TECHNICAL UNIVERSITY OF SOFIA

Faculty of Power Engineering and Power Machines

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PROCEEDINGS

of the

XVI National Scientific Conference with International Participation

Energy- Ecology- Comfort- Self- confidence

$17^{th} – 20^{th}$ September 2011 , Sozopol, Bulgaria

Vol. I I

Hydroaerodynamics and Hydraulic Machines

Textile Engineering

STATIC CHARACTERISTICS
OF PILOT OPERATED PRESSURE RELIEF VALVES
WITH COMPENSATING PISTON

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Abstract: In this paper a mathematical model of the static characteristics for a pilot operated pressure relief valve with compensating piston has been developed. The static characteristic has been presented in few diagrams. The influence of the compensating piston to the static characteristic has been emphasized as a manner for reducing the difference between pressure of opening of the pilot valve and the main valve.
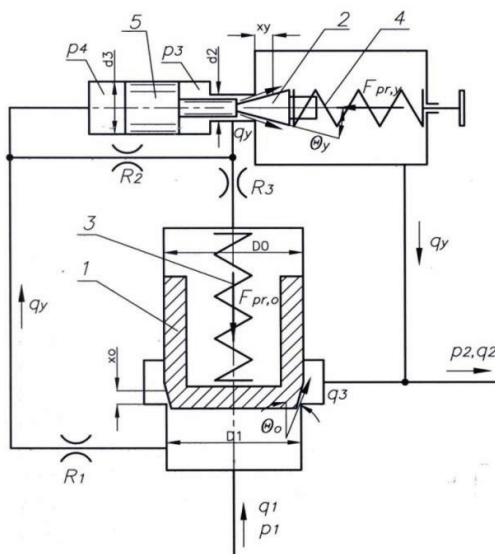
1. INTRODUCTION

There are few different designs of pilot operated pressure relief valves. The main feature of this type of valves is a difference between pressures of opening of the pilot valve and the main valve i.e. relative error at the beginning of opening of the valve. This error can be reducing with built-in compensating piston in front of the pilot valve [1].

Many authors [4], [5] have investigated the static characteristics of pilot operated pressure relief valves without compensating piston and different mathematical models have been obtained. But there has not been any significant investigation on influencing of the compensating piston to the static characteristics yet, although this type of valve has been a part of the standard production program of *Denison Hydraulics GmbH* [1] for long time. In this paper an attempt has been done, with contemporary mathematical approach, to determine theoretical static characteristics of pilot operated pressure relief valves with compensating piston and to emphasize influencing of the compensating piston to the static characteristics.

2. PRINCIPLE OF OPERATION

On *fig.1* a functional diagram of a pilot operated pressure relief valve with compensating piston is shown. This valve can be observed as a system consisted of three subsystems: the main valve *1,* the pilot valve *2* with the compensating piston *5* and the fixed orifices (R_1, R_2 and R_3). In initial position both the pilot and the main valve are closed under the influence of the springs *3* and *4*, and there is a balance of forces at the closing element of the main valve *1*. When inlet pressure p_1 will reach higher value than the preset spring force *4* of the pilot valve, the closing element of the pilot valve *2* is opening and through the orifices R_1 and R_2 beginning to flow some little amount of pilot flow q_y. The pressure p_3 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 continues to increase up to $p_{1,3} = F_{1,3}/A_1$ at which the main valve is opening and the flow $q_1 = q_3 + q_y$ is flowing to the tank. By changing the flow q_1 the pressure drops $p_{1,2}$ and $p_{1,3}$ also change which leads to moving the closing element of the main valve *1*.

Fig.1. Functional diagram of the pilot
   operated pressure relief valve

3. MATHEMATICAL MODELING OF THE STATIC CHARACTERISTIC

The static characteristic of a pressure relief valve shows changing of the control parameter - pressure drop $p_{1,2}$ depending on the inlet parameter- flow q_1.

The static characteristics of the pilot operated pressure relief valves have been described with following equations [3]:

➢ *Flow equation across the pilot valve*

$$q_y = \mu_y \cdot d_2 \cdot \pi \cdot x_y \cdot \sin\theta_y \cdot \sqrt{\frac{2}{\rho} \cdot p_{3,2}} \qquad (1)$$

where: q_y – the flow across the pilot valve; μ_y - the flow coefficient of the pilot valve; d_2 – the seat diameter of the pilot valve; x_y - the displacement of the closing element of the pilot valve; θ_y - the angle of oil flowing at the pilot valve, ρ – the oil density; $p_{3,2} = p_3 - p_2$ - the pressure drop in the pilot valve.

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

$$c_y \cdot (h_y + x_y) = p_{3,2} \cdot A_2 + p_{4,3} \cdot A_3 - r_y \cdot x_y \cdot p_{3,2}$$

or

$$x_y = \frac{p_{3,2} \cdot A_2 + p_{4,3} \cdot A_3 - c_y \cdot h_y}{c_y + r_y \cdot p_{3,2}} \qquad (2)$$

where: A_3- the area of the compensating piston; $p_{4,3} = p_4 - p_3 = R_{2l} \cdot q_y + R_{2m} \cdot q_y^2$ –the pressure drop through the orifice R_2; A_2-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; $r_y = 2 \cdot \mu_y \cdot \pi \cdot d_2 \cdot sin\theta_y \cdot cos\theta_y$-the hydrodynamic force coefficient of the pilot valve, if we express the velocity of flow through the pilot valve as $v_y = \sqrt{(2/\rho) \cdot p_{3,2}}$ and the flow $q_y = \mu_y \cdot v_y \cdot A_y = \mu_y \cdot v_y \cdot d_2 \cdot \pi \cdot x_y \cdot sin\theta_y$.

If we solve the equations *(1)* and *(2)*, the static characteristic of the pilot valve will be obtained:

$$q_y = \mu_y \cdot d_2 \cdot \pi \cdot \sin\theta_y \cdot \sqrt{\frac{2}{\rho} \cdot p_{3,2}} \cdot \frac{p_{3,2} \cdot A_2 + p_{4,3} \cdot A_3 - c_y \cdot h_y}{c_y + r_y \cdot p_{3,2}} \qquad (3)$$

➢ *Pressure drop at the fixed orifices*

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

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

where : $p_{1,3} = p_1 - p_3$ – the pressure drop at the pilot chain, $R_l = R_{1l} + R_{2l} = \frac{128 \cdot v \cdot \rho \cdot l_1}{\pi \cdot d_{dr1}^4} + \frac{128 \cdot v \cdot \rho \cdot l_2}{\pi \cdot d_{dr2}^4}$ – the linear hydraulic resistance in the orifices R_1 and R_2; $R_m = R_{1m} + R_{2m} = \xi_1 \cdot \frac{\rho}{2 \cdot A_{dr1}^2} + \xi_2 \cdot \frac{\rho}{2 \cdot A_{dr2}^2}$ – the local quadratic resistance in the orifices R_1 and R_2; $A_{dr} = \frac{d_{dr}^2 \cdot \pi}{4}$-the area of the orifice R_1 or R_2; d_{dr1} and d_{dr2} -the diametar of the orifice R_1 and R_2 ; l_1 and l_2 - the length of the orifice R_1 and R_2; v –the oil viscosity.

➢ *Pressure drop at the main valve*

$$p_{1,2} = p_{1,4} + p_{4,3} + p_{3,2} \tag{5}$$

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*

$$p_{1,3} \cdot A_o - p_{1,2} \cdot \Delta A = c_o \cdot (h_o + x_o) + r_o \cdot x_o \cdot p_{1,2}$$

or

$$x_o = \frac{p_{1,3} \cdot A_o - p_{1,2} \cdot \Delta A - c_o \cdot h_o}{c_o + r_o \cdot p_{1,2}} \tag{6}$$

where : A_o –the area of the closing element of the main valve; ΔA –the unbalanced area at the closing element of the main valve; h_o – the previous deformation of the spring of the main valve; x_o – the displacement of the closing element of the main valve; $r_o = 2 \cdot \mu_o \cdot \pi \cdot D_1 \cdot sin\theta_o \cdot cos\theta_o$ - the hydrodynamic force coefficient of the main valve, if we express the velocity of flow through the main valve as $v_o = \sqrt{(2/\rho) \cdot p_{1,2}}$ and the flow $q_y = \mu_o \cdot v_o \cdot D_1 \cdot \pi \cdot x_o \cdot sin\theta_o$; μ_o- the flow coefficient of the main valve; D_1 –the diameter of the seat of the main valve; θ_o –the angle of oil flowing at the pilot valve.

➢ *Flow across the main valve*

$$q_3 = \mu_o \cdot D_1 \cdot \pi \cdot x_o \cdot \sin\theta_o \cdot \sqrt{\frac{2}{\rho} \cdot p_{1,2}} \tag{7}$$

where: q_3 –the flow across the main valve.

➢ *Flow through pilot chain*

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

If we solve the system of equations *(1)* to *(8)*, we obtain the static characteristic of the pilot operated pressure relief valve with a compensating piston. For a pressure relief valve with the next parameters: $d_2 = 4\,[mm], d_3 = 5.5\,[mm], \quad D_1 = 17\,[mm], \quad D_o = 17.3\,[mm], \quad c_y = 140\,[N/mm], \quad c_o = 15.5\,[N/mm], \; d_{dr1} = 0.8\,[mm], \; d_{dr2} = 0.8\,[mm], \; d_{dr3} = 0.6\,[mm]$, the static characteristic of the valve is shown in *fig.2*. The diameter of the compensating piston corresponds to the *Denison's* pilot operated pressure relief valve. As a comparison, at the same figure the static characteristic of the pilot operated pressure relief valve without compensating piston is shown. For setting pressure of opening of the pilot valve $p_{1oy} = 150\,[bar]$, it is obtain $p_{1oo} = 155.5\,[bar]$-the pressure of opening of the main valve, $x_{yoo} = 0.0318\,[mm]$-the displacement of the pilot valve at the moment of opening of the main valve, $q_{yoo} = 0.73\,[l/min]$-the pilot flow at the moment of opening of the main valve, $p_{3oo} = 147.6\,[bar]$-the pressure in front of the pilot valve, $p_{4oo} = 151.7\,[bar]$-the pressure in front of the compensating piston.
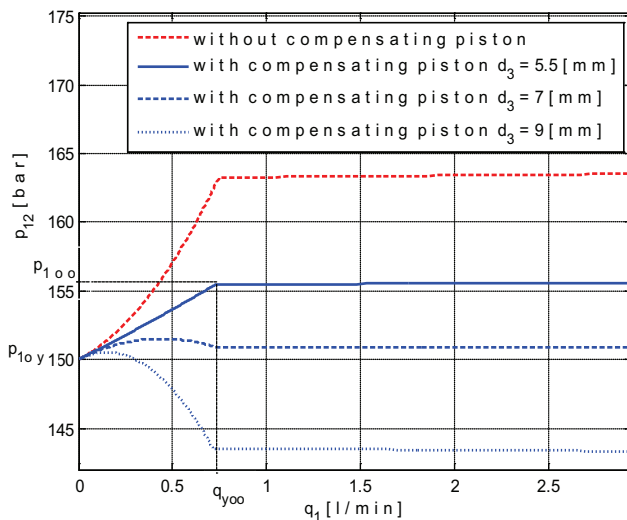

Fig.2. Static characteristics of
the pilot operated pressure relief valve

In *fig.3* it is shown the displacement of the pilot valve x_y and the main valve x_o depending on the flow q_1 for pressure relief valve with $d_3 = 5.5 \, [mm]$. This diagram shows that first the pilot valve opens until it reaches x_{yoo} and q_{yoo} and then the main valve opens.
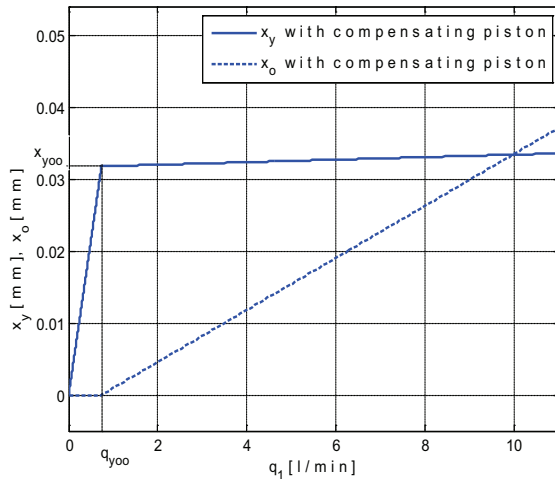

Fig.3. The displacement of the pilot and the main valve

From the above diagram it can be noticed that the pressure of opening of the main valve for pilot operated pressure relief valve without compensating piston is higher than the pressure of opening of the main valve for pressure relief valve with compensating piston. The static characteristic of the pilot operated pressure relief valves show relatively high difference between the pressure of opening of the pilot valve and the pressure of opening of the main valve i.e. there is high relative error at the beginning of the opening of the valve. As a consequence of this pressure difference there is a lot of waste of energy which is converted into heat which increases working temperature of the hydraulic oil and decrease the efficiency coefficient of the hydraulic system where the relief valve is built-in.

From the system of equations (1-8) it can be expressed the pressure of opening of the main valve for pilot operated pressure relief valve with compensating piston [2]:

$$p_{1oo} = \frac{1}{1-\varphi}\left(p_{1oy} + \frac{c_o \cdot h_o}{A_o} + \frac{x_y \cdot (c_y + r_y \cdot p_{3o})}{A_2} - \frac{A_3}{A_2} \cdot \left(R_{2l} \cdot q_y + R_{2m} \cdot q_y^2\right)\right) \qquad (9)$$

where: $\varphi = \Delta A / A_o$-the unbalanced area of the closing element of the main valve, $p_{1oy} = c_y \cdot h_y / A_2$-the pressure of opening of the pilot valve.

From Equ. (9) it is notable that a compensating piston with higher diameter d_3 i.e. higher area A_3, will reduce pressure difference $p_{1o} - p_{1y}$. For the compensating piston diameter $d_3 = 7 \, [mm]$, the static characteristic of the valve is shown on *fig.2*. But, it can be careful when increasing the compensating piston diameter d_3 because it can cause unstable work of the pressure relief valve. So, when designing pilot operated pressure relief valves it is necessary simultaneously to investigate the static and the dynamic characteristics. It is out of scope of this investigation and it will be included in the next work.

From the diagram in *fig.2* it can be seen that higher diameter of the compensating piston d_3, the pressure of opening of the main valve p_{1oo} can be reduce to near the pressure of opening of the pilot valve p_{yoo}. At the beginning the pressure p_1 rises slowly up to the maximum value of around $p_{1max} = 151.6 \, [bar]$, than falls down until the value of the opening of the main valve. For setting pressure of opening of the pilot valve $p_{1oy} = 150 \, [bar]$, it is obtained $p_{1oo} = 150.9 \, [bar]$-the pressure of opening of the main valve, $x_{yoo} = 0.0319 \, [mm]$-the displacement of the pilot valve at the moment of opening of the main valve, $q_{yoo} = 0.726 \, [l/min]$-the pilot flow at the moment of opening of the main valve, $p_{3oo} = 143.2 \, [bar]$-the pressure in front of the pilot valve, $p_{4oo} = 147.3 \, [bar]$-the pressure in front of the compensating piston.
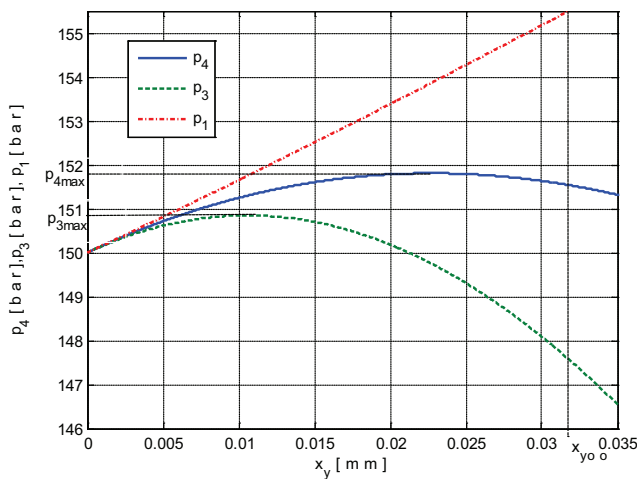
This improving of the static characteristic will increase the efficiency coefficient of the hydraulic system where the relief valve is built-in, it will provide better working conditions for the others components in the system, it will increase the accuracy of the system, and it will save a lot

of energy. At the diagram in the *fig. 2* it is shown, also, the static characteristic of the valve with compensating piston's diameter of $d_3 = 9\,[mm]$. It can be seen that the pressure of opening of the main valve is lower than the pressure of opening of the pilot valve (around $p_{1oo} = 144\,[bar]$). This feature of the pilot operated pressure relief valve has been used in the counterbalance pressure valves [2]. With appropriate ratio of the diameters d_3, D_o and D_1, the pressure of opening of the main valve can be much lower from the pressure of opening of the pilot valve.

It can be noticed that the pressure in front of the pilot valve p_{3oo} and the pressure in front of the compensating piston p_{4oo} at the moment of opening of the main valve are lower than the pressure of opening of the main valve p_{1oo}. It is important to analyze the values of the pressure p_3, the pressure p_4 and the pressure p_1 depending on the displacement of the pilot valve x_y.

In *fig.4* it is shown pressure p_1, p_3 and p_4 depending on x_y for the pressure valve with $d_3 = 5.5\,[mm]$.

As mentioned above, the main valve opens when the displacement of the pilot valve is $x_{yo} = 0.0318\,[mm]$. The maximum value of the pressure p_3 has been reached at around $x_y = 0.01\,[mm]$ and then begins to fall. But, the main valve is still closed. The pressure p_4 is still rising and influencing the pilot valve to increase displacement x_y, although p_3 is falling down. The maximum value of the pressure p_4 has been reached at around $x_y = 0.023\,[mm]$, and then begins to fall, but the main valve is still closed because there is not enough force difference acting on the main valve. The pressure p_1 is still rising until the force acting in a direction of opening of the main valve is higher than the force acting in a direction of closing of the main valve.

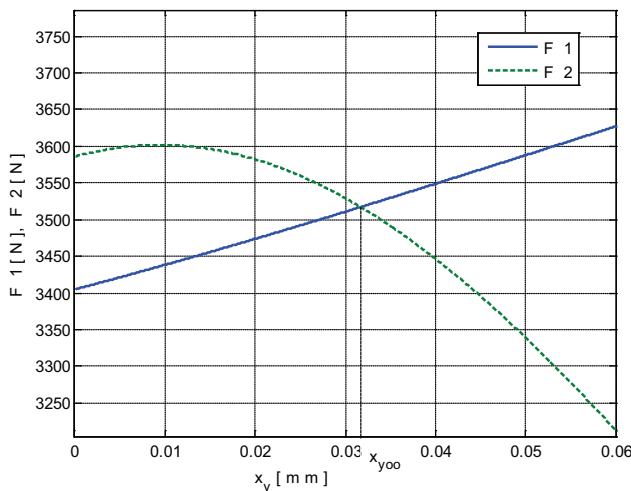Fig.4. The pressure p_1, p_3 and p_4 depending on x_y

Before to open the main valve, the force acting in a direction of opening of the main valve *F1* is: $F1 = p_1 \cdot A_1$. The force acting in a direction of closing of the main valve is: $F1 = p_3 \cdot A_o + c_o \cdot h_o$. In *fig.5* it is shown the ratio of the force acting in direction of opening of the main valve *F1* and the force acting in direction of closing of the main valve *F2* depending on the displacement of the pilot valve x_y. It can be noticed that until around $x_{yoo} = 0.032\,[mm]$, the force *F2* is higher than the force *F1*. So, the main valve is closed. At x_{yoo} the force *F1* reaches the force *F2* and the main valve opens.

Fig.5. Ratio forces acting on the main valve

4. CONCLUSION

In this paper a mathematical model of the static characteristics for a pilot operated pressure relief valve with compensating piston has been developed. The Influence of the compensating piston to the static characteristic has been emphasized as a manner for reducing the difference between pressure of opening of the pilot valve and main valve. It has been proved that increasing of the compensating piston diameter reduces the pressure difference of opening of the pilot and the main valve.

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