Research Of Lateral Assembly Of The Belt In Flat-Belt Transmissions And Transport Mechanisms

Bakhtiyorjon Davidboev, Yunus Mirzakhanov, Ikrorkhon Makhmudov, Nargizakhon Davidboeva

Abstract: In the given scientific article results of scientific researches belting of transfer with centering tension rollers by the device are resulted (brought). Are certain (determined) of the basic parameters centering tension rollers of the device, durability of belts and to avoid this we recommend to use a new tensioner with asymmetrical section. Non-uniform distribution of a load along the belt width, asymmetry of axes of leading and driven tensioners, as well as excessive or loose tension result in occurrence of shifting forces. Under the action of these forces belt deviates from its central position. To avoid this, we recommend to use a new tensioner with asymmetrical section.

Index Terms: Belt, circumferential force, diagram, force, lateral assembly, transmissions, transport mechanisms.

1 INTRODUCTION

Tension rollers are used mainly in flat-belt gears with large gear ratios and small interaxal distances. They make it possible to transmit large power with the same overall dimensions of the transmission. Flexible connection is in more difficult conditions due to additional bending in the opposite direction. In this case, the variable component of the stress cycle will be greater than in a transmission without a tension roller [1].

2 METHODS OF RESEARCH

Fig. 1 shows the kinematic diagram of a flat belt transmission with non-parallelism of the axes of rotation of the pulleys and the angle of inclination of the axis of rotation of the tension roller. Knowing that the power and frequency of rotation of the electric motor are known, we can determine the estimated speed of movement for various diameters of the driving pulley [2,3,4]:

\[ V_1 = \frac{\pi \cdot D_1 \cdot n}{60}; \quad V_2 = \frac{\pi \cdot D_2 \cdot n}{60}; \]  

where \( D_1, D_2 \) diameters of driving and driven pulleys.

Circumferential force \( F_c \), and resultant \( F_b \) belt drive forces according to the force diagram shown in Fig. 1, we determine by the formulas:

\[ F_b = \frac{60 \cdot k \cdot P}{\pi \cdot D \cdot n_{DB}}; \]  

where \( P \) – power consumption of the electric motor; \( k \) – coefficient taking into account the type of belt.

According to the force diagram in Fig. 1, the shear force \( F_{SD} \) arising from the non-parallelism of the axis of rotation of the pulleys is determined:

\[ F_{CD} = \frac{60 \cdot k \cdot P}{\pi \cdot D \cdot n} \cdot \sin \beta; \]  

Design scheme of flat belt transmission.

a - general view; b - driven pulley; in - a tension roller

\[ F_{SD} \]

Fig. 1. Shear force FSD arising from the non-parallelism of the axis

Where \( \beta \) - angle of deviation of the axis of rotation of the driven pulley relative to the axis drive pulley.

In the future, we consider the derivation of the formula for determining the shear force for the conveyor belt.

Shear force in a conveyor belt

\[ F_{CD} = \frac{k \cdot P_T}{\eta \cdot V} \cdot \sin \beta \]  


Where, - \( \alpha \) - overall efficiency transmission drive;
\( V \) - linear speed of the conveyor belt;
\( P_T \) - power requirement on the pulley shaft;
\( P_T = \frac{\lambda \cdot T_6 \cdot \omega}{1000} \) \hspace{5pt} (5)

Where - \( \alpha \) - coefficient taking into account the nature of the transmission.

The torque is determined according to:
\[ T_6 = (S_{Hb} - S_{Cb}) \cdot \frac{D_5}{2} \] \hspace{5pt} (6)

Where - \( S_{Hb}, S_{Cb} \) - tension forces in the running and running branches of the transmission. For the existing design of the conveyor of the raw cotton picker, taking into account the initial data, we calculated the shear force of the belt at various speed modes, angles of deviation of the axes of rotation of the drums and load conditions. The calculation results are given in table 1. In the table, Figure 2 shows similar calculations for determining the shear force in a flat belt transmission according to (3). Analysis of the calculation results given in Table 1 shows that the value of the shear force varies between 60 ... 170 N. With an increase in the linear speed of the conveyor belt of the raw cotton rippers, the shear force decreases, so at \( V = 5.63 \) m/s \( F_{sd} = 80 \) H, and at \( V = 6.28 \) m/s \( F_{sd} = 67 \) H. It was revealed that the load from raw cotton does not significantly affect the change in the shear force. The main parameter affecting a significant change in the shear force of the raw cotton riot pick-up conveyor is the change in the angle of inclination of the rotational axes of the working drums, so at \( \beta = 4^\circ \), \( V = 6.28 \) m/s, \( F_{sd} = 65 \) H, and at \( \beta = 10^\circ \), \( F_{sd} = 161 \) H. Analysis of the results of calculating the shear force for a flat belt transmission showed that a change in the magnitude of the shear force not only depends on a change in the angle of inclination of the pulley axis and speed conditions, but also on the type of belt. In other cases, the difference between the pulleys shifting at the same misalignment angles does not exceed the belt types under consideration 3.5...4.0 H. The maximum value of the shear force at \( \beta = 4^\circ \) (rubberized strap), \( V_1 = 5.58 \) m/s. This is due to the fact that the coefficient of friction of a belt with pulleys is greater for rubberized relative to "Habasit". The table shows that changing the angle of non-parallelism of the axes of the pulleys significantly affects the magnitude of the shear force. It was also revealed that with an increase in the linear speed of the belt, the magnitude of the shear force decreases, the kinetic energy increases, as well as the inertia of the system.

### Table 1. The main parameters affecting the lateral gathering of the conveyor belt at idle and load conditions

| № | \( V \), m/s | \( b \), grad | \( F_{sd} \), H | \( F_{sH} \), H | \( F_{sV} \), H | \( F_{sH} \), H | \( F_{sV} \), H |
|---|---|---|---|---|---|---|
| 1 | 5.63 | 4 | 1096 | 2286 | 77 | 113 | 1131 | 2359 | 80 |
| | | 6 | | | | | | |
| | | 8 | | | | | | |
| | | 10 | | | | | | |
| 2 | 6.28 | 4 | 925 | 2151 | 65 | 96 | 96 | 130 | 133 |
| | | 6 | | | | | | |
| | | 8 | | | | | | |
| | | 10 | | | | | | |

### Table 2. Theoretical dependence of the shear force \( F_{sd} \) on the change in the angle of deviation of the axis of the driven pulley relative to the axis of the driving pulley at various high-speed driving modes

<table>
<thead>
<tr>
<th>Belt types</th>
<th>Deflection angle</th>
<th>Shear force ( F_{sd} ) (H) at</th>
</tr>
</thead>
<tbody>
<tr>
<td>Habasit</td>
<td>( V_1 )</td>
<td>( V_2 )</td>
</tr>
<tr>
<td>1</td>
<td>8</td>
<td>7</td>
</tr>
<tr>
<td>2</td>
<td>15,8</td>
<td>14</td>
</tr>
<tr>
<td>3</td>
<td>23,7</td>
<td>20,85</td>
</tr>
<tr>
<td>4</td>
<td>31,63</td>
<td>27,83</td>
</tr>
<tr>
<td>Poly Belt</td>
<td>( V_1 )</td>
<td>( V_2 )</td>
</tr>
<tr>
<td>1</td>
<td>8,62</td>
<td>7,6</td>
</tr>
<tr>
<td>2</td>
<td>17,2</td>
<td>15,13</td>
</tr>
<tr>
<td>3</td>
<td>25,8</td>
<td>22,7</td>
</tr>
<tr>
<td>4</td>
<td>34,4</td>
<td>30,3</td>
</tr>
<tr>
<td>Rubberized</td>
<td>( V_1 )</td>
<td>( V_2 )</td>
</tr>
<tr>
<td>1</td>
<td>8,4</td>
<td>7,6</td>
</tr>
<tr>
<td>2</td>
<td>18,6</td>
<td>16,34</td>
</tr>
<tr>
<td>3</td>
<td>27,8</td>
<td>24,5</td>
</tr>
<tr>
<td>4</td>
<td>37,13</td>
<td>32,7</td>
</tr>
</tbody>
</table>

From table.2 it is seen, that the magnitude of the shear force depends little on the type of belts. When the angle of deviation of the axis of rotation \( \beta \) driven belt pulley "Habasit" \( \beta = 4^\circ \), \( F_{sd} = 31,63 \) H, and for the rubberized belt \( F_{sd} = 37,13 \) H. In order to balance or eliminate the arising shear force due to the non-parallelism of the axes of rotation of the drums (pulleys), the axis of the recommended tension roller must be installed obliquely in the opposite direction from the skew angle of the axes of the drums. The formula for determining the angle of inclination of the axis of the tension roller relative to the vertical plane has the form:
\[ \alpha \geq \arccsc \left( \frac{1 + \sin^2 \beta}{f + \frac{\sin 2 \beta}{2}} \right) \] \hspace{5pt} (7)

According to the obtained condition, for calculating the angle of inclination of the axis of the tension roller, a calculation was performed for different types of belt, the results of which are given in Table 3. The calculation results established the necessary values of the angle of the axis with a curved profile of the tension roller, depending on the angle of non-parallelism of the pulleys of the conveyor of the picker of raw cotton riots. So, with \( \beta = 5^\circ \), \( \alpha = 22^\circ \); at \( \beta = 7^\circ \), \( \alpha = 24^\circ 26' \) for Habasit and accordingly \( \alpha = 21^\circ 22^\circ 30' \) for rubberized belt. It should be noted that the results obtained make it possible to select the necessary parameters of the tensioner to prevent the belt (belt) from coming off the pulleys, depending on the driving conditions and the angle of parallelism of the drums.
Table 3. Dependencies of the angle of the axis \( \alpha ^{0} \) tension roller critical angle belt slip \( \beta ^{0} \) under normal load \( T = T_{\text{nom}} \)

<table>
<thead>
<tr>
<th>Belt types</th>
<th>The coefficient of friction, ( \mu )</th>
<th>( \beta , \beta \equiv 0 \times 4X )</th>
<th>( \alpha , \alpha \equiv 0 \times 4X A )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Habasit</td>
<td>0.34</td>
<td>5 ( 22^0 20' )</td>
<td>8 ( 5^0 06' )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6 ( 23^0 42' )</td>
<td>9 ( 5^0 45' )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>7 ( 24^0 26' )</td>
<td>5 ( 22^0 06' )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>8 ( 25^0 06' )</td>
<td>6 ( 22^0 45' )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>9 ( 25^0 45' )</td>
<td></td>
</tr>
<tr>
<td>Paulie Belt</td>
<td>0.32</td>
<td>5 ( 22^0 06' )</td>
<td>8 ( 23^0 30' )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>7 ( 23^0 30' )</td>
<td>9 ( 24^0 00' )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>8 ( 24^0 00' )</td>
<td>5 ( 21^0 00' )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>9 ( 24^0 36' )</td>
<td>6 ( 21^0 36' )</td>
</tr>
<tr>
<td>Rubberized</td>
<td>0.3</td>
<td>7 ( 22^0 30' )</td>
<td>8 ( 23^0 12' )</td>
</tr>
<tr>
<td></td>
<td></td>
<td>8 ( 23^0 12' )</td>
<td>9 ( 24^0 00' )</td>
</tr>
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For each type and material of belt (flat, wedge, leather, cotton, rubber) there is a certain ratio of the payload \( F_t \) (circumferential force) and the initial tension \( S_0 \), at which the highest traction ability of the transmission is manifested at the greatest K.P.D. This ratio is called traction coefficient:

\[
\phi = \frac{F_t - S_0}{S_0} = \frac{F_t}{2S_0} = \frac{\sigma_t}{2\sigma_0} \tag{8}
\]

Relative slip

\[
\xi = \frac{V_1 - V_2}{V_1} \cdot 100\% \tag{9}
\]

where \( V_1 \) and \( V_2 \) peripheral speeds on the driving and driven pulleys.

Permissible useful voltage

\[
\left[ \sigma_1 \right]_0 = 2 - \phi R \cdot \alpha_0 = a - \alpha - \frac{\delta}{D_{\text{min}}} \tag{10}
\]

Useful permissible voltage under actual operating conditions.

\[
\left[ \sigma_1 \right] = \left[ \sigma_1 \right]_0 \cdot C_{\alpha} \cdot C_{V} \cdot C_{\mu} \cdot C_{0} \tag{11}
\]

Where \( C_{\alpha} = 1 - 0.003 \cdot \left( 180^0 - \alpha^0 \right) \) coefficient taking into account the influence of the girth angle;

\( C_{V} \) - coefficient taking into account the influence of speed, for flat belt gears \( C_{V} = 1.04 - 0.0004 \cdot V_1 \);

\( C_{\mu} \) - coefficient taking into account the influence of the operating mode;

\( C_{0} \) - coefficient taking into account the influence of the angle of inclination \( \alpha \) center line to the horizon, as well as the method of belt tension

Table 4. Dependencies of the angle of the axis \( \alpha ^{0} \) tensioner pulley critical angle of descent belt \( \beta ^{0} \) at normal load \( T = T_{\text{nom}} \)

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<td>7 ( 24^0 26' )</td>
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<td></td>
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<td></td>
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</tr>
</tbody>
</table>

We will calculate the belts for durability, while the approximate value of the resource of the belt (tape) in hours:

\[
L_i = \left( \frac{45}{\sigma_{\text{max}}} \right) \times \left( \frac{1000}{C_{V}} \right) \times \left( \frac{150}{\sigma_{\text{y}}} \right) \times \left( \frac{1}{C_{\mu}} \right) \times \left( \frac{1}{C_{0}} \right) \times \left[ \frac{L_i}{2V} \right] \tag{12}
\]

where \( \sigma_{\text{max}} \) - is determined by the equation; \( m = 5 \) - for flat belts; \( m = 8 \) - for wedges; \( \sigma_{\text{y}} \) - endurance limit, for rubberized belts without interlayers - 7, for belts - 9, for cotton belts \( \sigma_{\text{y}} = 4 ... 5 \) n / mm\(^2\); \( C_{\mu} \) - coefficient taking into account the influence of the gear ratio; for \( u = 1; 2; 4 \) respectively 1; 1.7; 1.9; \( C_{H} \) - coefficient taking into account the variability of the load: at constant load \( C_{H} = 1 \), when the load changes from 0 to the calculated value \( C_{H} = 2 \).

\[
V = \frac{V}{t} = \frac{6}{1.4321} = 4.19 \tag{13}
\]

3 RESULT

Theoretical work on the study of the characteristics of the interaction of the belt with the roller and their influence on the stability of rotation of belt drives can be divided into groups. The first group includes works in which the belt is considered rigid, and the condition of rolling the roller with the belt without slipping is displayed by the classical equations of nonholonomic relations. The authors of the works belonging to the second group also consider the belt to be absolutely rigid, but give it the property of lateral withdrawal. In this case, when compiling the equations of motion of the belt drives, only the force shifting the belt from the pulley associated with the lateral deformation of the belt is taken into account. Therefore, the lateral displacement of the belt resulting from this is called lateral abduction, and the theory that studies this phenomenon is called the abstraction hypothesis. Numerous experiments have shown that the lateral force of the \( F_{SD} \) is proportional to...
the angle of withdrawal $\beta$, i.e. $F_{sd} = K Q \beta$. Where $K_Q$ called the drag coefficient and is determined experimentally.

4 CONCLUSION
Consequently, determine the coefficient of friction between the contacting surfaces:

$$f = \frac{2 \cdot \Delta \cdot (1-\mu)}{0.8225 \cdot RKP (2-\mu)_{\text{max}}}.$$ (14)

where $[\Delta]$ the magnitude of the limiting displacement of the surface relative to another; $R_{\text{max}}$ - maximum surface roughness; $\mu$ - Poisson's ratio, for rubberized belt $\mu = 0.5$; for other cases is determined by the formula:

$$\mu = \frac{E}{2 \cdot G} - 1$$ (15)

where: $E$ - modulus of elasticity of the first kind (modulus of longitudinal elasticity); $G$ - is the elastic modulus of the second kind (shear modulus).

$\Delta_{kp}$ - critical deformation displacement from the radii of the curved profile of the tension roller, which is determined by the formula:

$$\Delta_{kp} = \sqrt{R_1^2 + R_2^2 - 2 \cdot R_1 \cdot R_2 \cdot \cos \gamma}$$ (16)

Where $R_1$ and $R_2$ are the radius of the arc from the side of the smaller and larger diameter of the tension roller; $\gamma$ - the angle formed by the horizontal and the line connecting the points $D_1$ and $D_2$ in the plane of the roller axis.

$$R_1 = \frac{L}{4 \sin \frac{\alpha_1}{2} \cos \gamma}; \quad R_2 = \frac{L}{4 \sin \frac{\alpha_2}{2} \cos \gamma}.$$ (17)

where: $L$- is the length of the tension roller; $\alpha_1$ is the angle formed by the axis of the roller and the tangent drawn to the arc from the side of the larger diameter of the roller $D_2$; $\alpha_2$ is the angle formed by the axis of the roller and the tangent drawn to the arc from the side of the smaller diameter of the roller $D_1$ [3]; Analyzing all the dependences of the shear force $F_{sd}$ that we have carried out on the change in the deflection angle of the driven pulley $\beta$ relative to the axis of the driving pulley at various high-speed driving modes. For various tapes, physicomechanical properties resulted in the dependences (Fig. 2) of the angle of inclination of the axis of the tension roller $\alpha$ on the friction coefficient $f$ between the contacting surfaces.

Fig. 2. Theoretical dependence of the angle of inclination of the axis $\alpha^\circ$ of the tension roller relative to the coefficient of friction $f$ at the angle of non-parallelism $\alpha^\circ= 3^\circ; 6^\circ; 9^\circ$.

After preliminary calculations using elastic elements of centering tensioning devices, the productivity of the raw cotton loader increases 10...15%.

5 REFERENCES