Modeling of the flexible belt motion on the drive pulley of woodworking equipment

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Abstract. Purpose of the work. Establishing the patterns of a flexible belt movement on a drive pulley, to determine the parameters of power and kinematic interaction of the belt with the drive pulley, taking into account mechanical characteristics of the drive, belt and their contact interaction. Method of the work. Computer simulation of belt movement on the drive pulley as a multi-body system using the Universal Mechanism software package. Results. A computer-based multi-body model of the belt movement on the drive pulley has been developed. The model allows determining the parameters of the force and kinematic interaction of the belt with drive pulleys taking into account the viscoelastic properties of the belt, the nonlinear dependence of the friction force on the sliding speed and the elasticity of the contacting surfaces. The model includes a closed belt covering the drive and brake pulleys. The belt represented by 36 discrete bodies connected by elastic-dissipative elements. The computer model was verified using theoretical dependences. A technique for analyzing the belt movement on the drive pulley is demonstrated. The computer model and methodology are intended to analyze the operation of friction belt drives, belt conveyors, belt grinding machines in the design of woodworking equipment.

1. Introduction

The forestry complex includes harvesting, machining and wood processing. It consists of industries that differ in production technology, but use the same raw materials. In the woodworking industry, the friction drive of flexible belts is widely used in belt drives, belt conveyors, and belt grinding machines. The uneven movement of the belt on the drive pulley leads to slipping and increased wear of the belt, reducing the quality of grinding. The slipping of the belt on a drive pulley causes fires.

For the first time, the law of force transfer by a flexible belt by means of friction forces was introduced in 1775 by Leonard Euler for a weightless, absolutely flexible and inextensible thread, thrown over a fixed pulley with an angle of coverage \( \alpha_0 \). The thread under the action of the forces \( S_1 \) and \( S_2 \) applied to its ends evenly slides in the direction of the force \( S_1 \), exceeding the force \( S_2 \) by the amount of the friction force arising between the thread and the pulley surface [1].

Numerous theoretical studies of the interaction of a belt with a pulley are devoted to the creation of mathematical models that take into account the forces of inertia of moving bodies, the flexural rigidity of the belt, the dependence of friction on the contact parameters, the aerodynamic effect. These studies were carried out under the assumption that the belt monotonously moves along the surface of the pulley, although during the operation of belt drives and belt conveyors there are jerks of the traction member in the longitudinal direction, worsening the operating performance of the machines.
In [2], the transverse oscillations and lateral displacements of the sanding belt are considered, and the requirements for the belt drive mechanisms are formulated. It is shown that the effectiveness of the grinding depends on the belt transverse vibrations parameters.

The monograph [3] discusses the improvement of grinding processes of chipboards and profile milling of medium density fiberboards. Process improvement is based on improving the quality of the surface of wood boards by reducing its roughness. The reduction in surface roughness of the plates is ensured by the use of milling cutters with certain angles and the required speed of rotation of the belt drive.

The equations of motion for an extensible belt on a pulley with all effects of inertia were derived in [4]. Ignoring both centrifugal and stretching accelerations results in an over-prediction of the maximum moment that can be transmitted, and for a given transmitted moment, under-prediction of the slip angles on the driven pulley.

One reason of the uneven belt movement is its frictional self-oscillations on the drive pulley. In [5], the motion of a discrete belt element over the surface of a pulley is considered. The dependence of the friction force of the belt at the pulley surface on the sliding speed is represented by a power series. The equation of the belt self-oscillations with considering of centrifugal and Coriolis forces at constant speed of the pulley rotation is made. A numerical solution of the self-oscillations equation is obtained. It showed that the belt slip speed increases with the rise of angular velocity of the pulley. This model allows a qualitative representation of the longitudinal motion of a belt single element on the drive pulley, taking into account the nonlinear dependence of the frictional force on the sliding speed.

In [6], a system of two nonlinear differential equations describing self-oscillations of a belt and a drive pulley with an asynchronous electric drive is obtained. An analytical solution is obtained for the stationary operating mode of the drive without regard for nonlinearity.

Analytical model and solutions for the tensions and the relative displacements of the belt over the pulleys are established in [7] for the stationary regime. During the belt slip over the pulleys, two sliding zones situate at the onset and the end of the contact. Their lengths become greater when the transmitted torque increases. For the transient regime, differential equations for the belt tensions and the driven pulley velocity are derived. It is noted that the belt tensions and the driven pulley velocity have damped oscillations before the stationary regime is reached. The response of the pulley-belt system to an engine acyclism demonstrate the losses due to the belt slip in the transient regime. The model does not take into account the inertial and mechanical characteristics of the drive, the viscoelastic properties and the bending stiffness of the belt, the dependencies of the belt-pulley contact interaction.

In [8], an exact solution of the nonlinear steady-state equations of a string is obtained which includes elastic extension, Coulomb friction, radial and tangential accelerations, and the power loss due to friction between the belt and the pulleys. An example of a two pulleys system is considered, which shows that approximations of compatibility conditions used in previous solutions lead to an underestimation of the maximum transmitted moment, an overestimation of efficiency, an underestimation of both low and high values of stresses at full slip.

These analytical models do not allow to study the belt movement on the drive pulley effectively, taking into account the inertial and mechanical characteristics of the drive, the viscoelastic properties of the belt, the various dependencies of the belt movement resistance and nonlinear contact interaction.

The purpose of the work is establishing regularities of a flexible belt movement on a drive pulley, determination of the force and kinematic parameters of the belt interaction with the drive pulleys taking into account mechanical characteristics of the drive, belt and their contact interaction.

2. Description of the computer model

Computer simulation of the belt movement on the drive pulley is possible with the use of software packages Universal Mechanism, Adams, etc. Figure 1 presents a computer model of a belt covering the drive and brake pulleys.
Figure 1. The model of a closed belt loop covering the drive and brake pulleys created within Universal Mechanism software.

The model includes a closed belt represented by 36 discrete bodies connected by linear elastic-dissipative elements. Moments of inertia of discrete bodies and the belt bending rigidity are not taken into account.

The X axis of the model coordinate system coincides with the axis of rotation of the drive pulley. The Y axis passes through the axes of rotation of the both pulleys and the Z axis is directed vertically upwards.

The shaft of the drive pulley doesn’t move along the Y axis. The shaft of the brake pulley is connected to the take-up device and can move along the Y axis. The model includes the friction force of the brake pulley slider along the guide and the elastic-dissipative tensile force of the plummet rope.

The contact interaction of the belt with the pulley is represented by the point-to-circle contact forces. A ductile contact allows the introduction of one contacting body into the surface of another body. The transition from static friction to kinetic friction occurs if the slipping speed reverses direction. The magnitude of the static friction force is determined by the rigidity of the contacting surfaces [9].

In the case of sliding, the friction force $F$ is proportional to the normal force $N$ between the contacting bodies and is directed against the speed of slipping.

$$ F = fN, $$

where $f$ is the friction coefficient, the value of which in the general case depends on the speed of slipping $V_s$.

The variety of materials used to make the cover of conveyor belts and pulleys, different relief of covers, as well as substances contaminating the contacting surfaces, causes different types of dependence $f(V_s)$.

Rubber coated belts running over pulleys in practice display friction coefficients between 0.3 and 0.8 [10]. This paper studies the causes of the friction variations. It has been found that variations in the belt manufacture method are significant in explaining differences in friction behavior.

In [11], an increase in the friction coefficient was noted with increasing sliding speed of the rubber-fabric belt along the steel pulley and a decrease in the pressure of the contacting surfaces. In [12], data are given that, at negative temperatures, there is an increase in $f$ with increasing temperature, and at positive temperatures, a decrease. So for the pair SKS-50 rubber and polished steel at temperature of 40 °C, the greatest value of $f$ reaches at $V_s = 0.06$ mm/s and decreases by 1.1 times at $V_s = 18$ mm/s.

The classical model of the dependence of dry friction on speed $V_s$ is not convenient for numerical simulation of the adhesion mode, since it leads to jumps in the friction force and acceleration in the numerical solution if velocity $V_s$ is close to zero. In the presence of moisture or contamination in
contact and taking into account the Stribeck effect, $f$ can be represented as a continuous function of $V_S$ [13].

The dependence of the coefficient $f$ on the value of $V_S$ is realized by a combination of an exponential model that takes into account the Stribeck effect and viscous friction

$$f(V_S) = f_x + (f_0 - f_x) \exp\left(-\frac{V_S}{v_{st}}\right)^\delta + v V_S.$$  \hspace{1cm} (1)

Here $f_0, f_x$ are the friction coefficients at zero slipping speed and at an infinite slipping speed; $v_{st}$ is the Stribeck speed determining the slipping speed interval at which the effect of the exponential decrease of the friction coefficient takes place; $\delta \in [0.5; 1]$ is an empirical exponent; $v$ is the coefficient of viscous friction.

With an asynchronous electric motor, the torque of the drive pulley $M_1$ is represented by the dependence

$$M_1 = M_{nom} \omega_k \left(1 - \frac{\omega_1}{\omega_s}\right),$$  \hspace{1cm} (2)

where $M_{nom}$ is the nominal torque of the drive pulley, $\omega_k$ is the nominal sliding of the electric motor, $\omega_1$ is the angular velocity of the drive pulley, $\omega_s$ is the synchronous angular velocity of the electric motor, $I$ is the gear ratio of the drive.

The linear relation (2) is well suited for the quasi-steady-state operation of an asynchronous motor. The belt movement resistance is realized with the brake pulley. The braking torque of the brake pulley is represented by the linear dependence

$$M_2 = M_{C0} + M_{C1} \omega_2,$$  \hspace{1cm} (3)

where $M_{C0}$ and $M_{C1}$ are the resistance coefficients, $\omega_2$ is the angular velocity of the brake pulley.

The developed computer model, in contrast to the theoretical models [4-8], allows determining the characteristics of the conveyor friction pulley drive more accurately, taking into account the inertial and mechanical characteristics of the drive and the resistance to belt motion, mass and viscoelastic properties of the belt, parameters of nonlinear friction contact.

3. Verification of the computer model

To verify the computer model, we use the formulas of J. Poncelet and M. Kretz [14]. According to the Poncelet formula, the ratio of the tension forces of the upper strand $S_1$ and the lower strand $S_2$ is

$$2S_o = S_1 + S_2,$$

where $S_o$ is the pre-tension force of the belt (before the drive is turned on).

According to the Kretz formula, the ratio of the angular velocities $\omega_1$ and $\omega_2$ of the driving and the brake pulleys looks like

$$\frac{\omega_1 R_1}{1 + S_1/E_o} = \frac{\omega_2 R_2}{1 + S_2/E_o},$$

where $E_o$ is the tensile rigidity of the belt, $R_1$ and $R_2$ are the radii of the drive and brake pulleys.

We also verify the formula of V.A. Dobrovolsky (p. 61 [14]).

$$\frac{V_1}{1 + S_1/E_o} = \frac{V_2}{1 + S_2/E_o},$$

where $V_1$ and $V_2$ are the speed of the belt at the points of run-in and run-off the drive pulley.

In addition, we compare the values of the tension $S_1, S_2$ and the torque of the driving pulley $M_1$ using formula
Verification of the computer model is carried out with the following values of the belt conveyor drive parameters: the mass of the belt \( \rho = 16 \) kg/m; the length of the belt represented by a discrete belt body (BB) \( l_0 = 0.1 \) m; \( E_0 = 600 \) kN; the rigidity of the elastic-dissipative element \( C_I = 6 \) MN; the coefficient of dissipation of the elastic-dissipative element \( \mu_1 = 5.6 \) kN·s/m; \( \omega_k = 5.2 \) rad/s; \( \varepsilon = 0.04 \); \( M_{nom} = 5000 \) N·m; \( M_{C0} = 2500 \) N·m; \( M_{C1} = 500 \) N·m; \( R_1 = R_2 = 0.4 \) m; the central angle of the belt and the drive pulley contact \( \alpha_o = \pi, f_0 = f_\infty = 0.35 \); the coefficient of contact stiffness \( C_n = 1.0 \) MN/m; the coefficient of contact dissipation \( C_d = 500 \) N·s/m; \( \mu = 0 \); the mass of take-up device \( M_t = 5000 \) kg, the friction force of the brake pulley slider \( F_n = 192 \) N.

In the quasi-steady-state mode of operation, the following average simulated results were obtained: \( S_0 = 48.86 \) kN; \( S_1 = 30.74 \) kN; \( S_2 = 18.25 \) kN; \( \omega_1 = 4.99 \) rad/s; \( \omega_2 = 4.81 \) rad/s; \( M_1 = 4991 \) N·m; \( V_1 = 1.999 \) m/s; \( V_2 = 1.948 \) m/s. The relative error of the Poncelet formula is 0.2%. The relative error of the Kretz formula is 1.7%. The relative error of the Dobrovolsky formula is 0.6%. The relative error of the difference between the tension forces \( (S_1 - S_2) \) and the \( M_1/R_1 \) value is 0.1%. Thus, the results of the computer model verification can be considered satisfactory.

4. Technique of the belt movement analysis

The aim of the analysis is to establish the regularities of the belt quasi-steady motion on the drive pulley, including the dimensions of the adhesion and sliding zones, the belt speed slipping along the pulley surface, the presence of tension-compression waves in the belt, the determination of the pulley thrust.

Preliminary approbation of the computer model allowed to formulating the following provisions of the methodology.

1) We shall consider the quasi-steady state of the belt motion when the changes in the tension of the take-up device rope and the angular velocity of the drive pulley do not exceed 1% during the simulation.
2) The \( V_S \) speed of slipping of a BB is calculated as the difference between the absolute values of the circumferential velocity of the pulley surface and the \( V_i \) velocity of a BB

\[
V_S = R_i \omega_i - V_i ,
\]

where \( R_i \) is the actual distance from a BB to the axis of rotation of the drive pulley, taking into account the contact deformations.

3) The dimensions of the adhesion and sliding zones are determined using the calculated friction coefficient of a BB

\[
f_c = \frac{F}{N}.
\]

If \( f_c < f_0 \) for \( f_0 > f_\infty \), there is no slip of the belt along the pulley surface, and the relative displacement occurs due to elastic deformation.

If \( f_c > f_0 \) for \( f_0 > f_\infty \), then the belt slips over the pulley surface.

For \( f_\infty < f_0 \), the values of \( f_c \) must be compared with the \( f(V_S) \) values calculated with the formula (1).

Slip of the belt along the pulley surface is absent, if \( f_c < f(V_S) \) and the relative displacement occurs due to elastic deformation.

For comparison with the theoretical value, the central angle \( \alpha_{sl} \) of the belt sliding zone on the drive pulley is calculated with the formula

\[
\exp(f_c \alpha_{sl}) = S_i / S_2.
\]

The change in the angle dimension of the pulley belt wrap due to the oscillations of the upper and lower strands of the belt is neglected.
Due to oscillations, BBs have different sizes of the adhesion and sliding zones, so the resulting values will be calculated as an average for 12 successively moving BBs.

4) Waves of tension-compression of the belt at the contact zone can be identified through the distances between adjacent BBs. If the distances of different BB pairs change identically, then a monotonic longitudinal tension of the belt without longitudinal waves occurs. Otherwise, there are longitudinal waves of tension-compression.

5) The thrust of the driving pulley $F_T$ is determined through the torque $M_1$, averaging its values during the simulation:

$$F_T = M_1 / R_1.$$ 

6) To estimate the wear of the belt and the drive pulley, the power of the friction force is determined at the relative displacement of the belt at the contact zone.

7) To simulate the slipping conditions of the drive pulley, we vary the mass value of the take-up device. The degree of the drive pulley slippage $P_{sl}$ is estimated through the ratio of the projection on the Y axis of the $V_Y$ speed of the oncoming strand of the belt to the peripheral speed of the pulley:

$$P_{sl} = V_Y / R_1 \omega_1.$$ 

5. Simulation results

The quasi-steady-state operating mode of the drive pulley is considered with the same parameter values as in the verification.

Even in the quasi-steady state, when the axis of the take-up pulley does not move after decay of the transient start processes, the angular velocity of the drive pulley with an asynchronous drive oscillates within 0.5% due to longitudinal and transverse vibrations of the belt, nonlinearity of the contact interaction forces.

At the angle of the belt coverage of the drive pulley $\alpha_0 = \pi$ rad, 12 BBs simultaneously placed at the contact zone. In the model, BBs are numbered sequentially from No. 1 to No. 36.

The belt and drive pulley contact includes the adhesion and sliding zones. The adhesion zone is the part of the contact surface at which the relative displacements of the contacting points of the belt and the pulley are absent. The adhesion zone realizes static friction.

The transition from static friction to kinetic friction marks the beginning of the sliding zone. The sliding zone is the part of the contact surface where the relative displacements of the contacting points of the belt and the pulley occur. At $f_0 = f_\infty = 0.35$, the theoretical value of the central angle of the sliding zone $\alpha_{sl} = 1.5$ rad. That is $\alpha_{sl} = 0.477 \alpha_0$.

The simulation results showed that all BBs have the sliding zone at the end of the contact. The central angle of the sliding zone for different BBs varies from $0.29\alpha_0$ to $0.35\alpha_0$ and is, on the average, $0.326\alpha_0$. This means that the drive pulley transmits to the belt 15% of the traction force on the adhesion zone and 85% on the sliding zone.

Figure 2 shows the graphs of changes in the distance between neighboring BBs. So the distances between the adjacent BBs Nos. 6 … 14, 16 and 17 are reduced monotonically, correspondingly to a decrease in the tensile force of the belt. The distances between the BBs Nos. 14 and 15, 15 and 16 vary sinusoidally, which corresponds to a longitudinal tension-compression wave with a length equal to the contact length and amplitude of about 0.005 m. The time interval $[4.68, 5.88]$ (s) is needed so that BBs Nos. 6 … 17 cross the contact zone. The initial point (4.68 s) depends on initial conditions of modelling.
The graphs in figure 2 show that the movement of the belt relative to the drive pulley surface is nonmonotonic in nature. There are irregular longitudinal tension-compression waves. So, on a piece of the belt between BBs No. 14 and No.16 a longitudinal traveling wave of tension-compression has sprung up.

The intensity of the longitudinal waves of the belt at the drive pulley contact changes significantly when the contact parameters change. Thus, at \( f_0 = 0.35, f_\infty = 0.38, v_{str} = 0.01 \text{ m/s}, \nu = 0.2, \delta = 1 \), a discontinuous motion of the belt with short-term stops on the adhesion zone is observed.

Figure 3 shows the graphs of changes in the calculated friction coefficient \( f_c \) of the BBs at the drive pulley contact.

Based on the results of modeling, the friction forces of the BBs Nos. 7, 9, 12, 14 and 16 change directions on the adhesion zone, and the friction forces of the remaining BBs do not. This corresponds to a change in the sign of \( f_{12}, f_{14} \) and \( f_{16} \) in figure 3. (The graphs of the variation of the coefficients \( f_7 \) and \( f_9 \) are not shown in figure. 3.) The \( f_c \) value varies from 0.35 to 0.404 on the sliding zone. The
central angle of the sliding zone $\alpha_{sl}$ for different BBs varies from 0.21$\alpha_0$ to 0.36$\alpha_0$ and is, on average, 0.28$\alpha_0$. Thus, at the adhesion zone, the drive pulley transmits to the belt 13% of the traction force and 87% on the sliding zone. Despite the reduction in the sliding zone size compared to the example in figure 2, it realizes a larger traction force due to an increase in $f_c$.

Experimental studies [15] have shown that the size of the sliding zone increases almost linearly with increasing thrust. A six-ply rubber-fabric belt with a thickness of 12 mm, enclosing half the circumference of a steel pulley with a diameter of 0.4 m, had the adhesion zone of 0.15% of the contact and transmitted 15% of the traction force [1].

Figure 4 shows graphs of the BBs sliding speed at the drive pulley contact. The average sliding speed of the BBs at the contact zone is 0.02 m/s. Although the projection of the belt speed on the Y-axis is less than the circumferential speed of the pulley at the point of on-run, the sliding speed of all BBs (Nos. 6...17) change direction at the adhesion zone. With positive values of $V_{ck}$, the surface of the pulley overtakes the belt. With negative values of $V_{ck}$, the belt moves faster than the surface of the pulley.

**Figure 4.** Graphs of changes in the sliding speed of BBs Nos. 6, 8, 10, 12, 14, 16 and 17 at the contact zone (i.e. $V_{ck6}$ is the sliding speed of BB No. 6).

The relative displacements of BBs have different values at the end of the contact zone due to longitudinal waves of tension-compression of the belt and reach amplitude of 0.005 ... 0.0095 m.

An important practical result of the simulation is the ability to determine the friction power of the belt relative displacement at the contact. The average power of the drive pulley during the simulation is 24.730 W. The average power of the friction force of the belt relative displacement at the contact is 74.8 W. This is 0.3% of the power realized by the drive pulley. Thus frictional losses due to slip of the belt on the drive pulley are less than 1% under normal operating conditions and do not have a noticeable effect on the efficiency of the electromechanical drive, but, as noted in [16], are important for evaluating the wear and temperature of the belt and pulley.

Consider the process of the belt quasi-stationary sliding on the drive pulley, when the changes in the angular velocity of the drive pulley do not exceed 1%.

The skidding mode is called the operating mode of the drive pulley when along the entire contact zone the belt slides relative to the surface of the pulley. The beginning of skidding is accompanied by a change in the torque of the electric motor and the thrust transmitted by the drive pulley to the belt [16].

The causes of the skidding mode are insufficient tension of the belt and weak friction of the belt at the pulley. Thus, in [17] it was experimentally established abrupt drop in $f_0$ coefficient at the contact of the belt with the pulley even within insignificant water content (3 g/m²).
Consider the skidding mode at the following values of the parameters: \( C_1 = 6 \text{ MN} \); \( \mu_1 = 5.6 \text{ kN·s/m} \); \( \omega_s = 5.2 \text{ rad/s} \); \( \varepsilon_k = 0.04 \); \( M_{\text{com}} = 5000 \text{ N·m} \); \( M_C = 2500 \text{ N·m} \); \( R_1 = R_2 = 0.4 \text{ m} \); \( \alpha_\omega = \pi \), \( f_0 = f_\infty = 0.3 \); \( C_0 = 1.0 \text{ MN/m} \); \( C_d = 500 \text{ N·s/m} \); \( \nu_{\text{str}} = 0.01 \text{ m/s} \); \( \nu = 0.2 \); \( F_{\text{tu}} = 192 \text{ N} \).

Figure 5 shows the graphs of the speed of BB No. 6 at the contact zone at different tension of the belt caused by different mass of take-up device. The graphs in figure 5 show that reducing the \( M_{\text{tu}} \) mass to 2700 kg results in slippage with longitudinal auto-oscillations of the belt. The amplitude of the belt speed oscillations is 0.04 m/s (about 2% of the average belt speed), the frequency is, approximately, 10 Hz.

![Figure 5. Graphs of the change in the speed of BB No. 6 (m/s) at the contact zone (V6_6000 is speed of the BB No. 6 at \( M_{\text{tu}} = 6000 \text{ kg} \) and V6_2700 is speed of the BB No. 6 at \( M_{\text{tu}} = 2700 \text{ kg} \).)](image)

The model value of the slip angle \( \alpha_{sl} \) is 2.63 rad, i.e. \( \alpha_{sl} = 0.837\alpha_\omega \). With an average slip speed of 0.04 m/s, according to formula (2), \( f_c = 0.309 \) and the theoretical value of the slip angle \( \alpha_{sl} \) is 3.09 rad, i.e. \( \alpha_{sl} = 0.984\alpha_\omega \). The time interval \([5.14, 5.80]\) (s) is needed so that BBs No. 6 crosses the contact zone.

The model value of the slip angle is less than the theoretical one, since the belt slides along the pulley unevenly, and the friction force retains large values in the adhesion zone.

The nonmonotonic slipping of BBs relative to the pulley takes place at the contact zone. All BBs that are currently in contact with the pulley or most of them simultaneously change the slipping direction with frequency nearby 10 Hz.

At the end of the contact zone, the BB sliding speed reaches 0.08 ... 0.1 m/s, and the relative displacement is 0.019 m. The sliding speed of the belt \( V_S \) at the beginning of the contact zone is calculated as the average value for 12 or more BBs

\[
V_S = R_1\omega_\omega - V_Y .
\]

The used computer model corresponds to a stable mechanical system, since a decrease in the speed of the belt, according to (3), leads to a decrease in resistance to its movement. Therefore, there is no pronounced transition from normal operation to the sliding mode of the drive pulley during simulation. As the tension of the belt decreases, the belt velocity, the sliding speed, the torque and the angular velocity of the drive pulley change gradually.

Reducing the mass of take-up device \( M_{\text{tu}} \) from 6000 to 2200 kg results in a slight increase in the average angular velocity of the drive pulley from 4.99 to 5.04 rad/s, reducing the torque by 1.31 times and reducing the sliding speed of the belt at the point of onset on the drive pulley in 2.6 times, an increase in the degree of slippage \( P_{sl} \) from 0.983 to 0.992.
It is impossible to determine visually the presence of skidding due to the low speed of the belt slipping (0.1 ... 0.2 m/s). Therefore, the conveyor control system must have sensors for controlling the speed of the drive pulley rotation and the belt speed with the corresponding sensitivity.

Average during the simulation of skidding, the power of the drive pulley is 23452 W at $M_{tu} = 2700$ kg. The average power of the friction force due to the belt relative displacements at the contact zone is 262.8 W. This is 1.1% of the power realized by the drive pulley. That is, skidding increases the energy loss by 3.7 times and rises temperature of the contacting surfaces. Therefore, in addition to the speed sensors of the drive pulley and belt, the drive control system can use the surface temperature sensors of the pulley and belt.

6. Conclusion
The computer-based multi-body model of the flexible belt movement on the drive pulley has been developed. The model was verified using theoretical dependencies. The model allows determining the parameters of the force and kinematic interaction of the belt with drive pulleys taking into account the viscoelastic properties of the belt, the nonlinear dependence of the friction force on the sliding speed and the elasticity of the contacting surfaces. The technique for analyzing the movement of a belt on a drive pulley has been developed. The computer model and methodology allows determining the nature of the belt motion on the drive pulley, confirms the presence of the adhesion and sliding zones. The theoretical size of the sliding zone is exceeded the model value by the fact that the adhesion zone transmits part of the traction force.

The movement of the belt on the drive pulley is nonmonotonic in nature with irregular wave displacements relative to the surface of the pulley. When skidding, longitudinal self-oscillations of the belt at the contact zone take place. Frictional losses due to slip of the belt on the drive pulley are less than 1% under normal operating conditions and do not have a noticeable effect on the efficiency of the electromechanical drive, but the losses values are important for evaluating the wear and temperature of the belt and pulley.

When the traction capacity of the drive pulley is reduced, the transition from normal operation to skidding has a smooth character. It is impossible to determine the presence of skidding visually due to the low slip speed of the belt; therefore, the drive control system must have sensors for monitoring the speed of the drive pulley rotation and the belt speed of the corresponding sensitivity for fire safety of woodworking equipment.

When skidding, frictional energy losses increase in 3 or more times. This makes it possible, in addition to the speed sensors of the drive pulley and the belt, to use the surface temperature sensors of the drive pulley and the belt in the drive control system.

The model lets assess the loss of power due friction, the wear of the belt and quality of grinding at design stage. It is advisable to use the computer model, methodology and patterns of belt movement in the design and assessment of fire safety of woodworking equipment.

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