The modified drive of a metal-cutting machine with the V-belt transmission of increased resource

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Abstract. The process of the design improving of the V-belt transmission for the metal-cutting machine drive according to the criterion of increased resource is considered. A three-dimensional modeling of the main motion drive structure for multi-operation machine was carried out using the applied library computer aided design KOMPAS-3D. A new approach to the parametric representation of the procedure for constructing a unified profile of a V-belt pulley based on the syntax of the APM WinMachine system is proposed. The idea of constructive change of the standard operating V-belt contour with the corresponding analytical apparatus, which provides increased drawbar power of the transmission, is proposed. Experimental researches on typical sections of V-belts, confirming the increase in the operation reliability of the belt drive have been carried out. provided that the cross-sectional area of the belt remains unchanged

1. Introduction

The productivity of metal cutting equipment is limited by many factors, including the rather poor durability of belt-driven drives. V-belt drives are widely used in drilling-milling-boring machines, the research of the structures of which in the direction of increasing their operational reliability is an urgent direction.

The advantages of belt drives include low noise level, smooth operation and smoothing of high overloads in the starting mode. This is especially true for V-belt and poly-V-belts).

The ultimate goals of any frictional belt drive modernization are to increase belt drawbar capacity and durability. One of the reasons for the insufficient durability of V-belts and poly-V-belts belts is the significant bending stress caused by the relatively large thickness of these belts.

In turn, the reliability of functioning is directly related to the level of the stress-strain state of the belt drive elements. This level is estimated using the finite element method used in works on 3D modeling [1–3].

The analysis of works in the field of reliability of mechanical transmissions [4–6] showed the main directions on which the efforts of researchers are focused.

So, in [4], a study of the effect of loads on the intensity of belt wear and parameters reflecting this process was carried out. One of the consequences of wear is a decrease in belt cross-section, which results in increased stress levels and reduced service life. A classification of several factors affecting belt wear is given. The importance of such factors as the design of the belt transmission, the shape and geometric characteristics of its elements are noted. The authors of [4] point out the importance of early
detection of possible damage, which will allow preparing a more accurate schedule of current repairs. This work proposes an effective device for measuring belt thickness and, as a consequence, assessing changes in the transverse and longitudinal profile of the belt.

Along with the control of the belt geometry, it is necessary to record the level of forces occurring in the belt drive. In [5], algorithms were developed for calculating the primary resistance to the movement of the conveyor belt, considering the parameters of the belt, as well as the design and operational parameters of the conveyor. The authors of [5] have developed and implemented an Information Technology (IT) system for collecting complex data (technical, operational, diagnostic), and on this basis, determining energy characteristics.

In [6], the effect of belt lengthening caused by an increase in the pitch of the toothed belt, which occurs during operation and which persists after unloading the belt, is considered. The authors argue that the greatest amount of movement and power is transmitted by the form, and much less by friction. In process of research the mechanisms of this phenomenon were discovered, the main factor of which is the plastic deformation of the belt. It is about 70%.

At the same time, in these works, insufficient attention is paid to improving the design (shape) of the belt and its effect on drawbar characteristics and durability.

Statement of the research task: The purpose of the report is to perform a computer simulation to optimize the profile of the belt drive to stabilize the operation of the metal cutting machine. All studies aimed at increasing the stability of the equipment during metal cutting lead to increased productivity, improved quality of treated surfaces, which justifies the importance of the proposed topic.

2. 3D modelling main motion drive of metal-cutting machine tools

The drive of the main motion occupies a special position in the structure of the metal-cutting machine tools (MT). The main drives must ensure the high-efficient performance of various operations when changing the speed in a wide range. For a wide range of MT, the main drive includes an Alternating Current (AC) motor, a frequency converter and in most cases 2-3 mechanical transmissions, to increase the range of speeds and torque. The main advantages of such drive are relatively low cost compared to alternative solutions and a wide range of technical data, and its final specification can be changed by choosing another motor or a different belt gear ratio. To analyze the design features and assess the stress-strain state of the MT mechanical transmissions [7–9], a three-dimensional model of the drilling-milling-boring machine drive was built in the KOMPAS-3D CAD environment (Figure 1) using the specialized application "Shafts and mechanical transmissions-3D" [10-12].

![Figure 1. 3D models of MT drive: a – general view (transparency); b – kinematics.](image)

Similarly, using the above application, 3D models of V-belt pulleys were built (Figure 2, a) with normalized geometric characteristics (Figure 2, b).
The pulleys used in machine tools in their form in many cases do not differ from pulleys common in other machines. The variety of nomenclature used in pulley drives, on the one hand, and the availability of regulatory documentation governing the shape and size of their structures, makes it promising to use the parameterization mechanism [13–15]

The versatility of the parametric model is provided by considering the various options for implementing the design of the belt drive [16, 17]. The choice of the graphic profile for the formation of the pulley design will depend on the transmitted power, the belt section and the coordinates of the base point of configuration groove symmetry (Table 1). When designing drives for metal-cutting machines, it is not recommended to choose more than 6 belts in one V-belt transmission, which is associated with an increase in the level of vibrations caused by the difference in the lengths of the belts used. Moreover, in production conditions there is a recommendation – no more than 4 belts in one transmission. As a consequence of this limitation, when constructing a parametric model of a pulley, 4 variants of determining the coordinates of the base point of the pulley groove for the command "mirroring" are used (Table 1).

| Initial groove profile | Belt section | Transferred power N, kW | Coordinates base point |
|------------------------|-------------|-------------------------|------------------------|
| a                      | A           | N≤1                     | x = f; y = d_p-h        |
|                        | A           | 1<N<1.9                 | x = f+e/2; y = d_p+b    |
| c                      | A           | 1.9<N<2.8               | x = f+e;               |
| B                      |            | 3.6<N<5.3               | y = d_p-h              |
| C                      |            | 7.6<N<9.3               |                        |
|                        | A           | 2.8<N<3.5               | x = f+3e/2;            |
| B                      |            | 5.3<N<7.0               | y = d_p+b              |
| C                      |            | 9.3<N<12.5              |                        |
3. Modification of the V-belt transmission profile

Driving V-belts of normal cross-sections, GOST 1284.1-89 (conforms to European standard ISO 1081-90), widely used in modern devices, have insufficient durability. One of the main factors affecting this performance indicator is the high level of bending stresses $\sigma_b$ in the belt [18–20]. Calculations show that the share $\sigma_b$ in the total belt tension $\sigma_{\text{max}}$ is (67…73)% [21–23]. And since the durability of V-belts $L_d$ inversely proportional to the value $\frac{\sigma_{\text{max}}}{\sigma_b}$, the task of reduction $\sigma_{\text{max}}$ by reduction $\sigma_b$ is urgent.

A trapezoid $OABC$ with known parameters $S, W, W_p, \alpha, S$ is given, Figure 3:

$$OA=T; \ AB=W/2; \ P_0P=W_p/2; \ \alpha=40^\circ; \ S,$$

where $S$ – is the area of a full trapezoid; (hereinafter, the value $S_0=S/2$ referring to half of the trapezoid $OABC$ is used.

Figure 3. Construction of a modified section of a V-belt: a – geometrical parameters; b – modified pulley.

It is required to determine the height $OA_1=T_1$ of the modified trapezoid $OA_1B_1C_1$, Figure 3, equal in area of the trapezoid $OABC$, i.e. $S_{01}=S_0$ – this is the first calculation condition $T_1$.

The sides of the modified trapezoid $OA_1B_1C_1$ are formed by an arc of a circle in such a way that it touches the rectilinear side of the trapezoid $OABC$ at point $P$ [24, 25] (Figure 3 is the second calculation condition $T_1$). The radius of this circle:

$$R_p = \frac{T_p}{\sin(\alpha/2) - \sin(\alpha/4)},$$

where $T_p = T - 0.5 \cdot (W - W_p) \cdot \text{ctg}(\alpha/2)$ – height from the lower base of a given trapezoid to the line $P_0P_0$, i.e. a segment $OP_0$, Figure 3.

4. Discussion

The radius $R_p$ is determined from the condition that the tangent to the arc of the circle at its lowest point $C_1$ forms an angle $\alpha/4 = 10^\circ$ with the axis of symmetry $OY$.

The coordinates $[X_0, Y_0]$ of the center of a circle with a radius $R_p$ in the coordinate system $S[X, Y]$ are defined below.

In this case the equation of a circle with a radius $R_p$ in a coordinate system $S_p[X_0, Y_0]:$
\[ X_p = R_p \cdot \cos \beta; \]
\[ Y_p = R_p \cdot \sin \beta. \]  

(1)

Distance between centers \( O_P \) and \( O \) coordinate systems \( S_P\{X_p, Y_p\} \) and \( S\{X, Y\} \), Fig. 1:
\[ O_P O_0 = X_0 = 0.5 \cdot W_p + R_p \cdot \cos(\alpha/2); \quad OO_0 = Y_0 = R_p \cdot \sin(\alpha/4). \]

The connection between coordinate systems \( S_P\{X_p, Y_p\} \) and \( S\{X, Y\} \) is established directly from Figure 3:
\[ X = X_0 - X_p = X_0 - R_p \cdot \cos \beta; \]
\[ Y = Y_p - Y_0 = R_p \cdot \sin \beta - Y_0. \]  

(2)

Using relations (2) and (1), we represent the equation of a given circle in the coordinate system \( S\{X, Y\} \) in the form:
\[ X = f(Y) = X_0 - \sqrt{R_p^2 - (Y + Y_0)^2}. \]  

(3)

The area \( S_{01} \) of the modified trapezoid \( OA_1B_1C_1 \) is limited by three lines – a circular arc (3) and two straight lines: \( Y = T_1 \) and \( Y = 0 \). The calculation \( S_{01} \) is reduced to finding the antiderivative function (3):
\[ S_{01} = \int f(Y) \cdot dY = \left[ X_0 \cdot dY \right] - \int \sqrt{R_p^2 - (Y + Y_0)^2} \cdot dY \]  

(4)

Let us consider separately each term of function (4):
\[ J_1 = \int X_0 \cdot dY \quad \text{and} \quad J_2 = \int \sqrt{R_p^2 - (Y + Y_0)^2} \cdot dY. \]

To find the integral function \( J_2 \), we will use trigonometric substitution: \( Y + Y_0 = R_p \cdot \sin t \), [6].

Then \( \sqrt{R_p^2 - (Y + Y_0)^2} = R_p \cdot \cos t \)

From the introduced substitution follows: \( Y = R_p \cdot \sin t - Y_0 \). Then: \( dY = R_p \cdot \cos t \cdot dt \). As a result:
\[ J_1 = R_p \cdot X_0 \cdot \left[ \cos t \cdot dt \right] = R_p \cdot X_0 \cdot \sin t; \]
\[ J_2 = R_p^2 \cdot \left[ \cos^2 t \cdot dt \right] = 0.5 \cdot R_p^2 \cdot (t + 0.5 \cdot \sin 2t). \]

Returning to the original variable \( Y \), we find:
\[ t = \arcsin \left( \frac{Y + Y_0}{R_p} \right); \quad 0.5 \cdot \sin 2t = \sin t \cdot \cos t = \frac{Y + Y_0}{R_p} \cdot \sqrt{R_p^2 - (Y + Y_0)^2}. \]

After substituting these expressions into the right-hand sides of the functions \( J_1 \) and \( J_2 \) the initial dependence for the calculation \( S_{01} \) takes the form [26–28]:
\[ S_{01} = S_{01}(X) = J_1 - J_2 = (Y + Y_0) \cdot \left[ X_0 - 0.5 \cdot \sqrt{R_p^2 - (Y + Y_0)^2} \right] - 0.5 \cdot R_p^2 \cdot \arcsin(Y + Y_0)/R_p. \]

Substitute in \( S_{01} \) successively two values of the variable \( Y \), that correspond to the boundary points of the arc \( B_1C_1 \): \( Y = T_1 \) and \( Y = 0 \):
\[ S_{01(1)} = S_{01}(Y = T_1); \quad S_{01(2)} = S_{01}(Y = 0). \]

In this case, the area of the trapezoid \( OA_1B_1C_1 \) is determined by the dependence:
\[ S_{01} = S_{01(1)} - S_{01(2)}. \]
The condition for the equality of the areas of two trapezoids \( S_0 = S_{01} \):
\[
S_{01} = (T_1 + Y_0) \cdot \left[ Y_0 - 0.5 \cdot \sqrt{R_p^2 - (T_1 + Y_0)^2} \right] - Y_0 \cdot \left[ X_0 - 0.5 \cdot \sqrt{R_p^2 - Y_0^2} \right] - 0.5 \cdot R_p^2 \cdot \arcsin\left(\frac{T_1 + Y_0}{R_p}\right) + \arcsin\left(\frac{Y_0}{R_p}\right). \tag{5}
\]

From here one unknown quantity \( T_1 \) is determined [29, 30]. Equation (5) was solved by an iterative method, the calculation error was less than 0.1%.

The width of the upper base of the modified section, Figure 1:
\[
W_1 = 2 \cdot A_0 B_1 = 2 \cdot \left[ X_0 - \sqrt{R_p^2 - (T_1 + Y_0)^2} \right].
\]

The central angle \( \alpha_i \) of the arc \( B_i C_i \), which outlines the lateral sides of the modified section:
\[
\alpha_i = \arcsin\left(\frac{T_1 + Y_0}{R_p}\right) - \alpha / 4.
\]

The calculation results for 4 standard sections of V-belts \( A, B, C, D \) and the corresponding modified sections with concave lateral sides are presented in Table 2.

| Table 2. Comparison of standard and modified V-belts parameters. |
|---------------------------------------------------------------|
| Parameters | Belt cross-section as per GOST 1284.1-89 (ISO 1081-80) |
| --- | --- | --- | --- | --- |
| \( T_i / T \) | A | B | C | D |
| --- | --- | --- | --- | --- |
| 7.863/8 | 10.490/11 | 13.386/14 | 18.622/19 | |
| \( W_i / W \) | 13.173/13 | 17.028/17 | 21.812/22 | 32.457/32 |
| \( \alpha_i^0 \) | 15.19° | 15.49° | 13.69° | 15.60° |

In the lines of table 1: \( T_i / T \) and \( W_i / W \) the values in the numerator refer to the modified section, in the denominator – to the standard one.

The decrease in height from \( T \) to \( T_i \) for the belt sections indicated in the table was about 1.7 ... 4.6%.

To assess the change in belt durability in relative units with a specified change in belt height, we represent the maximum stress in a standard belt as:
\[
\sigma_{max} = \sigma + \sigma_b
\tag{6}
\]
where \( \sigma = \sigma_0 + \sigma_v + \sigma_t \) – stresses from pre-tension, from centrifugal forces and external load; (value \( \sigma = const \), since changing the height of the belt section while maintaining the sectional area will not change the values \( \sigma_0, \sigma_v, \sigma_t \).

Let us express \( \sigma_b \) it infractions \( \sigma_{max} : \sigma_b = k \cdot \sigma_{max} \) and the bending stress in the modified section \( \sigma_{b'} \) – infractions \( \sigma_b \) of the standard section: \( \sigma_{b'} = c \cdot \sigma_b \). By analogy with (6) the maximum stress in the modified belt \( \sigma_{max}' = \sigma + \sigma_{b'} \). Let’s find the relation \( \sigma_{max}' / \sigma_{max} \):
\[
\frac{\sigma_{max}'}{\sigma_{max}} = \frac{\sigma + \sigma_{b'}}{\sigma + \sigma_b} = \frac{\sigma + c \cdot \sigma_b}{\sigma + \sigma_b} \cdot \frac{\sigma_{max}}{\sigma_{max}} = \frac{(\sigma / \sigma_{max}) + c \cdot k}{(\sigma / \sigma_{max}) + k} = \frac{1 - k}{1 - k + c} = 1 - k \cdot (1 - c).
\tag{7}
\]

Relation (7) makes it possible to estimate the decrease in the maximum stress in the modified belt in comparison with a standard belt equal in area to it. The numerical values of the coefficient
The determination of these parameters is based on the analysis of the geometrical parameters of \( V_0 \). The coefficient 
\[
\frac{\sigma_b}{\sigma_{max}} \approx 0.67...0.73, \quad \text{indicated above are averaged: } \ k \approx 0.7.
\]
The dependence for the coefficient 
\[
c = \frac{\sigma_b}{\sigma_{max}} \text{ is transformed using the known dependence for the bending stresses in the belt: } \ \sigma_b = E \cdot T / d
\]
(for a belt with a modified section, respectively: 
\[
\sigma'_{b} = E \cdot T' / d
\]
) holds:
\[
c = \frac{\sigma_b}{\sigma_{max}} = \frac{E \cdot T'/d}{E \cdot T/d} = T'/T
\]

Numerical values \( T' \) and \( T \) are taken from table 1. As a result:
\[
c = \frac{T_i}{T} = \begin{cases} 
7.863/8 = 0.98 & \text{section A;}
10.49/11 = 0.95 & \text{section B;}
13.386/14 = 0.96 & \text{section C;}
18.622/19 = 0.98 & \text{section D.}
\end{cases}
\]

The parameters \( E \) and \( d \) in the formula (8) denote the elastic modulus of the belt and the calculated pulley diameter, respectively.

Find the ratio of the durability \( L'_h \) of the modified belt to the durability \( L_h \) of the standard belt:
\[
\frac{L'_h}{L_h} = \left( \frac{\sigma_{max}}{\sigma^*_{max}} \right)^8 = \frac{1}{(1-k \cdot (1-c))^8} = \begin{cases} 
1/[1-0.7 \cdot (1-0.98)]^8 = 1.12;
1/[1-0.7 \cdot (1-0.95)]^8 = 1.33;
1/[1-0.7 \cdot (1-0.96)]^8 = 1.26;
1/[1-0.7 \cdot (1-0.98)]^8 = 1.12.
\end{cases}
\]

Thus, the increase in the durability of the V-belts by giving its lateral sides a concave shape, outlined by an arc of a circle, ranged from 12% to 33% without changing their cross-sectional area.

5. Conclusion

A 3D model of the main drive with a belt drive has been developed for multioperational drilling-milling-boring machines in the integrated CAD KOMPAS-3D environment. At the same time, effective options of this system were used, associated with the presentation of the structure in a transparent and wireframe form, the formation of kinematic diagrams, which a more complete picture of the design and features of the designed products are created. To increase the productivity of the designer, a special application program "Shafts and mechanical transmissions-3D", with such graphic primitives (involute profile, trapezoidal profile, etc.) that are characteristic of stepped shafts, gear, belt transmissions) was used.

For various nomenclature of belt drive elements, the parameterization toolkit in CAD/CAE “APM WinMachine” is used, which makes it possible to implement an express-procedure for developing the design of individual product elements in a multivariate mode. A new approach to forming the working profile of V-belt transmission pulleys using analytical dependencies in the APM WinMachine syntax ("rectangular array", "mirror reflection") for the range of V-belts (from 1 to 6) is proposed.

The analysis of the geometrical parameters of V-belts of the modified section is carried out. Analytical dependencies are obtained for constructing the contour of such a section: the radius \( R_p \) of the circle arc describing its lateral sides, the coordinates of the center of this circle, the coordinates of the boundary points on the arc with the radius \( R_p \). The determination of these parameters is based on two conditions: 1) equality of cross-sectional areas for standard and modified belts; 2) tangency to the arc with a radius \( R_p \) at a certain point on the rectilinear side of the section for the standard belt.

Analytical calculations have shown that the cross-sectional height of the modified belt is 10...18% less than that of the standard one (with the same cross-sectional area). At the same time, the length of the curved profile in the longitudinal section is 11...19% larger, and as a result, the bending stress for the section \( B \) of the modified type belt decreases by about 5%. A comparative calculation of the durability of 4 V-belts of standard sections (A, B, C, D) and 4 modified sections of the same size has been made, which showed that the latter gives an increase in durability from 12% to 33%.
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