EXPERIMENTAL RESEARCH ON SINGLE--ACTING HYDRAULIC DRIVE FOR VALVES OF INTERNAL COMBUSTION ENGINES

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Summary

Experimental research on single-acting hydraulic drive for valves of internal combustion (IC) piston engines and the test setup used for this purpose have been described. The test setup presented consisted of typical poppet valve assembly of a high-speed IC engine, commercial hydraulic actuator, and electrically operated hydraulic distributor, which controlled the flow of the working fluid supplied to the hydraulic actuator. In the test setup, a Caterpillar solenoid valve of the HEUI fuel system was used as the distributor. Time histories of the current control signal, valve displacement, and system working pressures for different values of the hydraulic actuator supply pressure and for different distributor opening time intervals were measured and analysed, with results of the measurements and analyses having been presented in this paper. Based on the valve displacement vs. time curves recorded, the valve kinematics has been analysed in details. The properties of the valve drive system of this type have been discussed and values of the basic motion parameters characterizing this drive have been given. Special attention has been drawn to the valve motion delay measured from the beginning of the current control signal, valve opening and closing times, velocities, and accelerations, and impact of the actuator supply pressure on the valve motion. The results obtained were taken as a basis for developing a model of the valve drive system of this type. Further simulation tests will be needed to assess the applicability of such a valve drive system to IC piston engines. They would also enable the comparing of the solution proposed with the hydraulically operated engine valve actuation systems known from the literature.

Keywords: internal combustion engine, camless valve drive, hydraulic drive

1. Introduction

The idea of hydraulic actuation of engine valves has been known for a long time. Research on electrohydraulic valve timing systems is also being done by various academic and industrial research and development centres [1, 2, 3, 4, 5].

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During authors' own work [6, 7, 8, 9, 10] carried out on the hydraulic actuation and timing of IC piston engine valves, a single-acting hydraulic drive system was tested as one of the possible options. A necessary mathematical and simulation model of such a system was developed [6]. An experimental functional model of the system [8] was also built for experimental and verification needs. Thanks to that, some preliminary test results could be obtained. However, the limitations and simplifications of the model as against a practically usable system made it impossible to determine all the parameters that would be necessary for the work to be continued. Therefore, a decision was made to modify the existing test setup, to repeat selected tests, and to analyse the tests in details. The tests carried out were aimed at exploring the properties of the single-acting hydraulic drive system, especially the system made with the use of components of diesel engine fuel injection systems. Thanks to them, the basic parameters of motion of a typical (but hydraulically operated) poppet valve assembly of a high-speed IC engine could be determined in quantitative terms, which was indispensable for verification of the mathematical model developed previously. The model thus verified will be subsequently used for simulation tests necessary to assess the applicability of such valve drive systems to IC piston engines. It will also make it possible to compare the solution proposed with the hydraulically operated engine valve actuation systems known from the literature.

2. Test setup

A concept of the test setup has been schematically shown in Fig. 1.

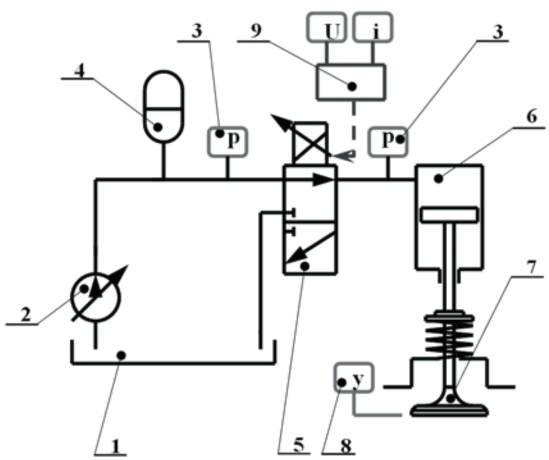

Fig. 1. Concept of the test setup:
1 – tank; 2 – supply pump; 3 – pressure sensor; 4 – hydro-pneumatic accumulator;
5 – solenoid valve (distributor); 6 – actuator; 7 – engine valve; 8 – valve displacement sensor;
9 – triggering system

The test setup (Fig. 2) was intended to produce single strokes of the engine valve and to measure basic parameters of operation of the single-acting hydraulic drive system, such as voltage and amperage of the distributor control signal, working fluid pressure, and valve displacement. The basic component of the valve drive system under investigation was a single-acting hydraulic actuator **6,** which opened the IC engine valve **7.** The valve was closed by a return spring. The actuator operation was controlled by distributor **5,** which connected the actuator with the supply or return line, depending on the control signal. The other major components of the test setup included tank **1,** supply pump **2,** and hydro-pneumatic accumulator **4** used to reduce the supply pressure fluctuation.

In the test facility shown in Fig. 2, oil is drawn by a Bosch high-pressure supply pump **2** and forced via the distribution pipe of a Common Rail system to accumulator **4** (of 1 dm^3 capacity) and then, via distributor **5,** to hydraulic actuator **6,** which controls the motion of

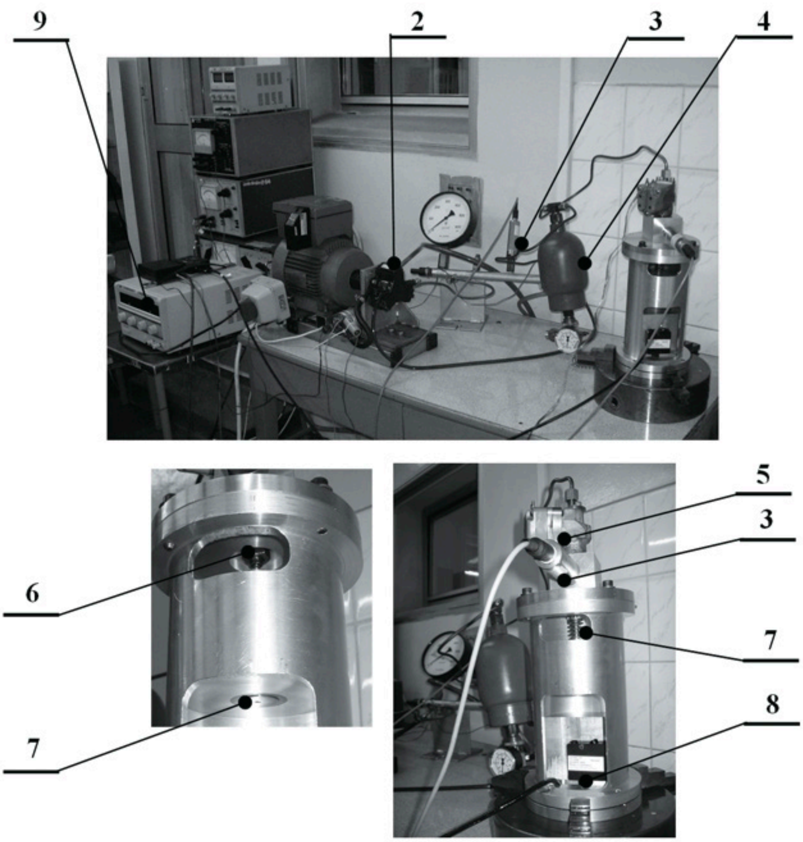

Fig. 2. The setup for testing the single-acting hydraulic drive system:
2 – supply pump; 3 – pressure sensor; 4 – hydro-pneumatic accumulator; 5 – solenoid valve (distributor); 6 – actuator; 7 – engine valve; 8 – valve displacement sensor; 9 – triggering system

valve **7.** The system was provided with a hydro-pneumatic accumulator filled with nitrogen in order to reduce pressure fluctuation in the oil supply line.

Since the electrohydraulic valve timing system was to be built with the use of components of fuel supply systems of diesel engines, according to one of the concepts considered [10], a Caterpillar solenoid valve of the high-pressure HEUI (Hydraulically actuated Electronic Unit Injector) system of a diesel engine was chosen to be used as distributor **5.** It is supplied with diesel oil at a pressure kept within limits from 4 to 21 MPa. As the hydraulic actuator, a pump and plunger unit of an in-line injection pump was adapted. The valve with a return spring was replaced with a system of identical functionality. The distributor was controlled by a pulse control system based on the discharging of a capacitor [10, 8].

The pulse control signal was generated by discharging capacitors of various capacitance values with the use of triggering system **9.** The triggering system additionally made it possible to measure the voltage and amperage of the distributor control current. The valve displacement was measured with optical displacement sensor **8.** Pressure sensors **3** were used to measure the oil pressure. Individual system components were connected together by means of standard pipes normally used in diesel engine fuelling systems. The working medium was engine oil of a popular grade.

When the test setup was modified, the pump and plunger unit of an injection pump was replaced with a commercial hydraulic actuator; moreover, a typical poppet valve assembly complete with a valve spring of a high-speed IC engine was used in place of the functional model [8]. The whole system was housed in a test sleeve discussed in details in [9]. The Caterpillar distributor **5** was installed on the test sleeve.

The basic technical specifications of selected components of the test setup have been given bellow.

High-pressure supply pump

A high-pressure pump used in diesel engine fuel systems, model BOSCH 117CN8, driven by an electric motor.

Hydraulic actuator

ROEMHELD B 1.458 single-acting hydraulic cylinder with a return spring. For the tests, the return spring was removed from the cylinder and the piston was returned by the engine valve, which was closed by the valve spring.

The technical characteristics of the critical measuring equipment items have been specified below.

Pressure sensors

Two piezoelectric membrane sensors, manufactured by HYDROTECHNIK GmbH, model PR 15, with a measuring range from 0 to 400 bar.

Displacement sensor

Laser sensor model LD 1607 20, with a measuring range of up to 20 mm, measurement frequency of up to 10 kHz, and resolution of 6 μm.

Data acquisition system

Personal Daq/3000 Series IOtech multifunction module, computer with a Windows XP Professional operating system. The module had 16 analog inputs, 4 digital inputs, and 2 analog outputs. Maximum measuring frequency for one channel: 1 MHz.

The measuring equipment used made it possible to measure and archive the following quantities:
- Time history of the voltage of the distributor control signal;
- Time history of the amperage of the distributor control signal;
- Valve displacement;
- Pressure in the working chamber of the actuator;
- Oil supply pressure.

3. Bench test results with an analysis

The tests of the single-acting hydraulic drive system with a Caterpillar distributor were carried out with the essential test parameters being as specified below.
- Valve stroke: 8 mm;
- Wide range of oil supply pressure values: from 2 to 16 MPa;
- Initial deflection of the valve spring, constant for all the tests: 3.9 mm;
- Different lengths of the distributor control signal that was induced by pulse discharging of capacitors of various size, i.e.: 330, 530, 800, and 1 000 μF (at the valve displacement tests, the energy accumulated in the 330 μF capacitor was insufficient to open the distributor).

An example set of test results has been given in Fig. 3. The graphs represent time histories of valve displacement (H), velocity (V), and acceleration (A), as well as oil supply pressure (Z), oil pressure in the actuator (S), and amperage of the current control signal (I). Additionally, the data presented in the figure include calculated valve velocity and acceleration values, in particular maximum valve opening velocity and acceleration (v_o and a_o, respectively), maximum valve closing velocity and acceleration (v_z and a_z, respectively), and valve impact velocity (i.e. velocity of the valve hitting its seat when being closed, v_u).

When analysing the curves shown in Fig. 3, we can notice that at the oil supply pressure of about 10 MPa, the time shift top between the beginning of the current control signal (curve I in Fig. 3) and the beginning of valve motion (curve H in Fig. 3) was about 10 ms. The valve opening time to until the valve displacement reached full valve lift value was about 4 ms and the valve closing time was about 3 ms. The maximum valve velocity and acceleration values were v_o = 3.4 ms and a_o = 3 300 m/s^2, respectively. Similarly during the return movement of the valve, the maximum valve velocity and acceleration values were

v_z = 4.5 ms and a_o = 3 200 m/s_2, respectively. The velocity of the valve hitting its seat during the return movement was vu = 1.4 m/s.

In the final valve opening phase, the valve opening displacement can be seen to be slightly bigger than that allowed by the actuator stroke. This may be explained by a loss of contact between the actuator piston rod and the valve stem. The current control signal amperage vs. time curve was discussed in details in the previous publication [8].

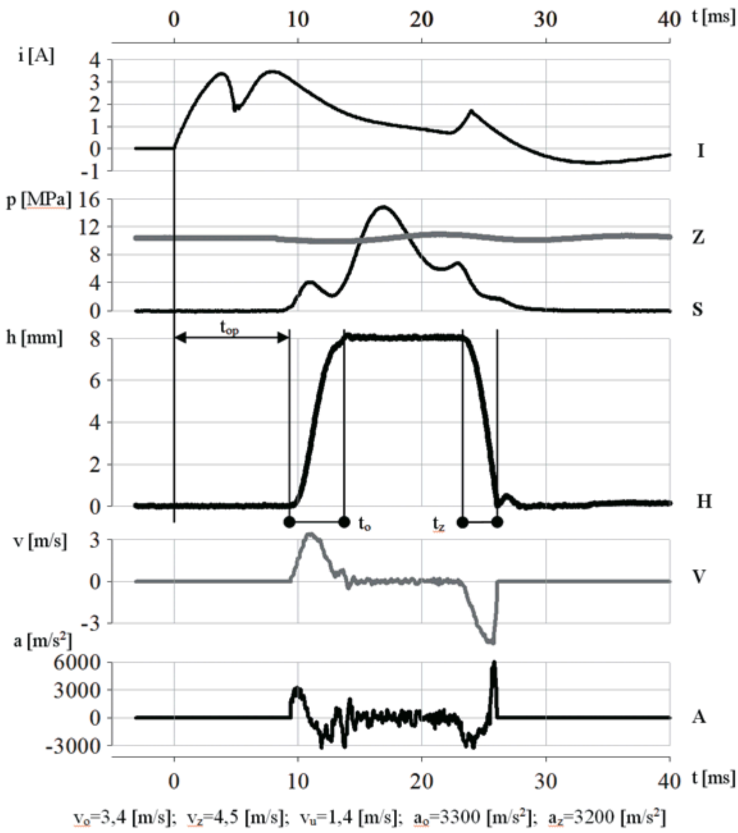

Fig. 3. Example time histories of current control signal, parameters describing the valve kinematics, and oil pressures recorded during the bench tests (U_0 = 24 V, C = 1 000 μF, p_0 = 10 MPa): I – current control signal; Z – oil supply pressure; S – oil pressure in the actuator; H – valve displacement; V – valve velocity; A – valve acceleration

The results of the tests carried out have been brought together in a comprehensive form in Table 1. When analysing the measured test results, we can notice the following.

Table 1. Time and valve kinematics parameters recorded during the tests

p_0	h_{max}	t_{op}	t_o	v_o	a_o	t_z	v_z	a_z
[MPa]	[mm]	[ms]	[ms]	[m/s]	[m/s²]	[ms]	[m/s]	[m/s²]
16	8	6	2.5	4.8	3 800	3	4.0	5 200
15	8	7	2.5	4.5	4 200	3	4.0	5 400
14	8	7	2.5	4.3	3 400	3.5	4.0	4 900
13	8	7	3	4.2	4 100	3	4.1	4 900
12	8	7	3	4.0	3 700	3.5	4.3	3 500
11	8	7	4	3.4	3 400	3.5	4.2	3 500
10	8	7.5	4	3.4	3 300	3	4.5	3 200
9	8	9	4.5	3.0	3 400	3	4.3	4 000
8	8	9	5	2.9	2 300	3	4.6	3 900
7	8	9	5	2.4	2 400	3	4.3	3 400
6	8	10	7	2.1	2 200	3	4.4	2 800
5	6	11	10	1.7	2 000	2.5	3.9	2 700
4	5	12	10	1.3	1 500	2.0	3.4	2 600

Legend: p_0 – oil supply pressure; h_{max} – maximum valve stroke; t_{op} – valve motion delay from the beginning of the current control signal; t_o – valve opening time; v_o – maximum valve opening velocity; a_o – maximum valve opening acceleration; t_z – valve closing time; v_z – maximum valve closing speed; a_z – maximum valve closing acceleration

Delay in the valve motion, measured from the beginning of the current control signal

Along with growing oil supply pressure, the delay from the beginning of the current control signal to the beginning of the valve motion increased, which has been shown in Fig. 4.

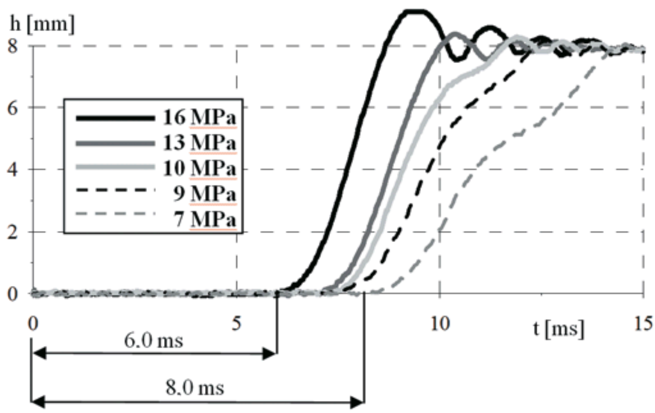

Fig. 4. Influence of the oil supply pressure on the valve motion delay (C = 1 000 μF)

In particular, this delay could be 6 ms at the oil supply pressure of about 15÷16 MPa; when the pressure was 6 MPa (at which full valve lift still could be achieved), it could extend to about 10 ms. When the supply pressure dropped below a level of about 5÷6 MPa (and, in consequence, the full valve lift could not be developed), the delay exceeded 10 ms.

Maximum valve displacement

The oil supply pressure was sufficient for the full valve lift of 8 mm to be achieved (see Table 1). Only the supply pressure values lower than 6 MPa were inadequate for the valve displacement to develop to the full valve lift.

Valve opening time and velocity

The valve opening time declined with increasing oil supply pressure, as it can be seen in Fig. 5.

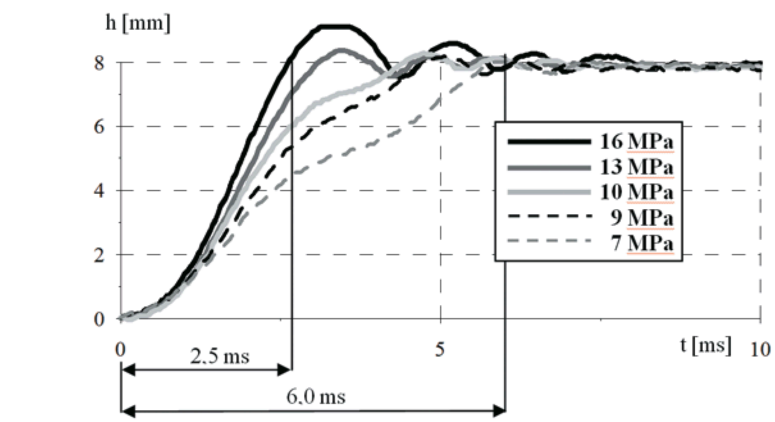

Fig. 5. Influence of the oil supply pressure on the valve opening process (C = 1 000 μF)

At high oil supply pressures (about 15÷16 MPa), the valve opening time happened to be even as short as 2.5 ms. When the supply pressure values were low (about 6 MPa), the valve opening time could increase to 7 ms. The maximum valve opening velocities and accelerations changed proportionally to the supply pressure. For lower supply pressures, a distinct inflection can also be seen in the valve displacement vs. time curve (P). This can be explained by insufficient flow through the instantaneous gap in the distributor supply path in relation to the actuator absorbing capacity resulting from the actuator piston movement. As an example: for the supply pressure below 10 MPa, the inflection point appeared for a valve displacement of about 7 mm; when the supply pressure was 7 MPa, the position of the inflection point corresponded to a valve displacement being as low as about 5 mm. This resulted in a distinct increase in the valve opening time, i.e. this time grew by about 1÷2 ms.

The influence of the oil supply pressure on the valve motion delay and the valve opening time has also been shown in Fig. 6. The valve motion delay and valve opening time vs. oil supply pressure curves were approximated with linear functions (within the range of the preset test pressures).

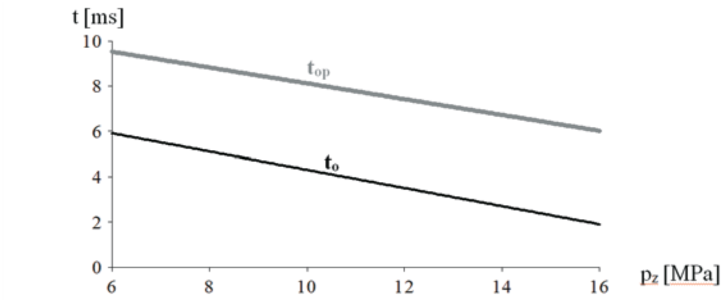

Fig. 6. Linearly approximated influence of the oil supply pressure on the valve motion delay and valve opening time: t_{op} – valve motion delay in relation to the current control signal; t_o – valve opening time

Valve closing time and velocity

The valve closing time was approximately constant and remained on a level of about 3÷3.5 ms. Only at low supply pressures, when the full valve lift could not be achieved, the closing time shortened. Example time histories of the valve closing process have been shown in Fig. 7.

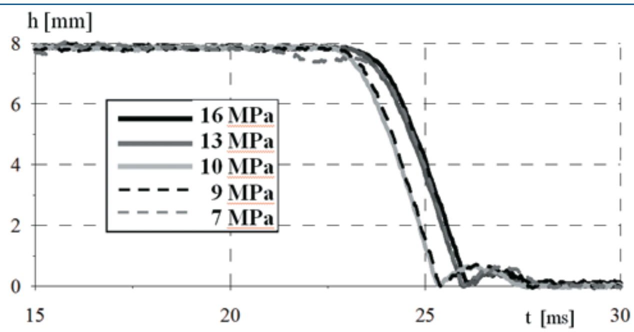

Fig. 7. Influence of the oil supply pressure on the valve opening process (C = 1 000 μF)

Fig. 7 shows that the oil supply pressure virtually did not affect the valve closing process, the kinematic parameters of which did not show significant differences. The velocity of the valve hitting its seat during the return movement was almost identical, and so was the course of the valve bounce. The minute differences and certain phase shifts resulted from non-repeatability of successive realizations of the valve displacement process.

Influence of the control signal

The influence of the control signal on the valve displacement vs. time curves can be observed in Fig. 8.

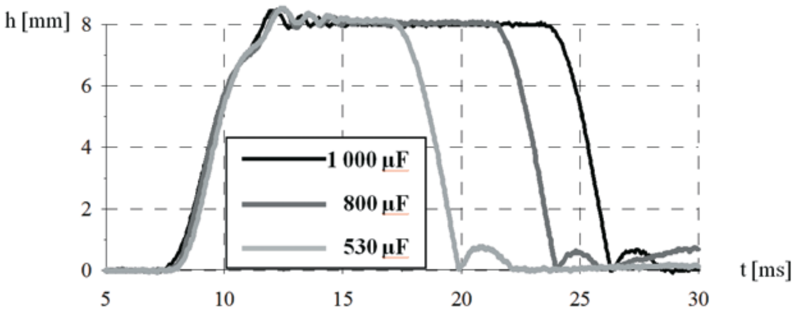

Fig. 8. Influence of the control signal on the valve displacement vs. time curves (p_0 = 10 MPa)

The pulse control system based on the discharging of capacitors of various capacitance values (actually, an RLC-type system) made it only possible for the operator to have an impact on the time of duration of the current control pulse (and within a limited scope at that). It can be very clearly seen in Fig. 8 that the opening slopes (leading edges) of the valve displacement vs. time curves plotted for different capacitors at a constant oil supply pressure virtually coincided with each other. The smaller capacitors were used, the less energy was accumulated in them and, in consequence, the shorter the periods were during which the valve remained open. The closing slopes (trailing edges) were also virtually identical, if superimposed on each other. For the smallest capacitor (330 µF), the energy accumulated in it was so small that no valve displacement was detected.

The velocity of the valve hitting its seat during the return movement

When the full valve lift (8 mm) was achieved, the valve head hit its seat with a velocity of about 1÷1.3 m/s, or even 1.4÷1.5 m/s in isolated cases, during the return movement, according to the test results. Such valve impact velocities were confirmed by the noise clearly heard during the tests. The recorded valve impact velocities only declined below 1 m/s at lower maximum valve displacement values (i.e. at partial valve lifts). As an example, for maximum valve lift values of about 3 mm and 1.2 mm, the valve impact velocity was 0.8 m/s and 0.4 m/s, respectively. These very high valve impact velocity values will significantly affect the durability and noise of valve train operation; therefore, a system to slow down the valve motion when approaching the seat will have to be applied. Such a system should reduce the valve head impact velocity below a level of 0.1 m/s.

4. Recapitulation

The paper presents results of testing a selected solution of the electrohydraulic system actuating an IC engine valve in an engine-less test setup. This model electrohydraulic valve actuation system should by no means be considered a prototype solution. Nevertheless, the information gained from the testing of such a system has made it possible to determine the basic properties of the valve actuation system type under investigation, which is important from the point of view of further work aimed at the development of a prototype electrohydraulic system of actuation of an engine valve.

The material presented provides grounds for determining the impact of selected parameters of an electrohydraulic system actuating an IC engine valve on the properties of such a system. Thanks to the tests carried out, the following general conclusions may be formulated:

- The valve drive system under investigation was characterized by valve actuation delay, which consisted of the time elapsing from the beginning of the control signal to the start of movement of the solenoid valve spool and the time of building up an actuator pressure necessary to move the engine valve. For specific oil supply pressure values, this time was constant and totalled from six to even more than ten milliseconds. For this reason, such a system cannot be controlled in real time, after the control signal is applied at an instant corresponding to a specific position of the engine crankshaft.
- The best dynamic properties of the valve drive system were obtained when the oil supply pressure was not less than 10 MPa. Along with growing values of this pressure, the period of full valve opening and the valve actuation delay shortened. On the other hand, no significant impact of this pressure on the valve closing time was observed.

Only the most important general conclusions drawn from the investigations carried out have been presented above. The detailed analyses described in this paper may constitute a basis for further research on the hydraulic drives for IC engine valves.

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