Justification of the parameters of the effective design of the centering tension roller of the conveyor belt

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1 Introduction

In mechanical engineering, belt conveyors are used to transport various goods [1,2]. At the same time, depending on the loading of the conveyor belt, in particular, the change in the frequency and amplitude of the change in the transporting load, the parameters of the belt are selected [3,4].

Due to the uneven load on the belt in conveyors, the belt jumps out of the pulley and guide rollers. Therefore, it is considered expedient to center the tape to eliminate the sidetwist of the tape [5, 6].

The well-known design of the centering lever-hinged tension roller contains automatic adjusting devices with asymmetric configurations to eliminate the lateral exit of the tape or belt [7].

Another design of the centering lever-hinged tension roller is equipped with an additional pair of intersecting bracket-shaped connecting rods, each of which is hingly connected by one end to the other end of the corresponding mentioned connecting rod with an attached spring holder, and the other end to the shackle of the other part of the tension roller, and the axes of the hinged connections of the connecting rods with each other and the connecting rods with shackles are located parallel to each other[8]. The disadvantage of these structures is the limited functionally opportunity and low reliability: due to the large pulling force at high speed conditions, automatic control devices and limitness rotation speed of the tension roller; reducing the accuracy of transferring movement between the rollers due to high values

Abstract

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of reaction forces in the kinematic pairs of connecting rods and in the supports of the tension roller, and others.

In another known design of the centering tension roller [9], in order to increase the functionality by increasing the reliability of operation, it is provided with a spring attachment with at least one elastic element connected to the links of the intermediate device or to the corresponding shackle of one of the half of the tension roller, and the driven roller is installed with the possibility axial displacements. The disadvantage of this design is the low reliability of the system due to large deformation of the springs and a decrease in elasticity effects at high rates of withdrawal forces. This leads to instability and a decrease in the reliability of the system.

In the centering lever-hinge roller [9], the connecting rods are additionally connected to each other by telescopic connecting rods and spring attachments. The disadvantage of this mechanism is the complexity of the design and the limited speed modes of operation.

The task of the new design is to increase the reliability of the transmission of the elimination of lateral mixing of the conveyor belts from the central axis.

In belt conveyors with a tension roller, containing a driving and driven drums, a tension roller with a symmetrical curved profile and a conveyor belt covering them, the roller is made of two symmetrical parts with concave curved surfaces connected by a tension spring. In addition, the half-shafts of the roller parts can perform rotational and reciprocating movements along the axis and, accordingly, have compression springs between the roller parts and the body (worn on the half-shafts of the roller parts). Plastic bushings are fixed at the inner ends of the roller parts [10].

In this work, a new design of a centering tension roller has been developed, with a new adjusting device with a flexible element. This recommended new design serves to prevent the belt from coming off the pulley sideways or for production to prevent the belt from coming off the side of the drum. A new design has been developed and the operating principle of the design with a description has been proposed.

2 Development of an efficient idler roller design

The proposed design with a tension roller lies in the fact that with an increase in the shear force, one or another part of the roller moves axially in the direction of the shear force, thereby increasing the working contact area of the roller surface with the conveyor belt and, due to the concave curved surface of the roller parts, the belt descent is eliminated from the drums. At the same time, due to the selection of the required stiffness of the tension and compression springs, the roller automatically adapts to changes in the asymmetric forces of the belt coming off the belt conveyor.

The belt conveyor with a centering tensioning device contains a driving 1 and a driven 2 drums, covering their belts 3 a tension roller 4 consisting of two left 5 right 6 parts with curved surfaces, interconnected by a tension spring 7 installed on the semi-axes 8 and 9, rotating and reciprocating translational motion, between the side surfaces of parts 5 and 6 of roller 4 and supports 10 and 11, compression springs 12 and 13 are installed. On the inner parts 5 and 6 of roller 4, plastic bushings 14 and 15 are fixed (Fig. 1 and Fig. 2).

The driving drum 1, rotating, transmits the movement to the driven drum 2 through the covering belts 3. The shear force arising in this case is directed unilaterally (or a combination of asymmetric forces) contributes to the lateral descent of the belt 3. When this force is applied, the left 5 or right 6 parts of the roller 4 move axially from the middle of the belt 4. This increases the total length of the tension roller 4. Due to the concave curved surface of parts 5 and 6 of the roller 4, an additional axial force of the opposite shear force acts on the tapes 3, thereby eliminating the lateral descent of the tape 3 from the drums 1 and 2. The
spring 7 during the axial movement of parts 5 and 6 of the roller 4 is stretched and always tends to the initial position. Compression springs 12 and 13 under the influence of shear forces are compressed and always tend to their original positions, additionally helping the restoring force 7. Supports 10 and 11 are kinematic pairs of the IV-the class, that is, they have two degrees of freedom with half-ax 8 and 9 can rotate and move along the axes. In the process of work, to eliminate significant bending deformations of the tape 3 in the middle, as well as to prevent contacts of the tape 3 with the spring 7, plastic 14 and 15 are installed.

Fig. 1. Kinematic diagram of a belt conveyor with a tension roller.

Fig. 2. Scheme of the centering tension roller.

Choosing the characteristics of the springs 7, 12 and 13, as well as the parameters of the curvilinearity of the surfaces (geometric parameters) of parts 5 and 6 of the roller 4, it is possible to eliminate the slipping of the tape 3 from the drums 1 and 2 at different values of the shear forces (Fig. 2) [10].

3 Design scheme and mathematical model of the belt conveyor

In order to study the law of motion of the conveyor mechanism, we will draw up a calculation scheme. This diagram is shown in Fig. 3. It is a two-mass elastic transmission system. When compiling the differential equations of motion of this system, we use the Lagrange equation of the second kind [12]:

$$\frac{\partial}{\partial t} \left( \frac{\delta T}{\delta \dot{q}_i} \right) - \frac{\delta U}{\delta q_i} = \mathbf{M}_i$$ (1)

Where \( q_i \) - generalized coordinate and generalized speed;
T - Kinetic energy of the system;
\( \mathbf{M}_i \) - the moment from the generalized forces.
For the conveyor mechanism of the raw cotton riot puller, the kinetic energy will be:

\[
\mathbf{K} = \frac{1}{2} I_1 \dot{\varphi}_1^2 + \frac{1}{2} I_2 \dot{\varphi}_2^2
\]  (2)

Where \( I_1, I_2 \) - reduced moments of inertia to the shafts of the driving and driven drums.

The work of external forces acting on the system with an infinitesimal change in the generalized coordinates:

\[
\Delta A_{bn} = -(M_{rc} + M_{rt}) \cdot \delta \cdot \varphi_2 + M_d \cdot \delta \cdot \varphi_1
\]  (3)

Where \( M_{rc} \) - moment of resistance from raw cotton on the drum and during its transportation;
\( M_{rt} \) - additional resistance due to the tensioner;
\( M_d \) - the driving moment of the asynchronous electric drive (drive motor).

The work of internal forces acting on a machine unit with a conveyor mechanism for a raw cotton riot with an infinitely small change in the generalized coordinates:

\[
\Delta A_{bn} = [c(\varphi_1 - i \varphi_2) - b(\varphi_1 - i \varphi_2)] \delta \varphi_1 - [ic(\varphi_1 - i \varphi_2) - i^{-1} b(\varphi_1 - i \varphi_2)] \delta \varphi_2
\]  (4)

Where \( c, b, i \) - respectively, the stiffness coefficients of the viscous resistance and the gear ratio of the elastic transmission.

The moment \( M_i \) from the generalized force, corresponding to the coordinate \( \varphi_i \), which is the ratio of the work \( \Delta A_i \), produced by all forces (internal and external) acting on the system with an infinitesimal change in the generalized coordinate \( \varphi_i \), by the value \( \delta \varphi_i \) to this infinitesimal change in the coordinate \( \varphi_i \) equals

\[
M(\varphi_i) = \frac{\Delta A_i}{\delta \varphi_i}
\]

Substituting the obtained values of kinetic energy and moments of generalized forces into the Lagrange equation of the second kind, we obtain a system of differential equations for a machine unit with a raw cotton conveyor mechanism:

\[
I_1 \cdot \varphi_1 = M_d - c \cdot (\varphi_1 - i \cdot \varphi_2) - b \cdot (\varphi_1 - i \cdot \varphi_2)
\]

Fig. 3. Kinematic (a) and calculated (b) diagram of the mechanism of the raw cotton conveyor:
1 - Driven drum; 2 - driving drum; 3 - conveyor belt.

The moment \( M_i \) from the generalized force, corresponding to the coordinate \( \varphi_i \), which is the ratio of the work \( \Delta A_i \), produced by all forces (internal and external) acting on the system with an infinitesimal change in the generalized coordinate \( \varphi_i \), by the value \( \delta \varphi_i \) to this infinitesimal change in the coordinate \( \varphi_i \) equals

\[
M(\varphi_i) = \frac{\Delta A_i}{\delta \varphi_i}
\]
The resulting system of second-order differential equations (7) describes the motion of the considered machine unit. Its solution mainly depends on the mathematical description of the motor characteristics and resistances from raw cotton.

Analysis of the system of differential equations (7) shows that to simplify the problem, it is advisable to solve it analytically without taking into account the viscous resistance in the elastic transmission. In this case, according to the results of the solution, it is possible to obtain the maximum values of the amplitude of the load fluctuations in the elastic transmission and the coefficients of unevenness of the angular velocities of the driving and driven drums.

**4 Main tasks and analysis of results**

Let us determine the driving moment of the electric motor $Md$ according to its mechanical characteristics [13]:

$$M_d = \frac{M_n \omega_n}{\omega_0 - \omega_n} \frac{M_n \varphi_D}{\omega_0 - \omega_n}$$  \hspace{1cm} (8)

Technological resistance from raw cotton to the working drum and from transporting it by an elastic band is described as follows:

$$M_{rc} = M_1 + M_0 \sin \alpha t$$ \hspace{1cm} (9)

In this case, taking into account the above, (8), and (9), the system of differential equations of the machine unit of the raw cotton riot picker will be:

$$\begin{cases} l_1 \cdot \varphi_1 = \frac{M_n \omega_1}{\omega_0 - \omega_n} - \frac{M_n \omega_1}{\omega_0 - \omega_n} \cdot \varphi_1 - \frac{M_n}{\omega_0 - \omega_n} \cdot \varphi_1 \cdot \frac{1}{i} \cdot \varphi_1, \\ l_2 \cdot \varphi_2 = -M_1 - M_0 \cdot \sin \alpha t - M_{rc} + \frac{1}{i} \cdot \varphi_1 \cdot \varphi_2. \end{cases}$$ \hspace{1cm} (10)

From system (10), we define $\varphi_2$:

$$\varphi_2 = \frac{l_1}{i} \cdot \varphi_1 \frac{M_n}{\omega_0 - \omega_n} \cdot \varphi_1 - \frac{M_n}{\omega_0 - \omega_n} \cdot \varphi_1 + \frac{1}{i} \cdot \varphi_1$$ \hspace{1cm} (11)

Let us execute the first and second derivatives of $\varphi_2$ with respect to time and expression and substitute it into the second equation of system (10):

$$\begin{aligned} \frac{l_1 l_2}{i} \cdot \varphi_1^IV + \frac{l_2 l_1 M_n}{i (\omega_0 - \omega_n)} \cdot \varphi_1^IV + \frac{l_2}{i} \cdot \varphi_1 &= -M_1 - M_0 \cdot \sin \alpha t - M_p + \\
\frac{c}{i} \cdot \varphi_1 - \frac{l_2}{i} \cdot \varphi_1 - \frac{M_n}{i (\omega_0 - \omega_n)} \cdot \varphi_1 + \frac{M_n}{i (\omega_0 - \omega_n)} \cdot \varphi_1 + \frac{1}{i} \cdot \varphi_1. \end{aligned}$$ \hspace{1cm} (12)

To solve (12), we first solve the homogeneous equation without its right-hand side. Its characteristic equation is:

$$\lambda^4 + D_1 \cdot \lambda^3 + D_2 \cdot \lambda^2 + D_3 \cdot \lambda = 0$$

$$D_1 = \frac{M_n}{l_1 (\omega_0 - \omega_n)}; \quad D_2 = \frac{(l_1 + l_2)}{l_1 (l_1 + l_2)}; \quad D_2 + \frac{c M_n}{l_1 l_2 (\omega_0 - \omega_n)}$$ \hspace{1cm} (13)

Wherein

$$\lambda = \lambda^3 + D_1 \cdot \lambda^2 + D_2 \cdot \lambda + D_3; \quad \lambda_0 = 0$$ \hspace{1cm} (14)
We solve the cubic equation by the Cardano method [13, 14]. In this case, substituting the value in (14), we have:

**Equation roots**

\[ \lambda = \sqrt[3]{-\frac{p}{3} + \frac{q^2}{27}}; \quad z^3 + p \cdot z + q = 0 \]

Where:

\[ \gamma = \sqrt[3]{\frac{p}{3} + \frac{q^2}{27}} \quad \beta = -\frac{p}{3} \quad \epsilon = -\frac{q}{3} \]

**Equation roots**

\[ z_1 = \gamma + \beta; \quad z_2 = \gamma \cdot \epsilon + \beta \cdot \epsilon^2; \quad z_3 = \gamma \cdot \epsilon^2 + \beta \cdot \epsilon \]

Where:

\[ \gamma = \sqrt{-\frac{p}{9} + \frac{q^2}{27}} \quad \beta = -\frac{p}{9} \quad \epsilon = -\frac{q}{9} \]

**Equation roots**

\[ \lambda_1 = z_1 - \frac{D_1}{3}; \quad \lambda_2 = z_2 - \frac{D_1}{3}; \quad \lambda_3 = z_3 - \frac{D_1}{3}. \]

Substituting the roots of equation (13) into (14), the solution of the differential equation without the right-hand side is found:

\[ \varphi = A_1 \cdot e^{\lambda_1 \cdot t} + A_2 \cdot e^{\lambda_2 \cdot t} + A_3 \cdot e^{\lambda_3 \cdot t} \]

Where:

\[ A_1 = \frac{c}{l_1 \cdot l_2} \left( \frac{M_p \cdot \omega_o}{\alpha - \omega_n} - M_1 \cdot i - M_p \cdot i \right) \]

\[ A_2 = \frac{M_p \cdot \omega_o - \alpha \cdot \omega_n}{l_1 \cdot l_2 \cdot \left( D_1 \cdot \alpha^2 - D_3 \cdot \alpha \right) + \left( \alpha^2 - D_2 \cdot \alpha \right)^2} \]

In this case, the general solution of the differential equation (12) will be:

\[ \varphi = A_1 \cdot e^{\lambda_1 \cdot t} + A_2 \cdot e^{\lambda_2 \cdot t} + A_3 \cdot e^{\lambda_3 \cdot t} \]

Substituting the values \( A_1, A_2, A_3 \) in (20), taking the time derivative and taking into account the initial conditions at \( t = 0 \), \( \varphi_0 = 0 \), \( \dot{\varphi}_0 = 0 \) according to expression (21), it is possible to determine the law of motion of the rotor of an asynchronous electric motor. Taking the time derivative of \( \varphi_0 \), one can obtain an expression for determining the
angular velocity of the rotor. Taking into account the gear ratio of the transmission, it is possible to obtain the angular displacement and the angular velocity of the conveyor drum of the raw cotton riot separator[7].

Determine the values of the coefficients

\[
\alpha_i = \frac{\lambda_i \cdot \lambda_i + \lambda_i \cdot \lambda_i + \lambda_i}{\lambda_i} - \lambda_i
\]

\[
\beta_i = \frac{\alpha_i \cdot \alpha_i + \lambda_i \cdot \lambda_i}{\lambda_i} - \lambda_i
\]

\[
\gamma_i = \frac{\lambda_i \cdot \lambda_i + \lambda_i \cdot \lambda_i + \lambda_i}{\lambda_i} - \lambda_i
\]

(22)

The calculations were carried out with the following values of the parameters of the electric motor and the conveyor of the raw cotton riot separator:

<table>
<thead>
<tr>
<th>Table 2.</th>
<th>Values of the parameters of the electric motor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electric motor brand</td>
<td>4A112M6UZ</td>
</tr>
<tr>
<td>Engine characteristics</td>
<td></td>
</tr>
<tr>
<td>power $P$, kW</td>
<td>3.0</td>
</tr>
<tr>
<td>rotation speed $n$, min$^{-1}$</td>
<td>955</td>
</tr>
<tr>
<td>$M_{\text{START}} / M_{\text{NOM}}$</td>
<td>1.0 ... 1.2</td>
</tr>
<tr>
<td>$M_{\text{MAX}} / M_{\text{NOM}}$</td>
<td>1.6 ... 2.2</td>
</tr>
<tr>
<td>Conveyor characteristics</td>
<td></td>
</tr>
<tr>
<td>drive ratio $i$</td>
<td>4</td>
</tr>
<tr>
<td>coefficient of stiffness of elastic transmission $C$, N·m / rad</td>
<td>$290 \cdot 10^4$</td>
</tr>
<tr>
<td>nominal torque $M_{\text{NOM}}$, N·m</td>
<td>29.8</td>
</tr>
<tr>
<td>moment of additional resistance due to the tensioner $M_{\text{P}}$, N·m</td>
<td>35.76</td>
</tr>
<tr>
<td>initial moment $M_0$, N·m</td>
<td>29.8</td>
</tr>
<tr>
<td>moment of inertia of the driving drum $I_1$, N·m·s$^2$</td>
<td>$4.71 \cdot 10^{10}$</td>
</tr>
<tr>
<td>moment of inertia of the driven drum $I_2$, N·m·s$^2$</td>
<td>$0.3 \cdot 10^{10}$</td>
</tr>
<tr>
<td>starting torque $M_{\text{P}}$, N·m</td>
<td>29.8</td>
</tr>
<tr>
<td>maximum moment $M$, N·m</td>
<td>59.6</td>
</tr>
<tr>
<td>initial angular velocity $\omega_0$, s$^{-1}$</td>
<td>100.5</td>
</tr>
<tr>
<td>nominal angular velocity $\omega_N$, s$^{-1}$</td>
<td>25</td>
</tr>
<tr>
<td>Significant coefficients:</td>
<td></td>
</tr>
<tr>
<td>$P_1$, s$^{-1}$</td>
<td>$1028 \cdot 10^{-6}$</td>
</tr>
<tr>
<td>$q$, s$^{-3}$</td>
<td>$5.23 \cdot 10^{-15}$</td>
</tr>
<tr>
<td>$D_1$, s$^{-1}$</td>
<td>$0.084 \cdot 10^{-16}$</td>
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<tr>
<td>$D_2$, s$^{-2}$</td>
<td>$1028 \cdot 10^{-6}$</td>
</tr>
<tr>
<td>$D_3$, s$^{-3}$</td>
<td>$81 \cdot 10^{-16}$</td>
</tr>
</tbody>
</table>
Fig. 4. Dependence of angular displacements (a), angular velocities (b) and driving moments (c) of the raw cotton conveyor drum over time.

Production test results for conveyor belt with recommended centering idler roller. The technical and economic efficiency of the proposed belt conveyor with a tensioning device is to increase the reliability and efficiency of the conveyor by eliminating the side slip of the belt.

Fig. 5. General view of the tensioner prototype.

To study the efficiency of the effective design of the centering tensioner, a prototype was made from the Kaprolon-V material and installed on a belt conveyor of the TLH-18 type. Figure 5. shows a general view of the installation of a prototype of the recommended design on a conveyor belt of the TLH-18 type.

Comparative tests of the modernized belt conveyor with the recommended centering tension roller show high performance and reliability in operation. At the same time, there is no actual slipping of the tape from the drums, even if the axis of the drums is not aligned (7-8 degrees), the slaughter of raw cotton during transportation is eliminated. Productivity has increased by 12% but in relation to the existing structure. The service life of the recommended structure has increased (10-15) percentage.

5 Conclusion
The effective design of the centering tension roller of the belt conveyor is recommended. The regularities of the movement of the conveyor drums are obtained, the parameters of the system are substantiated. Production tests of the modernized conveyor are presented.

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