Select the Best Model for Two-Phase Relief Sizing

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A variety of methods exist for sizing valves, but not all give the best predictions for certain conditions.

The proper nozzle area for a safety relief valve is determined by the required relief mass-flow rate $\dot{m}$, which is determined by the specific overpressure scenario and the mass flux capacity of the valve at these conditions:

$$A = \frac{\dot{m}}{G_v} \tag{1}$$

The valve mass flux G_v is the calculated value for an ideal (isentropic) nozzle G_n from an appropriate model, multiplied by a discharge coefficient K_d, which accounts for any discrepancies between the (ideal) nozzle model and flow in a real nozzle, as well as any differences between the flow in the nozzle and the actual valve:

$$G_v = K_d\, G_n \tag{2}$$

As shown in Figure 1, there are obvious similarities between an ideal nozzle and an actual relief valve, but there are also significant differences.

Proper sizing of a safety relief valve requires knowledge of the conditions upstream and downstream of the device, the physical and thermal properties of the fluid(s) at these conditions, and a model that accurately predicts the mass flux through the valve G_v as a function of the fluid properties and flow conditions. Methods for valve sizing for single-phase (gas or liquid) flows are well established, and are based upon simple single-phase isentropic flow models, together with discharge coefficients that are measured by the valve manufacturer and reported in the "Red Book" *(1)*.

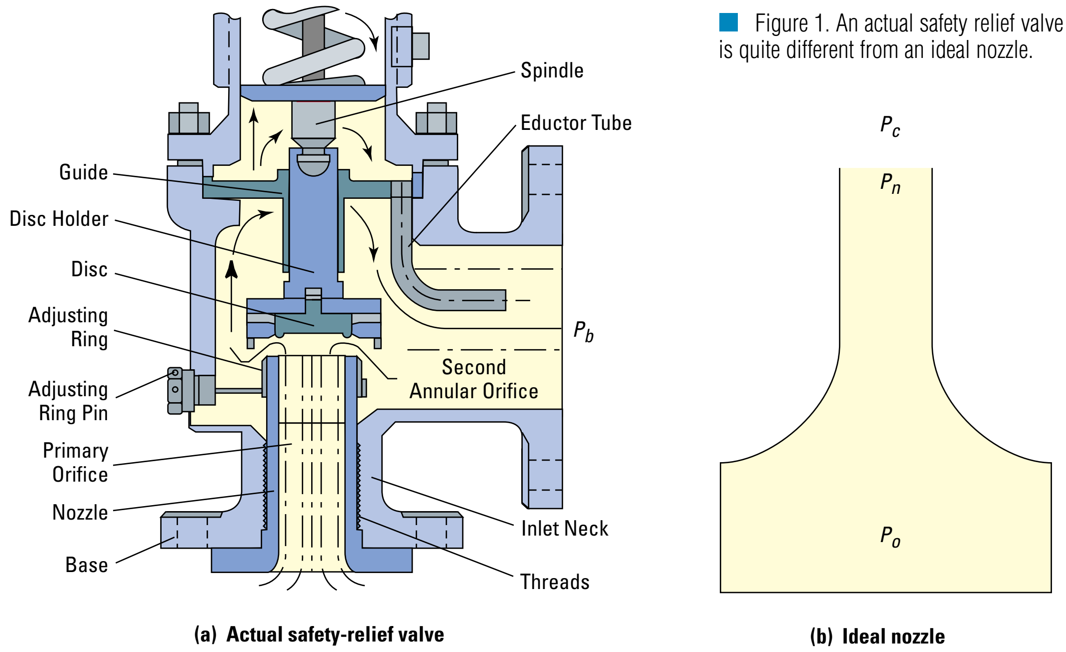


Figure 1. An actual safety relief valve is quite different from an ideal nozzle.

However, two-phase flow is frequently encountered in various relief scenarios and there are no data or Red Book coefficients, or even an accepted and verified two-phase flow model that may be used to size valves for such conditions. One reason for this is that two-phase flow is considerably more complex than single-phase, since there is a large number of variables associated with the fluid properties, distribution of the fluid phases, interaction and transformation of the phases, etc.

Consequently, there is a variety of models, each of which is based on a specific set of assumptions that may be valid for certain specific conditions, but may not be accurate for others. For example, the models that best describe the flow behavior of "frozen" gas-liquid flows (*i.e.,* those having no phase change) may be different from those for flashing flows, and the "best" model may be different for nonequilibrium flow in short nozzles than that for equilibrium flow in long nozzles.

Nomenclature

A = area, ft^2
C_{po} = specific heat of liquid at stagnation conditions, $ft\text{-}lb_f / lb_m \cdot °F$ or $Nm/kg \cdot °C$
e = pipe roughness, dimensionless
G = mass flux, $lb_m/ft^2 \cdot s$ or $kg/m^2 \cdot s$
G^* = dimensionless mass flux, $G_c/(P_o \rho_o)^{1/2}$
h_{LGo} = heat of vaporization at stagnation conditions, $ft\text{-}lb_f/lb_m$ or Nm/kg
K_d = discharge coefficient, dimensionless
K_f = friction loss coefficient, dimensionless
k_{NE} = empirical nonequilibrium parameter in Eq. 8, dimensionless
k_s = empirical slip coefficient in Eq. 7, dimensionless
L = nozzle length, ft or m
N_{NE} = nonequilibrium parameter defined by Eqs. 11 and 12, dimensionless
P = pressure, lb_f/ft^2 or Pa
S = ratio of gas phase velocity to liquid phase velocity, dimensionless
T = temperature, °R or K
V = velocity, ft/s or m/s
x = quality — mass fraction of gas or vapor in mixture, dimensionless
y = mole fraction in vapor
Z = vertical elevation, ft or m

Greek letters
α_o = volume fraction of gas phase at stagnation conditions
ν = specific volume, ft^3/lb_m or m^3/kg
ν_{LGo} = specific volume of gas minus specific volume of liquid at saturation conditions, ft^3/lb_m or m^3/kg
ν_o = specific volume of liquid at stagnation conditions, ft^3/lb_m or m^3/kg
ρ = density, lb_m/ft^3 or kg/m^3
ω = parameter defined by Eq. 9, dimensionless

Subscripts
G = gas or vapor phase
L = liquid phase
o = stagnation state
s = saturated state

Undersizing or oversizing

An inaccurate model for flow through a valve/nozzle could result in either undersizing or oversizing the valve. The consequences of undersizing are obvious — the valve will have insufficient capacity to prevent overpressuring the vessel.

However, the consequences of oversizing can also be severe, although maybe not as obvious. An oversized valve results in a higher than expected flow rate through the relief system, which also yields a greater than expected pressure drop both upstream and downstream of the valve. Either or both of these extreme pressure drops can negatively affect the stable operation of the valve, with possible damage or destruction to it. Unexpectedly high flow due to oversizing also results in undersized discharge piping and the effluent handling system downstream of the valve, also with negative consequences.

Many models, methods, and procedures have been proposed for analyzing two-phase flows in relief systems (*2–6*). However, none of these models has been systematically verified by comparison with extensive reliable experimental data. Here, we will review the predictions of what appear to be the most useful and appropriate models for two-phase flow in valves, nozzles, and tubes, and discuss the conditions under which each gives reliable results by comparison with the best data that have been found for two-phase flashing flows, in terms of the entering fluid quality and geometry of the system.

The models

The two-phase flow methods that are currently considered to be the most appropriate for relief valve sizing are based on either the homogeneous equilibrium (HEM) or the homogeneous nonequilibrium (HNE) models. There are various forms of each of these models in the literature, which vary primarily depending on the method used to evaluate the properties of the two-phase mixture. Both models assume that the two-phase mixture is homogeneous, *i.e.,* the two phases are sufficiently well mixed that they can be described as a "pseudo-single phase" fluid with properties that are a weighted average of those of each phase.

The equilibrium assumption implies that both phases are in thermodynamic and mechanical equilibrium, that is, any phase change (*e.g.,* flashing) occurs under equilibrium conditions at the saturation pressure, and both phases move at the same velocity (no slip). The Omega method *(3)* is a special case of the HEM model in which the two-phase density is represented as a linear function of pressure and the thermal/physical properties of the fluid at the stagnation state. This permits an analytical solution of the isentropic nozzle equation. A version of this model is being recommended by the American Petroleum Institute (API RP 520, "Sizing, Selection, and Installation of Pressure-Relieving Devices in Refineries," under revision). The nonequilibrium models may account for either the possibility of delayed flashing after the fluid mixture reaches the saturation pressure or the occurrence of slip between the phases.

It is generally assumed that the HEM model is adequate in most cases for two-phase flow in relatively long nozzles/pipes, for both frozen (constant quality) and flashing flows, when the fluid properties are properly evaluated. For frozen flows, however, large errors can result from neglecting slip in short nozzles.

Slip occurs as a result of expansion of the gas/vapor phase as the pressure drops, producing a corresponding increase in the gas-phase velocity. Hence, the gas accelerates relative to the liquid, resulting in a velocity difference and a corresponding drag force between the two phases.

The result is that the local *in situ* mixture density and mass flux (*i.e.*, the holdup) in the presence of slip are greater than they would be without it. Slip is expected to be most pronounced when the pressure gradient is large, such as in the entrance region of a pipe or nozzle. This can occur for both frozen or flashing flows, and has been shown to be especially important in frozen air/water mixtures in short nozzles *(7)*.

The most difficult situation to reproduce accurately is when the fluid entering the valve is either saturated liquid, liquid just above the saturation pressure (slightly subcooled), or a two-phase saturated mixture with very low quality, for which considerable flashing occurs within or near the exit of the nozzle. For such flows, nonequilibrium effects can result in a mass flux that is many times as great as predicted by the HEM model *(8)*.

This is because flashing is actually a rate process, rather than an equilibrium one. Nucleation of vapor bubbles only occurs when the pressure drops below the equilibrium saturation pressure (corresponding to a finite superheat), and the bubbles then grow at a finite rate controlled by the rate of heat and mass transfer from the liquid to the vapor phase. Even though this process can be fast, the high velocities experienced in a typical relief scenario can result in a significant distance of fluid travel before the flashing is complete. At a velocity of several hundred feet per second, the fluid will travel several inches in just one millisecond.

The HNE model accounts for this delayed flashing by assuming that vaporization is not complete until the fluid has traveled at least 10 cm (4 in.) along the nozzle. This nonequilibrium effect is more pronounced as the relief pressure rises, since the higher the pressure the higher the velocity through the nozzle, and the further the fluid will travel before flashing is complete. Both slip and nonequilibrium effects result in a higher mass flux and holdup relative to that predicted by the HEM model, since both of these effects result in a local quality (*i.e.*, mass fraction of gas) within the tube/nozzle that is smaller than would occur if the flow were at equilibrium.

TPHEM model

This model is based on work by Simpson *(9, 10)* and is described in a recent CCPS Guidelines book *(4)* that includes a CD containing a program for implementing the model. It is based on a numerical integration of the isentropic nozzle equation:

$$G_n = \rho_n \left(-2\int_{P_o}^{P_n} \frac{dP}{\rho} \right)^{1/2} \tag{3}$$

where G_n is the mass flux through the nozzle, P_o is the (upstream) stagnation pressure, P_n is the pressure at the nozzle exit, ρ is the local (two-phase) fluid density within the nozzle, and ρ_n is the fluid density at the nozzle exit. The two-phase density is related to the densities of the gas and liquid phases by:

$$\rho = \alpha\rho_o + (1 - \alpha)\rho_L \tag{4}$$

where α is the volume fraction of the gas phase. This, in turn, is related to the mass fraction (quality) of the gas phase x and the slip ratio $S = V_G/V_L$ by:

$$\alpha = \frac{x}{x + S(1-x)\rho_G/\rho_L} \tag{5}$$

There are many models for slip (or holdup (1 - α)) that relate S to the properties of the gas and liquid phases, *(6)*, although there are no clear guidelines for determining the conditions under which these models might be valid for nozzle flow. The TPHEM program includes an option that includes the effect of slip through the following expression:

$$G_n^2 = \frac{\int_{P_o}^{P_n} -2\left[x/\rho_G + (1-x)/\rho_L\right] dP}{\left\{[x/(\rho_G S) + (1-x)/\rho_L]^2\,[xS^2 + 1 - x]\right\}_t} \tag{6}$$

The subscript t represents conditions at the throat (exit) of the nozzle. The model also includes the following formula for the slip ratio in terms of an empirical parameter k_s, which is input by the user (while there are no guidelines for choosing the appropriate value of k_s, Ref. 8 offers some experimental guidelines):

$$S = \left(1 - x + x\frac{\rho_L}{\rho_G}\right)^{k_s} \tag{7}$$

The TPHEM program also includes an option to account for nonequilibrium behavior by replacing the local equilibrium quality x by a "nonequilibrium" quality x_{NE}, where $x_{NE} < x$:

$$x_{NE} = x_A + k_{NE}\left(x^2 - x_A^2\right) \tag{8}$$

Here, k_{NE} is an empirical parameter (input by the user), and x_A is the quality at the upstream state for saturated inlet conditions (or zero for subcooled inlet conditions). According to Simpson *(9)*, a value of k_{NE} = 1 gives results comparable to the HNE model (see below), but there are no further guidelines for the appropriate value to use for k_{NE}.

A critical aspect of any model is the method used to evaluate the density of the two-phase mixture that appears in the integral in Eq. 1. The TPHEM program incorporates a choice of a variety of possible (two- or three-parameter) empirical models for this density *(10)*. The user can select any of these models, and must input data for the liquid- and gas-phase densities and the quality at the upstream stagnation pressure at one or two downstream pressures, depending upon whether a two- or three-parameter density model is chosen.

The program evaluates the model parameters by fitting the pressure/density data provided, and uses the resulting model to determine the mass flux by numerical integration of Eq. 3 from the stagnation pressure to the discharge pressure. If the flow is choked, the mass flux will reach a limiting value before the discharge pressure is reached, at a pressure corresponding to the critical choke pressure, which can be determined by varying the discharge pressure. The program includes options for nozzle flow with or without friction (*i.e.*, an entrance loss coefficient) and pipe flow with friction, as well as for viscous or inviscid (zero-viscosity) liquid properties.

The one-parameter density model in TPHEM is equivalent to the Omega method *(3)*, which is an analytical solution of the nozzle mass-flux integral. The model parameter is equivalent to the parameter ω of the original Omega method, which can also be evaluated from the thermal/physical properties of the two-phase mixture at the stagnation conditions:

$$\omega = \alpha_o\left(1 - \frac{2P_o v_{LGo}}{h_{LGo}}\right) + \frac{C_{po}T_oP_o}{v_o}\left(\frac{v_{LGo}}{h_{LGo}}\right)^2 \tag{9}$$

Alternatively, this parameter can be determined from known values of the two-phase density at the stagnation pressure and one additional pressure between the stagnation and discharge pressures, as specified by the TPHEM program. This is usually more accurate than using Eq. 9.

HNE model

This model has been expressed in various ways *(5)*. For flashing flows, it can be represented by the following set of equations:

$$\frac{G_c}{G_1} = \left[\frac{\left(\frac{G_o}{G_1}\right)^2 + \frac{1}{N_{NE}}}{\left(1+K_f\right)}\right]^{1/2} \tag{10}$$

$$N_{NE} = \left(\frac{G_1}{G_3}\right)^2 + \frac{L}{L_e} \tag{11}$$

for $L \le L_e = 10$ cm

$$N_{NE} = 1 \qquad \text{for } L > L_e \tag{12}$$

$$G_o = \sqrt{2\rho_{Lo}\left(P_o - P_s\right)} \tag{13}$$

$$G_1 = \frac{h_{LGo}}{v_{LGo}\sqrt{T_oC_{pLo}}} = G_{ERM} \tag{14}$$

$$G_3 = \sqrt{2\rho_{Lo}\left(P_s - P_2\right)} \tag{15}$$

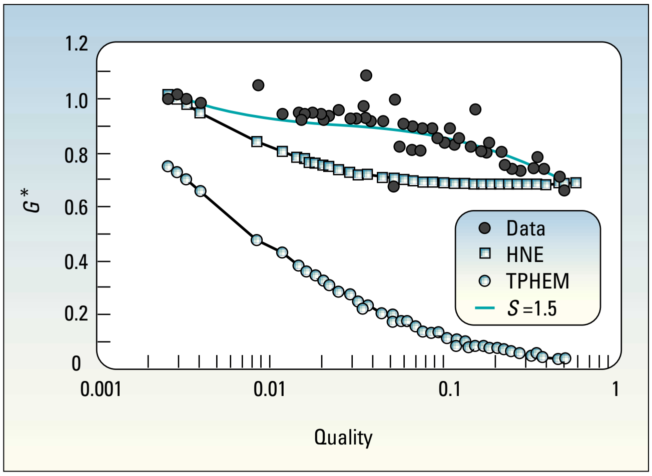


Figure 2. Graham Nozzle No. 1 with air/water data vs. predictions.

$G_1 = G_{ERM}$ represents the equilibrium rate model (ERM), which describes the critical (choked) mass flux resulting from the phase change (flashing). N_{NE} is the nonequilibrium parameter, which represents delayed flashing for lengths less than 10 cm. G_o and G_3 are the liquid components of the flow from P_o (stagnation) to P_s (saturation) for subcooled inlet conditions, and for P_s to P_2 (discharge). For all-liquid flow (no flashing), P_s is replaced by P_2 and $G_c = G_o$. For choked flow, P_2 is replaced by the choke pressure P_c.

The friction loss coefficient K_f can include entrance and fitting losses, as well as pipe friction for which $K_f = 4fL/D$; f is the Fanning friction factor. This model does not include a provision for accounting for slip.

The predictions of these various models were compared with the following data sets from the literature to test their validity.

Data: Frozen flow

The most consistent set of data for frozen flow appears to be that of Graham *(11)* for air/water mixtures through short (ASME-type) nozzles. Jamerson and Fisher *(7)* compared the TPHEM computer predictions with these data, and found that good agreement is obtained using a constant slip value S of 1.5–1.8. An example of these data is shown in Figure 2, in which the dimensionless mass flux, defined as:

$$G^* = \frac{G_c}{\sqrt{P_o\rho_o}} \tag{16}$$

is plotted vs. the inlet quality x_o for the Graham Nozzle No. 1 (⅝ in.), along with a comparison of the predictions of the TPHEM homogeneous equilibrium model, the TPHEM model with a constant slip ratio of $S = 1.5$, and the HNE model. The quality is defined as the mass fraction of the vapor (gas) phase, *i.e.*:

$$x = \frac{\nu - \nu_L}{\nu_G - \nu_L} = \frac{\nu - \nu_L}{\nu_{GL}} = \frac{\nu - \nu_s}{\nu_{GLs}} \quad (17)$$

where $\nu = 1/\rho$ is the two-phase specific volume, subscripts L and G refer to the liquid and gas phases, and s refers to saturation conditions for flashing flows (subcooled inlet conditions correspond to a negative quality, since $\nu < \nu_{Ls}$). It is evident that the HEM model drastically underpredicts the data, whereas including slip in the model with $S = 1.5$ results in close agreement with the data.

The HNE model is much better than the HEM model, and agrees with the data at very low and very high values of quality, but, underpredicts G^* by up to 20% for qualities near zero. Jamerson and Fisher *(7)* also analyzed the data of Toner *(12)* for air/water mixtures in longer converging/diverging nozzles, and found that slip ratios of 1.1–1.2 gave a reasonable fit of these data.

Data: Flashing flow in nozzles

The most extensive data in the literature for flashing two-phase flow are for steam/water mixtures, mostly from the nuclear power industry. In an Electric Power Research Institute (EPRI) Report, Ilic *et al. (13)* compiled over 70 data sets from 20 different sources for steam/water flows, which were culled for inconsistent and incomplete data. Thus, this represents what is claimed to be a comprehensive reliable database for steam/water in various nozzles and pipes.

The most extensive and consistent of these data sets are those of Sozzi and Sutherland *(14)* for flows in various lengths of straight and converging/diverging nozzles and tubes. Of particular interest are the data for their Nozzles 2 and 3. Nozzle 2 has a well-rounded entrance that converges from 1.75 in. to 0.5 in. over a distance of 1.75 in., with ten lengths of 0.5 in. straight tubing from 0 to 70 in. long attached. Steam/water flow rates were measured at inlet pressures from 800–1,000 psig and inlet quality conditions from -0.004 to 0.007.

Nozzle 3 was a square-entrance, 0.5 in. dia. tube, with lengths of 0.185–25 in., and inlet conditions similar to those of Nozzle 2. The database includes mass flux, inlet and exit pressures, quality, and enthalpy for each data set for a given pipe/nozzle length. The fluid densities and quality of the mixture downstream of the entrance were determined from steam tables assuming isenthalpic flow.

The Sozzi and Sutherland data for Nozzle 2 and Nozzle 3 were compared with the HNE and the various TPHEM options by Darby *(8)* and Darby *et al. (15)*. The Nozzle 2 data were compared with the predictions of the TPHEM equilibrium model, TPHEM with the slip option using the value of the parameter k_s that gave the best average fit of the data set (to the closest 0.1 unit of k_s), the TPHEM nonequilibrium option with the k_{NE} parameter adjusted to give agreement with each data point (to within 1%), the TPHEM nonequilibrium option with the one value of k_{NE} representative of the average value of k_{NE} for the data set, and the HNE model.

For the HNE calculation, the value of the nozzle choke pressure (as given in the database or determined by the TPHEM program) was used in place of P_2 for the exit pressure, since choked conditions prevailed for all runs. The three-parameter Model F for two-phase density was used for each case. Examples of the results for $L = 0$ and 1.5 in. are shown in Figures 3 and 4.

The results demonstrated that the value of k_{NE} required to fit the Nozzle 2 data increases with the nozzle length, from $k_{NE} = 1$ for $L = 0$ to $k_{NE} > 20$ for $L = 9$ in. The best fit value of k_{NE} depends upon both the nozzle length L and the stagnation quality x_o, increasing with both the nozzle length and inlet quality for superheated entrance conditions. It also increases with increasing subcooling, with a minimum value at or near saturated inlet conditions.

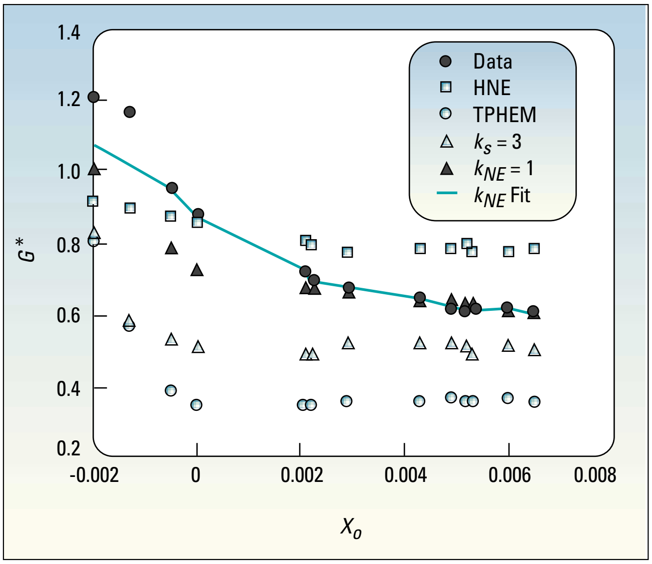
However, an average value of k_{NE} for all data points for each data set (*i.e.*, nozzle length) also gave a reasonable fit. An empirical equation was developed to represent this dependence *(8)*, although it is not recommended that this expression be extrapolated beyond the range of these data.

The TPHEM equilibrium model was consistently low for the shorter nozzle lengths, but gave good results for the longer nozzles/pipes (L = 7.5 in. and longer), and improved as the inlet quality increased. It was not possible to get agreement with the data by adjusting the slip parameter k_s in the TPHEM model for the shorter nozzles or low quality, but a reasonable fit was possible as the quality increased. For the longer nozzles ($L > 2.5$ in.), a good fit was obtained for the higher quality flows with $k_s = 1.5$.

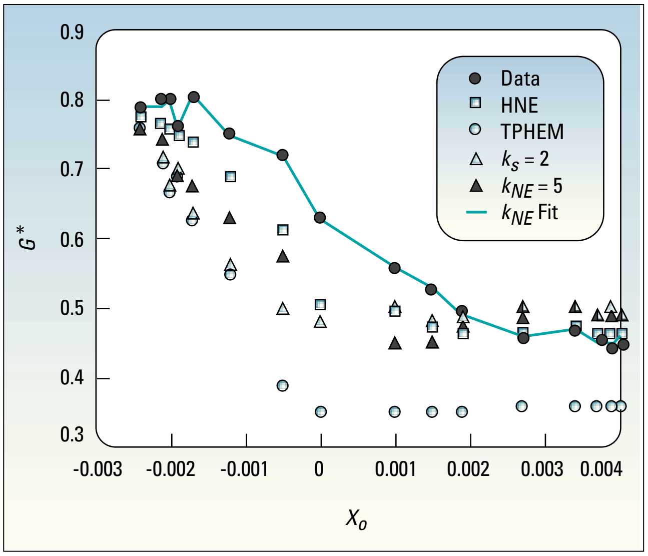
The goodness of fit was measured by the average and standard deviation of the ratio G_{obs}/G_{calc} for each model and for each data set. A value of $G_{obs}/G_{calc} = 1$ indicates a perfect match between the model and the data, and the lower the standard deviation the more consistent the prediction.

Except for the nonequilibrium TPHEM model option, which was a forced-fit of the data, the HNE model with no arbitrary adjustable parameters gave the most consistent results overall. The agreement of this model was consistently good for all inlet quality conditions for all nozzles from 1.5–9 in. The values of G_{obs}/G_{calc} varied from about 0.92–1.04, with a standard deviation as low or lower than the equilibrium models, except for $L = 0$. For the $L = 0$ nozzle, the fit was excellent near saturation, and within about 15% for higher and lower quality (see Figure 3).

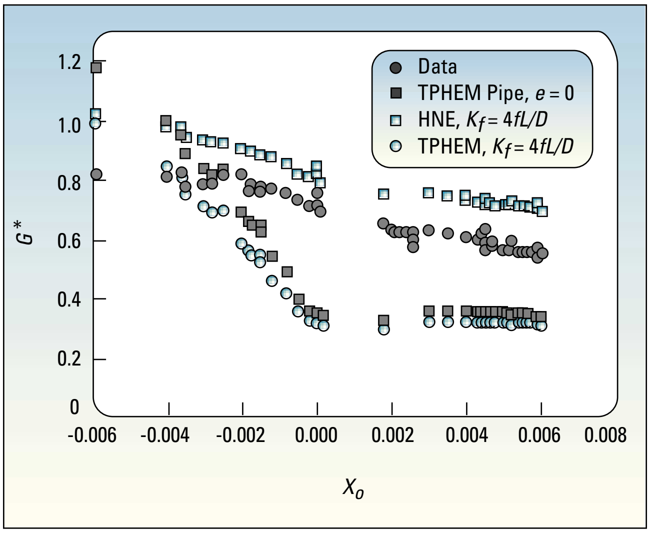
For the square entrance Nozzle 3, the only data set that corresponded to "nozzle" conditions (*i.e.*, $L < 4$ in.) was L = 0.185 in. (Figure 5). The nonequilibrium and slip options in TPHEM did not give consistent results for the Nozzle 3 data, so only the TPHEM nozzle option with friction, the TPHEM pipe option, and the HNE models were compared with these data. For this geometry, the TPHEM nozzle and pipe models agreed with the data for high subcooling (liquid flow), but underpredicted the data by 200–250% for



■ Figure 3. Sozzi and Sutherland Nozzle No. 2 data; $L = 0$.



■ Figure 4. Sozzi and Sutherland Nozzle No. 2 data; $L = 1.5$ in.



■ Figure 5. Sozzi and Sutherland Nozzle No. 3 data, $L = 0.185$ in.

two-phase low-quality flow, while the HNE model overpredicted this data set consistently by about 15–25% over the whole range of quality.

Data: Flashing flow in tubes

Nozzles 2 and 3 with straight lengths from 7.5–70 in. were considered to be pipes, with an appropriate wall roughness and entrance loss coefficient. These data were fit by the TPHEM nozzle flow option with a loss coefficient to account for both the entrance loss and the pipe friction loss, the pipe flow option, and the HNE model. The pipe flow TPHEM option uses the following equation for pipe flow, which is the equivalent of Eq. 3 for nozzles:

$$G^2 = \left(\frac{\int_{P_o}^{P_2} -\rho dP + g\left(Z_o - Z_2\right)\rho_{avg}^2}{\ln\left(\rho_2/\rho_o\right) + \left(\Sigma K_f + 1\right)/2} \right)_{max} \quad (18)$$

For the TPHEM nozzle option, the pipe friction loss coefficient is given by $4fL/D$. The Fanning friction factor f was taken to be that for fully turbulent flow in a 0.5 in. dia. tube with a roughness of 0.0004 in., typical of that for stainless steel (*i.e.*, $f = 0.00465$). The TPHEM pipe flow option requires an input of the pipe roughness (which was also taken to be 0.0004 in. for these data); the friction factor and loss coefficient are computed by the program. The same value of $K_f = 4fL/D$ was also used in the HNE model for pipe flow. A square entrance loss coefficient of 0.4 was included with the Nozzle 3 computations (the entrance loss coefficient is zero for Nozzle 2).

For the $L = 9$ in. No. 2 nozzle, the agreement of the HNE model was excellent for saturated and subcooled conditions, but, for higher inlet qualities, the model overpredicts the data by about 10%. Sample results are shown in Figures 6 and 7 for Nozzle 2 tubes 12.5 and 70 in. long, and in Figure 8 for a Nozzle 3 tube 20.2 in. long.

All of the models (TPHEM nozzle, TPHEM pipe, and HNE) gave reasonable agreement with the pipe data when the friction loss was included in the model. The HNE model gave as good or better agreement than the other models, with values of G_{obs}/G_{calc} of 0.98–1.08 for the Nozzle No. 2 tubes, and 0.86–0.97 for the Nozzle No. 3 tubes, with a standard deviation as low or lower than the TPHEM models. A summary of the various TPHEM program options that were run, along with the corresponding values of the various program parameters, is shown in Table 1.

Table 1 - TPHEM Options Run and Program Parameters

	TPHEM Option Parameters*					
Case Description	**IU**	**IC**	**IPTS**	**IV**	**INES**	**X**
TPHEM Nozzle w/ friction	3	1	3	1	--	--
TPHEM Nozzle w/o friction†	3	1	3	1	--	--
TPHEM Nozzle w/ slip†	3	3	3	1	2	1.5
TPHEM Nozzle K_{NE} fit†	3	3	3	1	11	Various‡
TPHEM Pipe w/ friction	3	1	3	-3	--	--
HNE w/ friction	--	--	--	--	--	--
HNE w/o friction†	--	--	--	--	--	--

* TPHEM Parameters:

- IU — Units: 3 = Metric
- IC — Case: 1 = gives flow rate output; 3 = gives flow rate output and also activates INES, advanced options
- IPTS — Two-phase density model and number of data states: 3 = Three-Parameter Model F (3 input data states)
- IV — Input Options: 1 = simple nonviscous nozzle input; -3 = pipe input without viscosity correction
- INES — Advanced Options: 2 = user inputs slip ratio, S; 11 = user inputs k_{NE} nonequilibrium parameter.
- X — Value of Advanced Option parameter (S or k_{NE})

† These cases gave poor results, and were eliminated from the final comparison.
‡ The k_{NE} parameter was varied to determine the value that fit each data point, and varied from 0 to 75.

Data: Flashing flow in valves

There is a paucity of suitable data in the literature for two-phase flow in actual relief valves. A comparison of the above models with limited data for flashing steam/water flows in valves was given by Darby *et al. (15)*. A summary of these results is given below, with a description of the valve/nozzle used in each study.

1. Data were provided by Anderson Greenwood Crosby, Inc. for flashing flow in a modified Crosby Series 900 valve, with two different straight uniform-bore nozzles with a smoothly contoured entrance. Both nozzles had a bore of 0.4 in. dia., with the shorter nozzle being 0.5 in. long and the longer 5.0 in. Two data points were taken with each nozzle, one at a low temperature (all liquid flow) and one at high temperature (saturated water at 365°F).

Data were also provided by Anderson Greenwood Crosby, Inc. from tests on the following two valves:

2. Crosby 9511 valve — The nozzle had a ⅝ in. dia. inlet straight section that tapered to a ⅓ in. straight section, 0.956 in. long. The diameter and length of this straight section were taken to be the governing nozzle dimensions. Four data points were recorded with this valve, two of which were all-liquid discharge and two were flashing two-phase discharge at 330 psig.

3. Crosby JLT/JBS valve — Five data points were reported over a range of 200–500 psig with (subcooled) inlet qualities from -0.0099 to -0.123. The nozzle tapered from a dia. of 1½ in. over a distance of 3⅞ in. to a uniform straight section with a dia. of 1.065 in. and a length of 0.875 in. The dimensions of this straight section were taken to be the controlling nozzle dimensions for the HNE model.

4. Data were reported by Bolle *et al. (16)* for a Crosby JLT/JOS-15A valve, with a nozzle that converged to a straight section 0.409 in. long and 0.409 in. dia. The entering fluid was subcooled water and 16 data points were reported (six of which were all-liquid discharge), over a pressure range of 2–6 bar.

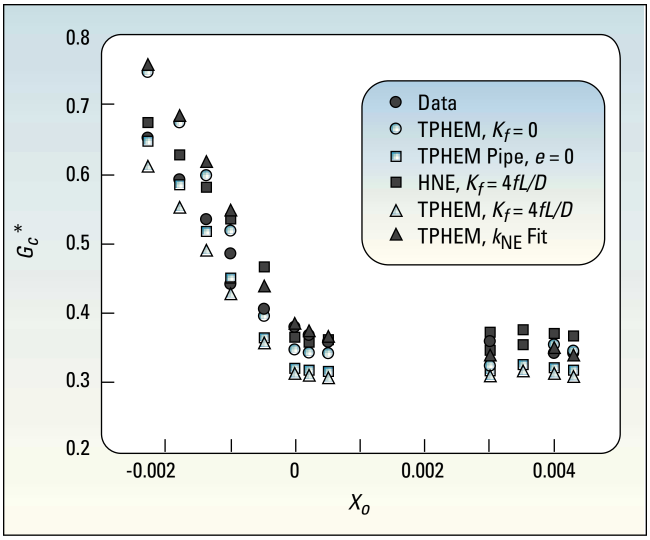
5. Two-phase data were reported by Lenzing *et al. (17)* for flow in five different valves: Albert Richter (ARI) DN25/40, Babcock Sempell SC 01, Bopp & Reuther Si 63 DN25/40, Crosby JLT/JBS, and Leser 441 DN25/40, DN40/65. The only data reported for flashing flow were for water in the Leser DN25/40 valve, at a pressure of 10.6 bar, over a range of inlet quality of 0.001–0.05. The nozzle minimum dia. was given as 23 mm, but no nozzle geometry or other dimensions were reported.

The calculated mass flux through each valve nozzle was compared with the reported measured values, and the results were expressed as an equivalent discharge coefficient:

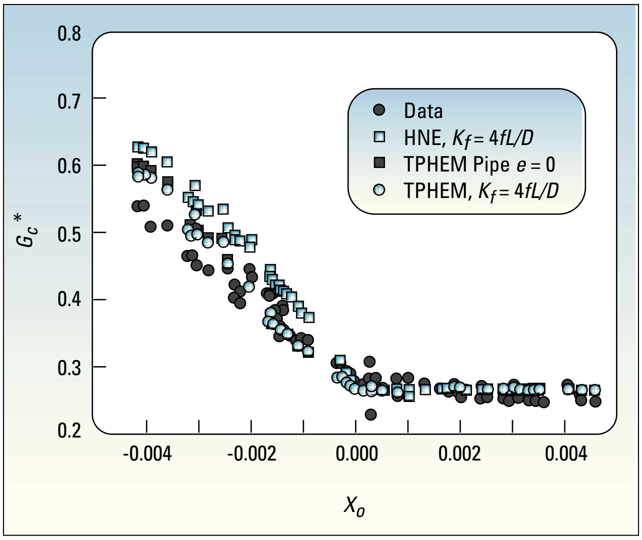
$$K_d = G_{exp}/G_{calc} \tag{19}$$

The models compared with these data were the equilibrium TPHEM nozzle model with the three-parameter F density model, $(K_d)_F$; the one-parameter homogeneous TPHEM nozzle model (comparable to the Omega method), $(K_d)_{omega}$; the nonequilibrium TPHEM nozzle option with an empirically adjusted value of k_{NE}, $(K_d)_{kne}$; and the HNE model, $(K_d)_{HNE}$. The HNE model has no adjustable parameters, and is the only model that is sensitive to the length of the nozzle.

The results of fitting the various models to each data point for each of the five data sets were reported by Darby *et al. (15)*. Comparison of the various model predictions with the valve capacity data showed mixed results, *i.e.*, the model that gave the best fit of the data varied from valve to valve. This could be attributed to the limited range and scope of the data, undefined uncertainties in these data, or the deviation of the valve nozzle geometry from the "ideal" straight bore rounded entrance nozzle.



■ Figure 6. Sozzi and Sutherland Nozzle No. 2, pipe data; L = 12.5 in.



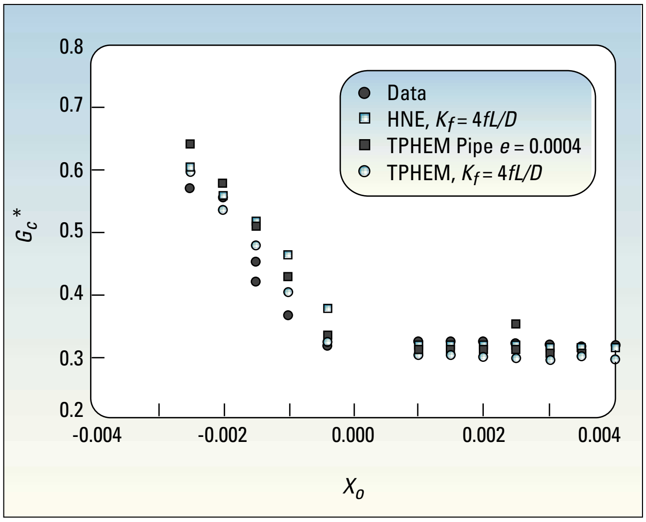
■ Figure 7. Sozzi and Sutherland Nozzle No. 2 data, L = 70 in.

Thus, any conclusions based upon these data should be considered to be tentative, at best. This being said, most of the models gave reasonable results for most of the valve data (all K_d values were between 0.7–1.3), with the exception of the Bolle data, for which all models consistently underpredicted the values by an average of about 30% (which would seem to indicate some systematic error or offset in these data).

The equilibrium TPHEM model with the 3-parameter (F) density model gave consistently somewhat better results than the one-parameter (Omega) density model, although for the majority of the data points, these values agreed to within about 10%. Overall, however, the HNE model consistently gave the best results, except for the Lenzing data at the highest quality, for which it was about 15–20% low (for an assumed nozzle length of 4 in.). In general, the homogeneous TPHEM model with the F density model gave results almost as good as the HNE model, except for the valves with the shortest nozzles.

As an example of the sensitivity of the data to entering conditions, the fluid entering the Crosby 900 valve was reported to be saturated at 365°F and 150 psig. However, the saturation temperature at this pressure is actually 365.87°F. This difference of less than 1°F translates to a variation of about 10% in the predicted mass flux and the corresponding values of K_d (*i.e.*, a small error in T can result in a large error in pressure).

It was possible to get a good fit of most of the valve data using the TPHEM nonequilibrium option by adjusting the parameter k_{NE}. However, the values of k_{NE} needed to fit the data varied from about 2.5–16, with no apparent correlation with the fluid properties or conditions. For some data points, the best fit was not very sensitive to the actual value of k_{NE} as this value increased.



■ Figure 8. Sozzi and Sutherland Nozzle No. 3 data, L = 20.2 in.

Recommendations

The Sozzi and Sutherland database for flashing flow of water in nozzles and tubes at approximately 1,000 psia is much more extensive and consistent than the available data for flashing flow in valves. Based on these data, the HNE model gives reasonably accurate values over the widest range of inlet quality and nozzle length conditions, when the effect of friction loss is included for the longer nozzles and tubes.

The equilibrium TPHEM models significantly underpredict the mass flux for subcooled and saturated inlet conditions in short nozzles, but are in good agreement with the data for higher inlet quality conditions and longer tubes when friction loss is included. It is possible to get good agreement with the data for the short Nozzle No. 2 (rounded entrance) data with the nonequilibrium option of

TPHEM by adjusting the k_{NE} parameter, but this does not work very well for the Nozzle No. 3 (square-entrance) nozzle and pipe data. Although it has been shown that slip can be important in frozen flows, it should not be significant in flashing flows relative to the effects of flashing.

The available data for valve flow are much more limited, and are complicated by the variable geometry of the valve nozzle. Because the flashing pressure for these data is considerably below that of the Sozzi and Sutherland data (typically about 100 psia), nonequilibrium effects are much less pronounced. Considering the limited extent and the uncertainty in these data, the tentative conclusion with regard to the suitability of the models is roughly the same as for the nozzle data. Within the uncertainty of the data, the HNE model gave the most consistent predictions for nozzles, pipes, and valves, as well as for rounded and square entrance configurations. It is also the most consistent over the range of inlet quality from subcooled to two-phase inlet and for short nozzles to long pipes and valves.

For flashing flow in valves, a discharge coefficient K_d of 1.0 is tentatively recommended for use with the HNE model based on these results. This is reasonable, since flashing flows are inevitably choked, so that the valve capacity is determined by flow in the nozzle only and is independent of the flow conditions downstream of the nozzle in the valve body, etc. Values of K_d that are significantly lower than 1.0 normally result for gases under subcritical conditions and liquid flows, for which the flow in the valve body has a significant effect on the valve capacity.

It should be emphasized that to obtain reliable results from the models, it is necessary to have a complete understanding of the thermodynamic and physical state of the fluid entering and leaving the nozzle, which implies accurate values of temperature and pressure, as well as fluid transport and thermodynamic properties at these conditions. For example, for an entering fluid near saturation, a variation of 1°F in the temperature can have an effect as large as 20% on the calculated mass flux through the valve.

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