

PARAMETRIC AND 3D MODELING OF A MULTI-OPERATION MACHINE DRIVE WITH INNOVATIVE TOOTHED BELT TRANSMISSION

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Abstract - This paper proposes a comprehensive approach to developing a multi-operation machine tool drive and creating an innovative belt drive design using 3D and parametric modeling toolings. A preliminary design of a two-stage drive with an integrated horizontal spindle unit is developed. The application of parametric modeling toolings for toothed belt transmission (TBT) in the specialized APM Graph module is considered. During the construction of parametric models, an analytical formula for creating a parameterized tooth profile and a rotary pulley tooth array was generated. 3D modeling of the main drive components in the integrated Creo Parametric computer aided design environment was performed. During drive-housing design, a parametric link was established between the design model and the future casting pattern. A belt and pulley system is created as the basis for TBT modeling and calculation. The fundamental concept behind the design improvement for a toothed belt drive, supported by a patent, is to modify the standard trapezoidal geometry by using a chevron tooth profile. This geometric change increases load capacity and ensures the belt self-centers on the pulleys. The presented 3D models of the toothed belt and pulley provide essential information on the chevron profile's design features. Key analytical relationships for determining the key geometric and dimensional characteristics of the belt and pulleys in this innovative design are presented. Based on the derived analytical relationships for standard toothed-belt drive profiles, it has been demonstrated that the width (and, consequently, the overall dimensions) of a chevron-toothed belt is 30% less than that of the standard version, while the load-carrying capacity of the innovative solution increases by approximately 40%.

Keywords: Machining Centre Drive, Parametric Modeling, 3D Models, Timing chevron belt.

1. Introduction

The widespread use of toothed (timing) belt transmission (TBT), which belongs to the class of gear transmissions, is due to its unique capabilities, combining the positive aspects of both gear and belt drives – relatively high load capacity, compact drive, comparatively low pre-tensioning and loads on shafts and supports [1, 2]. These transmissions successfully compete with chain-and-gear transmissions and, in several respects, significantly surpass them. These respects include: stability of gear ratio (reaching values up to high belt circumferential speeds (from 0.5 to 50 m/s; in rare cases, a wide range of power transmission (from 0.2 to 100 kW); relatively high efficiency value (up to $\eta = 0.98$).

Basic considerations for selecting spindle electric drives with DC and AC motors are discussed in [34]. The use of TBT in machine tool drives can even compete with direct drive, implemented in the form of a built-in electric motor integrated into the spindle unit without additional mechanical transmissions [3, 4]. This applies in particular to the direct drive option using clutches, which is much more difficult to implement [5]. In this sense, the belt has the advantage of offering a gear ratio that can be adjusted in the future, which, in any case, can lead to greater precision. Furthermore, the ability to adjust the backlash with the belt acts as a shock absorber during the machine tool motor start-up process, when sharp mechanical impact and vibration occur. Oscillations (vibrations, hum, jerkings) during machine tool motor start-up are a common problem, which is usually associated with high overload

currents, mechanical faults, or drive settings. When starting an asynchronous motor directly, the starting current exceeds the rated current by 5-8 times, which causes sharp mechanical impact and vibration.

The prospects of using TBT facilitate the research and development of new design options for such transmissions that meet and predict market needs in various industries and their expanding production.

The well-known Italian company Megadyne [6] produces a line of toothed belt drives. The Isoran model – rubber endless-toothed belts with a trapezoidal tooth profile – is promising. Along with the classic form, belts with a parabolic tooth profile are being introduced, which provides graduated pressure along the tooth height and, consequently, increased load capacity. Megadyne is conducting extensive research into new variants of TBT cord inserts. The use of fiberglass reinforcement ensures minimal elongation and high belt flexibility, while chloroprene rubber improves wear resistance and increases belt service life. Thanks to the innovative “Dual Core” hybrid cord technology, another Isoran Platinum model eliminates the problems inherent in belts with steel and aramid cords (in particular, lower noise levels). Furthermore, increased strength is achieved – 40% more compared to the previous belt model.

Another major manufacturer of drive belts, Gates (40% of global production) [7], conducts research and development to improve designs and manufacturing technology. Carbon fiber reinforcement is used to increase strength and maintain length stability. Nylon fabric coverings are used to protect teeth and improve wear resistance. A curved tooth profile is being introduced for the PCV 8MGT series of belts, providing improved stress distribution and higher overall load capacity. This design is claimed to increase efficiency to 99%.

However, TBT is characterized by several disadvantages, such as sensitivity to shaft misalignment, which leads to problems with self-centering of belts on pulleys; limitations in transmitting very high loads, which complicates load capacity increases; and the inability to slip to protect the drive from overloads.

An analysis of the development trends of timing belts allows us to identify several areas related to the improvement and development of new polymeric materials for the belt base and for reinforcement, on the one hand, and the optimization of tooth geometry, on the other.

By optimizing geometry using computer modeling, the foundations are laid for creating teeth with optimal strain distribution and increased load capacity. Historically, the geometric shape of the tooth has undergone several stages of improvement [8]:

1. Standard trapezoidal tooth profile (STD; T) with profile angle $\alpha = 40^\circ$;

2. HTD – profile (High Torque Drive) with a semi-circular tooth shape, which reduces strain concentration at the root of the tooth;

3. Advanced Trapezoidal (AT) – a trapezoidal profile with a rounded root, which increases the transmitted torque and service life of the belt. Increased profile angle $\alpha = 50^\circ$ creates a balance between the transmitted power and stability associated with tooth jumping under overloads;

4. GT (Gates Tooth) – a profile with a parabolic tooth contour for a more evenly distributed load and reduced wear;

5. RPP (Reinforced Power Profile) – a profile with a parabolic tooth reinforced with carbon fiber or Kevlar, designed for use in high-load drives.

An analysis of research and development in the field of the use of toothed belt drives in machine tool manufacturing and other industries has shown the potential of work in the field of modernization and improvement of materials, designs and production technology of drive belt drives with various tooth geometries.

On this basis, the following problem statement is carried out in this work: improving the design of the tooth belt transmission of the machine drive based on the search for the optimal geometry of the teeth.

2. Literature Review

The issues of load capacity and strength of the toothed belt associated with the transmission of maximum power, as well as the search for the optimal geometry of the belt teeth and pulley, in which the TBT can function without slipping, are considered in a large number of studies by various authors.

For belts with classic and modified trapezoidal profiles T 10 and AT 10 (according to ISO 5296), a study was conducted on the properties and interactions of individual tension layers of the belt material on the output parameters of the belt-pulley pair [9]. A new approach to solving the problem of increasing energy capacity and strength during torque transmission from the electric motor to the drive input shaft is the selection of mechanical stabilization factors and the orientation of individual layers of the belt material. These are the factors that determine the level of energy loss during long-term operation of the belt drive. At the same time, in the machine-tool industry, the “energy loss” criterion considered here is not critically important for long-term operation. The integrated criterion is the transmission load-carrying capacity, which encompasses the energy loss characteristic.

A different approach to analyzing the load capacity of a toothed belt associated with maximum power transmission is presented in [10]. This approach is based on the development of an innovative design for a toothed belt drive using two

pairs of driving and driven pulleys. These pairs of pulleys are characterized by a special spatial tooth arrangement: an increased pitch between the teeth of a pulley with a larger diameter and the teeth positioning of the smaller pulley precisely midway between the larger pulley teeth. This results in reduced angular velocity fluctuations, uniform transmission ratio, and, consequently, two-phase torque compensation. The ideas proposed in the work are intended for highly specialized applications involving large-scale machine tools; for small and medium-sized machines, the issue of transmission ratio non-uniformity is not a significant concern.

In [11], the problems of increasing the rigidity of toothed belts in mechanisms for transmitting power from a rotating shaft are considered. This increase is achieved by replacing the straight sections of the toothed belt with highly rigid connecting links. The toothed belt is modified using two connecting sections. The belt, reinforced in rigidity, consists of one connecting section, allowing for length adjustment for tension control, and another fastening section connected to the toothed belt. The authors created an analytical model for the reinforced belt design and performed simulations in the Matlab/Simulink environment. The authors did not examine how increased stiffness affects sensitivity to tension levels or the wear process at the belt-pulley interface.

The issues of searching for new design solutions related to the profile of belt teeth and the spatial position of the TBT pulleys are widely covered in the literature.

The paper [12] considered the problem of estimating angular error in power transmission, which is particularly relevant in precision engineering. The authors propose a system with a belt tension adjustment mechanism and a corresponding measuring platform. The optimal belt tension, which minimizes angular errors, was experimentally determined. A T10 trapezoidal belt was selected as the toothed belt under study. The experimental results show that the maximum angular error was 48 arc seconds at a belt tension of 7.5 N. With this angular error, the optimal belt tension was significantly lower than the values used in typical power transmission systems. Conclusions drawn for precision engineering are not necessarily applicable to the wide range of small, medium, and large machine tool sizes.

Some effective solutions have been proposed at the patent level. Patent [13] proposes a new TBT design, with belt teeth having a convex profile and pulley grooves being concave. The height of the belt teeth exceeds the depth of the pulley groove. Disadvantages of this patented solution include increased belt tooth deformation and the presence of flanges on the pulley end surfaces, which leads to significant frictional power losses.

The additional costs of tooth profile manufacturing technology should also be taken into account.

The patent [14] discusses an interesting approach to studying modified tooth geometry with the aim of increasing the strength of a toothed belt and tooth root, thereby increasing the belt's service life. The basic idea is to represent the tooth profile as two regions in the form of an arcuate section. The main graphic primitive is a common tangent through the arcuate section of the first toothed part and a similar arcuate section of the second toothed part. The second graphic primitive is a tangent to the tooth edge at the starting point of the circular arc forming the first part of the tooth. The research found a range of variation in the angles between the two tangents from 6° up to 14.4° , which achieves the effect of increased strength and durability.

An analysis of the above-mentioned works demonstrated the importance of research related to the search for modified designs of belt transmission components and the potential for finding optimal tooth profile geometry, taking into account their correct orientation within the machine tool's working space. This direction of research is consistent with the main trends in improving belt drives in modern mechanical engineering.

3. Parametric Models Formation for Main Motion Drives of a Multi-operation Machine

3.1. Conceptual Drafting of a Drive with a Belt Transmission

As stated above, the service life of the machine directly depends on the design and characteristics of the drive belt transmission.

Let's consider a two-stage speed box for the main drive of a multi-operation machine tool, which is mounted in a cast housing with an integrated horizontal spindle unit [15]. This drive is equipped with interchangeable modular tooling [16] (vertical and slotting heads) and a camshaft that transmits rotation to the horizontal or vertical spindle via an automatic pinion shift device. A sketch variant of a two-stage machine drive design is shown below (Fig. 1).

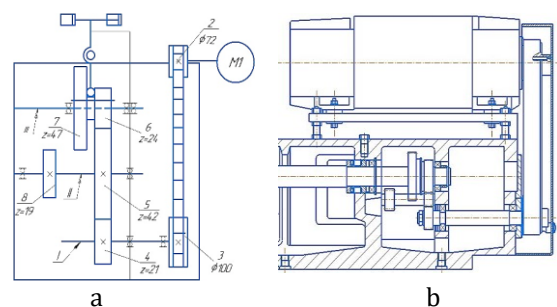


Figure 1: Sketch version of a drive with a tooth belt transmission: a) kinematics; b) section

Spindle rotation is transmitted from an electric motor via a toothed belt transmission to shaft I (Fig. 1, a). Then, through a two-stage speed box, to shaft III, from which rotation is transmitted to the clutch of the vertical head or slotting head, or to the gear of the horizontal spindle. Tool blocks are changed using a hydraulic station, a hydraulic cylinder, and a pack of disc springs [17].

Depending on the machine size, a transverse layout (convolution) is designed, which predetermines the overall dimensions of the machine itself. Figure 2 shows a layout that ensures maximum rigidity of the spindle unit and the machine as a whole. A special feature of this design is the implementation of a joint on the right side of the housing for mounting a gear in a spatial layout [18, 19]. At the same time, the overall technologically feasible rectangular shape of the housing with minimal dimensions is maintained [20]. In addition, a spherical cavity is made in the lower part of the housing, which allows for minimizing the dimensions of the housing with fixed initial data.

For the efficient design of a whole range of TBT for multi-operation machines with a two-stage main drive (shown in Fig. 1), we will use the parameterization tools of the APM WinMachine CAD system [21, 22].

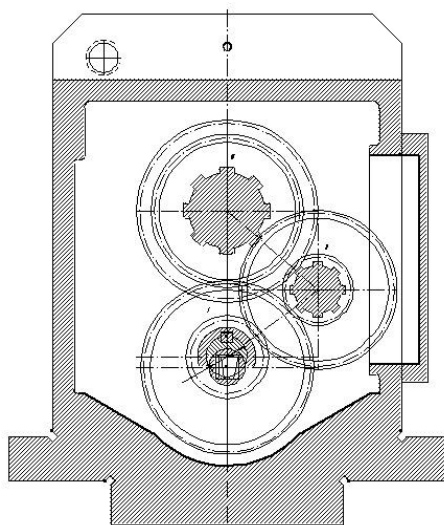


Figure 2: Speed box transverse layout

3.2. Construction of Parametric Models of Toothed Belts and Pulleys

To evaluate the main geometric, thermal and kinematics characteristics of the toothed belt transmission, a traditional calculation method was used [23]. We will calculate the belt transmission with the following parameters: module (m) and the number of teeth (z) driving (index 1) and driven (index 2) pulleys: $m = 4$ mm; $z_1 / z_2 = 18 / 25$. The calculation results are presented in Table 1.

Parametric models of a toothed belt and pulley are constructed in accordance with the following algorithm [21, 24].

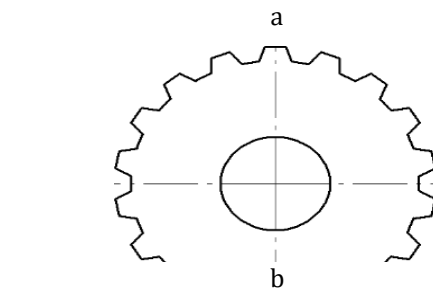
At the first stage, input data is entered in the form of variables in the “Variables” dialog box (Fig. 3).

At the second stage, a graphical assignment of a sequence of commands is implemented in the “Commands” window, leading to the construction of the required model of the toothed belt (Table 2).

Table 1. Calculation results

Pulleys parameters	Pulley number	
	2	3
Reference diameter of the pulley, $d = m \cdot z_p$ (mm)	72	100
Correction K , taking into account the load level $K = 0.2 \cdot (F_r/b) \cdot \lambda \cdot z_p$	0.0336	0.0562
Circular pitch of pulley P_p in the midline of the teeth $P_p = (d_a - h_p) / z_p$ (mm)	11.596	11.855
Angle between the branches of the belt γ (deg)	40	40
Tooth pulley length $B = b + m$ (mm)	54	54

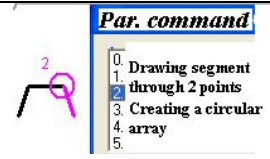
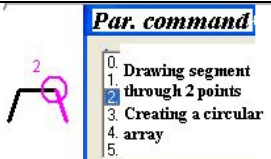
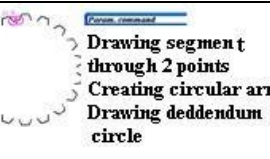
Variables		
Variable	Expression	Value
m		4
beta		25
tp	$6.283 \cdot (m) = 2 \cdot \pi \cdot (3) + 9.425 \cdot (r)$	12.566
h	$8 \cdot m$	2.4
H	$m + 1$	5
dt	$4 \cdot (m) = 2 \cdot \pi \cdot (4) + 8 \cdot (m) = 5 \cdot \pi$	0.4
n		3
x4	$tp / 2 \cdot (m / 2 + h \cdot \tan(\text{rad}(25)))$	3.1638616204
delta	$6 \cdot (dt) = 4 \cdot \pi \cdot (8) + 1.3 \cdot (dt) =$	0.6
b	$10 \cdot (m) = 2 \cdot \pi \cdot (3) + 20 \cdot (m) = 3 \cdot \pi$	30
n1		7



Variables		
Variable	Expression	Value
N		1
n1		1450
u		2
k		1.65
cp		1.7
T1	$9550 \cdot N / n1$	6.5962068966
m1	$k \cdot (cp \cdot T1)^{1/3}$	3.6912946037
m	$2 \cdot (m1) = 1 \cdot \pi \cdot (2.5) + 3 \cdot (m1)$	4
z1	$11 \cdot (m) = 1.5 \cdot \pi \cdot (2.5) + 14 \cdot (m)$	18
d1	$m \cdot z1$	72
v	$M \cdot P1 \cdot d1 \cdot n1 / 50000$	5.4663712172
h1	$1.8 \cdot (m) = 1.5 \cdot \pi \cdot (2.5) + 2.7 \cdot (m)$	3.8
Fp	$2000 \cdot cp \cdot T1 / d1$	311.0153256705
del	$6 \cdot (m) = 1.5 \cdot \pi \cdot (3) + 8 \cdot (m) = 3 \cdot \pi$	0.8
b	$12 \cdot (m) = 1.5 \cdot \pi \cdot (2.5) + 20 \cdot (m)$	30

Figure 3: Windows variable: a) belt; b) pulley; c) expression

Table 2. Elements of the parametric pulleys model

Creates a tooth shape	Creating a circular array of tooth addendum	Creating a circular array of tooth dedendum
		
$L = \frac{h_p}{\cos(\text{rad}(2\beta))};$ $\alpha = 270 + \beta$	$n = z_p$	$t = \frac{2 \cdot \pi \cdot \left(\delta + \frac{h_1}{2} \right)}{z_p}$

The parametric command list specifies parameters for commands, if needed. Calculation expressions use variables specified in the command window or auxiliary variables created during model construction. Each graphical procedure (comprising a graphical formula) corresponds to an analytical expression (analytical form). Similarly, a model of a toothed belt drive pulley is constructed in accordance with the data in Table 1.

In the process of constructing a graphical model of a toothed belt transmission ($m = 4 \text{ mm}$; $z_p = 18$) the problems of parametric representation are solved: a) a pulley tooth with a side length L , mm; pulley tooth height h_p , mm; tooth profile angle β , grad; b) the formation of circular arrays of tooth peaks and c) a circular array of cavities determined by the height of the belt tooth h_1 , mm; In the process forming array the distances from the tops of the teeth to the center line of the belt cord δ , mm are determined and also the distance from the bottom of the tooth groove to the belt cord t , mm and the number of pulley tooth z_p .

The third stage involves analyzing the resulting model for compliance with the required one, and, if necessary, correcting the command parameters or modifying the construction method for the entire model or part of it. Fig. 4 shows a sketch of a toothed belt drive with lateral and radial clearances.

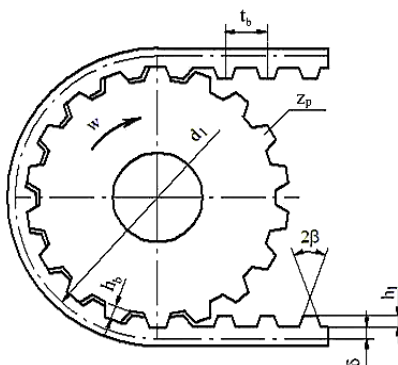


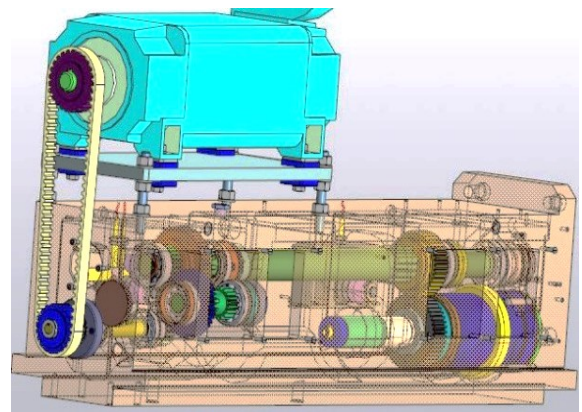
Figure 4: Sketch of a parameterized model of a toothed belt transmission

4. 3D Modeling of a Toothed Belt Drive for Multi-operation Machine

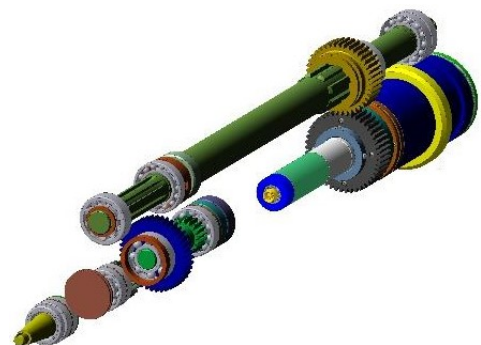
The study of the designs of various mechanisms and the designs of their stress-strain state is carried out using a three-dimensional modeling tooling [25, 26].

To analyze the working capacity of the gear transmission for a multi-operation machine main drive, a set of three-dimensional models was developed in the Creo Parametric integrated CAD environment [27].

In the two-stage machine drive under consideration (Fig. 5), solid models of the housing, shafts and gears have been developed.



a



b

Figure 5. 3D modelling of a two-stage machine drive: a) transparent release; b) kinematics

During the connection establishment process, important aspects such as shrinkage, required pattern draft angle, and fillets are taken into account, along with the geometry of the housing structure, which requires machining after casting. Another advantage of this relationship is the mechanism used to "Inherit" geometry and elements from the design part to the casting pattern. Furthermore, the developed method for identifying the geometry and data of a design element for transformation in the inherited element, without changing the original part, is very effective.

When constructing a shaft model (Fig. 5), the Shaft command is used to define the axis of symmetry, sketch the sections of revolution, and then place them on the model. Stepped shafts require a step-by-step addition of material as they transition from one section to another. The key element of a stepped shaft is the "Neck". The construction algorithm specifies an angle that determines the number of degrees of rotation. A reference plane is then selected, and an open cross-section of the neck with the ends aligned with the silhouette edge of the part is sketched. When creating the neck, the section rotates around the part at a specified angle, removing material within the section.

The pinion-shaft toothed surface (Fig. 5) is created in the Creo Mechanism Design module using the "Gear engagement" tab and the "Basic Gear Pair" tooling. The effectiveness of this tooling is due to its universal representation of various types of gearing as the interaction of a pair of rigid bodies connected by an axis of motion.

A wide range of modeled designs of gear elements is supported by a consideration method that is not based on the geometry of the part and on the ability to independently specify the gear ratio and the value of the pitch circle [28]. Visually, this construction method displays two circles centered relative to the selected axis of motion (in this case, it coincides with the shaft axis (Fig. 5)). These circles are connected by smaller segments representing the helix angle. If the Pitch Circle Diameter defined by the user is selected, then the number of gear teeth for the parameters (Gear 1, Fig. 5) and gear wheel (Gear 2) should be entered in the Gear Ratio section. In this case, the diameters of the pitch circles on both tabs are calculated automatically, which allows the express method to implement the procedure of multivariate design.

The parameter values (Pressure Angle) are entered in a similar manner (default value $\alpha = 20^\circ$) and the helix angle of the gear connection. The helix angle is applied to the pinion, and the corresponding helix angle of the gear wheel is automatically adjusted to match the helix angle of the pinion.

For the development and 3D modeling of belt drives, Creo Parametric has the appropriate 3D tooling and command sets that provide creating

both a timing belt and pulleys that transmit torque from the engine to the gearbox. To create a circular array of teeth, use the Import Data Doctor module. To do this, with a belt tooth (pulley) object selected, use the Transform command, rotate item to create a circular array of teeth. It's convenient to enter a teeth pitch in the Move window and use the Smooth command to move this geometric object smoothly.

As a result of using the Creo Mechanism Design module, a belt and pulley system was created. This system is a parameterized design element of the drive transmission assembly for a multi-operation machine tool.

The innovative nature of the proposed solution at the design stage also lies in an integrated approach combining 3D and parametric modeling. This enables the rapid design of various synchronous belt drive configurations across the full range of cross-sections for these new belt designs.

5. Creation of an Innovative Toothed Belt Transmission Design

Increasing the load capacity and durability of toothed belt transmission, solving vibration resistance issues through vibration damping, is one way to improve the technical characteristics of mechanical drives [29].

Issues related to assessing the vibration level of machine tool components when machining a blade-type part in a turning-boring process cycle are presented in [30]. A calculation formula and a dynamics model have been developed for high-precision machining – fine boring, in which intense bending-torsional vibrations occur [31].

To solve these types of issues, the article presents a patent-level approach [32] that develops a design modification of the machine tool drive belt transmission to solve this problem. The basic idea behind the improvement is to modify the standard trapezoidal geometry by using a chevron-tooth (double-helical) profile along the longitudinal direction. The effect of this geometry change is an increase in load capacity.

A 3D design of the components of a modified belt transmission was developed in the Creo Parametric environment (Fig. 6). This design shows a drawing of a fragment of a double-helical belt (DHB) with longitudinal teeth and a pulley for this type of belt. The elements of the modified belt's teeth and its profile are shown in Fig. 6.

The teeth of a belt with a trapezoidal profile in a normal cross-section are made in the longitudinal direction in the form of chevrons [32]. The teeth and cavities located on the outer surface of the pulleys with a similar trapezoidal profile in a normal cross-section are also made in the form of chevrons.

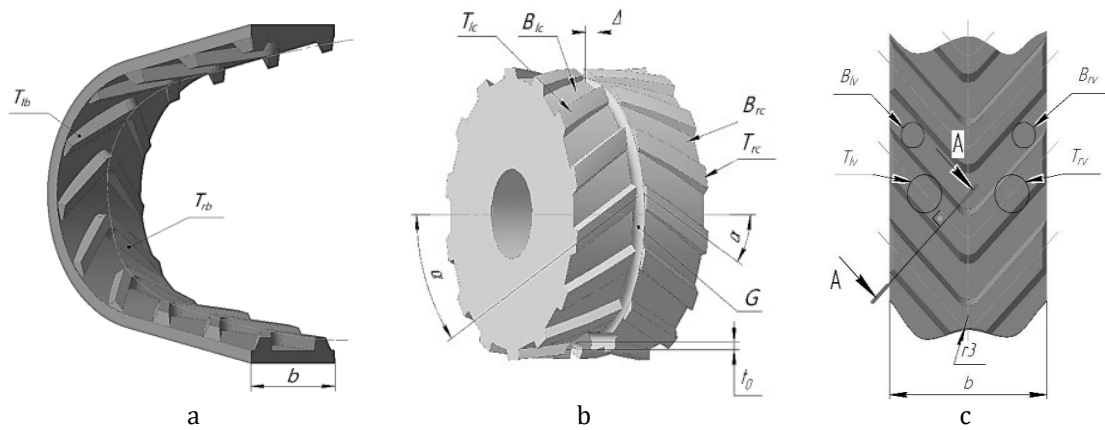


Figure 3. 3D models of toothed belt drive components: a) belt; b) pulley; c) chevron belt profile

Let us consider the design features according to the received patent solution [32]. The belt teeth on the left (T_{lb}) and right half-chevrons (Fig. 6, a) have the same angles of inclination (T_{rb}) and are directed in different directions, Fig. 6, b. The cavities between the belt teeth on the left and right half-chevrons are indicated respectively B_{lc} and B_{rc} .

Fig. 6, b shows a pulley with teeth (T_{lc}) and cavities (B_{lc}) left half of the chevron, with teeth (T_{rc}) and cavities B_{rc} right half-chevron with angles of the same size and direction α , as on the belt, and also an circular groove G between the half-chevrons of the pulley width Δ and depth t_0 .

The trapezoidal profile of the belt teeth in the normal section and its main parameters, shown in Fig. 6, coincide with the profile of the teeth and the numerical values of the parameters of a standard toothed belt [33].

The presence of a circular groove on the pulley with a width Δ (Fig. 6, b), allows to exclude contact of the pulley teeth with non-working areas of the belt teeth, where their left half-chevrons intersect with the right half-chevrons, which is formed by a rounding radius $r = 0.2 \cdot m$; (m – module of teeth in normal section $A-A$).

Chevron belt width b_c , found according to the criterion of belt traction capacity will be (mm): $b_c = F_t \cdot \cos \alpha / (W - q \cdot V^2)$. Here F_t – calculated circumferential force transmitted by the belt (N); w – calculated specific circumferential force (N/mm); q – linear density (kg/(m mm)) of 1 m of belt with a width of 1 mm; V – peripheral speed of the belt (m/s).

The numerical value of the width b_c in $(\cos \alpha)$ times smaller than the width b standard toothed belt, [3]. For the angle of inclination of the teeth on the half-chevrons $\alpha = \pi / 4$ modified belt width b_c all other things being equal, approximately 30% less than the width of a standard timing belt b .

Accordingly, if these belts are equal in width: $b_c = b$, the load capacity of a chevron belt in $(1 / \cos \alpha)$ times more (that is, 40% for $\alpha = \pi / 4$), than a standard timing belt.

Furthermore, the chevron arrangement of the teeth on the belt and pulleys prevents lateral belt displacement on the pulleys, which occurs in standard toothed belt drives [31, 34], especially under external vibration loads. That is, the chevron configuration of the belt in the longitudinal direction ensures its self-centering on the pulleys.

The design modification of timing belts discussed here offers two common technical advantages over standard timing belts. First, due to the increased tooth length, the modified belts have a higher load capacity with the same overall dimensions. Second, the curved longitudinal shape of the teeth eliminates axial displacement of the modified belts from the pulleys, which occurs with standard timing belts, especially under vibration loads.

6. Conclusions

Based on the conducted research, the following results were obtained.

1. A preliminary design was developed for a two-stage gear drive for a multi-purpose, multi-function machine tool, equipped with interchangeable modular tooling and an integrated two-support horizontal spindle unit. A new approach is proposed for determining the optimal transverse drive layout based on the criterion of maximum spindle unit rigidity, utilizing the parametric kernel of the APM WinMachine CAD system. For this variant, the housing component design was adapted, featuring a slot on the right side of the housing for mounting the gear and a spherical cavity, minimizing the housing dimensions while maintaining the same initial specifications.

2. For the toothed belt drive, parametric models of the toothed pulley and belt using the tooling and syntax of the specialized APM Graph module were generated. The authors incorporated novel features into the process of generating the belt tooth profile and the circular array of teeth, utilizing combined graphical primitives constructed from analytical formulas via the syntax of the APM Graph module. This basic approach enables an express procedure for the preliminary, multi-variant design of a wide range of toothed belt drives.

3. A set of 3D models of the main drive components: housing, shaft, and gear, was created. The powerful functionality of Creo Parametric CAD software provided the foundation for a holistic understanding of the design and its assembly, enabling the start of a manufacturability analysis. New design solutions were implemented during the 3D modeling of a complex housing component for a machine tool's main drive, utilizing the inheritance mechanism offered by the Creo system and a parametric link between the design and casting models. The Shaft command was used to implement a step-by-step material addition procedure when transitioning from one section of a stepped shaft to another using groove cross-sections. Using the "Basic Gear Pair" command, various designs of gear wheels, gear clusters, and pinion-shafts were constructed using input data for the pitch circle and gear ratio.

4. An innovative toothed belt drive with a longitudinally chevron surface is created and supported by the author's patent. By reducing the belt width with chevron teeth by approximately 30%, the modified belt's load capacity is approximately 40% higher than that of a standard toothed belt. A new design solution is proposed for the configuration of a chevron-profile groove on a toothed pulley with a radius $r = 0.2 \cdot m$. The result of which is the elimination of contact between the pulley teeth and the non-working sections of the belt teeth. Equally important is the ability to prevent lateral belt movement on the pulleys, meaning the chevron arrangement of the belt teeth ensures its self-centering on the pulleys.

Thus, the proposed new construction solutions, the design of a machine drive based on parametric and solid modeling in the Creo Parametric CAD environment, implement an effective, competitive main drive option for a multi-operation machine with an innovative belt drive.

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