# Profiling the cam of the centrifugal screwdown structure of the V-belt variator

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**Abstract.** We consider the problem of profiling the cam of centrifugal pressure device of the driving pulley of V-belt variator which parameters and, first of all, profile would allow to create the optimal axial forces corresponding to the required mode of the transmission operation. A methodology for determining the required optimum axial forces from the condition to ensure the best traction ability of the belt transmission is given.

## 1 Introduction

In a number of mechanical systems, e.g. vehicle transmissions, the transmission ratio has to vary according to the speed and force conditions. For example, V-belt variators are used for infinitely variable transmission ratio control. This ensures that the optimum automatic control of the transmission is realised in such a way that the combustion engine works at a constant output, the highest torque, or the lowest fuel consumption rate. In the V-belt variator, the automation of its control is achieved by using the dependence of the axial forces on the pulleys on the mode of operation. Theoretical and experimental studies of such mechanisms are reflected in various works, e.g. [1-7].

A centrifugal cam device (Fig.1), in which the cam creates the axial force, can be used as a thrust device to generate the axial force on the pulleys of a V-belt variator. However, analytical profiling of the required cam profile is not available in the literature. This raises the challenge of designing a cam whose parameters, especially its profile, ensure that the axial forces required are optimal and in line with the operating mode of the transmission.

## 2 Theoretical part

To obtain a cam profile that corresponds to optimum axial thrust $F_x$, Consider the fact that for each position of the pressure plate (Fig. 2), determined by the amount of its movement *x,* corresponds to a certain force $F_x = f(x)$, and each movement of the disc $x$, equal to the axial movement of the cam, corresponds to a certain angle $\alpha$ its rotation. In turn, the axial force developed by the centrifugal force on the cam also depends on the position of the

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cam. Thus, it is possible to obtain the following relationship $\alpha = f(x; F_x)$ cam angle from its axial movement with known axial force $F_x$.

When profiling the cam we apply the motion reversal method - we mentally stop the cam by giving the entire cam mechanism an additional angular velocity $-\omega_k$ and linear velocity $-v_k$, equal in magnitude and opposite in direction to, respectively, the angular $\omega_k$ and linear $v_k$ to the speed of the cam (fig.2). As a result, the pusher with the centre $O_2$ will rotate around the centre of the cam with an angular velocity of $-\omega_k$, having a linear velocity $-v_k$. Each rotation angle α of the tappet axis will correspond to a certain displacement *x* of the tappet along its axis.

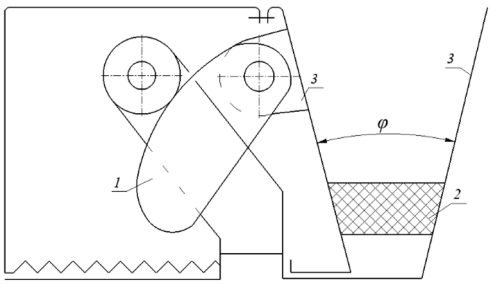


**Fig. 1.** 1 - cam, 2 - belt, 3 - pulleys.

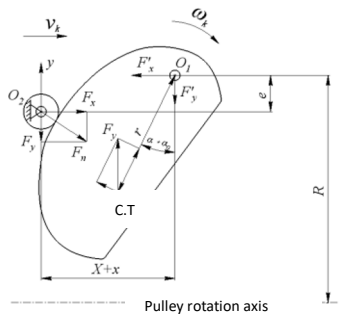


**Fig.2.** Forces acting on the cam.

Angle value $\alpha$ and the radius is a vector

$$\ell = \sqrt{e^2 + (X + x)^2}\,, \qquad (1)$$

equal to the distance from the centre of rotation of the cam to the centre of the roller (Fig.2), unambiguously defines the cam centre profile. Here $X$ – axis distance between centres $O_1$ and $O_2$ when they are as close as possible, which corresponds to the greatest depth of the belt into the pulley groove; $e$ – cam eccentricity; $F_x$ – required optimum axial pressing force; $r$ and $\alpha_0$ – coordinates of the centre of gravity of the cam at the greatest depth of the belt into the pulley groove, selected by design; $R$ – distance from the axis of rotation (pulley axis) to the centre $O_1$ cam rotation; $F_c = m\,\Omega^2[R - r\cos(\alpha_0 + \alpha)]$ – centrifugal force developed by the cam; Ω – angular speed of the pulley; $m$ – cam mass.

Dependency $\alpha = f(x; F_x)$, we obtain from the condition of equality of work of the active and reactive forces acting on the cam (Fig.2):

$$\sum A_{F_c} = \sum A_{F_n}\,. \qquad (2)$$

Decompose the force $F_c$ subdivided $F_c \sin(\alpha_0 + \alpha)$ and $F_c \cos(\alpha_0 + \alpha)$. Reaction $F_n$ The tappet on the cam, normal to both surfaces (disregarding friction), will be moved to the point of $O_2$ and also decompose $F_x$ and $F_y$.

Elementary work of forces $F_y$ and $F_c \cos(\alpha_0 + \alpha)$ on the elementary displacements is zero, since there is no displacement in the direction of these forces. Then equality (2) is written in the form

$$rF_c\, sin(\alpha_0 + \alpha)\, d_\alpha = F_x d_x, \qquad (3)$$

*or*

$$\int_{\alpha=0}^{\alpha} F_c\, r sin(\alpha_0 + \alpha)\, d_\alpha = \int_{x=0}^{x} F_x d_x\ . \qquad (4)$$

After substituting $F_c$ in (4) and further integration we have

$$m\,\Omega^2 r[R\cos\alpha_0 - R\cos(\alpha_0+\alpha) - \frac{r}{2}\cos^2\alpha_0 + \frac{r}{2}\cos^2(\alpha_0+\alpha)] = \int_{x=0}^{x} F_x d_x \quad (5)$$

Designating $\cos(\alpha_0+\alpha) = y$ and solving equation (5) with respect to y, we obtain

$$y_{1,2} = \frac{1}{r}\left[R \pm \sqrt{(R - r\cos\alpha_0)^2 + \frac{2}{m\,\Omega^2}\int_{x=0}^{x} F_x d_x}\,\right], \quad (6)$$

Where from

$$\alpha = arccos\left\{\frac{1}{r}[R - \sqrt{(R - r\cos\alpha_0)^2 + \frac{2}{m\,\Omega^2}\int_{x=0}^{x} F_x d_x}]\right\} - \alpha_0. \quad (7)$$

Sequence for determining the optimum force $F_x$ is discussed below. This produces discrete values $F_{x_i}$ for another pulley position determined by the coordinate $x_i$, and in view of the fact that to obtain an analytical function $F_x = f(x)$, and therefore the integral $\int F_x d_x$ difficult, the value $\int F_x d_x$ it is more convenient to calculate in an approximate way

$$\int_{x=0}^{x} F_x d_x = \sum_{i=0}^{i=n} \Delta x \frac{F_{x_i} + F_{x_{i+1}}}{2} = \frac{\Delta x}{2}\sum_{i=0}^{i=n}(F_{x_i} + F_{x_{i+1}}), \quad (8)$$

where the force $F_{x_i}$ is defined at points with coordinates $x_i = i\Delta x$ ($i$ =1; 2; . . . $n$); $n$ – the number of equal segments into which the stroke is divided $x_n$ the pressure plate from one end position to the other: $\Delta x = x_n/n$.
As a result

$$\ell_i = \sqrt{e^2 + (X + x_i)^2} \quad (9)$$

and

$$\alpha_i = arccos\left\{\frac{1}{r}[R - \sqrt{(R - r\cos\alpha_0)^2 + \frac{\Delta x}{m\,\Omega^2}\sum_{i=0}^{i=n}(F_{x_i} + F_{x_{i+1}})}]\right\} - \alpha_0. \quad (10)$$

Since the operating mode in which the motor is automatically loaded must, by its nature, correspond to the maximum utilisation of torque and power, some optimum speed must be substituted in (10) $\Omega_1 < \Omega < \Omega_2$ (Figure 3), found, as pointed out by Yu.P. Pozdnyakov, from the condition

$$P_p(\Omega) + \mu\, T_p(\Omega) = max, \quad (11)$$

where $P_p$ and $T_p$ define optimum power and torque values; $\mu_1 \le \mu \le \mu_2$; $\mu_1 = \frac{P}{T}$ by the $\Omega = \Omega_1$ and $\mu_2 = \frac{P}{T}$ by the $\Omega = \Omega_2$.

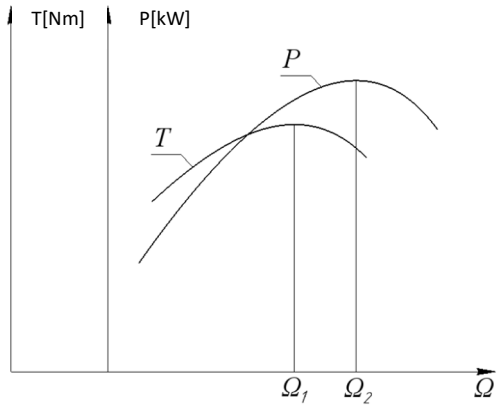


**Fig.3.** Selection of optimum speed.

Thus, with a known optimum force $F_{x_i}$, appropriate to a certain position $x_i$ of the pressure plate, by (10) and (9) it is possible to find the rotation angle $\alpha_i$ and the radius is a vector $\ell_i$, defining the cam centre profile.

The optimum axial force developed by the centrifugal cam follower of the V-belt variator can be calculated from the condition that the best traction capacity of the belt transmission is achieved, which is the traction factor of ψ = 0.6 . . . 0.75.

## 3 Calculations

Following the recommendations of B.A. Pronin and G.A. Revkov [1], the sequence of calculations for the case in question is given below. We will assume that the smallest diameter has already been selected $d_1$ drive pulley, largest diameter $D_2$ the driven pulley, the regulation range is known, the belt length is selected and the cross-sectional dimensions of the belt are known.

1. Set the optimum torque value $T_1$ on the drive sheave.
2. Determine the current design diameter of the drive pulley

$$d_{x_{1_i}} = d_1 + \frac{x_{1_i}}{tg\,\varphi/2}, \qquad (12)$$

where $\varphi$ – groove profile angle (Fig.1).

3. Determine the circumferential force on the drive sheave

$$F_t = 2T_1 / d_{x_{1_i}} \qquad (13)$$

4. Set the average value of the optimum value of the traction coefficient ψ = 0.675.
5. Determine the value of $m = (1 + \psi)/(1 - \psi)$.
6. Find the relative axial force on the drive pulley [1]

$$\frac{F_{x_{dp}}}{F_t} = Y_{dp} = \frac{cos(\frac{\varphi}{2})}{2f} + \frac{m}{m-1}\,\frac{\Theta_{п_1}}{2tg(\rho+\frac{\varphi}{2})}, \qquad (14)$$

where $\varphi$ – pulley groove angle; $f$– belt friction coefficient against the pulley; $f_1' = f/\sin 0.5\varphi$ – reduced coefficient of friction for the driving pulley; $\Theta_{п_1} = \Theta_1 - (\ln m)/f_1'$ – prime mover angle; $\Theta_1$ – angle of belt wrap around the drive pulley; ρ = arctg $f$.

7. Determine the required axial force on the traction sheave

$$F_{x_{dp}} = Y_{dp}\, F_t. \qquad (15)$$

8. Find the optimum axial force developed by each of the drive sheave cams,

$$F_{x_i} = \frac{1}{z}\,(F_{x_{dp}} + F_{sp}), \qquad (16)$$

where z – number of push cams; $F_{sp}$ – axial force developed by the spring on the drive pulley.

After determining the axial force at the initial position of the traction sheave, find $F_{x_1}$ by the $x_1$, after $F_{x_2}$, by the $x_2$ and etc.

Since motor vehicles are regulated by both engine crankshaft speed and drag torque, in this case it is necessary to find the axial force $F_{x_{rm}}$ on the idler pulley as a function of torque $T_2$ and slave disc movement $x_{rm}$, which is used to select the characteristic and type of pressure device for the driven pulley.

Sequence of subsequent axial force determination $F_{x_{rm}}$ and its corresponding torque $T_2$ on the idler pulley will be as follows:

1. Determining the current estimated diameter of the driven pulley

$$d_{x_{2_i}} = \sqrt{(\pi a - d_{x_{1_i}})^2 + 4a(L-2a) - (2\pi a - d_{x_{1_i}})d_{x_{1_i}}} - \pi(a - d_{x_{1_i}}), \qquad (17)$$

where $a$ – centre-to-centre distance (including belt extension), $L$ – strap length.

2. Determination of idler pulley displacement

$$x_{2_i} = (D - d_{x_{2_i}})tg(\varphi/2). \qquad (18)$$

3. Determining the relative axial force of the driven pulley [1]

$$\frac{F_{x_{rm}}}{F_t} = Y_{rm} = \frac{1}{2f_2' tg(\rho_r + \frac{\varphi}{2})} + \frac{1}{m-1}\frac{\Theta_{п_2}}{2tg(\rho + \frac{\varphi}{2})}, \qquad (19)$$

where $\Theta_{п_2} = \Theta_1 - (\ln m)/\, f_2'$ – idle angle at the idler pulley; $\alpha_2$ – angle of belt wrap around the idler pulley; $f_r = f sin\beta$ – individual coefficient of friction; $\beta$– the angle between the sliding speed and the tangent (usually take $\beta = 20°$); $\rho_r$=arctg$f_r$;

$$f_2' = \frac{f cos\beta}{sin\left(\frac{\varphi}{2}\right) + f sin\beta cos\left(\frac{\varphi}{2}\right)}$$

– reduced angle of friction for the driven pulley.

4. Determining the required axial force on the idler pulley

$$F_{x_{rm}} = Y_{rm} F_t. \qquad (20)$$

5. Determining the resisting torque to be overcome $T_2$ on the idler pulley

$$T_2 = F_t\, d_{x_2}/2. \qquad (21)$$

Based on this relationship $F_{x_{rm}} = f(x_{rm};\ T_2)$ design the pressure device for the idler pulley.

## 4 Conclusion

In the presented work the problem of designing the cam of centrifugal pushing device of pulley of stepless V-belt transmission (variator) which profile allows to create optimal axial forces corresponding to the required mode of transmission operation is solved. Methodology is given to determine the optimum axial forces required from the condition to provide the best traction ability of the belt transmission.

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