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Kinematic analysis and synthesis of cutter movement of slotting machine

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Abstract

The kinematic characteristics of the links and individual points of the mechanism are determined by the method of closed geometric contours and the method of designing plans. The forces of interaction between the links of the mechanism are determined by the method of kinetostatics, and the balancing moment by considering the dynamic equilibrium of the crank and the method of power balance. The drawbacks of the structural scheme of the mechanism are revealed and the ways of their elimination are offered.

The results of research are presented in the form of graphical dependences of the kinematic parameters of the cutter, reactions or hodographs of forces in kinematic pairs. The moments of resistance forces and inertia forces are determined and the dynamic and mathematical models of the movement of the mechanism are constructed. The technology of determining the power of the electric motor is shown, and its stable area of operation is approximated by a straight line.

Kinematic synthesis was performed and a modernized mechanism was obtained in which the cutter can move according to a predetermined law, in particular, without soft shocks at the boundaries of the kinematic cycle and with quasi-constant speed (error up to 5%) in the middle of the kinematic cycle.

Key words: linkage mechanisms, optimization, synthesis, kinematic analysis, law of motion, cams

Introduction

Slotting machines are used both in serial production and in repair shops to obtain grooves, flat and shaped surfaces of small height, but significant transverse dimensions, through and blind holes and cavities. The main mechanism of slotting machines is a rocker with a two-lead group attached to it, the slide of which moves in a vertical plane. The productivity of the machine is limited by the planing speed (70-80 m/min), due to the reciprocating movements of the slide with which the cutter is rigidly connected. The movement of the cutter on the working stroke is not even, which degrades the quality of the planing surface, the durability of the cutter. The presence of cutter acceleration jumps at the edges of the kinematic cycle causes soft impacts, which worsens the dynamic state of the mechanism as a whole and, as a result, further impairs the quality of the planing surface.

Literature review

Analysis of modern publications in the papers on Mechanism and Machine Science showed that the existing methods for the synthesis and analysis of mechanisms of slotting machines, both analytical and numerical, do not provide exact parameters to synthesize multi-link linkage mechanisms that could be characterized by no acceleration jumps at the boundaries of the kinematic cycle and quasi-constant velocity. cycle [1-3], including a number of modern works on this topic [8-10].

In particular, in the work [1] the methods of synthesis of linkage and cam mechanisms using unified subroutines to calculate the main kinematical parameters of linkage mechanisms are shown. It is also considered the problems of kinematic synthesis of linkage mechanisms, synthesis and analysis of cam mechanisms: a number of examples of the usage of the mathematical package Mathcad for this purpose are also provided.

Additionly, in our opinion, in the methods that are described in the existing works [1-3], there are some drawbacks that do not allow to conduct correct calculations for the entire range of initial data and according to all the neccesary requirements of the designer. For example, no analytical dependences are given for all variants of the angles of inclination of vectors of kinematic quantities and reactions in kinematic pairs.

Several calculation schemes of the structural groups are described not in the most general form. For example, for the structural group of the 2nd type, when the position of the slider will be in different quadrants, the angles of the shuttle without additional calculations can not be determined. Using modern CAD-systems, it is possible to solve numerically many problems of kinematic and dynamic analysis of technical systems, as shown in [10], but the problems of the kinematic synthesis, taking into account a number of additional criterai, require special methods, which is the subject of this publication. In this paper we will use some methods that were previously described in [4-7].

Purpose

The aim of the research is to analyze the kinematic state of the slotting machine mechanism and on its basis to suggest the ways to improve the dynamic parameters of the mechanisn and the quality of the planing surface; to synthesize such displacement of the additional slide (fixed cam profile) relative to the coupling link, where the cutter will not have soft impacts at the boundaries of the kinematic cycle and move at a constant velocity in the middle.

The kinematic analysis

The scructural scheme of the mechanism is shown in the Fig. 1 and contains: crank *OA* – 1, rocker slider – 2, coupling link (rocker) *ABC* – 3, shuttle *CD* – 4 and slider DD_1 – 5 to which the cutter is perpendicularly attached and which is affected by the planing force F_p .

The analysis is carried out on the basis of the structural qualification of mechanisms according to Assur, where structural formula for constructing the mechanism is as follows:

$$I(0-1)\rightarrow II(2-3)\rightarrow II(4-5) \tag{1}$$

Kinematic analysis is performed from the beginning of the structural formula according to the dependences obtained in [1-3]. We connect the right coordinate system *xOy* with the center of rotation of the crank. We consider the frequency of rotation of the crank and the geometric characteristics of the links as a known one.

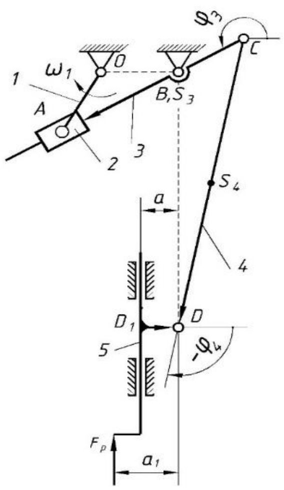

Fig.1. Structural scheme of the mechanism

Determination of the kinematic characteristics of the mechanism is carried out in groups from the beginning of the structural formula (1). Crank *OA*.

$$v_A=\omega_1 l_{OA};\ \gamma_a=\varphi_1-\pi,\ a_A=\omega_1^2 l_{OA},\ \psi_a=\varphi_1+\pi,,$$

where v_A i γ_a – modulus and angle of inclination of the velocity vector $\vec{v}_A$ to the abscissa, a_A i ψ_a – modulus and angle of inclination of the acceleration vector $\vec{a}_A$ to the abscissa.

Structural Group ABC. Calculate the kinematic characteristics of the rocker *AB*:

$$x_B=l_{OB};\ y_B=0;\ l_{AB}=\sqrt{(x_A-x_B)^2+(y_A-y_B)^2},\ \varphi_3=\text{arctg}\left(\frac{y_A-y_B}{x_A-x_B}\right);$$

$$\omega_3=v_A\sin(\gamma_A-\varphi_3)/l_{AB},\ v_{A_3A}=-v_A\cos(\gamma_A-\gamma_3);$$

$$\varepsilon_3=\left[a_A\sin(\psi_A-\varphi_3)+2\omega_3 v_{A_3A}\right]/l_{AB},\ a_{A_3A}=-a_A\cos(\psi_A-\varphi_3)-\omega_3^2 l_{AB}.$$

Calculate the kinematic characteristics of the kinematic pair *C*:

$$x_C = x_B + l_{BC}\cos(\varphi_3+\pi), y_C = y_B + l_{BC}\sin(\varphi_3+\pi);\;\; l_C = \sqrt{x_C^2 + y_C^2},\; \varphi_C = \mathrm{arctg}(y_C / x_C);$$
$$v_C = l_{BC}\left|\omega_3\right|,\; \gamma_C = \varphi_3 + \pi + 0{,}5\pi\cdot\sin(\varphi_3),\;\; a_C = l_{BC}\sqrt{\omega_3^4+\varepsilon_3^2},\; \psi_C = \varphi_3 - \mathrm{arctg}(\varepsilon_3/\omega_3^2).$$

Structural group CD. ξ – the angle of the guide *5* to the abscissa, $x_B=1$ i $y_B=0$ – sign of the point of intersection of the perpendicular dropped from the beginning of the coordinate system on the guide of the slider, $ze = \mathrm{sign}\left\{\mathrm{sgn}\left[y_E\cos(\xi)\right] - \mathrm{sgn}\left[x_E\sin(\xi)\right]\right\}$,

$$\varphi_4 = \arcsin\left[e\cdot ze + a - l_C\sin(\varphi_C-\xi)\right]/l_4 + \xi,\; e = l_{OB} - a;$$
$$y_D = l_C\sin(\varphi_C) + l_4\sin(\varphi_4) - a,\; s_D = y_{D\max} - y_D;$$
$$\omega_4 = \frac{v_C\sin(\xi-\gamma_C)}{l_4\cos(\varphi_4-\xi)},\; v_D = \frac{v_C\cos(\varphi_4-\gamma_C)}{\cos(\varphi_4-\xi)},\; a_D = \frac{a_C\cos(\psi_C-\varphi_4) - \omega_4^2 l_4}{\cos(\varphi_4-\xi)}$$
$$\varepsilon_4 = \frac{-a_C\sin(\psi_C-\xi) + \omega_4^2 l_4\sin(\varphi_4-\xi)}{l_4\cos(\varphi_4-\xi)},\; \gamma_D = \xi + \frac{\pi\left[1-\mathrm{sgn}(v_D)\right]}{2},\;\; \psi_D = \xi + \frac{\pi\left[1-\mathrm{sgn}(a_D)\right]}{2}.$$

The analysis was performed for such parameters of the mechanism:

$n = 120$ rpm; $l_1 = 0.11$ m; $l_{OB} = 0.05$ m; $a = 0.01$ m; $b = 0.02$ m; $l_{BC} = 0.11$ m; $l_4 = 0.45$ m; $l_5 = a$; $e = l_{OB} - a$. (2)

The kinematic characteristics of the cutter are shown in the Fig. 2.

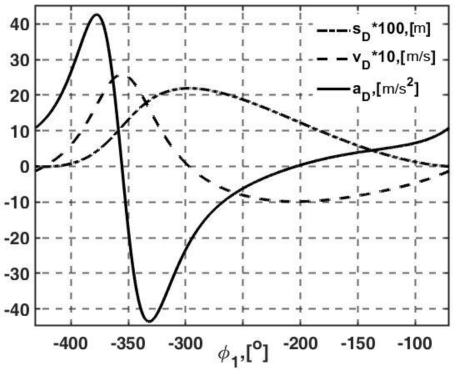

Fig. 2. Kinematic characteristics of the slider (cutter)

The results of the analysis confirm the fact that the linkage mechanisms do not provide movement of the moving link with a quasi-constant speed. The beginning and end of the planing process occurs with an acceleration jump, which causes a soft impact effect and leads to a deterioration of the dynamic state of the whole machine, its productivity and the quality of the planing surface. To avoid side effects, we need:

– to synthesize or use the known laws of periodic motion [2, 6, 7], in which there are no soft impacts, and in the middle of the kinematic cycle we have a section of quasi-constant velocity;

– to ensure the movement of the cutter according to the synthesized or selected law, replace one of the links of constant length with a link of variable length. In this case, the problem is reduced to the synthesis of such a variable length of the link at which the cutter will move according to the synthesized or selected law.

The kinematic synthesis

In the conclusions to [5] it is stated that the cutter on the working stroke has no interval of movements at a constant speed, and at the ends of the kinematic cycle acceleration is not equal to zero. Such kinematic characteristics of the cutter cannot be considered satisfactory, as soft impacts lead to a sudden action of inertia on the cutter, and variable speed degrades the quality of the planing surface. Thus, occurs the question, is it possible to modernize the mechanism to avoid such shortcomings?

It is known [1, 2] that the geometry of linkage mechanisms with one degree of freedom completely determines the qualitative kinematic characteristics of the links and it is impossible to change them. To improve the known structural scheme of the slotting machine (Fig. 3, a), it is proposed to add one more link in the kinematic chain in order to obtain a mechanism with two degrees of freedom (Fig. 3, b). The additional link of the slider *6* is connected to the shuttle 4 by a rotating kinematic pair *C* and by a translational C_3 to the slide 3. The roller *p* rotates around the axis of the pair *C* and rolls a fixed cam, which changes the length l_{BC}.

With such changes, the degree of freedom of the modernized structural scheme *W=2*. Kinematic synthesis includes choosing the law of motion of the slider *6*, according to which the cutter (same as point *D*) will move according to a predetermined law.

If the crank *1* is driven by an electric motor, the additional slider can be moved in different ways. At the stage of discussions step electric motors and cam mechanisms are considered. Today, the disadvantages of step motors are that changing the position of the shaft is not a smooth line, but stepped. In addition, you need another power supply and automated control system. The use of a cam mechanism (Fig. 3, c) allows theoretically to obtain any law of periodic motion, but it is necessary to obtain a profile of a fixed cam *k*, the pressure angles of which will not exceed the allowable. Since the cam mechanism has a higher kinematic pair, this may impose additional restrictions on its use.

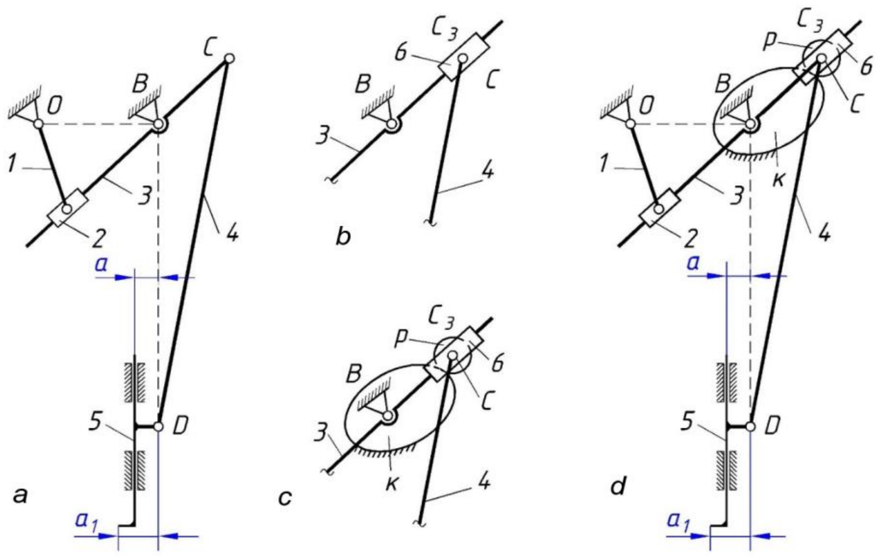

Fig. 3. The steps of modernization of the structural scheme of the mechanism of the slotting machine

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In our opinion, the use of the cam mechanism is simpler, cheaper and does not require special maintenance. Its use in various industries from printing engineering to internal combustion engines has proven its wide potential. The block diagram of the improved mechanism is shown in Fig. 3, d.

The essence of kinematic synthesis is to synthesize such a movement of the additional slide *6* (fixed cam profile *k*) relative to the backstage, at which the movement of the cutter 5 will occur without soft impacts at the boundaries of the kinematic cycle and at a constant speed in the middle

Theoretical grounds of kinematic synthesis

Consider links 3-5 and 6 separately (Fig. 4). Obviously, the point D and the cutter belong to the same body, and therefore the motion of the point *D* is also translational. The contour of the BC can be interpreted as a crank-shuttle mechanism (CSM) in which the length of the conditional car of the aircraft varies depending on the angle $\varphi_3' = \varphi_3 - \pi$; where φ_3 – the angle of rotation of the link *ABC* to the abscissa in the *xOy* coordinate system.

The input parameters of the synthesis, in addition to geometric dimensions, includes the law of motion s_D of the cutter (point *D*), which will achieve the objective of synthesis. The initial parameter of the synthesis is the radius vector of the fixed cam.

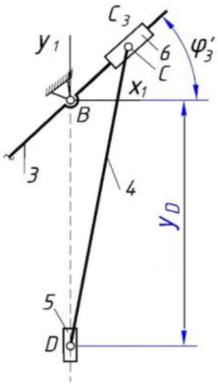
Denote the variable length of the conditional box $r_C \equiv l_{BC}$. Each position of point *C* must correspond to a specific given position of point *D*, and the distance between them must always be constant. This problem can be reduced to a purely mathematical one: to determine the coordinates of the point of intersection of the conditional crank of radius $r_C = \sqrt{x_C^2 + y_C^2}$ and a circle of radius l_{CD} conducted from a given point *D*:

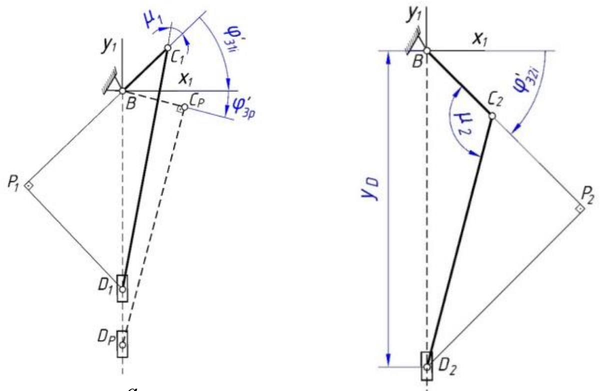
$$\left.\begin{aligned} x_C &= r_C \cos(\varphi_3'), \\ y_C &= r_C \sin(\varphi_3'), \\ x_C^2 + (y_C - y_D)^2 &= l_{CD}^2, \end{aligned}\right\}$$

where x_C, y_C – coordinates of point *C* in the coordinate system x_1By_1.

However, we will determine in a more visual way. We show the positions of links 3 and 4 in cases where the angle μ between the links *BC* and *CD* is acute, obtuse and equal to 90º (Fig. 5). In the Fig. 5, and shows the case when the angle μ_1 is acute and straight.

Variable radius of the conditional crank in the case of an acute angle μ_1 $r_C = C_1P_1 - BP_1$, where $C_1P_1 = \sqrt{l_4^2 - D_1P_1^2} = \sqrt{l_4^2 - \left[y_{D_1}\sin(\varphi'_{31}+\xi)\right]^2}$, $BP_1 = y_{D_1}\sin(\varphi'_{31}+\xi)$, $l_4 = l_{CD}$ – length, shuttle, $\xi = -90^\circ$ – the angle of the cutter guide to the abscissa Ox_1.





a *b*

Fig. 4. Structureal scheme of the combined mechanism with variable lwngth of the conditional crank

Fig. 5. For the determination of the length of the of the conditional crank *BC*

Therefore, for an acute angle, the radius is (Fig. 5, a):

$$r_C = y_{D_1}\sin(\varphi'_{31}+\xi) + \sqrt{l_4^2 - \left[y_{D_1}\sin(\varphi'_{31}+\xi)\right]^2}. \tag{3}$$

For a right angle $\mu = 90^\circ$ the radius is

$$r_C = y_{D_p}\sin\left(\varphi'_{3p}+\xi\right), \tag{4}$$

where y_{D_p} i φ'_{3p} – moving the point D and the angle of inclination *BC* in the position when $\mu = 90^\circ$.

For an obtuse angle μ_2 (Fig. 5, b) $r_C \equiv BC_2 = BP_2 - C_2P_2$, where $BP_2 = y_{D_2}\sin\left(\varphi'_3+\xi\right)$, $C_2P_2 = \sqrt{l_4^2 - \left[y_{D_2}\cos\left(\varphi'_3+\xi\right)\right]^2}$. The radius of the conditional crank for an obtuse angle:

$$r_C = y_{D_1}\sin\left(\varphi'_3+\xi\right) - \sqrt{l_4^2 - \left[y_{D_2}\cos\left(\varphi'_3+\xi\right)\right]^2}. \tag{5}$$

The obtained three dependences (3)-(5) are valid when the angle μ is obtuse, straight or acute. We apply the sign function and write the value of the variable length of the conditional crank for any angle φ'_3 of the coordinate of the point *D* - y_D:

$$r_C = y_D\sin\left(\varphi'_3+\xi\right) - \operatorname{sgn}\left(\varphi'_{3p}-\varphi'_3\right)\sqrt{l_4^2 - \left[y_D\cos\left(\varphi'_3+\xi\right)\right]^2}. \tag{6}$$

Obviously, the root expression cannot be negative. But if the expression is positive, then the trajectory of point *C* has a gap of the first kind, which also does not satisfy us. Therefore, additional studies of sub-root function were performed $z = l_4^2 - \left[y_D\cos\left(\varphi'_3+\xi\right)\right]^2$ and built its graphs for three cases $z > 0$, $z < 0$ i $z = 0$ (Fig. 6). It turned out that to synthesize the radius vector $\vec{r}_C$ possible only if in a mutually perpendicular position ($\mu = 90^\circ$) of links *BC* and *CD* function *z*=0. From this condition we determine the angle of rotation of the conditional crank φ'_{3p}, for which $\mu = 90^\circ$. In addition, another synthesized parameter is the length of the crank *CD*:

$$l_{4s} \equiv l_{CDs} = y_D(\varphi'_{3p})\cos\left(\varphi'_{3p}+\xi\right). \tag{7}$$

For the given geometrical characteristics, the conditional crank of *BC* and shuttle *CD* are mutually perpendicular when the angle of turn $\varphi'_3 = 77.79^\circ$. The length of the synthesized shuttle $l_{4s} = 0{,}4611$ m.

With this addition, the variable length of the conditional crank (6) is equal to

$$r_C = y_D\sin\left(\varphi'_3+\xi\right) - \operatorname{sgn}\left(\varphi'_{3p}-\varphi'_3\right)\sqrt{l_{4s}^2 - \left[y_D\cos\left(\varphi'_3+\xi\right)\right]^2}. \tag{8}$$

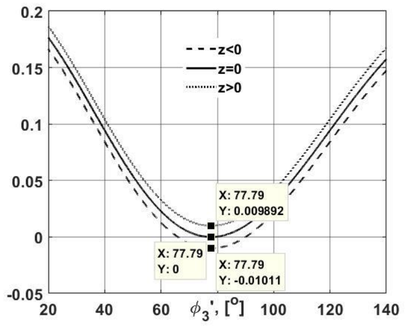


Fig. 6. To determine the synthesized length of the shuttle *CD*

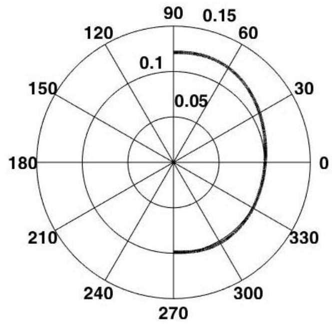


Fig. 7. The trajectory of point C of the synthesized mechanism at the stage of operation

Choice of the law of motion for the cutter

To eliminate the drawbacks of the mechanism of the slotting machine, it is necessary that the law of periodic motion (LPM) of the cutter at the edges of the stroke would not have acceleration jumps, and in the middle of the kinematic cycle there would be a constant velocity interval.

To choose a specific LPM of a cutter, we can use already known, or synthesize a new combined law. Among the known LPM [2, 6, 7] a large number of cycloid laws in which there is no acceleration at the boundaries. Among them are laws with quasi-constant velocity in the middle of the kinematic cycle.

The authors chose the law [6] that has the widest interval of quasi-constant velocity with 5% error. Invariant of moving this law:

$$a_k = (70k^3 - 245k^4 + 378k^5 - 280k^6 + 80k^7)/3, \tag{9}$$

where $k = t/T$ – dimensionless time, $0 \le t \le T$ – current time, T –working period. The invariants of velocities and accelerations are calculated according to known dependencies $b_k = \dfrac{da_k}{dk}$, $c_k = \dfrac{b_k}{dk}$.

The chosen law is an algebraic law of the second type, for which the kinematic invariants are B = 1.46; C = 6.51, and the interval of quasi-constant velocity is 36.92%.

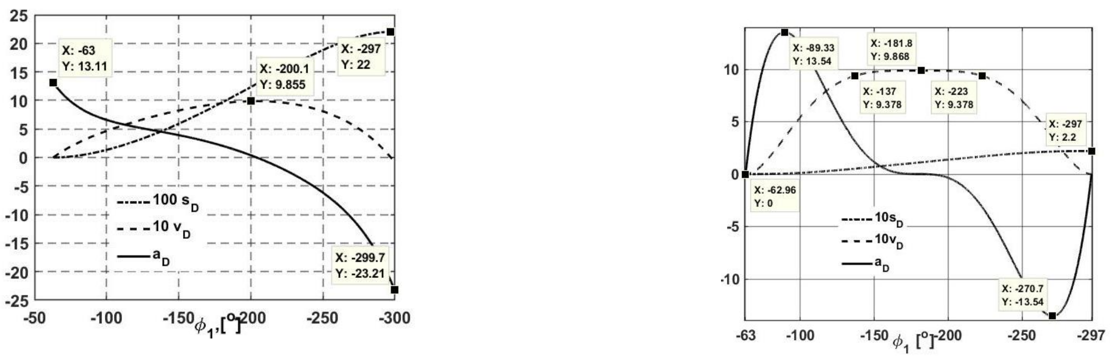
In the Fig. 7 is shown the trajectory of point C is built using (6) (fixed cam profile), at which the cutter on the working stroke will move according to the selected law (9). Visual analysis shows that the profile is smooth, the largest angles of pressure are observed at angles close to 30° and not exceeding about 20°. It will be possible to know more specifically after calculation of corresponding angles of pressure.

It is important to note that at half of the planing interval, the angles from $\varphi_3' = 0^\circ$ to $\varphi_3' = -90^\circ$, the radius of the trajectory is close to the arc of the circle, which practically unloads the highest kinematic pair of the cam.

The kinematic characteristics of the cutter of the non-synthesized mechanism are shown in the Fig. 8, a, and the synthesized one – in the Fig. 8, b. Calculations of actual values were performed by [6]:

$$s_D = a_k[S],\ v_D = b_k\frac{[S]}{[T]},\ a_A = c_k\frac{[S]}{[T^2]},$$

where [S]=0.22 m – stroke of the cutter; $[T] = 2\pi/|\omega_1|(\varphi_e - \varphi_s)/(\varphi_s + \varphi_e) = 0{,}3251\,\text{sec.}$ – period of the kinematic cycle (operating time), $\varphi_e = -297.036^\circ$ i $\varphi_s = -63.964^\circ$ – the angle of rotation of the crank at the beginning and end of the stroke.



a *b*

Fig. 8. Kinematic characteristics of the cutter: a - not synthesized and b - synthesized mechanism

There are no acceleration jumps at the boundaries of the kinematic cycle, and there is a quasi-zero velocity interval in the middle. The quasi-zero velocity is almost equal to the maximum velocity of the non-synthesized mechanism. In the mechanisms of real planing machines operation does not begin at the beginning of the stroke and does not end at the finish. There is always a certain angle of rotation of the handle at the boundaries of the kinematic cycle, after which the planing process begins and ends. Therefore, the constant velocity interval of 36.92% is minimal.

Conclusions

– for the first time a six-linked linkage mechanism with variable link length was synthesized;

– changing the length of the link is proposed to provide by a fixed cam;

– the synthesized cam profile is a smooth curve and the pressure angles are clearly less than maximum that are allowed $[\gamma]\cong 40^{\circ}-45^{\circ}$ [2];

- it is received the movement of the cutter with zero accelerations at the beginning and at the end of the working stroke and with a quasi-constant speed interval of not less than 36.92% of working stroke;

– the quasi-constant velocity interval can be increased by reducing the chipping interval.

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Пасіка В.Р., Роман Д.А., Харжевський В.О. Кінематичний аналіз та синтез руху різця довбального верстату

Кінематичні характеристики ланок і окремих точок механізму визначені методом замкнутих геометричних контурів та методом проектування планів. Сили взаємодії між ланками механізму визначені методом кінетостатики, а зрівноважувальний момент розглядом динамічної рівноваги корби і методом балансу потужностей. Виявлено недоліки структурної схеми механізму і запропоновано способи їх усунення.

Результати досліджень подані у вигляді графічних залежностей кінематичних параметрів різця, реакцій або годографів сил у кінематичних парах. Визначені зведені до корби моменти сил опору і сил інерції та побудована динамічна і математична моделі руху механізму. Показано технологію визначення потужності електродвигуна, а його стійку ділянку роботи апроксимовано прямою лінією.

Проведено кінематичний синтез і отримано модернізований механізм у якого різець може рухатись за наперед заданим законом, зокрема, без м’яких ударів на границях кінематичного циклу і з квазісталою швидкістю (похибка до 5 %) у середині кінематичного циклу.

Ключові слова: важільні механізми, оптимізація, синтез, кінематичний аналіз, закони руху, кулачкові механізми