DYNAMIC BEHAVIOUR OF HYDRAULIC PRESSURE RELIEF VALVE

B. J. Patil¹, Dr. V. B. Sondur²

¹ Asst. Prof, Mechanical Engineering Department, Maratha Mandal Engineering College, Belgaum, Karnataka, India
² Founder Director, Sondur’s Academy, Belgaum, Karnataka, India

ABSTRACT

This paper discusses the influence of the radial clearance of the poppet of the direct spring operated pressure relief valve type DPRS06K315 on the dynamic behaviour of the valve. The mathematical model of the valve has been developed; these mathematical terms have been represented in Matlab/SIMULINK. The results obtained by a simulation describe dynamic behaviour of the valve and its influence on the system dynamic with respect to the poppet clearance.

Keywords: Radial Clearance, Simulation, Dynamic, Mathematical Model, SIMULINK, Valve Type DPRS06K315.

I. INTRODUCTION

The pressure relief valves are used to safeguard the hydraulic components from greater pressure[1]. This is one of the most important elements of a hydraulic system and is essentially required for safe operation of the system. Its main function is to limit the system pressure within a specified limit. It is normally a closed type and it opens when the pressure exceeds a specified maximum value by diverting pump flow back to the reservoir. The simplest type valve consists of a poppet held in a seat against the spring force as shown in fig 1 [2]. The oil enters from the opposite side of the poppet. As system pressure exceeds the set value of pressure, the poppet lifts and the oil is escaped through the orifice to the storage tank directly. It decreases the system pressure and as the pressure reduces to the set limit again the valve closes.
Relief and safety valves are fundamental equipments for oil and gas pipelines and load/unload terminals\[3\]. The installation integrity and workers safety depend on the appropriate design and performance of these equipments. In spite of the importance of relief valves, there is lack of information about the dynamic behaviour of these equipments.\[1\] Thus, users are forced to work using valve characteristics supplied only by manufactures. Further, the information supplied by manufactures is generally restricted to situations of maximum pressure relief flow. The full dynamic behaviour of the relief valves during their opening stage, which is fundamental for analysis of transients during their actuation, is usually not available.\[3\]

In spite of the importance of relief valves, only a few works about its dynamic behaviour has been published. Catalani (1984) performed a dynamic stability analysis of a relief valve and identified the effects of its components on its stability\[5\]. The undesired phenomenon named chatter (abrupt oscillations of the disc) was studied by MacLeod (1985) who modeled, using differential equations, the dynamic of a relief valve and identified the conditions to avoid it\[7\]. In 1991 Shing made a study about the dynamic and static characteristics of a two stage pilot relief valve and determined the governing parameters of the valve response which could be improved.

The dynamic of a direct operated relief valve with directional damping was studied by Dasgupta et al (2001) using the bondgraph technique. Maiti et al (2002) studied the dynamic characteristics of a two-stage pressure relief valve with proportional solenoid control of its pilot stage\[6,13\]. According to their results, the overall dynamic behaviour is dominated by the solenoid characteristic relating force to applied voltage. Boccardi et al (2004) analyzed experimentally the water/vapour two phase flows through a relief valve [3]. A new correlation for the discharge coefficient was developed, by comparing the experimental data with the solution of the flow based on a homogeneous model. The objective of this work is to simulate the dynamic behaviour of a direct acting spring loaded pressure relief valve (PRV) during its actuation. The identification of its governing parameters will allow the extension of the analysis to more general and real cases.\[3\]

II. MATHEMATICAL MODEL

Considering the fig.2 the total mass of the moving parts \(m\) is equal to the mass of the plunger plus one third mass of the spring, the differential equation of the dynamic behavior is derived as follows
The dynamic behaviour of the valve is described by the following set of mathematical relations. The effect of the transmission lines is neglected.

**The Poppet Valve Throttling Area.**

The following mathematical expression for the poppet valve area $A_p$

$$A_p = \pi (R^2 - (R - X \tan(\alpha))^2)$$

Where $R$ - radius of the poppet, $X$ – poppet displacement

**Equation of Motion of the Poppet**

$$M \frac{d^2x}{dt^2} + F_v + F_s - F_{SR} - A_p \cdot P_d - Qs^2 \rho / A = 0$$

where: $x$ – poppet displacement, $t$ – time, $F_v$ – viscosity friction force, $F_s$ – spring force, $F_{SR}$ – Seat reaction force, $A_p$ – poppet area normal to pressure, $P_d$ – system pressure,

**$F_v$. Viscosity friction force is given as**

$$F_v = C \frac{dx}{dt}$$

where: $C$ – viscous force coefficient.

**$F_s$. The spring force which acts on the poppet**

$$F_s = K^* (X_0 + X)$$

Where $K$ – spring stiffness, $X_0$- pre-compressed spring length, $X$ – poppet displacement.

**Seat Reaction Force**

The poppet displacement in the closure direction is limited mechanically. When reaching its seat, a seat reaction force takes place due to the action of the spring stiffness.
\[
F_{SR} = \begin{cases} 
0 & X > 0 \\
K(X_0 + X) & X < 0 
\end{cases}
\]  \hspace{1cm} (5)

**Flow Rate Through the Clearance of the Damping spool**

\[
Q_d = \frac{\pi D d^3 c^3 (P - P_d)}{(12 \mu u L)}
\]  \hspace{1cm} (6)

**Flow Rate Through the Poppet Valve**

\[
Q_s = C_d A_x \sqrt{\frac{2(P - P_0)}{\rho}}
\]  \hspace{1cm} (7)

Where \(A_x\) – poppet valve area

**Continuity Equation Applied to the Damping spool Chamber**

\[
Q_d - A_d \frac{dX}{dt} = \frac{V_0 + A_d X}{B} \frac{dP_d}{dt}
\]  \hspace{1cm} (8)

\[
\frac{dP}{dt} = \frac{B}{v_p} (Q_p - Q_s - Q_d - Q_{leak})
\]  \hspace{1cm} (9)

**III. DAMPING COEFFICIENT**

To find the damping coefficient of the system, we should consider the poppet as piston and the surrounding area of the poppet as a cylinder which is called as dash pot as shown in fig 3. The poppet is moving to and fro in a cylinder full of viscous fluid as shown in fig 3. We should consider the damping resistance due to the pressure difference on the two sides of the piston. The pressure difference is caused by the restriction to the fluid flow due to the piston motion.

![Fig. 3: Dash pot](image-url)
It can be shown that if clearance between the piston and cylinder is small, the first two components of the damping are negligible and the total damping is wholly due to the third component, and is given by

$$C = \frac{12 \mu}{\pi} \frac{A_p^2 \cdot L}{D_m \cdot c^3}$$

Where $C = $ Viscous damping coefficient N.s/m.
$\mu = $ Coefficient of viscosity of the fluid Pa.s
$A_p = $ Area of the piston m$^2$
$L = $ Length of the Piston m.
$D_m = $ Mean diameter of the piston and the cylinder
$c = $ clearance between the piston and the cylinder.

IV. SIMULINK MODEL

Based on the derived differential equation of the pressure relief valve poppet movement, fluid flow rate, pressure at inlet of the valve and the damping chamber the simulation model in MATLAB/SIMULINK has been obtained as shown in fig.4. A scope block is connected to monitor the time response of the pressure relief valve. The connections to the various blocks in the model have been made by considering the Equations 1, 2, 3, 4, 5, 6, 7, 8, 9, 10 which are obtained in chapter 4 mathematical modelling.

![Fig. 4: Simulink Model](image)

V. SIMULATION RESULTS

The transient response of the valve is calculated for different values of the poppet radial clearance. The transient response of valve input is calculated and plotted as shown in fig 5-9. The simulation results show that the radial clearance has a significant effect on the valve response. For smaller radial clearance, flow rate into the damping spool chamber is throttled and the pressure building in this chamber is delayed. The poppet takes a longer time to open which results in greater pressure overshoot. For larger radial clearance, the damping effect weakens and the poppet takes a short time to open which results in lesser pressure overshoot.
Fig. 5: Simulation results of response of the pressure relief valve pressure for damping poppet radial clearance of 10e-6 m

Fig. 6: Simulation results of response of the pressure relief valve pressure for damping poppet radial clearance of 15e-6 m
Fig.7: Simulation results of response of the pressure relief valve pressure for damping poppet radial clearance of 20e-6 m

Fig.8: Simulation results of response of the pressure relief valve pressure for damping poppet radial clearance of 25e-6 m
VI. CONCLUSION

The effect of the poppet clearance geometry on the system dynamic characteristics has been examined by means of the shown mathematical model. Under the same operating conditions, the behaviour of the pressure relief valve is varied, by chasing the poppet clearance. The simulation results are presented through system Pressure, Fig. 5-9. An optimum value of the radial clearance is estimated from the graph as shown in Fig. 9. This figure shows that the settling time is within 0.02 seconds and the maximum percentage of overshoot is considerably reduced.

By using the simulation model, it is possible to predict the pressure relief valve behaviour during its working. In addition, by using the mathematical model it is possible to select the optimal setting points of the pressure relief valve (poppet radial clearance with respect to the system requirements

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Fig.9: Simulation results of response of the pressure relief valve pressure for damping poppet radial clearance of 30e-6 m
VIII. REFERENCE


