The paper considers the method of computer simulation of the stress-strain state of the drive drum shell in the NASTRAN integrated environment. Due to the complexity of determining stresses and deformations in the drum sections by the analytical method, it is proposed to solve this important problem using the numerical finite-element method. At the preliminary stage of computer modeling, a mechanical design scheme was developed, including a variable pressure that changes depending on the angle of rotation of the drum, the deterministic relations describing the variable force factors are based on the Euler ratio. It is also proposed to take into account the pressure from the variable friction force, which depends on the changing coefficient of adhesion of the belt to the drum.

As a result of the computer calculation, the equivalent Mises stresses of 65 MPa were determined, the safety factor was 4.2 and the components of the tangential stresses were determined using the stress tensor marker, the shear stress reached the level τ=16 MPa for fabric tape and τ=3.14 MPa for rubber tape. According to the results of the calculation, the dependence of the tangential stresses on the angle of rotation of the drum was constructed. A diagram of the change in the component of tangential stresses along the forming shell of the drum was constructed.

Analysis of stress-strain state allowed us to determine the factor of safety of the drum shell. Based on the analysis of equivalent stresses, it is proposed to further calculate the durability of the drum using the method of long-term fatigue. The computer calculation of shear stresses in the component allows choosing the rational parameters of the lining, based on such indicators as peel strength and break, as well as determining the angle 61° of the slab lining required to improve the reliability and traction ability of the pipeline

Keywords: conveyor belt, durability, drive drum, stresses, deformation, finite-element method, lining

UDC 62-251

DOI: 10.15587/1729-4061.2021.229213

DEVELOPMENT OF A TECHNIQUE FOR COMPUTER SIMULATION OF THE STRESS STATE OF THE DRIVE DRUM SHELL OF A BELT CONVEYOR TO OPTIMIZE ITS DESIGN PARAMETERS

Olzhas Jassinbekov
PhD*
E-mail: olzhas_9114@mail.ru

Madina Isametova
PhD, Associate Professor*
E-mail: isametova69@mail.ru

Gabit Kaldan
Postgraduate Student*
E-mail: usenuly@list.ru

*Department of Industrial Engineering
Satbayev University
Satpayev str., 22a, Almaty,
Republic of Kazakhstan, 050013

Received date 10.03.2021
Accepted date 12.04.2021
Published date 26.04.2021

How to Cite: *Jassinbekov, O., Isametova, M., Kaldan, G. (2021). Development of a technique for computer simulation of the stress state of the drive drum shell of a belt conveyor to optimize its design parameters. Eastern-European Journal of Enterprise Technologies, 2 (7 (110)), 31–39. doi: https://doi.org/10.15587/1729-4061.2021.229213*

1. Introduction

Belt conveyors are widely used in mining enterprises, as the introduction of in-line technology in mines, pits and quarries increases the technical level and efficiency of mining production. A characteristic trend of the modern development of belt conveyors is a significant increase in their length and productivity. At the same time, the requirements for the reliability and durability of belt conveyors and, consequently, for the most responsible and loaded components are constantly increasing [1–3].

The belt conveyor is a continuous transport machine, the main elements are: the belt-traction and load-bearing body; drive, deflecting, bypass and tail drums; conveyor stav with guide structures of roller supports; upper and lower rollers; drive blocks; auxiliary and protective devices. Fig. 1 shows the belt conveyor diagram.

The modern development of conveyor transport is characterized by an increase in its productivity and the length of transportation, which increases the load on the drive and deflection drums, as well as the number of drums in the conveyor. This reduces the overall reliability of the conveyors and worsens the technical and economic performance of their work. A large number of drums are required not only to newly manufacture equipment conveyors, but also to replace the drums installed on existing transporting cars, the service life is significantly below the service life of the pipeline, especially in mining. Therefore, in order to meet the needs for replacing failed drums, it is necessary to find ways to increase the production of drums, as well as to use them more efficiently. The complexity of this task requires finding ways to further improve the design, calculation methods, and increase the reliability and durability of belt conveyor drums. In this regard, the scientific task of creating a computer model for calculating the stress-strain state of belt conveyor drums to reduce their metal consumption and increase their durability and reliability is relevant.

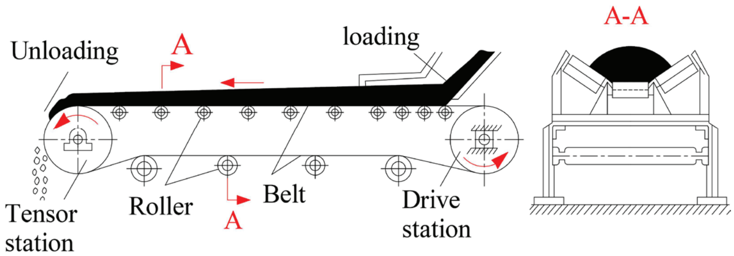


Fig. 1. Belt conveyor diagram

The stress-strain state of the elements of the drive drum of a belt conveyor is of a complex character: a normal load combined with significant tangential, and they are variable angle strap tape drum, being, however, is cyclical [4].

In the manufacture of the drum, the thickness of its shell is assumed to be too high, which often does not increase the durability, but increases the metal consumption of the drum and its supporting structure.

As the experience of operation shows, one of the reasons for the failure of drums is the fatigue failure of the drum elements. This leads to downtime of the conveyors and significant economic costs, since the repair of the drive drum is a long and expensive operation.

In order to predict fatigue and extend the service life of the drum, using the method of applying lining to the drum shell, it is necessary to create an adequate computer model of the stress-strain state of the shell.

System fatigue in automated design systems can be calculated using two methods: low-cycle fatigue and multi-cycle fatigue, and at the stage of computer-aided design, the correct choice of method is to determine the correctness of determining the durability and recommendations for the maintenance of conveyors.

Only taking into account the expansion of the stress tensor in the co-computer calculation, it is possible to more accurately determine the magnitude and actual direction of tangential stresses, as a result of which it becomes possible to determine the rational angle of inclination of the lining grooves.

To solve these problems, it is necessary to create a computer model that would allow us to determine the contribution of tangent stresses to the SSS of the drum rim.

2. Literature review and problem statement

In [5–7], methods for calculating the structural elements of conveyor drums for strength are proposed. The calculation of the drive drum shell of a belt conveyor is based on the beam theory. It considers the bending of a closed cylindrical shell for the cases of the following loads: contour, radial and tangent (evenly distributed along the length of the shell and on a small section of its width).

In [8], a system of equations for determining the displacements and force factors in the disk is presented. It is believed that the bending of the drive drum shaft at the girth angle of the belt 1800 is caused by the forces P and P' acting respectively in the planes of the landing places of the disks. It should be noted that it is impossible to agree with such a simplified system of forces acting on the drum – it contradicts the methods of bringing forces and moments acting on a solid body, accepted in mechanics.

In [9], we also consider the issues of calculating the drums of belt conveyors in the case when the shell is reinforced with an annular frame of width bk and thickness hk. Studies have shown that the reinforcing ring reduces internal forces and stresses in the shell.

The effect of the reinforcing ring on the stress state of the end disks is less significant. By tightening the shell, the ring reduces the forces transmitted to the disks from the shell side, but does not change the picture of the stress state of the disks. The forces acting on the drum were considered as concentrated, which greatly simplifies the actual calculation scheme.

There are also a few studies of the SSS of the drum, using modern computer-aided design systems. In [10], the forces applied to the drum surface were modeled and the stress-strain state of the drum shell was determined. To do this, a computer model was created to simulate the Abaqus complex. Using this model, the influence of the size and shape of the final element on the SSS of the drum was studied. The author notes that the load distribution along the axis of the drum affects the maximum stress both in size and location. The overall maximum can be up to 15 % lower, while for a specific node, the reduction can be even higher – up to 20 %. However, the real picture of the stresses arising from the variable pressure on the surface of the shell is not presented.

It should be noted that in the works analyzed above, when calculating the SSS, the components of the stress tensor, namely the tangent components, which are decisive, are not considered in the issues of plastic deformation of the shell surface and the selection of rational parameters of the lining to increase the service life and increase the traction capacity of the drums [11].

3. The aim and objectives of the study

The aim of this study is to develop a computer methodology for assessing the stress-strain state of the drum for further recommendations on the choice of the method for calculating the durability of the drum and the selection of lining parameters.

To achieve this aim, the following objectives were set:

– to develop a calculated mechanical scheme that simulates the action of variable pressure and variable friction force;

– to develop an adequate computer model of the drum and belt assembly, which will maximally reflect the condition of the conveyor operation;

– to perform a computer calculation in the NASTRAN integrated environment for two versions of the tape: fabric and rubber and determine the influence of the mechanical properties of the tape material on the stress-strain state of the drum casing;

– to determine the overall strength of the drum and propose a method for further evaluating the durability of the shell;

– to determine the tangential stresses along the drum generator to develop a method for selecting rational parameters of the lining of the drum shell.

4. Materials and methods for studying the basic deterministic relationship that mathematically describes the stresses and deformations in the drum shell from the action of the belt pressure

The drum under study is a physical model of the drive drum of a high-power belt conveyor. The design of the drum of belt conveyors for mining enterprises is shown in Fig. 2.

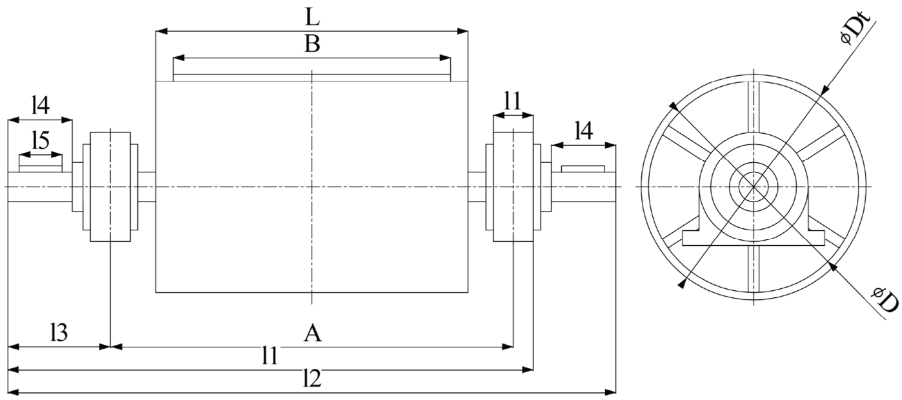


Fig. 2. Legend for dimensions of drum elements

Parameters of the drum under study:

– D=500 mm – drum diameter;

– B=1,000 mm – width of the tape;

– P=575 kN – circumferential effort;

– S=137 kN – load on the drum;

– μ=0.34 – coefficient of friction.

Drum material 10G2S1.

Initially, we consider the main deterministic relations that mathematically describe the stresses and deformations in the drum shell from the action of the belt pressure.

The stress-strain state of the drive drum is determined mainly by its design, the tension on the incoming S_{inc} and cascading S_{cas} branches of the belt, and the nature of the tension change along the arc of the girth. When determining the relationship between the angle of the belt circumference of the drum surface and the tension on the incoming and cascading branches of the drive drum, the well-known ratio obtained by L. Euler is commonly used:

$$\frac{S_{inc}}{S_{cas}} = e^{\mu\alpha}, \tag{1}$$

where α – is the angle of the belt girth of the drive drum, μ – coefficient of adhesion between the belt and the drum.

To calculate the SSS of the shell according to the Vlasov theory, the model shown in Fig. 3 is used as the design calculation [12]. The calculations are performed under the following assumptions: the drum is firmly planted on the shaft, the distance between the supporting disks is equal to the width of the tape tension constantly on the width of the tape, grip tape drum happening across the arc of circumference, the force is transferred in accordance with the Euler formula. When solving the problem of deformation of the drum shell, disks and shaft are considered together.

The distributed radial and tangential loads on the drive drum shell are decomposed into a trigonometric Fourier series relative to the middle of the arc of the belt girth of the drum and are presented in the form of rows:

$$z(\alpha,\beta) = -\left[\frac{S_{cas}e^{\mu\beta_0}\,\mathrm{sh}\beta}{\pi\mu LR} + \int\limits_{n=0}^{\infty} z'_n \cos n\beta + z'' \sin n\beta\right], \tag{2}$$

$$y(\alpha,\beta) = -\mu z(\alpha,\beta), \tag{3}$$

where

$$z'_n = \frac{2S_{cas}e^{\mu\beta_0}}{\pi\left(n^2+\mu^2\right)LR}\left(\mu\,\mathrm{sh}\mu\beta_0 \cos n\beta_0 + n\,\mathrm{ch}\mu\beta_0 \sin n\beta_0\right), \tag{4}$$

$$z''_n = \frac{2S_{cas}e^{\mu\beta_0}}{\pi\left(n^2+\mu^2\right)LR} \times \times \begin{pmatrix} \mu\,\mathrm{sh}\mu\beta_0 \sin n\beta_0 + \\ + n\,\mathrm{ch}\mu\beta_0 \cos n\beta_0 \end{pmatrix}, \tag{5}$$

where α, β – dimensionless coordinates.

Here the following designations are introduced: R – the radius of the circle, the median surface of the cylindrical shell; X, Y, Z – positive directions of surface forces; u, y, ω – positive directions of displacement along the generatrix of the drum, along the arc of its circumference and along the radius, respectively.

To solve the system of differential equations of the theory of elasticity, a generalized stress function f(α, β) is introduced, which satisfies the differential equation:

$$\nabla^2\nabla^2\nabla^2\nabla^2\phi - \frac{1-\nu^2}{c^2}\frac{\partial^4\phi}{\partial\alpha^4} = \frac{R}{D_{tot}}z(\alpha,\beta) - \frac{R^4}{D_{tot}}y(\alpha,\beta), \tag{6}$$

by

$$\nabla^2 = \frac{\partial^2}{\partial\alpha^2} + \frac{\partial^2}{\partial\beta^2};\quad D_{tot} = \frac{Eh_{tot}^3}{12\left(1-\nu^2\right)};\quad c = \frac{h_{tot}^2}{12R^2}, \tag{7}$$

where υ – Poisson's coefficient.

The general solution of the equation is sought in the form:

$$\phi = \phi_0 + \phi_z + \phi_y, \tag{8}$$

where ϕ_0 – general solution of the homogeneous equation; ϕ_z, ϕ_y – partial solutions.

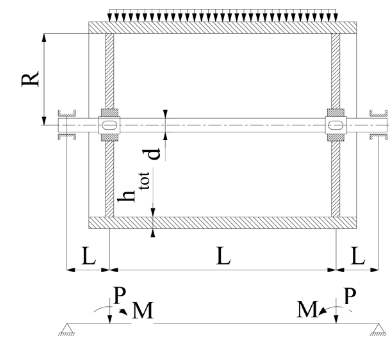


Fig. 3. Calculation model of the drum [33]

The displacements and force factors in the shell are determined by the dependencies:

$$\begin{gathered} u = \left(\frac{\partial^3}{\partial\alpha\partial\beta^2} - \upsilon\frac{\partial^3}{\partial\alpha^3}\right)\left(\phi_0+\phi_z\right) - \\ -\frac{\left(\partial^2\phi_y\right)}{\partial\alpha\partial\beta} - \frac{1+\upsilon}{1-\upsilon}c^2\frac{\partial^2}{\partial\alpha\partial\beta}\nabla^2\nabla^2\phi_y, \end{gathered} \tag{9}$$

$$\begin{gathered} \upsilon = -\left(\frac{\partial^3}{\partial\beta^2} + (2+\upsilon)\frac{\partial^3}{\partial\alpha^3\partial\beta}\right) + 2(1+\upsilon)\frac{\partial^2\phi_y}{\partial\alpha^2} + \\ + \frac{\partial^2\phi_y}{\partial\beta^2} + c^2\left(\frac{\partial^2}{\partial\beta^2} + \frac{2}{1-\upsilon}\frac{\partial^2}{\partial\alpha^2}\right)\nabla^2\nabla^2\phi_y, \end{gathered}$$

$$\omega = \nabla^2\nabla^2\left(\phi_0+\phi_z\right)-\left(2+\upsilon\right)\frac{\partial^3\phi_y}{\partial\alpha^2\partial\beta}-\frac{\partial^3\phi_y}{\partial\beta^3} \text{ [13].} \tag{10}$$

Such tasks require significant capital and time costs for experimental implementation. However, these difficulties can be circumvented by using computer mathematical modeling.

5. Results of the study of the stress-strain state of the conveyor drum shell

5. 1. Developing a mechanical design scheme to simulate the action of variable pressure and variable friction force

The preliminary stage of computer modeling is the creation of a mechanical design scheme. By simulating the belt and drum assembly, we created a mechanical design scheme corresponding to the scheme in Fig. 4.

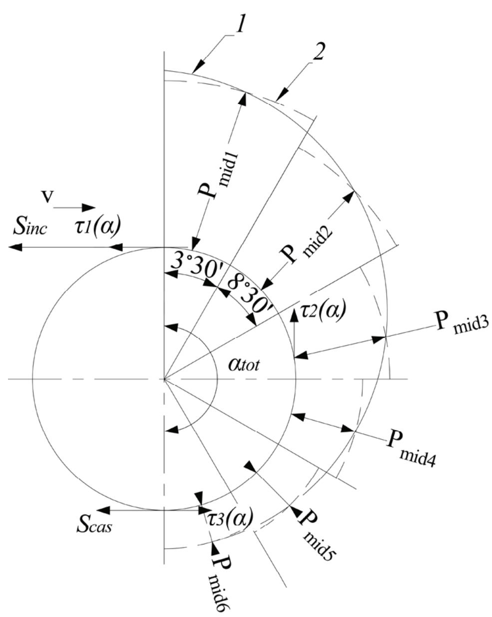


Fig. 4. Character of change in pressure in the drive drum

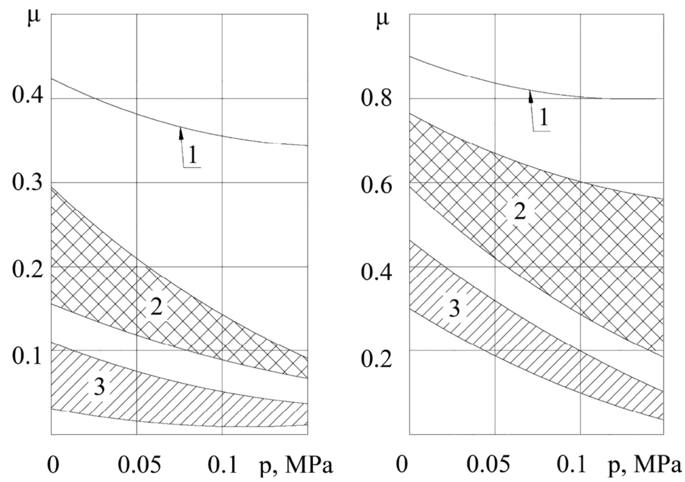
The diagram indicates the occurrence of two force factors when the tape acts on the drum – these are variable pressure force and friction force.

The case of variable pressure and variable friction force was modeled. According to the theory of K. Grimmer, D. Torman, the variable friction force arises from the condition of the occurrence of a variable coefficient of adhesion μ from pressure p. Fig. 5 shows the graphs [13].

As can be seen from the graphs, the coefficient of adhesion μ significantly depends on the pressure of the belt on the drum, and, consequently, on the variable tension of the belt along the arc of the girth. Thus, to determine the stress state of the drum elements, it is necessary to determine the nature of the change in tension over the surface of the drum with a variable coefficient of adhesion of the tape to the surface of the drum [14].

Table 1 shows the dependencies that determine the variable nature of the force factors acting on the drum shell.

The formulas in the table are used further in modeling the loading process of the drive drum in the pressure range p=0.1÷0.4 MPa. At higher pressures (p>0.4 MPa), it is advisable to take into account the change in the coefficient $m(p)$ as the circumference angle increases and the corresponding increase in pressure $p(\alpha)$.



a *b*

Fig. 5. Graphs of variable pressure and variable friction: *a* — smooth steel drum; *b* — rubber-lined drum: 1 — dry; 2 — wet; 3 — wet with the presence of clay

Table 1

Data for the boundary conditions of the computer model of the SSS drum

Power factor	Function
P	$p(\alpha)=\left[\left(p_{cas}^{1,1}+\frac{\mu_1}{\mu_0}\right)e^{1,1\mu_0\alpha}-\frac{\mu_1}{\mu_0}\right]^{\frac{1}{1,1}}$ MPa P – the force of pressure from the force in the incoming branch μ_1 – the coefficient of the variable clutch μ_0 – constant coupling coefficient $-\pi\leq\alpha\leq\pi$
T	$\tau(\alpha)=\left[\left(p_{cas}^{m}+\frac{\mu_1}{\mu_0}\right)e^{m\mu_0\alpha}-\frac{\mu_1}{\mu_0}\right]^{\frac{1}{m}}\times$ $\times\left[\mu_0+\mu_1\left(\left(p_{cas}^{m}+\frac{\mu_1}{\mu_0}\right)e^{m\mu_0\alpha}-\frac{\mu_1}{\mu_0}\right)^{-1}\right]$ P – the force of pressure from the force in the incoming branch μ_1 – the coefficient of the variable clutch μ_0 – constant coupling coefficient $-\pi\leq\alpha\leq\pi$ $p\leq0.6$ MPa, $m\cong1.25$

5. 2. Developing an adequate computer model of the drum and tape assembly

The problem of interaction between the tape and the drum was modeled as an assembly, which entailed some technical complexity, however, this formulation most adequately reflects the real physical model. To solve this problem, in the NASTRAN program, the shell and the tape were divided into groups, their contact was carried out by the Equivalence operation, 2,561 nodes belonging to the tape and drum grid were combined along the half-surface of the shell. The drum belt and assembly models are shown in Fig. 6 [15].

Fig. 7 shows the finite-element grid (position 1). The number of finite elements for the shell of a drum with a tape width B=1,000 mm and a diameter D=500 mm was about 4.000. The end element type is Quad4 Isomesh grid view.

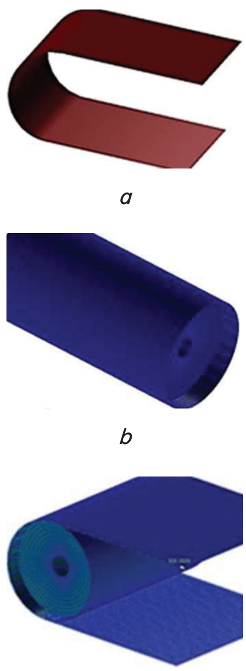


Fig. 6. Drum belt and assembly models: *a* — tape model; *b* — drum model; *c* — built model

The force boundary conditions were set as variables using the Fields function, which includes the built-in PCL programming language. Table 1 shows the laws of the dependence of the variable normal pressure and the friction force of the belt, as a function of the variable coefficient of adhesion.

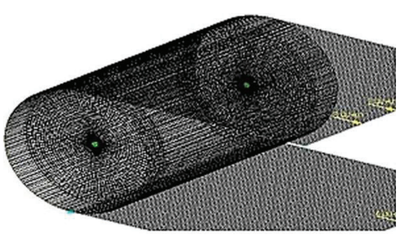


Fig. 7. Finite-element grid of the digital drive drum model

Fig. 8 shows a computer model of the design scheme of interaction between the tape and the drum.

The problem was solved in a static formulation under the assumption that the tape is an elastic linearly deformable body obeying Hooke's law.

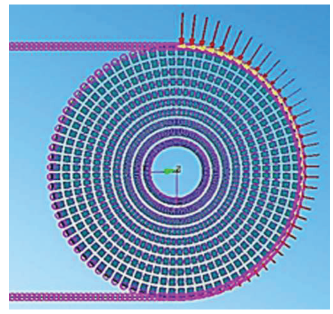


Fig. 8. Computer model of the design scheme of interaction between the tape and the drum

5. 3. Results of computer calculation in the NASTRAN integrated environment for two versions of the tape: fabric and rubber

Determination of the influence of the mechanical properties of the tape material on the stress-strain state of the drum casing.

Fig. 9 shows the diagrams of the Mises stress distribution in both the tape and the drum. Colored isolines indicate the occurrence of maximum stresses in the area of the drum landing on the shaft, here the maximum stresses reach σ_{max}=65 MPa. At this stress level, the coefficient of safety margin is sufficient for the steel grade 10G2S1 k=4.2.

The obtained results of computer calculations reveal new quantitative effects of the stress-strain state of the drum shell on the verification calculations of the durability of structures with different levels of accuracy, as well as on recommendations for the use of rational means of lining in order to increase the reliability and traction capacity of conveyors.

Despite the excessive strength ratio, it would seem, leading to increasing metal construction, do not forget about the fact that the service life before technical inspection must be 1,000 hours.

According to the results of a computer calculation of the stress tensor components τ_x, τ_y (Fig. 12) on the surface of the drum, the diagram was constructed based on the maximum shear stresses on the angle of rotation of the drum in increments of α=30° for two variants of the tape: cloth tape with a modulus of elasticity E=40 MPa and rubber tapes with a characteristic E=5.8·10^3 MPa.

The graph (Fig. 10) shows that the elastic coefficient of the tape affects the level of tangential stresses, the tape with less elasticity gives a higher level of stress, with a fabric tape τ=16 MPa, with a rubber tape τ=3.14 MPa.

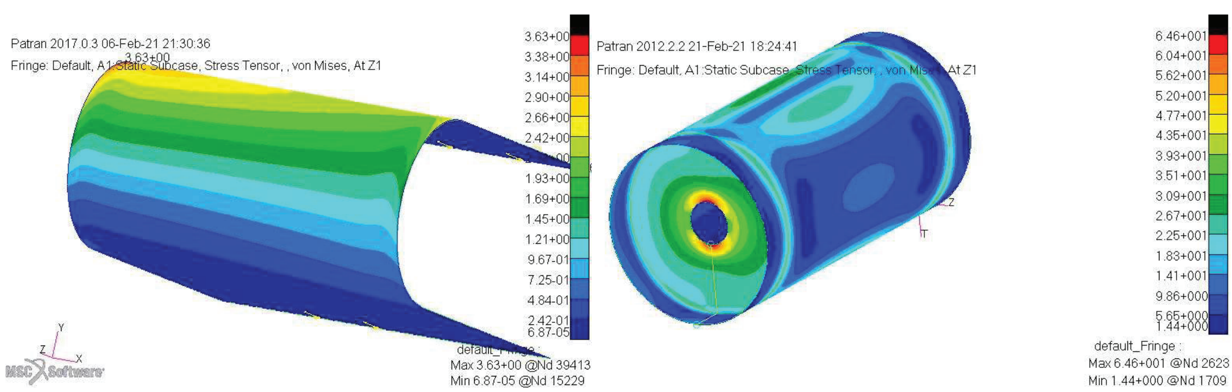


Fig. 9. Diagram of the stresses in the tape and drum

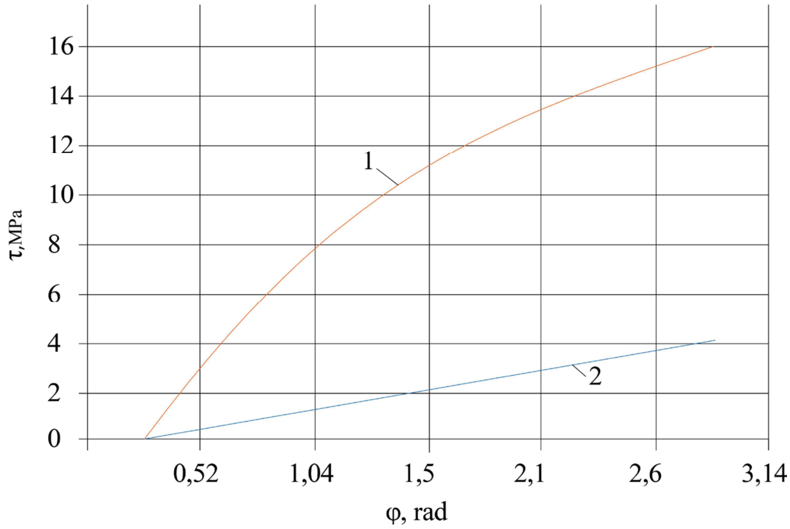


Fig. 10. Graph of the dependence of tangential stresses on the angle of rotation: 1 – with fabric tape τ=16 MPa, 2 – with rubber tape τ=3.14 MPa

The graph of changes in tangential stresses shows that when using more rigid (rubber-wire) belts, the rest arc zone increases and more favorable conditions are created for transmitting a significant portion of the traction force without sliding the contacting surfaces, which also reduces the mutual wear of the tape and lining.

The stresses determined by the virtual experiment were compared with the results of the experiment [16].

Fig. 11 shows the installation diagram for determining the SSS of the drum. On the frame 1, a drum 4 is fixed in the brackets 5, the stress state of which is subject to investigation. On its shaft 6, a lever 7 is rigidly fixed, the end of which is connected to the frame 1 through a dynamometer 8. The shell of the drum 4 is covered by a tape 10, the ends of which are also connected to the frame 1 by means of devices 9 and dynamometers 2 and 3.

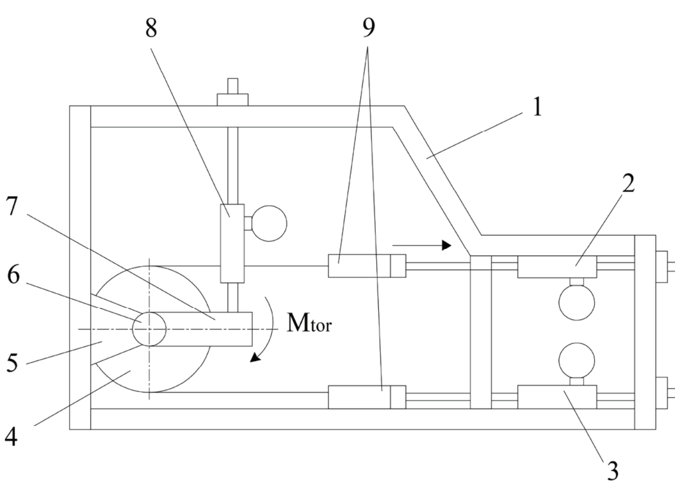


Fig. 11. Experimental installation for the study of the SSS of the drum

5. 4. Evaluation of the shell durability

In the first assessment of the durability of the drum, we can talk about a further calculation for fatigue, which should be carried out according to the method of long-term fatigue. Based on the results of the calculation of the stress-deformable state of the shaft given above, it is possible to reasonably choose the S-N method of calculating multi-cycle fatigue, since multi-cycle fatigue occurs at stresses significantly below the yield strength ($\sigma_{max}<0.6\sigma_{0.2}$), as the condition is met $\sigma_{max}<0.6\sigma_{0.2}$ [17].

When analyzing the stress level along the width of the belt, a group of nodes on the full surface in the most stressed zone of the drum was selected from the total array of solutions obtained. The result of the stress change is shown in Fig. 12.

The stress peaks σ=24 MPa along the cylinder generatrix indicate the critical points of possible crack formation, which are localized at the junction of the disks with the surface of the shell, which serves as recommendations for further calculation of the welds for strength and fatigue.

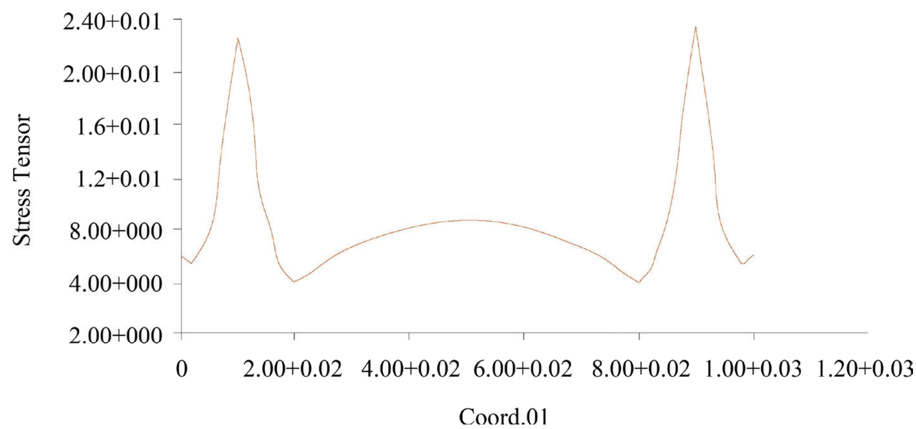


Fig. 12. Graph of stress changes along the cylinder generatrix

5. 5. Components of tangential stresses along the drum generatrix

Determination of tangential stresses for the selection of rational parameters of the drum shell lining.

To determine the contribution of tangential stresses to the level of equivalent stresses, the Marker option of the stress tensor of the element lying on the shell surface was used. Fig. 13 shows the stress tensor on the surface of the drum rim.

Certain values of tangential stresses allow you to correctly solve the important issue of selecting the material of the drum lining.

It is known that the minimum value of the shear stiffness of the lining is limited by the fatigue wear of the lining material during its repeated loading. Therefore, it is recommended to select such parameters of the lining, at which high traction properties of the drive are achieved with acceptable performance friction wear of the belt lining and the drum lining when they interact [18, 19].

A theoretical analysis of a conveyor belt with a lining, taking into account its volumetric deformation, shows that the transfer of significant traction shear stress (and transverse deformation) to the width of the drum allows measuring the parameters that are necessary when choosing the parameters of the lining and determining more accurate values and actual direction of total shear stresses [20, 21].

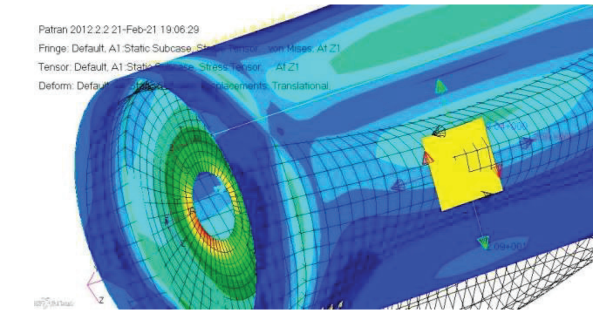


Fig. 13. Stress tensor marker

The dependence of the level of tangential stresses τ_y on the width of the tape was analyzed (Fig. 14).

According to the analysis of the diagram, the interaction of the conveyor belt with the shell, taking into account its volumetric deformation, showed that when transmitting significant traction forces, the components of tangential stresses along the width of the drum reach certain values of τ_y=5 MPa, which can lead to plastic deformations.

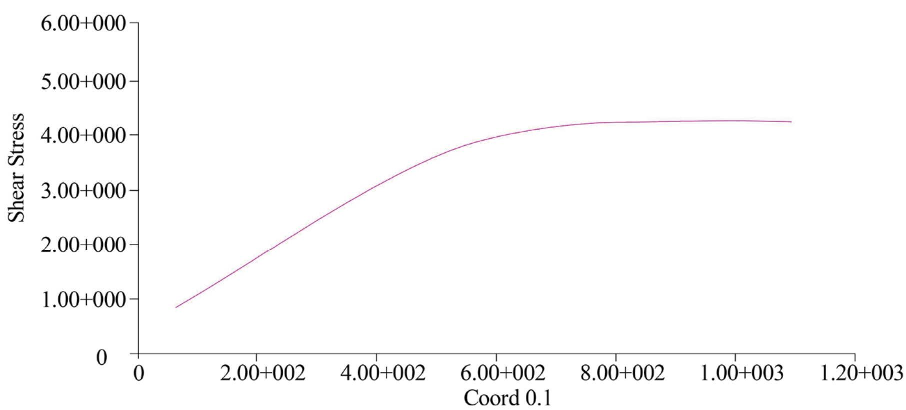


Fig. 14. Graph of the change in the component of the stress tensor over the width of the tape

For further selection of rational parameters of the lining, the parameters of the tangential stress component τ_x, τ_y, determined from the results of computer modeling of the stress-strain state of the shell, are used.

Using the values of the computer calculation, it is possible to determine such important parameters of the lining as:

1. Specific breakaway strength of the rubber tape [22]

$$\sigma_{br} = \tau_{\max} k_3 \mu \frac{T_{inc} \cdot h_{\phi}}{B(R_0 + h_l) l_{\partial}^3} \leq [\sigma]_{br}. \tag{11}$$

2. Specific exfoliation strength.

$$\sigma_{exf} = k_3 \tau_{\max} \leq [\sigma]_{exf}, \text{ MPa}, \tag{12}$$

where $[\sigma]_{exf}$ – is the permissible peel strength of rubber mixtures for the drive drum linings.

3. Rational angle of inclination of the corrugated lining.

For normal operation of the grooved lining, the angle of inclination of the grooves α to the longitudinal axis of the conveyor must not exceed a certain limit.

The elastic slip velocity of the tape V_{sl}^T consists of two components V_x and V_y, which depend on the value of the tangential stresses τ_x and τ_y.

$$\frac{\tau_y}{\tau_x} = \frac{V_y}{V_x}. \tag{13}$$

Then the angle of the grooves is defined as

$$\tan g = \frac{\tau_y}{\tau_x}. \tag{14}$$

For the drum under study, the angle of inclination of the groove æ=61°.

6. Discussion of experimental results

The obtained results of computer calculations reveal new quantitative effects of the stress-strain state of the drum shell on the verification calculations of the durability of structures with different levels of accuracy, as well as on recommendations for the use of rational means of lining in order to increase the reliability and traction capacity of conveyors.

The simulated design model includes variable force factors such as pressure and friction. The difference of the design scheme given in this work is that in the model, when forming the loading scheme of the drum by the tape, a modified Euler formula is used, which takes into account the dependence of the coefficient of adhesion of the tape to the drum on its tension force in the range of 0.1÷0.4 MPa. This model makes it possible to assess the influence of the variable friction force on the level of shear stresses on the surface of the drum shell, which is missing in the works on modeling given in the sources [11, 12].

The formulation of the problem of the interaction between the tape and the drum was modeled as an assembly, which most adequately reflects the real physical model of the interaction between the tape and the drum and the forces arising from the contact.

To assess the computer stresses arising in the drum of a belt conveyor of a conventional design, it is proposed to use a simulation of the assembly of a belt and a drum using a quadrangular four-node flat finite element with six degrees of freedom at each node using the accepted boundary conditions and the described load schematization. The Equivalence operation allows combining common assemblies to simulate the transfer of forces from the belt to the drum.

The result of stresses along the drum generatrix on the stress versus coordinate graph indicated stress peaks σ=24 MPa corresponding to the critical points of possible cracking, which are localized at the junction of the discs with the shell surface, which serves as recommendations for further calculation of welded joints for strength and fatigue.

Despite the excessive strength factor, seemingly leading to an increase in metal structures, it is not worth reducing the thickness of the shell, since the service life before inspection should be 1000 hours, which corresponds to prolonged fatigue, and an increase in stresses will lead to an increase in the stress range when calculating for durability.

The condition $\sigma_{max}<0.6\sigma_{0.2}$ at a calculated stress level of 64 MPa in the most dangerous section and the yield point of steel 245 MPa assumes, when calculating for fatigue, the choice of the S-N method for calculating high-cycle fatigue.

In the course of calculations, the places most susceptible to fatigue failure were identified: the middle part of the shell and the welded seams connecting the shell and support disks, which corresponds to the experimental data given in [16] and computer simulation given in [23].

An analysis of the diagram of the dependence of tangential stresses on the coordinate along the generatrix of the shell shows that the interaction of the conveyor belt with the shell, taking into account its volumetric deformation during the transfer of significant tractive forces, the components of tangential stresses along the width of the drum reach certain values τ_y=5 MPa, which can lead to plastic deformations.

The parameters of the tangential component of stresses τ_x, τ_y, determined from the results of computer modeling of the stress-strain state of the shell, will determine the further choice of rational parameters of the lining. According to the level of certain tangential stresses, the lining for the investigated drum will have an angle of inclination of the grooves of 61°.

Despite the importance of the results obtained in the studies, the limitation of such a solution to the problem lies in the fact that the problem of determining the SSS of the drum was solved in a linear setting in the elastic zone. For further development of the study of the problem of strength and fatigue of the drum shell, it is necessary to improve approaches to computer modeling and take into account the physical nonlinearity of the tape material, as well as consider the zone of contact between the tape and the drum in the formulation of the Hertz problem, which will allow determining the contact stresses leading to the appearance of defects on the shell surface.

7. Conclusions

1. A mechanical design scheme has been developed that simulates the effect of variable pressure and variable friction force on the drum rim. The scheme takes into account the variable nature of the friction force as a function of the variable adhesion of the belt to the drum.

2. The approach to modeling an adequate computer model of the drum and belt assembly, which reflects the condition of the conveyor functioning as much as possible, has been developed. The assembly of a belt and a drum was modeled using a quadrangular four-node flat finite element with six degrees of freedom at each node using the accepted boundary conditions and the described load schematization, which made it possible to simulate the transfer of forces from the belt to the drum.

3. A computer calculation was carried out in the NASTRAN integrated environment for two versions of the tape: fabric and rubber, the levels of tangential stresses with a fabric tape τ=12 MPa, with a rubber tape τ=4 MPa.

4. The total strength of the drum σ_{max}=65 MPa is determined, the coefficient of the safety margin is sufficient for the steel grade 10G2S1 k=4.2, the long-term fatigue S-N method is proposed for further evaluation of the durability of the shell.

5. The dependence of the level of tangential stresses τ_y on the width of the tape is determined, the maximum value for the rubber tape τ_y=4 MPa. According to the method of selecting rational parameters of the lining of the drum shell, the angle of inclination of the groove is determined æ=61° for the standard size of drive drums with a belt width of 1,000 mm.

References

1. Mihailidis, A., Bouras, E., Athanasopoulos, E. (2015). FEM analysis of a belt conveyor driving drum. 6th BETA CAE International Conference. Available at: https://www.researchgate.net/publication/283500471
2. Solovykh, D. Ya. (2014). Modelirovanie na EVM napryazhennogo sostoyaniya privodnogo barabana lentochnogo konveyera dlya otsenki dolgovechnosti svarnykh shvov [Computer-aided modeling of stress state of belt driving drum to estimate endurance of joint welds]. Moscow: Gornaya kniga, 12. Available at: https://www.litres.ru/d-solovyh/modelirovanie-na-evm-napryazhennogo-sostoyaniya-privodnogo-barabana-lentochnogo-konveyera-dlya-ocenki-dolgovechnosti-svarnyh-shvov-25285151/
3. Ananth, K. N. S., Rakesh, V. (2013). Design and Selecting the Proper Conveyor-Belt. International Journal of Advanced Engineering Technology, 4 (2). Available at: https://www.technicaljournalsonline.com/ijeat/VOL%20IV/IJAET%20VOL%20IV%20ISSUE%20II%20APRIL%20JUNE%202013/Vol%20IV%20Issue%20II%20Article%2012.pdf
4. Dmitriev, V. G., Asaenko, V. V. (2011). Harakter nagruzheniya obechayki privodnogo barabana lentochnogo konveyera pri peremennom koeffitsiente stsepleniya lenty s ego poverhnost'yu. Gorniy informatsionno-analiticheskiy byulleten' (GIAB), 2, 375–378.
5. Conveyor Belt Basic Rules and Procedures for Tracking and Training (2006). Besser Service Bulletin.
6. Conveyor Handbook (2009). Fenner Dunlop. Available at: http://www2.hcmuaf.edu.vn/data/dangnh/file/5_Fenner%20Dunlop_%202009_%20Conveyor%20Handbook.pdf
7. Rulmeca. Technical Information (2009). Project and Design Criteria for Belt conveyors.
8. Lodewijks, G., Nuttall, A. J. G. (2007). Dynamics of decentralized driven belt conveyors. 9th International Conference on Bulk Materials Storage, Handling and Transportation, ICBMH 2007.
9. Solovyh, D. Ya. (2014). Matematicheskaya model' formirovaniya davleniy na privodnom barabane lentochnogo konveyera s uchetom izmenyayuschegosya koeffitsienta stsepleniya. Gorniy informatsionno-analiticheskiy byulleten' (GIAB), spetsvypusk, 12.

10. Fedorko, G., Beluško, M., Hegedűš, M. (2015). FEA Utilization for Study of the Conveyor Belts Properties in the Context of Internal Logistics Systems. Czech Republic. Available at: http://konsys2.tanger.cz/files/proceedings/24/papers/4619.pdf
11. Michalik, P., Zajac, J. (2012). Using of computer integrated system for static tests of pipe conveyer belts. Proceedings of the 13th International Carpathian Control Conference (ICCC). doi: https://doi.org/10.1109/carpathiancc.2012.6228691
12. Affolter, C., Piskoty, G., Koller, R., Zgraggen, M., Rütti, T. F. (2007). Fatigue in the shell of a conveyor drum. Engineering Failure Analysis, 14 (6), 1038–1052. doi: https://doi.org/10.1016/j.engfailanal.2006.11.071
13. Gulak, M. L. (1998). Issledovanie napryajennogo sostoyaniya i optimizaciya konstruktivnih parametrov barabanov lentochnih konveierov gornih predpriyatii. Moscow, 148. Available at: https://www.dissercat.com/content/issledovanie-napryazhennogo-sostoyaniya-i-optimizatsiya-konstruktivnykh-parametrov-barabanov
14. Solovih, D. Y. (2018). Razrabotka metoda rascheta ustalostnoi dolgovechnosti privodnih barabanov lentochnih konveierov dlya gornoi promishlennosti. Moscow, 125. Available at: https://misis.ru/files/9582/Solovyh_dis.pdf
15. Zhilkin, V. A. (2013). Azbuka injenernih raschetov v MSC Patran-Nastran-Marc. Sankt-Peterburg: Prospekt Nauki, 576. Available at: http://www.iprbookshop.ru/35886.html
16. Kim, J. K., Shim, H. J., Kim, C. S. (2006). Durability Analysis of the Pulley in the Power Steering System Considering the Variation of the Fatigue Strength. Key Engineering Materials, 306-308, 429–434. doi: https://doi.org/10.4028/www.scientific.net/kem.306-308.429
17. Povetkin, V. V., Isametova, M. E., Isayeva, I. N., Bukayeva, A. Z. (2018). Dynamic modeling of ball mill drive with regard to damping properties of its elements. Mining Informational and Analytical Bulletin, 5, 184–192. doi: https://doi.org/10.25018/0236-1493-2018-5-0-184-192
18. Fatemi, A. (2004). Fatigue behavior and life predictions of notched specimens made of QT and forged microalloyed steels. International Journal of Fatigue, 26 (6), 663–672. doi: https://doi.org/10.1016/j.ijfatigue.2003.10.005
19. Ragan, P., Manuel, L. (2007). Comparing Estimates of Wind Turbine Fatigue Loads Using Time-Domain and Spectral Methods. Wind Engineering, 31 (2), 83–99. doi: https://doi.org/10.1260/030952407781494494
20. Solovih, D. Ya. (2014). Matematicheskaya model' formirovaniya davleniy na privodnom barabane lentochnogo konveyera s uchetom izmenyayuschegosya koeffitsienta stsepleniya. Moscow, 7. Available at: https://globalf5.com/Knigi/Nauka-Obrazovanie/Inzhnnerno-tehnicheskie-nauki/prikladnaya-geologiya-i/Gornoe-delo/Matematicheskaya-model-formirovaniya_176123
21. Zenkov, R. L., Ivashkov, I. I., Kolobov, L. N. (1987). Mashini neprerivnogo transporta. Moscow: Mashinostroenie, 432. Available at: https://www.studmed.ru/zenkov-rl-ivashkov-ii-kolobov-ln-mashiny-nepreryvnogo-transporta_2ff26c75477.html
22. Ushakov, Ye. N., Kostrykin, A. P., Shaidulin, K. V., Merzliakov, P. Ye. (2011). Methods of belt conveyor drive drum rubber lining laboratory tests conducting. Vestnik nauchnogo tsentra po bezopasnosti rabot v ugol'noy promyshlennosti. Available at: https://cyberleninka.ru/article/n/metodika-provedeniya-laboratornyh-ispytaniy-rezinovoy-futerovki-privodnyh-barabanov-lentochnyh-konveyerov
23. Affolter, C., Piskoty, G., Koller, R., Zgraggen, M., R tti, T. F. (2007). Fatigue in the shell of a conveyor drum. Engineering Failure Analysis, 14 (6), 1038–1052. doi: https://doi.org/10.1016/j.engfailanal.2006.11.071