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# Development of the methods of kinematic analysis of elliptic drum of vertical-spindle cotton harvester

Gayrat Bahadirov<sup>3</sup>, Shavkat Ravutov<sup>2</sup>, Abdusalam Abdukarimov<sup>3</sup> and Elyor Toshmatov<sup>1</sup>

<sup>1</sup>Tashkent Institute of Irrigation and Agricultural Mechanization Engineers, 39, st. Kari-Niyazi, 100000, Tashkent, Uzbekistan

<sup>2</sup>Tashkent State Technical University, Universitetskaya str., 2, Tashkent, Uzbekistan

<sup>3</sup>Institute of Mechanics and Seismic Stability of Structures of the Academy of Sciences of the Republic of Uzbekistan, Durmon yuli str., 31, 100125, Tashkent, Uzbekistan

E-mail: tohir-zs@mail.ru, instmech@rambler.ru, t.elyor85@mail.ru

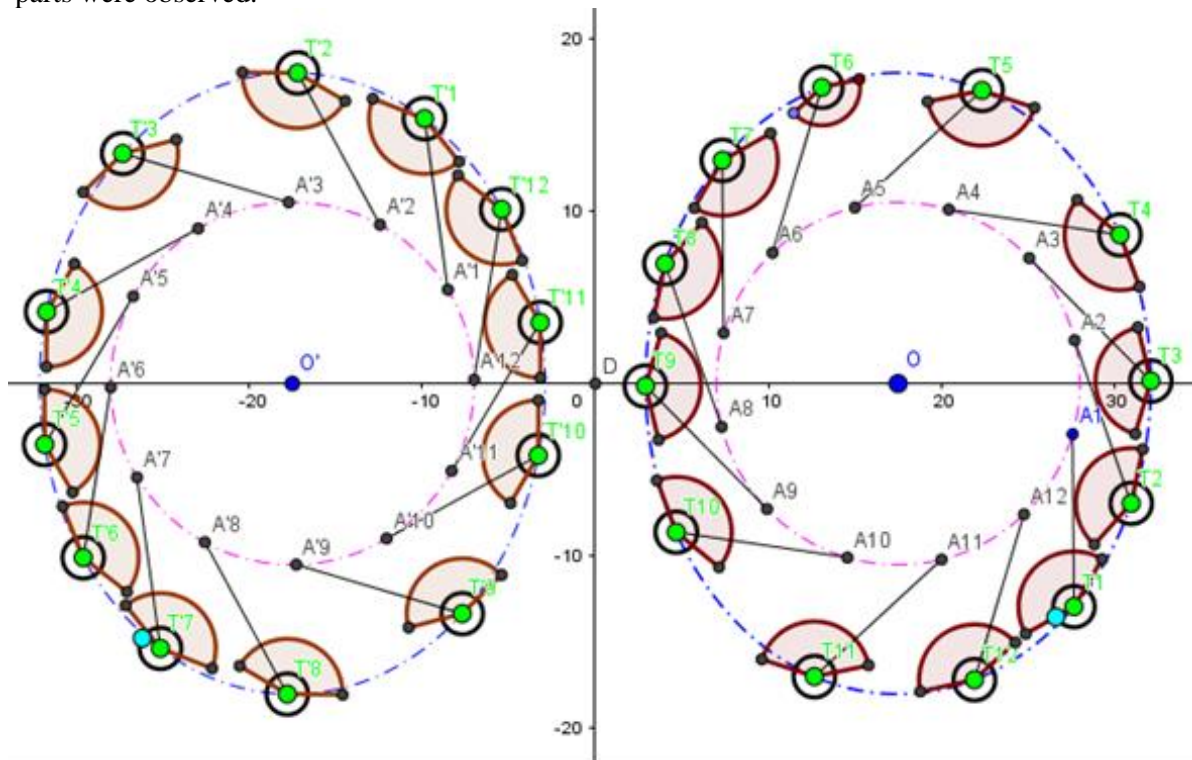
**Abstract.** The paper is devoted to the development of a graphic-analytical method for studying the lever mechanism of an elliptical drum of a cotton harvester, which favorably differs from a serial drum in the completeness of cotton picking from bushes and the quality of the harvested raw cotton. The aim of the study is to define the degree of the effect of the structure and geometry of individual links of the elliptical drum mechanism on changes in kinematic parameters of the characteristic points of the mechanism links, in particular, of the spindles centers. The kinematics of the mechanism was investigated using classical methods of theoretical mechanics, the theory of mechanisms and mechanics of machines, combined with modern methods of computer graphics. The developed research method was described, its usefulness and advantages were shown. As an example, computational and graphic procedures were performed to determine the linear and angular velocities and accelerations of the characteristic points of the mechanism of an elliptical drum in its specific position. The method is clear, easy to control the results, allows one to quickly solve applied design problems, and develops an engineering intuition to evaluate the capabilities of the mechanism according to its kinematic scheme. The research results are of interest for a wide practical application of elliptical spindle drums in the construction of vertical-spindle cotton harvesters.

## 1. Introduction

One of the promising directions for increasing the efficiency of vertical-spindle cotton harvesting machines (VS CHM) is the improvement of its main working body - the spindle drum (SD). A new design of the vertical-spindle drum was obtained earlier [1,2] by structural development of the mechanism of existing vertical-spindle drum. The main distinguishing features of this spindle drum in comparison with the serial one are the ellipticity of the spindle orbit, the increased contact area of the spindles with a cotton plant during operation, and an increased number of simultaneously operating spindles at the same number of spindles in one drum (Figure 1). In addition to these properties, it is possible to change the kinematic parameters of the spindles by changing the geometrical dimensions of the drum mechanism links, which is very important for determining the optimal kinematic modes of the spindles [3-9]. In particular, in [5], the study is limited to determining the linear and angular



velocities of the spindle, which is not enough for an objective assessment of drum operation. To increase the efficiency of the spindles used in a vertical-spindle apparatus, in [6] it was proposed to introduce a plant mechanism (PWM) into the drum design. This method allows some increase (up to 5-5.5%) in the completeness of raw cotton harvesting. However, according to the authors' statement, in field experiments with cotton picker equipped with BWM, frequent breakdowns of the machine parts were observed.



**Figure 1.** Diagram of a cotton picking machine with elliptical drums

Another disadvantage of the drum is that when the PWM was introduced into the drum design, its mass increased almost 2 times. A thorough structural-kinematic analysis of the BWM showed an excessive complication of the drum design [11, 12] and the authors of that study came to the conclusion that high accuracy must be observed in order to increase the reliability of the drum operation in the manufacture and assembly of drum parts.

This study is a continuation of our earlier work [3, 4, 5] and the goal is to develop an effective method for determining the kinematic parameters of the elliptical drum mechanism and to determine the degree of influence of the structure and geometry of individual links of the mechanism on its kinematics. At the same time, the classical methods of theoretical mechanics, the theory of mechanisms and mechanics of machines [13-17] were applied, in combination with modern methods of computer graphics, which actually eliminates the errors in graphic and computational work, while increasing the visualization of the results obtained [17-21].

## 2. Methods

The work addresses the following tasks:

1. With account for the complexity of kinematic study of the mechanism with higher kinematic pairs, to replace it with an equivalent mechanism with lower kinematic pairs [913].
2. Analysis of the features of the replacing mechanism structure and the possibility of using the grapho-analytical method to study its kinematics.

3. Graphic representation of the velocity changes in characteristic points of the mechanism links, with simultaneous calculation of their numerical values in various positions using the computer graphics program "Compass-Graph".

2.1. Structural analysis and replacement of higher pair in the mechanism with lower one.

The structurally developed mechanism of the drum is a cam-lever mechanism, which differs from

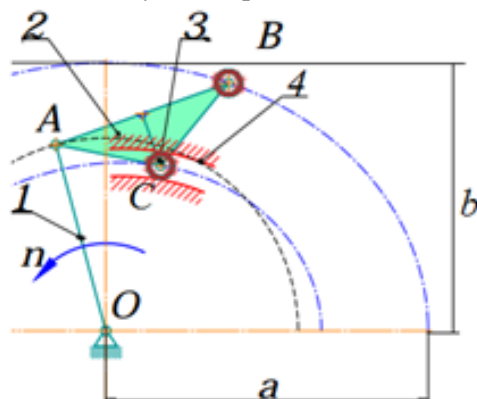


Figure 2. Kinematic diagram of the elliptical drum mechanism

traditional cam mechanisms by the cam immobility (Figure 2). In this mechanism, the rocker arm 2 moves around the fixed cam, performing a complex plane-parallel motion, and one end of the rocker arm A makes a circular motion with the crank 1, and the other - B - moves along an ellipse. This point coincides with the center of the spindle in the elliptical drum. To ensure the motion of point B along an ellipse, a finger with a freely rotating roller 3 is attached to point C of the rocker arm, which, in contact with the walls of the guide track of the fixed cam 4, forms a geometric constraint with it — a higher kinematic pair. The crank design is made in the form of a disk pivotally connected to the rocker arms, the number of which is equal to the number of spindles on the drum.

The mechanism is flat, as its links enter only kinematic pairs of IV (the roller with the walls of the cam guide track) and V classes (rotational pairs O, A and C) (Figure 2). Using the Chebyshev formula, we determine the degree of freedom of the mechanism,

$$W = 3n - 2p_5 - p_4 = 2, \tag{1}$$

where  $n = 3$  is the number of moving links: crank - 1, rocker arm - 2, and roller - 3;  $p_5 = 3$  – the number of lower kinematic pairs: rotational pairs O, A and C;  $p_4 = 1$  – the number of higher kinematic pairs - the connection of the roller with the walls of the cam guide track. As can be seen, the degree of freedom of the mechanism is equal to two.

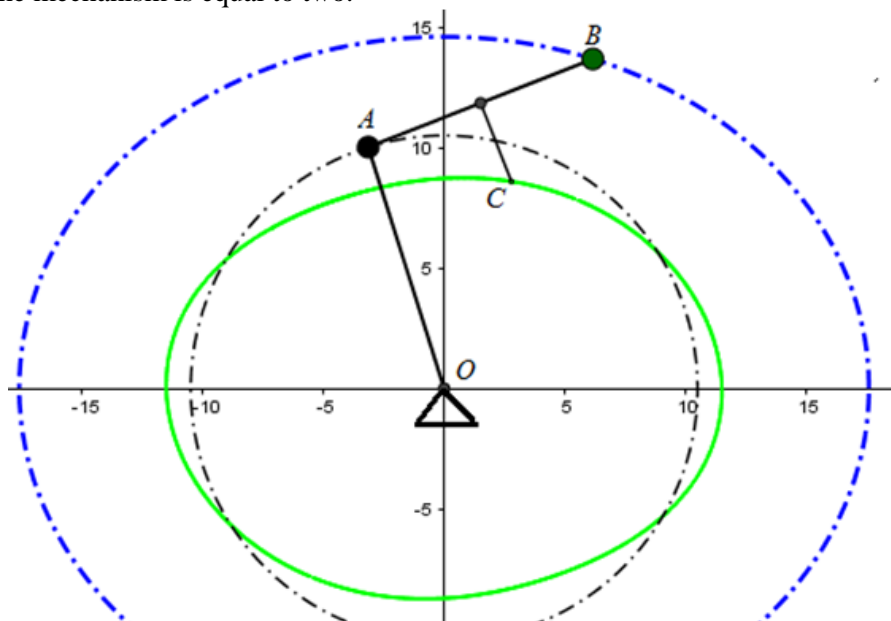
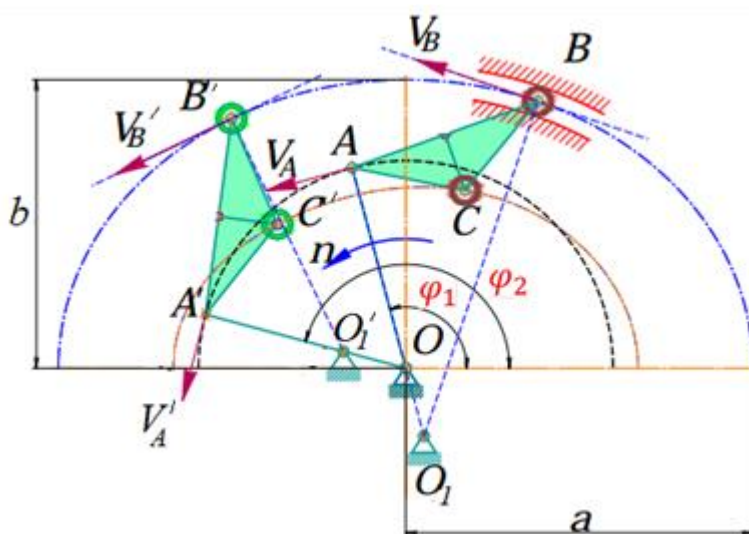


Figure 3. Trajectory of point C motion

However, if to consider the position of point  $C$ , depending on the position of the crank, then it can be seen that any position of the crank corresponds to a defined position of point  $C$ , that is, the degree of freedom of the mechanism should definitely be equal to one -  $W = 1$ . This result is typical for cam mechanisms, the reason was described in detail in [14]. To replace the mechanism with the one that has only lower, rotational kinematic pairs, we carry out some analysis of the mechanism structure [13]. It can be seen from the diagram that, from a structural and technological point of view, points  $B$  and  $C$  are the most characteristic for the mechanism of an elliptical drum: point  $B$  in the mechanism coincides with the center of the spindle, and point  $C$  determines the trajectory of point  $B$  along an ellipse with semi-axes  $a$  and  $b$  (Figure 3). But the trajectory of motion of point  $C$ , differs in shape from the ellipse and has a rather complex analytical description [14], which complicates the application of the graphic-analytical method for studying the mechanism (Figure 3). Therefore, it is possible to artificially move the guide track from point  $C$  to point  $B$  with an elliptical trajectory, which does not change the structure of the mechanism and the kinematics of its links, and of the points of these links (Figure 3).

A flat hinged four-bar linkage is one of the most widespread and most studied mechanisms in engineering, that has only rotational kinematic pairs [14]. We will replace the mechanism with a hinged four-bar linkage. Then the rocker arm 2 will turn into a connecting rod  $AB$ , which at point  $B$  forms a rotational pair with the rocker arm  $BO_1$  [3].

The peculiarity of this hinged four-bar linkage is that, depending on the position of point  $B$  on the ellipse, the location of the rack  $O_1$  changes, and the length of the rocker arm  $BO_1$  changes accordingly



**Figure 4.** Diagram of the replacement mechanism

(Figure 4). The coordinates of the position of the rack  $O_1$  coincides with the instantaneous center of velocities (ICV) of the points of hinged four-bar linkage. To determine the location of the racks corresponding to different positions of point  $B$ , we use the rule, widely known in theoretical mechanics, for determining the instantaneous center of velocities of a rigid body [15]. To do this, we draw perpendiculars to the directions of linear velocities  $V_A$  and  $V_B$  and find the point of their intersection  $O_1$ . Point  $O_1$  is the location of the rocker arm rack

of the replacing mechanism in its given position, and the distance  $BO_1$  is the length of the rocker arm of the replacing mechanism. Figure 4 shows two positions of the replacing mechanism with racks  $O_1$  and  $O_1'$ . Thus, a conditional hinged four-bar linkage with variable geometrical parameters is obtained

Determine the degree of freedom of the replaced conditional mechanism.

From the diagram (Figure 4) we see that in the replacing mechanism there are  $n = 3$  movable links:  $OA$  - crank,  $AB$  - connecting rod, and  $BO_1$  - rocker arm;  $p_5 = 4$  rotational pairs of the fifth class: points  $O, A, B$  and  $O_1$ . So,

$$W = 3n - 2p_5 = 1 \tag{2}$$

From expression (2) it follows that the replacing mechanism has the same degree of freedom and retains all the conditions of relative motion of the mechanism links.

2.2. Velocity determination of characteristic points of the mechanism by constructing velocity and acceleration diagrams.

The analytical formulas obtained by the authors for calculating the parameters of an elliptical drum are too cumbersome and require a huge amount of calculations, and they do not have clearness [1, 2]. A number of scientists also indicate these shortcomings in their works [17-21]. Graphoanalytical and graphic methods for studying lever mechanisms are free from these disadvantages. The main disadvantage of these methods is the comparative inaccuracy of the results obtained. The emergence of a number of computer graphics programs, such as AutoCAD, SolidWorks, Compass-Graph, Mathcad, etc., eliminates this draw-back. In addition, graphoanalytical and graphical research methods, due to their clearness and ease of control, are invaluable for prompt checking the correctness of analytical calculations, visualizing the mechanisms performance characteristics and evaluating their work; they are especially valuable for developing engineering intuition, and finally, they can be used as resulting curves in analytical calculations.

2.3. Mechanism diagram.

In a serial vertical-spindle cotton harvester, the drum radius ( $R = 146 \text{ mm}$ ) is limited by the condition of the device blocks capacity in the cotton row-spacing of  $90 \text{ cm}$  (with the frontal arrangement of the apparatus blocks) and  $60 \text{ cm}$  (with the tandem arrangement of the apparatus blocks). In this regard, at the initial stage of designing a cotton harvester with new drums, it is necessary to preserve all dimensions that ensure the capacity of the apparatus blocks in the row-spacing, including the transverse dimension of the ellipse at the centers of the spindles of the elliptical drum. Therefore, we take  $b=R=146 \text{ mm}$ . To study the tendencies in changes of kinematic dimensions depending on the value of the major semi-axis  $a$  of the ellipse, it is possible to vary them within  $a = 156 \div 196 \text{ mm}$  with a step of  $5 \div 10 \text{ mm}$ . The remaining dimensions of the mechanism are selected by design reasons as:  $OA = 0.105 \text{ m}$ ;  $AB = 0.10 \text{ m}$ .

Figure 5, a shows a diagram of a hinged four-link mechanism of an elliptical drum, where the major semi-axis is  $a = 176 \text{ mm}$ . The position of the mechanism in this scheme is determined by the position of the crank  $\varphi_{OA} = 105^\circ$  (further all graphic and computational work is performed for  $\varphi_{OA} = 105^\circ$ ). The construction of the mechanism diagram was conducted using the computer graphics program "Compass-Graph". In this case, the scale factor of linear dimensions is taken as  $k_l = 1$ , which greatly simplifies the design work when calculating the kinematic parameters of the links and characteristic points of the mechanism by constructing velocity and acceleration diagrams.

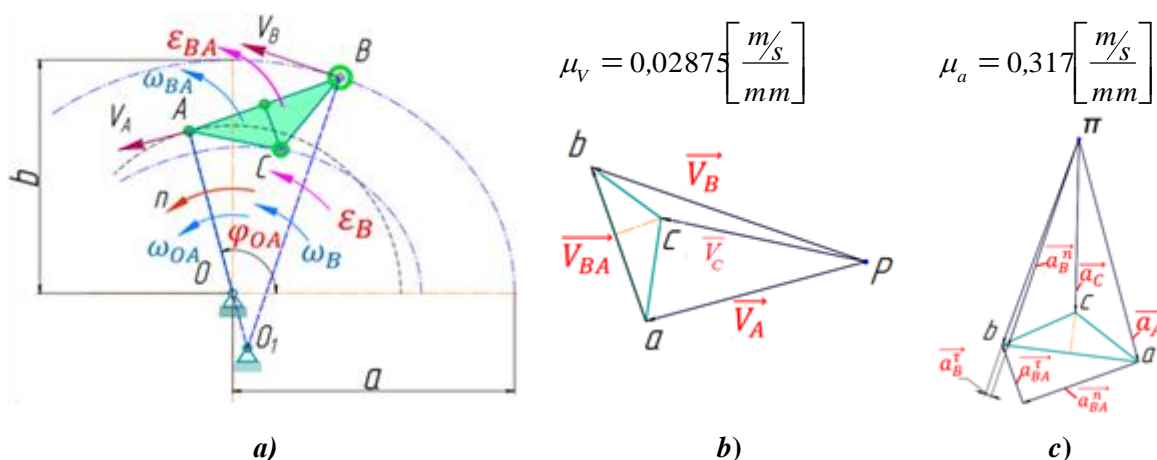


Figure 5. Construction of velocity and acceleration diagrams.

#### 2.4. Velocity diagram.

The construction of the velocity diagram is carried out with the semiaxes of the ellipse with dimensions  $b = 0.146 \text{ m}$  and  $a = 0.176 \text{ m}$ . Dimensions  $OA = 0,105 \text{ m}$ ;  $AB = 0,10 \text{ m}$  are known. The dimension of  $BO_I$  is obtained by measurement from the mechanism diagram for a given position. The number of revolutions of the drum shaft is taken to be equal to the number of revolutions of the serial drum  $n = 105 \text{ rpm}$ . Then the angular velocity of the drum shaft is  $\omega_A = \frac{\pi \cdot n}{30} = 10,99 \text{ s}^{-1}$

The velocities and accelerations of points  $A$ ,  $B$ ,  $C$ , and the angular velocities of links  $AB$ ,  $BO_I$  and point  $C$  are determined.

The velocity value of point  $A$  is determined by the well-known formula [9],

$$V_A = OA \cdot \omega_A \quad (3)$$

accounting the values of  $OA$  and  $\omega_A$ :  $V_A = 1,154 \text{ m/s}$ . The direction of the velocity vector  $V_A$  coincides with the direction of rotation of the shaft  $n$ .

From the diagram (Figure 5, a) it can be seen that the relationship between the velocities of points  $B$  and  $A$  is determined by the vector equation

$$\vec{V}_B = \vec{V}_A + \vec{V}_{BA}, \quad (4)$$

where  $V_{BA} \perp BA$  and  $V_B \perp BO_I$

There are two unknown elements in the vector equation (4): the  $V_B$  and  $V_{BA}$  velocity values. These values are determined by constructing a velocity diagram. To do this, a segment  $\overline{pa}$  40 mm long is marked off on an arbitrary point  $p$ . The scale factor of the velocity is chosen

$$\mu_V = \frac{V_A}{pa} = \frac{1,154 \text{ m/s}}{40 \text{ mm}} = 0,02875 \left[ \frac{\text{m/s}}{\text{mm}} \right].$$

Through the end of vector  $\overline{pa}$  (point  $a$ ) we draw a perpendicular to the  $AB$  link, and through the point  $p$  a perpendicular to  $BO_I$ . At the intersection of these perpendiculars, mark point  $b$ . The obtained segment  $\overline{pb}$  represents the velocity vector  $V_B$ , and the segment  $\overline{ab}$  represents the velocity vector  $V_{BA}$ .

Points  $A$ ,  $B$ , and  $C$  belong to the same link – a connecting rod. Therefore, to determine the speed  $V_C$  we use the proportionality principle of such figures in mechanism diagram and in velocity diagram. Through point  $a$  draw a line perpendicular to  $AC$ , and through point  $b$  — a line perpendicular to  $BC$ . At the intersection of these lines, we find point  $c$ , then a segment  $\overline{pc}$ , which shows velocity  $V_C$ . Now, taking into account the scale factor of velocities, we calculate the values of linear  $V_B$ ,  $V_{BA}$ ,  $V_C$  and angular velocities  $\omega_B$ ,  $\omega_{BA}$ ,  $\omega_C$ :

$$V_B = \mu_V \cdot \overline{pb} = 0,02875 \left[ \frac{\text{m/s}}{\text{mm}} \right] \cdot 52 \text{ mm} = 1,495 \text{ m/s};$$

$$V_{BA} = \mu_V \cdot \overline{ba} = 0,02875 \left[ \frac{\text{m/s}}{\text{mm}} \right] \cdot 31 \text{ mm} = 0,89 \text{ m/s};$$

$$V_C = \mu_V \cdot \overline{pc} = 0,02875 \left[ \frac{m/s}{mm} \right] \cdot 37 \text{ mm} = 1,06 \text{ m/s} ;$$

$$\omega_B = \frac{V_B}{BO_I} = \frac{1,495 \text{ m/s}}{0,182 \text{ m}} = 8,22 \text{ s}^{-1} ;$$

$$\omega_{BA} = \frac{V_{BA}}{AB} = \frac{0,89 \text{ m/s}}{0,1 \text{ m}} = 8,9 \text{ s}^{-1} ;$$

$$\omega_{BC} = \frac{V_C}{CO_I} = \frac{1,06 \text{ m/s}}{0,133 \text{ m}} = 7,97 \text{ s}^{-1} .$$

where  $CO_I$  is selected by measurement from the mechanism diagram for its given position. Transferring the vectors  $pb$ ,  $ba$  and  $pc$  from the velocities diagram (Figure 5, *b*) to points  $B$  and  $C$  in the mechanism diagram, it is possible to determine the directions of angular velocity  $\omega_B$ ,  $\omega_{BA}$  and  $\omega_C$  (Figure 5, *a*).

The same results can be obtained by constructing velocity diagrams for other positions of the mechanism.

When assessing the tendencies of changes in the velocity magnitude and direction of the link points of the mechanism, the hodographs of velocity vectors are sometimes used, which have a particular clearness of the results obtained and are usually more effective than plotting the graphs [9]. Therefore, the results of computational and graphic works to study the pattern of changes in velocity modes of the drum are presented in the form of hodographs of the velocity vectors of points  $B$  and  $C$  (Figure 6). In addition to the clearness of the results obtained, this approach makes it possible to determine the acceleration direction of a point in its given position.

### 2.5 Acceleration diagram.

The crank  $OA$  makes a circle with a constant angular velocity  $\omega_A$ . Therefore, the acceleration of a point consists of the normal acceleration  $a_A^n$  only, which is directed along the straight line  $OA$  and is determined by the formula

$$a_A^n = OA \cdot \omega_A^2 \quad (5)$$

$$a_A^n = 0,105 \text{ m} \cdot (10,99 \text{ s}^{-1})^2 = 12,68 \text{ m/s}^2 .$$

From an arbitrary point  $\pi$ , we mark off  $\overline{\pi a}$  as a segment  $\overline{\pi a} = 40 \text{ mm}$  (Figure 5, *c*).

Determine the acceleration scale factor

$$\mu_a = \frac{a_A^n}{\pi a} = \frac{12,68 \text{ m/s}^2}{40 \text{ mm}} = 0,317 \left[ \frac{m/s^2}{mm} \right] .$$



To determine the linear acceleration of point  $B$  according to the scheme (Figure 5, c), the following equation is written:

$$\vec{a}_B = \vec{a}_B^n + \vec{a}_{BA}^n + \vec{a}_{BA}^\tau, \tag{6}$$

where  $\vec{a}_{BA}^n$  is the vector of normal acceleration of point  $B$  relative to point  $A$  (directed along the straight line  $BA$  from point  $B$  to point  $A$ ),  $a_{BA}^n = BA \cdot \omega_{BA}^2 = 0,10 \cdot (8,9 s^{-1})^2 = 7,921 m/s^2$ ;  $\vec{a}_{BA}^\tau$  – is the vector of tangential acceleration of point  $B$  relative to point  $A$ ,  $a_{BA}^\tau \perp BA$  but its direction is unknown.

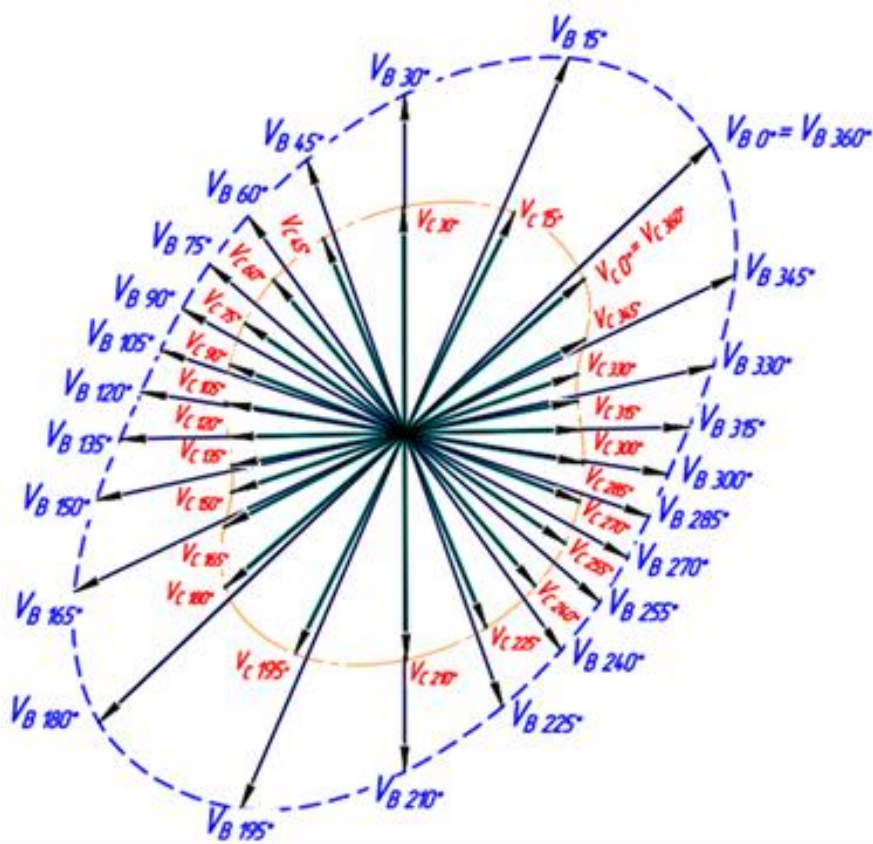


Figure 6. Hodographs of velocity vectors of points B and C.

Since point  $B$  also moves relative to point  $O_1$ , its acceleration  $a_B$  is the sum of the normal acceleration  $a_B^n$  and tangential acceleration  $a_B^\tau$ . Therefore, equation (6) is rewritten as:

$$\vec{a}_B^n + \vec{a}_B^\tau = \vec{a}_B^n + \vec{a}_{BA}^n + \vec{a}_{BA}^\tau \tag{7}$$

Normal acceleration  $a_B^n$  is directed along the segment  $BO_1$  from point  $B$  to point  $O_1$  and is defined as:  $a_B^n = BO_1 \cdot \omega_B^2 = 0,182 m \cdot (8,22 s^{-1})^2 = 12,3 m/s^2$ .  $a_{BA}^\tau \perp BO_1$  but its direction is still unknown.

The lengths of the segments representing the vectors  $a_{BA}^n$  and  $a_B^n$  through the scale factor of accelerations are determined:

$$\bar{a}_{BA}^n = \frac{a_{BA}^n}{\mu_a} = \frac{7,921m/s^2}{0,317 \left[ \frac{m/s^2}{mm} \right]} = 24,98mm;$$

$$\bar{a}_B^n = \frac{a_B^n}{\mu_a} = \frac{12,3m/s^2}{0,317 \left[ \frac{m/s^2}{mm} \right]} = 38,8mm.$$

$a_{BA}^n = 24,98 \text{ mm}$  is marked off from the end of the segment  $\overline{\pi a}$  and a perpendicular to this segment is drawn. Next, from the point  $\pi$ , we mark off  $\bar{a}_B^n = 38,8 \text{ mm}$  and a perpendicular to this segment. The point of intersection of the perpendiculars is marked with letter  $b$  and is connected to the point  $\pi$ . In accordance with formula (7), the segment  $\overline{\pi b}$  is the vector of full acceleration  $a_B$  in point  $B$  (Figure 5, c).

The acceleration diagram shows the directions of tangential accelerations  $a_B^\tau$  and  $a_{BA}^\tau$ . Point  $a$  on the acceleration diagram is connected to point  $b$  and vector  $\overline{ab}$  is obtained, which represents the overall relative acceleration  $a_{BA}$ .

We calculate the numerical values of the accelerations  $a_B$ ,  $a_B^\tau$ ,  $a_{BA}^\tau$ ,  $a_{BA}$  for the investigated position:

$$a_B = \mu_a \cdot \overline{\pi b} = 0,317 \left[ \frac{m/s^2}{mm} \right] \cdot 38mm = 12,046m/s^2;$$

$$a_B^\tau = \mu_a \cdot \bar{a}_B^\tau = 0,317 \left[ \frac{m/s^2}{mm} \right] \cdot 1,2mm = 0,38m/s^2;$$

$$a_{BA}^\tau = \mu_a \cdot \bar{a}_{BA}^\tau = 0,317 \left[ \frac{m/s^2}{mm} \right] \cdot 1,01mm = 3,2m/s^2;$$

$$a_{BA} = \mu_a \cdot \bar{a}_{BA} = 0,317 \left[ \frac{m/s^2}{mm} \right] \cdot 23,0mm = 7,291m/s^2;$$

We calculate the angular acceleration of links  $OB_1$  and  $AB$  for a given position using the formula well-known from theoretical mechanics:

$$\varepsilon_{BO_1} = \frac{a_B^\tau}{BO_1} = \frac{3,8m/s^2}{0,182m} = 2,01s^{-1};$$

$$\varepsilon_{BA} = \frac{a_{BA}^\tau}{BA} = \frac{3,2m/s^2}{0,10m} = 32,0s^{-1}.$$

The direction of angular accelerations  $\varepsilon_{BO_1}$  and  $\varepsilon_{BA}$  is determined by the conceptual transfer of vectors  $\vec{a}_B^\tau$  and  $\vec{a}_{BA}^\tau$  from the acceleration diagram to point  $B$  of the mechanism diagram (Figure 5,  $a$ ).

We find the acceleration of point  $C$ . Information on the tendency of changes in acceleration of point  $C$  depending on its position is of particular interest when choosing the shape of the guide track, taking into account the load on its walls acting from the roller during drum operation. This acceleration can be determined from the proportionality principle of the sides of similar figures in the mechanism diagram and the acceleration diagram. To do this, we calculate the segments

$$\overline{ac} = AC \frac{AB}{ab} \text{ and } \overline{bc} = BC \frac{AB}{ab},$$

and describe them as the radii of a circle around points  $a$  and  $b$ . At the intersection of these circles we find two points  $c$ . The correct one is the point  $c$ , which lies within the contour of the acceleration diagram. We connect the selected point with a point  $\pi$  and get a segment  $\overline{\pi c}$ , representing the acceleration vector  $a_c$ . In this position of the mechanism  $\overline{\pi c} = 30.6 \text{ mm}$ . Let's calculate the numerical value of the acceleration of point  $C$ :

$$a_c = \mu_a \cdot \overline{\pi c} = 0,317 \left[ \frac{m/s^2}{mm} \right] \cdot 30,6 \text{ mm} = 9,7 \text{ m/s}^2.$$

### 3. Results and Discussions

Thus, all the graphic and numerical values of kinematic parameters of characteristic points of the elliptical drum mechanism links are obtained for its one position. The proposed method can be used to determine with a sufficiently high accuracy the kinematic parameters of any position of the mechanism for its practical application in the design of promising vertical-spindle cotton harvesters.

Based on the research conducted, we would like to highlight the following results:

1. The cam mechanism of an elliptical drum can be replaced with a conventional hinged four-bar linkage without compromising the structure and mobility of the mechanism.
2. The developed method for determining velocity parameters of the spindles with a non-circular orbit allows carrying out a kinematic study of an elliptical drum.
3. The constructed hodograph of the velocities of characteristic points makes it possible to observe visually the changes in the spindle velocity depending on its location and to determine the acceleration directions of the spindle center and the guide roller.
4. The proposed method can be used to determine the linear and angular accelerations of the spindle and the guide roller, which is very important for the dynamic study of the mechanism of elliptical drum.
5. The obtained research results will undoubtedly prove useful for wide practical application of elliptical spindle drums in the structures of vertical spindle cotton harvesters.

### References

- [1] Turapov A.T. et al. The spindle drum of the cotton harvester. Inventor's certificate USSR № 1419582 // Discovery, invention, 1988, Bul. No. 32.
- [2] Turapov A.T., Akimbaev E.S., Ravutov Sh.T. Cotton drum reverse rotation shoe. Inventor's certificate USSR № // Discovery, invention, 1991, Bul. No. 40
- [3] Ravutov Sh.T. On the study of the structurally developed planetary mechanism of the vertical-

- spindle drum of the cotton harvester. Problems and prospects of innovative techniques and technology. // Resp. Sci.-Tech. conf. -Tashkent. Tashkent State Technical University, 2019, p. 114.
- [4] Ravutov Sh.T. Problems of increasing the efficiency of the vertical spindle cotton picker. // Problems of mechanics. Tashkent.-2019.-№4.-p. 51-55.
- [5] Ravutov Sh.T., Ulzhaev E., Turaev Kh. Grapho-analytical study of an elliptical drum of a vertical-spindle cotton harvester. International scientific and practical conference on: "Modern problems of innovative development of science, education and production". Andidzhan-2020, Uzbekistan. P.29-39.
- [6] Zhang, J., & Qin, L. (2019). A tunable frequency up-conversion wideband piezoelectric vibration energy harvester for low-frequency variable environment using a novel impact- and rope-driven hybrid mechanism. *Applied Energy*, 240, 26–34. doi:10.1016/j.apenergy.2019.01.261.
- [7] Tian, J., Zhang, X., Zhang, W., Li, J., Yang, Y., Dong, H., Jiu X., Yu Y., Zhao Z., Xu Sh., Zuo, W. (2018). Fiber damage of machine-harvested cotton before ginning and after lint cleaning. *Journal of Integrative Agriculture*, 17(5), 1120–1127. doi:10.1016/s2095-3119(17)61730-1.
- [8] Amanov, A., Bahadirov, G., Amanov, T., Tsoy, G., & Nabiev, A. 2019 Determination of Strain Properties of the Leather Semi-Finished Product and Moisture-Removing Materials of Compression Rolls. *Materials*, 12(21), 3620. doi:10.3390/ma12213620.
- [9] Bakhadirov G.A. Khusanov K. 2020 Construction of automatic lines connecting several mechanical operations in processing leather semi-finished product. *International Journal of Psychosocial Rehabilitation*. Volume 24 - Issue 4, 2079-2090. doi: 10.37200/IJPR/V24I4/PR201318.
- [10] Abdazimov A.D., Sadridinov A.S., Tulaev A.R. Phase discrete modeling of harvesting apparatus processes with controlled movement of cotton bolls. Tashkent: Nat. Library of Uzbekistan named after A. Navoi, 2011, 180 p.
- [11] Kh. Turanov, A. Abdazimov, M. Shaumarova, S. Siddikov. Incorrect application of the epicycloid equation to the planetary mechanism of the cotton harvester. *E3S Web of Conferences* 157, 01008 (2020) KTTI-2019.
- [12] Kh. Turanov, A. Abdazimov, M. Shaumarova, S. Siddikov. Mathematical modeling of a multiloop coulisse mechanism of a vertical spindle cotton harvester. *International Scientific Energy Management of Municipal Facilities ...* P. 306-321.
- [13] Frolov. K.V. et al. *Theory of mechanisms and mechanics of machines*. Textbook for technical colleges. - M.: Higher school, 2001. pp. 442-444.
- [14] Baranov G.G. *Course of the theory of mechanisms and machines*. Teaching aid. M.: "Mechanical engineering", 1975, pp. 31-32.
- [15] Drong V.I. et al. *Course of theoretical mechanics*. Textbook for universities. Moscow: Ed. Moscow State Technical University named after N.E. Bauman, 2005. P.90-95.
- [16] Dobronravov V.V., Nikitin N.N. *Course of Theoretical Mechanics: Textbook for Mechanical Engineering Special Universities - 4th ed.* - M.: Higher school, 1983. - pp. 98-102.
- [17] Levitsky, N.I. et al. (1974). *Theory and application of gear-lever mechanisms*, pp. 5-8, pp. 135-136, Ed. By N.I. Levitsky. M.: Nauka, -140p.
- [18] Fateev, N.A. Determination of angular velocities and angular accelerations of links of flat gear-lever mechanisms. *Proceedings of the XXXVIII scientific and technical conference*. Volume one. Technology. Stavropol: Sev.Kav.GTU, 2009-218p., Wwww.ncstu.rue.
- [19] Karimov. R.I., (2012). On one method of controlling the kinematic characteristics of the slider of the cam-lever mechanism. *Applied problems of mathematics and mechanics*. Proceedings of the XX International Scientific and Technical Conference. Sevastopol, September 10-14, 2012. SevNTU, 2012, pp. 172-176.

- [20] Abdukarimov A., Bakhadirov G. Graphic-analytical study of tooth-lever differential gear mechanisms. *International journal of modern manufacturing technologies*. Vol. V, No.1 / 2013.
- [21] Zheng, Q. J., & Yu, A. B. (2015). Modelling the granular flow in a rotating drum by the Eulerian finite element method. *Powder Technology*, 286, 361–370. doi:10.1016/j.powtec.2015.08.025.
- [22] Bahadirov, G., Sultanov, T., Umarov, B., & Bakhadirov, K. 2020 Advanced machine for sorting potatoes tubers. *IOP Conference Series: Materials Science and Engineering*, 883, 012132. doi:10.1088/1757-899x/883/1/012132.