Budapest University of Technology and Economics
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Predicting Pipeline and Safety Valve Dynamics in Multiphase Flow

Booklet of the PhD Dissertation

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Introduction

Safety valves (e.g., direct spring-operated pressure relief valve, see Fig.1.1), are devices that protect the pipeline system and reservoirs from excess pressure by venting the unnecessary fluid to maintain the system pressure beneath a prescribed maximum pressure and vent the unnecessary fluid in an emergency scenario. Safety valves are operated with different working mediums, for example, single-phase service (e.g., oil or air) and multiphase process fluids (e.g., steam-water). The nature of the fluid passing through the valve and the phase change that might occur during the relief process increases the complexity of the valve operation, and undoubtedly, the inconsistent performance of the valve becomes significant. Correspondingly, inappropriate operation of the safety-critical devices, such as insufficient venting capacity, vibration (chattering or fluttering), and the inability to open due to stuck parts, leads to catastrophic consequences.

A safety valve is a single degree of freedom (1 DoF) oscillator coupled with the fluid dynamics inside the piping system, resulting in surprisingly rich dynamical behaviour, see, e.g., [2, 3]. Furthermore, the inlet pipeline dynamics imposes additional investigation in the design phase due to the acoustic coupling between the valve body and the inlet (upstream) pipeline. Thus, correct PRV sizing is critical for optimal protection of the pressure vessel and pipeline system.

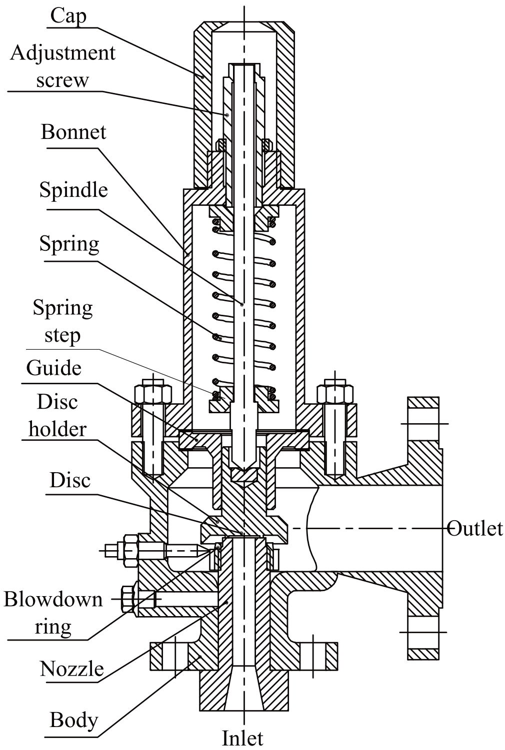

Figure 1.1: Direct spring-operated pressure relief valve [1].

A pressure relief valve, Fig.1.1 consists of an inlet or nozzle mounted on the pressurized system, a disc held against the nozzle to prevent flow under standard system operating conditions, a spring to hold the disc closed, and a body/bonnet to contain the functional elements. The movable parts are the seat disc (to ensure leakage-free closure), the disc holder loaded by an adjustable pre-compressed spring (to define the set pressure), and the blowdown adjustment ring to adjust the desired reseat pressure, [4].

Research plan

The research aims to address the safety valve and pipeline dynamics in the case of frozen mixture, assuming constant gas mass fraction in homogeneous mixture. Examples of frozen mixture can be: (a) humid air, (b) well-mixed bubbly flow (in vertical pipes), in the absence of phase separation or interphase slip, and (c) oil with small air content.

First, our primary concern is to estimate the opening time of a safety valve directly mounted on a reservoir (no inlet pipe between the reservoir and the valve). To ease the prediction of opening time parameter, I have derived an analytical approximation for this purpose (Thesis #**1**) where the mixture sonic velocity is the crucial parameter which results in drastic changes in valve opening time.

Next, exploring the effect of acoustic inlet piping on the valve stability are preformed to predict the instability thresholds of PRV (defining the stability map) (Thesis #**2**) where the quarter-wave criterion developed for single-phase flow is applicable for homogenous frozen mixtures to define the critical pipe length (instability borders).

After that, restricted valves are discussed extensively in air and water services for different set pressure to discover the implications of restriction solution on improving the valve stability. The discussed results reveals the difficulty of formulating a criterion to predict the valve instability of restricted valves

(Thesis #**3**).

Finally, the flow forces exerted on the valve disc have been investigated experimentally in terms of "effective area" approach to highlight the importance of the momentum forces. In addition, the discharge coefficients are studied empirically (Thesis #**4**). The main purpose to facilitate the prediction of these static parameters of safety valves by providing correlations which assist to estimate the effective area and discharge coefficient of different valve geometries with different sizes (different orifice diameters).

Developing an accurate multiphase flow physical model of a safety valve via multiphase flow (i.e., frozen mixture) models of safety valves and experiments was our main objective aiming to aid the researchers in comprehending the safety valve and the pipeline dynamics effects in designing a relief system.

Summary of the results

Valve opening time

In particular, the dynamical valve models discussed in [5, 6] were developed to cope with frozen mixture flow passing through a PRV connected directly to a pressure vessel. Utilising numerical simulation of the valve and reservoir dynamics at the equilibrium position, we have shown that upon varying the gas mass fraction, the valve opening time (i.e. the time for reaching 95% of the equilibrium valve lift) changes in a surprisingly wide range, from a few tenths of seconds up to as much as a few seconds showing a strong correlation with the sonic velocity. The valve opening time is a valve selection parameter crucial in studying valve stability. In terms of instability predictions, there are some criteria, such as pressure surge criterion, where the valve opening time is an input parameter, see [7, 8]. Hence, to ease the prediction of valve opening time, I derive an analytical approximation for this purpose with two simplified formulae that allow an order-of-magnitude estimation by simple means.

Thesis #1

In the case of a safety valve mounted directly on a reservoir and operating with frozen mixture flow, the valve opening time can be estimated by

$$t_{op} = \frac{\psi}{\omega_v} \quad \text{and} \quad \omega_v = \sqrt{\frac{s}{m}}, \tag{1}$$

where ψ is the solution of

$$\sigma := \frac{0.95 x_e \omega_v^3 V_r m}{A_v a_r^2 \dot{m}_{in}} = \frac{1}{\psi}\left(1 + \frac{\psi^2}{2} - \cos(\psi) - \psi\sin(\psi)\right), \tag{2}$$

where ω_v m, s, x_e, a, $\dot{m}_{in}$, V_r, and A_v denote valve eigenfrequency, valve mass, spring stiffness, equilibrium lift, sonic velocity, inlet mass flow, reservoir volume, and valve seat area, respectively. The above solution assumes constant discharge coefficient and effective area, and low valve damping (1% relative damping). The error of the opening time provided by the above formula is less than 30% compared to numerical predictions.

Moreover, if $\sigma < 0.09$, then $\psi = 2\sqrt[3]{\sigma}$ (slow valve) with a maximum relative error of 5% (compared to (2)). If $\sigma > 10.5$, then $\psi = 2\sigma$ (fast valve) with a maximum relative error of 10% (compared to (2)).
Related publication: [P1, P2, P3].

Fig.3.1 indicates that the solution of $\psi = 2\sigma$ (highlighted with solid black line) has a good agreement with simulation results (highlighted with red dashed line) with a maximum relative error of 5.4% for $s > 4.5$ kN/m ($\sigma > 10.5$ and the condition $\psi \gg 1$ holds). On the other hand, the simplified formula $\psi = 2\sqrt[3]{\sigma}$ for $\psi \ll 1$ (highlighted with solid cyan line) matches the simulation results significantly with a maximum relative error of 3.3% for $s < 1$ kN/m and $\sigma < 1.1$. The accurate solution of (2) (highlighted with solid magenta line) notably still agrees with the simulation results (note that both have the same tendency). However, using the simplified versions ($\psi = 2\sigma$ and $\psi = 2\sqrt[3]{\sigma}$) is reasonable and still more effective in terms of time

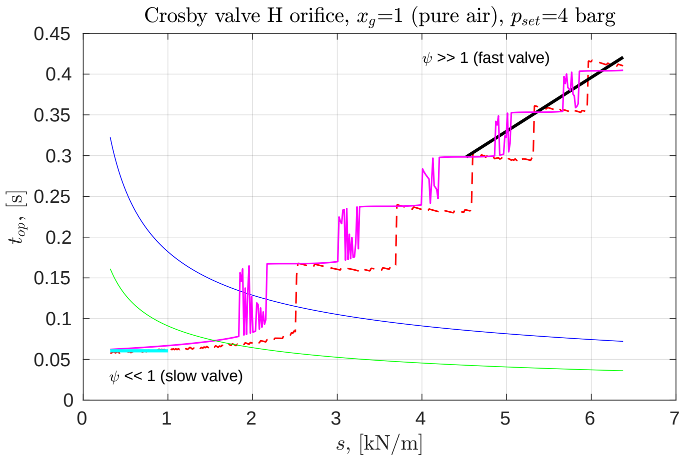

Figure 3.1: Valve with H orifice opening time vs spring stiffness where the lines: red dashed line stands for the simulation opening time, the solid magenta line refers to the accurate solution by (2), the solid black line is the solution of $\psi = 2\sigma$ for $\psi \gg 1$, the solid cyan line represents the solution of $\psi = 2\sqrt[3]{\sigma}$ for $\psi \ll 1$, the solid green line denotes the Grolmes expression $t_{op,Grolmes} \simeq \frac{1}{2}t_{valve}$, and the solid blue line represents the characteristic valve timescale t_{valve}.

cost rather than using the accurate formula (2) (which takes a long time to be solved). Eventually, comparing the results of $t_{op,Grolmes} \simeq \frac{1}{2}t_{valve}$ (see the solid green line) against the simulation results show a lack of its accuracy in terms of evaluating the valve opening time (the error increases with spring stiffness increasing) and our analytical approximation depicted in (2), $\psi = 2\sqrt[3]{\sigma}$, and $\psi = 2\sigma$ formulae has a superior performance than Grolmes model.

Dynamic of a pipe-valve system

In order to explore the effect of frozen mixture and lift restriction on the dynamic behaviour and stability of direct spring-operated pressure relief valves with upstream piping, I employed numerical prediction techniques. DIER's omega technique to model a two-phase homogeneous frozen mixture was used to predict the mass flow rate passing through the safety valve. Additionally, the Lax-Wendroff method was utilized to predict the dynamical behaviour inside the pipeline for unsteady mixture flow, coupled with the dynamic model of the valve and reservoir. In addition, the change in the critical pipe length (beyond which the valve chatters, see, e.g., [2]) as a function of the gas mass fraction was studied for the entire range between pure gas and pure liquid, see Fig.3.2. The investigation reveals that for mixtures of intermediate mass fractions, i.e. extremely low sonic velocity give rise to heavy valve chattering, even with short inlet piping. On the other hand, comparing the numerical simulation against the quarter-wave instability model developed in [9] showed that the latter is applicable in the case of homogenous frozen mixture applications, provided that the mixture sonic velocity is properly adjusted, see Thesis #**2**.

Thesis #2

For a relief system that comprises a reservoir, inlet pipe, and a safety valve, the quarter-wave criterion developed for single-phase flow is applicable for homogenous frozen mixtures. The mixture sonic velocity is the primary parameter governing the onset of instability. The formula was verified against simulations, providing a maximum error of 50% for the x_g range (0.1-1) and 12% near the liquid-dominated region ($x_g < 10^-3$) assuming constant effective area, constant discharge coefficient, and small-amplitude oscillations around the equilibrium. Hence

$$L_{crit} = \frac{\pi a(x_g)}{2\omega_v} \frac{1}{\sqrt{2\frac{A_v p_e}{x_e s} + 1}}, \qquad (3)$$

where ω_v denotes the valve natural frequency, x_e and p_e are the equilibrium valve lift and (absolute) pressure at the valve inlet, $a(x_g)$ is the *mixture sonic velocity*, s is the spring stiffness and A_v is the valve (bore) area, on which p_e acts.
Related publication: [P3, P4, P5].

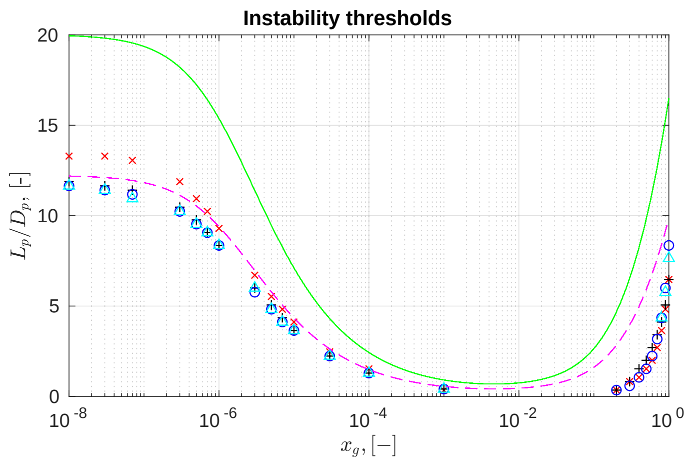

Figure 3.2: Critical relative pipe length (L_p/D_p vs. gas mass fraction x_g for different safety valve geometries. Symbols: $\times$: $A_{eff} \approx 1$ & $C_d = const$, $+$: $\theta = 120°$ poppet valve, $\bigcirc$: $\theta = 90°$ disc valve, $\triangle$: $\theta = 0°$ disc valve, dashed line is the solution of (3), and the solid green line represents the solution of $L^*_{crit} = \frac{\pi a}{2\omega_v}\frac{1}{\sqrt{\frac{x_e+x_0}{x_e}}}$.

At first glance, Fig.3.2 reveals that for all geometries, the instability border (in terms of L_p/D_p) decreases once the gas mass fraction x_g decreases within the interval starting from 1 up to 0.1. For $0.1 \leq x_g \leq 1$, the local speed of sound drops from 343 m/s to roughly 50 m/s, resulting in decreasing L_p/D_p from 8 up to a minimal value of around 0.25. In contrast, for x_g range from 0.1 to 0 (pure water), the instability borders are reversely proportional to x_g, which is due to the fact that the sonic velocity of the mixture increases again since the mixture density becomes greater (i.e., liquid content increasing).

The valve geometry does not seem to play a significant role in valve stability: the general valve geometry with $A_{eff} \approx 1$ (highlighted with red $\times$ sign in Fig.3.2) provides almost identical results to the rest of the geometries in gas service ($3\times10^{-5} \leq x_g \leq 0.6$. In the liquid-dominated region $x_g \leq 10^{-6}$, the poppet valve provides slightly better results (longer allowable inlet pipe lengths), but still irrelevant from the viewpoint of practical applications. In addition, comparing the simulations results with the quarter wave model depicted in (3)(see the magenta dashed line) shows a significant matching with a maximum relative error of 12% for x_g smaller than 10^{-3}, while the maximum error increases for $0.1 \leq x_g \leq 1$ up to 50%. Nevertheless, the trends clearly coincide. Eventually, the Izuchi model given by $L^*_{crit} = \frac{\pi a}{2\omega_v}\frac{1}{\sqrt{\frac{x_e+x_0}{x_e}}}$ (see the solid green line) shows larger magnitudes of critical pipe length L_{crit} compared with the simulation results. Consequently, (3) has a superior performance over Izuchi model.

Next, I explore the possibility of improving the stability properties of the system by applying larger but restricted valves while keeping the vented mass flow rate constant. The numerical experiments suggest that the additional restoring force emerging from the valve restriction improves the stability behaviour in the case of gas applications, see Fig.3.3 but has hardly any effect in the case of valves in liquid service, see Thesis #**3**.

Thesis #3

For a direct spring-operated pressure relief valve, employing a larger valve orifice (larger nozzle diameter) with lift restriction leads to improved stability (less chattering) in the case of gas

service. The restriction must be chosen in such a way that the capacity of the restricted, larger valve equals the non-restricted capacity of the original valve. However, in the case of liquid service, restricted valves do not show enhanced stability. In the case of gas service, up to 50 barg set pressure, the critical inlet pipe length (beyond which the valve chatters) is improved by a factor of min. 5, if the area ratio of the original and restricted (larger) valve is 1.4. For higher set pressures, area ratios beyond 2 result in a negative effect on the critical pipe length.
Related publication: [P4, P5].

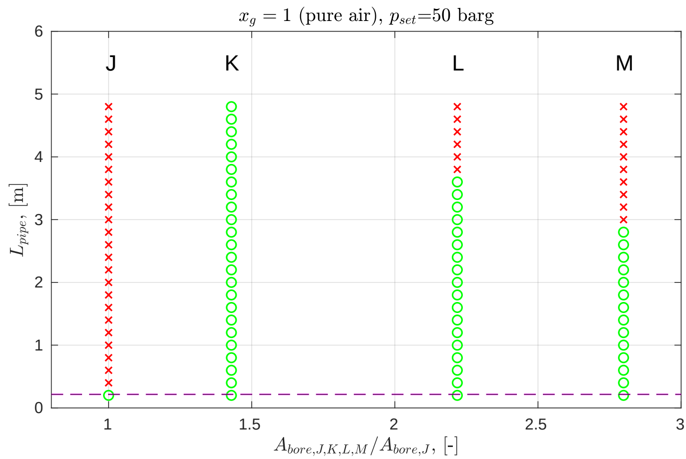

Figure 3.3: Stability map of valve with orifices J, K, L and M at $p_{set} = 50$ barg, in air service.

Flow force and discharge coefficient

In order to study the flow forces and find the discharge coefficients of a safety valve, experimental work was conducted to measure the total (overall) force exerted on the valve disc and the mass flow rates passing through the valve area in the case of frozen mixtures for different valve geometries. The measured forces were represented in terms of the "effective area" concept to highlight the importance of the momentum force. Through the measurement runs, we have shown that the added water (liquid mass fraction) plays a minor role in changing the momentum force for water mass fractions ranging between 0 (pure air) and 0.41, which is consistent with the corresponding literature [10] and [11]. We have highlighted frozen mixtures effects by comparing the effective area curves of three different geometries of a safety valve. In the case of the poppet (conical) valve, the fluid force decreases with increasing the valve lift, while for disc valves, the fluid force increases with the increase of the valve lifts. The effect of increasing mass fraction ratios can be combined into a single correlation model, see Fig.3.4. Moreover, the discharge coefficients show the same tendencies for the tested geometries, decreasing with the lift increasing and, even in the frozen mixture region, do not change significantly, allowing one single correlation to be used over a wide range of mass fractions. This result provides a reasonable estimation for the required discharge factor in the process of valve sizing in the case of two-phase flow.

Thesis #4

For three different geometries of safety valves (a poppet valve and two disc-shaped valves with different jet angles), the dependence of the A_{eff} effective area curve and the C_d discharge coefficient was experimentally studied for various inlet pressures in the case of frozen mixture flow of air and water. It was found that A_{eff} and C_d can be approximated by the following correlations, independently of the liquid mass fraction.

For a poppet valve with jet angle $\theta = 2\pi/3$,

$$A_{eff} = 0.37\tilde{x}^2 - 0.56\tilde{x} + 1 \qquad \text{and} \qquad C_d = 1.56\tilde{x}^2 - 1.67\tilde{x} + 1. \tag{4}$$

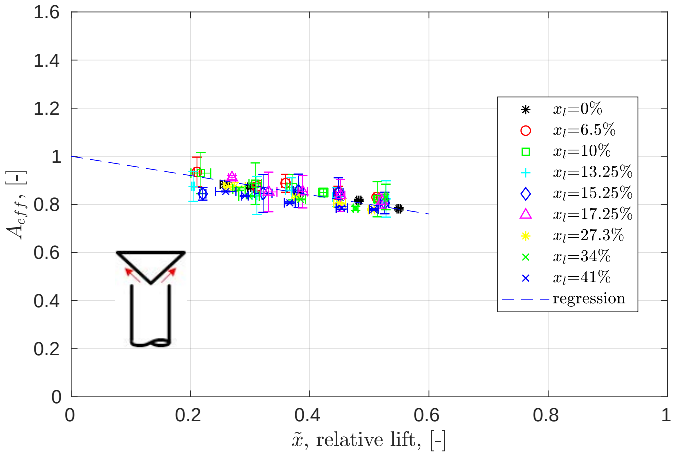

Figure 3.4: Effective area vs. relative lift of poppet valve, $\theta = 2\pi/3$.

For a disc valve with jet angle $\theta = 0$,

$$A_{eff} = 1 + 0.56\tilde{x} \qquad \text{and} \qquad C_d = 1 - 0.59\tilde{x}. \tag{5}$$

For a disc valve with jet angle $\theta = \pi/2$,

$$A_{eff} = 1 + 0.1\tilde{x} \qquad \text{and} \qquad C_d = 1 - 0.45\tilde{x}. \tag{6}$$

In the above equations $A_{eff} = \frac{F_{fluid}}{A_v \Delta p}$ with F_{fluid} being the total fluid force exerted on the valve body, $A_v = \pi D_v^2/4$ is the area on which the inlet pressure acts for a closed valve and Δp stands for the pressure drop through the valve. Moreover, the relative valve lift is $\tilde{x} = \frac{4x}{D_p}$, where x denotes the valve lift.

The above correlations hold in the range of $1 \leq \Delta p \leq 6.6$ bars (the valve vents to ambient pressure) and $0 \leq x_l \leq 0.41$ (x_l is the liquid mass fraction). The maximum relative error between the experimental results and the above predictions is 16% for the effective area and 12% for the discharge coefficient. Related publication: [P2, P6, P7].

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