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Oleksandra Trokhaniak*
PhD in Technical Sciences, Associate Professor
National University of Life and Environmental Sciences of Ukraine
03041, 15 Heroiv Oborony Str., Kyiv, Ukraine
https://orcid.org/0000-0002-4671-5824

Substantiation of the rational parameters of the hinged working bodies of the flexible screw conveyor

Abstract. The relevance of this study lies in the need to improve, develop innovative designs, and determine the best structural and kinematic parameters of the hinged sections of the screw working bodies of the conveyors, to obtain better functional and operational characteristics in transporting bulk and lumpy materials and granular fertilizers. Therefore, the purpose of this study was to improve the performance of flexible screw conveyors by developing and determining the best parameters of hinged flexible screw sectional working bodies for transloading bulk materials along curvilinear transportation branches. The study was conducted using the methods of the theory of mathematical and computer modelling, the method of mathematical planning of the experiment. The paper presents innovative designs of hinged screw working bodies and substantiates their best parameters for ensuring the movement of bulk materials along curvilinear routes. The study found the correlation between the magnitude of the torque that occurs between the loaded screw sections and their spatial arrangement. Analytical dependences were obtained for the stiffness conditions of a separate section with a hinged connection, depending on the load and design parameters of the conveyor. It was found that the angular displacement of the sections according to the state of dependence helps to significantly (up to 4-5 times) reduce the amplitude of the change in the torque of the drive and lowers its maximum value. It was proved that in the pushing mode, energy consumption is 1.06...1.4 times higher than in the pulling mode. The practical significance of the results lies in the ability to choose the optimal structural, kinematic, and technological parameters of the developed sections of the screw working body for transporting bulk or lumpy materials and mineral fertilizers

Keywords: screw conveyors; curvilinear branch of transportation; power parameters; bulk material; transloading gear

INTRODUCTION

The transportation of bulk materials requires compliance with standards related to maximum productivity in the execution of the technological process, minimization of energy consumption and damage to the bulk material. The execution of such technological operations during their movement on curvilinear branches in closed elastic fixed casings, carried out by screw conveyors, allows for high mobility of loading and unloading operations (Bulgakov *et al.*, 2017; Çelik & Güneş, 2017; Baranovsky *et al.*, 2018).

V.G. Gorobets *et al.* (2018), B.M. Hevko *et al.* (2018), V. Kuvachov *et al.* (2021) noted the specific features of using screw conveyors for transporting or lifting bulk materials over short and medium distances. These studies also analysed the influence of such parameters as the rotating speed of the screw on the performance of the screw conveyor; angle of inclination of the conveyor feeder; volumetric filling level of bulk material. Operating conditions that affect the performance of a screw conveyor are considered by applying the Discrete Element Method (DEM) to simulate a single-stage screw conveyor with periodic boundary conditions. S. Xiaoxia *et al.* (2017), Y. Tian *et al.* (2018) established that the uniformity of the flow depends on the

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*Corresponding author

design of the screw working body and the hopper, the shape of the particles, and the force of friction against the walls. The results of this study improve the representation of the use of screw conveyors for particular materials.

R.B. Hevko *et al.* (2017), R.B. Hevko *et al.* (2018), V. Kaminskyi *et al.* (2021) proposed mathematical models of the screw conveyor loading process. The use of the proposed models creates conditions for the development of innovative designs of screw conveyor devices with substantiation of their rational parameters. H. Li & W.F. Liu (2012), M.R. Hevko (2013), O.L. Lyashuk *et al.* (2015) demonstrated the operating modes of an inclined screw conveyor that contains a helical working element with constant parameters. The kinematics of the grain load was studied based on the equations of motion in the screw conveyor. The movement of the freight in the established high-speed mode was analysed.

The studies of A.S. Merritt (2008), P.J. Owen & P.W. Cleary (2009), P.J. Owen & P.W. Cleary (2010) showed damage to grains when processing in a screw conveyor and estimated the moisture content of the bulk material (at three levels of 8%, 11%, and 14%), the gap between the working body and the casing (at four levels of 6 mm, 9 mm, 12 mm, and 15 mm) and screw rotation speed (at five levels of 100 rpm, 200 rpm, 300 rpm, 400 rpm, and 500 rpm). J.W. Fernandez *et al.* (2011) proposed a theoretical model of power consumption of a screw conveyor with flexible discreteness of spiral elements. Simulation results suggest that increasing the spiral angle will increase the particle transport rate. Thus, it is necessary to improve and develop innovative designs of screw working bodies and substantiate the best structural and kinematic parameters of hinged sections of conveyor screws, which can ensure the improvement of functional and operational indicators of transporting bulk and lumpy materials and mineral fertilizers.

The purpose of this study was to improve the functional and operational characteristics of transporting bulk and lumpy materials and mineral fertilizers by improving and developing innovative designs of screw working bodies with a hinged connection of their sections and substantiating the best structural and kinematic parameters of the screws.

The originality lies in the establishment of patterns of changes in the kinematic and power parameters of the hinged screw working bodies of the augers for various load conditions and operating modes, as well as the distribution of stresses and deformations in the screw sections and the influence of the ratio of the parameters of the screw elements and axial plates on the twist angle and, accordingly, the bearing capacity of the section shaft and regularities.

MATERIALS AND METHODS

To improve the performance of screw conveyors, which lie in ensuring the transportation of bulk materials along curved routes, increasing the load-bearing capacity, as well as improving their maintainability, an auger with hinged sections was developed. Its scheme is presented in Figure 1.

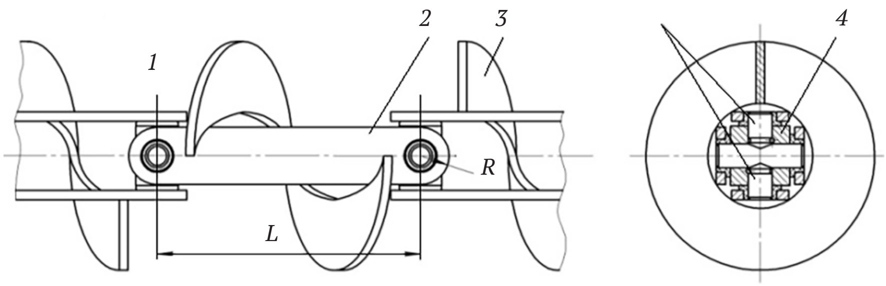

Figure 1. Design scheme of the developed auger with hinged sections
Note: *1* – plate; *2* – spiral rib; *3* – radial trunnion of the finger; *4* – intermediate frame, *L* – length of the hinged section; *R* – radius of the side surfaces of the hinged plates
Source: R.B. Hevko *et al*. (2018)

During the rotation of the working element in the elastic fixed casing between the sections, the torque is transmitted using plates 1 with a rib 2, which are rigidly fixed on a pair of radial pins of the finger 3 and the intermediate frame 4. During the operation of the conveyor along curvilinear routes, the radial fingers rotate, which ensures transmission of torque and helps transport bulk material with helical spirals.

The analysis of the technical characteristics of various brands of screw conveyors suggests that for the same structural dimensions (diameter d=100 mm and screw pitch t=70...80 mm), kinematic parameters (screw rotation speed n=400...700 mm) and technological characteristics (the angle of inclination of the working body to the horizon α⩽40o) the productivity of the conveyor during hopper loading of grain material for its bulk mass of 0.75 t/m3 is 20...25% higher compared to a self-loading nozzle (Roberts, 2015).

To increase the length of the technological branch in the central part of the conveyor, a transloading nozzle is installed, as well as drive nodes of the flexible working bodies of the loading and unloading tracks. This helps substantially reduce the torques on the working bodies at the points of attachment to the drive shaft. To conduct experimental research, a test bench was developed and manufactured, its design scheme is presented in Figure 2.

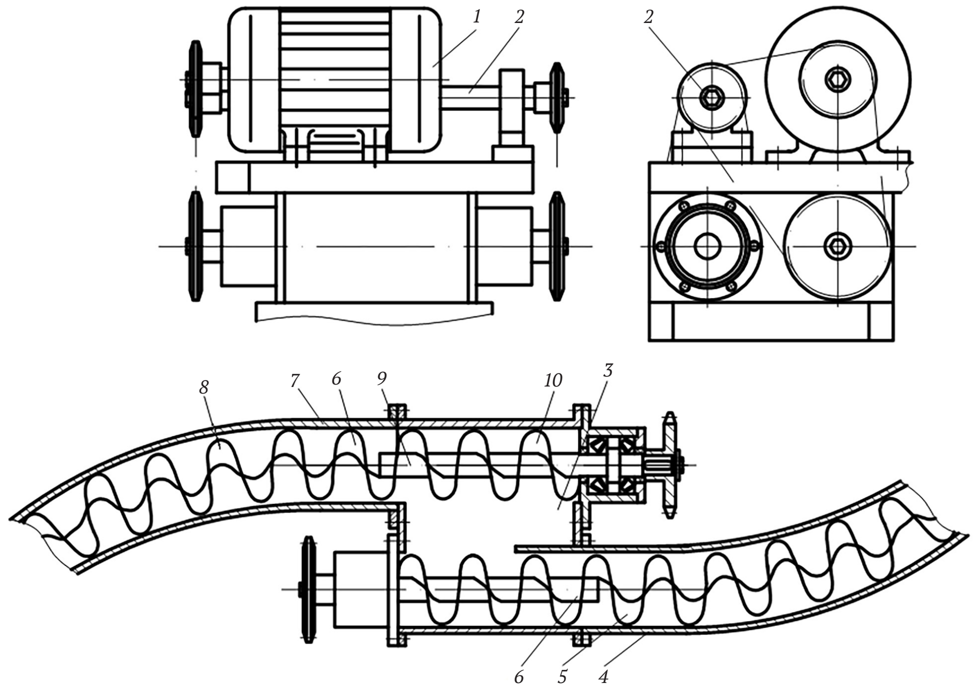

Figure 2. Schematic diagram of the flexible screw conveyor test bench
Source: A.W. Roberts (2015)

The overloading feeder of the test bench of the flexible screw conveyor (Fig. 2) contains the loading and unloading branches, which are designed in the form of casings 7 and 4, and the loading 8 and unloading 5 flexible screw bodies are placed in them in parallel. Screws, respectively, are fixed on the drive shafts 6 and 9, which are located in the overload feeder 3. Therewith, the intermediate shaft 2 is located parallel to the drive shafts and fixed on the overload feeder. The drive and intermediate shafts outside the overload feeder are kinematically connected to the shaft of electric motor 1.

During operation, the drive shafts are kinematically connected, the electric motor shaft and the intermediate shaft allow the working flexible working bodies to rotate. When feeding bulk or lumpy material, auger 8 transports it in a stationary casing towards the overload feeder. Because the drive shafts are located in a horizontal plane, the material to the unloading branch is moved via radial flat plates. The basis of the development of the hinged working bodies of flexible screw conveyors is the principle of the functioning of Hooke's spatial hinges of the polycardan type (Hevko, 2013). The angular displacement of the following after the first section is determined by the equations as follows:

$$\begin{aligned}\varphi_2 &= \operatorname{arctg}\left[\left(\cos\alpha_{1,2}\right)^{-1}\operatorname{tg}(\varphi_1+\psi_1)\right]-\\ &-\operatorname{arctg}\left[\left(\cos\alpha_{1,2}\right)^{-1}\operatorname{tg}(\psi_1)\right];\\ \varphi_2 &= \operatorname{arctg}\left[\left(\cos\alpha_{2,3}\right)^{-1}\operatorname{tg}(\varphi_2+\psi_2)\right]-\\ &-\operatorname{arctg}\left[\left(\cos\alpha_{2,3}\right)^{-1}\operatorname{tg}(\psi_2)\right];\\ \varphi_{n+1} &= \operatorname{arctg}\left[\left(\cos\alpha_{n,n+1}\right)^{-1}\operatorname{tg}(\varphi_n+\psi_n)\right]-\\ &-\operatorname{arctg}\left[\left(\cos\alpha_{n,n+1}\right)^{-1}\operatorname{tg}(\psi_n)\right]; \qquad (1)\end{aligned}$$

where φ_i is the angle of rotation of the section in the selected frame of reference; $\alpha_{i,i+1}$ is the angle between the axes of the sections in the plane that passes through the axes of the sections; i is the section number, $i=1,2...n$; ψ_i is the phase angle of the section hinge.

The angular velocity and angular acceleration were equal to:

$$\omega_{n+1}=\frac{ac\omega_1}{a^2\sin^2\varphi+(b\sin\varphi+c\cos\varphi)^2}, \qquad (2)$$

where ω_{n+1} is the angular speed of the next section; ω_1 is the angular speed of the driving section.

In the case when ω_1=const, the angular acceleration of the section is determined according to the following expression:

$$\varepsilon_{n+1}=-\frac{ac\omega^2[(a^2+b^2-c^2)\sin 2\varphi+2bc\cos 2\varphi]}{[a^2\sin^2\varphi+(b\sin\varphi+c\cos\varphi)^2]^2}, \qquad (3)$$

where a, b, c are parameters of the model, which a determined through quantitative and spatial placement of sectional overcut elements:

$$a=a_1a_2...a_n;\ b=a_1a_2...a_{n-1}b_n+a_1a_2...a_{n-2}b_nc_{n-1}+...$$
$$+a_1b_2c_3...c_n+b_1c_2..c_nb;\ c=c_1c_2...c_n;\ a_1=\cos\alpha_{i,i+1}\ 1+tg^2\phi_i);$$
$$c_1=\cos\alpha_{i,i+1}+tg^2\psi_i.$$

The considered power parameters during the twisting of the first and, respectively, the second section of the auger (Fig. 3):

The torque applied to the driven section from the driving section M_{21} is determined from the known dependence:

$$M_{21} = M_k\sqrt{1+\tan^2\alpha_{1,2}\cos^2\phi}, \quad (4)$$

where M_k is the torque required to move the freight by one section.

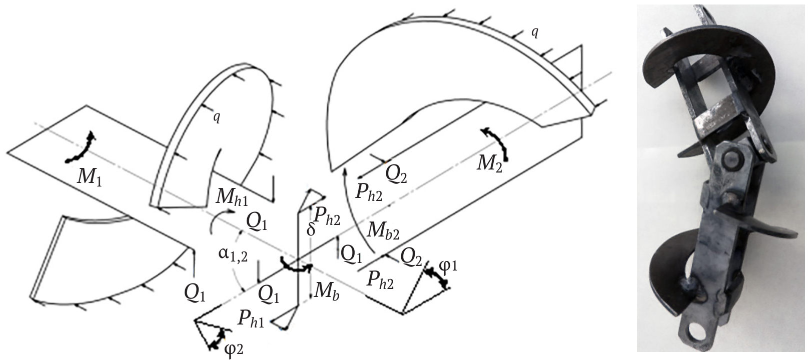

Figure 3. Placement of forces during a static load on the crosspiece of the interfacial cross-shaped hinged rotary mechanism of the sectional working body

Note: M_1 is the torque of the driving section; M_2 – torque of the driven section; M_b is the bending moment; M_{h1} is the torque of the interfacial flexible rotary mechanism of the driving section; M_{h2} is the torque of the interfacial flexible rotary mechanism of the driven section; Mb1 is the bending moment that occurs in the plane of the misalignment of the driving section; M_{b2} is the bending moment that occurs in the plane of the misalignment of the driven section; Q_1 is the cutting force of the driving section; Q_2 is the cutting force of the driven section; P_{h1} is the concentrated force of the interfacial flexible rotary mechanism of the driving section; P_{h2} is the concentrated force of the interfacial flexible rotary mechanism of the driven section; q is the distributed load affecting the spiral; $\alpha_{1,2}$ is the angle between the axes of the driving and driven sections in the plane that passes through the axes of the sections; φ_1 is the angle of rotation of the driving section; φ_2 is the angle of rotation of the driven section; δ is the lever where a pair of forces is applied

Source: M.R. Hevko (2013)

Accordingly, the torque from the drive M_{n1}, applied to the n^{th} (driving) section on the section from the track of constant curvature and, therefore $\alpha_{n+1,n}=\alpha_{n,n-1}$:

$$M_{n1} = M_k\{1 + \\ + \prod_{m=1}^{n-1}\sqrt{1+\tan^2\alpha_{1,2}\cos^2[\phi - \pi\lambda(m-2)/(n-2)]}\}, \quad (5)$$

where λ is the parameter of the angular displacement of one section relative to another, *n* is the number of sections.

If the axes of each of the sections are placed parallel to each other, then the maxima of the alternating torque for all sections will coincide, and the drive will have maximum torque values with the maximum amplitude of their oscillations (Liashuk *et al.*, 2015). Figure 4 shows the graphs of the torque applied to a flexible screw, from the angle of its rotation along a curved path with a constant angular displacement between sections $\alpha_{n,n-1}=\pi/6$. The number of moving sections in the auger here is 8. The eighth section is fixedly connected to the drive, accordingly M_{81} is the drive moment. The moment needed to move the freight by one section was assumed to be M_k=100 (units or %). Accordingly, in Figure 4a sections were placed without angular displacement, while in Figure 4b – with an offset $\Delta\varphi_{n,n-1}=\pi/6$ in relation to the previous section. We assume that between the first and second sections $\Delta\varphi_{2,1}=0$.

It was found that the rational values of the angular displacements of the sections depend on their number in the transportation screw and is determined by the dependence $\alpha_{n,n-1}=\pi/(n-2)$.

Given that, according to dependencies (2) and (3), the kinematic and dynamic parameters of the sectional screw conveyor change, the costs of transporting freight along curvilinear paths associated with its periodic acceleration increase accordingly. And therefore, the effect of rational angular displacement of sections will be even greater.

The force is calculated according to the following formula:

$$P_{h1}=0{,}5M_{h1}/\delta, \quad (6)$$

where δ is the lever where the pair of forces is applied (Fig. 3).

As a result of the inclination of the axes of the overcut of the sections, there is a bending moment M_{b1}, which is applied in the planes of the skew of the sections, it is found from $M_{b1}=M_1 \mathrm{tg}\alpha_{1,2}\cos\varphi$.

The bending moment acting in the overcut of the driven section was determined according to the formula:

$$M_{b2}=M_1\,\mathrm{tg}\alpha_{1,2}\sin\varphi(1-\sin^2\alpha_{1,2}\sin^2\varphi)^{0.5}. \quad (7)$$

In the case when it is necessary to increase the rigidity of the sections of the auger, it is proposed to use a design with rigid fastening of the screw strip to the shaft plates, which forms a kind of frame that enables a considerable increase in the rigidity of the section. The general appearance of the specified section and the model for finding its torsion angle are presented in Figure 5.

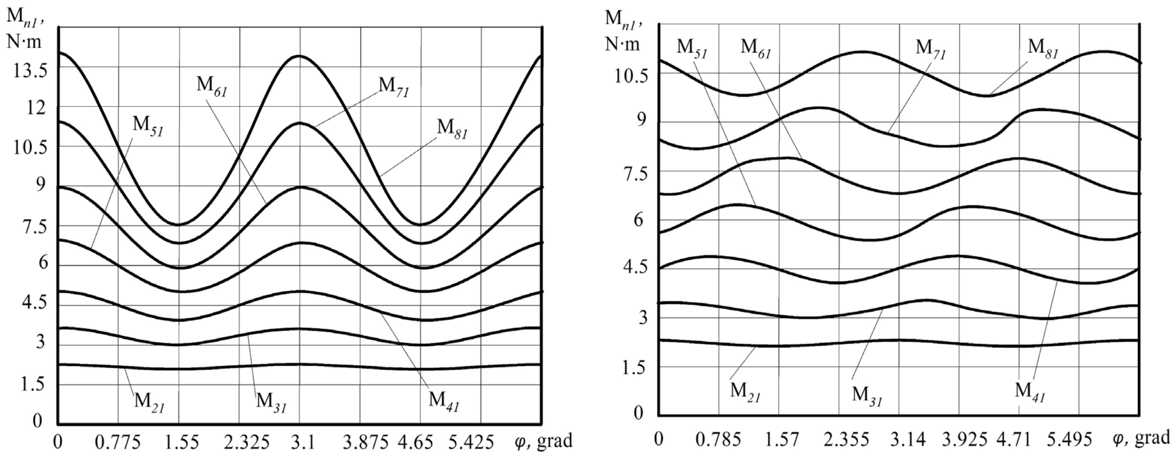

Figure 4. Change in torque on the drive 8 of the sectional shaft of the flexible screw from the angle of rotation of the screw, without angular displacement of sections, and with angular displacement $\alpha_{n,n-1}=\pi/6$

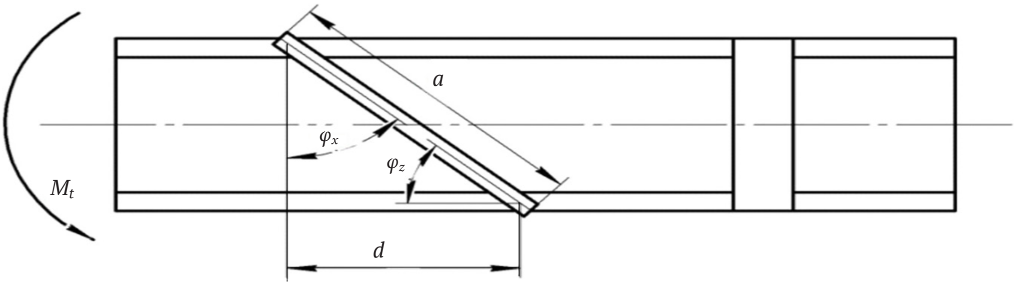

Figure 5. Design model for the determination of its torsion angle

Source: M.R. Hevko (2013)

Section stiffness condition for the permissible torsion angle:

$$\varphi = \frac{M_t l}{G J_d\left(1+\frac{E F_p (2\omega \cos\varphi_z - ed\cos\varphi_x)^2}{m d G J_d}\right)} \leq [\varphi], \tag{8}$$

where M_t is the applied torque; l is the length of the sectional connection; m is the length of the rod between its attachment points; d is the distance between the points of the inclined rod; E, G are the elasticity moduli of the first and second kind, respectively; J_d is the moment of inertia of the rod section; F_p is cross-sectional area of an inclined rod; ω is sectoral coordinate of the attachment point, m^2; φ_z, φ_x are the angles of inclination of the rod to the base of the section; e is distance from the centre of the bend to the plane.

A test bench was developed for conducting experimental research, the general view of which is presented in Figure 6.

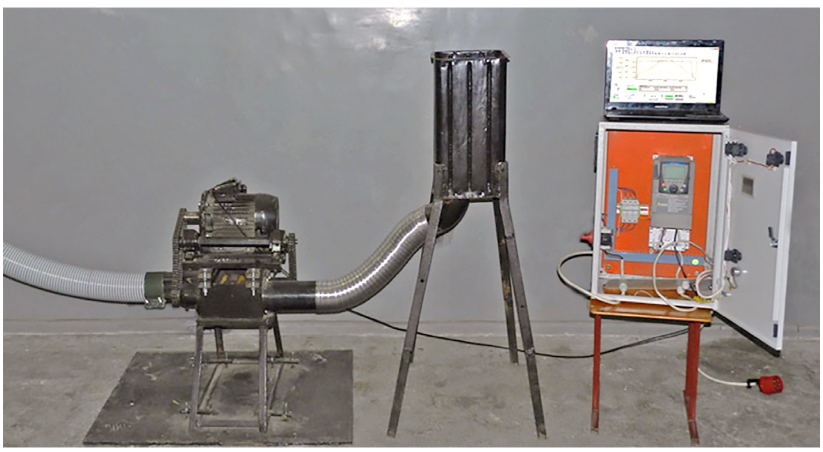

Figure 6. Test bench for conducting research

Source: developed by the author

The work of the test bench was as follows. Grain or bulk material was poured into the hopper and then moved to the unloading area. To start the engine and control its angular velocity, rotation was carried out using the Altivar 71 frequency converter and the Power Suite v.2.5.0 software. After the end of the material overload process, the curves of changes in torque and engine power over time were obtained on the computer display in the Power Suite program window.

The distance between the ends of the adjacent edges of the spiral Δ is adjusted by changing the length of the base of the shaft. It consists of hinged helical sections that have the form of two parallel flat plates, and helical ribs are attached to them. Flat plates, via anti-friction bushings, interact with the hinge element 4, which is made in the form of a square. A number of radial holes are made in mutually perpendicular planes in the hinge element, and the fingers are placed in them, with the fingers also placed on flat plates.

In the case of repositioning the fingers in the radial holes in the hinge element with their subsequent fixation, we can obtain adjustment of the length Δ of the screw pitch.

Thus, the control of the length Δ between the screw pitch can be obtained by using square bases of variable length, on the sides of which mutually perpendicular holes are made.

Parameters of the working element: external diameter of the rib – 96 mm; the inner diameter of the auger rib is 46 mm; rib pitch – 80 mm.

The screw rotation speed was adjusted using the Altivar 71 frequency converter. Tests were performed for three of its sizes: 14 mm, 28 mm, and 42 mm. With an increase in the length Δ of the pitch and the same length of the working body, its mass decreased and was 7.38 kg, 7.04 kg, and 6.64 kg, respectively. Therewith, the distance of overloading of agricultural material was 1.5 m.

RESULTS AND DISCUSSION

According to the research results, the curves of changes in the peak values of the torque *T* and engine power *N* depending on the increase in the rotation frequency of the screw *n* at different values of Δ in the pulling (Fig. 7a) and pushing (Fig. 7b) modes were obtained.

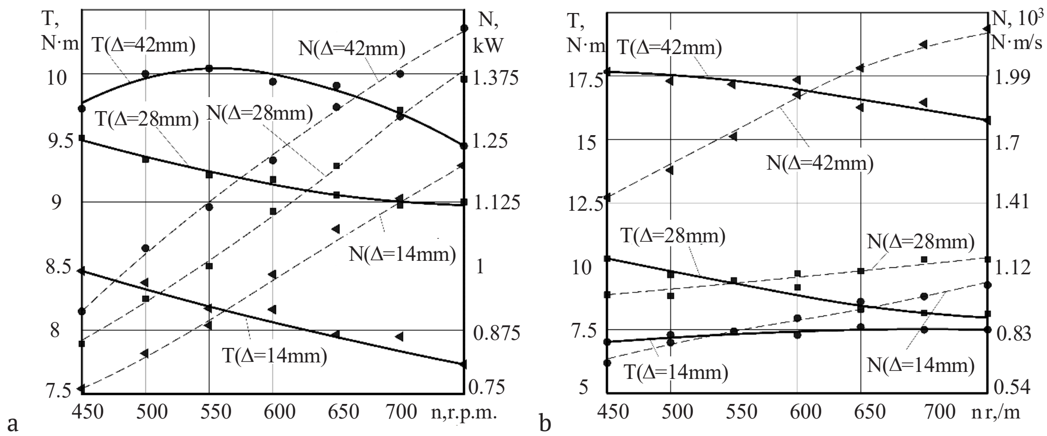

Figure 7. Graphs of changes in power *N* and torque *T* as a function of the rotation frequency of the flexible screw *n*, in steady (a) and pushing (b) modes during grain movement

Source: developed by the author

The analysis of the obtained curves shows that in the case of an increase in the rotation frequency *n* of the screw in the range from 450 to 750 rpm, the torque *T* tends to decrease. In the case of consideration of power *N*, the graphs were obtained with a linear nature of the growth of *N* as the value of *n* increases.

In this range of rotation frequency of the working body, bulk or lumpy material passes the length Δ not only due to kinetic energy, which is obtained through the interaction with the screw ribs, but due to the constant supply of material from the intake zone as well. An increase in the distance Δ between the ends of adjacent helical ribs leads to a relative increase in the values of torque *T* and power *N* because in transportation, part of the material accumulates between adjacent helical ribs, which adversely affects energy consumption.

Tests were also performed to establish the influence of the radius of curvature of the technological branch R_k and the height of the lifting of bulk material *h* on the value of the torque *T* and power *N* on the drive of the flexible conveyor during the movement of such bulk and granular materials as industrial salt, wheat, and peas.

Since the influence of the value of the rotation frequency of the flexible conveyor screw on the value of the torque had already been determined, the rotation frequency of the flexible screw was not changed, and its value was 450 rpm.

Based on the obtained results of the experimental tests, graphical dependences of the torque on the drive of the flexible auger on the height *h* of the transportation of agricultural material (Fig. 8) and on the radius of curvature R_k of the technological branch of the transportation (Fig. 9) were constructed.

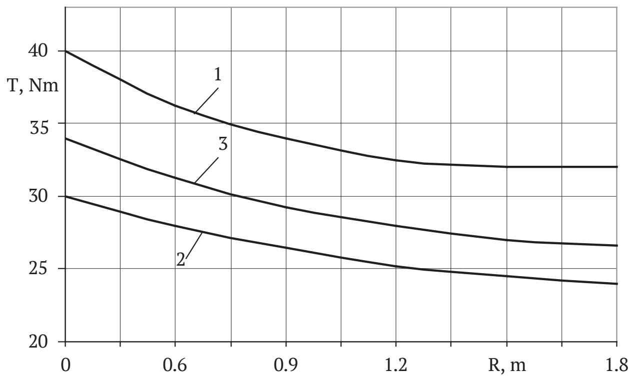

Figure 8. Variation of the torque T on the flexible screw from the radius of curvature of the transloading branch R_k for n=450 rpm for varied materials

Note: 1 – industrial salt; 2 – grain; 3 – peas

Source: developed by the author of this study

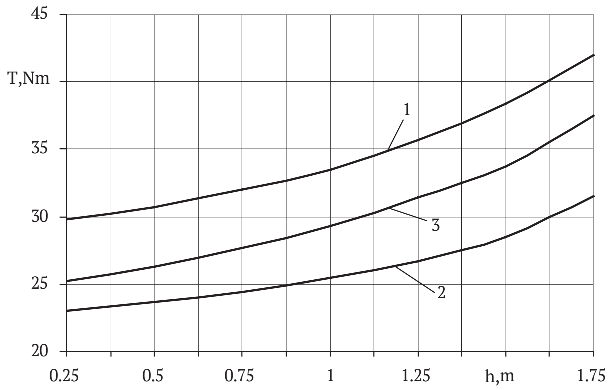

Figure 9. The change in the torque T on the flexible auger from the height of the agricultural material transloading h for n=450 rpm

Note: 1 – industrial salt; 2 – grain; 3 – peas

Source: developed by the author of this study

The analysis of the obtained graphs (Fig. 8) suggests that an increase in the curvature radius of the technological branch of transportation R_k in the range of 0.6…1.8 m leads to a decrease in the torque T; therewith, for wheat ΔT=14%, for peas – ΔT=15%, and for technical salt ΔT=10.2%.

The analysis of the graphs (Fig. 9) suggests that an increase in the height of transportation of agricultural material h within the range of 0.25…1.75 m leads to an increase in the torque T; therewith, for wheat ΔT=24.5%, for peas ΔT=30%, and for technical salt ΔT=29%.

From the data for conducting a multifactorial experiment, a regression dependence was derived to establish the influence of n, Δ, and h (height of lifting of agricultural material) on the torque T during grain transloading (Molodetska, 2021).

$$T=-0{,}667442+0{,}012691n+17{,}064586h+0{,}118525\Delta-0{,}000005n^2-0{,}005277nh-0.000438n\Delta-0{,}827612h^2-0{,}023019h\Delta+0{,}014866\Delta^2. \quad (9)$$

Figure 10 shows the response surfaces of changes in T values due to changes in two factors.

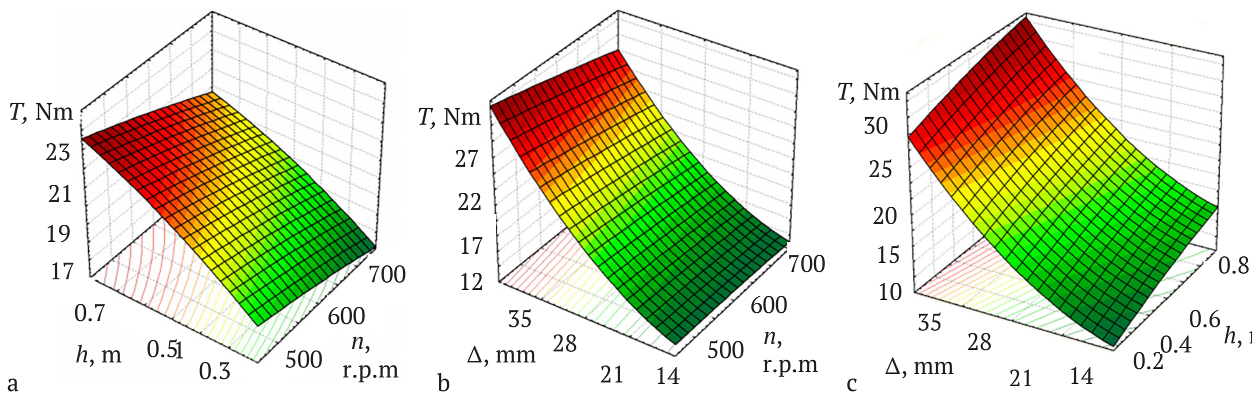

Figure 10. Response surfaces of a drive shaft torque from variable factors

Note: a: $T=f(h,n)$; b: $T=f(\Delta,n)$; c: $T=f(\Delta,h)$

Source: developed by the author of this study

It was determined that the dominant factor affecting the value of the torque T is the value of the gap Δ and n; however, the height of lifting the load h also has a substantial effect on the value of T.

The factor field was determined by the following range of parameter changes: 400 (rpm)<n<700 (rpm); 0.2 (m)<h <1.2 (m); 0.014 (m)<Δ<0.042 (m).

A test was conducted to determine the efficiency of a screw conveyor with a flexible working element. The parameters of the developed working element were as follows: D_k=100 mm; D=96 mm; d=46 mm; p=80 mm. The research was carried out using bulk materials of the appropriate bulk mass: wheat – 720 kg/m^3; peas – 728 kg/m^3; compound feed – 555 kg/m^3; bran – 252 kg/m^3.

It was determined that the maximum efficiency of the conveyor for transporting materials of larger bulk mass (peas, wheat) corresponds to the rotation frequency of the screw working body within 650…670 rpm and their values are similar: (7…7.2 m^3/h; 5…5.2 t/h). A further increase in the rotation frequency n of the working element causes a decrease in the efficiency of the conveyor, which can be explained by a smaller volume of material intake. Having a greater bulk weight, the material reacts more slowly and partially returns to the hopper. In the case of lighter materials (combined feed, bran) in this range of changes in the frequency of rotation of the working element, an increase in efficiency factor is observed and their nature is linear.

The analysis of theoretical studies of the working bodies of flexible screw conveyors (I. Çelik & E.C. Güneş (2017), V. Kuvacov *et al.* (2021)) suggests that most authors derived analytical dependencies to establish operational, structural, and functional parameters of flexible screws. H. Li & W.F. Liu (2012), O.L. Lyashuk *et al.* (2015) proposed mathematical models of the vertical screw conveyor loading process. An algorithm for solving differential equations with partial derivatives of the movement of a fluid medium in a screw channel has been developed. The use of the proposed model helps develop innovative designs of intake devices of screw conveyors with substantiation of their rational parameters.

Therewith, the issues regarding the substantiation of the parameters of the working bodies, which ensure their high operational functional and characteristics at the minimum permissible weight of the auger structure, are not resolved. Solving these problems will help significantly reduce energy costs for the process of moving, the degree of damage to bulk and lumpy agricultural material.

Studies presented in the studies by M.R. Hevko (2013), V.M. Baranovsky *et al.* (2018), and the proposed type of screw working body are characterized by 8-12 times lower energy consumption for the transportation of bulk materials.

Comparing the performance of a rigid (horizontal, inclined, and vertical) screw working body of the conveyor, the study of which is described in the articles of A.S. Merritt (2008), S. Xiaoxia *et al.* (2017), Y. Tian *et al.* (2018), it can be stated that the proposed design of the working body is different in that it helps transport materials along curvilinear tracks, which considerably increases the productivity of the flexible screw conveyor. H. Li & W.F. Liu (2012), M.R. Hevko (2013), O.L. Lyashuk *et al.* (2015) proposed the use of a flexible tubular screw conveyor to extend the range of self-loading. The main shaft and outer tube are made of rubber, and the propeller blade is made of continuously rigid plastic. The conveyor is commonly used in a wide range for flexibility. A method of reducing the wear of the screw conveyor is proposed. Furthermore, the change in energy consumption of the conveyor transmission under the influence of the design parameters of the screw was analysed.

In the study of D. Mondal (2018), the filling factor of the screw conveyor is a key point for the transmission efficiency. It is proposed to determine it using a ribbon helix with different screw speeds. Design parameters and rotation speed are optimized using EDEM simulation software. The theoretical results are then verified experimentally.

İ. Çelik & E.C. Güneş (2017) suggested a flexible discrete screw conveyor with spiral blades. The parameters of energy consumption when using a flexible helical blade structure were investigated employing theoretical analysis and simulation experiments.

The study results presented in this paper relate to helical spirals without the use of a flexible shaft. However, based on the practical observations of S. Xiaoxia *et al.* (2017), P.J. Owen & P.W. Cleary (2010), during the transportation of material along a curvilinear route (especially with small radii of curvature equal to about 1.5 m) due to the presence of cyclic deformations caused by the presence of different signs during the screw rotation, destruction occurs, which significantly limits the use of such screws due to non-performance operational requirements. Whereas the proposed working body can work at a minimum radius of curvature in the technological line of about 0.5 m, which considerably increases the mobility of the flexible screw conveyor. Therefore, to increase the reliability of the technological process during the development and research of working bodies, it is necessary to use a sectional method of manufacturing flexible screws, which will help eliminate sign-changing cyclic loads; to increase the contact area between the torque transmission nodes, which allows reducing the internal stresses and forces in the friction pairs due to the increase in the torque transmission arm.

Based on the obtained analytical dependencies, for the selection of rational design parameters of flexible sectional screw conveyor working bodies, the interrelationship of power characteristics was established, proceeding from the conditions of its operation, the laws of influence of structural nature and force. The dynamic and technological parameters of the developed flexible sectional screw working bodies were established according to the functional and performance indicators of the flexible screw conveyor process.

CONCLUSIONS

To improve the performance of screw conveyors, namely, to ensure the transportation of bulk materials along curvilinear routes, to increase the load capacity, as well as to increase their maintainability, a hinged screw working element was developed. The performed theoretical calculations of the structural and kinematic characteristics of the hinged sections of the screw helped determine the relationship between the magnitude of the torque that occurs between the loaded screw sections and their spatial location. Analytical equalities of the condition of stiffness of a separate section and elements of the hinged connection depending on the forces and structural dimensions of the working body were derived.

It was found that the angular displacement of the sections according to the dependence obtained in the study allows reducing both the amplitude of the change of the drive torque (up to 4…5 times) and its maximum value. A study was conducted using the developed test bench and a hinge-screw working body to determine the torque T and the power N. Based on the conducted multifactorial experiment, a regression relationship was obtained to determine the effect of the rotation frequency of the working body n, the value Δ and the height of the material lift h on the value of the torque moment T during grain material transportation. Its analysis strongly suggests that the dominant factor affecting the value of T is the setting of the clearance Δ, followed by the value of n and the height of the material lift h.

Based on the conducted set of theoretical and experimental studies, a competitive sectional screw working body was created, which allows improving the functional and performance characteristics of screw conveyors for transporting bulk and lumpy agricultural materials. The use of the proposed sectional screw working bodies in the constructions of conveyors, considering the recommended parameters, will allow transporting agricultural materials efficiently along curvilinear tracks.

In the future, it is necessary to perform calculations and analyse the change in the efficiency factor of the hinged mechanism depending on the main design parameters. To establish the feasibility of using long conveyor screws with a significant number of elements, to determine the limits of increasing the bending radius of the screw in this case to improve the overall efficiency factor of the entire conveyor.

CONFLICT OF INTEREST

The author declares no conflict of interest.

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Олександра Миколаївна Троханяк

Кандидат технічних наук, доцент
Національний університет біоресурсів і природокористування України
03041, вул. Героїв Оборони, 15, м. Київ, Україна
https://orcid.org/0000-0002-4671-5824

Обґрунтування раціональних параметрів шарнірно з'єднаних робочих органів гнучкого гвинтового конвеєра

Анотація. Актуальність дослідження полягає в необхідності вдосконалювати, розробляти нові конструкції та визначати оптимальні конструктивні та кінематичні параметри шарнірно з'єднаних між собою секцій шнекових робочих органів конвеєрів, з метою отримання кращих функціональних й експлуатаційних характеристик процесу транспортування сипких та кускових матеріалів і гранульованих добрив. Тому метою проведеного дослідження було покращення експлуатаційних показників гнучких шнекових конвеєрів шляхом розроблення й визначення оптимальних параметрів шарнірно з'єднаних між собою гнучких гвинтових секційних робочих органів для перевантаження сипучих матеріалів по криволінійних гілках транспортування. Дослідження виконано із використанням методів теорії математичного та комп'ютерного моделювання. методу математичного планування експерименту. У статті представлено нові конструкції шарнірних гвинтових робочих органів та обґрунтовано їхні оптимальні параметри для забезпечення переміщення сипких матеріалів по криволінійних трасах. Визначено взаємозв'язок між величиною крутного моменту, який виникає між навантаженими гвинтовими секціями та їх просторового розміщення. Отримано аналітичні залежності для умов жорсткості окремої секції із шарнірним з'єднанням в залежності від навантаження та конструктивних параметрів транспортера. Встановлено, що кутові зміщення секцій згідно встановленої в статі залежністі, дозволяє суттєво (до 4-5 раз) зменшити амплітуду зміни крутного моменту приводу, а також понижує його максимальне значення. Було доведено, що при штовхаючому режимі енерговитрати є більшими у порівнянні з тягнучим у 1,06...1,4 рази. Практичне значення результатів полягає в можливості обрати оптимальні конструктивні, кінематичні та технологічні параметри розроблених секцій гвинтового робочого органу для транспортування сипких чи кускових матеріалів і мінеральних добрив

Ключові слова: шнекові механізми; криволінійна гілка транспортування; силові параметри; сипкий матеріал; перевантажувальні механізми