Determining efficient values of continuous technological machines parameters

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Abstract. The paper deals with the issues of increasing the continuous mixers efficiency. The solution to this problem is known to be possible both by increasing the volume of the working chamber and by increasing the velocity characteristics of the work member. Consequently, there is a need to investigate continuous mixers with an oscillation mechanism. The article presents the calculation scheme of the continuous mixer as well as the algorithm of the mixer kinematic parameters calculation considering the reciprocating movement of the work members. We obtained the graphical dependence of the working shafts path, velocity and acceleration in the axial direction on the eccentric rotation angle as a result of mathematical modeling of these parameters. The work provides the dependence of the axial velocity and acceleration on the axial and tangential velocities ratio, as well as the dependence of tangential and axial load on the blade fitting angle for some values of the axial and tangential velocities ratio. We prove that with the increase in the blade rotation angle the axial loading increases, while tangential load decreases, thus the constant value of the resultant force remains constant within the specified interval of values. The increase in the velocities ratio leads to the increase in tangential load and reduction of axial load, which should be taken into account in the synthesis of a new mechanism.

1. Introduction

The main direction of technological progress in most industries is the introduction of advanced technologies, automated production lines and high-performance equipment for large enterprises; and complete plants facilitating the mechanization of the basic production, auxiliary and transport operations at small enterprises [1-5]. This improves the production lines performance and product quality, minimizes labor-intensive manual operations, reduces raw material losses and enhances the overall production culture.

The development and introduction of competitive high-performance machines becomes of primary importance owing to the transition to the domestic equipment of the majority of Russian enterprises. Therefore, improving the efficiency of process equipment is an urgent task [6-10].

Mixing units are widely exploited in many industries and agriculture. The automatic lines usually employ continuous mixers. Mixing units of various designs, including blade mixers, are used to obtain loose mixtures. The work members of such mixers make rotary movements, but they are also imparted with additional reciprocating movement (oscillation) for the increase in intensity of mixing and "dead zones" elimination.

Considering this, there is a need to investigate continuous mixers with an oscillation mechanism.

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2. Theory, materials and methods

The continuous mixers efficiency can be presented in the following form:

$$E = \frac{T_f \cdot P}{E_{pr}}, \tag{1}$$

where ***E*** is the efficiency of a process machine, kg/rub; $\boldsymbol{T_f}$ is the factual labor time, h; ***P*** is the machine production capacity, kg/h; $\boldsymbol{E_{pr}}$ is the production expenditures, rub.

The factual labor time depends on the number of work shifts and is determined at the stage of production organization. Since expenditures reduction is not always possible, the efficiency can be raised mainly by increasing production capacity.

The production capacity of mixers is expressed by [7]:

$$P = 0.25nD^2\left(\frac{L}{\tau}\right)\rho k, \tag{2}$$

where ***n*** is the rotating frequency of the blade shaft, $\min^{-1}$; ***D*** is the mixer working chamber diameter, m; ***L*** is the mixer working chamber length, m; $\boldsymbol{\tau}$ is the duration of the mixing operation, s; $\boldsymbol{\rho}$ – product density, kg/m^3; ***k*** is the feed coefficient.

By analyzing the dependence (2), you can identify two ways to improve production capacity:

1. The increase in the volume of the working chamber by changing its geometric dimensions. However, it is often impossible because the units are built into automatic lines.

2. The increase in the velocity of product transportation by increasing the work member rotating frequency. Yet, this imposes a number of problems, such as loss of product quality and reduction of shafts vibration resistance.

With the increase in the mixer velocity characteristics accompanied by additional oscillation movement of work members, there appears a problem of defining the optimum parameters for enhancing production capacity of the machine and maintaining the high quality of the mixture.

The power (W) can be calculated by [7]:

$$W = \frac{(P_t \cdot V_t + P_a \cdot V_a)z}{\eta}, \tag{3}$$

where $\boldsymbol{P_t}$ is the tangential component of forces acting on a blade, N; $\boldsymbol{P_a}$ is the axial component of forces acting on a blade, N; $\boldsymbol{V_t}$ is the tangential velocity, m/s; $\boldsymbol{V_a}$ is the axial velocity, m/s; ***z*** is the number of blades; $\boldsymbol{\eta}$ is the drive efficiency.

Tangential and axial velocities are calculated by the following formulas:

$$V_t = 2 \cdot \pi \cdot n \cdot R, \tag{4}$$

$$V_a = V_t \cdot i, \tag{5}$$

where ***R*** is the rotation radius of the blade plane center, m; ***i*** is the transmission ratio.

However, the research interest to this issue is stipulated by the fact that the existing methods of the technological load calculating are valid only for the mixers with work members' rotary movement without additional reciprocating movement. All this considered, the following calculation dependences were obtained for the analyzed mixers:

$$P_t = F\left[g \cdot R \cdot \rho \cdot tg^2\left(45 + \frac{\gamma}{2}\right) + 2 \cdot C \cdot tg\left(45 + \frac{\gamma}{2}\right)\right]\left(\cos(\alpha - \delta) + \mu \cdot \sin(\alpha - \delta)\right) \tag{6}$$

$$P_a = F\left[g \cdot R \cdot \rho \cdot tg^2\left(45 + \frac{\gamma}{2}\right) + 2 \cdot C \cdot tg\left(45 + \frac{\gamma}{2}\right)\right]\left(\sin(\alpha - \delta) - \mu \cdot \cos(\alpha - \delta)\right), \tag{7}$$

where ***F*** is the area of the blade immersed in the mixture, m^2; ***g*** is the free fall acceleration, m/c^2; $\boldsymbol{\rho}$ is the product density, kg/m^3; ***C*** is the specific resistance to mixing, $\boldsymbol{P}_a$; $\boldsymbol{\gamma}$ is the angle of the mixture

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internal friction; $\boldsymbol{\alpha}$ is the blade obliquity angle relative to the radius, deg; $\delta = arctg\left(\frac{V_a}{V_t}\right)$ is the entry angle of the blade into the mixture, deg; $\boldsymbol{\mu}$ is the coefficient of mixture friction at the blade.

The calculation of such machines is complicated by the fact that, on the one hand, the shafts movement in the axial direction changes the place of the load relative to the supports. On the other hand, the change of the resulting velocity in both direction and magnitude leads to the change of power characteristics in size and direction (Figure 1), and this makes the calculation of the mechanism vibration resistance more sophisticated.

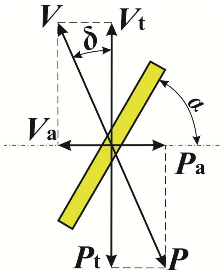

Figure 1. The direction of velocities and process load

We developed the design model (Figure 2) of a continuous mixer with rotational and reciprocating movement of work members and performed its kinematic analysis to determine the optimum parameters values.

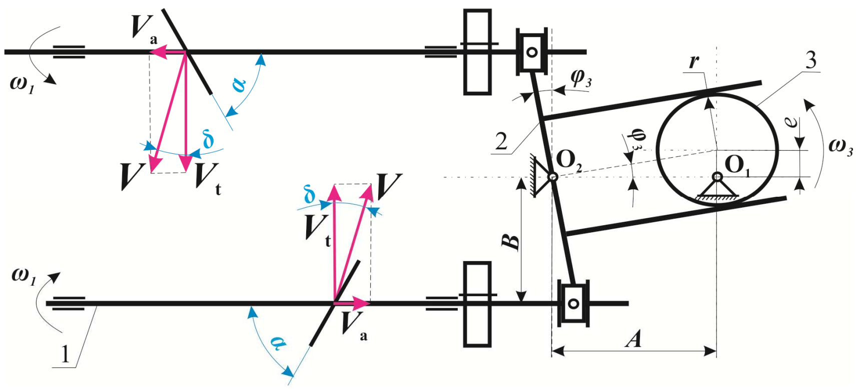

Figure 2. The design model of continuous mixer

Reciprocating movement of working shafts 1 with the blades installed on them is executed through the system of levers 2 and eccentric mechanism 3. Rotational movement of shafts 1 is imparted from the engine by a V-belt drive and a tooth gear; therefore, tangential velocity of blades ($\boldsymbol{V}_t$) at stationary operation mode of the machine is constant. Consequently, we investigated only the following kinematic parameters: path ($\boldsymbol{S}$), velocity ($\boldsymbol{V}_a$) and acceleration ($\boldsymbol{a}_a$) of the working shafts in the axial direction. This required identification of their correlation with the mechanism geometric parameters.

The path S traversed by the shaft in the axial direction in the first approximation was determined by the formula:

$$tg\varphi_3 = \frac{S}{B}, \tag{8}$$

where φ_3 is the lever rotation angle at the eccentric turn at the angle of φ, deg; $\boldsymbol{B}$ is the distance from the levers rotation axis to the shafts rotation axis, m.

On the other side, the angle φ_3 is determined by the expression [6]:

$$tg\varphi_3 = \frac{e \cdot \sin\varphi}{A + e \cdot \cos\varphi}, \qquad (9)$$

where ***e*** is the eccentricity, m; ***A*** is the distance from the eccentric rotation axis to the levers rotation axis, m.

Equating the right parts of the equations (8) and (9) yields the formula of the shaft path in the axial direction analogue:

$$S' = \frac{B \cdot e \cdot \sin\varphi}{A + e \cdot \cos\varphi} \ . \qquad (10)$$

In order to move from analogues to real values it is necessary to consider the following:

$$\varphi = \omega_3 t \ , \qquad (11)$$

where ω_3 is the eccentric angular velocity, rad/s; ***t*** is time, s.

Then the path is:

$$S = \frac{B \cdot e \cdot \sin\omega_3 t}{A + e \cdot \cos\omega_3 t} \ . \qquad (12)$$

Velocity is the rotation angle or time first derivative of the path, therefore:

$$V_a = \frac{A \cdot B \cdot e \cdot \omega_3 \cdot \cos\omega_3 t + B \cdot e^2 \cdot \omega_3}{(A + e \cdot \cos\omega_3 t)^2} \ . \qquad (13)$$

Acceleration is the time first derivative of the velocity or the second derivative of the path:

$$a_a = \frac{B \cdot e \cdot \omega_3^2 \cdot \sin\omega_3 t\left(-A^3 + A \cdot e^2 \cdot \cos^2\omega_3 t + 2 \cdot A \cdot e^2 + 2 \cdot e^3 \cdot \cos\omega_3 t\right)}{(A + e \cdot \cos\omega_3 t)^4} \ . \qquad (14)$$

The next stage of research is mathematical modeling of kinematic and power parameters. The geometrical parameters of the mixer are set: ***A*** = 0.125 m; ***B*** = 0.220 m; ***e*** = 0.020 m.

The eccentric angular velocity is defined by the expression:

$$\omega_3 = \omega_1 \cdot i_{3-1}, \qquad (15)$$

where ω_1 is the angular velocity of work members, ω_1 = 6.074 rad/s; ***i*** is the transmission ratio, ***i*** = 27.381 · 10^{-3}.

3. Results and discussion

We obtained the graphical dependence of the working shafts path, velocity and acceleration in the axial direction on the eccentric rotation angle as a result of mathematical modeling of these parameters (Figure 3).

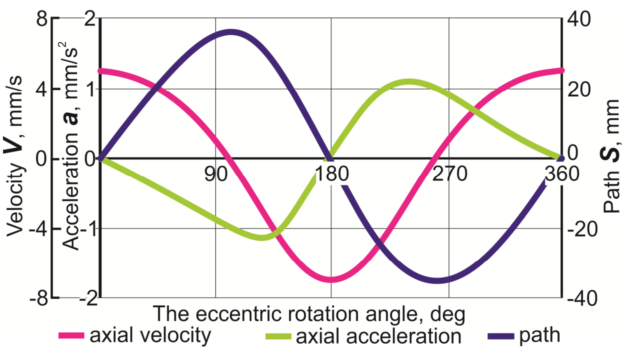

Figure 3. Kinematic characteristics of the mixer at the V_a/V_t = 0.0015

The maximum values of velocity and acceleration are of practical interest. According to the graph, the axial velocity has the maximum value at the eccentric rotation angle of 180°, and the acceleration has the maximum value at 120° or 240°.

This data allows determining the dependence of the maximum axial acceleration on the maximum axial velocity when the latter changes from zero to V_t. In this case, the acceleration is calculated by the formula (14) at the considered eccentric rotation angle and the angular velocity ω_3 which corresponds to a certain:

$$\omega_3 = \frac{V_a(A-e)}{B \cdot e^2 - A \cdot B \cdot e} \quad . \tag{16}$$

Thus, the calculations resulted in a graphical dependence of axial velocity and acceleration on the ratio of the velocities V_a/V_t (Figure 4).

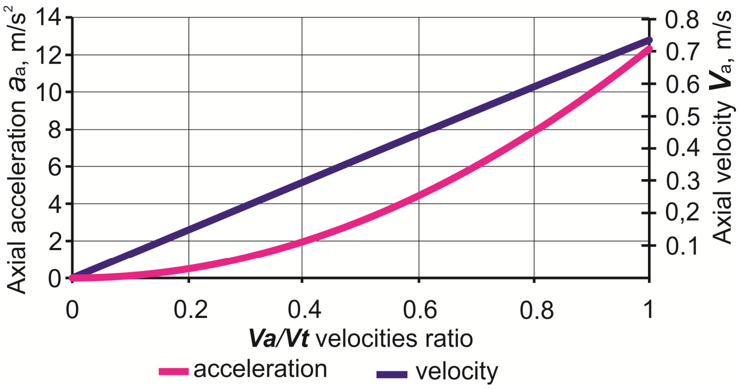

Figure 4. The dependence of axial velocity and acceleration on the V_a/V_t velocities ratio

The analysis of the obtained dependence shows that it is possible to avoid large dynamic loads by running mixers within the range of the axial velocity change characterized by a slight acceleration change. In Figure 4, this range is within the velocity ratio change interval from 0 to 0.2.

We performed mathematical modelling of the technological load according to the formulas (13) and (14) depending on the blade fitting angle within the specified interval. The dependence of the tangential load on the blade fitting angle within the range from 0 to 90 ° is shown in Figure 5.

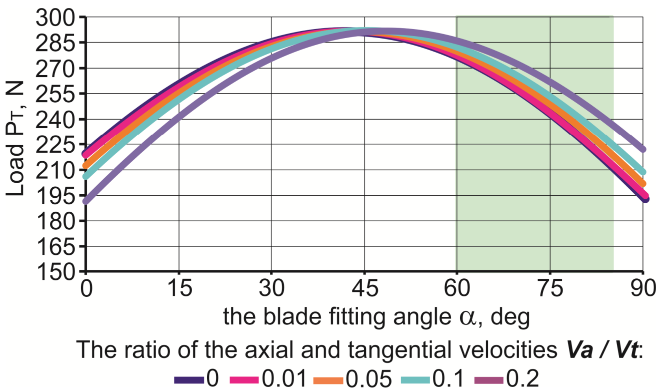

Figure 5. The dependence of the tangential load on the blade fitting angle for some values of the axial and tangential velocities ratio

These dependencies indicate that the optimal range of the blade rotation angle is between 60° and 85°. This can be explained by the following facts: at small angles the product moves slowly, and the angles close to 45 ° cause significant loads.

The nature of the axial load change is shown in Figure 6.

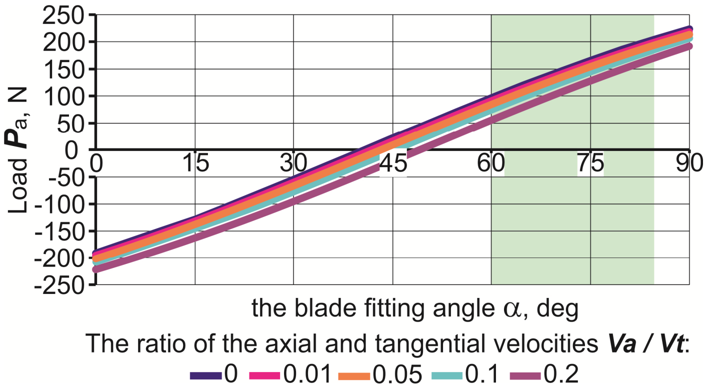

Figure 6. The axial load for some values of axial and tangential velocity ratio

With the increase in the blade rotation angle, the axial load increases, while the tangential load decreases within the selected interval. This is the consequence of the resulting force magnitude uniformity. However, the increase in the velocities ratio leads to the increase in tangential load and the reduction of axial load, which must be taken into account in the synthesis of a new mechanism.

4. Conclusion

The conducted research resulted in the following conclusions:

1. the increased axial velocity leads to a sharp increase in acceleration owing to dynamic loads;
2. the influence of velocities ratio on technological loading must be considered;
3. with the increase in the velocities ratio there is a growth of tangential and reduction of axial load within the specified range of the blades fitting angle change.

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References

[1] Hejma P et al. 2017 *Procedia Engineering* **177** 3–10
[2] Lobanov D V et al. 2016 *IOP Conf. Ser.: Mater. Sci. Eng.* **142(1)** 012081
[3] Lobanov D V et al. 2017 *Key Engineering Materials* **736** 81-85
[4] Hejnová M 2014 *Procedia Engineering* **96** 157-163
[5] Hsieh J F 2014 *Mechanism and Machine Theory* **81** 155-165
[6] Podgornyj Yu I et al. 2017 *IOP Conf. Ser.: Earth Environ. Sci.* **87(8)** 082039
[7] Antipov S T, Kretov I T, Ostrikov A N and Panfilov V A 1975 *Machines and devices of food production* (Moscow: High School Press) p 703
[8] Podgornyj Yu I et al. 2013 *Obrabotka metallov* **3** 68–73
[9] Podgornyj Yu I et al. 2016 *Obrabotka metallov* **2** 41–50
[10] Ptitsyn S V et al. 2013 *Obrabotka metallov* **2** 33–38