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Safety devices for protection against excessive pressure —

Part 10:
Sizing of safety valves and bursting discs for gas/liquid two-phase flow

Dispositifs de sécurité pour protection contre les pressions excessives —

Partie 10: Dimensionnement des soupapes de sûreté et des disques de rupture pour les débits diphasiques gaz/liquide

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Foreword

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This document was prepared by Technical Committee ISO/TC 185, *Safety devices for protection against excessive pressure,* in collaboration with the European Committee for Standardization (CEN) Technical Committee CEN/TC 69, *Industrial valves*, in accordance with the Agreement on technical cooperation between ISO and CEN (Vienna Agreement).

This second edition cancels and replaces the first edition (ISO 4126-10:2010), which has been technically revised.

The main changes are as follows:

— opening of the method for sizing of bursting discs;

— more thorough iteration for the calculation of the flow rate;

— allowing for slip;

— allowing for velocity in the outlet line and pressure losses in front and after the safety device;

— added an example for flow rate to be discharged (Annex B);

— added an example for dischargeable mass flow rate added and method to estimate pressure drop in pipe flow (Annex C);

— various correction.

A list of all parts in the ISO 4126 series can be found on the ISO website.

Any feedback or questions on this document should be directed to the user’s national standards body. A complete listing of these bodies can be found at www.iso.org/members.html.

Introduction

Well-established recommendations exist for the sizing of safety valves and bursting discs and the connected inlet and outlet lines for steady-state, single-phase gas/vapour or liquid flow. However, in the case of a two-phase vapour/liquid flow, the required relieving area to protect a system from overpressure is larger than that required for single-phase flow when the same vessel condition and heat release are considered. The requirement for a larger relief area results from the fact that, in two-phase flow, the liquid partially blocks the relieving area for the vapour flow, by which most of the energy is removed by evaporation from the vessel.

This document includes a widely applicable method for the sizing of the most typical safety valves and bursting discs in fluid services encountered in various industrial fields (see Table 1). It is based on the omega parameter method, which is extended by a thermodynamic non-equilibrium parameter. A balance is attempted between the accuracy of the method and the unavoidable uncertainties in the input and property data under the actual sizing conditions.

In case of two-phase flow, the safety device size can influence the fluid state and, hence, the mass flow rate to be discharged. Furthermore, the two-phase mass flow rate through a safety device essentially depends on the mass flow quality (mass fraction of vapour) of the fluid at the inlet of the device. Because these parameters are, in most cases, not readily at hand during the design procedure of a relief device, this document also includes a comprehensive procedure that covers the determination of the fluid-phase composition at the safety device inlet. This fluid-phase composition depends on a scenario that leads to the pressure increase. Therefore, the recommended sizing procedure starts with the definition of the sizing case and includes a method for the prediction of the mass flow rate required to be discharged and the resulting mass flow quality at the inlet of the safety device.

The formulae of ISO 4126-7:2013/Amd 1:2016 for single-phase flow up to the narrowest flow cross-section are included in this document, modified to SI units, to calculate the flow rates at the limiting conditions of single-phase gas and liquid flow.

In this document, the unit bar for pressures is being used 100 000 Pa = 1 bar.

Table 1 — Possible fluid state at the inlet of the safety valve or bursting disc that can result in two-phase flow

Fluid state at device inlet	Cases	Examples
liquid	subcooled (possibly flashing in the safety device) saturated with dissolved gas	cold water boiling water CO_2/water
gas/vapour	near saturated vapour (possibly condensing in the safety device)	steam
gas/liquid	vapour/liquid non-evaporating liquid and non-condensable gas (constant quality) gas/liquid mixture, when gas is desorbed or produced	steam/water air/water

Safety devices for protection against excessive pressure —

Part 10:
Sizing of safety valves and bursting discs for gas/liquid two-phase flow

1 Scope

This document specifies the sizing of safety valves and bursting discs for gas/liquid two-phase flow in pressurized systems such as reactors, storage tanks, columns, heat exchangers, piping systems or transportation tanks/containers, see Figure 2. The possible fluid states at the safety device inlet that can result in two-phase flow are given in Table 1.

NOTE The pressures used in this document are absolute pressures, not gauge pressures.

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements 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.

ISO 4126-7:2013/Amd 1:2016, *Safety devices for protection against excessive pressure — Part 7: Common data*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 4126-7:2013/Amd 1:2016 and the following apply.

ISO and IEC maintain terminology databases for use in standardization at the following addresses:

— ISO Online browsing platform: available at https://www.iso.org/obp

— IEC Electropedia: available at https://www.electropedia.org/

3.1 General

3.1.1
pressurized system
equipment being protected against excessive pressure accumulation by a safety device

EXAMPLE Equipment can be reactors, storage tanks, columns, heat exchangers, piping systems and transport tanks/containers, etc.

3.1.2
critical filling threshold
ϕ_{limit}
maximum initial liquid filling threshold (liquid hold-up) in the *pressurized system* (3.1.1) at sizing conditions, up to where vapour disengagement occurs and single-phase gas or vapour flow can be expected

Note 1 to entry: The critical filling threshold is expressed as a ratio of the total volume of the system.

Note 2 to entry: For filling levels above the critical filling threshold, two-phase flow is assumed to occur.

3.1.3
initial liquid filling level
ϕ_0
liquid hold-up in the *pressurized system* (3.1.1) at the sizing conditions

Note 1 to entry: The initial liquid filling level is expressed as a ratio of the total volume of the system.

3.1.4
inlet line
piping and associated fittings connecting the *pressurized system* (3.1.1) to the safety device inlet

3.1.5
outlet line
piping and associated fittings connecting the safety device outlet to a containment system or the atmosphere

3.1.6
vent line system
combination of safety device, *inlet line* (3.1.4) and *outlet line* (3.1.5)

3.1.7
cryogenic vessel
vacuum jacketed vessel intended for application at low temperature involving liquefied gases

3.2 Pressure

3.2.1
maximum allowable working pressure
p_{MAW}
maximum pressure permissible at the top of a *pressurized system* (3.1.1) in its operating position for designated temperature

3.2.2
maximum allowable accumulated pressure
p_{MAA}
sum of the *maximum allowable working pressure* (3.2.1) and the *maximum allowable accumulation* (3.2.3)

Note 1 to entry: The maximum allowable accumulation is established by applicable code for operating and fire contingencies.

3.2.3
maximum allowable accumulation
Δp_{MAA}
pressure increase over the *maximum allowable working pressure* (3.2.1) of a *pressurized system* (3.1.1) during discharge through the safety device

Note 1 to entry: The maximum allowable accumulation is expressed in pressure units or as a percentage of the maximum allowable working pressure.

3.2.4
opening pressure
p_{open}
predetermined absolute pressure at which a safety valve under operating conditions at the latest commences to open

3.2.5
absolute overpressure
Δp_{over}
pressure increase over the *opening pressure* (3.2.4), p_{open}, of the safety device

Note 1 to entry: The maximum absolute overpressure is the same as the maximum accumulation, Δp_{MAA}, when the opening pressure of the safety valve is set at the *maximum allowable working pressure* (3.2.1) of the *pressurized system* (3.1.1).

Note 2 to entry: The absolute overpressure is expressed in pressure units or as a percentage of the opening pressure.

3.2.6
overpressure
p_{over}
maximum pressure in the *pressurized system* (3.1.1) during relief, i.e. pressure less or equal to the maximum accumulated pressure

3.2.7
sizing pressure
p_0
pressure at which all property data, especially the compressibility coefficient, ω, are calculated for sizing the safety device

Note 1 to entry: In the case of tempered and hybrid reactive systems, the sizing pressure shall be as low as reasonable possible, but should not affect the normal operation. In the case of non-reactive and *gassy systems* (3.5.3), the designer may choose a higher value for the sizing pressure, but it shall not exceed the *maximum allowable accumulated pressure* (3.2.2).

3.2.8
critical pressure
p_{crit}
fluid-dynamic critical pressure occurring in the narrowest flow cross-section of the safety valve and/or at an area enlargement in the *outlet line* (3.1.5)

Note 1 to entry: At this pressure, the mass flow rate approaches a maximum at a given sizing condition in the *pressurized system* (3.1.1). Any further decrease of the downstream pressure does not increase the flow rate further. Usually, the critical pressure occurs in the safety valve, either in the valve seat, inlet nozzle and/or valve body. In the bursting disc, critical pressure can occur downstream of the device at a minimum flow area, at the exit of the vessel or a change in pipe diameter. In long safety device outlet lines, multiple critical pressures can also occur.

3.2.9
stagnation condition
condition when fluid is at rest

EXAMPLE Fluid in large vessels, where the flow velocity is almost zero, even in case of a discharge of mass.

3.2.10
critical pressure ratio
η_{crit}
ratio of *critical pressure* (3.2.8) to the *sizing pressure* (3.2.7)

3.2.11
thermodynamic critical pressure
p_c
state property, together with *thermodynamic critical temperature* (3.6.1), at the thermodynamic critical point

3.2.12
back pressure
p_b
pressure that exists at the outlet of a safety device as a result of pressure in the discharge system

Note 1 to entry: Back pressure can be either constant or variable; it is the sum of superimposed and *built-up back pressure* (3.2.13).

3.2.13
built-up back pressure
pressure existing at the outlet of the safety device caused by flow through the valve or bursting disc and discharge system

3.2.14
inlet pressure loss
Δp_{loss}
irrecoverable pressure decrease due to flow in the piping from the equipment that is protected to the inlet of the safety device

3.2.15
blowdown
Δp_{BD}
difference between *opening pressure* (3.2.4) and reseating pressure of a safety valve

Note 1 to entry: Blowdown is normally stated as a percentage of the opening pressure.

3.2.16
dimensionless reduced pressure
p_{red}
local pressure divided by the *thermodynamic critical pressure* (3.2.11) of the substance

3.3 Flow rate

3.3.1
mass flow rate required to be discharged from a pressurized system
$Q_{m,out}$
mass flow rate required to avoid that the pressure exceeds the *maximum allowable accumulated pressure* (3.2.2) in the *pressurized system* (3.1.1) during relief

3.3.2
feed mass flow rate into the pressurized system
$Q_{m,feed}$
maximum mass flow rate through a feed line or control valve fed into the *pressurized system* (3.1.1) being protected

3.3.3
dischargeable mass flux through the safety device
$\dot{m}_{SD}$
mass flow rate per area through a safety device at the sizing conditions calculated by means of the certified discharge coefficients for gas and liquid flow

Note 1 to entry: See Formula (48).

3.3.4
certified valve discharge coefficient for single-phase gas/vapour respectively liquid flow
$K_{dr,g}$ ⟨gas⟩
$K_{dr,l}$ ⟨liquid⟩
correction factor defined by the ratio of the theoretically *dischargeable mass flux through the safety device* (3.3.3) to an experimentally determined mass flux through a device of the same manufacturer's type

Note 1 to entry: The discharge coefficient of a safety valve is related to the valve seat cross-section and accounts for the imperfection of flow through the device compared to that through a reference model (ideal nozzle). Certified values for gas and liquid flow, K_d, are usually supplied by valve manufacturers or determined by experiment. Rated discharge coefficients K_{dr}, equal to 0,9 K_d, are used to calculate the safety valve sizing area.

Note 2 to entry: The discharge coefficient of a bursting disc is related to the disc cross-section and accounts for the imperfection of flow through the device compared to that through a reference model.

3.4 Flow area

3.4.1
safety device sizing area
A_0
most essential result of the sizing procedure in accordance with this document required to select an adequately sized safety device and defined as the minimum cross-section of flow area

Note 1 to entry: It is important that the *dischargeable mass flux through the safety device* (3.3.3) be related to this specific area.

3.4.2
effective flow area of the feed line or the control valve
A_{feed}
discharge flow area of a feed line or control valve in the line to the *pressurized system* (3.1.1)

3.5 Fluid state

3.5.1
gas/liquid mixture
fluid mixture composed of both a liquid part and a gas part, in which the gas is not necessarily of the same chemical composition as the liquid

3.5.2
tempered system
fluid system in which some energy is removed from the liquid phase by evaporation or flashing

3.5.3
gassy system
fluid system in which permanent gas is generated (e.g. by chemical reaction or by evolution from solution) and in which no significant amount of energy is removed from the liquid by evaporation at the sizing conditions

3.5.4
hybrid system
fluid system that exhibits characteristics of both tempered and *gassy systems* (3.5.3) to a significant extent at the sizing conditions

3.5.5
thermal runaway reaction
uncontrolled or undesired exothermic chemical reaction

3.6 Temperature

3.6.1
thermodynamic critical temperature
T_c
state property, together with *thermodynamic critical pressure* (3.2.11), at the thermodynamic critical point

3.6.2
sizing temperature
T_0
temperature of the *pressurized system* (3.1.1) at the sizing conditions

3.6.3
overtemperature
T_{over}
maximum temperature in the *pressurized system* (3.1.1) during relief

3.6.4
saturation temperature difference
ΔT_{over}
difference between the saturation temperature at the maximum pressure during relief, p_{over}, and the saturation temperature at the *sizing pressure* (3.2.7), p_0

3.6.5
dimensionless reduced temperature
T_{red}
local temperature divided by the *thermodynamic critical temperature* (3.6.1) of the substance

4 Symbols and abbreviated terms and figures

4.1 Symbols

Variable	Definition	Unit
A_{feed}	effective flow area of the feed line or the control valve	m^2
A_{fire}	The wetted surface area to be considered for the heat transfer due to fire. In detail, it is the partial surface area of a vertical cylindrical vessel wetted by internal liquid and located within 7,5 m vertically from ground or from any surface capable of sustaining a pool fire. Depending on the fire case considered it may either include the wall of the bottom or the bottom wall of the vessel is not included.	m^2
A_{heat}	area of heat exchange in the pressurized system in case of external heat input	m^2
A_0	minimum required safety device area (safety device sizing area). In general, for safety valves it is the safety valve seat area and for bursting discs the minimum net flow area.	m^2
A_R	cross-sectional area in a vertical cylindrical vessel	m^2
B_{heat}	(maximum) overall heat transfer coefficient, see Formula (24)	$W/(m^2 \cdot K)$
C	dimensionless mass flow rate	—
C_1	flow conversion factor 1	
C_2	flow conversion factor 2	
c_p	specific heat capacity at constant pressure	$J/(kg \cdot K)$
D	inner vessel diameter of a vertical cylindrical vessel	m
d	diameter	m
$\frac{dp}{dt}$	rate of pressure increase in the pressurized system	Pa/s
$\frac{dT}{dt}$	reaction self-heat rate inside the pressurized system	K/s
F	environmental factor for heat input from fire (see 6.4.3.2)	—
g	acceleration due to gravity	m/s^2
H_l	height of liquid level in a vertical cylindrical vessel (bottom of vessel to liquid level)	m
H_{fire}	maximum height of flames above ground	m
H_{vessel}	height of the bottom of the vessel flames above ground	m
k_∞	correlating parameter to calculate the characteristic bubble-rise velocity	—
$K_{dr,2ph}$	two-phase flow valve discharge coefficient	—
$K_{dr,g}$	certified valve discharge coefficient for single-phase gas/vapour flow	—
$K_{dr,l}$	certified valve discharge coefficient for single-phase liquid flow	—
K_R	velocity head loss for bursting disc	—

Variable	Definition	Unit
K_{vs}	Liquid discharge factor for fully opened control valve in the feed line, which characterizes A_{feed} of the feed line or control valve of a frictionless valve with the same pressure difference for the same flow rate.	m^3/h
L	Vertical length of a flow restriction to account for potential energy change. For safety valves and bursting discs L may be set to 0. For inlet and outlet lines the heights of the system shall be considered.	m
$\dot{m}$	mass flux	$kg/(m^2 \cdot s)$
$\dot{m}_{SD}$	dischargeable mass flux through the safety device	$kg/(m^2 \cdot s)$
M_0	total liquid mass in the pressurized system at the sizing conditions	kg
M	molecular mass	kg/kmol
N	boiling delay factor accounting for thermodynamic non-equilibrium	—
p	pressure in the pressurized system	Pa
p_b	back pressure	Pa
p_c	thermodynamic critical pressure	Pa
p_{crit}	fluid-dynamic critical pressure	Pa
p_{MAW}	maximum allowable working pressure	Pa
p_{MAA}	maximum allowable accumulated pressure	Pa
p_0	sizing pressure	Pa
p_{over}	maximum pressure in a pressurized system during relief, see Figure 1	Pa
p_{open}	opening pressure	Pa
$\dot{q}_{fire}$	dimensionless fire exposure flux	—
$Q_{m,out}$	mass flow rate required to be discharged from a pressurized system	kg/s
$Q_{m,feed}$	feed mass flow rate into the pressurized system	kg/s
$Q_{m,SD}$	dischargeable mass flow rate through the safety device	kg/s
$\dot{Q}$	heat input into the pressurized system, either by runaway reaction or by external heating	W
$\dot{Q}^*_{acc}$	ratio of the sensible heat to the latent heat	—
$\dot{Q}^*_{in}$	ratio of total heat input to energy flow removed by evaporation	—
R	universal gas constant (8 314,2 J/(kmol·K))	J/(kmol·K)
R_{2ph}	two-phase multiplier	—
T	temperature in the pressurized system	K
T_c	thermodynamic critical temperature	K
T_{heat}	maximum possible temperature of the external heat source	K
T_0	temperature of the pressurized system at the sizing conditions	K
T_{over}	maximum temperature in the pressurized system during relief	K
$u_{g,0}$	superficial gas velocity in the free-board gas volume of a vertical cylindrical vessel at the sizing conditions	m/s
u_∞	characteristic bubble-rise velocity of the gas/vapour in the liquid	m/s
u^*	dimensionless bubble-rise velocity	—
v	specific volume in the pressurized system	m^3/kg
V	volume of the pressurized system	m^3
$\dot{x}$	mass flow quality, i.e. the ratio of the gas mass flow rate to the total mass flow rate of a two-phase mixture	—
Z	real gas factor	—
β	ratio of the vent inlet diameter to the throat diameter	—

Variable	Definition	Unit
ε_0	void fraction in the pressurized system at the sizing conditions for a homogeneous two-phase mixture	—
ε_{seat}	void fraction in the narrowest cross-section, see Formula (50)	—
$\zeta_{v,ref}$	Resistance coefficient of reference, either inlet or outlet	—
η	pressure ratio, either η_{crit} or η_b	—
η_b	ratio of the safety valve back pressure to the sizing pressure	—
η_{crit}	critical pressure ratio	—
η_S	ratio of the saturation pressure corresponding to the sizing temperature and the sizing pressure (measure of liquid subcooling), see Formula (64)	—
θ	Angle of a vent line to the horizontal	°
κ	isentropic coefficient	—
ρ	fluid density	kg/m^3
ρ_{H2O}	density of water during experiments to measure the K_{vs} value at a temperature of 5 °C	kg/m^3
σ	surface tension	N/m
ϕ_{limit}	critical filling threshold	—
ϕ_0	initial liquid filling level at the sizing conditions, i.e. the liquid volume divided by the total volume of the pressurized system considered	—
ω	compressibility coefficient	—
ω_{eq}	compressibility coefficient at equilibrium condition ($N = 1$)	—
Γ_g	gas production rate per liquid mass, i.e. the gas mass flow rate per liquid mass inventory in the pressurized system	kg/(s·kg)
Γ	dimensionless velocity ratio	—
Δh_v	latent heat of vaporization	J/kg
Δp	pressure drop in the inlet or outlet line	Pa
Δp_{MAA}	maximum allowable accumulation	Pa
Δp_{feed}	pressure loss between the outlet of the control valve in the feed line and the pressurized system	Pa
Δp_{H2O}	pressure drop across a control valve during experiments to measure the K_{vs} value defined at a pressure difference of 10^5 Pa	Pa
Δp_{loss}	inlet line pressure loss	Pa
Δp_{over}	absolute overpressure	Pa
ΔT_{over}	saturation temperature difference	K
Ψ	Boiling area ratio	—
Ω	dynamic viscosity	Pa·s

4.2 Abbreviated terms

Index	Meaning
0	sizing condition
2ph	two-phase flow
b	back
CV	upstream of the control valve
c	thermodynamic critical property
crit	critical condition with respect to flow
ct	churn turbulent
feed	into the pressurized system
fire	heat externally by fire

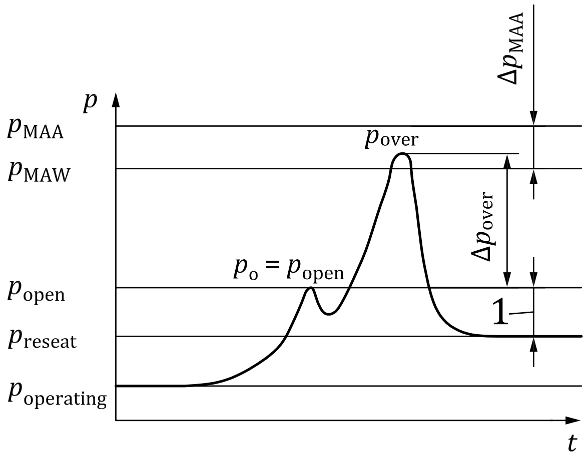
Index	Meaning
g	gas phase
HNE	Homogenous Non Equilibrium
H_2O	water
heat	external heat input (source)
i	liquid mixture component
ideal	theoretically perfect (adiabatic, frictionless)
in	inlet
is	isentropic
j	identification of a particular liquid mixture component
l	liquid phase
limit	limiting value (threshold)
loss	usually pressure loss
MAA	maximum allowable accumulated condition
MAW	maximum allowable working condition
max	maximal value
mean	average between set condition and condition at maximum allowable accumulated pressure
operating	operating condition
out	dischargeable from the pressurized system
over	overpressure or overtemperature
r	derated value (see 3.3.4)
red	reduced condition, i.e. condition related to the thermodynamic critical property
ref	reference cross-section, either inlet or outlet
reseat	reseating condition
SD	through the safety device
sat	saturation condition of the liquid phase
seat	condition in the narrowest flow cross-section of the device, e.g. the valve seat or minimum flow area of a bursting disc
open	condition, at which a safety valve under operating conditions at the latest commences to open
s	subcooling
th	throat
vessel	bottom of the vessel
∞	characteristic bubble-rise property in the liquid phase

Exponent	Meaning
a	exponent in the formula for the boiling delay coefficient, *N*, see Formula (62)
*	dimensionless

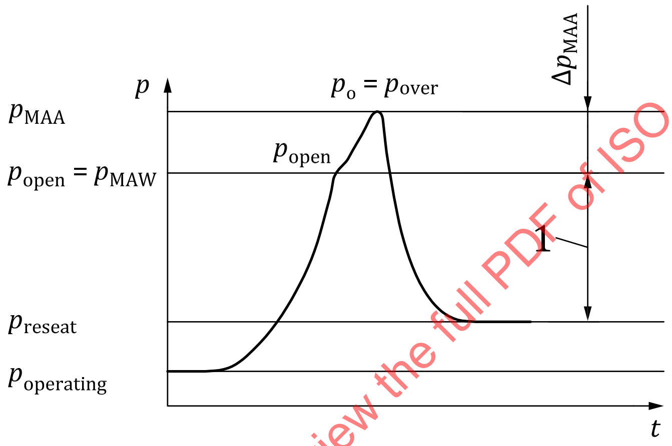
4.3 Figures

See Figures 1 a) and 1 b) for an illustration of the relationship of the pressures defined in 3.2.

In contrast to the definition used in other parts of the ISO 4126 series (e.g. ISO 4126-7) all pressures are absolute pressures and not gauge pressures.



a) Pressure history of a typical tempered reaction system that is adequately sized

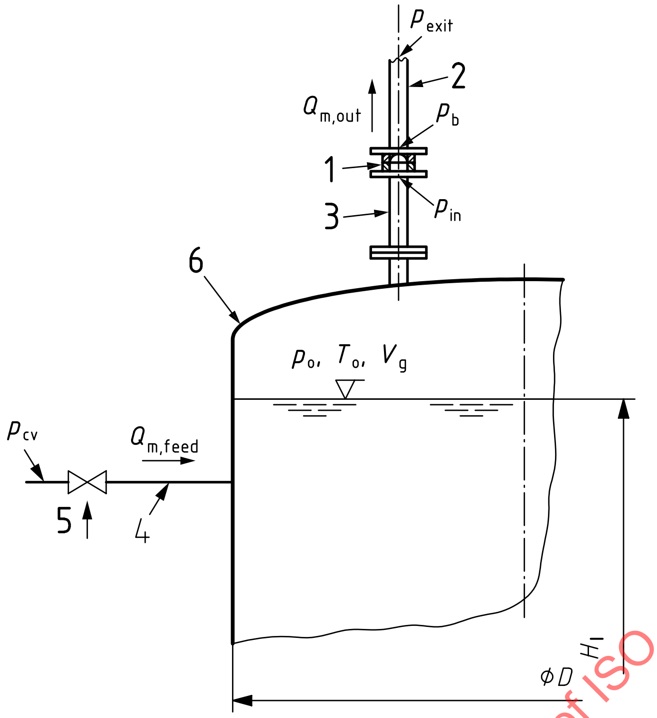


b) Typical pressure history for an externally heated gas vented system

Key

p_{MAA}	maximum allowable accumulated pressure
p_{MAW}	maximum allowable working pressure
p_{open}	opening pressure
p_{reseat}	reseating pressure
$p_{operating}$	operating pressure
1	Δp_{BD} blowdown
p_0	sizing pressure equal to p_{open} as shown in Figure 1 a) and equal to p_{over} as shown in Figure 1 b)
p_{over}	overpressure
Δp_{MAA}	maximum allowable accumulation
Δp_{over}	change in overpressure

Figure 1 — Relationship of the defined pressures



Key

1 safety device
2 outlet line
3 inlet line
4 feed line
5 control valve
6 pressurized system

Figure 2 — Safety device in a pressurized system

5 Application range of the method

5.1 General

A homogeneous, non-equilibrium flow model for a single-component gas/liquid mixture is used for the safety device sizing. The flow is assumed to be in a quasi-steady-state. It is recommended to remain within the application range of the method, as given in 5.2 and 5.3.

5.2 Limitations of the method for calculating the two-phase mass flux in safety devices

5.2.1 Flashing flow

The method is accurate for vapour/liquid flashing systems in which either one or both of the following conditions is/are true.

a) The overtemperature is less than 90 % of the fluid's thermodynamic critical temperature as given in Formula (1):

$$T_{\text{red}} = \frac{T_{\text{over}}}{T_{\text{c}}} < 0{,}9 \quad (1)$$

b) The overpressure is less than 50 % of the fluid's thermodynamic critical pressure as given in Formula (2):

$$p_{\text{red}} = \frac{p_{\text{over}}}{p_{\text{c}}} < 0{,}5 \quad (2)$$

where

T_{red} is the dimensionless reduced temperature;

p_{red} is the dimensionless reduced pressure;

T_{over} is the maximum temperature during relief, in K;

p_{over} is the maximum pressure during relief (maximum accumulated pressure or less), in Pa;

T_c is the thermodynamic critical temperature of the fluid, in K;

p_c is the thermodynamic critical pressure of the fluid, in Pa.

If both the reduced pressure and temperature of a vapour/liquid flashing system are above the specified limits, the property data usually change too rapidly, which can lead to unacceptable errors.

The formulae used in the sizing method to predict the properties of the gas/liquid flow lead, in general, to an overestimation of the required flow area, when conditions above the limits in Formulae (1) and (2) up to the thermodynamic critical point are considered[2]. Additional comments relative to the range of applicability of the method are given in 6.5.1[36].

The limitations on the model due to the linear approximation of the pressure density relation are discussed in Reference [26].

5.2.2 Condensing flow

Due to pressure drop in the safety device, even if there is single-phase gas flow at the inlet, condensation may occur resulting in two-phase flow. A formula of state for real gases should be used and the evaporation enthalpy should be considered. Further information is given in Reference [36].

5.2.3 Flashing flow for multi-component liquids

This method may be applied to multi-component flashing systems whose saturation temperature range does not exceed 100 K, as given in Formula (3) for mixtures of chemically similar liquids:

$$T_{sat,i} - T_{sat,j} < 100\text{ K} \tag{3}$$

where $T_{sat,i}$ and $T_{sat,j}$ are the saturation temperatures at the sizing pressure for components i and j, in K.

Component i exhibits the highest saturation temperature, and component j the lowest saturation temperature of the mixture.

5.2.4 Dissolved gases

The method proposed shall not be applied directly to cases where significant quantities of gases are dissolved in the fluid being discharged, which is typical in cases of high pressure with gases such as nitrogen or hydrogen dissolved in a liquid.

The presence of dissolved gases can profoundly affect the mixture properties and the mass flow rate through the safety device and shall be taken into account. For example, the thermodynamic critical conditions of the mixture can be very different from the thermodynamic critical conditions of the pure components in a mixture. This leads to a change of the saturation line. Also, other properties, such as the heat of vaporization, the liquid mixture density (e.g. of polymers), or the liquid mixture viscosity, can be affected. Even small amounts of dissolved gases, which desorb due to the pressure decrease in a non-evaporating liquid flow through a safety device, can markedly reduce the mass flow rate compared to that in liquid flow only.

If it is required to consider the desorption of gases, because they affect the thermodynamic critical properties or the saturation line, it is possible to use the design equations for gassy systems if no other reliable information is available. The time delay (in reaching equilibrium) during the desorption of gases should be neglected. For a conservative calculation, the required safety device area should be estimated with a mass flow quality that would belong to the two-phase gas/liquid mixture in equilibrium at the lowest pressure in the narrowest flow cross-section (homogeneous equilibrium flow between device inlet and narrowest device area). This can be, for example, the critical pressure in the safety device area. Because the desorption time delay has been neglected, mass flow quality is overestimated and, hence, the required device area is oversized.

5.2.5 Compressibility coefficient ω

The method is accurate if the compressibility coefficient ω satisfies the relation: $0 \le \omega \le 100$.

5.3 Limitations of the method for calculating the mass flow rate required to be discharged

5.3.1 Rate of temperature and pressure increase

In case of a runaway reaction, the reaction self-heat rate at the maximum pressure, p_{over}, during relief should be limited to 2 K/s, as given in Formula (4):

$$\left.\frac{\mathrm{d}T}{\mathrm{d}t}\right|_{over} < 2\ \mathrm{K/s} \tag{4}$$

This is limited by the method for the flow rate required to be discharged for tempered reactions in 6.4.4.2. Further, the rate of pressure increase is restricted to 20 kPa/s (12 bar/min), as given in Formula (5):

$$\left.\frac{\mathrm{d}p}{\mathrm{d}t}\right|_{over} < 20\ \mathrm{kPa/s} \tag{5}$$

If the reaction self-heat rate and the rate of pressure increase are considerably higher than these values, unfeasible relief sizes can result.

Safety valves should not be used if there is an excessive pressure increase during valve-opening. Typical spring-loaded valve-opening times are between 80 ms to 120 ms. It is necessary that the rate of pressure increase is slow enough to allow for a full opening of the valve before p_{over} is reached. Therefore, the rate of pressure increase should be limited to at least 10 % absolute overpressure above p_{open} of the safety valve within the valve-opening time. In most cases, this leads to values higher than the recommended limiting value of 20 kPa/s. For even faster pressure increases, bursting discs can be suitable.

5.3.2 Immiscible liquids

The sizing method might not be directly applicable if immiscible liquids are present in the system, e.g. in the case of the venting of an emulsion polymerization reactor. In this case, the reaction kinetics developed from dedicated laboratory experiments depend on the particular agitation state, which, in general, cannot be scaled to a technical size[22].

6 Sizing steps

6.1 General outline of sizing steps

The sizing of a vent line system includes the following essential steps (Figure 3).

Step 1: Identification of the sizing case, see 6.2.

All reasonably conceivable deviations from normal plant operation shall be considered to identify the sizing case for the valve. Whether or not it is required to consider malfunction as reasonably possible can depend on the hazard potential (local regulations can have requirements regarding this topic). The sizing case is of key importance for the dimensioning of a safety valve and often more important than the type of calculation method. Although it is beyond the scope of this document to outline the details of hazard analysis, an introduction is given in 6.2.

Step 2: Flow regime at the inlet of the vent line system, see 6.3.

Step 3: Calculation of the mass flow rate required to be discharged, see 6.4.

Step 4: Calculation of the dischargeable mass flux through and pressure change in the vent line system, see 6.5.

Step 5: Ensure proper operation of safety valve vent line systems under plant conditions, see 6.6.

In the preceding steps, the safety valve has been sized without considering any influence of the connected inlet and/or outlet line, which can lead to a deterioration of safety valve capacity or function. Low-frequency opening and closing (pumping) of the valve or a higher-frequency flutter (chatter) can occur. In the latter case, the valve and/or the inlet and outlet line can be damaged.

For the following sizing steps, recommendations and a computational procedure applicable to common engineering practice are given in 6.2 to 6.6.

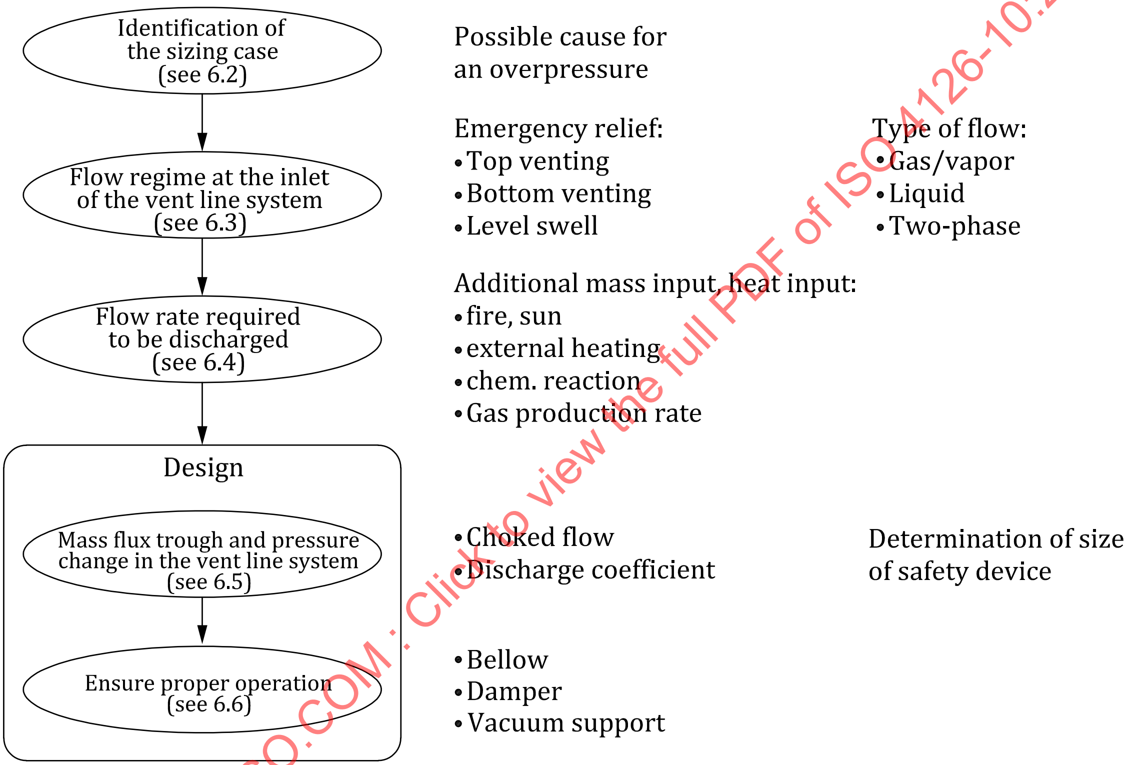


Figure 3 — Procedure of safety device sizing

6.2 Step 1 — Identification of the sizing case

The sizing case is of key importance for the dimensioning of a safety device and often more important than the type of calculation method.

Reasonably conceivable deviations from normal plant operation should be identified and evaluated relative to their hazardous risk potential (local regulations can have requirements regarding this topic) using a process hazard assessment and safety evaluation (PHASE)[3]. Several well-established procedures, e.g.

— HAZOP (hazard and operability study)[4],

— PHA (preliminary hazard analysis)[5],

— LOPA (layer of protection analysis), or

— what-if analysis,

or the more quantitative methods, e.g.

— fault-tree analysis[6] and

— event-tree analysis[7]

are used in practice. These methods are often supplemented by checklists[8] with several levels of detail. All these procedures provide a means to assess the causes for a pressure increase. These causes can be related to changes in mass and energy transfer to or from the pressurized system, or a deviation from the normal reaction system, e.g. when a thermal runaway reaction occurs.

In Annex A, the most common causes for a pressure increase are summarized. Some of these can occur simultaneously or immediately after one another and, subsequently, it can be necessary to take one or more independent process deviations into account as possible sizing cases for the valve or bursting disc. The actual outline of the possible sizing cases depends on the risk and loss potential that can result from the deviations from normal plant operation. However, the deviations should not be quantified in a too conservative manner. Otherwise, the valve seat area or bursting disc is oversized. This can result in unnecessarily large release flow rates, overloads for the downstream process equipment and, possibly, additional environmental risks.

For further information, see ISO 23251.

6.3 Step 2 — Flow regime at the inlet of the vent line system

6.3.1 General

For the sizing, it is essential to know whether a two-phase flow occurs at the inlet of the vent line. Subsequently, criteria are given for the identification of the occurrence of two-phase flow.

6.3.2 Phenomenon of level swell

Safety devices are often mounted on the top (gas side) of a vessel. Consider, for example, a system capable of generating vapour during venting (tempered system): the pressure drops immediately when the device opens due to the release of vapour from the free-board vessel volume. An initially subcooled or saturated liquid superheats, i.e. the temperature remains above the saturation temperature corresponding to the actual pressure. If the vapour production rate exceeds the rate of vapour disengagement across the interface, after a boiling delay time (typically 0,1 s to 1,0 s), bulk evaporation of liquid starts and forces the level to swell. If the mixture reaches the valve inlet, the flow regime changes from single-phase gas to two-phase flow.

Level swell primarily occurs due to limited bubble-rise velocity; the characteristic rise velocity of the gas/vapour in the liquid is designated as u_∞. This phenomenon is especially common in highly viscous liquids and foam systems. Vessels with such contents are therefore emptied almost completely during emergency venting. For foaming and highly viscous fluids, this applies even for initial liquid filling levels down to about 15 %[9].

In gas-desorbing or -producing systems, analogous phenomena, i.e. desorption delay, level swell, etc., also occur.

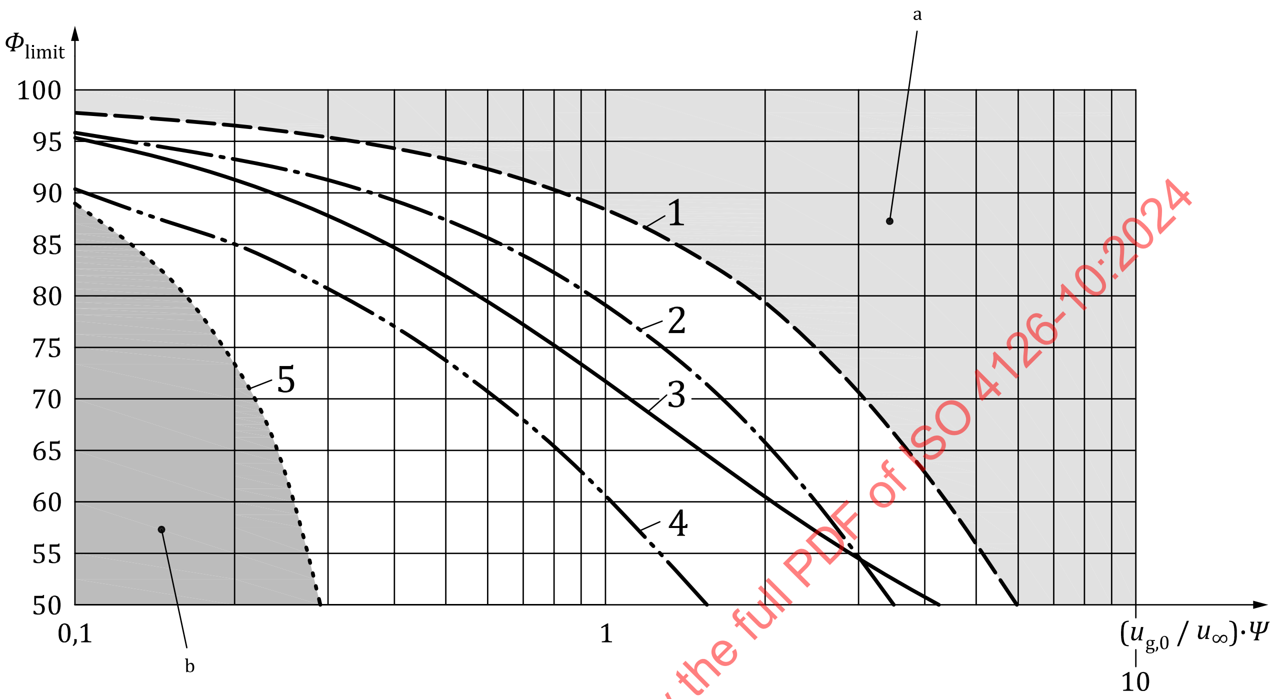
In externally heated vessels (e.g. non-reacting, non-heat-generating systems), the vapour bubbles are formed at the vessel wall, not in the bulk liquid. This can result in less level swell, with the possible consequence that single-phase flow relief occurs at higher initial liquid-filling levels[37].

6.3.3 Influence of liquid viscosity and foaming behaviour on the flow regime

A criterion to distinguish between single-phase and two-phase flow makes use of initial liquid filling level, gas or vapour production rate, liquid viscosity and foaming behaviour. The liquid viscosity shall be considered at the conditions when p_{MAA} in the pressurized system is reached. For highly viscous liquids ($\Omega_{l,0}$ > 0,1 Pa·s), homogeneous venting shall be assumed of the pressurized system; see Figure 4. In case of homogeneous venting of the pressurized system, the void fraction of the effluent is equal to the average void fraction in the vessel. Therefore, two-phase flow is to be considered at the safety device inlet.

The prediction of the flow regime at the safety device inlet is then required in case of a *non-homogeneous* vessel model. In this case, Figure 4 shall be used to determine whether single or two-phase flow is present at the safety device inlet at sizing conditions.

In case the foaming behaviour of the liquid is not known from common practice, experiments are required to assess this property at the sizing conditions of the safety device, as it is difficult to predict the foaming behaviour merely from physical properties alone.



Key

1 external heat input due to fire, H/D = 1

2 external heat input due to fire, H/D = 2

3 non-foaming and low-viscous ($\Omega_{l,0} \leq 0{,}1$ Pa·s) (churn turbulent flow, $k_\infty = 1{,}53$)

4 external heat input due to fire, H/D = 5

5 foaming or viscous, $\Omega_{l,0} > 0{,}1$ Pa·s (bubbly flow, $k_\infty = 1{,}18$)

a Two-phase flow (below considered curve).

b Gas/vapour (above considered curve).

Figure 4 — Flow regime at safety device inlet (single-phase or two-phase flow)[23] — Critical filling threshold, ϕ_{limit}, as a function of the dimensionless bubble-rise velocity

The dimensionless bubble-rise velocity is calculated according to Formula (6):

$$u^* = \frac{u_{g,0}}{u_\infty} \cdot \psi \tag{6}$$

where

$$u_{g,0} = Q_{m,g,out} \cdot \frac{v_{g,0}}{A_R} \tag{7}$$

$$u_{\infty} = k_{\infty} \frac{\left[\sigma_{l,0} \cdot g\left(\rho_{l,0} - \rho_{g,0}\right)\right]^{1/4}}{\sqrt{\rho_{l,0}}} \tag{8}$$

If an internal heat release or bulk heat input is considered, e.g. due to a chemical reaction, the boiling area ratio Ψ is by Formula (9), $\Psi = 1$.

In case of an external heating of the vessel, e.g. due to steam, a conservative estimate of the boiling area ratio is $\Psi = 1$.

For an external heat input due to a fire beside the vessel but no underfireing of the vessel, Formulae (15) and (16) may be applied to determine a level swell. Here, A_{fire} shall be determined as the partial surface area (wetted mantle area) of a vertically cylindrical vessel with the heights between vessel bottom and fundament (ground) H_{vessel} and a maximum height of the fire $H_{\text{fire}} \leq 7{,}5\ \text{m}$.

$$\psi = \frac{A_R}{A_{\text{fire}}} = \frac{1}{4 \cdot \left(\frac{H_{\text{fire}} - H_{\text{vessel}}}{D_R}\right)} \tag{9}$$

6.3.4 Prediction of the flow regime (gas/vapour or two-phase flow)

6.3.4.1 General

For a top-mounted safety device, if ϕ_0 exceeds ϕ_{limit} a two-phase mixture is most probably vented for part of the venting time. When ϕ_{limit} is not exceeded, sizing should be based on single-phase flow. ϕ_{limit} is correlated with $u_{g,0}$ and u_{∞} (see Figure 4), which has been verified on a wide range of (non-foaming) fluids[9],[10].

Two calculation procedures for the determination of the flow regime at the inlet of the vent line system are distinguished according to the type of heat generation or input:

a) formation of homogeneously distributed bubbles in the liquid, e.g. in case of a thermal runaway reaction, see 6.3.4.2;

b) formation of bubbles at the inner vessel wall of the pressurized system, e.g. in case of fire or external heating, see 6.3.4.3.

6.3.4.2 Recommended calculation procedure in case of generation of homogeneously distributed bubbles (e.g. thermal runaway reaction)

To distinguish between single-phase and two-phase flow the following procedure is recommended.

a) Assume that the flow regime at the inlet of the vent line is single-phase gas/vapour flow.

b) Determine $Q_{m,\text{out}}$ according to the liquid reaction system, as follows:

1) For a tempered system, $Q_{m,\text{out}}$ is given by Formula (10):

$$Q_{m,\text{out}} = \frac{\dot{Q}}{\Delta h_{v,0}} \cdot \frac{1}{\dot{Q}_{\text{in}}^{*}} \tag{10}$$

where $\dot{Q}_{\text{in}}^{*}$ is calculated iteratively from Formula (11):

$$0 = \dot{Q}_{\text{acc}}^{*} - \dot{Q}_{\text{in}}^{*} + v^{*}\left(1 - \ln\left[\frac{v^{*}}{\dot{Q}_{\text{in}}^{*}}\right]\right) \tag{11}$$

where

$Q_{m,out}$ is the mass flow rate required to be discharged from a pressurized system;

$\dot{Q}$ is the heat input in the pressurized system by a runaway reaction, see Formula (35), in W;

$\dot{Q}_{in}^{*}$ is the ratio of total heat input to energy flow removed by evaporation;

$\dot{Q}_{acc}^{*}$ is the ratio of sensible heat to latent heat; see Formula (69);

v^{*} is the dimensionless specific volume; see Formulae (66), (67) or (68);

$\Delta h_{v,0}$ is the latent heat of vaporization at the sizing condition, in J/kg.

2) For a gassy system, $Q_{m,out}$ is as given by Formula (12):

$$Q_{m,out} = \Gamma_{g,0} \cdot M_0 \tag{12}$$

where

$\Gamma_{g,0}$ is the gas production rate per liquid mass, i.e. the gas mass flow rate per liquid mass inventory in the pressurized system at sizing conditions, in kg/(s·kg);

M_0 is the total liquid mass in the pressurized system, in kg.

For reacting systems, Γ_0 should be determined by experiment. For example, the pressure increase during a runaway reaction in an adiabatic reaction calorimeter is a measure of Γ. Γ shall be the maximum rate expected under adiabatic conditions, given that the system is not tempered and that the temperature increases during venting.

3) For hybrid systems (gas and vapour production), $Q_{m,out}$ is as given by Formula (13):

$$Q_{m,out} = \frac{\dot{Q}}{\Delta h_{v,0}} + \Gamma_{g,0} \cdot M_0 \tag{13}$$

where

$\dot{Q}$ is the heat input in the pressurized system by a runaway reaction, in W; see Formula (35);

$\Gamma_{g,0}$ is the gas production rate per liquid mass, i.e. the gas mass flow rate per liquid mass inventory in the pressurized system at the sizing conditions, in kg/(s kg);

M_0 is the total liquid mass in the pressurized system, in kg.

Although the system is tempered by the evaporating liquid, p_0 should be set equal to p_{MAA}, i.e. $\dot{Q}_{in}^{*} = 1$.

c) Using Figure 4, determine ϕ_{limit} from $u_{g,0}$ and u_∞. In case of a non-foaming and low-viscous ($\Omega_{l,0} \leq 0,1$ Pa·s) liquid, the solid line curve with k_∞ = 1,53 applies. For a viscous liquid system ($\Omega_{l,0}$ > 0,1 Pa·s) and for a foaming system, the dotted line curve with k_∞ = 1,18 applies. These values apply to vertical vessels with a height-to-diameter ratio of approximately 2:1 to 3:1. Special consideration shall be given for higher height-to-diameter ratios, horizontal or spherical vessels[10].

d) Compare ϕ_0 to ϕ_{limit}.

1) If $\phi_0 \geq \phi_{limit}$, two-phase flow should be assumed.

2) If $\phi_0 < \phi_{limit}$, single-phase gas flow should be assumed.

This is true only for a safety device with a discharge capacity $Q_{m,SD}$ less than or equal to $Q_{m,g,out}$. In practice, often a safety device with a nominal size larger than required is chosen from the standard selection of a manufacturer. If the device size is larger than required, a higher $u_{g,0}$ results and leads to a lower ϕ_{limit}; see Figure 4. As a consequence, this can result in a two-phase flow. A_0 originally sized for single-phase gas flow can then be too small and it can be necessary to re-evaluate it.

In this case, $Q_{m,out}$ should be set equal to $Q_{m,SD}$, as given in Formula (14):

$$Q_{m,SD} = \dot{m}_{SD,g} \cdot A_0 \tag{14}$$

where

$\dot{m}_{SD,g}$ is the dischargeable gas mass flux through the safety device, in kg/(m²·s);

A_0 is the narrowest flow cross-section of the safety device, in m².

$\dot{m}_{SD,g}$ should be calculated by using Formula (48) for gas/vapour flow. On using the new $Q_{m,SD}$, steps a) to d) should be repeated until the re-evaluated estimated flow regimes correspond. If two-phase flow is identified as the flow regime at the device inlet, no iteration is necessary.

With respect to Figure 4, ϕ_{limit} for highly viscous or foaming fluids are typically below 10 %. This most often leads to the assumption of two-phase flow.

6.3.4.3 Recommended calculation procedure in case of external heating due to a fire exposure of a vessel containing a non-foaming and low-viscous liquid

When boiling is due to external heating of the contents of the pressurized system and the fluid is non-foaming, bubbles can be formed mainly at the walls rather than in the bulk of the liquid. Provided that there are no internal baffles within the equipment, recirculation patterns form in the vessel and two-phase flow is less likely than in the case of homogeneously distributed bubble generation, which is addressed in 6.3.4.2[11]. These empirical formulae are valid only for vertical cylindrical vessels with no fire-heat transfer to the vessel bottom.

For example, Formulae (15) and (16) are given for fire exposure[12]:

$$\phi_{limit} = 1 - \left[2{,}279\,4 \cdot 10^{-4} \cdot \dot{q}_{fire}^{0{,}667} \left(0{,}089 + 1{,}000\,31 \cdot 10^{-7} \cdot \dot{q}_{fire}\right) \cdot \left(\frac{H_l}{D}\right)\right] \tag{15}$$

$$\dot{q}_{fire} = \frac{3{,}218 \cdot 10^5}{\left(\rho_{g,0} \cdot \Delta h_{v,0} \cdot u_\infty\right)} \frac{\dot{Q}}{A_{fire}} \tag{16}$$

where

$\dot{q}_{fire}$ is the dimensionless fire exposure flux in non-foaming and low-viscous fluids due to fire exposure transferred through the area, A_{fire}, into a vertical cylindrical vessel with no fire heat transfer to the vessel bottom at the sizing conditions;

A_{fire} is the partial surface area of a vertical cylindrical vessel wetted by internal liquid and located within 7,5 m vertically from ground or from any surface capable of sustaining a pool fire, including only the wall and not the bottom of the vessel, in m²;

u_∞ is the characteristic rise velocity of the gas/vapour in the liquid. The value of this variable is calculated using Formula (8) (see Figure 4) using a value of k_∞ = 1,53;

$\rho_{g,0}$ is the gas phase density in the pressurized system at the sizing conditions, in kg/m³;

$\Delta h_{v,0}$ is the latent heat of vaporization at the sizing condition, in J/kg;

$\dot{Q}$ is the heat input into the pressurized system (a vertical cylindrical vessel in this case) by fire, in W; see Formula (25);

H_l is the height of liquid level in a vertical cylindrical vessel, in m;

D is the inner vessel diameter in the free-board gas volume of a vertical cylindrical vessel, in m;

ϕ_{limit} is the critical filling threshold for the fire exposure.

If $\phi_0 \geq \phi_{limit}$, two-phase flow should be assumed.

Also, they neglect any two-phase flow due to droplet entrainment rather than due to level swell. This is important only for low-design-pressure tanks (unpressurized vessels). A method for deciding whether it is required to consider two-phase flow due to entrainment for low-design-pressure tanks is given in Reference [12].

6.4 Step 3 — Calculation of the mass flow rate required to be discharged

6.4.1 General

In 6.4.2 to 6.4.4, formulae are given for predicting the required flow rate to ensure that the vessel pressure remains below p_{MAA}. The particular set of formulae depends on the type of process deviation causing the pressure increase. It is not feasible to cover the whole range of possible deviations within this document; therefore, only some that are typically encountered in the industry are discussed as examples.

Pressure increase can be caused by:

a) excess flow into the equipment to be protected (e.g. feed control valve malfunction), see 6.4.2;

b) increased heat input due to fire exposure or external heating, see 6.4.3;

c) runaway reaction, see 6.4.4.

6.4.2 Pressure increase caused by an excess in-flow

A typical cause for excess in-flow is a control valve malfunction in a feed line. In this case, if there is a large pressure difference, flashing can occur in the pressurized system and a discharge of a two-phase mixture is required in order to prevent excess pressure increase.

So long as the incoming fluid is a pure liquid and does not cause boiling of the liquid already in the pressurized system, $Q_{m,out}$ can then be determined via a simplified mass balance formula, whereby credit may be taken for flow out of the protected system via other channels not affected by the feed control valve malfunction, such that $Q_{m,out}$ is as given by Formula (17) (see also the guidelines in Table A.1):

$$Q_{m,out} = \sum Q_{m,feed} \tag{17}$$

$Q_{m,feed}$ can be calculated with a representative discharge factor for a fully opened control valve, e.g. the aid of K_{vs}. For non-evaporating liquid flow through the control valve, this should be calculated as given by Formula (18):

$$A_{feed} \cong \frac{K_{vs}}{t1} \cdot \sqrt{\frac{\rho_{H_2O}}{2 \cdot \Delta p_{H_2O}}} = K_{vs} \cdot C_1 \tag{18}$$

where

$$C_1 = 1{,}964 \cdot 10^{-5} \tag{19}$$

C_1 is a constant including the conversion from typical units of K_{vs} into SI units, in $\frac{h}{m}$;

$t_1 = 3\,600$

t_1 is a constant for conversion into SI units, in s;

A_{feed} is the effective flow area, in m^2, of the feed line or control valve;

K_{vs} is the liquid discharge factor for fully opened control valve in the feed line, in m^3/h, as usually given by valve manufacturers;

ρ_{H2O} is the density of water at the temperature during experiments to measure the K_{vs} value, i.e. at the temperature of 5 °C to 50 °C, i.e. equal to about 1 000 kg/m^3;

Δp_{H2O} is the pressure drop across a control valve during experiments to measure the K_{vs} value defined at a pressure difference of 10^5 Pa.

K_{vs} equals the water volume flow rate for Δp_{H2O} of 10^5 Pa at a temperature of 5 °C to 50 °C; see IEC 60534-2-1.

$Q_{m,feed}$ for non-flashing liquid flow can then be calculated as given by Formula (20):

$$Q_{m,feed} = A_{feed} \cdot \sqrt{2 \cdot \rho_{l,0} \cdot (p_{CV} - [p_0 + \Delta p_{feed}])} \tag{20}$$

Substituting Formula (18) into Formula (20) allows the expression of the $Q_{m,feed}$, as given in Formula (21):

$$Q_{m,feed} = K_{vs} \cdot C_2 \cdot \sqrt{\rho_{l,0} \cdot (p_{CV} - [p_0 + \Delta p_{feed}])} \tag{21}$$

where

$$C_2 = 2{,}778 \cdot 10^{-5} \tag{22}$$

C_2 is a constant including the conversion from typical units of K_{VS} into SI units, in $\frac{h}{m}$;

$\rho_{l,0}$ is the liquid phase density in the pressurized system at the sizing conditions, in kg/m^3;

p_{CV} is the pressure upstream of the control valve in the feed line, in Pa;

p_0 is the sizing pressure of the equipment being protected, in Pa;

Δp_{feed} is the pressure loss between the outlet of the control valve in the feed line and the pressurized system, in Pa.

p_{CV} can be either the maximum operating pressure upstream of the control valve or p_{MAW} in the upstream pressurized system. The pressure being used should have been specified in the specification of the sizing case according to 6.2.

Formulae (18) to (22) are valid only for pure liquid flows through the control valve. If flashing is expected in the control valve, the method given in Reference [33] is recommended.

NOTE The effective area for compressible fluids can differ from that estimated by Formula (18) because the flow contraction in the control valve is generally lower and, hence $Q_{m,feed}$ can increase. It is most conservative to consider the geometrically smallest flow cross section in the control valve neglecting movable flow restrictions and flow contractions.

6.4.3 Pressure increase due to external heating

6.4.3.1 General

External heating resulting in two-phase flow includes heat input due to process heat transfer, fire exposure, or solar radiation, etc. Pressure can increase as a result of vaporization, desorption, or expansion of an enclosed liquid (e.g. liquefied gases).

The fire case is one of the main design cases for refinery industry installations and for the equipment for the storage/transport of liquefied gases on land or at sea. Details are included in the codes[13],[14],[15] and in Reference [41]. The key parameter for the safety device sizing is the heat flux. Depending on the field of application, this heat flux is modified, which results in different sizes of the safety device. In 6.4.3.2, the API recommended practice is used as an example.

6.4.3.2 Mass flow rate required to be discharged

The formulae to calculate $Q_{\mathrm{m,out}}$ are included in Table 3, based on different heat input mechanisms and fluid compositions, i.e. two-phase flow and single-phase gas/vapour or liquid flow.

$Q_{\mathrm{m,out}}$ should be calculated as given in Formula (10) in the case of vapour/liquid two-phase flow and external heating:

$$Q_{\mathrm{m,out}} = \frac{\dot{Q}}{\Delta h_{\mathrm{v,0}}} \cdot \frac{1}{\dot{Q}_{\mathrm{in}}^*} \tag{10}$$

where $\dot{Q}_{\mathrm{in}}^*$ is calculated iteratively as given by Formula (23):

$$\frac{1}{\dot{Q}_{\mathrm{in}}^*}\left(v^* - \dot{Q}_{\mathrm{in}}^* - \dot{Q}_{\mathrm{acc}}^*\right) - \ln\left(\frac{v^*}{\dot{Q}_{\mathrm{in}}^*}\right) = 0 \tag{23}$$

where

$\Delta h_{\mathrm{v,0}}$ is the latent heat of vaporization at the sizing condition, in J/kg;

$\dot{Q}_{\mathrm{in}}^*$ is the ratio of total heat input to energy flow removed by evaporation;

$\dot{Q}_{\mathrm{acc}}^*$ is the ratio of sensible heat to latent heat; see Formula (69);

v^* is the dimensionless specific volume of the homogeneous two-phase mixture at the sizing conditions in the pressurized system; see Formulae (66), (67) or (68);

$\dot{Q}$ is the heat input into the pressurized system; see Formula (24) or Formula (25).

For cases where $\dot{Q}_{\mathrm{acc}}^* = 0$, i.e. cases where $p_0 = p_{\mathrm{over}}$ that means ΔT_{over} = 0, and Formula (23) reduces to $\dot{Q}_{\mathrm{in}}^* = v^*$.

Depending on the type of heat transfer, $\dot{Q}$ is calculated as follows:

a) In the case of process heat transfer, $\dot{Q}$ is calculated as given by Formula (24):

$$\dot{Q} = B_{\mathrm{heat}} \cdot A_{\mathrm{heat}} \cdot \left(T_{\mathrm{heat}} - T_{\mathrm{sat}}\left(p_{\mathrm{over}}\right)\right) \tag{24}$$

where

B_{heat} is the maximum overall heat transfer coefficient (without fouling in the equipment), in W/(m^2.K), for the sizing case considered;

A_{heat} is the area of heat exchange in the pressurized system in case of external heat input, in m^2;

T_{heat} is the maximum possible temperature of the heat source, in K;

T_{sat} is the saturation temperature, calculated at p_{over} (the maximum pressure during the relief, in Pa) instead of p_0 (the sizing pressure, in Pa); credit is taken on Δp_{MAA}, see Figure 1.

b) In case of fire exposure, $\dot{Q}$ is calculated by using Formula (25):

$$\dot{Q} = 43\ 200 \cdot F \cdot A_{fire}^{0,82} \text{ for open pool fires} \tag{25}$$

$$\dot{Q} = 66\ 300 \cdot F \cdot A_{fire} \text{ for confined fires} \tag{26}$$

where

F is the environmental factor for heat input from fire;

A_{fire} is the total surface area wetted by internal liquid and located within 7,5 m vertically from ground or from any surface capable of sustaining a pool fire, in m^2.

The factor 43 200 is based on experiments[14], and the assumption of prompt fire fighting and adequate drainage. Where these conditions do not exist, a factor of 70 900 should be used. F is listed in Table 3 for insulated and non-insulated pressurized systems. A_{fire} including only the mantle and not the bottom of the vessel is raised to the power 0,82 to give credit to the fact that large equipment cannot be completely engulfed by a fire.

As described in 6.3.4.3, two-phase flow is less likely to occur in the case of external heating compared with the generation of homogeneously distributed bubbles.

If a fire cannot be excluded as a source of energy input into the pressurized system, where further energy sources are available like chemical reaction or heat input, the overall sum of energy release should be considered for sizing the safety device.

6.4.3.3 Heating of a cryogenic vessel in case of loss of vacuum

In the case of loss of vacuum, the space between the inner and outer shells may be filled either with air or with the gas stored. The heat input will depend on the sizing temperature and the type of insulation.

If the sizing temperature, T_0, is equal to or above 75 K, whatever the type of insulation, the heat input should be calculated as given in Formula (27):

$$\dot{Q} = \left(\frac{k_v A_s}{\delta_{ins}} + k_p \sum_{1}^{n} \frac{A_n}{l_n} \right) (T_A - T_0) \tag{27}$$

If the sizing temperature, T_0, is below 75 K, which implies that air or nitrogen may condense in the space between the inner and outer shells and the insulation is in perlite, the heat input should be that given by Formula (27) but with the value of thermal conductivity of insulation doubled.

If the sizing temperature, T_0, is below 75 K, but the insulation is multilayer, the heat input should be that given by Formula (27) or Formula (28), whichever is higher.

$$\dot{Q} = \frac{38\ 400 + X^{0,73}}{0,96 + X^{0,73}} A_i + k_p \sum_{1}^{n} \frac{A_n}{l_n} (T_A - T_0) \tag{28}$$

where

$\dot{Q}$	is the heat input to the pressurized system, in W;
k_v	is the mean thermal conductivity of insulating material on loss of vacuum, based on the highest of the values given in Table 2 for air and for the gas stored inside the cryogenic vessel;
A_s	is the average of the areas of the inner and outer shells of the cryogenic vessel, in m^2;
A_i	is the area of the inner shell of the cryogenic vessel, in m^2;
δ_{ins}	is the thickness of insulation expected to remain in case of loss of vacuum, in m;
T_A	is the ambient temperature, in K;
T_0	is the sizing temperature, in K;
k_p	is the mean thermal conductivity of piping and supports between T_A and T_0, in W/(m·K);
n	is the number of pipes and supports connected to the cryogenic vessel;
A_1, A_2,…A_n	is the area of cross-section of pipe or support 1 to n respectively, in m^2;
l_1, l_2,…l_n	is the length, inside the vacuum interspace, of pipe or support 1 to n respectively, in m;
X	is the number of insulation layers.

6.4.3.4 Heating of a cryogenic vessel in case of fire with insulation

In case of exposure to fire, if it is considered that the shell and the insulation inside will survive, the heat input will depend on the sizing temperature and the type of insulation.

If the sizing temperature, T_0, is equal to or above 75 K, whatever the type of insulation, the heat input should be calculated as given in Formula (29):

$$\dot{Q} = \left(2,6 \frac{k_f A_s^{0,82}}{\delta_{ins}} + k_f \sum_{1}^{n} \frac{A_n}{l_n} \right) (922 - T_0) \tag{29}$$

If the sizing temperature, T_0, is below 75 K, which implies that air or nitrogen may condense in the space between the inner and outer shells and the insulation is in perlite, the heat input should be that given by Formula (29) but with the value of thermal conductivity doubled.

If the sizing temperature, T_0, is below 75 K, but the insulation is multilayer, the heat input should be that given by Formula (29) or Formula (30), whichever is higher:

$$\dot{Q} = 1,95 \frac{92\ 160 + X^{0,73}}{0,96 + X^{0,73}} A_i^{0,82} + k_f \sum_{1}^{n} \frac{A_n}{l_n} (922 - T_0) \tag{30}$$

where k_f is the mean thermal conductivity of insulating material under fire conditions, based on the highest of the values given in Table 2 for air and the gas stored inside the cryogenic vessel, according to ISO 21013-3:2016, 4.2.3.

Table 2 — Thermal conductivity coefficients to be used in the cases of loss of vacuum (k_v) and fire engulfment (k_f)

Fluid	k_v W/(m·K)	k_f W/(m·K)
Air	0,019	0,043
Argon	0,013	0,027
Carbon dioxide	0,017	0,039
Carbon monoxide	0,020	0,039
Ethane	0,016	0,064
Ethylene (ethene)	0,015	0,056
Helium	0,104	0,211
Hydrogen	0,116	0,217
Krypton	0,007	0,015
Methane	0,024	0,074
Neon	0,034	0,067
Nitrogen	0,019	0,040
Nitrous oxide	0,014	0,038
Oxygen	0,019	0,043
Trifluoromethane	0,012	0,027
Xenon	0,005	0,009

6.4.3.5 Heating of a cryogenic vessel in case of fire without insulation

In case of exposure to fire, if it is considered that the shell and the insulation inside will not survive, the heat input should be calculated as given in Formula (31).

$$\dot{Q} = 70\ 900 A_i^{0,82} \tag{31}$$

where

$\dot{Q}$ is the heat input to the pressurized system, in W;

A_i is the area of the inner shell of the cryogenic vessel, in m^2.

In this case the heat transferred through piping and supports can be neglected.

6.4.4 Pressure increase due to thermal runaway reactions

6.4.4.1 General

For the calculation of $Q_{m,out}$, the type of (homogeneous) liquid-phase reaction system generating the pressure shall be identified first. These are vapour pressure (tempered), gassy, and hybrid systems; see 3.5.

In a tempered system, the majority of heat released by the chemical reaction is discharged in the form of vapour originating from the evaporation of one or more of the components. The rate of temperature increase is reduced or even stopped in case of adequate relief design. In gassy systems, gas is produced by chemical reaction and the vapour pressure of the reaction components is negligibly small in comparison. In this case, the temperature and reaction rate continue increasing despite pressure relief because relatively small heat energy is released with the gas. The temperature continues to increase until the reaction is almost complete. For a hybrid system, the pressure is due to both evolution of a permanent gas and increasing vapour pressure with rising temperature.

The kinetics and rates of heat or gas release are seldom available at the conditions of emergency relief. Therefore, the determination of these shall be experimentally determined in a laboratory with dedicated equipment (e.g. reaction calorimeters). Indeed, it is possible that the type of liquid-phase reaction system changes during the relieving process.

Depending on the type of liquid-phase reaction system, different formulae apply to the calculation of the mass flow rate required to be discharged. Formulae are given for two-phase gas/liquid mixtures and single-phase gas or liquid flow.

6.4.4.2 Tempered (vapour pressure) systems

$Q_{m,out}$ can be calculated from Formula (32):

$$Q_{m,out} = \frac{\dfrac{\dot{Q}}{\Delta h_{v,0}}}{\left(\sqrt{v^*} + \sqrt{\dot{Q}^*_{acc}}\right)^2} \tag{32}$$

where

$\dot{Q}$	is the heat input into the pressurized system, in W, see Formula (35);
$\Delta h_{v,0}$	is the latent heat of vaporization at the sizing condition, in J/kg;
v^*	is the dimensionless specific volume of the two-phase mixture in the pressurized equipment at the sizing pressure, see Formulae (66), (67) or (68);
$\dot{Q}^*_{acc}$	is the ratio of the sensible heat to the latent heat, see Formula (69).

In Formula (32), $\dot{Q}^*_{acc}$ characterizes an allowable further temperature increase, ΔT_{over}, as the difference between T_{sat} at p_{over} and T_{sat} at p_0, as given in Formula (33):

$$\Delta T_{over} = T_{sat}(p_{over}) - T_{sat}(p_0) \tag{33}$$

where

ΔT_{over}	is the difference between the saturation temperature at the maximum pressure during the relief, p_{over}, and the saturation temperature at the sizing pressure, p_0, in K;
T_{sat}	is the saturation temperature of the liquid, in K;
p_{over}	is the maximum pressure during the relief, in Pa;
p_0	is the sizing pressure, in Pa.

Here, p_{over} should not exceed p_{MAA}, so that Δp_{over}, is as given by Formula (34):

$$\Delta p_{over} = p_{over} - p_0 \le p_{MAA} - p_0 \tag{34}$$

where

p_{over}	is the maximum pressure during the relief, in Pa;
p_0	is the sizing pressure, in Pa;
p_{MAA}	is the maximum allowable accumulated pressure of the pressurized system, in Pa.

Δp_{over} is often chosen in the range of 10 % to 30 % of p_{open}. If $\Delta p_{over} = 0$, then $\dot{Q}^*_{acc} = 0$. By allowing for a pressure difference, the necessary size of the safety device is usually significantly reduced.

In case of reacting systems, p_{open} should usually be less than p_{MAW} of the equipment. By keeping p_{open} low, the discharge occurs at lower temperatures so that the reaction rate remains relatively low.

The heat input $\dot{Q}$ from the chemical reaction into the pressurized system is obtained by averaging the adiabatic rate of temperature increase corresponding to p_0 and p_{over}, as given in Formula (35):

$$\dot{Q} = \frac{M_0 \cdot c_{p,l,0}}{2} \left(\left. \frac{dT}{dt} \right|_0 + \left. \frac{dT}{dt} \right|_{over} \right) \tag{35}$$

where

M_0 is the total liquid mass in the pressurized system, in kg;

$c_{p,l,0}$ is the specific heat capacity of the liquid mixture at the sizing condition, in J/(kg·K);

$\left. \frac{dT}{dt} \right|_0$ is the rate of temperature increase in the pressurized system at the sizing conditions, in K/s;

$\left. \frac{dT}{dt} \right|_{over}$ is the rate of temperature increase in the pressurized system at the maximum pressure during relief, in K/s. If the maximum rate of temperature increase occurs between p_0 and p_{over}, this maximum reaction self-heat rate should be applied instead of the temperature increase at maximum pressure during relief.

Formula (35) is not applicable if the maximum rate of temperature increase occurs between p_0 and p_{over}. In this case, it is recommended to use the maximum reaction self-heat rate instead of that at p_{over}. It is important that the reaction self-heat rate correspond to that in an adiabatic reaction system.

6.4.4.3 Gassy and hybrid systems

$Q_{m,out}$ can be calculated from Formula (36):

$$Q_{m,out} = \Gamma_{g,0}\ M_0\ v_{g,0} / v_0 \tag{36}$$

where

$\Gamma_{g,0}$ is the gas production rate per liquid mass, i.e. the gas mass flow rate per liquid mass inventory in the pressurized system at the sizing conditions, in kg/(s·kg);

M_0 is the total liquid mass in the pressurized system, in kg;

v_0 is the specific volume of a homogeneous two-phase mixture in the system at the sizing conditions, in m^3/kg; see Formula (38);

$v_{g,0}$ is the specific volume of the gas phase at the sizing conditions, in m^3/kg; see Formula (73).

In contrast to the sizing for tempered systems, a “mass-loss allowance” is not included for gassy and most of the hybrid systems in order to keep the sizing procedure simple and yet conservative. A more accurate and realistic prediction is possible, if the loss of reactive mass from the vessel during the relief period is considered. This mass loss is considerably influenced by the swell-up behaviour in the vessel, which again influences the quality of the fluid at the inlet of the safety device. It is not conservative to assume a homogeneous flow regime in the vessel, since the mass loss would be overestimated and therefore the pressure increase underestimated. Because the consideration of the mass loss is, therefore, strongly dependent on a realistic model of the flow regime, the method described here does not consider a mass loss during the relief period. Further information on this topic is given in References [26], [27], [28], [29] and [30].

It is important that the maximum Γ correspond to that in an adiabatic runaway with equal starting pressure, concentrations, etc., postulated for the sizing case from step 1; see 6.2. Often, Γ shall be developed from dedicated experiments[16]. The use of Γ can sometimes lead to an oversizing if, during the relief process, the reactor is emptied before the maximum Γ occurs.

The relief of mixtures identified as hybrid systems may be conservatively treated by using the formulae for gassy systems.

For hybrid systems which are tempered during the whole relief time, Formula (37) is less conservative compared to Formula (36)[40]:

$$Q_{\mathrm{m,out}} = \frac{\Gamma_{\mathrm{g},0} M_0 v_{\mathrm{g},0}}{v_0} + \frac{\dot{Q}}{\Delta h_{v,0} \left(\sqrt{v^*} + \sqrt{\dot{Q}^*_{\mathrm{acc}}} \right)^2} \tag{37}$$

6.4.4.4 Stagnation mass flow quality and specific volume of the mixture at the inlet of the vent line system

$\dot{x}_0$ and v_0 entering the vent line system are to be calculated. For high viscous and for foaming fluids and if the properties of the reactor inventory are not known in detail, homogeneous vessel venting shall be assumed.

The mean value of v_0 of the homogeneous two-phase mixture is as given in Formula (38).

$$v_0 = \dot{x}_0 \cdot v_{\mathrm{g},0} + (1 - \dot{x}_0) \cdot v_{\mathrm{l},0} \tag{38}$$

$\dot{x}_0$ is determined from Formula (39):

$$\dot{x}_0 = \frac{\varepsilon_0 \cdot v_{\mathrm{l},0}}{(1 - \varepsilon_0) \cdot v_{\mathrm{g},0} + \varepsilon_0 \cdot v_{\mathrm{l},0}} \tag{39}$$

$$\varepsilon_0 = 1 - \phi_0 = 1 - \frac{V_{\mathrm{l},0}}{V} \tag{40}$$

where

ε_0 is the void fraction in the pressurized system at sizing conditions;

$V_{\mathrm{l},0}$ is the specific liquid volume at sizing conditions in the pressurized system, in m^3;

$v_{\mathrm{g},0}$ is the specific gas volume at sizing conditions in the pressurized system, in m^3/kg.

In case the liquid viscosity $\Omega_{\mathrm{l},0} > 0{,}1$ Pa·s, or is not known, homogeneous vessel venting regarding Formula (39) shall be assumed to calculate at the inlet of the safety device vent line[37].

For a non-foaming liquid with a viscosity $\Omega_{\mathrm{l},0} \leq 0{,}1$ Pa·s at sizing conditions, or in case experimental evidence has shown a significant vapour/liquid disengagement during relief, $\dot{x}_0$ may be determined from Formula (41).

$$\dot{x}_0 = \frac{v_\mathrm{l} - v_{0,\mathrm{ct}}}{v_\mathrm{l} - v_{\mathrm{g},0}} \tag{41}$$

$$v_{0,\mathrm{ct}} = v_{\mathrm{l},0} + \phi_\infty \left(\frac{v_{\mathrm{g},0} - v_{\mathrm{l},0}}{v_{\mathrm{g},0}} \right) \frac{1}{\dfrac{Q_{\mathrm{m,out}}}{A_\mathrm{R} \cdot u_\infty}} \tag{42}$$

$$\phi_\infty = \frac{2(1 - \phi_0)}{1 - C_0 (1 - \phi_0)} \tag{43}$$

where

Φ_0	is the initial maximum liquid level at sizing conditions;
$v_{0,ct}$	is the specific volume of the mixture for churn turbulent flow at sizing conditions, in m^3/kg;
A_R	cross-sectional area in a vertical cylindrical vessel.

The distribution factor C_0 is typically between 1 and 1,5. In case a churn turbulent level swell can be reasonably assumed, the mass flow rate to be discharged has to be known to calculate $v_{0,ct}$. Hence, $Q_{m,out}$ can only be calculated iteratively[38].

6.5 Step 4 — Calculation of the dischargeable mass flux through and pressure change in the vent line system

6.5.1 General

The dischargeable mass flow rate $\dot{m}_{SD}$ through a vent line system depends on the restriction of the fluid within all elements of the vent line system until either critical flow or the end of the vent line is reached. State of technology is to simultaneously calculate $\dot{m}_{SD}$ and Δp in the vent line comprising of the inlet line, safety device and outlet line. In safety valve vent line systems often the inlet pressure drop Δp is of minor importance as choking occurs typically in the safety valve. Historically, this led to a separate sizing of safety valve and the connected piping for gas service. This is not always conservative in the case of two-phase flashing flow systems, especially at low inlet mass flow qualities. Therefore, the vent line system should be sized simultaneously. In bursting disc vent line systems the resistance is generally governed by the piping elements, not the bursting disc, hence, simultaneous calculation of inlet line bursting disc and outlet line is mandatory.

Annex C gives an example of a method for the determination of Δp in a connected vent line system. Alternative methods are available, however, it is important to ensure that any method selected is relevant to the particular application and is correctly applied by those appropriately qualified and experienced.

The basis for sizing the safety device is the formula for the conservation of energy for a steady-state flow through an adiabatic restriction. Thus, $\dot{m}_{ideal}$ is as given in Formula (44)[36]:

$$\dot{m}_{ideal} = \sqrt{2 \cdot \frac{p_0}{v_0}} \cdot \frac{1}{v_{th}^*} \cdot \sqrt{\frac{-\int_{\eta_{in}}^{\eta_{th}} v^* \cdot d\eta - \frac{g \cdot \sin\theta \cdot L}{v_0 \cdot p_0}}{1 - \Gamma + \zeta_{v,ref} \cdot \left(\frac{A_{th}}{A_{in}} \cdot \frac{v_{ref}^*}{v_{th}^*} \right)^2}} \quad (44)$$

$$\eta = \frac{p}{p_0}$$

$$v^* = \frac{v}{v_0}$$

$$\beta th = \frac{dth}{d_{in}}$$

$$\Gamma = \left(\frac{A_{th}}{A_{in}} \cdot \frac{v_{in}}{v_{th}} \right)^2 = \left(\frac{\beta_{th}^{\ 2}}{v_{th}^*} \right)^2 \quad (45)$$

where

v_{seat}	is the specific volume in the narrowest flow cross-section between the pressurized system and the valve seat and minimum flow area for bursting discs, in m^3/kg;
p_{seat}	is the pressure in the narrowest flow cross-section (usually the valve seat), in Pa;
p_0	is the sizing pressure, in Pa;
A_{th}	is the flow cross-section at narrowest cross-section; In the case of a pipe or a bursting disc where the bursting disc has a larger available area on bursting than the cross-section of the vent line, $A_{th} = A_{in}$;
A_{in}	is the flow cross-section at inlet conditions;
L	is the vertical length of a flow restriction to account for potential energy change. For safety valves and bursting discs, L may be set to 0. For inlet and outlet lines, the heights of the system shall be considered;
v	is the specific volume in the nozzle between the inlet and the narrowest flow cross-section, in m^3/kg;
$\zeta_{v,ref}$	is the resistance coefficient at the reference cross-section;
p	is the pressure in the nozzle, in Pa;
Γ	is the dimensionless velocity ratio between the inlet and the throat of a nozzle;
g	is the acceleration due to gravity;
θ	is the angle of the discharge line to the horizontal.

Formula (44) is a direct integration method allowing the use of a formula of state for the specific volume of the mixture v between the inlet and the throat of a nozzle. It includes the inlet velocity, friction, heat exchange and the hydrostatic energy change in case of upward flow. Thermal, mechanical and chemical non-equilibrium effects due to very short relaxation times of the fluid in the nozzle like boiling or condensation delay and slip are implicitly incorporated in the specific volume v. For an ideal (adiabatic, frictionless) flow from a plenum through a nozzle, Formula (44) yields[36]:

$$\dot{m}_{ideal} = \frac{1}{v_{th}^*} \sqrt{-\int_1^{\eta_{th}} v^* \, d\eta} \qquad (46)$$

In order to evaluate this integral, it is necessary to know v as a function of pressure along the nozzle path from p_0 to p_{seat}. A number of models have been proposed in the literature for approximating this $v(p)$ relationship. In general, they are based on the assumption of a homogeneous equilibrium flow and empirical corrections are made for the thermal and mechanical non-equilibrium[25].

In industry, thermodynamic property data can be available from a database or else measured. These data are measured for each phase separately under equilibrium conditions. Then, typically, two-phase specific volume data are modelled by a linear summation of the single-phase specific volumes weighted by the quality. If homogeneous equilibrium flow is assumed, the integral can be evaluated numerically, i.e. neglecting boiling delay and slip.

Equilibrium models might not be fully appropriate for the flow through short nozzles and low-quality fluid flow. That is the reason why the thermal non-equilibrium is included empirically[18],[24],[26],[27] and[38]. Typically, the non-equilibrium effect is more pronounced than the change of property data between inlet and nozzle throat[34].

In the case that a linear relation for $v(p)$ is assumed, fluid thermo-physical property data at only one pressure point, e.g. p_0, are required, and this allows for evaluating the integral analytically, e.g. the Omega method of Leung[21]. Diener and Schmidt[18] modified this method to account for non-equilibrium effects and formed the basis for the method presented in 6.5. The latest modification was published as the HNE-CSE omega method[36], mainly to overcome some weaknesses of the classical omega methods, namely to

a) perform consistent results with ISO 4126-7:2013/ Amd 1:2016 for gas and liquid flow,

b) account for the inlet velocity to the safety device,

c) include the potential energy change in case of upward or downward flow,

d) account for frictional pressure drop between inlet and outlet,

e) cover the inlet conditions of initially subcooled liquid and low mass flow qualities,

f) include a model for the critical pressure ratio accounting for boiling delay which is applicable for integrated sizing of inlet line, safety device and outlet line (see 6.7).

The HNE-CSE method is included in this standard. Due to its enlargement it can be applied to all kinds of throttling devices as safety valves and bursting discs and also to straight pipes and pipe elements.

A simplified formula for the compressibility coefficient ω at the sizing conditions for safety valves and bursting discs with a significant flow restriction is given in Formula (47):

$$\omega = \frac{\left(\dfrac{v_{\text{seat}}}{v_0} - 1\right)}{\left(\dfrac{p_0}{p_{\text{seat}}} - 1\right)} \tag{47}$$

where

v_{seat} is the specific volume of the two-phase gas/liquid mixture in the narrowest flow cross-section of the safety valve, or the specific volume in the minimum flow area of a bursting disc, in m^3/kg;

v_0 is the specific volume in the pressurized system at the sizing conditions, in m^3/kg;

p_0 is the sizing pressure, in Pa;

p_{seat} is the pressure in the narrowest flow cross-section of the safety valve, or the pressure in the minimum flow area of a bursting disc, in Pa.

If η_{crit} has been established in the safety device, $p_{\text{seat}} = p_{\text{crit}}$. Otherwise, $p_{\text{seat}} = p_{\text{b}}$.

The above definition of ω involves property data at two pressure points and is more accurate than evaluating it at one pressure point. It is also valid for single- or multi-component systems close to the limitations of p_{red} and T_{red}. This definition has the disadvantage that it is necessary to know the property data in the narrowest flow cross-section of the safety device, which depend on the degree of thermodynamic non-equilibrium (boiling delay, slip of gas phase) in the valve. Generally, these data are not available. For simplification, ω can be based on Formula (47) with p_{seat} set to 80 % or 90 % of p_0 or on the sizing conditions (see 6.5.4), i.e. typically η_{crit} and assuming thermodynamic equilibrium conditions to calculate v_{seat} at p_{crit}. A comparison of the accuracy of the methods is given in References [34] and [26].

In case the flow is not significantly restricted, e.g. in straight pipes or wide opening bursting discs, Formula (47) cannot be applied. Here, the more general definition in 6.5.4 should be used.

The calculation of $\dot{m}_{\text{SD}}$ is given by Formula (48).

$$\dot{m}_{\text{SD}} = K_{\text{dr}} \cdot \dot{m}_{\text{ideal}} = K_{\text{dr}} \cdot C \cdot \sqrt{\frac{2 p_0}{v_0}} \tag{48}$$

where

K_{dr}	is the valve discharge coefficient; see Table 4;
$\dot{m}_{ideal}$	is the mass flux through a theoretically perfect (adiabatic, frictionless) nozzle, in kg/(m²·s);
C	is the dimensionless mass flow rate; see Formula (53);
p_0	is the sizing pressure, in Pa;
v_0	is the specific volume of the homogeneous two-phase mixture at sizing conditions in the pressurized system, in m³/kg; see Formula (38).

The calculation is performed in two steps. First, $\dot{m}_{ideal}$ is calculated through a theoretically perfect (adiabatic frictionless) nozzle. It is composed of C (see 6.5.3), which is a function of p_0, and v_0. Second, $\dot{m}_{ideal}$ is then reduced by using K_{dr}, related directly to this model.

$\dot{m}_{SD}$ is related to A_0.

In Table 4, formulae are given for liquid, two-phase mixture and single-phase gas/vapour flow.

6.5.2 Two-phase flow discharge coefficient, $K_{dr,2ph}$

NOTE 1 $K_{dr,2ph}$ is strongly dependent on the ideal mass flux model and the flow behaviour inside the safety device.

$K_{dr,2ph}$ shall either be determined experimentally or according to the following method as the weighted average of the certified discharge coefficients for single-phase flow according to Formula (49)[36]:

$$K_{dr,2ph} = K_{dr,g} \cdot \varepsilon_{seat} + (1 - \varepsilon_{seat}) \cdot K_{dr,l} \cdot K_v \quad (49)$$

where

ε_{seat} is the void fraction in the narrowest cross-section of the safety device at the sizing conditions for a homogeneous two-phase mixture, as given by Formula (50):

$$\varepsilon_{seat} = 1 - \frac{v_{l,0}}{v_0 \cdot \left[\omega \cdot \left(\frac{1}{\eta} - 1\right) + 1\right]} \quad (50)$$

$K_{dr,g}$	is the certified discharge coefficient for single-phase vapour/gas flow;
$K_{dr,l}$	is the certified discharge coefficient for single-phase liquid flow;
η	is the pressure ratio, either the critical pressure ratio η_{crit} (if $\eta_b < \eta_{crit}$) or the back-pressure ratio η_b (if $\eta_b \geq \eta_{crit}$);
$v_{l,0}$	is the specific volume of the liquid phase at the sizing conditions, in m³/kg;
v_0	is the specific volume of a homogeneous two-phase mixture in the system at the sizing conditions, in m³/kg;
ω	is the compressibility coefficient;
K_v	is the viscosity correction factor.

If the liquid viscosity at sizing conditions $\Omega_{l,0} \leq 0{,}1$ Pa·s:

$K_v = 1$

otherwise for $10 < Re < 100\,000$:

$$K_v = \left(0{,}993\,5 + \frac{2{,}878}{Re^{0{,}5}} + \frac{342{,}75}{Re^{1{,}5}}\right)^{-1{,}0} \tag{51}$$

where

$$Re = \frac{Q_m \cdot (1 - \dot{x}_{seat})}{3{,}6 \cdot \Omega_{l,0}} \sqrt{\frac{4}{\pi \cdot A_{th} \cdot (1 - \varepsilon_{seat})}} \tag{52}$$

NOTE 2 This method is based on the assumption that the value of the discharge coefficient for two-phase flow is in between those values for single-phase liquid and vapour flow. The result of this averaging procedure is in substantial agreement with the method presented in Reference [24]. More adequate results can be obtained by experiments for a specific safety device type.

NOTE 3 The use of this discharge coefficient is indispensably coupled with the homogeneous, partially non-equilibrium flow model used for calculation of C as given in 6.5.3.

NOTE 4 At thermodynamic critical condition of a fluid, $v_{l,0} = v_{g,0}$. Hence, $K_{dr,g} = K_{dr,l}$. To the current knowledge, no information is available about the compressibility effect on the discharge coefficient. If the flow in the narrowest flow cross-section of the safety device approached, it is possible that the thermodynamic critical point the discharge coefficient for gases is not valid anymore.

6.5.3 Dimensionless mass flow rate, C

C shall be calculated by using Formula (53).

NOTE 1 C is a function of ω. C also accounts for the presence of critical flow in the safety device.

$$C = \frac{\eta_{th}}{\omega(N) \cdot (1 - \eta_{th}) + \eta_{th}} \cdot \sqrt{\frac{(\eta_{in} - \eta_{th}) \cdot (1 - \omega(N)) + \omega(N) \cdot \ln\left(\frac{\eta_{in}}{\eta_{th}}\right) - \frac{g \cdot \sin\theta \cdot L}{v_0 \cdot p_0}}{1 - \Gamma + \zeta_{v,ref} \cdot \left(\frac{A_{th}}{A_{ref}}\right)^2 \cdot \left(\frac{\omega(N) \cdot (1 - \eta_{ref}) + \eta_{ref}}{\omega(N) \cdot (1 - \eta_{th}) + \eta_{th}} \cdot \left(\frac{\eta_{th}}{\eta_{ref}}\right)\right)^2}} \tag{53}$$

$$\Gamma = \left(\frac{A_{th}}{A_{ref}}\right)^2 \cdot \left(\frac{\omega(N) \cdot (1 - \eta_{in}) + \eta_{in}}{\omega(N) \cdot (1 - \eta_{th}) + \eta_{th}} \cdot \left(\frac{\eta_{th}}{\eta_{in}}\right)\right)^2 \tag{54}$$

where

η_{in} is the ratio between the static pressure at inlet and the sizing pressure at inlet. In case of a pressure vessel, $\eta_{in} = 1$;

η_{th} is the ratio of the pressure in the narrowest cross-section to the sizing pressure;

N is the non-equilibrium factor;

$\omega(N)$ is the compressibility factor accounting for non-equilibrium effects;

A_{ref} is the reference flow cross-section (either inlet or throat), in m^2;

η_{ref} is the ratio of the pressure in the reference cross-section to the sizing pressure.

In short, vertically directed nozzles or in bursting discs, the hydrostatic term may be neglected ($L=0$). The friction at the nozzle wall may be linked to a two-phase multiplier[36] but is often only of minor importance ($\zeta_{v,ref}=0$) in short nozzles.

NOTE 2 Γ represents the kinetic energy in front of the safety device.

6.5.4 Compressibility coefficient, ω (numerical method)

ω shall be specified according to Formula (55):

$$\omega(N)=\omega_{frozen}+\omega_{flash}\cdot N \tag{55}$$

NOTE In case of low mass flow qualities at the inlet of the safety device, the flow is typically isothermal, whereas at very high mass flow qualities the property change is almost isentropic. Therefore, ω changes depending on the mass flow quality between both states:

$$\omega_{frozen}=\dot{x}_0\cdot\frac{v_{g,is}}{v_0}\cdot\left[(1-k)+\frac{k\cdot\eta_{th}}{(1-\eta_{th})}\left(\frac{1}{\eta_{th}}^{\frac{1}{\kappa}}-1\right)\right] \tag{56}$$

$$v_{g,is}=\frac{1}{\eta_S}\cdot\frac{T_{is}}{T_0}\cdot v_{g,0} \tag{57}$$

$$T_{is}=T_0\cdot\eta_{th}^{\frac{\kappa-1}{\kappa}} \tag{58}$$

$$\omega_{flash}=\frac{cp_{l,0}\cdot T_0\cdot p_0\cdot\eta_s}{v_0}\cdot\frac{(v_{g,0}-v_{l,0})^2}{\Delta h_{v,0}^2} \tag{59}$$

where

$v_{g,is}$ is the isentropic specific volume, in m³/kg;

η_S is the subcooling ratio;

T_{is} is the isentropic temperature;

k is the factor for calculation of the ratio of the actual temperature difference to the maximum achievable temperature difference in isentropic gas flow;

κ is the isentropic coefficient.

The actual temperature difference between inlet and throat related to the maximum achievable temperature difference in isentropic gas flow shall be calculated according to Formula (60):

$$k=\frac{\dot{x}_0}{\dot{x}_0+(1-\dot{x}_0)\cdot\frac{c_{p,l,0}}{c_{p,g,is}}};\ k\in(0..1) \tag{60}$$

N reflects the boiling delay at high velocity flow through a short device. The exponent a is equally valid for initially subcooled, boiling or two-phase flow.

$$N=\left[\dot{x}_0+\omega_{Flash}\cdot\frac{v_0}{v_{g,is}-v_{l,0}}\cdot\ln\left(\frac{\eta_s}{\eta_{th}}\right)\right]^a \tag{61}$$

$$a=(\eta_s)^{-0,6}-\frac{3}{5}\cdot\Gamma_{HNE} \tag{62}$$

$$\text{if } \dot{x}_0 < 0{,}003\text{: } \Gamma_{\text{HNE}} = \frac{\dot{x}_0}{0{,}003}\text{, else: } \Gamma_{\text{HNE}} = 1 \tag{63}$$

$\eta_s = 1$ for saturated liquids and two-phase flow at the inlet of the safety device and yields for initially subcooled liquids:

$$\eta_s = \frac{p_S(T_0)}{p_0} \tag{64}$$

where

$c_{p,l,0}$ is the specific heat capacity of the liquid at sizing conditions, in J/(kg·K);

$c_{p,g,is}$ is the ideal specific heat capacity of the gas in an isentrop change of condition, in J/(kg·K);

Γ_{HNE} is the factor for consideration of the gas content in the non-equilibrium exponent;

a is the non-equilibrium exponent.

Formulae (53) to (64) shall be solved iteratively by varying η_{th} between 1 and η_b to get the maximum value for C either at a calculated pressure ratio in between 1 and η_b (critical flow) or at the pressure ratio η_b (subcritical flow).

It is recommended to start the iteration at η_{th} = 1 and subtract values of 0,01 or less until the maximum coefficient is computed. Modern mathematical solvers are more effective than this stepwise method.

$Q_{m,SD}$ shall be calculated based on an analytical solution of Formula (48).

$$\dot{m}_{SD} = \frac{Q_{m,SD}}{A_0} = K_{dr} \cdot C \cdot \sqrt{\frac{2p_0}{v_0}} \tag{48}$$

6.5.5 Calculation of the downstream stagnation condition

When applying the HNE-CSE model to calculate the pressure change through a vent line system, calculation of the fictitious stagnation downstream conditions after a certain pipe element is required. The calculated stagnation conditions are then used as input conditions to the following element of the vent line system.

A calculation method for the downstream stagnation conditions consists of the application of the HNE-CSE model for a frictionless flow through an ideal nozzle. After the outlet pressure ratio of a certain pipe element is calculated, the dischargeable mass flux $\dot{m}_{SD}$ through the same element can also be determined.

Therefore, the model is computed in such a way that the calculated outlet pressure ratio of the pipe element is used to determine the fictitious stagnation conditions at the same location in the vent line. A simultaneous calculation of the mass flow quality at the same location is required by means of an isenthalpic flash.

6.5.6 Slip correction for non-flashing two-phase flow

If the flow at the inlet of the safety device comprises of a non-flashing liquid and a non-condensable gas (e.g. air and water at ambient conditions), the slip between gas and liquid has a significant effect and the slip factor shall be calculated according to Formula (65).

$$\phi_{slip} = \sqrt{\frac{v_0}{v_{eq,0}}} = \frac{\sqrt{\dfrac{v_0}{v_{l,0}}}}{\sqrt{1+\dot{x}_0 \cdot \left[\left(\dfrac{v_{g,0}}{v_{l,0}}\right)^{1/6} - 1\right] \cdot \left[1+\dot{x}_0 \cdot \left[\left(\dfrac{v_{g,0}}{v_{l,0}}\right)^{5/6} - 1\right]\right]}} \tag{65}$$

where ϕ_{slip} is the slip factor.

In case of flashing flow, ϕ_{slip} = 1.

C shall be multiplied by ϕ_{slip}[34].

6.5.7 Slip correction for two-phase flow in straight pipes

The HNE-CSE method is based on the assumption of a homogeneous flow through a restriction like a valve, bursting disc or pipe, empirically improved for phenomena of non-equilibrium and slip. A slip correction may also be applied for pressure drop calculation in the inlet and/or outlet line (straight pipe) to adapt the here proposed method to certain pressure drop correlations including slip flow (e.g. Chisholm, HTFS, Friedel, or other correlation).

6.6 Step 5 — Ensure proper operation of safety valve vent line systems under plant conditions

This subclause applies to safety valves only. The piping and fittings from the protected equipment to the safety valve and from the safety valve to the ultimate fluid discharge location can have a significant effect on the performance of the safety valve. Proper design of these piping segments is essential to the proper pressure protection of the plant equipment.

In the case of single-phase flow (liquid, gas or vapour), ISO 4126-9 provides guidance on the design and installation of these piping systems connected to safety valves. The methods and limitations provided in ISO 4126-9 have been developed for and applied to safety valves in single-phase (liquid or vapour) service. The guidance provided in ISO 4126-9 (limitation of inlet pressure loss and back pressure to certain values) is generally specified by national codes or regulations but is based on experience with single-phase fluids.

In the case of two-phase flow, the limiting values developed for single-phase flow (3 % Δp_{loss} and 10 % p_b) are not necessarily applicable. Experimental work is required on this subject.

Proper operation of a safety valve integrated into a vent line system includes ensuring that the safety valve discharges at least the minimum required discharge mass flow rate while avoiding unstable operation of the safety valve. For example, valve chatter can occur during relief conditions if the forces acting on the valve piston lead to a high frequency opening and closing. Chattering of the valve can dramatically reduce the flow capacity and the valve body and/or the piping could fail[20].

The stability of the operation of the safety valve shall be considered. Dampers and balanced bellows are examples of possible solutions to limit the effects of instability. Additionally, the manufacturer should be consulted with respect to trim configuration.

Because the relationship between inlet pressure loss and safety valve instability is not definitively understood, detailed requirements for an engineering analysis are the responsibility of the user. The user's engineering analysis shall be documented. An engineering analysis shall not be used to accept a safety valve installation that has experienced chatter.

6.7 Simultaneous calculation of the dischargeable mass flux and pressure change in the vent line system

The pressure change through the vent line system shall be simultaneously calculated and the built-up back pressure evaluated.

Annex B shows an example of calculation of the mass flow rate to be discharged $Q_{m,out}$ from a pressurized system after evaluating the flow regime at the vent line inlet as a two-phase flow.

After evaluated $Q_{m,out}$, a simultaneous calculation of $\dot{m}_{SD}$ and pressure changes in the inlet line, safety device and outlet is required. Annex C shows an example of calculation on which a vessel is connected by an inlet pipe to a safety valve, followed by an outlet pipeline to the atmosphere. The same procedure is shown to be applicable also for the case of a bursting disc vent line system. The method consists of a first estimation of the $\dot{m}_{SD}$ through the safety device, only. The $\dot{m}_{SD}$ is then recalculated though the complete vent line system.

6.8 Summary of calculation procedure

Table 3 — Formulae used to calculate $Q_{m,out}$, the mass flow rate required to be discharged and $\dot{Q}$

	Known parameters	Liquid flow (bottom venting)	Two-phase flow	Gas/vapour flow
Inlet void fraction	ϕ_0	$\varepsilon_0 = 0$	$\varepsilon_0 = 1 - \phi_0 = 1 - \frac{V_{l,0}}{V}$ (40)	$\varepsilon_0 = 1$
Inlet mass flow quality	$v_{g,0}$ $v_{l,0}$	$\dot{x}_0 = 0$	$\dot{x}_0 = \frac{\varepsilon_0 \cdot v_{l,0}}{(1-\varepsilon_0)\cdot v_{g,0} + \varepsilon_0 \cdot v_{l,0}}$ (39) [a] $\dot{x}_0 = \frac{v_l - v_{0,ct}}{v_l - v_{g,0}}$ (41) [b]	$\dot{x}_0 = 1$
Inlet specific volume		$v_0 = v_{l,0}$	$v_0 = \dot{x}_0 \cdot v_{g,0} + (1-\dot{x}_0)\cdot v_{l,0}$ (38) [a]	$v_0 = v_{g,0}$
		$v^* = \frac{v_{l,0}}{v_{g,0} - v_{l,0}}$ (66)	$v^* = \frac{v_0}{v_{g,0} - v_{l,0}}$ (67)	$v^* = \frac{v_{g,0}}{v_{g,0} - v_{l,0}}$ (68)
		—	$v_{0,ct} = v_{l,0} + \phi_\infty \left(\frac{v_{g,0} - v_{l,0}}{v_{g,0}}\right) \frac{1}{\frac{Q_{m,out}}{A_R \cdot u_\infty}}$ (42) [c] $\phi_\infty = \frac{2(1-\phi_0)}{1 - C_0(1-\phi_0)}$ (43)	—
Dimensionless accumulated heat	ΔT_{over} $c_{pl,0}$ $\Delta h_{v,0}$	$\dot{Q}^*_{acc} = \frac{c_{pl,0} \cdot \Delta T_{over}}{\Delta h_{v,0}}$ (69)		
		$\Delta T_{over} = T_{sat}(p_{over}) - T_{sat}(p_0)$ (33)		
Tempered runaway reaction	M_0 $c_{p,l,0}$ $\Delta h_{v,0}$ $\left.\frac{dT}{dt}\right\|_0$ $\left.\frac{dT}{dt}\right\|_{over}$	$Q_{m,out} = \frac{\dot{Q}}{\Delta h_{v,0}} \cdot \frac{1}{\dot{Q}^*_{in}}$ (10) [d] $0 = \dot{Q}^*_{acc} - \dot{Q}^*_{in} + v^* \left[1 - \ln\left(\frac{v^*}{\dot{Q}^*_{in}}\right)\right]$ (11)	$Q_{m,out} = \frac{\frac{\dot{Q}}{\Delta h_{v,0}}}{\left(\sqrt{v^*} + \sqrt{\dot{Q}^*_{acc}}\right)^2}$ (32)	$Q_{m,out} = \frac{\dot{Q}}{\Delta h_{v,0}} \cdot \frac{1}{\dot{Q}^*_{in}}$ (10) [d] $0 = \dot{Q}^*_{acc} - \dot{Q}^*_{in} + v^* \left[1 - \ln\left(\frac{v^*}{\dot{Q}^*_{in}}\right)\right]$ (11)
		$\dot{Q} = \frac{M_0 \cdot c_{pl,0}}{2}\left(\left.\frac{dT}{dt}\right\|_0 + \left.\frac{dT}{dt}\right\|_{over}\right)$ (35)		

Table 3 *(continued)*

<table>
<tr><th></th><th>Known parameters</th><th>Liquid flow (bottom venting)</th><th>Two-phase flow</th><th>Gas/vapour flow</th></tr>
<tr><td>Gassy runaway reaction</td><td>M_0
v_0
$v_{g,0}$</td><td>—</td><td>$Q_{m,out} = \Gamma_{g,0} M_0 v_{g,0} / v_0$ (36) [e]</td><td>$Q_{m,out} = \Gamma_{g,0} M_0 v_{g,0} / v_0$ (36) [e]</td></tr>
<tr><td rowspan="2">Hybrid runaway reaction</td><td rowspan="2">M_0
v_0
$v_{g,0}$
$c_{p,l,0}$
$\Delta h_{v,0}$
$\left.\frac{dT}{dt}\right\|_0$
$\left.\frac{dT}{dt}\right\|_{over}$</td><td>—</td><td>$Q_{m,out} = \frac{\Gamma_{g,0} M_0 v_{g,0}}{v_0} + \frac{\dot{Q}}{\Delta h_{v,0}\left(\sqrt{v^*} + \sqrt{\dot{Q}_{acc}^*}\right)^2}$ (37)
NOTE Formula (37) is simplified compared to the original[23].</td><td>$Q_{m,out} = \frac{\Gamma_{g,0} M_0 v_{g,0}}{v_0} + \frac{\dot{Q}}{\Delta h_{v,0}}$ (70)</td></tr>
<tr><td colspan="3">$\dot{Q} = \frac{M_0 \cdot c_{pl,0}}{2}\left(\left.\frac{dT}{dt}\right\|_0 + \left.\frac{dT}{dt}\right\|_{over}\right)$ (35)</td></tr>
<tr><td rowspan="3">Abnormal external heat input</td><td rowspan="3">M_0
V
$c_{p,l,0}$
$\Delta h_{v,0}$
$v_{g,0}$
$v_{l,0}$
ΔT_{over}
see Formula (33)</td><td>$Q_{m,out} = \frac{\frac{\dot{Q}}{\Delta h_{v,0}}}{v^* + \frac{\dot{Q}_{acc}^*}{2,303}}$ (71)</td><td>$Q_{m,out} = \frac{\dot{Q}}{\Delta h_{v,0}} \cdot \frac{1}{\dot{Q}_{in}^*}$ (10) [f]</td><td>$Q_{m,out} = \frac{\dot{Q}}{\Delta h_{v,0}}$ (72)</td></tr>
<tr><td>—</td><td>$\frac{1}{\dot{Q}_{in}^*}\left(v^* - \dot{Q}_{in}^* - \dot{Q}_{acc}^*\right) - \ln\left(\frac{v^*}{\dot{Q}_{in}^*}\right) = 0$ (23)</td><td>—</td></tr>
<tr><td colspan="3">$\dot{Q} = B_{heat} A_{heat} \left[T_{heat} - T_{sat}\left(p_{over}\right)\right]$ (24)</td></tr>
<tr><td rowspan="3">Fire</td><td rowspan="3">M_0
V
$c_{p,l,0}$
$\Delta h_{v,0}$
$v_{g,0}$
$v_{l,0}$
ΔT_{over}
see Formula (33)</td><td>$Q_{m,out} = \frac{\frac{\dot{Q}}{\Delta h_{v,0}}}{v^* + \frac{\dot{Q}_{acc}^*}{2,303}}$ (71)</td><td>$Q_{m,out} = \frac{\dot{Q}}{\Delta h_{v,0}} \cdot \frac{1}{\dot{Q}_{in}^*}$ (10) [f]</td><td>$Q_{m,out} = \frac{\dot{Q}}{\Delta h_{v,0}}$ (72)</td></tr>
<tr><td>—</td><td>$\frac{1}{\dot{Q}_{in}^*}\left(v^* - \dot{Q}_{in}^* - \dot{Q}_{acc}^*\right) - \ln\left(\frac{v^*}{\dot{Q}_{in}^*}\right) = 0$ (23)</td><td>—</td></tr>
<tr><td colspan="3">$\dot{Q} = 43\,200 \cdot F \cdot A_{fire}^{0,82}$ for F, see Annex D (25) [g]</td></tr>
</table>

Table 3 *(continued)*

	Known parameters	Liquid flow (bottom venting)	Two-phase flow	Gas/vapour flow
Loss of vacuum in a cryogenic vessel	M_0 V $c_{p,l,0}$ $\Delta h_{v,0}$ $v_{g,0}$ $v_{l,0}$ ΔT_{over} see Formula (33)	$Q_{m,out} = \dfrac{\dfrac{\dot{Q}}{\Delta h_{v,0}}}{v^* + \dfrac{\dot{Q}^*_{acc}}{2,303}}$ (71)	$Q_{m,out} = \dfrac{\dot{Q}}{\Delta h_{v,0}} \cdot \dfrac{1}{\dot{Q}^*_{in}}$ (10) [f]	$Q_{m,out} = \dfrac{\dot{Q}}{\Delta h_{v,0}}$ (72)
		—	$\dfrac{1}{\dot{Q}^*_{in}}\left(v^* - \dot{Q}^*_{in} - \dot{Q}^*_{acc}\right) - \ln\left(\dfrac{v^*}{\dot{Q}^*_{in}}\right) = 0$ (23)	—
		$\dot{Q} = \left(\dfrac{k_v A_s}{\delta_{ins}} + k_p \sum_1^n \dfrac{A_n}{l_n}\right)(T_A - T_0)$ (27) [h] $\dot{Q} = \dfrac{38\ 400 + X^{0,73}}{0,96 + X^{0,73}} A_i + k_p \sum_1^n \dfrac{A_n}{l_n}(T_A - T_0)$ (28) [i]		
Fire in a cryogenic vessel with insulation	M_0 V $c_{p,l,0}$ $\Delta h_{v,0}$ $v_{g,0}$ $v_{l,0}$ ΔT_{over} see Formula (33)	$Q_{m,out} = \dfrac{\dfrac{\dot{Q}}{\Delta h_{v,0}}}{v^* + \dfrac{\dot{Q}^*_{acc}}{2,303}}$ (71)	$Q_{m,out} = \dfrac{\dot{Q}}{\Delta h_{v,0}} \cdot \dfrac{1}{\dot{Q}^*_{in}}$ (10) [f]	$Q_{m,out} = \dfrac{\dot{Q}}{\Delta h_{v,0}}$ (72)
		—	$\dfrac{1}{\dot{Q}^*_{in}}\left(v^* - \dot{Q}^*_{in} - \dot{Q}^*_{acc}\right) - \ln\left(\dfrac{v^*}{\dot{Q}^*_{in}}\right) = 0$ (23)	—
		$\dot{Q} = \left(2,6\dfrac{k_f A_s^{0,82}}{\delta_{ins}} + k_f \sum_1^n \dfrac{A_n}{l_n}\right)(922 - T_0)$ (29) [j] $\dot{Q} = 1,95\dfrac{92\ 160 + X^{0,73}}{0,96 + X^{0,73}} A_i^{0,82} + k_f \sum_1^n \dfrac{A_n}{l_n}(922 - T_0)$ (30) [k]		

Table 3 *(continued)*

	Known parameters	Liquid flow (bottom venting)	Two-phase flow	Gas/vapour flow
Fire in a cryogenic vessel without insulation	M_0 V $c_{p,l,0}$ $\Delta h_{v,0}$ $v_{g,0}$ $v_{l,0}$ ΔT_{over} see Formula (33)	$Q_{m,out} = \dfrac{\dfrac{\dot{Q}}{\Delta h_{v,0}}}{v^* + \dfrac{\dot{Q}^*_{acc}}{2,303}}$ (71)	$Q_{m,out} = \dfrac{\dot{Q}}{\Delta h_{v,0}} \cdot \dfrac{1}{\dot{Q}^*_{in}}$ (10) [f]	
		—	$\dfrac{1}{\dot{Q}^*_{in}}\left(v^* - \dot{Q}^*_{in} - \dot{Q}^*_{acc}\right) - \ln\left(\dfrac{v^*}{\dot{Q}^*_{in}}\right) = 0$ (23)	—
		$\dot{Q} = 70\ 900 A_i^{0,82}$ (31)		

[a] For liquid viscosity $\Omega_{l,0} > 0{,}1$ Pa·s, or in case of homogeneous vessel venting.
[b] For non-foaming low viscous flow.
[c] For non-foaming low viscous flow, where $Q_{m,out}$ is calculated iteratively.
[d] Where $\dot{Q}^*_{in}$ is calculated iteratively from Formula (11).
[e] Where $\Gamma_{g,0}$ is experimentally determined.
[f] Where $\dot{Q}^*_{in}$ is calculated iteratively.
[g] Where $F = 1$ indicates no insulation; for $F < 1$, see Annex D.
[h] For $T_0 \geq 75$ K, use Formula (27).
[i] For $T_0 \geq 75$ K and in case of multilayer insulation, use the higher $\dot{Q}$ value between Formulae (27) and (28).
[j] For $T_0 \geq 75$ K, use Formula (29).
[k] For $T_0 \geq 75$ K and in case of multilayer insulation, use the higher $\dot{Q}$ value between Formulae (29) and (30).

Table 4 — Formulae to calculate the mass flux through the safety device

Parameter		Highly subcooled liquid	Two-phase flow	Gas/vapour flow
Mass flux (all cases)	$\dot{m}_{SD}$		$\dot{m}_{SD} = \frac{Q_{m,SD}}{A_0} = K_{dr} \cdot C \sqrt{\frac{2 p_0}{v_0}}$ (48)	
Inlet void fraction	ε_0	$\varepsilon_0 = 0$	$\varepsilon_0 = 1 - \phi_0$ $= 1 - \frac{v_{l,0}}{V}$ (40)	$\varepsilon_0 = 1$
Mass flow quality	$\dot{x}_0$	$\dot{x}_0 = 0$	$\dot{x}_0 = \frac{\varepsilon_0 \cdot v_{l,0}}{(1-\varepsilon_0) \cdot v_{g,0} + \varepsilon_0 \cdot v_{l,0}}$ (39)	$\dot{x}_0 = 1$
Specific volume	v_0	$v_0 = v_{l,0}$	$v_0 = \dot{x}_0 \cdot v_{g,0} + (1-\dot{x}_0) \cdot v_{l,0}$ (38)	$v_0 = v_{g,0}$ $= \frac{Z \cdot R \cdot T_0}{p_0 \cdot M}$ (73)
Discharge coefficient	K_{dr}	$K_{dr} = K_{dr,l}$	$K_{dr,2ph}$ $= \varepsilon_{seat} \cdot K_{dr,g} + (1-\varepsilon_{seat}) \cdot K_{dr,l} \cdot K_V$ (49) $\varepsilon_{seat} = 1 - \frac{v_{l,0}}{v_0 \cdot \left[\omega \cdot \left(\frac{1}{\eta} - 1\right) + 1\right]}$ (50)	$K_{dr} = K_{dr,g}$
Critical pressure ratio	η_{crit}	$\eta_{crit} = \frac{p_{sat}(T_0)}{p_0}$ (74)	$0 = \eta_{crit}^2 + (\omega^2 - 2\omega)(1-\eta_{crit})^2 + ...$ $... + 2\omega^2 \cdot \ln(\eta_{crit}) + 2\omega^2 (1-\eta_{crit})$ (75) [a] $\eta_{crit} = 0{,}55 + 0{,}217 \cdot \ln(\omega) - 0{,}046 \cdot [\ln(\omega)]^2 + 0{,}004 \cdot [\ln(\omega)]^3$ (77) [b] $\omega = \omega(N=1) = \frac{\dot{x}_0 \cdot v_{g,0}}{\kappa \cdot v_0} + \frac{c_{p,l,0} \cdot p_0 \cdot T_0}{v_0} \cdot \left(\frac{v_{g,0} - v_{l,0}}{\Delta h_{v,0}}\right)^2$ (78)	$\eta_{crit} = ...$ $... = \left[\frac{2}{(\kappa+1)}\right]^{\frac{\kappa}{\kappa-1}}$ (76)
Flow coefficient [c]	C		$C = \frac{\eta_{th}}{\omega(N) \cdot (1-\eta_{th}) + \eta_{th}} \cdot \sqrt{\frac{(\eta_{in} - \eta_{th}) \cdot (1-\omega(N)) + \omega(N) \cdot \ln\left(\frac{\eta_{in}}{\eta_{th}}\right) - \frac{g \cdot \sin\theta \cdot L}{v_0 \cdot p_0}}{1 - \Gamma + \zeta_{v\_ref} \cdot \left(\frac{A_{th}}{A_{ref}}\right)^2 \cdot \left(\frac{\omega(N) \cdot (1-\eta_{ref}) + \eta_{ref}}{\omega(N) \cdot (1-\eta_{th}) + \eta_{th}} \cdot \left(\frac{\eta_{th}}{\eta_t}\right)\right)^2}}$ (53)	

Table 4 *(continued)*

Parameter	Highly subcooled liquid	Two-phase flow	Gas/vapour flow
		$$\Gamma = \left(\frac{A_{\text{th}}}{A_{\text{in}}}\right)^2 \left(\frac{\omega(\text{N})\cdot(1-\eta_{\text{in}})+\eta_{\text{in}}}{\omega(\text{N})\cdot(1-\eta_{\text{th}})+\eta_{\text{th}}}\cdot\left(\frac{\eta_{\text{th}}}{\eta_{\text{in}}}\right)\right)^2$$	(54)
		$$\omega(N) = \omega_{\text{Frozen}} + \omega_{\text{Flash}} \cdot N$$	(55)
		$$\omega_{\text{Frozen}} = \dot{x}_0 \cdot \frac{v_{\text{g,is}}}{v_0} \cdot \left[(1-k) + \frac{k\cdot\eta_{\text{th}}}{(1-\eta_{\text{th}})}\left(\left(\frac{1}{\eta_{\text{th}}}\right)^{\frac{1}{\kappa}} - 1\right)\right]$$	(56)
		$$v_{\text{g,is}} = \frac{1}{\eta_S}\cdot\frac{T_{\text{is}}}{T_0}\cdot v_{\text{g0}}$$	(57)
		$$T_{\text{is}} = T_0 \cdot \eta_{\text{th}}^{\frac{\kappa-1}{\kappa}}$$	(58)
		$$\omega_{\text{Flash}} = \frac{c_{\text{p,l,0}}\cdot T_0 \cdot p_0 \cdot \eta_{\text{s}}}{v_0}\cdot\frac{(v_{\text{g,0}} - v_{\text{l,0}})^2}{\Delta h_{\text{v0}}^2}$$	(59)
		with $\eta_{\text{th}} = \eta_{\text{g,crit}}$ if $\eta_{\text{b}} \leq \eta_{\text{g,crit}} = \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}}$ or $\eta_{\text{th}} = \eta_{\text{b}}$ if $\eta_{\text{b}} > \eta_{\text{g,crit}}$	(60)
		$$k = \frac{\dot{x}_0}{\dot{x}_0 + (1-\dot{x}_0)\cdot\frac{c_{\text{p,l,0}}}{c_{\text{p,g,is}}}};\quad k \in (0..1)$$	(61)
		$$N = \left[\dot{x}_0 + \omega_{\text{Flash}} \cdot \frac{v_0}{v_{\text{g,is}} - v_{\text{l,0}}}\cdot \ln\left(\frac{\eta_{\text{s}}}{\eta_{\text{th}}}\right)\right]^a$$	(62)
		$$a = \eta_s^{-0,6} - \frac{3}{5}\cdot\Gamma_{\text{HNE}}$$	(63)
		If $\dot{x}_0 < 0{,}003 : \Gamma_{\text{HNE}} = \frac{\dot{x}_0}{0{,}003}$, else: $\Gamma_{\text{HNE}} = 1$	(64)

a For flashing flow, with $\omega \leq 2$.

b For flashing flow, with $\omega > 2$.

c For critical flow: $\eta_{\text{b}} \leq \eta_{\text{crit}} \Rightarrow \eta_{th} = \eta_{\text{crit}}$, $\eta_{\text{crit}} = \frac{p_{\text{crit}}}{p_0}$ (79)

For subcritical flow: $\eta_{\text{b}} > \eta_{\text{crit}} \Rightarrow \eta_{th} = \eta_{\text{b}}$, $\eta_{\text{b}} = \frac{p_{\text{b}}}{p_0}$ (80)

Annex A
(informative)

Identification of sizing scenarios

Table A.1 — Possible causes for overpressure in a system to be protected

Reasons for a pressure increase	Calculation of the mass flow rate required to be discharged
Changed feed/exit mass flow rate:	
— inlet line: valve malfunction (e.g. sticking valve cone)	Fully opened process valve on an inlet line Take into account any constriction of the inlet, e.g. an orifice (pressure loss in process valve in inlet line is not taken into account). Maximum operating pressure or, in case of high hazard potential, maximum pressure possible on the high-pressure side, for example, relieving pressure of safety valve. (Pressure reducer is not acceptable as a safety system.) Maximum possible pumping pressure/flow rate
— pump/compressor operation failure	Fully closed outlet valve
— heat exchanger: leakage/pipe crack	
— unintentionally closed valve (power outage, wiring/operation failure)	
— blockage of downstream pipelines or equipment	
— overfilled tank	Maximum possible pumping pressure/flow rate
Increased energy input:	
— solar radiation	
— fire	Heat flux according to Table 3 and Reference [14]
— unusual/abnormal heating:	Maximum conductivity coefficient for clean heat exchange surfaces
thermal expansion of enclosed liquids	Temperature in the pressurized system equals the smaller of:
shutdown of steam controller	a) mixture boiling temperature at p_{open};
cleaning: heating of solvents	b) maximum heating temperature (e.g. steam temperature).
Decreased energy transfer:	Decreased energy removal
— cooler shutdown (cooling water/condenser)	Total energy shutdown
— mixer shutdown (broken mixer shaft)	
— poor mixing/separation	
— high viscosity	
Deterioration of the heat transfer (fouling)	Total or partial loss of energy removal

Table A.1 *(continued)*

Reasons for a pressure increase	Calculation of the mass flow rate required to be discharged
Changed reaction:	
— runaway reaction due to:	
cooler shutdown (electrical circuit/pump failure);	Characteristic quantities in safety engineering (e.g. temperature increase rate and also reaction enthalpy)
charging failure (sequence, amount, substances);	Measurement of the temperature and pressure changes during a step reaction of the operation in an adiabatic reaction calorimeter
dosing failure (sequence, substances, dosing rate); impurities (starter inactive, catalysis); side reactions (pH difference); incorrect coupling/interruption of flows;	Heat capacity of the steel of the pressurized system with the coolers included, if relevant. Pay attention to accumulation of reactive components (reaction stops due to a cooler breakdown or inactive catalyst; overdoses of an input).
back flow or flow from other systems; starting reaction or operating time; gas producing reaction (e.g. decomposition reaction) resulting from pollution (catalysis).	Gas production rate per initial mass inventory from laboratory tests, e.g. reaction calorimeter, see Formula (36)

Annex B
(informative)

Example calculation of the mass flow rate to be discharged

B.1 General

This annex provides an example of the calculations of the mass flow rate to be discharged though a safety device from an 8 m^3 polymerization reactor following a runaway reaction. Any flow restriction according to an inlet and outlet vent line can be accounted by the method given in Annex C.

B.2 Step 1 — Identification of the sizing case

B.2.1 General

NOTE See 6.2.

B.2.2 Input data

a) for the pressurized system:

$V = 8\ m^3$	volume of the pressurized system,
$M_0 = 6\ 000\ kg$	total liquid mass in the pressurized system at the sizing conditions,
$p_0 = 1\ MPa\ (10\ bar)$	sizing pressure,
$p_b = 0{,}1\ MPa\ (1\ bar)$	back pressure,
$p_{open} = p_0$	opening pressure,
$p_{over} = 1{,}2\ MPa\ (12\ bar)$	maximum pressure in a pressurized system during relief,
$p_{MAA} = 1{,}5\ MPa\ (15\ bar)$	maximum allowable accumulated pressure,
$\phi_0 = 0{,}85$	initial liquid filling level,
$A_v = 4\ m^2$	cross-sectional area of the vessel (pressurized system);

b) result of laboratory experiments:

$T_0 = 453{,}05\ K$	sizing temperature,
$p_{sat}(T_0) = 1\ MPa\ (10\ bar)$	saturation pressure,
$\Delta T_{over} = 20\ K$	saturation temperature difference (overtemperature),
$\frac{dT}{dt_0} = 0{,}083\frac{K}{s}$	reaction self-heat rate inside the pressurized system at the sizing condition,
$\frac{dT}{dt_{MAA}} = 0{,}183\frac{K}{s}$	reaction self-heat rate inside the pressurized system at maximum allowable accumulated conditions,

$\frac{dp}{dt_0} = 0{,}400 \frac{\text{MPa}}{\text{min}} \left(4 \frac{\text{bar}}{\text{min}} \right)$ maximum rate of pressure increase inside the pressurized system at the sizing condition;

The reaction mixture is a non-foaming system.

c) property data from appropriate sources:

$T_c = 647$ K thermodynamic critical temperature,

$p_c = 22{,}1$ MPa (221,3 bar) thermodynamic critical pressure,

$c_{pl,0} = 4\,650 \frac{\text{J}}{\text{kg} \cdot \text{K}}$ specific heat capacity at constant pressure (liquid phase),

$M = 36 \frac{\text{kg}}{\text{kmol}}$ relative molecular mass,

$\Delta h_{v,0} = 1\,826\,000 \frac{\text{J}}{\text{kg}}$ latent heat of vaporization,

$v_{l,0} = 0{,}001\,193 \frac{\text{m}^3}{\text{kg}}$ specific volume in the pressurized system (liquid phase),

$v_{g,0} = 0{,}198\,4 \frac{\text{m}^3}{\text{kg}}$ specific volume in the pressurized system (gas phase),

$\sigma_{l,0} = 0{,}000\,1 \frac{\text{N}}{\text{m}}$ surface tension of liquid inside the pressurized system at the sizing condition,

$\Omega_{l,0} = 0{,}01 \text{ Pa} \cdot \text{s}$ viscosity of liquid at the sizing condition,

$\kappa_0 = 1{,}3$ isentropic coefficient (gas phase);

d) device data: safety device data for pre-selected device type (given by manufacturer):

$K_{dr,g} = 0{,}77$ certified device discharge coefficient for single-phase gas/vapour flow,

$K_{dr,l} = 0{,}5$ certified device discharge coefficient for single-phase liquid flow.

B.2.3 Sizing a safety device — Application range of the method

NOTE See Clause 5.

The method is accurate for systems in which either or both of the following conditions are true.

a) The overtemperature is less than 90 % of the fluid's thermodynamic critical temperature:

$T_{over} = \Delta T_{over} + T_0$ — $T_{over} = 473{,}05$ maximum temperature during relief, in K

$$T_{red} = \frac{T_{over}}{T_c} \quad (1)$$

$T_{red} = 0{,}731$ reduced temperature

b) The overpressure is less than 50 % of the fluid's thermodynamic critical pressure:

$$p_{red} = \frac{p_{over}}{p_c}$$
(2)

$p_{red} = 0{,}054$ reduced pressure

In the case of a runaway reaction, the reaction self-heat rate in the pressurized system at the maximum pressure during relief should be less than 2 K/s. The maximum rate of pressure increase is restricted to 20 kPa/s (12 bar/min). The method is applicable as Formulae (B.1) and (B.2) hold:

$$\left.\frac{dT}{dt}\right|_0 = 0{,}083\frac{K}{s} \tag{B.1}$$

$$\left.\frac{dp}{dt}\right|_0 = 4\frac{bar}{min} \tag{B.2}$$

B.3 Step 2 — Flow regime at the inlet of the vent line system

Calculations for a non-foaming system and low-viscosity mixture at the sizing conditions are carried out as follows.

NOTE See 6.3.

$$\dot{Q} = \frac{M_0 \cdot c_{p,l,0}}{2}\left(\left.\frac{dT}{dt}\right|_0 + \left.\frac{dT}{dt}\right|_{over}\right)$$
(35)

$\dot{Q} = 2{,}316 \cdot 10^6$ W heat input into the pressurized system at the sizing condition

$$\dot{Q}^*_{acc} = \frac{c_{pl,0} \cdot \Delta T_{over}}{\Delta h_{v,0}}$$
(69)

$\dot{Q}^*_{acc} = 0{,}05$ ratio of the sensible heat to the latent heat

$$v^* = \frac{v_{g,0}}{v_{g,0} - v_{l,0}}$$
(68)

$v^* = 1$ dimensionless specific volume for single-phase vapour flow

$$0 = \dot{Q}^*_{acc} - \dot{Q}^*_{in} + v^*\left(1 - \ln\left[\frac{v^*}{\dot{Q}^*_{in}}\right]\right)$$
(11)

$\dot{Q}^*_{in} = 0{,}714$ Ratio of total heat input to energy removed per unit time by evaporation

$$Q_{m,g,out} = \frac{\dot{Q}}{\Delta h_{v,0}}\frac{1}{\dot{Q}^*_{in}}$$
(10)

$Q_{m,g,out} = 1{,}777$ vapour mass flow rate required to be discharged, in kg/s

$$u_{g,0} = \frac{Q_{m,g,out} \cdot v_{g,0}}{A_R}$$
(7)

$u_{g,0} = 6{,}29$ superficial gas velocity in the dome of the pressurized system, in cm/s

$$u_\infty = \frac{1{,}53 \cdot \left[\sigma_{l,0} \cdot g \cdot \left(\frac{1}{v_{l,0}} - \frac{1}{v_{g,0}}\right)\right]^{0{,}25}}{\sqrt{\frac{1}{v_{l,0}}}}$$
(8)

$u_\infty = 5{,}0\frac{cm}{s}$ bubble-rise velocity in churn turbulent flow at the sizing conditions

$u^* = \frac{u_{g,0}}{u_\infty} \cdot \psi$ (6)	$u^* = 1{,}252$	dimensionless bubble-rise velocity

The dimensionless bubble-rise velocity leads to a critical threshold of about 67 % (see Figure 4). Hence, the initial filling level of 85 % in the pressurized system under sizing conditions is larger than the filling threshold and two-phase discharge should be taken into consideration.

B.4 Step 3 — Calculation of the mass flow rate required to be discharged

Calculations for the flow rate required to be discharged (gas/liquid mixture) are carried out as follows.

NOTE See 6.4.

$\dot{Q} = \frac{M_0 \cdot c_{p,l,0}}{2} \left(\left.\frac{dT}{dt}\right\|_0 + \left.\frac{dT}{dt}\right\|_{over} \right)$ (35)	$\dot{Q} = 3{,}711 \cdot 10^6$ W	average heat input into the pressurized system
$\dot{Q}^*_{acc} = \frac{c_{pl,0} \cdot \Delta T_{over}}{\Delta h_{v,0}}$ (69)	$\dot{Q}^*_{acc} = 0{,}051$	ratio of the sensible heat to the latent heat
$\varepsilon_0 = 1 - \phi_0$ (40)	$\varepsilon_0 = 0{,}15$	initial gas hold-up
$\dot{x}_0 = \frac{\varepsilon_0 \cdot v_{l,0}}{(1-\varepsilon_0) \cdot v_{g,0} + \varepsilon_0 \cdot v_{l,0}}$ (39)	$\dot{x}_0 = 1{,}06 \cdot 10^{-3}$	mass flow quality
$v_0 = \dot{x}_0 \cdot v_{g,0} + (1-\dot{x}_0) \cdot v_{l,0}$ (38)	$v_0 = 1{,}402 \cdot 10^{-3} \frac{m^3}{kg}$	specific volume of the mixture
$v^* = \frac{v_0}{v_{g,0} - v_{l,0}}$ (67)	$v^* = 7{,}109 \cdot 10^{-3}$	dimensionless specific volume
$Q_{m,out} = \frac{\dot{Q}}{\Delta h_{v,0} \cdot \left(\sqrt{v^*} + \sqrt{\dot{Q}^*_{acc}}\right)^2}$ (32)	$Q_{m,out} = 21{,}147 \frac{kg}{s}$	mass flow rate (tempered)

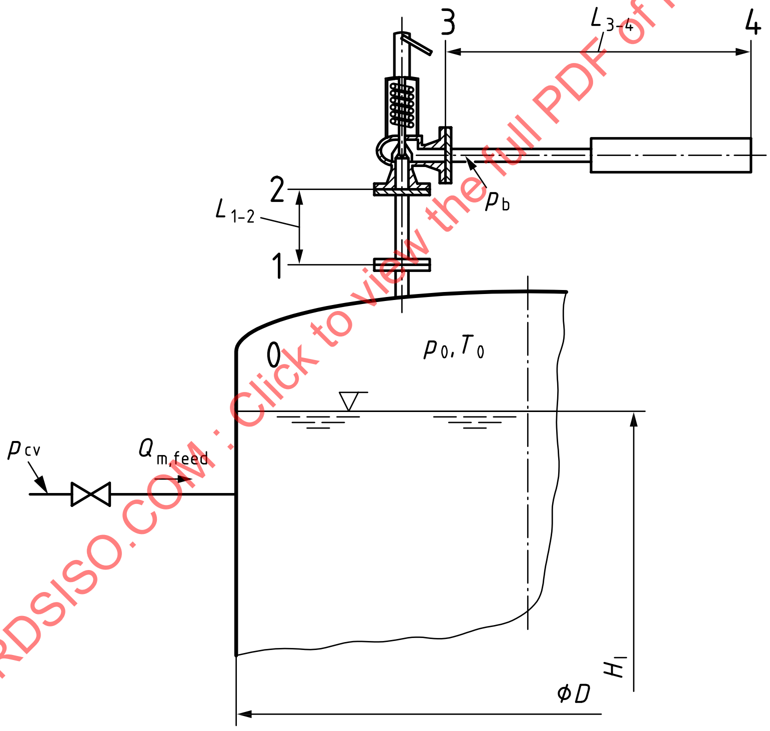
Annex C
(informative)

Example of calculation of the dischargeable mass flux and pressure change through connected vent line systems

C.1 General

An example of calculation of the dischargeable mass flux and pressure change through safety valve and bursting disc connected vent line systems is presented. Before estimating the dischargeable mass flux through a vent line system, the fluid phase composition at the inlet of the vent line, as well as the flowrate required to be discharged can be accessed according to Annex B.

In the present example, the pressurized vessel is connected by an inlet pipe to a safety device, followed by an outlet pipeline to the atmosphere as depicted in Figure C.1. In this example a safety valve was considered as the safety device.



Key

0	is the vessel
1	is the nozzle
2	is the valve inlet
3	is the valve outlet
4	is the end of line
L_{1-2}	is the inlet pipe length of the safety device, in m
L_{3-4}	is the outlet pipe length of the safety device, in m

Figure C.1 — Schematic of the sizing case including an inlet line, safety device and outlet line on top of a pressurized system

C.2 Input data

a) for the pressurized system:

Water and steam	medium,
p_0 = 1 MPa (10 bar)	vessel total inlet pressure,
p_b = 0,1 MPa (1,013 bar)	back pressure,
T_A = 20 °C	ambient temperature,
$\dot{x}_0$ = 0,000 1	vessel total mass flow quality,
T_0 = 180 °C	sizing temperature,
η_0 = 1	initial pressure ratio.

b) geometric parameters of the vent line system:

SV DN 25 × 40	safety valve type,
d_{th} = 23,0 mm	throat diameter,
d_{1-2} = 28,5 mm	inlet pipe inner diameter,
d_{3-4} = 85,5 mm	outlet pipe inner diameter,
L_{1-2} = 1 m	inlet pipe length,
L_{3-4} = 2 m	discharge pipe length.
$f = 0{,}01$	fanning friction factor,
$\zeta_{nozzle,inlet}$ = 0,5	resistance coefficient of the vessel exit nozzle.

c) device data: safety device data for pre-selected device type (given by manufacturer):

$K_{dr,g} = 0{,}7$	certified valve discharge coefficient for single-phase gas/vapour flow,
$K_{dr,l} = 0{,}45$	certified valve discharge coefficient for single-phase liquid flow.

C.3 Step 4 – Calculation of the dischargeable mass flux and pressure change through the vent line system

C.3.1 First estimate of the dischargeable mass flow rate

A first estimate for $Q_{m,SD}$ is obtained by calculation of the critical mass flow rate through the safety device without inlet and outlet line. Exceptions may occur at conditions close to the saturation line[39].

The HNE-CSE model is applied to each vent line segment by iterating the outlet pressure ratio of the respective segment until the previously estimated mass flow rate is reached.

After the outlet pressure ratio of a certain piping element is calculated, the downstream mass flow quality is determined assuming isenthalpic flash.

To calculate the critical pressure ratio in the narrowest flow cross-section of the safety device (throat), the following Formulae are iterated in such a way that $\frac{dC}{d\eta_{th}} = 0$. Formula (C.1) is a simplification of Formula (53) in case of flow through a vertically directed nozzle $\left(\sin\theta = 90°; L = 0\text{m}; \zeta_{v,ref} = 0; A_{in} = \infty\ m^2\right)$.

$$C(\eta_{th}) = \frac{\eta_{th}}{\omega(N)\cdot(1-\eta_{th})+\eta_{th}} \cdot \sqrt{(1-\eta_{th})\cdot(1-\omega(N))+\omega(N)\cdot\ln\left(\frac{1}{\eta_{th}}\right)} \tag{C.1}$$

$$\omega(N) = \dot{x}_0 \cdot \frac{v_{g,is}}{v_0} \cdot \left[(1-k)+\frac{k\cdot\eta_{th}}{1-\eta_{th}}\left[\left(\frac{1}{\eta_{th}}\right)^{\frac{1}{\kappa}}-1\right]\right] + \frac{cp_{l,0}\cdot T_0\cdot p_0\cdot\eta_s}{v_0} \cdot \frac{\left(v_{g,0}-v_{l,0}\right)^2}{\Delta h_{v,0}^2} \cdot N \tag{C.2}$$

$$k = \frac{\dot{x}_0}{\dot{x}_0 + (1-\dot{x}_0) \cdot \frac{c_{p,l,0}}{c_{p,g,is}}} \tag{60}$$

$$N = \left[\dot{x}_0 + \frac{cp_{l,0}\cdot T_0\cdot p_0\cdot\eta_s}{v_0} \cdot \left(\frac{v_{g,0}-v_{l,0}}{\Delta h_{v,0}}\right)^2 \cdot \frac{v_0}{v_{g,is}-v_{l,0}} \cdot \ln\left(\frac{\eta_s}{\eta_{th}}\right)\right]^a \tag{C.3}$$

$$v_{g,is} = \frac{1}{\eta_S} \cdot \frac{T_{is}}{T_0} \cdot v_{g,0} \tag{57}$$

$$T_{is} = T_0 \cdot \eta_{th}^{\frac{\kappa-1}{\kappa}} \tag{58}$$

where

C	is the dimensionless mass flow rate;
$\omega(N)$	is the compressibility factor at inlet stagnation conditions;
k	is the factor for calculation of the flashing/frozen flow;
N	is the non-equilibrium factor N;
a	is the non-equilibrium exponent (see Formula (62));
$\dot{x}_0$	is the mass flow quality at sizing conditions;
p_0	is the total pressure at sizing conditions, in Pa;
T_0	is the temperature at sizing conditions, in K;
v_0	is the specific volume at the inlet of the vent line at sizing conditions, in m^3/kg;
η_{th}	is the ratio of the static pressure at the throat to the stagnation inlet pressure;
$c_{p,l,0}$	is the specific heat capacity of the liquid at sizing conditions, in J/(kg·K);
$c_{p,g,is}$	is the isentropic specific heat capacity of the gas, in J/(kg·K);
$v_{g,is}$	is the isentropic specific volume, in m^3/kg;
$v_{l,0}$	is the specific volume of the liquid phase at the sizing conditions, in m^3/kg;
$v_{g,0}$	is the specific volume of the gas phase at the sizing conditions, in m^3/kg;

$\Delta h_{v,0}$ is the latent heat of vaporization at the sizing condition, in J/kg;

η_S is the subcooling ratio;

T_{is} is the isentropic temperature;

Γ_{HNE} is the factor for consideration of the gas content in the non-equilibrium exponent (see Formula (63));

κ is the isentropic coefficient.

The results of the first estimate of the dischargeable mass flowrate are obtained as follows:

$$C_{crit} = 0{,}453$$

$$\eta_{crit} = 0{,}688$$

where

C_{crit} is the critical dimensionless mass flow rate;

η_{crit} is the critical pressure ratio.

The dischargeable mass flow rate $Q_{m,estimated}$ is recalculated as follows (see also Formulae (48) to (50)):

$$\varepsilon_{seat} = 1 - \frac{v_{l,0}}{v_0 \cdot \left[\omega(N) \left(\frac{1}{\eta_{crit}} - 1 \right) + 1 \right]} = 0{,}252 \tag{50}$$

$$K_{dr,2ph} = K_{dr,g} \cdot \varepsilon_{seat} + (1 - \varepsilon_{seat}) \cdot K_{dr,l} \cdot K_v = 0{,}513 \tag{49}$$

$$Q_{m,estimated} = C_{crit} \cdot \sqrt{2 \cdot \frac{p_0}{v_0}} \cdot A_{th} \cdot K_{dr,2ph} = 4{,}055 \tag{C.4}$$

where

ε_{seat} is the void fraction at the safety valve seat;

$K_{dr,2ph}$ is the discharge coefficient of the safety device for two-phase flow;

$Q_{m,estimated}$ is the dischargeable mass flow rate, in kg/s;

A_{th} is the narrowest flow cross-section in the safety device, i.e. the valve seat diameter A_0 or the minimum net flow area (MNFA) of a bursting disc;

K_v is the viscosity correction factor.

C.3.2 Calculation of the pressure drop between location 0 and 1

The pressure drop after the inlet nozzle (location 0 to location 1 in Figure C.1) can be calculated by applying the HNE-CSE model with a resistance coefficient $\zeta_{nozzle,inlet}$ = 0,5 (sharp edged inlet flow).

The ratio between the static pressure at location 1 and the inlet pressure at sizing conditions ($\eta_1 = p_1 / p_0$) is calculated starting at η_1 = 1 and reducing this value until one of the following clauses is verified (see 6.5.4 for the generic form of the formulae):

a) the dimensionless mass flow rate $C(\eta_1) = \frac{(Q_{m,estimated} / A_1)}{\sqrt{\frac{2 \cdot p_0}{v_0}}}$, where A_1 is the cross-sectional area at location 1, or

b) a maximum of the dimensionless mass flow rate is reached at the critical pressure ratio. In this case, the estimated mass flow rate is reduced, and the above-described procedure is repeated.

The following assumptions/boundary conditions apply:

$$A_0 = \infty$$

$$\zeta_{v,ref} = \zeta_{nozzle,inlet} = 0{,}5$$

$$\theta = 90°$$

$$L = 0$$

where

A_0	is the cross-sectional area at location 0, in m^2;
$\zeta_{v,ref}$	is the resistance coefficient at the nozzle inlet;
θ	is the angle of the inlet line to the horizontal;
L	is the length of the nozzle, in m.

The application of the HNE-CSE model between location 0 and 1 yields:

$$\eta_1 = 0{,}991$$

$$p_1 = 9{,}909$$

where

η_1	is the pressure ratio at location 1;
p_1	is the static pressure at location 1, in bar.

C.3.3 Calculation of the downstream stagnation conditions

By means of an isenthalpic flash, the static mass flow quality can be deduced from the previously calculated p_1 and from the stagnation enthalpy h_t:

$$h_t = 762\ 716$$

$$\dot{x}_1 = \frac{h_t - e_{pot,1} - e_{kin,1} - e_{diss,1} - h_{l,1}}{h_{g,1} - h_{l,1}} \quad \text{(C.5)}$$

$$e_{diss,1} = \zeta_{nozzle,inlet} \cdot e_{kin,1} \tag{C.6}$$

$$e_{kin,1} = \frac{\dot{x}_0}{2}\left[\frac{Q_{m,estimated}}{A_1} \cdot \dot{x}_0 \cdot v_{g,1}\right]^2 + \frac{1-\dot{x}_0}{2}\left[\frac{Q_{m,estimated}}{A_1} \cdot (1-\dot{x}_0) \cdot v_{l,1}\right]^2 \tag{C.7}$$

$$e_{pot,1} = L \cdot g \cdot \sin\theta \tag{C.8}$$

where

h_t is the stagnation enthalpy at sizing conditions, in J/kg;

$\dot{x}_1$ is the static mass flow quality at location 1;

$v_{l,1}$ is the specific volume of the liquid phase at location 1, in m^3/kg;

$v_{g,1}$ is the specific volume of the vapour phase at location 1, in m^3/kg;

$h_{g,1}$ is the static enthalpy of the vapour phase at location 1, in J/kg;

$h_{l,1}$ is the static enthalpy of the liquid phase at location 1, in J/kg;

$e_{kin,1}$ is the kinetic energy at location 1, in J/kg;

$e_{diss,1}$ is the dissipation energy at location 1, in J/kg;

$e_{pot,1}$ is the potential energy at location 1, in J/kg;

Applying Formulae (C.5) to (C.8), the static mass flow quality at location 1 yields:

$$\dot{x}_1 = 0{,}000\ 96$$

where $\dot{x}_1$ is the static mass flow quality at location 1;

The stagnation flow conditions are required at the inlet of the next segment. As a first estimate, the stagnation mass flow quality can be approximated by means of an isenthalpic flash.

$$\dot{x}_{t,1} = \frac{h_t - e_{pot,1} - h_{l,t,1}}{h_{g,t,1} - h_{l,t,1}} \tag{C.9}$$

$$p_{t,1} = p_0 - \zeta_{nozzle,inlet} \cdot \left(\frac{Q_{m,estimated}}{A_1}\right)^2 \cdot \frac{v_1}{2} \tag{C.10}$$

where

$\dot{x}_{t,1}$ is the stagnation mass flow quality at location 1;

$p_{t,1}$ is the stagnation pressure at location 1, in Pa;

$h_{g,t,1}$ is the stagnation enthalpy of the vapour phase at location 1, in J/kg;

$h_{l,t,1}$ is the stagnation enthalpy of the liquid phase at location 1, in J/kg;

v_1 is the specific volume of the two-phase fluid at location 1, in m^3/kg;

Applying Formulae (C.9) and (C.10), the stagnation pressure at location 1 yields:

$$p_{t,1} = 9{,}965$$

$$\dot{x}_{t,1} = 0{,}000\ 43$$

where

$p_{t,1}$ is the stagnation pressure at location 1, in bar;

$\dot{x}_{t,1}$ is the stagnation mass flow quality at location 1, in bar.

An alternative calculation method for the downstream stagnation conditions is described under 6.5.5. It comprises a procedure on which the outlet pressure ratio η_1 is used to determine the fictitious stagnation conditions at location 1 by means of an isenthalpic flash.

C.3.4 Calculation of the pressure drop between location 1 and 2

The pipe loss coefficient may be determined by Formula (C.11), where ζ_i represents the additional loss coefficients, e.g. for bends, if existing in the considered section. In the following the pressure drop through the pipe is calculated using a one-step calculation method. In order to receive more accurate results for two-phase flashing flow the pipe should be discretized and solved numerically.

The pipe is considered long enough for the thermal equilibrium to be reached, so that boiling delay can be neglected in this case. Alternatively, to obtain more accurate results, the non-equilibrium exponent a should be adjusted to match experimental results.

$$\zeta_{pipe} = \frac{4 \cdot f \cdot L}{d} + \sum \zeta_i \qquad \text{(C.11)}$$

where

ζ_{pipe} is the pipe loss coefficient;

f is the fanning friction factor;

L is the length of the considered pipe element;

d is the diameter of the considered pipe element;

ζ_i are the additional loss coefficients, e.g. for bends, if existing in the considered section.

The frictional pressure loss of the pipe should be determined by an appropriate two-phase pressure drop model. In the literature are several models available, typically given as a two-phase multiplier[36]. Slip and flow pattern dependencies should be covered by this multiplier. To fit into the HNE-CSE methodology, any of the two-phase multiplier can be rearranged into a dimensionless frictional pressure drop Γ^*_{fric}, which is a function of the two-phase multiplier R_{2ph}. According to the Continuity law this relationship can be represented as following:

$$\zeta_{pipe} = \frac{\Delta p_v}{\dfrac{Q_{m,D}^{\,2} \cdot v_{ref}}{2 \cdot A_{ref}^{\,2}}} = \frac{\dfrac{\Delta p_v}{p_0}}{C^2 \cdot \dfrac{A_{out}^{\,2}}{A_{ref}^{\,2}} \cdot v^*_{ref}} = \frac{\Gamma^*_{fric}\left(R_{2ph}^{\,2}\right)}{C^2 \cdot \dfrac{A_{out}^{\,2}}{A_{ref}^{\,2}} \cdot v^*_{ref}}$$

$$\Gamma^*_{fric} = R_{2ph}^{\,2} \cdot C^2 \cdot \frac{L}{D} \cdot \lambda_l \cdot \frac{v_l}{v_0}$$

$$v^*_{ref} = \omega(N)_{ref}\left(\frac{1}{\eta_{ref}} - 1\right) + 1$$