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Synthesis of the drive mechanism of the continuous production machine

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ABSTRACT

Introduction. Existing mixing devices operate at a constant angular velocity of the working body. During this process, there are zones in which there may be no movement of material, which leads to a decrease in the quality of the finished product. When the working body moves with a variable angular rate, the inertia forces, when changing its sign, contribute to the creation of conditions under which the mixture will lose contact with the blade and move to new levels of movement, and this helps to improve the quality and intensity of the mixing process. The *purpose of the work* is to improve the quality of the processed mixture on horizontal blade (kneading) machine. **Methods**. Theoretical studies are carried out using the basic provisions of the theory of machines and mechanisms, structural and parametric synthesis, kinematic analysis, mathematical and computer simulation. **Results and discussion**. In accordance with the proposed method, the synthesis of the cam-rocker mechanism is carried out, which made it possible to select the main dimensions for the cam mechanism: the minimum radius and center distance. For the synthesis of the rocker group, the parameters of the synthesized cam mechanism are used and, using the main parameters for the rocker group (the size of the input link, the angle of the second arm initial position and rocker centre line, equal to 90°). The rocker arm span angle is obtained equal to 103°. As a result of the kinematic calculation, it is found that the dwell time of the working shafts is within 80°. The quality of the mixture can be assessed by the angle of the stagnation zone, which is formed during the movement of granular material. Under static conditions, it is equal to 0.846°, and at variable angular rate – 0.550°. It is theoretically confirmed that inertial forces that change sign four times in one cycle will provide shaking and rebound of the mixed mass from the blades, which, in turn, will significantly improve the quality of the mixture.

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Introduction

In works [1–7], much attention is paid to the process of a product mixing and it is noted that the process itself occurs under stationary conditions, i.e. at constant angular rates of the mixing devices. And this is a main roadblock on the way to obtaining a high-quality mixture, and it is noted that during the mixer operation, after a while, the speed of the mixture becomes equal to the speed of the working body of the mixer. As a result, the mixture moves in layers: particles of larger mass components move along orbits of a larger radius, particles of smaller mass move along orbits of a smaller radius. At the same time, there are zones in the mixers where there is little or no material movement; as a result, the quality of the finished product is reduced. When the variable angular velocity is imparted to the working body, the mass of the product passes from one layer to another, which contributes to an increase in the quality and intensity of the mixing process [2]. There are various designs of drive mechanisms for continuous mixers [8–10]. One of it [8] is a working chamber made in the form of a half-cylinder, inside which a working shaft with blades is placed along its axis. The mixture fills the chamber evenly across its entire width. The drive to the working shaft is carried out from the electric motor by means of V-belt and gear drives and has a constant rotation speed. Known design with two working shafts [9], which perform a complex movement due to a combination of rotational and reciprocating movements. The rotation is transmitted from the engine to the working shafts by means of a belt drive and a double-reduction gear unit; and the reciprocating motion is transmitted through a single-reduction gear and worm-gear and an eccentric mechanism. The disadvantages of these designs of mixers include the following: during machine downtime, the mixture in the working chamber is compacted, the machine restarting is difficult, and in some cases it becomes impossible due to increased loads on the kneading blades during its progressive motion. The loads become so heavy that it leads to significant deformations of the blades, and therefore, repair of the working bodies is required. This problem was encountered at a pasta factory in Novosibirsk, where a two-shaft continuous mixer (kneading mixer) is operated as part of an automatic line. One of the solutions to this problem was proposed in [10], according to which the drive to the working shafts includes a motor, a mechanism for imparting rotational motion to the working shafts, and a transmission mechanism for imparting reciprocating motion to it. At the same time, an overload release clutch is installed on the shaft between the worm and the gear wheel of the single-reduction gear. Such a design of the kneading mixer allows increasing its productivity by reducing downtime due to the absence of the need to unload the compacted dough mass from the working chamber and reload it.

The presence of uneven rotation of the working bodies, and, consequently, of the product, will also improve the quality of the product due to the elimination of zones of non-mixing [1–7]. The design of mechanisms that provide uneven rotation of the working shafts is a complex problem and depends on a number of factors, such as the raw material being processed, its density, and the shape of the elements interacting with the raw material. In this paper, it is proposed to use a cam-rocker mechanism, including a cam group and an Assur group of the second class of the third type, as a kinematic scheme for driving the working shafts of a kneading machine. It should be noted that the rotational movement from the motor shaft is transmitted to the crank carrying a two-arm lever, on one arm of which there is a roller located in the groove of a fixed cam, and on the other there is a collet, which is located in the groove of the rocker, having an axis of rotation coinciding with axis of rotation of the working shaft of the machine.

The ***purpose of the work*** is to improve the quality of the processed mixture on horizontal blade mixers (kneading machines).

To achieve this purpose, the following tasks were solved:

1. Development of a technique for synthesizing a drive to the working shafts of a machine, including:

– structural synthesis and development of the kinematic scheme of the mechanism;

– parametric synthesis, which consists in choosing the main dimensions of the cam and rocker mechanisms, which ensure the uneven movement of the working shafts;

– determination of the necessary and sufficient kinematic characteristics of the working shafts of the machine.

2. Refinement of the quality characteristics of the mixture.

Methods

The first task to achieve this purpose is to develop a technique for synthesizing the machine shaft drive, which allows designing a mechanism that improves the quality of the processed mixture. Below are the components included in the technique in the order of its solution. The first is the structural synthesis of the mechanism, which provides a variable angular velocity of the working shafts of the kneading machine. The synthesis was carried out in the following order: as the first group, providing a variable velocity of the working shafts of the machine, a cam group with a fixed cam *1*, a roller *2*, a two-arm lever *3* was adopted; as the second, a rocker group was adopted, carrying a collet *4*, movably fixed on the second arm of the lever *3* and placed in the groove of the rocker *5* (Fig. 1). Due to the fact that in the proposed design the cam is fixed, and the axis of the two-arm lever moves around the circumference, the synthesis of such a mechanism presents a certain difficulty. In this regard, for the synthesis of this mechanism, it is proposed to bring a new model, assuming that the cam is movable, and the two-arm lever freely rotates relative to the fixed axis (Fig. 1).

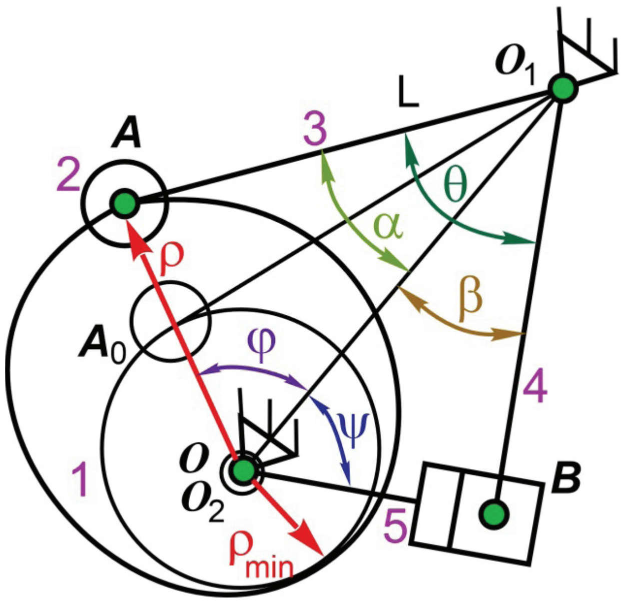


Fig. 1. General block scheme of the mechanism

To check the existence of a mechanism, the degree of its mobility using the *Chebyshev* formula [11] is determined. The degree of mobility of this mechanism was $W = 2$, which indicates the correct choice of the block scheme (an additional degree of mobility appeared due to the rotation of the roller around its axis).

Getting to the second task of synthesis, we will carry it out as a parametric one. It is thought that in order to move particles of a crumbly mass, the working shaft with blades should be able to dwell in the upper position to create favorable conditions when moving the product from the upper layers down. Since the mechanism consists of a number of elements of kinematic pairs, the dwell time should be evaluated by the last link – the link, which sets the working shafts in rotation. Therefore, taking as a basis its angle of rotation, as well as displacement, speed and acceleration, the rational option that will satisfy the goal, can be chosen. For the case under consideration, this means the presence of a dwell, smoothness and continuity of the kinematic characteristics of the machine working shaft.

The choice of the scheme of the mechanism is due to some already known design solutions of the drive, for example: the design of the working shafts, the location of the gears, the position of the engine and the design of the system carriers. Due to the fact that the cam mechanism is the first to the rocker group, let's begin the synthesis with it. In accordance with the works [12–14, 21], it is possible to accept the displacement of the center of the roller along the cycloid with the pusher journey $H = 25$ mm; the length of the rocker equal to $L = 60$ mm; cycloid angle $\beta = 180°$. It was adopted on the assumption that the period of the variable angular rate of the kneading rolls should be equal to half of its full revolution. The minimum radius of the cam ρ_{min}, center distance $a = OO_1$; initial angle ψ_0 are proposed to be determined as the desired parameters in the process of the synthesis.

Then

$$h = \frac{H}{\pi}\left(\frac{\pi\varphi}{\beta} - \frac{1}{2}\sin\left(\frac{2\pi\varphi}{\beta}\right)\right), \tag{1}$$

where h is the current value of displacement; H is the maximum value of displacement; φ is the current value of the angle of the cam rotation; β is the angle of the cam profile equal to 180°. This issue is presented in more detail in [12].

The kinematic characteristics of the center of the roller of the cam mechanism can be found by differentiating the obtained displacement function (Fig. 2).

$$v(i) := \frac{d}{di} h(i);$$

velocity and acceleration:

$$v(i) := \left| \begin{array}{l} v(i) \mathit{if}\ 0 \le i \le \pi \\ -v(i-\pi) \mathit{if}\ \pi \le i \le 2\pi \\ 0\ \mathit{otherwise} \end{array} \right. ;$$

$$a(i) := \frac{d}{di} v(i);$$

$$a1(i) := \left| \begin{array}{l} a(i) \mathit{if}\ 0 \le i \le \pi \\ -a(i-\pi) \mathit{if}\ \pi \le i \le 2\pi \\ 0\ \mathit{otherwise} \end{array} \right. .$$

Fig. 2. Program listing determining the kinematic characteristics of the cam mechanism:

$v(i)$ – roller center velocity analogue; $h(i)$ – current value of roller center displacement; $a(i)$ – differential of $v(i)$; $a1(i)$ – acceleration analogue; $i = \varphi$ – cam rotation angle

This synthesis algorithm is also used for further calculations, but with a change in some input parameters that do not affect the program, but lead to a change in the kinematic characteristics of Assur groups. The values and nature of the change in velocity analogs are shown in Figure 3.

The size of the rocker is set based on the design parameters of the roller, its axis, as well as the dimensions of the hub. The maximum pressure angle is chosen taking into account the efficiency of the entire mechanism. To determine the missing dimensions of the cam mechanism, let's mark the position of point A of the rocker arm 3. Further, on the rays connecting point O_1 and point A, segments equal to the values of velocity analogues in certain intervals of rotation angles are intercept (Fig. 4). Markings are made both for the growing phase and for the lowering phase. In our case, 8 values are given, which determined the hodograph of analogues of the velocity of point A of the mechanism.

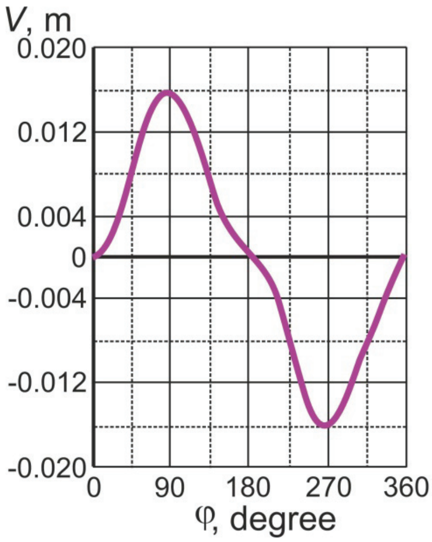


Fig. 3. Cam roller center velocity analogs

Drawing tangents to points A at an angle of $90° - \delta_{max}$, we got a family of tangents that form a shaded area in Figure 4, which determine the position of the cam axis point. Figure 4 shows the point of intersection of the tangents only for the case of maximum analogues of velocities. Then the distance from the point O to the beginning of the trajectory of the roller center will be equal to the least radius of the cam $\rho_{min} = R = 90$ mm. After the basic parameters for the cam mechanism are received, let's proceed to the synthesis of the rocker group. The parametric synthesis of the Assur group of the second class of the third type is proposed to begin with the definition of input parameters and conditions that should be set in this case. The kinematic characteristics of this group depend on the dimensions of the rocker arm L, the angle of its location with respect to the shoulder of the rocker arm of the cam group θ, which should be determined from the condition that at the moment the collet enters the groove of the rocker, the angle O_1BO_2 is equal to 90° (Fig. 1).

Taking the size of the arm, on which the collet is located, equal to $L = 60$ mm, let's determined the angle between the arms θ, for which it is necessary to consider the scheme of the mechanism shown in Fig. 1.

Then

$$\alpha = \arccos\left(\frac{a^2 + L^2 - \rho^2}{2LA}\right). \tag{2}$$

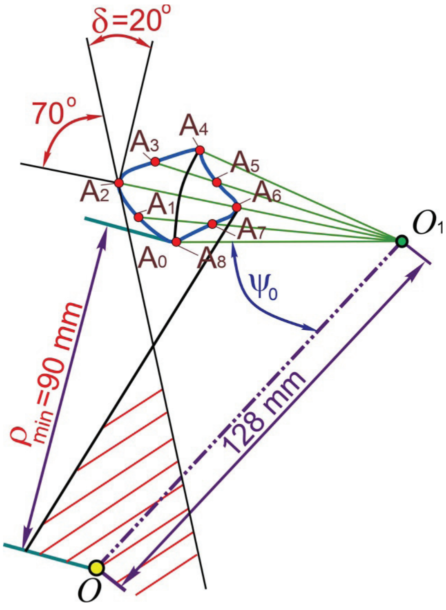


Fig. 4. To determine the missing parameters of the cam mechanism

Angle

$$\beta = \arccos\left(\frac{L}{a}\right). \tag{3}$$

The total angle, determined by the arms rotation angle, will be determined:

$$\theta = \alpha + \beta, \tag{4}$$

As a result of calculations, the total angle is θ = 103°.

A rocker swing angle is determined as follows:

$$OB = \sqrt{a^2 + O_1B^2 - 2aO_1B\cos(\beta)}, \tag{5}$$

and

$$O_1B^2 = a^2 + OB^2 - 2aOB\cos(\psi), \tag{6}$$

$$\psi = \arccos\left(\frac{a^2 + OB^2 - O_1B^2}{2aOB}\right), \tag{7}$$

Thus, as a result of the synthesis of the mechanism, the main dimensions were determined: center distance a = OO_1 = 128 mm; rocker swing angle θ = 103°; the initial angle providing the entry of the collet into the rocker O_1BO_2 = 90°; cam mechanism starting angle

$$\psi_0 = 47°.$$

The qualitative characteristics of the mixing is determined in accordance with the equation, given in [22]:

$$\mu = \mu_0 e^{-kV}, \tag{8}$$

where μ is the reduced angle of the stagnation zone; μ_0 is the coefficient of friction of the mixture on the blade in static conditions; *e* is the base of the natural logarithm; *k* is the experimental coefficient; *V* is the peripheral speed of the blade, which can be determined according to:

$$V = 5\sin(\omega t)L_b, \tag{9}$$

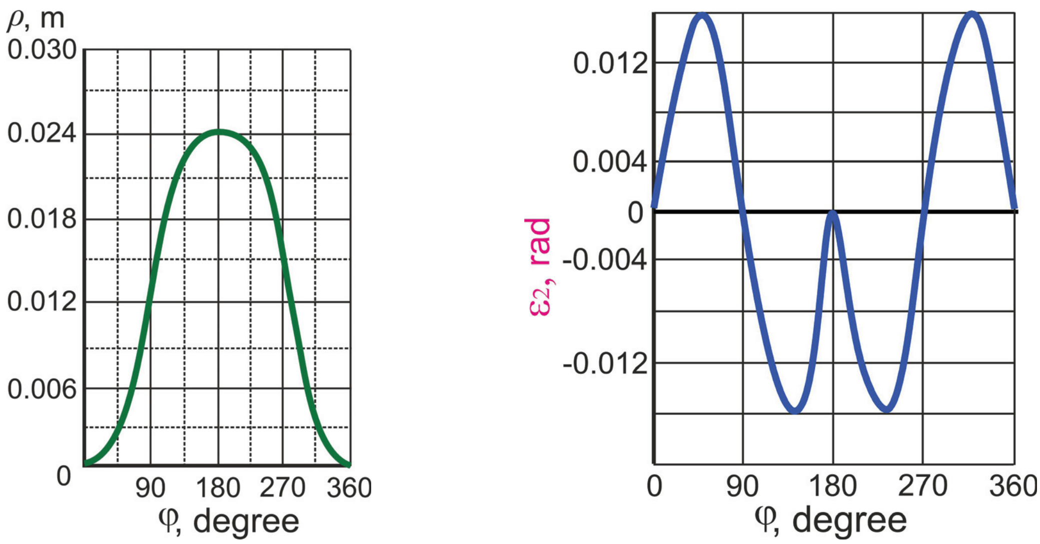
where ω is the angular frequency of revolution of the crank; L_b is the length of the blade.

Results and discussion

In accordance with the algorithm shown in Fig. 2, analogues of the kinematic characteristics of the center of the roller for the cam mechanism are calculated (Fig. 5).

When synthesizing the cam mechanism, several options were considered. The analysis showed that the selected parameters of this mechanism mainly affect the amplitude values of the kinematic characteristics, but at the same time it remains smooth and continuous without spikes. Therefore, it was decided to carry out further research on the general reduced model of the mechanism, which will allow evaluating the synthesis of the cam-rocker mechanism in terms of choosing its dimensions and kinematic characteristics.

Several synthesis options for this mechanism have been proposed. For clarity, let's analyze the influence of various parameters on changes in the kinematic characteristics of the output link – rocker.



a *b*

Fig. 5. Kinematic characteristics of the center of the roller for the cam mechanism:
a – displacement; *b* – acceleration

Option 1. Mechanism parameters have the following values: $a = OO_1 = 0.128$ m; minimum radius of the cam $\rho_{min} = 0.09$ m; arms $L = O_1A = O_1B = 0.06$ m; rocker arm span angle $\theta = 103°$. In this case, changes in the theoretical values of the radius vectors of the cam are shown in the graph (Fig. 6).

The behavior of the swing angle for the wings is shown in Fig. 7, analogue of angular accelerations in Fig. 8.

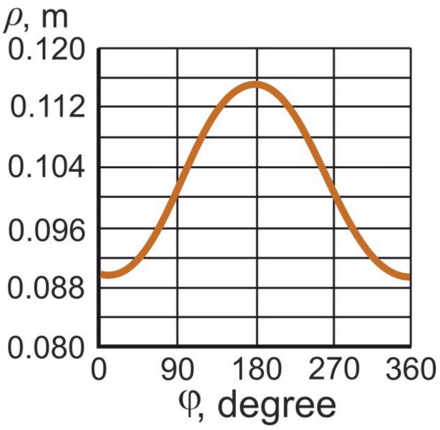


Fig. 6. Graph of the change in the numerical values of the radius vectors at $\rho_{min} = 90$ mm:
ρ – cam radius vector values; φ – cam rotation angles

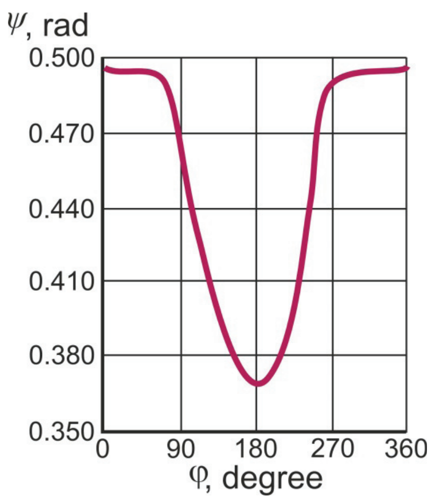


Fig. 7. Rocker swing angle graph ψ:
φ – cam rotation angle

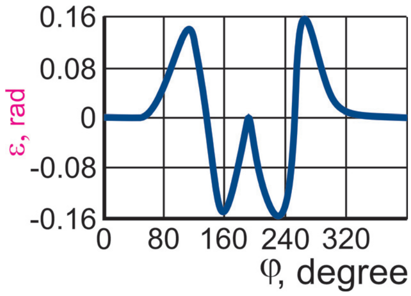


Fig. 8. Graph of rocker angular accelerations

As can be seen from the graphs shown in Fig. 7 and 8, the swing and acceleration angles have smooth and continuous functions without oscillation. In Fig. 7, 8, there are dwells at the beginning and end of the graphs. Its total value is about 80°.

Option 2. Let's change one size of the link of the mechanism. To do this, we take the minimum radius of curvature ρ_{min} = 70 mm. The remaining dimensions are left as in Option 1. Let's carry out a kinematic calculation. In this case, the values of the kinematic characteristics can be seen on the graphs: the values of the radius vectors of the cam are shown in Fig. 9, and the values of the swing angles of the rocker and analogues of angular accelerations are shown in Fig. 10, 11, respectively.

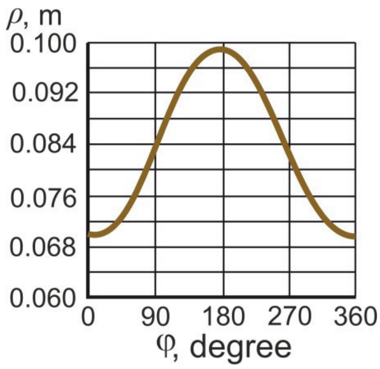


Fig. 9. Graph of the change in the numerical values of the radius vectors at ρ_{min} = 70 mm:
ρ – cam radius vector values; φ – cam rotation angles

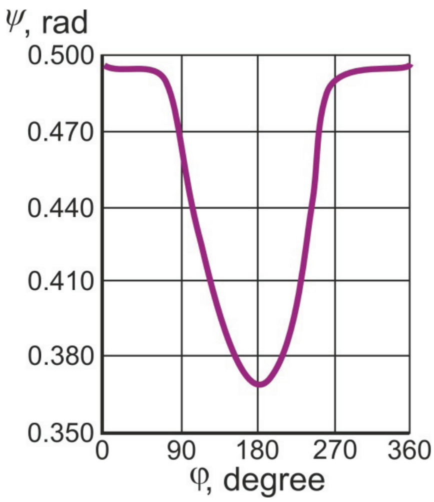


Fig. 10. Graph of the change in the swing angle of the rocker ψ at ρ_{min} = 70 mm

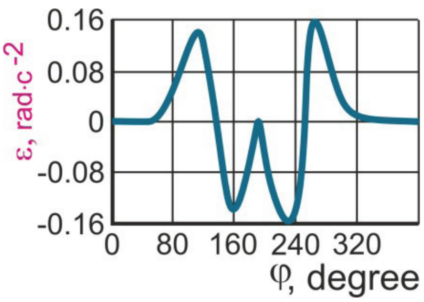


Fig. 11. Graph of the change in the accelerations of the rocker at ρ_{min} = 70 mm

As can be seen from the graphs shown in Fig. 10 and 11, the swing angles and analogues of accelerations have smooth and continuous functions without oscillations. On the graph (Fig. 10, 11), the dwells at the beginning and end of the graphs are well defined. The dwell value is about 80°.

Option 3. Let's change one size of the link of the mechanism: ρ_{min} = 50 mm. The remaining dimensions are left as in Option 1. Let's carry out a kinematic calculation. In this case, the values of the radius vectors of the cam are shown on the graph in Fig. 12. The graph of the rocker swing angle values is shown in Fig. 13.

For the rocker, the behavior of swing angle does not meet one of the main synthesis criteria: there is no smoothness and continuity of the swing angle graphs and a pronounced law of swing angle change. The beginning of the graph is displaced by more than 100° from the origin of coordinates, and it ends at approximately 245° of the cam shaft revolution (Fig. 13).

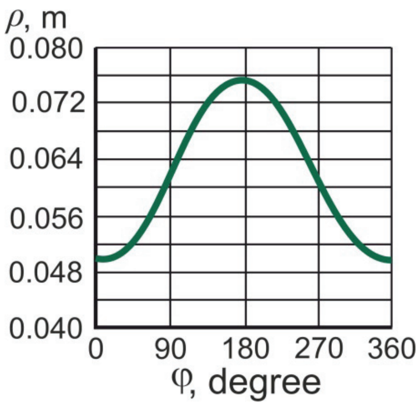


Fig. 12. Graph of the change in the numerical values of the radius vectors at ρ_{min} = 50 mm

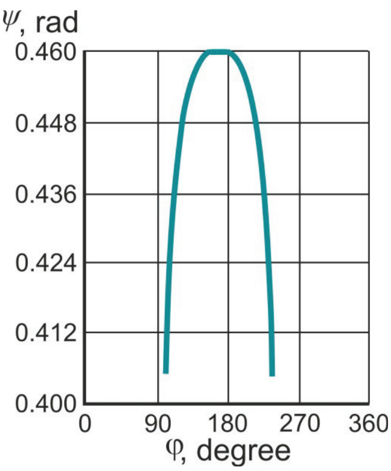


Fig. 13. Graph of the change in the rocker swing angle ψ at ρ_{min} = 50 mm

As can be seen from the above Option, the laws of change of kinematic parameters for the cam-rocker mechanism do not satisfy the set tasks of synthesis. Several other options were considered. The results of theoretical calculations are summarized in Table.

The values of the angle of rotation of the rocker arm θ were calculated using analytical dependences (1–11) and, by setting some numerical values for individual parameters of the drive mechanism. In addition, the smoothness and continuity of the graphs of kinematic characteristics and the presence of a dwell of the driven link of the cam-rocker mechanism, which has a kinematic connection with the working shaft of the machine, were taken into account.

The quality of the mixture can be estimated in accordance with equation (8). Having the numerical values of the friction coefficient of the mixture μ_0 = 0.789 and the velocity determined according to (9), the maximum total angle 0.9, and the minimum – 0.6 were obtained.

The results of theoretical calculations for the synthesis of the cam-rocker mechanism

No.	*H*, m	*a*, m	*L*, m	ρ_{min}, m	θ, deg.	Analogue of the angular acceleration ε, s^{-2}	Analog of the angular velocity ω, s^{-1}	Characteristics of curves, deg.
1	0.025	0.128	0.06	0.09	103	–0.160	0.098	dwell ≈80
2	0.025	0.128	0.06	0.07	103	–0.172	0.101	dwell ≈80
3	0.025	0.128	0.06	0.05	103	–	–	discontinuities of function
4	0.025	0.128	0.08	0.09	103	0.046	0.067	no dwell
5	0.025	0.128	0.09	0.09	103	0.040	0.060	no dwell
6	0.025	0.130	0.06	0.09	103	0.055	0.040	no dwell
7	0.025	0.140	0.06	0.08	103	0.024	0.018	no dwell
8	0.025	0.128	0.06	0.09	110	0.040	0.024	no dwell
9	0.025	0.128	0.06	0.08	120	0.172	0.10	dwell ≈80
10	0.025	0.128	0.06	0.09	105	0.174	0.12	dwell ≈80

Conclusion

The main purpose, set in the work, is to improve the quality of the processed product, which is obtained through the developed mechanism, including the synthesis of cam and rocker groups, providing the necessary degree of mobility and link sizes.

So for the cam mechanism, the rational parameters of the links are: center distance $a = OO_1 = 128$ mm; rocker swing angle θ = 103°; the initial angle $\psi_0 = 470°$ for given dimensions of the rocker arm $L = 60$ mm, the roller diameter equal to 60 mm, and the use of the law of motion of the roller center along the cycloid with the curve angle β = 180° and the pusher journey $H = 25$ mm.

To obtain the length of the working shaft of the machine, the synthesis of the rocker group provided the angle of the initial position of the collet and the rocker equal to $O_1BO_2 = 90°$.

The quality of the mixture was evaluated by the angle of the stagnant zone, which is formed during the movement of bulk material. In static conditions, it is equal to –0.846, and with a variable angular velocity –0.550. In addition, inertial forces, which in present case will change sign four times in one cycle, will provide shaking and rebound of the crumbly mass from the blades. All these activities will improve the quality of the mixture.

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Conflicts of Interest

The authors declare no conflict of interest.