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Experimental and theoretical analysis of flashing water flow through a safety valve

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Abstract

A set of experiments has been carried out on the model of the CROSBY safety valve for flashing and nonflashing water flow. The pressure distribution inside the valve, the inlet and outlet temperatures and the mass-flow rate have been measured. The characteristics of the valve giving the mass-flow rate as a function of the square root of the pressure drop are given. The experimental results were compared with equilibrium and relaxation two-phase flow models. The nonequilibrium character of fast evaporation and its substantial influence on the two-phase flow behaviour has been emphasized. This could constitute a comprehensive base for a better understanding of two-phase flow through safety valves.

Keywords: Flashing flow; Safety valve; Two-phase flow; Relaxation model; Choked flow

1. Introduction

Safety relief valves play an important role in industrial emergency systems. They are installed in pressurized systems to ensure operating pressure does not exceed unsafe limits that could result in an accident. It may happen, however, that due to off-normal or accidental conditions, the valve initially sized for highly subcooled liquid may be required to discharge a liquid with the temperature not far from the saturation one. The occurrence of rapid evaporation (flashing) during a discharge of initially subcooled liquid essentially limits the valve capacity. So, the pressure relief system may not operate as it is designed because the safety valve, selected by means of the traditional sizing methods generally based on the Bernoulli equation, may produce the mass-flow rate too small to adequately reduce the pressure inside an installation. The present knowledge about this phenomenon is insufficient and no theoretical

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model exists that can accurately predict the total pressure drop for the subcritical flashing liquid flow through a safety valve. Only a few experimental data concerning this problem are available in the literature. Most of them report only inlet and outlet parameters, which makes investigations of all the complicated phenomena inside the valve body rather difficult.

2. Experimental

A scheme of the experimental facility is shown in Fig. 1. A vessel is filled with hot pressurized water. The pressure is maintained constant during the whole experiment by means of compressed air. Water is heated up to the required temperature by circulating through the boiler; the closed circuit linking the boiler to the vessel is interrupted during the discharge of the vessel.

The discharge line consists of the following elements: a ball valve, a vertical pipe (1 in. diameter), a model of safety valve, a horizontal 2 in. pipe and a catch tank connected to the atmosphere. A 2 m long pyrex glass section enables to visualize the flow at the exhaust of the safety valve model. There is the possibility to control the pressure in the catch tank by filling it initially with compressed air and adjusting the valve situated on the connection pipe with the atmosphere. The discharge line is preheated by means of the auxiliary circuit fed with a very slow flow of water originating from the vessel.

The mass-flow rate is measured by means of the electromagnetic flowmeter located at the bottom of the vertical pipe. Temperature measurements are made by chromel–alumel thermocouples (three thermocouples are located in the vessel, two on the vertical pipe, two on the horizontal pipe, one in the catch tank). Pressure measurements are made at 18 locations, mainly concentrated inside the valve model (see Fig. 1). A data acquisition system (ASYST) enables, among other features, the observation of instantaneous values of mass-flow rates, pressures and temperatures on PC, during the experiment itself.

Maximum characteristics of water in the vessel are 6 bar and 150 °C. Limiting factors are related to the flowmeter: it exhibits a 150 °C temperature limit and the flow is restricted to pure liquid. That excludes, presently, the possibility of feeding the valve model with two-phase mixture.

Fig. 1 presents the model of safety valve (CROSBY 1D2 JLT-JOS-15-A) with indicated positions of the pressure taps. During experiments, the disc of the valve model is fixed to any desired lift. The curves giving the area of the smallest cross-section open to the flow versus lift are shown in Fig. 2, where S_{min} is the area of the last cylindrical section of the nozzle (10.4 mm diameter).

3. The models

The most significant feature of flashing flow is the ease with which critical flow conditions are attained. The classical homogeneous equilibrium model (HEM) was

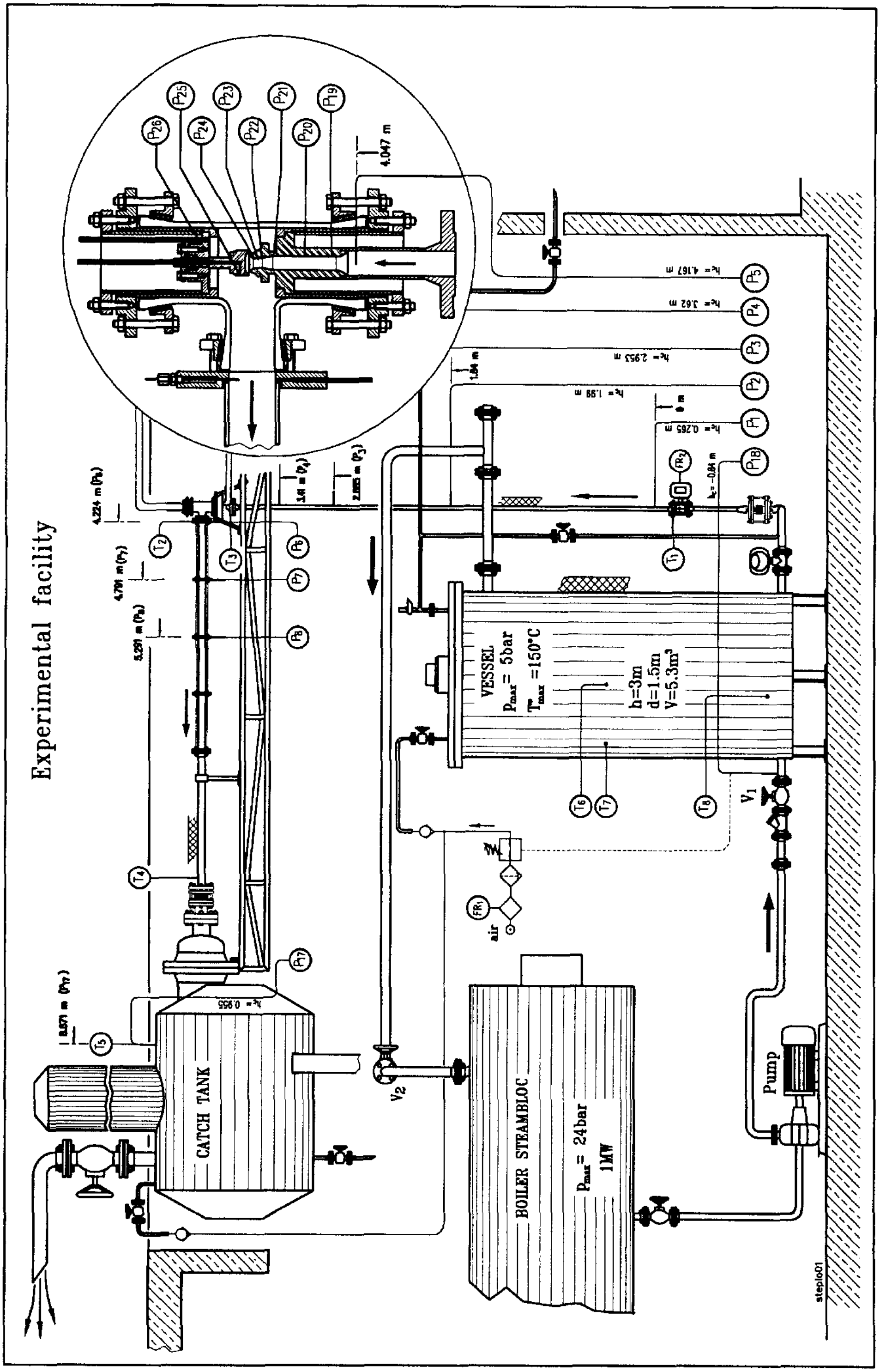

Fig. 1. Scheme of the experimental facility.

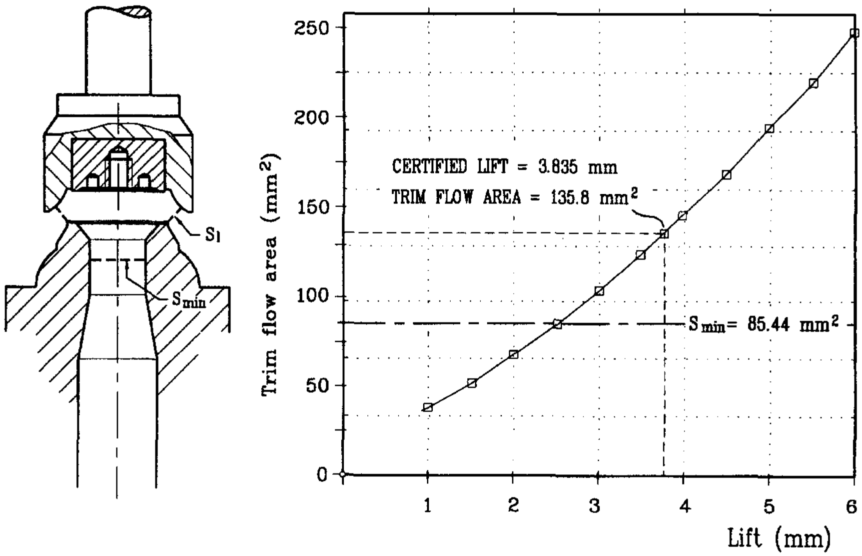

Fig. 2. Trim flow area versus lift.

recently chosen by the AIChE (DIERS) for emergency relief sizing design [1, 2]. It is based on the assumption that both liquid and vapour follow the saturation line during evaporation. The same idea was accepted in the ASME recommendations [3] which propose the theoretical curve based on HEM, giving the critical mass-flow rate only for saturated water. For two-phase inlet conditions, very conservative dry saturated vapour sizing methods are suggested.

The critical velocity for water versus void fraction arising from HEM is presented in Fig. 3. One can at once notice a big difference between its values for low void fractions and the speed of sound of pure liquid. As a consequence, for the high speed water flow, which appears in the safety valve nozzle, the flow is choked, according to the HEM, exactly in the same point where it meets saturation conditions. This enables further calculations of pressure and void fraction distribution. An assumption like this seems to be also rather far from reality.

It was observed by Reocreux [4] and in many other experiments that during fast evaporation the two-phase mixture does not follow the saturation conditions which results in overheating of the liquid phase. Taking this into account, the relaxation model [5], with developed correlation for the relaxation time [6], has been chosen. Its critical velocity tends to the speed of sound of pure liquid when void fraction goes to zero (Fig. 3). The relaxation model belongs to the family of nonequilibrium models represented also by the delayed equilibrium model [7]. These two models are based on fundamental thermodynamic equations of mass, momentum and energy conservation. No slip and pressure equilibrium between the phases are assumed. The main concern is on the constitutive equation that describes nonequilibrium mass exchange

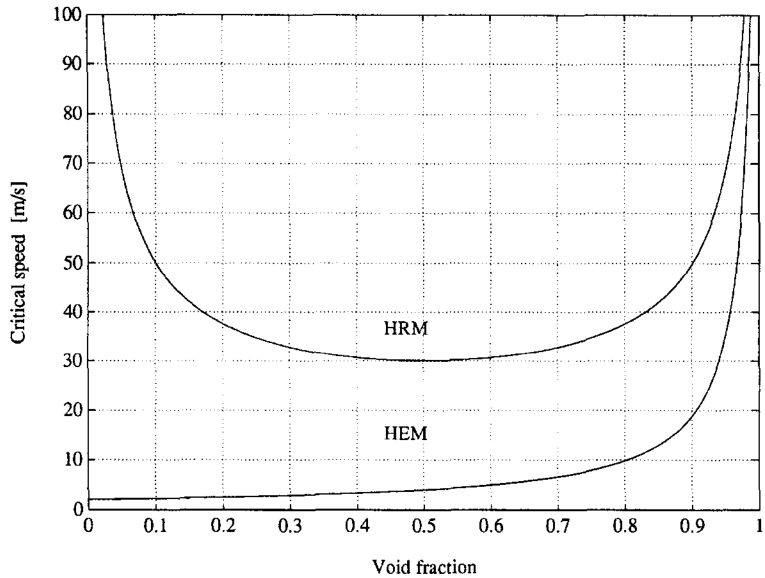

Fig. 3. The critical velocity for water ($P = 2$ bar) predicted by equilibrium (HEM) and relaxation (HRM) models for the saturated conditions.

between superheated liquid and saturated vapour. The concept of this equation makes the above-mentioned models different. Since they are rather simple and they do not require a lot of troublesome parameters, they may be easily applied for industrial calculations.

4. Results and calculations

A typical example of experimental results for a test with decreasing back pressure is presented in Figs. 4 and 5, reporting the time variation of inlet, nozzle and outlet pressures as well as the mass-flow rate for the flashing flow that is finally choked. The measurements were recorded every 5 s. Some instability of inlet pressure comes from manual regulation of the mass-flow rate of compressed air that maintains the pressure in the vessel. The decrease of back pressure was kept slow enough to be close to the steady-state flow regime. The occurrence of choking, manifesting itself by insensibility of mass-flow rate for the changes of back pressure, appears to be evident.

Figs. 6 and 7 show calculated pressure and void fraction distribution inside the valve model for the flow which is assumed to be choked in the nozzle (the nozzle shape is marked at the bottom of the figure). Calculations were performed using PIF (possible–impossible flow) procedure [8]. Fig. 6 gives the comparison between calculated and measured (lift 6 mm) pressure profiles.

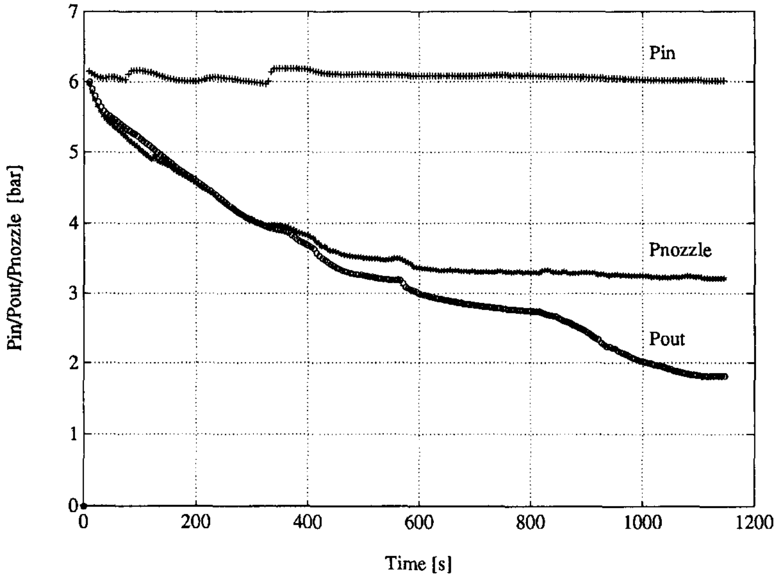

Fig. 4. Measured pressure variation with time for a test with decreasing back pressure (T_{in} = 150 °C).

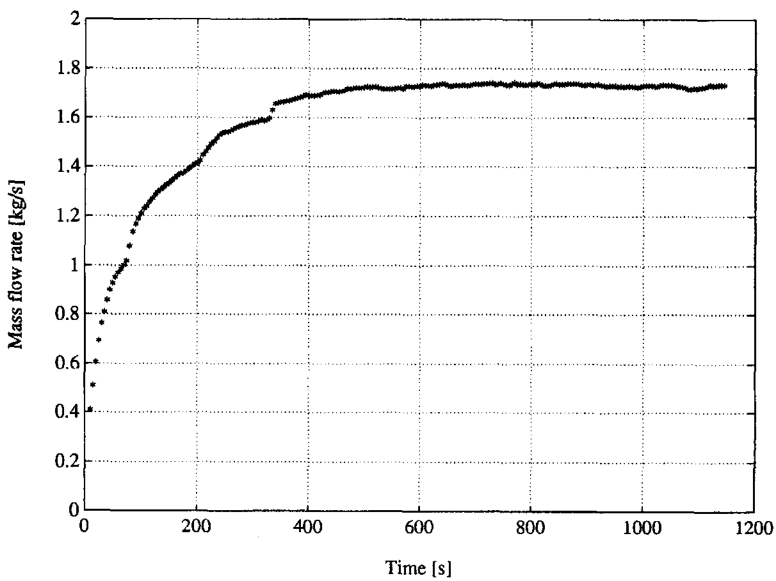

Fig. 5. Measured mass-flow rate variation with time for a test with decreasing back pressure.

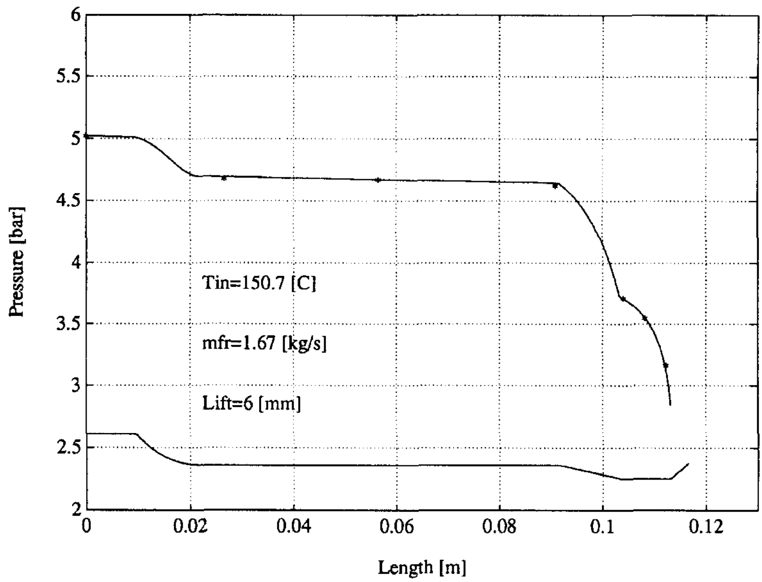

Fig. 6. Measured and calculated pressure profiles.

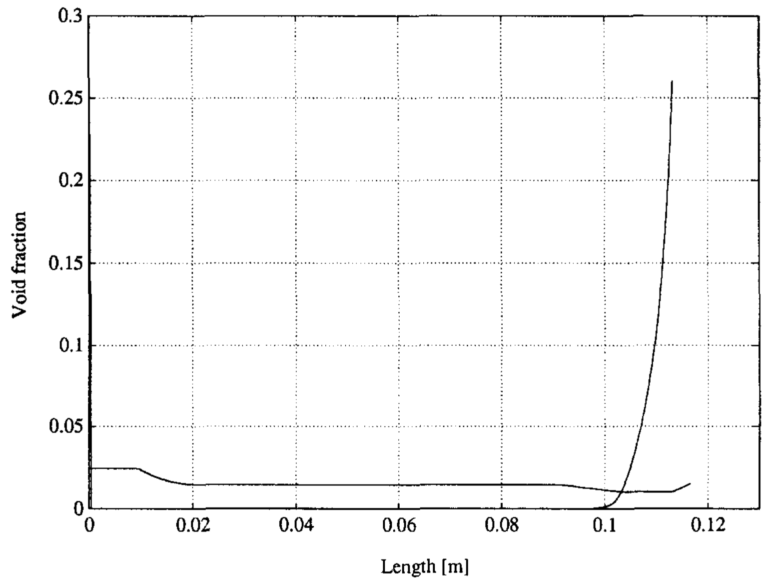

Fig. 7. Calculated void fraction profile.

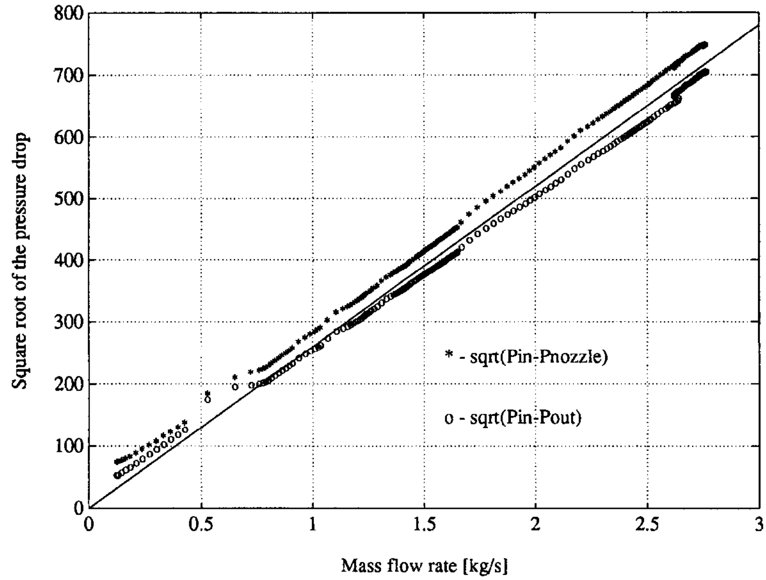

Fig. 8. Characteristics of the valve for cold water.

Fig. 8 presents the measured characteristics of the valve for the cold water ($T_{in} = 17.5\,°C$) with the lift 6 mm. Stars and circles represent square roots of the difference between inlet pressure (P_5) and respectively outlet (P_6) and nozzle (P_{23}) pressures. The straight line corresponds to the Bernoulli equation calculated for the nozzle ($d = 10.4$ mm). The comparatively good agreement between the valve characteristics for cold water and the Bernoulli equation comes from large mechanical energy dissipation downstream the valve nozzle. Thus, the pressure increase between the nozzle and the outlet of the valve is not high and it has the same order of magnitude as irreversible pressure losses in the nozzle itself. As a consequence, assuming frictionless flow and taking outlet pressure equal to the nozzle one, we can calculate the mass-flow rate that is close to its measured value.

The significant influence of flashing on the valve characteristics is demonstrated in Fig. 9, where we can find the comparison between cold ($T_{in} = 17.5\,°C$) and hot ($T_{in} = 150\,°C$) water flows. The occurrence of choking for hot water appears clearly, which essentially limits available mass-flow rate.

The measured values of the critical mass-flow rates for the lift 5 mm were exactly the same as that one for the disc removed. However, for the lift smaller than 5 mm the mass-flow rate started to be influenced by the disc even in this region, where the nozzle cross-section area remained the smallest one. Fig. 10 shows the influence of the lift on the valve characteristics for inlet conditions close to $P_{in} = 6$ bar, $T_{in} = 150\,°C$.

For a subcritical flow the mass-flow rate is a growing function of the pressure difference between inlet and outlet of a valve. The pressure drop inside the valve may

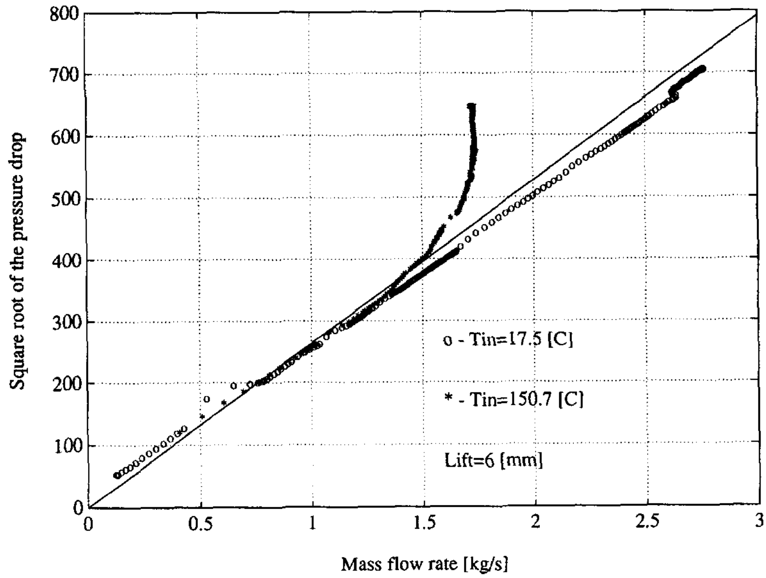

Fig. 9. Characteristics of the valve for cold and hot water.

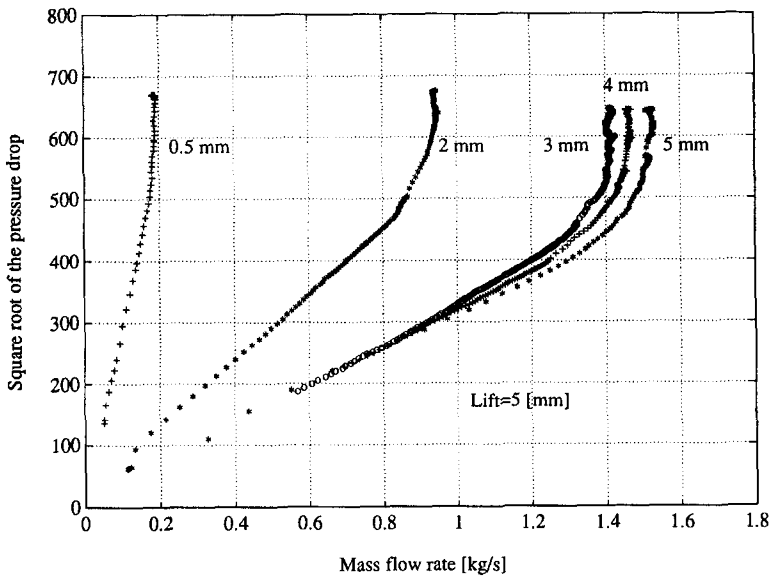

Fig. 10. Characteristics of the valve for different lifts.

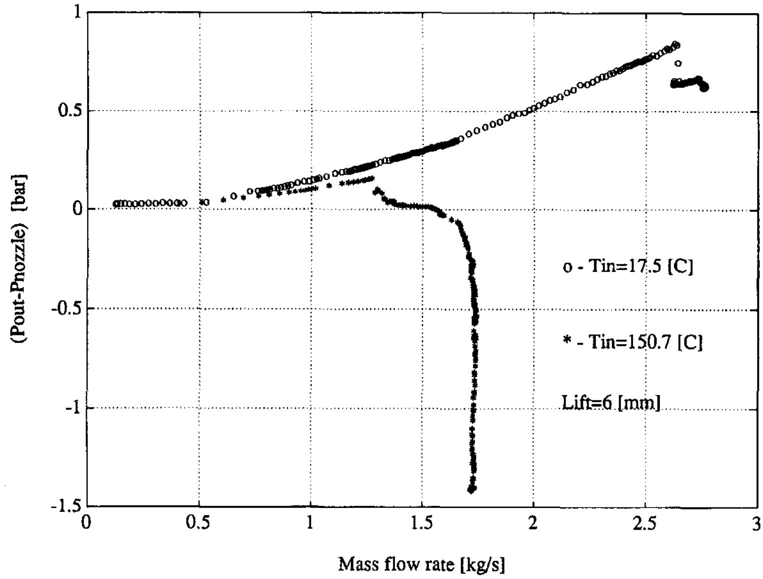

Fig. 11. The pressure difference between the nozzle and the valve outlet for cold and hot water.

be accurately calculated as long as the flow remains one-dimensional, that is up to the valve nozzle. The comparison of the pressure difference between the nozzle and the outlet of the valve, as a function of mass-flow rate, for both cold (T_{in} = 17.5 °C) and hot (T_{in} = 150 °C) water is given in Fig. 11. Some discontinuity that can be noticed for the cold water for large mass-flow rates is probably a result of cavitation. In this region the flow is three-dimensional and a strong dissipation of mechanical energy can be observed. The first attempt to calculate the total pressure drop between the inlet and the outlet of the safety valve for two-phase two-component flow was made by Friedel [9]. It may be interesting to adapt this model for the case of flashing flow.

For the critical flow the nozzle pressure remains constant and is not influenced by the outlet pressure (see Fig. 11). Table 1 shows the measured values of the critical mass-flow rate for different inlet conditions and for the lift 5 mm, so the critical cross-section is probably located in the nozzle. Following this assumption the PIF procedure has been used to calculate the critical mass-flow rate for the same inlet conditions by means of both relaxation (HRM) and equilibrium (HEM) models. The discrepancies in percent between measured and calculated values are also presented. It can be seen that the equilibrium model generally tends to underestimate the values of the critical mass-flow rates. Comparatively good agreement appears for only highly subcooled inlet water, when the assumption that the flow is choked when it reaches the saturation pressure could be accepted.

Table 1
A comparison of measured and calculated values of the critical mass-flow rates

No.	P_{in} (bar)	T_{in} (°C)	$T_{sat}(P_{in}) - T_{in}$ (°C)	mfr (kg/s)	ΔHRM (%)	ΔHEM (%)
1	6.14	149.2	10.05	1.77	− 0.3	− 19.7
2	6.08	150.7	8.66	1.67	1.1	− 22.6
3	5.60	149.7	6.47	1.61	− 4.0	− 32.4
4	5.02	149.9	2.10	1.31	− 0.9	− 53.3
5	5.78	150.0	7.39	1.52	0.4	− 22.6
6	5.56	148.4	7.49	1.50	0.8	− 22.0
7	5.62	149.3	7.00	1.50	0.2	− 24.0
8	5.28	149.5	4.41	1.36	0.2	− 34.3
9	4.05	138.5	5.59	1.23	− 1.1	− 27.1
10	4.70	137.6	11.94	1.46	3.9	− 7.3
11	4.68	135.6	13.78	1.50	4.2	− 4.1
12	5.59	120.4	35.70	2.21	1.7	1.3
13	6.10	120.0	39.49	2.35	1.7	1.9
14	4.05	119.7	24.39	1.75	0.5	− 2.4
15	5.12	119.7	33.64	2.12	0.7	0.4
16	5.72	119.1	37.88	2.27	1.5	1.5

5. Conclusions

The main objective of this paper is to provide new experimental results, reporting the pressure distribution inside the valve, and to compare them with some existing and developed calculational methods. The Bernoulli equation, calculated for the nozzle cross-section area, may be a good estimation for the cold (nonflashing) water flow. The relaxation model seems to predict with a good accuracy both the pressure profile along the nozzle and the critical mass-flow rate. So, it appears to be adequate to represent the flashing flow for high lifts, when the flow is choked in the injection nozzle. The equilibrium model tends to underestimate the values of critical mass-flow rates for more than 20%, when the inlet subcooling is not large.

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References

[1] J.E. Huff, Plant/Oper. Prog., 4 (1985) 191.
[2] H.G. Fisher, Plant/Oper. Prog., 10 (1991) 1.

[3] ASME Boiler and Pressure Vessel Code, Appendix 11: Capacity Conversions for Safety Valves, American Society of Mechanical Engineers, New York, 1986, p. 610.
[4] M. Reocreux, Contribution a l'etude des debits critiques en ecoulement dipdasique eauvapeur, Ph.D. Thesis, Université Scientifique et Medicale de Grenoble, 1974.
[5] Z. Bilicki and J. Kestin, Proc. Roy. Soc. London Ser. A, 482 (1990) 379.
[6] P. Downar-Zapolski, An influence of the thermodynamic disequilibrium on the pseudo-critical flow of one-component two-phase mixture, Ph.D. Thesis, Institute for Fluid Flow Machinery, Gdansk, 1993.
[7] V. Feburie, M. Giot, S. Granger and J.M. Seynhaeve, Int. J. Multiphase Flow, 19 (1993) 541.
[8] F. Yan and M. Giot, in: X.J. Chen et al. (Eds.), Multiphase Flow and Heat Transfer, Hemisphere, Washington, DC, 1991.
[9] L. Friedel, Two-Phase Pressure Drop Model for Safety Valves, Eur. Two-Phase Flow Group Meeting, Trondheim, 1987.