A Practical Approach to Capacity Certification

By using certified single-phase flow information in an asymptotic equation form, plants can certify the capacity of pressure-relief valves in two-phase flow

By Dr. Hans K. Fauske

Although certified all-liquid and gas discharge coefficients — as well as all-liquid viscosity correction factors — generally are provided by the valve manufacturers, currently no recognized procedure exists for certifying the capacity of pressure-relief valves (PRVs) in two-phase flow service.

A practical approach to alleviate this problem is to make direct use of the certified single-phase information in the following asymptotic equation form:

$$G_o = \left[\frac{1-x_o}{G^2_{\ell,o}} + \frac{x_o}{G^2_{g,o}} \right]^{-1/2} \qquad (1)$$

Where:
x_o is the stagnation quality.
Single-phase flow rates $G_{\ell,o}$ (kg m^{-2} s^{-1}) and $G_{g,o}$(kg m^{-2} s^{-1}) are given by:

$$G_{1,o} = k_v C_{D,\ell} \sqrt{2(P_o - P_b)\rho_\ell} \qquad (2)$$

and:

$$G_{g,o} = C_{D,g} P_o \left(\frac{M_w}{RT}\right)^{1/2} \left[k \left(\frac{2}{k+1}\right)^{(k+1)/(k-1)} \right]^{1/2} \qquad (3)$$

Where:
$C_{D,\ell}$ = all-liquid discharge coefficient provided by valve manufacturers.
$C_{D,g}$ = all-gas discharge coefficient provided by valve manufacturers.
P_o = stagnation pressure (Pa).
P_b = backpressure (Pa).
ρ_ℓ (kg m^{-3}) = liquid density.

M_w = gas molecular weight.
R (8,314 Pa-m^3/K-kg mole) = gas constant.
T (K) = temperature.
k = isentropic coefficient.

The liquid viscosity correction factor, k_v, can be estimated from the Darby and Molavi correlation (1):

$$k_v = \left(\frac{170\mu_\ell}{G_\ell d} + 1 \right)^{-1/2} \qquad (4)$$

Where:
μ_ℓ (Pa-s) is the liquid viscosity.
d(m) is the valve nozzle diameter.

The form of Equation 1 permits evaluation not only in the limits of $x_o = 0$ and $x_o = 1$, but also in the transition region or two-phase region. A critical test of Equation 1 is provided by the two-component air-liquid valve data reported by Friedel and coworkers (Lenzing and Friedel, 1997; Wieczorek et al., 2001). Typical comparisons between data and predictions from Equation 1 are illustrated in Fig. 1.

Figure 1. Comparison Between Measured Data and Equation 1

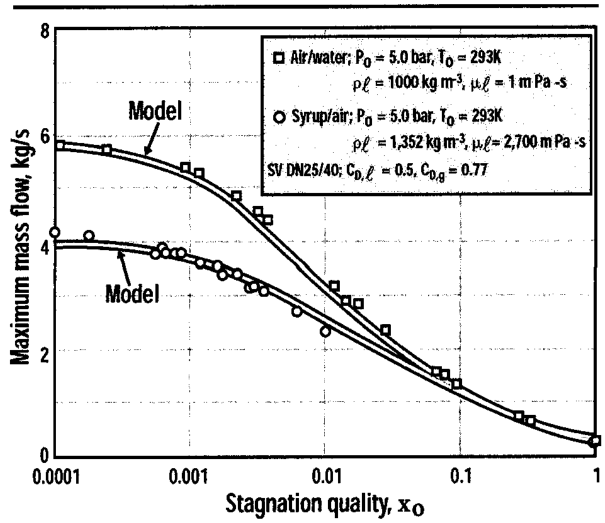
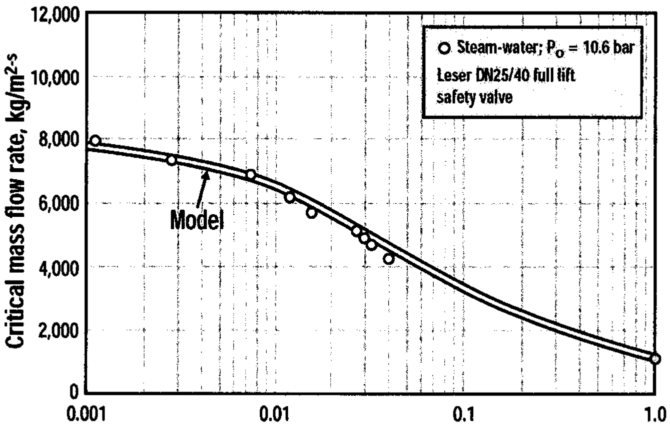


Figure 2. Steam-Water Flashing Data — $C_{D,\ell} = 1.0$, $C_{D,g} = 0.77$ (data from Friedel and co-workers)



For the water case in the figure, with its low viscosity of 1 mPa-s, the value of $k_v = 1.0$. The syrup case shown in the figure ($\mu_\ell = 2{,}700$ m Pa-s) results in $k_v = 0.57$. For both cases, $C_{D,\ell} = 0.5$ and $C_{D,g} = 0.77$, as specified by the valve manufacturer. The mass flows (kg s^{-1}) shown in Fig. 1 correspond to a valve nozzle diameter of d = 0.023 m.

The above approach can be extended to low viscosity ($k_v = 1.0$)* flashing two-phase flows by setting (Fauske, 1999):

$$G_{\ell,o} = \rho_g \lambda \,(Tc)^{-1/2} \qquad (5)$$

Where:

ρ_g (kg m^{-3}) = stagnation vapor density.
λ (J kg^{-1}) = latent heat of evaporation.
c (J kg^{-1} K^{-1}) = liquid specific heat.

The all-liquid discharge coefficient, $C_{D,\ell}$, is set equal to 1.0, consistent with the small pressure drops for asymptotic flashing flows for $x_o = 0$. Fig. 2 provides a comparison with non-viscous steam-water flashing data. This is a key test of the asymptotic equation form with $C_{D,\ell} = 1.0$ (and not 0.5) and $C_{D,g} = 0.77$.

Making direct use of available single-phase information such as certified discharge coefficients and the liquid viscosity correction factor, the asymptotic form of Equation 1 provides consistent and improved comparison with data, and is much easier to use than other proposed models. Furthermore, it provides a practical approach to certifying the capacity of PRVs in two-phase flow service.

Sizing PRV nozzles

Given a specified relief vent rate W (generally obtained from a volumetric balance consideration) and PRV inlet conditions, the minimum orifice flow area is given by:

$$A_{min} = \frac{W}{G_o} \qquad (6)$$

Where:

G_o is the nozzle discharge mass flux (Equation 1) evaluated at 10 percent overpressure. See Fig. 3.

For example, the PRV orifice size needed for a required rate of 0.85 kg s^{-1} of a 1 percent ($x_o = 0.01$) steam-water mixture with a relief set pressure of 35 psig can be determined. At an inlet pressure of 53.2 psia (accounting for 10 percent overpressure), the critical flashing flow rate is $G_o = 2{,}784$ kg m^{-2} s^{-1}. Equation 6 yields $A_{min} = 0.85/2{,}784 = 3.05 \times 10^{-4}$ m^2 (0.475 sq in.). The next available orifice is a G with an area of 0.503 sq in. (3.245×10^{-4} m^2). See Fig. 4.

It should be noted that for subsequent sizing of inlet and outlet piping, the actual discharge rate based on the PRV orifice area should be used as shown in Equation 7:

$$W = \frac{A_n G_o}{F}$$

$$= \frac{3.245 \times 10^{-4} \times 2{,}784}{0.9} \qquad (7)$$

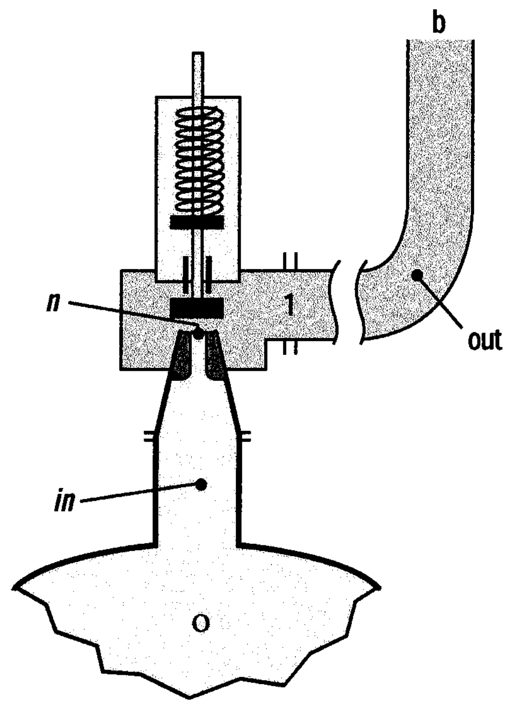
$$= 1.0 \text{ kg s}^{-1}$$

Note that F = 0.9 is a derating factor and is recommended if the American Petroleum Institute (API) nominal orifice areas are used.

Sizing inlet piping

API recommends limiting the inlet nonrecoverable losses to 3 percent or less of the gauge set pressure. The pressure loss can be estimated from:

Figure 3. PRV Schematic: Use of Subscripts



*A possible approach to high-viscosity flashing flows is to treat the flows as turbulent for the entire Reynolds number range (Fauske, 1999).

$$\Delta P_{in.} = \frac{1}{2} K_{in.} \left(\frac{W}{A_{in.}}\right) V_{in.} \quad (8)$$

Where:
$K_{in.}$ is the combined entrance, fitting and straight pipe resistances.
$A_{in.}$ is the inlet pipe flow area.
$V_{in.}$ is the specific volume in the inlet pipe section.

Using the example, the standard inlet pipe connection for a G orifice is 1.5 in. with $A_{in.} = 1.313 \times 10^{-3}\ m^2$. Considering the 3 percent allowable pressure drop, a vapor quality of 0.011 is estimated based on isenthalpic expansion of a steam-water mixture at a pressure of 53.2 psia, and results in a value of $V_{in.} = 6.51 \times 10^{-3}\ m^3kg^{-1}$. The allowable value of $K_{in.}$ follows from:

$$(35 \times 0.03 \times 10^5)/14.5 = \frac{1}{2} K_{in.} \left(\frac{1.0}{1.313 \times 10^{-3}}\right)^2 (6.51 \times 10^{-3}) \quad (9)$$

This equation results in $K_{in.} = 3.8$. Using a friction factor of about 0.005, this is equivalent to about 24 feet of straight pipe, suggesting that the 3 percent inlet rule is easily satisfied.

"API recommends limiting the inlet nonrecoverable losses to 3 percent or less of the gauge set pressure."

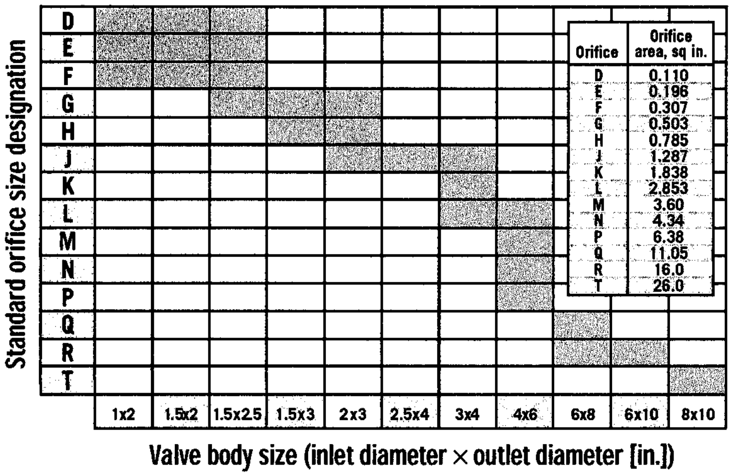
Sizing outlet piping

Given the maximum allowable backpressure, the limitation is the length of the outlet pipe run. The allowable backpressure typically is 10 percent of the gauge set pressure for conventional (unbalanced) valves. An acceptable design that can pass the actual flow, G_{ac}, can be estimated from:

$$G_{ac} \leq G_1 \quad (10)$$

Where G_1 is given by:

Figure 4. Generally Available Relief-Valve Body and Orifice Combinations



Orifice	Orifice area, sq in.
D	0.110
E	0.196
F	0.307
G	0.503
H	0.785
J	1.287
K	1.838
L	2.853
M	3.60
N	4.34
P	6.38
Q	11.05
R	16.0
T	26.0

$$G_1 = \left[\frac{1-x_1}{G_{\ell,1}^2} + \frac{x_1}{G_{g,1}^2} \right]^{-1/2} \quad (11)$$

$$G_{\ell,1} = (1+K_{out})^{-0.39} \rho_{g,1} \lambda_1 (T_1 C_1)^{-1/2} \quad (12)$$

$$G_{g,1} = (1+K_{out})^{-0.39} C_1 (P_1 \rho_{g,1})^{1/2} \left[k \left(\frac{2}{k+1} \right)^{(k+1)/(k-1)} \right]^{1/2} \quad (13)$$

Where:

K_{out} represents the outlet pipe resistances.

C_1 = 1.0 for sonic flows or given by:

$$C_1 = \frac{b^{0.185}}{[1+0.0283\, b^{-3.173}]^{0.1}} \quad (14)$$

Where:

$$b = \left[1 - \frac{P_b}{P_1} \right] \Bigg/ \left[1 - (1+K_{out})^{-0.39} \left(\frac{2}{k+1} \right)^{k/(k-1)} \right] \quad (15)$$

Continuing with the above example, with the G orifice, the smallest standard outlet pipe is 2.5 in. ($A = 3.08 \times 10^{-3}\ m^2$), resulting in a value of G_{ac} = 325 kg $m^{-2}\ s^{-1}$. The maximum allowable backpressure, $P_1 = 35 \times 0.1 + 14.7 = 18.2$ psia. Considering an isenthalpic flash to this pressure from 53.2 psia results in $x_1 = 0.075$.

Given a design requirement of $K_{out} = 5.5$, Equation 12 results in $G_{\ell,1}$ = 621 kg $m^{-2}\ s^{-1}$.

From Equation 13, $G_{g,1}$ = 68.3 kg $m^{-2}\ s^{-1}$, and the corresponding G_1 value is obtained from Equation 11:

$$G_1 = \left[\frac{1-0.075}{621^2} + \frac{0.075}{68.3^2} \right]^{-1/2} = 233\ \text{kg}\ m^{-2}\ s^{-1} \quad (16)$$

This result does not satisfy Inequality 10 and, therefore, the 10 percent rule.

Given that the plant layout calls for a value of $K_{out} = 5.5$, two options are available. The first option is the larger 3-in. standard outlet pipe ($A = 4.77 \times 10^{-3}\ m^2$) available with the G orifice, where the actual flow rate is reduced to $G_{ac} = 1.0/4.77 \times 10^{-3} = 210$ kg $m^{-2}\ s^{-1}$. With this pipe size, Inequality 10 is satisfied, as is the 10 percent rule.

Another option is to consider a balanced valve with an allowable backpressure of about 40 percent of the gauge set pressure ($P_1 = 35 \times 0.4 + 14.7 = 28.7$ psia) and the smallest available 2.5-in. outlet pipe. In this case, $x_1 = 0.049$, $G_{\ell,1}$ = 920 kg $m^{-2}\ s^{-1}$, and $G_{g,1}$ = 139 kg $m^{-2}\ s^{-1}$. It follows from Equation 11 that G_1 = 553 kg $m^{-2}\ s^{-1}$, which satisfies Inequality 10 with a considerable margin. **CP**

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Key:

in. = inches
K = Kelvin
kg = kilograms
ℓ = liquid
m = meters
Mw = molecular weight
Pa = Pascals
psia = pounds per square inch absolute
psig = pounds per square inch gauge
s = seconds
sq in. = square inches

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