Determination of the Internal Combustion Engine Valve Gear Characteristics

Alexander Vasilyev¹ and Yulia Bakhracheva ²

¹Department of heat engineering and hydraulics, Volgograd State Technical University, Prosp. Lenin, 28, Volgograd, 400005, Russian Federation
²Department of Information Security, Priority Technologies Institute, Volgograd State University, Prosp. Universitetskyy, 100, Volgograd, 400062, Russian Federation

E-mail: vasilyev@vstu.ru, bakhracheva@volsu.ru

Abstract. Method of the Internal Combustion Engine (ICE) valve gear characteristics determination by using the numerical differentiation of tappet movement law is submitted. The method of converting these characteristics for different valve types is proposed. Based on the methods of calculation of the motion law of the follower, characteristics of the timing and conversion of the data during the transition from one kinematics to another developed computer software included in the software complex for synthesis of the characteristics of the valve timing. Their use allows to identify resources for improving timing mechanism of an internal combustion engine.

1. Introduction
Availability, reliability and efficiency of the mechanism of gas-distribution (MG) of internal combustion engines greatly depend on the kinematic characteristics [1 - 5]. The movement of the link mechanism is set by the cam profile, which determines the law of motion of the follower. Therefore, for calculation of numerous characteristics of the cam-follower (strength, contact-hydrodynamic, tribology, etc.), as well as the laws of motion of all slave units need information about the laws of displacement, velocity and acceleration of the follower on the cam rotation angle [6, 7].

In the most favorable case, a known method of profiling the cam and all the source data defining configuration profiles and the law of motion of the follower, including the above kinematic characteristics of the displacement, velocity and acceleration. If only the dependence of the acceleration of the follower on the cam rotation angle can be used as the initial data. In this case, the required characteristics of speed and acceleration of the follower can be determined by numerical differentiation of tabular predetermined law of movement of the pusher on the cam rotation angle, which causes some scatter of data due to insufficient stability of the procedure of the numerical differentiation. However, this calculation is only possible in the case of calculating the characteristics of the real cam, the configuration of which is
determined experimentally and tabulated for a given law of movement of the pusher shown in the drawing of the camshaft.

It is also important to note that while calculating the kinematic and other characteristics of the pairing of the cam-follower and valve gear in General is important to consider the type of the used pusher and the kinematic scheme of the mechanism. In this regard, it is often necessary to do the recalculation of characteristics, as the law of movement of the pusher can be defined in one scheme (for example, when the forward movement of the pusher with flat contact surface), and in the timing mechanism of an internal combustion engine implements a different kinematic scheme (for example, the same cam works in conjunction with lever pusher having a cylindrical contact surface). The recalculation of characteristics is also necessary for the manufacture and control of the cam profile, as the scheme of the technological process of processing of the cam profile, as a rule, also different from the kinematic scheme applied in the gas distribution mechanism of internal combustion engines.

2. Mathematical model

In this regard, in the course of the present study was tasked with developing mathematical models, methods and relevant elements of the software system for computers that allows you to calculate the required characteristics of the timing mechanism of an internal combustion engine. And to perform these conversions for different kinematic schemes.

To characterize the velocity and acceleration of the follower (or their analogues) based on the table given the values of the movement used the technique of double numerical differentiation with smoothing. It should be noted that the step angle of the cam driven original table of values of displacement of the pusher is often too large for numerical differentiation. Therefore, the first step is an interpolation function using an interpolation process of the Aitken-Lagrange [8]. At the same time, the subroutine calculates the value of $Y$ uniquely the function $y(x)$ for a given value $X$ of its argument on a given table $x_p, y_j$ ($j = 1, 2, ..., n$) the corresponding values of argument and function. In this case, the step, the values $X$ and $Y$ get can be quite small.

The first approximation $Y_1$ of the calculated value $Y$ is the value of the function to be interpolated in $X$ to the nearest node interpolation $x_i$, i.e., $Y_1 = y(x_i) = y_i$. As more accurate approximations are used sequentially computed values of $L_0(X)$ polynomials of Lagrange, the degree of which increases at each new step of the interpolation. The calculations are performed by a known interpolating triangular scheme of Aitken. The interpolation process is terminated when performing, as a rule, the following conditions: the absolute value of the difference between two successive interpolated values is less than the specified $\varepsilon$, that is achieved by the specified accuracy of the iteration.

The selection nearest to a given $X$ value of the argument (on the cam rotation angle) and function (movement of the pusher) from the source table is based on the use of the sub-ordering of the arrays. For a given value of $X$ the function argument $y(x)$ and a given table $x_p, y_j$ ($j = 1, 2, ..., n$) values of a function and its argument are formed arrays of values $x_i, y_i$ ($i = 1, 2, ..., n$), are arranged in such a way that the modules of the difference $|X - x_i|$ increased with the growth of index $i$:

$$|X - x_i| \geq |X - x_k|, \text{ if } i > k. \quad (1)$$

The next step of calculating the characteristics of the velocity and acceleration of the follower is the differentiation of the functions of its displacement, given a table of values at equally spaced points according to equations of Lagrange with five nodes. Used by the routine allows for a function given a set of $y_1, y_2, ..., y_m$ of its values $y_k$ c step change in the argument (on the cam rotation angle) $h = x_{k+1} - x_k$ ($k = 2, ..., m$), to calculate the set $z_1, z_2, ..., z_m$ values of derivatives in the given $m$ points. To calculate values of derivatives used the Lagrange interpolating polynomial of the fourth degree, built on five successive points $x_{k+p}, y_{k+p}$ ($p = -2, -1, 0, 1, 2$). The required values of the derivative are calculated by the known formulas [8].

As already mentioned, the operation of numerical differentiation is quite stable and leads to some scatter results. In this regard, as shown by numerous calculations, it is advisable after each procedure of numerical differentiation to carry out a two-fold smoothing. For this purpose, a special
routine smoothing function given a table of values at equally spaced points, with the first-degree polynomial constructed on three consecutive points by the method of least squares. This subroutine computes a set of \( y_1, y_2, \ldots, y_m \) smoothed values of the function \( z(x) \), given many \( z_1, z_2, \ldots, z_m \) of its values at \( m \) equally spaced points with step \( h = x_k - x_{k-1} \) \((k = 2, \ldots, m)\).

Thus, applying the developed methods and software package, it is possible to calculate the characteristics of the velocity and acceleration of the follower (or their analogues) on the basis of the table specified values of its movement. However, the results obtained can be further used to calculate other characteristics of the timing mechanism only when the measured displacement of the follower produced by the same kinematic scheme, which is used in a real test the mechanism. Otherwise, as already noted, there is a need of recalculating the values of displacement, velocity and acceleration of the follower depending on on the cam rotation angle to conform the pattern and size of engine timing.

**Figure 1.** Schematic of cam mechanisms with flat (a), a centrally located cylindrical (b) tappets and a lever having a cylindrical surface of the pusher (c).
Let the scheme of the mechanism corresponds to Fig. 1.(c), and the law of motion of the follower for the cam profile is set depending on the angle $\beta$ according to the scheme in Fig. 1.(b). Then the analogues of velocity $\frac{dR}{d\phi}$ and acceleration $\frac{d^2R}{d\phi^2}$ pusher can be calculated by the formulas:

$$\frac{dR}{d\phi} = \frac{dR}{d\beta} \frac{d\beta}{d\phi}$$

(2)

$$\frac{d^2R}{d\phi^2} = \frac{d^2R}{d\beta^2} \left( \frac{d\beta}{d\phi} \right)^2 + \frac{dR}{d\beta} \frac{d^2\beta}{d\phi^2}$$

(3)

Differentiating the ratio

$$\varphi = \beta \mp (\arccos B_0 - \arccos B), \text{ where}$$

(4)

$$B = \frac{A^2 - R_i^2 + R_t^2}{2AR}; \quad B_0 = \frac{A^2 - R_i^2 + (R_0 + R_i)^2}{2A(R_0 + R_i)}$$

(5)

As a result, will receive

$$\frac{d\beta}{d\phi} = \frac{1}{1 \mp \frac{dB}{d\beta} \sqrt{1 - B^2}}$$

(6)

$$\frac{d^2\beta}{d\phi^2} = \pm \frac{\frac{d\beta}{d\phi} \left[ \frac{d^2B}{d\phi^2} \sqrt{1 - B^2} + \left( \frac{dB}{d\beta} \right)^2 \frac{B}{\sqrt{1 - B^2}} \right]}{\left( \sqrt{1 - B^2} \mp \frac{dB}{d\beta} \right)^2}$$

(7)

In these formulas the upper sign corresponds to a positive sign $\xi$, and the lower - negative. Thus, substituting (6) and (7) into (2) and (3), we obtain the relations for computing $\frac{dR}{d\phi}$ and $\frac{d^2R}{d\phi^2}$ in terms of the law of motion of the follower, given by the diagram on Fig. 1(b), for the circuit in Fig. 1.(c). The developed software package also provides recalculations for other schemes. For example, participants discussed common case, when the scheme of the mechanism corresponds to Fig. 1.(c), but the movement of the pusher and its derivatives for the cam profile defined by the corner $\alpha$ according to Fig. 1.(a).

3. Discussion

Figure 2 illustrates the differences in the characteristics of the same symmetrical profile of the intake cam tractor engine with the tappets of different types: centrally located (Fig. 1.b) and lever (Fig. 1.C) having a cylindrical contact surface. In the latter case, a clockwise direction of the cam rotation is selected when using the lever pusher engine. The maximum value of the analog acceleration on the side of the lift tappet is 57.3 mm/rad$^2$, and on the side of lowering – 62.1 mm/rad$^2$, which is 8.4% more.
Figure 2. Analogue acceleration of the follower on the cam rotation angle for symmetric kinematic schemes of the lever (1) and centrally located (2) cylindrical tappets

4. Conclusion
Based on the submitted methods of calculation of the follower motion law, timing characteristics and data recalculation for other kinematics schemes computer software is developed and included in the software complex for synthesis of the valve timing characteristics. Their use allows to identify resources for improving timing mechanism of an internal combustion engine.

5. References
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