ANNALS OF FACULTY ENGINEERING HUNEDOARA
– International Journal of Engineering
Tome XI (Year 2013) – FASCICULE 2 (ISSN 1584 – 2665)

[1.] Sasko S. DIMITROV

INVESTIGATION OF STATIC CHARACTERISTICS OF PILOT OPERATED PRESSURE RELIEF VALVES

[1.] TECHNICAL UNIVERSITY OF SOFIA, BULGARIA

ABSTRACT: Theoretical and experimental investigation of the static characteristics of the pilot operated pressure relief valve is presented in this article. A mathematical model of pressure drop vs. flow depending for pilot operated pressure relief valve is developed. An experimental test stand was created for experimental determination of the static characteristics and compared with each other which confirm the mathematical model. The results of solving the mathematical model and experimental investigation are presented in few diagrams. A few directions for improving the static characteristics are given, especially at the moment of opening of the main valve. Advantages and disadvantages of the static characteristics are discussed.
KEYWORDS: Pilot operated pressure relief valve, flow, pressure, static characteristic, mathematical model, experiment

INTRODUCTION

Main feature of the static characteristic of the relief valves is its slope. The slope of the static characteristic is called pressure-flow coefficient. For the same setting pressure and flow, the pressure-flow coefficient is higher for direct operated pressure relief valves comparing with pilot operated pressure relief valves [1],[3]. Additional decrease of the pressure-flow coefficient of the conventional pilot operated pressure relief valves can be obtain with built-in compensating control piston in front of the pilot valve. Another deficiency of the static characteristic of the pilot operated pressure relief valve is the difference between the pressures of opening of the pilot valve and the main valve. This error can be reduced also with a compensating control piston in the pilot valve [9].

In [7] the authors have shown theoretically and experimentally the advantage of the pilot operated pressure relief valves-lower pressure-flow coefficient of the static characteristic of the valve. But it cannot be clearly seen the pressure difference between opening of the pilot and the main valve. In [6] a special attention has been taken to the influence of the hydrodynamic reaction force to the valve characteristics. Also, the author has investigated different designs of the resistance orifices in the pilot chain to improve the characteristics of the valve. Most comprehensive theoretical mathematical model is presented in [4], in details explained and included all the factors influencing the quality of the static characteristics. This model was developed for valve with "pilot flow through main valve". In this study this model has been little modified for valve with "round about pilot flow". To improve the static characteristic of the valve the built-in compensating control piston with resistance orifice in it in the pilot chain is taken into consideration. Expressions for the pressure-flow coefficient of the valve without compensating control piston and with compensating control piston are obtained numerically to determine the pressure-flow coefficient of the valve. The influence of the diameter of the compensating control piston to the static characteristics of the valve is presented, as well. This study has experimentally proved the validity of this model.

FUNCTIONAL DIAGRAMS OF THE VALVES

On figure 1 a functional diagram of the pilot operated pressure relief valve without compensating control piston a) and with compensating control piston b) is shown.

This valve can be observed as a system consisted of three subsystems: main valve, pilot valve and fixed orifices (R_1 and R_3). In neutral position both pilot and main valves are closed under the influence of the springs, and there is a balance of forces at the closing element of the main valve. When inlet pressure p_1 will reach higher value than the preset spring force of the pilot valve, the closing element of the pilot valve is opening and through the orifices R_1 and R_3 beginning to flow some little amount of pilot flow q_y. The pressure p_4 in the upper part of the main valve is maintaining approximately constant by the pilot valve. With further increase of the inlet pressure p_1 the pressure drop $p_{1,4} = p_1 - p_4$ continues to increase until the main valve opens and the flow $q_1 = q_3 + q_y$ is flowing to the tank.

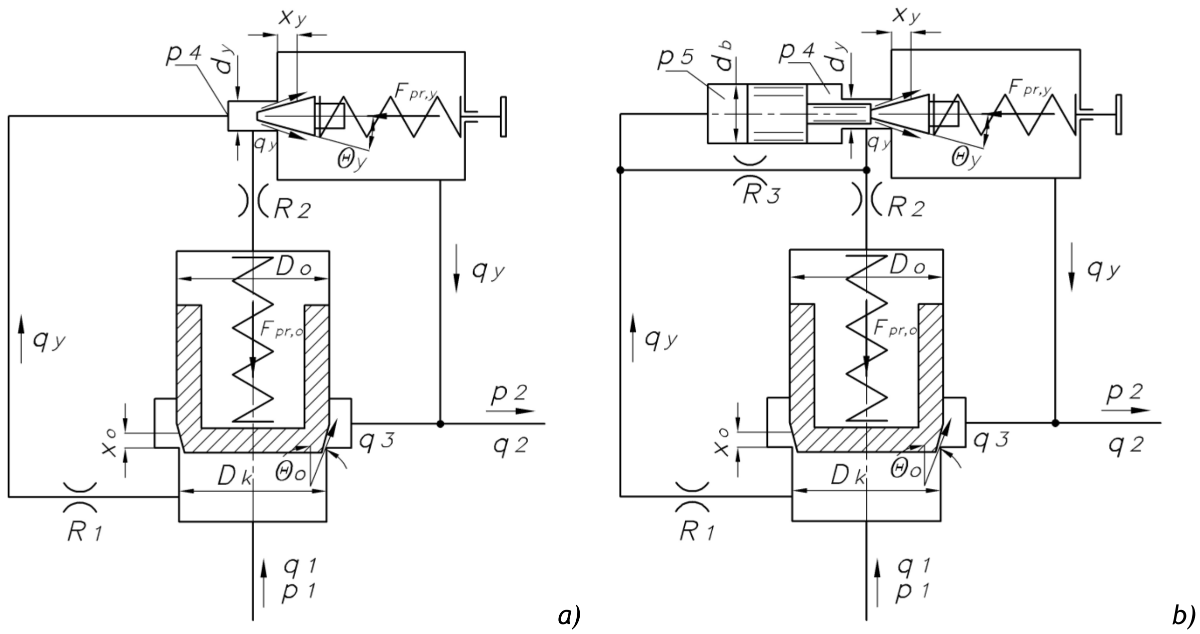


Figure 1. Functional diagram of a pilot operated pressure relief valve

MATHEMATICAL MODEL OF THE STATIC CHARACTERISTICS

Static characteristic of a pressure relief valve shows changing of the control parameter (pressure drop $p_{1,2}$) depending of the inlet parameter (inlet flow q_1).

For theoretical determination of the static characteristics a methodology presented in [4] is used in this study. For this type of pressure relief valve-"with pilot flow through main valve", the mathematical model in [4] is slightly modified. There is, also, an additional equation for balance forces on the compensating control piston.

The static characteristics of the pilot operated pressure relief valves are described with following equations:

- *Flow equation across the pilot valve*

$$q_y = \mu_y \cdot d_y \cdot \pi \cdot x_y \cdot \sin\theta_y \cdot \sqrt{\frac{2}{\rho} \cdot p_{4,2}} \qquad (1)$$

where: q_y - the flow across the pilot valve; μ_y - the flow coefficient of the pilot valve; d_y - the seat diameter of the pilot valve; x_y - the displacement of the closing element of the pilot valve; θ_y - the angle of flowing of the oil at the pilot valve, ρ - the density of the oil; $p_{4,2} = p_4 - p_2$ - the pressure drop in the pilot valve.

- *Balance of forces acting on the closing element of the pilot valve*

$$c_y \cdot (h_y + x_y) = p_{4,2} \cdot A_y + p_{5,4} \cdot A_b - r_y \cdot x_y \cdot p_{4,2}$$

or

$$x_y = \frac{p_{4,2} \cdot A_y + p_{5,4} \cdot A_b - c_y \cdot h_y}{c_y + r_y \cdot p_{3,2}} \qquad (2)$$

where: A_y - the area of the seat of the pilot valve; c_y - the spring constant of the pilot valve; h_y - the previous deformation of the spring of the pilot valve; $r_y = 2\ \mu_y \cdot \pi \cdot d_2 \cdot \sin\theta_0 \cdot \cos\theta_0$ - the hydrodynamic force coefficient of the pilot valve; A_b - the area of the compensating control piston; d_b - diameter of the compensating control piston; $p_{5,4}$ - pressure drop in the compensating control piston orifice.

If we solve the equations (5) and (6), the static characteristic of the pilot valve will be obtained:

$$q_y = \mu_y \cdot d_y \cdot \pi \cdot \sin\theta \cdot \sqrt{\frac{2}{\rho} \cdot p_{4,2}} \cdot \frac{p_{4,2} \cdot A_y + p_{5,4} \cdot A_b - c_y \cdot h_y}{c_y + r_y \cdot p_{3,2}} \qquad (3)$$

- *Pressure drop at the fixed orifices*

$$p_{1,4} = R_{1l} \cdot q_y + R_{1m} \cdot q_y^2 + R_{3l} \cdot q_y + R_{3m} \cdot q_y^2 \qquad (4)$$

$$p_{1,4} = (R_{1l} + R_{3l}) \cdot q_y + (R_{1m} + R_{3m}) \cdot q_y^2 = R_l \cdot q_y + R_m \cdot q_y^2$$

where: $p_{1,3} = p_1 - p_3$ - the pressure drop at the pilot chain, $R_l = R_{1l} + R_{3l}$ - the linear hydraulic resistance in the orifices R_1 and R_2; $R_m = R_{1m} + R_{3m}$ - the local quadratic resistance in the orifices R_1 and R_3; A_{dr} - the area of the orifice R_1 and R_3; d_{dr1} - the diametar of the orifice R_1; l_1 - the length of *the orifice R_1; d_{dr3} - the diameter of the orifice R_3, l_3 - the length of the orifice R_3; υ - the viscosity of oil.*

□ *Pressure drop at the main valve*

$$p_{1,2} = p_{1,4} + p_{4,2} \tag{5}$$

where: $p_{1,2} = p_1 - p_2$ - the pressure drop at the main valve

□ *Balance of forces acting on the closing element of the main valve*

$$p_{1,4} \cdot A_k - p_{1,2} \cdot \Delta A = c_0 \cdot (h_0 + x_0) + r_0 \cdot x_0 \cdot p_{1,2}$$

or

$$x_0 = \frac{p_{1,4} \cdot A_k - p_{1,2} \cdot \Delta A - c_0 \cdot h_0}{c_0 + r_0 \cdot p_{1,2}} \tag{6}$$

where: A_k - the area of the closing element of the main valve; ΔA - the unbalanced area at the closing element of the main valve; $\varphi = \Delta A / A_k$ - geometric parameter of the valve; h_0 - the previous deformation of the spring of the main valve; x_0 - the displacement of the closing element of the main valve; $r_0 = 2\,\mu_o \cdot \pi \cdot D_k \cdot \sin\theta_0 \cdot \cos\theta_0$ - the hydro-dynamic force coefficient of the main valve; μ_0 - the flow coefficient of the main valve; D_k - the diameter of the closing element of the main valve; θ_0 - the angle of flowing of the oil at the pilot valve.

□ *Flow across the main valve*

$$q_3 = \mu_0 \cdot D_0 \cdot \pi \cdot \sin\theta \cdot \sqrt{\frac{2}{\rho} \cdot p_{1,2}} \tag{7}$$

where: q_3 - the flow across the main valve.

□ *Flow through pilot chain*

$$q_1 = q_3 + q_y \tag{8}$$

The static characteristics of the pilot operated pressure relief valves are fully described by the equations (1) to (8). From equations (1) – (8) theoretically can be expressing the pilot flow (9) and the difference between the pressure of opening of the pilot valve and the main valve (10) if there is built-in compensating control piston [5], [9]:

$$(q_y)_{00} = \sqrt{\frac{p_{1,00} \cdot \varphi + \dfrac{c_0 \cdot h_0}{A_k} - (R_{1l} + R_{3l}) \cdot (q_y)_{00}}{R_{1m} + R_{3m}}} \tag{9}$$

According to equation (9), the pilot flow directly depends of the geometric parameter of the valve φ and the resistance in the pilot orifices R_1 and R_2. Pilot oil flow is always higher for pilot operated pressure relief valve without compensating control piston.

$$p_{1,00} = \frac{1}{1-\varphi}\left(p_{1,0y} + \frac{c_0 \cdot h_0}{A_k} + \frac{x_y \cdot (c_y + r_y \cdot p_{4,0})}{A_y} - \frac{A_b}{A_y}\left(R_{3l} \cdot (q_y)_{00} + R_{3m} \cdot (q_y)_{00}^{\;2} \right) \right) \tag{10-a}$$

For pilot operated pressure relief valve without compensating control piston, the pressure of opening of the main valve is:

$$p_{1,00} = \frac{1}{1-\varphi}\left(p_{1,0y} + \frac{c_0 \cdot h_0}{A_k} + \frac{x_y \cdot (c_y + r_y \cdot p_{4,0})}{A_y} \right) \tag{10-b}$$

Equations (10-a) and (10-b) shows that the pressure difference between pilot and main valve is lower when there is built-in compensating control piston in the pilot valve. With increasing the diameter of compensating control piston d_b, i.e. the area of the compensating piston, the last part of the equation (10-a) is getting greater and the pressure difference is getting lower. The effect of reducing the difference between the pressure of opening of the pilot and the main valve experimentally is shown on Figure 7.

With linearization of the equations (1) - (8) around the steady state values of pressure p_0 and flow q_0, the statism can be expressed as [4]:

$$k_{st} = \frac{c_0 + r_0 \cdot (p_{1,2})_0}{\mu_0 \cdot \pi \cdot D_0 \cdot \sin\theta_0 \sqrt{\dfrac{2}{\rho} \cdot (p_{1,2})_0} \cdot A_k} \cdot \left[1 + \frac{k_{st,y}}{(R_{1l} + R_{3l}) + (R_{1m} + R_{3m}) \cdot 2 \cdot q_{y,0}} \right] \tag{11}$$

The pressure-flow coefficient of the pilot valve without compensating control piston is:

$$k_{st,y} = \frac{c_y + r_y \cdot (p_{4,2})_0}{\mu_y \cdot \pi \cdot d_y \cdot \sin\theta_y \sqrt{\dfrac{2}{\rho} \cdot (p_{4,2})_0} \cdot A_y} \tag{12-a}$$

The pressure-flow coefficient of the pilot valve with compensating control piston is:

$$k_{st,y} = \frac{c_y + r_y \cdot (p_{4,2})_0}{\mu_y \cdot \pi \cdot d_y \cdot \sin\theta_y \sqrt{\frac{2}{\rho} \cdot (p_{4,2})_0} \cdot A_y} - (R_{3l} + 2 \cdot q_{y,0} \cdot R_{3m}) \cdot \frac{A_b}{A_y} \quad (12\text{-}b)$$

From equations (12-a) and (12-b) it can be clearly seen the difference in the pressure-flow coefficient of the static characteristic of the valve without compensating piston (12-a) and with compensating piston (12-b). Decreasing of the pressure-flow coefficient can be achieved by increasing the diameter of the compensating piston d_b, i.e. by increasing the area of the compensating piston A_b. If the second part of the equation (12-b) is equal to the first part, the pressure-flow coefficient would be zero and it would be obtained ideally horizontal static characteristic. But, if the second part of the equation (13) is greater than the first part, the pressure-flow coefficient would be negative and it is possible the valve turn to work unstable in the steady state regime.

On Figure 3 a pressure-flow coefficient dependence of the diameter of the compensating piston for different pressure settings and flows across the valve is presented. It can be seen that valve with compensating control piston with diameter d_b = 65.5 mm and flow across the valve q_1 = 30 l/min will have negative pressure-flow coefficient above 65 bar. If the setting pressure is below 65 bar, the pressure-flow coefficient is positive and the valve is stable in steady state regime. Experimental confirmation of this claim is shown on Figure 4.

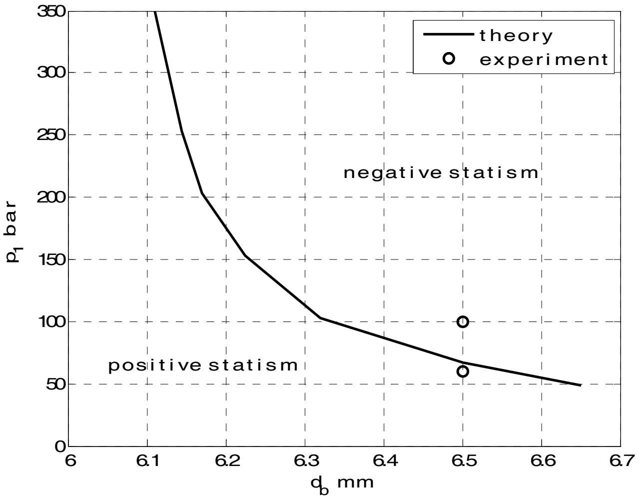


Figure 3. Pressure-flow coefficient dependence of the d_b for different pressure settings and flows

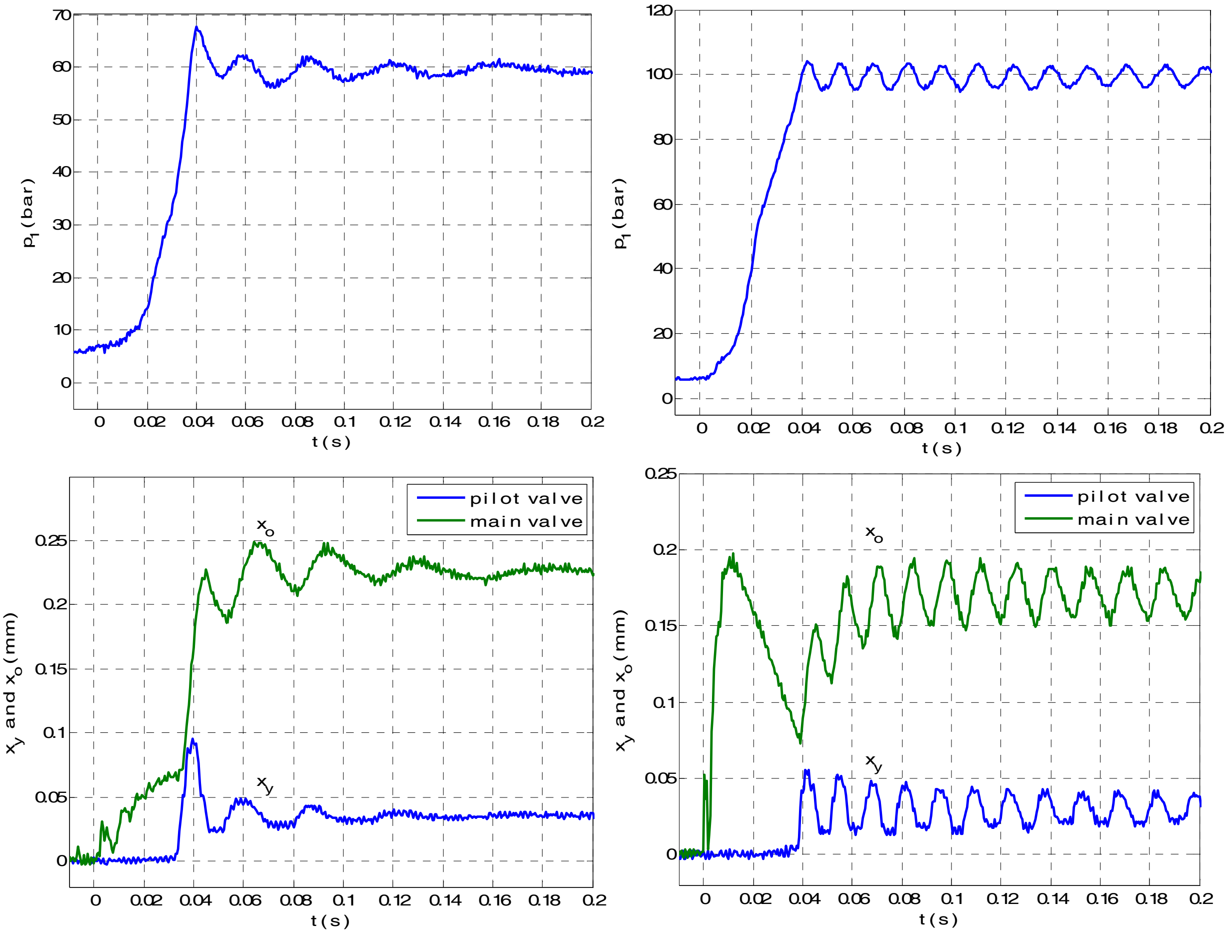


Figure 4. Stable and unstable work of the pilot operated pressure relief valve with compensating piston

On Figure 4-a a stable work of the pilot operated pressure relief valve with compensating control piston with diameter d_b = 65.5 mm, working pressure of 60 bar and flow across the valve 30 l/min is shown. On Figure 4-b an unstable work of the same valve at the same condition but with increased working pressure of 100 bar is shown. At every working pressure above 65 bar the valve is working unstable because the pressure-flow coefficient is negative, according to Figure 3.

EXPERIMENTAL AND THEORETICAL CHARACTERISTICS

The measurement instruments were previously calibrated. A pressure transducer type HM17 manufactured by BoschRexroth was used for pressure measurement. For displacement of the valve a position sensor manufactured by BoschRexroth was used. The data are stored in the computer through 14 bit data acquisition card manufactured by National Instruments.

The subject of investigation was Denison pressure relief valve type R4V 06, shown on figure 6 [8]. The parameters of the specified pressure relief valve are: d_y = 5 mm, c_y = 250 N/mm, d_b = 5.5 mm, μ_y = 0.65, υ = 34 cSt, ρ = 890 kg/m^3, d_{dr1} = d_{dr3} = 0.8 mm, d_{dr2} = 0.68 mm, l_{dr1} = l_{dr3} = 1 mm, D_k = 28.5 mm, D_0 = 28 mm, c_0 = 7 N/mm, h_0 = 16.5 mm, μ = 0.6. Theoretical and experimental static characteristics are presented on figure 7.

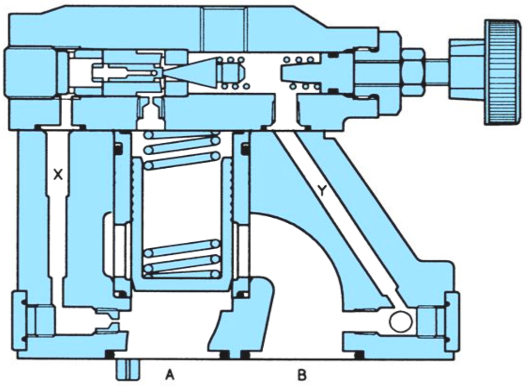
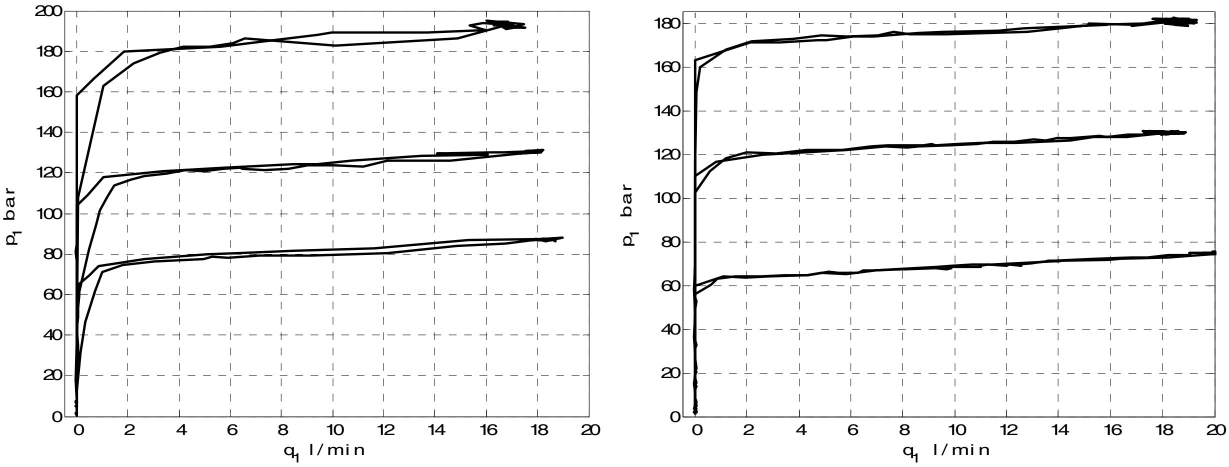
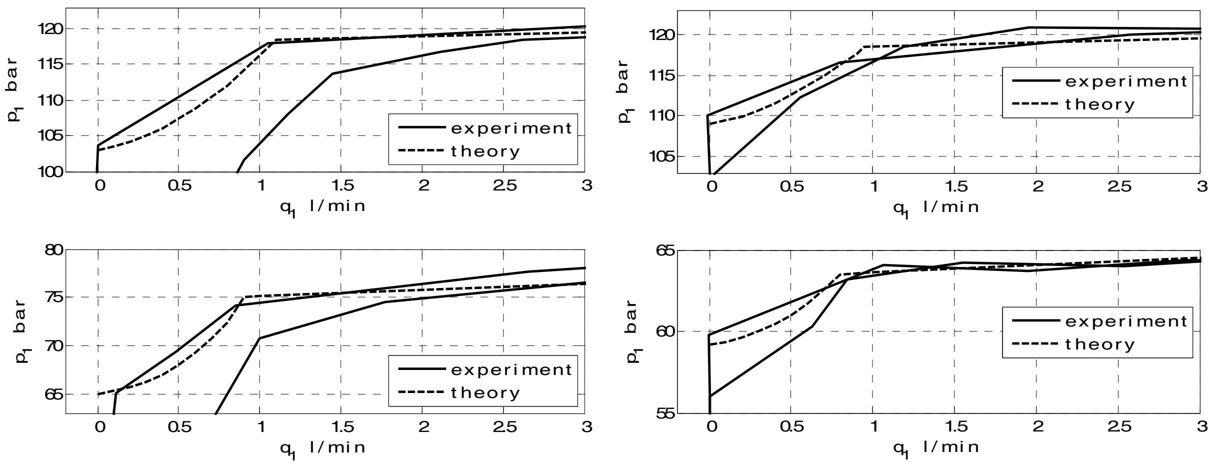


Figure 6. The investigated pressure relief valve [7]



a) without compensating piston *b) d_b=5.5 mm*



c) without compensating piston *d) d_b=5.5 mm*

Figure 7. Theoretical and experimental static characteristics of the specified pressure relief valve

On figure 7-a static characteristic of the pilot operated pressure relief valve without compensation control piston is shown. On figure 7-b static characteristic of the pilot operated pressure relief valve with compensation control piston d_b = 5.5 mm is shown. A comparison between experimental and theoretical static characteristic of the specified pressure relief valve, for two pressure settings-around 60 bar and around 100 bar, without compensating control piston (figure7-c) and with compensation control piston d_b = 5.5 mm (figure7-d) is shown zoomed for lower flows across the valve. It can be noticed that the presence of the compensating control piston reduces the pilot flow and the pressure difference, according to (10). The experimental investigation of the valve proved the *proposed mathematical model for theoretical determination of the static characteristics of the pilot operated pressure relief valves with and without compensating control piston.*

At the diagrams on figure 7 can be seen that pressure difference between opening of the pilot and the main valve is higher for the valve without compensating piston and it is decrease with the increasing of the compensating control piston diameter d_b. Also, the pilot oil flow and the pressure-flow coefficient of the valve are decreasing with increasing of the compensating control piston diameter d_b and higher without compensating control piston.

CONCLUSIONS

Experimental investigation proved the theoretical examination (eq. 10-a and eq. 10-b) that inserting a compensating control piston in front of the pilot valve can reduce the pilot flow and the pressure difference between opening of the pilot valve and the main valve. The error at the beginning of opening of the valve can be reduced if the relative ratio of the areas of the compensating piston and the seat of the pilot valve is increased. Figure 7 shows that a valve without compensating control piston has pressure error of around 10 bar at 60 bar pressure setting and around 15 bar at 100 bar pressure setting (Figure 7-a and Figure 7-c). Inserting the compensating control piston with d_b=5.5 mm, the pressure error is around 4 bar at 60 bar pressure setting and 8 bar at 100 bar pressure setting (Figure 7-b and Figure 7-d). Theoretical investigation says that higher diameter of the compensating control piston d_b can much more reduce the pressure error (10), but it is possible the pressure-flow coefficient of the static characteristic of the valve to be negative (12) in that case. Graphically it is presented the values of the pressure-flow coefficient depending of the diameter of the compensating control piston d_b, Figure 3, and the limit values of d_b when valve is working unstable. This claim is also experimentally confirmed and presented on Figure 4. According to eq. (9) and eq. (10) additional improvement – decreasing of the pilot oil flow and pressure difference between opening of the pilot and the main valve can be achieved by minimizing the geometric parameter φ of the main valve.

REFERENCES

[1.] Backé W., Murrenhoff H. Grundlagen der Ölhydraulik. Institut für fluidtechnische Antriebe und Steuerungen. Technische Hochschule Aachen. 1994

[2.] Will D., Ströhl H., Gebhardt N. Hydraulik. Springer-Verlag Berlin. 1999.

[3.] Brodowski W. Beitrag zur Klärung des stationären und dynamischen Verhaltens direktwirkender Druckbegrenzungsventile. Dissertation. 1974.

[4.] Komitovski M. Elements of hydro- and pneumo-drives (in Bulgarian). Technika, Sofia, 1985.

[5.] Komitovski M. Pressure controlling in hydraulic systems with pressure-unloading valves (in Bulgarian). Scientific Conference EMF99, Sozopol, Bulgaria, 1999

[6.] Wobben, G. D. Statisches und dynamisches Verhalten vorgesteuerter Druckbegrenzungsventile unter besonderer Berücksichtigung der Strömungskräfte, Dissertation, RWTH Aachen, 1978;

[7.] Takenaka, T.,Urata, E. Static and Dynamic Characteristics of Oil-Hydraulic Control Valves, Fluid Power International Conference, Tokyo, 1968

[8.] DENISON HYDRAULICS GmbH, Pressure relief valve R4V, Catalogue sheet 3-EN 2400-A;

[9.] Dimitrov, S. Static characteristics of pilot operated pressure relief valves with compensating control piston. Scientific Conference EMF2011, Sozopol, Bulgaria, 2011