Synthesis of irregular motion mechanisms for production machine drives

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Abstract. Irregular motion of the working bodies is widely used in mixing equipment. It helps to achieve homogeneity of the mixtures, since the uniform motion may lead to the "dead" zones formation. The purpose of this work is to synthesize the mechanism for driving the mixer's working shafts which includes an epicyclic transmission with a swing. This paper reports the studied pathways, velocities, and accelerations for different gear ratios at the final link, i.e. the swing. The research methods are analysis and synthesis of irregular motion mechanisms. It was found that in the mechanisms with the lowest pulsation, the amplitude of the angular acceleration change at the distance from the planet gear rotation axis to the pin equal to 14 mm is the smallest of the considered ones. The performance increases by 22 % with this coefficient of non-uniformity. As a result of the conducted research, we determined the main rational parameters for the synthesis of the considered mechanism.

1. Introduction

In many industries, mechanisms that impart irregular motion to working bodies are widely used. Such mechanisms are especially important in the mixing equipment for obtaining mixtures with uniform consistency.

The use of uniform motion in the mixers working bodies leads to the formation of the so-called "dead" zones in the working chamber, and, as a result, low quality of the finished product [1].

When an irregular motion is imparted to the working body, the actual instantaneous speed of the mixed raw material is composed of the average value of the flow rate over time and the deviation of the actual speed from the average (pulsation). This mode of particle movement allows achieving uniform consistency of the product [2-4].

This problem can be solved by the use of various drive mechanisms providing for irregular motion of the mixer working body [5-12].

In this paper, we consider the synthesis of the drive mechanism of the mixer's working shafts, which includes an epicyclic transmission with a swing.

2. Materials and methods

As a rule, mixers operate at a constant angular velocity of the working body. Under these conditions, over time the fluid velocity becomes equal to the speed of the blades of the mixer working body, i.e. there is a stratified fluid flow: particles of larger mass components move in orbits of a larger radius;
particles of smaller mass move in orbits of a smaller radius, thus creating laminar motion which negatively affects the results of the mixing process.

When a variable angular velocity is imparted to the working body, the actual instantaneous velocity of the fluid \( \omega_f \) will be composed of the average value of the flow velocity over time \( \omega_a \) and the deviation of the actual velocity from the averaged \( \Delta \omega \):

\[
\omega_f = \omega_a + \Delta \omega,
\]

In this case, pulsations occur due to the deviation of the actual speed from its average value, which will differ in magnitude and direction. Consequently, turbulent phenomena occur, i.e. elementary liquid masses transition from one layer to another, which contributes to improving the quality and intensity of the mixing process.

The turbulent mode of particle motion allows achieving homogeneity of the mixture. Thus, to improve the quality of the mixture, it is advisable to use mixers with a variable angular rotation speed of the kneading body. This problem can be solved by using lever and gear drive mechanisms that ensure irregular motion of the mixer working body.

For these purposes, we propose to consider epicyclic transmission.

To synthesize such a mechanism, the geometric parameters of the constituent elements must be determined. The scheme of the mechanism is shown in Figure 1.

![Figure 1. Epicyclic transmission: 1 – Sun gear; 2 – Planet gear; M – Pin tracing an epicycloid; \( d_1 \) – Diameter of the sun gear; \( d_2 \) – Diameter of the planet gear; \( l \) – Interaxial distance; \( e \) – Distance between the planet gear rotation axis and the pin \( M \).](image)

According to [2, 4, 6 - 12], the gear ratio is determined by the formula:

\[
u = \frac{d_2}{d_1}, \tag{1}\]

The planet gear rotation angle

\[
\psi = \frac{\alpha}{u}, \tag{2}\]

with \( \alpha \) – the carrier rotation angle \( O_1O_2 \); \( u \) – gear ratio.

The angle \( \gamma = \angle O_1O_2M \) is determined by:

\[
\gamma = 180 \cdot \text{deg} - \psi, \tag{3}\]

The value of the radius vector \( R \) connecting points \( O_1 \) and \( M \):
\[ R = \sqrt{l^2 + e^2 - 2 \cdot l \cdot e \cdot \cos \gamma}, \]  \hspace{1cm} (4)

The angle \( \beta = \angle O_2 O_1 M \) is found by the sine theorem:

\[ \beta = \arcsin \left( \frac{e \cdot \sin (\psi)}{R} \right), \]  \hspace{1cm} (5)

The angle \( \varphi \) is made up of the angle \( \alpha \) and \( \beta \):

\[ \varphi = \alpha + \beta \]  \hspace{1cm} (6)

The angular velocity of the epicyclic mechanism pin can be found by differentiating expression 6:

\[ \omega_2 = \frac{d}{dt} \varphi \]  \hspace{1cm} (7)

The angular acceleration of the epicyclic mechanism pin is determined by differentiating the angular velocity equation (7):

\[ \varepsilon_2 = \frac{d}{dt} \omega = \frac{d^2}{dt^2} \varphi \]  \hspace{1cm} (8)

Pulsations are received from the equation:

\[ \lambda_2 = \frac{d}{dt} \varepsilon = \frac{d^2}{dt^2} \omega = \frac{d^3}{dt^3} \varphi \]  \hspace{1cm} (9)

In order to impart the motion to the slave link, it is necessary to transfer the motion from the point \( M \) to the swing, whose axis coincides with the axis of the carrier rotation. Thus, the law of change in angular velocity and acceleration is transmitted to the working shaft.

Based on this, the pathways of the slave link with the transfer ratios \( u = 1; u = 1.5; u = 2; u = 3; u = 4 \) were obtained. The most characteristic pathways of pin motion (point \( M \), see Figure 1) are shown in Figure 2.
Angular velocities and accelerations are shown in Figure 3.

Figure 2. Pin motion pathways (M): a - \( u = 1 \); b - \( u = 1.5 \); c - \( u = 2 \); d - \( u = 4 \)

Figure 3. Diagram of angular velocities, accelerations, and pulsations: the transfer ratio \( u = 2 \).
3. Results and discussion

We demonstrate that the most rational gear ratio is \( u = 2 \) based on the analysis of graphs representing pathways, velocities, and accelerations at the final link (swing) at different gear ratios with the lowest angular accelerations.

Then the coefficient of non-uniformity \( \delta \) is determined by the formula [1]:

\[
\delta = \frac{\omega_{\text{max}} - \omega_{\text{min}}}{\omega_{\text{av}}},
\]

with \( \omega_{\text{max}}, \omega_{\text{min}}, \omega_{\text{av}} \) being the maximum, minimum, and average angular speeds of the swing, respectively.

The average angular velocity is calculated according to the expression:

\[
\omega_{\text{av}} = \frac{\omega_{\text{max}} + \omega_{\text{min}}}{2},
\]

The capability of a continuous paddle mixer \( C \) (kg·h\(^{-1}\)) is determined by the formula [13]:

\[
C = 1435 \cdot \varphi \cdot \rho \cdot \left( \frac{b}{H} \right) \cdot D^3 \cdot \omega \cdot \sum_{i=1}^{n} j_i \cdot \sin 2\alpha_i ,
\]

with \( \varphi = 0.2 \) – feed coefficient [13]; \( \rho = 1200 \) kg/m\(^3\) – dough density [13]; \( b = 48 \) mm – blade width; \( H = 300 \) mm – blade pitch; \( D = 0.5 \) m – blade outer diameter; \( \omega \) – angular speed of the working shaft; \( j_i \) – the number of blades mounted at one angle; \( \alpha_i \) – the angle between the blades and the shaft center line; \( i \) – variants of the blades location (Figure 4).

Calculations for epicyclic transmission were made in accordance with formulas (1 - 11). On the basis of these values, we obtained the dependence of the machine performance on the nonuniformity coefficient (Figure 5).
Figure 5. Dependence of the machine capability on the coefficient of rotation non-uniformity of the working body

It is obvious that there is a positive correlation between performance and the non-uniformity coefficient. Based on this, we determined the law of performance change related to the coefficient of non-uniformity. For this purpose, we build a trend line (Figure 5). The graph shows that when the non-uniformity coefficient δ increases by 1 %, the machine's performance increases by an average of 5.583 kg/h.

Based on the range of the recommended values of the non-uniformity coefficient [14], we choose the optimal values: δ₁ = 43.907 %, δ₂ = 50.749 %, δ₃ = 57.848 % . These values correspond to the O₂M value (Figure 1): e₁ = 14 mm, e₂ = 16 mm, e₃ = 18 mm, respectively. Point M (Figure 1) traces an epicycloid.

The swing angular velocity is determined by the formula (7); the angular velocity of the carrier ω₁ equals to 1 rad/s (Figure 6).

Figure 6. Graph of the change in the angular velocity of the swing

Angular acceleration is obtained by differentiating the angular velocity (formula 7); the graph of angular acceleration is shown in Figure 7.

Figure 7. Graph of the change in the angular acceleration of the swing

The pulsation graph (Figure 8) is obtained by differentiating the angular acceleration (formula 9).
4. Conclusion
In the course of the conducted research we investigated the epicyclic mechanisms with transfer ratios from 1 to 4 with a step of 0.5.

The mechanism with the lowest pulsation $\delta = 43.907\%$ was chosen from the presented variants, since in this case the amplitude of the angular acceleration change at $e = 14$ mm is the smallest of the considered ones.

The performance increases by 22 % with this coefficient of non-uniformity.

Thus, the most rational basic parameters for the synthesis of an epicyclic mechanism with a swing are: the diameter of the sun gear $d_1 = 176$ mm; the diameter of the planet gear $d_2 = 88$ mm; the gear ratio $u = 2$; the interaxial distance $l = 132$ mm; the distance from the planet gear rotation axis to the pin $e = 14$ mm.

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