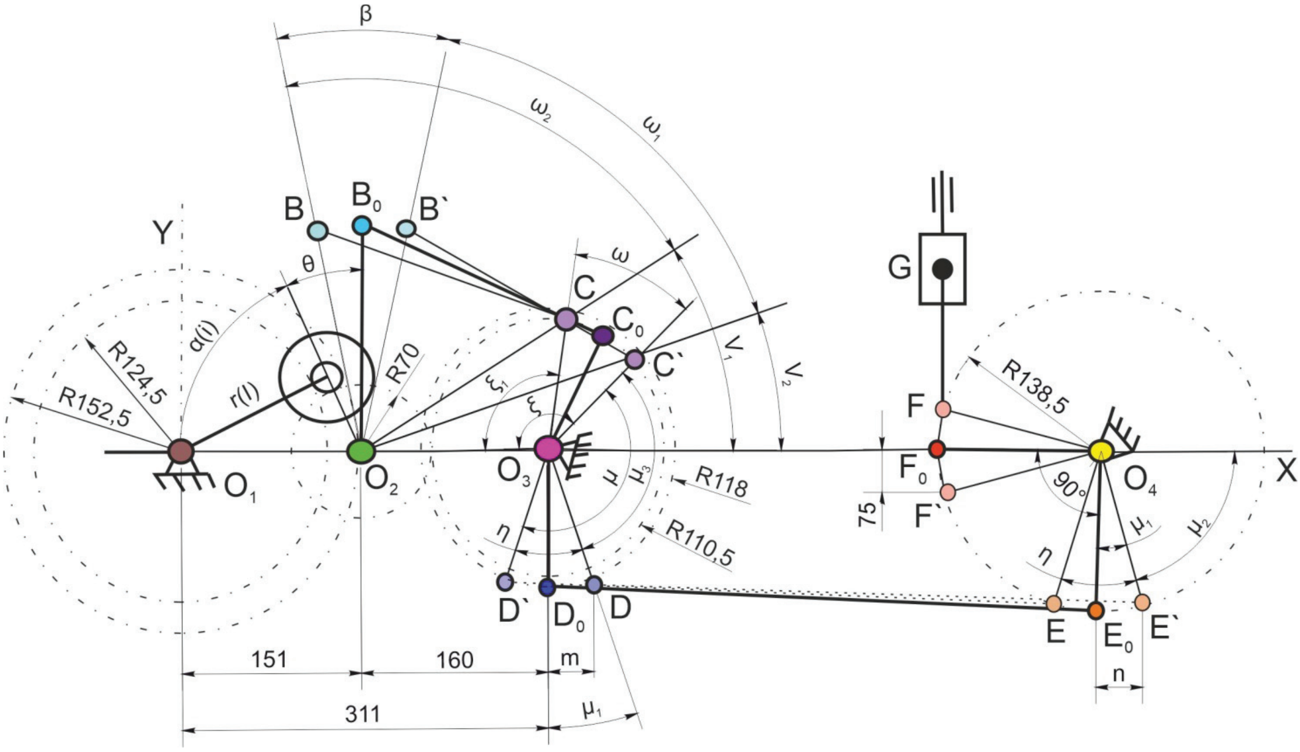


Fig. 2. Kinematic scheme of the heddle drive mechanism

The angle value is $\mu_1 = 15.15°$.

Further synthesis of the lever mechanism is carried out on the assumption that the angle of the lever rotation O_3DD' is equal to the angle O_4EE', while the hard angle for the lever CO_3D is assumed to be 155°. In this case (see Fig. 2), the angle ξ is determined by:

$$\xi = 180° - (\mu - (\mu_1 + 90°)), \tag{2}$$

Here, the angle $\xi = 130.15°$.

To determine the angle ξ_1, it is necessary to consider the triangle O_2CO_3. First of all, from an oblique triangle, we define the side O_{21}:

$$O_2C_1 = \sqrt{O_2O_3^2 + O_3C'^2 - 2O_2O_3 \cdot O_3C' \cdot \cos\xi}, \tag{3}$$

We received $O_2C_1 = 270.849$ mm.

Then the angle ξ_1 is determined from the expression:

$$\xi_1 = 180° - (\mu - (90° - \mu_1)), \tag{4}$$

Its values were $\xi_1 = 99.85°$.

From the oblique triangle O_2CO_3', we define the side O_2C'

$$O_2C' = \sqrt{O_2C_3^2 + O_3C'^2 - 2 \cdot O_2O_3 \cdot O_3C' \cdot \cos\xi_1}, \tag{5}$$

The side size is $O_2C' = 228.832$ mm.

Similarly, the rod length $VS = 225$ mm is found from the oblique triangles $O_2C_0O_3$ and $O_2B_0C_0$.

To determine the angles v_1 and v_2, consider the expression

$$v_2 = \arcsin\left(\sin\xi \cdot \frac{O_3C'}{O_2C'}\right), \tag{6}$$

The value of this angle is $v_2 = 23.008°$.

$$\nu_1 = \arcsin\left(\sin \xi_1 \cdot \frac{O_3C}{O_2C}\right), \tag{7}$$

The angle value is $v_1 = 36.607°$.

The angles ω_1 and ω_2 are determined from the triangles $O_2B'C'$ and O_2BC:

$$\omega_1 = \arccos\left(\frac{O_2B'^2 + O_2C'^2 - C'B'^2}{2 \cdot O_2B' \cdot O_2C'}\right), \tag{8}$$

Then the angle is $\omega_1 = 55.014°$.

$$\omega_2 = \arccos\left(\frac{O_2B^2 + O_2C^2 - CB^2}{2 \cdot O_2B \cdot O_2C}\right), \tag{9}$$

Then $\omega_2 = 63.874°$.

The swing angle of the roller shaft is determined by:

$$\beta = \omega_2 + \nu_1 - (\omega_1 + \nu_2), \tag{10}$$

Its value is $\beta = 22.46°$.

Under such conditions, the stroke of the roller center is 27.44 mm.

The dimension of the connecting link *DE* was determined by the position of the points *DE* and amounted to 1133 mm for the fourth heddle shaft.

Based on the data from the technical documentation of the weaving machine manufacturer (Sibtextilmash plant), the minimum and maximum radius vectors of the cam were r_{min} = 124.5 mm and r_{max} = 152.5 mm; in this case, the stroke of the roller center along the chord is 28 mm. In order to leave these parameters unchanged, we changed the dimensions of the lever O_3C, and interpolating the values obtained, we found the necessary size for the lever, equal to 142.5 mm, which provided the necessary stroke of the center of the roller (28 mm). The main dimensions of the lever system obtained as a result of synthesis are summarized in Table.

Link dimensions

Link dimensions, mm						
AO_2	BO_2	BC	CO_3	DO_3	DE	EO_4
70	192.5	225	142.5	138.5	1133	138.5

To ensure an interlacing pattern based on 10 heddle shafts, the heddle lifting mechanism must allow for determining the stroke of each shaft [10]. For this purpose, consider the diagram shown in Fig. 3. Where h_i is the height of the shed; t is the stroke between the heddle shafts; Δh_i is the increments of the shafts stroke; α_p is half of the angle of the shed, representing only a part of the shed. In this case, the amount of opening for a full shed (the stroke of heddle shafts) can be determined by the formula:

$$H_n = \left[(h_1 + (n-1) \cdot t \cdot \tan(\alpha_p)\right] \cdot 2, \tag{11}$$

To implement dependence (11), it is necessary that the dimensions of the lever DO_3 correspond to the specified motion of the heddle.

Consider the kinematic scheme shown in Fig. 2. The angle μ_1 for the arm DO_3 is left unchanged, and the chord D_0D takes a value equal to half the stroke of the heddle shaft. Taking into account the expression (11), we obtain:

$$L_n = \frac{H_n}{2} \cdot \tan(\mu_1), \tag{12}$$

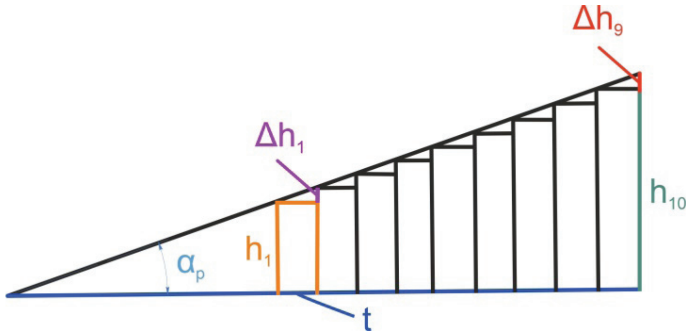


Fig. 3. A fragment of a half-open shed of a loom

where L_n is the dimension of the lever; H_n is the full stroke of the heddle shaft corresponding to its number; μ_1 is the angle determining the position of the lever relative to the vertical axis.

After determining the required dimensions of the lever system, the synthesis of the cam mechanism becomes possible. The synthesis of the mechanism begins with determining the main parameters and the law of motion for the roller center [10–12, 24–39, 45, 46]. It was necessary to determine the law for the roller center motion because we were only given a table of radius vector values. We considered power-law, straight-line, simple harmonic, double harmonic and cycloidal laws of motion. There is no need to dwell on this in more detail, as it is well described in [10–12, 19, 24–28, 35–38, 45, 46]. The law of motion of the roller center along the cycloid was found to be the most acceptable for the case under consideration.

For the synthesis of the cam mechanism, the following calculated data were used: minimum cam radius $r_{\min}$ = 124.5, maximum cam radius $r_{\max}$ = 152.5; roller radius R = 75 mm; phase angles: heddle lifting φ_1 = 70°, delay in the upper position – φ_2 = 110°, lowering of the heddle – φ_3 = 70°, delay in the lower position is φ_4 = 110°; the interaxial distance is O_1O_2 = 151 mm.

The coefficients for the cycloid calculations are taken from [10–12, 19, 24–28, 35–38, 45, 46]:

$$k_1 = S_{\max} \cdot \frac{2\pi}{\varphi_1^2}; \qquad k_3 = S_{\max} \cdot \frac{2\pi}{\varphi_3^2}.$$

The calculation of acceleration analogues was performed using the formula:

$$a(\varphi) = \left| \begin{array}{l} k_1 \cdot \sin\left(\varphi \cdot \dfrac{2\pi}{\phi_1}\right) if\, 0 \le \varphi \le \varphi_1 \\ 0 if\, \varphi_1 \le \varphi \le \varphi_2 + \varphi_1 \\ k_3 \cdot \sin\left(\varphi \cdot \dfrac{2\pi}{\phi_3}\right) if\, \varphi_1 + \varphi_2 \le \varphi \le \varphi_1 + \varphi_2 + \varphi_3 \\ 0 if\, \varphi_1 + \varphi_2 + \varphi_3 \le 360° \end{array} \right|, \tag{13}$$

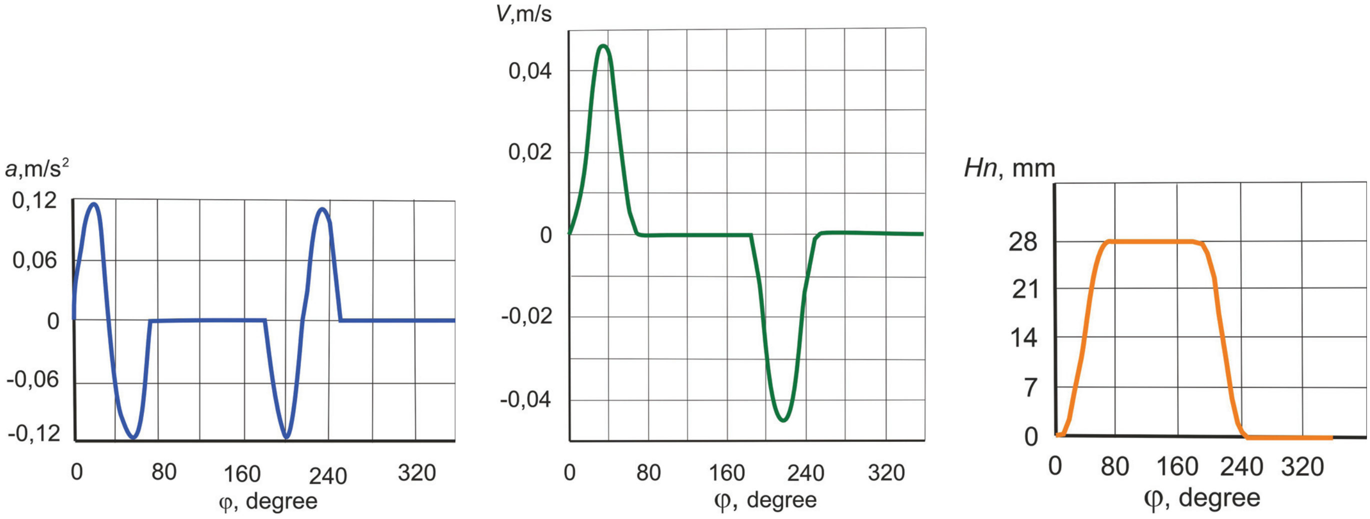
To determine the speed of the roller center, we integrated accelerations from 0° to 360° of cam rotation.

$$V(\varphi) = \int_0^{\varphi} a(\varphi)\, d(\varphi), \tag{14}$$

To determine the motion of the roller center, we integrated the speeds from 0° to the 360° cam rotation.

$$S(\varphi) = \int_0^{\varphi} V(\varphi)\, d(\varphi), \tag{15}$$

Graphs of kinematic characteristics for the roller center of the cam mechanism are shown in Fig. 4.



a *b* *c*

Fig. 4. Graphs of kinematic characteristics' analogs for the center of the roller:
(*a*) acceleration; (*b*) speed; (*c*) motion

To prevent jamming and ensure the strength of the cam pair of the mechanism, it is necessary to know the numerical values of the pressure angles, which in our case should not exceed 45 $^{\circ}$. The program developed for this purpose calculated the pressure angle values. Because the determination procedure is cumbersome, it is not presented in the paper. They did not exceed acceptable values in the entire range of studies conducted. To confirm the accuracy of the selected roller dimension, its value was compared with the actual radius of curvature, determined by the formula:

$$\rho_i = \frac{\left[y_i^2 + (\dot{y}_2)^2\right]^{\frac{3}{2}}}{y_i + 2\cdot(\dot{y}_i)^2 - y_i \cdot \ddot{y}}, \tag{16}$$

where $\rho_i, y_i, \dot{y}_i, \ddot{y}_i$ are the radius vectors of the center profile of the cam and the derivatives at the *i*-th point.

A program for the mathematical software package was developed for this purpose. The calculation results are shown in Fig. 5.

The conditions agree well with the expression:

$$r \le 0,7\cdot\rho_{\min}; r \le 0,4\cdot r_0, \tag{17}$$

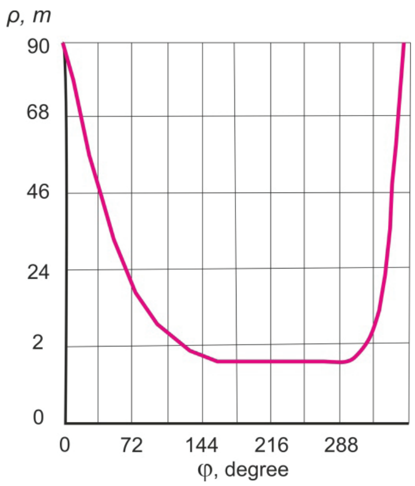


Fig. 5. Pressure angles for the cam mechanism

where $\rho_{\min}$ is the minimum radius of curvature of the cam's center profile.

The data analysis results suggest that the roller radius choice for the cam mechanism is accurate.

Next, the cam profile (radius vectors of the cam *r(i)*) was determined using equation (15). The calculation was carried out in a mathematical sotware package; the matrix of values of radius vectors and the shape of the cam profile are shown in Fig. 6.

To confirm the accuracy of the selected link dimensions, it is necessary to conduct a kinematic analysis for individual Assur groups. If their graphs have smooth and continuous characteristics, we assume that the synthesis was accurate. For kinematic analysis, the dimensions of the links obtained as a result of the lever system synthesis were used (see Table 1). Kinematic analysis began with a first-class first-order mechanism, which was used as a variable radius vector shown in the table (Fig. 6) [10–12, 19, 24–28, 35–

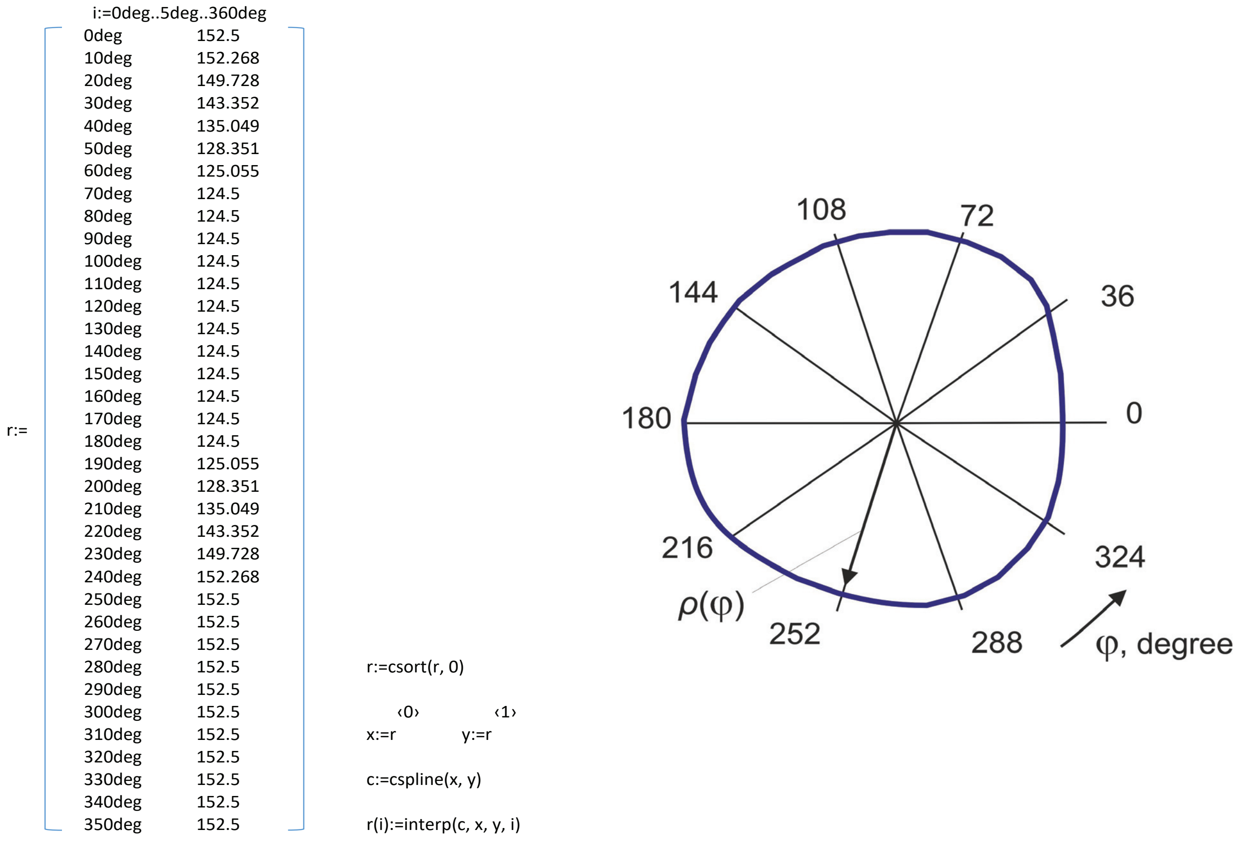


Fig. 6. The matrix of values of radius vectors and the shape of the cam profile: (*a*) radius vectors of the cam; (*b*) shape of the cam profile

38, 45, 46]. Fig. 7 shows a diagram for determining the coordinates of point *B*. In this case, it is necessary to have the radius vectors of the cam *r(i)*, as well as the lengths of the links *AB*, BO_2, coordinates O_1 and O_2 (see Table), and a hard angle θ.

According to the cosine theorem, we find the angle α (Fig. 8) from the triangle AO_1O_2

$$\alpha = \arccos\left(\frac{O_1O_2^2 + AO_2^2 - r(i)^2}{2 \cdot O_1O_2 AO_2}\right), \tag{18}$$

The angle δ is determined by:

$$\delta = \pi - (\alpha + \theta). \tag{19}$$

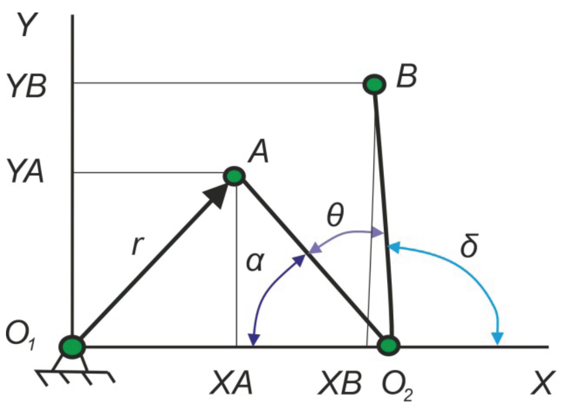


Fig. 7. The second-class first-type Assur group attached to the first-class first-type mechanism

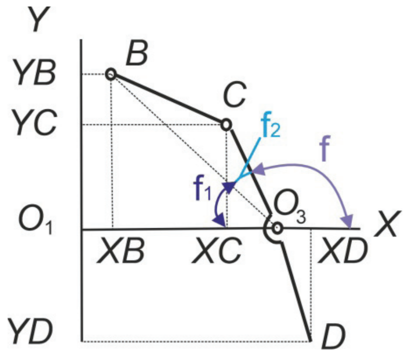


Fig. 8. The second-class first-type Assur group

The coordinates of point *B* are found as projections on the *X* and *Y* axes:

$$XB = O_1O_2 - BO_2 \cdot \cos(\delta), \quad (20)$$

$$YB = BO_2 \cdot \sin(\delta). \quad (21)$$

By determining the coordinates of point *B,* you can find the resulting value according to the expression:

$$B = \sqrt{XB^2 + YB^2}. \quad (22)$$

For the second-class first-type Assur group (Fig. 8), the following values are set: coordinates of point *B* and O_3, lengths of links *BC*, CO_3, and *CD* (Table 1), as well as the angle determining the position of vector *B*. To determine the coordinates of point *D*, it is necessary to determine the angle *f* of the lever *DC* and the value of the segment BO_3 from the triangle BO_2O_3 according to the cosine theorem.

$$f_1 = \arctan\left(\frac{y}{O_1O_3 - x}\right), \quad (23)$$

$$BO_3 = \sqrt{BO_2^2 + O_2O_3^2 - 2 \cdot BO_2 \cdot O_2O_3 \cdot \cos(f_1)}, \quad (24)$$

$$f_2 = \arccos\left(\frac{BO_3^2 + CO_3^2 - BC^2}{2 \cdot BO_3CO_3}\right), \quad (25)$$

$$f = \pi - (f_1 + f_2). \quad (26)$$

The coordinates of points *C* and *D* are found as projections on the *OX* and *OY* axes:

$$XC = O_1O_3 + CO_3 \cdot \cos(f), \quad (27)$$

$$YC = CO_3 \cdot \sin(f), \quad (28)$$

$$XD = O_1O_3 - DO_3 \cdot \cos(f), \quad (29)$$

$$YD = DO_3 \cdot \sin(f). \quad (30)$$

Further, according to the Pythagorean theorem, their resulting values were found:

$$C = \sqrt{XC^2 + YC^2}, \quad (31)$$

$$D = \sqrt{XD^2 + YD^2}. \quad (32)$$

Consider the second Assur group, which is part of the general scheme of the heddle motion mechanism. This is also second-class first-order group (Fig. 9).

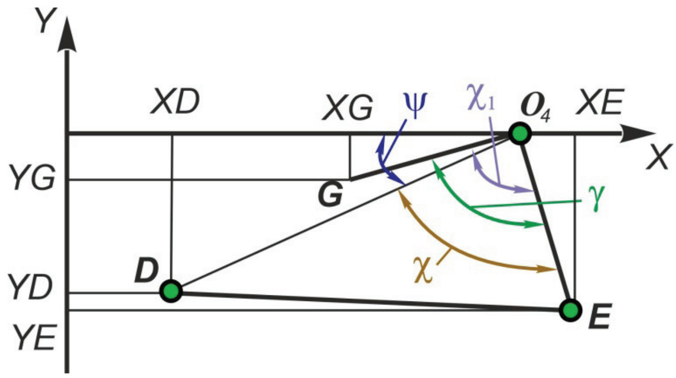


Fig. 9. The second-class first-type Assur group, which is part of the general scheme of the heddle motion mechanism

The following values should be set for this group: the coordinates of points *D* and O_4, the lengths of the links *DE*, GO_4 and the hard angle between EO_4 and GO_4.

From the triangle DEO_4, according to the Pythagorean theorem, we determine the hypotenuse DO_4 by the formula:

$$DO_4 = \sqrt{(O_1O_4 - XD)^2 + YD^2}\,. \tag{33}$$

From the triangle DEO_4, we find the angle *s* by the cosine theorem

$$\chi = \arccos\left(\frac{DO_4^2 + EO_4^2 - DE^2}{2 \cdot DO_4 \cdot EO_4}\right). \tag{34}$$

The angle ψ is determined by:

$$\psi = \arcsin\left(\frac{YD}{DO_4}\right). \tag{35}$$

Then the angle χ_1 is found as:

$$\chi_1 = \psi + \chi\,. \tag{36}$$

The coordinates of the point *E* are found as projections on the axes *OH* and *OY*:

$$XE = O_1O_4 - EO_4 \cdot \cos(\chi_1)\,, \tag{37}$$

$$YE = EO_4 \cdot \sin(\chi_1)\,. \tag{38}$$

The value of motion *E* is determined by:

$$E = \sqrt{XE^2 + YE^2}\,. \tag{39}$$

The angle of the lever position GO_4E is found as the angle difference:

$$\varepsilon = \chi_1 - \gamma\,. \tag{40}$$

The coordinates and the length of the vector of the point *G* are defined as:

$$XG = O_1O_4 - GO_4 \cdot \cos(\varepsilon)\,, \tag{41}$$

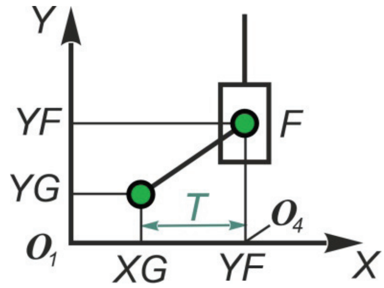
$$YG = GO_4 \cdot \sin(\varepsilon)\,. \tag{42}$$

The resultant is determined by:

$$G = \sqrt{XG^2 + YG^2}\,. \tag{43}$$

Consider the last Assur group for our mechanism. It is a second-class second-type group (Fig. 10). The length of the link *GF*, the *x* coordinate of the guide along which the slider *F* moves (in this case, it is zero) are needed to determine the trajectory of the point *F*. The length of the projection of the *GF* link on the *OX* axis is equal to the difference between the coordinates of the *G* point and the guide for the slider.

Fig. 10. The second-class second-type Assur group



Based on Fig. 10, the value of *T* is determined by:

$$T = XF - XG\,. \tag{44}$$

From the *GFT* triangle, according to the Pythagorean theorem, we define:

$$YF = \sqrt{GF^2 - T^2}\,. \tag{45}$$

Then the total motion of the point *F* is determined:

$$(YF)_o = EG + YF\,. \tag{46}$$

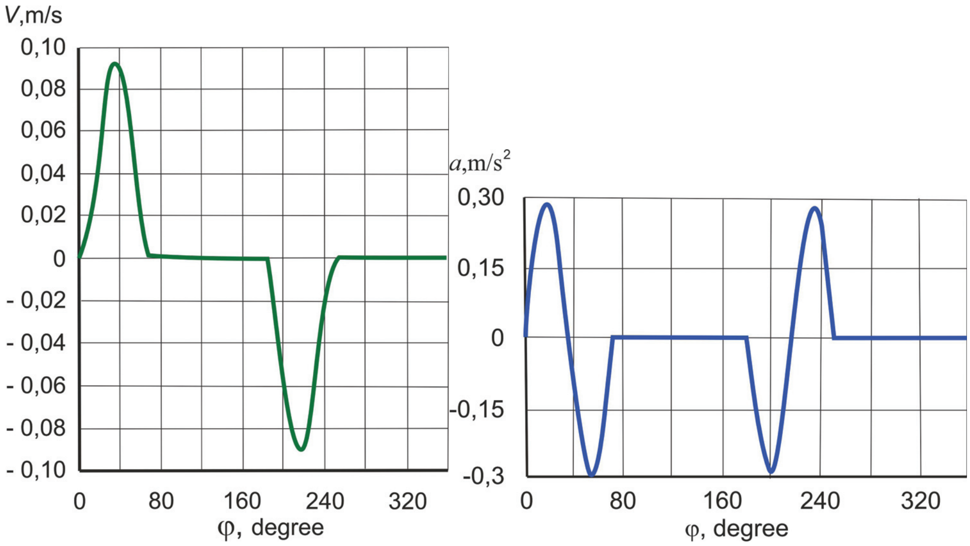
Results and discussion

The analysis revealed that the heddle fixing mechanism can be placed outside the heddle frame. As a result, the values of the axial distance O_2O_3 were reduced by 100 mm. Due to the fact that the heddle shaft stroke is a known value obtained as a result of calculations of the shed geometry [1] (point *G* in Fig. 2), the methodology of synthesizing the mechanism [9, 29, 35, 36, 45–49] for moving the heddle suggests starting it from the last Assur group. The motion of the fourth heddle shaft equal to 75 mm is accepted as a known parameter [1, 9, 10, 29, 35, 36, 45–49]. The synthesis condition for this group is the equality of chords $E'E = F'F$ relative to the horizontal axis. The angles of rotation of these levers are also equal and amount to $\mu_1 = 15.15°$. They were given previously and defined by formula (1). Further synthesis was carried out for the fourth second-class first-type Assur group. Significantly, the main condition for synthesis is the equalization of arcs (chords) $E'E = D'D$, $EE_0 = DD_0$ and arm lengths $O_4E = O_3D$. Further synthesis of the mechanism consisted of determining the swing angle of the lever with rollers, which is calculated by the formula (10). The swing angle of this lever depends, among other things, on the dimension of the arm O_3D. The dimensions of this lever were taken within the range of 138.5–143.5 mm. By interpolating the values of angle β, we determined that β = 22.926°, corresponding to a chord length of 28 mm. We then calculated the length of the arm O_3D of the lever O_3DC to be 143.5 mm. When tackling the loom for manufacturing a variety of fabrics, up to ten heddles may be used, and their movement is determined by their position within the machine. Therefore, the dimension of one of the levers in the kinematic scheme, which allows the adjustment of the heddles' stroke, was chosen as a variable parameter. In our case, it was the DO_3 lever. Using the analytical dependences (11) and (12), it is possible to calculate the length of the DO_3 lever and the value of the heddles' stroke.

After the synthesis of the lever mechanism, which enabled calculating the motion of the roller center equal to 28 mm, the main parameters for the synthesis of the cam mechanism were determined. The synthesis justified the law of motion of the roller center along the cycloid and determined the radii of curvature (see Fig. 5). This led to the conclusion that a roller radius of 37.5 mm satisfies condition (17). The calculated pressure angles are within acceptable limits throughout the entire range of rotation of the main shaft, from 0° to 360°. The radius vectors of the cam are shown as a matrix of values (Fig. 6). After processing the tabular values of the radius vectors with splines, we conducted a kinematic analysis of the mechanism for the characteristic points of the Assur groups. The purpose of this analysis was to confirm the accuracy of the synthesis and the smoothness and continuity of the kinematic parameters graphs of the Assur groups characteristic points. For the latter group, it was necessary to confirm the value of the heddle shaft stroke for point *G*, since it was the basis for calculating and constructing the synthesis methodology.

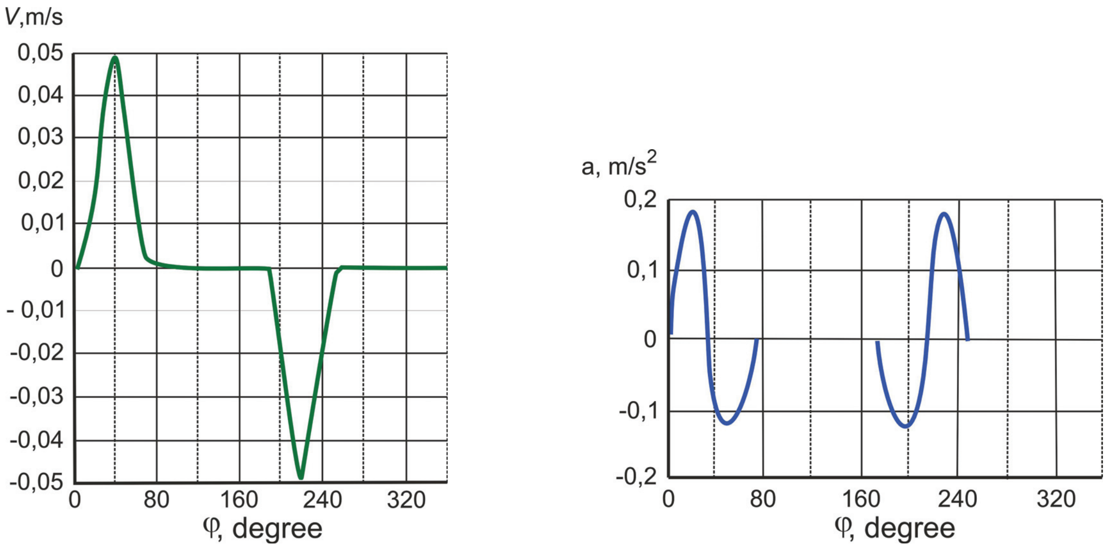
Thus, for point *B*, the kinematic characteristics are shown in Fig. 12, and for point *C,* in Fig. 11.

For point *D*, the kinematic characteristics are shown in Fig. 13.



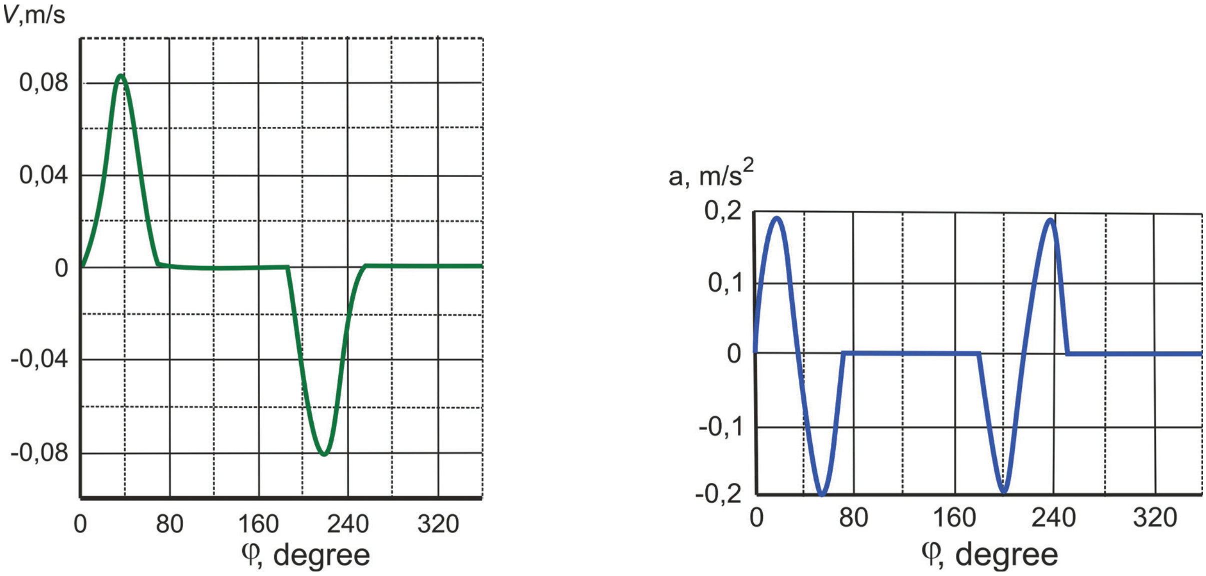
a *b*

Fig. 11. Kinematic characteristics for point *B*:
(*a*) velocity; (*b*) acceleration



a *b*

Fig. 12. Kinematic characteristics for point *C*:
(*a*) velocity; (*b*) acceleration

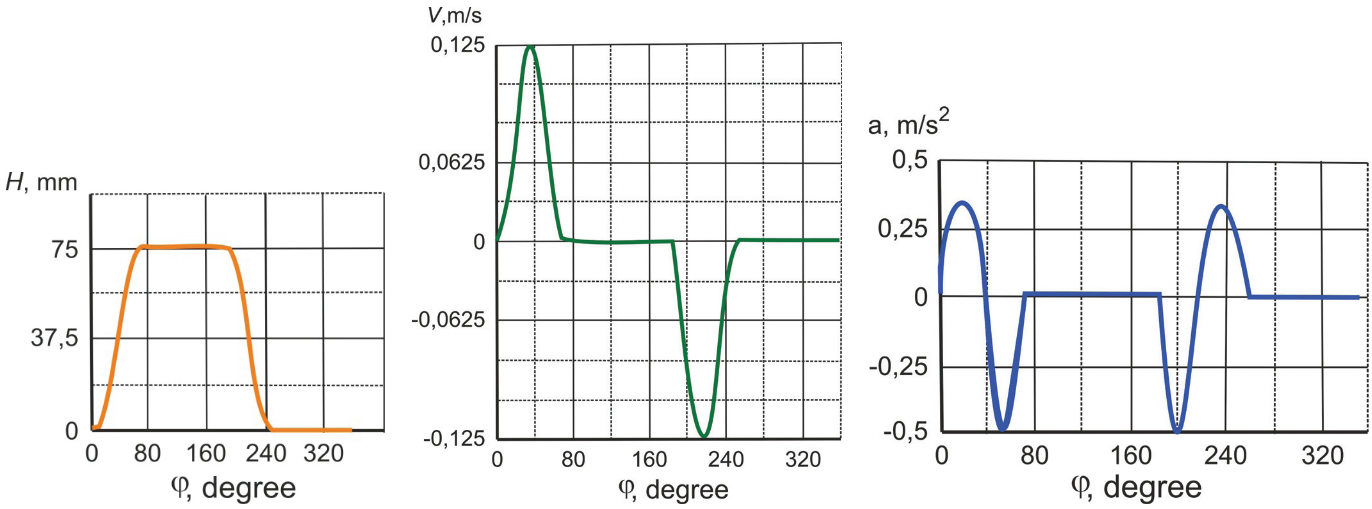


a *b*

Fig. 13. Kinematic characteristics for point *D*:
(*a*) velocity; (*b*) acceleration

The analysis showed that the velocities and accelerations for points *A, B, C, D, E, G, F* have smooth and continuous graphs, which indicates a properly conducted synthesis of the lever system for individual Assur groups. The kinematic characteristics for points *G* and *E* are not given in the work.

Motion for point *F* is shown in Fig. 14.



a *b* *c*

Fig.14. Kinematic characteristics for point *F*:
(*a*) motion; (*b*) velocity; (*c*) acceleration

The economic effect of the proposed solution implementation was determined based on the data of the work [10], where the removal of products from 1 m^2 of the production area, adjusted for the rotation frequency of the main shaft equal to 300 min^{-1}, is 1.035 m^2/hour.

Conclusion

The main purpose of the work was to reduce the dimensions of the loom by changing the design parameters of the heddle lifting mechanism. As a result of placing the heddle fixing mechanism outside of the heddle frame, the dimension of O_2O_3 was reduced by 100 mm. In this regard, all dimensions for the elements included in the Assur groups were changed, with the exception of the last one. The presented synthesis methodology assumes that it is carried out in the reverse sequence of their connection. Synthesis for the lever system should be carried out for the fourth heddle shaft, for which a motion of point *G* was set to be 75 mm. First of all, the swing angle of the lever CO_3 D was determined, which was equal to the swing angle of the other lever EO_4F. Then the dimension of the CO_3 arm was determined, which was 143.5 mm. As a result, the stroke of the center of the roller was equal to 28 mm. Since the stroke of the heddle is different in the depth of the loom, the value of the lever DO_3 was chosen as the variable parameter. The variable parameters of the DO_3 lever and the heddle stroke were calculated using the analytical dependencies presented in [10] and a mathematical software package of application programs. The results are shown in Fig. 10.

As a result of the synthesis, the dimension of the connecting link BC was calculated to be 225 mm, the link O_3D was 138.5 mm, and the angle between the arms O_3D and CO_3 was 155 °. The connecting rod *DE* assumed the value of 1133 mm. The objectivity of the synthesis is confirmed by the results of the conducted studies for Assur groups. The kinematic characteristics for individual points of the mechanism are presented in the form of graphs and have smooth, continuous functions, which indicates the quality of the synthesis performed.

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Conflicts of Interest

The authors declare no conflict of interest.