# Optimization of the operation of the mechanism of the cross planer

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**Abstract.** The subject of the research is the optimization of the mechanism of the cross planer. In the article, on the basis of the condition of closed contours, the equations of the projection of the links on the corresponding coordinate axes are compiled. A functional relationship has been established between the kinematic parameters characterizing the movement of the input and output links of the mechanism. The obtained numerical calculations are presented in the form of graphic dependences. The aim of the work is the optimal metric synthesis of the mechanism of a cross-planer, with parametric restrictions. The conclusions obtained in the article can be used to improve and design modern technological equipment with mechanisms similar to the mechanism of a cross-cutting machine.

## 1 Introduction

Among the many priority areas for the development of Uzbekistan, adopted by presidential decree for 2022-2026, the key goal is to increase GDP per capita by 1.6 times, due to high economic growth rates, including mining, agriculture, and most importantly, engineering. In the implementation of this task, a special place is played by the creation of modern technological equipment. A large group of this equipment consists of grooving and planing machines, which are used in machine shops, at all large factories in Uzbekistan, including a mining and smelting plant in the city of Almalyk, a mining and smelting plant in the city of Navoi, an aggregate plant for agricultural machinery and a mechanical plant in the city of Tashkent, as well as in small-scale production. Today, machine tools are imported from other countries. The supply of cross planing machines to Uzbekistan is carried out by the UZSTANEX company, which is part of the StanexGroup holding. However, in the future, with an increase in trade turnover, and the output of production to a new level, Uzbekistan will need a large amount of machine tools, and its own plant for their production. Complete independence from suppliers will give our Republic a solid basis for raising its competitiveness. Starting with Leonardo Davinci, many scientists and engineers are engaged in the creation and improvement of planing machines. At the present stage, research in this area is carried out by [1-8]. This article is devoted to the metric synthesis of the mechanism of a cross-planing machine, and the optimization of its work by determining the optimal dimensions of the links.

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## 2 Objects and methods of research

Slotting and planning machines are designed for processing horizontal, inclined, vertical and shaped surfaces, cutting grooves and grooves. Cross planers are classified according to the type of main drive. They are hydraulic, gear, crank and rocker.

In the metric and kinematic analysis and synthesis of any mechanism, a schematic - kinematic scheme of the mechanism is used. If a specific mechanism is given, then the diagram is drawn to scale. Since in the article, the optimal link sizes are selected, the machine diagram is drawn without scale.

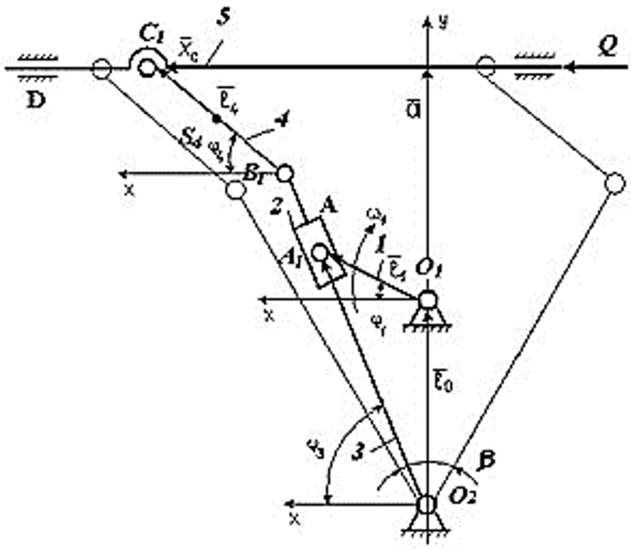


**Fig. 1.** Cross planer mechanism

Figure 1 shows the kinematic diagram of a cross planer with a rocker drive mechanism. The mechanism consists of racks 0, crank 1, connecting rods 4.6, rocker-scenes 2, and wings 3, 5. The machine is powered by an electric motor that transmits the rotation of a six-speed gearbox to a shaft equipped with a sliding dowel. The shpanka holds a triple block of gears, which guarantees a three-speed mode to the shaft. Consequently, during the movement of the block along the axis of the shaft, the gears are alternately captured with the gears fixed on the key on the shaft [1]. The work of the cross-planing machine (Fig. 1.) is as follows. The cutter performs a horizontal reciprocating motion with speeds Vr.x (working stroke) and Vx.x (idle). This movement is characterized by the number of double strokes per minute of the slider. One double stroke consists of a working stroke, in which the cutter cuts off a layer of metal with a section f = ts = ab mm 2, and an idle stroke, in which the cutter returns to its original position [2].

The main geometric characteristic of a cross planer is the slider stroke, since the dimensions of the processed product directly depend on this. When creating new mechanisms and improving the operation of existing ones, the designer takes into account not only the performance and strength of the mechanism as a whole, but also the interaction of its nodes in the process of operation, for which kinematic analysis is usually used. The values of the kinematic characteristics of the mechanism (displacement, speed, acceleration) are needed both to determine the position of the mechanism and for subsequent dynamic research.

To cut a new layer of metal, a feed is made with a new working stroke. The movement of the cutter is the main movement for the planer, and the feed movement is the movement of the workpiece in the transverse direction. The rocker mechanism serves to convert the rotational movement from the box into the rectilinear reciprocating movement of the slider. The backstage swings and informs the slider of an uneven travel speed. The speed of the slider varies from zero to the maximum, in the middle position of the wings the slider reaches the maximum speed, and at the extreme positions of the wings the speed is zero. Since the idling of the slider has a much higher speed than the working one, this saves time on unproductive idling of the machine. The gearbox has steps of double stroke numbers with large differences. All this makes it difficult to select the optimal cutting mode. The rocker mechanism makes it possible to set the stroke length of the slider depending on the length of the workpiece, but the mechanism itself thereby limits the stroke length.

Kinematic studies of the mechanism are carried out by two methods: graphical and analytical. The analytical method is used in cases where high accuracy in determining the displacements, velocities and accelerations of the points of the mechanism links is required. Using modern programming methods, both a simple and a complex task of studying multi-link mechanisms can be solved. In turn, the analytical method is conditionally divided into the method of closed vector contours, which was proposed by V.A. Zinoviev and the coordinate transformation method proposed by Yu.F. Moroshkin.

To compile analytical dependencies, we will use the condition of closedness of the contours of their kinematic chains, since it is more convenient for flat mechanisms. Composing the projection equations of the links on the corresponding coordinate axes, they establish a functional relationship between the kinematic parameters characterizing the movement of the input and output links of the mechanisms.

The six-link mechanism depicted in Figure 1 consists of a crank 1, a rocker 2, a swinging rocker 3, a connecting rod 4 and a slider 5 that reciprocates relative to the rack 6. The initial link is a crank that rotates with an angular velocity $\omega_1$. The crank is also a leading link, since it has a generalized coordinate-angle $\varphi_1$. We choose a rectangular coordinate system XOY, the origin of which coincides with the center of the hinge $O_1$, the X axis passes through the points $O_2$ and B, and the Y axis is drawn parallel to the movement of the slider 5. The angles $\varphi_1$, $\varphi_3$ and $\varphi_4$ are counted from the positive direction of the X axis in the direction of rotation of the crank 1. We write down the condition for the closedness of the contour, composed of the vectors $l_{AB}$, $l_{AC}$ and $l_{BC}$ links 1, 6 and 3 in the form of a vector equation:

$$l_{AB} = l_{AC} + l_{BC}$$
$$E_B sin\varphi_3 = l_1 sin\varphi_1 + l_0$$
$$E_B cos\varphi_3 = l_1 cos\varphi_1 + l_1$$
$$l_0 + \varphi = l_4 sin\varphi_4 + l_3 sin\varphi_3$$
$$x_D = l_4 cos\varphi_4 + l_3 cos\varphi_3$$
$$l_3 cos\varphi_4 = l_1 cos\varphi_1 + (l_3 - EB) cos\varphi_3$$
$$cos\varphi_4 = \frac{l_1 cos\varphi_1}{EB} - EB^2 = l_o^2 + l_1^2 - 2l_1 l_0 cos(90 + \varphi_1) = l_o^2 + l_1^2 + 2l_1 l_0 sin\varphi_1$$
$$EB = \sqrt{l_o^2 + l_1^2 + 2l_1 l_0 sin\varphi_1}$$
$$cos\varphi_3 = \frac{l_1 cos\varphi_1}{\sqrt{l_o^2 + l_1^2 + 2l_1 l_0 sin\varphi_1}}$$
$$X_D = l_4 cos\varphi_4 + \frac{l_3 l_1 cos\varphi_1}{\sqrt{l_o^2 + l_1^2 + 2l_1 l_0 sin\varphi_1}}$$
$$sin\varphi_3 = \sqrt{1 - \frac{l_1^2 cos\varphi_1}{l_o^2 + l_1^2 + 2l_1 l_0 sin\varphi_1}}$$

Equation differential: 3 and 4 in $\varphi_1$ we get

$$0 = l_4 cos\varphi_4 \cdot d\varphi_4 + l_3 cos\varphi_3 d\varphi_4$$

$$dx_D = -l_4 \cdot sin\varphi_4 d\varphi_4 - l_3 sin\varphi_4 \cdot d\varphi_4$$

$$0 = U_{43} l_4 cos\varphi_4 + l_3 cos\varphi_3$$

$$\frac{dx_D}{d\varphi_3} = -l_4 \cdot U_{43} l_4 sin\varphi_4 - l_3 sin\varphi_4 = U_{53}$$

$$U_{43} = \frac{l_3 cos\varphi_3}{l_4 cos\varphi_4} = \frac{l_3 l_1 cos\varphi_1}{\sqrt{l_o^2 + l_1^2 + 2l_1 l_0 sin\varphi_1}} \cdot \frac{1}{\sqrt{l_4 - \left(l_0 + a - l_3 \cdot \sqrt{1 - \frac{l_1^2 cos^2\varphi_1}{l_0^2 + l_1^2 + 2l_1 l_0 sin\varphi_1}}\right)^2}}$$

$$U_{53} = -l_4 \cdot U_{43} \frac{\left(l_0 + a - l_3 \sqrt{1 - \frac{l_1^2 cos^2\varphi_1}{l_0^2 + l_1^2 + 2l_1 l_0 sin\varphi_1}}\right)}{l_4} - l_3 \cdot \sqrt{1 - \frac{l_1^2 cos^2\varphi_1}{l_0^2 + l_1^2 + 2l_1 l_0 sin\varphi_1}}$$

$$sin\varphi_4 = \frac{(l_0 + a - l_3 cos\varphi_3)}{l_4} = \left(\frac{l_0 + a - l_3 \sqrt{1 - \frac{l_1^2 cos^2\varphi_1}{l_0^2 + l_1^2 + 2l_1 l_0 sin\varphi_1}}}{l_4}\right)$$

$$cos\varphi_4 = \sqrt{1 - sin^2\varphi_4} = \sqrt{1 - \left(\frac{l_0 + a - l_3 \sqrt{1 - \frac{l_1^2 cos^2\varphi_1}{l_0^2 + l_1^2 + 2l_1 l_0 sin\varphi_1}}}{l_4}\right)}$$

$$X_D = l_4 cos\varphi_4 + l_3 cos\varphi_3$$
$$= l_4 \cdot \sqrt{1 - \left(\frac{l_0 + a - l_3 \sqrt{1 - \frac{l_1^2 cos^2\varphi_1}{l_0^2 + l_1^2 + 2l_1 l_0 sin\varphi_1}}}{l_4}\right)^2} + \frac{l_3 \cdot l_1 cos\varphi_1}{l_0^2 + l_1^2 + 2l_1 l_0 sin\varphi_1}$$

at $l_1$; $l_0$; $l_3$; $l_4$; $a$

Divide the second equation by the first and find:

$$tg\ \varphi_3 = \frac{l_1 sin\varphi_1 + l_0}{l_1 cos\varphi}$$

The resulting equation is differentiable with respect to $\varphi_1$ and get

$$\frac{d\varphi_3}{d\varphi_1} = \frac{cos^2\varphi_3}{l_1^2 cos^2\varphi_1} \cdot (l_1^2 cos^2\varphi_1 + l_1^2 sin^2\varphi_1 + l_0 l_1 sin\varphi_1) = \frac{cos^2\varphi_3}{l_1^2 cos^2\varphi_1} \cdot (l_1^2 + l_0 l_1 sin\varphi_1) =$$
$$= \frac{l_1^2 cos^2\varphi_1}{l_1^2 cos^2\varphi_1} \cdot \frac{(l_1^2 + l_0 l_1 sin\varphi_1)}{EB^2} = \frac{l_1^2 + l_0 l_1 sin\varphi_1}{l_0^2 + l_1^2 + 2l_1 l_0 sin\varphi_1} = U_{31}$$

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The given trigonometric equations are the basis for the analytical method of kinematic analysis of the mechanism of a cross-cutting machine. Having compiled an algorithm and a calculation program in the Mathcad computer environment [9, 12], the authors of the article obtained a mathematical model of the dependence of the change in intermediate gear ratios depending on the length of the links of the mechanism.

## 3 Results and their discussion

The performed calculations showed the following results:

1. After analyzing the graph of the dependence of the change in the gear ratio of the crank - the link $U_{31}$ on the length of the crank (Figure 2), the authors of the article concluded that the optimal values of the crank are in the range from 250 to 300 millimeters, since it is here that the maximum values of the gear ratio $U_{31}$ occur. Calculations are made for one revolution of the crank.

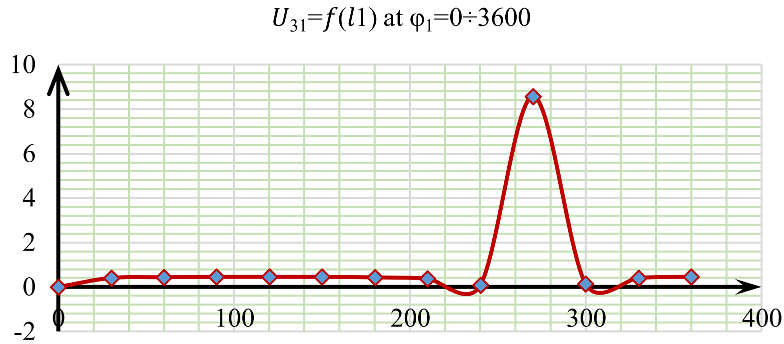


**Fig. 2.** $U_{31}$ gear ratio versus crank length, per revolution

2. According to the graph, in Figure 3, the dependence of the change in the gear ratio of the crank - the link $U_{31}$ on the distance $l_0$ (Figure 1), we can conclude that the optimal distance between the points $O_1$ (the place where the crank is attached to the rack) and $O_2$ (the place where the link is attached to the rack) are in range from 275 to 325 millimeters.

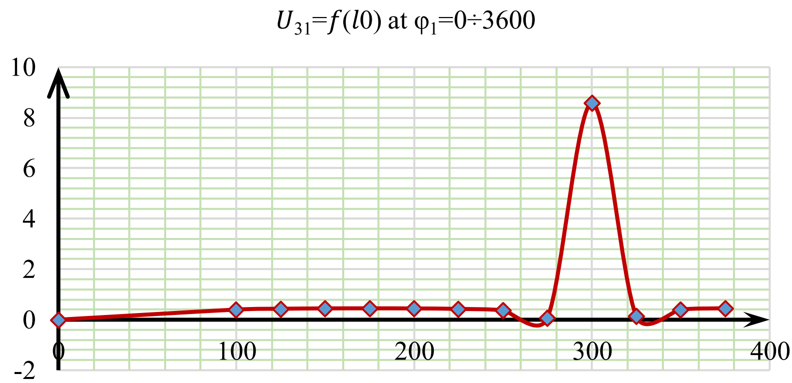


**Fig. 3.** Graph of the change in gear ratio $U_{31}$ from the distance $l_0$.

3. Figure 4 shows a graph of the change in gear ratio $U_{43}$ (link - connecting rod) on the length of the connecting rod. When calculating, the distance from point $O_1$ (the point of

connection of the crank with the rack) to the line of action of the cutter - DQ, will be considered constant, equal to $l_{35}$ millimeters. The graph is drawn for one revolution of the connecting rod. The graph shows that the optimal value, the gear ratio reaches when the connecting rod length is from 80 to 180 millimeters.

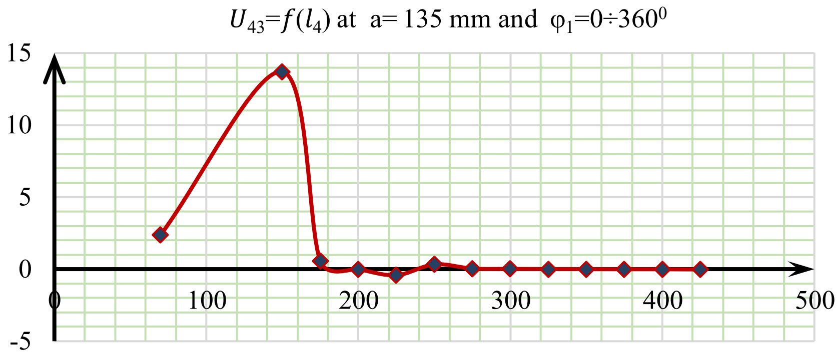


**Fig. 4.** Graph of the change in gear ratio $U_{43}$ (scene - connecting rod) on the length of the connecting rod.

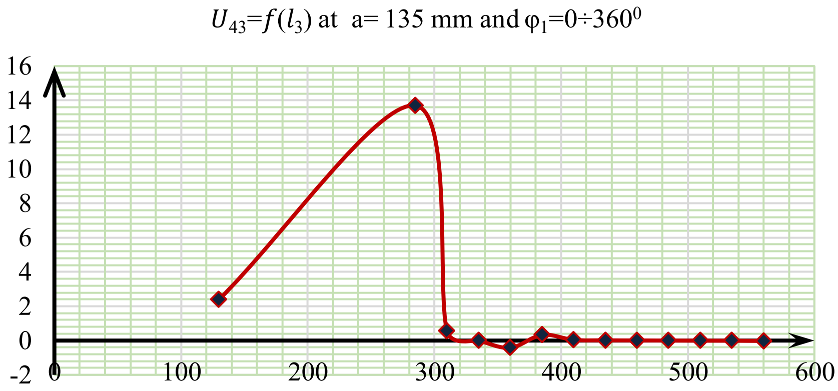


**Fig. 5.** Graph of the change in gear ratio $U_{43}$ (link - connecting rod) on the length of the link.

4. Figure 5 shows a graph of the change in gear ratio $U_{43}$ (link - connecting rod) on the length of the link. When calculating, the distance from point $O_1$ (the point of connection of the crank with the rack) to the line of action of the cutter - DQ, will be considered constant, equal to $l_{35}$ millimeters. The graph is drawn for one revolution of the connecting rod. It can be seen from the graph that the gear ratio takes on the optimal value for backstage sizes from 130 to 310 millimeters.

5. Figure 6 shows a graph of the change in gear ratio $U_{53}$ (link - slider) on the length of the connecting rod. When calculating, the distance from point $O_1$ (the point of connection of the crank with the rack) to the line of action of the cutter - DQ, will be considered constant, equal to $l_{35}$ millimeters. The graph is drawn for one revolution of the connecting rod. It can be seen from the graph that the gear ratio takes on an optimal value with connecting rod sizes from 100 to 180 and from 340 to 375 millimeters. Comparing the results obtained with the graph in Figure 2, we leave the range of connecting rod values from 100 to 180.

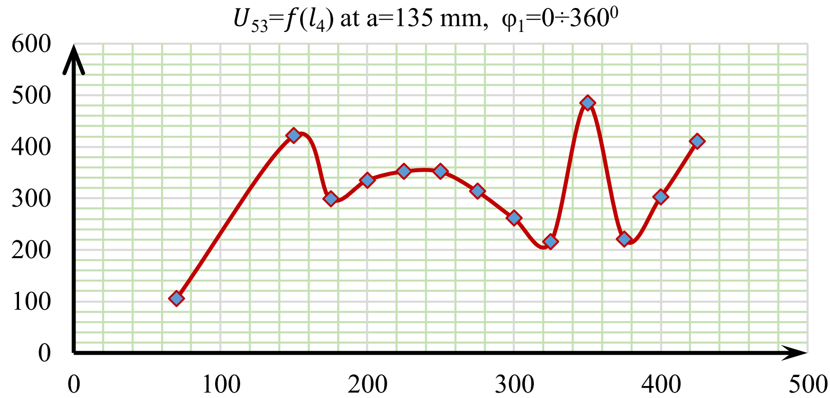


**Fig. 6.** Graph of the change in gear ratio U53 (scene - slider) on the length of the connecting rod.

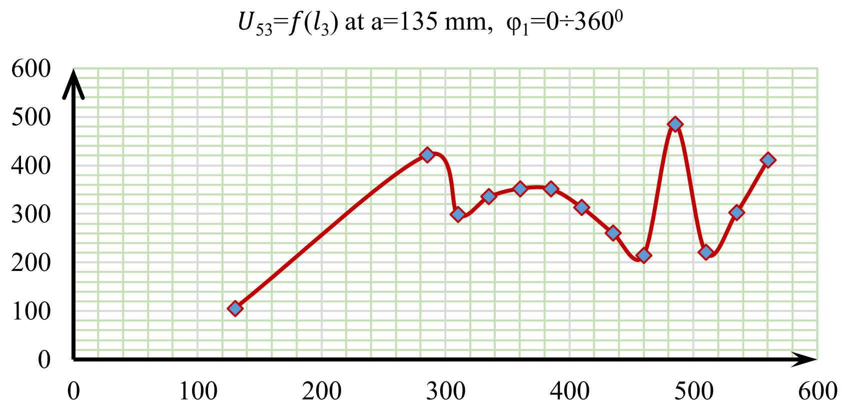


**Fig. 7.** Graph of the change in gear ratio U43 (link - connecting rod) on the length of the link.

6. Figure 7 shows a graph of the change in gear ratio $U_{53}$ (link - slider) on the length of the link. When calculating, the distance from point $O_1$ (the point of connection of the crank with the rack) to the line of action of the cutter - DQ, will be considered constant, equal to $l_{35}$ millimeters. The graph is drawn for one revolution of the connecting rod. It can be seen from the graph that the gear ratio takes on the optimal value for backstage sizes from 200 to 310 and from 460 to 510 millimeters. Comparing the results obtained with the graph in Figure 4, we leave the range of connecting rod values from 200 to 310 millimeters.

## 4 Conclusions

A mathematical model of work and analytical study of the kinematic analysis of the mechanism of a cross-cutting machine has been developed. In the work, mathematical expressions are obtained that describe the movement of the output - working link, in the form of functions of the rotation angles of the input and intermediate links.

Calculations of the optimal dimensions of the crank, backstage and connecting rod, the mechanism of the cross-planer, for a given value of the distance between the line of movement of the cutter and the place of attachment of the crank to the rack, using the MathCAD application program.

The prospect of improving existing machines, which include rocker mechanisms similar to a cross-cutting machine, and the invention of new ones are proposed.

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