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Dynamic behaviour of direct spring loaded pressure relief valves: III valves in liquid service

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ABSTRACT

Previous studies into direct-spring pressure relief valves connected to a tank via a straight pipe are adapted to take account of liquid sonic velocity. Good agreement is found between new experimental data and simulations of a coupled fluid-structure mathematical model. Upon increasing feed mass flow rate, there is a critical pipe length above which a quarter-wave instability occurs. The dependency is shown to be well approximated by a simple analytical formula derived from a reduced-order model. Liquid service valves are found to be stable for longer inlet pipes than for the gas case. However, the instabilities when they do occur are more violent and the valve is found to jump straight into chatter, in which it impacts repeatedly with its seat. Flutter-type oscillations are never observed. These observations are explained by finding that the quarter-wave Hopf bifurcation is subcritical. Water hammer effects can also be observed, which result in excessive overpressure values during chatter. In addition a new, Helmholtz-like instability — not encountered in gas service — is identified for short pipes with small reservoir volumes. This can also be predicted analytically and is shown to explain a valve-only instability found in previous work that incorporated significant mechanical damping.

1. Introduction

This paper continues the previous work by the present authors in Hős et al. (2014, 2015) on practical considerations of mechanisms of instability in direct spring operated pressure relief valves (PRVs). Here we specifically consider valves in liquid, rather than gas, service. Current guidelines (such as the API RP520) for avoiding valve flutter and chatter in both gas and liquid cases, refer to the need to avoid inlet pressure losses due to pipe friction of more than 3% of the set pressure of the valve. The logic of this criterion is to avoid so-called rapid cycling motion, in which pressure loss causes the valve to shut prematurely, only to open again soon afterwards. The believed sufficiency of this criterion can be traced back to work of Frommann and Friedel (1998) who studied valve vibrations in pneumatic systems both numerically and experimentally. However, as shown in Hős et al. (2014) for gas and in Figs. 3 and 4 below for liquids, this criterion does not seem to capture the correct parameter trends especially for low mass flow rates.

Instead, in Hős et al. (2015) we found an accurate stability criterion which we tested against experimental results for three different commercially available gas service PRVs. The key is to recognize that the fundamental instability causing valve flutter is a flow-induced *Hopf bifurcation* caused by an interplay between the valve natural dynamics and the fundamental *quarter-wave* acoustic vibration mode in the pipe. Effectively, the valve can supply *negative damping* to the acoustic mode; which is an explanation that does not rely on there being any internal resonance. The trend for this instability was found to be such that for each mass flow rate up to full capacity, there is a critical inlet pipe length beyond which the valve is unstable, with an approximately square-root dependence between the critical pipe length and the mass flow rate. For a yet longer pipe, the limit-cycle flutter behaviour will undergo a transition into large-amplitude chatter motion in which the valve impacts with its seat. Moreover, through reduced-order modelling, in Hős et al. (2015) we were able to produce a close analytical approximation to this curve which depends only a few pipe and valve parameters. In addition, specific features of the valve geometry can cause static jumps, or *fold bifurcations*, but these do not necessarily lead to flutter and shall not be considered in the present

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work.

The purpose of this paper is to extend that work to deal with the case of liquids. The paper Hős et al. (2014) contains a comprehensive literature review on instabilities of pressure relief valves, focusing mostly on the case of gas pipes. We give here only additional references relevant to the liquid case.

Licsko et al. (2009) assembled a simple system of ordinary differential equations describing the motion of a valve connected to a hypothetical hydraulic chamber. They performed linear and nonlinear stability analyses and found parameter ranges where flutter-induced limit cycles turn into chattering motion through *grazing bifurcations*. At the opposite end of the spectrum of computational complexity, Song et al. (2010), produced a 3D CFD model with deforming mesh for investigating the precise flow physics of transient valve motion at instability. Moussou et al. (2010) investigated both static and dynamic instabilities through a mixture of analytical calculations, CFD and experiments. They also introduced the concept of what is referred to in our work as the *effective area* of the valve at a given lift (called the *equivalent surface* by Moussou et al. (2010)) and investigated the appearing limit cycle. Similar concepts were developed by Viel and Imagine (2011) who studied stability with the help of Nyquist plots, and by Beune et al. (2012) who calculated flow force versus displacement using more computationally efficient 2D CFD. Further developments in full CFD simulations for liquid service PRVs have been undertaken by various authors showing just how complex the flow field can be, depending on the valve geometry and turbulence model adopted: see Dossena et al. (2013); Qian et al. (2014); Song et al. (2014, 2013); Wu et al. (2015).

Bazsó and Hős (2012) investigated the need for unsteady CFD for predicting the point of onset of the instability by, comparing 2D simulations with full 3D deforming mesh computations for a simple conical valve body. They found that both were able to capture the same point of instability, even reaching agreement on the nature of the appearing limit cycle, sufficiently close to the instability point. These predictions were also found to match well with experiments. Furthermore, recent work by Erdődi and Hős uses 2D CFD computations with both stationary and moving meshes to justify the use of simple effective-area-versus-lift curves and single discharge coefficients. Those results appear to justify the use of reduced-order modelling for instability prediction, even for liquids where the unstable motions tend to be much more violent than for gas flows. This violence is due to the significant additional momentum in the fluid, which would otherwise be dissipated in compressible fluids.

Such reduced-order models were studied in a series of papers: Hős and Champneys (2012); Bazsó et al. (2015, 2014). These works produce a simple dimensionless models of liquid service PRVs without fitting parameters to a particular set of test valves. Another form of Hopf bifurcation also leading to flutter and chatter was discovered, referred to as a *valve only* or *Helmholtz-like* instability. This instability was found to occurs for finite mass-flow rates even in the limit that the pipe length tends to zero. It is of important to note though that those studies assumed significant mechanical valve damping, which is unrealistic for commercial valves.

The rest of this paper shall follow a similar approach to that in Hős et al. (2014, 2015). In Sec. 2 we present the results of dedicated tests of valves in liquid service connected to a reservoir via a straight pipe of variable length to document cases that lead to instability, and the nature of the instabilities observed. Then, Sec. 3 presents our previous mathematical models and their modification to the problem at hand. We also present a new analysis on the cause of the Helmholtz-like instability. Sec. 4 presents numerical simulations of the model, along with parameter studies and comparison with both experiments and the analytical predictions of the quarter-wave and Helmholtz-like instabilities. Finally, Sec. 5 draws conclusions and discusses practical significance of the work.

2. Experimental results

2.1. Test procedure

A series of tests were conducted in the Pentair test facility using the rig depicted in Fig. 1 in which pressurized water is connected to a valve via a straight inlet pipe. The length of the pipe can be varied from 0 to 12 feet by carefully fitting together pipe segments of different lengths. The pressure in the feeder tank is closely controlled by means of a supply from a larger tank whose pressure can be increased via a drive of nitrogen gas. A nitrogen vent can be opened in order to reduce the tank pressure. Two different industry standard valve types were used, a 2J3 and a 1E2. It should be stressed that neither of these valves were fitted with specific liquid trims. The gas trim used was identical to that used for the gas-service tests Hős et al. (2014), so as to enable a direct comparison.

A total of over fifty tests were performed for different pipe lengths and different desired mass flow rates. In each run, the drive pressure (labelled PD in Fig. 1) was slowly ramped up to a fixed value that enables a mass flow rate of the desired percentage of the valve's capacity. The drive pressure was held steady for about 20 s before the nitrogen valve was opened and the pressure ramped down. All tests were performed with a nominal opening set pressure of 120 psig (8.27 bar). The actual opening pressure was measured from the data at the valve inlet (labelled VI), at the instant when the lift first reaches 1% of its maximum possible lift. This pressure was typically found to be within a few percent of the nominal set pressure. Blowdown pressure was nominally set to be 114 psig (7.86 bar, or 5% of set pressure), but was similarly measured as the VI pressure at which the lift falls beneath 1% after the valve has been open for at least 1 s. However, as can be seen from the example time traces in Fig. 2 below, the measurement of blowdown in particular was found to be problematic due to the noisy nature of the pressure response in the pipe.

The capacity (flow rate at full lift and 10% above set pressure) of the two valves was nominally 400 USgpm (25.23 kg/s) for the 2J3 and 70 USgpm (4.41 kg/s) for the 1E2 valve. These values and measurements of the rate of pressure loss at capacity allowed us to determine the discharge coefficient C_d corresponding to the *curtain area* $D_{bore}\pi x_{valve}$ of each valve. We thus obtained C_d-values of 0.36 and 0.32 respectively for the 2J3 and the 1E2 valves.

Accurate measurement of sonic velocity is important for matching with simulations, because it plays a crucial role in the dynamic interaction between the pipe and the valve. The standard value for fresh water in an infinitely rigid pipe or reservoir would be about 4700 ft/s (1430 m/s) but both pipe wall elasticity and air content within the water reduce this value in practice. We measured sonic velocity using the following procedure. Upon valve closing, pressure waves are generated in the pipe, which are then reflected back from the reservoir end of the pipe. The time needed for one full cycle is

$$T_{\text{pipe}} = \frac{2L}{a}, \tag{1}$$

with a being the sonic velocity and L the length of the pipe. By recording the time needed for 10 such reflections to occur, and averaging over several tests, we were able to estimate the actual sonic velocity to be

$$a = 2811\ \text{feet/s} \quad = 857\ \text{m/s}$$

Due to the relatively long pipe lengths, pressure loss due to pipe