~~Technical Report on~~ Pressure-relief System Calculations

API TECHNICAL REPORT 522

SECOND EDITION, XXXX 202X

1 Scope

This technical report is not a design code. It only provides equations and examples for performing relief system calculations. Users are responsible for performing their own calculations and using appropriate references for equations. This report contains a variety of calculation examples for equations ~~and~~, methods, and topics found in API ~~Standard 520, *Sizing, Selection, and Installation of Pressure-relieving Devices, Part II—Installation*~~publications on pressure-relief systems.

2 Normative References

~~The following referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.~~

~~API Standard 520, *Sizing, Selection, and Installation of Pressure-relieving Devices, Part II–Installation*, 7th Edition~~

There are no normative references in this document.

3 Terms and Definitions

3.1
physical acoustic line length
The axial linear distance from the PRV inlet flange to the first significant acoustic reflection point. See API 520, Part II, Annex C.

3.2
speed of sound
The distance traveled per unit time by a sound wave as it propagates through an elastic medium.

3.3
spring constant

A characteristic of a spring that is the ratio of the force affecting the spring to the displacement caused by it.

4 Force Balance Assessment—Vapor Example

4.1 General

This is an example of a force balance assessment from API 520, Part II, Section 7.3.6.d for a simple installation in vapor service using reference [1] and [[4].

4.2 Example Calculation

1) Obtain the valve and installation information.

— Valve: 1½·F·2 bellows, vapor certified.

— Relief fluid phase at inlet of PRV: Vapor.

— Certified orifice area (A_N)(A_N) = 0.3568 in^2.

— Certified coefficient of discharge value (K_d)(K_d) = 0.855 unitless.

— Disc backpressure area (A_{pop})(A_{pop}) = 0.4638 in^2.

— Lift (x_{max})(x_{max}) = 0.182 in.

— Set pressure (P_{set})(P_{set}) = 50 psig.

— Vessel MAWP = 50 psig.

— Blowdown (P_{BD})(P_{BD}) = 7 % of set pressure = 3.5 psi.

— Overpressure (P_{OP})(P_{OP}) = 10 % of set pressure = 5.0 psi.

— PRV mounted to a 2-in. Schedule 40 [inner diameter (ID) of 2.067 in. and cross-sectional area (A_p)(A_p) of 3.354 in^2] inlet line pipe that is routed directly to a pressure vessel with no bends. PRV is not mounted on a process line.

— Physical inlet pipe length from vessel to equipment (L)(L) = 15 ft.

— Physical acoustic line length (L_{pa})(L_{pa}) = 15 ft.

— Non-recoverable frictional inlet pressure loss (ΔP_f)(ΔP_f) = 6 % of set pressure = 3.0 psi.

— Rated capacity of relief valve (based on hexane and certified values) = 2860 lb/hr.

— The total backpressure (P_{back})(P_{back}) = 2.0 psig. This example assumes no effects on set pressure from superimposed backpressure, so the built-up backpressure is equal to the total backpressure. The built-up backpressure is equal to 4 % of the set pressure.

— Total mass of the PRV (M_{PRV})(M_{PRV}) = 45 lb$_m$.

NOTE This is a hypothetical relief valve for which the spring constant, mass in motion, and damped opening time will be estimated.

2) Determine the fluid properties.

At PRV inlet pipe entrance:

— Fluid composition = 100 % n-hexane vapor.

— Relief temperature (T)(T) = 300 °F.

— Relief pressure (P)(P) = (50 psig *1.1+14.7) = 69.7 psia.

— Vapor density @ 55 psig = 0.8323 lb/ft³.

— Compressibility (Z)(Z) = 0.8852.

— Isentropic expansion coefficient (K_v)(K_v) = 0.963.

3) Determine the speed of sound in the vapor at the PRV inlet pipe.

Assume that n-hexane behaves as a real gas and ignore the elasticity of the pipe.

The speed of sound for a real gas is given by Equation (1):

USC units:

$$c = \sqrt{\frac{K_v\, Z\, R \times 144 in^2 ft^{-2} \times 32.174 lb_m fts^{-2} lb_f^{-1} \times T}{M}} = 68.06\sqrt{\frac{K_v\, Z\, R\, T}{M}} \qquad (1)$$

$$c = \sqrt{\frac{K_v Z R \times 144 in^2 ft^{-2} \times 32.174 lb_m ft\ ^{-2} lb_f^{\ -1} \times T}{M}} = 68.06\sqrt{\frac{K_v Z R T}{M}} \qquad (1)$$

SI units:

$$c = \sqrt{\frac{K_v\, Z\, R\, T}{M}}$$

$$c = \sqrt{\frac{K_v Z R T}{M}}$$

Where:

c c	speed of sound (USC units: $ft.s^{-1}$, SI units: $m.s^{-1}$)
K_v K_v	isentropic expansion coefficient (dimensionless)
z z	compressibility factor (dimensionless)
R R	gas const. (USC units: $10.731\ ft^3 psi R^{-1} lb\text{-}mol^{-1}$, SI units: $8314.46\ m^3 Pa K^{-1} kmol^{-1}$)
T T	absolute temperature (USC units: R, SI units: K)
M M	molecular weight (USC units: $lb_m.lbmol^{-1}$, SI units: $kg.kmol^{-1}$)

$$c = 68.06\sqrt{\frac{0.963(0.8852)10.731\, ft^3\ psi\, R^{-1}\, lbmol^{-1}(300\,F + 459.67)}{86.18\, lb_m / lbmol}} = 611\, ft\, s^{-1}$$

4) Estimate the spring constant using Grolmes' equations [1].

$$k_S = \left(\frac{P_{full\ flow}}{P_{set}}\right)\left(\frac{A_{pop}}{A_N}\right)\left[\frac{P_{set}A_N}{x_{\max}}\right]$$

$$\left(\frac{P_{full\ flow}}{P_{set}}\right) = 1.1 \text{ and } \left(\frac{A_{pop}}{A_N}\right) = 1.3$$

$$k_S = (1.1)(1.3)\left[\frac{P_{set}A_N}{x_{\max}}\right] = 1.43\frac{P_{set}A_N}{x_{\max}}$$

$$c = 68.06\sqrt{\frac{0.963(0.8852)10.731ft^3psiR^{-1}lbmol^{-1}(300F + 459.67)}{86.18lb_m/lbmol}} = 611fts^{-1}$$

4) Estimate the spring constant using Grolmes' equations[1].

$$k_S = \left(\frac{P_{fullflow}}{P_{set}}\right)\left(\frac{A_{pop}}{A_N}\right)\left[\frac{P_{set}A_N}{x_{max}}\right]$$

$$\left(\frac{P_{fullflow}}{P_{set}}\right) = 1.1 \text{ and } \left(\frac{A_{pop}}{A_N}\right) = 1.3$$

$$k_S = (1.1)(1.3)\left[\frac{P_{set}A_N}{x_{max}}\right] = 1.43\frac{P_{set}A_N}{x_{max}}$$

Where:

k_S k_S Valve spring constant (USC units: $lb_f in^{-1}$, SI units: $N.m^{-1}$)

$P_{fullflow}$ $P_{fullflow}$ Pressure which rated capacity was calculated (USC units: $psig$, SI units: $Pa(g)$)

T T Set pressure of the relief valve (USC units: $psig$, SI units: $Pa(g)$)

A_N A_N Minimum flow area (USC units: in^2, SI units: m^2)

A_{pop} A_{pop} Disc backpressure area (USC units: in^2, SI units: m^2)

x_{max} x_{max} Maximum valve lift, or restricted lift (USC units: in, SI units: m)

$$k_S = 1.43 \frac{50\, psig * \left(0.3568\, in^2\right)}{0.182\, in} = 140.2\, lb_f / in$$

$$k_S = 1.43 \frac{50\, psig\, \left(0.3568\, in^2\right)}{0.182\, in} = \frac{140.2\, lb_f}{in}$$

5) Estimate the mass in motion from the Grolmes' equations [1].

$$m_D \approx \frac{M_{PRV}}{100}\left(1.8 + 0.022 \times M_{PRV}\right)$$

$$m_D \approx \frac{M_{PRV}}{100}\left(1.8 + 0.022 \times M_{PRV}\right)$$

Where:

m_D m_D mass in motion (USC units: lb_m)

M_{PRV} M_{PRV} total mass of PRV (USC units: lb_m)

$$m_D \approx \frac{45\, lb_m}{100}\left(1.8 + 0.022 \times 45\, lbs\right) = 1.26\, lb_m$$

$$m_D \approx \frac{45 lb_m}{100}\left(1.8 + 0.022 \times 45 lbs\right) = 1.26\ lb_m$$

6) Estimate the PRV opening time.

USC units:

$$\tau_n = \frac{2\pi}{\omega_n} = 2\pi \sqrt{\frac{m_D}{K_S \times 12\, in\, ft^{-1} \times 32.174\, lb_m fts^{-2} lb_f^{-1}}} = 0.3198 \sqrt{\frac{m_D}{K_S}}$$

$$f_n = \frac{1}{\tau_n} = \frac{1}{0.3198}\sqrt{\frac{K_S}{m_D}}$$

$$f_n = \frac{1}{0.3198} = \sqrt{\frac{140.2\, lb_f / in}{1.26\, lb_m}} = 33\, s^{-1}$$

$$\tau_n = \frac{2\pi}{\omega_n} = 2\pi\sqrt{\frac{m_D}{K_S \times 12i \quad {}^{-1} \times 32.174 \quad {}_m ft \quad {}^{-2} lb_f{}^{-1}}} = 0.3198\sqrt{\frac{m_D}{K_S}}$$

$$f_n = \frac{1}{\tau_n} = \frac{1}{0.3198}\sqrt{\frac{K_S}{m_D}}$$

$$f_n = \frac{1}{0.3198} = \sqrt{\frac{140.2 lb_f/in}{1.26 lb_m}} = 33 s^{-1}$$

SI units:

$$\tau_n = \frac{2\pi}{\omega_n} = 2\pi\sqrt{\frac{m_D}{K_S}}$$

$$f_n = \frac{1}{\tau_n} = \frac{1}{2\pi}\sqrt{\frac{K_S}{m_D}}$$

$$\tau_n = \frac{2\pi}{\omega_n} = 2\pi\sqrt{\frac{m_D}{K_S}}$$

$$f_n = \frac{1}{\tau_n} = \frac{1}{2\pi}\sqrt{\frac{K_S}{m_D}}$$

Where:

τ_n τ_n undamped natural period (s)

ω_n ω_n undamped natural frequency ($radians.s^{-1}$)

f_n f_n natural frequency of the valve (s^{-1} or Hz)

K_S K_S spring constant (USC units: $lb_f in^{-1}$, SI units: $N.m^{-1}$)

m_D m_D mass in motion (USC units: lb_m, SI units: kg)

$$t_{open} \approx \frac{1}{2\pi f_n}\sqrt{\frac{2}{A_{pop}/A_N - 1}} \approx \frac{1}{2 f_n}$$

t_{open}

$$t_{open} \approx \frac{1}{2\pi f_n}\sqrt{\frac{2}{A_{pop}/A_N-1}} \approx \frac{1}{2f_n}$$

t_{open} undamped valve opening time (s)

~~$t_{open} \approx \frac{1}{2\left(33s^{-1}\right)} \approx 0.01515\,s$~~

$$t_{open} \approx \frac{1}{2(33\ \ ^{-1})} \approx 0.01515s = 15.15\text{ ms}$$

~~$t_{open,d} = \frac{t_{open}}{\sqrt{1-\zeta^2}}$~~

~~$t_{open,d}$~~ $t_{open,d} = \frac{t_{open}}{\sqrt{1-\zeta^2}}$

$t_{open,d}$ damped valve opening time (s)

~~ζ~~ ζ damping coefficient (dimensionless) = 0.5 based on best fit [1]

~~$t_{open,d} = \frac{0.01515s^{-1}}{\sqrt{1-0.5^2}}$~~

$$t_{open,d} = \frac{0.01\ \ ^{-1}}{\sqrt{1-0.5^2}} = 0.0175\text{ s} = 17.5\text{ ms}$$

7) Calculate the inlet line pressure drop [1, 4] ~~(ΔP_{inlet}).~~ (ΔP_{inlet}).

~~$\Delta P_{inlet} = \Delta P_{wave} + \Delta P_{f,wave}$~~

$$\Delta P_{inlet} = \Delta P_{wave} + \Delta P_{f,wave}$$

Where:

~~ΔP_{inlet}~~ ΔP_{inlet} inlet line pressure drop (USC units: psi, SI units: Pa)

~~ΔP_{wave}~~ ΔP_{wave} wave pressure drop (USC units: psi or SI units: Pa)

~~$\Delta P_{f,wave}$~~ $\Delta P_{f,wave}$ frictional wave pressure drop (USC units: psi or SI units: Pa)

USC units:

~~$\Delta P_{wave}(psi) = \tau \frac{c_0 \times \dot{M}_{open}}{A_p \times 32.174 lb_m fts^{-2} lb_f^{-1}} + \tau^2 \frac{\dot{M}_{open}^{\ 2} \times 144 in^2 ft^{-2}}{2\rho_0 \times 32.174 lb_m fts^{-2} lb_f^{-1} \times A_p^{\ 2}}$~~

~~$$\Delta P_{wave}(psi) = 3.108\times10^{-2}\tau\frac{c_0\times\dot{M}_{open}}{A_p} + 2.2378\tau^2\frac{\dot{M}_{open}{}^2}{\rho_0 A_p{}^2}$$~~

$$\Delta P_{wave}(psi) = \tau\frac{c_0\times\dot{M}_{open}}{A_p\times32.174 lb_m fts^{-2} lb_f{}^{-1}} + \tau^2\frac{\dot{M}_{open}{}^2\times144 in^2 ft^{-2}}{2\rho_0\times32.174 lb_m fts^{-2} lb_f{}^{-1}\times A_p{}^2}$$

$$\Delta P_{wave}(psi) = 3.108\times10^{-2}\tau\frac{c_0\times\dot{M}_{open}}{A_p} + 2.2378\tau^2\frac{\dot{M}_{open}{}^2}{\rho_0 A_p{}^2}$$

SI units:

~~$$\Delta P_{wave}(Pa) = \tau\frac{c_0\times\dot{M}_{open}}{A_p} + \tau^2\frac{\dot{M}_{open}{}^2}{2\rho_0 A_p{}^2}$$~~

~~$\dot{M}_{open}$~~ $$\Delta P_{wave}(Pa) = \tau\frac{c_0\times\dot{M}_{open}}{A_p} + \tau^2\frac{\dot{M}_{open}{}^2}{2\rho_0 A_p{}^2}$$

$\dot{M}_{open}$ mass flow rate of the relief valve at 10 % overpressure, during PRV opening (USC units: $lb_m s^{-1}$, SI units: $kg.s^{-1}$)

~~A_p~~ A_p inlet line inside pipe area (USC units: in^2, SI units: m^2)

~~c_0~~ c_0 speed of sound of the fluid at the pressure source (USC units: $ft.s^{-1}$, SI units: $m.s^{-1}$)

~~ρ_0~~ ρ_0 density of the fluid at the pressure source (USC units: $lb_m ft^{-3}$, SI units: $kg.m^{-3}$)

~~τ~~ τ ratio of wave travel time (t_{wave}) to damped valve opening time ($t_{open,d}$)

~~$$\tau = min\left(\frac{t_{wave}}{t_{open,d}}, 1\right)$$~~

~~$$0 \le \tau \le 1$$~~

~~$$t_{wave} = 2L_{pa}/c_0$$~~

~~L_{pa}~~ $$\tau = min\left(\frac{t_{wave}}{t_{open,d}}, 1\right)$$

$$0 \le \tau \le 1$$

$$t_{wave} = \frac{2L_{pa}}{c_0}$$

L_{pa} acoustic inlet line length (USC units: ft, SI units: m)

~~c_0~~ c_0 speed of sound in relieving fluid (USC units: $ft.s^{-1}$, SI units: $m.s^{-1}$)

~~$t_{open,d}$~~ $t_{open,d}$ damped opening time (s)

~~$t_{wave} = 2(15\,ft)/611\,fts^{-1}$~~ $t_{wave} = 2(15ft)/611fts^{-1} = 0.0491\ \text{sec}$

~~$\tau = min\left(\frac{0.0491}{0.0175} = 2.806, 1\right) = 1$~~

~~$\dot{M}_{open} = \frac{2860\,lb_m\,hr^{-1}}{3600\,s\,hr^{-1}}$~~

$$\tau = min\left(\frac{0.0491}{0.0175} = 2.806, 1\right) = 1$$

$$\dot{M}_{open} = \frac{2860\quad _m hr^{-1}}{3600 s hr^{-1}} = 0.794\ \text{lb}_\text{m}\text{s}^{-1}$$

~~$\Delta P_{wave}(psi) = 3.108\times10^{-2}1\frac{611\,ft\,s^{-1}(0.794lbm\,s^{-1})}{3.354\,in^2} + 2.2378(1)^2\frac{(0.794^2\,lb_m\,s^{-1})^2}{0.8323\,lb_m\,ft^{-3}(3.354\,in^2)^2}$~~

~~$\Delta P_{wave}(psi) = 4.65\,psi$~~

~~$\Delta P_{f,wave} = \tau^2 \Delta P_f$~~

~~$\Delta P_{f,wave} = 1^2(3\,psi) = 3\,psi$~~

~~$\Delta P_{inlet} = 3\,psi + 4.65\,psi = 7.65\,psi$~~

$$\Delta P_{wave}(psi) = 3.108 \times 10^{-2} 1 \frac{611fts^{-1}(0.794lbms^{-1})}{3.354in^2} + 2.2378(1)^2 \frac{(0.794^2 lb_m s^{-1})^2}{0.8323 lb_m ft^{-3}(3.354in^2)^2}$$

$$\Delta P_{wave}(psi) = 4.65psi$$

$$\Delta P_{f,wave} = \tau^2 \Delta P_f$$

$$\Delta P_{f,wave} = 1^2(3psi) = 3psi$$

$$\Delta P_{inlet} = 3psi + 4.65psi = 7.65psi$$

8) Calculate the force balance.

From API 520, Part II, Section 7.3.6:

The total inlet pressure loss + 0.1 * the built-up backpressure ≤ overpressure + blowdown. The adjustment to the built-up backpressure term recognizes that the bellows area isolates a large percentage of the disk area from the backpressure.

In equation form:

~~$\Delta P_{inlet} + 0.1 P_{back} \leq P_{OP} + P_{BD}$~~

$\Delta P_{inlet} + 0.1 P_{back} \leq P_{OP} + P_{BD}$ for bellows valves

Where:

~~ΔP_{inlet}~~ ΔP_{inlet} total inlet pressure loss, including both the wave component and the frictional (non-recoverable) component of pressure loss (USC units: psi; SI units: Pa)

~~P_{OP}~~ P_{OP} overpressure (USC units: psi; SI units: Pa)

~~P_{BD}~~ P_{BD} blowdown (USC units: psi; SI units: Pa)

~~P_{back}~~ P_{back} built-up backpressure at the relief device discharge (USC units: psi, SI units: Pa)

~~$7.65\ psi + 0.1 \cdot 2\ psi \leq 5\ psi + 3.5\ psi$~~

~~$7.85\ psi \leq 8.5\ psi$~~ $7.65psi + 0.1 \cdot 2psi \leq 5psi + 3.5psi$

$7.85psi \leq 8.5psi$ is true; therefore, the PRV passes the force balance assessment.

Note that the force balance may only be one criterion of many in an engineering analysis for the acceptability of elevated inlet pressure drops. Refer to API Standard 520, Part II, Section 7.3.6 for more information.

5 Force Balance Assessment—Liquid Example

5.1 General

This is an example of a force balance assessment from API 520, Part II, Section 7.3.6.d for a simple installation in liquid service ~~using reference~~ [1].

5.2 Example Calculation

1) Obtain the valve and installation information.

- Valve: 1·D·2 bellows, liquid certified.
- Relief fluid phase at inlet of PRV: Liquid.
- Certified orifice area ~~(A_N)~~(A_N) = 0.128 in^2.
- Certified coefficient of discharge value ~~(K_d)~~(K_d) = 0.67 unitless.
- Disc backpressure area ~~(A_{pop})~~(A_{pop}) = 0.1664 in^2.
- Lift ~~(x_{max})~~(x_{max}) = 0.095 in.
- Set pressure ~~(P_{set})~~(P_{set}) = 50 psig.
- Vessel MAWP = 50 psig.
- Blowdown ~~(P_{BD})~~(P_{BD}) = 10 % of set pressure = 5.0 psi.
- Overpressure ~~(P_{OP})~~(P_{OP}) = 10 % of set pressure = 5.0 psi.
- PRV mounted to a 2-in Schedule 40 [inner diameter ~~(ID)~~(ID) of 2.067 in. and cross-sectional area ~~(A_p)~~(A_p) of 3.354 in^2] inlet line pipe that is routed directly to a pressure vessel with no bends. PRV is not mounted on a process line.
- Physical inlet pipe length from vessel to equipment ~~(L)~~(L) = 15 ft.
- Physical acoustic line length ~~(L_{pa})~~(L_{pa}) = 15 ft.

— Non-recoverable frictional inlet pressure loss ~~(ΔP_f)~~(ΔP_f) = 8 % of set pressure = 4.0 psi.

— Rated capacity of relief valve (based on liquid hexane and certified values) = 29 USGPM = 9287 lbm/hr.

— The total backpressure ~~(P_{back})~~(P_{back}) = 4.0 psig. This example assumes no effects on set pressure from superimposed backpressure so the built-up backpressure is equal to the total backpressure. The built-up backpressure is equal to 8 % of the set pressure.

— Total mass of the PRV ~~(M_{PRV})~~(M_{PRV}) = 40 lbm.

NOTE This is a hypothetical relief valve for which the spring constant, mass in motion, and damped opening time will be estimated.

2) Determine the fluid properties.

At PRV inlet pipe entrance:

— Fluid composition = 100 % n-Hexane liquid.

— Relief temperature ~~(T)~~(T) = 100 °F.

— Relief pressure = (50 psig *1.1+14.7) = 69.7 psia

— Liquid density @ 55 psig ~~(ρ)~~(ρ) = 40.3 lb/ft^3

— Specific gravity (60 F/60 F) ~~(S)~~(S) = 0.647

— Real heat capacity ratio ~~(C_p / C_v)~~(C_p/C_v) = 1.31

3) Determine the speed of sound of the liquid at the PRV inlet pipe.

Ignore the elasticity of the pipe.

The speed of sound for a liquid is given by the following equation (API 520, Part II, C.4).

USC units:

~~$$c = 8.62 \times \sqrt{\frac{\kappa_S}{S}}$$~~

$$c = 8.62 \times \sqrt{\frac{\kappa_S}{S}}$$

SI units:

$$c = \sqrt{\frac{\kappa_S}{S}}$$

$$c = \sqrt{\frac{\kappa_S}{S}}$$

The isentropic bulk modulus of elasticity may be calculated as:

$$\kappa_S = \frac{C_P}{C_V} \cdot \kappa_T = \frac{C_P}{C_V} \cdot \frac{1}{\beta_T}$$

$$\kappa_S = \frac{C_P}{C_V} \cdot \kappa_T = \frac{C_P}{C_V} \cdot \frac{1}{\beta_T}$$

Where:

~~κ_s~~ κ_s isentropic bulk modulus of elasticity (USC units: psi, SI units: Pa)

~~C_p / C_v~~ C_p / C_v real heat capacity ratio (dimensionless)

~~κ_T~~ κ_T isothermal bulk modulus of elasticity (USC units: *psi*, SI units: Pa)

~~β_T~~ β_T isothermal compressibility (USC units: psi^{-1}, SI units: Pa^{-1})

~~S~~ S specific gravity of the fluid

The isothermal compressibility can be obtained from a process simulator directly or by calculating the slope of the density to pressure at constant temperature, then dividing by the density. A recommended range is ±20 % of the set pressure at the relief temperature. See Figure 1.

$$\beta_T = -\frac{1}{V}\left(\frac{\delta V}{\delta P}\right)_T = \frac{1}{\rho}\left(\frac{\delta \rho}{\delta P}\right)_T$$

$$\beta_T = -\frac{1}{V}\left(\frac{\delta V}{\delta P}\right)_T = \frac{1}{\rho}\left(\frac{\delta \rho}{\delta P}\right)_T$$

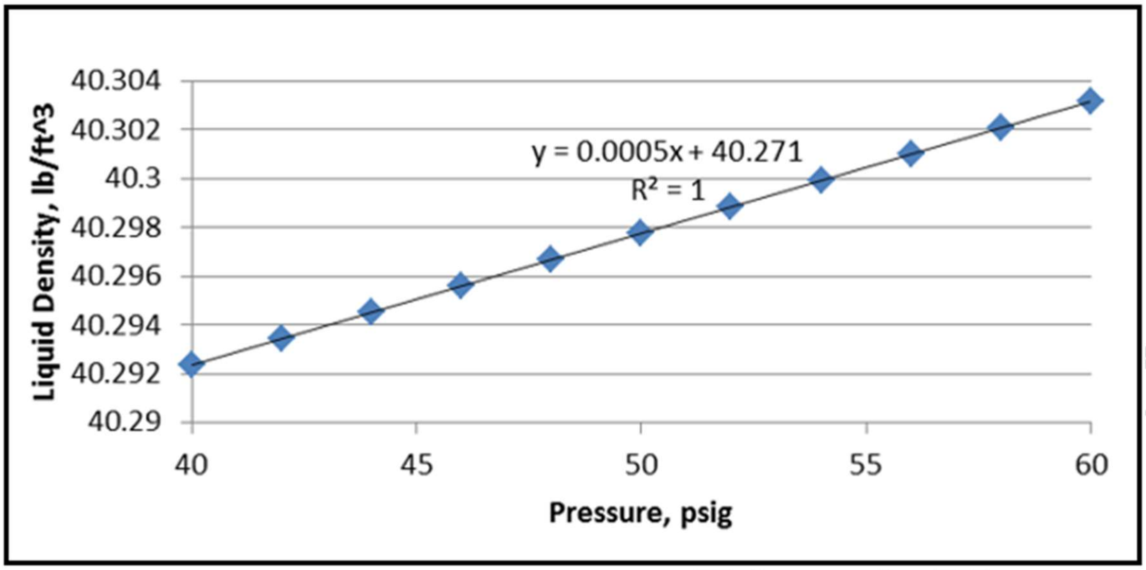


FIGURE ~~1—ISOTHERMAL BULK MODULUS~~1 - ISOTHERMAL BULK MODULUS OF ~~ELASTICITY~~ELASTICITY AT 100-°F

$$\beta_T = \frac{1}{40.3lbft^{-3}}(0.0005)_T = 1.24 \times 10^{-5} psi^{-1}$$

$$\kappa_S = 1.31 \frac{1}{1.24 \times 10^{-5} psi^{-1}} = 105{,}586 psi$$

$$c = 8.62 \times \sqrt{\frac{105{,}586 psi}{0.647}} = 3{,}482 ft\ s^{-1}$$

4) Estimate the spring constant using Grolmes' equations [1].

~~$$\beta_T = \frac{1}{40.3\, lb\ ft^{-3}}(0.0005)_T = 1.24 \times 10^{-5}\ psi^{-1}$$~~

~~$$\kappa_S = 1.31 \frac{1}{1.24 \times 10^{-5}\ psi^{-1}} = 105{,}586\ psi$$~~

~~$$c = 8.62 \times \sqrt{\frac{105{,}586\ psi}{0.647}} = 3{,}482\ ft\ s^{-1}$$~~

~~4) Estimate the spring constant using Grolmes' equations [1].~~

~~$$k_S = \left(\frac{P_{full\ flow}}{P_{set}}\right)\left(\frac{A_{pop}}{A_N}\right)\left(\frac{P_{set} A_N}{x_{max}}\right)$$~~

$$\left(\frac{P_{full\ flow}}{P_{set}}\right) = 1.1 \quad \text{and} \quad \left(\frac{A_{pop}}{A_N}\right) = 1.3$$

$$k_S = (1.1)(1.3)\left[\frac{P_{set}A_N}{x_{max}}\right] = 1.43\frac{P_{set}A_N}{x_{max}}$$

$$k_S = \left(\frac{P_{fullflow}}{P_{set}}\right)\left(\frac{A_{pop}}{A_N}\right)\left(\frac{P_{set}A_N}{x_{max}}\right)$$

$$\left(\frac{P_{fullflow}}{P_{set}}\right) = 1.1 \text{ and } \left(\frac{A_{pop}}{A_N}\right) = 1.3$$

$$k_S = (1.1)(1.3)\left[\frac{P_{set}A_N}{x_{max}}\right] = 1.43\frac{P_{set}A_N}{x_{max}}$$

Where:

k_S k_S valve spring constant (USC units: $lb_f in^{-1}$, SI units: $N.m^{-1}$)

$P_{full\ flow}$ $P_{full\ flow}$ Pressure which actual capacity was calculated (USC units: *psig*, SI units: *Pa(g)*)

P_{set} P_{set} Set pressure of the relief valve (USC units: *psig*, SI units: *Pa(g)*)

A_N A_N Minimum flow area (USC units: in^2, SI units: m^2)

A_{pop} A_{pop} Disc backpressure area (USC units: in^2, SI units: m^2)

x_{max} x_{max} Maximum valve lift, or restricted lift (USC units: *in*, SI units: *m*)

$$k_S = 1.43\frac{50\,psig * \left(0.128\,in^2\right)}{0.095\,in} = 96.3\,lb_f in^{-1}$$

$$k_S = 1.43\frac{50psig * (0.128in^2)}{0.095in} = 96.3lb_f in^{-1}$$

5) Calculate the mass in motion from the ~~Grolmes~~Grolmes' equation [1].

$$m_D \approx \frac{M_{PRV}}{100}\left(1.8 + 0.022 \times M_{PRV}\right)$$

$$m_D \approx \frac{M_{PRV}}{100}\left(1.8 + 0.022 \times M_{PRV}\right)$$

Where:

m_D m_D mass in motion (USC units: lb_m)

M_{PRV} M_{PRV} total mass of PRV (USC units: lb_m)

$$m_D \approx \frac{40\,lb_m}{100}\left(1.8 + 0.022 \times 40\,lbs\right)$$

$$m_D \approx \frac{40lb_m}{100}\left(1.8 + 0.022 \times 40lbs\right) = 1.07\ \text{lb}_\text{m}$$

6) Estimate the PRV opening time.

USC units:

$$\tau_n = \frac{2\pi}{\omega_n} = 2\pi\sqrt{\frac{m_D}{K_S \times 12\,in\ ft^{-1} \times 32.174\,lb_m\,ft\,s^{-2}\,lb_f^{-1}}} = 0.3198\sqrt{\frac{m_D}{K_S}}$$

$$f_n = \frac{1}{\tau_n} = \frac{1}{0.3198}\sqrt{\frac{K_S}{m_D}}$$

$$f_n = \frac{1}{0.3198}\sqrt{\frac{96.3\,lbf\,/\,in}{1.07\,lbm}} = 29.6\,s^{-1}$$

$$\tau_n = \frac{2\pi}{\omega_n} = 2\pi\sqrt{\frac{m_D}{K_S \times 12in\ ft^{-1} \times 32.174lb_m fts^{-2}lb_f^{-1}}} = 0.3198\sqrt{\frac{m_D}{K_S}}$$

$$f_n = \frac{1}{\tau_n} = \frac{1}{0.3198}\sqrt{\frac{K_S}{m_D}}$$

$$f_n = \frac{1}{0.3198}\sqrt{\frac{96.3\ lbf/in}{1.07lbm}} = 29.6s^{-1}$$

SI units:

$$\tau_n = \frac{2\pi}{\omega_n} = 2\pi\sqrt{\frac{m_D}{K_S}}$$

$$f_n = \frac{1}{\tau_n} = \frac{1}{2\pi}\sqrt{\frac{K_S}{m_D}}$$

$$\tau_n = \frac{2\pi}{\omega_n} = 2\pi\sqrt{\frac{m_D}{K_S}}$$

$$f_n = \frac{1}{\tau_n} = \frac{1}{2\pi}\sqrt{\frac{K_S}{m_D}}$$

Where:

τ_n τ_n	undamped natural period (s)
ω_n ω_n	undamped natural frequency ($radians.s^{-1}$)
f_n f_n	natural frequency of the valve (s^{-1} or Hz)
k_S k_S	spring constant (USC units: $lb_f in^{-1}$, SI units: $N.m^{-1}$)
m_D m_D	mass in motion (USC units: lb_m, SI units: kg)

$$t_{open} \approx \frac{1}{2\pi f_n}\sqrt{\frac{2}{A_{pop}/A_N - 1}} \approx \frac{1}{2 f_n}$$

$$t_{open} \; t_{open} \approx \frac{1}{2\pi f_n}\sqrt{\frac{2}{A_{pop}/A_N - 1}} \approx \frac{1}{2 f_n}$$

t_{open}	undamped valve opening time (s)

$$t_{open} \approx \frac{1}{2(29.6)} \approx 0.0169\,s = 16.9\,ms$$

$$t_{open,d} = \frac{t_{open}}{\sqrt{1-\zeta^2}}$$

$$t_{open,d} \; t_{open} \approx \frac{1}{2(29.6)} \approx 0.0169s = 16.9ms$$

$$t_{open,d} = \frac{t_{open}}{\sqrt{1-\zeta^2}}$$

$t_{open,d}$ damped valve opening time (s)

~~ζ~~ ζ damping coefficient (dimensionless) = 0.5 based on best fit

$$\sout{t_{open,d} = \frac{0.0169s^{-1}}{\sqrt{1-0.5^2}}} \; t_{open,d} = \frac{0.0169s^{-1}}{\sqrt{1-0.5^2}} = 0.0195 \text{ s} = 19.5 \text{ ms}$$

7) Calculate the inlet line pressure drop ~~(ΔP_{inlet})~~ (ΔP_{inlet}) [1].

$$\sout{\Delta P_{inlet} = \Delta P_{wave} + \Delta P_{f,wave}}$$

$$\Delta P_{inlet} = \Delta P_{wave} + \Delta P_{f,wave}$$

Where:

~~ΔP_{inlet}~~ ΔP_{inlet} inlet line pressure drop (USC units: psi, SI units: Pa)

~~ΔP_{wave}~~ ΔP_{wave} wave pressure drop (USC units: psi or SI units: Pa)

~~$\Delta P_{f,wave}$~~ $\Delta P_{f,wave}$ frictional wave pressure drop (USC units: psi or SI units: Pa)

USC units:

$$\sout{\Delta P_{wave}(psi) = \tau \frac{c_0 \times \dot{M}_{open}}{A_p \times 32.174 lb_m fts^{-2} lb_f^{-1}} + \tau^2 \frac{\dot{M}_{open}^{\;2} \times 144 in^2 ft^{-2}}{2\rho_0 \times 32.174 lb_m fts^{-2} lb_f^{-1} \times A_p^{\;2}}}$$

$$\sout{\Delta P_{wave}(psi) = 3.108 \times 10^{-2} \tau \frac{c_0 \times \dot{M}_{open}}{A_p} + 2.2378 \tau^2 \frac{\dot{M}_{open}^{\;2}}{\rho_0 A_p^{\;2}}}$$

$$\Delta P_{wave}(psi) = \tau \frac{c_0 \times \dot{M}_{open}}{A_p \times 32.174 lb_m fts^{-2} lb_f^{-1}} + \tau^2 \frac{\dot{M}_{open}^{\;2} \times 144 in^2 ft^{-2}}{2\rho_0 \times 32.174 lb_m fts^{-2} lb_f^{-1} \times A_p^{\;2}}$$

$$\Delta P_{wave}(psi) = 3.108 \times 10^{-2} \tau \frac{c_0 \times \dot{M}_{open}}{A_p} + 2.2378 \tau^2 \frac{\dot{M}_{open}^{\;2}}{\rho_0 A_p^{\;2}}$$

SI units:

$$\sout{\Delta P_{wave}(Pa) = \tau \frac{c_0 \times \dot{M}_{open}}{A_p} + \tau^2 \frac{\dot{M}_{open}^{\;2}}{2\rho_0 A_p^{\;2}}}$$

$$\Delta P_{wave}(Pa) = \tau \frac{c_0 \times \dot{M}_{open}}{A_p} + \tau^2 \frac{\dot{M}_{open}^{\;2}}{2\rho_0 A_p^{\;2}}$$

Where:

$\dot{M}_{open}$ $\dot{M}_{open}$ mass flow rate of the relief valve at 10 % overpressure, during PRV opening (USC units: $lb_m s^{-1}$, SI units: $kg.s^{-1}$)

A_p A_p inlet line inside pipe area (USC units: in^2, SI units: m^2)

c_0 c_0 speed of sound of the fluid at the pressure source (USC units: $ft.s^{-1}$, SI units: $m.s^{-1}$)

ρ_0 ρ_0 density of the fluid at the pressure source (USC units: $lb_m ft^{-3}$, SI units: $kg.m^{-3}$)

τ τ ratio of wave travel time (t_{wave}) to damped valve opening time ($t_{open,d}$)

$$\tau = min\left(\frac{t_{wave}}{t_{open,d}},1\right) \; 0 \le \tau \le 1$$

$$t_{wave} = 2L_P / c_0$$

$$\tau = min\left(\frac{t_{wave}}{t_{open,d}},1\right) \; 0 \le \tau \le 1$$

$$t_{wave} = 2L_P / c_0$$

Where:

L_P L_P Acoustic inlet line length (USC units: ft, SI units: m)

c_0 c_0 speed of sound in relieving fluid (USC units: $ft.s^{-1}$, SI units: $m.s^{-1}$)

$t_{open,d}$ $t_{open,d}$ damped opening time (s)

$$t_{wave} = 2(15\,ft)/3482\,ft\,s^{-1} \quad t_{wave} = 2(15ft)/3482fts^{-1} = 0.0086 \text{ sec}$$

$$\tau = min\left(\frac{0.0086}{0.0195} = 0.44,1\right) = 0.44$$

$$\dot{M}_{open} = \frac{9287\,lbm\,hr^{-1}}{3600\,s\,hr^{-1}} \quad \tau = min\left(\frac{0.0086}{0.0195} = 0.44,1\right) = 0.44$$

$$\dot{M}_{open} = \frac{9287 \quad ^{-1}}{3600shr^{-1}} = 2.58 \text{ lb}_\text{m}\text{s}^{-1}$$

$$\Delta P_{wave}(psi) = 3.108\times10^{-2}(0.44)\frac{3482\,ft\,s^{-1}\left(2.58\,lb_m\,s^{-1}\right)}{3.354\,in^2} + 2.2378(0.44)^2\frac{\left(2.58\,lb_m\,s^{-1}\right)^2}{40.3\,lbm\,ft^{-3}\left(3.354\,in^2\right)^2}$$

$$\Delta P_{wave}(psi) = 36.64\,psi$$

$$\Delta P_{f,wave} = \tau^2 \Delta P_f$$

$$\Delta P_{f,wave} = 0.44^2(4\,psi) = 0.78\,psi$$

$$\Delta P_{inlet} = 36.64\,psi + 0.78\,psi = 37.4\,psi$$

$$\Delta P_{wave}(psi) = 3.108\times10^{-2}(0.44)\frac{3482fts^{-1}(2.58lb_ms^{-1})}{3.354in^2} + 2.2378(0.44)^2\frac{(2.58lb_ms^{-1})^2}{40.3lbmft^{-3}(3.354in^2)^2}$$

$$\Delta P_{wave}(psi) = 36.64psi$$

$$\Delta P_{f,wave} = \tau^2 \Delta P_f$$

$$\Delta P_{f,wave} = 0.44^2(4psi) = 0.78psi$$

$$\Delta P_{inlet} = 36.64psi + 0.78psi = 37.4psi$$

8) Calculate the force balance.

From API 520, Part II, Section 7.3.6:

The total inlet pressure loss + 0.1 * the built-up backpressure ≤ overpressure + blowdown. The adjustment to the built-up backpressure term recognizes that the bellows area isolates a large percentage of the disk area from the back pressure.

In equation form:

$$\Delta P_{inlet} + 0.1P_{back} \le P_{OP} + P_{BD}$$

$$\Delta P_{inlet} + 0.1P_{back} \le P_{OP} + P_{BD}$$

Where:

~~ΔP_{inlet}~~ ΔP_{inlet} total inlet pressure loss, including both the wave component and the frictional (non-recoverable) component of pressure loss (USC units: psi; SI units: Pa)

~~P_{OP}~~ P_{OP} overpressure (USC units: psi; SI units: Pa)

~~P_{BD}~~ P_{BD} blowdown (USC units: psi; SI units: Pa)

~~P_{back}~~ P_{back} built-up backpressure at the relief device discharge (USC units: psi, SI units: Pa)

~~$$37.4\ psi + 0.1 \cdot 4\ psi \leq 5\ psi + 5\ psi$$~~

~~$$37.8\ psi \leq 10\ psi$$~~ $$37.4psi + 0.1 \cdot 4psi \leq 5psi + 5psi$$

$37.8psi \leq 10psi$ is false; therefore, the PRV does not pass the force balance assessment.

6 Critical Line Length—Vapor Example

6.1 General

This is an example of a critical line length calculation for a simple installation in vapor service ~~using references~~ [1] ~~and~~ [2].

6.2 Example

The critical line length calculation is calculated based on the Izuchi criteria [1][2]. It can be expressed as:

~~$$L_{crit} = \frac{c_0}{4 f_n}\sqrt{\frac{x}{x + x_0}}$$~~ $$L_{crit} = \frac{c_0}{4f_n}\sqrt{\frac{x}{x+x_0}}$$ (USC units: ft, SI units: m)

Where:

~~c_0~~ c_0 speed of sound in fluid (USC units: $ft.s^{-1}$, SI units: $m.s^{-1}$)

~~f_n~~ f_n natural frequency of the valve (s^{-1} or Hz)

~~x~~ x valve lift (USC units: in, SI units: m)

~~x_o~~ x_o initial compression of the spring (USC units: in, SI units: m)

When the overpressure is set at 10 %, the initial compression of the spring (x_o) can be simplified. As a result, the critical length calculation becomes:

~~$$L_{crit} = \frac{c_0}{4 f_n} \sqrt{\frac{1.43x}{1.43x + x_{max}}}$$~~

$$L_{crit} = \frac{c_0}{4 f_n} \sqrt{\frac{1.43x}{1.43x + x_{max}}}$$

The equation above shows how the critical length (L_{crit}) is dependent on the valve lift (x).

Where:

~~c_0~~ c_0 speed of sound in fluid (USC units: $ft.s^{-1}$, SI units: $m.s^{-1}$)

~~f_n~~ f_n natural frequency of the valve (s^{-1} or Hz)

~~x_{max}~~ x_{max} maximum valve lift (USC units: in, SI units: m)

~~x~~ x valve lift (USC units: in, SI units: m)

1) Determine the properties.

~~c_0~~ c_0 611 ft s^{-1} from Section 4 example

~~f_n~~ f_n 33 s^{-1} from Section 4 example

~~x_{max}~~ x_{max} 0.182 in. from Section 4 example

Assume that the valve is at full lift.

~~$$x = x_{max} = 0.182 in$$~~

$$x = x_{max} = 0.182 in$$

2) Calculate critical line length [1].

~~$$L_{crit} = \frac{c_0}{4 f_n} \sqrt{\frac{1.43x}{1.43x + x_{max}}}$$~~

~~$$L_{crit} = \frac{611\ ft\ s^{-1}}{4(33\ s^{-1})} \sqrt{\frac{1.43(0.182\ in)}{1.43(0.182\ in) + 0.182\ in}}$$~~

$$L_{crit} = \frac{c_0}{4 f_n} \sqrt{\frac{1.43x}{1.43x + x_{max}}}$$

$$L_{crit} = \frac{611fts^{-1}}{4(33s^{-1})}\sqrt{\frac{1.43(0.182in)}{1.43(0.182in)+0.182in}}$$

$$L_{crit} = 3.55 \text{ ft}$$

$$L_p = 15 \text{ ft}$$

Since $L_p > 1.2 * L_{crit}$, $L_p > 1.2 * L_{crit}$, the valve is expected to cycle at full lift.

7 Flash Calculation Example

7.1 General

Figure 2 is an example of an equilibrium-phase diagram for a given liquid having the following evolution.

a) Initial conditions:
— 0 % of the liquid is vaporized;

— T_a = 256 °C (492 °F);

— absolute pressure p_a = 2861 kPa (415 psi).

b) The conditions after Step n = 1 are as follows:
— 5 % of the liquid is vaporized and 95 % of the liquid remains;

— T_1 = 252 °C (486 °F);

— absolute pressure p_1 = 2496 kPa (362 psi).

c) The development of the diagram continues stepwise until depressuring is completed at the following conditions after Step n = 5 at Item b):
— 30 % of the liquid is vaporized and 70 % of the liquid remains;

— T_b = 238 °C (461 °F);

— absolute pressure p_b = 1482 kPa (215 psi).

For convenience, the mass percent vaporized was assumed to be equal to the volume percent vaporized. By assuming that an incremental part of the liquid (e.g. 5 %) was vaporized during each step, the change in the liquid temperature can be computed using Equation (2).

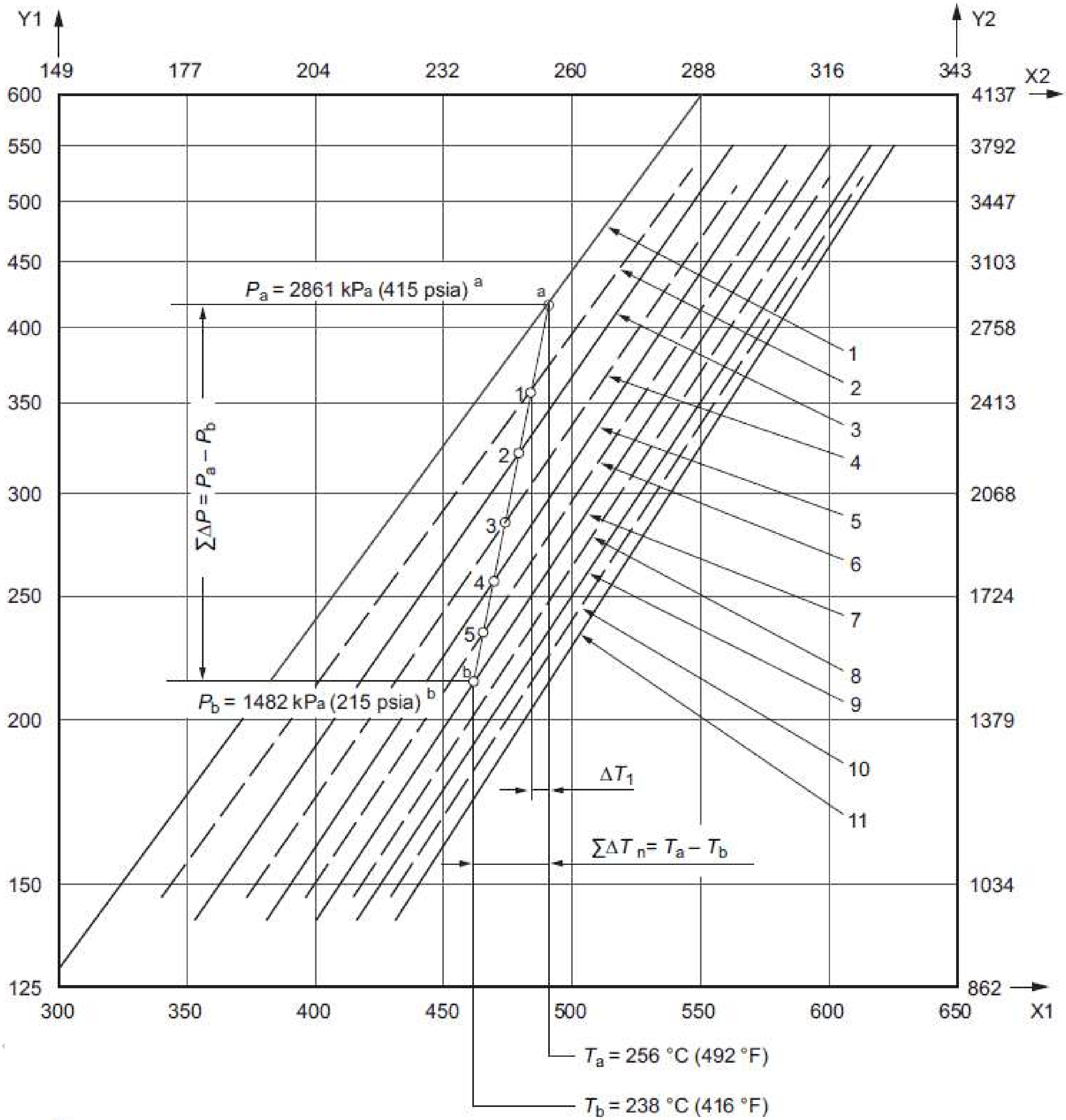
$$(\Delta T_n)_i = \left[\frac{L_n(\Delta q_{\mathrm{m,v}} \times t)_n}{\left((q_{\mathrm{m,L}} \times t)_{n-1} - (\Delta q_{\mathrm{m,v}} \times t)_n\right) \times (C_p)_n}\right]_i \quad (2)$$

Because the remaining liquid has a saturation temperature and pressure along the 5 %-vaporized line of the phase diagram and the temperature change has been determined using Equation (2), the pressure change is also known. The process is repeated in incremental steps until the pressure, p_b, at the end of the depressuring period is obtained. In Figure 2, the desired end pressure is reached when the mass fraction, X_i, of the initial liquid in the vessel, i, that has been vaporized is ≈0.30. Substituting this value of X_i into Equation (4) for the last term in Equation (3)

$$(q_{\mathrm{m,v}} \times t)_i \approx \left[(q_{\mathrm{m,a}} \times t)_i - \frac{Q_i \times t}{2\lambda_i}\right] \times \left[\frac{2(C_p)_i (T_a - T_b)_i}{2\lambda_i + (C_p)_i (T_a - T_b)_i}\right] \quad (3)$$

gives the estimated mass of liquid flashed as a result of the depressuring from the vessel, i, of the system during a simultaneous fire.

$$(q_{\mathrm{m,v}} \times t)_i \approx \left[(q_{\mathrm{m,a}} \times t)_i - \frac{Q_i \times t}{2L_i}\right] w_i \quad (4)$$



Key

X1	liquid temperature, expressed in °F	1	0 % vaporized	7	30 % vaporized
X2	liquid temperature, expressed in °C	2	5 % vaporized	8	35 % vaporized
Y1	pressure, expressed in psig	3	10 % vaporized	9	40 % vaporized
Y2	pressure, expressed in kPa	4	15 % vaporized	10	45 % vaporized
[a]At start.		5	20 % vaporized	11	50 % vaporized
[b]At end.		6	25 % vaporized		

Figure 2—Equilibrium Phase Diagram for a Given Liquid

7.2 Flash Calculation Nomenclature

The variables used in the preceding equations are defined as follows:

C_p is the average specific heat of the liquid, expressed in kJ/kg·K (Btu/lb·°R);

L is the average latent heat of the liquid, expressed in kJ/kg (Btu/lb);

m is the mass of liquid or vapor, expressed in kg (lb);

$\dot{m}$ is the mass flow rate per unit time;

M is the relative molecular mass of the vapor;

p is the absolute pressure, expressed in kPa (psi);

q_m is the vapor mass flow rate, expressed in kg/h (lb/h);

Q is the total heat absorption (input) to the wetted surface, expressed in kJ/h (Btu/h);

T is the absolute temperature of the liquid or vapor, expressed in K (°R);

t is the depressuring time interval, expressed in hours (usually assumed to be 0.25 h);

V is the volume available for the vapor, expressed in m^3 (ft^3);

V_T is the total number of vessels in the depressuring system;

w is the mass fraction of the initial liquid in the system vaporized as a result of depressuring, dimensionless;

Z is the compressibility factor, dimensionless;

l is the latent heat of the liquid, expressed in kJ/kg (Btu/lb);

D represents a difference, for example, as in $DT_n = T_{n-1} - T_n$.

The subscripts used in the equations are defined as follows:

a is the original condition at the start of the depressuring time interval, assumed to be the saturated vapor-liquid equilibrium condition with respect to temperature and pressure;

b is the depressurized condition at the end of the depressuring time interval;

d relates to the density change of the vapor due to pressure reduction;

f relates to vaporization from the fire;

i relates to an individual vessel of the system if more than one vessel is involved and requires separate consideration because of differing fluid properties, insulation for fire effect, or related factors;

L relates to liquid;

n is the nth depressuring step of many steps between the original condition and the depressurized condition;

n – 1 is the depressuring step preceding Step n;

v relates to liquid flash or vapor generated from pressure reduction;

8 Flare Knockout Drum Sizing Example

The following sample calculations have been limited to the simplest of the designs: horizontal drum with the vapor entering one end of the vessel and exiting at the top of the opposite end (no internal baffling); and vertical drum with the vapor inlet nozzle entering the vessel radially and the outlet nozzle at the top of the vessel's vertical axis (with the inlet stream baffled to direct the flow downward).

The calculations for a horizontal drum with the vapor entering at each end on the horizontal axis and a center outlet; and a horizontal drum with the vapor entering in the center and exiting at each end on the horizontal axis are similar, with one-half the flow rate determining one-half the vessel length.

The normal calculations are used for vertical vessel with a tangential nozzle and are not duplicated here.

The following conditions are assumed.

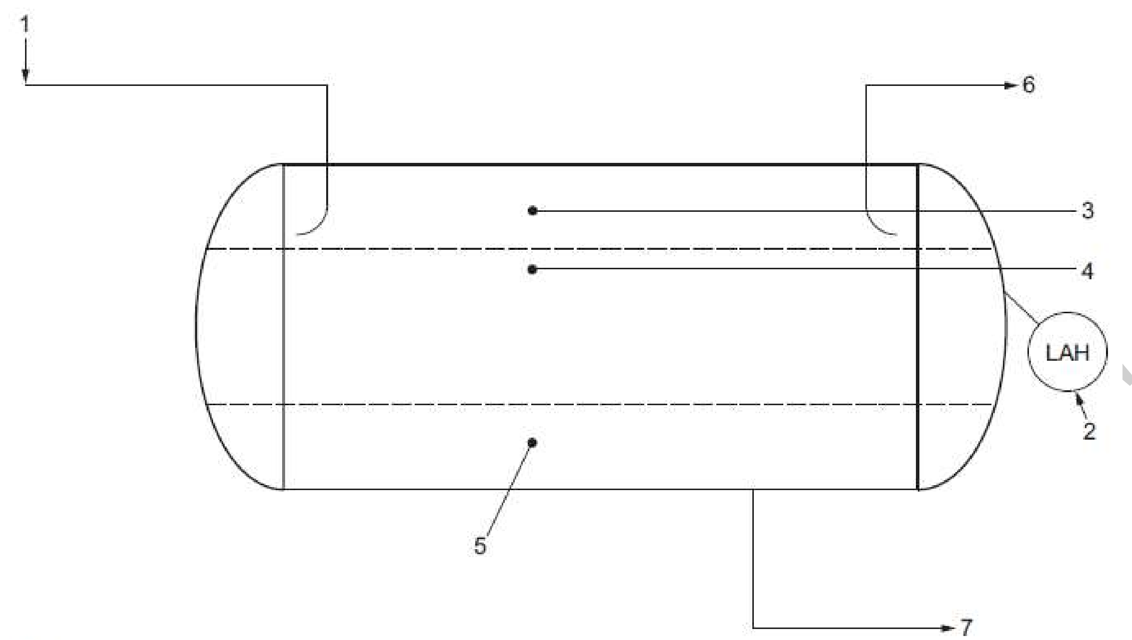
1. A single contingency results in the flow of 25.2 kg/s (200,000 lb/h) of a fluid with a liquid density of 496.6 kg/m^3 (31 lb/ft^3) and a vapor density of 2.9 kg/m^3 (0.18 lb/ft^3), both at flowing conditions.

2. The gauge pressure is 13.8 kPa (2 psi), and the temperature is 149 °C (300 °F).

3. The viscosity of the vapor is 0.01 mPa·s (0.01 cP).

4. The fluid equilibrium results in 3.9 kg/s (31,000 lb/h) of liquid and 21.3 kg/s (169,000 lb/h) of vapor.

In addition, 1.89 m^3 (500 U.S. gal) of storage for miscellaneous drainings from the units is desired. The schematic in Figure 3 applies. The droplet size selected as allowable is 300 µm (0.012 in.) in diameter.



Key

1 vapor and liquid pressure relief valve releases
2 level instrument to indicate when the stipulated slop and drain volume becomes liquid full
3 minimum vapor space for dropout velocity
4 liquid holdup from pressure relief valves and other emergency releases
5 slop and drain liquid
6 to flare
7 pump-out

Figure 3—Flare Knockout Drum

The vapor rate, R_v, in actual m³/s (ft³/s), is determined as follows.

In SI units:

$$R_v = \frac{21.3}{2.9} = 7.34\ m^3/s$$

In USC units:

$$R_v = \frac{169{,}000}{3600 \times 0.18} = 261\ ft^3/s$$

The drag coefficient, C, is determined from Figure 4 using Equation (5) or Equation (6).

In SI units:

$$C(\mathrm{Re})^2 = \frac{0.13 \times 10^8 \times 2.9 \times (0.0003)^3 \times (496.6 - 2.9)}{(0.01)^2} = 5025 \qquad (5)$$

In USC units:

$$C(Re)^2 = \frac{0.95 \times 10^8 \times 0.18 \times (0.000984)^3 \times (31 - 0.18)}{(0.01)^2} = 5021 \qquad (6)$$

From Figure 4, C = 1.3.

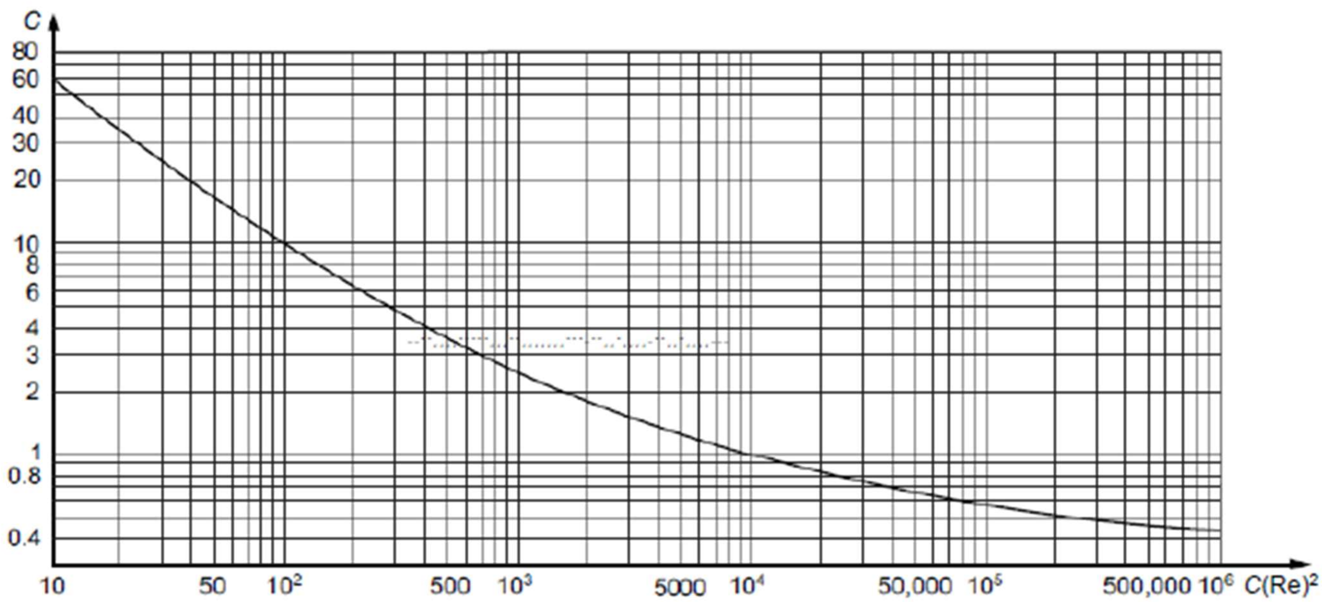


Figure 4—Determination of Drag Coefficient

A horizontal vessel with an inside diameter, D_I, and an assumed cylindrical length, L. This gives the following total cross-sectional area, A_t:

$$A_t = \frac{\pi}{4} \cdot (D_i)^2 \qquad (7)$$

Liquid holdup for a 30 min release from the single contingency, in addition to the slop and drain volume, is desired. The volume in the heads is neglected for simplicity. The liquid holdup required, A_{L1} in m³ (ft³), is calculated as follows.

The slop and drain volume of 1.89 m³ (500 US gal, 66.8 ft³) occupies a bottom segment as follows.

In SI units:

$$A_{L1} = 1.89\left(\frac{1}{L}\right) \qquad (8)$$

In USC units:

$$A_{L1} = 66.8\left(\frac{1}{L}\right) \qquad (9)$$

A total of 3.9 kg/s (31,000 lb/h) of condensed liquids with a density of 496.6 kg/m³ (31 lb/ft³) accumulated for 30 min occupies a cross-sectional segment as follows.

In SI units:

$$A_{L2} = \frac{3.9}{496.6} \times (30 \times 60)\left(\frac{1}{L}\right) \qquad (10)$$

In USC units:

$$A_{L2} = \frac{31{,}000}{31} \times \frac{30}{60}\left(\frac{1}{L}\right) \qquad (11)$$

The cross-sectional area remaining for the vapor flow is expressed as follows:

$$A_v = A_t - (A_{L1} + A_{L2}) \tag{12}$$

The vertical depths of the liquid and vapor spaces are determined using standard geometry and the total drum diameter, h_t, calculated by Equation (13):

$$h_t = h_{L1} + h_{L2} + h_v \tag{13}$$

Where:

h_{L1} is the depth of slops and drains;

$(h_{L1} + h_{L2})$ is the depth of all liquid accumulation;

h_v is the remaining vertical space for the vapor flow.

The adequacy of the vapor space is verified by determining the liquid dropout time, θ, using Equation (14) or Equation (15).

In SI units:

$$\theta = \left(\frac{h_v}{100}\right)\left(\frac{1}{u_c}\right) \tag{14}$$

In USC units:

$$\theta = \left(\frac{h_v}{12}\right)\left(\frac{1}{u_c}\right) \tag{15}$$

Where:

θ is the liquid dropout time, expressed in s;

h_v is the vertical drop available for liquid dropout, expressed in cm (in.);

u_c is the dropout velocity, expressed in m/s (ft/s).

The velocity of N vapor passes, based on one vapor pass, is determined from Equation (16) and Equation (17) for a volume flow rate of 7.34 m^3/s (260 ft^3/s).

In SI units:

$$u_v = \left(\frac{7.34}{N}\right)\left(\frac{1}{A_v}\right) \tag{16}$$

In USC units:

$$u_v = \left(\frac{260}{N}\right)\left(\frac{1}{A_v}\right) \tag{17}$$

Where:

A_v is the cross-sectional area, expressed in m^2 (ft^2);

N is the number of vapor passes;

u_v is the vapor velocity, expressed in m/s (ft/s).

The flow path length required, L_{min}, is determined as follows:

$$L_{\min} = u_v \times \theta \times N \quad (18)$$

To ensure an adequate flow path length, the resulting estimated L_{min} would be less than or equal to the above assumed flow path length (i.e. distance between the inlet and outlet nozzle), L; otherwise, the calculation is repeated with a newly assumed flow path length.

Table 1 and Table 2 summarize the preceding calculations for one pass for horizontal drums with various inside diameters to determine the most economical drum size. Drum diameters in 15 cm (6 in.) increments are assumed, in accordance with standard head sizes. It can be concluded from Table 1 and Table 2 that:

1. all of the drum sizes above fulfill the design requirements;

2. the most suitable drum size is selected according to the design pressure, material requirements, and corrosion allowance as well as layout, transportation, and other considerations;

3. the choice of two-pass flow, as shown in Figure 3, is optional.

Table 1—Optimizing the Size of a Horizontal Knockout Drum (SI Units)

Trial No.	D_1 [a]	L [b]	Cross-sectional Area m²				Vertical Depth of Liquid and Vapor Spaces cm				θ [c]	u_v [d]	L_{min} [e]
			A_t	A_{L1}	A_{L2}	A_v	h_{L1}	$h_{L1} + h_{L2}$	h_v	h_t			
1	2.44	5.79	4.67	0.33	2.45	1.89	30	140	104	244	1.45	3.9	5.6
2	2.29	6.25	4.10	0.30	2.27	1.53	29	137	91	229	1.28	4.8	6.2
3	2.13	6.86	3.57	0.28	2.07	1.22	28	133	81	213	1.13	6.0	6.7
4	1.98	7.62	3.08	0.25	1.86	0.98	27	128	70	198	0.98	7.5	7.4

1. The data in this table are in accordance with the example given in text for one-pass vapor flow.
1. The values in this table are rounded-off conversions of the values in Table 2.

[a] D_1 is the assumed drum inside diameter, expressed in meters.
[b] L is the assumed drum cylindrical length, expressed in meters.
[c] θ is the liquid dropout time, expressed in seconds.
[d] u_v is the vapor velocity, expressed in meters per second.
[e] L_{min} is the required flow path length, expressed in meters.

Table 2—Optimizing the Size of a Horizontal Knockout Drum (USC Units)

Trial No.	D_1 [a]	L [b]	Cross-sectional Area ft²				Vertical Depth of Liquid and Vapor Spaces in.				θ [c]	u_v [d]	L_{min} [e]
			A_t	A_{L1}	A_{L2}	A_v	h_{L1}	$h_{L1} + h_{L2}$	h_v	h_t			
1	8.0	19.0	50.26	3.52	26.32	20.43	11.75	55.0	41.0	96	1.45	12.76	18.5
2	7.5	20.5	44.17	3.26	24.39	16.52	11.43	54.0	36.0	90	1.28	15.78	20.2
3	7.0	22.5	38.49	2.97	22.22	13.29	11.00	52.3	31.7	84	1.13	19.62	22.1
4	6.5	25.0	33.18	2.67	20.00	10.51	10.5	50.4	27.6	78	0.98	24.82	24.3

1. The data in this table are in accordance with the example given in the text for one-pass vapor flow.

[a] D_1 is the assumed drum inside diameter, expressed in feet.
[b] L is the assumed drum cylindrical length, expressed in feet.
[c] θ is the liquid dropout time, expressed in seconds.
[d] u_v is the vapor velocity, expressed in feet per second.
[e] L_{min} is the required flow path length, expressed in feet.

If a vertical vessel is considered, the vapor velocity is equal to the dropout velocity, which is 0.71 m/s (2.35 ft/s). The volume flow rate is 7.34 m³/s (260 ft³/s). The required cross-sectional area, A_{cs}, of the drum, in m² (ft²) is determined as follows.

In SI units:

$$A_{cs} = \frac{7.34}{0.71} = 10.3\ m^2 \qquad (19)$$

In USC units:

$$A_{cs} = \frac{260}{2.35} = 110.6\ \text{ft}^2 \qquad (20)$$

The drum diameter, D, is determined as follows.

In SI units:

$$D = \sqrt{10.3 \times \frac{4}{\pi}} = 3.6\ m \qquad (21)$$

In USC units:

$$D = \sqrt{110.6 \times \frac{4}{\pi}} = 11.9\ \text{ft} \qquad (22)$$

9 Vent Stack Sizing Example

For this calculation, the following conditions are assumed: The maximum-relief rate, $\dot{m}$, is 31.5 kg/s (250,000 lb/h). The relative molecular mass of the vapor, M, is 44. The absolute temperature of the vapor just inside the exit point from the vent stack, T, is 361 K (650 °R). The exit velocity, v, is 150 m/s (500 ft/s). The absolute pressure of the vapor just inside the exit point from the vent stack, p, is 101 kPa (14.7 psi). The gas constant, R, is 8.3 in SI units (10.7 in USC units). The density, □, is then calculated as in Equation (23) or Equation (24).

In SI units:

$$\rho = \frac{M \times p}{R \times T} = \frac{44 \times 101}{8.3 \times 361} = 1.48\ \frac{\text{kg}}{m^3} \qquad (23)$$

In USC units:

$$\rho = \frac{M \times p}{R \times T} = \frac{44 \times 14.7}{10.7 \times 650} = 0.1\ \frac{\text{lb}}{\text{ft}^3} \qquad (24)$$

The tip area, A_T, is determined by Equation (25) and Equation (26).

In SI units:

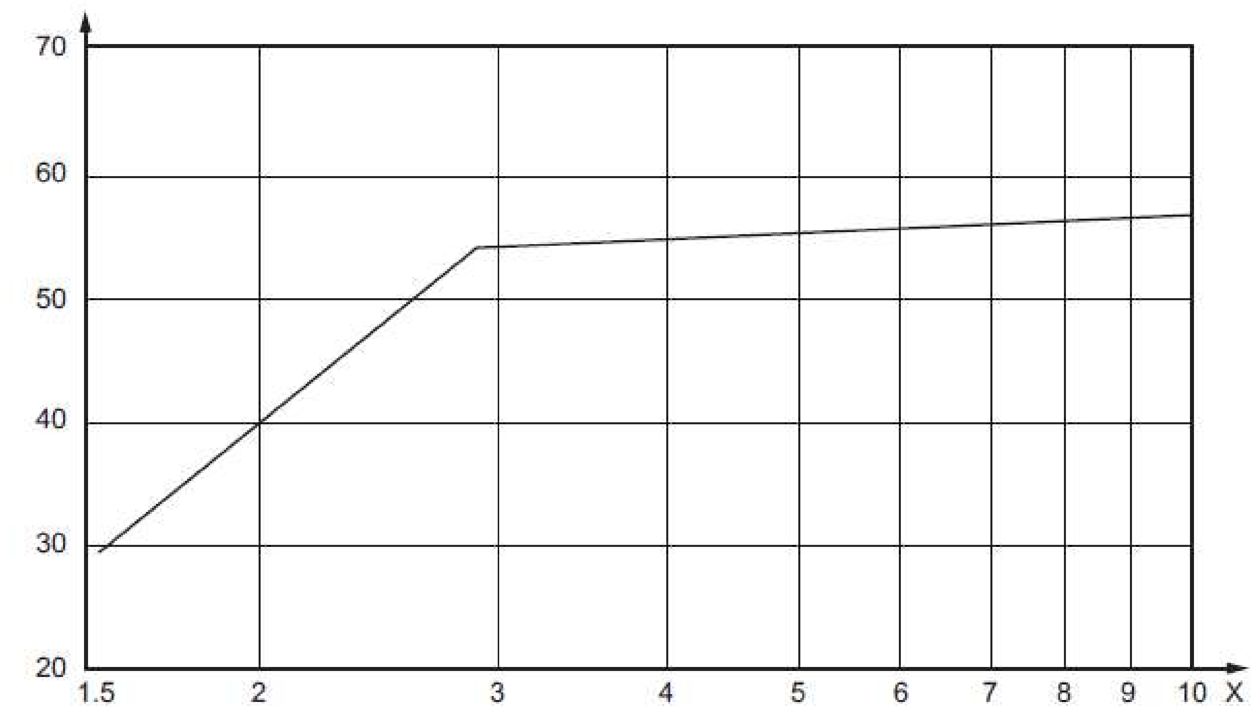
$$A_T = \frac{\dot{m}}{\rho \times v} = \frac{31.5}{1.48 \times 150} = 0.14\ m^2 \quad (25)$$

In USC units:

$$A_T = \frac{\dot{m}}{3600 \times v} = \frac{250{,}000}{3600 \times 0.1 \times 500} = 1.39\ \text{ft}^2 \quad (26)$$

Thus, the pipe diameter is approximately DN 400 (NPS 16).

10 Noise Calculation Example



Key

X pressure ratio, PR

Y sound pressure level, $L = L_{30(100)} - 10\ \lg(0.5q_m \cdot c^2)$, decibels

NOTE: The pressure ratio is defined as the absolute static pressure upstream from the restriction (e.g. pressure-relief valve nozzle) divided by the absolute pressure downstream of the restriction while relieving. In the case of calculations involving pressure-relief devices, the upstream pressure is the inlet pressure to the pressure-relief device during the relieving event. The downstream pressure is typically atmospheric pressure when venting to atmosphere. In some cases, critical flow can occur not only in the pressure-relief nozzle but also at the discharge pipe outlet to atmosphere. In this case, the noise level is additive (logarithmic). In the case of the discharge pipe, the pressure ratio is the absolute pressure within the pipe at the outlet divided by atmospheric pressure.

Figure 5—Sound Pressure Level at 30 m (100 ft) from the Stack Tip

10.1 Noise Calculation Example in SI Units

The noise level at 30 m (100 ft) from the point of discharge to the atmosphere can be calculated in SI units as follows.

1) Calculate ($0.5q_m \times c^2$) in watts.

2) Calculate $10 \times \lg(0.5q_m \times c^2)$.

3) In Figure 5, find the value of PR on the abscissa and read the corresponding ordinate.

4) Add Items b) and c) to obtain $L_{30(100)}$, which is the average sound pressure level at 30 m (100 ft), expressed in decibels.

Consider the following relieving scenario:

– q_m = 14.6 kg/s;

– k = 1.4;

– M = 29;

– T = 311 K;

– inlet pressure to the relief device during the relieving event = 330 kPa;

– downstream pressure = atmospheric = 101 kPa.

Then:

PR = pressure ratio = 330/101 = 3.3;

$$c = 91.2 \times \left(\frac{1.4 \times 311}{29}\right)^{0.5} = 353 \ \frac{m}{s} \tag{27}$$

The results are as follows, referring to the numbered list items:

a) ($0.5q_m \times c^2$) = (0.5)(14.6)(353)^2 = 910,000;

b) $10 \times \lg(0.5q_m \times c^2)$ = 60;

c) from Figure 5, the ordinate corresponding to PR = 3.3 is 54;

d) $L_{30(100)}$ = 54 + 60 = 114 dB.

10.2 Noise Calculation Example in USC Units

The noise level at 30 m (100 ft) from the point of discharge to the atmosphere can be calculated in USC units as follows.

1) Calculate ($0.5q_m \times c^2$) in watts as follows: Divide the mass flow (lb/s) by 32.2 ft-lb/lbf-s^2 to obtain q_m. Multiply ($0.5q_m \times c^2$) feet times pound-force per second by 1.36.

2) Calculate $10 \times \lg(0.5q_m \times c^2)$.

3) In Figure 5, find the value of PR on the abscissa and read the corresponding ordinate.

4) Add Items b) and c) to obtain $L_{30(100)}$, which is the average sound pressure level at 30 m (100 ft), expressed in decibels.

Assume the following:

– mass flow = 32.2 lb/s;

– q_m = (32.2 lb/s)/(32.2 ft-lb/lbf-s^2) = 1 ft·lbf/s;

– k = 1.4;

– M = 29;

– T = 560 °R;

– inlet pressure to the relief device during the relieving event = 48 psia;

– downstream pressure = atmospheric = 14.7 psia.

Then:

PR = pressure ratio = 48/14.7 = 3.3;

$$c = 223 \times \left(\frac{1.4 \times 560}{29}\right)^{0.5} = 1\ 159\ \frac{\text{ft}}{s} \qquad (28)$$

The results in USC units are as follows, referring to the list items above:

a) $(0.5q_m \times c^2) = (0.5)(1)(1159)^2(1.36) = 910{,}000$;

b) $10 \times \lg(0.5q_m \times c^2) = 60$;

c) from Figure 5, the ordinate corresponding to PR = 3.3 is 54;

d) $L_{30(100)} = 54 + 60 = 114$ dB.

11 Corrected Hydrotest Pressure Examples

11.1 Overpressure Scenario Temperature Equals Design Temperature

Material allowable stress values generally decrease with an increase in temperature (i.e. vessel becomes weaker as the temperature increases). This effect needs to be considered if evaluating the overpressure scenarios defined in 4.2.2. Also, most pressure design codes require this effect to be considered when determining the hydrotest pressure for new vessels. For example, say a vessel has the following design:

— designed in accordance with ASME *BPVC*, Section VIII, Division 1 (2008);

— fabricated from ASTM A515 [5] Grade 70 carbon steel;

— design pressure of 75 psig [517 kPa (gauge)];

— MAWP is the same as the design pressure;

— design temperature of 650 °F (343 °C).

ASME *BPVC*, Section VIII, Division 1 (2008) UG-99(b) requires the hydrotest be performed at 130 % of the MAWP. In addition to the 130 %, the hydrotest pressure needs to be corrected for temperature differences between the design temperature and temperature that the hydrotest is conducted (typically ambient temperature). This temperature correction is done by multiplying the MAWP by the ratio of allowable stress at test temperature to the allowable stress at design temperature. ASTM A515 Grade 70 carbon steel has an allowable stress of 18,800 psi (130 MPa) at the stipulated design temperature. Say the hydrotest was conducted at an ambient temperature of 70 °F (21 °C). At this ambient temperature, the allowable stress of ASTM A515 Grade 70 carbon steel is 138 MPa (20,000 psi). The actual hydrotest pressure is then as follows.

In SI units:

$$517 \times (138/130) \times 1.3 = 713 \text{ kPa (gauge)} = 138 \text{ \% of the MAWP}$$

In USC units:

$$75 \times (20{,}000/18{,}800) \times 1.3 = 103.7 \text{ psig} = 138 \text{ \% of the MAWP}$$

When evaluating an overpressure scenario, it is a common and generally conservative practice to use the design temperature to determine the material stress value when the overpressure occurs. For the example above, the corrected hydrotest pressure used for the overpressure scenario evaluations is simply 130 % of the MAWP. In this example, a temperature correction is not required because the MAWP is based on the design temperature. The actual hydrotest pressure is not used in this example unless a temperature correction is applied because the hydrotest pressure is determined at ambient temperature, not at design temperature. The temperature correction is the ratio of the material stress at the overpressure scenario temperature (i.e. design temperature for this example) to the material stress at the temperature at which the hydrotest was conducted. In other words:

In SI units:

$$\text{Corrected Hydrotest Pressure} = 713 \times (130/138) = 672 \text{ kPa (gauge)} = 130 \text{ \% of the MAWP}$$

In USC units:

$$\text{Corrected Hydrotest Pressure} = 103.7 \times (18{,}800/20{,}000) = 97.5 \text{ psig} = 130 \text{ \% of the MAWP}$$

11.2 Correction for Other Temperatures

The preceding example can be generalized using the following expressions, depending upon whether the MAWP or the hydrotest pressure is used as the basis:

$$P_{\text{corrected}} = \text{MAWP} \times H_f \times \frac{\sigma_{\text{scenario}}}{\sigma_{\text{design}}} \tag{29}$$

or

$$P_{\text{corrected}} = P_{\text{hydrotest}} \times \frac{\sigma_{\text{scenario}}}{\sigma_{\text{hydrotest}}} \quad (30)$$

Where:

$P_{corrected}$ is the corrected hydrotest pressure, expressed in psig (kPa gauge);

MAWP is the maximum allowable working pressure, expressed in psig (kPa gauge);

H_f is the pressure design code hydrotest factor (e.g. 1.3 if 130 %);

$\sigma_{scenario}$ is the material stress at the overpressure scenario temperature, expressed in psi (MPa);

σ_{design} is the material stress at the vessel design temperature, expressed in psi (MPa);

$P_{hydrotest}$ is the vessel hydrotest pressure, expressed in psig (kPa gauge);

$\sigma_{hydrotest}$ is the material stress at the vessel hydrotest temperature, expressed in psi (MPa).

The hydrotest pressure can be corrected for temperatures other than the design temperature using these equations. For example, assume the same vessel design as in 11.1, that is, ASTM A515 Grade 70 carbon steel with a design pressure (MAWP) of 75 psi (517 kPa) and design temperature of 650 °F (343 °C), but the overpressure scenario temperature is, say, 100 °C (212 °F). The allowable stress is determined to be 20,000 psi (138 MPa) at 212 °F (100 °C). Using the MAWP, the corrected hydrotest pressure to use when evaluating this overpressure scenario is then as follows.

In SI units:

$P_{corrected}$ = 517 × 1.3 × (138/130) = 713 kPa (gauge)

In USC units:

$P_{corrected}$ = 75 × 1.3 × (20,000/18,800) = 103.7 psig

Note that allowable stress values for other materials and other temperatures can be found in ASME *BPVC*, Section II, Part D [6], for example.

12 Capacity Evaluation of Rupture Disk and Piping System – 100% Vapor Flow and Constant Pipe Diameter

12.1 General

The following method can be used to estimate the vapor capacity of a rupture disk/piping system of constant diameter. The method is based on compressible pipe flow equations contained in Crane Technical Paper 410 [7] and the application of standard resistance factors (K values) from API 521.

The method assumes that C_p/C_v is equal to 1.4. This assumption provides conservative results. The method can be applied to a piping system with varying diameters by treating each section of constant diameter separately.

NOTE The method presented in Crane 410 is based on graphical evaluation of several parameters. Curve fits of the graphical data are also presented below to allow direct solution without the graphical data. The use of the curve fitting equations introduces negligible error relative to the accuracy of the K factors.

12.2 Example Problem

Figure 6 shows the arrangement of the vessel and rupture disk/piping system for the example problem.

a) Step 1—Determine required information:

— MAWP = 100 psig;

— P_1 = relieving pressure = 110 % = 124.7 psia;

— T_1 = relieving temperature = 200 °F + 460 °F = 660 °R;

— Z_1 = relieving compressibility = 1.0;

— M = molecular weight = 20;

— P_2 = backpressure = 14.7 psia.

b) Step 2—Determine overall piping resistance factor, K, from Table 3.

c) Step 3—Determine Y_{sonic} and $\frac{dP_{sonic}}{P_1}$ based on total system K.

The charts on page A-22 of Crane 410 [7] can be used to obtain Y_{sonic} and $\frac{dP_{sonic}}{P_1}$. From the chart, where k (C_p/C_V) = 1.4, the following values are determined:

$Y_{sonic} = 0.653$

$\frac{dP_{sonic}}{P_1} = 0.70$

Table 3—Determination of Overall Piping Resistance Factor, K

Description	K Value	Source of K Value Data
Sharp-edged entrance	0.50	Crane [7], Page A29
Rupture disk	1.50	Consult the rupture disk manufacturer
15 ft NPS 3 Schedule 40 pipe	1.04	$K = \frac{fL}{D}$ $f = 0.0178$ $L = 15$ ft $D = \frac{3.068}{12} = 0.2557$ ft
Sudden expansion (exit loss)	1.00	API 521 (see NOTE)
Total system K	4.04	

NOTE When the pressure drop calculation method used (e.g. the Isothermal Method in API 521) accounts for kinetic energy, the exit loss coefficient can be neglected when discharging compressible fluids into a large reservoir such as the atmosphere. The user is cautioned that some piping pressure drop calculation methods do not account for kinetic energy in which case the exit losses are included in the pressure drop equation.

As an alternate to the chart method, curve fit equations for obtaining Y_{sonic} and $\frac{dP_{sonic}}{P_1}$ have been provided as Equation (31) through Equation (34).

For $\frac{dP_{sonic}}{P_1}$:

If $1.2 < K \leq 10$, then $\frac{dP_{sonic}}{P_1} = 0.1107\, In(K) + 0.5352$ (31)

If $10 < K \leq 100$, then $\frac{dP_{sonic}}{P_1} = 0.0609\, In(K) + 0.6513$ (32)

For Y_{sonic}:

If $1.2 < K \leq 20$, then $Y_{sonic} = 0.0434\, In(K) + 0.5889$ (33)

If $20 < K \leq 100$, then $Y_{sonic} = 0.710$ (34)

Based on $K = 4.04$:

$$\frac{dP_{sonic}}{P_1} = 0.69$$

$$Y_{sonic} = 0.65$$

d) Step 4—Compare $\frac{dP_{sonic}}{P_1}$ to $\frac{dP_{actual}}{P_1}$.

$$\frac{dP_{actual}}{P_1} = \frac{(124.7 - 14.7)}{124.7} = 0.88$$

Since $\frac{dP_{sonic}}{P_1} < \frac{dP_{actual}}{P_1}$, the flow will be sonic (critical).

Use Y_{sonic} and $\frac{dP_{sonic}}{P_1}$ and skip to Step 6 (if subsonic, proceed to Step 5).

e) Step 5—Evaluate Y_{actual} (subsonic cases only).

Using the Crane 410—Chart A-22 Method to obtain Y_{actual}:

1) at $\frac{dP_{actual}}{P_1}$ and K determine Y_{actual} from Chart A-22;

2) use $\frac{dP_{actual}}{P_1}$ and Y_{actual} in Step 6.

Using the Curve Fit Method for obtaining Y_{actual}:

1) calculate Y_{actual} from the Equation (35):

$$Y_{actual} = 1 - \frac{(1 - Y_{sonic})}{dP_{sonic}/P_1}\left(\frac{dP_{actual}}{P_1}\right) \tag{35}$$

2) use $\frac{dP_{actual}}{P_1}$ and Y_{actual} in Step 6 in place of $\frac{dP_{sonic}}{P_1}$ and Y_{actual}.

f) Step 6—Calculate capacity based on Crane 410—Equation (3-20)

$$W = 0.9\left(1{,}891 Y d^2 \sqrt{\frac{dP}{KV_1}}\right) \tag{36}$$

g) Step 7—Using the Chart Method values and Equation (36):

1) $Y = Y_{sonic} = 0.65$;

2) d = pipe ID (in.) = 3.068 in.;

3) $dP = \left(\frac{dP_{sonic}}{P_1}\right)(P_1) = 87.3$ psi;

4) K = overall resistance = 4.04;

5) V_1 = vapor specific volume = 2.84 ft^3/lb (obtained using ideal gas law and compressibility).

6) $W = 0.9\left(1{,}891 \times 0.65 \times 3.068^2 \sqrt{\frac{87.3}{4.04 \times 2.84}}\right) = 28{,}700\,\frac{lb}{hr}$

h) Step 8—Using the Curve Fit Method values from Figure 7 and Equation (36):

1) $Y = Y_{sonic} = 0.65$;

2) d = pipe ID (in.) = 3.068 in.;

3) $dP_{sonic} = \left(\frac{dP_{sonic}}{P_1}\right)(P_1) = 86.0$ psi;

4) K = overall resistance = 4.04;

5) V_1 = vapor specific volume = 2.84 ft^3/lb (obtained using ideal gas law and compressibility).

6) $W = 0.9\left(1{,}891 \times 0.65 \times 3.068^2 \sqrt{\frac{86.0}{4.04 \times 2.84}}\right) = 28{,}508\,\frac{lb}{hr}$

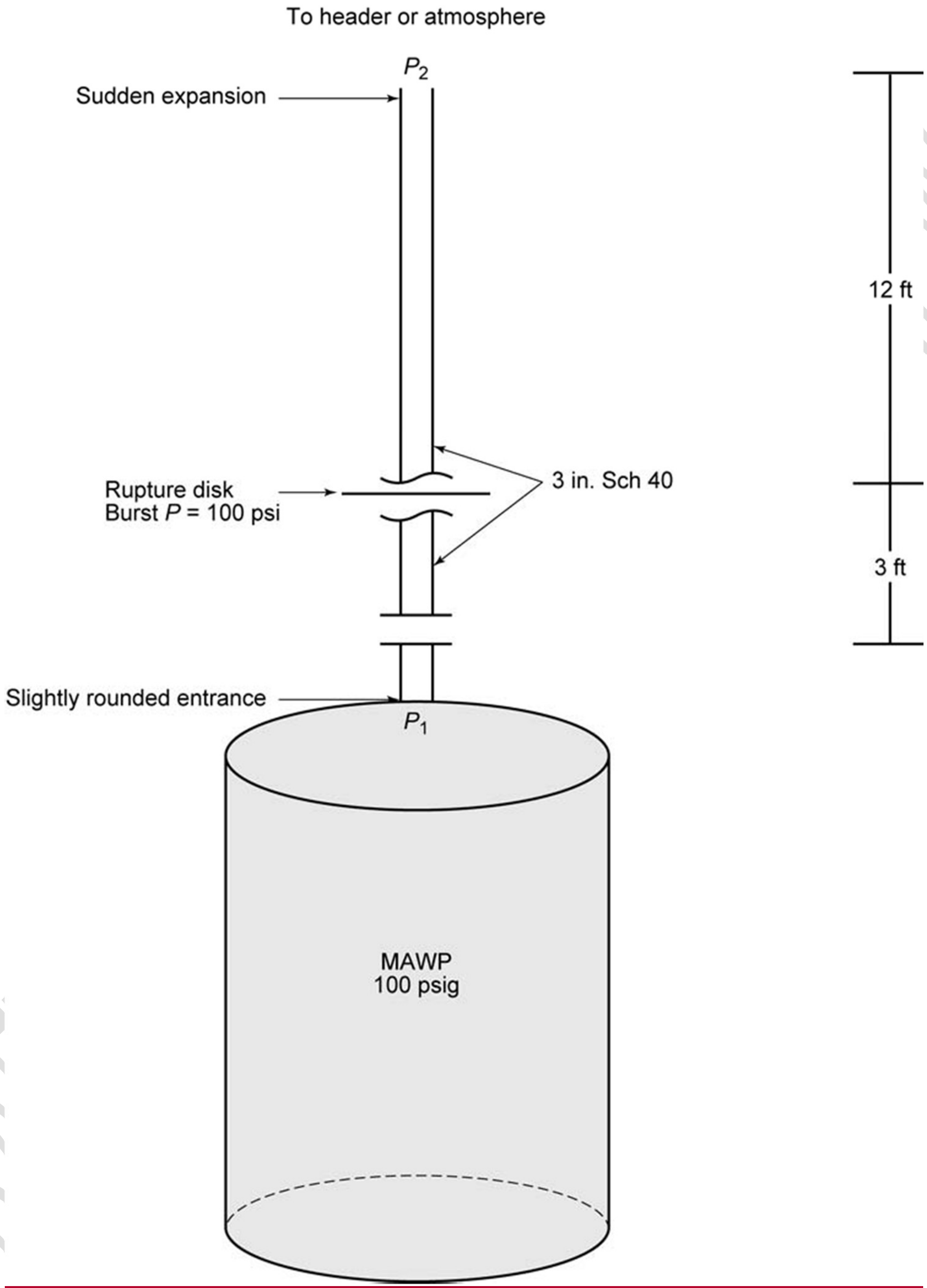


Figure 6—Pressure-relief System for Example Problem

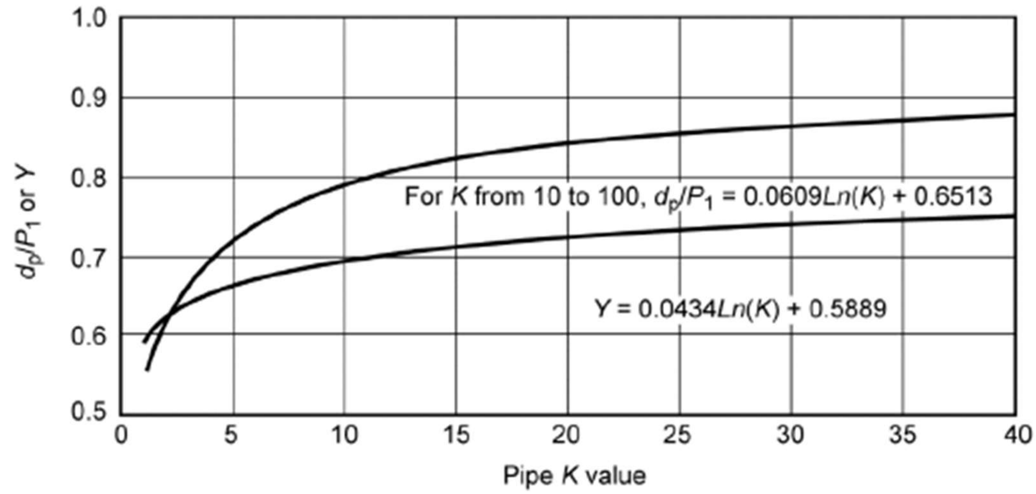


Figure 7—Curve Fit for C_p/C_v = 1.4 (Crane 410, Chart A-22)

Bibliography

[1] Melhem, G.A., "Analysis of PRV Stability in Relief Systems, Part II—Screening," ~~ioMOSAIC~~ioMosaic white paper, November 2016

[2] Hisao Izuchi, "Stability analysis of safety valve,"10th Topical Conference on Gas Utilization, AIChE, 2010.

[3] NB-18,[1] *Pressure Relief Device Certification*

[4] Darby R. "The dynamic response of pressure relief valves in vapor or gas service, part I: Mathematical model". *Journal of Loss Prevention in the Process Industries*, November 2013, 26(6):1262-1268.

[5] ASTM A515 [2], *Standard Specification for Pressure Vessel Plates, Carbon Steel, for Intermediate- and Higher-Temperature Service*

[6] ASME *Boiler and Pressure Vessel Code (BPVC)*[3]*, Section II: Materials; Part D: Properties*

[7] Crane Engineering Division, *Flow of Fluids Through Valves, Fittings, and Pipe,* Technical Paper 410

[1] The National Board of Boiler and Pressure Vessel Inspectors, 1055 Crupper Avenue, Columbus, Ohio 43229, ~~www.nationalboard.org.~~www.nationalboard.org.

[2] ASTM International, 100 Barr Harbor Drive, West Conshohocken, Pennsylvania 19428, www.astm.org.

[3] American Society of Mechanical Engineers, Two Park Avenue, New York, New York 10016-5990, www.asme.org.

[8] API Standard 520, *Sizing, Selection, and Installation of Pressure-relieving Devices, Part I–Sizing and Selection*

[9] API Standard 520, *Sizing, Selection, and Installation of Pressure-relieving Devices, Part II–Installation,* 7th Edition

[10] API Standard 521, *Pressure-relieving and Depressuring Systems*