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BELT TENSION ANALYSIS AND KINEMATIC CALCULATION

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ANNOTATION

When we analyze these graphs, the number of transmissions is greater than the largest, The number of transmissions reaches the smallest value. So, from the graph it turns out that with the growth of the eccentricity of the tensile roller, the change in the number of transmissions of the stretch occurs as the interval increases in one straight line. This article covers the solution of the problem.

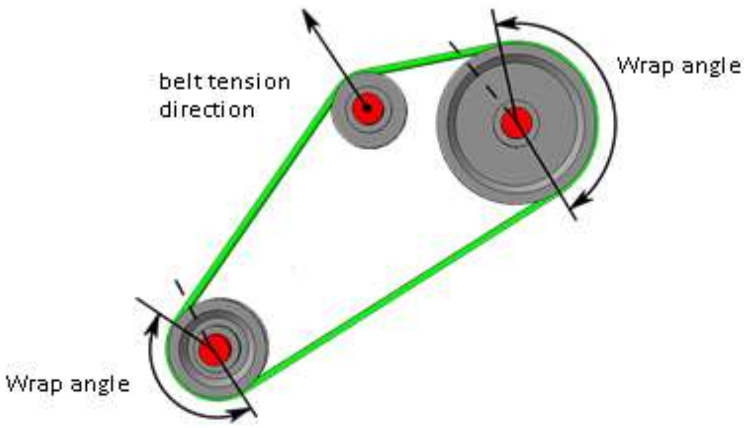
Keywords: tension roller, consistency, eccentricitet, sinusoidal, angle speed, lead pulley, graphs, dressing the number of passes, distance between arrows.

INTRODUCTION

It is known that cotton consists of an elongated belt drive, which drives the working bodies of the cleaning machine from small impurities. Studies have shown that if a variable angle of inclination of a drum with a pile is quickly processed, then the efficiency of cleaning cotton from small impurities increases. It is noteworthy that we have achieved this by laying the eccentricity of the stretching roller on the design of the working bodies of the strip extension for quick movement of the working bodies of the cotton gin machine from small impurities to a variable angle. The kinematic diagram of this transmission is presented, which consists of the following working parts.

The key element in this system is the idler pulley. It is also called an idle pulley. Idler rollers are of two types, direct and reverse. The figure below shows an example of a straight idler pulley.

This type of roller is installed inside the belt. It reduces the angle of the pulley wrap. Because of this, it must be positioned close to a large pulley.



(Fig. 1)

The profile of the working surface of the pulley must match the profile of the belt (Fig. 1).

The reverse idler roller works from the back of the belt. It increases the wrap angle on the pulleys. Typically, this roller arrangement is used for drives with a large gear ratio and small center distance. It is located on the driven branch of the belt, near the small pulley. The working surface of such a roller is flat (like a regular cylinder).

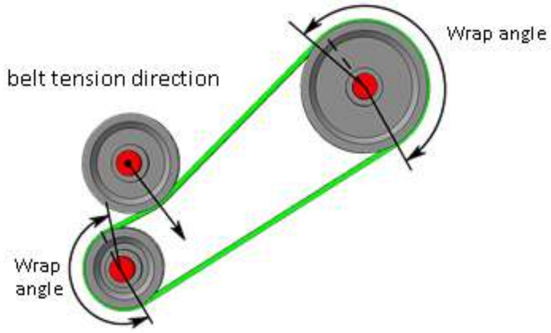


Fig. 2.

For each type of drive belt, there are standards that determine the minimum diameter of the idler pulley. Moreover, the dimensions of the forward and reverse rollers for the same type of belt will be different (Fig. 2.). The type of the projected belt drive is provided for by the terms of reference. Select the belt section according to the nomogram (Fig. 3) depending on the power transmitted by the drive pulley, P_1 = P/nom, kW (where P/nom is the rated power of the engine) and its speed n_1 = n/nom, rpm (where n/nom is the rated frequency rotation of the motor shaft)

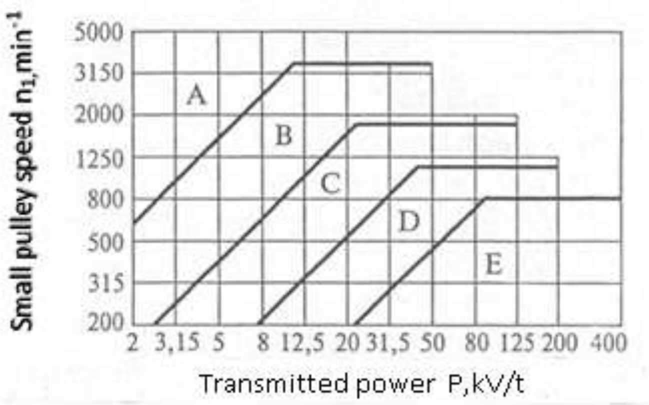


Fig. 3. Graph for selecting a section of a V-belt with a normal section.

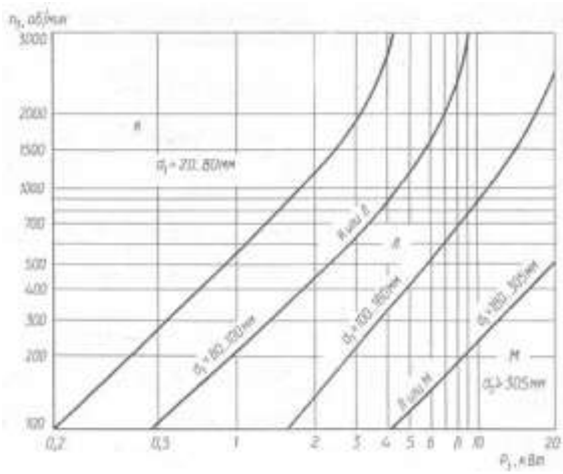


Fig. 4. Graph for selection of poly V-belt section.

It can be seen that the drive pulley transmits a variable angular velocity motion to the essence of the pulley to the drive pulley. We use well-known theorems and laws to find the laws of motion of working bodies and transmission bodies.

As you know, the number of passes with the tape is equal to $u_{12} = \frac{\omega_1}{\omega_2} = \frac{r_2}{r_1}$(1)

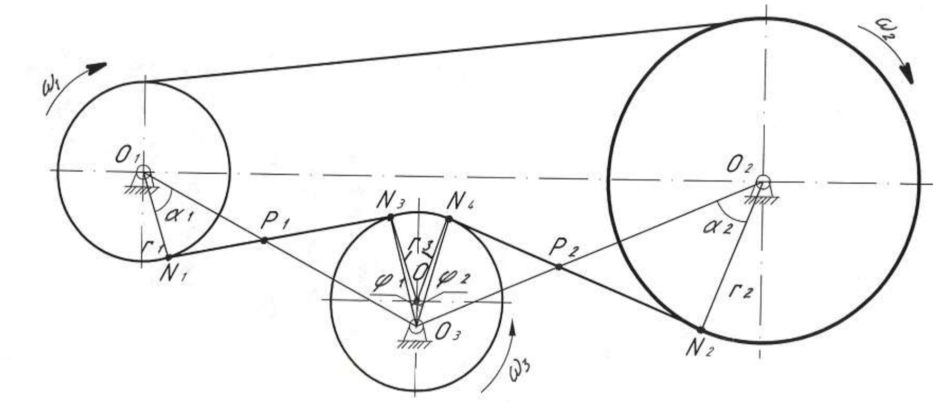


Figure 5. Kinematic diagram of the transmission.

1-drive pulley, 2-driven pulley, 3-belt clad in pulleys, 4-roller eccentric stet tensioner.

(1) we write the number of cited transmissions using the equality

$$u_{12} = \frac{\omega_1}{\omega_2} = \frac{u_{13}}{u_{23}} \quad (2)$$

the ratio of the eccentricity of the tension roller transmission with the drive pulley can be written as follows,

$u_{13} = \frac{\omega_1}{\omega_2}$ (3)

the ratio of the eccentricity of the passes of the tensile roller with the driving pulley can be written as follows,

$u_{23} = \frac{\varpi_2}{\omega_3}$ (3)

Where, ω_1 is the angular velocity of the driving pulley,

ϖ_2- angular speed of the driven pulley,

ω_3-angular speed of the idler roller

u_13-gear ratio of the drive pulley and idler pulley

u_23-gear ratio of the driven pulley and the transmission of the idler roller

From the presented kinematic diagrams, we write down the laws of rotational motion of the working parts of the strip extension,

$$\frac{\omega_1}{\varpi_2} = \frac{O_3P_1}{O_1P_1} \qquad \text{и} \quad \frac{\varpi_2}{\omega_3} = \frac{O_3P_2}{O_2P_2} \quad (4)$$

$$\text{where,} \quad O_1P_1 = \frac{r_1}{\cos\alpha_1}; \text{ and } O_3P_1 = A_1 - O_1P_1 (5)$$

$$O_2P_2 = \frac{r_2}{\cos\alpha_2}; \qquad \text{и } O_3P_2 = A_2 - O_2P_2 (6)$$

Here, r_1 is the radius of the drive pulley;

r_2- radius of the driven pulley;

A_1 is the distance between the arrows of the drive pulley and the tension roller;

A_2 is the distance between the drive pulley and the arrows of the tension roller;

α_1 is the angle between r_1 and A_1;

α_2 is the angle between r_2 and A_2.

If the eccentricity of the idler pulley is $e = 0$

$$r_1 + r_3 = A_1 \cdot cos\,\alpha_1 \qquad и \;\; r_2 + r_3 = A_2 \cdot cos\,\alpha_2 \; (7)$$

It is known from the kinematic diagrams that since the eccentricity of the stretching roller is equal to e ≠ 0, then we define the poles of the stretching roller r_3^I and r_3^{II}..I and r II. from this value of the radii of the tensile roller poles r_r_3^I and r_3^{II}II can be written as follows,

$$r_3^I = e \cdot cos\,\varphi_1 + \sqrt{r_3^2 - e^2 \cdot sin^2\,\varphi_1} \qquad (8)$$

$$r_3^{II} = e \cdot cos\,\varphi_2 + \sqrt{r_3^2 - e^2 \cdot sin^2\,\varphi_2} \qquad (9)$$

here r_3 is the real radius of the idler roller;

e - eccentricity;

φ_1-angle between and r_3^I;

φ_2-angle between e and r_3^{II}

Now, based on the above, we write

$$A_1 \cdot cos\,\alpha_1 = r_1 + e \cdot cos\,\varphi_1 + \sqrt{r_3^2 - e^2 \cdot sin^2\,\varphi_1} \qquad (10)$$

$$A_2 \cdot cos\,\alpha_2 = r_2 + e \cdot cos\,\varphi_2 + \sqrt{r_3^2 - e^2 \cdot sin^2\,\varphi_2} \qquad (11)$$

In equations (10) and (11) the angle between r_1 and $A_1$is α_1, and the angle between $r_2$and $A_2$is α_2

$$cos\,\alpha_1 = \frac{e \cdot cos\,\varphi_1 + \sqrt{r_3^2 - e^2 \cdot sin^2\,\varphi_1} + r_1}{A_1} \qquad (12)$$

$$cos\,\alpha_2 = \frac{e \cdot cos\,\varphi_2 + \sqrt{r_3^2 - e^2 \cdot sin^2\,\varphi_2} + r_2}{A_2} \qquad (13)$$

If we put the values (10), (11) of the equalities $A_1 \cdot$because α_1 and $A_2 \cdot$ cos $\alpha_2$in (5) and (6) of the equation, we will have the following expression

$$\frac{\omega_1}{\omega_3} = \frac{e \cdot cos\,\varphi_1 + \sqrt{r_3^2 - e^2 \cdot sin^2\,\varphi_1}}{r_1} \qquad (14)$$

$$\frac{\varpi_2}{\omega_3} = \frac{e \cdot cos\,\varphi_2 + \sqrt{r_3^2 - e^2 \cdot sin^2\,\varphi_2}}{r_2} \qquad (15)$$

In order to find the number of gears of this belt transmission, characterized by a variable angle of speed, we set (2) equality (14), (15) equality $\frac{\omega_1}{\omega_2}$ and $\frac{\omega_2}{\omega_3}$

$$\overline{u}_{12} = \frac{r_2}{r_1} \cdot \frac{e \cdot cos\,\varphi_1 + \sqrt{r_3^2 - e^2 \cdot sin^2\,\varphi_1}}{e \cdot cos\,\varphi_2 + \sqrt{r_3^2 - e^2 \cdot sin^2\,\varphi_2}} \qquad (16)$$

In this case, the angular velocity of the leaderboard can be written as follows

$$\varpi_2 = \frac{\omega_1 \cdot r_1}{r_2} \cdot \frac{e \cdot cos\,\varphi_2 + \sqrt{r_3^2 - e^2 \cdot sin^2\,\varphi_2}}{e \cdot cos\,\varphi_1 + \sqrt{r_3^2 - e^2 \cdot sin^2\,\varphi_1}} \qquad (17)$$

If this variable angle can be deduced from the number of gears of a variable speed transmission, then we get the following equation:

$$\bar{u}_2^{I} = \frac{(-r_2 \cdot e \cdot \sin\varphi_1 - \dfrac{r_2 \cdot e^2 \cdot \sin\varphi_1 \cdot \cos\varphi_1}{\sqrt{r_3^2 - e^2 \cdot \sin^2\varphi_1}}) \cdot (r_1 \cdot e \cdot \cos\varphi_2 + r_1 \cdot \sqrt{r_3^2 - e^2 \cdot \sin^2\varphi_2}) +}{}$$

$$\frac{+(r_1 \cdot e \cdot \sin\varphi_2 + \dfrac{r_1 \cdot e^2 \cdot \sin\varphi_2 \cdot \cos\varphi_2}{\sqrt{r_3^2 - e^2 \cdot \sin^2\varphi_2}}) \cdot (-r_2 \cdot e \cdot \cos\varphi_1 + r_2 \cdot \sqrt{r_3^2 - e^2 \cdot \sin^2\varphi_2})}{(r_1 \cdot e \cdot \cos\varphi_2 + r_1 \cdot \sqrt{r_3^2 - e^2 \cdot \sin^2\varphi_2})^2} \quad (18)$$

Angular acceleration is the product of angular velocity

$$\varepsilon = \frac{d\varpi}{dt} = \frac{\left(-r_1 \cdot \omega_1 \cdot e \cdot \sin\phi_2 - \frac{r_1 \cdot \omega_1 \cdot e^2 \cdot \sin\varphi_2 \cdot \cos\varphi_2}{\sqrt{r_3^2 - e^2 \cdot \sin^2\varphi_2}}\right) \cdot \left(r_2 \cdot e \cdot \cos\phi_1 + r_2 \cdot \sqrt{r_3^2 - e^2 \cdot \sin^2\varphi_1}\right) -}{(r_2 \cdot e \cdot \cos\phi_1 + r_2 \cdot \sqrt{r_3^2 - e^2 \cdot \sin^2\phi_1})^2}$$

$$\frac{-(-r_2 \cdot e \cdot \sin\phi_1 - \frac{r_2 \cdot e^2 \cdot \sin\phi_1 \cdot \cos\phi_1}{\sqrt{r_3^2 - e^2 \cdot \sin^2\phi_1}}) \cdot (r_1 \cdot \omega_1 \cdot e \cdot \cos\phi_2 + r_2 \cdot \sqrt{r_3^2 - e^2 \cdot \sin^2\phi_2})}{(r_2 \cdot e \cdot \cos\phi_1 + r_2 \cdot \sqrt{r_3^2 - e^2 \cdot \sin^2\phi_1})^2} \quad (19)$$

The origin of (16), (17), (18) and (19) by placing numerical values instead of parameters in the expressions of the computer in the Excel program, we obtained graphs of changes in the number of gears, angular speed, changes in the number of gears and angular accelerations of the transfer.

The limited number of standard sizes of standard V-belts made it possible to determine the permissible load for each standard size of the belt, and the transmission calculation was reduced to the selection of the type and number of belts according to tables or graphs (see Appendix 2 to GOST 1284-68).

The type and number of belts are selected and calculated using the formula (5.28) and the graphs in Fig.

$$N = N_0 \cdot K_\alpha \cdot K_{\text{HZ}} \ (5.26)$$

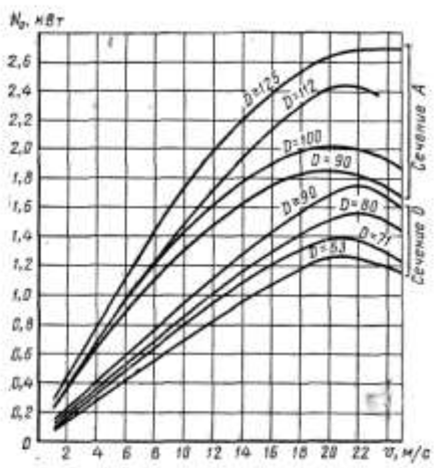


Fig. 6

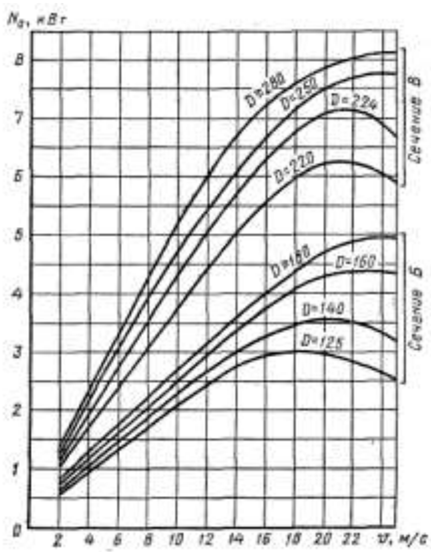


Fig. 7

Where, N - transmission power; N_0 - power transmitted by one belt in a typical transmission (α= 1800, uniform load) - according to the graphs in Fig.5.19, 5.20; Kα- coefficient of the angle of wrap; Kн - load mode factor; **Z**- is the number of belts.

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