MODEL OF HYDRAULIC DOUBLE-ACTING DRIVE FOR VALVES OF INTERNAL COMBUSTION ENGINE

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Abstract

The article presents the idea and the model of electrohydraulic double-acting valve drive for internal combustion engine. In this solution to control the position of the engine valve used Rexroth's servovalve and the double-acting actuator. This drive should give free control of valve lift, valve open and valve close time. In the article the principles of the operation and the mathematical model of the drive are described. Submitted model gives bases to the initial value definition of the main parameters of the hydraulic drive. It lets execute the analysis of the features of such drive and point necessary modifications for the correct realization of the process of the gas exchange in internal combustion engine. The first results from the simulation showed that proposed solution allows obtaining desirable run of lift of valve in the wide range of engine speed. Comparisons of valve lift proceedings driven hydraulically and mechanically by the use of harmonic cam, for two engine speeds, obtained from simulations are presented in the article. These comparisons show that at greatest engine speeds it is indispensable the use of pressures of the working liquid exceeding 10 MPa to obtain nearing results to the classical mechanical timing gear. Preliminary simulation test results indicate that it may ensure desired run of lift of valve in the wide range of engine speed.

Keywords: *internal combustion engine, servovalve, hydraulic actuator, camless engine*

1. Introduction

The most of presently produced internal combustion engines are constructed with a mechanical cam actuation of valves. However, the need of realization the determinate way of gas exchange in different engine working conditions causes the necessity of the continuous lift and angles of the opening and the closing regulation of intake and exhaust valves. In that case the need of elaboration of solutions appeared that makes possible optimal valve control in the full range of engine work parameters. One of such solution is the replacement of the classical mechanical timing gear by camless electrohydraulic valves drive.

2. Conception of electrohydraulic double-acting valve drive

The essence of electrohydraulic valve drive relies on actuating the engine valves by means of hydraulic actuators controlled electronically. In the article one refers to the drive with the use of the double-acting actuator without the spring. The schema of such solution is shown in Fig. 1. The basic element of such system is double-acting actuator with double-side piston rod. One of its piston rod is connected with engine valve, however the second is used to trace the position of valve. In this solution, the movement in double directions of valve (piston) is forced by the pressure lead to cylinder of working agent. Controlling the lead of working agent is realized by the use of four ported spool distributor – servovalve. Through its controlling, it is possible to realize the connections of one of actuator's chamber while the second is connected with the tank. Reversing the connections allows creating engine valve backwards movement. The remaining elements of the system are: supply pump, safety valves, filter, cooler and hydropneumatic accumulator used to reducing the pressure fluctuation.

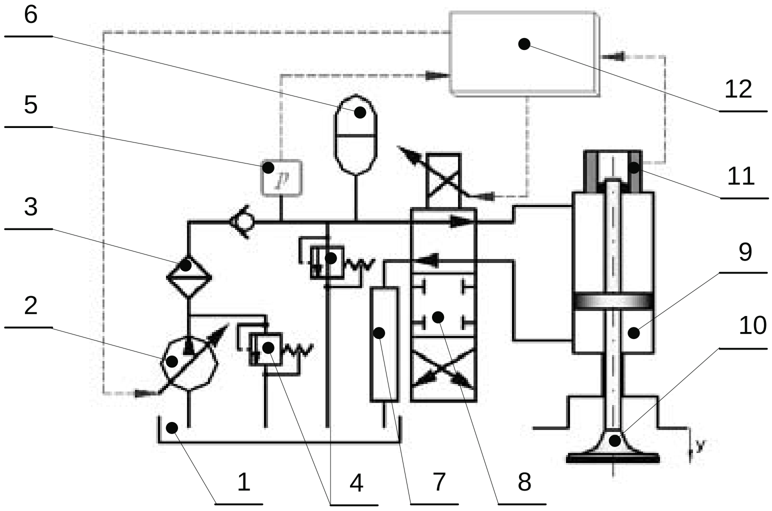

Fig. 1. Functional schema of electrohydraulic valve drive: 1 - tank, 2 - supply pump, 3 – filter, 4 - safety valves, 5 – pressure sensor, 6 - hydropneumatic accumulator, 7 – cooler, 8 – servovalve, 9 – actuator, 10 – engine valve, 11 – position sensor, 12 – control unit

3. Structure and operations of servovalve

In the analyzed drive to control the movement of actuator piston (engine valve) there was used Rexroth's proportional valve, whose schema is shown in Fig. 2. It consists of two major elements:

- torque motor cooperating (5, 6) with the system nozzle – flapper (3, 4),
- spool (1).

These elements are connected by mechanical feedback in the spring form (2).

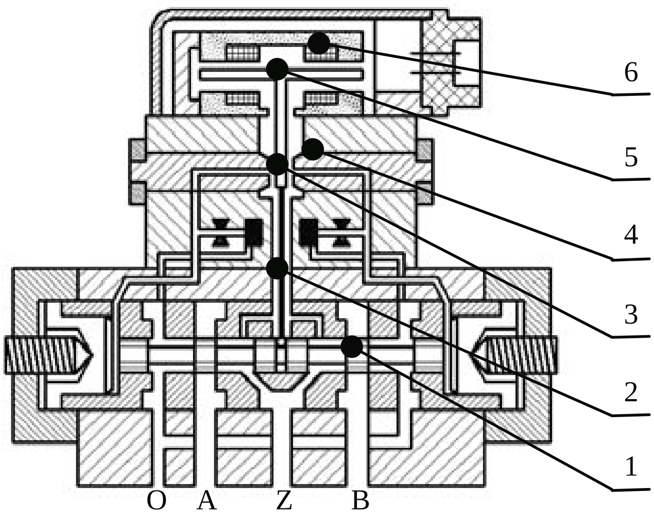

Fig. 2. Schema of Rexroth's servovalve type 4WS2EM10: 1– spool 2 – feedback spring, 3 – nozzle, 4 – flapper, 5 – armature, 6 – coil, O – inlet port, A – port A, B – port B, Z – return port

In the neutral position (the lack of electric signal) the armature of torque motor (5), and thanks to it, nozzle flapper are put in the middle position. The pressure of oil constantly flowing through nozzles is the same, which effects the four-edged spool put in the middle position. Electric steering signal causes the creation of magnetic moment by coils that are why the armature is magnetised to appropriate pole of torque motor. Magnetic moment is proportional to the current intensities flowing through windings. As a result of armature's movement, flapper deflects from its original middle position covering one of the nozzle and at the same time revealing the second one (Fig. 3a).

The consequence of it is the increase of oil pressure in front of covered nozzle and oil pressure decrease in front of uncovered nozzle. Owing to this situation, the pressures working on spool's head area are not the same. The difference in pressure causes the fact that the resultant of forces working on the spool is different from zero, which forces spool's movement (Fig. 3b). It is given on flapper and armature by feedback spring (2). The movement of spool finishes in the position, in which moment of force on armature deriving from feedback spring (2) and the magnetic moment are equalized. Flapper will take then the position, in which resultant of forces working on the spool equals zero (Fig. 3c). Feedback spring causes that the movement of spool is proportional to steering signal. As a result of moving the spool there is a connection one of ports with supply when the second is connected to the tank (Fig. 3c).

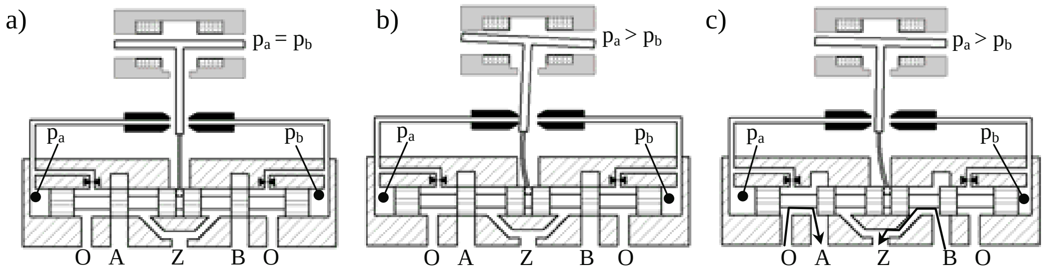

Fig. 3. The servovalve working phases

4. Model of valve drive

For the model of hydraulic components of the drive, the following assumptions are applied:

- working oil do not change of physical properties (temperature is not changed, no air is present),
- inlet pressure is equal to supply pressure and constant,
- return pressure is equal to atmospheric pressure (due to conduits with large diameters),
- resistance of flow is small (due to big work pressure),
- speed of pressure propagation is infinite - related events are simultaneous (due to short distances and big pressure wave propagation speed),
- no oil leakage between the components,
- walls expansion is not taken into consideration,
- oil compressibility is taken into consideration.

Model of the set giving movement to the four-edged spool

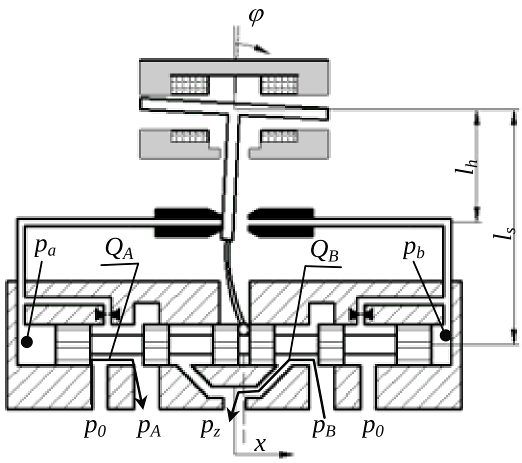

Fig. 4. Schema of servovalve

The rotary motion equation of the armature, at the foundation of her not large dislocations, is (signatures in compliance with the Fig. 4):

$$J_z \frac{d^2\varphi}{dt^2} = M_z - F_s \cdot l_s - F_h \cdot l_h - c_\varphi \frac{d\varphi}{dt}, \tag{1}$$

where:

J_z – representative inertia of armature,

φ – armature turning angle,

$M_z = k_i \cdot i$ – armature torque, where: k_i – current gain coefficient, i – torque motor current,

$F_s = k_s \cdot (x + \varphi \cdot l_s)$ – spring force, where: k_s – feedback spring stiffness, x – spool displacement,

$F_h = \chi \cdot \frac{\pi \cdot {d_d}^2}{4} \cdot (p_a - p_b)$ – hydrodynamic force acting on flapper, where: χ - hydrodynamic coefficient, d_d – nozzle diameter, p_a, p_b – pressure in nozzles channels (on ends of spool),

c_φ – viscosity friction coefficient of armature.

In two stage nozzle-flapper servovalve the pressure differential is linear function of armature turning angle. So accordantly with the graph in Fig. 5, this dependence can be described as:

$$p_a - p_b = k_\kappa \cdot \varphi,$$

where:

k_φ – coefficient of pressure differential,

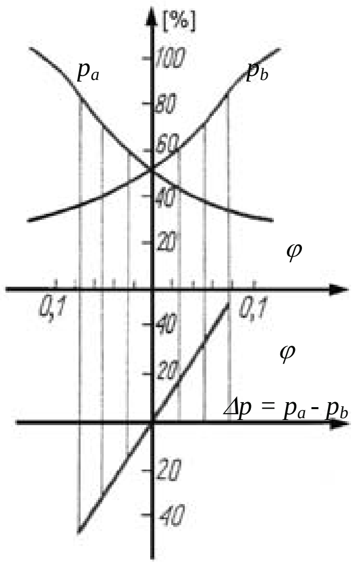

Fig. 5. Characteristic of nozzle-flapper servo

The motion equation of the spool can be expressed as:

$$m_s \cdot \frac{d_s^2 x}{dt^2} = \Delta p_s \cdot A_s - F_s - F_{hd} - F_{ts} \cdot sign\left(\frac{dx}{dt}\right) - c_s \cdot \frac{dx}{dt}, \tag{2}$$

where:

m_s – mass of spool,

$\Delta p_s = (p_a - p_b)$ – pressure differential on ends of the spool (also in nozzles inlets),

$A_s = \dfrac{\pi \cdot d_s^2}{4}$ – spool end area, where: d_s - spool diameter,

$F_{hd} = \dfrac{0{,}72}{\sqrt{\xi}} \cdot \pi \cdot d_s \cdot x \cdot \Delta p_s$ – hydrodynamic force acting on spool, where: ξ - hydrodynamic resistance coefficient,

F_{ts} – stiction force,

c_s – viscosity friction coefficient of spool.

Model of the hydraulic distributor

Described with the equation (2) the movement of the spool changes the area of orifices in controlling ports through the working liquid flow (Fig. 3). The value of the flowrate one can describe with the dependence obtained from the Bernoulli's equation:

$$Q = c_q \cdot \sqrt{\frac{2}{\rho}} \cdot S \cdot \sqrt{\Delta p} = c_q \cdot \sqrt{\frac{2}{\rho}} \cdot \pi \cdot d_s \cdot x \cdot \sqrt{\Delta p} = K_Q \cdot x \cdot \sqrt{\Delta p}\ , \tag{3}$$

where:

c_q – flow coefficient dependent on Reynolds number,

$S = \pi \cdot d_s \cdot x$ – cross-sectional area of orifice,

$K_Q = c_Q \cdot \sqrt{\dfrac{2}{\rho}} \cdot \pi \cdot d_s$ – flow coefficient dependent on orifices geometry, Reynolds number and working fluid density,

Δp – pressure drop on orifices.

For analysed servovalve the flowrate to the A port of the actuator, at the foundation of zero-covering of the spool is (Fig. 4):

$$Q_a = \begin{cases} K_Q \cdot x \cdot \sqrt{p_0 - p_A} & for \quad x_{\max} \geq x \geq 0 \\ K_Q \cdot x \cdot \sqrt{p_A - p_z} & for \quad -x_{\max} \leq x < 0 \end{cases}. \tag{4}$$

where:

p_o – inlet pressure,

$p_{A/B}$ – pressure associated with port A/B of the servovalve,

p_z – return pressure.

Flowrate associated with side B of the servovalve is:

$$Q_b = \begin{cases} K_Q \cdot x \cdot \sqrt{p_B - p_z} & for \quad x_{\max} \geq x \geq 0 \\ K_Q \cdot x \cdot \sqrt{p_0 - p_B} & for \quad -x_{\max} \leq x < 0 \end{cases}. \tag{5}$$

Model of actuator

The model of the electrohydraulic valve actuator is the composition of the servovalve model and the model of the actuator with engine valve (Fig. 6). In that case, foundations to its building were coincident with assumed in the preceding subsection.

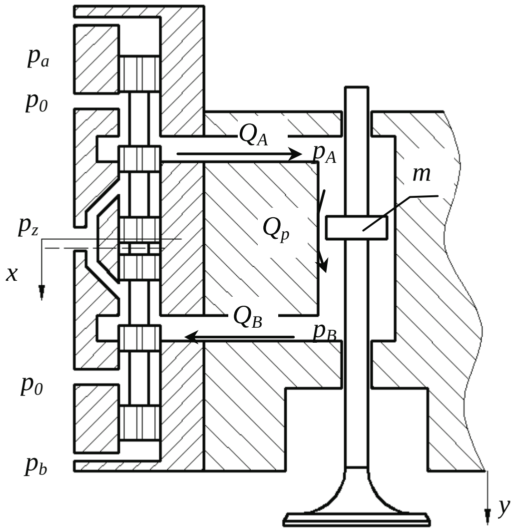

Fig. 6. Schema of hydraulic valve drive

Flowrate equation for system hydraulic distributor - the actuator is:

$$\begin{aligned} Q_A &= Q_h + Q_p + Q_{sA} \\ Q_B &= -Q_h - Q_p + Q_{sB} \end{aligned}, \tag{6}$$

where:

$Q_h = A\dfrac{dy}{dt}$ – flowrate for actuator absorbency, where: A – hydraulic amplifier, piston working area,

y – engine valve displacement

$Q_p = k_v(p_A - p_B)$ – leak flowrate, where: k_v - leak coefficient,

$Q_{sA/B} = \dfrac{V_{A/B}}{E_c}\dfrac{dp_{A/B}}{dt}$ – flowrate associated with oil compressibility, where: E_c - oil bulk modulus,

$V_{A/B}$, – volume of chamber A/B.

The motion equation of the piston can be expressed as:

$$m \cdot \frac{d^2 y}{dt} = A \cdot \Delta p - F_g - F_t \cdot sign\left(\frac{dy}{dt}\right) - C \cdot \frac{dy}{dt}, \tag{7}$$

where:

m – mass of piston and element connected with it,

y – engine valve displacement,

$F_g = A_z \cdot p_s(t)$ – gas force acting on engine valve head, where: A_z - area of engine valve head,

$p_s(t)$ – gas pressure in engine cylinder,

F_t – stiction force,

C – viscosity friction coefficient.

5. Preliminary simulation tests of the drive

Mathematical model of hydraulic double-acting valve drive showed above, allowed to create a simulation model in Simulink. Detailed description of this model together with the way of defining its parameters will be described in further papers.

Preliminary test results show that suggested solution can ensure the demanded proceedings of valve lift in the wide range of engine speed. In Fig. 7 there is a comparison of valve lift proceedings driven hydraulically and mechanically by the use of harmonic cam. For the assumed time of the valve opening equals 14 ms (it corresponds 248^{o} crank-shaft turn for 3000 rpm) and the pressure equals 6 MPa. Received results show that in case of hydraulic drive it is possible to get a bigger cross-section time of 25%.

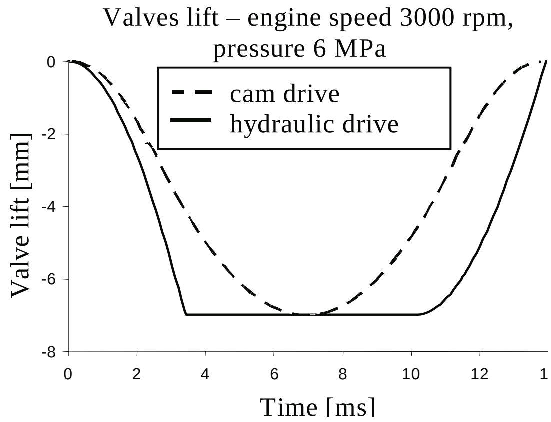

Fig. 7. Lifts of valve driven hydraulically and mechanically by the use of harmonic cam for engine speed 3000 rpm

In case of higher engine speed, to get similar cross-section time for cam and hydraulic valve drive, it is necessary to increase the oil working pressure. For example at the engine speed 6000 rpm and the same angle of valve opening – $248\,^{o}$ crank-shaft turn, it is necessary to increase the pressure to 10 MPa (Fig. 8).

Preliminary simulation test results show that the proceeding of valve lift in hydraulic drive is characterized by larger accelerations in relation to the cam drive. It leads to appearance of unfavourable hits in final position of a valve. While opening, there is a hit of piston into a buffer, while closing valve head into a valve seat.

The submitted mathematical drive model, after a completion of solutions allowing to smooth brake of valve before its final position, will let to estimate the solutions quickly. It will be beneficial for design processing and facilitate examination on such drive.

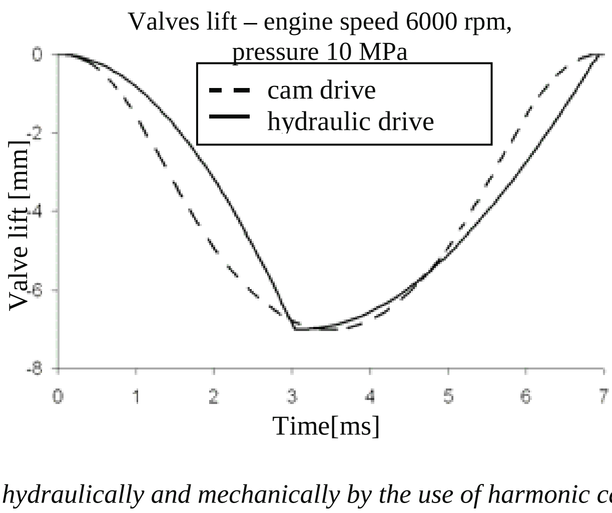

Fig. 8. Lifts of valve driven hydraulically and mechanically by the use of harmonic cam for engine speed 6000 rpm

6. Conclusion

Model of electrohydraulic double-acting valve drive for internal combustion engine presented in the article gives the solid basis for further works on its development. Preliminary simulation test results indicate that it may ensure desired run of lift of valve in the wide range of engine speed. Further tests will be aimed at verification of established model parameters. Next, having the model verified, it will be possible to define basic parameters of the drive and analyse their impact on the drive characteristics. This will allow pointing out any modification needed to correct realization of the gas exchange process.

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