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MATHEMATICAL MODEL FOR RENDERING THE FROFILES OF THE SHOCK-FREE CAMS USED FOR THE CAMSHAFTS OF THE ROAD VEHICLES ENGINES

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ABSTRACT : The paper presents a study on the Kurz cam profiles, ended with the creation of a software which enables the statement of a complex motion law allowing the good functioning, without shocks or detachment, of the cam-displacing shoe tappet mechanism, at high speed, typical of the camshafts used on the internal-combustion engines of the road vehicles.

1 INTRODUCTION – DESCRIPTION OF THE KURZ CAM USED IN THE MANUFACTURING OF CAMSHAFTS FOR INTERNAL COMBUSTION ENGINES

The cam mechanisms are used for an accurate construction of some complex functions for the motion transmission and are very widely used in the structure of modern cars, of internal-combustion engines, mechanical automatic machines, manipulators. The advantages of these mechanisms are offered by the upper couples enabling a wide variety of motions of the routed element. They are more compact and are easier to design for any required motion law.

The Kurz profiles ensure a smooth, shock-free variation of the acceleration which determines a decrease in mechanical stress, noise and vibration level, along with an increase in reliability and the improvement of the intake and exhaust processes. The advantages of this profile and the existence of several techniques needed to easily manufacture it using numerical control machine tools, have made this profile an interesting them for the present paper.

The Kurz cam is divided into 10 areas, two of the corresponding to the base and top circles, and two groups of four areas corresponding to the elevation and descending areas of the tappet, including the appropiate connections to compensate the thermal clearance.

Within the paper the symmetrical profile of the cam was considered, so that the laws determining the tappet upward motion in the four elevation areas are similar to these defining its downward motion.

The upward motion in the first area represents a cosine curve, has the size of a quarter of wave and corresponds to an interval marked as Φ_0, (Figure 1)

It provides the connection of the primal, enlarged cercle with the thermal clearance and the real elevation profile and is expressed by the

relation::

$$h_0 = h_\rho \left(1 - \cos\frac{\pi}{2\Phi_0}\alpha_0^*\right) \qquad (1)$$

The elevation in the second area represents half a sine curve and corresponds to an interval noted with Φ_1. It provides positive acceleration and is expressed by :

$$h_1 = h_\rho + c_{11}\alpha_1^* - c_{12}\sin\frac{\pi}{\Phi_1}\alpha_1^* \qquad (2)$$

The upward movement in the third area is a quarter of a sine curve, corresponds to an interval marked as Φ_2 and enables the transition from positive to negative acceleration. The mathematical relation of the tappet movement in this area is the following :

$$h_2 = h_{1f} + c_{21}\alpha_2^* + c_{22}\sin\frac{\pi}{\Phi_2}\alpha_2^* \tag{3}$$

The tappet elevation in the last area corresponds to an interval noted with Φ_3 and is defined by a fourth-degree polynom according to the relation :

$$h_3 = h_{2f} + c_{31}(\Phi_3 - \alpha_3^*)4 - c_{32}(\Phi_3 - \alpha_3^*)2 + c_{33} \tag{4}$$

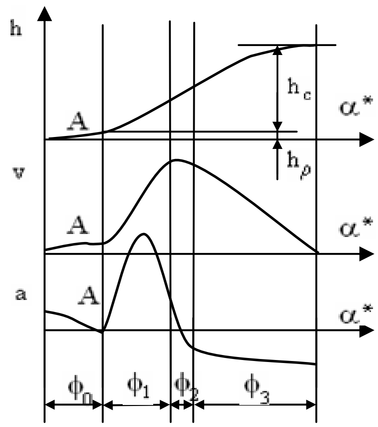


Figure1 Kinematic parameters of the cam-tappet mechanism for Kurz profiles

In order to determine the C_{jk} values, the determination by differentiation was required of the mathematical functions of the acceleration and speed levels during the tappet elevation [2], [4].

$$\tilde{v}_0 = h_\rho\frac{\pi}{\Phi_0}\sin\frac{\pi}{2\Phi_0}\alpha_0^* \tag{5}$$

$$\tilde{v}_1 = c_{11} - c_{12.}\ \frac{\pi}{\Phi_1}\cos\frac{\pi}{\Phi_1}\alpha_1^* \tag{6}$$

$$\tilde{v}_2 = c_{21} + c_{22}\frac{\pi}{2\Phi_2}\cos\frac{\pi}{2\Phi_2}\alpha_2^* \tag{7}$$

$$\tilde{v}_3 = -4c_{31}(\Phi_3 - \alpha_3^*)^3 + 2c_{32}(\Phi_3 - \alpha_3^*) \tag{8}$$

$$\tilde{a}_0 = h_\rho\left(\frac{\pi}{2\Phi_0}\right)^2\cos\frac{\pi}{2\Phi_0}\alpha_0^* \tag{9}$$

$$\tilde{a}_1 = c_{12}\left(\frac{\pi}{\Phi_1}\right)^2\sin\frac{\pi}{\Phi_1}\alpha_1^* \tag{10}$$

$$\tilde{a}_2 = -c_{22}\left(\frac{\pi}{2\Phi_2}\right)^2\sin\frac{\pi}{2\Phi_2}\alpha_2^* \tag{11}$$

$$\tilde{a}_3 = 12c_{31}(\Phi_3 - \alpha_3^*)^2 - 2c_{32} \tag{12}$$

where: - * represents the rotation angle of the camshaft

- $\omega^* = \pi n/60$ [rad/s]
- $\tilde{v}$ =corresponding speed $\qquad \tilde{v} = v/\omega^* = dh/d\alpha^*$ (13)
- $\tilde{a}$ =corresponding acceleration $\tilde{a} = a/\omega^{*2} = d\tilde{v}/d\alpha^* = d^2h/d\alpha^{*2}$ (14)

After applying the minimal condition which provide the functions continuity at the starting and final moment of the interval, a system of 6 equations are obtained [4] :

$$c_{11}\Phi_1 + c_{21}\Phi_2 + c_{22} + c_{33} - h_c = 0 \qquad (15)$$

$$c_{31}\Phi_3^4 - c_{32}\Phi_3^2 + c_{33} = 0 \qquad (16)$$

$$c_{11} - c_{12}\frac{\pi}{\Phi_1} - \tilde{v}_{0f} = 0 \qquad (17)$$

$$c_{11} + c_{12}\frac{\pi}{\Phi_1} - c_{21} - c_{22}\frac{\pi}{2\Phi_2} = 0 \qquad (18)$$

$$c_{21} + 4\,c_{31}\Phi_3^3 - 2c_{32}\Phi_3 = 0 \qquad (19)$$

$$c_{22}\left(\frac{\pi}{2\Phi_2}\right)^2 + 12c_{31}\Phi_3^2 - 2c_{32} = 0 \qquad (20)$$

where *i* and *f* values represent the starting and the final moments of the interval.

2 PROFILE SYNTHESIS OF THE ROTATING CAM WITH DISPLACING SHOE TAPPET

The designing of the cam mechanism must take into account issues regarding the tappet motion laws as well as the calculus of the optimal sizes, when trying to obtain the tolerable values of some functional parameters.

The motion laws are determined by the functioning conditions of the cam-tappet mechanism. The synthesis of the cam mechanisms involves the synthesis of the motion laws, the calculus of the base circle radius and the cam profile synthesis.

In case of a shoe tappet, the cam profile must be strictly convex, so that the shoe should be tangential to any of the profile points, during both the elevation and descending movements. The convexity condition of the cam profile-curve is that the curving centre C_1 should be on the same side of the curve, that is, the curving radius ρ_1 be strictly positive ($\rho_1 > 0$).

The base circle radius r_0 (Figure 2) is derived from the requirement that the curving radius $\rho_{A_1} = A_1C_1$ of the active profile Γ_1 should not change its sign [1].

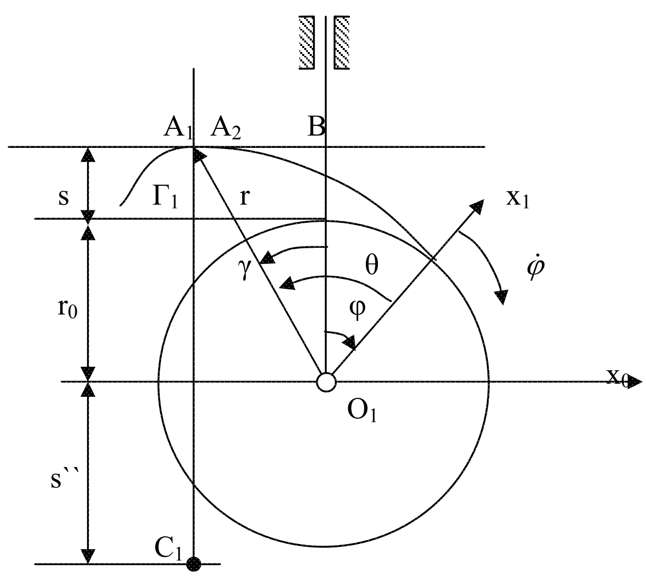


Figure 2 Kinematic chart of the rotating cam-displacing shoe tappet mechanism

Thus, the curve , representing the cam profile, must not have inflection points and this is expressed by the relation :

$$\rho_{A_1} = r_0 + s + s'' > 0 \qquad (21)$$

from which the following inequality can be deducted :

$$r_0 > -(s + s'') \qquad (22)$$

For s″ > 0 his requirement is met with no restriction. But for the negative values of acceleration(s″ > 0),the inequation (22) can be written [1] :

$$r_0 > \left| s''^{(-)}_{max} \right| - h \qquad (23)$$

because the acceleration maximum negative value corresponds to the maximum movement of the displacing shoe tappet $s_{max} = h$ (the tappet stroke).

In Figure 2, the cam is considered in momentary position, being rotated clockwise with angle $\dot{\varphi}$ and angular speed. Semi-axis O_1x_1 belongs to the rotating cam profile, passing through the connection point of the base circle with convex curve Γ_1.

The displacement direction of the tappet is perpendicular to the stationary guide path and its shoe meets the cam in point A, represented by 2 geometrically overlapped points (A_1 and A_2), but kinematically different (they have different speeds and accelerations).

The positions vector of point A_1 was marked by O_1A_1 = r, whichin the mobile system of coordinates $O_1x_1y_1$ is located by angle $x_1O_1A_1 = \theta$.

By way of the vectorial equation of the linear speeds [1],

$$\vec{v}_{A_2} = \vec{v}_{A_1} + \vec{v}_{A_2A_1} \qquad (24)$$

it is deducted that the variable segment AB corresponds to the tappet small speed, that is AB='s.

Thus, using the already calculated motion and speed functions s(φ), s'(φ), we can next calculate the length of vector radius r in point A1 within the right triangle O_1AB (Figure 2)

$$r = \sqrt{(r_0 + s)^2 + (s')^2} \qquad (25)$$

The other polar coordinate, angle θ , can be obtained depending on the cam rotation angle φ and angle y, representing the deviation of the vector radius r from the stationary guide path of the tappet:

$$\theta = \varphi + \gamma \qquad (26)$$

where the deviation angle can be calculated using the triangle O_1AB:

$$\gamma = \operatorname{arctg}\left(\frac{s'}{r_0 + s}\right) \qquad (27)$$

The carthesian coordinates of point A_1 in the mobile system fixed by the cam are the following :

$$x = r\cos\theta \quad ; \quad y = r\sin\theta. \qquad (28)$$

3 SOFTWARE PROGRAMME FOR RENDERING THE SHOCK-FREE CAM PROFILES OF THE CAMSHAFTS

The cam profiles were determined starting from the valve elevation laws of the distribution system. To create these laws a software programme was made up in Matlab, whose logical scheme is presented in Figure 3.

The data offered by this programme were used to calculate the polar and carthesian coordinates and to mark points on the cam profile. The descent and elevation profiles are considered symmetrical and their joining was achieved through the circular arcs corresponding to the cam base circle and top circle.

The values of the parameters that define each variant of elevation law were stored in Excel files named ''variante_sa'' for the intake valve and ''variante_se'' for the exhaust valve.

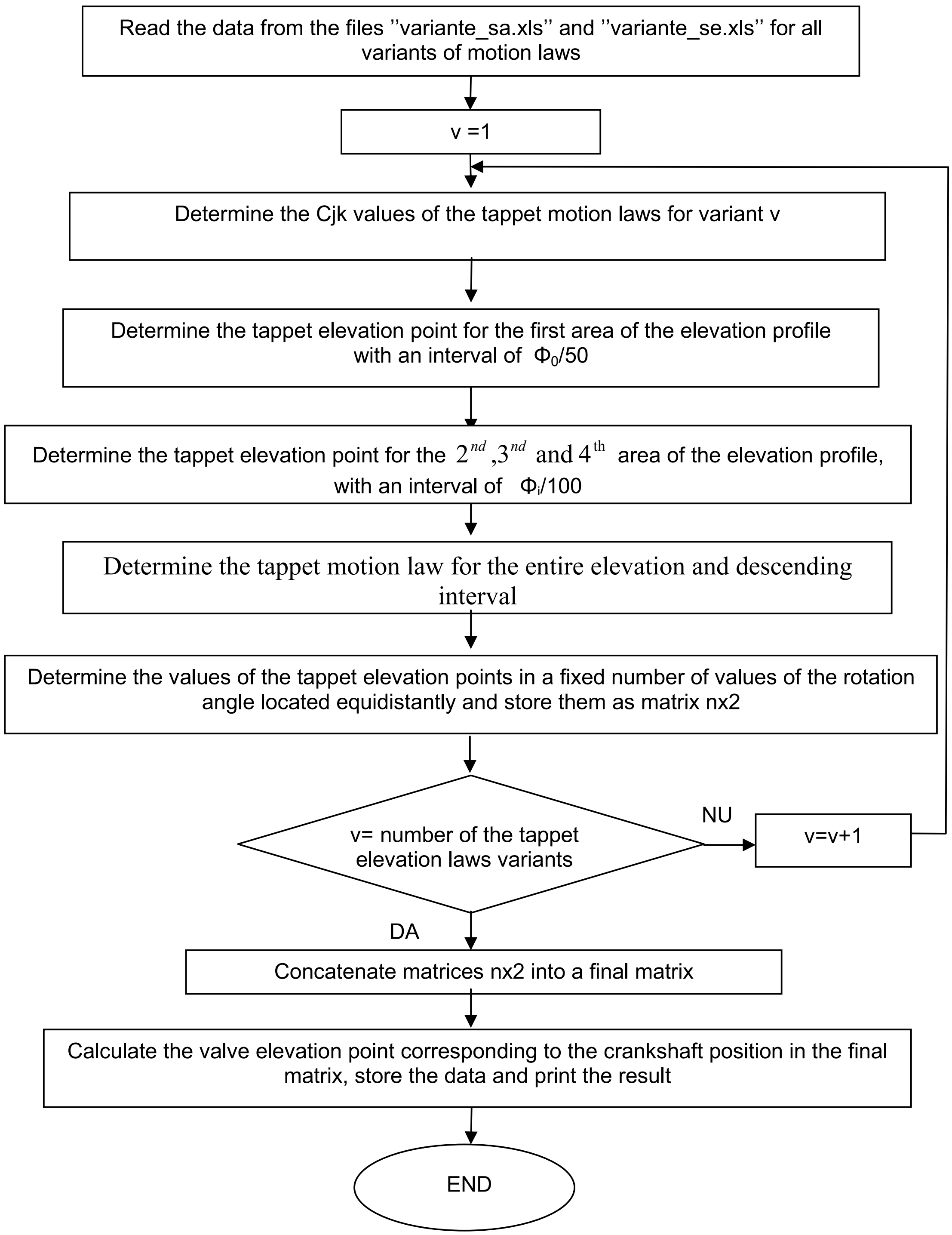
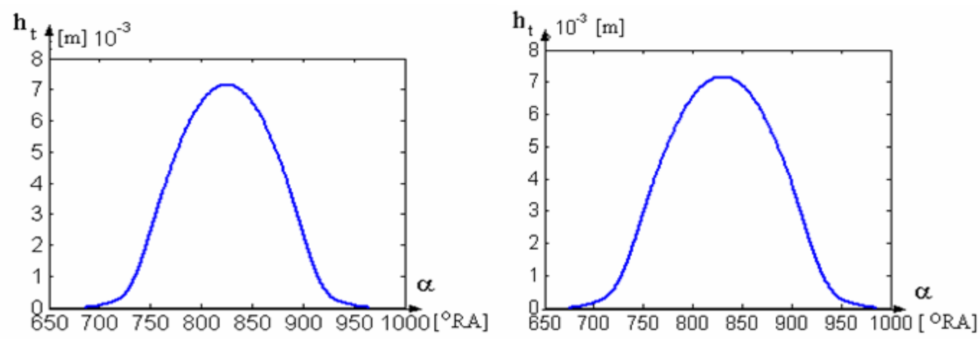


Fig. 3 Logical scheme of the ''Lege_sa.mat'' and ''Lege_se.mat'' programme for rendering the valve motion laws

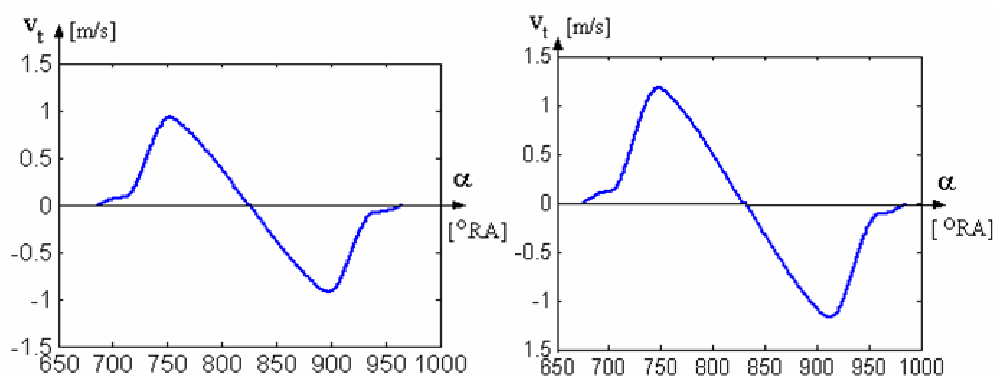
4 RESULTS AND CONCLUSIONS OBTAINED BY MODELING THE TAPPET MOTION LAWS AND RENDERING THE SHOCK-FREE CAM PROFILE FOR D118 ENGINE

The mathematical model was used for the D118 engine, at rotation speed of 1400/min and 2000/min, and the results are presented below:



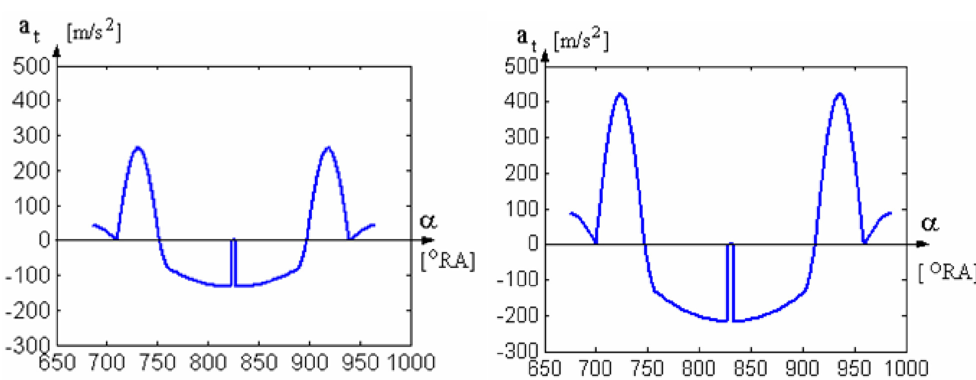
a) *b)*

Figura 4 Elevation law of the tappet which drives the intake valve:
a) at a rotation speed of 1400/min b) at a rotation speed of 2000/min



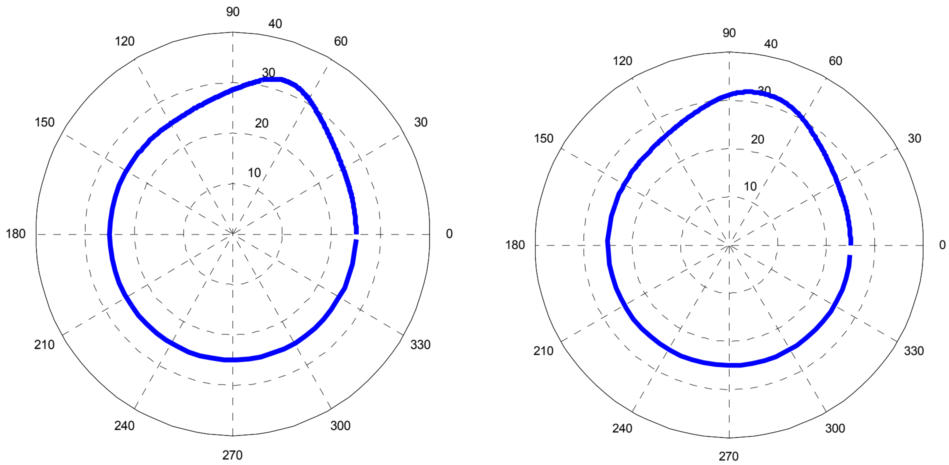
a) *b)*

Figura 5 Motion speed variation of the tappet which drives the intake valve
a) at a rotation speed of 1400/min b) at a rotation speed of 2000/min



a) *b)*

Figura 6 Acceleration variation of the tappet which drives the intake valve
a) at a rotation speed of 1400/min b) at a rotation speed of 2000/min



a) **b)**

Figura 7 Optimal profile of the cam which drives the intake valve tappet
a) at a rotation speed of 1400/min b) at a rotation speed of 2000/min

It is easily noticeable that the profiles of the rendered cams enable smooth acceleration and speed variation, without sudden jumps from positive to negative values. These profiles prevent the possible tappet detachment from the cam and ensure a long-lasting functioning without shocks or vibrations.

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