A KINEMATIC STUDY OF A MECHANICAL VARIABLE VALVE TIMING MECHANISM WITH CONTINUOUS VALVE LIFT

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ABSTRACT: An innovating solution for throttle-free load control for spark-ignition engines is variable valve timing system with continuous valve lift (VVTL System, or VVA - Variable Valve Actuation System). A variable valve timing mechanism which can provide continuous valve lift (VVL System) represents a good solution for internal combustion engines to become more efficient, with an improved dynamic performance, to generate fewer emissions by reducing fuel consumption and, furthermore, allows technologies like gasoline direct injection (GDI), homogeneous charge compression ignition (HCCI) / controlled auto ignition (CAI), etc. to perform an optimized operation.
In this paper is presented a kinematic analysis, using the analytic method, of a valve timing mechanism with continuous valve lift variation and constant duration. This type of mechanism ensures a continuous valve lift between two extreme valve heights. It is also presented a numerical example, in which the solving principle is based on a numerical method.

1. INTRODUCTION

For the traditional spark-ignition engines the timing configuration represents a compromise (1) which does not allow the best engine performance to be achieved for all regime and load. Infinitely variable inlet valve lift and timing is used to control engine load, reducing throttling losses and fuel consumption (2). It also improves low-end torque and transient response.
Variable valve timing (VVT) technology allows better engine performance by reducing fuel consumption and therefore low emissions (3), higher efficiency, highly precise responsiveness of the powertrain (4).
The key parameter for petrol engine combustion, and therefore efficiency, emissions and fuel consumption is the quantity and characteristics of the fresh air charge in the cylinders. In conventional petrol engines, the throttle-based air control wastes about 10% of the input energy in pumping the air (5). VVT systems with continuous valve lift variation, usually combined with cam phasers, are designed to eliminate the classical throttle and its inconvenience.

2. MECHANISM DESCRIPTION AND OPERATION

The proposed mechanism is a mechanical, continuously VVL timing system, Fig. 1, designed to ensure for the intake valves a continuous range of lifts comprised between a minimum lift which corresponds to the engine idling and a full lift for the maximum engine load. The mechanism's main idea belongs to Professor Vasile Dumitrescu who adapted a standard valve timing mechanism, which perform a unique valve lift, Fig. 2, into a mechanical variable valve timing mechanism with continuous variable valve lift. Therefore the classical roller rocker 2 was divided in two parts as follows: the oscillating roller cam follower 2 and the lever 7 which acts the engine valve, each one linked on the initial joint O.

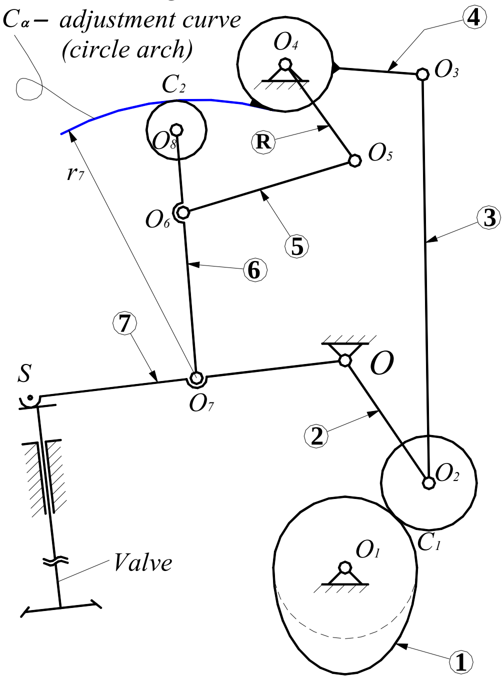

Fig. 1. The kinematic schema of the mechanism

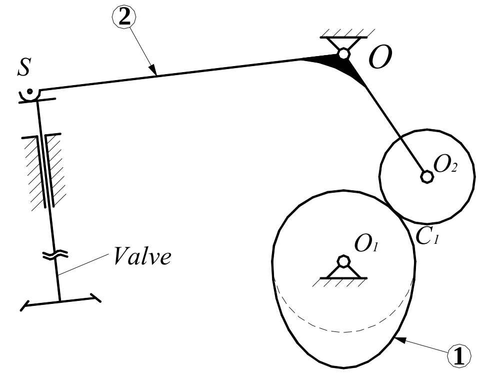

Fig. 2. The scheme of classical valve timing mechanism which underpins the Dumitrescu VVL mechanism

Further more, some elements were introduced: 3 – push rod, 4 – oscillating rocker jointed to base in O_4, 6 – push rod equipped with a roller in O_8 and linked to the lever 7 in O_7, 1 – cam. Hence the mechanism (Fig. 2) is planar, composed with $n = 7$ active elements, the adjustment element R being stationary (actually, its movement is controlled). The connection between elements is via Class IV couplings denoted with C_1, C_2 and via Class V couplings denoted O, O_1, O_2, …, O_7; as consequence $c_4 = 2$ and $c_5 = 9$. It results that the mechanism's mobility is unique (6).

$$M = 3n - c_4 - 2c_5 = 21 - 2 - 18 = 1. \qquad (2.1)$$

The adjustment of the valves height is made by positioning the push rod 6 via the adjustment element R wherewith is linked by the lever 5. The adjustment curve's shape is thus circular, being formed with a circle arch with the radius r_7, and the center in O_7 when the contact C_1 belongs to the cam's base circle.

The mechanism's minimum adjustment is made by rotating the adjustment element R until O_8 is collinear with O_4 and O_7. We mention that in the present configuration, the mechanism can not provide valves deactivation, maximum valve lift related to minimum adjustment being above zero.

The motion is transmitted by the camshaft to the oscillating roller cam follower 2, push rod 3, the rocker arm 4 which is equipped with an adjustment curve, C_α. Further on, the rocker arm acts on the push rod 6 which rotates the levers 5 and 7. Finally the movement of the lever arm 7 is transmitted to the engine valve.

3. THE KINEMATIC ANALYSIS

3.1. Kinematic aspects. Notations

In order to realize the mechanism's kinematic analysis, we define the main dimensions and nomenclature (Fig. 3):

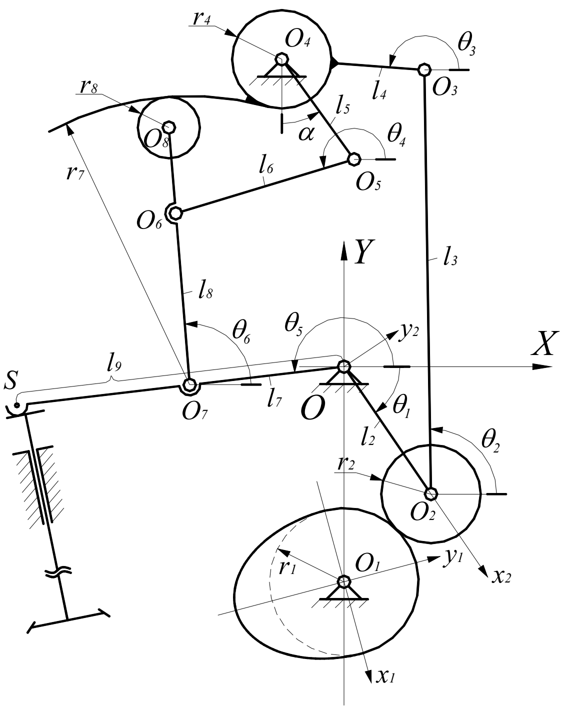

Fig. 3. Kinematic schema with nomenclature

- The general coordinate system OXY, and local coordinate systems $O_1x_1y_1$, $O_2x_2y_2$,
- The fixed points coordinates in OXY: $O_1(X_1, Y_1)$, $O_4(X_4, Y_4)$, and $O(0,0)$,
- The adjustment of valve height is made by angular positioning of the adjustment element R around the fixed joint O_4, the amount of rotation being quantified by angle α,
- r_1 is the radius of the cam base circle which rotates around $O_1(0, Y_1)$, therefore we impose $X_1 = 0$,
- r_2, r_4, r_7, r_8, the radii of the circles point-centered respectively in: O_2, O_4, O_7, O_8,
- The lengths denoted with: $l_2 = OO_2$, $l_3 = O_2O_3$, $l_4 = O_3O_4$, $l_5 = O_4O_5$, $l_6 = O_5O_6$, $l_7 = OO_7$, $l_8 = O_6O_7$, $l_9 = OS$,
- The angle θ_1 between OX axis and OO_2 segment, in clockwise direction,
- The angle θ_2 between OX axis and O_2O_3 segment,
- The angle θ_3 between OX axis and O_3O_4 segment,
- The angle θ_4 between OX axis and O_5O_6 segment,
- The angle θ_5 between OX axis and OO_7 segment,
- The angle θ_6 between OX axis and the line defined by O_7 and O_8,
- The angle α between OY axis and O_4O_5 segment,
- The angle φ between OX axis and O_1x_1 axis solidar with the cam.

Is an indication that the situation in which $\varphi = 0$, the mechanism is in the reference position, and notation data will be accompanied by the character *, resulting the angles $\varphi^* = 0$, θ_1^*, $\theta_2^*,\ldots,$ θ_6^*, respectively the coordinates of the mobile points $O_2(X_2^*, Y_2^*)$, $O_3(X_3^*, Y_3^*)$, $O_6(X_6^*, Y_6^*)$, $O_7(X_7^*, Y_7^*)$, $O_8(X_8^*, Y_8^*)$.

3.2. The reference position

We consider that the mechanism's reference position is when the cam rotation angle $\varphi = 0$. In this position the contact joint C_1 between the cam and the next element is made on the cam's base circle; actually, any situation in which the contact joint C_1 belongs to the cam's base circle is similar with the reference position. Therefore, writing elementary mathematical relations we get:

$$X_2^* = l_2 \cos\theta_1^* ;\ Y_2^* = -l_2 \sin\theta_1^* , \tag{3.1}$$

$$X_3^* = X_2^* + l_3 \cos\theta_2^* ;\ Y_3^* = Y_2^* + l_3 \sin\theta_2^* , \tag{3.2}$$

$$X_5 = X_4 + l_5 \sin\alpha ;\ Y_5 = Y_4 - l_5 \cos\alpha , \tag{3.3}$$

$$X_7^* = l_7 \cos\theta_5^* ;\ Y_7^* = l_7 \sin\theta_5^* , \tag{3.4}$$

$$X_S^* = l_9 \cos\theta_5^* ;\ Y_S^* = l_9 \sin\theta_5^* , \tag{3.5}$$

$$X_6^* = X_5 + l_6 \cos\theta_4^* ;\ Y_6^* = Y_5 + l_6 \sin\theta_4^* , \tag{3.6}$$

$$X_8^* = X_7 + (r_7 - r_8)\cos\theta_6^* ;\ Y_8^* = Y_7 + (r_7 - r_8)\sin\theta_6^* . \tag{3.7}$$

3.3. A certain position

In order to write the cam definition, let us consider that the cam's parametric equations in its local coordinate system $O_1x_1y_1$ are $[x_1(\xi_1), y_1(\xi_1)]$.

Also, we consider the local coordinate system Ox_2y_2 solidar with the oscillating roller cam follower 2 (OO_2 segment) so point O_2 belongs to the system abscissa, Fig. 3. The parametric equations of the circle point-centered in O_2, written in Ox_2y_2 are:

$$x_2 = l_2 + r_2 \cos\xi_2 ;\ y_2 = -r_2 \sin\xi_2 . \tag{3.8}$$

On the other hand, in the general reference system OXY, the contact joint between the cam and the O_2 point centered circle is written:

$$x_2 \cos\theta_1 + y_2 \sin\theta_1 - x_1 \cos\varphi + y_1 \sin\varphi = 0 , \tag{3.9}$$

$$- x_2 \sin\theta_1 + y_2 \cos\theta_1 - Y_1 - x_1 \sin\varphi - y_1 \cos\varphi = 0 . \tag{3.10}$$

Having the same tangent, we deduce:

$$\frac{x_{1p}\cos\varphi - y_{1p}\sin\varphi}{x_{1p}\sin\varphi + y_{1p}\cos\varphi} = \frac{\lambda(x_{2p}\cos\theta_1 + y_{2p}\sin\theta_1)}{\lambda(x_{2p}\sin\theta_1 + y_{2p}\cos\theta_1)}, \tag{3.11}$$

wherefrom

$$(x_{1p}x_{2p} + y_{1p}y_{2p})\sin(\varphi + \theta_1) + (y_{1p}x_{2p} - x_{1p}y_{2p})\cos(\varphi + \theta_1) = 0 , \tag{3.12}$$

wherein x_{1p}, y_{1p} are the derivatives of x_1, y_1 with respect to the cam rotation angle φ, and x_{2p}, y_{2p} are the derivatives of x_2, y_2 with respect to the ξ_2 angle.

For the reference position:

$$\xi_1^* = \frac{\pi}{2} - \cos^{-1}\frac{Y_1^2 + (r_1 + r_2)^2 - l_2^2}{2|Y_1|(r_1 + r_2)} , \tag{3.13}$$

$$\xi_2^* = \pi - \cos^{-1}\frac{l_2^2 + (r_1 + r_2)^2 - Y_1^2}{2l_2(r_1 + r_2)}, \tag{3.14}$$

$$\theta_1^* = \frac{\pi}{2} - \cos^{-1}\frac{l_2^2 + Y_1^2 - (r_1 + r_2)^2}{2l_2|Y_1|}. \tag{3.15}$$

Further on, the position of the point O_3 is given by

$$X_3 = l_2\cos\theta_1 + l_3\cos\theta_2;\ Y_3 = l_2\sin\theta_1 + l_3\sin\theta_2 \tag{3.16}$$

and

$$X_3 = X_4 + l_4\cos\theta_3;\ Y_3 = Y_4 + l_4\sin\theta_3, \tag{3.17}$$

hence

$$l_2\cos\theta_1 + l_3\cos\theta_2 - X_4 - l_4\cos\theta_3 = 0, \tag{3.18}$$

$$-l_2\sin\theta_1 + l_3\sin\theta_2 - Y_4 - l_4\sin\theta_3 = 0. \tag{3.19}$$

For the reference position:

$$\theta_2^* = \frac{\pi}{2} - \sin^{-1}\frac{X_3 - X_2}{l_3}, \tag{3.20}$$

$$\theta_3^* = \sin^{-1}\frac{Y_3 - Y_4}{l_4}. \tag{3.21}$$

The positions of the points O_6, O_8, O_7 expressed in the general reference system OXY are:

$$X_6 = X_5 + l_6\cos\theta_4;\ Y_6 = Y_5 + l_6\sin\theta_4, \tag{3.22}$$

$$\begin{aligned} X_8 &= X_4 + (r_4 + r_7)\cos(\gamma - \theta_3^* + \theta_3) + (r_7 - r_8)\cos\theta_6; \\ Y_8 &= Y_4 + (r_4 + r_7)\sin(\gamma - \theta_3^* + \theta_3) + (r_7 - r_8)\sin\theta_6, \end{aligned} \tag{3.23}$$

$$X_7 = l_7\cos\theta_5;\ Y_7 = l_7\sin\theta_5 \tag{3.24}$$

wherein

$$\gamma = \frac{3\pi}{2} - \sin^{-1}\frac{X_4 - X_7}{r_4 + r_7} \tag{3.25}$$

and expressing the lengths O_7O_8, O_6O_8, O_6O_7, it results

$$(X_8 - X_7)^2 + (Y_8 - Y_7)^2 - (r_7 - r_8)^2 = 0, \tag{3.26}$$

$$(X_8 - X_6)^2 + (Y_8 - Y_6)^2 - (r_7 - r_8 - l_8)^2 = 0, \tag{3.27}$$

$$(X_7 - X_6)^2 + (Y_7 - Y_6)^2 - l_8^2 = 0. \tag{3.28}$$

In the reference position we get:

$$\theta_4^* = \pi + \tan^{-1}\frac{Y_5 - Y_7}{X_5 - X_7} - \cos^{-1}\frac{l_6^2 + (X_7 - X_5)^2 + (Y_7 - Y_5)^2 - l_8^2}{2l_6\sqrt{(X_7 - X_5)^2 + (Y_7 - Y_5)^2}}, \tag{3.29}$$

$$\theta_5^* = \pi + \tan^{-1}\frac{Y_4}{X_4} + \cos^{-1}\frac{l_7^2 + X_4^2 + Y_4^2 - (r_4 + r_7)^2}{2l_7\sqrt{X_4^2 + Y_4^2}}, \tag{3.30}$$

$$\theta_6^* = \frac{\pi}{2} - \sin^{-1}\frac{X_6 - X_7}{l_8}. \tag{3.31}$$

Solving the equations (3.9), (3.10), (3.12), (3.18), (3.19), (3.26), (3.27) and (3.28) in which the unknown parameters are ξ_1, ξ_2, θ_1, θ_2, θ_3, θ_4, θ_5, θ_6, for $\varphi = 1 \div 360°$, it results the movement of the mechanism.

3.4. Valve displacement

Knowing the variation of θ_5 angle for the entire kinematic cycle, the valve lift function is found using the relation (3.32), Fig. 4:

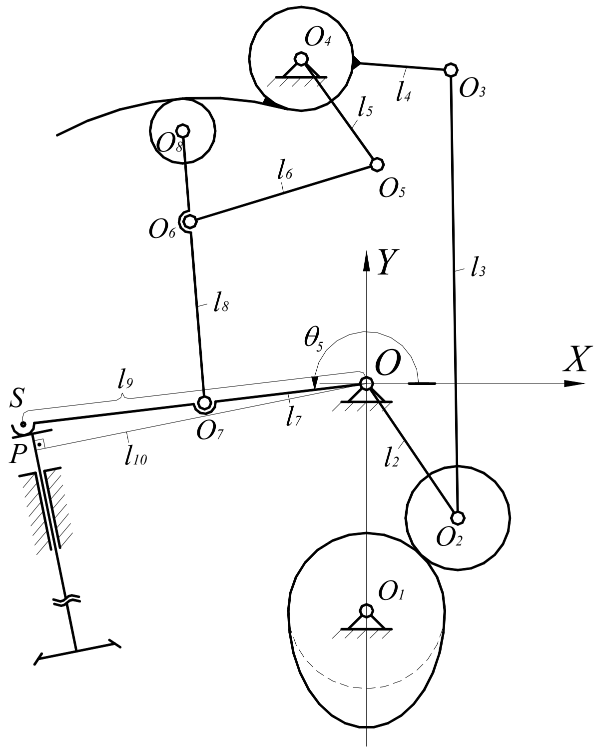

Fig. 4. The valve displacement schema

$$s(\varphi) = l_9\left[\sin\psi - \sin\left(\psi - \theta_5 + \theta_5^*\right)\right], \tag{3.32}$$

where

$$\psi = \cos^{-1}\frac{l_{10}}{l_9}. \tag{3.33}$$

4. IMPLEMENTATION AND RESULTS

Let us consider the approximate dimensions used to validate the computational model, as follows: $\alpha = 18 \div 62.2°$, $O_1(0,-35)\,\text{mm}$, $O_4(-10, 50)\,\text{mm}$, and the constant parameters given by the values: $l_2 = OO_2 = 25\,\text{mm}$, $l_3 = O_2O_3 = 69\,\text{mm}$, $l_4 = O_3O_4 = 23\,\text{mm}$, $l_5 = O_4O_5 = 20\,\text{mm}$, $l_6 = O_5O_6 = 30\,\text{mm}$, $l_7 = OO_7 = 25\,\text{mm}$, $l_8 = O_6O_7 = 28\,\text{mm}$, $l_9 = OS = 53\,\text{mm}$, $l_{10} = OP = 52.8344\,\text{mm}$, $r_1 = 12\,\text{mm}$, $r_2 = 8\,\text{mm}$, $r_4 = 8\,\text{mm}$, $r_7 = 47\,\text{mm}$, $r_8 = 5\,\text{mm}$. The final dimensions of the mechanism will be established after the mechanism optimization procedure.

For the mechanism's reference position and $\alpha = 18°$ we obtain: $\xi_1 = 45.585°$, $\xi_2 = 78.463°$, $\theta_1 = 55.952°$, $\theta_2 = 90.882°$, $\theta_3 = -4.295°$, $\theta_4 = 193.789°$, $\theta_5 = 186.811°$, $\theta_6 = 106.882°$, $\gamma = 254.364°$.

In order to realize the mechanism kinematic simulation, we consider the cam is half circle half ellipse, Fig. 5, its equation being written in $O_1x_1y_1$, as follows:

$$\begin{cases} x_1 = r_1 \cos \xi_1 \\ y_1 = \bar{r}_1 \sin \xi_1 \end{cases}, \tag{3.34}$$

where $\bar{r}_1 = \begin{cases} r_1, \text{ for } \xi_1 \in [0, \pi] \\ 1.5 r_1, \text{ for } \xi_1 \in (\pi, 2\pi) \end{cases}$.

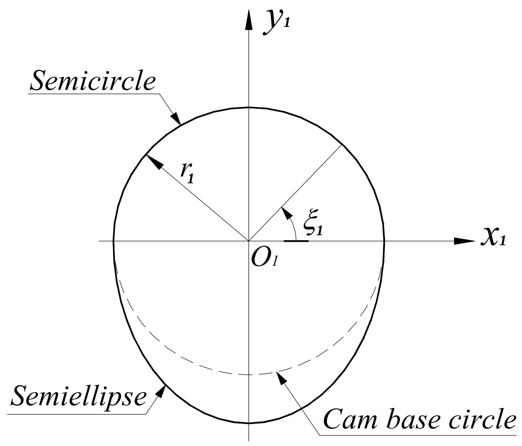

Fig. 5. Semi ellipse cam design

Solving the equations (3.9), (3.10), (3.12), (3.18), (3.19), (3.26), (3.27), (3.28) requires a numerical computing method. Therefore we use the Newton-Raphson numerical method, in which the initial approximation results from the reference position of the mechanism, i.e. for $\varphi = 0$. The obtained results represents the initial approximation for $\varphi = 1°$. The procedure is repeated until $\varphi = 360°$. By modifying the adjustment angle α within the mentioned range the mechanism performs different valve heights.

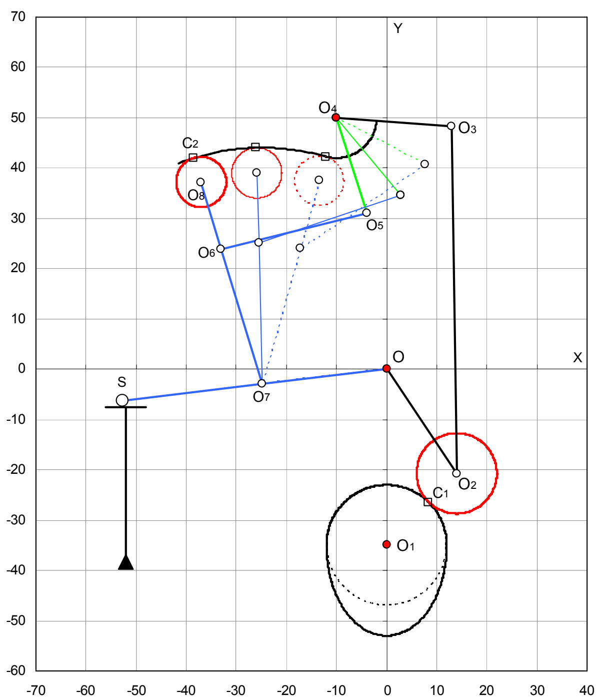

Fig. 6. The adjustment of mechanism in the reference position ($\varphi = 0$)

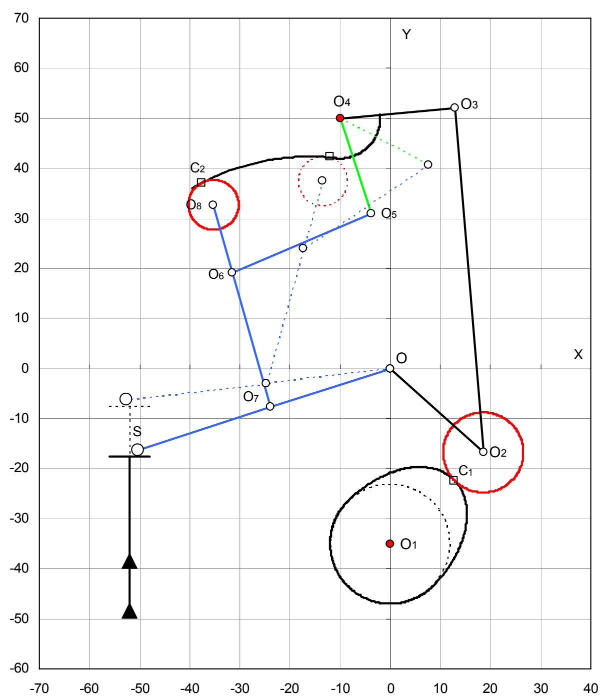

Fig. 7. The extreme adjustment of mechanism in a certain position ($\varphi = 135°$)

In Fig. 6 is represented the mechanism adjustment procedure. It is easy to observe that in this case, when the contact C_1 belongs to cam base circle, any variation of the adjustment angle α does not affect the valve position, i.e. the valve remains closed, and in the same time contacts C_1 and C_2 are maintained. This fact is due to the shape of the adjustment curve C_α, which is a circular shape.
In Fig. 7 is represented the mechanism in a certain position, when the contact C_1 belongs to cam profile. In this case, by varying the adjustment angle α it results different valve heights (in this diagram is represented the extreme adjustment for $\alpha = 18°$ when the mechanism performs maximum valve lift, and $\alpha = 62{,}2°$ when the mechanism performs minimum valve lift).

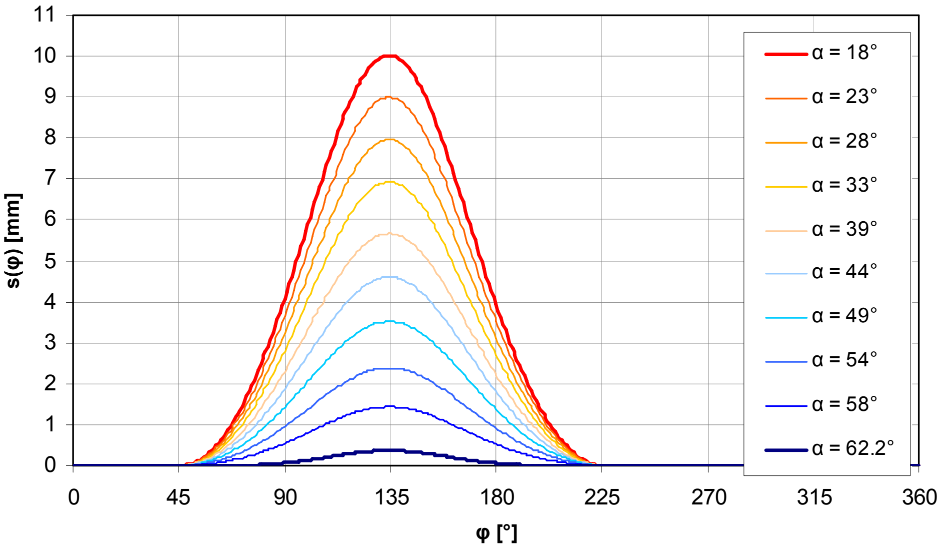

Fig. 8. The valve lift family of laws obtained with a semi-ellipse cam

If we maintain a fixed adjustment angle and vary the cam rotational angle φ we get a valve lift law. In Fig. 8 is designed the family of valve lift laws which the mechanism can generate, with respect to relation (3.32). The maximum valve lift is around $10\,\text{mm}$ and the minimum valve lift is $0.35\,\text{mm}$. The mechanism performs continuous valve lift, so any valve lift can be achieved in this range.
Also, the valve opening and closing points remain unchanged during the adjustment variation, which means that the duration of effective valve opening remains the same. Therefore this mechanism is a Constant Duration Variable Valve Actuation (CDVVA), according to reference (7).

5. CONCLUSIONS

After this study on the variable valve lift mechanism with continuous valve lift the following conclusions were reached:
– the valve's position must remain unchanged during the rotation of the adjustment lever. This condition must be respected by all the VVL mechanism, no matter the type of the valve lift mechanism or the way the lift is set;
– the kinematic analysis shows that the mechanism offers a continuous variation of the valve lift, with valve displacement starting from around $0.4\,\text{mm}$ lift to a maximum lift around

$10\,\mathrm{mm}$, without changing the valve opening and closing points. The minimum valve lift must be set on a precise value, around $0.4 \div 0.5$ millimeters, correlated with the engine idling.
– a disadvantage of this mechanism is that it does not allow zero valve lift. This disparity will be removed after a complete mechanism optimization, in which the valve displacement at minimum adjustment will be one of the optimization criteria.
The future work will be concentrated on the mechanism's cam synthesis and optimization.

ACKNOWLEDGEMENT
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