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Self-excited Oscillations of Direct Spring Operated Pressure Relief Valves

Booklet of the PhD Dissertation

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Introduction

Pressure relief valves serve as the final safety net against overpressure, hence their reliable operation is of utmost importance. In some special cases, such as in the oil industry — where most of the protected pipelines are located in remote areas — they should also be able to operate with low maintenance needs even in harsh environment. The direct spring operated construction, which consists of a body pressed against the seat by a pre-compressed spring (Fig. 1.1), fulfils these needs, as its operating principle is relatively simple, and it only has a few moving parts. The two significant parameters of a pressure relief valve from a sizing viewpoint are the *set pressure*, that is the pressure difference at which the valve opens; and the *capacity*, which is by definition the flow rate through the valve at set conditions.

From industrial experience [1] it is known that this type of valves is prone to a number of instabilities arising from various physical phenomena. Indeed, as the valve is essentially a one degree-of-freedom oscillator with an external excitation from the fluid forces, the non-linearity of this system — especially if other connecting elements, such as pipelines and vessels, are considered as well — is expected to introduce instabilities in certain operating regions. One of the possible causes is the so-called *quarter-wave instability*, which is the direct result of the acoustic coupling between the valve and the inlet piping. Its

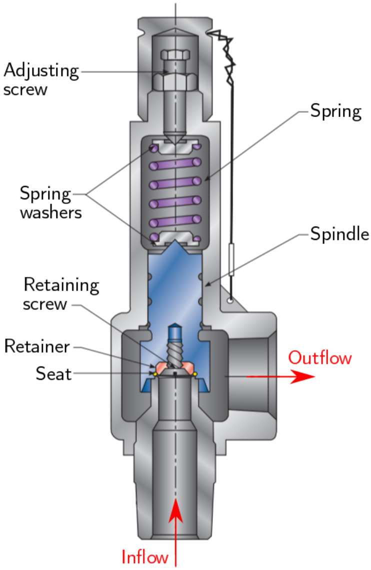

Figure 1.1: A typical direct spring operated valve [2].

name originates from the fact that if this instability appears, one typically encounters the first quarter-wave harmonics of the pipe, that is a standing wave of wavelength of four times the pipe length. As a direct continuation of the extensive research provided by Hős et al. [3, 4, 5, 6], the main focus of this thesis is to provide further validation of the so-called *quarter-wave model* by performing accurate CFD simulations, and to extend both their *quarter-wave model* and their one-dimensional *gas dynamical model* with the concept of the effective area. Finally, an application of the extended *gas dynamical model* will be shown with experimental validation for a configuration in which the valve discharges to the atmosphere through an additional outlet pipe (contrary to the previously cited articles, in which the valve always discharged directly to the atmosphere).

Summary of the results

2.1 Blowdown prediction

Many pressure relief valves exhibit a phenomenon called *blowdown*. It means that the reseating pressure of the valve (i.e. the pressure drop at which the valve closes) is *lower* than the set pressure. This behaviour is widely known in the process industry, and the terminology, sizing guidelines and required amount of blowdown are therefore specifically treated in industrial standards and codes of practice (e.g. see [1, 7]). However, the required steps to achieve a given level of blowdown typically involve CFD simulations or empirical measurement, for example, through adjustment of so-called blowdown rings *in situ*. As both the experimental and numerical methodology for blowdown analysis has already been discussed in detail in the literature, this work presents a rather simple analytical approach for explaining the underlying physical reasons for this phenomenon, and also for the purpose of quick qualitative analysis. The modelling methodology is based on the research of Bazsó et al. [8], however, in the present work stability analysis is based on the vessel total pressure instead of the static pressure at the valve, thus providing a more systemwise point of view. It will be shown that blowdown is essentially a special case of linear static instability.

Thesis #1

In the case of direct spring operated pressure relief valves with incompressible Newtonian liquid, the blowdown (i.e. the difference between the opening and closing pressures) can be predicted analytically by calculating the equilibrium total pressure drop vs. lift curve from

$$\Delta p_{\mathrm{t}} = \frac{\left(1 + C_{\mathrm{D}}^2 B^2(\tilde{x})\,\tilde{x}^2\right) s\,(\tilde{x} + \tilde{x}_0)}{D\pi \tilde{A}_{\mathrm{eff}}(\tilde{x})},$$

where Δp_{t} is the total pressure drop on the valve, C_{D} is the discharge coefficient, $\tilde{x} = 4x/D$ is the dimensionless lift, $\tilde{x}_0 = 4x_0/D$ is the dimensionless pre-compression of the spring, s is the spring stiffness, D is the seat diameter, and $\tilde{A}_{\mathrm{eff}}(\tilde{x}) = A_{\mathrm{eff}}(\tilde{x})/(D^2\pi/4)$ is the dimensionless effective area. Additionally, $B(\tilde{x}) \equiv 1$ for disc shaped valve bodies, and $B(\tilde{x}) = \left(1 - \frac{\tilde{x}}{4}\cos\alpha\sin\alpha\right)\sin\alpha$ for conical valve bodies, where α is half of the opening angle of the cone.

With the help of this characteristic curve, the extent of blowdown can be evaluated based on the $\tilde{A}_{\mathrm{eff}}$ effective area function and the C_{D} discharge coefficient of any given valve.

Thesis #2

For direct spring operated pressure relief valves with incompressible Newtonian liquid, let $\Delta p_{\mathrm{t}}(\tilde{x})$ denote the equilibrium lift curve (that is, the $\tilde{x}$ equilibrium valve lift for a given Δp_{t} total pressure difference). Then, the condition for linear static stability is

$$s - D\pi \frac{\mathrm{d}\tilde{A}_{\mathrm{eff}}(\tilde{x})}{\mathrm{d}\tilde{x}} \frac{1}{1 + C_{\mathrm{D}}^2 B^2(\tilde{x})\,\tilde{x}^2} \Delta p_{\mathrm{t}} +$$
$$D\pi \frac{2C_{\mathrm{D}}^2 \tilde{A}_{\mathrm{eff}}(\tilde{x})\, B(\tilde{x})\,\tilde{x}\left(B(\tilde{x}) + \frac{\mathrm{d}B(\tilde{x})}{\mathrm{d}\tilde{x}}\tilde{x}\right)}{\left(1 + C_{\mathrm{D}}^2 B^2(\tilde{x})\,\tilde{x}^2\right)^2} \Delta p_{\mathrm{t}} > 0,$$

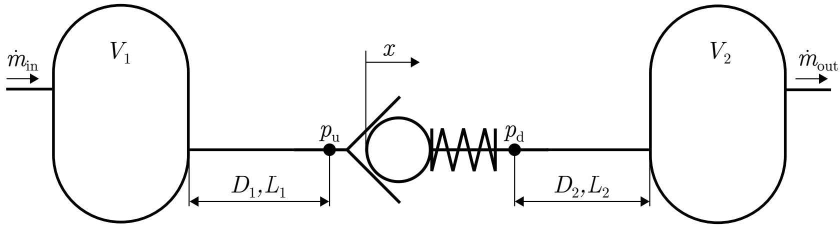

Figure 2.1: The gas dynamical model.

where s is the spring stiffness, D is the seat diameter, $\tilde{A}_{\mathrm{eff}}(\tilde{x}) = A_{\mathrm{eff}}(\tilde{x})/(D^2\pi/4)$ is the dimensionless effective area, C_{D} is the discharge coefficient, $\tilde{x} = 4x/D$ is the dimensionless lift, and Δp_{t} is the total pressure drop on the valve. Additionally, $B(\tilde{x}) \equiv 1$ for disc shaped valve bodies, and

$$B(\tilde{x}) = \left(1 - \frac{\tilde{x}}{4}\cos\alpha \sin\alpha\right)\sin\alpha$$

for conical valve bodies, where α is half of the opening angle of the cone.

This condition is equivalent to

$$\frac{\mathrm{d}\Delta p_{\mathrm{t}}(\tilde{x})}{\mathrm{d}\tilde{x}} > 0$$

for the equilibrium $\Delta p_{\mathrm{t}}(\tilde{x})$ curve, providing a visual method for stability analysis based on the $\tilde{A}_{\mathrm{eff}}$ effective area and the C_{D} discharge coefficient.

2.2 Valve boundary condition

The general, extended gas dynamical model can be seen in Figure 2.1. It consists of an upstream vessel, an upstream pipe, a direct spring operated valve (based on its operation, it can either be a relief or a check valve), a downstream pipe, and a downstream vessel. The mass conservation equation is solved for the vessels with various assumptions for the change of state based on the operation, the one-dimensional mass, momentum and energy conservation equations are solved for the pipes, while the valve is modelled as a one degree-of-freedom oscillator. Appropriate boundary conditions must be defined between the listed components, which were based on the work of Hős et al. [3, 4, 5, 6]. Contrary to the cited articles, the current setup includes a downstream pipe after the valve, for which a new boundary condition will be presented.

Thesis #3

Using the isentropic method of characteristics, a physically consistent boundary condition was developed for a direct spring operated pressure relief valve connecting two straight pipe sections, and operating with compressible Newtonian fluid. The unkown variables to be computed are the state variables (pressure, temperature, density, specific energy, velocity) at the upstream and the downstream pipe connections, as well as the pressure at the valve seat and directly downstream of the valve.

The method assumes ideal gas, that the flow in the vicinity of and through the valve is isentropic, and that choking (if any) occurs at the valve body-seat cross section. It also takes into account the cross-sectional area changes between the upstream pipe and the seat, and the downstream valve chamber and pipe.

The equations to be solved for modelling the valve itself and the effect of the cross-sectional changes from the upstream pipe to the seat, and from the valve to the downstream pipe are the following: the flow-through equation for the valve (separated for choked and non-choked cases), the continuity equation, the conservation equation from the incompressible method of characteristics, the energy equation, the equation for isentropic change of state, and the ideal gas law.

If the valve is closed, a wall boundary condition should be applied for both the upstream and the downstream pipe solvers.

Backflow can be treated by simply exchanging the upstream and downstream sides.
Related publications: [E1].

2.3 Quarter-wave instability

To validate the results of the gas dynamical model and the assumptions of the quarter-wave model, unsteady CFD deforming mesh simulations were set up. As a relief valve operates in a relatively wide lift range, the expected mesh deformations are significant, leading to a decrease in mesh quality, or in some cases the mesh can even “fold” onto itself. To avoid this, dynamic remeshing was employed, i.e. once the mesh deforms beyond a prescribed limit, a new mesh is generated.

Of the gas dynamical, the quarter-wave, and the deforming mesh models presented in this work, the deforming mesh CFD simulations can be regarded as the most accurate. This technique is able to resolve all components of the system, as it couples the motion of the oscillating valve body with the transient flow field for any arbitrary valve geometry.

Comparing the results of the gas dynamical and the CFD models shows that the pipe is indeed in the quarter-wave mode for small amplitude oscillations, i.e. just beyond the loss of stability (which means slightly different parameters for the two models), as seen in Figure 2.2. The author would like to emphasize that the goal was not to fit the GDM and CFD curves as close as possible, but to show that both models are in the quarter-wave mode during the small amplitude oscillations following the loss of stability. A quarter sine wave fitted on the first and the second to last pressure value (dashed line in Figure 2.2) confirms that the assumptions were true. Note that the last pressure value does not fit on this curve as it is directly at the seat, resulting in a strong “stopping” effect from the valve.

Up to the author’s knowledge, this is the first case when a quarter-wave instability was capture by means of CFD.

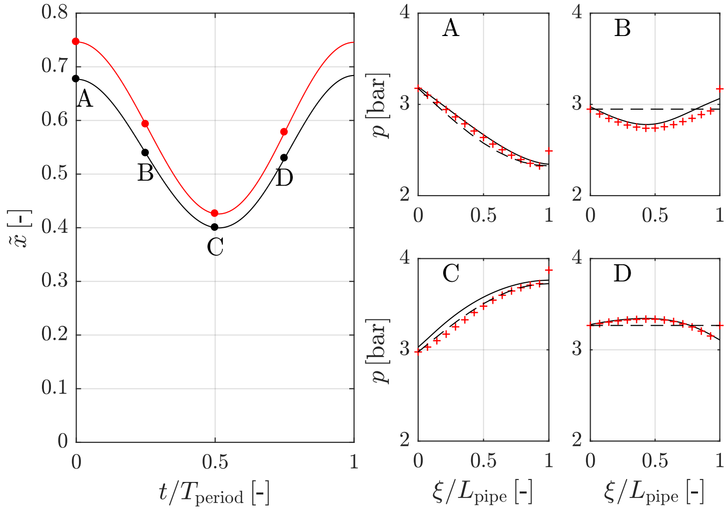

Figure 2.2: The dimensionless valve lift during an oscillation period (left, black: GDM, red: CFD) and the pressure distributions in the pipe at the marked time instants (right, black cont. line: GDM, red markers: CFD, dashed line: quarter sine wave).

Thesis #4

For a direct spring operated pressure relief valve installed on a straight pipe section, and operating with compressible Newtonian fluid, it was verified by two-dimensional unsteady coupled RANS CFD simulations that upon loss of dynamic stability, the pressure distribution in the pipe exhibits oscillations in the quarter-wave form. This proves that the primary mechanism causing the quarter-wave instability is the acoustic coupling between the pipe and the valve.
Related publication: [E2, E3, E4, E5, E6].

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