

Sizing of Throttling Device for Gas/Liquid Two-Phase Flow

Part 1: Safety Valves

Ralf Diener^a and Jürgen Schmidt^b

^a BASF AG, Inorganic Chemicals Europe, Ludwigshafen, Germany

^b BASF AG, Safety Engineering, Ludwigshafen, Germany; juergen.schmidt@onlinehome.de (for correspondence)

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The calculation of the mass flow rate through throttling devices is difficult when handling two-phase flow, especially when boiling liquids flow into these fittings. Safety valves are typically oversized by a significant extent, if sizing methods like the ω -method (originally developed by J. Leung), are used in case of low-quality inlet flow. Within this method the boiling delay of the liquid and the influence of the boiling delay on the mass flow rate are not considered. In this paper the HNE-DS model is proposed, where the compressibility coefficient ω is extended by adding a boiling delay coefficient. It includes the degree of thermodynamic nonequilibrium at the start of the nucleation of small mass fractions of vapor upstream of the fitting.

In Part 1 the sizing of safety valves is described. Additionally, the derivation of the HNE-DS method is given in detail. Part 2 considers the mass flow rate through short nozzles, orifices, and control valves. The HNE-DS model can be used for all those fittings. A comparison with experimental results on safety valves with steam/water and air/water flow has emphasized the excellent accuracy of the new model. © 2004 American Institute of Chemical Engineers Process Saf Prog 23: 335–344, 2004

INTRODUCTION

The sizing of safety valves for the flow of gases, noncondensing vapor, and nonvaporizing liquids is described in the standard EN-ISO 4126 [1]. In the future this European law will replace the national standards, such as the German AD-2000 Merkblatt A2 [2]. In the United States the API RP 520 standard is still predominantly used and will be adapted. The current standards are sufficiently accurate for the sizing of safety valves

for single-phase flow, although they contain no reliable recommendations for two-phase mixtures composed of vapor and liquid. At present there is no appropriate standard either nationally or internationally.

In the chemical and petrochemical industries, and also in power plants and offshore facilities, liquids are often pumped from tanks or pipelines into parts of plants having relatively low pressures. In doing so the volume flow rate is controlled by means of a control valve (see Figure 1). The plant under low pressure must, as a general rule, be protected by a safety valve. In practice the flow rate through the control valve is often limited for safety reasons by an additional orifice, which is fitted downstream from the control valve. The design engineer is thus confronted with the task of estimating the flow rate through the control valve or orifice to determine the size of the safety valve on the low-pressure equipment (Figure 1). This sizing task is divided into two steps:

1. Sizing a relief valve for two-phase flow (Part 1)
2. Sizing a control valve or orifice for two-phase flow (Part 2)

Both steps are based on the same method: the HNE-DS (homogeneous nonequilibrium) method developed by the authors Diener and Schmidt). It is an extended ω -model using the well-known Design Institute for Emergency Relief Systems (DIERS) methodology. In this article the HNE-DS method is described for sizing safety valves.

LITERATURE REVIEW

In larger companies numerical methods are used to determine the mass flow rate discharged through safety valves. Several computer codes, such as Safire, Relief, and AspenDynamics, or in-house products such as REACTOR

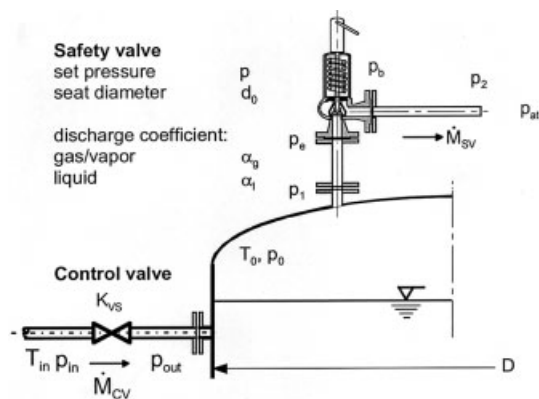


Figure 1. Typical layout of production vessel fed by a control valve with a safety valve on top to avoid an inadmissible vessel overpressure. The size of the safety valve is determined by the maximum feed through the control valve.

from BASF, are used. To apply these computer codes very special expertise is needed and the calculations are complex and expensive. In addition, physical properties data such as densities, viscosities, enthalpies, and entropies of the vapor and liquid phases as a function of pressure and temperature over a broad range are needed. In many industrial applications these are not available or must first be determined experimentally.

In a review paper on the sizing of safety valves for two-phase flow, Schmidt and Westphal [3, 4] recommended a method including a two-phase discharge coefficient and the ω -method developed by Leung [5, 6]. This sizing method is simple and only requires the physical properties at the stagnation point. In contrast with the homogeneous equilibrium model only the densities of the two phases, the specific heat capacity of the liquid and its enthalpy of vaporization are needed. These are available for almost all fluids. A key disadvantage with the ω -method, however, is that predictions of the mass flow rate are too low, when a vapor-liquid flow has a low vapor content. If, for example, this method is used to estimate the mass flow rate of a boiling liquid through the control valve or the orifice, the mass flow rate that actually occurs can be almost an order of magnitude as high as that calculated. Any safety valve sized on the basis of this mass flow rate would then be seriously undersized. On the other hand, the safety valve would be substantially oversized if the required flow rate is known, from measurements for example, and the ω -method is used for sizing the safety valve.

A working group of the Technical Committee 185 of the ISO (ISO TC185/WG1 "Flashing Liquids in Safety Devices") is currently working on an international standard for the sizing of safety valves for gas/vapor-liquid flow (Part 10 of ISO 4126). The publications by Schmidt and Westphal [3, 4] are being used as the basis for the standard. The sizing method originally developed by J. Leung is extended for low mass flow qualities at the entrance of the safety valve [7]. The draft standard was presented and published at the "Loss Prevention in the

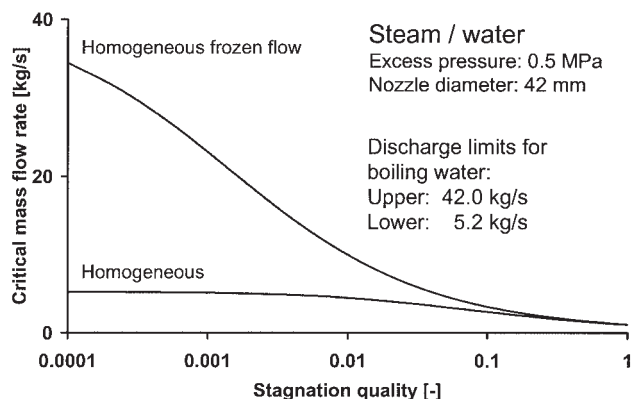


Figure 2. Discharge limits of a homogeneous steam/water flow.

Process Industries" Symposium in Stockholm in June 2001 [8]. The method in this ISO standard is called the HNE-DS method and is described below.

UNCERTAINTIES OF THE ω -METHOD

The effect of hydrodynamic nonequilibrium and nonvaporizing flows, such as air/water, may lead to uncertainties in the mass flow calculation of up to 50%. Nevertheless, it is generally much less pronounced than the effect of thermodynamic nonequilibrium in vaporizing fluids. Vaporization initiates almost spontaneously after the liquid has been superheated by the rapid fall in pressure down into the narrowest cross section. The temperature of the liquid can adapt to the drop in pressure attributed to evaporation only after a certain delay. As an example, Figure 2 shows the expansion of boiling water from a tank having an excess pressure of 0.5 MPa (5 bar) through a nozzle having a diameter of 42 mm. The water is released into the atmosphere. The mass flow rate through the nozzle has been calculated by the homogeneous equilibrium model (HEM, model for the lower limit of the flow rate) and the homogeneous frozen flow model (model for the upper flow rate limit), in which the stagnation mass fraction of vapor was varied over the entire range of two-phase flow. In frozen flow (that is, maximum thermodynamic nonequilibrium flows), the calculated mass flow rate is almost an order of magnitude greater than the flow rate calculated for expansion from thermodynamic equilibrium. The difference is greatest for a boiling liquid or a liquid-vapor mixture having very low vapor content. It decreases very rapidly, however, as the stagnation vapor mass content increases. When the stagnation vapor mass flow quality exceeds a value of approximately 15% the differences are negligible small. In that case, the void fraction is much higher than 90% in depressurization systems typically encountered in industry.

SIZING OF SAFETY VALVES

In general, the mass flow rate through safety valves may be calculated by

Table 1. Determination of mass flux for frictionless flow through an adiabatic throttling device (such as nozzle, orifice, control valve, safety valve).

State variables and property data	$p_0, T_0, p_b, \Delta h_{v,0}, cp_{l,0}, v_{g,0}, v_{l,0}$	
Pressure ratios	$\eta_b = \frac{p_b}{p_0} \quad \eta_{crit} = \frac{p_{crit}}{p_0} \quad \eta = \frac{p_{VC}}{p_0}$	(3)
Homogeneous specific volume of mixture	$v_0 = \dot{x}_0 v_{g,0} + (1 - \dot{x}_0) v_{l,0}$	(4)
Compressibility factor (equilibrium condition, $N = 1$)	$\omega = \omega_{N=1} = \frac{\dot{x}_0 v_{g,0}}{v_0} + \frac{cp_{l,0} T_0 p_0}{v_0} \left(\frac{v_{g,0} - v_{l,0}}{\Delta h_{v,0}} \right)^2$	(5)
Critical pressure ratio	$\eta_{crit} = 0.55 + 0.217 \ln \omega - 0.046(\ln \omega)^2 + 0.004(\ln \omega)^3$ $\eta_{crit}^2 + (\omega^2 - 2\omega)(1 - \eta_{crit})^2 + 2\omega^2 \ln(\eta_{crit}) + 2\omega^2(1 - \eta_{crit}) = 0$	(6)
Compressibility factor (nonequilibrium condition, $N \leq 1$)	$\omega = \frac{\dot{x}_0 v_{g,0}}{v_0} + cp_{l,0} T_0 p_0 \left(\frac{v_{g,0} - v_{l,0}}{\Delta h_{v,0}} \right)^2 N$	(7a)
	$N = \left[\dot{x}_0 + cp_{l,0} T_0 p_0 \left(\frac{v_{g,0} - v_{l,0}}{\Delta h_{v,0}} \right) \ln \left(\frac{1}{\eta_{crit}} \right) \right]^a$	(7b)
	$a = 3/5$ orifices, control valves, short nozzles (see Part 2) $a = 2/5$ safety valves, control valve (high lift) $a \approx 0$ long nozzles, orifice with large area ratios	
Critical pressure ratio	$\eta_{crit} = 0.55 + 0.217 \ln \omega - 0.046(\ln \omega)^2 + 0.004(\ln \omega)^3$ $\eta_{crit}^2 + (\omega^2 - 2\omega)(1 - \eta_{crit})^2 + 2\omega^2 \ln(\eta_{crit}) + 2\omega^2(1 - \eta_{crit}) = 0$	(8)
Outflow function	$\eta_b \leq \eta_{crit} \Rightarrow \eta = \eta_{crit}$ $\eta_b > \eta_{crit} \Rightarrow \eta = \eta_b$	(9)
	$\psi = \frac{\sqrt{\omega \ln \left(\frac{1}{\eta} \right) - (\omega - 1)(1 - \eta)}}{\left[\omega \left(\frac{1}{\eta} - 1 \right) + 1 \right]}$	
Mass flux for isentropic frictionless flow	$\dot{m}_{id} = \psi \sqrt{\frac{2p_0}{v_0}}$	(10)

Table 2. Calculation of the required relief area of safety valves.

Data from Table 1	$v_{l,0}, v_0, \omega, \dot{m}_{id}, \eta = \eta_{crit} \text{ or } \eta_b$	
Discharge coefficients	$\alpha_b \alpha_g$	
Mean void fraction	$\varepsilon = \frac{V_g}{V_g + V_l} = 1 - \frac{v_{l,0}}{\omega v_0 \left(\frac{1}{\eta} - 1 \right) + 1}$	(11)
Discharge coefficient	$\alpha_{SV} = \varepsilon \alpha_g + (1 - \varepsilon) \alpha_l$	(12)
Required cross-sectional area	$A_{req} = \frac{\dot{M}_{SV}}{\alpha_{SV} \cdot \dot{m}_{id}} = \frac{\dot{M}_{CV/orif}}{\alpha_{SV} \cdot \dot{m}_{id}}$	(13)

$$\dot{M}_{SV} = \Gamma_{corr} A_0 \dot{m}_{id} \quad \text{with} \quad \Gamma_{corr} = \Phi \alpha_{SV} / S \quad (1)$$

where A_0 is the seat area of the fitting, $\dot{m}_{id}$ is the mass flux through an ideal nozzle, α_{SV} is the discharge coefficient of the safety valve, and S is a safety factor (recommended values 1–1.3). Although a slip correction factor ϕ was derived for control valves and orifices, it is not used here

because of the lack of data to precisely determine it for safety valves ($\phi = 1$).

In Table 1 the calculation procedure for the mass flow rate through an ideal nozzle $\dot{m}_{id}$ is summarized. Details of the derivation of the HNE-DS method are given below in the Appendix.

In practice the sizing scenario for the safety valve is

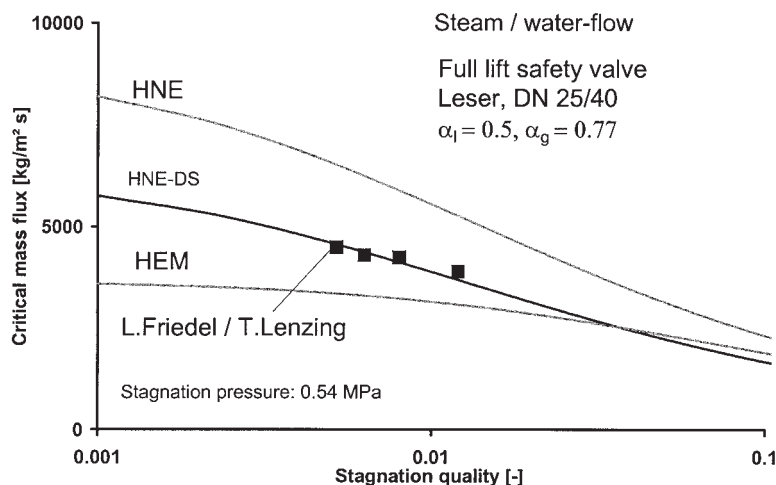


Figure 3. Comparison of measurements with calculated results from the HEM model, the HNE-DS method, and the Leung model.

often a blocked outlet line of the pressurized vessel to be protected. Rearranging Eq. 1 and taking into account that the mass flow rate through the safety valve at the opening conditions must be larger than or equal to the mass flow rate through the control valve or orifice at the maximum permissible inlet pressure, the minimum required seat area of a safety valve is to be calculated by

$$A_0 \geq A_{req} = \frac{\dot{M}_{SV}}{\alpha_{SV} \cdot m_{id}} \quad \text{with} \quad \dot{M}_{SV} = \dot{M}_{cv/orif} \quad (2)$$

The equations for determining the relief area for safety valves are presented in Table 2. Most of the required input data are defined in Table 1. In case of choked flow in the safety valve (most often in practice) the critical pressure ratio η_{crit} is used. The discharge coefficients for pure gas (α_g) and liquid flow (α_l) are generally determined experimentally. As a general rule, valve manufacturers specify those measured values in company catalogs. The mean volumetric vapor content in the narrowest flow cross section is then determined (Eq. 11), based on the stagnation conditions upstream of the safety valve. It is multiplied by the discharge coefficient for two-phase flow α_{SV} (Eq. 12), to calculate the minimum required cross-sectional area of the safety valve (Eq. 13).

There are several methods available in the literature to calculate the discharge coefficient for two-phase flow. Leung [9] investigated the effect of compressibility on the discharge coefficient for the flow of gases through orifices and proposed a compressibility correction based on the force defect model of Jobbson [10]. A constant value of 0.975 is recommended by the API. Generally, the discharge coefficient contains an adjustment for the flow through a safety valve in comparison to an ideal nozzle flow and the uncertainties of the nozzle flow model, such as real gas effects or the dependency on the viscosity of the mixture. Hence, it must be measured. A constant value is not recommended.

Typical safety valves from American and European valve manufactures differ in such specifications as in the height of the valve lift and the contour of the inlet nozzle. As a result, the discharge coefficients are quite different. At elevated pressures, the choking area can move from the valve seat to a location below the valve disc. This can lead to a dramatic change in discharge coefficient. Furthermore, typical high-pressure valves are often built with conical discs and, thus, the discharge coefficient can be much lower than that typically encountered for liquid flow at lower pressures. Consequently, the discharge coefficient greatly depends on the valve geometry and cannot be calculated theoretically. Even a change in valve size can lead to a modification in the discharge coefficient because the valves are in general not built geometrically similar—manufacturers try to decrease the valve body with increasing valve size to lower the manufacturing cost.

Within BASF the compressibility effect on the discharge coefficient was intensively studied. Measurements were made on valves with different inlet pressures ranging from 0.4 to 30 MPa (4 to 300 bar). The discharge coefficient did not change significantly with the inlet pressure for the same type of valve. On the other hand, the liquid discharge coefficient is generally lower than the gas discharge coefficient for the same inlet conditions and the same valve. The contraction of the flow must, therefore, be density dependent. It can also be flow pattern dependent, as was experimentally shown for pipe contractions [11]. At high mass flow qualities the vena contracta vanishes and is again formed when pure gas flow is reached. Comparable data for safety valves are not given. In the absence of more specific data, Schmidt and Westphal [3, 4] proposed a linear weighting factor with the void fraction based on both, the measured discharge coefficients of gas and liquid flow. They considered the two-phase mixture as a single-phase fluid with the two-phase homogeneous density and assumed a continuous

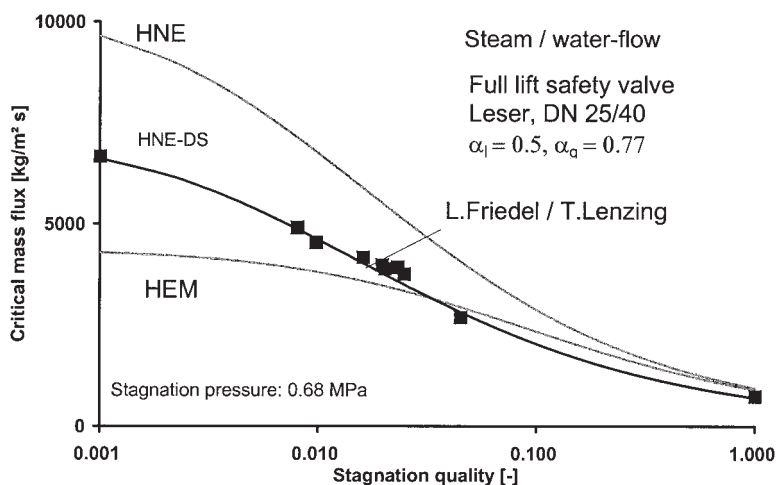


Figure 4. Comparison of measurements with calculated results from the HEM model, the HNE-DS method, and the HNE model of Leung.

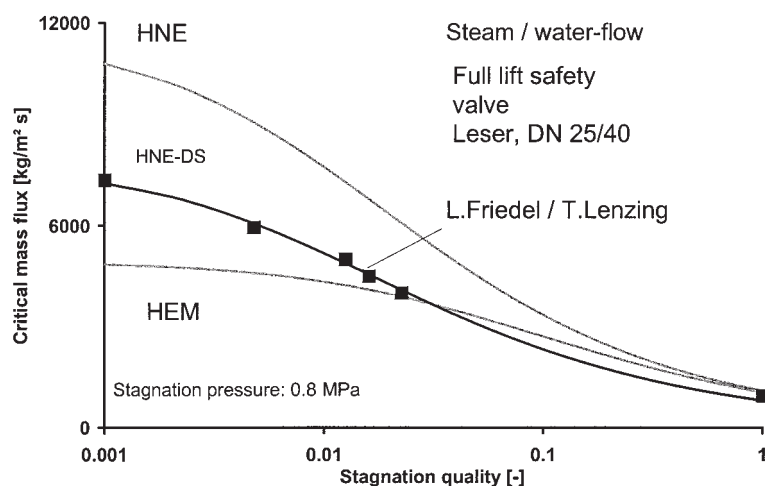


Figure 5. Comparison of measurements with calculated results from the HEM model, the HNE-DS method, and the Leung model.

change of the streamlines between inlet and narrowest cross section starting with a single-phase liquid and increasing the void fraction up to pure gas flow.

In the case of viscous liquid flow, it is recommended as a first estimate to include the viscosity correction factor of Wiecezorek [12, 13] or of Darby and Molavi [14] into the discharge coefficient:

$$\alpha = \varepsilon \alpha_g + (1 - \varepsilon) \alpha_l K_\eta \quad (14)$$

At liquid viscosities less than 100 mPa·s the viscosity correction K_η is almost equal to 1.

Overall, the combination of a measurement based discharge coefficient and a frictionless mass flow rate through an adiabatic nozzle, including boiling delay and slip correction, fit into the HNE-DS method.

ACCURACY OF THE HNE-DS METHOD

In contrast with the ω -method, the HNE-DS method more generally determines accurate sizing of safety valves. In Figures 3–6 calculated critical mass flux in the narrowest cross section of a full-lift safety valve from the Leser company, and corresponding values measured by Lenzing and Friedel [15,17], are plotted against stagnation vapor mass flow qualities for set pressures of 0.54 MPa (5.4 bar), 0.68 MPa (6.8 bar), 0.8 MPa (8 bar), and 1.06 MPa (10.6 bar). The data were determined with a two-phase steam and water flow through the valve.

In general, the HNE-DS model fits the measured data very well, irrespective of the stagnation quality and the set pressure of the valve. The overall trend of critical mass flux vs. stagnation pressure follows exactly the measured trend, starting at almost boiling liquid up to pure vapor

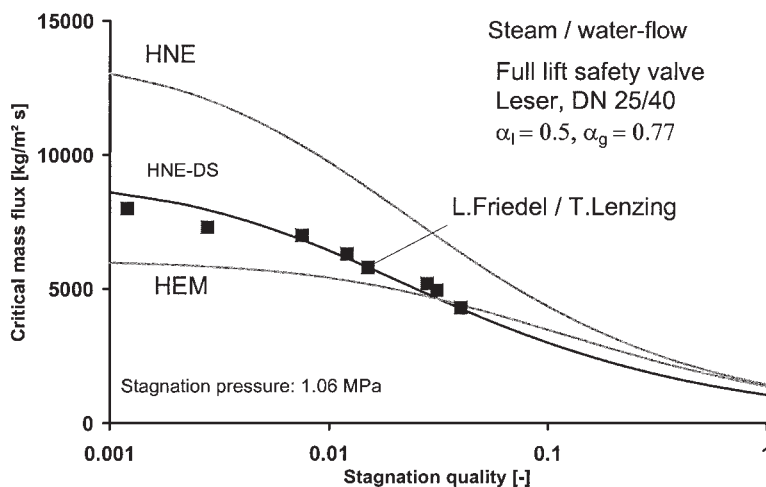


Figure 6. Comparison of measurements with calculated results from the HEM model, the HNE-DS method, and the Leung model.

flow. Compared with the HNE-DS model, the results of the HEM method (homogeneous equilibrium model [9]) are significantly lower. In contrast, the results calculated with the proposed HNE model from Leung are significantly larger. The overall trend of both the HEM and the HNE Leung method are not in agreement with the measurements. This is because these two methods use a constant discharge coefficient of 0.975 and 1, respectively. The measured discharge coefficient given by the valve manufacturer for pure vapor flow is 0.77.

The accuracy of the HNE-DS model is even better for nonflashing air/water flow through the same safety valve.

Overall, the recommended HNE-DS method is more precise than the original ω -method and easier to handle than the method of Henry and Fauske: in the Henry-Fauske method, several equations have to be solved simultaneously and detailed property data are necessary over the whole range of temperature and pressure down to the hydrodynamic critical pressure. In particular, entropy data are not readily available for many chemical substances. The recommended HNE-DS method is based on stagnation properties only and does not need any iteration.

SUMMARY

The well-known and easy-to-use ω -method developed by Leung for the sizing of safety valves for two-phase flow is extended by a term to take account of boiling delay (thermodynamic nonequilibrium). The HNE-DS method — an extended ω -method — is just as easy to use as that developed by Leung, and the HNE-DS method requires physical properties only at the stagnation condition. Resource-intensive equations of state and derivations of physical property functions are not needed; nor, as a rule, are iterations necessary. Only in the case of very low compressibility factors ($\omega < 2$) is it advisable to determine the critical pressure ratio by means of the implicit equation.

The accuracy of the model has been checked against

measurements for safety valves with steam/water and air/water flow. Only the discharge coefficients for pure gas and pure liquid flow from the valve manufacturers' catalogs have been used to correct the ideal nozzle mass flow rate. The HNE-DS model provides excellent results even at very low mass flow qualities at the valve inlet.

A major advantage of the HNE-DS model is the use of the same method not only for safety valves but also for control valves, orifices, and nozzles. This will be described in the companion paper (Part 2).

NOMENCLATURE

- α_g = discharge coefficient for single-phase vapor/gas flow
- α_l = discharge coefficient for single-phase liquid flow
- α_{SV} = discharge coefficient for two phase flow
- a = boiling delay exponent
- A_{req} = minimum required cross-sectional area of the safety valve for a defined mass flow rate
- $cp_{l,0}$ = specific heat capacity of the liquid at stagnation state
- $cp_{g,0}$ = specific heat capacity of the vapor/gas at stagnation state
- d_0 = safety valve seat diameter
- ϵ = void fraction in the narrowest flow cross section, e.g., valve seal area
- η = pressure ratio (ratio of real pressure in the narrowest flow cross section and the sizing pressure p_0)
- η_b = back pressure ratio (ratio of back pressure and the sizing pressure p_0)
- η_{crit} = critical pressure ratio (ratio of critical pressure in the narrowest flow cross section and the sizing pressure p_0)
- $\Delta h_{v,0}$ = latent heat of vaporization at stagnation state
- ψ = outflow function

- $\dot{m}_{id}$ = mass flux through an adiabatic frictionless nozzle
 $\dot{M}_{SV}$ = mass flow rate to be discharged from the pressurized system
 $\dot{M}_{CV/orif}$ = maximum permissible mass flow rate through control valve or orifice at vessel entrance (see Figure 1)
 N = boiling delay factor
 p_0 = stagnation or sizing pressure (the pressure at which all property data, especially the compressibility factor ω , is calculated for sizing the throttling device, i.e. up to 110% of the design pressure of the vessel (according to German pressure vessel regulation))
 p_b = back pressure of the safety valve (the pressure that exists at the outlet of a throttling device as a result of pressure in the discharge system)
 p_{crit} = critical pressure (fluid-dynamic pressure occurring in the narrowest flow cross section of the throttling device when the pressure behind this section is decreased to such an extent that a maximum mass flow rate is reached at a given stagnation state in the pressurized system)
 S = safety margin (values of 1 – 1.3 are typically used)
 T_0 = stagnation temperature
 $\dot{x}$ = mass flow quality
 $\dot{x}_0$ = stagnation or sizing mass flow quality, that is, the ratio of the gas mass flow rate to the total mass flow rate of a two-phase mixture at stagnation state
 $\Delta\dot{x}_e$ = changing of the mass flow quality at homogeneous equilibrium flow
 $\dot{x}_e$ = mass flow quality at homogeneous equilibrium flow
 V = volume of two-phase flow
 V_g = gas volume
 V_l = liquid volume
 $v_{g,0}$ = specific gas volume at stagnation state
 v = mixture-specific volume for homogeneous two-phase flow
 v_0 = mixture-specific volume at stagnation state
 $v_{l,0}$ = specific liquid volume at stagnation state
 ω = compressibility factor

APPENDIX: DEVELOPMENT OF THE HNE-DS MODEL

To extend the ω -method developed by Leung, the compressibility factor ω is described for two-phase flow with an evaporating liquid as well as for frozen flow, that is, for flow without phase transition (see Leung [6]). From the definition of the critical mass flow rate in any flow cross section, in accordance with the law of conservation of momentum, this yields for isentropic flow:

$$\dot{m}_{crit} = \frac{-1}{\sqrt{\left(\frac{dv}{dp}\right)_s}} \quad (A1)$$

where v is the mixture-specific volume for homogeneous two-phase flow, which is derived with respect to pressure as follows:

$$v = \dot{x}v_g + (1 - \dot{x})v_l \quad \frac{dv}{dp} = \dot{x} \frac{dv_g}{dp} + (v_g - v_l) \frac{d\dot{x}}{dp} \quad (A2)$$

The first term in the equation characterizes the expansion of the vapor as a result of the change in pressure and the second term corresponds to the change in vapor volume as a result of the evaporation of liquid. Delayed boiling of the liquid must be taken into account using the second term. This becomes smaller with increasing thermodynamic nonequilibrium and assumes the value of zero in frozen flow. By analogy with the procedure developed by Henry and Fauske, delayed boiling is reflected by the boiling delay factor N :

$$\frac{d\dot{x}}{dp} = \frac{d\dot{x}_e}{dp} N \quad (A3)$$

where $d\dot{x}_e/dp$ is the rate of evaporation at thermodynamic equilibrium (without boiling delay) and $d\dot{x}/dp$ is the actual rate of evaporation. As a consequence, the boiling delay factor N is equal to 1 in equilibrium flow and zero by definition in a frozen flow:

$$\begin{aligned} \lim_{\Delta\dot{x} \rightarrow \Delta\dot{x}_e} N &= 1 \text{ (equilibrium flow)} \\ \lim_{\Delta\dot{x} \rightarrow 0} N &= 0 \text{ (frozen flow)} \end{aligned} \quad (A4)$$

Using the law of conservation of energy at thermodynamic equilibrium conditions,

$$\frac{d\dot{x}_e}{dT} = - \frac{cp_l}{\Delta h_v} \quad (A5)$$

the Clausius–Clapeyron equation for a single component system in thermodynamic equilibrium,

$$\frac{dT}{dp} = \frac{(v_g - v_l)}{\Delta h_v} T \quad (A6)$$

and Eq. A3, Eq. A2 can be rearranged to

$$\frac{dv}{dp} = \dot{x} \frac{dv_g}{dp} - Ncp_l T \left(\frac{v_g - v_l}{\Delta h_v} \right)^2 \quad (A7)$$

Equation A7 is valid for flashing two-phase flow. In the case of a (nonflashing) frozen flow the vapor quality $\dot{x}$ remains constant and, accordingly, the term $d\dot{x}/dp$ in Eq. A2 becomes zero. Hence, the second term in Eq. A7 would vanish. To determine the specific volume of the gas phase v_g the vapor expansion is assumed to follow an isothermal change of state, which is usually a good approximation for a two-phase flow. Then, the

ideal gas law is used to calculate the gas specific volume (real gas effects are not taken into consideration), whereas the specific volume of the liquid is neglected by comparison to that of the vapor. Additionally, the enthalpy of vaporization Δh_v is described by a constant average value. These assumptions are not valid close to the thermodynamic critical point, where the compressibility factor has only a limited validity.

$$\int_{v_0}^v dv = p_0 v_{g,0} \int_{p_0}^p - \frac{\dot{x} dp}{p^2} - N$$

$$cp_l T_0 p_0^2 \left(\frac{v_{g,0} - v_{l,0}}{\Delta h_{v,0}} \right)^2 \int_{p_0}^p - \frac{dp}{p^2} \quad (\text{A8})$$

In principle the second term on the right-hand side is dominant, except in the case where the vapor quality is almost equal to one. At large vapor qualities, however, the vapor quality varies to only an insignificant extent with pressure. Accordingly, in the first term of Eq. A8 the vapor quality may be taken to be constant and be replaced by the stagnation vapor mass flow quality $\dot{x}_0$ without this producing any appreciable error in the calculation. Rearranging the integrated Eq. A8 leads to an extended compressibility factor ω , as follows:

$$\omega = \frac{\frac{v}{v_0} - 1}{\frac{p_0}{p} - 1} = \frac{\dot{x}_0 v_{g,0}}{v_0} + \frac{cp_l T_0 p_0}{v_0} \left(\frac{v_{g,0} - v_{l,0}}{\Delta h_{v,0}} \right)^2 N \quad (\text{A9})$$

which differs by comparison with the original ω only by the boiling delay factor N .

In the case of flashing two-phase flow the magnitude of the ω -parameter and thus the mass flow rate is mainly determined by the boiling delay factor N . From a physical perspective, the boiling delay factor N is defined between 0 (thermodynamic equilibrium) and 1 (total boiling delay without vaporization of the liquid). To determine the boiling delay factor N , the nucleation of bubbles, the bubble growth, and the bubble distribution within the liquid must be specified, depending on the flow conditions for different geometries of the throttling device. Friction losses, vortex detachments, and local flow contractions are device specific and have to be taken into consideration. Additionally, the boiling delay depends on the physical property data of the fluid. In the literature extensive measurements can be found regarding this topic. For industrial applications the detailed modeling of the nonequilibrium vaporization is currently not possible. Further investigations are necessary.

From a less rigorous perspective, the main parameters for determining the boiling delay are identified as the inlet mass flow quality, the pressure drop, and the relaxation time between inlet and narrowest cross section. Depending on the relaxation time, both phases may be in total nonequilibrium or reach the equilibrium state during the flow up to the narrowest cross

section. In typical nozzles and orifices with a small area ratio (as well as control valves) the depressurization is very fast within a very short length of the flow path. There is not sufficient time for heat transfer between both phases. Hence, a large nonequilibrium is expected. In contrast, the flow stream up to the seat area in typical safety valves contracts only moderately. As a consequence, a less-pronounced boiling delay is expected. If a straight pipe is considered, the flow would not contract; and because of the frictional pressure drop, quite a long section is needed to reach the critical pressure. Enough time for heat exchange between vapor and liquid is the result that leads to almost thermodynamic equilibrium conditions (or a boiling delay factor of 1). Compared to those industrial applications it might be possible to widely suppress the nucleation of bubbles by the evacuation of the liquid and with very smooth pipe walls. A flow of two phases in a highly nonequilibrium system will be the result of this "laboratory" flow condition. The boiling delay factor reaches its minimum value close to 0. The larger the relaxation time up to the narrowest cross section, the less pronounced will be the nonequilibrium effect and, thus, the larger will be the boiling delay factor.

A measure for the rate of vaporization with a certain depressurization rate is the change in mass flow quality between inlet and narrowest cross section. In a long nozzle it will be equal to the change of mass flow quality in equilibrium flow. Thus, the boiling delay factor may be defined as

$$N = \frac{d\dot{x}}{d\dot{x}_e} \quad (\text{A10})$$

For the special case of short throttling devices with large depressurization rates, the acceleration pressure drop dominates the frictional pressure change. Here, even the equilibrium mass flow quality in the narrowest cross section is a measure of the boiling delay shown by Henry and Fauske [16] in comparison to measurements.

In accordance with experimental experience from more than 1300 data of valves and orifices a power law, Eq. A11, is recommended for the boiling delay factor. The power-law exponent a must be less than 1, to account for the disproportional increase in the phase boundary surface area with increasing mass flow quality, which impedes the heat transfer into the liquid at rapid depressurization:

$$N = [\dot{x}_e(p_{crit})]^a \quad (\text{A11})$$

Equation A11 allows the integration of Eq. A10 by taking both limiting flow models—the thermodynamic equilibrium flow model and the frozen flow model—into consideration (see Eq. A4). With the new definition of N , the boiling delay factor is between the values of 0 and 1 for the whole mass flow quality range between pure liquid and pure gas.

In industrial applications, in general, choking in the narrowest flow cross section must be taken into consideration. The vapor mass flow quality at a critical

pressure ratio is obtained from the sum of the vapor quality at stagnation conditions $\dot{x}_0$ and the increase in vapor content attributed to evaporation during the expansion process $\Delta\dot{x}_e(p_{crit})$:

$$\dot{x}_e(p_{crit}) = \dot{x}_0 + \Delta\dot{x}_e(p_{crit}) \quad (A12)$$

In doing this the last term can be determined by integrating Eqs. A5 and A6:

$$\Delta\dot{x}_e(p_{crit}) = c p_l T_0 p_0 \left(\frac{v_{g,0} - v_{l,0}}{\Delta h_{v,0}^2} \right) \ln \left(\frac{p_0}{p_{crit}} \right) \quad (A13)$$

The critical pressure ratio $\eta_{crit} = p_{crit}/p_0$ can be determined iteratively by means of

$$\begin{aligned} \eta_{crit}^2 + (\omega^2 - 2\omega)(1 - \eta_{crit})^2 + 2\omega^2 \ln(\eta_{crit}) \\ + 2\omega^2(1 - \eta_{crit}) = 0 \end{aligned} \quad (A14)$$

When the compressibility factor ω has values between 2 and 100 the critical pressure ratio can also be determined without significant errors using the following explicit equation:

$$\begin{aligned} \eta_{crit} = 0.55 + 0.217 \ln \omega - 0.046(\ln \omega)^2 \\ + 0.004(\ln \omega)^3 \end{aligned} \quad (A15)$$

Taking the nonequilibrium state into account yields the following relationship for ω :

$$\omega = \omega_{N=1} \left[\dot{x}_0 + c p_{l,0} T_0 p_0 \left(\frac{v_{g,0} - v_{l,0}}{\Delta h_{v,0}^2} \right) \ln \left(\frac{1}{\eta_{crit}} \right) \right]^a \quad (A16)$$

The exponent a is determined by approximating the critical mass flux

$$\dot{m}_{crit} = \frac{\sqrt{\omega \ln \left(\frac{1}{\eta_{crit}} \right) - (\omega - 1)(1 - \eta_{crit})}}{\left[\omega \left(\frac{1}{\eta_{crit}} - 1 \right) + 1 \right]} \sqrt{\frac{2p_0}{v_0}} \quad (A17)$$

using the method of Henry and Fauske [16]. The results of this method are extensively compared with measurements and widely accepted. Beside this, our own experimental results were used to estimate a best value of the power-law exponent a . At a value of $a = 3/5$ the calculated values from the extended ω -method agree well with the results from the method of Henry and Fauske for short nozzles. Accordingly, this value is recommended for the calculation of the mass flow rate through throttling devices with large depressurization rates and short flow lengths, that is, nozzles, orifices, and control valves typically encountered in industry. For devices with less-pronounced flow contraction, such as safety valves, a value of $a = 2/5$ is most appropriate. This value was found from comparison to experimental results with safety valves charged with two-phase steam/water and air/water flow.

In throttling devices with long flow paths the exponent a approaches $a = 0$.

Overall, the HNE-DS method uses values for the exponent a that are the result of an extensive comparison with experimental data of different throttling devices. Nevertheless, further investigations and experimental results may bring up minor improvements for devices, which are not considered here in detail, such as long nozzles, venturies, or orifices with very large area ratios.

The application of this new method has shown that an iterative determination of the compressibility factor (Eq. A16), including several iteration steps, is not required. It is initially recommended that the critical pressure ratio should be estimated using the compressibility factor without taking delayed boiling into account ($N = 1$) (Eq. A9), and then determine an improved value for this factor with the aid of Eq. A16 and taking delayed boiling into account.

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