MATHEMATICAL MODELING OF THE BREAKTHROUGH CAM PROFILE OF THE ENGINE GAS DISTRIBUTION MECHANISM

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Abstract. The equation of movement of a flap with allowance for of elastic deformations of a valve actuator permitting is maneuvered to calculate a profile of a shock-free cam supplying a desired law of movement of a flap. It ensures maximal filling up of the barrel and conservation of the marked phases of a valve timing, does not enable oscillating a flap, the breakoff it from a saddle on landing and does not enable a undue wear of parts of a valve operating mechanism.

Key words: gas distribution mechanism, camshaft, computer-aided design, valve, Cam profile section, deformation

Recently, methods of designing shockless Cams with elastic deformations of gas distribution mechanism parts have become more and more common. One of these methods, often called the "polydine" method, is summarized below.

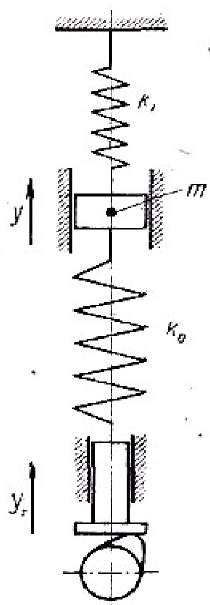

Fig. 1. Simplified system of gas distribution mechanism

When calculating the actual system of the gas distribution mechanism, we replace it with a simplified system equivalent to it with respect to elastic vibrations [1]. The scheme of a simplified single-mass system consisting of four parts is shown in Fig. 1. Where k_1 — is the stiffness of the valve spring (springs), kg/sm (N/m); k_0 — is the total stiffness of the parts of the valve timing mechanism, kg/ cm (H/м), determined from the condition of equality of the potential energy of the elastic forces of the actual and reduced mechanisms; m — given to the axis of the valve equivalent mass of the valve train components (valves, rocker arms and rods) kg·sek^2/sm (kg), determined by the condition of conservation of kinetic energy of oscillations of parts of the mechanism of the system; y — the actual path of the valve, sm (m); $y_т$ — the path of the pusher, sm (m); $y_0=i_{кор}\cdot y_т$ — way valve in the assumption that the elements of the system have absolute rigidity; $i_{кор}=l_{кл}/l_т$ — ratio rocker arms.

Frequency (1/сек) of natural oscillations of the reduced single-mass system

$$\mu = \sqrt{\frac{k_0 + k_1}{m}} \qquad (1)$$

If the stiffness of the camshaft and rocker support, as usually happens in practice, is so great that the deformations of these parts can be ignored, then the values m and μ can be determined by the formulas:

$$\left.\begin{aligned} & m = m_{кл} + \frac{1}{3}\cdot m_{пр} + \frac{J_{кор}}{l_{кл}^2} + \frac{1}{3}\cdot\frac{m_{шт}}{\left(i_{кор} + \dfrac{k_{шт}}{i_{кор}\cdot k_{кор}}\right)^2}; \\ & \frac{1}{k_0} = \frac{i_{кор}^2}{k_{шт}} + \frac{1}{k_{кор}}; \\ & \mu = \sqrt{\frac{1}{\dfrac{i_{кор}^2}{k_{шт}} + \dfrac{1}{k_{кор}}}\cdot\frac{1}{m_{кл} + \dfrac{m_{пр}}{3} + \dfrac{J_{кор}}{l_{кл}^2} + \dfrac{m_{шт}}{3\left[i_{кор} + k_{шт}/(i_{кор}\cdot k_{кор})\right]^2}}} \end{aligned}\right\}. \tag{2}$$

here $m_{кл}$ — the mass of the valve and the details of the valve spring attachment; $i_{кор}$ — the moment of inertia of the rocker relative to its axis of rotation; $m_{пр}$ — the mass of the valve spring; $k_{кор}$ — the rigidity of the rocker; $m_{шт}$, $k_{шт}$ — the mass and stiffness of the rod. Rod stiffness (kG/sm), made of a single material,

$$k_{шт} = E\cdot\frac{f_{шт}}{l_{шт}} \tag{3}$$

($k_{шт}$ is expressed in N/m, if E — in N/m^2, $l_{шт}$ — is in т and $f_{шт}$ — m^2),
here E — is the elastic modulus of the first kind of rod material, kG/sm ^{2}; $f_{шт}$, $l_{шт}$ — cross-sectional area of the rod, sm^2, and its length, sm.

The equilibrium condition for the forces applied to the equivalent system has the form

$$P_{пр} + P'_j + P_{упр} + P_г = 0 \tag{4}$$

here $P_{пр} = P_{пр1} + k_1\cdot y$ — the force developed by the valve spring (s) when lifting the valve (equivalent mass) on y (sm), kg; $P_{пр}$ — force preliminary tightening of the spring; $P'_j = m(d^2y/dt^2)$ – the force of inertia is equivalent to mass; $P_{упр} = k_0\cdot(y_0 - y)$— the force that occurs when the drive parts are deformed; $P_г = p_ц\cdot f_{кл.ц.} - p_п\cdot f_{кл.п.}$ — force of gases acting on the valve; $p_ц$, $p_п$ — gas pressure in the cylinder and valve branch, kg/cm^2; f кл.ц., f кл.п. — the area of the valve plate on the cylinder side and on the side of the valve branch, cm^2. From equation (4), taking into account the direction of forces and neglecting the gas pressure force to simplify conclusions, we obtain:

$$m\frac{d^2y}{dt^2} = -P_{пр1} - k_1\cdot y - k_0\cdot y + k_0\cdot y_0,$$

from this

$$y_m = \frac{l_m}{l_{кл}}\cdot y_0 = \frac{l_m}{l_{кл}}\cdot\left(\frac{P_{пр1}}{k_0} + \frac{k_0 + k_1}{k_0}\cdot y + \frac{m}{k_0}\cdot\frac{d^2y}{dt^2}\right). \tag{5}$$

Equation (5), derived taking into account the elastic deformations of the valve drive and linking the movement of the tappet with the movement of the valve, allows you to calculate the profile of the shockless cam, which provides the desired diagram of the valve rises [3, 4].

Changes in the path, speed and acceleration of the valve and pushrod in computer-aided design are more convenient to Express not depending on time, but depending on the angle of rotation of the camshaft.

Based on the ratios

$$x = \frac{360°}{60}\cdot n_{р.в.}\cdot t,\quad d_x = 6\cdot n_{р.в.}dt,$$

$$\frac{dy}{dt} = 6\cdot n_{р.в.}\frac{dy}{dx},\quad \frac{d^2y}{dt} = 36\cdot n_{р.в.}^2\frac{d^2y}{dx},$$

here $n_{р.в.}$ — the number of revolutions of the camshaft per minute, we get the path (sm)

$$y_m = \frac{l_m}{l_{кл}} \cdot \left(\frac{P_{пр1}}{k_0} + \frac{k_0 + k_1}{k_0} \cdot y + \frac{m}{k_0} \cdot 36 \cdot n^2_{р.в.} \frac{d^2 y}{dx^2} \right); \qquad (6)$$

speed (sm/deg)

$$w_m = \frac{l_m}{l_{кл}} \cdot \left(\frac{k_0 + k_1}{k_0} \cdot \frac{dy}{dx} + \frac{m}{k_0} \cdot 36 \cdot n^2_{р.в.} \frac{d^3 y}{dx^3} \right); \qquad (7)$$

valve acceleration (sm/deg^2)

$$j_m = \frac{l_m}{l_{кл}} \cdot \left(\frac{k_0 + k_1}{k_0} \cdot \frac{d^2 y}{dx^2} + \frac{m}{k_0} \cdot 36 \cdot n^2_{р.в.} \frac{d^4 y}{dx^4} \right). \qquad (8)$$

To determine the values of y_T, w_T, j_T it must first set the law of movement of the valve $y = f(x)$.

In order to obtain a smooth change in the path, speed, and acceleration of the valve, the first four derived functions must be continuous functions.

The y = f (x) function must also ensure that the cylinders are filled to the maximum and that the intended timing phases are maintained, that the valve does not oscillate or detach from the seat during landing, that the valve springs do not oscillate to a minimum, and that the gas distribution mechanism does not run noisy or wear out excessively.

Table 1

The number of function	p	q	r	s
1	6	10	14	18
2	8	14	20	26
3	10	18	26	34
4	12	20	32	42
5	14	22	38	50

Dudley's research has found that the above requirements are most satisfied by the function

$$y = h \cdot y_1 = h \cdot \left[1 + C_2 \cdot \left(\frac{x}{a} \right)^2 + C_p \left(\frac{x}{a} \right)^p + C_q \left(\frac{x}{a} \right)^q + C_r \left(\frac{x}{a} \right)^r + C_s \left(\frac{x}{a} \right)^s \right] \qquad (9)$$

here h — is the maximum valve lift; p, q, r, s — exponents of the monomials, the values of which are given in table.1; C_2, C_p, C_q, C_r, C_s – are the coefficients defined by

$$\left.\begin{aligned}
C_2 &= \frac{-pqrs}{(p-2)\cdot(q-2)\cdot(r-2)\cdot(s-2)};\\
C_p &= \frac{2qrs}{(p-2)\cdot(q-p)\cdot(r-p)\cdot(s-p)};\\
C_q &= \frac{-2prs}{(q-2)\cdot(q-p)\cdot(r-q)\cdot(s-q)};\\
C_r &= \frac{2pqs}{(r-2)\cdot(r-q)\cdot(r-p)\cdot(s-r)};\\
C_s &= \frac{-2pqr}{(s-2)\cdot(s-p)\cdot(s-q)\cdot(s-r)},
\end{aligned}\right\} \qquad (10)$$

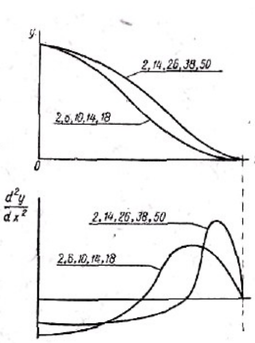

Fig. 2. Diagrams of rises and accelerations of the valve calculated using the polydine method for exponents of the first and fifth groups of polynomial degrees

here a — is the angle of rotation of the cam from the moment y = h the moment y = 0 (Fig. 2); for automobile gasoline engines a = 60-72°.

When selecting the y function, keep in mind that with increasing indicators p, q, r, s positive accelerations and the "time – section" factor of the valve increase [2, 5]. In this regard, for high-speed engines of passenger cars, sports cars, and racing cars, you should choose the y functions with the indicators of the first three groups of table 1 and with the indicators of the fourth and fifth groups for truck engines.

Differentiating equation (9) four times, we get:

$$\frac{dy}{dx} = h \cdot \frac{dy_1}{dx} = h \cdot \left[\frac{2 \cdot C_2}{a} \cdot \left(\frac{x}{a}\right) + \frac{p \cdot C_p}{a} \cdot \left(\frac{x}{a}\right)^{p-1} + \frac{q \cdot C_q}{a}\left(\frac{x}{a}\right)^{q-1} + \frac{r \cdot C_r}{a}\left(\frac{x}{a}\right)^{r-1} + \frac{s \cdot C_s}{a}\left(\frac{x}{a}\right)^{s-1} \right] \tag{11}$$

$$\frac{d^2 y}{dx^2} = h \cdot \frac{d^2 y_1}{dx^2} = h \cdot \left[\begin{array}{l} \frac{2 \cdot C_2}{a^2} + \frac{p \cdot (p-1) \cdot C_p}{a^2} \cdot \left(\frac{x}{a}\right)^{p-2} + \frac{q \cdot (q-1) \cdot C_q}{a^2}\left(\frac{x}{a}\right)^{q-2} + \\ + \frac{r \cdot (r-1) \cdot C_r}{a^2}\left(\frac{x}{a}\right)^{r-2} + \frac{s \cdot (s-1) \cdot C_s}{a^2}\left(\frac{x}{a}\right)^{s-2} \end{array} \right] \tag{12}$$

$$\frac{d^3 y}{dx^3} = h \cdot \frac{d^3 y_1}{dx^3} = h \cdot \left[\begin{array}{l} \frac{p \cdot (p-1) \cdot (p-2) \cdot C_p}{a^3} \cdot \left(\frac{x}{a}\right)^{p-3} + \frac{q \cdot (q-1) \cdot (q-2) \cdot C_q}{a^3}\left(\frac{x}{a}\right)^{q-3} + \\ + \frac{r \cdot (r-1) \cdot (r-2) \cdot C_r}{a^3}\left(\frac{x}{a}\right)^{r-3} + \frac{s \cdot (s-1) \cdot (s-2) \cdot C_s}{a^3}\left(\frac{x}{a}\right)^{s-3} \end{array} \right] \tag{13}$$

$$\frac{d^4 y}{dx^4} = h \cdot \frac{d^4 y_1}{dx^4} = h \cdot \left[\begin{array}{l} \frac{p \cdot (p-1) \cdot (p-2) \cdot (p-3) \cdot C_p}{a^4} \cdot \left(\frac{x}{a}\right)^{p-4} + \\ + \frac{q \cdot (q-1) \cdot (q-2) \cdot (q-3) \cdot C_q}{a^4}\left(\frac{x}{a}\right)^{q-4} + \\ + \frac{r \cdot (r-1) \cdot (r-2) \cdot (r-3) \cdot C_r}{a^4}\left(\frac{x}{a}\right)^{r-4} + \\ + \frac{s \cdot (s-1) \cdot (s-2) \cdot C_s}{a^4}\left(\frac{x}{a}\right)^{s-4} \end{array} \right] \tag{14}$$

In formulas (9)–(14), the gap and elastic deformations of the drive mechanism are not taken into account.

Equation (9) for the design of the Cam profile section corresponding to the choice of a gap, usually called the escape profile, is not suitable enough [6, 7]. The increase in the lift of the pusher at the junction of the gap selection curve in the curve of the main Cam profile is very slow, while the increase in speed and acceleration is very fast. Thus, even a slight change in the clearance or stiffness of the drive parts will lead to a noticeable change in the timing phases, as well as (when the gap increases) to the rise and landing of the valve with significant speeds and accelerations. In high-speed automobile engines, therefore, the main Cam profile is added to the gap selection section profile (run-off profile). At present, profiles that ensure the movement of the pusher at first with a constant acceleration, and then with a constant speed have become common for escape profiles. Sharp change in accelerations at the beginning of the gap selection is not a disadvantage of the profile, since the forces of inertia of the pusher do not affect the operation of the gas distribution mechanism (the valve is in the closed position at this time).

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