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THEORETICAL JUSTIFICATION OF BELT TRANSMISSION PARAMETERS

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Abstract: *This article provides information on the types of belt transmissions, advantages, disadvantages, areas of use, and theoretical calculation of belt transmission parameters.*

Keywords: *mechanical, transmission, geometric, kinematic, parameter, belt, force, length, diameter.*

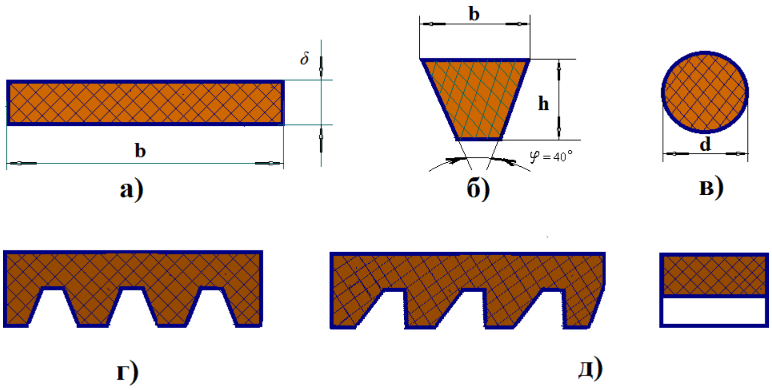
Introduction

The reason for the fact that the active working bodies produced in the agricultural sector do not work at the required level, fail before the deadline, do not transmit enough power during operation, that is, the transmission quality is low (it is not properly fixed, not stretched, the material for the transmission is incorrectly selected, etc.) will be.
Belt transmissions are mainly used in active working bodies in the agricultural sector. Different types of belt transmissions are used in active working bodies, depending on the working conditions and the work they perform. This, in turn, prevents various defects or inconveniences. In order not to cause many inconveniences, it is necessary to correctly choose the type of belt transmission and theoretically justify its parameters.

Research method. A belt transmission is a mechanism consisting of a leading and driven pulley and a belt worn on them, and which transmits movement due to friction (Fig. 1). The load is transmitted by the frictional forces created between the pulleys and the belt as a result of the tension of the belt.

Belt transmissions are divided into the following types: According to the shape of the cross section of the belt: a) flat; b) pone-like; c) circular (round); g) semi-pone-shaped; d) gear (Figure 1).

The advantages of belt transmission include: the ability to transmit movement over long distances (15 m and more); works smoothly, evenly and without noise; ability to work at high speeds due to the elasticity of the belt; protection of mechanisms from sudden changes in the load due to the elasticity of the belt; protection of mechanisms from excessive loads due to belt slippage; simplicity of construction; relative ease and simplicity of use (no lubrication required, ease of assembly and adjustment).



Picture 1. Types of tapes.

The disadvantages of the belt transmission include the following: increased external dimensions (the diameter of the pulleys is five times larger than the diameter of the gear wheels under the same conditions); non-constant transmission ratio due to sliding of the belt on the pulley under the influence of load; due to the high initial tension of the belt, the load on the shafts and supports is high (the load on the shafts is 2...3 t times greater than on gears); low belt continuity (within 1000...5000 hours).

The power of the belt drive usually does not exceed 50 kW. In modern engineering, the use of belt drives is more widespread. New types of flat belts (made of plastic) are finding their place in high-speed transmissions. Round belts are used in small capacities, for example, in tools, tape recorders, household sewing machines.

Research results. The theoretical basis of calculation is common for all types of tapes.
The workability of belt transmissions and their calculation criteria include the traction capacity determined by the friction force between the belt and the pulley; includes belt life limited by fatigue wear under normal service conditions.
The basic calculation of belt drives is their towing capacity. The life of the belt is taken into account by choosing the main parameters of the transmission in accordance with the recommendations developed as a result of experience.
Kinematic parameters. Rotational speeds of pulleys:

$$V_1 = (\pi \cdot d_1 \cdot n_1) / 60 \ \ m/c; \qquad V_2 = (\pi \cdot d_2 \cdot n_2)/60 \ \ m/c. \qquad (1)$$

Given the presence of belt slip on the pulley V2 □ V1 in the transmission. So,

$$V_2 = (1 - □)\ V_1 , \qquad (2)$$

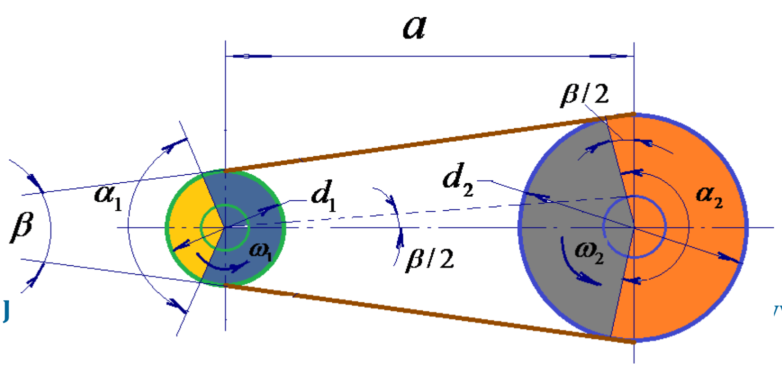
where: □-sliding coefficient (□=0.01 in rubber band; □ =(0.01 □ 0.02) in rubber band).

Transmission ratio:

$$U = n_1 / n_2 = V_1 \cdot d_2 / V_2 \cdot d_1 = d_2 / (d_1 (1 - □)) \approx d_2 / d_1. \qquad (3)$$

Geometric dimensions of the belt drive (Fig. 2): d_1, d_2 - diameters of the leading and driven pulleys; *a* - distance between axes; □□is the angle between the webs of the tape; a_1, a_2 are the coverage angles of the belt on the small and large pulley. In geometric calculations, d_1, d_2 and α are usually known, and the coverage angle α and the length of the strip *l* are determined.

2- picture. Geometric dimensions of the belt drive.



The values of α and *l* are approximately determined due to tensioning (pulling) and hanging of the tape:

$$\alpha = 180 - \beta; \qquad Sin\ (\beta /2) = (d_2 - d_1) / 2\ \alpha. \tag{4}$$

In practice, assuming that the value of b/2 is greater than 15º, the value of the sine can be equal to its argument:
In that case,

$$\beta = [(d_2 - d_1) / a]\ rad \approx [(d_2 - d_1) / a]\ 57^0 \tag{5}$$

And so,

$$\alpha_1 = 180^0 - [57^0(d_2 - d_1) / a] \tag{6}$$

If *l* is known, the distance between the axes can be determined as follows:

$$a = [2a\text{-}\pi(d_2 + d_1) + \sqrt{((2a\text{-}\ \pi(d_2 + d_1))^2 - 8(d_2 - d_1)^2)}] / 8 \tag{7}$$

In the extension, it should be possible to reduce the distance between the axles by 0.01 *l* for easy installation of the belt on the pulleys, and to increase the distance between the axles by 0.025 *l* to ensure the necessary tension on the belts.
Forces in strip networks. In order to solve the problem of the forces in the belt networks and the connections between them, we will compare the transmissions without load $T_1 = 0$ and with load $T_1 > 0$ (Fig. 3, a, b). In the picture, F_0 is the tension force in the webs of the belt without loading; F_1, F_2 is the generated tension force in the leading and driven sectors of the loaded transmission.
Rotational force in extension:

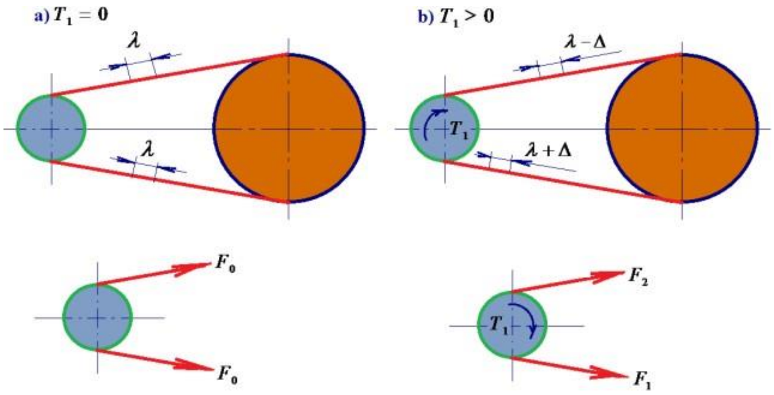
$$F_t = 2\ T\ / d_1.$$

Equilibrium condition for the drive pulley $T_1 = 0{,}5 d_1\ (F_1 - F_2)$ or

$$F_1 - F_2 = F_t \tag{8}$$

will be.
The interaction of forces F_0, F_1 and F_2 can be determined based on the following. The geometric length of the belt is independent of the load value and remains the same whether the extension is loaded or unloaded. Based on this, it can be concluded that the tension (fatigue) of the leading network is compensated by the corresponding reduction of the leading network (Fig. 3). In this case, the stretching of the leading network is balanced by the shortening of the leading network, that is:

$$F_1 = F_0 + \Delta F, \quad F_2 = F_0 - \Delta F \ \text{ yoki } \ F_1 + F_2 = 2 F_0, \tag{9}$$



3 - picture. Forces affecting extension.

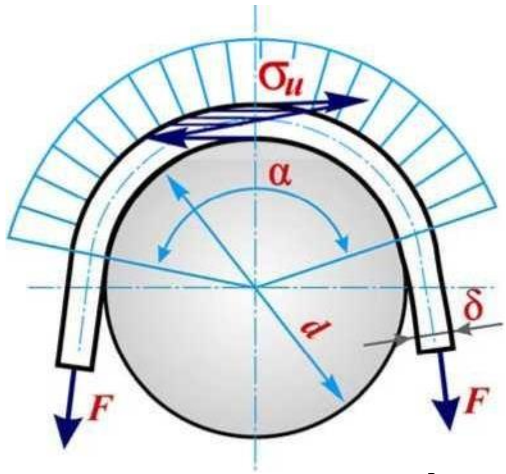
From equations (8) and (9):

$$F_1 = F_0 + F_t / 2; \quad F_2 = F_0 - F_t / 2\ . \tag{10}$$

As a result, two systems of equations with three unknowns: F_0, F_1 and F_2 are formed. Although these equations determine the tension of the leading and driven chains from the load F_t, they cannot reveal the ability to transmit this load or the traction capacity of the transmission, which is related to the magnitude of the friction forces between the belt and the pulley. L.Euler was able to analyze this relationship in his scientific research.

Tensions in the belt. The largest tension in the webs of the belt - σ_{max} is created in the leading web, which is the tension created by the force in the leading web of the belt - σ_1, the tension created by the centrifugal force - σ_V and the bending created in the place where the belt covers the pulley voltage is formed from the sum of segs and is determined as follows:

4 – picture. Bending stresses in the strip.



$$\sigma_1 = F_1 / A; \quad \sigma_V = F_V / A = \rho V^2; \tag{11}$$

Taking into account the formula (9), the voltage σ_1 can be expressed as follows:

$$\sigma_1 = F_0 / A + F / 2A = \sigma_0 + \sigma_t / 2; \tag{12}$$

where: σ_t = F_t / A- useful voltage, A= b · d - cross section of the strip; σ_0 is the stress resulting from the initial tension.

Bending stress appears in the part of the belt covering the pulley (Fig. 4). Based on Hooke's law

$$\sigma_{eg} = \varepsilon \cdot E, \tag{13}$$

here: e - the relative elongation of the surface fibers of the tape; E is the modulus of elasticity.
To find ε, we determine by observing the part of the arc bounded by the angle dφ.
If $\varepsilon = \delta/d$, the bending stress is equal to:

$$\sigma_{eg} = E\ \delta/d\ , \tag{14}$$

From the formula (14), it can be seen that the main factor determining the magnitude of the corrosive stresses is the ratio of the belt thickness to the pulley diameter.

Conclusion. According to the given data, belt drives are mainly subjected to bending stress. Therefore, belt drives can be subjected to bending force. When calculating the belt transmissions under the influence of bending force, their main parameters are the pulley diameters, the distance between the axles, and the friction force between the belt and the pulley. At the same time, the allowable stress of the metal is also one of the important factors in calculating the bending stress.

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