EFFECTS OF ADJUSTABLE AND CONTROL PARAMETERS ON PERFORMANCE CHARACTERISTICS OF MULTI-CONTROLLED VARIABLE DISPLACEMENT PUMP

UDC:62-82
Original scientific paper https://doi.org/10.46793/aeletters.2025.10.1.2

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Abstract:
Energy efficiency and improving the performance of high-power hydraulic systems in machinery are essential trends. Various methods can be employed, and the widespread use of multi-controlled hydraulic pumps is one of the most popular approaches. In this work, the effects of adjustable and control parameters on the pump's performance characteristics are investigated. Regarding this, based on the operation principle of hydraulic pumps with a multi-function controller, a dynamic model within the mathematical equations and the corresponding simulation model are developed. An experiment study is also conducted to confirm the accuracy of the theoretical model. From the theoretical and experimental results with a given pump configuration, it is seen that the pump flow is regulated according to the changing the control signal within a range of 0 to 500 mA and varying load between 40 and 350 bar, and the pump power is adjusted according to the initial setting force of the pressure compensator from 50 to 150 bar which is guaranteed to correspond to the working load. The initial setting force adjustment of the load sensing valve should not exceed 400 N due to flow fluctuations and pump overload. The research results provide a foundation for properly installing and adjusting the pump to enhance its operational efficiency in hydraulic systems.

ARTICLE HISTORY
Received: 3 June 2024
Revised: 4 November 2024
Accepted: 15 November 2024
Published: 31 March 2025

KEYWORDS
Hydraulic pump,
Multi-function controller,
Control parameters,
Pump characteristics,
Multi-function test bench

1. INTRODUCTION

Recently, energy and fuel savings have become increasingly important. This not only has significant economic implications but also holds great significance for environmental protection. To achieve these goals, besides improving hydraulic components, optimizing system diagrams, and utilizing high-quality hydraulic oils, hydraulic systems with various control types are employed to drive working equipment on medium and large machines. Among these, the load-sensing (LS) hydraulic pump with adjustable control is widely used in many different versions [1,2]. Consequently, different research directions on pumps and pump types have been developed.

The group of authors from Komatsu by Fukushima et al. (2009) [3] introduced the schematic diagram of CLSS (Closed-Center Load Sensing System) hydraulic systems for forklifts to enhance energy efficiency and meet safety standards. Similarly, Bondar's research (2003) [4] analyzed the operating principles, advantages, and disadvantages of the LS and FS (Flow Sharing) hydraulic systems, as well as the improved LS, for a detailed review can be seen in Li et al. (2022) [5]. Galukhin (2015) [6] identified the drawbacks of hydraulic systems using pressure-compensated pumps with spring-loaded compensator valves and demonstrated the necessity of replacing them with systems featuring pressure-compensated valves without springs to enhance system efficiency. Furthermore, Galukhin proposed and analyzed the

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theoretical aspects of a new hydraulic system employing electronic control and springless pressure compensator valves. However, these studies only focused on analyzing the working principles of system diagrams without the dynamic characteristics of pumps and systems.

Lettini et al. (2010) [7] presented an electrically controlled LS hydraulic system, which employed a Casappa MVP pump with electronically controlled pressure and an integrated tilt-disc angle sensor. Tests and evaluations of the pump's operation have been conducted on the small excavator Kubota KX 161-3. However, a comprehensive dynamic analysis of the pump and hydraulic system has not been performed to evaluate their functional behaviors under operational conditions. Nevertheless, no research on the hydraulic pump and system dynamics was conducted to assess their working characteristics. Currently, numerous studies are focusing on LS-PC (Load Sensing - Pressure Compensator) pumps. Casoli et al. [8-11] conducted several pieces of research on the multibody mathematical model of hydraulic excavators, including the basic LS-PC control pump. They investigated the variations in the system's dynamic parameters during operation. Garciano et al. (2014) [12] studied the high-frequency oscillations of LS-PC hydraulic pumps and proposed solutions to reduce these oscillations. Using the Bond graph method [13] or power linkage graph method [14], researchers established comprehensive models of LS-PC controlled pumps, exploring the theoretical and experimental aspects of dynamic parameters such as pressure and flow rate over time. Borovin et al. (2009) [15] investigated the LS-PC-controlled hydraulic system on a six-legged walking machine considering the variations in cylinder displacement speed, system efficiency, and pump power changes. They evaluated the effectiveness of using the basic LS-PC pump to improve machine performance.

Research on variations of LS-PC pumps, conducted by Nguyen et al. (2021) [16], focused on the hydraulic dynamics of LS-PC controlled pumps with mechanical reverse linkage and CLSS hydraulic systems in the process of remote-controlled bomb handling on an explosive ordnance disposal machine. The study analyzed the changes in dynamic parameters throughout the entire bomb deployment process. Andersson (2009) [17] evaluated hydraulic systems with common pre-compensators and pre-compensators with anti-saturation, using electronically controlled pumps. The study compared the energy efficiency of this system with that of a conventional LS system. Banaszek and Petrovic (2019) [18] studied the LS systems on board ships and offshore to reduce volumetric-type energy losses. Wagner (2014) [19] investigated the hydraulic dynamics of LS-controlled pumps by constructing mathematical models and performing simulation and experimental calculations to evaluate the operational stability of the pump.

This paper focuses on the hydraulic dynamics of a multi-controlled pump. Based on building mathematical models and simulations in the AMESim environment, the dynamic characteristics of the pump will be analyzed and evaluated. Additionally, the influence of pump adjustment parameters on its working characteristics will also be studied. The experiment working on the developed testing bench verifies the present theoretical model.

2. MATHEMATICAL MODEL OF MULTI-CONTROLLED HYDRAULIC PUMP

The structure principle of the multi-controlled variable displacement pump is shown in Fig. 1.

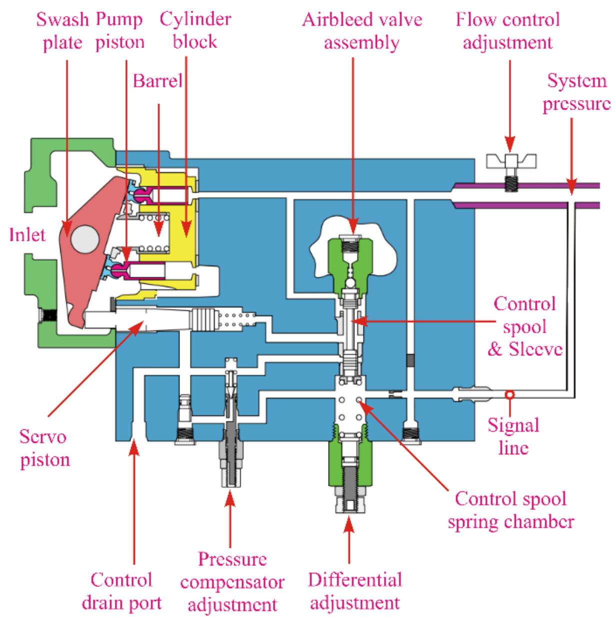

Fig. 1. Structure principle of a multi-controlled variable displacement pump

The dynamic hydraulic model of the pump (Fig. 2) is developed based on the following assumptions: The return line pressure is constant and can be considered negligible. The viscosity and elastic modulus values do not change during operation, and the influence of temperature is not taken into account. Leakage of the working fluid through the radial clearances of the distributor slide valve, actuating cylinder, and pump is ignored. The compressibility of the oil in the control chambers of the main distributor valve can be disregarded due to the small volume of the end face chamber. The hydrodynamic forces acting on the control slides are considered negligible and can be neglected. The load torque on the pump's inclined plate is generated by the mass of the moving parts, leading to servo piston deflection.

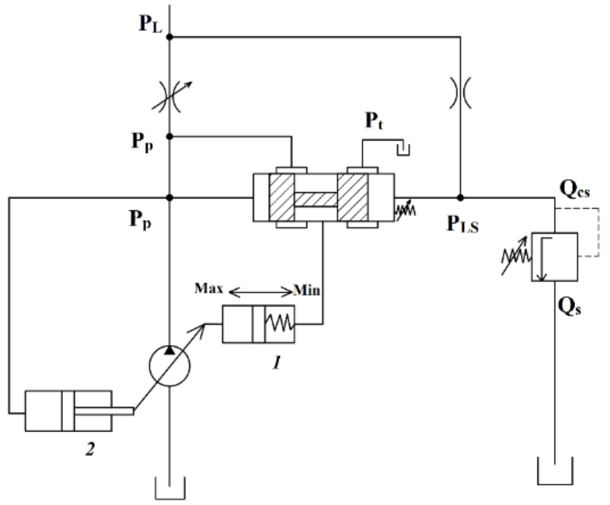

Fig. 2. Mathematical model of the multi-controlled variable displacement pump

The dynamics of the hydraulic system are built based on equations for flow rate, pressure, and system interconnections using the Newton-Euler equations, the law of conservation of flow, Pascal's law, and the compressibility of the working fluid [16,17]. From the continuity condition of the fluid flow, the flow equation in the high-pressure pipeline of the hydraulic system is expressed as follows:

$$k_n \frac{dp_p}{dt} = Q_p - Q_{sy} - Q_c, \tag{1}$$

where are: $k_n = V_{ht}/B_{com}$ - is the compression coefficient of the working fluid; B_{com} - is the elastic modulus of the working fluid; p_p - is the output pressure of the pump; Q_p - is the pump output flow rate; Q_{sy} - is the required system flow rate; Q_c - is the flow rate entering the control line.

$$Q_{sy} = \sum_{i=1}^{n} Q_i, \tag{2}$$

where are: Q_i - is the required flow rate for each hydraulic subsystem; *n* - is the number of hydraulic subsystems connected to the pump (in this case, *n* = 1).

The flow rate entering the control line is equal to the flow rate reaching the control valve LS of the pump:

$$Q_c = Q_{ls}. \tag{3}$$

The mathematical model of the multi-controlled pump is constructed based on the structural characteristics and operation of each pump component. The flow rate of the pump is calculated according to the following formula:

$$Q_p = \frac{q_p}{\gamma_{max}}, \tag{4}$$

where are: q_p - is the specific displacement volume of the pump; γ_{max} - is the maximum tilt angle of the inclined plate; ω - is the angular velocity of the pump shaft; γ - is the instantaneous tilt angle of the inclined plate.

The relationship between the displacement of the servo piston and the tilt angle of the inclined plate is as follows:

$$\gamma = \arctan \frac{y_{max}}{R}, \tag{5}$$

where are: *y* - is the coordinate determining the position of the servo piston; y_{max} - is the maximum value of *y*; *R* - is the radius of the inclined plate of the pump.

The equation of motion for the LS spool valve:

$$m_{sp-ls} \frac{d^2 x_{ls}}{dt^2} + k_{fr-ls} \frac{dx_{ls}}{dt} + c_{sp-ls} x_{ls} + P_{0sp-ls} = (p_p - p_{ls}) F_{ls}, \tag{6}$$

where are: m_{sp-ls} - is the mass of the LS spool valve; k_{fr-ls} - is the coefficient of wet friction in the radial clearance of the LS spool valve; c_{sp-ls} - is the stiffness of the LS valve spring; P_{0sp-ls} - is the initial set force of the LS valve spring; p_{ls} - is the control pressure of the LS valve; F_{ls} - is the cross-sectional area of the LS valve head; x_{ls} - is the displacement of the LS spool valve.

Suppose the x_{cl} initial clearance value of the LS valves. The flow rate equations through the LS valves in different operating conditions are as follows.

If $x_{ls} < x_{cl}$ (x_{ls} - is very small, which means the pump pressure P_p is very low and the pump flow rate is at its maximum Q_{pmax}), then the flow rate through the LS valve is given by:

$$Q_{ls} = \mu_{ls} \pi d_{ls} |x_{cl} - x_{ls}| \sqrt{\frac{2}{\rho} |p_p - p_c|} \operatorname{sign}(p_p - p_c), \tag{1}$$

where are: Q_{ls} - is the flow rate through the LS valve; μ_{ls} - is the flow coefficient of the LS valve; d_{ls} - is the diameter of the LS valve spool; p_c, p_p are the control pressure and the pump pressure, respectively.

If $x_{ls} > x_{cl}$ (x_{ls} - is very large), then the control chamber is connected to the return line (corresponding to Q_{pmin}).

$$Q_{ls} = \mu_{ls}\pi d_{ls}|x_{ls} - x_{cl}|\sqrt{\frac{2}{\rho}|p_c - p_t|}\mathrm{sign}(p_c - p_t). \quad (2)$$

If $x_{ls} < x_{cl}$ and the pump pressure P_p is at a fixed value (the pump operates in normal mode), then:

$$Q_{in-ls} = \mu_{ls}\pi d_{ls}|x_{cl} - x_{ls}|\sqrt{\frac{2}{\rho}|p_p - p_c|}\mathrm{sign}(p_p - p_c) \quad (3)$$

$$Q_{out-ls} = \mu_{ls}\pi d_{ls}|x_{cl} - x_{ls}|\sqrt{\frac{2}{\rho}|p_c - p_t|}\mathrm{sign}(p_c - p_t). \quad (4)$$

In the normal operating mode, due to the structural characteristics and working principles of the LS valve, the pressure value p_c can take two values (p_p or p_t), depending on the displacement value x_{ls}. Therefore, the value of Q_{c1} - is determined as follows:

$$\begin{cases} Q_{c1} = Q_{in-ls} + Q_{out-ls} \text{ when } p_c = p_p \\ Q_{c1} = Q_{in-ls} - Q_{out-ls} \text{ when } p_c = p_t. \end{cases} \quad (5)$$

The equation for the flow rate into the servo pistons:

$$\left.\begin{aligned} Q_{c1} = F_{c1}\frac{dy}{dt} + \frac{V_{c1}}{B_{com}}\frac{dp_c}{dt} \\ Q_{c2} = F_{c2}\frac{dy}{dt} + \frac{V_{c2}}{B_{com}}\frac{dp_p}{dt} \end{aligned}\right\}, \quad (6)$$

where are: $F_{c1} = \pi d_1^2/4$, F_{c2} are the cross-sectional area of the servo piston faces; V_{c1}, V_{c2} are the volume of the servo cylinders; B_{com} - is the hydraulic oil bulk modulus.

The equation of motion for the inclined disc pump and servo piston.

$$m_c\frac{d^2y}{dt^2} + k_{fr-c}\frac{dy}{dt} + P_{0sp-pc} = p_cF_{c1} + c_{sp}y - p_pF_{c2}, \quad (7)$$

where are: m_c - is the mass of the control pistons; k_{fr-c} - is the coefficient of wet friction between the piston and the servo cylinder; c_{sp} - is the stiffness of the spring inside the control cylinder; P_{0sp-c} - is the initial set force of the spring inside the control cylinder 1.

The common mathematical description is usually based on Bernoulli's equation and leads to the form:

$$Q_{th} = C_qA\sqrt{\frac{2}{\rho}|\Delta p|}, \text{ with } \Delta p = p_p - p_L, \quad (8)$$

where are: Q - is the flow rate; C_q - is the flow coefficient; A - is the cross-section of the orifice; Δp - is the differential pressure; ρ - is the density at mean pressure. The flow rate through the flow control valve is determined as follows:

$$Q_{th} = Q_{sy} = Q_p - Q_{c1} - Q_{c2}. \quad (9)$$

Assuming that the opening and closing process of the flow control valve (or distribution valve) is instantaneous, the opening degree of the valve can be considered linear dependent on the control current flow.

$$A = K_{tl}i, \quad (10)$$

in which i represents the control current flow, and K_{tl} denotes the proportionality constant.

The evolution of the pressure from the control chamber of the valve poppet, p_{cs}, results from the continuity equation attached to this volume:

$$Q_{cs} - A_{cs}\dot{x}_s = \frac{V_c}{\varepsilon_e}\dot{p}_{cs}, \quad (11)$$

where are: A_{cs} - is the area of the poppet control piston surface; $\dot{x}_{cs}$ - is the poppet speed; V_c - is the average liquid volume from the valve control chamber; ε_e - is the equivalent bulk modulus of the liquid included in volume V_c.

The distance between the conical poppet and the seat x_s (valve opening) results from the poppet motion equation:

$$m_s\ddot{x}_s = F_{cs} + F_{hs} + F_{ht} + F_e, \quad (12)$$

where are: m_s - is the equivalent mass of the spring and the poppet; F_{cs} - is the control pressure force on the damper piston attached to the poppet; F_{hs} - is the hydrodynamic force on the poppet generated by the fluid-speed variation inside the valve (steady-state component); F_{ht} - is the pressure force on the poppet generated by the fluid-speed variation inside the valve (transient component); F_e - is the elastic force generated by the helical spring. The flows exiting from the valve can be computed by Bernoulli's equation and the law of conservation of mass:

$$Q_s = K_sx_s\sqrt{p_{cs}}, \quad (13)$$

$$Q_{cs} = K_{cs}(p_{cs} - p_s). \quad (14)$$

3. INVESTIGATION OF PUMP CHARACTERISTICS

In this section, the dynamic model, proposed in Section 2, is studied for the PAVC65 pump [20] in the LMS AMESim simulation environment [21], see Fig. 3. The initial setting parameters for simulation are list in Table 1.

Table 1. Structure and operating parameters of the pump

Factors	Symbol	Units	Value
Structure parameters			
Diameter of servo piston	d_1	mm	40
Mass of servo piston	m_c	kg	0.5
Diameter of LS valve	d_{ls}	mm	10
Mass of LS valve spool	$m_{sp\text{-}ls}$	kg	0.1
Stiffness of LS valve spring	$c_{sp\text{-}ls}$	N/m	2000
Initial setting force of LS valve spring	$P_{0sp\text{-}ls}$	N/m	0
Initial gap of LS valve	x_{cl}	mm	1
Stiffness of spring in servo piston	c_{sp}	N/mm	10
Operating parameters			
Pump displacement	q_p	cc/rev	65
Pump shaft speed	ω	rpm	1000
Setting pressure	p_{cs}	bar	200
Specific gravity of hydraulic oil	ρ	kg/m^3	850

A study on the operational properties of the pump was conducted under two conditions: Firstly, by varying the intensity of the current flowing through the pressure relief valve (or main distribution valve); secondly, by changing the pressure of the load within the hydraulic system (using cylinders to generate the load). Following this, an examination was made into the effects of adjusting the load sensing valve and pressure compensator on the pump's operational characteristics while maintaining the opening of the flow control valve and the load pressure constant.

In the first scenario, the value of the flow control valve opening current intensity is used to evaluate the pump control parameters and the system flow capacity. In this case, the load pressure is kept constant at p_L = 120 bar, and the electrical control signal applied to the proportional flow control valve ranges from 0 to 500 mA in a stepwise manner.

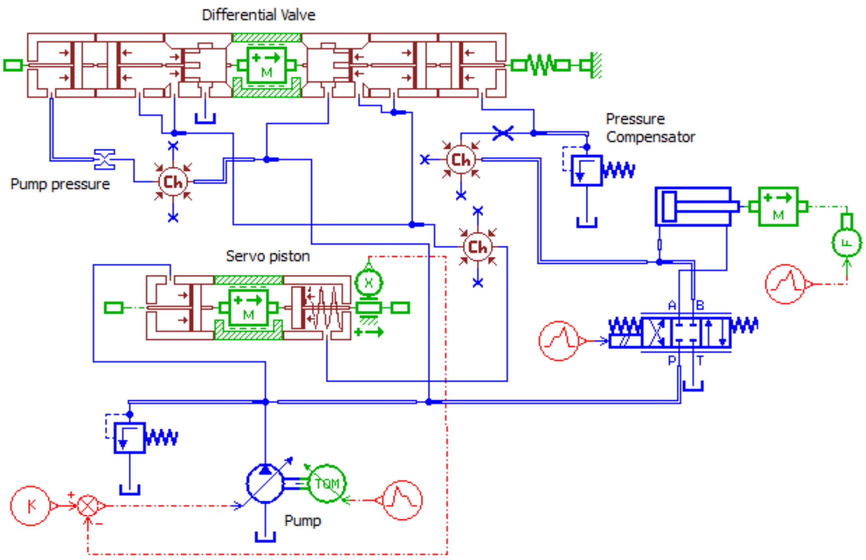

Fig. 3. Simulation model in the AMESim environment

Based on the research results in Figs. 4, 5, 6, and 7, it is observed that as the system control signal (controlling the opening degree of the distribution valve) increases from 0, 100, 200, to 500 mA under constant load conditions, the pump flow rate gradually increases from below 10 L/min to 36 L/min, corresponding to control parameters increasing from 0.2 to 0.56. Additionally, the pressure at the pump inlet decreases from 350 bar to 140 bar. These changes are entirely consistent with the operational requirements of the hydraulic pump, meaning that as the system's flow demand increases, the hydraulic pump will provide a larger flow rate.

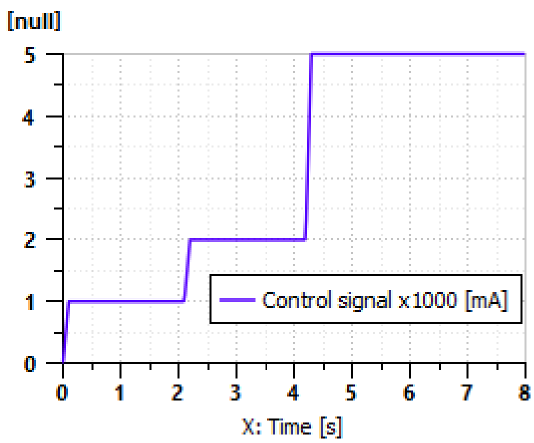

Fig. 4. Electrical signal controlling the flow control valve

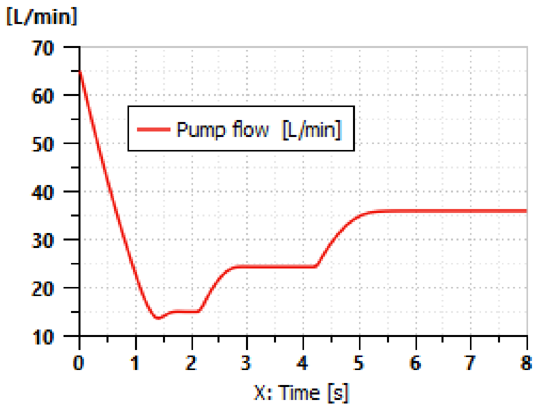

Fig. 5. Flow rate at the pump outlet

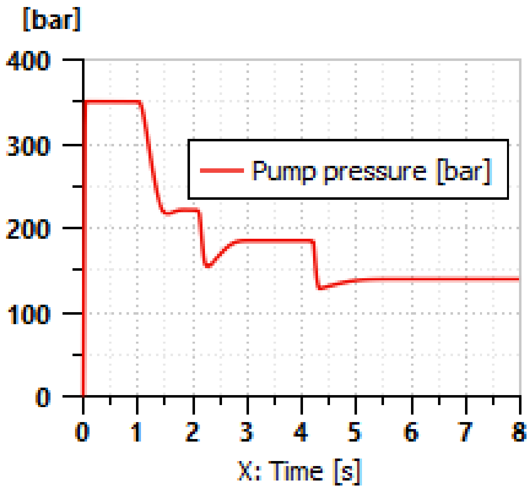

Fig. 6. Pressure at the pump outlet

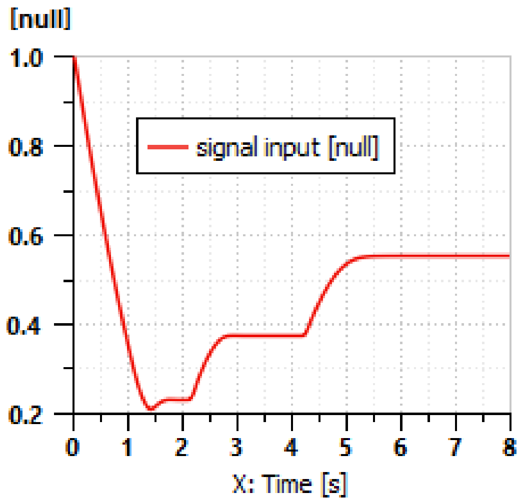

Fig. 7. Pump control parameters

In the second scenario, the electrical control signal for the flow control valve is maintained at 200 mA, while the load pressure varies from 300 bar to 0 bar in a stepwise manner. This setup is employed to assess the pump control parameters and the corresponding changes in pump flow rate in response to the load.

Based on the research results in Figs. 8, 9, 10, and 11, it is observed that with a constant system control signal (controlling the opening degree of the distribution valve), when the load pressure decreases from 350 bar to 40 bar, the pressure at the pump inlet also decreases from 350 bar to 140 bar, while the pump flow rate increases from 0 to 35 L/min. This corresponds to the control signal increasing from 0 to 0.52. These changes are entirely consistent with the operational requirements of the hydraulic pump, meaning that when the load pressure increases, the flow rate of the hydraulic pump must decrease accordingly to ensure constant pump power and improve hydraulic system efficiency.

In the third scenario, the study investigates the impact of adjusting the LS valve on the pump's performance characteristics. Under constant load pressure of p_L = 140 bar and a constant opening degree of the flow control valve, the control current of the flow control valve is maintained at i = 200 mA. Specifically, the initial setting force of the LS valve spring is adjusted within the range from $P_{0sp\text{-}lsN}$ = 100 to 1500 N. In this case, the flow rate and pressure characteristics of the pump will change correspondingly, as depicted in the subsequent figures.

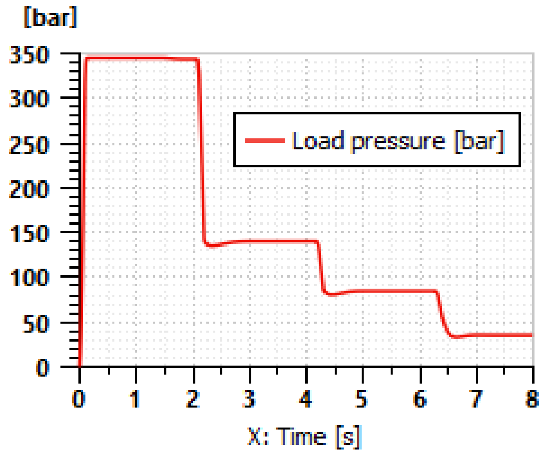

Fig. 8. Load pressure

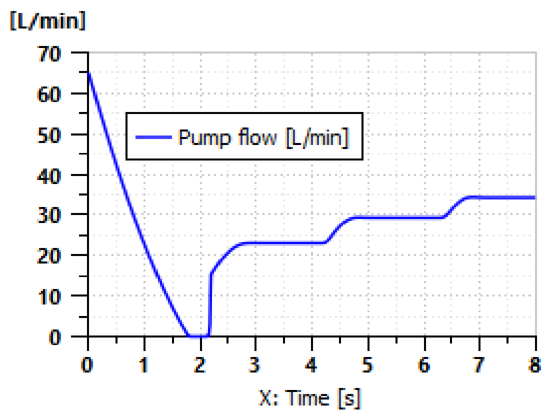

Fig. 9. Flow rate at the pump outlet

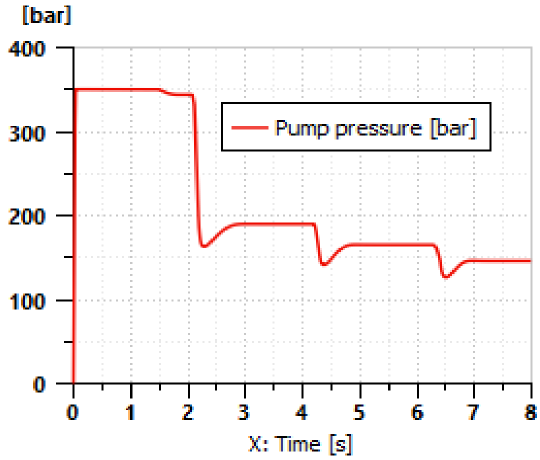

Fig. 10. Pressure at the pump outlet

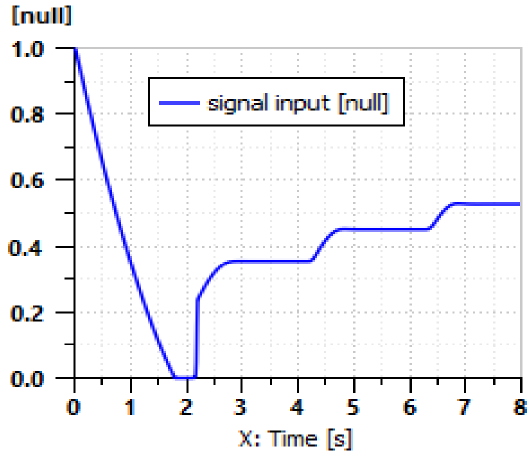

Fig. 11. Pump control parameters

According to Figs. 12 and 13, it is observed that at the initial setting force values of the LS valve of 100 N and 300 N, the flow rate and pressure characteristics of the hydraulic pump completely coincide with each other. However, at an initial setting force value of the LS valve of 400 N, oscillations appear in the pressure and flow rate at the pump inlet, adversely affecting the pump's operation. At an initial setting force value of the LS valve of 1500 N, both the pressure and flow rate of the pump reach their maximum values, corresponding to 350 bar and 65 L/min, leading to overloading of the hydraulic pump's driving motor. Therefore, adjusting the initial setting force of the LS valve significantly affects the performance characteristics of the hydraulic pump. To ensure the reasonable operation of the hydraulic pump, this setting value should not exceed 300 N.

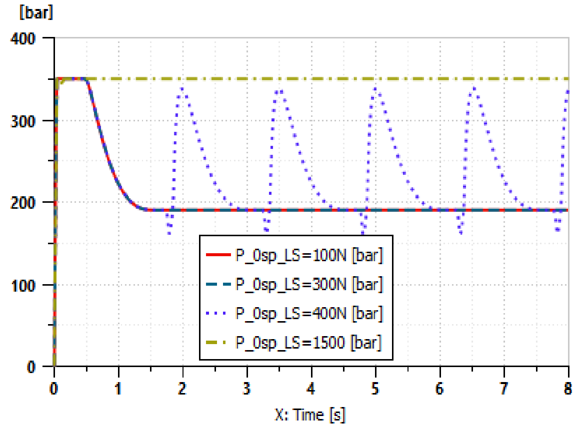

Fig. 12. Pressure at the pump outlet for different initial setting force values of the LS valve spring

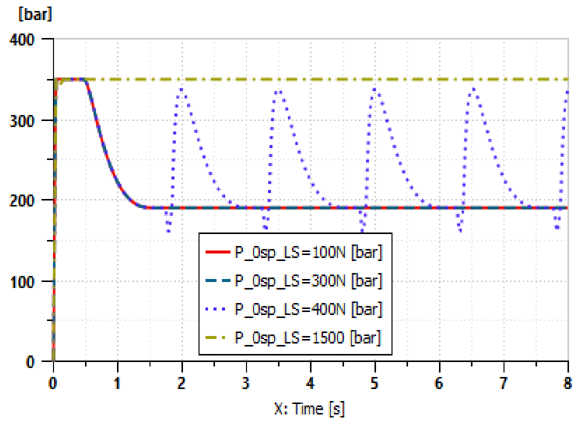

Fig. 13. Flow rate at the pump outlet for different initial setting force values of the LS valve spring

In the fourth scenario, the study examines the impact of adjusting the pressure compensator on the pump's performance characteristics. Under constant load pressure of 140 bar and a constant opening degree of the flow control valve, with a control current of the flow control valve set at i = 800 mA and an initial setting force of the LS valve at $P_{0sp\text{-}ls}$ = 250 N. Specifically, the pressure setting of the pressure compensator is varied within the range from 50 bar to 150 bar. In this case, the flow rate and pressure characteristics of the pump will change correspondingly, as depicted in the subsequent figures.

In this case, according to Figs. 14 and 15, it is observed that when the pressure setting of the pressure compensator decreases from 150 bar to 50 bar, the pressure at the pump inlet also decreases from 150 bar to 50 bar, while the pump flow rate remains unchanged and stays at 65 L/min. This indicates that the pump supply flow rate remains constant, but the power of the hydraulic pump has decreased correspondingly with the pressure setting of the pressure compensator.

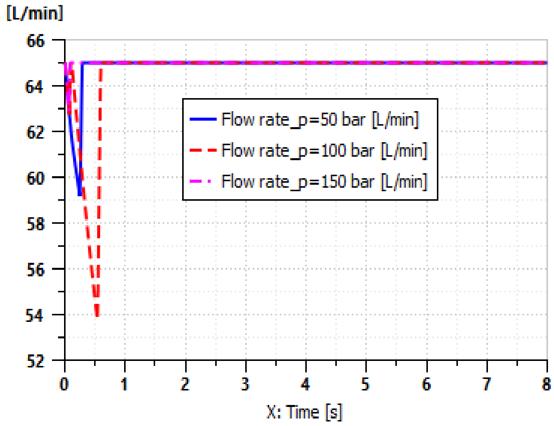

Fig. 14. Flow rate at the pump outlet for different initial setting force values of the spring

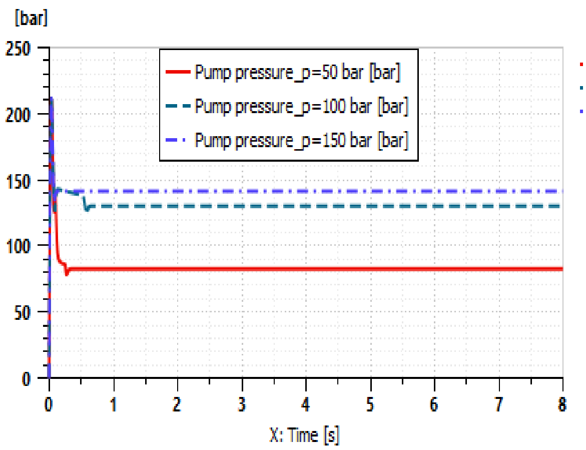

Fig. 15. Pressure at the pump outlet for different initial setting force values of the spring

Therefore, adjusting the pressure compensator is essentially adjusting the power of the hydraulic pump. The greater the power demand of the hydraulic system, the higher the pressure setting of the pressure compensator needs to be adjusted.

4. EXPERIMENT VERIFICATION

To verify the simulation results presented in the previous section, experiments on the pump under consideration will be conducted in this section.

Experimental research on pump characteristics is conducted on the hydraulic test bench (Fig. 16) in two cases: changing the pressure at the pump port and the pressure compensator's setting force. Equipment used for the experiment includes a hydraulic test bench (Fig. 16.a) with some main components including the LS pump (Parker PAVC65, Fig. 16c), flow sensor (AW LAKE R4B-6HV-25, Fig. 16g), pressure sensor (Huba Control 511, Fig. 16g), pressure compensator (Parker RS10R35S4SN1JW, Fig. 16h), a signal acquisition and processing device National Instrument, NI6009. Herein, the pressure and flow measuring devices are installed on the discharge line of the main pump in front of the load control valve or pressure compensator. The oil line after the load control valve leads to the tank. It should be noted that to test other system parameters and cross-compare the measured data, several additional sensors and measuring instruments are also integrated with the test bench.

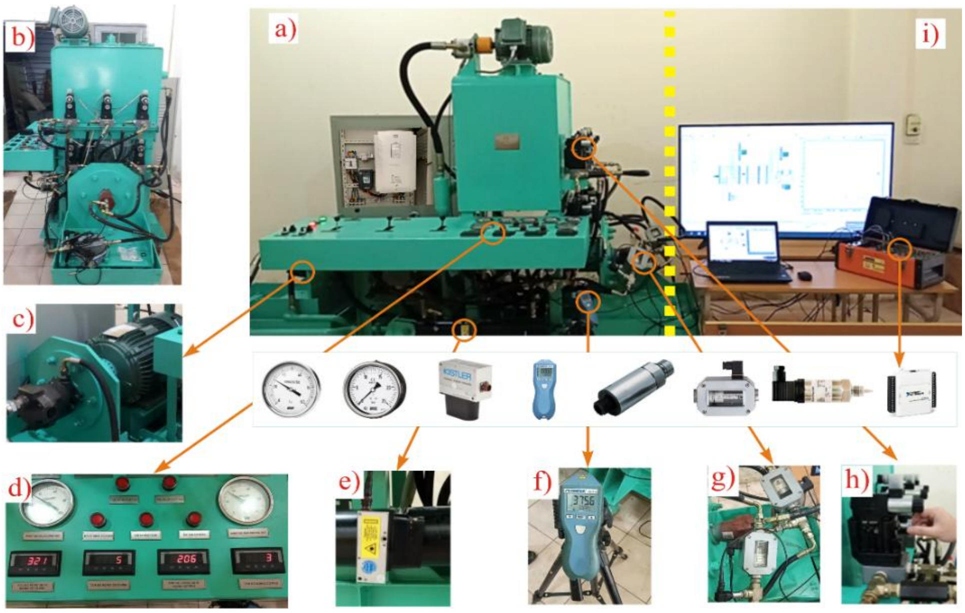

Fig. 16. Test bench with the main components: a) General layout of the test bench; b) Side view of the bench; c) Main electric motor and pump; d) Display panel; e) Distance sensor for measuring the cylinder displacement; f) Laser tachometer for measuring the rotational speed of the electric motor; g) Pressure/temperature sensor and flow rate meter; h) Relief valves for load generation; i) System for receiving, converting, and displaying the measured signals/data

Adjusting the load pressure at the pump port according to the rules of theoretical research is very difficult to do because manually adjusting the load pressure makes it very difficult to control the variation of pressure according to the set time. Therefore, the comparison of the results of theoretical studies and experimental studies was performed at relatively stable load pressure values.

In the first experimental case: Compare the results of theoretical and experimental research at the load pressure levels at the pump port of 140 bar and 190 bar (corresponding to Fig. 10). According to experimental research results, at load pressure values of 140 bar and 190 bar (Fig. 17a), the corresponding flow rates are 32 L/min and 22 L/min (Fig. 17b). Thus, the pump flow decreases corresponding to the increase of load pressure, however, compared to the theoretical research results, the pump flow values are lower (2÷3 L/min, corresponding error from 10 to 13%), due to measurement errors and pump internal leak.

In the second experiment case: When adjusting the pressure compensator at two pressure levels of 50 bar and 100 bar (Fig. 18a), the corresponding pump pressure is 80 bar and 130 bar, and the pump flow has a small change from 63 L/min to 60 L/min (lower than theoretical value 2÷5 L/min, corresponding error from 3 to 7%, Fig. 18b). Thus, this result is relatively consistent with theoretical research (65 L/min), the deviation is explained by the fact that as the pump pressure increases, the flow always leaks more. Thus, through two experimental cases, the experimental research results are completely consistent with the theoretical research results.

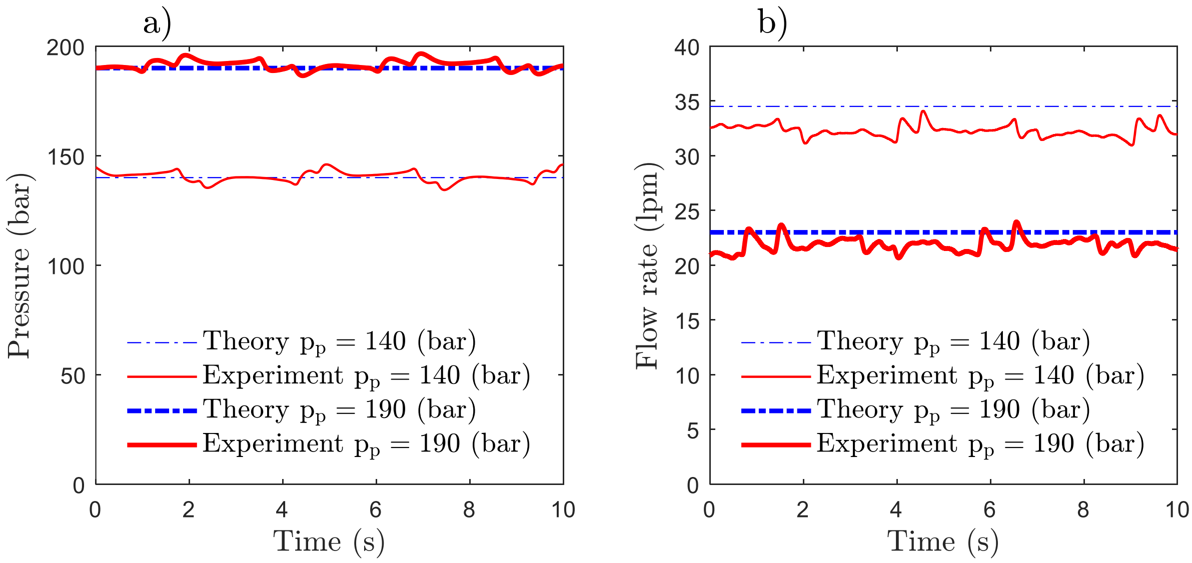

Fig. 17. Pump pressure and flow rate for the case of changing the pressure at the pump port

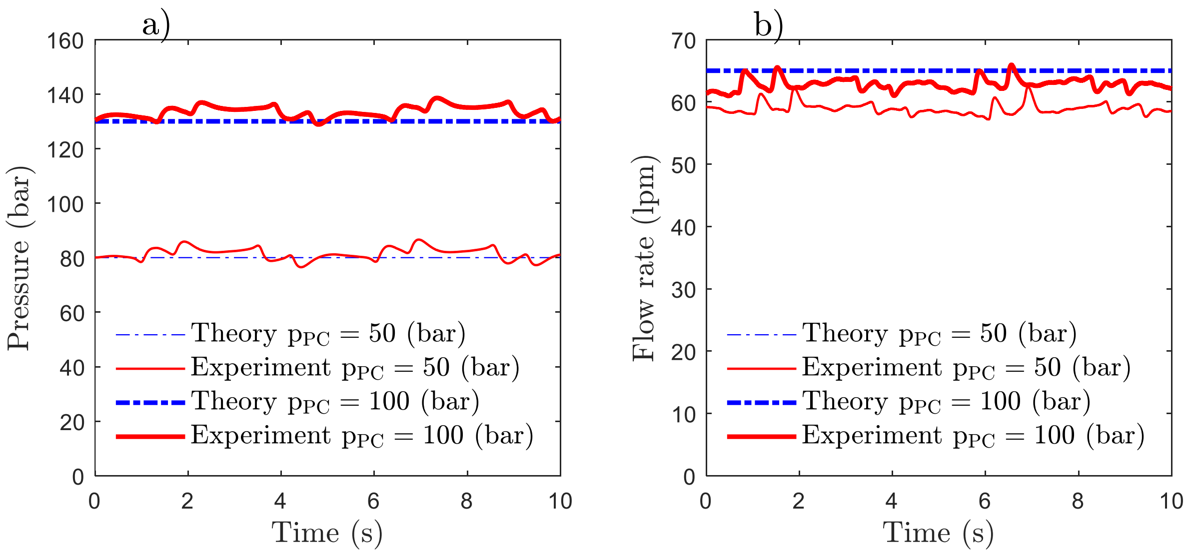

Fig. 18. Pump pressure and flow rate according to the pressure compensator setting

5. CONCLUSION

In the present work, a dynamic model of the multi-controlled variable displacement pump is developed based on differential equations describing the dynamics of each component of the hydraulic pump and hydraulic system to provide a comprehensive study of the dynamic characteristics of the pump. As a result, the hydraulic pump can quickly adapt to changing conditions of the system control signal (such as the opening degree of the distribution valve or flow control valve) and load to ensure the provision of appropriate flow rates for the system (cases 1 and 2). Additionally, both the initial setting force of the LS valve spring and the pressure setting of the pressure compensator significantly impact the hydraulic pump's dynamic characteristics (cases 3 and 4). The errors between theoretical and experimental research results are less than 13%. These research findings serve as crucial foundations for the rational setup and adjustment of hydraulic pumps, suitable for the capacity and technical condition of the pump, thereby enhancing their operational efficiency when integrated into hydraulic systems in many application fields (e.g., machinery, vehicles, industrial systems, and aviation).

ACKNOWLEDGEMENTS

This work is supported by the Le Quy Don Technical University (Grant No. 23.1.34).

CONFLICTS OF INTEREST

The authors declare no conflict of interest.

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