EXPERIMENTAL VERIFICATION OF THE MODEL OF ELECTROHYDRAULIC DRIVE FOR INTERNAL COMBUSTION ENGINE VALVES

Tomasz Szydłowski

Technical University of Lodz, Department of Mechanical Engineering
Zeromskiego Street 116, 90-924 Lodz, Poland
tel.: +48 42 6312395, fax: +48 42 6312398
email: tomasz.szydlowski@p.lodz.pl

Abstract

The paper presents experimental electrohydraulic valve drive for internal combustion engine. The design is a single-acting hydraulic actuator with spring return controlled by servovalve. Such solution should give free control of valve lift, valve open and valve close time. In the article the principles of the operation of the drive are described. The paper focuses on exploring the dynamic of experimental electrohydraulic engine valve actuation system working in open-loop. For investigations the mathematical and simulation models were elaborated. The values of unknown coefficients of the models were intercepted on two test stands. First of them served to servovalve research. Second was built on the basis of the prototype drive and let on the research of the dynamics of the all drive. Chosen results of these investigations are placed in the article. The drive was explored for different oil supply pressures, control signals and spring mounting lifts. Experimental dates let to verified dynamic mathematical model. Comparisons of valve lift proceedings obtained from experiments and simulations are presented in the article. Verified model let to 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.

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

1. Introduction

Flexible valve actuation systems can improve the fuel economy, emission and torque of internal combustion engine. Continuous regulation of lift and angles of the opening and the closing intake and exhaust valves can be achieved for example with electromagnetic, electropneumatic or electrohydraulic valve drive systems.

In this paper electrohydraulic single-acting valve drive was analyzed. In this solution opening of valve is forced by the pressure lead to the actuator. The return movement of the valve causes the spring force. Direction of valve movement is controlled by servovalve.

2. Structure and operations of electrohydraulic single-acting valve drive

The system outlined below is capable of actuating a single engine valve. The schema of laboratory electrohydraulic valve drive is shown in Fig. 2. The basic elements of such system are: single-acting actuator with one-side piston rod and servovalve – Fig. 1. The piston rod is connected with engine valve. Controlling the lead of working agent is realized by servovalve. The servovalve consists of two major elements – Fig. 1:

- 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).

From the schematic it can be seen that 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 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.

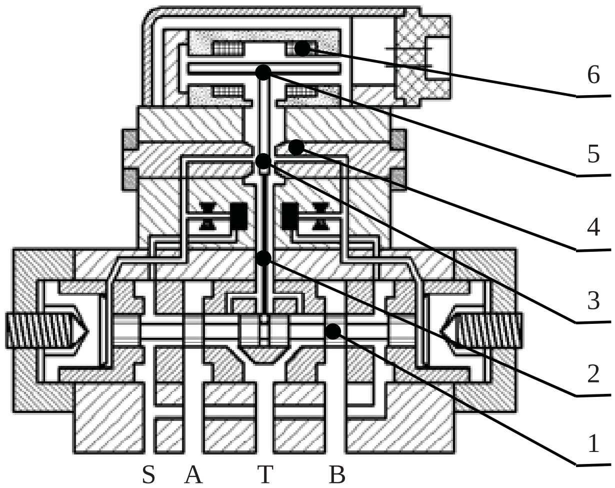

Fig. 1. Schema of Rexroth's servovalve type 4WS2EM10: 1– spool 2 – feedback spring, 3 – nozzle, 4 – flapper, 5 – armature, 6 – coil, S – inlet port, A – port A, B – port B, T – return port

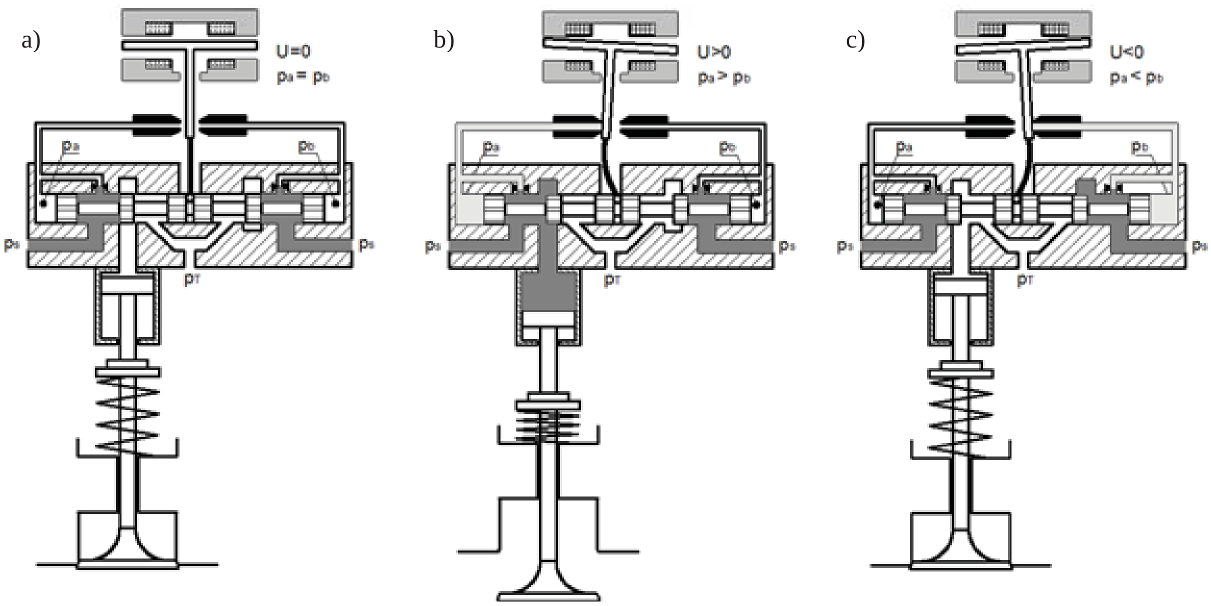

Fig. 2. The working phases of electrohydraulic valve drive

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. 2b). 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. 2b). 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. Feedback spring causes that the movement of spool is proportional to steering signal. As a result of moving the spool there is a connection the supply port with actuator and the piston move down. Opposite electric steering signal causes analogical situation with the difference that the spool moves in opposite direction. The result of such moving is connection actuator with the tank (Fig. 2c). Then the return spring moves the piston up. Since the piston is in direct contact with the engine valve its displacement is equal to the displacement of the engine valve.

Application of two stage servovalve causes the movement of engine valve is generated by the input voltage ranges ±10 V and current not exceed 100 mA. In such solution small electric power is enough to move the valve.

3. Model of valve drive

The mathematical model of servovalve is recount in [4]. In accordance with [4] the rotary motion equation of the armature can be expressed as:

$$J_z \frac{d^2\varphi}{dt^2} + c_\varphi \frac{d\varphi}{dt} + \left(k_s \cdot l_s^2 + \chi \cdot \frac{\pi \cdot {d_d}^2}{4} \cdot l_h \cdot k_\varphi \right) \cdot \varphi = k_i \cdot i - k_s \cdot x \,, \tag{1}$$

where:
J_z - representative inertia of armature,
φ - armature turning angle,
c_φ - viscosity friction coefficient of armature,
k_s - feedback spring stiffness,
l_s - the distance between the rotation axis of the armature and axis of the slider,
χ - hydrodynamic coefficient,
d_d - nozzle diameter,
l_h - the distance between the rotation axis of the armature and axis of the flapper,
k_φ - coefficient of pressure differential,
i - torque motor current,
k_i - current gain coefficient,
x - spool displacement.

At the foundation of not large armature dislocations the current equation can be expressed as:

$$L \cdot \frac{di}{dt} + R \cdot i = U \,, \tag{2}$$

where:
L - inductance of the solenoid,
R - resistance of the solenoid.

The motion equation of the spool can be expressed as [4]:

$$m_s \cdot \frac{d^2x}{dt^2} + c_s \cdot \frac{dx}{dt} + F_{ts} \cdot sign\left(\frac{dx}{dt}\right) + \left(k_s + \frac{0{,}72}{\sqrt{\xi}} \cdot \pi \cdot d_s \cdot \Delta p \right) \cdot x = \left(k_\varphi \cdot A_s - k_s \cdot l_s \right) \cdot \varphi \,, \tag{3}$$

where:
m_s - mass of spool,
c_s - viscosity friction coefficient of spool,
F_{ts} - stiction force,
ξ - hydrodynamic resistance coefficient,
Δp - pressure drop on servovalve orifice,
d_s - spool diameter,
$A_s = \frac{\pi \cdot d_s^2}{4}$ - spool end area.

3.1. Model of the hydraulic distributor

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

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

where:

c_q - flow coefficient dependent on Reynolds number,

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

x_r - real orifice open,

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

Δp - pressure drop on orifices.

Flowrate equation for system hydraulic distributor – the actuator, according to Fig. 3 is:

$$Q_S = Q_T + Q_h + Q_l + Q_c\ , \tag{4}$$

where:

$Q_S = K_{QS} \cdot x_{rS} \cdot \sqrt{p_S - p_A}$ - inlet flowrate,

$Q_T = K_{QT} \cdot x_{rT} \cdot \sqrt{p_A - p_T}$ - return flowrate,

where:

p_s - inlet pressure,

p_A - pressure in the actuator,

p_T - return pressure,

$x_{rS} = \begin{cases} x - x_{0S} & \text{for } x \ge x_{0S} \\ 0 & \text{for } x < x_{0S} \end{cases}$ - inlet real orifice open, where x_{0S} - spool overlap,

$x_{rT} = \begin{cases} 0 & \text{for } x > x_{0T} \\ x + x_{0T} & \text{for } x \le x_{0T} \end{cases}$ - return real orifice open, where x_{0T} - spool overlap,

$Q_h = \begin{cases} A\frac{dy}{dt} & \text{for } y \le y_{p\max} \\ 0 & \text{for } y > y_{p\max} \end{cases}$ - flowrate for actuator absorbency,

where:

A - hydraulic amplifier piston working area,

y - engine valve displacement,

y_p - piston displacement,

$Q_l = k_v \cdot p_A$ - leak flowrate, where: k_v - leak coefficient,

$Q_c = \frac{V}{E_c}\frac{dp_A}{dt}$ - flowrate associated with oil compressibility,

where:

E_c - oil bulk modulus,

V - volume of actuator chamber.

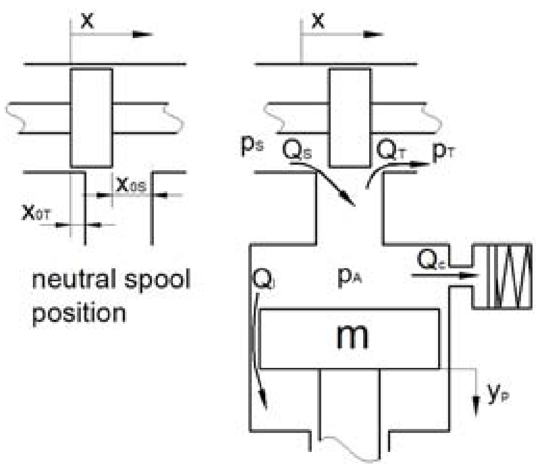

Fig. 3. Schema of hydraulic part of valve drive

The motion equation of the engine valve can be expressed as:

$$\begin{cases} m \cdot \dfrac{d^2 y}{dt^2} + C \cdot \dfrac{dy}{dt} + F_t \cdot sign\left(\dfrac{dy}{dt}\right) + k \cdot y = A \cdot p_A + m \cdot g - k \cdot y_0 - F_g & \text{for} \quad y < y_{p\max}, \\ m_v \cdot \dfrac{d^2 y}{dt^2} + C_v \cdot \dfrac{dy}{dt} + F_{tv} \cdot sign\left(\dfrac{dy}{dt}\right) + k \cdot y = m \cdot g - k \cdot y_0 - F_g & \text{for} \quad y > y_{p\max}, \end{cases} \tag{5}$$

where:

m - the sum of masses of: piston, engine valve and element connected with it,
m_v - mass of engine valve and element connected with it,
y - engine valve displacement,
y_0 - spring mounting lift,
C, C_v - viscosity friction coefficients,
F_t, F_{tv} - stiction forces,
k - stiffness of the spring,
A - area of piston,
p_A - actuator pressure,
$F_g = A_z \cdot p_s(t)$ - gas force acting on engine valve head,

where:

A_z - area of engine valve head,
$p_s(t)$ - gas pressure in engine cylinder.

4. Investigation objects

For the purpose of intercept the values of unknown coefficients of the model one built two test stands. First of them was necessary to obtain no loaded flow rate characteristics. These characteristics let to determine values of inlet and return spool overlaps. The schema and the photo of the test stand are shown in Fig. 4.

Second test stand was built to research dynamic of complete drive. The schema and photo of this test stand are shown in Fig. 5. The test stand consist of: proportional valve (servovalve), hydraulic actuator, engine valve, spring, supply pump, safety valves, filter, cooler, hydropneumatic accumulator and input signal generator. The hydraulic pump provides the high pressure fluid up to 10 MPa. The hydropneumatic accumulator filled with nitrogen is used to reduce the supply pressure fluctuation. The generator realizes open loop control, by delivering rectangular voltage signal. The displacement of the engine valve was measured with utilization of the optical distance sensor.

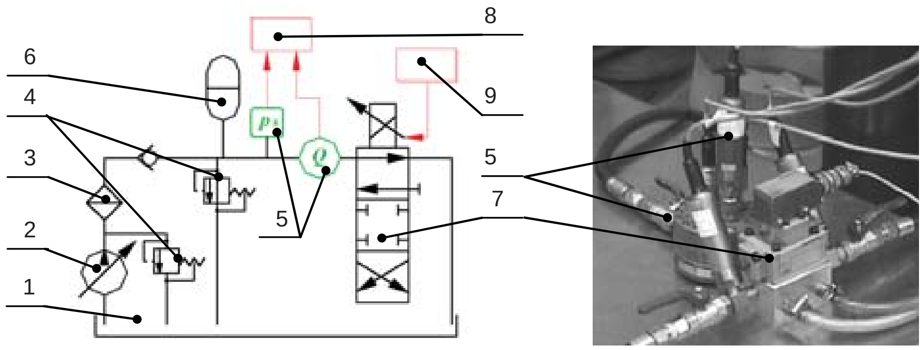

Fig. 4. The test stand for servovalve researches: 1 - tank, 2 - supply pump, 3 – filter, 4 - safety valves, 5 – pressure and flowrate sensors, 6 - hydropneumatic accumulator, 7 – servovalve, 8 – acquisition system, 9 – generator

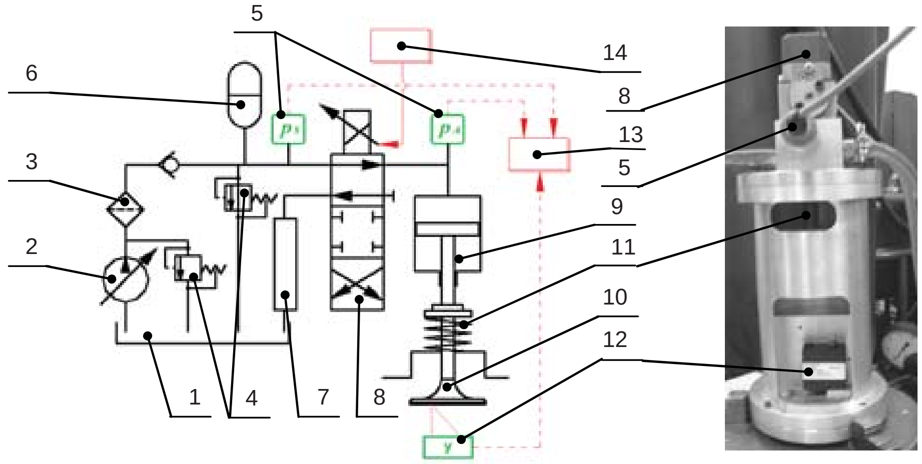

Fig. 5. Functional schema of electrohydraulic valve drive: 1 - tank, 2 - supply pump, 3 – filter, 4 - safety valves, 5 – pressure sensors, 6 - hydropneumatic accumulator, 7 – cooler, 8 – servovalve, 9 – actuator, 10 – engine valve, 11 – valve spring, 12 – position sensor, 13 – acquisition system, 14 – generator

5. Experimental tests results

The flowrate characteristics were obtained for pressures range 2-6 MPa. On their base one counted values of flow coefficients and spool overlap. Chosen characteristics obtained on this stage of investigations are presented in Fig. 6.

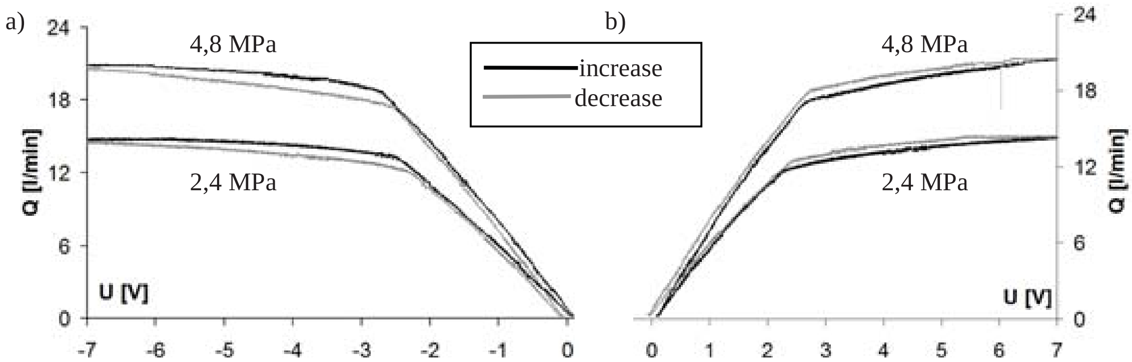

Fig. 6. Flowrate characteristics of servovalve: a) the outflow from the actuator b) the inflow to the actuator

Dynamic of the drive was research on the test stand shown in Fig. 5. The investigations were led for pressures of the power supply within 6-10 MPa. One changed also spring mounting lift. The best dynamic parameters of hydraulic engine valve drive were obtained for highest pressures. Exemplary valve lift proceedings, his speeds and accelerations, for supply pressure 10 MPa, are presented in Fig. 7. One examined two ways of closing of the engine valve. The first, through leaving of the return orifice opening equal the spool overlap – Fig. 7a. The second, through maximal opening of return orifice – Fig. 7b. In this ways obtained different speeds of closing of the valve and especially different valve seating velocities. Such control of drive is considering as one of ways of valve braking.

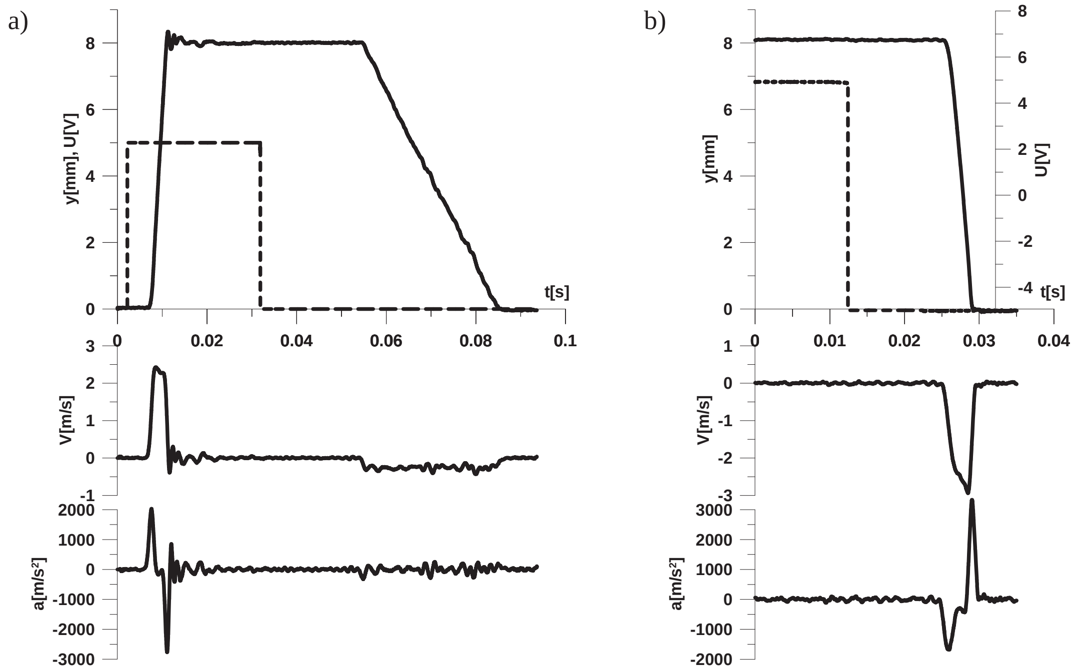

Fig. 7. Valve displacement, velocity and acceleration: a) return orifice opening equal the spool overlap, b) full opening of return orifice

6. The verification of the model

The mathematical model of hydraulic single-acting valve drive described above, allowed creating a simulation model in Simulink. This model was experimentally verified. The comparative assessments were made for different pressure supply and spring mounting lifts. The verification example is illustrated in Fig. 8. Below presented graphs show good accordance simulation and experimental test dates.

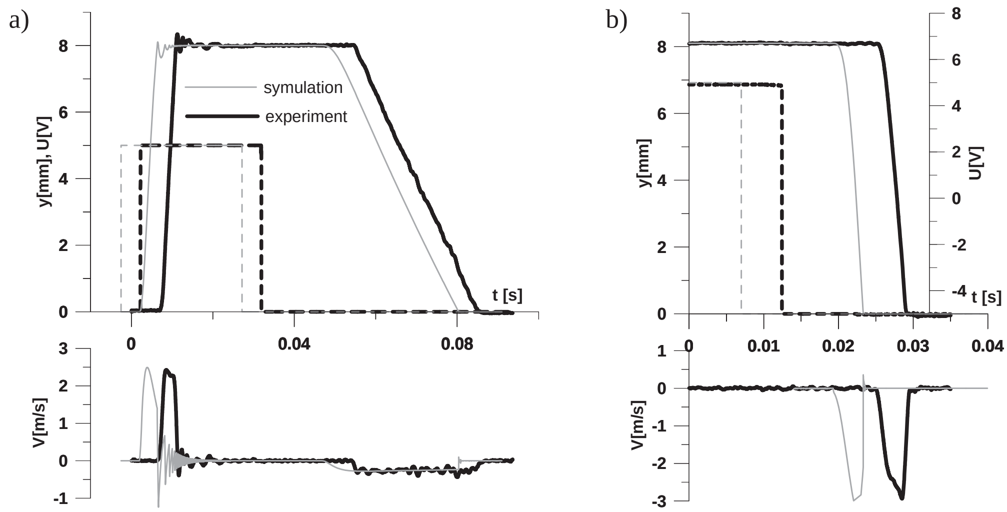

Fig. 8. Valve displacement, velocity and acceleration – test dates obtained from simulation and experiment: a) return orifice opening equal the spool overlap b) full opening of return orifice

7. Conclusion

Results of verifying researches of the model of electrohydraulic single-acting valve drive show that it gives the solid basis for further works on such drive development. Following tests will be aimed at elaboration an appropriate control system of the drive. Such control system connected with small mechanical changes should give required valve displacement, especially concerning valve seating velocity. Verified model let to execute the analysis of the features of presented engine valve drive and point necessary modifications for the correct realization of the process of the gas exchange in internal combustion engine.

Ministry of Science and Higher Education supports the work as a research project number NN509292835.

References

[1] Pawelski, Z., *Napęd hybrydowy dla autobusu miejskiego*, Monografie Politechniki Łódzkiej 1996.
[2] Pawelski, Z., *Modelowanie i obliczanie napędu hydrobusu*, Monografie Politechniki Łódzkiej 2000.
[3] Stryczek, S., *Napęd hydrostatyczny*, WNT, Warszawa 1997.
[4] Szydłowski T., Smoczyński M., *Model of hydraulic double acting drive for valves of internal combustion engine*, Journal of KONES Powertrain and Transport, Vol. 16, No. 1, pp. 487-494, 2009.
[5] Tomczyk J., *Modele dynamiczne elementów i układów napędów hydrostatycznych*, WNT, Warszawa 1999.
[6] Zbierski, K., Szydłowski, T., *Napęd hydrauliczny zaworu rozrządu tłokowego silnika spalinowego. Istota, możliwości, własne koncepcje – cz. I*, Napędy i Sterowanie, 10/2007.
[7] Zbierski, K., Szydłowski, T., *Napęd hydrauliczny zaworu rozrządu tłokowego silnika spalinowego. Model i podstawowe parametry napędu jednostronnego działania – cz. II*, Napędy i Sterowanie, 7-8/2008.