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EXPERIMENTAL RESEARCH OF PROPERTIES OF HYDRAULIC DRIVE FOR VALVES OF INTERNAL COMBUSTION ENGINES

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Abstract

This paper describes a research stand and results of experimental research of single-acting hydraulic drive for valves of internal combustion engines. The research stand of the hydraulic valve drive was consisted of: typical valve drive for high-speed internal combustion engine, commercial hydraulic actuator, commercial hydraulic accumulator and electrically controlled hydraulic distributor, which controlled the flow of oil supply to the hydraulic actuator. Rexroth commercial servovalve was used as hydraulic distributor for this valve drive. Components of hydraulic valve drive were mounted to specially designed research sleeve. On this sleeve complete drive, which consisted of a hydraulic cylinder and servovalve, was mounted. Drive control was performed in an open loop with a use of rectangular control signal. Displacement of the valve was measured by an optical displacement sensor. Tests were performed for constant lift of the valve, supply pressure set in the range of 6 to 10 MPa and at fixed temperature of the working medium and different initial deflections of valve spring. The behaviour of the drive was researched for bipolar and a rectangular control signals. Such range of measurements was to determine inter alia: the impact of the supply pressure and the control signals values on the drive work. Obtained results were used to verify the simulation model in a wide range of variation of the characteristic parameters of the electro-hydraulic actuator. The results of experimental measurements like valve movement and pressures in the drive were shown and analysed in this paper. With a reference to the valve movement, a detailed calculations of the valve kinematics were performed. Special attention was given to the opening and closing velocities of the valve, depending on the servo control signal. Subsidence valve velocity during its return movement was calculated. This subsidence valve velocity is an important parameter in terms of the applicability of this drive for the internal combustion engines. Valve opening time delay in opposition to the current control signal was also specified and discussed. Obtained results allowed to conclude that the proposed drive provides acceptable kinematic parameters for high-speed engines at supply pressures of at least 8 to 10 MPa. During the measurements acceptable valve subsidence speeds were obtained. It was found that there is a possibility of adjustment of this parameter by setting the slider servovalve negative overlap. Results became the basis of development of the model of this type of the drive. Further simulation studies will allow to evaluate the applicability of such valve drive for internal combustion engines. Further simulation studies allow to compare the proposed solution with known literature hydraulic valve drives.

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. This paper is focused on electrohydraulic valve drive. The idea of hydraulic valve drive for internal combustion engine is known for many years. Research of such kind of timing gear is conducted by many research and industrial centres [1-5].

In this paper a single-acting hydraulic actuator with a return spring, controlled by a servovalve was analysed. Opening of the engine valve was forced by the pressure leading to the actuator. The return movement of the valve was forced by the valve spring. Direction of valve movement was controlled by a servovalve. Such solution should give free control of valve lift, valve open and valve close time [6-8, 10].

As part of this paper presents results of research of an experimental electrohydraulic valve drive. The study was aimed at understanding the properties of a single-acting hydraulic actuator, particularly the drive built with a use of the commercial servovalve produced by Bosch Rexroth. They enabled the quantitative determination of the basic parameters of the motion of a typical valve hydraulically actuated to verify the mathematical model developed earlier. Such a verified model will be used to carry out simulation studies necessary to evaluate the possibility of using such drives in internal combustion engines. It will also enable a comparison of the proposed solution with hydraulic drives known from literature.

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

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

From the diagram it can be seen that in the neutral position (the lack of electric signal) the armature of torque motor (5), and, as a result, nozzle flapper are put in the middle position. The pressure of oil constantly flowing through nozzles is the same, which affects the spool put in the middle position. Electric steering signal causes the creation of magnetic moment by coils that is why the armature is magnetised to appropriate pole of torque motor. Magnetic moment is proportional to the current intensities flowing through windings.

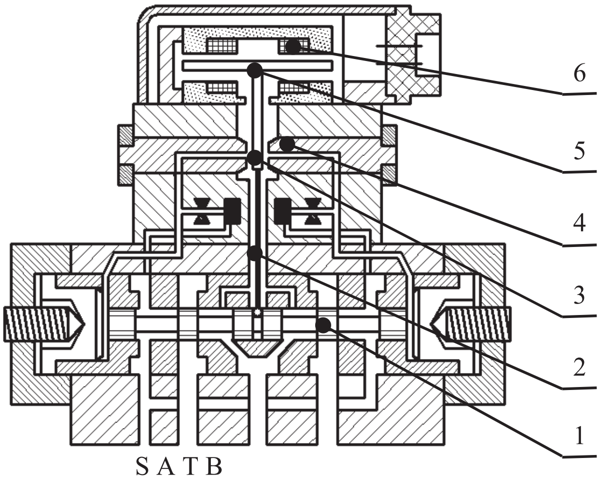

Fig. 1. Schema of Rexroth's servovalve type 4WS2EM10: 1– spool 2 – feedback spring, 3 – nozzle, 4 – flapper, 5 - armature, 6 – coil, S – inlet port, A – port A, B – port B, T – return port

As a result of armature 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. Consequence of this situation is that the pressures working on spool head area are not the same. The difference in pressure causes the fact that the resultant force working on the spool is different from zero, which forces spool movement (Fig. 2b). It is given on flapper and armature by feedback spring (2).

The movement of spool ends 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 force 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, connection between the supply port and the actuator is created and the piston moves down. Opposite electric steering signal causes analogical situation with the difference that the spool moves in opposite direction. Such movement of the spool links the actuator and 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.

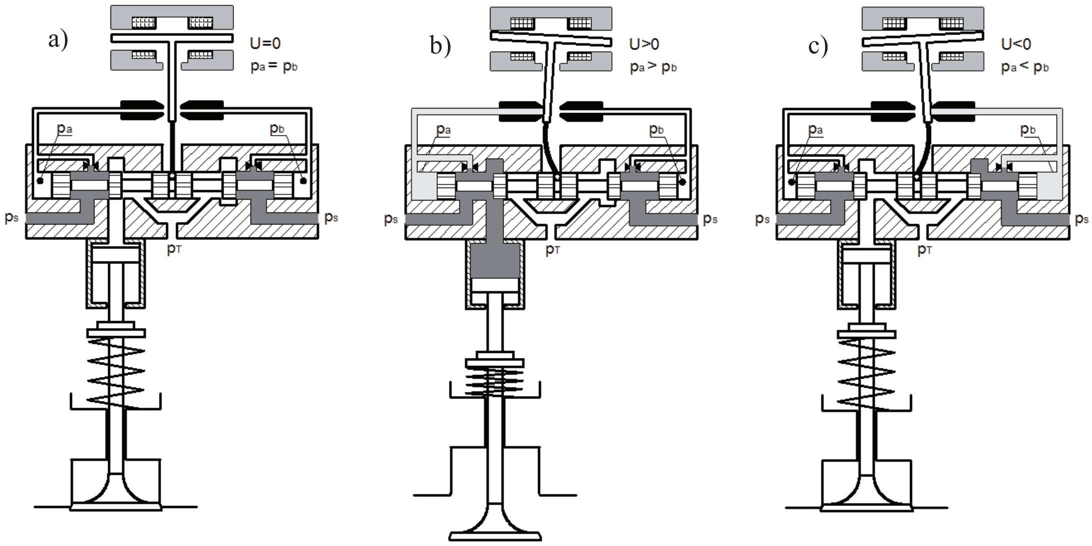

Fig. 2. The working phases of electrohydraulic valve drive

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

3. Investigation objects

The research stand was built to study the dynamic of complete drive. The schema and photo of this stand are shown in Fig. 3. The research stand was consisted of: proportional valve (servovalve), hydraulic actuator, engine valve, spring, supply pump, safety valves, filter, cooler, hydropneumatic accumulator and input signal generator. The hydraulic pump provided the high pressure of the fluid up to 10 MPa. The hydropneumatic accumulator filled with nitrogen was used to reduce the supply pressure fluctuation. The generator realized open loop control, by delivering rectangular voltage signal. The displacement of the engine valve was measured with utilization of the optical displacement sensor.

4. Experimental tests results

The research stand described in the previous section allowed further research, which let to identification of the main characteristics of the analysed drive and determination of the values of coefficients for developed and described in earlier publications mathematical model of hydraulic drive of valve [6, 7, 9].

Laboratory research was performed:

- for valve lift equal to 8 mm,
- for the supply pressure 6, 7, 8, 9 and 10 MPa,
- at a temperature of hydraulic oil 38±1°C,
- for three different valve spring deflections y_0: 3.9 mm, 10.2 mm and 14.0 mm,
- for servovalve control signals:
  - voltage step signals: 5 and 10V,
  - rectangular bipolar signals with amplitudes 5 and 10V.

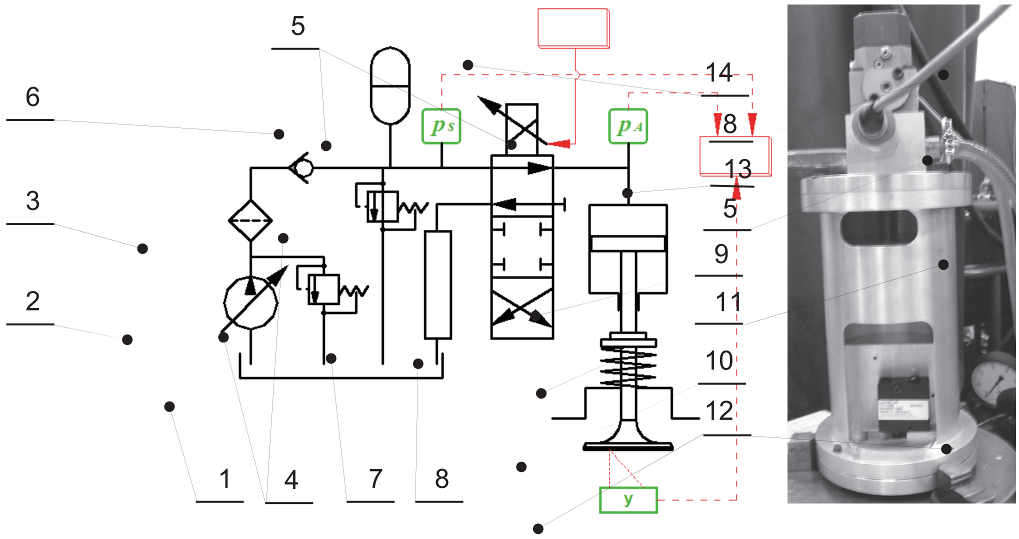

Fig. 3. Functional schema of electrohydraulic valve drive: 1 – tank, 2 – supply pump, 3 – filter, 4 – safety valves, 5 – pressure sensors, 6 – hydropneumatic accumulator, 7 – cooler, 8 – servovalve, 9 – actuator, 10 – engine valve, 11 – valve spring, 12 – displacement sensor, 13 – acquisition system,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

Such range of research was to determine inter alia the impact of: the supply pressure value, the control signals and servovalve control methods on work of whole the drive. At the same time results of research were used to verify the simulation model.

The crucial point of the study was also to determine the closing speed of the engine valve depending on control of the servovalve.

Below are shown and shortly characterized chosen results of research.

Figure 4 shows waveforms of: valve lift, the supply pressure value, the pressure in the actuator for step control signal of value 10 V, the supply pressure equal 10 MPa and the valve spring deflection equal 14 mm. For such parameters, the drive obtain the valve opening time equal to 3.7 ms, which is close to the desired from the viewpoint of the applicability of the drive in a high-speed internal combustion engine. In the final phase of the valve movement one can see that the opening of valve is slightly larger than possible movement of piston. This is due to valve stem detachment from the piston rod. The increase in pressure in the cylinder and the subsequent movement of the valve began around 5 ms after the signal.

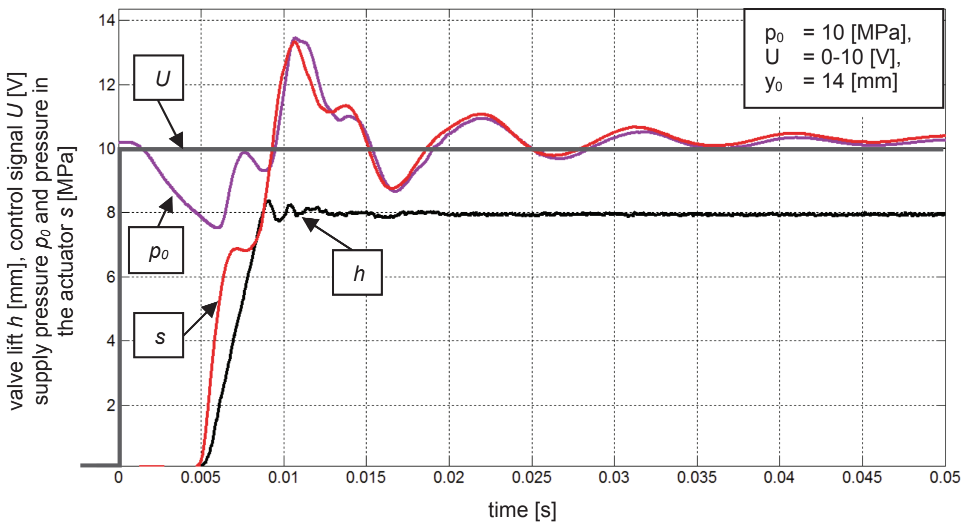

Fig. 4. Waveforms of: valve lift "h", the supply pressure "p0" control signal "U", the pressure in the actuator "s" for step control signal of value 10 V, the supply pressure equal 10 MPa and the valve spring deflection "y0"equal 14 mm

Figure 5 shows the influence of supply pressure value on the waveforms of valve lift. As can be seen from waveforms of valve lift in the studied range of supply pressure the opening times were in the range 3.7-5.0 ms. For supply pressures below 8 MPa occurred incomplete opening of the valve. At lower feed pressure considerable overdrives were observed. They were related mainly to the inertial force acting on the valve causing the detachment the valve stem from the piston rod. Another factor of the magnifying overdrive was propagation of hydraulic pressure wave.

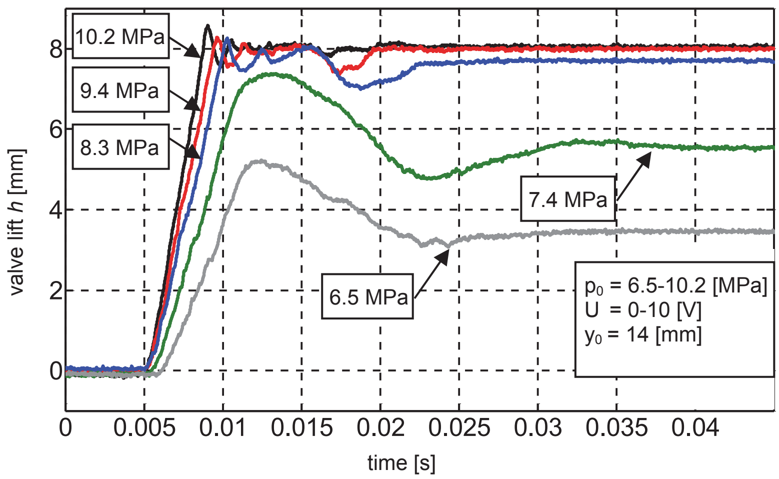

Fig. 5. Influence of supply pressure value on the waveforms of valve lift

Figure 6 shows the waveforms of closing the valve. As one can see it is possible to get the valve closing speed of about 0.27 m/s. Such speed, from the viewpoint of valve subsidence in the seat, seems acceptable. There is an additional possibility of adjusting this parameter by setting the slider negative overlap in servovalve.

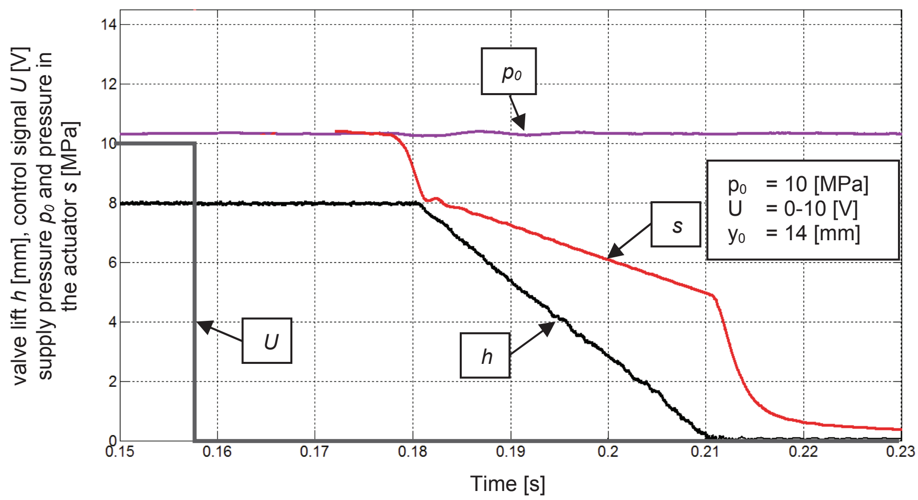

Fig. 6. Waveforms of: valve lift "h", the supply pressure "p0" control signal "U", the pressure in the actuator "s" during valve closing, for step control signal of value 10 V, the supply pressure equal 10 MPa and the valve spring deflection "y0" equal 14 mm

Figure 7 shows waveforms of valve lift during it closing for the different supply pressures. There was no important influence the supply pressure on the valve closing speed. One can see only a slight increase in the velocity as a function of the supply pressure.

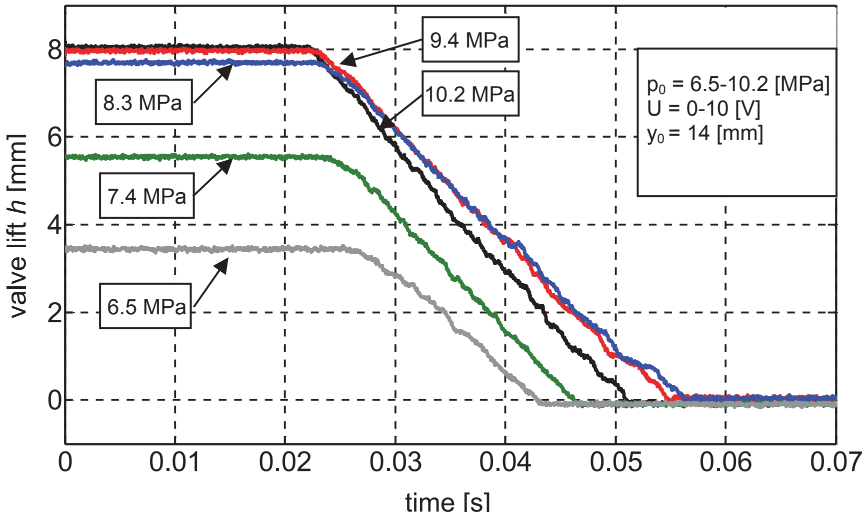

Fig. 7. Influence of supply pressure on waveforms of valve closing

Naturally, presented waveforms of valve closing were not optimal because of too long closing time. Servovalve used in the analysed drive allows to achieve higher closing speed of the engine valve, depending on the control method. Suitable overdrive of servovalve allows to achieve opening proportional to the control signal of the runoff gap, thus obtaining different closing speeds of valve. Fig. 8 shows the waveforms of valve closing depending on supply pressure for maximally opened runoff gap. In this case valve closing times were around 5 ms.

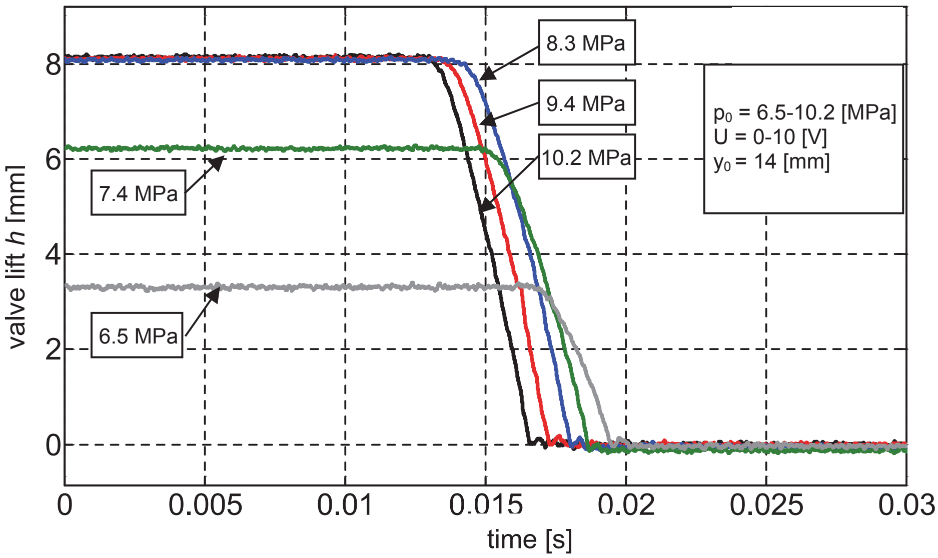

Fig. 8. Influence of supply pressure on waveforms of valve closing for bipolar signal

Rexroth servovalve it is a proportional valve in which the opening of gaps is proportional to the control signal. The study shows that the value of the control signal does not affect the valve opening time, but has great significance from the point of view of the time delay - from the start of control signal to the start of valve movement. For example for discrete control signals 5 and 10 V obtained delay difference equal to 1 ms, as shown in Fig. 9.

5. Conclusions

The paper presents the results of the research of chosen solution of electrohydraulic valve drive for internal combustion engine. This research was made on no-engine stand. This model of electrohydraulic valve drive was not a prototype solution, but the information gained during the research allowed to determine the basic properties of a drive of this type, which is important from the point of view of further work intended to lead to the development of prototype of the electrohydraulic valve drive for the internal combustion engines.

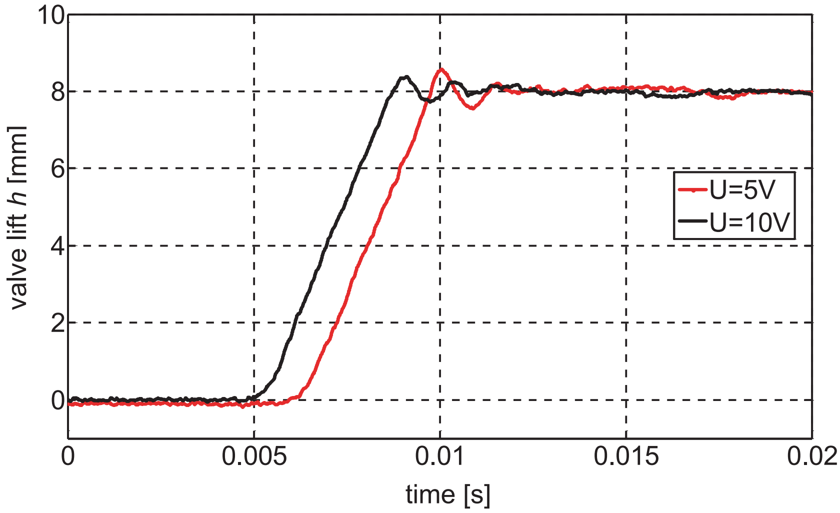

Fig. 9. Influence of the input signal on waveform of valve opening

This paper, due to the limitations of its volume, presents only selected results of research. However, they allowed for assessment of the analysed valve drive, and complete documentation of research was used to verify the developed earlier the mathematical model of this valve drive.
From the research the following general conclusions can be drawn:

– analysed drive was characterized by response delay. This response delay is consisted of: time between start of control signal and beginning the movement of the servovalve spool and the rise time of the corresponding actuator pressure, needed to move off the engine valve. The time for a specific pressure value is constant and the total is from no less than five milliseconds. For this reason, the control of this valve drive cannot be done in real time, after start of the control signal at the time corresponding to one position of the crankshaft,
– The best dynamic properties of this drive were obtained at pressures equal or greater than 10 MPa. With the increase of supply pressure the time to achieve full opening of the valve and actuator response delay were shortened,
– better dynamic properties of this drive were obtained at higher control signal voltages.

This general conclusions are only the most important stemming from the research. Detailed analyses of the work may be a basis for further research on the hydraulic valve drive for internal combustion engines.

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