SOI: 1.1/TAS DOI: 10.15863/TAS
International Scientific Journal
Theoretical & Applied Science

p-ISSN: 2308-4944 (print) **e-ISSN:** 2409-0085 (online)

Year: 2023 **Issue:** 06 **Volume:** 122

Published: 09.06.2023 **http://T-Science.org**

Issue Article

Jasur Kholmamatovich Beknazarov
Navoi State Mining and Technological University
Docent of the department Mechanical Engineering, Navoi, Uzbekistan

Shokhrukh Khurshid ugli Ashurov
Navoi State Mining and Technological University
Master student of the Department of Mechanical Engineering Technology, Navoi, Uzbekistan

Jakhongir Shavkat ugli Oblokulov
Navoi state mining and technological university
student of the department Mechanical engineering

CALCULATION OF TOOTHED BELT TRANSMISSIONS IN TECHNOLOGICAL MACHINES

Abstract: *In recent years, toothed belt transmission has been increasingly used in light industry. Progress in the field of such machines and devices as embroidery and sewing machines, automated cutting systems, etc. not conceivable without a wide, scientifically substantiated use of a toothed belt drive.*

Preliminary studies have shown that the use of a toothed belt drive leads to a change in the dynamic characteristics of technological equipment, which entails the appearance of factors that have a negative impact on the working conditions of working and maintenance personnel, as well as a violation of sanitary and technical standards established for the relevant types of equipment.

Key words: *toothed belt transmission, rotary feed mechanism, design, kinematic and power parameters.*

Language: *English*

Citation: Beknazarov, J. Kh., Ashurov, Sh. Kh., & Oblokulov, J. Sh. (2023). Calculation of toothed belt transmissions in technological machines. *ISJ Theoretical & Applied Science, 06 (122),* 69-74.

Soi: http://s-o-i.org/1.1/TAS-06-122-14 ***Doi***: crossref https://dx.doi.org/10.15863/TAS.2023.06.122.14

Scopus ASCC: *2200.*

Introduction

In order to determine these factors, a number of scientific studies were carried out, the results of which are the current standards. Calculation methods, rules for installation and operation of a toothed belt drive, regulated by these standards, are published in the work and represent the following provisions.

The main parameter of the toothed belt is the modulus m, the values of which are selected according to the standard depending on the transmitted power and the speed of the input shaft. For example, for 1022M class sewing machines, the maximum speed of the main shaft, which is 4500 rpm, the ratio of the toothed belt module and the power transmitted to it is as follows:

Table 1.

P, kW	0,05…0,18	0,27…1,5	2,2…5,5	7,0…17
m, mm	2;3	3;4	4;5	5;7

The main parameters of toothed belts are indicated in (Fig. 1) and are given in Table 1. The width of the toothed belt is also selected depending on the module, and the estimated length of the belt is determined by the ratio:

$$L = m \cdot z_p$$

where: z_p - number of belt teeth.

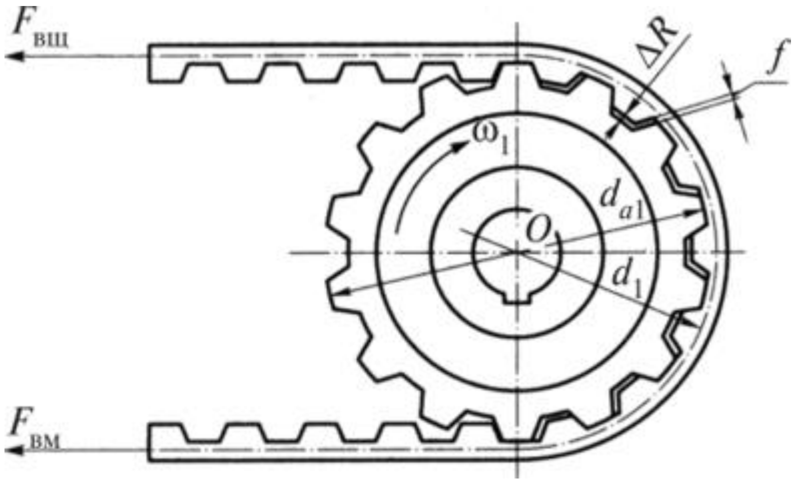

Figure 1. Toothed belt drive.

Toothed pulleys are made from cast iron, steel, light alloys, plastics (depending on peripheral speed). To prevent the belt from running off, the smaller pulley is provided with two or one flanges. With a gear ratio i > 3, both pulleys are made with flanges. The dimensions of the pulley depend on the modulus and the number of teeth. The number of teeth of the smaller pulley is taken within 12 ... 28 depending on the speed and module, and the number of teeth of the larger pulley is determined by the expression:

$$z_2 = i \cdot z_1$$

Table 2. The main parameters of the toothed belt

Parameters	Belt module m, mm					
1	2	3	4	5	6	7
Belt pitch $p = \pi \cdot m$, mm	6.28	9.42	12.57	15.71	21.99	31.42
Total belt thickness H, mm	3.00	4.00	5.00	6.50	11.00	15.00
Tooth height h, mm	1,50	2.00	2.50	3.50	6.00	9.00
Smallest tooth thickness S	1.80	3.20	4.40	5.00	8.00	12.00
Tooth profile angle 2 γ grad	50	40	40	40	40	40
Rope diameter, mm	0.36	0.36	0.36	0.65	0.65	0.65
belt width	8;10;	12,5;	20;25	25;32	50;63	50;63
b, mm	12.5	16;20	32;40	40; 50	80	80
Distance from the cable axis to the belt cavity 8, mm	0.6	0.6	0.8	0.8	0.8	0.8

The diameters of the pitch circles and the outer diameters of the pulleys (Fig. 1) are found from the following relationships:

$$d_1 = m \cdot z_1$$
$$d_2 = m \cdot z_2$$

and the outer diameters of the pulleys are determined based on the expressions:

$$d_1 = m \cdot z_1 + 2 \cdot \delta$$
$$d_2 = m \cdot z_2 - 2 \cdot \delta$$

where δ = 0.6 mm with a cable diameter of 0.36 mm and δ = 1.3 at cable diameter 0.65 mm. The width of the toothed pulley is determined from the expression:

$$B = b + m$$

The depression angle is: (50 + 2) degrees, at m = 2 mm and (40 + 2) degrees, at m>2 mm. The pitch of the teeth on the outer diameter is found from the expression:

$$p = \pi \cdot \frac{d_a}{z}$$

Toothed belt drives fail due to wear and shearing of the belt teeth. Therefore, the strength reliability is evaluated in the form of limiting the specific (referred to the width of the belt) circumferential force p_t on the belt:

$$p_t \leq [p_t]$$

where p_t – permissible specific circumferential force.

The calculated value of the specific circumferential force, taking into account the forces of inertia:

$$p_t = \frac{F_t}{b} + q \cdot v^2$$

where q is the mass of 1 m of a belt 1 cm wide;

Table 3.

m, mm	2	3	4	5	7	10
$q \cdot 10^2\ kg/(m \cdot sm)$	0.3	0.4	0.6	0.7	0.8	1.1

v ·- belt speed, m/s;

F_t - circumferential force transmitted by the belt,

$$F_t = C_p \frac{P}{v}$$

where P is the transmitted power, W;

P is the coefficient of dynamism (see Table 1.2.);

v·-belt speed, m/s;

For example, for sewing machines 1022M class, the coefficient $C_p = 1{,}1$.

Permissible specific circumferential force is determined by:

$$|P_t| = |P_0| \cdot C_i \cdot C_н \cdot C_т$$

where $|P_0|$ - permissible specific circumferential force (see table. 1.1.);

C_i - gear ratio, entered only for overdrive, with i>1 ratio $C_i = 1$

Table 4.

i	1...0.8	0.8...0.6	0.6...0.4	0.4...0.3	less 0.3
C_i	1	0.95	0.9	0.85	0.8

For 1022M class sewing machine coefficient $C_i = 1$.

$C_н$ - coefficient taking into account the use of a tension or guide roller,

$C_н = 0{,}9$ with one roller and $C_н = 0{,}8$ with two rollers;

$C_т$- coefficient taking into account the uneven distribution of the load between the coils of the cable.

Coefficient value $C_т$ accept depending on the width b of the belt:

Table 5.

b, mm	8	10	12.5	16	20	25	40	63	100
$C_т$	0.67	0.77	0.83	0.91	0.94	1	1.04	1.09	1.2

For 1022M class sewing machine coefficient $C_т = 0{,}91$.

Table 6. Dynamic factor

The nature of the load	Machine type	C_p
Calm. Starting, load up to 120% of normal	Electric generators, centrifugal pumps and compressors machines with intermittent cutting process fans; belt conveyors	1

Moderate load fluctuations. Starting load up to 150% of normal	Piston pumps and compressors with three or more cylinders; machine tools and automatic machines; plate conveyors	1.1
Significant load fluctuations. Starting load up to 200% of normal	Reversible drives; piston pumps and compressors with one and two cylinders; planing and slotting machines; screw and scraper conveyors; elevators; eccentric and screw presses with heavy flywheels	1.25
Uneven load. Starting load up to 300% of normal	excavators, dredges; eccentric and screw presses with light flywheels	1.5

To ensure the engagement of the belt with the pulley, a preload F_0 is assigned, which is taken depending on the belt module m and its width b:

Table 7.

m, mm	2	3	4	6	7	10
F_0/b, N/m	0.4	0.6	0.8	1.0	1.4	2.0

For 1022M class sewing machine initial tension $F_0 = 9{,}6\ H$. The forces acting on the transmission shafts are determined by the relation:

$$F_r = (1 \dots 1{,}2)F_t$$

In this work, it is noted that the pretension in such gears is created using spring-loaded rollers or due to the mobility of the support of one of the toothed pulleys. However, in sewing machines, the distance between the supports of the main and camshafts always remains unchanged, and the installation of a spring-loaded roller in the conditions of using synthetic toothed belts will accelerate the relaxation processes in them. In addition, the design features of sewing machines are such that the replacement of a worn toothed belt is associated with labor-intensive repair and restoration work, as a result of which there is a violation of the tolerances for the installation of individual machine mechanisms, which leads to an increase in its noise and vibration. toothed belt transmission, made in accordance with the instructions of the standard, in relation to sewing machines did not meet the requirements for them, and the use of known tension systems to stabilize the pretension of the toothed belt in sewing machines cannot be recommended. Therefore, additional studies were carried out on the toothed belt drive used in light industry sewing machines.

The paper presents the results of studies of the physical and mechanical characteristics of toothed belts, the geometry of the toothed-belt gear, provides a description of the most well-known methods for the production of toothed belts, and also carried out calculations to determine the natural vibration frequencies of toothed belts used in sewing machines of the 97th class. The main results of the work are the dependence of the magnitude of the longitudinal deformation of the toothed belt on the load when it is stretched, the optimal profile of the belt tooth, theoretical and experimental studies of the natural frequencies of the toothed belt as part of industrial sewing equipment.

In the process of research, a toothed belt branch was considered as a string with a uniformly distributed mass. Based on the application of the differential equation of free transverse vibrations of a string with a uniformly distributed mass, the author obtained expressions for the frequencies of natural vibrations of a toothed belt branch:

for the first tone $p_1 = \frac{\pi}{A}\sqrt{\frac{S_0}{\mu}}$

for the second tone $p_2 = \frac{2\pi}{A}\sqrt{\frac{S_0}{\mu}}$

for the third tone $p_3 = \frac{3\pi}{A}\sqrt{\frac{S_0}{\mu}}$

where A is the center-to-center distance;

S_0 - tension;

μ - mass per unit length of the belt.

For a branch of the drive belt of a 97-class sewing machine, the value of the center distance is A=22,4 cm, and the value of the mass of a unit length of the belt μ = 3,93 10^{-4} kg.

The values of the obtained frequencies of transverse oscillations of the belt branch, depending on the tension, are given in tab. 3.:

Table 3. Frequency of transverse oscillations of the toothed belt branch

S_0, N	5	8	15	24	30	50	70	100
p1, Hz	24.95	31.56	43.22	54.67	1.13	78.91	93.37	111.60
p2, Hz	49.9	63.12	86.94	109.3	122.3	157.8	186.7	223.20
P3, Hz	74.95	94.68	129.7	164.0	183.4	236.7	280.1	334.80

For the experimental determination of these frequencies, two types of sensors were used: a wire strain gauge glued to the belt, and an inductive sensor that recorded a change in the gap between the metal cord of the toothed belt and the sensor itself.

The inductive sensor is a horseshoe-shaped magnet with a winding. When the magnetic field of the sensor is crossed by the metal cables of the belt cord, EMF is induced in the windings, the value of which is determined by known dependencies. Both sensors were connected to a loop oscilloscope. According to the obtained oscillograms, one can judge the oscillation frequency, and the amplitude value on the oscillograms characterizes the amplitude of the belt oscillation speed, but not the amplitude of its movements.

An analysis of the oscillation frequencies obtained in the experiment allowed the author to establish that the sources of disturbances in the gear belt transmission of a 97 class sewing machine are:

1. The error introduced into the pitch of the toothed belt during its manufacture, and the collisions at the entrance to the meshing of the same tooth of the belt with the same tooth of the pulley.

2. Pulley eccentricity, mass imbalance and sewing machine body vibration.

3. Collisions between the teeth of the belt and the teeth of the pulley, when each successive tooth engages.

In this work, the issue of weakening the pre-tensioning of toothed belts during their operation was not considered.

However, observations of the operation of the toothed belt drive in sewing machines of equal classes showed that over time, the pretensioning of the toothed belts can decrease to values that are dangerous for the operation of the sewing machine, and it needs automatic stabilization of the belt tension during its operation. In addition, in this work, no studies were carried out for timing belts made on the basis of synthetic cord.

Therefore, the results obtained in this work cannot be directly used to develop a system for automatically stabilizing the tension of toothed belts during their operation as part of existing sewing machines.

In conclusion, the calculation of toothed belt transmissions plays a crucial role in the design and operation of technological machines. Toothed belt transmissions are widely used in various industries to transmit power and motion between different machine components.

The calculation process involves several important factors, including the selection of the appropriate belt type, determining the required belt length, calculating the speed ratio, and assessing the load capacity and torque requirements. These calculations are essential for ensuring the efficient and reliable operation of the toothed belt transmission system.

When selecting the belt type, factors such as the application requirements, environmental conditions, and the load to be transmitted must be considered. Different belt materials and designs have varying load capacities, flexibility, and resistance to wear and tear, so choosing the right belt is crucial.

The calculation of the belt length is based on the distance between the pulleys and the desired center distance. It is important to account for factors such as tensioning devices and the potential for belt elongation over time. Proper tensioning is essential for maintaining the desired power transmission efficiency and preventing belt slippage or premature wear.

The speed ratio calculation involves determining the required rotational speed of the driven pulley based on the input speed and the desired output speed. This calculation helps in selecting the appropriate pulley sizes to achieve the desired speed ratio and power transmission.

Load capacity calculations consider the maximum torque and load that the belt transmission system will experience. This involves assessing factors such as the weight of the transmitted load, shock loads, and the desired safety factor. It is important to ensure that the selected belt can handle the anticipated loads without exceeding its maximum load capacity.

In summary, the calculation of toothed belt transmissions in technological machines is a complex process that involves considering various factors such as belt selection, length calculation, speed ratio determination, and load capacity assessment. Proper calculations and considerations are essential for ensuring the efficient and reliable operation of the toothed belt transmission system, ultimately contributing to the overall performance and longevity of the technological machine.

References:

1. Egamberdiev, I. P., et al. (2020). Research of vibration processes of bearing units of mining equipment. *International Journal of Advanced Trends in Computer Science and Engineering*, 2020, T. 9, №. 5, pp. 7789-7793.
2. Akhmedov, K. I., et al. (2020). Influence of integrated machining on the cutting capacity of fast-cutting steel R6M5. *Theoretical & Applied Science*, 2020, №. 9, pp. 322-327.
3. Ahmedov, H. I., et al. (2017). Analiz drobil`nogo agregata kak linejnaja staticheskaja sistema. *Luchshaja studencheskaja stat`ja* 2017, pp. 73-75.
4. Yaxshiyev, S.N., Ashurov, Kh. K., & Mamadiyarov, A.J. (n.d.). Dynamics of Spindle Assembly of Metal-Cutting Machine. *International Journal of Engineering and Advanced Technology (IJEAT)*, pp. 2249-8958.
5. Karimov, N. K., et al. (2020). Povyshenie udarnoj vjazkosti konstrukcionnyh stalej termociklicheskoj obrabotkoj. *Estestvennye i tehnicheskie nauki: problemy transdisciplinarnogo sinteza*, 2020, pp. 40-43.
6. Ibragimov, A. A., et al. (2017). Interval`no-analiticheskie metody reshenija polnoj i chastichnoj problemy sobstvennyh znachenij. *Sovremennye tehnologii: aktual`nye voprosy, dostizhenija i innovacii*, 2017, pp. 16-22.
7. Jegamberdiev, I. P., Jahshiev, Sh. N., & Mamadijarov, A. Zh. (2021). *Prognozirovanie tehnicheskogo sostojanija podshipnikovyh opor metallorezhushhih stankov po spektral`nym harakteristikam.* Sostav redakcionnoj kollegii i organizacionnogo komiteta.
8. Egamberdiev, I., Sharafutdinov, U., & Ashurov, K. (2021). *Investigation of the possibility of increasing the durability of steel castings 110G13L*. Glavnyj redaktor: Ahmetov Sajranbek Mahsutovich, d-r tehn. nauk; Zamestitel` glavnogo redaktora: Ahmednabiev Rasul Magomedovich, kand. tehn. nauk; Chleny redakcionnoj kollegii, p.27.
9. Juraev, A. et al. (2021). *Modification of the structure of the Bitter separator machine*. Strategija sovremennogo nauchno-tehnologicheskogo razvitija rossii: problemy i perspektivy realizacii, 2021, pp. 12-16.
10. Kulmuratov, N. R., et al. (2020). Various issues in the field of setting nonstationary dynamic problems and analyzing the wave stress state of deformable media. *Theoretical & Applied Science*, 2020, №. 9, pp. 365-369.
11. Ashurov, K., et al. (2020). Applications metallographic and X-ray structural analysis. *Studencheskij vestnik*, 2020, №. 20-14, pp.19-21.
12. Jahshiev, Sh. N., et al. (2021). *Formirovanie vibroakusticheskogo signala v podshipnikovyh oporah metallorezhushhih stankov.* European science forum, 2021, pp. 19-23.
13. Ulug'ov, G. O., et al. (2021). Investigation of the possibility of increasing the durability of steel castings 110G13L. *Universum: tehnicheskie nauki*, 2021, №. 11-6 (92), pp. 27-34.
14. Jegamberdiev, I. P., et al. (2021). Ocenka vyhodnoj tochnosti shpindel`nogo uzla tokarnogo stanka NT-250 I. *Innovacionnyj diskurs razvitija sovremennoj nauki*, 2021, pp. 103-107.