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Research and justification of parameters for a flexible sectional screw working body

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Abstract. The article presents a new design of the hinged-sectional working body of a flexible screw conveyor intended for transporting bulk agricultural materials. The influence of the hinge mechanism's structural parameters on its operational efficiency is investigated, particularly in terms of ensuring the required angular displacement of axes, preventing jamming, and minimizing energy losses. Analytical dependencies are proposed to determine the arc displacement of the ball along friction surfaces, its velocity, and the efficiency coefficient of the hinge mechanism. It is established that the maximum efficiency coefficient ranges between 0.88 and 0.91 when the ratio of the ball radius to the rotation radius of ball centers is within 0.3–0.45. Calculations show that an increase in the conical socket inclination angle leads to a reduction in efficiency. To enhance the overall conveyor efficiency, it is recommended to reduce the hinge axis deviation angle, which results in a larger bending radius of the screw. Specifically, it is determined that the axis deviation angle should not exceed 20–25° while maintaining an efficiency coefficient of at least 0.9. To determine the optimal structural characteristics of the screw working body sections, computer modeling of force load influence on corresponding deformations was conducted. The results show that an increase in torque leads to a greater twisting angle of the section, with the most significant deformation occurring at a minimal number of axial rods. The proposed design improves the efficiency of loading and unloading operations while expanding technological capabilities in transporting grain and other bulk materials.

Key words: flexible screw conveyor, hinge mechanism, efficiency coefficient, force parameters, computer modeling, bulk material transportation.

INTRODUCTION

Due to significant technogenic impacts, thousands of hectares of Ukrainian land have suffered extensive damage, leading to soil quality deterioration and accelerated degradation processes. Various mechanical, physical, and chemical factors contribute to these effects, disrupting soil structure and ecosystem functions and negatively affecting its physico-geochemical properties.

One of the most significant impacts on agricultural lands and soil cover comes from intense external influences, including explosions and heavy machinery operations, which account for a substantial portion of the overall damage. As a result, fields are often left covered with craters and disrupted soil layers, varying in size and depth (Balyuk et al., 2022).

Therefore, the restoration of contaminated lands and the preservation of agricultural soil fertility are essential tasks for safeguarding Ukraine's food security. Rehabilitating explosion-affected areas requires the removal or stripping of the damaged surface soil layer. This is necessary not only due to chemical contamination with heavy metals but also because the fertile topsoil is often significantly degraded or completely destroyed.

For the technical reclamation of lands impacted by such disturbances, a method has been proposed that involves the removal of damaged and contaminated soil containing heavy metals and other hazardous substances. To facilitate the removal and disposal of the contaminated soil layer, a flexible screw conveyor equipped with a sectional auger working body can be employed. This design enables efficient transfer of soil directly into a truck bed.

Screw conveyors are commonly used for handling a wide range of bulk agricultural materials, such as grains, granular seeds, clay, sand, loose soil, animal feed, and granulated mineral fertilizers (Gill, 2003; Evstratov et al., 2015; Olanrewaju et al., 2017; Baranovsky et al., 2020; Karpeenko et al., 2021).

The foundation of comprehensive mechanization in loading and unloading operations, which enhances labor productivity and production efficiency, lies in technical means for the continuous transportation of bulk agricultural materials (Lyashuk et al., 2015; Rohatynska et al., 2015). Their performance and mobility in various technological processes expand their technological capabilities and contribute to reducing energy consumption (Rohatynskyi et al., 2016).

The primary drawback of rigid screw conveyors is the limited mobility of technological pipelines. Additionally, the placement of the screw shaft in bearings and a rigid guiding tube increases the material consumption of such conveyor designs (Bulgakov et al., 2022a; Hevko et al., 2016a).

Analysis of the literature indicates that the least resource-intensive method for transporting bulk agricultural materials along curved trajectories is the use of flexible screw, which are based on separate screw sections hingedly connected to each other (Lech, 2001; Hevko et al., 2018). Studies (Bulgakov et al., 2022b; Pylypaka et al., 2017; Hevko et al., 2016b) have substantiated the structural, kinematic, and technological parameters of screw working bodies in accordance with their functional applications. To eliminate the mentioned shortcomings, active research has recently been conducted to develop optimal designs of flexible screw conveyors with enhanced functional and operational characteristics (Hevko et al., 2020; Hevko et al., 2017).

In such conveyors, the flexible screw working body is freely positioned (without supports) inside an elastic casing, where its inner surface moves the material toward the discharge zone, which constitutes their primary advantage.

When rotating at speeds exceeding 550 rpm, the flexible screw spiral evenly distributes the bulk material along the periphery of the conveyor casing, ensuring self-centering and enabling efficient transport to the discharge zone (Tian et al., 2018; Trokhaniak et al., 2020).

The results of studies dedicated to the justification of design, kinematic, and technological parameters of the working bodies of flexible screw conveyors from the perspective of their functional purpose are also known (Loveikin & Rogatynska, 2011; Hevko et al., 2019). However, several unresolved issues remain regarding the operational and durability characteristics of such working bodies. This indicates that improving the efficiency of flexible screw conveyors remains an important and relevant challenge in agricultural production.

The aim of the study was to justify the design parameters of flexible screw working bodies in order to improve the functional and operational performance of the developed construction.

MATERIALS AND METHODS

To enhance the functional and operational performance of screw conveyors in transporting removed damaged soil, a flexible sectional screw working body has been developed. Its structural diagram is shown in Fig. 1.

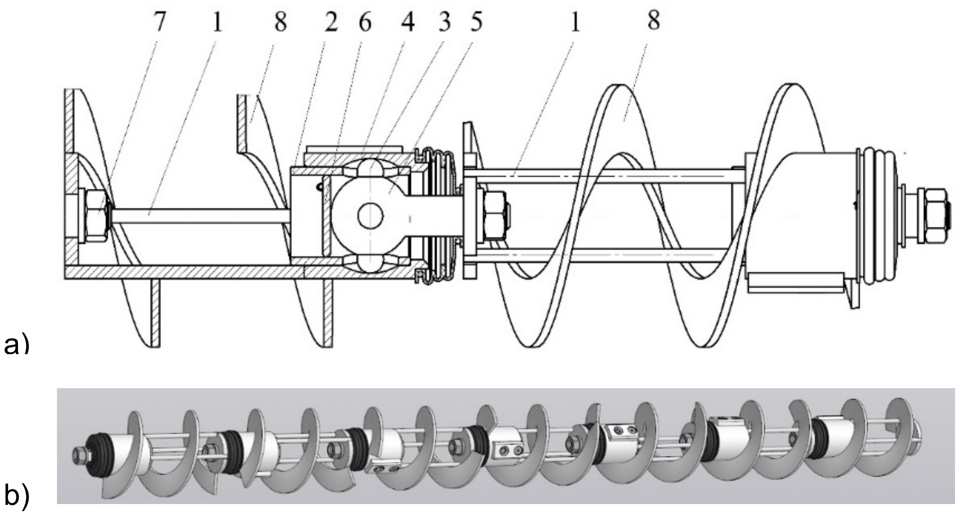

Figure 1. Structural diagram (a) and 3D model (b) of the flexible sectional screw working body.

This working body consists of individual sections of equal diameters and lengths, designed as welded structures. These sections include rods (1), with a cylindrical sleeve (2) rigidly attached to one end. The sleeve features a system of axially parallel grooves, evenly distributed around the circumference, which interact with balls (3). On the opposite side, the balls (3) engage with the spherical sockets of the housing (4), enabling axial and angular rotation of the spherical pin (5).

Additionally, the top of the cylindrical sleeve (2) is rigidly connected to the spherical housing (4), whose inner surface interacts with the outer surfaces of the balls (3). On the opposite end of the welded cylindrical section, a connecting sleeve (6) is rigidly installed perpendicularly to the axis within the internal opening. This sleeve interacts with the corresponding end of the spherical pin (5) of the adjacent section and is securely fastened using a bolted connection (7). A helical section (8) is rigidly welded to the outer surface of the cylindrical section.

During operation, as the screw working body rotates, the motion of the helical segments (8) is transmitted through the balls (3) to the spherical pin (5) and then to the adjacent sections of the screw assembly.

Fig. 2 presents the general view of the fabricated flexible sectional screw working body.

Figure 2. General view of the fabricated flexible sectional screw working body.

To ensure the proper functioning of the hinge joint between the sections of the working body, its structural parameters were studied. These parameters ensure mutual rotation, the required angular displacement of the axes, and the prevention of hinge joint jamming between the auger sections (Evstratov et al., 2020; Frey et al., 2012).

The structural diagram of the hinge joint of the screw working body sections is shown in Fig. 3.

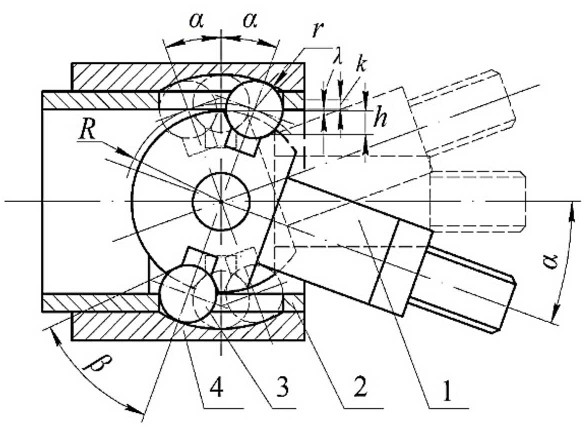

Figure 3. Structural diagram of the hinge joint of the screw working body: 1 – spherical pin; 2 – cylindrical sleeve; 3 – ball; 4 – sleeve with a spherical surface.

To determine the force parameters of the hinge joint between the sections of the screw working body, the forces acting on the ball during torque transmission were analyzed (Fig. 4).

The ball is subjected to force F_1, that originates from the conical surface of the socket and is directed perpendicularly to the surface, toward the ball's center. Force F_2 applies pressure to the tubular part of the hinge to transmit torque, while force F_3, exerted by the outer casing, acts on the ball to prevent it from disengaging.

When the hinge rotates with axes offset by an angle α, the ball moves along the surface of the casing and the groove of the tubular part. This movement occurs with sliding, leading to friction forces that depend on multiple factors. The friction force is directed along the instantaneous movement of the ball and perpendicular to the applied pressure forces (Bulgakov et al., 2024).

The friction forces F_{f1}, F_{f2} and F_{f3} (Fig. 4) change their direction during the rotation of the hinge.

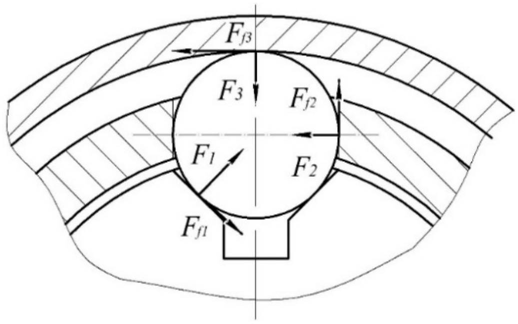

Figure 4. Schematic for determining the force parameters acting on the ball during torque transmission.

The equilibrium equation for each of the n balls during torque transmission T can be written in the first approximation as projection equations of the forces (Grote & Feldhusen, 2007), without considering the friction forces:

$$F_1 \cdot \cos\beta = F_2, \tag{1}$$

$$F_1 \cdot \sin\beta = F_3, \tag{2}$$

$$F_2 = \frac{T}{n \cdot R \cdot \cos\alpha}, \tag{3}$$

where β – the angle of inclination of the conical socket, deg; R – the radius of rotation of the centers of the balls, m.

Where:

$$F_1 = \frac{F_2}{\cos\beta} = \frac{T}{n \cdot R \cdot \cos\alpha \cdot \cos\beta}, \tag{4}$$

$$F_3 = F_2 \cdot \tan\beta = \frac{T \cdot \tan\beta}{n \cdot R \cdot \cos\alpha}. \tag{5}$$

From this, to account for the variable friction forces during the ball's sliding, it is necessary to determine the velocities and directions of its movement along the corresponding friction surfaces. For clarity, we assume that the ball in the socket is stationary, and the sliding occurs along the surfaces of the casing and the tubular part of the hinge (in the area where forces F_2 and F_3 act).

The friction forces F_{f1}, F_{f2} and F_{f3} (Fig. 4) change their direction during the rotation of the hinge. In general, the friction forces F_{f2} and F_{f3}, given the coefficients of friction for these surfaces f_2 and f_3, are determined by the following relationships:

$$F_{f2} = f_2 \cdot F_2, \tag{6}$$

$$F_{f3} = f_3 \cdot F_3. \tag{7}$$

As a result of the complex dependence between the directions of friction forces and the forces that perform useful work and ensure the transmission of the required torque T, the energy losses due to the friction was estimated. The coefficient of efficiency of the hinge was also determined at each moment in time and its average value over one full rotation of the hinge's drive shaft.

Due to the deviation of the hinge axes by an angle $\boldsymbol{\alpha}$, the ball, during its rotation, performs an oscillatory motion and slides along the friction surfaces of the tubular part of the hinge and the outer casing by an angle 2α. The radius of the arc through which the ball moves along both friction surfaces can be described using the following expressions:

$$x_2 = R \cdot \sin\alpha \cdot \cos\omega t, \tag{8}$$

$$x_3 = (R + r) \cdot \sin\alpha \cdot \cos\omega t. \tag{9}$$

where r – is the radius of the ball, m; θ – is the angular parameter of motion (the angle of rotation), deg.

The velocity of each ball's movement along the arc will be:

$$V_2 = -\omega R \cdot \sin\alpha \cdot \sin\omega t, \tag{10}$$

$$V_3 = -\omega(R + r) \cdot \sin\alpha \cdot \sin\omega t. \tag{11}$$

The power of the friction forces at each moment in time can be calculated as the product of the corresponding friction force and its speed in magnitude (the friction force always opposes the motion, so its work is negative):

$$W_2 = \left|F_{f2} \cdot V_2\right|, \tag{12}$$

$$W_3 = \left|F_{f3} \cdot V_3\right|. \tag{13}$$

The total power that the hinge must transmit is equal to the sum of the useful power ωT and the power of the friction forces that need to be overcome. The efficiency coefficient of the hinge can be calculated using the following expression:

$$\eta = \frac{\omega T}{\omega T + n(W_2 + W_3)} \tag{14}$$

From this, it follows that the torque on the pin of the flexible shaft section T_1 is greater than the drive torque $\boldsymbol{T}$ of the conveyor, according to the following expression:

$$T_1 = \frac{T}{\eta}. \tag{15}$$

Substituting the friction forces from Eqs (3) and (5), and the velocities from equations (10) and (11) into expressions (6) and (7), and after integrating, we will obtain the final expressions for determining the efficiency coefficient and the torque T_1 on the drive shaft.

$$\eta = \frac{2\pi}{2\pi + 4\left[f_2 + f_3 \cdot \left(1 + \frac{r}{R}\right) \cdot \tan\beta\right] \cdot \tan\alpha}, \tag{16}$$

$$T_1 = \frac{T\left\{2\pi + 4\left[f_2 + f_3\left(1 + \frac{r}{R}\right)\tan\beta\right]\tan\alpha\right\}}{2\pi}. \tag{17}$$

The analysis of expression (17) shows that when $\alpha = 0$, the term corresponding to the friction forces disappears, and the torque remains unchanged.

The minimum possible value of the angle of inclination of the conical socket is determined according to the expression (Hevko et al., 2017):

$$\sin\beta = \frac{\left(R\cdot\left(1-\cos\alpha\right)+k+\lambda+h\right)}{r}, \tag{18}$$

where k – required guaranteed overlap of the ball and the groove of the cylindrical bushing, m; λ – gap between the spherical pin and the cylindrical bushing, m; h – the specified depth of the ball position in the socket, m.

To select the optimal structural characteristics of the screw working body sections, computer modeling was conducted to analyze the influence of force loads on the corresponding deformations, depending on operating conditions. Using SolidWorks software package, a computer model of the working body section of a flexible screw conveyor was created, incorporating jointed transmission mechanisms (Fig. 5).

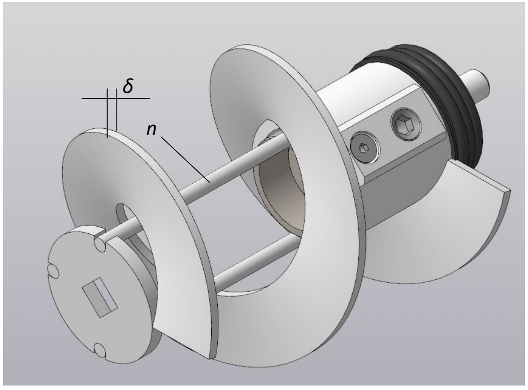

Figure 5. Section model for calculating the twist angle *φ*.

RESULTS AND DISCUSSUION

Taking into account the obtained theoretical relationships, calculations and analysis were performed to examine the change in the coefficient of efficiency η of the hinge mechanism depending on its main design parameters:

- the ratio of the ball radius to the radius of rotation of the ball centers r/R;
- the angle of deviation of the hinge axes α, deg;
- the angle of inclination of the conical socket β, deg.

The calculations were carried out for the following constant numerical parameters of the hinge, which were considered unchanged when varying one of them:

- radius ratio $r/R = 0.2$;
- friction coefficient $f = 0.32$;
- hinge axis deviation angle $\alpha = 20°$.

The angle of inclination of the conical socket β was calculated separately.

Graphical dependencies of the efficiency coefficient η on the design parameters of the hinge mechanism are presented in Fig. 6.

The calculation results show that there is a minimum radius ratio (curve 1, Fig 6) at which the efficiency coefficient of the hinge drops sharply to zero. This is explained by the impossibility of the hinge transmission mechanism functioning when the ball radius is significantly reduced. In other words, if the ball disengages from the cylindrical bushing, the calculated angle of the conical surface of the socket increases significantly. In real mechanisms, the ratio of these radii can range from 0.3 to 0.45. In this case, the efficiency coefficient η is within the range of 0.88 to 0.91 when the above-mentioned parameters are constant.

Particular interest lies in determining the dependence of the efficiency coefficient on the angle of deviation of the hinge axes (curve 2). It is evident that when the angle α is zero, the efficiency coefficient will be equal to one, meaning there will be no losses in the hinge design, and all elements of the hinge will rotate as a single unit without mutual sliding. However, an increase in the angle of axis deviation leads to mutual sliding of the balls relative to the contact surfaces, which in turn results in a decrease in the efficiency coefficient.

For angle values of axis deviation α up to 23°, the efficiency coefficient decreases insignificantly, and this reduction is linear. This means that in this range of angles, the hinge can function without causing significant loads. Further increase in the angle between the axes initially leads to a significant decrease in the efficiency coefficient, and then to the inability of the mechanism to function due to jamming and disconnection of the hinge elements. The critical value of the deviation angle α depends significantly on other design parameters of the hinge connection. To determine the operating range of a specific hinge, it is necessary to use design parameters that ensure the efficiency coefficient is at least 0.9. For example, to increase the angle of axis deviation, it is advisable to increase the ball radius without changing the radius of the cylindrical bushing. However, in any case, increasing the angle to more than 25° is impractical.

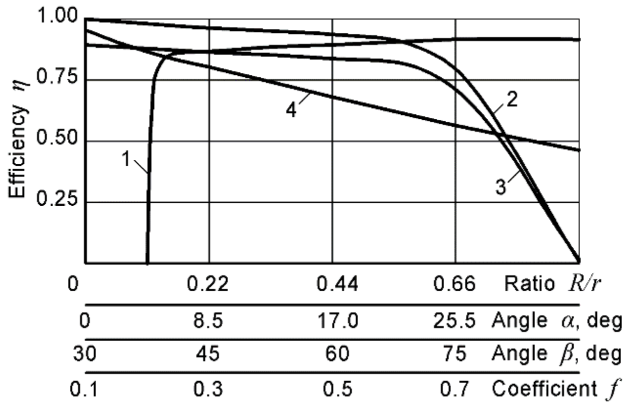

Figure 6. Dependence of the coefficient of efficiency η on: 1 – the ratio of the ball radius to the radius of rotation of the ball centers r/R; 2 – the angle of deviation of the hinge axes α; 3 – the angle of inclination of the conical socket β; 4 – the friction coefficient f.

Curve 3 in Fig. 6 describes the dependence of the efficiency coefficient on the change in the angle of inclination of the conical socket, which holds the ball in the spherical pin of the hinge. The value of this angle primarily depends on the geometric parameters of the connection.

Smaller values of the angle of inclination of the conical socket than those derived from this expression cannot be used, as in this case, the ball will not contact the conical surface but rather the edge of the socket itself. This explains the starting point of curve 3 at 30° – this is the minimum calculated value of the angle of inclination of the conical socket, at which the ball can contact the conical surface (given the specified geometric parameters of the hinge).

The calculation showed that an increase in the angle of inclination of the conical socket reduces the efficiency coefficient of the mechanism. This is related to the redistribution of forces in the transmission. Such redistribution is possible due to the increase in the radial component of the force, corresponding to an increase in frictional forces, which could lead to jamming. For the case we calculated, it was established that with the minimum value of the angle of inclination of the conical socket being 30°, it is acceptable to increase the angle up to 60°, which does not significantly reduce the efficiency coefficient and increases the radial component of the force. It is advisable to select the angle of inclination of the conical socket according to the minimum allowable value. This will allow for the maximum efficiency coefficient and minimize forces and contact stresses in the hinge.

Curve 4, shown in Fig. 6, characterizes the dependence of the efficiency coefficient on the variation of the friction coefficients *f* of the ball against the surfaces of the joint components. Calculations show that an increase in the friction coefficients *f* leads to a significant decrease in the efficiency coefficient. This dependence is nearly linear, especially in the range of low friction coefficient values. The actual values of the friction coefficients *f* in the joint, under good lubrication and sealing conditions, will not exceed 0.1. The ingress of foreign material into the joint and the absence of lubrication can increase friction forces several times, which will negatively affect the efficiency coefficient.

Based on the above, it can be stated that, from a structural perspective, the ratio of the ball radius to the radius of rotation of the ball centers r/R should be chosen as large as possible, with a minimum value of 0.35 – this has been identified as the optimum point of the two-zone curve (Nadykto & Velichko, 2014). At the same time, the inclination angle of the conical recess β should be selected as the minimum possible, considering the structural design of the joint. Additionally, during the design of the transmission, it is necessary to ensure that the deviation angle of the joint axes does not exceed 20–25°, verifying that the calculated efficiency coefficient is not less than 0.9.

When employing long transport augers composed of numerous elements, it is advisable to reduce the deviation angles between joint axes (i.e., increase the auger's bending radius). This approach enhances the overall efficiency coefficient of the entire conveyor system.

To calculate the twist angle φ of the section and obtain the corresponding graphical dependencies on the torque, experimental conditions were simulated. One intermediate element was rigidly fixed, while a torque was applied to the other. Three parameters were considered as variables: the number of base rods in the section n, the thickness of the section's screw spiral ∂, and the applied torque T.

Using MathCad software, graphical representations of the displacement of the working body elements under the applied torque were obtained. Based on the obtained values, the twist angle of the section was calculated while varying the number of rods n, the thickness of the section's screw spiral δ, and the applied torque T. The resulting graphical dependencies are shown in Figs. 7 and 8.

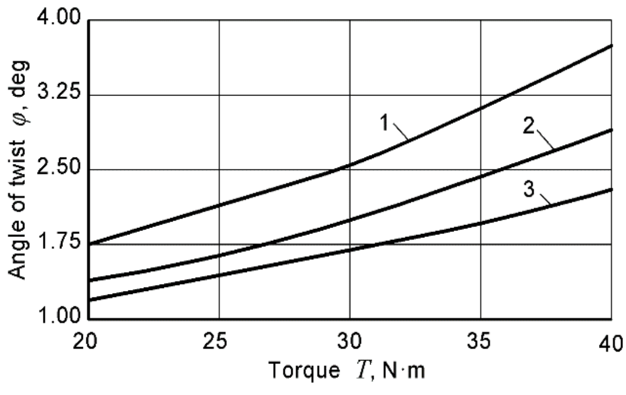

Figure 7. Dependence of the twist angle φ on the number of rods: 1 – n = 2; 2 – n = 3; 3 – n = 4.

The analysis of these dependencies indicates that as the applied torque increases, the twist angle of the sections also increases. The most significant increase is observed when the number of axial rods in the section's base is $n = 2$, while the least deformation occurs at $\delta = 4$ mm.

The research results confirm the high efficiency of the proposed design of the jointed-sectional working body of a flexible screw conveyor compared to known analogs. In particular, the obtained efficiency coefficient values (within the range of 0.88–0.91) exceed those recorded in the works of other researchers. For instance, according to Hevko et al. (2018), the efficiency of similar systems ranged from 0.82 to 0.85, which is explained by the use of less accurate methodologies and insufficient consideration of the geometric parameters of the joint.

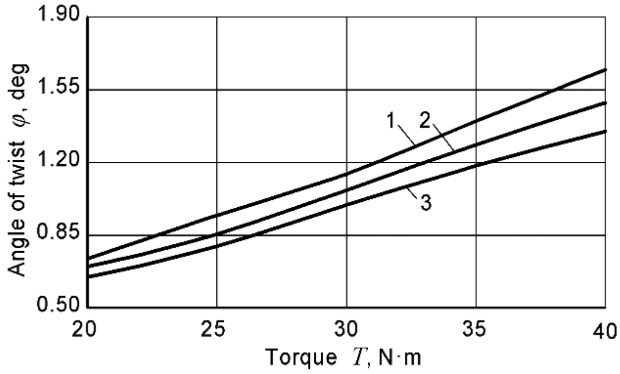

Figure 8. Dependence of the twist angle φ on the thickness of the helical spiral: $1 - \delta = 2$ mm; $2 - \delta = 3$ mm; $3 - \delta = 4$ mm.

In the study by Hevko et al. (2017), it was also noted that when the deflection angles of the axes exceed 25 degrees, there isa sharp decrease in the efficiency of the transmission, which fully correlates with the findings of this research. However, the mentioned work did not propose any constructive solutions to compensate for these losses, while our model recommends increasing the radius of the ball while maintaining the constant radius of the sleeve.

The findings of Bulgakov et al. (2022) also confirm the influence of the conical socket angle on the distribution of forces and the potential jamming of the mechanism. However, the study did not establish specific limit values for this parameter. In contrast, our study determines that the minimum permissible socket angle is 30 degrees, and it is recommended not to exceed 60 degrees, which helps avoid excessive growth of the radial component of the force.

Thus, the conducted study not only aligns with the conclusions of previous authors but also significantly complements them by offering optimization solutions based on numerical modeling and analytical dependencies.

As a result of this theoretical study, it can be concluded that the newly developed design of the articulated sectional working body of the flexible screw conveyor enhances operational efficiency and expands technological capabilities for transporting bulk agricultural materials, particularly grain. For the transportation of technological agricultural materials, it is advisable to use the developed flexible screw conveyor with a sectional auger working body, which improves the efficiency of loading and unloading operations from the truck body.

Further investigation is recommended into the dynamic behavior of articulated joints under variable load conditions, especially in high-frequency oscillatory regimes, to evaluate long-term wear and fatigue resistance of the contact elements. Additional finite element modeling (FEM) is suggested to simulate the stress-strain behavior of screw sections during dynamic loading, especially in transition zones between articulated elements, to prevent structural failure and optimize material distribution.

CONCLUSIONS

1. To ensure the proper functioning of the articulated joint, the influence of its structural parameters on enabling mutual rotation, the required angular displacement of the axes, and the prevention of jamming were investigated.

2. An analytical dependence was derived for determining the arc displacement of the ball along both friction surfaces. The velocity of each ball's movement along the arc of the spherical sleeve, as well as the efficiency coefficient of the joint and the torque magnitude on the drive shaft, were determined. Based on the obtained analytical dependencies, the variation in the efficiency coefficient of the articulated mechanism has been analyzed in relation to key structural parameters: the ratio of the ball radius to the cylindrical sleeve radius at the contact point, the angular deviation of the joint axes, and the conical surface angle of the socket. It was established that the transmission efficiency reaches its maximum value in the range of 0.88–0.91 when the ratio of the ball radius to the rotation radius of the ball centers is within 0.3–0.45.

3. To determine the operating range of a specific joint, structural parameters should be used that ensure an efficiency coefficient of at least 0.9. The calculations showed that increasing the conical socket angle reduces the mechanism's efficiency. The conical surface angle of the socket should be set to the minimum possible value for the selected joint design. For long transport screws with a large number of segments, it is recommended to reduce the inclination angles of the axes (i.e., increase the screw's bending radius) to enhance the overall efficiency coefficient of the entire conveyor system. A sharp bend in the working body's pipe, even at a single point, may lead to the entire mechanism jamming.

4. To select the optimal structural characteristics of the screw working body sections, computer modeling of the influence of force loads on the corresponding deformations under different operating conditions was conducted.

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