Sizing of safety valves for non-flashing gas-liquid flow to protect a heat exchanger – application of the HNE-CSE model

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Sizing of an overpressure protection device for a tube bundle heat exchanger is challenging, especially when safety devices have to be sized for two-phase flow. There is a lack of reliable calculation models to determine the mass flow rate to be discharged.

A risk analysis for a heat exchanger is presented, with a high pressurized air-filled shell and a low-pressure oil-filled tube. The scenario tube leakage – not tube rupture – is selected as the worst-case scenario for the sizing calculation of a safety valve. The calculation of the required discharge area of the safety valve is performed in two steps. First, the mass flow rate of air entering the oil-filled tube after leakage has to be determined. The mass flow rate depends on the leakage position, because friction losses and pressure drop through the pipe cannot be neglected. As a second step, the mass flow rate to be discharged through the safety valve is calculated.

The air content during discharge varies over the time. Starting with a pure (non-flashing) oil, the gas mass flow quality of air increases as the air is entering the heat exchanger tube through the bore hole. The maximum mass flow rate to be discharged through the safety valve needs to be calculated with a consistent set of sizing equations, applicable for both subcritical and critical flow conditions. Additionally, the transient from low quality isothermal two-phase flow into a rather isentropic fluid flow at the end of the discharge with low mass flow qualities has to be precisely modeled. Sizing models in the literature are typically based on energy balances and developed only for the isothermal change of fluid state. However, the HNE-CSE model based on a non-equilibrium ω(N) equation of state consists of both changes of fluid state, an entrance velocity and the velocity head in vertically upward flow. It has a much wider application range and yields more precise results for the calculated dischargeable mass flow rates.

1. Introduction

Sizing methods for overpressure protection devices on tube bundle heat exchangers, especially when operated with multiphase flow are hardly to be found in common standards. The sizing equations of safety devices are handled in the standard ISO 4126, Parts 1 [1] , 3 [2] and 10 [3]. ISO 4126-10 describes the procedure for sizing a safety valve in 5 steps:

1. Identification of the sizing case
2. Determination of the fluid-phase composition at the safety valve inlet (single- or two-phase flow)
3. Calculation of the mass flow rate required to be discharged from the pressurized system
4. Determination of the dischargeable mass flux through the safety valve
5. Check of the proper valve operation in vent line systems under plant conditions

In the first step, conceivable deviations from normal plant operation are identified and their hazardous risk potential is evaluated. Therefore, several well-established procedures like HAZOP (hazard and operability study), what-if analysis or fault tree analysis may be applied in practice, often supplemented by checklists in various levels of detail. For the identification of the fluid-phase state at the safety valve inlet (step 2), phenomena like level swell together with the foaming behaviour of the fluid have to be considered. In step 3, the mass flow rate to be discharged from the pressurized equipment is calculated according to the overpressure scenario evaluated in step 1. The dischargeable mass flux through the safety valve (also known as safety valve capacity pro m² of the valve seat) is determined in step 4. To enable a stable valve operation, the pressure loss in the inlet and outlet piping is restricted to limits dictated in the standards. The adherence to these criteria is verified in step 5.

According to typical sizing standards like ISO 4126-7 [4] and -10 [3] or AD 2000 A2 [5] the dischargeable mass flow rate through the safety valve without inlet and outlet piping is calculated. In a separate step, the stable operation of the safety valve depending on the installation of inlet and outlet lines of the valve is evaluated. This procedure does not represent the typical sizing of safety valve vent lines with current sizing tools and may lead to significant errors in low mass flow quality flashing two phase flow. Nowadays it is usual to simulate simultaneously the mass flow rate and the pressure drop within the whole inlet and outlet piping system including all fittings like pipes, bends, tees and the valve. This approach is e.g. applied within the European Industrial Sizing Group EURISG, which was founded in 2015 to discuss and evaluate sizing of safety devices in industrial applications. The group consists of 14 companies from the chemical and petrochemical industry, manufacturers and safety consultants [6]. In comparison to methods recommended in the standards ISO 4126-10 and ISO 23251 [7] for sizing safety valves in two-phase flow there is a lack of reliable calculation models to determine the mass flow rate to be discharged [8]. Initially subcooled or low mass flow quality inlet flow, almost isentropic gas flow and non-plenum flow at the inlet are not well represented by the recommended methods. For initially subcooled liquid flow Schmidt [9] proposed a method, which is recommended in ISO4126-10. He extended this method for flashing two-phase flow and validated the results by means of several different types of safety valves [10]. The latest improvement of the method lead to the **H**omogeneous **N**on-**E**quilibrium model with a **C**onsistent set of **S**izing **E**quations (HNE-CSE) for both pipes and restrictions with friction [4]. It is an extension of the HNE-DS model of Diener and Schmidt [11] to size not only safety valves for isothermal two-phase flow but further safety devices and pipping fittings for two-phase flow even with a high mass flow quality. The model and the recommended procedure was applied to size a safety valve on a tube bundle heat exchanger in a production plant for floor lining.

The here considered heat exchanger [12] contains heat transfer oil at 6 bar operating pressure at the tube side. The heat transfer oil is upstream filtered and preheated by an electrical heater with a power of 550 kW. The maximum allowable working pressure of the tubes is 10 bar at a temperature of about 300°C. The tubes are u-shaped and connected to the front head of the heat exchanger which has to be protected by a safety device to prevent overpressure. The hot oil is pumped in a loop into the heat exchanger and transfers the heat to high pressure air at 80 bar operating pressure in the shell side of the heat exchanger that can be pressurized to a maximum allowable working pressure of 100 bar. With a fan the air is passed through the shell around the u-shaped tubes and to the exit of the heat exchanger shell. After leaving the heat exchanger the air flows into a press to heat up glue.

2. Hazard analysis on a tube bundle heat exchanger

The hazard analysis provides the first step to determine the sizing case of a plant component. In the hazard analysis, all feasible hazards from substances inside the tube and in the shell, the range of operating conditions (system-inherent hazard potential), and other possible deviations from normal operation that may lead to impermissible process conditions (process maloperations and system failures) are identified. Various methods are available for this, for example the risk graph as a qualitative method, see IEC 61511, Appendix E, [13] or VDI/VDE 2180 [14]. Depending on the magnitude of the risk as a result of a deviation from normal operation, the Safety Integrity Level (SIL level 1 to 4) is obtained as the requirement for a safety measure. Alternatively, quantitative or semi-quantitative methods can be used, e.g. IEC 61511, Appendix A. For a proper performance of the hazard analysis only operating and monitoring devices should be considered without taking pre-installed safety devices into account. In a following step, the causes triggering these deviations are evaluated in accordance with their probability of occurrence.

2.1. Determination of the sizing case

For the determination of the sizing case, all conceivable scenarios which could generate a deviation from normal operation shall be taken into account. A widespread scenario in association with tube bundle heat exchangers is a leakage or a rupture of one ore multiple tubes. Tubes failure are more likely in case of induced vibrations or material failure. The consequences may be particularly pronounced if the substances inside the heat exchanger start an exothermic reaction when coming into contact. From the high-pressure side, the fast expansion of a gas or flashing liquid into the low-pressure side can evoke a pressure shock wave which may propagate through the equipment and affect the resistance of the heat exchanger shell. In addition, a shell leakage due to insufficient material resistance, corrosion or erosion, dynamic load changes or outside mechanical damage through a third party have to considered but are often of minor probability. Liquid blockage, external heating from the sun, a fire or a cooling malfunction could also cause an overpressure with an eventual damage due to the expansion of the liquid phase volume. The scenario overheating at maximum operation pressure shall be considered. Another typical cause for an overpressure inside of heat exchangers is an inadmissibly blocked outlet when pumps, compressors or plant grids are used in a supply system. Due to the blocked outlet, the pressure inside of the apparatus rises by means of the mass influx of a fluid. The same phenomenon occurs when return flow comes from an adjacent apparatus. Every single scenario of failure causes an undesired overpressure in the system that has to be prevented by a safety measure. For tube bundle heat exchangers, the protection of the tube and shell side are recommended if the heat exchanger is operated under overpressure conditions.

For the heat exchanger in the regarded case, a tube leakage or rupture may be taken into consideration even though the heat exchanger does not experience vibrations from fluid flow or adjacent machinery. Regarding the heat exchanger size, a rupture of multiple tubes is unlikely and can reasonably be excluded. Neither the oil nor the air are reactive components so a chemical reaction inside the heat exchanger can be excluded as well. When a leakage or a rupture appears on the oil-filled tube, high pressure air flows into the tube and displaces the liquid over a short amount of time. In the beginning, one small air bubble will enter the tube and cause a massive increase of pressure. The bubble is followed by even more air until the whole pipe is filled up. Depending on the diameter of the leak the air flow will choke and therefore restrict the mass flow rate entering the tubes. As a result, the sudden increase in pressure can cause a pressure shock wave in the liquid phase that may affect the resistance of the front head on the heat exchanger and may lead to the bursting of the apparatus. The heat exchanger is located inside a machinery hall and is not exposed to sun or other environmental thermal changes. In addition, the heat exchanger is not exposed to load cycles so the scenario shell leakage doesn't need to be considered. The machinery hall is an explosion-risk area and each apparatus inside is ex-protected certified, therefore a fire or an explosion may be excluded as hazard scenario. The scenario of blocked-in liquid with external heating can be neglected due to the exclusion of the fire case as potential hazard. An overpressure caused by a blocked outlet is not possible due to locked-open design and return flow from adjacent machinery can be also excluded due to the stand-alone position of the heat exchanger. The resulting worst-case scenario and sizing case of the heat exchanger in question is a tube leakage or rupture followed by a pressure shock wave that flows towards the front head and causes bursting. Therefore, the safety device for the low-pressure tube side has to be located on top of the front head to provide the optimum protection of the apparatus.

2.2. Evaluation of the consequences of heat exchanger failure

The size of leakages in heat exchangers depends on several parameters like the pipe material, the number of pipes, the pipe diameter, the length of the pipes, the operating conditions and the dynamic load cycles. A proper calculation of the leak size is only possible in a few isolated cases. On the basis of values from experience with industrial heat exchangers and measurements on model appliances [15], the leakage size can in most cases at least be conservatively estimated [15]. Often a leak size being twice the area of the inner pipe is suggested. This represents a tube rupture, which is assumed to occur close to the manifold base [16] [17]. A tube rupture is very rare at materials with sufficient ductility and can be reasonably excluded if the leak before break criteria (LBB) are fulfilled. Measurements, incident analysis and long-term experience under typical operating conditions at heat exchanger in the chemical and petrochemical industry have shown that a leak occurs most likely before a tube rupture (leak before break – LBB), if the following criteria are strictly fulfilled [18]:

1 Proper design of the heat exchanger, i.e. all operation loads are taken into account (pressure, temperature, dead weight, additional stresses, thermal expansion, vibrations)

2 Appropriate manufacture and adequate testing of the heat exchanger (e.g., testing of tube weld points, determination of the corrosion resistance)

3 Minimum nominal diameter DN50 for long single-tube heat exchangers (tube coils in vessels)

4 Adequate strength of tube connections, for example, minimum nominal pipe diameter DN50 for very hazardous substances – for example, toxic – and DN25 for the other hazardous substances, and/or protection against tube tear off due to external force

5 Ductile material (e.g., austenitic steel such as 1.4571)

6 No corrosion or erosion stress loads (e.g., stress concentrations near welds, concentration of components with corrosive action, degassing of oxygen, unsuitable material pairings, excessive flow velocities)

7 No periodic oscillation close to the resonance frequency

8 No regular, large dynamic loads, for example, temperature load cycles during start-up/shutdown of the heat exchanger

9 No major external mechanical stresses

10 Regularly repeated inspection program

11 Reliable and timely leak detection before serious consequences (e.g., pressure, temperature, level, and concentration monitoring) and possibility of suitable countermeasures, e.g., number 12

12 Possibility of separating the heat exchanger off from other plant components during incidents

The leak size must be estimated case by case. On the basis of incident studies, leaks with an equivalent diameter of 5 mm (20 mm^2 area) are usually assumed as large enough in tube bundle heat exchangers. Under moderate internal pressure and the above-stated preconditions, an estimated leak area of 20 mm^2 appears to be sufficiently conservative. The tube bundle heat exchanger in the regarded case fulfils all 12 points and an equivalent diameter of 5 mm seems to be an adequate consideration. To estimate the required discharge area, the mass flux of air entering the system has to be determined in the beginning. Therefore, it is necessary to know if the total air mass flux from the high-pressure side entering the low-pressure tube arrives at the safety device position or if the pressure drop inside the tubes reduces the mass flux. The worst case would occur when the total air mass flow rate through the leak arrives at the safety device, with disregard of any pressure drop through the flow path. This is the case when the leakage appears close to the distributor plate and the air mass flux entering the tube through the leakage is discharged through the tube diameter into the front head. For this, the air mass flux coming into the tube has to be compared to the dischargeable mass flux through the tube diameter into the front head to exclude potential choking at the tube outlet.

Inner leakages in heat exchangers with a high difference in pressure between the tube and shell side could lead to heavy pressure shock waves inside the apparatus. In particular, if the low-pressure side is filled with a liquid, the sudden rise of pressure provides as shock wave through the internal pipe when a large amount of gas enters the tube from the high-pressure shell side. In this case a safety valve is insufficient to protect the heat exchanger because it does not open fast enough. Bursting disks would be a better choice instead [19] [20]. To make a decision on which safety devices provides the adequate protection the valve opening time compared to the pressure shock wave distribution should be taken into account.

The air entering the tube through the leak can be calculated by means of a nozzle flow model as an isentropic discharge of high-pressure air (nozzle inlet) into the oil-filled low-pressure heat exchanger tube (nozzle outlet). The amount of air entering the tube varies over the time, so that the calculation of the required area of discharge has to be done for quasi-steady conditions with non-flashing liquids (frozen flow). The HNE-CSE model [21] is an extension of the previous HNE-DS model [22] [23] which is applied in the current standard ISO 4126-10 [24]. The HNE-CSE model represents todays state-of-the-art model for these conditions.

3. The HNE-CSE Model: Consistent set of Sizing Equations

The HNE-CSE Model is based on the fundamental equation for the calculation of the one-dimensional, steady state frictional flow through a fluid element where interfacial and wall friction, acceleration due to the cross-sectional area change and density change as well as the change in potential energy as a result of gravity are considered, c.f. Figure 1.

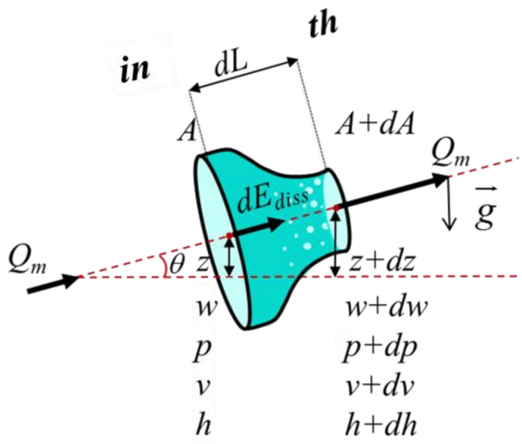

Figure 1: Flow channel with a nozzle-type contraction

An energy balance around the fluid element is performed by means of the first and second law of thermodynamics and the continuity equation. The integration of the specific volume is carried out between the inlet static location (***in***) and outlet static locations (***th***) to obtain the fundamental equation of the HNE-CSE Model [25] [26] [8]. Here the index "***th"*** stands for throat, the outlet location for the integration:

$$C = \frac{1}{v^*_{th}} \cdot \sqrt{\frac{-\int_{\eta_{in}}^{\eta_{th}} v^* \cdot d\eta - \frac{g \cdot \sin\theta \cdot L}{v_0 \cdot p_0}}{1 - \Gamma + \zeta_{v\_ref} \cdot \left(\frac{A_{th}}{A_{ref}} \cdot \frac{v^*_{ref}}{v^*_{th}} \right)^2}} \tag{1}$$

Equation **(1)** represents the general equation of the new HNE-CSE model. Industry fittings and devices like safety valves, control valves, orifices, nozzles and rupture discs can be therewith sized. In equation (1), v_0 and p_0 are the inlet specific volume and inlet pressure at stagnation conditions, ζ_{v_ref} is the resistance coefficient regarding wall and interfacial frictional effects related to a reference flow cross section (e.g. the inlet or outlet **[8]**), and ***Γ*** is the ratio of the squared of time and cross sectional averaged inlet velocity to outlet velocity:

$$\eta = \frac{p}{p_0}, \quad v^* = \frac{v}{v_0}, \quad \Gamma = \left(\frac{w_{in}}{w_{th}} \right)^2 = \left(\frac{A_{th}}{A_{in}} \cdot \frac{v_{in}}{v_{th}} \right)^2 \tag{2}$$

The flow coefficient C in equation (1) represents the calculated mass flow rate through the nozzle in dimensionless form. As a reference state, a (theoretical) maximum mass flow rate $Q_{m,max}$ is taken, which is the result of the flow of an incompressible fluid $v_{th} = v_0$; $v^*_{th} = 1$ from an infinite large reservoir ($A_{in} = \infty$) through a frictionless ideal nozzle ($\zeta_{v,ref} = 0$) with certain total conditions at the inlet p_0 and vacuum at the narrowest flow cross section $\eta_{th} = 0$.

$$C = \frac{Q_{m,nozzle}}{Q_{m,max}} = \frac{Q_{m,nozzle}}{A_{th} \cdot \sqrt{2 \cdot \frac{p_0}{v_0}}} \tag{3}$$

Flow resistance coefficients for various pipe fittings are published in the literature [27] [15]. Depending on the pipe fitting and its definition, the reference state can be related to the inlet cross section ***in*** or the outlet cross section (***th***). With the reference to the inlet cross section, the expression in the denominator in equation (1) turns to $1 - \Gamma(1 - \zeta_{v\_in})$ whereas in case the reference state to be the outlet cross section, the denominator becomes $1 - \Gamma + \zeta_{v\_th}$. The critical pressure ratio arises at the maximum mass flow coefficient. It can be derived from equation (1) as $\frac{dC}{d\eta_{crit}} = 0$. An analytical solution can be found in [8].

For the integration of the specific volume in equation (1) a homogeneous mixture of gas and liquid in a mechanical equilibrium (no slip) and a thermodynamic equilibrium (no boiling delay) is assumed. Following the derivation of the original omega method [28] and taking the boiling delay factor N of the HNE-DS Model [11] [29] [14] [10] into consideration yields an equation of state for the calculation of the specific volume of a two-phase flow including thermodynamic non-equilibrium effects, *N*.

$$v^* = \omega(N)\left(\frac{1}{\eta} - 1\right) + 1 \tag{4}$$

The omega compressibility parameter expression was defined in the form:

$$\omega(N) = \frac{1}{\kappa_0}\frac{\dot{x}_0 \cdot v_{g0}}{v_0} + \frac{cp_{l0} \cdot T_0 \cdot p_0 \cdot \eta_s}{v_0} \cdot \left[\frac{v_{g0} - v_{l0}}{\Delta h_{v0}}\right]^2 \cdot N \tag{5}$$

The boiling delay depends on inlet parameters and on the pressure reached at the narrowest flow cross section, η_{th}, which may be even critical. Hence, the coupling of equation (5) and (6) has to be iteratively solved:

$$N = \left(\dot{x}_0 + cp_{l0} \cdot T_0 \cdot p_0 \cdot \eta_0 \cdot \left(\frac{v_{g0} - v_{l0}}{\Delta h_{v0}^{\,2}}\right) \cdot ln\left(\frac{\eta_0}{\eta_{th}}\right)\right)^a \tag{6}$$

The exponent ***a*** (non-equilibrium coefficient) in equation (6)is a measure for the boiling delay and depends on the relaxation time to reach equilibrium conditions. A value of 0 represents the equilibrium conditions (HEM Model, [28] [30], spontaneous evaporation) and a value of infinity represents a flow under highly non-equilibrium conditions (frozen flow, no evaporation). The exponent is adjusted to experimental data. For safety valves, the non-equilibrium coefficient takes values of 2/5 and for orifices and control valves the coefficient a is 3/5.

Applying the omega equation of state for thermodynamic non-equilibrium (equation (4)) into equation (1), it yields:

$$C = \frac{\eta_{th}}{\omega(\mathrm{N}) \cdot (1 - \eta_{\mathrm{th}}) + \eta_{\mathrm{th}}} \cdot \sqrt{\frac{(\eta_{\mathrm{in}} - \eta_{\mathrm{th}}) \cdot (1 - \omega(\mathrm{N})) + \omega(\mathrm{N}) \cdot \ln\left(\frac{\eta_{\mathrm{in}}}{\eta_{\mathrm{th}}}\right) - \frac{g \cdot \sin\theta \cdot \mathrm{L}}{v_0 \cdot p_0}}{1 - \Gamma + \zeta_{\mathrm{v\_ref}} \cdot \left(\frac{A_{\mathrm{th}}}{A_{ref}}\right)^2 \cdot \left(\frac{\omega(\mathrm{N}) \cdot (1 - \eta_{ref}) + \eta_{ref}}{\omega(\mathrm{N}) \cdot (1 - \eta_{\mathrm{th}}) + \eta_{\mathrm{th}}} \cdot \left(\frac{\eta_{\mathrm{th}}}{\eta_{ref}}\right)\right)^2}} \tag{7}$$

with the velocity ratio in the form:

$$\Gamma = \left(\frac{A_{\mathrm{th}}}{A_{\mathrm{in}}}\right)^2 \left(\frac{\omega(\mathrm{N}) \cdot (1 - \eta_{\mathrm{in}}) + \eta_{\mathrm{in}}}{\omega(\mathrm{N}) \cdot (1 - \eta_{\mathrm{th}}) + \eta_{\mathrm{th}}} \cdot \left(\frac{\eta_{\mathrm{th}}}{\eta_{\mathrm{in}}}\right)\right)^2 \tag{8}$$

For a fluid flowing from an infinite large reservoir (($A_{in} = \infty$)) with total inlet conditions ($\eta_{in} = \eta_0 = 1$) with no friction ($\zeta_{v\_ref} = 0$) and neglectable nozzle length ($L = 0$), equation (1) reduces to:

$$C = \frac{\eta_{th}}{\omega(\mathrm{N}) \cdot (1 - \eta_{\mathrm{th}}) + \eta_{\mathrm{th}}} \cdot \sqrt{(1 - \eta_{\mathrm{th}}) \cdot (1 - \omega(\mathrm{N})) + \omega(\mathrm{N}) \cdot \ln\left(\frac{1}{\eta_{\mathrm{th}}}\right)} \tag{9}$$

Equation (9) is the basic equation to calculate the mass flow rate through an ideal nozzle.

3.1. Consistent omega compressibility parameter for non-equilibrium, ω(N)

For high mass flow qualities flows, the assumption of isothermal flow for the compressibility factor $\omega(N)$ and the non-equilibrium factor N is not valid anymore. Schmidt [29] [8] proposes a combined approach for the change of state of a two-phase flow through a nozzle. According to this model conception, the liquid and gas phases flow separately from the nozzle inlet up to an arbitrary location ***z*** in the nozzle duct. The gas expands following an isentropic behaviour up to the temperature T_{is} in the location 'z', whereas the liquid flows isothermally, T_0. At location x both phases mix spontaneously to reach the time and cross sectional averaged mixture temperature T_{mix} .

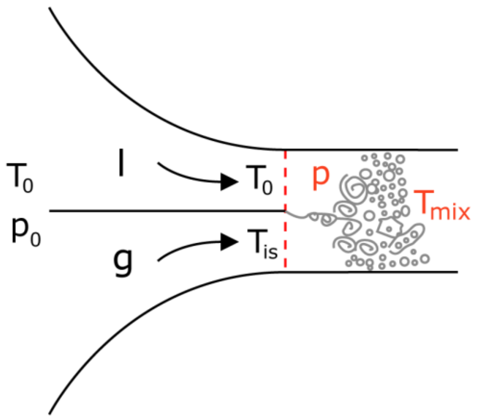

Figure 2 Phase separated two-phase frozen mixing model to calculate the local temperature of the two-phase gas liquid mixture.

By means of this new approach, liquid-only flows and gas-only flows as well as the transition into the two-phase region can be calculated without the need of implementing external models:

$$\omega(N) = \omega_{Frozen} + \omega_{Flash} \cdot N \tag{10}$$

The compressibility parameter $\omega(N)$ is divided into two terms. The component ω_{Frozen} describes the compressibility of the vapour as a consequence of the pressure change within the nozzle, whereas ω_{Flash} represents the compressibility due to the evaporation of the liquid or condensation of the vapour.

At the mixing cross sectional area, both phases are spontaneously mixed to get the mixing temperature T_{mix}:

$$T_{mix} = T_0 - k \cdot \left[T_0 - T_0 \cdot \eta^{\frac{\kappa-1}{\kappa}} \right] \tag{11}$$

Depending on the mass flow quality in a two-phase vapor/liquid flow, the two-phase mixing temperature T_{mix} may vary between the inlet temperature T_0 (isothermal liquid flow with small bubbles) and the isentropic temperature T_{is} (isentropic gas flow).

The isentropic temperature rate factor, k in equation (11) is derived from an energy balance and is defined as the ratio of reached temperature difference $(T_0 - T_{mix})$ to the maximum temperature difference for a gas-only flow $(T_0 - T_{is})$, where the change of specific heat ratios at this temperature differences are neglected:

$$k = \frac{T_0 - T_{mix}}{T_0 - T_{is}} \cong \frac{\dot{x}_0}{\dot{x}_0 + (1 - \dot{x}_0) \cdot \frac{cp_{l,0}}{cp_{g,is}}} \quad k \in (0..1) \tag{12}$$

The frozen part in the omega equation turns therefore to:

$$\omega_{Frozen} = \dot{x}_0 \cdot \frac{v_{g,is}}{v_0} \cdot \left[(1-k) + \frac{k \cdot \eta_{th}}{(1-\eta_{th})} \left(\left(\frac{1}{\eta_{th}} \right)^{\frac{1}{\kappa}} - 1 \right) \right] \text{ with } v_{g,is} = \frac{1}{\eta_{th}} \cdot \frac{T_{is}}{T_0} \cdot v_{g0} \text{ and} \tag{13}$$

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

$$\text{with } \eta_{th} = \eta_{g,crit} \text{ when } \eta_b \leq \eta_{g,crit} = \left(\frac{2}{\kappa+1} \right)^{\frac{\kappa}{\kappa-1}} \quad oder \quad \eta_{th} = \eta_b \text{ when } \eta_b > \eta_{g,crit} \tag{14}$$

The flashing part of the compressibility coefficient yields:

$$\omega_{Flash} = \frac{cp_{l0} \cdot T_0 \cdot p_0 \cdot \eta_s}{v_0} \cdot \frac{(v_{g0} - v_{l0})^2}{\Delta h_{v0}^2} \tag{15}$$

The mixture approach in the omega equation leads to consistent set of sizing equations for the whole range of two-phase flow, i.e. for a pure isothermal liquid flow until a pure isentropic gas flow, see [8]. It is recommended to substitute the current model in ISO 4126-10 by means of the HNE-CSE model, which is consistent with the sizing recommendations at the limits of liquid and gas flow in ISO 4126-7 [4].

4. Sizing a safety valve to protect the heat exchanger for non-flashing air/oil flow

4.1. Determination of the mass flow rate to be discharged by means of the HNE-CSE Model

The critical mass flow rate to be discharged through the safety valve has to be determined by calculating the maximum flow rate through a tube with a leakage. In the narrowest flow cross section of the leakage air from a pressure reservoir of 80 bar enter into a heat exchanger tube – location (1) in Figure 4 – and expands to the operating pressure of 6 bar within the tube side of the heat exchanger. Very fast the pressure in the tube side will increase to the opening pressure of the safety valve. If the cross section of the tube is not large enough, the mass-flow-rate-limiting location would establish at the entrance into the manifold – location (2) in Figure 4, instead of the narrowest flow cross section in the leakage.

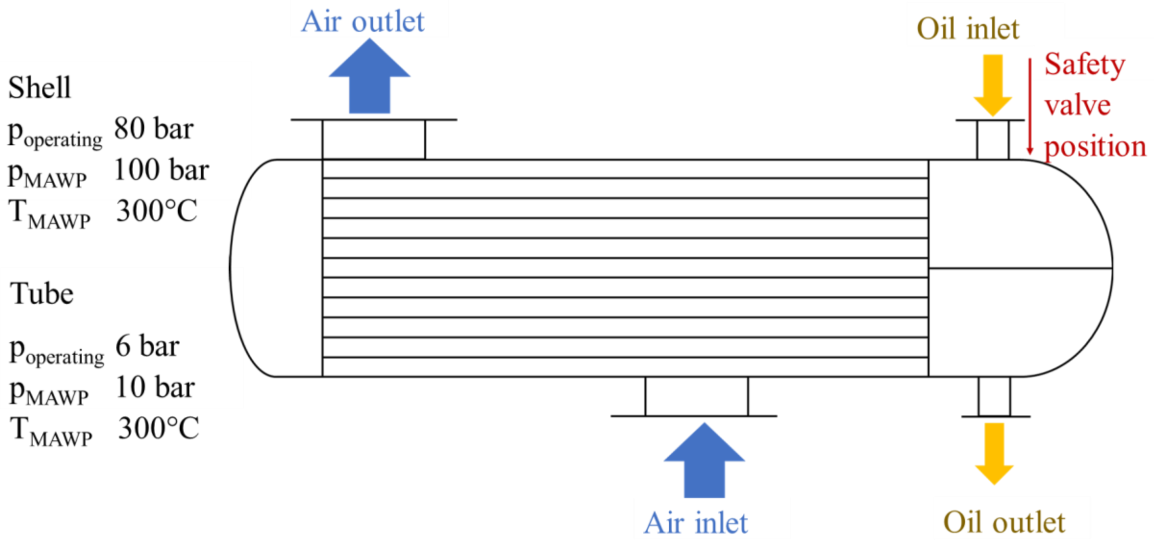

Figure 3 Heat exchanger schema

In the first step the leak is assumed to be a round bore hole with a diameter of 5 mm. For this purpose, the flow through the leak may be considered as an ideal nozzle flow and equation (9) can be applied. Air from the high-pressure shell side (***in***) will flow into the oil-filled tube-side (***th***) with the corresponding pressure conditions of the system when the safety valve is in open mode. Critical flow establishes at the leak. The resulting air critical mass flow rate must be converted into a volume flow rate via an isenthalpic flash from the shell side pressure conditions (80 bars) to the tube side opening pressure condition of the safety valve (10 bar). As a result, a mass flow rate of 965 kg/h is entering into the heat exchange tube through the leak. Expanding down to the opening pressure of the safety valve a volume flow rate of 124,23 m³/h need to be passed through the valve.

In a second step it should be proven if the critical mass flow rate of the entering air through the leak may be discharged through the tube inner diameter into the manifold of the heat exchanger. The pressure upstream of the outlet of a tube into the manifold cannot be larger than the stagnation critical pressure in the leakage. The HNE-CSE model is applied to calculate the critical mass flow rate throughout the tube area based on that upstream pressure and the opening pressure of the safety valve as the back pressure which lead to a volume flow rate at that back pressure of 322,61 m³/h. For the present case study, the critical air volume flow rate through the tube into the heat exchanger head is larger than the critical volume flow rate through the leak. The flow rate limiting location is therefore the leak location.

The air flow through the leakage expands to the opening pressure of the safety valve and displaces the same volume flow rate through the safety valve. But the mass flow quality of the flow through the safety valve changes throughout the discharge starting from pure oil flow at the first moment and running across all mass flow rates up to pure air flow at the end of the discharge. The typical sizing procedure for safety valve need to be rewritten for a constant superficial velocity j_{SV} at the entrance of the valve. The superficial velocity or volumetric flux is the volume flow rate at opening pressure of the safety valve related to the seat area of the valve:

$$j_{SV}(\dot{x}_0) = \dot{m}(\dot{x}_0) \cdot v_0(\dot{x}_0) = K_{dr,2ph} \cdot C(\dot{x}_0) \cdot \sqrt{2 \cdot \frac{p_0}{v_0(\dot{x}_0)}} \tag{16}$$

The superficial velocity j_{SV} depends on the mass flow quality at inlet stagnation condition $\dot{x}_0$. It can be calculated from the dimensionless mass flow rate $C(\dot{x}_0)$, if the two-phase derated discharge coefficient K_{dr_2ph} for the safety valve is known. Two contradicting effects lead to the minimum required seat area of the safety valve: The gas specific volume $v_0(\dot{x}_0)$ increases drastically when approaching a gas mass flow rate of 1 (pure air flow at the end), whereas simultaneously the dimensionless mass flow rate decreases. The largest required seat area is surprisingly not the area at the largest dimensionless mass flow rate at a mass flow quality of zero but establishes typically in the two-phase region at low mass flow qualities.

To size the safety valve, the minimum of the superficial velocity or volumetric flux j_{SV} must be determined:

$$\frac{d}{d\dot{x}_0} j_{SV}(\dot{x}_0) = 0 \tag{17}$$

This lead to the minimum required seat area of the safety valve, if the critical volumetric flow rate of the air in the leakage $\dot{V}_{air}$ is known:

$$A_{seat} \cdot K_{dr,2ph} = \frac{\dot{V}_{air}}{j_{SV}(\dot{x}_0)} \tag{18}$$

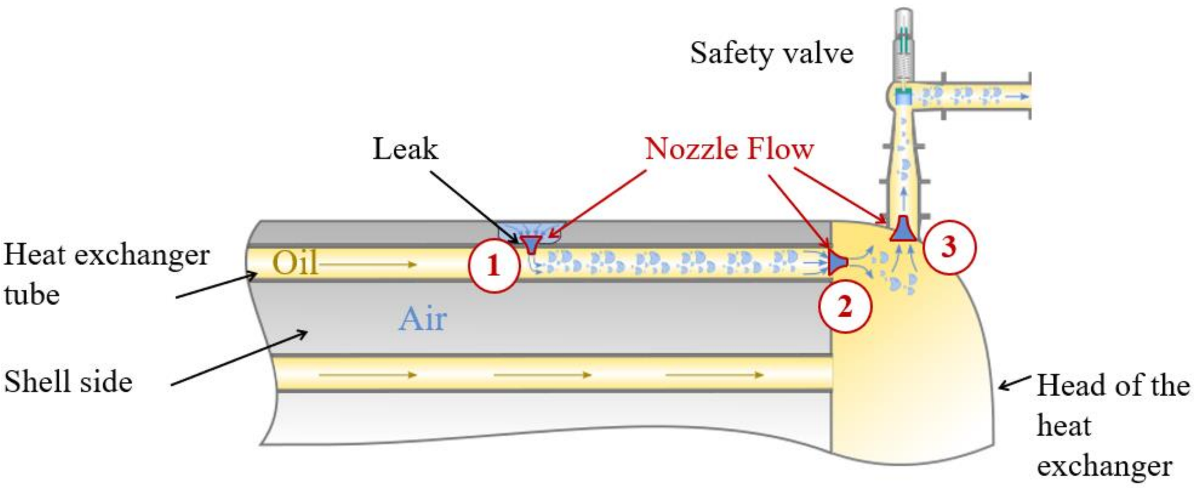

Figure 4: Simplified model of the flow through a leakage into the manifold of the heat exchanger

The two-phase valve discharge coefficient for the HNE-CSE model is calculated in accordance with the HNE-DS model and is based on the discharge coefficients for gas flow $K_{dr,g}$ and liquid flow $K_{dr,l}$, in general given by the valve manufacturer,

$$K_{dr,2ph} = K_{dr,g}\varepsilon_{th} + K_{dr,l}\left(1-\varepsilon_{th}\right) \tag{19}$$

here ε_{th} is the void fraction in the narrowest flow cross section for the fluid-dynamic critical flow.

$$\varepsilon_{th} = 1 - \frac{v_{l,0}}{v_0\left[\omega(N)\left(\frac{1}{\eta_{th}}-1\right)+1\right]} \tag{20}$$

v_0 represents the two-phase specific volume at total inlet conditions, which is calculated as the mass flow quality average between the specific volume of the gas and the liquid, $\omega(N)$ is the compressibility parameter and may be calculated by means of equation (13), η_{th} is the pressure ratio in the throat (valve seat) which most often corresponds to the critical pressure ratio and $v_{l,0}$ is the specific volume of the liquid at total inlet conditions.

4.2. Minimum volumetric flow rate through the safety valve by means of the HNE-CSE Model.

The sizing calculations for the safety valve were performed according to the model HNE-CSE (Homogeneous Non-Equilibrium Model based on a Consistent Set of Sizing Equations) for a fluid-dynamic critical flow consisting of air and heat transfer oil under the conditions of "frozen flow" (non-evaporating flow). All mass flow qualities of air and oil during the transient discharge through the safety valve are considered by means of the HNE-CSE Model to obtain the required valve seat.

At the beginning, only pure oil may be released, but shortly thereafter air may be increasingly mixing with the oil up to the end of the discharge, when only air will be relieved. With increasingly air content, the safety valve capacity is reduced. However, the specific volume of the two-phase mixture may simultaneously increase.

In Figure 5 the volumetric flux (superficial velocity) for a constant volume flow rate to be discharged at safety valve inlet and an air mass flow quality ranging between 0 and 1 is represented. The minimum volume flux through the safety valve during the two-phase discharge is hence 23,1 m/s.

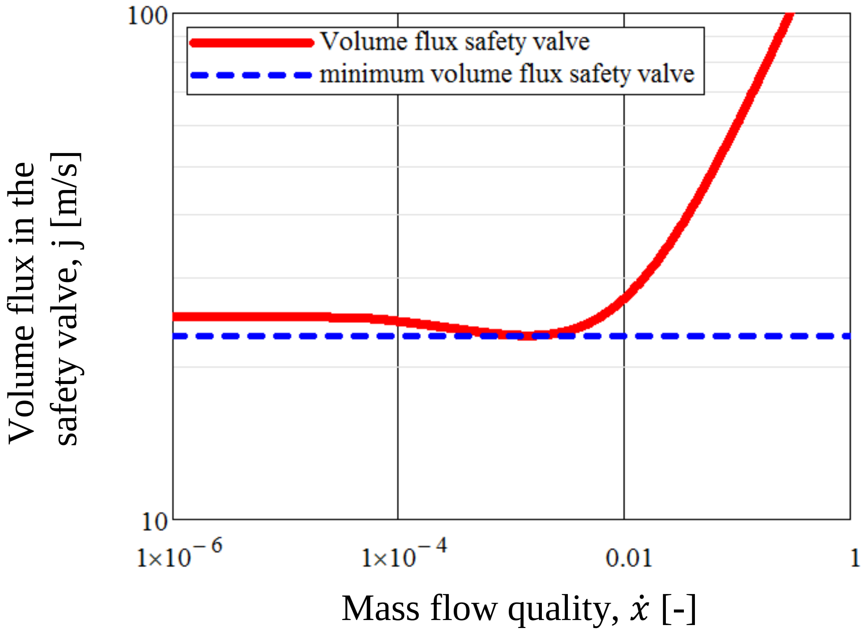

Figure 5: Dischargeable volume flux through the safety valve

From the dischargeable volume flux and the volume flow rate to be discharged, a required valve seat cross section can be calculated, A_{seat}:

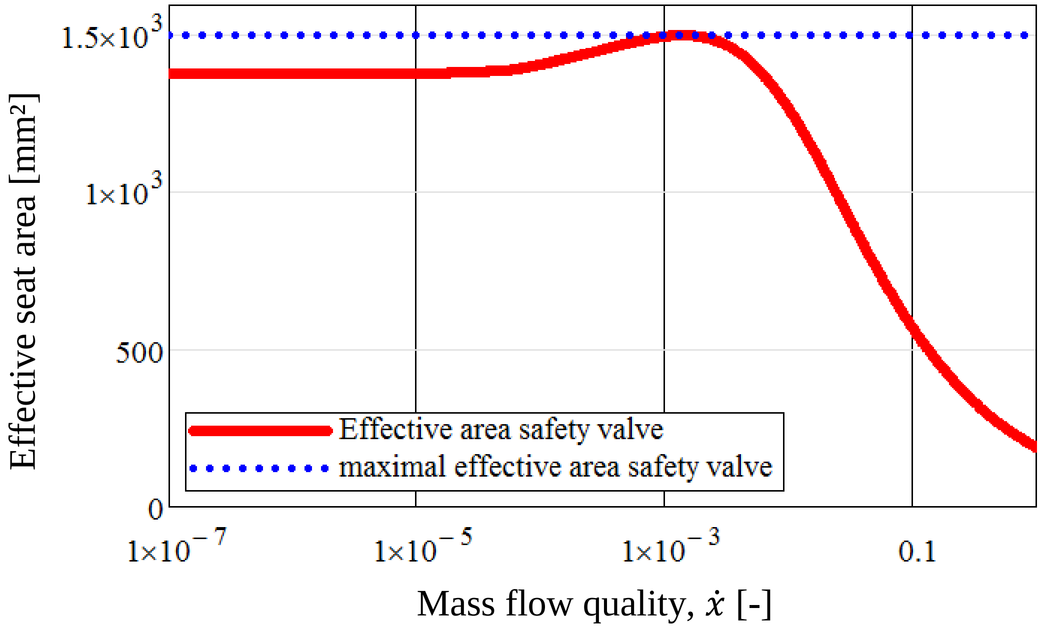

Figure 6: Required safety valve area

The minimum required effective seat area $\left(A_{seat} \cdot K_{dr,2\,ph}\right)$ is therefore 1500 mm².

Ensuring proper operation of the safety valve

In single phase flow the pressure loss in the inlet line as well as the back pressure at the safety valve outlet need to be considered to guarantee the proper operation of the valve. According to ISO 4126-9 [31] a pressure loss in the inlet line larger than 3% is not allowed for gases and liquids. The backpressure is limited to 10% or 15% depending on the valve manufacturer. For two-phase flow there are no criteria available to evaluate the stable operation of the safety valve. An application of the criteria from two-phase flow is physically not reasonable. The values for single-phase flow are a convention based on experience and a steady state consideration of the opening of a valve. In 4126-10 [3] sit is recommended to use friction damper and balanced bellows for two-phase flow.

Conclusions

The case study of the protection of a high pressurized air-filled shell and a low-pressure oil-filled tube bundle heat exchanger by means of a safety valve is resolved by following the sizing steps presented at the ISO 4126-10. After a hazards analysis study the worst-case scenario was determined to be the tube leakage followed by a pressure shock wave. The sizing calculations are performed for quasi-stationary conditions for an air-oil frozen flow with the HNE-CSE sizing model. The HNE-CSE model is the only model to consider not only an isothermal change of state typical for low mass flow quality flow but also the almost isentropic change of state of the flow at a large gas content in the two-phase mixture. It is recommended to add the HNE-CSE model as state-of-safety-technology into the revision of ISO 4126-10. The model lead to equal results as recommended for gas and liquid flow in ISO 4126-7.

In general, the assumption of a full-bore tube rupture lead to tremendous volume flow rates entering the low-pressure system and often to unreasonable large safety devices for heat exchangers. In addition, the transition from pure liquid to pure gas flow is very fast – often within seconds. For typical heat exchanger of the chemical and petrochemical industry is shall be evaluated in any case, if a leak for break assumption based on the criteria given in this paper is reasonable. It is in any case not recommended to apply this assumption in high consequence applications like liquefied gas heat exchangers.

Especially in heat exchangers where large pressure differences occur, the shock wave at the instant of the leak opening shall be evaluated. Often a rupture disk is more efficient to avoid a burst of the apparatus due to a short peak pressure than a safety valve. State of the art rupture disk manufacturer deliver highly reliable safety devices with a very fast opening characteristic. In addition to a prevention against shock waves, the required discharge area is generally much smaller compared with a safety valve. Start-up and shut-down processes must be carefully evaluated.

Beside the considered valve sizing other technical risks should be considered for a safety concept: if the flash point of the oil is below the maximum allowable working temperature, an ignition of the oil must be prevented. Liquid shall be separated in a retention system, e.g. a cyclone separator [32]. During the discharge process an explosion-risk area should be determined where ignition sources are prevented.

Nomenclature

Variable	**Definition**	**Unit**
a	Exponent of the non-equilibrium factor N	—
A_{in}	Flow cross section at inlet conditions	m²
A_{ref}	Reference flow cross section	m²
A_{seat}	Seat flow cross section	m²
A_{th}	Flow cross section at outlet conditions	m²
C	Flow coefficient	—
$cp_{g_{is}}$	Specific heat capacity of the gas after an isentropic change of state	J/(kg K)
cp_{l_0}	Specific heat capacity of the liquid at inlet total conditions	J/(kg K)
g	Gravity acceleration	m/s²
Δh_{v_0}	Specific heat of vaporization at inlet total conditions	J/kg
j_{sv}	Superficial velocity or volumetric flux	m/s
k	Isentropic temperature rate factor	—
$K_{dr,2ph}$	Rated discharge coefficient for two-phase flow	—
$K_{dr,g}$	Rated discharge coefficient for gas	—
$K_{dr,l}$	Rated discharge coefficient for liquid	—
L	Piping element length	m
$\dot{m}$	Mass flux	kg/s m²
N	Non-equilibrium factor	—
p	Pressure	Pa
p_0	Pressure at inlet total conditions	Pa
p_{th}	Pressure at the narrowest flow cross section (throat)	Pa
$Q_{m,nozzle}$	Mass flow rate through a nozzle	kg/s
$Q_{m,max}$	Calculated maximum feasible mass flow rate	kg/s
T_0	Temperature at inlet total conditions	K
T_{is}	Temperature after an isentropic change of state	K
T_{mix}	Averaged mixture temperature of the two-phase fluid	K
v	Specific volume of the fluid	m³/kg
v_0	Specific volume of the fluid at inlet total conditions	m³/kg
v_{g_0}	Specific volume of the gas at inlet total conditions	m³/kg
$v_{g_{is}}$	Specific volume of the gas after an isentropic change of state	m³/kg
v_{l_0}	Specific volume of the liquid at total inlet conditions	m³/kg
v^*	Dimensionless specific volume	—
v^*_{in}	Dimensionless specific volume at inlet conditions	—
v^*_{ref}	Dimensionless specific volume at reference flow cross section	—
v^*_{th}	Dimensionless specific volume at outlet conditions	—
$\dot{V}_{air}$	Critical volumetric flow rate of the air in the leakage	m³/s
$\dot{x}_0$	Mass flow quality at total inlet conditions	—
w	Two-phase flow velocity	m/s
w_{in}	Two-phase flow velocity at inlet conditions	m/s
w_{th}	Two-phase flow velocity at outlet conditions	m/s
ζ_{v_ref}	Flow resistance coefficient at reference conditions	—
Γ	Dimensionless velocity ratio	—
ε_{th}	Void fraction	—
η	Pressure ratio	—
η_{crit}	Ratio of the critical pressure to the total inlet pressure	—
η_{in}	Ratio of the inlet pressure to the total inlet pressure	—
η_s	Ratio of the inlet saturated pressure to the total inlet pressure	—

η_{th}	Ratio of the pressure in the narrowest nozzle flow cross section (throat) to the total inlet pressure (either the critical pressure ratio or the back-pressure ratio, whichever is larger)	—
θ	Pipeline inclination	rad
κ_0	Isentropic coefficient at inlet conditions	—
ω_{Frozen}	Compressibility factor of the non-flashing gas	—
ω_{Frlash}	Compressibility factor due to evaporation (flash) or condensation	
$\omega(N)$	Compressibility factor accounting for non-equilibrium effects	—

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