Dynamic Analysis of Hydraulic Pressure of Relief valve

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Abstract- Pressure relief valve is one of the most important devices used on the security of pipelines, since it is responsible to guarantee the integrity of the installations. Generally, the response and behaviour of a relief valve during its transient is unknown by users, who employ simplified and static analysis to design the pipeline, further, the information provided by manufactures is limited. In this work, a numerical dynamic model of a spring load pressure relief valve was developed using the principles of conservation of mass and momentum in combination with solid dynamics equation. The valve discharge coefficient was numerically determined. The dynamic characteristics of the valve were examined with regard to the pressure set point, disc lift and spring parameters, during the transient discharge flow. In this work we have analysed direct spring operated pressure relief valves DPR 06 and DPR 10 by using Matlab - SIMULINK.

Keywords: Pressure relief valve, dynamic characteristics, and transient discharge flow.

I. INTRODUCTION

Relief and safety valves are fundamental equipments for oil and gas pipelines and load/unload terminals. 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. 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. 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. 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 flow through a relief valve. 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]

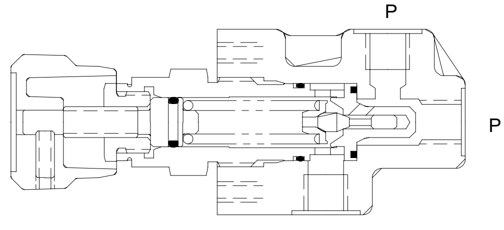


Figure 1. DPR 06 [2]

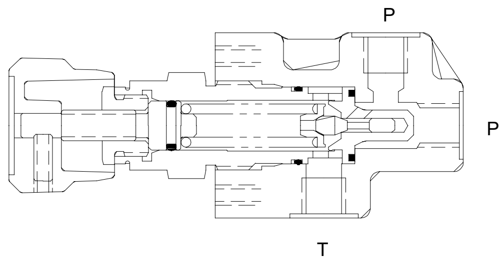


Figure 2 . DPR 10 [2]

II. MATHEMATICAL MODEL

Although the dynamic behaviour of a PRV is strongly influenced by its geometric configuration and dimensions, a simplified geometry, as shown in Fig. 3, was considered to the development of a mathematical model. The simplified system is composed of a spring, a cap or disc and a input flow pipe (valve wall). For the flow analysis through the PRV, the fluid was considered as incompressible and isothermal. Due to the cylindrical shape of the geometry, the flow was considered axi -symmetric.

A. Dynamic characteristics

The PRV starts opening when the operation pressure Pa exceeds the set point pressure Psp. During the disc displacement, the Newton's second law can be applied to the system illustrated in Fig. 4, resulting in the spring-disc dynamic system equation, Eq. (1).

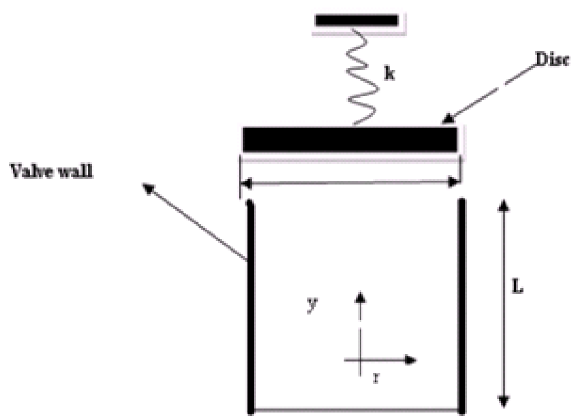


Figure 3. Simplified System.[3]

$$F_f - k(Y_D + Y_0) - c\frac{dY_D}{dt} - m_D g - P_0 A = m_D \frac{d^2 Y_D}{dt^2} \qquad (1)$$

where F_f is the force applied by fluid to disc, k is the spring constant, Y_D is the disc displacement, Yo is the spring initial deformation, c is the spring viscous damping coefficient, m_D is the disc mass, g is the gravity acceleration, Po is the external pressure (atmospheric pressure) and A is the cross section of the little pipe / disc area. Applying the principle of conservation of linear momentum in the y direction, to control volume inside the PRV illustrated in Fig. 3, neglecting the time y momentum variation inside the control volume, since it can be considered small in relation to the others quantities, results in[3]

$$\sum Fy = \frac{\partial}{\partial t}\int_{cv} u\rho \, d\forall + \frac{\partial}{\partial t}\int_{cs} u\rho \quad \acute{u}.d\overline{A} \rightarrow -\rho g A(L+Y_D) - F_f + P_a A = -u_e \rho u_e A \qquad (2)$$

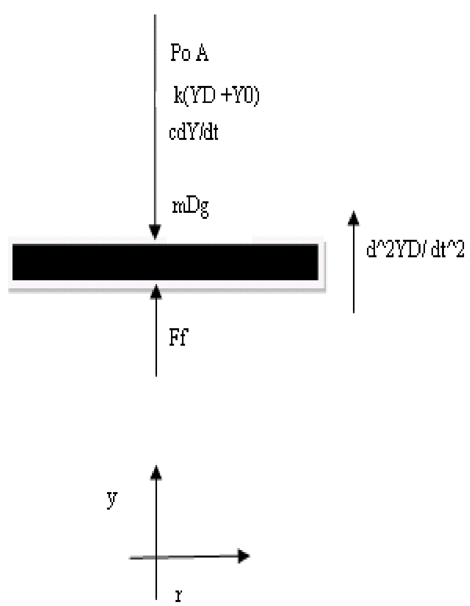
Where ρ is the fluid density, L is the length of the valve wall, Ff is the reaction applied by the disc to fluid, u_e is the average velocity coming into the control volume and Pa is the operation pressure or the PRV input pressure. Combining the spring-disc dynamic equation, Eq. (1), with the fluid momentum conservation equation, Eq. (2), the following expression is obtained

$$0 = m_D\frac{d^2 Y_D}{dt^2} + c\frac{dY_D}{dt} + (k+\rho gA)Y_D + kY_0 + \rho gAL + m_D g - (P_a - P_0)A - Q^2\frac{\rho}{A} \qquad (3)$$

Where Q = ρ ue A is the average flow rate coming into the control volume.

The initial spring deformation Yo can be determined as a function of the set point pressure Psp to open the relief valve, by applying Eq. (3) to the instant immediately before the valve opening, i.e., YD = 0 and Q=0, Pa=Psp.

$$Y_0 = \frac{1}{k}[(P_{sp} - P_0)A - \rho gAL - m_D g] \qquad (4)$$



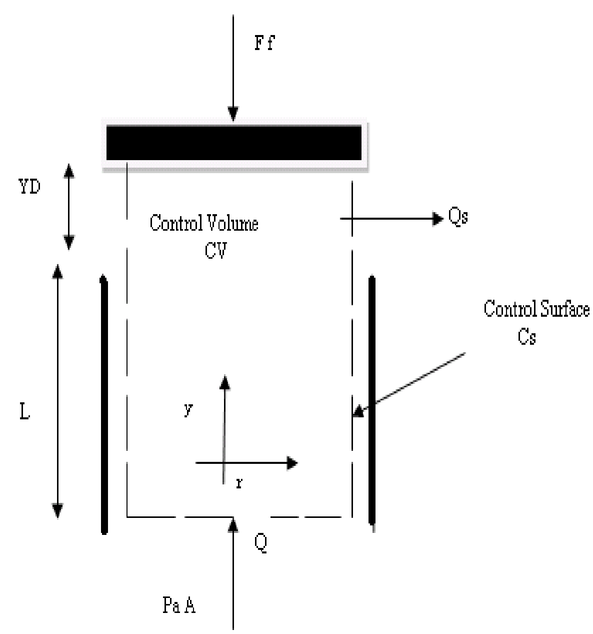


Figure 4, Free Body Diagram Figure 5. Control Volume inside of the PRV during its actuation

Equating (3) can be simplified with Eq. (4) as

$$0 = m_D\frac{d^2 Y_D}{dt^2} + c\frac{dY_D}{dt} + (k+\rho gA)Y_D - (P_a - P_{sp})A - Q^2\frac{\rho}{A} \qquad (5)$$

Applying the principle of mass conservation into the PRV control volume during its actuation, Fig. 5, the average flow rate that exits from the control volume Qs can be related with the inflow rate Q and the disc displacement Y_D as

$$0 = \frac{\partial}{\partial t}\int_{cv} u\rho \, d\forall + \frac{\partial}{\partial t}\int_{cs} u\rho \, \acute{u}.d\bar{A} \rightarrow 0 = A\frac{dY_D}{dt} - Q + Q_s \quad (6)$$

Further, the valve outflow rate Qs can be defined by the valve equation as[10]

$$Q_s = c_d A\sqrt{2\frac{(Pa - P0)}{\rho}} \quad (7)$$

Where Cd is the valve coefficient and A is a reference area, in this work it was considered as the disc area.
Finally, the equation that governs the dynamic behaviour of the PRV during its actuation can be obtained by combining Eqs. (7), (6) and (5) as

$$0 = m_D\frac{d^2Y_D}{dt^2} + (c - 2c_d A\sqrt{2\rho(P_a - P_0)})\frac{dY_D}{dt} + (k + \rho g A)Y_D - (P_a - P_{sp})A - (\frac{dY_D}{dt})^2 \rho A^{---} 2c_d^2 \; A(P_a - P_0) \quad (8)$$

B. Initial and Boundary Conditions

Initially it is considered that the PRV is closed (YD=0 and d YD /d t = 0). The flow through the valve begins when the operation pressure Pa exceed the set point pressure Psp. When the disc reaches its maximum position $Y_D$max, there is no more displacement, and the disc wall reaction RD for this situation can be obtained from Eq. (2) as

$$-\rho g A(L + L_{D\,Max}) - R_D + P_a A = - Q^2 \frac{\rho}{A} \quad (9)$$

and the inflow and outflow volumetric flow rate through the valve is the same, Q=Qs. The flow rate can be calculated using the Eq. (7). A similar situation occurs when the disc displacement reaches its minimum position, Y_{Dmin}=0, and the PRV is closed.

C. Dimensionless Parameters

The mathematical model, described by Eq. (8), can be normalized considering it as an oscillation-damping system, with $Y_{D*} = Y_D/D$, t* = t ν/D2 and P* = P/(ρν2/A) where ν is the cinematic viscosity as

$$0 = \frac{dt^{2}\,*}{d^2Y_D}^{*} + 2\xi^*\omega_0^{\,*}\left(1 - \sqrt{2\pi}\, c_d \,\frac{1}{c}\,\sqrt{Pa \quad + P0}\right)\frac{dt\,*}{dY_D}^{*} + \omega_0^{\,*\,2}(1 + \psi^*)\,Y_D^{\,*} - (P_a^{\,*} - P_{sp}^{\,*})\frac{1}{m_D} - \frac{\pi}{4}\left(\frac{dt\,*}{dY_D}^{*}\right)^2\frac{1}{m_D} - 2c_d^2\frac{1}{m_D}(P_a^{\,*} - P_{sp}^{\,*}) \quad (10)$$

The resulting dimensionless parameters that govern the valve behaviour are listed in Table 1

III. NUMERICAL METHOD

Equation (10) was solved numerically using a fourth-order Runge-Kutta method.

Table 1 - Dimensionless Parameters

Parameters	
$\omega_0^* = \omega_0 \frac{D^2}{v}$	Dimensional natural frequency, $\omega_0 = \sqrt{\frac{k}{m_D}}$
$\xi^* = \frac{1}{2}\frac{c}{\sqrt{k m_D}}$	Damping ratio
$\Psi^* = \frac{\rho A g}{k}$	Relation between the fluid weight per length unit and the spring constant
$m_D^* = \frac{m_D}{\rho D^3}$	Dimensionless disc mass
$C^* = \frac{c}{\rho v D}$	Dimensionless viscous damping coefficient
$\mathbf{C_d}$	Valve coefficient
$Y_{D\ max}^*$	Dimensionless disc maximum displacement

Among the several parameters listed on Table 1, the valve coefficient Cd is the critical parameter to be specified. It depends on the flow distribution inside the valve. Usually, it is determined experimentally, based on steady state flow with different valve openings. At the present work, the discharge coefficient was determined numerically, considering a steady state regime for different valve openings, as it is done experimentally.

A. Discharge Coefficient

The valve coefficient, Cd, was determined from the flow field inside the simplified valve, illustrated in Fig. 3,

$$\frac{\partial \acute{u}_j}{\partial x_j} = 0 \qquad (11)$$

$$\frac{\partial}{\partial x_j}(\rho \acute{u}_i \acute{u}_j) = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}[(\mu+\mu_t)(\frac{\partial \acute{u}_i}{\partial x_j} + \frac{\partial \acute{u}_i}{\partial x_j})] \qquad (12)$$

where u j is the time average velocity, μ and μt are the absolute and turbulent viscosity, and P is the pressure. The turbulent viscosity was determined with the κ−ω SST model (Menter, 1994), which was developed to blends theeffectively robust and accurate formulation of the standard κ−ω model in the near-wall region with the free-stream independence of the κ−ε model in the far field. The blending is designed to be one in the near-wall region, whichactivates the standard κ−ω model, and zero away from the surface, which activates the transformed κ−ε model. Theturbulent eddy viscosity μt is defined as

$$\mu_t = \rho \frac{\kappa}{\omega \epsilon} \qquad (13)$$

Where ω is the specific dissipation and ξ is the blending term. There is also a cross-diffusion term Dω included in the ω equation. The model requires the solution of two conservation equations, one is the standard κ equation, and the other is specific dissipation ω equation. These equations are given as

$$\frac{\partial}{\partial x_j}(\rho \acute{u}_i k) = \frac{\partial}{\partial x_j}[\Gamma_k \frac{\partial k}{\partial x_j}] + G_k - Y_k + S_k \ ; \quad \frac{\partial}{\partial x_j}(\rho \acute{u}_j \omega) = \frac{\partial}{\partial x_j}[\Gamma\omega \frac{\partial \omega}{\partial x_j}] + G_\omega - Y_\omega + S_\omega \qquad (14)$$

Where Gκ represents the production of turbulent kinetic energy due to mean gradients, while Gω is the production of ω. Γκ = μ+μt/σκ and Γω = μ+μt/σω are the effective diffusivity of κ and ω, where σκ and σω are the turbulent Prandtl numbers for of κ and ω, respectively. Yκ and Yω are the destruction of κ and ω, due to turbulence.

The operating pressure Pa = 50 bar was set at the inlet and the discharge pressure Po was set as 1 atm. Mineral oil was selected as the working fluid (ρ = 780 kg/m3 and μ= 10-3 Pa-s). From the converged flow field, the valve coefficient Cd was calculated using Eq.(7). Figure 6 presents the valve coefficient Cd as a function of the different opening, normalized by the maximum aperture Y_{Dmax} ($Y'_D=Y_D/Y_{Dmax}$). At the same figure, a third order polynomial adjusted to fit the data was plotted. This polynomial was included in Eq. (10) to determine the dynamic of the valve aperture

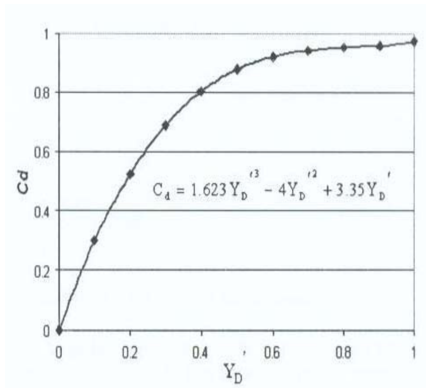


Figure 6. Valve Coefficient calculated numerically[3, 7]

IV. RESULTS

To validate the developed computational code, several test cases were performed. The pressure set point and operational pressure were defined as Psp = 50 bar and Pa = 55 bar. The spring dimensional parameters were specified as: k = 9.96N /mm, k=15.38 N/mm for DPR06 and DPR10 respectively and C= 29N-S/m, m_D = 43.186 g and C= 53.4N-S/m, m_D = 94.616 g for DPR06 and DPR10 respectively. For this situation the initial spring displacement Yo is 10.0 mm. Figure 7 and Figure 8 shows the PRV disc displacement along the time. On the beginning of the PRV opening the disc has reached its maximum displacement, followed by a strong oscillation, which was damped as time increased. The Figure 9 and Figure 10 show the pressure variation of the inlet and outlet flow rate through the PRV.

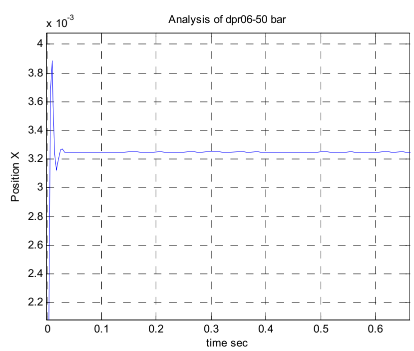


Figure 7. Simulink of DPR 06 50 bar

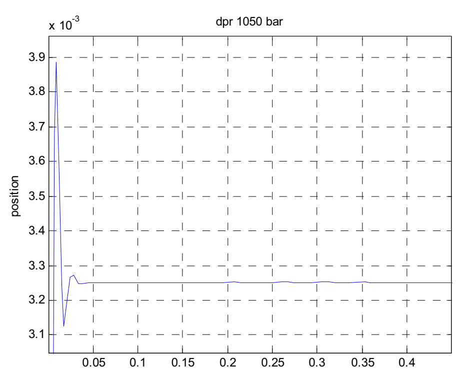


Figure 8. Simulink of DPR 10 50 bar

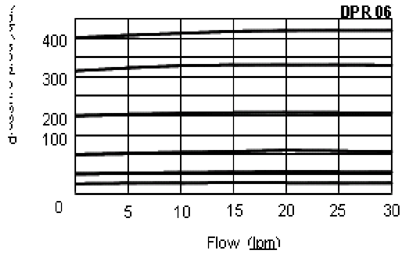


Figure 9. Flow rate of DPR 06

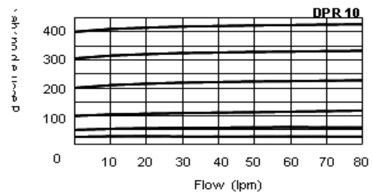


Figure 9. Flow rate of DPR 10

V. CONCLUSION

The present work has derived a mathematical model for a direct acting spring loaded pressure relief valve. The developed mathematical model predicts the disc behaviour and the input – output flow rate of the relief valve during its transient (dynamic characteristics) and equilibrium state. Although a simplified geometry was considered, the methodology can be applied to more complex geometries. A sensibility analysis was performed, by analyzing the influence of the several governing parameters in the valve disc displacement, and reasonable results were obtained.

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