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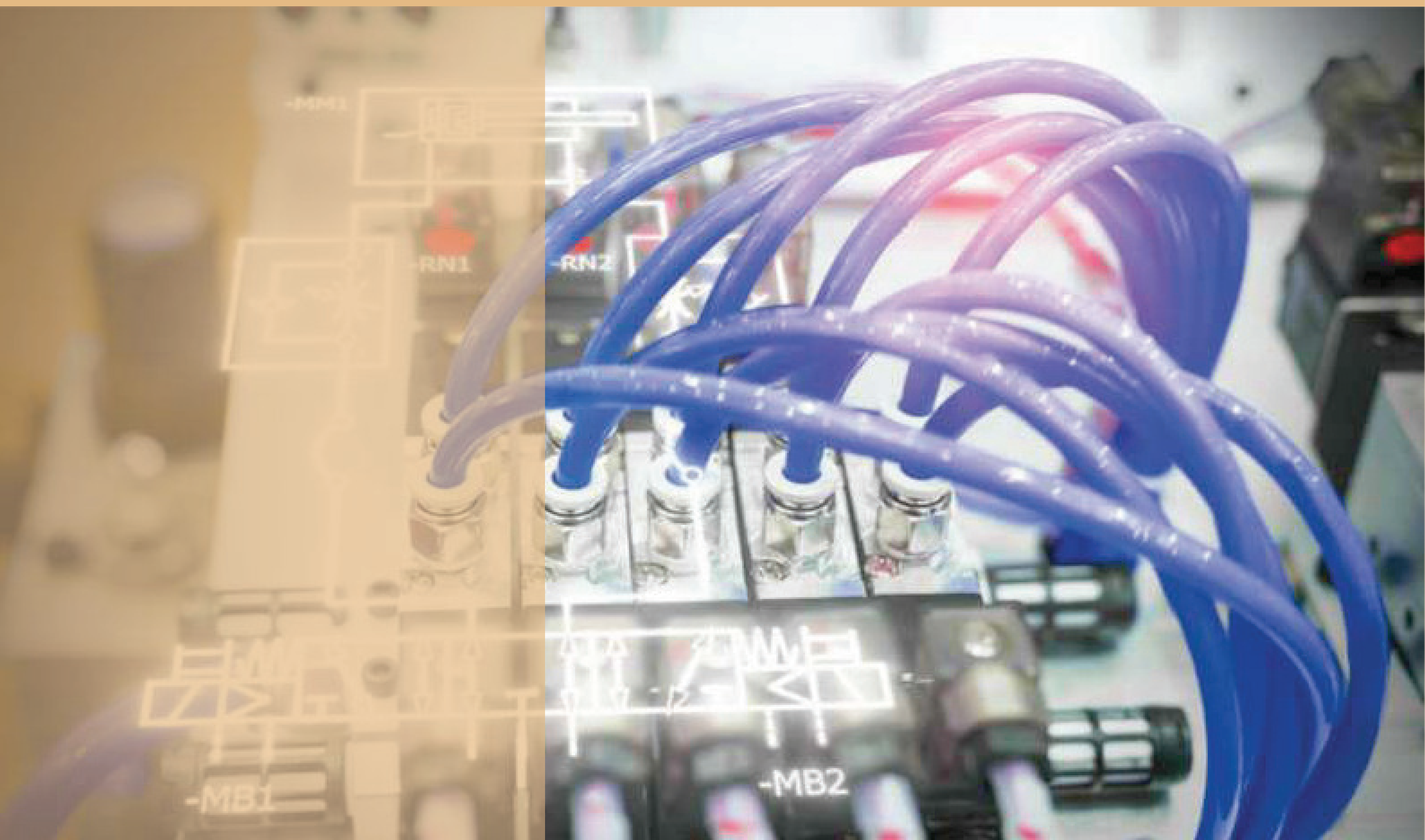

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Modelling and Simulation of the Transient Performance of a Direct Operated Pressure Relief Valve

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Abstract: *The dynamic characteristics determine variations of the inlet pressure in front of the valve in function of the flow through the valve in the time. In any hydraulic system, the valve is connected at least by a pipe at the outlet, and at its inlet there is some volume of compressible oil which influences the quality of the transient process. When switching the directional control valve in the hydraulic system with direct-operated pressure relief valves, a transient process occurs in which it is possible for the pressure to reach values many times higher than the set value. This causes the system to be overloaded with undesirable consequences.*

This paper examines experimentally and theoretically the transients in hydraulic systems with these valves. From the experimental static characteristics, the coefficient of hydrodynamic force acting on the valve poppet is determined.

Keywords: *Static and dynamic characteristics, pressure relief valve, transient response*

1. Introduction

The pressure control valves can perform different functions in the hydraulic systems, such as establish maximum pressure, reduce pressure in some circuit lines, and establish sequence movements, among other functions. The main operation of these valves consists of providing a balance between the pressure difference and the force load on a spring. Most of these valves can be positioned in many different levels, between totally open and totally closed, depending on the flow and the differential pressure. The pressure control valves are usually named according to their primary functions, and their basic function is to limit or to determine the pressure of the hydraulic system for the attainment of a certain function of the equipment in motion. In order to protect a hydraulic circuit against overloads and limit the work pressure, pressure relief valves are used. The main function of these valves is to limit the maximum working pressure in the hydraulic system. They are normally positioned after the hydraulic pump. In this case, the valve does not require an external power source, meaning that the fluid pressure is enough to open or close the valve. This means that the function of this class of valves is done automatically. This makes this class of valves indispensable for the hydraulic circuit function and operation.

There are two types of the pressure relief valves: direct operated and pilot operated. The direct operated pressure relief valves have higher deviation of the adjusted pressure in the static characteristics than pilot operated one, which leads to overloading of the hydraulic system [1]. The reason for this deviation is the hydrodynamic reaction force that acts on the valve poppet and always tends to close the poppet. To reduce the influence of the hydrodynamic reaction force to the slope of the static characteristics, shape modification of the valve poppet has been done.

Many authors have investigated the static and dynamic characteristics of direct operated pressure relief valves. Brodowski [3] has presented experimental and theoretical dynamic characteristics and shown that the magnitude of the pressure peak is far higher than the steady-state magnitude. He also has proved that the pressure peak depends on the size of the damping orifice. Many authors have worked on dependence of the discharge coefficient in the control orifice. During an unsteady process, the flow presumably passes in and out of laminar and turbulent regions. So, it is needed a model which describe both regimes simultaneously. That kind of model is recommended by Borutzky [8]. Another, empirical model for the discharge coefficient has been presented in [7]. For the pressure relief valve it is suitable to determine the discharge coefficient in the control orifice based on the experimental static characteristics [12]. That kind of model is presented and used in this paper to determine the discharge coefficient of the investigated pressure relief valve. High

impact to the static and dynamic characteristics has the hydrodynamic reaction force of the flow [2], [4], [10]. In the dynamic mode, it can even cause unstable work of the valve.
Dasgupta and Karmakar [5] studied the dynamics of a direct operated pressure relief valve with directional damping through bond graph simulation technique. The authors concluded that some significant parameters of the valve response are identified, which can be modified to improve the dynamic characteristics of the valve. Their theoretical research they have compared with the experimental dynamic characteristic presented by Watton [6].
Although this class of valves is indispensable for the function and operation of the hydraulic systems, a review of the available researches shows that their study is not well covered and there is a need for an in depth study of modeling and simulation of their performance. Therefore, a comprehensive study of the modeling and simulation of the performance of this class of valves, in the steady-state and transient modes of operation, is carried out in this paper. A comprehensive nonlinear mathematical model, taking into account most nonlinearities of the valve, is developed. The steady-state and transient performance of the studied valve are investigated theoretically and experimentally. The experimental study is also used to validate the simulation program of the studied valve in the steady-state and transient modes of operation.

2. Valve components description and schematic diagram

The objective of the pressure relief valve is to limit a system pressure downstream the valve. Fig. 1 shows the basic components of the studied valve, while its schematic diagram is shown on fig. 2. This valve basically consists of sleeve 1, adjusting spring 2, poppet with damping piston 3, and adjustment element 5. The system pressure setting can be infinitely varied by means of adjustment element 5. Spring 2 presses poppet 3 onto its seat. Port P is connected to the system. The system pressure acts on the poppet area. When the pressure in the port P rises above the value adjusted on spring 2, the poppet 3 moves against spring 2 and the valve is opening. Hydraulic oil can now flow from port P towards port T. The stroke of poppet 3 is limited by embossment 6.

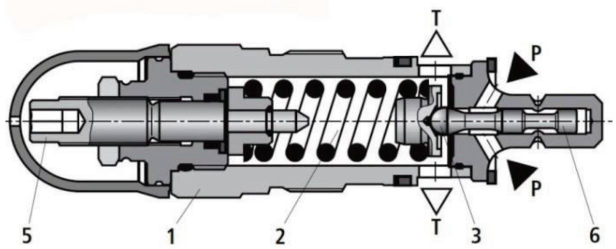

Fig. 1. Schematic diagram of the valve

On fig. 2 schematic diagram of the test rig with the studied pressure relief valve, volume of oil at its inlet V_0 and output pipeline with linear R_p and inertial L_p resistance is shown. To isolate the oil compressibility between the pump and the valve and for reducing pressure pulsation of the pump, it is included a throttle with high inertial resistance.

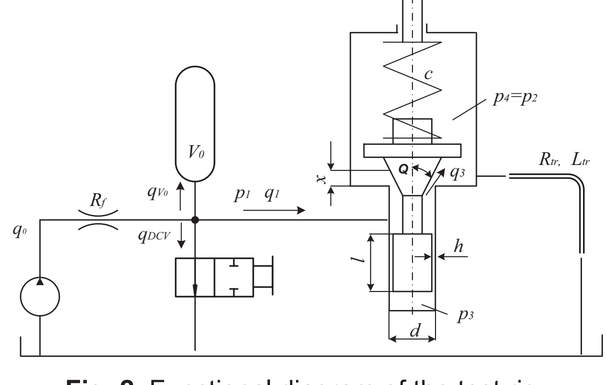

Fig. 2. Functional diagram of the test rig

The valve is normally closed. When the pressure p_1 is lower than the value necessary to move the poppet against the spring, the main valve throttling area remains closed and the valve poppet presses onto its seat. Rapidly activating the control valve V_1, the pressure begins to rises in the volume V_0 and in front of the valve and the transient process starts. When the pressure is high enough, the valve poppet lifts from its seat and the valve opens. Thus, the valve limits additional rising of the pressure downstream the valve.

3. Mathematical modelling of the studied valve

To model the studied valve, some assumptions are made in developing the nonlinear mathematical model. It is assumed that the tank pressure is constant at atmospheric pressure; the geometry and discharge area of the valve restriction usually change nonlinearly; the pressure losses in the short pipe lines are neglected; the oil temperature and viscosity are kept constant. During the transient mode of operation, the flow rate passing through the valve throttling area is of high Reynolds number. The discharge coefficient of this throttling area change with the Reynolds number in a complicated manner. For this reason, the discharge coefficient for the valve throttling is determined by the experimental static characteristics of the valve.

3.1. Coefficient of the hydrodynamic reaction force

The design of the poppet of this type of valve is characterized with turning the streaming flow of the oil with the ring 1, fig.3.
This turning of the streaming flow is leading to decreasing of the component of the hydrodynamic force F_h which, together with the spring force F_s, tends to close the valve. With this design modification, the error in the static characteristic is decreased. The value of the hydrodynamic force depends on the diameter and the shape of the ring 1 of the poppet and it is difficult to define it. Thus, it is necessary to use the experimental static characteristics with the displacement of the poppet of the valve measured, to determine the coefficient of the hydrodynamic reaction force.

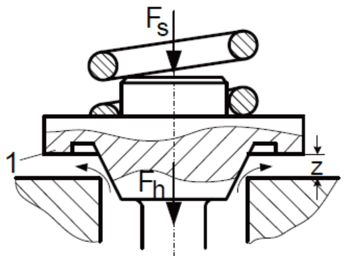

Fig. 3. Hydrodynamic force compensation

Experimental static characteristics of the specified direct operated pressure relief valve are presented on fig.4.

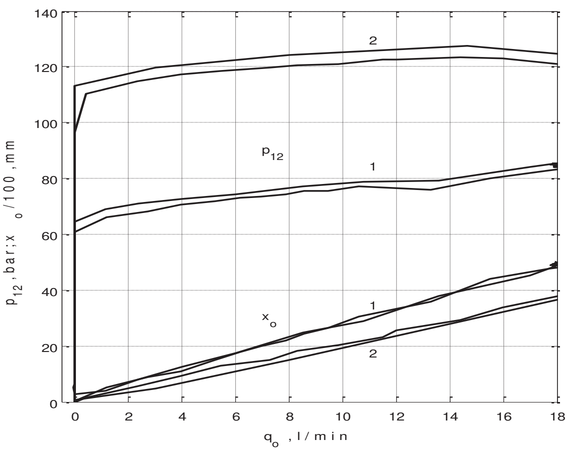

Fig. 4. Experimental static characteristic of the specified valve

For a given flow q_0, a pressure drop $p_{1,2}$, a poppet area A_k and a valve spring constant c and measured pressure – flow constant k_{st} of the static characteristic of the valve and displacement of the poppet x_0 of the valve (fig.3), the coefficient of the hydrodynamic force r_h has been calculated by the expression [12]:

$$r_h = \frac{\frac{k_{st} \cdot q_0 \cdot A_k}{x_0} - c}{p_{1,2}} \tag{1}$$

3.2. Discharge coefficient of the valve throttling area

As it is already mentioned, the discharge coefficient of the main throttling area of the studied pressure relief valve depends on the Reynolds number in a complicated manner. In the transient mode, the flow rate passing through the opening area of the valve restriction is assumed to be turbulent of unknown Reynolds number. Therefore, the flow rate q_3 passing through the main valve throttling area is given by the following equation:

$$q_3 = \mu \cdot \pi \cdot d \cdot x \cdot sin\theta \cdot \sqrt{\frac{2}{\rho} \cdot p_{1,2}} \tag{2}$$

where μ and $\pi \cdot d \cdot x \cdot sin\theta$ are the discharge coefficient and the opening area of the main throttling area of the valve. According to the eq. (2) it is possible to compute the discharge coefficient from the experimental static characteristics. The experimental discharge coefficient of the valve throttling area depending on the Reynolds number is expressed on fig.5.
The Reynolds number of the valve throttling area is given with the equation:

$$Re = \frac{v \cdot d}{\nu} = \frac{2 \cdot q_3}{\pi \cdot d \cdot \nu} \tag{3}$$

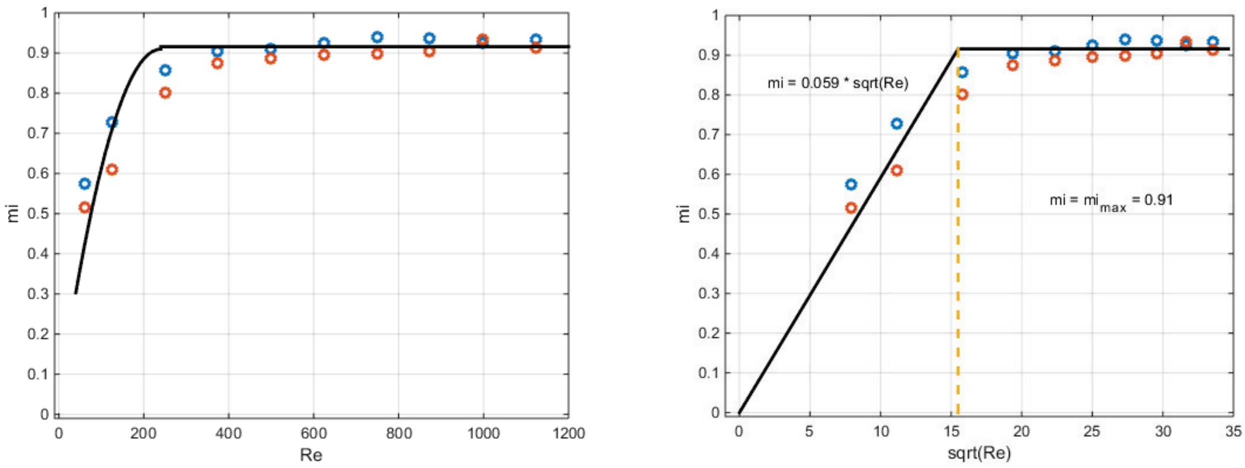

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Fig. 5. Discharge flow coefficient for the throttling area of the valve

As it is presented on fig.5 the discharge coefficient is changing linearly with the square root of Re up the limit value of $Re_{lim} = 240$ or $\sqrt{Re_{lim}} = 15.5$. After Re_{lim} the discharge coefficient is constant and it is around 0.91.
According to above statements the discharge coefficient can be expressed by the following equations:

$$\mu = \mu_{max} = 0.91 \quad for\ Re > Re_{lim}$$

$$\mu = 0.059 \cdot \sqrt{Re} \quad for\ Re < Re_{lim} \tag{4}$$

For the simplicity of calculation, in this paper the discharge coefficient is taken to be constant.

Woben [10] and Zehner [4] have presented the experimental discharge coefficient for different geometric parameters of the valve.

3.3. Mathematical model of the valve

Mathematical model of the system is described by the following equations:
According the fig. 2 the equation of continuity in front of the investigated pressure relief valve can be expressed as:

$$q_0 = q_{in} + q_v + q_1 \tag{5}$$

where q_{in}, q_v, and q_1 are the flow rate through restriction area in the directional control valve V1, the flow rate which enters in the volume V_0 and the flow rate entering in the valve, respectively. The transient variation of the restriction area of the directional control valve V1 affects the studied valve transient response. The flow rate q_{in} passing through the directional control valve is given by

$$q_{in} = (1 - \frac{t}{t_1}) \cdot \mu_v \cdot \pi \cdot d_v \cdot x_v \cdot \sqrt{\frac{2}{\rho} \cdot p_1} \tag{6}$$

where t_1 is closing time of the DCV, μ_v, d_v and x_v are the discharge coefficient, the diameter of the valve spool and valve spool displacement, respectively. The flow which enters in the volume V_0 can be expressed by the equation of the compressibility effect in the volume V_0:

$$q_v = \frac{V_0}{K} \cdot \frac{dp_1}{dt} \tag{7}$$

where K is the bulk modulus of the oil.
Equation of continuity in the valve in front of the control orifice and after it is

$$q_1 = q_2 = q_3 + A_k \cdot \frac{dx}{dt} \tag{8}$$

where: A_k – the area of the valve poppet; q_3 – the flow through the control orifice in the valve.
The equation of motion of the valve poppet is

$$m \cdot \frac{d^2x}{dt^2} + c \cdot (h_0 + x) + r_h \cdot x \cdot p_{1,2} = A_k \cdot (p_3 - p_4) - F_T \tag{9}$$

where: $m = m_k + \frac{1}{3} m_f$ – the equivalent mass of the valve poppet m_k and the spring m_f; c – the stiffness of the spring; h_0 – the deformation of the spring when $\mathrm{x} = 0$; r_h– the coefficient of the hydrodynamic force obtained by the expression (1); F_T – friction force between the valve poppet and the body of the valve.
The pressure in the lower chamber of the closing element of the valve p_3 depends on the losses in the orifice h between the piston of the valve poppet and the body of the valve:

$$p_3 = p_1 - R_{a,l} \cdot A_k \cdot \frac{dx}{dt} - R_{a,m} \cdot \left(A_k \cdot \frac{dx}{dt}\right)^2 - L_a \cdot A_k \cdot \frac{d^2x}{dt^2} \tag{10}$$

where: $R_{a,l}$, $R_{a,m}$ and $L_a = \rho \frac{l}{\pi dh}$ are linear, local and inertial resistances in the orifice with length l.
The pressure in the upper chamber above the valve poppet is obtain analogically when for this type of the valve is $p_4 = p_2$.
The pressure drop in the outlet pipeline is

$$p_2 = R_{p,l} \cdot q_2 + R_{p,m} \cdot q_2^2 + L_p \cdot \frac{dq_2}{dt} \tag{11}$$

where: $R_{p,l}, R_{p,m}$ and $L_{p,t}$ respectively linear, local and inertial resistance of the outlet pipeline with length l_p and diameter d_p.
Additional conditions were taken into account when solving the nonlinear system of the differential equations: the flow rate q_1 is zero when the valve is closed; the pressure p_1 cannot be less than the absolute vacuum; the displacement x of the valve poppet cannot be negative; the flow rate q_{in} is zero at $t > t_1$, etc., The mathematical model (2),(5)-(11) can be solved with computer programs for solving nonlinear differential equations. For solution of the system of the nonlinear differential

equations, the adaptive Runge-Kutta method has been used. This method based on the forth order Runge-Kutta method estimate the truncation error at each integration step and automatically adjust the time step size to keep the error within prescribed limits.

4. Experimental and theoretical characteristics of the researched valve

Fig. 6 presents the results of experimental and theoretical studies of a Bosch Rexroth type valve for a pressure of $60\,bar$ and oil volumes $V_0 = 52\,cm^3$ and $480\,cm^3$. The closing time t_1 of the directional control valve is less than $20\,ms$ and the flow rate of the pump is $q_0 = 25\,l/min$. The outlet pipe is $12\,mm$ in diameter and $1\,m$ long. The experiment was performed with pressure and displacement transducers and was recorded on a computer.
With a volume of oil at the inlet of $52\,cm^3$, a relatively large dynamic load is obtained, as the pressure reaches $100\,bar$ and the natural frequency is $742\,rad\,/\,s$. This leads to system overload, which in many cases is unacceptable. As the volume increases to $480\,cm^3$, the maximum pressure and the natural frequency of the transient process decrease to $85\,bar$ and $206\,rad\,/\,s$, respectively.

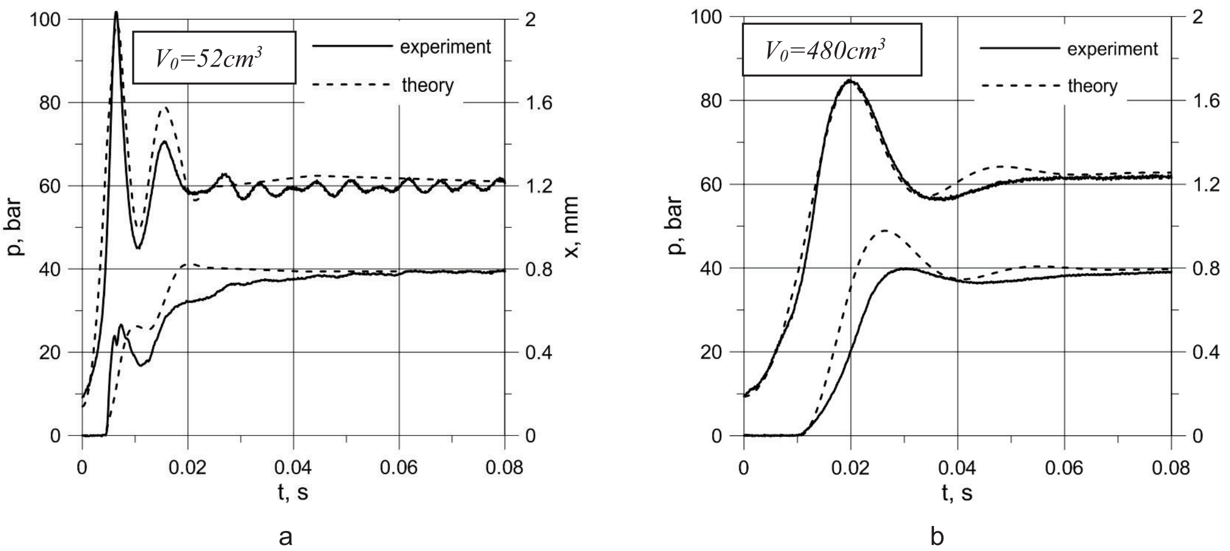

Fig. 6. Experimental and theoretical dynamic characteristics of the specified valve for different pressures and volumes at inlet port

The poppet of the valve opens when the spring-set pressure of $60\,bar$ is reached. The pressure in this phase of the transient process changes at a rate determined by the flow rate of the pump, the volume of oil at the inlet and the closing time of the directional control valve t_1. The difference between the experiment and the theoretical solution is due to the change in the hydrodynamic force and the slope of the static characteristic during the transient process, which are not taken into account in the mathematical model.

5. Conclusion

The steady-state and transient characteristics of a direct operated pressure relief valve are researched theoretically and experimentally. A comprehensive nonlinear mathematical model of the studied valve is deduced to predict the performance of the studied valve in the steady-state and transient modes of operation. The developed model, which takes into consideration most nonlinearities of the studied valve, is used to develop a computer simulation program. The steady-state and transient characteristics of the studied valve are simulated using this program. The experimental work aimed at validating the studied valve proposed model and the simulation program. The results showed good agreement between simulation and experimental results in the steady-state and transient modes of operation. The analysis of the simulation results showed that, when studying the performance of the hydraulic control valves, nonlinearity occurs due to the transient fluctuation in the valve operating pressures and the fluctuation in the throttling areas of the valve restrictions and their discharge coefficients. The transient fluctuation in the valve operating pressures causes nonlinear velocity changes of the fluid flow due to the high bulk modulus, which decreases during the valve operation. The throttling areas of the valve restrictions usually have nonlinear mathematical formulas. The discharge coefficients of these areas are assumed constant independent of flow rates and opening areas. They change in a complicated manner with the flow rates, Reynolds numbers, and the dimensions of the throttling areas. It was also found that the geometry of the throttling orifice, which connects the valve inlet port to the downstream port, plays an important role in the steady-state and transient performance of the studied valve. This result implies the need for further investigation the effect of the geometric parameters of the valve poppet on the steady-state and transient performance of hydraulic control valves.

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Main Constructive Solutions for Actual Wind Turbines Used for Green Power Generation

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Abstract: *It can be said that at the present time there is more than ever a need for energy at the global level to support the activities undertaken by the industrial branches as well as by the human communities. The possibilities of obtaining energy are limited and if we refer to the burning of fossil fuels, they have a direct effect on the environment, contributing decisively to the increase in global temperature values. Therefore, special attention must be paid to alternative methods that can be used to obtain energy; here we are talking about the action of the wind, the sun, or the force of sea waves. This paper presents the possibilities of obtaining energy from the wind action, on different models of wind turbines, with specific efficiency values for each of them, the main projects that have been carried out for capture both on land and offshore facilities, some of the most important onshore and offshore wind turbine parks established worldwide and for the Romanian area, but also the main results obtained at the present time in terms of energy amounts of these wind facilities.*

Keywords: *Wind action, turbine, constructive solutions, wind farm, power generation*

1. Introduction

The wind action can be used for power generation at the level of a turbine rotor that is able to convert wind mechanical energy into electric energy by means of an electric generator.
The turbine constructive solution result as an entire assembly that based on the wind force can generate energy in a constant manner.
The wind interaction with the turbine rotor takes place at the blades level, which are specially designed and constructed to be oriented directly towards the wind action direction to achieve a lift displacement and then rotation movement necessary for a continuous operation that coincides with the duration of the wind action.
Hence, the areas differentiation where the wind turbines facilities installation is indicated where the wind velocity values are constant and over a long period of time throughout the year.
The main types of wind turbines used mainly for wind farms with high production capacity (installed power) are three bladed horizontal axes (HAWT), while the vertical axis wind turbines (VAWT) are mainly used for smaller applications and where the wind speed is not very high, as they have the possibility to start and operate at low wind speeds.

2. The atmospheric masses movement and the winds action formation

Due to the atmospheric pressure values, in conjunction with the uneven heating of the atmospheric masses and the Earth rotational movement, a continuous movement of the air masses at the planetary level is created.
It should be emphasized that the Earth rotation movement involves concerted actions between the Coriolis forces, the centrifugal force, the frictional force between the air particles that act differently depending on the non-uniform character of the Earth's surface, the pressure gradient, the effects of turbulence, as well as the possibility of atmospheric air masses to transport water vapor (advection).

Table 1: Specific involved efforts in air masses displacement

Current number	Effort type	Specific relation
1.	Coriolis	$F_{Cor} = 2\omega v \sin\phi$
2.	Centrifugal	$F_{Cen} = m\omega^2 r$