ioMosaic®
Minimizing risk. Maximizing potential.®

Analysis of PRV Stability In Relief Systems Part VIII

Pilot Operated Pressure Relief Valves

An ioMosaic® Publication

G. A. Melhem, Ph.D., FAIChE

ioMosaic® Corporation

Analysis of PRV Stability In Relief Systems Part VIII
Pilot Operated Pressure Relief Valves

Process Safety and Risk Management Practices

authored by

G. A. Melhem, Ph.D., FAIChE

Printed October 18, 2024

Notice:

This document was prepared by ioMosaic® Corporation (**ioMosaic**) for Public Release. This document represents ioMosaic's best judgment in light of information available and researched prior to the time of publication.

Opinions in this document are based in part upon data and information available in the open literature, data developed or measured by ioMosaic, and/or information obtained from ioMosaic's advisors and affiliates. The reader is advised that ioMosaic has not independently verified all the data or the information contained therein. This document must be read in its entirety. The reader understands that no assurances can be made that all liabilities have been identified. This document does not constitute a legal opinion.

No person has been authorized by ioMosaic to provide any information or make any representation not contained in this document. Any use the reader makes of this document, or any reliance upon or decisions to be made based upon this document are the responsibility of the reader. ioMosaic does not accept any responsibility for damages, if any, suffered by the reader based upon this document.

Revision Log:

Revision 1: December 2023

...

Table of Contents

List of Figures

List of Tables

1 Introduction

Dynamic and static methods were previously developed for the modeling of spring loaded pressure relief valve stability in Parts 1 through 7 of this white paper series [2, 3, 4, 5, 6, 7, 8]. We extend those methods to pilot operated pressure relief valves in this 8th installment. Pilot operated pressure relief valves can be modeled dynamically using either a dual degree of freedom (DDOF) model or a lumped single degree of freedom (SDOF) model.

This paper provides guidance on how to select parameters for both DDOF and SDOF models for use with Process Safety Office® SuperChems Expert .

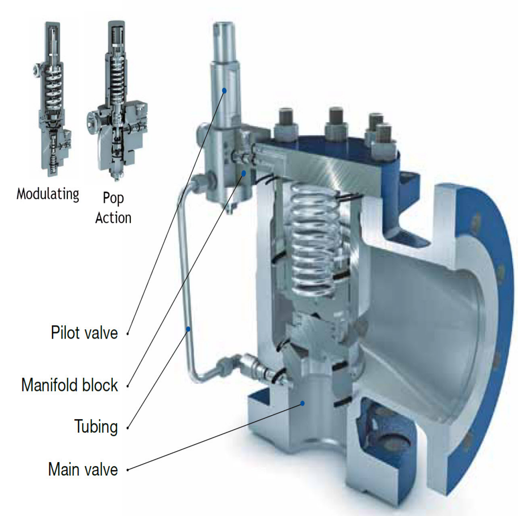
2 Pilot Operated Pressure Relief Valves

Pilot operated pressure relief valves (POPRV) consist of two major components, (a) the "pilot" valve, and (b) the "main" valve. This is shown in Figure 1.

Both the pilot valve and the main valve are spring loaded. The pilot valve, also called the "pilot", is the control unit of the POPRV. It controls the function and behavior of the main valve.

The pilot valve directs the main valve to open or close by allowing fluid to enter or exit the dome of the main valve. The main valve action follows the pilot valve action but can lag the pilot valve action by several milliseconds.

Figure 1: A typical pilot operated pressure relief valve



https://www.leser.com/en/products/pilot-operated-safety-valves/

3 How Pilot Operated Pressure Relief Valves Work

POPRVs work somewhat similarly to spring operated pressure relief valves but they rely on the system pressure to open and close the main valve.

During normal operation, the system fluid and pressure are routed to the main valve inlet and the main valve dome as illustrated in Figure 3(a). Since the main valve dome area, A_{piston} or A_2, is larger than the area of the main valve seat, A_{seat} or A_0, the closing force is greater than the opening force. This keeps the main valve tightly closed. The use of a piston makes the POPRV inherently balanced

Figure 2: Typical POPRV modulating behavior

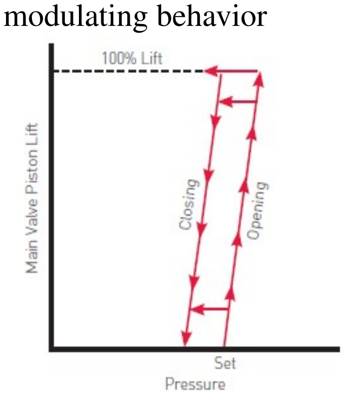
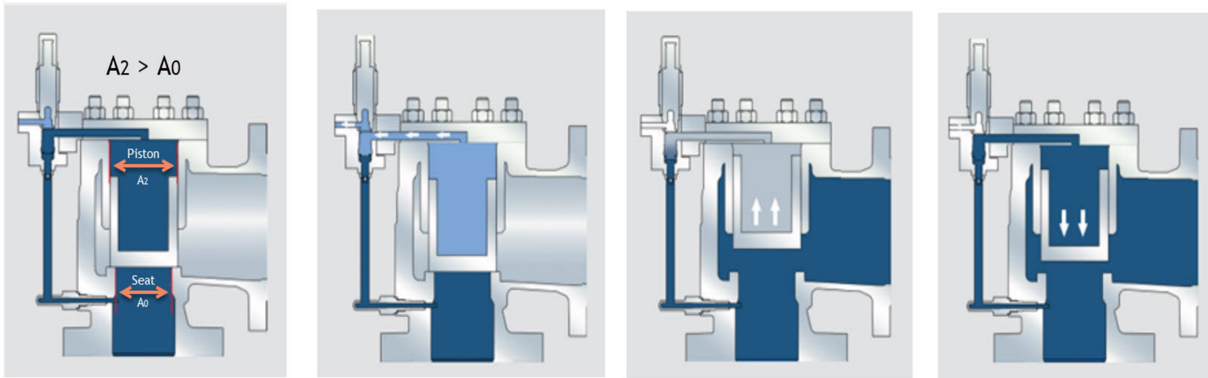


Figure 3: How pilot operated pressure relief valves work



(a) Normal Operation, Initially Closed State
(b) At Set Pressure, Actuating
(c) Opening, Main Valve Opening
(d) Closing, Refilling the Dome

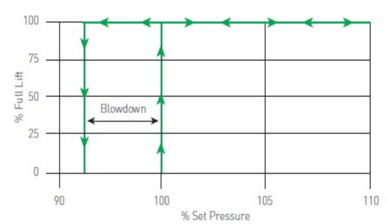
https://www.leser.com/en/products/pilot-operated-safety-valves/

against backpressure.

When the system pressure reaches the set pressure, the pilot valve actuates. The fluid is no longer routed to the dome as illustrated in Figure 3(b). This prevents any further increase in dome pressure. Because the dome is vented, the main valve closing force becomes less than the opening force causing the main valve to open. Note that pilots can also be non-flowing and do not flow while the main valve is opened and flowing. Non-flowing pilots will just flow the dome volume at each open-close cycle. This can be useful in cryogenic applications such as LNG.

As the main valve opens and depending on the pilot valve type, the opening can either be quick and complete (pop or snap action), or gradual and partial following system pressure (modulating action). Modulating POPRVs will flow the required system flow rate to keep the system pressure below a certain limit as shown in Figure 3(c). Because of their design, modulating POPRVs are less susceptible to chatter than spring loaded pressure relief valves.

Figure 4: Typical POPRV pop action behavior



If the system pressure drops to the closing pressure, the pilot valve actuates and again routes the fluid and fluid pressure to the dome. This is illustrated in Figure 3(d). The pressure in the dome builds up and the main valve recloses either rapidly and completely (pop action) or gradually and partially following the system pressure (modulating action).

4 Fluid Stiffness

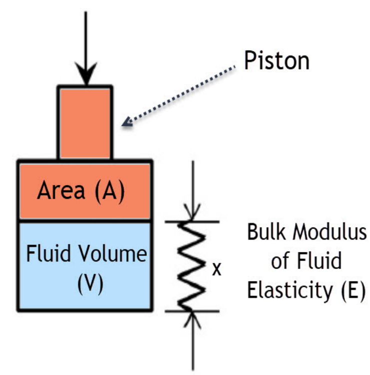
Because all fluids are compressible (including liquids), they can be treated as springs. The stiffness of a fluid is proportional to its change in volume due to a change in system pressure. A spring can store and deliver power similar to a capacitor in an electrical system. The stiffness of a spring is determined by its spring constant, K_{sf}. The larger the constant, the stiffer the spring. The fluid or hydraulic stiffness is related to its modulus of elasticity, E, or isothermal/adiabatic compressibility, κ.

A linear actuator containing a compressible fluid is shown in Figure 5. The actuator behaves like a spring because of the capacitance of the fluid. The stiffness of the actuator is given by:

$$K_{sx} = \frac{A^2 E}{V} \text{ where } E = \frac{1}{\kappa} \tag{1}$$

where A is the piston area, and V is the volume of the fluid.

Figure 5: Linear actuator containing a compressible fluid



The geometry of the cylinder in Figure 5 will determine the stiffness of the component. The stiffness will change as the cylinder expands and retracts.

Note that increasing the piston area will increase the stiffness of the system while increasing the fluid volume will decrease the stiffness of the system. Equation 1 shows that the stiffness increase is proportional to A^2. Increasing the area has a greater effect in increasing the responsiveness of the system than decreasing the volume or length.

A piston - volume (dome) system can be assumed to be a linear mass-spring system. This is a reasonable assumption even though the dome pressure will very during the piston stroke due to fluid compressibility:

$$F_x = K_{sx} x = PA \text{ or } K_{sx} = \frac{PA}{x} \tag{2}$$

where F_x is the fluid or hydraulic force. The frequency of the piston movement can be expressed as a function of the fluid spring constant and the mass of the piston:

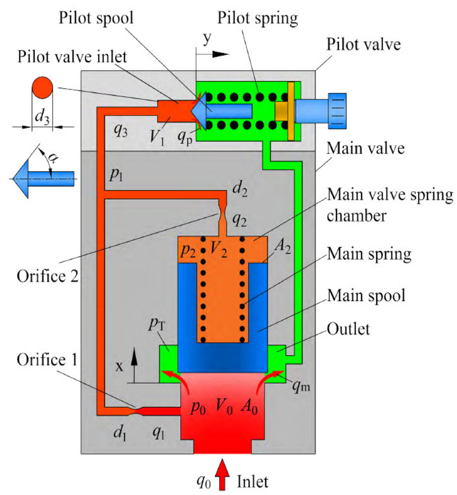
$$f_p = \frac{1}{2\pi}\sqrt{\frac{K_{sx}}{m}} = \frac{1}{2\pi}\sqrt{\frac{PA}{xm}} \tag{3}$$

where m is the mass of the piston and f_p is the frequency in Hz. f_p increases as x decreases (or a PORV main valve opens as pressure increases and its remaining lift decreases).

5 Modeling Pilot Pressure Relief Valve Dynamics

A typical two (double) degree of freedom system is illustrated in Figure 6. Two dynamic force balance equations are written, one for the pilot spring-mass system, and one for the main valve spring-fluid-mass system [1, 9, 10, 11, 12].

Figure 6: DDOF representation of a pilot operated pressure relief valve[1]



Pilot valve:

The force balance for the pilot spool can be written as follows:

$$P_1 A_3 = m_1 \frac{d^2 y}{dt^2} + C_1 \frac{dy}{dt} + K_y (y_0 + y) + K_{sy} y + m_1 g \quad (4)$$

$$A_3 = \frac{\pi d_3^2}{4} \quad (5)$$

where P_1 is the pressure at the pilot valve inlet, d_3 is the orifice diameter in pilot valve seat, A_3 is the flow area of d_3, m_1 is the mass of the pilot spool, y is the displacement of pilot spool, y_0 is the pre-compression displacement of the pilot spring, K_y is the pilot spring stiffness or spring constant, K_{sy} is the hydrodynamic force stiffness of the pilot valve port, g is the gravitational constant, and C_1 is the viscous damping coefficient of the pilot spool.

Main valve:

The force balance for the main valve can be written as follows:

$$P_0 A_0 - P_2 A_2 = m_0 \frac{d^2 x}{dt^2} + C_0 \frac{dx}{dt} + K_x (x_0 + x) + K_{sx} x + m_0 g + F_f \quad (6)$$

where P_0 is the pressure at the main valve inlet, P_2 is the pressure in the main valve spring chamber, A_0 is the effective area of the lower-end of the main spool, A_2 is the effective area of the upper-end of the main spool, m_0 is the mass of the main valve spool and is typically negligible, x is the displacement of main spool, x_0 is the pre-compression displacement of the main valve spring, K_x is the main valve spring stiffness or spring constant (typically a light or soft spring), K_{sx} is the hydrodynamic force stiffness of the main valve port, F_f is the static friction between the main spool and the seat which is typically negligible, and C_0 is the viscous damping coefficient of the main spool.

If P_0 increases just enough to open the main valve ($x \simeq 0$) and holds at that value, then:

$$P_0 A_0 > P_2 A_2 + K_x x_0 + m_0 g \quad (7)$$

$$P_0 > P_2 \frac{A_2}{A_0} + \frac{K_x x_0 + m_0 g}{A_0} \quad (8)$$

A volume balance is written for the main valve inlet volume where the volumetric flow rates in and out of the main valve inlet volume must balance:

$$\frac{V_0}{E} \frac{dP_0}{dt} = q_0 - q_m - q_1 - A_0 \frac{dx}{dt} \quad (9)$$

where V_0 is the volume of the main valve inlet, q_0 is the volumetric flow rate through the main valve, q_1 is the volumetric flow rate through orifice 1 which is positive if $P_0 \geq P_1$ and negative if $P_0 < P_1$, q_m is the volumetric flow rate through the main valve outlet, and E is the fluid bulk modulus of elasticity [1]. For liquid flow through the main valve, q_m and q_1 are given by:

$$q_0 = C_{d,0} A_0 \sqrt{\frac{2}{\rho}\left(P_{\text{src}} - P_0\right)} \tag{10}$$

$$q_m = C_{d,m} A_x \sqrt{\frac{2}{\rho}\left(P_0 - P_b\right)} \tag{11}$$

$$A_x = n\left[\frac{d_0^2}{4}\cos^{-1}\left(1 - \frac{2x}{d_0}\right) - \left(\frac{d_0}{2} - x\right)\sqrt{x\left(d_0 - x\right)}\right] \tag{12}$$

$$q_1 = C_{d,1}\frac{\pi d_1^2}{4}\sqrt{\frac{2}{\rho}\left(P_0 - P_1\right)} \tag{13}$$

where P_{src} is the vessel or protected equipment source pressure which can be time dependent, P_b is the backpressure at the main valve outlet, A_x is the curtain flow area of the main valve outlet which depends on main valve spring lift x, $C_{d,m}$ is the discharge coefficient of the main valve flow outlet, $C_{d,1}$ is the discharge coefficient of orifice 1, n is the number of drain holes on the main valve sleeve, d_0 is the diameter of drain hole on the main valve sleeve, d_1 is the diameter of orifice 1.

Additional volume balances are written for the main valve spring chamber and for the pilot valve inlet:

$$\frac{V_2}{E}\frac{dP_2}{dt} - A_2\frac{dx}{dt} = q_2 \tag{14}$$

$$\frac{V_1}{E}\frac{dP_1}{dt} = q_1 - q_P - q_2 \tag{15}$$

$$q_2 = C_{d,2}\frac{\pi d_2^2}{4}\sqrt{\frac{2}{\rho}\left(P_1 - P_2\right)} \tag{16}$$

$$q_p = C_{d,p}\pi d_3 y \sin\alpha\sqrt{\frac{2}{\rho}\left(P_1 - P_b\right)} \tag{17}$$

where V_1 is the volume of the pilot inlet, V_2 is the volume of the main valve spring chamber and q_2 is the volumetric flow rate through orifice 2, positive if $P_1 \geq P_2$ and negative if $P_1 < P_2$, q_p is the volumetric flow rate through the pilot valve outlet, d_2 is the diameter of orifice 2, and $C_{d,2}$ is the discharge coefficient of orifice 2, $C_{d,p}$ is the discharge coefficient of the pilot valve port, and α is the half cone angle of the pilot spool.

The equations described above for the pilot valve and main valve can be solved simultaneously to

[1] The isothermal value of E_T is equal to the inverse of the fluid isothermal compressibility κ_T, $E = 1/\kappa_T$

calculate x, y, u_y, u_x, $P0$, P_1, and P_2:

$$\frac{dy}{dt} = u_y \quad (18)$$

$$\frac{dx}{dt} = u_x \quad (19)$$

$$m_1 \frac{du_y}{dt} + C_1 u_y = P_1 A_3 - K_y (y_0 + y) - K_{sy} y - m_1 g \quad (20)$$

$$m_0 \frac{du_x}{dt} + C_0 u_x = P_0 A_0 - P_2 A_2 - K_x (x_0 + x) - K_{sx} x - m_0 g - F_f \quad (21)$$

$$\frac{V_0}{E}\frac{dP_0}{dt} = q_0 - q_m - q_1 - A_0 \frac{dx}{dt} \quad (22)$$

$$\frac{V_2}{E}\frac{dP_2}{dt} = q_2 + A_2 \frac{dx}{dt} \quad (23)$$

$$\frac{V_1}{E}\frac{dP_1}{dt} = q_1 - q_P - q_2 \quad (24)$$

$$(25)$$

The use of a pilot DDOF model requires detailed information about the pilot valve internals.

6 Modeling POPRV Dynamics Using a Lumped SDOF Model

DDOF model parameters are usually not immediately available and have to be obtained from the relief device manufacturer. Recent detailed dynamic studies using DDOF models demonstrate that the main valve action closely follows the pilot valve action (see Figure 7). If we assume $\frac{dy}{dt} = \frac{dx}{dt} = u_y = u_x$, $K_{sx} = K_{sy} = 0$, $P_1 \simeq P_0$, and $F_f = 0$, we can reduce the DDOF model to a lumped SDOF model:

$$\underbrace{(m_0 + m_1)}_{m_D} \frac{du_x}{dt} + (C_1 + C_0) u_x = \underbrace{P_0 A_0 + P_0 A_3}_{F_{Up}} \underbrace{- P_2 A_2 - (K_x + K_y) x - K_x x_0 - K_y y_0 - m_D g}_{F_{Dn}} = F_{NET} \quad (26)$$

Information about main valve lift as a function of overpressure and the opening time of the POPRV can be obtained from the relief device manufacturer. A lumped POPRV single degree of freedom model (see Figure 8) can be used based on the equations developed by Melhem [2, 3] for a spring loaded pressure relief valve.

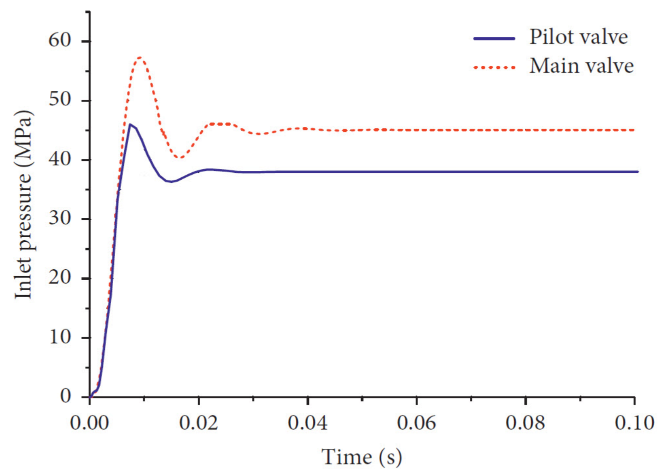
$K_x x_0$ and $K_y y_0$ can be related to fluid force:

$$K_x x_0 + m_0 g = P_{set} A_0 - P_2 A_2$$
$$K_y y_0 + m_1 g = (P_{set} - P_{atm}) A_3$$

P_{set} is the absolute set pressure of the POPRV and P_{atm} is the absolute ambient pressure. The lumped SDOF model can be further reduced to a form that is more equivalent to a spring loaded pressure relief device SDOF model:

$$\begin{aligned} m_D \frac{du_x}{dt} + C u_x &= (P_0 - P_{set}) A_0 \\ &\quad + (P_0 - P_{set} + P_{atm}) A_3 \\ &\quad - \tilde{K}_x x \\ &= F_{NET} \end{aligned} \tag{27}$$

Figure 7: DDOF solution for main valve and pilot valve pressures



where $\tilde{K}_x = K_x + K_y$, and $C = C_0 + C_1$. Using a single degree of freedom analysis to describe the motion of a spring loaded valve disc, we can write the following equations (see Melhem [2, 3]):

$$\frac{dx}{dt} = u_x \tag{28}$$

$$m_D \frac{du_x}{dt} + C u_x = F_{NET} \tag{29}$$

where C is the coefficient of viscous damping in $\mathrm{Ns/m}$, m_D is the spring-disk system mass in motion in kg, F_{NET} is the net force acting on the disk in N, and u_x is the speed of valve stem movement in $\mathrm{m/s}$.

The general form of F_{NET} for a pilot or spring loaded pressure relief valve is reproduced from Melhem [2]:

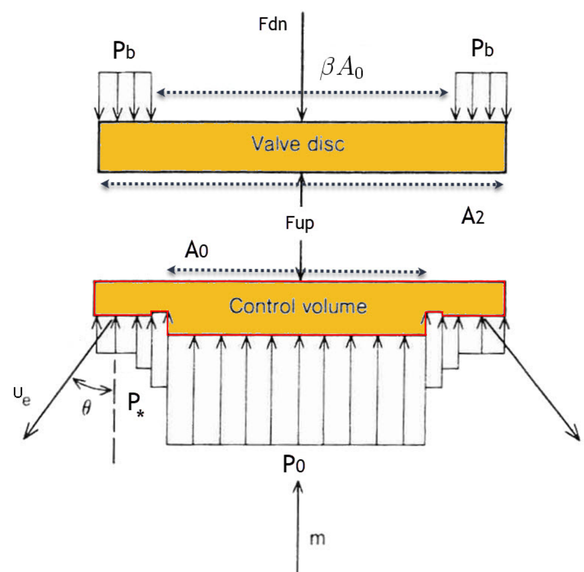
$$\begin{aligned} F_{NET} &= F_{Up} - F_{Dn} \\ &= (P_0 - P_{set}) A_0 + \underbrace{\eta P_* (A_2 - A_0)}_{\text{or } \eta(P_0 - P_{set} + P_{atm})} + \dot{m} u_e \cos\theta + \frac{\dot{m}^2}{\rho_0 A_0} \\ &\quad - \frac{1}{\psi_2} K_s x - P_{atm} A_0 (\beta - 1) - P_b (A_2 - \beta A_0) \\ 1 &\leq \beta \leq \frac{A_2}{A_0} \\ K_s &= \left(\frac{P_{\text{Full Flow}}}{P_{set}}\right) \left(\frac{A_2}{A_0}\right) \left(\frac{P_{set} A_0}{x_{max}}\right) \end{aligned}$$

where βA_0 represents the disk area protected from backpressure [2], $\dot{m}$ is the mass flow rate in $\mathrm{kg/s}$,

[2] Typically this is the area occupied by bellows in a spring loaded bellows PRV, $A_{\text{bel}} = \beta A_0$.

x_{max} is the maximum disk lift [3] in m, u_e is fluid exit velocity (typically sonic for gas flow) in m/s, θ is the fluid exit discharge angle with respect to the vertical in degrees, P_* is the pressure on the upstream side of the disk in Pa and η is a conversion efficiency factor used to calibrate the model. The value of P_* will depend on whether the flow is choked or not.

Figure 8: Forces acting on the valve disc and control volume



The value of βA_0 is used to isolate the disk area from backpressure. The value of β is selected such that the lumped SDOF model reproduces the specified blowdown pressure of the POPRV. The values of K_s and m_D are selected such that specified set pressure and opening time of the POPRV are matched. If $\beta = 1$, the valve disk is exposed to a backpressure force of $P_b\,(A_2 - A_0)$. If $\beta = \frac{A_2}{A_0}$, the valve disk is exposed to a backpressure force of $P_{atm}\,(A_2 - A_0)$.

Finally, the values of η and θ are regressed based on the known performance of the POPRV, i.e. fractional lift vs. overpressure which can be obtained from the manufacturer. Typical values for η and θ for a pilot operated pressure relief valve should be close to 0 and 90 degrees.

Using a lumped SDOF model for a pilot pressure relief device in SuperChems, the user can only specify the value of A_0. As a result, SuperChems allows the user to define a ψ_2 factor to derate the contribution $\tilde{K_x}x$ such that:

$$\frac{1}{\psi_2}K_s = \tilde{K_x}x \tag{30}$$

The value of ψ_2 will often be close to 1 but is larger than 1.

It is typical to relate the coefficient of viscous damping to the critical damping coefficient for a spring-mass system:

$$C = \zeta C_{cr} \tag{31}$$

$$C_{cr} = 2m_D\omega_n = \frac{2K_s}{\omega_n} = 2\sqrt{K_s m_D} \tag{32}$$

$$\omega_n = \sqrt{\frac{K_s}{m_D}} \tag{33}$$

where ζ is the viscous damping coefficient, dimensionless, typically set at 0.2, and ω_n is the undamped circular natural frequency in radians/s. Other variables used in the context of single

[3] Maximum disk lift x_{max} is greater than disk critical lift, x_c. Critical lift occurs when the curtain flow area equals the nozzle or seat flow area, $2\pi r x_c = A_0$, where $A_0 = \pi r^2$.

degree of freedom analysis include:

$$\tau_n = \frac{2\pi}{\omega_n} = 2\pi\sqrt{\frac{m_D}{K_s}} \quad (34)$$

$$f_n = \frac{1}{\tau_n} = \frac{\omega_n}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{K_s}{m_D}} \quad (35)$$

where τ_n is the undamped natural period in s, and f_n is the undamped natural frequency in Hz where one Hz equals 1 cycle/second.

Using the above equations, we can now write an overall description of how the valve disc will move:

$$\frac{dx}{dt} = u_d \quad (36)$$

$$\frac{du_d}{dt} = \frac{1}{m_D}\left[F_{NET} - \zeta C_{cr} u_d\right] \quad (37)$$

When the valve disk is on the seat or at the upper stop, a coefficient of restitution is used to reverse the spindle direction [13]:

$$\frac{dx^{'}}{dt} = -\beta\frac{dx}{dt} \quad (38)$$

where β is the coefficient of restitution (typically set at 0.01) and $x^{'}$ represents x after contact with the valve seat or upper stops.

7 Use of Pilot Operated Pressure Relief Valves

Pilot operated pressure relief valves are used in clean service (particle sizes of solids and suspensions less than 100 microns). Modulating pilot operated relief valves can provide a practical solution where conventional and bellows spring loaded devices are likely to chatter due to excessive inlet pressure loss and/or backpressure. They are also used in cryogenic service or in cold weather climates because of potential freezing of atmospheric moisture inside the bonnets of bellows spring loaded pressure relief valves. In addition, applications requiring set pressure independence of backpressure and applications with increased tightness requirements are good candidates for the use of pilot operated pressure relief valves.

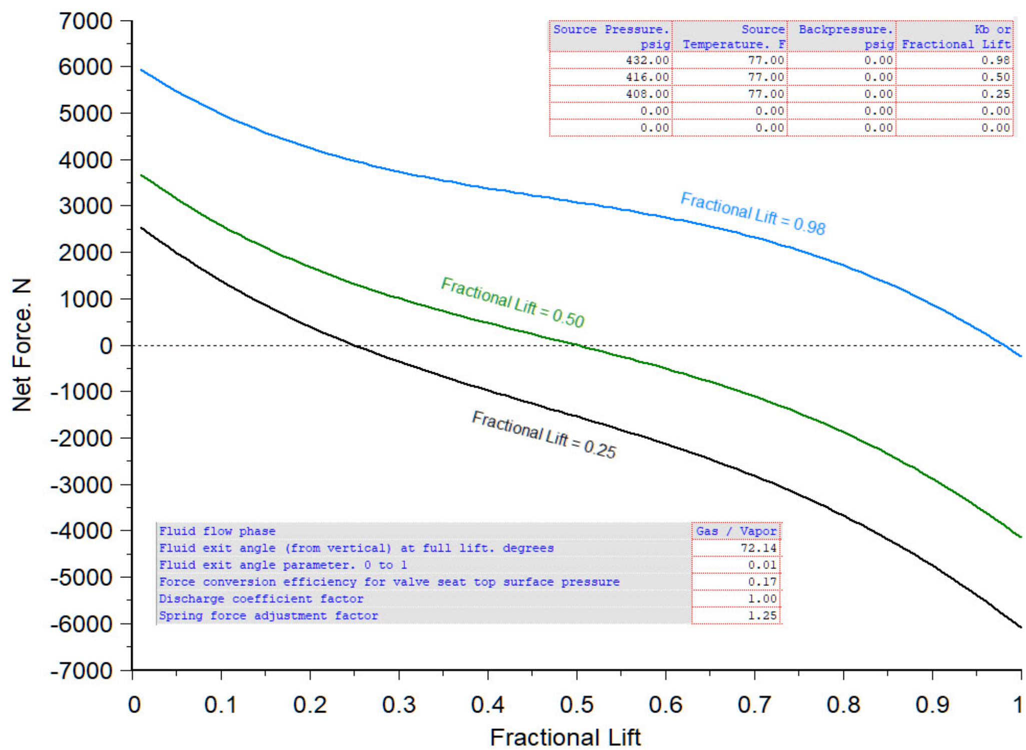
8 Specifying a POPRV SDOF Model using SuperChems

SuperChems Enterprise provides the ability to regress SDOF model parameters for conventional, bellows, and pilot operated pressure relief valves. Tools are also provided to test the goodness of fit and the performance of the developed models prior to use by the 1D fluid dynamics models. A numerical test bench tool is provided to quickly test the implied blowdown of an SDOF model as well as the relief device opening and closing times.

For example, a lumped SDOF model for a modulating 8T10 POPRV with a set pressure of 400 psig is developed using SuperChems Enterprise. The data provided by the manufacturer included an opening time of approximately 80 ms, an expected blowdown between 0 and 4 %, and the fraction open at three different levels of overpressure. The piston area protected from overpressure is set equal to the seat area (flow area), $\beta = 1$, and the piston area A_2 is set at 1.2 times the seat area, $A_2 = 1.2A_0$.

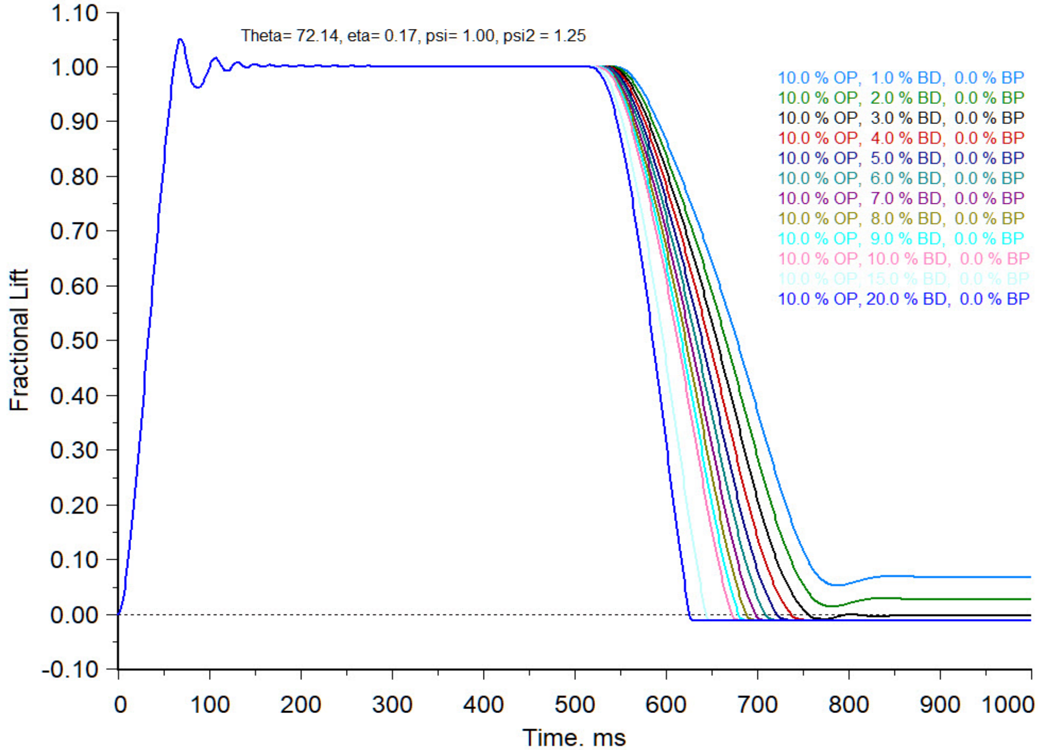
Figure 9 illustrates an excellent fit for a lumped SDOF model. Figure 10 illustrates the performance of the model yielding a blowdown of approximately 3 % and an opening time of approximately 80 ms.

Figure 9: SDOF best fit of model parameters for an 8T10 modulating POPRV set at 400 psig (Source: SuperChems Enterprise v11.92)



Lumped SDOF models are used to model the flow dynamics of systems where a single POPRV is installed as well as systems where multiple relief devices are installed. Information pertaining to the dynamic loading of reaction forces on system piping, potential unstable interactions of multiple device installations, optimal separation distances between relief devices, and the adequacy of set points staggering and sequencing can be evaluated for risk reduction.

Figure 10: SDOF best fit model performance testing for an 8T10 modulating POPRV set at 400 psig (Source: SuperChems Enterprise v11.92)



9 Conclusions

We have demonstrated that modeling the DDOF dynamics of pilot operated pressure relief valves can be achieved with a lumped SDOF mass-spring system.

A lumped SDOF model can be just as effective and often more practical to develop based on the limited availability of relief device data. The existing 1D Dynamics models in SuperChems Enterprise can be easily used to fit the required lumped SDOF model parameters and to model the dynamics of pilot operated pressure relief valves single device and multiple device installations.

References

[1] Q. Hao, W. R. Wu, X. Liang, and Z. Liu. Effects of structure parameters on abnormal opening of pilot-operated relief valve under alternating pressure. *IEEE Access*, 7:33932–33942, 2019.

[2] G. A. Melhem. Analysis of PRV stability in relief systems. Part I - Detailed dynamics. *ioMosaic Corporation White Paper*, 2014.

[3] G. A. Melhem. Analysis of PRV stability in relief systems. Part II - Screening. *ioMosaic Corporation White Paper*, 2014.

[4] G. A. Melhem. Analysis of PRV stability in relief systems. Part III - How to avoid the singing pressure relief valve problem. *ioMosaic Corporation White Paper*, 2014.

[5] G. A. Melhem. Analysis of PRV stability in relief systems. Part IV - On the estimation of speed of sound and thermodynamic properties for fluid flow and prv stability. *ioMosaic Corporation White Paper*, 2017.

[6] G. A. Melhem. Analysis of PRV stability in relief systems. Part V - Get a handle on PRV stability. *ioMosaic Corporation White Paper*, 2019.

[7] G. A. Melhem. PRV stability inlet line critical length. A short communication. *ioMosaic Corporation White Paper*, March 2021.

[8] G. A. Melhem. Analysis of PRV stability in relief systems. Part VII - Bridging the 3 % inlet line pressure loss rule gap. *ioMosaic Corporation White Paper*, 2022.

[9] K. Dasgupta and R. Karmakar. Dynamic analysis of pilot operated pressure relief valve. *Simulation Modelling Practice and Theory*, 10(1):35–49, 2002.

[10] S. Dimitrov. Static and dynamic characteristics of direct operated pressure relief valves. 2013.

[11] L. Wan, H. Dai, Q. Zeng, Z. Lu, Z. Sun, M. Tian, and Y. Lu. Characteristic analysis of digital emulsion relief valve based on the hydraulic loading system. *Shock and Vibration*, 2020.

[12] J. Peng, C. Youn, T. Takeuchi, and T. Kagawa. Stabilization of pilot valve system using linear flow resistance. *Advances in Mechanical Engineering*, 9(7), 2017.

[13] A. Singh, A. M. Hecht, and M. E. Teske. A model for predicting the performance of spring loaded safety valves. In *Testing and Analysis of Safety / Relief Valve Performance*, pages 47–54. American Society of Mechanical Engineers, ASME, 1983.

Index

About the Authors

Dr. Melhem is an internationally known pressure relief and flare systems, chemical reaction systems, process safety, and risk analysis expert. In this regard he has provided consulting, design services, expert testimony, incident investigation, and incident reconstruction for a large number of clients. Since 1988, he has conducted and participated in numerous studies focused on the risks associated with process industries fixed facilities, facility siting, business interruption, and transportation.

Prior to founding ioMosaic® Corporation, Dr. Melhem was president of Pyxsys Corporation; a technology subsidiary of Arthur D. Little Inc. Prior to Pyxsys and during his twelve years tenure at Arthur D. Little, Dr. Melhem was a vice president of Arthur D. Little and managing director of its Global Safety and Risk Management Practice and Process Safety and Reaction Engineering Laboratories.

Dr. Melhem holds a Ph.D. and an M.S. in Chemical Engineering, as well as a B.S. in Chemical Engineering with a minor in Industrial Engineering, all from Northeastern University. In addition, he has completed executive training in the areas of Finance and Strategic Sales Management at the Harvard Business School. Dr. Melhem is a Fellow of the American Institute of Chemical Engineers (AIChE) and Vice Chair of the AIChE Design Institute for Emergency Relief Systems (DiERS).

Contact Information

Georges. A. Melhem, Ph.D., FAIChE
E-mail. melhem@iomosaic.com

ioMosaic Corporation
93 Stiles Road
Salem, New Hampshire 03079
Tel. 603.893.7009, x 1001
Fax. 603.251.8384
web. www.iomosaic.com

How can we help?

Please visit www.iomosaic.com and www.iokinetic.com to preview numerous publications on process safety management, chemical reactivity and dust hazards characterization, safety moments, video papers, software solutions, and online training.

In addition to our deep experience in process safety management (PSM), chemical reaction systems, and the conduct of large-scale site wide relief systems evaluations by both static and dynamic methods, we understand the many nontechnical and subtle aspects of regulatory compliance and legal requirements. When you work with ioMosaic® you have a trusted ISO certified partner that you can rely on for assistance and support with the lifecycle costs of relief systems to achieve optimal risk reduction and PSM compliance that you can evergreen. We invite you to connect the dots with ioMosaic®.

We also offer laboratory testing services through ioKinetic® for the characterization of chemical reactivity and dust/flammability hazards. ioKinetic® is an ISO accredited, ultramodern testing facility that can assist in minimizing operational risks. Our experienced professionals will help you define what you need, conduct the testing, interpret the data, and conduct detailed analysis. All with the goal of helping you identify your hazards, define and control your risk.

ioMosaic®
Minimizing risk. Maximizing potential.®

About ioMosaic Corporation

Our mission is to help you protect your people, plant, stakeholder value, and our planet.

Through innovation and dedication to continual improvement, ioMosaic has become a leading provider of integrated process safety and risk management solutions. ioMosaic has expertise in a wide variety of disciplines, including pressure relief systems design, process safety management, expert litigation support, laboratory services, training, and software development.

As a certified ISO 9001:2015 Quality Management System (QMS) company, ioMosaic offers integrated process safety and risk management services to help you manage and reduce episodic risk. Because when safety, efficiency, and compliance are improved, you can sleep better at night. Our extensive expertise allows us the flexibility, resources, and capabilities to determine what you need to reduce and manage episodic risk, maintain compliance, and prevent injuries and catastrophic incidents.

Consulting Services

- Asset Integrity
- Auditing
- Due Diligence
- Facility Siting
- Fault Tree/SIL Analysis
- Fire & Explosion Dynamics
- Incident Investigation, Litigation Support, and Expert Testimony
- Hydrogen Safety
- LNG Safety
- LPG Safety
- Pipeline Safety
- Process Hazard Analysis
- Process Engineering Design and Support
- Process Safety Management
- Relief and Flare Systems Design and Evaluation
- Risk Management Program Development
- Quantitative Risk Assessment
- Software Solutions
- Structural Dynamics
- Sustainability Reporting Support
- Technology Transfer Package Development
- Process Safety Training

Laboratory Testing Services (ISO Accredited)

- Battery Safety Testing
- Chemical Reactivity Testing
- Combustible Dust Hazard Analysis and Testing
- Flammability Testing
- Physical Properties Testing
- Process Safety Services
- Specialized Testing

US Offices

Salem, New Hampshire
Houston, Texas
Minneapolis, Minnesota
Berkeley, California

International Offices

Al Seef, Kingdom of Bahrain
Bath, United Kingdom

Software Solutions

Process Safety Office® : A suite of integrated tools for process safety professionals and risk analysts.

Process Safety Enterprise® : Process Safety Management compliance made easy with enterprise workflows, dynamic forms, document management, key performance indicators and metrics, and more.

Process Safety Learning® : Build your process safety competencies incrementally using learning modules.

Process Safety tv® : The worldś first video streaming platform dedicated to process safety.

Contact us

www.ioMosaic.com
sales@ioMosaic.com
1.844.ioMosaic