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EVALUATION OF DYNAMIC PARAMETERS OF THE SELECTED COMPONENTS WITHIN DIGITALLY CONTROLLED HYDRAULIC SYSTEM

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Abstract

Dynamic parameters characterizing proportional hydraulic components are essential factors affecting the performance of the construction machinery automatic control system. Knowledge of these parameters is inevitable in modelling the automatically controlled hydraulic power systems as well as designing and analysing the control algorithms. This article presents a methodology of assessing the dynamic parameters of selected components being part of digitally controlled hydraulic power system. The analysed system comprised a typical for mobile hydraulics applications PVG32 proportional directional control valve (DCV) and a variable displacement piston pump. The control system utilised an MC088 PLC controller with specifically prepared and configured software. The measuring system was based on CANbus, which, combined with the PLC used, allowed for flexible configuration of the sensor variables and logging both analogue and digital signals. Among others, the DCV characteristics, the response to step and sinusoidal inputs and DCV hysteresis were examined. The goal was to gather the data required for system modelling and to assess the importance and influence of the investigated parameters onto the model being created.

Keywords: *mobile hydraulics, proportional directional control valve, hydraulic cylinder control, electro-hydraulic pilot valve, hydraulic system modelling*

1. Introduction

In the wide scope, this article applies to the hydraulic power control systems that are, or potentially can be, installed in excavators for performing those tasks that require precise leading of the bucket edge in accordance with a predefined trajectory. The usage of such systems can cause several benefits, some of which are listed below:
– increased accuracy,
– reduction of working cycle duration,
– obtaining high accuracy regardless of operator's skills and experience,
– possibility of energy consumption optimization,
– performing tasks with reduced or lack of tool visibility,
– performing repeatable operations such as grading maintaining high accuracy and efficiency.

Additionally, this kind of systems can be installed for research and development reasons on laboratory stands or actual machines to assist in the tests that require repeatable tool movement, such as soil cutting process analysis.

The trend of implementing automated systems into construction machinery is present both in the industry [1] and in the scientific publications [2, 4]. The publications on this matter commonly regard to complex control algorithms. However, the results of detailed studies that relate to the dynamic parameters of the hydraulic components constituting hydraulic power control system of excavator being controlled, as well as their impact on the system performance, are scarcely available. For example [5] present simulation results of a hydraulic cylinder position control system involving a complex control algorithm (hybrid fuzzy-PID) with the dynamics of the DCV neglected. Experimental estimation of static and dynamic parameters of selected proportional DCV is described in [6] together with examples of how these parameters can be used for modelling the performance of a given hydraulic system. The presented results apply to a high dynamics DCV with parameters typical for industrial servo-valves. They are therefore not applicable to mobile hydraulic components.

The main goal of the research presented in this article was to gather data required for modelling a given hydraulic power control system built from components typical for mobile application. The research also involved:

– verification an supplementing of data provided by vendors,
– assessing the components' parameters individually and as part of the system,
– quantitative and qualitative assessment of the importance and influence of the investigated parameters onto the model being created.

2. System description

The research presented herein was carried out on a laboratory test stand at Institute of Construction Machinery Engineering, Warsaw University of Technology. The hydraulic schematic of the tested system (limited to the components used for this research) is shown in Fig. 1.

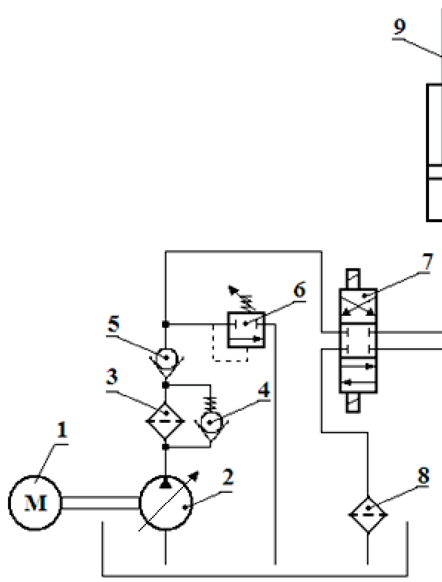

Fig. 1. Hydraulic system schematic: 1 – electric motor, 2 – variable displacement piston pump, 3, 4 – supply filter with bypass valve, 5 – check valve, 6 – relief valve, 7 – PVG32 DCV section, 8 – return filter, 9 – hydraulic cylinder

The graphical representation of the PVG32 [7] DCV is simplified, i.e. the stand-by, load sensing and pressure compensator circuits are not shown. Details are available in the manufacturer data sheet. Modular structure of these valves consists of replaceable electro-hydraulic PVES [8] pilot valves (one per each section of the valve block). The pilot valves are responsible for positioning of the valve spool in relation to the control signal voltage. The one used in this research provided also a spool position feedback signal, therefore allowed for spool position logging.

The hydraulic supply to the test stand was delivered by an A10VSO [9] variable displacement piston pump controlled by VT 5041 [10] amplifier card, which allowed for logging of the set and actual swash plate angles.

The electronic control system was based on the CANbus. It comprised a MC088 [11] PLC controller, with a specifically prepared and configured software. The measuring system was based on CANbus, which, combined with the PLC used, allowed for flexible configuration of the sensor variables and logging both analogue and digital signals.

The laboratory test stand was equipped with 6 pressure transducers (two of which were positionable with quick-connect couplings) and 2 flowmeters (turbine-type one fixed at the pump output and positionable gear-type one with quick-connect couplings), which additionally increased the configurability of the stand and allowed for variety of tests to be carried out. The hydraulic cylinder rod was attached to a position and velocity transducer, which combined with known diameters of the piston and rod, provided additional source of information regarding DCV output flowrates. A PC computer equipped with a CANbus interface worked as the data acquisition system.

3. Results

Fundamentally, the DCV behaviour is described by its oil flow characteristics. It is defined as the relation between the control signal and the output flow rate through the DCV. The characteristics shown in Fig. 2 was recorded when the DCV was supplying an unloaded hydraulic cylinder. The control signal (measured in relation to the neutral value of 50% of the PVES supply voltage) is shown on the horizontal axis. Vertical axes show flowrate, pressure and spool position as labelled. The pump flowrate was being kept constant and the control signal was being changed in 0.5 V increments.

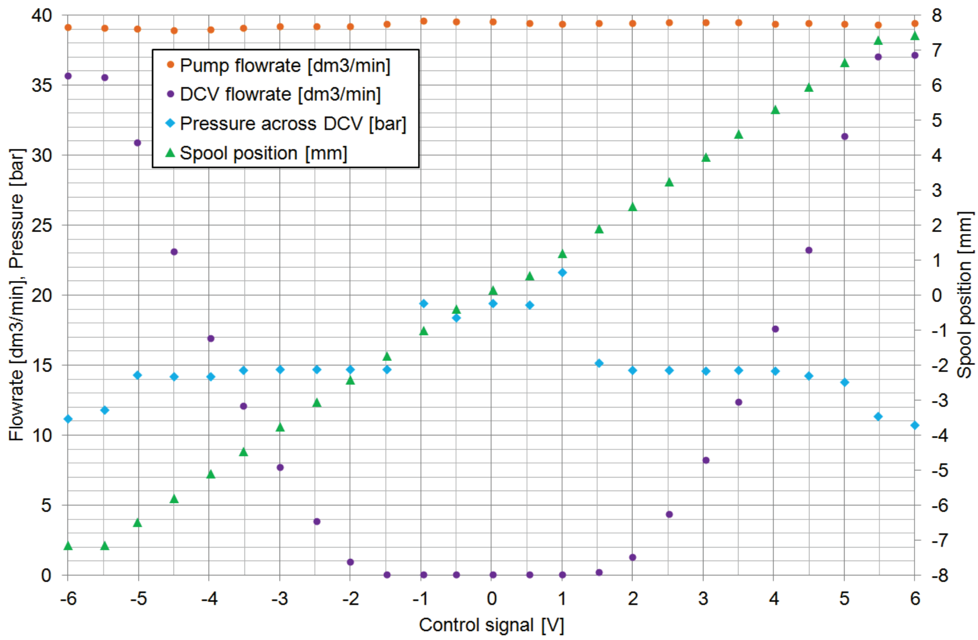

Fig. 2. DCV oil flow characteristics including spool position and pressure difference across DCV

The spool position adjusts to the control signal linearly except for the two extreme points where no further spool movement is observed. The control signal effective range is therefore limited to ±5.5 V. There is no DCV output flowrate for signals within -1.5 to 1.5 V, as this is the dead zone range for the spool type fitted to the DCV and the valve remains closed in this range. The flowrate characteristics outside the dead zone are highly non-linear. This also depends on the spool type fitted to the DCV (Note that, though linear spools are available, they are not recommended for human-controlled systems. The reason it that they cause the system response to joystick movement to be perceived by operator as sudden and erratic. An automated excavator needs to maintain the option of being controlled by human, therefore the use of a linear spool is not the preferred solution).

Figure 2 shows the pressure difference across the DCV obtained for each of the spool position and flowrate values. The values of the pressure across DCV for spool positions within the dead zone are not applicable as there is no flowrate through the valve in such cases.

The DCV oil flow characteristics need to be included in any model of the control system utilizing the DCV to accurately represent its performance. As the hydraulic cylinder was not loaded and the remaining sections of the DCV were also not loaded, it can be assumed that the pressure compensator within the DCV is fully open and the recorded characteristics are not affected by its operation.

Generally, the DCV characteristics can be represented in a model using the oil flow through orifice equation [12]:

$$Q(t) = C_d w x_v(t) \sqrt{\frac{2\Delta p}{\rho}}, \tag{1}$$

where:
Q – oil flowrate [m^3/s],
C_d – orifice discharge coefficient [–],
w – valve area gradient [m],
$x_v(t)$ – spool position [m],
t – time [s],
Δp – pressure difference across the valve [Pa],
ρ – oil density [kg/m^3].

Note that in accordance with the equation (1), the oil flow at a given Δp depends linearly from the spool position, which is not the case as per the results in Fig. 2 (see especially flowrate and pressure values for control signals between 2V and 4V). The non-linear characteristics can be represented by replacing the constant C_d and w parameters with a function of the spool position:

$$Q(t) = A(x_v) x_v(t) \sqrt{\frac{2\Delta p}{\rho}}, \tag{2}$$

where:
$A(x_v)$ – effective valve area gradient [m].

The $A(x_v)$ function can be established experimentally, by substituting the flowrate, pressure and spool position values from the results presented in Fig. 2. The $A(x_v)x_v(t)$ product represents the valve opening area combined with the discharge coefficient. The $A(x_v)$ function can be implemented into the model of the system to represent the characteristics of a given DCV. Similar approach can be used to assess the Q and Δp dependence for the return side of the valve. Note that, as stated above, the operation of the pressure compensator is not included in this relation, as only a single valve section was analysed and the effect of unevenly loaded DCV outputs was not relevant.

As the valve spool does not displace from one position to another instantaneously, therefore, to be able to represent the DCV performance in a dynamic model, one needs to know the time dependences of spool movement in relation to changing control signal. This can be assessed based on the step input response or sine wave input response of the spool position.

Figure 3 shows the response to the response of the spool position to the control signal at 3 different values. After a short period of less than 40 ms, the spool starts to displace at approximately the same velocity in all three cases. The response to step input can therefore be modelled by using two parameters: one describing the delay and the other describing the spool velocity. Alternatively, for simplicity of the model, this can also be modelled by a single parameters describing the averaged velocity (therefore ignoring the delay of the spool movement start). One can also represent the dynamics of the spool position response to control signal with a transfer function parameters of which can be assessed based on the plots shown in Fig. 3.

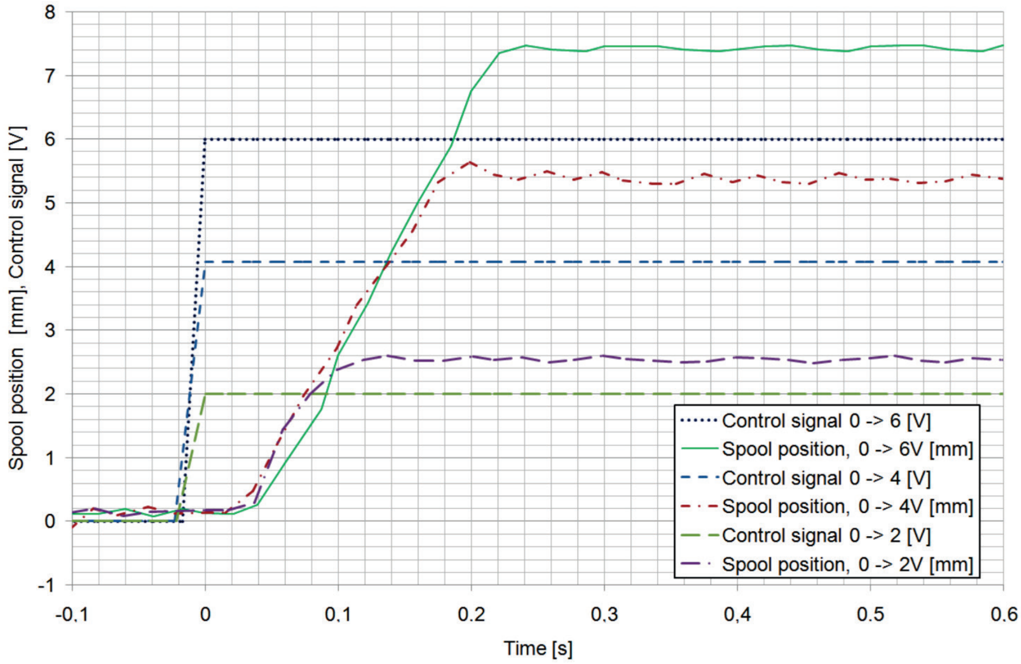

Fig. 3. Response of the spool position to step input of the control signal at from 0 to 2V, 4V and 6V

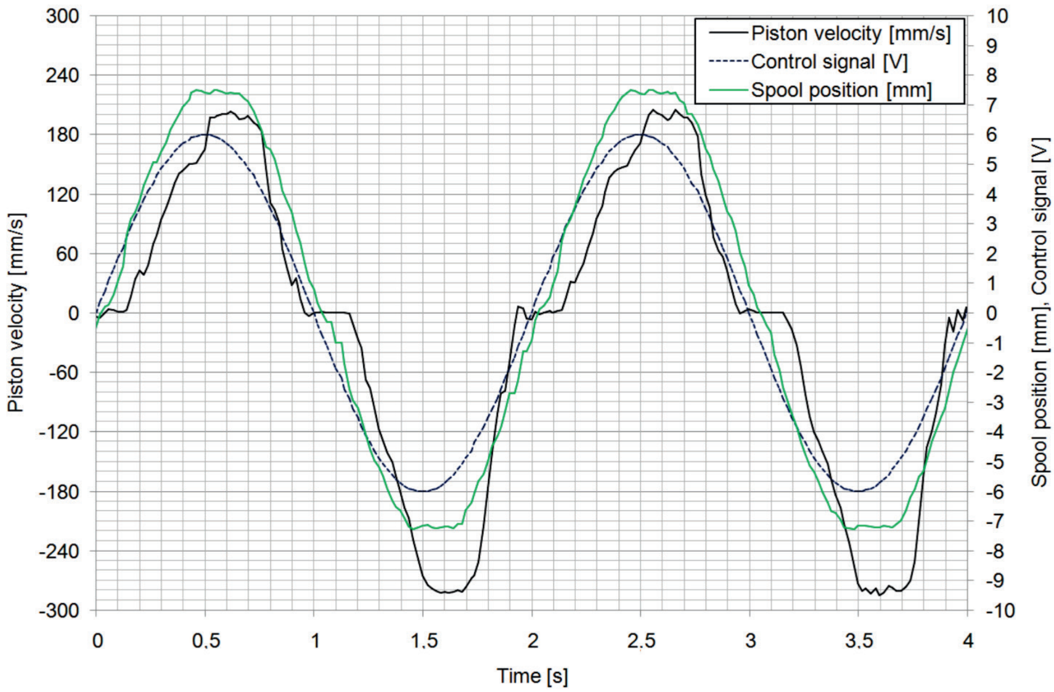

Fig. 4. Sine wave response of the spool position and piston velocity at 0.5 Hz control signal

Figure 4 shows the response of the spool position and piston velocity to a sine wave input at control signal frequency of 0.5 Hz. and piston velocity. The spool position plot is delayed in relation to the input signal. At this frequency, the spool is capable of reaching the full displacement. For higher frequencies, the amplitude decreases and the delay increases. Repeating this test for other frequencies can be used for plotting the frequency response characteristics of the DCV. These can then be used as data for working out what is the transfer function of the spool movement in response to control input and implemented into the model of the system. Fig. 5 shows the characteristics for spool position amplitude. The response is reduced by 3 dB at a frequency of

15.7 [rad/s] = 2.5 [Hz]. For comparison, the same plot shows the amplitude of piston velocity. It reduces more rapidly as the frequency increases than for the spool position. It is also different at positive and negative directions, which is caused by the asymmetry of the hydraulic cylinder.

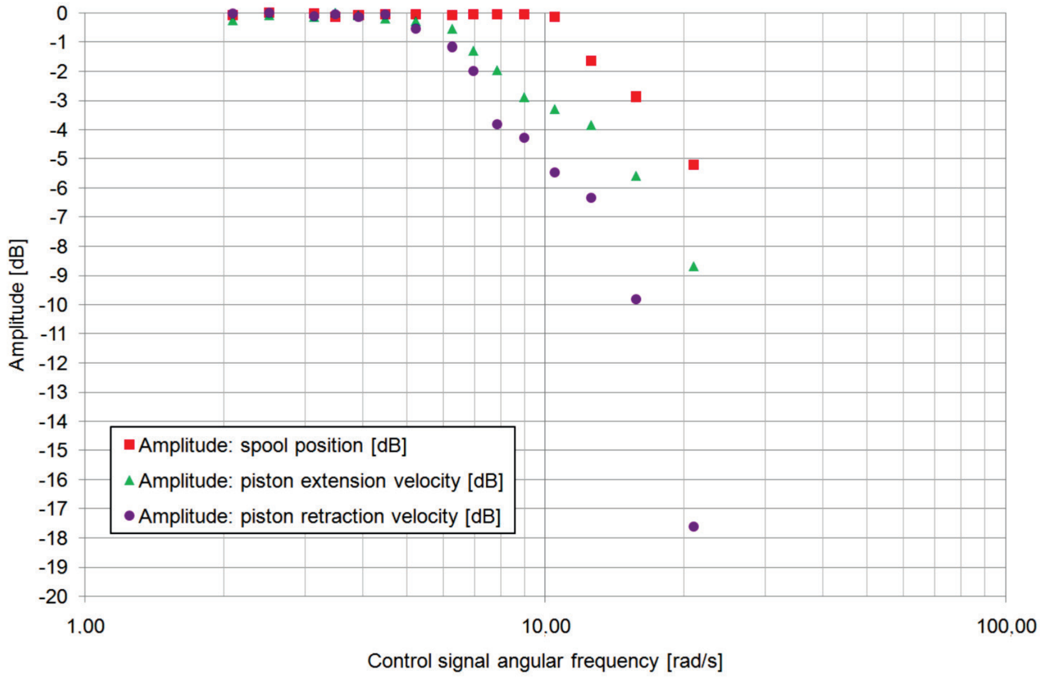

Fig. 5. Amplitude characteristics for spool position and piston velocity

The plot in Fig. 6 shows the hysteresis loop obtained for the tested DCV. The DCV outputs were looped together during this test. The control signal was being changed in cycles between the extreme values at frequency of 0.02 Hz. The obtained hysteresis loop width for the tested DCV is approximately 0.09 V (0.75% of the full control signal range). It is not a big number and in most cases, it could be neglected. However, if needed, it can easily be implemented into the model of the system as a single parameter. Note that in case of other DCVs the hysteresis effect may be more significant.

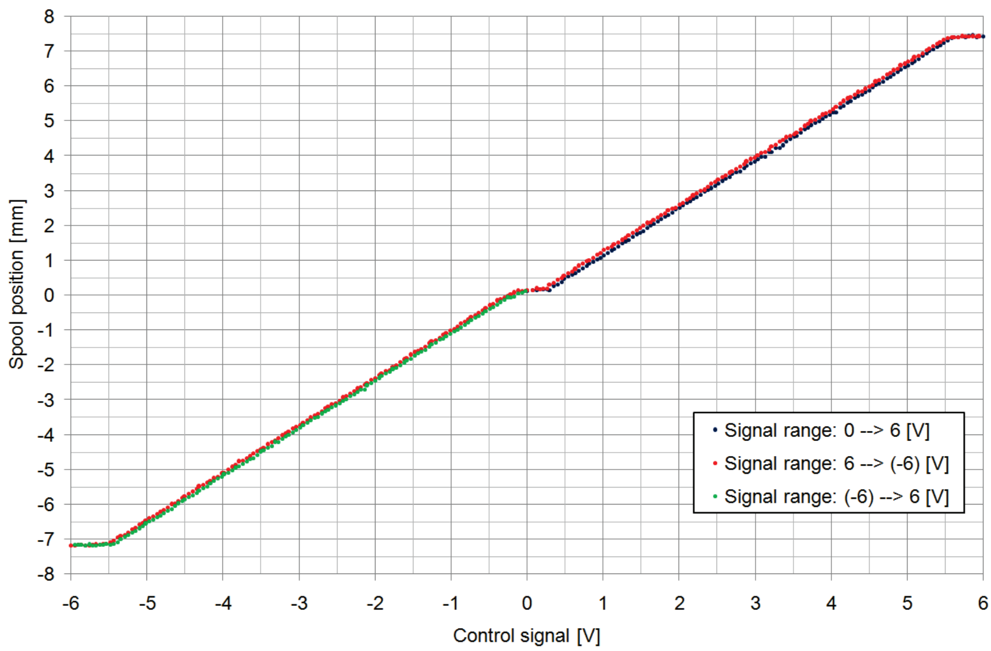

Fig. 6. Spool position hysteresis obtained for control input frequency of 0.02 Hz

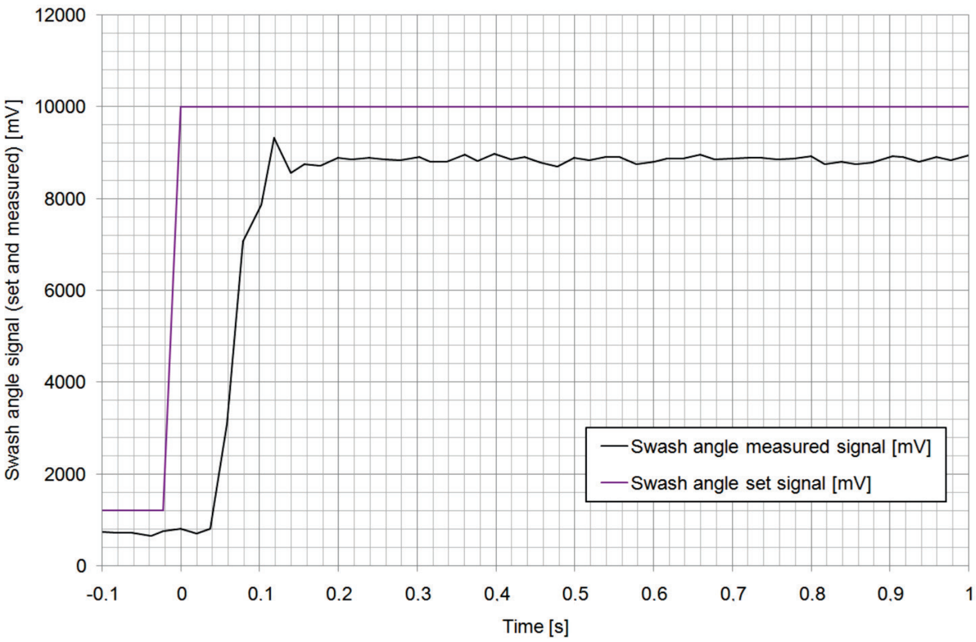

Fig. 7. Step input response of the variable displacement piston pump

Similarly as for the DCV spool position, the step input response was tested for the variable displacement pump. Fig. 7 shows the commanded and measured swash angle signal. In the analysed case, the pump response was quicker than it was for the DCV. Knowledge of the pump response is important in the modelling and analysis of control algorithms involving the variable displacement of the pump.

The results presented in Fig. 2 were obtained for an unloaded hydraulic cylinder. The test was repeated with external load applied to the rod of the hydraulic cylinder. The cylinder was extracting to lift the external load so the pressure caused by the pressure occurred at the piston side of the hydraulic cylinder. Fig. 8 shows the pressure values measured at the piston side with and without the external load. The difference between the two values indicates the value of the pressure caused by the load itself, i.e. the useable pressure. In this case, the useable pressure was approximately 30 bar, which for the piston diameter of 63 mm corresponds to 9352 N of external load at the rod. At the same time, the pump had to deliver oil flow at pressure in the range of 85 to 100 bars. The difference between the useable pressure and pump pressure, which is predominantly caused by friction losses within the pipes, hoses, fittings, etc. The disproportion between these values is significant, therefore it cannot be ignored in the model of the system.

The friction losses can be estimated analytically using methods available from fluid mechanics. However, doing so for an actual hydraulic circuit, it is a complex task because including in the estimates every single hydraulic fitting and change of pipes and hoses cross section takes a lot of effort. Additionally, it is difficult to accurately assume every data for the task, such as oil viscosity, pipe roughness, flow passage loss factors for fittings, the influence of temperature, etc. Therefore, provided that such opportunity exists, it is much easier to assess the losses experimentally.

The experimentally assessed dependence of pressure loss at the hose connecting the DCV output to piston side port of the hydraulic cylinder is shown in Fig. 9. Two pressure transducers were placed at each end of the hose. The oil flowrate was being increased in 10 increments from 0 to 38 dm^3/min. The values of measured pressure loss are placed along 2nd order curve. The equation representing this curve is shown on the plot. For simplicity of the model being created, this dependence can be approximated by a linear function (also shown on the plot) provided that the accuracy is deemed sufficient. The slope of this function can be implemented into the model as a parameter. The pressure loss for the given hose will therefore be:

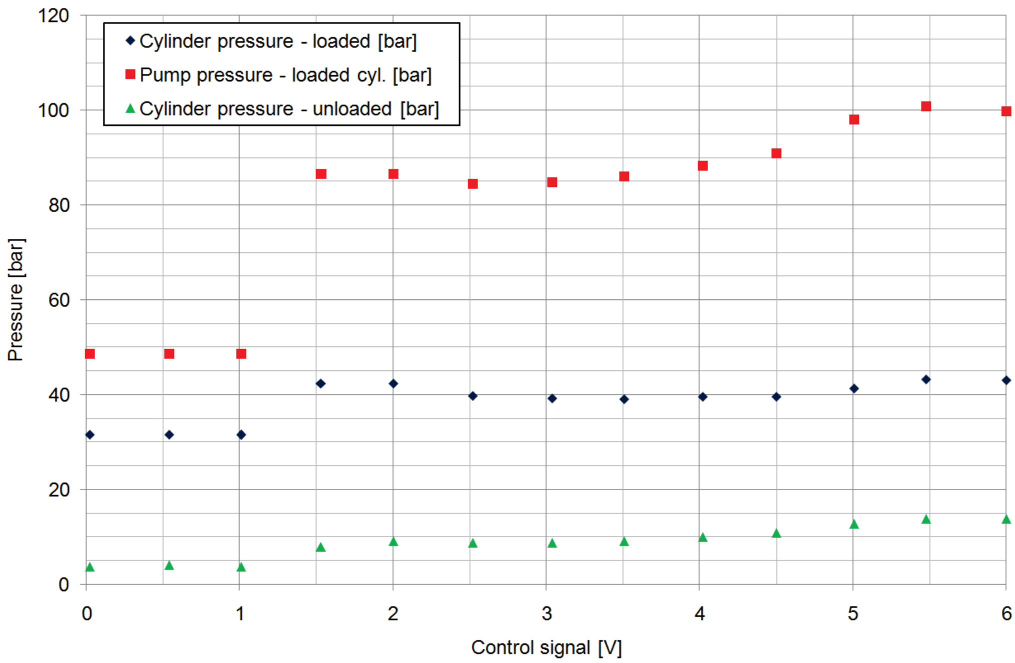

Fig. 8. Cylinder pressure for loaded and unloaded case and pump pressure for loaded case

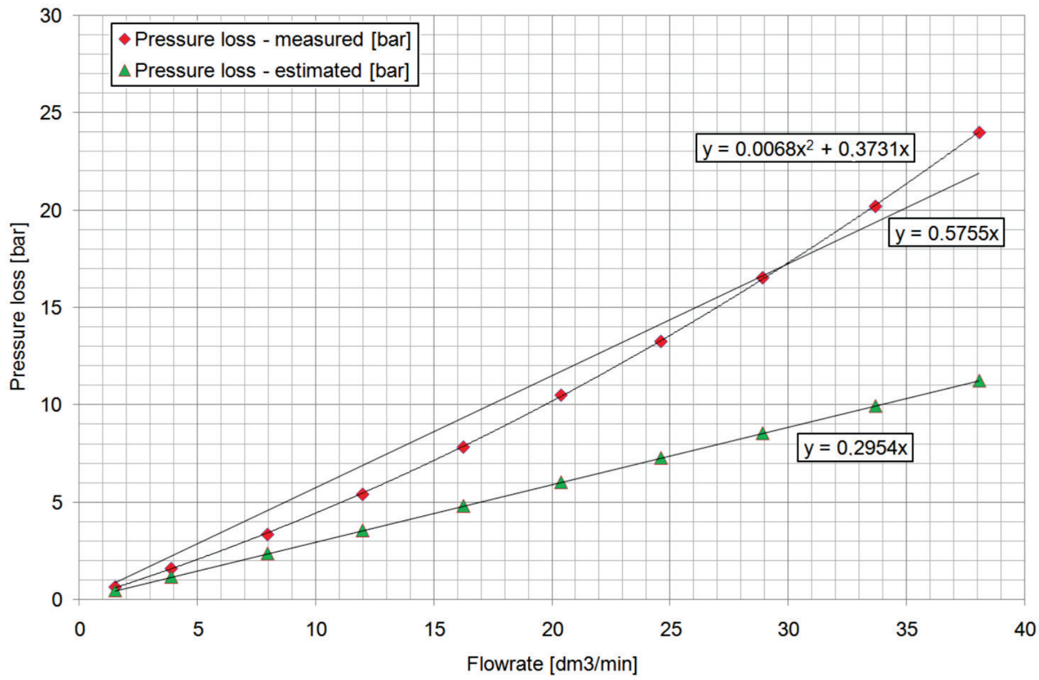

Fig. 9. Pressure losses in hose connecting DCV output to piston side of the hydraulic cylinder

$$p_{loss} = K_{loss} Q, \tag{3}$$

where:

p_{loss} – pressure loss [bar],
K_{loss} – total pressure loss factor for a hose, in this case equal to 0.5755 [bar/(dm^3/min)],
Q – oil flowrate [dm^3/min].

For comparison, Fig. 9 includes the pressure loss estimated analytically for input data being the same as during the actual measurement. The hose length was 4.1 m. The oil kinematic viscosity at test temperature of 28ºC was assumed as 100 mm^2/s and oil density as 875 kg/m^3. The internal diameter of the pipes and hoses on the test stand was 9.53 mm, which for L-HL46 hydraulic oil at test temperature of 28ºC and flowrates up to 38 dm^3/min results in Reynold's number well below

2300, therefore the flow was laminar. In accordance with the theory for laminar flow, the pressure drop in this case can be calculated as follows:

$$p_{loss} = \frac{fl\rho v^2}{2d} = \frac{128\mu lQ}{\pi d^4}, \tag{4}$$

where:
p_{loss} – pressure loss [Pa],
f – fluid friction factor (for laminar flow $f = 64/\mathrm{Re}$) [–],
l – hose length [m],
ρ – oil density [kg/m^3],
v – oil flow velocity [m/s],
d – hose internal diameter [m],
μ – absolute viscosity [Pa·s],
Q – oil flowrate [m^3/s].

Note that the estimated pressure loss slope is nearly 2 times lower than in case of the measure one. Pressure loss estimated using this simple analytical method is underestimated. Calculations that are more sophisticated would be required to obtain accurate relation, which proves the experimental method as useful.

4. Conclusion

The laboratory test stand used for carrying out the described tests allowed for gathering the data required for modelling of the system. The dynamic parameters of the selected components in the analysed hydraulic system were measured and their importance for the model of the system was assessed. The following properties were analysed:
– DCV oil flow characteristics,
– DCV spool position response to step input,
– DCV spool position response to sine wave input,
– variable displacement piston pump response to step input,
– DCV hysteresis
– pressure losses on hydraulic pipes and hoses.

Furthermore, discussion of how these parameters can be implemented into the model of the system was carried out.

Performing this type of tests and knowledge of the hydraulic components parameters such as those listed above is essential for the proper modelling of the hydraulic system and further analysis of the control system performance and its algorithms.

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