SAFETY RELIEF VALVE SIZING: COMPARISON OF TWO-PHASE FLOW MODELS TO EMPIRICAL DATA

A Senior Honors Thesis

By

PAUL ROBERT MEILLER

Submitted to the Office of Honors Programs
& Academic Scholarships
Texas A&M University
In partial fulfillment of the requirements
For the Designation of

UNIVERSITY UNDERGRADUATE
RESEARCH FELLOWS

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April 2000

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ABSTRACT

Safety Relief Valve Sizing: Comparison of Two-phase

Flow Models to Empirical Data. (April 2000)

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The proper sizing of safety relief valves is an important issue in chemical process safety. Many emergency relief scenarios require consideration of two-phase flow conditions. However, two-phase flow involves complex physics and is the subject of intensive on-going study. The objective of this research is to identify and verify simple yet accurate two-phase flow models which allow the design engineer to predict the mass flux of any given relief scenario. Two contemporary models were considered in this study: The Two-Phase-Homogenous-Equilibrium Model (TPHEM), proposed by the Center for Chemical Process Safety (CCPS), and the Homogenous-Nonequilibrium Model proposed by Fauske. These models were evaluated against steam/water data (both sub-cooled and two-phase entrance) from Sozzi and Sutherland. This research allowed the determination of what conditions were relevant when considering flashing flows, as well as verifying the models accuracy.

ACKNOWLEDGMENTS

The author gratefully and especially acknowledges the assistance of Dr. Ron Darby of the Chemical Engineering Department, Texas A&M University without whom this research would never have been possible.

Thanks also to Jared Stockton for his work on the shorter Sozzi and Sutherland nozzle #2 data and for his help troubleshooting (and solving) TPHEM problems.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

INTRODUCTION

IMPORTANCE

An emergency relief system is essential to ensuring the safe operation of process units. It is almost impossible to guarantee that some sort of overpressuring event will never occur. Therefore, an emergency relief system stands ready as a "last-resort" safety device to mediate an overpressure event. To operate properly, the emergency relief system must be designed properly, and relief valve sizing remains a topic of on-going study. A properly sized valve is crucial for successful management of emergency releases. Obviously, an under-sized valve will not allow adequate mass flow to handle the pressure build-up, ultimately leading to a tank/reactor rupture. Less evident is the hazard presented by an over-sized valve. In this case, there may be too high a pressure drop upstream and downstream of the valve to provide enough pressure difference to keep the valve open. Thus, an unsteady-state condition results, which can lead to damage to the valve (from excessive openings and closures). This undesired state of operation is referred to as valve chatter.

There are two major steps when sizing safety valves. First, the design engineer must estimate the amount of mass flow the valve must handle to maintain vessel integrity (e.g., estimating the amount heat and vapor generated by a runaway reaction). Second, the engineer must determine the mass flux for a certain valve configuration given the physical and thermodynamic properties of the system. The required cross-

This thesis follows the style and format of *Chemical Engineering Progress.*

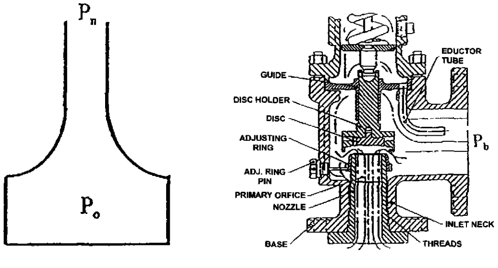
sectional area (and thus the diameter) of the valve is simply obtained by dividing the mass flow rate by the mass flux. This paper focuses on a special case of the second step, namely determining the mass flux through a nozzle under two-phase flow conditions.

Consideration of a two-phase flow system is often required in emergency relief design *(1)*. Two-phase flow, unlike single-phase flow, is poorly understood despite intensive research efforts. This is not surprising in view of the much greater complexity of two-phase flow. Successfully modeling two-phase flows has been the objective of a variety of models, methods, and procedures proposed in the literature (for example, see Ref. 2, 3, 4, and 5). This paper will attempt to verify two of the most contemporary two-phase flow models against an extensive two-phase flashing flow database. Systematic verifications of this type have not been attempted previously. The aim of this research is to: (1) determine the type and range of system conditions that the models can be successfully applied to, (2) gain understanding in what types of effects are most important and which effects can be safely ignored, and (3) ultimately, propose guidelines to aid the process engineer in successfully and safely applying these models to emergency relief valve design.

BACKGROUND

SINGLE-PHASE FLOW

By way of introducing the theory underlying mass flux problems, a brief explanation of single-phase flow is given below *(6)*. Single-phase flow is well understood, and is modeled by an ideal, adiabatic nozzle (Figure 1a) modified with a discharge coefficient, K_D. Note the drastic difference in flow geometry between the actual valve (Figure 1b) and the ideal nozzle. The discharge coefficient can range from 0.6-0.7 for liquids to close to unity for gases. The various valve manufacturers publish empirical discharge coefficients for single-phase flows **only** in the so-called "Red Book" *(7)*. No data on two-phase flow discharge coefficients are routinely published.



a) Ideal Nozzle b) Typical Spring-Loaded Relief Valve

Figure 1: Schematic of Actual Valve and Ideal Nozzle

To calculate the ideal mass flux, G_{nozzle}, the following information is required: stagnation (upstream) pressure, P_o; exit pressure, P_n; and the relationship between the fluid density and pressure. The actual mass flux, G_{valve}, is obtained by multiplying by K_D.

$$G_{valve} = K_D \, G_{nozzle} = K_D \rho_n \left[2 \int_{P_o}^{P_n} -\frac{dP}{\rho} \right]^{1/2} \qquad (1)$$

An important complication arises in gas/vapor flows. Due to the compressible nature of gases/vapors these systems can reach a maximum flux or "choke" point where the local velocity equals the speed of sound (i.e. the sonic velocity). The pressure that corresponds to the maximum velocity is referred to as the choke pressure or P_c. Once the choke pressure is reached, any further pressure drop that occurs in the system does not affect the mass flux calculation. This is an important consideration as it explains how the ideal nozzle results are in such good agreement with empirical gas flows. Most releases involve large pressure drops that tend to choke very quickly. If the choke pressure occurs at the nozzle exit, then the rest of the valve becomes irrelevant. Note that the valve nozzle itself closely resembles the ideal nozzle. In practice, when using Eq. 1 to calculate the maximum mass flux, the exit pressure is lowered until a maximum in G_{nozzle} is obtained. This final pressure is the choke pressure of the system.

TWO-PHASE FLOW

Although two-phase flow is substantially more complex, the basic approach to sizing relief valves is the same as single-phase flows: choosing the appropriate model for the pressure-density relationship so Eq. 1 can be evaluated, and determining the proper discharge coefficient to use. The discharge coefficient is of considerable concern since there is presently no published K_D data for two-phase flow through relief valves. In order to select an appropriate model, assumptions must be made with regard to a number of factors which are discussed by Darby *(1)*. The more important considerations are summarized below.

Phase Distribution

Differing relative flow rates and relative amounts of the two phases gives rise to a variety of flow regimes. Although a wide range of regimes are possible, the typical assumption for relief valve scenarios is homogeneous flow, where the gas and liquid phases are uniformly mixed. A common-day example of this is the flow out of a well-shook champagne bottle. When analyzing relief valves, the common assumption is that the two-phase mixture can be represented as a "pseudo- homogeneous single-phase fluid". The physical properties of this "homogenous fluid" are some appropriate weighted average of the properties of the separate phases. Both models discussed in this paper make use of the pseudo-homogeneous fluid assumption.

Thermodynamic State

How the thermodynamic state of the fluid changes across the valve body is a major consideration that must be accounted for. A number of different cases can exist. A "frozen" system is one where the quality (vapor mass fraction) of the fluid is not a function of pressure and therefore does not change (i.e. the relative amounts of gas/vapor and liquid remain constant). A cold air/water mixture is an example. "Flashing" systems (e.g. water/steam) do change quality as pressure drops across the valve and the liquid evaporates or boils. "Saturated" and "slightly sub-cooled" systems are a sub-set of the "flashing" condition. In the "saturated" case, the initial saturated liquid undergoes additional flashing through the valve. "Slightly sub-cooled" systems enter the nozzle as an all-liquid flow and do not flash until after the saturation pressure has been reached. It is of extreme importance for any model to account for these different conditions.

Thermodynamic Equilibrium

While it is tempting to simply assume local thermodynamic equilibrium when analyzing flashing flows, this assumption ignores the rate processes involved in the flashing process (i.e. nucleation site formation, heat transfer involved in bubble growth, etc.). These rate processes result in an effective delay in the initial generation of vapor after the saturation pressure has been reached. In systems where flashing has already generated a significant amount of vapor, the assumption of local thermodynamic equilibrium appears to be valid. Otherwise, when dealing with "saturated", "slightly sub-cooled", or "slightly superheated" systems (i.e. systems with low qualities) the non-equilibrium nature of flashing must be considered.

Mechanical Nonequilibrium (Slip)

As the pressure drops across the valve body, the gas/vapor phase will expand (i.e. density decreases, specific volume increases) while the liquid phase density remains constant. Thus, the gas phase will accelerate relative to the liquid phase, and a so-called "slip" condition is created. Slip affects the local volume fraction of gas/vapor or the gas/vapor "hold-up". Slip effectively increases the liquid "hold-up" for a given location within the system. General situations where slip might be important include: low quality flows (qualities less than 1%), frozen systems, and systems with large pressure gradients.

Choked Conditions

Like single-phase flow, two-phase flows reach a maximum mass flux when the sonic velocity is reached. Unlike single-phase flow, however, the speed of sound is less clearly defined. Still, the same basic technique can be used, namely lowering P_n to determine the maximum flux. Flashing flows will choke at much higher exit pressures than single-phase gas systems. For comparison, most single phase gas/vapor systems will choke at roughly half of the inlet pressure while two-phase systems may choke at up to 90% of the inlet pressure. Frozen flows may choke over a much wider range of pressures, but generally at lower pressures than flashing flows.

Fluid Composition

In addition to all the above considerations, the composition of the fluid can be very important. Many relief systems are installed on reactors that contain a complex mixture of chemicals. These chemicals could have either low or high viscosity. They could behave as a Newtonian, non-Newtonian, or even viscoelastic fluid. "Foamy" mixtures are also a concern *(8)*. Foaminess is especially hard to predict because of its extreme sensitivity to surface tension. Each of these factors must be considered to properly determine the fluid pressure-density relationship. The affect many of these factors have on two-phase flows is poorly understood and the subject of on-going research.

TWO-PHASE FLOW MODELS

Many computational methods and models for predicting two-phase mass flux through an ideal nozzle are available in the literature *(1)*. Many of these are designed to apply only to a limited number of the special cases discussed above. These models may be split into two rough categories: "two-fluid" and "homogenous". The "two-fluid" models rely on microscopic conservation equations of mass, energy, and momentum. The balances are generally applied locally to each phase and a model for inter-phase transport is also included. As might be expected, these models are quite computer intensive, require a large amount of system data, and can be applied to only limited cases. These models are more appropriate for detailed mechanism investigations then engineering design calculations. "Homogenous" models, on the other hand, make the "pseudo-homogenous-single-phase-fluid" assumption as discussed previously. This greatly simplifies the calculations and drastically reduces the amount of system information required. As such, these models are far easier to apply to relief system design. In addition, it is not at all evident that "two-fluid" models provide more accurate results than the "homogenous" models. This paper evaluates and verifies two recent "homogenous" models that are able to handle a wide range of the cases discussed above. A detailed discussion of each model follows.

Two-Phase-Homogenous-Equilibrium Model

The Two-Phase-Homogenous-Equilibrium-Model (TPHEM) is a computer model described recently by the Center for Chemical Process Safety (CCPS) *(5)*. TPHEM has two different calculation methods that are of interest: the nozzle case, and the pipe case. In the nozzle case, TPHEM uses numerical integration to solve the ideal nozzle equation:

$$G_{nozzle} = \rho_n \left[2 \int_{P_o}^{P_n} -\frac{dP}{\rho} \right]^{1/2} \tag{2}$$

where G_{nozzle} is the mass flux through the nozzle, P_o is the (upstream) stagnation pressure, P_n is the pressure at the nozzle exit, and ρ is the local two-phase fluid density (ρ_n is the exit density). The integration schemes are based on the papers of Simpson *(9, 10)*. The local two-phase density is calculated by making the "pseudo-homogenous-single-phase-fluid" assumption. The pseudo-fluid density is related to the densities of the gas and liquid phases by

$$\rho = \alpha\, \rho_G + (1-\alpha)\rho_L \tag{3}$$

where α is the volume fraction of the gas phase. The TPHEM model incorporates a number of possible models for the pressure-density relationship (Table 1). Depending on the model chosen, the user must input information from one, two, or three points in the system (referred to as point A, B, and C). For each point, the pressure, the gas/vapor quality (i.e. mass fraction), the gas/vapor density, and the liquid density must be given. The program evaluates the model parameters from the data entered and then numerically integrates the mass flux integral from stagnation to discharge pressure. If the flow is choked, as is often the case, a maximum flux will be found before the discharge pressure is reached at the critical choke pressure. The one-parameter model ("A" in Table 1) is equivalent to the Omega model proposed by Leung *(11)*.

MODEL	POINTS	EQUATION
A	1	$(\rho_o/\rho-1)=a(P_o/P-1)$
B	3	$(\rho_o/\rho-1)=a(P_o/P-1)^b$
C	3	$(\rho_o/\rho-1)=a[(P_o/P)^b-1]$
D	2	$(\rho_o/\rho-1)=a(P_o/P-1)+b(P_o/P-1)^2$
E	2	$x=a_o+a_1P$ $\rho_G=b_oP^{b_1}$ $\rho_L=c_o+c_1P$
F	3	$x=a_o+a_1P+a_2P^2$ $(\rho_{Go}/\rho_G-1)=b_o[(P_o/P)^{b_1}-1]$ $\rho_L=c_o+P+c_2P^2$

Table 1: TPHEM Density Models

TPHEM, Nozzle case also has the capability to account for both mechanical (slip) and thermal nonequilibrium. There are many slip models and little or no experimental confirmation of which is most suitable. Nevertheless, TPHEM includes two methods which require the user to either input the slip ratio ($S = V_G / V_L$) directly or to input a k_s parameter by which S is calculated:

$$S=\left(1-x+x\frac{\rho_L}{\rho_G}\right)^{ks} \qquad (4)$$

Nonequilibrium is included by replacing the local equilibrium quality (x) by a "nonequilibrium" quality (x_{NE}) where $x_{NE} < x$:

$$x_{NE} = x_o + k_{NE}(x^2 - x_o^2) \tag{5}$$

and k_{NE} is an empirical parameter (input by the user), and x_o is the quality at the stagnation condition (in this equation, x_o is zero for subcooled inlet conditions). Simpson asserts that a k_{NE}=1 gives results comparable to the Homogenous Nonequilibrium Model (see below).

As stated previously, TPHEM also has the capability of analyzing pipe flows. To calculate the mass flux, TPHEM, Pipe replaces Eq. 2 with Eq. 6 which is a rearrangement of the Bernoulli's equation for pipe flow:

$$G_{Pipe} = \left[\frac{\int_{P_o}^{P_n} -\rho\, dP + g\Delta Z \rho_{avg}^2}{\ln(\rho_o/\rho_n) + (K_f + 1)/2} \right]^{1/2} \tag{6}$$

where g is the acceleration of gravity, ΔZ is the elevation change, and K_f is the friction loss *(10)*. TPHEM, Pipe includes the effect of friction loss in the pipe and fittings, but does not include terms for slip or nonequilibrium.

Homogenous Nonequilibrium Model

This model has evolved over time and has been recently described by Fauske *(3)*:

$$\frac{G_C}{G_1} = \left[\frac{\left(\frac{G_o}{G_1}\right)^2 + \frac{1}{N_{NE}}}{1 + K_f} \right]^{1/2} \qquad (7)$$

$$N_{NE} = \left(\frac{G_1}{G_3}\right)^2 + \frac{L}{L_e} \quad \textit{for} \quad L \le L_e \qquad (8)$$

$$N_{NE} = 1 \qquad \textit{for} \quad L > L_e \qquad (9)$$

$$G_o = \sqrt{2\,\rho_{Lo}(P_o - P_s)} \qquad (10)$$

$$G_1 = \frac{h_{GLo}}{v_{GLo}\sqrt{T_o C_{p,Lo}}} = G_{ERM} \qquad (11)$$

$$G_3 = \sqrt{2\,\rho_{Lo}(P_s - P_2)} \qquad (12)$$

where,

G_c = two-phase mass flux

G_1 = G_{ERM} = equilibrium rate (see below)

N_{NE} = nonequilibrium parameter

K_f = loss coefficient = 4fL/D (for straight pipe)

L_e = equilibrium length = 10 cm

ρ_{Lo} = liquid density at stagnation condition

P_o = stagnation pressure

P_s = saturation pressure

h_{Glo} = heat of vaporization at stagnation conditions

v_{Glo} = specific volume of gas minus specific volume of liquid at stagnation conditions

T_o = stagnation temperature

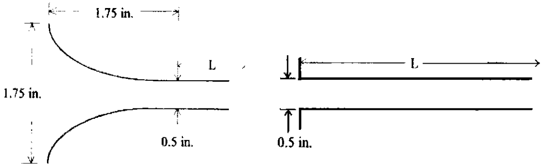
$C_{p,Lo}$ = specific heat of liquid at stagnation conditions

P_2 = discharge pressure (choke pressure if flow is choked)

G_o represents the liquid flow from stagnation to saturation for subcooled inlets. G_1 represents the critical (choked) mass flux resulting from the phase change, and G_3 is the liquid flow from saturation to discharge pressure. For choked flows, P_2 is replaced by the choke pressure, P_c. The nonequilibrium parameter, N_{NE}, represents the delayed flashing for lengths less than 10 cm. The 10 cm criteria comes from studies by Fauske that show that flashing is normally completed within 10 cm of the inlet *(12)*. Fauske's model does not include the effect of slip.

EMPIRICAL DATABASE

The great majority of two-phase flow data comes from studies conducted by the nuclear power industry. Most of this research was driven by work on boiling water reactors and predictions of flow rates in boiler tubes and broken pipes or leaks. One EPRI Report *(13)* compiled the available data sources and then screened the sources for inconsistent and incomplete data sets. Thus, this report gives a comprehensive and reliable database for steam/water two-phase flow in pipes and nozzles. The Sozzi and Sutherland database tabulated in this report provides the most consistent and useful data *(14)*. This database includes a number of different nozzle configurations each attached to a number of different lengths of exit piping. This paper considered nozzles #2 and #3. Nozzle #2 has a well-rounded entrance while nozzle #3 has a square entrance. Figure 2 shows the dimensions of the two configurations and Table 2 lists the length, number of data points, and run conditions (range of qualities and range of stagnation pressures) for each dataset considered in this paper. These datasets represent the "pipe" cases (i.e. those cases with long runs of exit piping). Darby, et. al. has used a methodology very similar to this paper's to compare both TPHEM and HNE to six shorter "nozzle" cases *(6)*. The "nozzle" cases were nozzle #2 with the following exit length: 0 in, 1.5 in, 2.5 in, 4.5 in, 7.5 in, and 9 in. For each run, the following data were reported: stagnation pressure, P_o; stagnation enthalpy, H_o; stagnation quality, x_o; mass flux, G_c; critical pressure, P_c; and critical quality, x_c. The Sozzi and Sutherland data were selected because of the large number of datasets, usefulness of the range of run conditions, and the consistency of the data. The range covers slightly sub-cooled to low quality entrance conditions, thus bracketing the saturation condition. Systems close to saturation are the most difficult to model.



a) Nozzle #2 b) Nozzle #3

Figure 2: Sozzi and Sutherland Nozzle Configurations

LENGTH (in.)	# OF POINTS	X_o RANGE (-)	P_o RANGE (psig)
NOZZLE #2 – Rounded Entrance			
12.5	19	-0.0023 to 0.0043	817 to 985
20.0	13	-0.0020 to 0.0020	831 to 975
25.0	96	-0.0043 to 0.0043	845 to 1023
70.0	81	-0.0043 to 0.0034	877 to 988
NOZZLE #3 – Square Entrance			
7.7	24	-0.0020 to 0.0050	863 to 996
12.8	24	-0.0020 to 0.0040	874 to 999
20.2	17	-0.0030 to 0.0040	879 to 1013
25.2	23	-0.0020 to 0.0040	867 to 1009

Table 2: Overview of Sozzi and Sutherland Database

METHODOLOGY

The eight datasets listed in Table 2 were used as inputs for both the TPHEM and HNE models. The empirical mass flux values were compared to the calculated mass flux for various cases of each of these models. In an effort to determine which parameters and conditions were important and which could be ignored, the datasets were run against a total of seven different cases. Table 3 lists these cases and also lists the numerical value of the switches used in the TPHEM cases. Refer to the CCPS guidelines for detailed instructions on using TPHEM *(5)*.

A number of assumptions were needed to apply the models to the Sozzi and Sutherland database. The quality (vapor mass fraction) was calculated by assuming the flow through the nozzle or pipe is isenthalpic (which gives results very close to the isentropic assumption). Quality is thus defined as

$$x = \frac{H_o - H_L}{H_G - H_L} \tag{13}$$

where, H_o is the initial enthalpy as reported in the dataset, and H_G and H_L refer to the enthalpy of the gas phase and liquid phase, respectively, at the local pressure. TPHEM requires the entrance and exit pressures. Eq. 13, along with given thermodynamic data and steam table properties, was used to generate the three additional data points required by TPHEM. The first data point, point A, was taken to be at saturation conditions. The pressures at points B and C were taken to be 0.75 P_A and 0.50 P_A, respectively. It was

CASE DESCRIPTION	TPHEM PARAMETER[1]					
	IU	IC	IPTS	IV	INES	X
TPHEM, Nozzle w/ friction	3	1	3	1	--	--
TPHEM, Nozzle w/o friction[2]	3	1	3	1	--	--
TPHEM, Nozzle w/ slip[2]	3	3	3	1	2	1.5
TPHEM, Nozzle, KNE Fit[2]	3	3	3	1	11	Variable[3]
TPHEM, Pipe w/ friction	3	1	3	-3	--	--
HNE, w/ friction	--	--	--	--	--	--
HNE, w/o friction[2]	--	--	--	--	--	--

[1] TPHEM Parameters:
- IU – Units
- IC – Case. Option (1) give flow rate output. (3) also gives flow rate output and in addition activates INES, the advanced option flag.
- IPTS – Model and number of data states. (3) corresponds to model F in Table 1 and three data points.
- IV – Input Options. (1) is simple non-viscous input and (-3) is pipe input without viscosity correction
- INES – Advanced Options. Option (2) allows the user to input a slip ratio, S. (11) is used to input a KNE value for nonequilibrium corrections.
- X – Advanced Options Value. The value for either S or k_{NE}, depending on the value of INES (see above).

[2] These cases were so grossly inaccurate they were eliminated from the study.

[3] The KNE fit case attempted to find the value of k_{NE} that would fit the empirical result. Thus, the k_{NE} value varied from 0 to 75.

Table 3: Cases Run and TPHEM Parameters

then straightforward to determine the other required data, local qualities, gas phase densities, and liquid phase densities, by using Eq. 13 and the steam tables. The steam table used was a Microsoft Excel plug-in based on the 1967 ASME code. A consistent method for calculating the friction loss was also needed. Several methods were tried and rejected before a final choice was made. The TPHEM Pipe case calls for a roughness factor. The pipe was assumed to have a roughness of 0.0004 mm, typical of stainless steel. To maintain consistency with the other cases, this roughness value was used to calculate an equivalent loss coefficient, K_f, which is required for the TPHEM, nozzle case and HNE. The Churchhill equation was used to relate the friction factor to the roughness and Reynolds number *(15)*.

$$f = 2\left[\left(\frac{8}{N_{Re}}\right)^{12} + \frac{1}{(A+B)^{3/2}}\right]^{1/12} \qquad (14)$$

$$A = \left[2.457\ \ln\left(\frac{1}{\left(\frac{7}{N_{Re}}\right)^{0.9} + \frac{0.27\varepsilon}{D}}\right)\right]^{16} \qquad (15)$$

$$B = \left(\frac{37{,}530}{N_{Re}}\right)^{16} \qquad (16)$$

$$N_{re} = \frac{DV\rho}{\mu} = \frac{DG_c}{\mu_L} \qquad (17)$$

and

$$K_f = \frac{4fL}{D} \qquad (18)$$

where D is the diameter of the straight pipe, G_c is the mass flux as taken from the dataset, μ_L is the viscosity of water (taken to be constant at 1 cP), and L is the length of the straight pipe. This method was simple to apply and gave consistent results. For nozzle #3, an entrance loss of 0.4 was added to the calculated value from Eq. 18. Finally, to correctly apply the HNE model a value for the choke pressure, P_c was needed for Eq. 12. This value was taken from the Sozzi database when provided, otherwise the value was taken from the output of the TPHEM, Nozzle w/ friction case.

The TPHEM, KNE fit case was different from the other cases in that it required an iterative procedure to arrive at an answer. The aim of the KNE fit test was to find the k_{NE} value that would most closely match the empirical results. k_{NE} values from 0 to 75 were input. This test was later eliminated for the "pipe" cases in this study for reasons discussed below.

With the exception of the KNE fit test, all other cases result in a calculated mass flux, G_{calc}. TPHEM also gave final values for critical pressure and quality. The calculated mass flux value was compared with the empirical value, G_{obs}, for each data point. For each dataset a value for the average G_{calc}/G_{obs} was calculated as well as the standard deviation. These results are shown in Table 4. A plot of the dimensionless mass flux (G_c^*) vs. stagnation quality (x_o) was also produced, where

$$G_c^* = \frac{G_c}{\sqrt{P_o \rho_o}} \tag{18}$$

Refer to Figures 3 - 10 for these plots.

RESULTS AND DISCUSSION

Several of the cases listed in Table 3 were found to be inappropriate and were eliminated from the study. The TPHEM, nozzle without friction and the HNE without friction did not account for the frictional loss and therefore over-predicted the mass flux quite significantly (greater than 20%). This was especially true for the longer pipes. TPHEM, nozzle with slip was shown to return results almost identical to the TPHEM, Nozzle without friction case. This led to the conclusion that, in flashing systems, slip effects are negligible compared to the flashing effects. The TPHEM, KNE fit test was shown to be useless for longer nozzles. In sub-cooled and saturated cases (and even a few two-phase cases) the TPHEM model would become insensitive to the k_{NE} value and would exhibit a limiting behavior at mass fluxes much higher than the empirical data (greater than 15%) for very large values of k_{NE}. It was shown that in cases exhibiting this limiting behavior any k_{NE} value over 75 would have negligible effect on the output. The results from all of the above cases were deemed physically meaningless and are not reported.

The results of the three remaining cases for nozzle #2 and nozzle #3 are shown in Table 4 and Table 5, respectively. The mass flux vs. quality plots are contained in Figures 3 - 10.

DATASET	$(G_{calc} / G_{obs})_{avg}$	STANDARD DEVIATION
NOZZLE #2 – Rounded Entrance – L = 12.5 in.		
TPHEM, Nozzle w/ friction	0.904	0.039
TPHEM, Pipe w/ friction	0.935	0.045
HNE, w/ friction	1.058	0.059
NOZZLE #2 – Rounded Entrance – L = 20.0 in.		
TPHEM, Nozzle w/ friction	0.884	0.066
TPHEM, Pipe w/ friction	0.931	0.071
HNE, w/ friction	1.038	0.041
NOZZLE #2 – Rounded Entrance – L = 25.0 in.		
TPHEM, Nozzle w/ friction	0.938	0.061
TPHEM, Pipe w/ friction	0.997	0.083
HNE, w/ friction	1.109	0.077
NOZZLE #2 – Rounded Entrance – L = 70.0 in.		
TPHEM, Nozzle w/ friction	0.973	0.063
TPHEM, Pipe w/ friction	1.008	0.074
HNE, w/ friction	1.006	0.076

Note: TPHEM, Pipe used a roughness factor of ε=0.0004 mm and other cases used an equivalent K_f (see *Methodology*).

Table 4: Goodness of Fit of Sozzi and Sutherland Nozzle #2

DATASET	$(G_{calc} / G_{obs})_{avg}$	STANDARD DEVIATION
NOZZLE #3 – Square Entrance – L = 7.7 in.		
TPHEM, Nozzle w/ friction	0.853	0.051
TPHEM, Pipe w/ friction	0.929	0.057
HNE, w/ friction	1.004	0.041
NOZZLE #3 – Square Entrance – L = 12.8 in.		
TPHEM, Nozzle w/ friction	0.941	0.143
TPHEM, Pipe w/ friction	1.054	0.075
HNE, w/ friction	1.108	0.095
NOZZLE #3 – Square Entrance – L = 20.2 in.		
TPHEM, Nozzle w/ friction	0.951	0.141
TPHEM, Pipe w/ friction	1.030	0.072
HNE, w/ friction	1.037	0.079
NOZZLE #3 – Square Entrance – L = 25.2 in.		
TPHEM, Nozzle w/ friction	0.968	0.165
TPHEM, Pipe w/ friction	1.1061	0.140
HNE, w/ friction	1.072	0.099

Note: *TPHEM, Pipe* used a roughness factor of ε=0.0004 mm and other cases used an equivalent K_f plus 0.4 for entrance loss (see *Methodology).*

Table 5: Goodness of Fit of Sozzi and Sutherland Nozzle #3

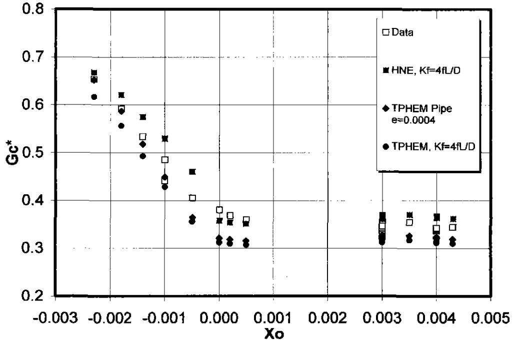


Figure 3: Sozzi and Sutherland Nozzle #2, L = 12.5 in

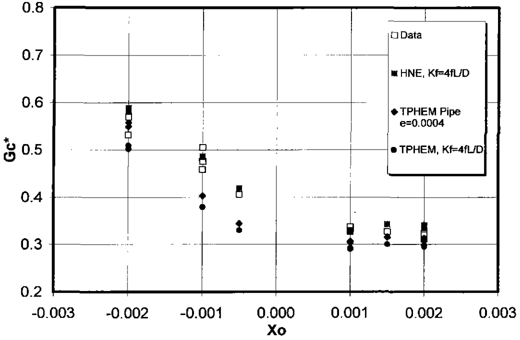


Figure 4: Sozzi and Sutherland Nozzle #2, L = 20.0 in

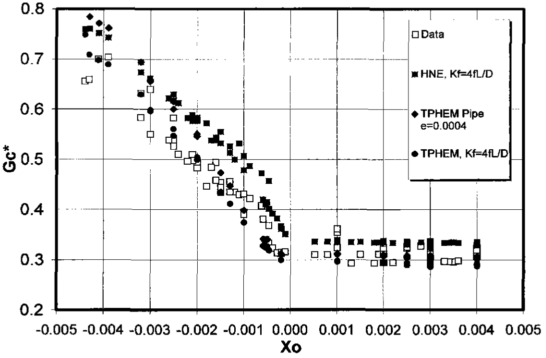


Figure 5: Sozzi and Sutherland Nozzle #2, L = 20.0 in

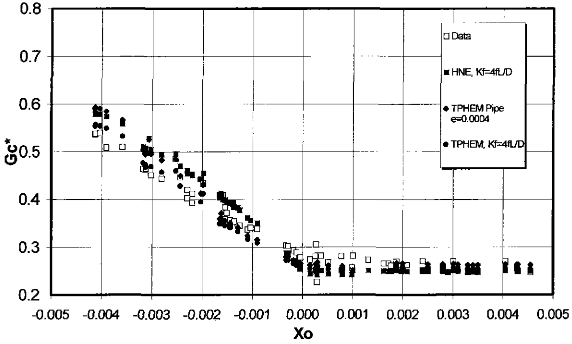


Figure 6: Sozzi and Sutherland Nozzle #2, L = 70.0 in

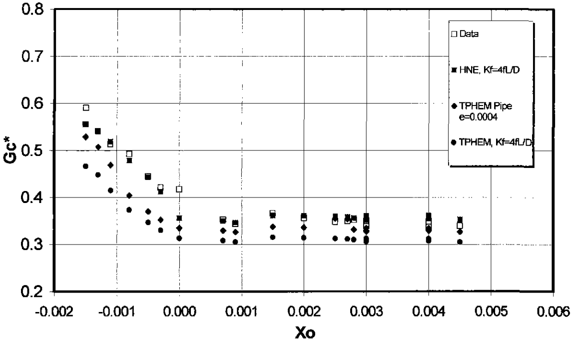


Figure 7: Sozzi and Sutherland Nozzle #3, L = 7.7 in

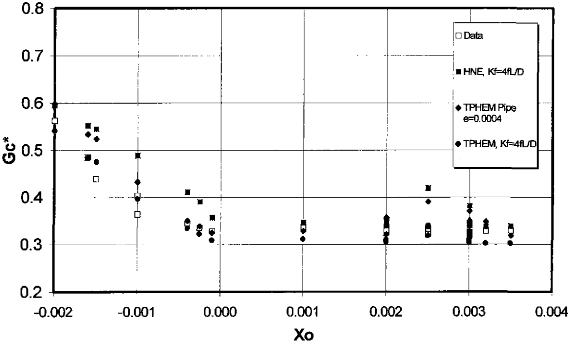


Figure 8: Sozzi and Sutherland Nozzle #3, L = 12.8 in

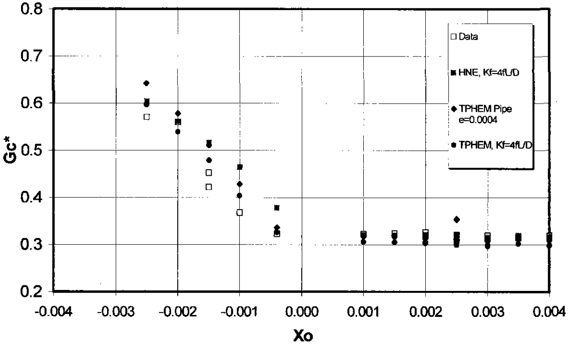


Figure 9: Sozzi and Sutherland Nozzle #3, L = 20.2 in

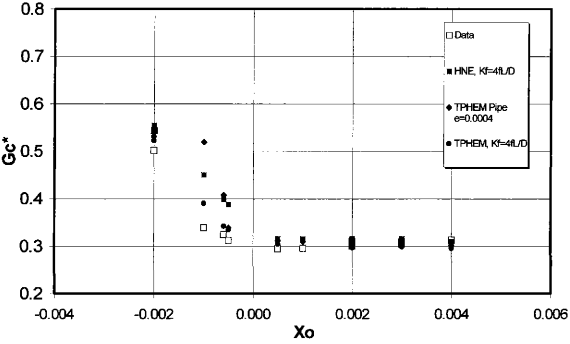


Figure 10: Sozzi and Sutherland Nozzle #3, L = 25.2 in

Inspection of the tables and the plots reveals some general trends. First, all cases seem to improve in accuracy as the exit piping length increases. This might be due to some non-equilibrium effects not accounted for in the models. Second, in all cases two-phase entrance conditions gave better agreement than sub-cooled conditions. All cases tend to over-predict the mass flux for sub-cooled entrance. This inaccuracy worsens as the degree of subcooling increases. Therefore, it would seem both TPHEM and HNE are not accounting for all the vapor generation under sub-cooled conditions. As can be seen in Table 6, overall, TPHEM, Pipe was the most accurate model based on average fit, but it gave the highest deviation (i.e. scatter). HNE tended to over-predict the mass flux under all conditions, while the TPHEM, nozzle case tended to under predict.

MODEL	OVERALL PERCENT DIFFERENCE	STANDARD DEVIATION
TPHEM, nozzle w/ friction	-7.3% (under)	4.2%
TPHEM, pipe w/ friction	0.1% (over)	6.5%
HNE, w/ friction	-5.4% (over)	4.1%

Table 6: Overall Percent Difference for Models

CONCLUSIONS

As previously stated, an understanding of what conditions are important and which can be neglected is vital to applying these models correctly. The results of this paper have shed some light on this area. It was shown that slip is completely dominated by flashing effects in both two-phase and sub-cooled conditions for long nozzles and pipes. Thus, it can be safely eliminated for flashing flows **only** (for frozen flows slip can be very important and should not be eliminated). As previously stated by Fauske, nonequilibrium effects are not appreciable when the exit piping is much greater than 10 cm *(11)*. Friction loss is a very important consideration (especially as the exit piping length increases) and should not be neglected.

The above results show that all three models give results in good agreement with empirical data over the entire range of inlet quality and piping length. As such, these models are acceptable for most design calculations especially relief valve sizing. However, in order to obtain these results a complete knowledge of the fluid's thermodynamic and phase state must be known. In a design problem, the ideal nozzle model would need to corrected by adding a discharge coefficient. Although there is only sparse amounts of two-phase flow through valve data available, what little there is suggests that discharge coefficients close to unity are acceptable *(6)*. Regardless, to better predict this two-phase discharge coefficient further research is needed to investigate the relationship between the ideal nozzle and actual valve geometry.

NOMENCLATURE

a, b, c	empirical parameters for TPHEM density models in Table 1
$C_{p,Lo}$	specific heat of liquid at stagnation conditions (ft lb_f / lb_m °F or N m / kg °C)
D	straight pipe diameter (ft or m)
f	friction factor (-)
g	acceleration of gravity = 32.17 ft / s^2 or 9.8 m / s^2
G	mass flux (lb_m / ft^2 s or kg / m^2 s)
G^*	dimensionless mass flux = $G_c / (P_o \rho_o)^{1/2}$ (-)
h_{Glo}	heat of vaporization at stagnation conditions (ft lb_f / lb_m or N m / kg)
K_d	discharge coefficient (-)
K_f	friction loss coefficient (-)
k_{NE}	TPHEM nonequilibrium parameter (-)
k_s	TPHEM slip parameter (-)
L	nozzle length (ft or m)
L_e	equilibrium length for HNE model = 10 cm
N_{ne}	nonequilibrium parameter defined by Eq. 7 and 8 (-)
P	pressure (lb_f / ft^2 or Pa)
S	slip ratio, ration of gas phase velocity to liquid phase velocity (-)
T	temperature (°F or °C)
x	quality, mass fraction of gas or vapor in mixture (-)
x_{NE}	TPHEM nonequilibrium quality
V	velocity (ft / s or m / s)

ΔZ elevation change (ft or m)

Greek

α volume fraction of gas phase in mixture (-)

ε roughness factor (-)

υ specific volume (ft^3 / lb_m or m^3 / kg)

υ_{Glo} specific volume of gas minus 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)

μ viscosity (lb_m / ft s or kg / m s)

Subscripts

c critical (choked) state

G gas or vapor phase

L liquid phase

n discharge (exit) state.

o stagnation (upstream) state

s saturated state

REFERENCES

1. Darby, R., "Pressure Relief Device Sizing: What's Known and What's Not", presented at the 1998 Plant Process Safety Symposium, Houston, TX (October 26-27, 1998).

2. Fisher, H.G., H.S. Forrest, S.S. Grossel, J.E. Huff, A.R. Muller, J.A. Noronha, D.A. Shaw, and B.J. Tilley, "Emergency Relief System Design Using DIERS Technology", AIChE, NY (1992).

3. Fauske, H.K., "Determine Two-Phase Flows During Releases", *Chemical Engineering Progress*, pp. 55-58 (February 1999).

4. Darby, R., "Viscous Two-Phase Flow in Relief Valves", Phase I Report to DIERS/AIChE (November 1997).

5. Center for Chemical Process Safety, "Guidelines for Pressure Relief and Effluent Handling Systems", AIChE, NY (1998).

6. Darby, R., P. Meiller, J. Stockton, "Relief Sizing for Two-Phase Flow", presented to the 34th Annual Loss Prevention Symposium, Atlanta, GA (March 5-9, 2000).

7. National Board of Boiler and Pressure Vessel Inspectors, "Pressure Relief Device Certifications", Columbus, OH (1997).

8. Fauske, H.K., "Properly Size Vents for Nonreactive and Reactive Chemicals", *Chemical Engineering Progress*, pp. 17-29 (February 2000).

9. Simpson, L.L., "Estimate Two-Phase Flow in Safety Devices", *Chemical Engineering*, pp. 98-102 (August 1991).

10. Simpson, L.L., "Navigating the Two-Phase Maze". *Proceedings of the International Symposium on Runaway Reactions and Pressure Relief Design.* G.A Melhem and H.G. Fisher, eds. AIChE/DIERS, NY (1995).

11. Leung, J.C., "Easily Size Relief Devices and Piping for Two-Phase Flow", *Chemical Engineering Progress*, pp. 28-50. (December 1996).

12. Fauske, H.K., "Flashing Flows or: Some Guidelines for Emergency Releases", *Plant Operations Progress*, v. 4, no. 3, pp. 132-134 (1985).

13. Bannerjee, S., and S. Behring, "A Qualified Database for the Critical Flow of Water", EPRI Report NP-4556 (May 1986).

14. Sozzi, G.L. and W.A. Sutherland, "Critical Flow of Saturated and Subcooled Water at High Pressure", Report NEDO-13418, General Electric Company, San Jose, CA (July 1975).

15. Darby, R. *Chemical Engineering Fluid Mechanics*. Marcel Dekker Press. New York (1996).

VITA

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