

VERIFICATION OF THE MODEL OF SINGLE-ACTING HYDRAULIC DRIVE FOR CYLINDER VALVES OF INTERNAL COMBUSTION ENGINES

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Summary

An extended and verified model of single-acting hydraulic drive for cylinder valves of internal combustion (IC) piston engines has been presented. In a physical model of the valve drive system proposed, a Caterpillar electric control valve from the HEUI fuel system was used as a distributor and the function of an actuator was performed by a commercially available hydraulic cylinder. Experimental and verification tests were carried out on the physical model built on a test stand. In the model prepared, electrical, mechanical, and hydraulic parts may be discerned. The model of the valve drive system was divided into sub-models representing the control system, distributor electromagnet solenoid with variable inductance, distributor spool with the electromagnet armature, hydraulic actuator, valve with valve spring, distributor working chamber, flows through the supply gap and drain gap in the distributor, and the actuator working chamber. The models, whether originally prepared or extended during the verification tests, as well as the verification of the sub-models representing the control system and distributor electromagnet solenoid have been discussed in detail. The structure of the entire model and the relationships between input and output data of individual sub-models have been shown. The profiles of voltage and current of the control signal as well as the profiles of valve displacement and other valve motion parameters obtained from experiments have been compared with the corresponding profiles obtained from simulations. Good consistence between the experimental and simulation profiles of the valve displacement was obtained. The model thus prepared and verified became a tool for examining the properties of a single-acting hydraulic drive for the valves of internal combustion engines.

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

1. Introduction

During the work carried out on the hydraulic actuation and timing of IC piston engine valves [1], a single-acting hydraulic drive system was tested as one of the possible options [7, 8,

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9, 10]. A mathematical model of such a system was developed, which was presented in one of the papers published previously [2]. The basic component of such a valve drive system was a single-acting hydraulic actuator, which opened the IC engine valve. As a control valve, a solenoid valve taken from the high-pressure injection unit of a diesel engine was selected. The valve was closed conventionally, i.e. by a return spring. The system was also provided with a hydro-pneumatic accumulator to reduce pressure fluctuation in the oil supply line.

For the needs of experiments and model verification tests, a test stand was built, thanks to which the model prepared could be verified. The test stand as well as the test results and conclusions were presented in another one of the previous papers [3]. The test stand made it possible to produce single strokes of the engine valve and to measure basic parameters of operation of the single-acting hydraulic drive system. In the test setup, a Caterpillar solenoid valve of the high-pressure HEUI (Hydraulically actuated Electronic Unit Injector) system of a diesel engine was used as a distributor. As the hydraulic actuator, a commercially available hydraulic cylinder was applied. Individual system components were connected together by means of standard pipes normally used in diesel engine fuelling systems. The working medium was engine oil. Finally, a typical poppet valve assembly of a high-speed IC engine was used in the test setup.

During the research work, it was found necessary to modify and extend the model originally prepared [2]. This paper deals with the extended and verified model of single-acting hydraulic drive for valves of internal combustion engines.

2. Model of single-acting hydraulic drive

Consistently with the hydraulic drive concept having been worked out and the test stand having been built, the model prepared was composed of the following parts:

- Electrical part (control system and distributor electromagnet solenoid);
- Mechanical part (distributor electromagnet armature with the distributor spool as well as hydraulic actuator with the engine poppet valve and valve spring);
- Hydraulic part (oil supply line, distributor chamber and gaps, hydraulic actuator chamber, hydraulic passages, and connecting pipes).

In consideration of the hydraulic drive concept adopted and the sequence of operations of the experimental verification of the entire model and its component sub-models, the following model structure was adopted for further analysis, with using the symbols as boldfaced below:

- Model of the control system (used at the tests) - **RLC**;
- Model of the electromagnet solenoid (with variable solenoid inductance) - **LC**;
- Model of the electromagnet (representing the cooperation between the solenoid and the armature) - **EM**;
- Model of the distributor spool (connected with the electromagnet armature) - **SU**;

- Model of the supply gap (conical) of the distributor – **SO**;
- Model of the drain gap (cylindrical) of the distributor – **SZ**;
- Models of the oil flow through the supply gap and drain gap – **QO** and **QZ**, respectively;
- Model of the working chamber (distributor and connecting pipe with actuator) – **RA**;
- Model of the engine valve (connected with the hydraulic actuator piston rod) – **SH**.

The mathematical and simulation sub-models being components of the entire model of the single-acting hydraulic drive system have been discussed below. The sub-models and issues that were not covered in the previous publications have been discussed here in detail.

Model of the control system (RLC) used at the tests

During the tests, the hydraulic distributor (the Caterpillar solenoid valve) was controlled by an electric system, the schematic diagram of which has been shown in Fig. 1. The system was designed with an assumption having been made that the voltage necessary to open the distributor with a short distributor opening time would be generated by discharging a capacitor when the electric circuit would be closed by switch *W*. Thus, the test control system is an RLC circuit (with solenoid resistance of R [Ω], internal (additional) resistance of R_w [Ω], solenoid inductance of L [H], and capacitor capacitance of C [F]), where the time histories of the distributor control current and the voltage across capacitor terminals depend on characteristics of the circuit components and voltage U_0 of the capacitor charging source.

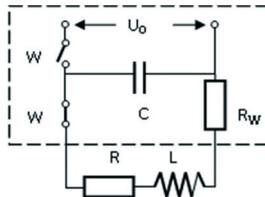


Fig. 1. Electrical diagram of the test system used to control the hydraulic distributor: C – capacitor; R – solenoid resistance; R_w – additional resistance; L – solenoid inductance; W – switch

If the capacitor C charging time is long enough, the following equation holds in the **RLC** series circuit without the source of the electromotive force:

$$u_C(t) + u_L(t) + u_R(t) = 0 \text{ [V]} \quad (1)$$

If the values of voltage drop across the resistance, inductance, and capacitance are:

$$u_R(t) = R \cdot i \quad u_L(t) = L \cdot \frac{di}{dt} \quad u_C = \frac{1}{C} \cdot \int i \cdot dt \text{ [V]} \quad (2)$$

where C is the capacitor resistance [F], then the following integro-differential equation may be formulated for the **RLC** series circuit (which makes it possible to take into account the variability of the solenoid inductance L and the induction electromotive force e_b generated by solenoid armature movements):

$$\frac{d(L \cdot i)}{dt} + R \cdot i = U_0 - \frac{1}{C} \cdot \int i \cdot dt - R_w \cdot i - e_b \text{ [V]} \quad (3)$$

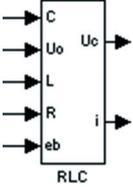
The right side of equation (3) represents the decreasing voltage across the distributor solenoid terminals when the capacitor is discharged.

Obviously, the initial conditions are as follows:

$$\begin{aligned} u_C(0) &= U_0 \text{ [V]} \\ i(0) &= 0 \text{ [A]} \\ q(0) &= C \cdot U_0 \text{ [C]} \end{aligned} \quad (4)$$

The inputs and outputs of the model of the control system (**RLC**) have been shown in Table 1.

Table 1. Inputs and outputs of the model of the control system (RLC)

	<p>Inputs:</p> <ul style="list-style-type: none"> C - capacitor capacitance U_0 - capacitor charging voltage L - solenoid inductance (varying with armature movements) R - solenoid resistance e_b - electromotive force in the solenoid (generated by armature movements) <p>Outputs:</p> <ul style="list-style-type: none"> U_c - voltage across the distributor terminals (auxiliary parameter) i - current in the distributor solenoid
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Model of the electromagnet solenoid (LC)

The dynamic behaviour of the solenoid was described with the use of the magnetic non-linear Cheung model, which was employed when diesel engine injectors were modelled [5, 6].

The solenoid force F_e and inductance L_e , which depend on the current and electromagnet armature displacement, may be expressed with the use of the following equations:

$$F_e(x, i) = \frac{\mu_0 \cdot N^2 \cdot A \cdot i^2}{x_e^2} \text{ [N]} \text{ and} \quad (5)$$

$$L_e(x) = \frac{\mu_0 \cdot N^2 \cdot A}{\frac{s}{\mu_r} + x_e} \text{ [H]}, \quad (6)$$

where: x_e – displacement of the solenoid armature (and distributor spool as well) [m]; i – current in the electromagnet solenoid [A]; N – number of solenoid coils [-]; A – solenoid cross-section area [m²]; s – length of the return magnetic path [m]; μ_0 – magnetic permeability of free space [H/m]; μ_r – relative magnetic permeability [-].

If the following symbols are used to denote the constant values:

$$A_e = \mu_0 \cdot N^2 \cdot A, \quad (7)$$

$$B_e = \frac{S}{\mu_r} \quad (8)$$

and in consideration of the fact that a solenoid without an armature has an inductance of a non-zero value L_0 , the formulas necessary to calculate the inductance may be modified to the form:

$$F_e(x, i) = A_e \cdot \frac{i^2}{x_e^2} \text{ and} \quad (9)$$

$$L_e(x) = \frac{A_e}{B_e + x_e} + L_0 \quad (10)$$

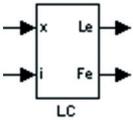
Obviously, the relationship between the distributor spool displacement x and the distance between the armature and the electromagnet solenoid x_e is:

$$x_e = x_0 + x_{mks} - x \quad (11)$$

where: x_0 – distance between the armature and the electromagnet if the distributor spool is in its open position [m]; x_{mks} – distance between the armature and the electromagnet if the distributor spool is in its closed position [m].

The inputs and outputs of the model of the electromagnet solenoid (**LC**) have been shown in Table 2.

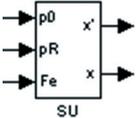
Table 2. Input data and results obtained for the model of the electromagnet solenoid (LC)

	<p>Inputs:</p> <p>x – displacement of the solenoid armature (and distributor spool) from the closed position</p> <p>i – current in the solenoid</p> <p>Outputs:</p> <p>L_e – variable solenoid inductance</p> <p>F_e – variable force attracting the armature with spool to the solenoid</p>
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Model of the distributor spool (SU)

The model of the distributor spool was discussed in detail in a paper published previously [2]. The inputs and outputs of the model of the distributor spool (**SU**) have been shown in Table 3.

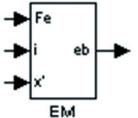
Table 3. Inputs and outputs of the model of the distributor spool (SU)

	<p>Inputs:</p> <p>p_0 – distributor supply pressure</p> <p>p_R – pressure in the working chamber of the spool</p> <p>F_e – force generated in the electromagnet solenoid</p> <hr/> <p>Outputs:</p> <p>x' – spool velocity</p> <p>x – spool displacement</p>
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Model of the electromagnet (EM)

The model of the electromagnet was discussed in detail in a paper published previously [2]. The inputs and outputs of the electromagnet model (**EM**) have been shown in Table 4.

Table 4. Inputs and outputs of the electromagnet model (EM)

	<p>Inputs:</p> <p>F_e – force generated in the solenoid</p> <p>i – current in the solenoid</p> <p>x' – spool velocity</p> <hr/> <p>Outputs:</p> <p>e_b – induction electromotive force generated during movements of the steel armature in the magnetic field of the solenoid</p>
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Model of the mechanical and electrical part of the distributor (RO)

The model of the mechanical and electrical part of the distributor consists of the sub-models presented above, i.e. models of the test control system (**RLC**), solenoid (**LC**), electromagnet (**EM**), and distributor spool (**SU**). The structure of the model and the relationships between the component sub-models have been shown in Fig. 2. The inputs and outputs of the model denoted by **RO** have been shown in Table 5.

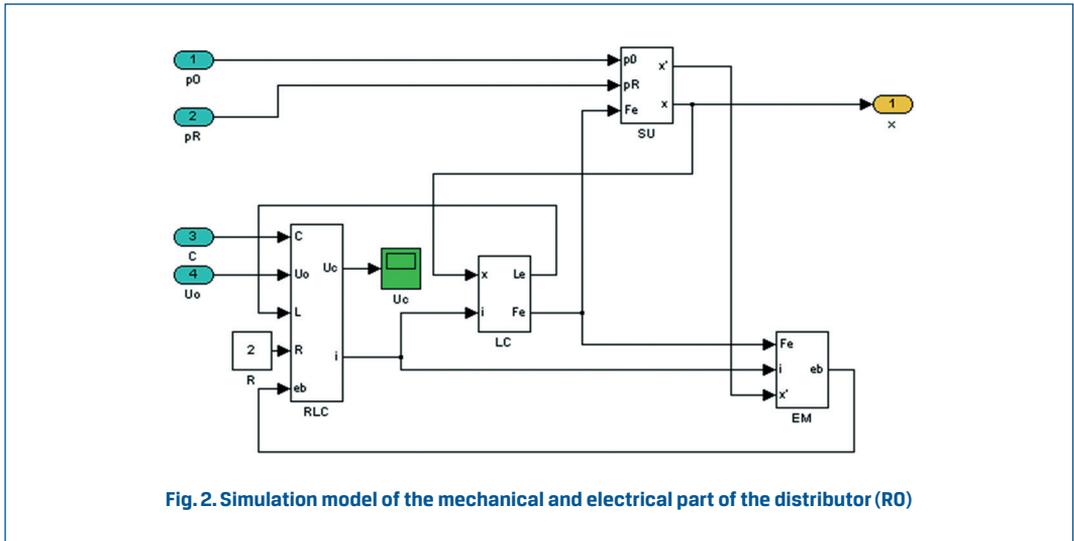


Fig. 2. Simulation model of the mechanical and electrical part of the distributor (R0)

Table 5. Inputs and outputs of the model of the mechanical and electrical part of the distributor (R0)

	<p>Inputs:</p> <ul style="list-style-type: none"> p_0 – distributor supply pressure p_R – pressure in the working chamber of the spool C – capacitor capacitance U_0 – capacitor charging voltage <p>Outputs:</p> <ul style="list-style-type: none"> x – displacement of the distributor spool
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Model of the engine valve with the actuator piston (SH)

The model of the engine valve with the piston of the hydraulic actuator was discussed in detail in paper [3]. In this model, the movement of the actuator piston together with the engine valve is calculated, with the effect of the valve hitting its seat during the return movement being also taken into account. The inputs and outputs of the model denoted by SH have been shown in Table 6.

Table 6. Inputs and outputs of the model of the engine valve with the actuator piston (SH)

	<p>Inputs:</p> <ul style="list-style-type: none"> p_R – pressure in the working chamber of the spool <p>Outputs:</p> <ul style="list-style-type: none"> y – displacement of the actuator piston with the engine valve y' – velocity of the actuator piston with the engine valve
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Model of the hydraulic part of the valve drive system (HY)

A one-dimensional model based on the mass conservation equation was used, where the compressibility of liquids was taken into account. This model and its component sub-models were described in detail in a paper published previously [2]. In the test stand having been built, the arrangement of system components made it possible to avoid the use of a connecting pipe, thanks to which the model was simplified – see Fig. 3.

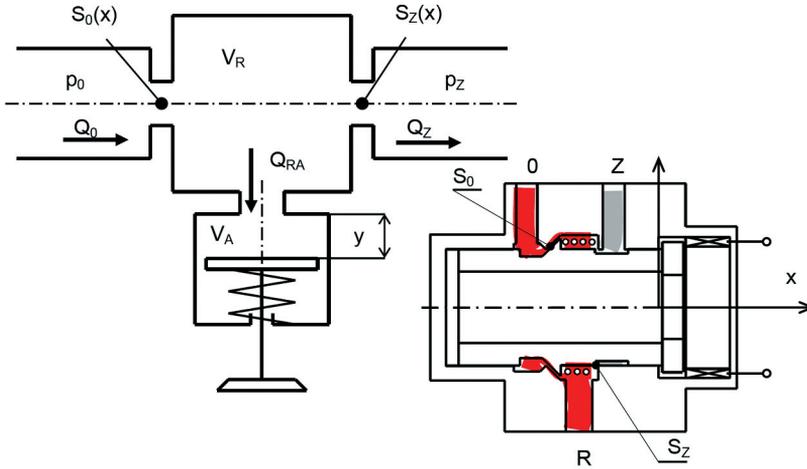


Fig. 3. Schematic diagram of the hydraulic part of the physical model of the single-acting valve drive:
V – volumes; p – pressures; S – gaps; Q – oil flow rates;
Subscripts: A – actuator; 0 – supply; R – outlet to the actuator (working chamber); Z – drain

Models of the distributor gaps (S0 and SZ)

The distributor supply line is connected with the working chamber in the distributor via a conical seat of the closing part of the spool (with the gap cross-section area S0 varying with the spool displacement). The working chamber in the distributor is connected with the drain line via a cylindrical gap (with the gap cross-section area SZ varying with the spool displacement). The gap models were discussed in detail in a paper published previously [2]. The inputs and outputs of the gap models have been shown in Table 7.

Table 7. Inputs and outputs of the models of gaps (S0 and SZ)

		Inputs: x – displacement of the distributor spool Outputs: S – gap cross-section area
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Models of the oil flow through distributor gaps (QO and QZ)

The models of the oil flow through distributor gaps were discussed in detail in a paper published previously [2]. The inputs and outputs of the models of oil flow through distributor gaps have been shown in Table 8.

Table 8. Inputs and outputs of the models of oil flow through distributor gaps (QO and QZ)

	<p>Inputs:</p> <ul style="list-style-type: none"> μ_0 – coefficient of effective cross-section area of the supply gap μ_z – coefficient of effective cross-section area of the drain gap S_0 – cross-section area of the supply gap S_z – cross-section area of the drain gap p_0 – distributor supply pressure p_R – pressure in the working chamber of the spool p_z – pressure in the drain line (atmospheric)
	<p>Outputs:</p> <ul style="list-style-type: none"> Q_0 – rate of the oil flow through the supply gap Q_z – rate of the oil flow through the drain gap

Model of the working chamber (RA)

Since adequate delay in the growth of pressure in the working chamber could not be achieved during the verification of research results, the model of the working chamber was modified so that the small leakage of the working fluid from the distributor chamber could be taken into account. Until the working chamber is 100% filled, the working fluid flows into it without causing a growth in the fluid pressure. Such a phenomenon was observed during the tests when the fluid was continuously leaking to the distributor drain. The models of the working chamber were discussed in detail in a paper published previously [2]. The inputs and outputs of the model of the working chamber have been shown in Table 9.

Table 9. Inputs and outputs of the model of the working chamber (RA)

	<p>Inputs:</p> <ul style="list-style-type: none"> Q_0 – rate of the oil flow through the supply gap Q_z – rate of the oil flow through the drain gap y' – velocity of the piston of the hydraulic actuator y – displacement of the piston of the hydraulic actuator wyp – working chamber fullness factor <p>Outputs:</p> <ul style="list-style-type: none"> p_R' – rate of changes in the working chamber pressure
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The models of gaps (**SO** and **SZ**), fluid flows through the gaps (**QO** and **QZ**), and working chamber (**RA**) taken together constitute a model of the hydraulic part of the valve drive system (**HY**). The structure of the simulation model of the hydraulic part of the system has been shown in Fig. 4. The gap models make it possible to calculate the rates of fluid flow through the supply gap and drain gap. Based on the flow rates and parameters of actuator

piston motion, the working chamber pressure p_R is calculated. The inputs and outputs of the model of the hydraulic part of the system have been shown in Table 10.

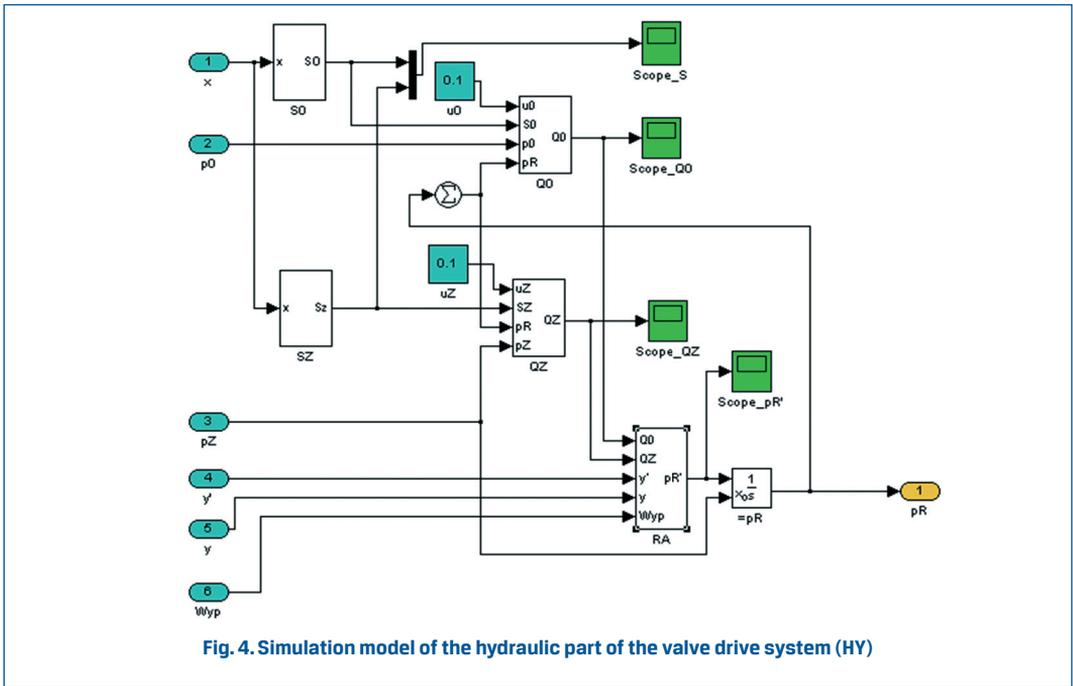
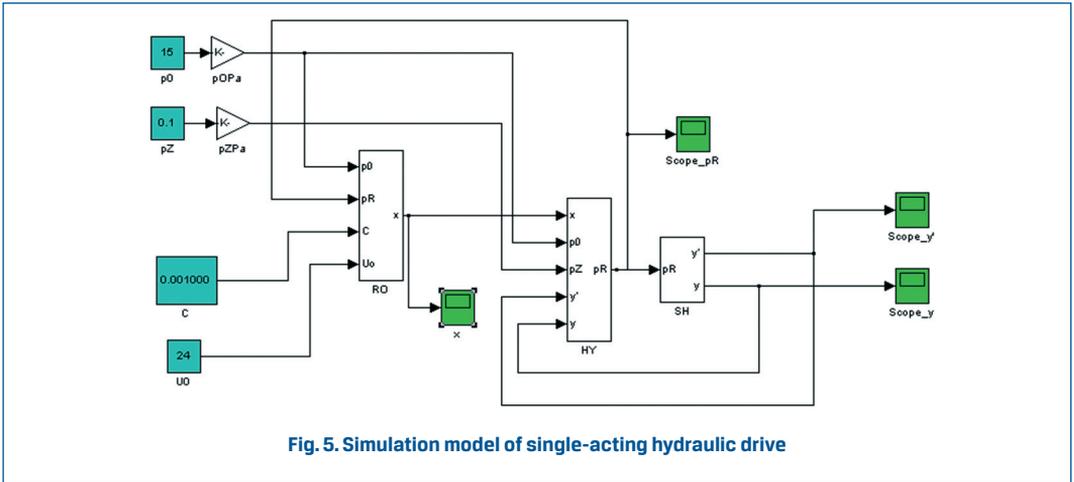


Table 10. Inputs and outputs of the model of the hydraulic part of the valve drive system (HY)

	<p>Inputs:</p> <ul style="list-style-type: none"> x – displacement of the distributor spool p_0 – distributor supply pressure p_z – pressure in the drain line (atmospheric) y' – velocity of the piston of the hydraulic actuator y – displacement of the piston of the hydraulic actuator wyp – working chamber fullness factor <p>Outputs:</p> <ul style="list-style-type: none"> p_R – pressure in the working chamber
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The entire model of single-acting hydraulic drive consists of the models discussed above and representing mechanical and electrical part of the distributor (**RO**), hydraulic part (**HY**), and hydraulic actuator (**SH**). The model structure has been shown in Fig. 5. The model of the mechanical and electrical part of the distributor (**RO**) returns the displacement of distributor spool (x) as an output. The output of the model of the hydraulic part (**HY**) is the pressure in the working chamber (p_R), which is taken by the actuator model (**SH**) as a basis for determining the actuator motion parameters (y and y'), which simultaneously are the parameters of motion of the engine valve.



3. Verification of the model of single-acting hydraulic drive

In this section, some verification examinations of the model of single-acting hydraulic drive and its sub-models have been discussed. The setups built to test the single-acting hydraulic drive system with a Caterpillar distributor as described previously in publication [3] made it possible to carry out the tests thanks to which the values of the basic coefficients of the mathematical model having been prepared could be determined and the verification tests of the model could be carried out on these grounds.

Verification of the model of the electromagnet solenoid (LC)

The constant initial solenoid inductance, independent of armature position, was experimentally determined as $L_0=4.46$ mH; the other constants were determined by approximation of equation (10) with using the least-squares method and their values were $A_e=4.91$ and $B_e=0.39$. The constant distance between the armature and the electromagnet x_0 was determined on the grounds of the measured inductance $L_{e0}=16.00$ and $L_{ez}=10.06$ of the distributor in its open and closed position, respectively, from the inverse calculation of distance x_e from equation (10).

Based on the calculations carried out, the initial armature distance was determined as $x_0=0.035$ mm and the calculated stroke of the armature (i.e. the spool as well) between the closed and open positions of the Caterpillar distributor was $x_{mks}=0.45$ mm, which simultaneously confirmed the data obtained from an analysis of the distributor design.

Verification of the model of the mechanical part of the distributor (RM)

During the verification, both the supply pressure p_0 and working pressure p_r were assumed as being equal to 0.1 MPa (the measurements were carried out without switching on the

supply pump). An example comparison between measurement and simulation test results has been shown in Fig. 6. With the supply system used at the tests, ideal conformity between the current profiles could not be achieved. However, the time discrepancies between the instants of both the opening and closing of distributor spool did not exceed 1–2 ms, which was considered as indicating the accuracy to be acceptable for the purposes of model testing, especially in consideration of the fact that the capacitor-based triggering system used at the tests was a provisional measuring system.

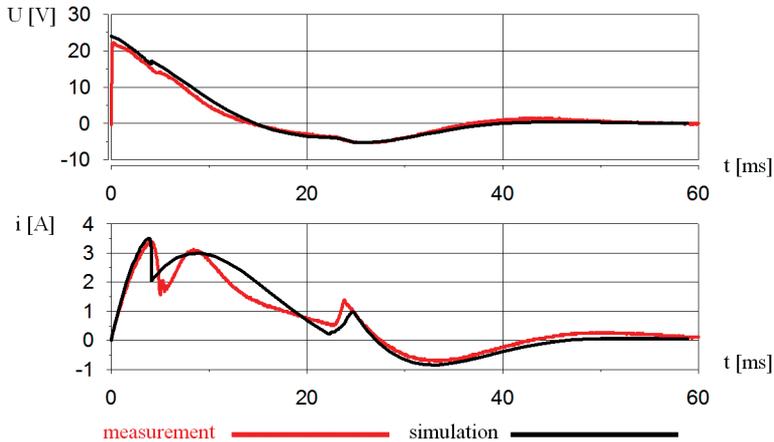


Fig. 6. Comparison between example results of simulation and experimental [3] tests of the mechanical part of the distributor (1 000 μ F, 24 V)

Verification of the model of single-acting hydraulic drive

Example comparisons between bench and simulation test results have been shown in Figs. 7 and 8.

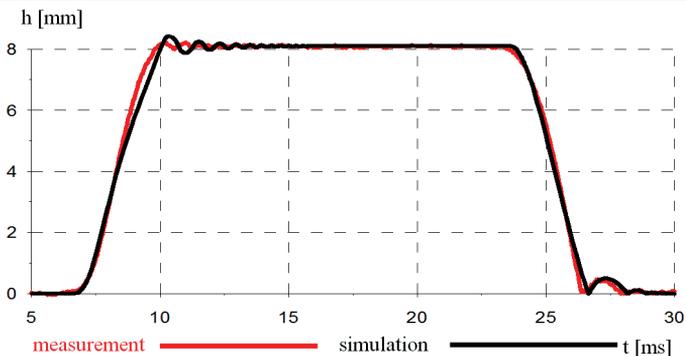


Fig. 7. Comparison between the valve displacement vs. time curves obtained from bench [3] and simulation tests ($p_0=10.0$ MPa, $C=1$ 000 μ F)

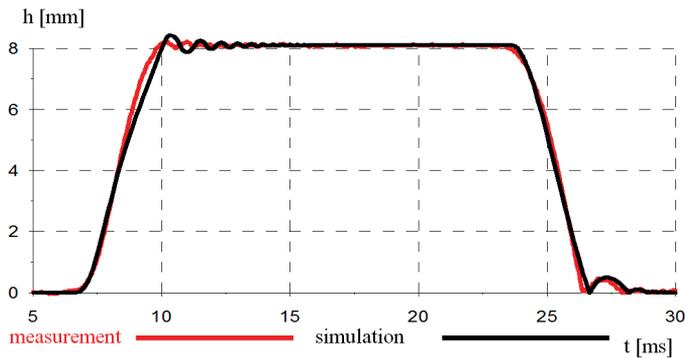


Fig. 8. Comparison between the valve displacement vs. time curves obtained from bench [3] and simulation tests ($p_0=15.0$ MPa, $C=1\ 000$ μ F)

The valve overshoot occurring when the valve displacement reached the maximum lift value and the valve impact against its seat during the valve return movement could not be perfectly reproduced. A detailed review of the results of bench tests revealed that the valve overshoots significantly differed from each other depending on the time of carrying out the measurements (measurement series). At some tests, the overshoot amplitudes were relatively small (like those shown in Fig. 7) while they were considerably bigger (like those shown in Fig. 8) at other tests. Similar differences were observed in the valve bouncing when the valve head hit its seat: in isolated cases, the bouncing was definitely smaller than usually and sometimes it was imperceptible at all. Therefore, the model coefficients were not individually determined for each measurement; instead, efforts were made to achieve high conformity of the valve displacement vs. time curves for the valve opening and closing processes.

During the verification tests, differences were found to occur in the curves representing changes in the fluid pressure in the actuator working chamber during the simulation tests and bench tests. The differences in the curve profiles resulted from the fact that the model was built without taking into account the wave phenomena that occur on the oil supply side when the engine valve is suddenly opened and reaches its maximum lift and the hydraulic actuator piston rod is violently stopped due to hitting its stop. The said wave phenomena strongly depend on pipe lengths and shapes and the observations are greatly affected by the location of measuring sensors. Although the impact of the wave phenomena was ignored in the model, however, satisfactory conformity was obtained between the valve displacement vs. time curves (during both the valve opening and closing process) recorded during the bench and simulation tests. The valve displacement vs. time curve was considered most important for the assessment of the hydraulic system of actuation and timing of IC engine valves; therefore, the model was considered satisfactory and was used for subsequent simulation tests.

4. Recapitulation

In this paper, a verified model of single-acting hydraulic drive for IC engine valves has been presented. The model was verified and modified in result of bench tests carried out with the use of a test stand specially prepared. The paper includes selected results of experimental verification of this model. The presented comparisons of selected parameters show good conformity between the time histories of the parameters recorded during the bench and simulation tests. Adequate conformity of the profiles of electric parameters was obtained for the control system. Similarly, the valve displacement vs. time curves representing the valve opening and closing processes during the simulation and bench tests were sufficiently close to each other and this was the most important criterion of evaluation of the model examined that is to represent the hydraulic drive for engine cylinder valves.

The model verification revealed discrepancies between the curves representing changes in the fluid pressure in the actuator working chamber during the simulation tests and bench tests. The differences in the curve profiles resulted from the fact that the model was built without taking into account the wave phenomena that occur when the engine valve is suddenly opened and the hydraulic actuator piston rod is violently stopped due to hitting its stop. A modification to the model for the wave phenomena to be taken into account may be expected to improve the model accuracy.

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