14th International Research/Expert Conference
"Trends in the Development of Machinery and Associated Technology"
TMT 2010, Mediterranean Cruise, 11-18 September 2010

INFLUENCE OF THE HYDRAULIC RELIEF VALVE POPPET GEOMETRY ON VALVE PERFORMANCE

Zdeslav Jurić, Zlatan Kulenović
University of Split, Faculty of Maritime Studies
Zrinsko – Frankopanska 38, Split
Croatia

Darko Kulenović
University of Split, Faculty of Electrical Engineering, Mechanical Engineering and Naval Architecture
Ruđera Boškovića bb, Split
Croatia

ABSTRACT
This paper discusses the influence of the poppet geometry of the direct spring operated pressure relief valve on the dynamics of its motions. Using derived differential equations, a mathematical model in Matlab –Simulink has been obtained. The results obtained by a computer simulation describe dynamic properties of the poppet and its influence on the system dynamic with respect to the poppet geometry.
Keywords: direct spring operated pressure relief valve, poppet valve geometry, simulation model

1. INTRODUCTION

The pressure relief valve, as part of either a hydraulic or pneumatic system, is a necessary safety element that protects the system and its elements from excessive pressure [1, 2]. The occurrence of pressure above the upper limit can cause the failure or malfunction of the system and/or elements in the hydraulic system. In order to avoid this, the system is fitted with the pressure relief valve that maintains pressure in the system below the upper limit value.
The design (construction) of the pressure relief valve [2] and the fluid characteristics used in the hydraulic system cause oscillations [3] of the pressure relief valve poppet at its opening (these are also affected, to a lesser but not negligible extent, by the air quantity in the hydraulic system and piping material [4]). Due to these oscillations, the valve flow area is constantly changing. This causes a permanent flow change through the relief valve, and thus the system pressure oscillation [5]. Various relief valve poppet shapes can lead to various dynamic characteristics, responses and working stability, which can reduce vibration and noise in the system. Based on derived differential equations the Matlab-Simulink model has been obtained for examining the impact of poppet geometry on the system pressure and displacement, velocity and acceleration of the valve poppet.

2. MATHEMATICAL MODEL

The pressure relief valve is an important element of the hydraulic system. An arrangement of a pressure relief valve in a ship's steering gear is shown in Figure 1 [6]. Figure 2 shows the structural model of a direct spring operated pressure relief valve. When the system pressure in the hydraulic system p_s is below the upper limit, the spring keeps the poppet on its seat, the valve is closed. If the pressure in the system rises above the permissible value, pressure force that acts on the poppet overcomes the spring forces and the poppet is moved from its seating, the valve is open. Now, through the relief valve, fluid from the system is redirected into the tank, thus reducing the pressure in the system. After the initial pressure drop, the valve poppet starts to move toward its seating, thus reducing the valve flow area through the valve. At this moment, the system pressure is increasing. These flow and pressure oscillations in the system cause unwanted vibrations and noises, which affects its proper functioning and system elements. These unwanted appearances can be decreased by selecting a proper valve poppet geometry.

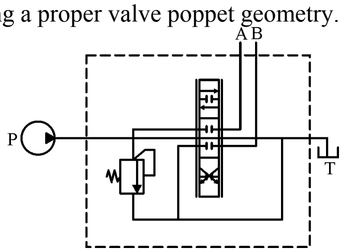

Figure 1. Part of ship's steering gear

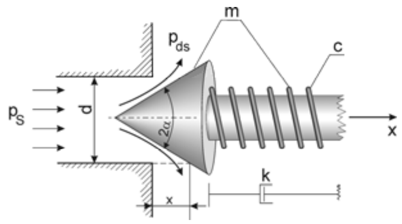

Figure 2. Pressure relief valve model

Considering that the total mass of the moving parts *m* is equal to the mass of the plunger plus one-half mass of the spring, the differential equation of the dynamic behaviour is derived as follows:

$$m \cdot \frac{d^2x}{dt^2} + F_v + F_s + F_{uf} \pm F_c - A_v \cdot \left(p_s - p_{ds}\right) = 0 \qquad ... (1)$$

where: x – poppet displacement, t – time, F_v – viscosity friction force, F_s – spring force, F_{us} – unsteady flow force, F_c – Coulomb friction force, A_v – poppet area normal to pressure, p_s – system pressure, p_{ds} – discharge port pressure (assumed to be equal to atmospheric pressure).
Viscosity friction force is given as in Eq. (1) [3,4].

$$F_v = k \cdot \frac{dx}{dt} \qquad ... (2)$$

where: k – viscous force coefficient.
The spring force which acts on the poppet is [4]:

$$F_s = k \cdot \left(x_0 + x\right) \qquad ... (3)$$

where: k – spring stiffness, x_0 – pre-compressed spring length, x - poppet displacement.
Due to the fluid flow through the valve poppet, unsteady flow force act on it by force [7]

$$F_{uf} = \pi \cdot \rho \cdot L \cdot C_d \cdot \sin\alpha \cdot \left[\left(d - x \cdot \sin 2\alpha\right) \cdot \frac{dx}{dt} + \frac{x \cdot \left(d - 0{,}5 \cdot x \cdot \sin 2\alpha\right)}{\sqrt{2 \cdot \rho \cdot \left(p_s - p_{ds}\right)}} \cdot \frac{d\left(p_s - p_{ds}\right)}{dt} \right] \qquad ... (4)$$

where: ρ - fluid density, L – damping length, C_d – discharge coefficient, α – poppet half angle, d – feed canal diameter.
Coulomb friction force acts in a direction opposite to that of the poppet motion, given in Eq. (5).

$$F_c = F \cdot \left[-sign\left(\frac{dx}{dt}\right) \right] \qquad ... (5)$$

where: F – intensity of Coulomb friction force. This force contributes to the hysteresis exhibited by the valve during its opening and reseating.
The system pressure ratio is given by the differential equation [7]:

$$\dot{p}_s = \frac{\beta_{ef}}{V_c} \cdot \left(Q_{in} - Q_{out} - Q_v - A_v \cdot \frac{dx}{dt} \right) \qquad ... (6)$$

where: β_{ef} – effective bulk modulus, V_c – pressure sensing volume, Q_{in} – inlet flow rate, Q_{out} – outlet flow rate, Q_v – flow rate through the relief valve.
The influence of the air content in fluid and the pipeline material on bulk modulus is expressed as the effective bulk modulus [4,8]

$$\frac{1}{\beta_{ef}} = \frac{1}{\beta_l} + \frac{1}{\beta_p} + \frac{1}{\beta_g} = \frac{1}{\beta_l} + \frac{1}{\beta_p} + \frac{a}{\kappa \cdot p_s} \qquad \text{... (7)}$$

where: β_l – bulk modulus of the fluid, β_p – bulk modulus of the pipeline, β_p – bulk modulus of the air, a – relative air content at atmospheric pressure, κ – specific heat ratio (adiabatic exponent) of air.
The fluid volume flow through the pressure relief valve is

$$Q_v = C_d \cdot A_{vf} \cdot \sqrt{2 \cdot \frac{(p_s - p_{ds})}{\rho}} \qquad \text{... (8)}$$

where: A_{vf} – valve flow area.
The flow area through the valve is proportional to the poppet displacement from the seating and the poppet half–angle $A_{vf} = \pi \cdot x \cdot \sin\alpha \cdot (d - x \cdot \sin\alpha \cdot \cos\alpha)$, where: d – pressure sensing port diameter.
According to the above set expressions, the differential equation that describes the motion of the valve poppet is derived.

3. SIMULATION RESULTS AND CONCLUSION

Based on the derived differential equation of the pressure relief valve poppet movement, the simulation model in Matlab–Simulink has been obtained.

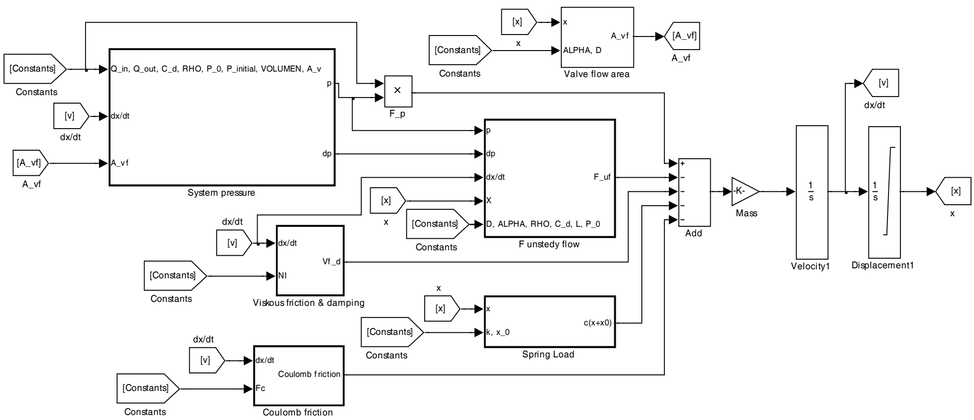

Figure 3. Simulation model of the direct spring operated pressure relief valve

The influence of the poppet half-angle on the system dynamic characteristics has been examined by means of the shown mathematical model. Under the same operating conditions the performance of the pressure relief valve is changed, by altering the poppet half–angle. The simulation results are presented through displacement, velocity and acceleration of the poppet and through the system pressure, Fig. 4, 5, 6 and 7.
At the time t = 0,02 s, the driven element reaches its end position. Further pump operation increases the system pressure until it reaches the cracking pressure so that the pressure relief valve opens. During the pressure relief valve operation, the pressure oscillations occur. They are especially pronounced at the opening time, see Figures 4 and 5. The stabilisation of the initial pressure

oscillations in the system is fastest at the lower poppet half-angle ($\alpha = 30°$), while at the higher value ($\alpha = 90°$) the stabilisation requires more time.

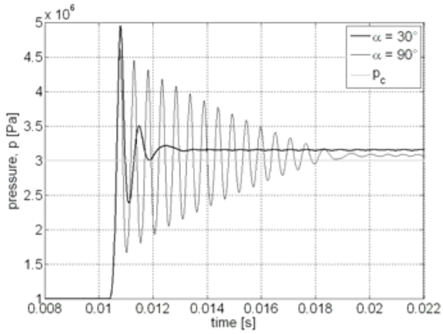

Figure 4. System pressure diagram

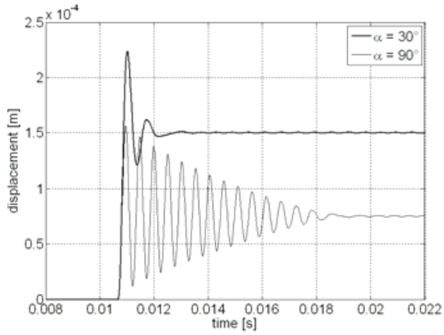

Figure 5. Poppet displacement diagram

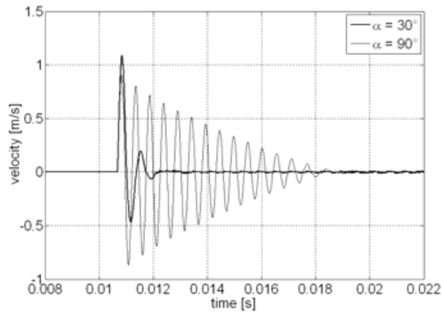

Figure 6. Poppet velocity diagram

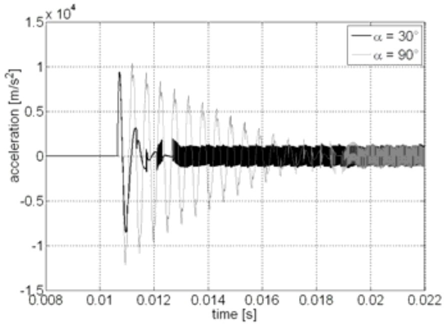

Figure 7. Poppet acceleration diagram

Another notable feature is the influence of the poppet geometry on the maximum system pressure. Figure 4 shows that at the poppet half-angle of 30° the system pressure is higher than for the higher tested half–angles. This indicates that it is necessary to properly select and set the direct spring operated relief valve with respect to its geometry and shape in order to protect the system from the moment of a sudden initial pressure increase to the system's stabilization.

By using the simulation model, it is possible to predict the pressure relief valve performance during its construction. In addition, by using the mathematical model it is possible to select the optimal setting points of the pressure relief valve (poppet shape, spring stiffness, spring pre-compressed length etc.) with respect to the system requirements.

ACKNOWLEDGEMENT

The results presented in the paper have been derived from the scientific research project „New Technologies in Diagnosis and Control of Marine Propulsion Systems“ supported by the Ministry of Science, Education and Sports of the Republic of Croatia.

4. REFERENCES

[1] Gallal Rabie, M.: Fluid Power Engineering, McGraw-Hill, 2009.
[2] Smith, P., Zappe, W. R.: Valve Selection Handbook, Elsevier, Oxford, 2004.
[3] White, F.: Fluid Mechanics, McGraw-Hill, 1987.
[4] Merrit, H. E.: Hydraulic control system, John Wiley and Sons, New York, 1967.
[5] Jurić Z., Kulenović Z.: A Simulation Model of Ship's Hydraulic valve, 8th International Research/Expert Conference TMT 2004, Neum, Bosnia and Herzegovina, 2004
[6] Taylor, D. A.: Introduction to marine engineering, Butterworth Heinemann, Boston, 1996.
[7] Fitch, E. C., Hong, I. T.: Hydraulic Component Design and Selection Manual, BarDyne, Stillwater, Oklahoma, 1998.
[8] Jelali, M., Kroll, A.: Hydraulic Servo-System: Modelling, Indentification and Control, Springer – Verlag London Ltd., 2003.