# SAFETY LIFE CYCLE ANALYSIS APPLIED TO THE ENGINEERING OF PRESSURE RELIEF VALVES IN PROCESS PLANTS

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## Confidentiality of acquired field information

To a large extent, information coming from a worldwide petrochemical company has been used within this thesis as, for example, in the statistical analysis and case studies. The author has gathered this information to implement the new methodology. Because company-related safety issues and safety policies are considered to be confidential information, the name of the involved company has been withheld

# Abstract

Chemical plants and other industrial installations process and store hazardous materials, which represent a certain risk to people, equipment and the environment. Overpressure is one of the most common upsets in process plants and relief devices (pressure relief valves, rupture discs, overpressure-vacuum valves) are required on process equipment to prevent internal pressures from rising to levels, which could cause catastrophic equipment failure. They are the ultimate line of protection against equipment rupture and are therefore extremely critical safety elements. However, this criticality is not always accounted for in existing plants and in the engineering of new ones, as a lot of pressure relief systems audits performed recently, especially in US, have demonstrated. As an example, a 2014 report from Siemens Energy, Inc. of Houston, pointed out that, after performing 1,197 pressure relief systems audits between 2005 and 2014, consisting of 174,943 pieces of equipment and 80,372 pressure relief devices, 47% of the relief devices had a deficiency, e.g. 13,4% were undersized. In another statistical study focused on the maintenance process of pressure relief valves conducted in 1995 and based on 13,000 items inspected in the workshop, it was found that 18% of the valves only opened at a pressure higher than 110% of the set pressure and 3% did not open at a pressure of twice the set pressure.

Therefore, it is clear that a methodology for increasing the reliability of pressure relief devices would be very useful. A new methodology has been developed in this thesis based on an extension of the safety life cycle concept developed for the safety instrumented systems, according to the IEC 61511. This new methodology covers all the steps of the life cycle of a pressure relief valve: a) risk analysis; b) safety requirements specification; c) design; d) reception, installation and checking; e) operation, maintenance and revision; f) management of change; g) decommissioning h) verification and i) documentation and technical audits.

This methodology has been applied to three existing petrochemical plants, obtaining results that validate it as a useful tool in the target of increasing the reliability of pressure relief valves.

# Contents


Chapter 1. Introduction.......... 1

1.1 Safety layers in the process industry .......... 1

1.2 Pressure relief valves types .......... 2

1.3 Flaws in installed safety valves .......... 3

1.4 Objectives of the thesis.......... 4

Chapter 2. Critical analysis of the need, design, installation and maintenance of the pressure relief valves 5

2.1 Historical analysis of accidents .......... 5

2.2 Statistical analysis of deficiencies in pressure relief systems. Literature survey .......... 7

Chapter 3. Development of a new methodology to increase the reliability of pressure relief valves based on the Safety Life Cycle analysis .......... 17

3.1 Phase 1. Risk analysis .......... 18

3.2 Phase 2. Safety requirements specification .......... 21

3.3 Phase 3. Design .......... 25

3.4 Phase 4. Reception, installation and checking.......... 28

3.5 Phase 5. Operation, maintenance and revision .......... 29

3.6 Phase 6. Management of change .......... 32

3.7 Phase 7. Decommissioning.......... 35

3.8 Phase 8. Verification .......... 35

3.9 Phase 9. Documentation and technical audits .......... 38

3.10 Commercial software available. Critical review .......... 40

Chapter 4. Detailed development of the design phase .......... 43

4.1 Relief load analysis (contingency analysis) .......... 43

4.1.1 Basic design philosophy.......... 43

4.1.2 Specific design basis .......... 47

4.2 Relief load calculation including vapor/liquid disengagement study.......... 50

4.2.1 External fire.......... 50

4.2.2 Liquid filled systems .......... 54

4.2.3 Distillation equipment .......... 55

4.2.4 Heat exchanger tube rupture.......... 57

4.2.5 Vapor/liquid disengagement.......... 58

4.3 Selection and design (required area) of the pressure relief valve ..... 59
4.3.1 Gases/Vapors ..... 61
4.3.2 Steam ..... 62
4.3.3 Liquids ..... 62
4.3.4 Two phase-flow ..... 63
4.3.5 Supercritical flow with retrograde condensation ..... 65
4.3.6 Liquid thermal expansion ..... 66
4.4 Inlet pressure drop ..... 66
4.4.1 Compressible gases/vapors ..... 68
4.4.2 Liquids and two-phase flow ..... 68
4.5 Outlet pressure drop ..... 68
4.5.1 Vapor phase pressure drop in flare networks ..... 69
4.5.2 Two-phase pressure drop in flare networks ..... 70
4.6 Stability calculations: engineering analysis ..... 72
4.7 Forces and moments imposed on the pressure relief valve ..... 80
4.7.1 Discharge to atmosphere ..... 80
4.7.2 Discharge to flare network ..... 81
4.8 Acoustic induced vibration ..... 81
4.9 Noise ..... 82
4.10 Body bowl choking ..... 83
4.11 Assignation of the first turnaround interval ..... 84
4.11.1 Probability of failure ..... 85
4.11.2 Consequence of failure ..... 86
Chapter 5. Application of the new methodology to three existing petrochemical plants ..... 93
5.1 Design basis of the relief system ..... 93
5.2 Scope of the work. Data source ..... 96
5.3 Actions taken and difficulties encountered in implementing the new methodology. ..... 97
5.4 Deficiencies encountered in the design phase. Statistical analysis ..... 99
5.5 Evaluation of the deficiencies and mitigation actions ..... 101
Chapter 6. Special case studies related to the design phase ..... 103
6.1 Total backpressure calculations in the flare network of existing plants ..... 103
6.2 Comparison of different methods for the design of PRVs with Z < 0.8 or Z > 1.1 ..... 120

6.3 Comparison of different methods for the design of pressure relief valves with two-phase flow... 125
6.4 Relieving of supercritical fluids with retrograde condensation ... 132
6.5 Calculation of relieving loads in distillation columns ... 148
6.6 Tube rupture in heat exchangers ... 150
6.7 Application of the engineering analysis methodology according to API 520 Part II-2015 ... 161
6.8 Optimization of the revision interval of pressure relief valves ... 177
6.9 Inspection methodology applied to the planned turnarounds ... 179
Chapter 7. Conclusions and future research ... 191
References ... 193
Annexes ... 209
Annex A: Standards and guidelines ... 209
Annex B: Pre start-up safety review checklist for the new methodology ... 213
Annex C: List of pressure relief valves studied ... 217
Annex D: Calculation of relief rate due to fluid expansion and external heat for YS 702/01 ... 229

## Acronyms

| Acronym | Meaning |
|---|---|
| AIV | Acoustic Induced Vibration |
| ALARP | As Low As Reasonably Practicable |
| API | American Petroleum Institute |
| ASME | American Society of Mechanical Engineers |
| CCPS | Center for Chemical Process Safety |
| CSB | U.S. Chemical Safety Board |
| DI | Direct Integration method |
| DIERS | Design Institute for Emergency Relief Systems |
| EEMUA | Engineering Equipment Material Users Association |
| EOS | Equation Of State |
| EPR | Emergency Response Plan |
| HDI | Homogeneous Direct Integration |
| HIPS | High-Integrity Protection System |
| HNDI | Homogeneous Nonequilibrium Direct Integration |
| HNE | Homogeneous NonEquilibrium model |
| HNE-DS | Homogeneous NonEquilibrium model-Diener/Schmidt |
| HNEM | Homogeneous NonEquilibrium model |
| HSE | Health Safety and Environmental |
| IEC | International Electrotechnical Commision |
| IP | Institute of Petroleum |
| IPL | Independent Protection Layer |
| ISO | International Organization Standardization |
| LOPA | Layers of Protection Analysis |
| MAWP | Maximum Allowable Working Pressure |
| MDMT | Minimum Design Metal Temperature |
| MHIDAS | Major Hazard Incident Data Service |
| MOC | Management of Change |
| MSDS | Material Safety Data Sheet |
| NCSR | National Centre of Systems Reliability |
| NFPA | National Fire Protection Association |
| OSHA | Occupational Safety and Health Administration |
| PED | Pressure Equipment Directive |
| PFD | Probability of Failure on Demand |
| PHA | Process Hazard Analysis |
| P&ID | Piping & Instrument Diagram |
| POPRV | Pilot-Operated Pressure Relief Valve |
| PRV | Pressure Relief Valve |
| PSI | Process Safety Information |
| PS PPM | Process Safety Pressure Protection Manager |
| PSSR | PreStart-up Safety Review |
| RV | Relief Valve |
| SAFed | Safety Assesment Federation |
| SEM | Slip Equilibrium Model |
| SIL | Safety Instrumented Level |
| SIS | Safety Instrumented System |
| SOP | Standard Operating Procedures |
| SP | Set Point |
| SRV | Safety Relief Valve |
| SV | Safety Valve |
| TPHEM | Two-Phase Homogeneous Equilibrium Model |
| UK-HSE | UK-Health and Safety Executive |
| VCE | Vapour Cloud Explosion |

# Chapter 1. Introduction

## 1.1 Safety layers in the process industry

The chemical industry handles, processes, stores and transports hazardous materials. This is why adequate safety measures are required at all stages: in the design, the operation and the logistics of installations. These measures can be associated with layers of protection: an external emergency plan, an internal emergency plan, mitigation, prevention (safety valves/bursting discs, safety instrumented systems, etc.) and control (Marszal and Scharpf, 2002; Melhem, 2006). Figure 1-1 shows the different layers of protection of a chemical process.

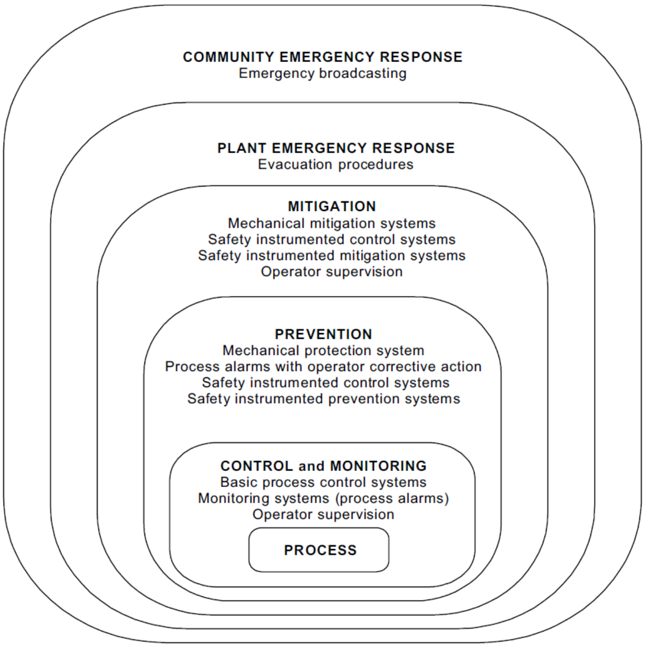



**Figure 1-1.Typical layers of protection of an industrial process (taken from IEC61511, 2003).**

Each layer of protection includes equipment, control systems and actions devised to reduce risk. The protection layers can be divided into:

- Layers of Prevention. These act before the loss of material or energy within the plant. For example, safe design (inherent safety design concept (CCPS, 2009)), basic process control, safety instrumented systems, pressure relief devices, etc.
- Layers of Mitigation. These act after the loss of material or energy from the plant. For example, passive protection, Fire & Gas systems, evacuation plan, etc.

One of the prevention layers named “mechanical protection system” in Figure 1-1 is related to pressure relief devices: pressure relief valves, bursting discs and overpressure-vacuum valves. The Layer of Protection Analysis (LOPA) emphasizes the importance of pressure relief devices to decrease the risk associated with overpressure (Dowell, 2001). However, any failure of these valves can lead to major accidents.

Pressure relieving systems, comprising of pressure relief devices, are required to be installed on process equipment to protect them in case of overpressure situations. They are intended to operate only in emergency situations, when a failure, accident or misoperation has caused an overpressure condition that the basic system design and its associated control system cannot resolve. It should relieve the excess pressure safely so as to protect personnel working in the facility from accidents, prevent damage to equipment and adjoining property, and comply with government regulations.

This thesis will study pressure relief valves, as they are the most representative item of pressure relief devices. According to Hellemans (2009) 80% of pressure relief devices are pressure relief valves. The experience of the author of this thesis shows that in some plants the figure could be 95% or higher.

## 1.2 Pressure relief valves types

According to API 520 (Part I, 2008) there are different types of pressure relief valves:

**Pressure relief valve (PRV)**: a pressure relief device designed to open and relieve excess pressure and to reclose and prevent the further flow of fluid after normal conditions have been restored.

**Safety valve (SV)**: a spring-loaded pressure relief valve actuated by the static pressure upstream of the valve and characterized by rapid opening or pop action. A safety valve is normally used with compressible fluids.

**Relief valve (RV)**: a spring-loaded pressure relief valve actuated by the static pressure upstream of the valve. The valve opens normally in proportion to the pressure increase over the opening pressure. A relief valve is used primarily with incompressible fluids.

**Safety relief valve (SRV)**: a spring-loaded pressure relief valve that may be used as either a safety or relief valve depending on the application.

**Pilot-operated pressure relief valve (POPRV)**: A pressure relief valve in which the major relieving device or main valve is combined with and controlled by a self actuated auxiliary pressure relief valve (pilot).

However, the European Pressure Equipment Directive (PED, 1997) uses the overall term “safety valve” for every pressure-relieving device subject to the PED code (Hellemans, 2009). In this thesis the terms pressure relief valve and safety valve are interchangeable.

This thesis deals with PRVs. The bursting discs and the overpressure-vacuum valves, although they share related problems, have been excluded. Figure 1-2 shows the internal configuration of pressure relief valves.

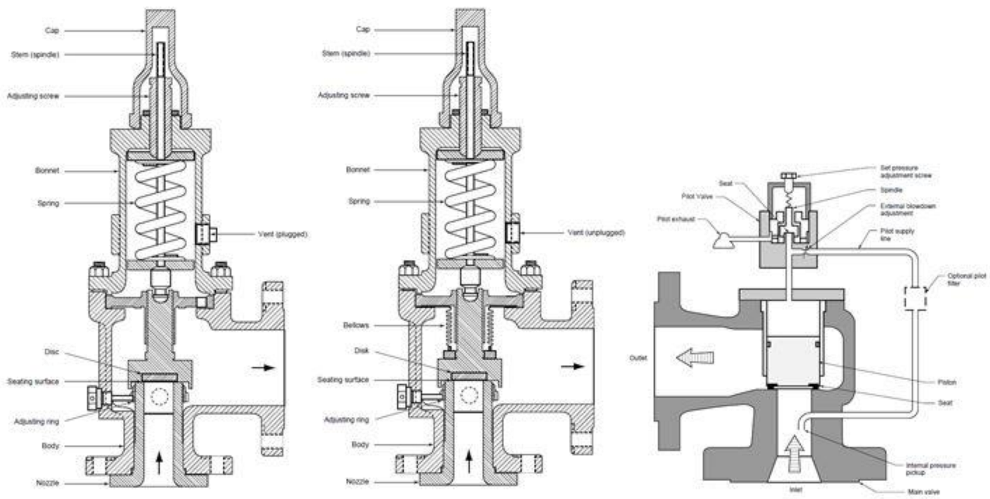



**Figure 1-2. Internal configuration of PRVs: conventional (left); balanced (center) and pilot-operated (right).**

## 1.3 Flaws in installed safety valves

Having exposed the importance of the PRVs in avoiding accidents, a historic survey on accidents related to safety valves has been performed. Historic surveys of accidents have been used in the chemical industry as a source of information about the hazards associated with a specific chemical process (Vilchez et al., 1995). The survey was performed on accidents that could be attributed to overpressure plus the simultaneous failure of a safety valve. Among the 48 cases found from 1944 to 2005 (MHIDAS, 2007), 35 were associated with “mechanical failure of a safety valve”, with considerable associated damage: 56 fatalities, 292 injured and more than 8000 evacuated. Consequently, the correct engineering of safety valves is obviously an important issue for any industrial plant.

Of course, most latent failures do not lead to accidents; some safety valves will never need to actuate under an overpressure situation in their entire life. However, such failures can be decisive in certain circumstances. The DIERS Institute (CCPS, 1998) observed that, amongst the 100 worst major accidents that occurred in the process industry between 1956 and 1986, twenty-five could be attributed, at least partly, to the inadequate design or maintenance of pressure relief systems. Several authors have studied these aspects and the results, showed here, have been divided between design and maintenance faults.

**Technical design faults**

Berwanger et al. (2000) analyzed the adequacy of pressure relief systems in 272 process plants in the US. In this important study, 14,873 devices were analyzed and the main conclusion was that: approximately 40% of process equipment has at least one error in its pressure relief system (no relief device 15%; undersized device 7%; improper installation 17%; undersized and improperly installed device 2%). Kumana and Aldeeb (2014) in a very wide study comprising 80,372 pressure relief devices, taken over 1197 audits performed between 2005 and 2014, have found that: the number of relief devices with at least one major issue is 47% and 13,4% of the equipment was unprotected. In Europe, Westphal and Köper (2003), performed a survey of 4,000 safety valves, and found faults in 17%, including undersizing, pressure drop in the inlet pipe higher than 3%, and total backpressure higher than 15% for conventional safety valves.

**Technical maintenance faults**

Aird (1982) observed that 44% of safety valves opened outside the range +/- 10% of the set pressure in the pre-test. Smith (1995) analyzed the behavior of 13,000 safety valves: 18% opened at a pressure higher than 110% of the set pressure and 3% did not open at a pressure of twice the set pressure. On the basis of an analysis of 750 complaints concerning faulty operation of safety valves, Hellemans (2009) found that 10% were due to under- or over-sizing, 8% to bad maintenance, 33% to incorrect installation, 29% to incorrect transportation or handling, 12% to a manufacturing default and 7% to various other reasons. In a pre-test inspection of 292 valves, Chien et al. (2009) found that 4% opened at a pressure higher than 119% of the set pressure.

These figures show that there is an impressive quantity of latent deficiencies in the engineering of pressure relief valves. As pointed out before, the majority of these deficiencies will never be discovered because the valve will never need to open.

Having this wide problem identified, the main research question of the thesis emerged: how can the engineering of pressure relief valves be improved? The answer came from the framework developed for the safety instrumented systems according to IEC 61511 (2003) and, specifically with the functional safety life cycle concept.

Therefore, the motivation for this thesis is to improve the reliability of the pressure relief valves, through the application of the safety life cycle approach which, although ideal for the proposed objective, has not been used in pressure relief systems before.

## 1.4 Objectives of the thesis

General objective: to develop a new methodology that allows the reliability of pressure relief valves to be increased; this will be done by applying the safety life cycle analysis concept of IEC 61511 to the engineering process of protecting pressure vessels from damage due to an overpressure through these valves.

Specific objectives:

1. To develop and apply a new methodology based on IEC 61511 to all phases of the engineering process.
2. To analyse and recommend the best methods for calculating the required relief load and required area for each contingency.
3. To develop and apply a new procedure for the “engineering analysis” concept of API 520 Part II, to check for instability of pressure relief valves.
4. To improve the methodology for assigning the revision intervals of pressure relief valves through a quantitative approach as that of API 581.
5. To develop and apply a new procedure for the revision of pressure relief valves in a turnaround.

# Chapter 2. Critical analysis of the need, design, installation and maintenance of the pressure relief valves

## 2.1 Historical analysis of accidents

As pointed out in the introduction, a survey of accidents involving pressure relief valves has been performed with the database MHIDAS. Although there are other databases available (FACTS, MARS, ARIA, etc.), MHIDAS was selected because it offers more information (Vilchez et al., 1995). This point is very important because the interest in each accident relies on the sequence: upset, overpressure and consequences of the accident because there was no relief valve or the relief valve had not worked properly (undersized, mechanical failure, etc.). The version of MHIDAS used, has data on accidents since the beginning of the last century until 2007. The survey was based on three key search words:

“reliefvalv”: 54 records were found

“overpressure”: 248 records were found

“relief”: 142 records were found

From the information generated in each entry, the code used for the cause of the accident was the basis for considering it or not. The information is in the codes GC (general causes) and SC (specific causes). Inside SC there is the mechanical failure with the case “reliefvalv”.

After carefully examining key words “relief” and “overpressure”, it was clear that the information gained could not be used to quantify the concept “accidents attributable to a relief valve deficiency” because the description in a lot of cases of overpressure did not clarify whether the cause was no relief valve present or another design operation or mechanical problem.

The only information that could give some light on whether the accidents were due a pressure relief valve was under the key word “reliefvalv”. From the 54 records, 2 were due to a disk rupture and 4 were counted twice (one for each substance involved). The final number of usuable records was therefore 48. From this quantity the following results were obtained (see Table 2.1).

**Table 2-1. Summary of the incidents recorded in MHIDAS database related to safety valves.**

| Record No. | Date of incident | Country | People affected | Specific cause |
|---|---|---|---|---|
| 13404 | 11.01.05 | US | 0 | mechanical failure |
| 12124 | 25.03.03 | US | 1 injured | undersized |
| 11470 | 25.04.02 | Sudafrica | 0 | mechanical failure |
| 11090 | 27.07.01 | US | 0 | mechanical failure |
| 11073 | 19.07.01 | US | 1000 evacuated | mechanical failure |
| 11021 | 28.06.01 | US | 10 evacuated | mechanical failure (external) |
| 11002 | 18.06.01 | US | 0 | mechanical failure (external) |
| 9903 | 12.08.99 | Great Britain | 0 | mechanical failure |
| 9351 | 29.08.98 | US | 127 injured | incorrect design |
| 9076 | 06.04.98 | US | 0 | mechanical failure |
| 8976 | 13.02.98 | Australia | 0 | mechanical failure |
| 8426 | 18.04.97 | Great Britain | 0 | mechanical failure |
| 8013 | 08.05.96 | Great Britain | 0 | mechanical failure |
| 6960 | 07.07.94 | US | 14 injured | mechanical Failure |
| 4592 | 09.01.90 | Great Britain | 0 | mechanical failure |
| 4482 | 31.03.62 | Great Britain | 0 | undersized |
| 4453 | 18.06.90 | US | 0 | mechanical failure |

| Record No. | Date of incident | Country | People affected | Specific cause |
|---|---|---|---|---|
| 4384 | 31.01.90 | Venezuela | 0 | undersized |
| 4224 | 18.05.90 | US | 34 injured, 100 evacuated | |
| 4113 | 02.07.90 | Great Britain | 0 | mechanical failure |
| 3519 | 22.07.88 | US | 1 injured, 500 evacuated | mechanical failure |
| 3495 | 17.02.82 | US | > 200 evacuated | mechanical failure |
| 3484 | 25.06.84 | US | > 200 evacuated | mechanical failure |
| 3036 | 1987 | Great Britain | 0 | mechanical failure |
| 3027 | --.07.87 | Great Britain | 0 | mechanical failure |
| 2994 | 18.08.75 | Great Britain | 0 | undersized |
| 2895 | 17.04.68 | Great Britain | 0 | mechanical failure |
| 2872 | 20.07.49 | US | 0 | mechanical failure |
| 2469 | 15.09.77 | US | >2 killed | mechanical failure |
| 2442 | 05.12.83 | US | 0 | mechanical failure |
| 2239 | 02.09.83 | Canada | 0 | mechanical failure |
| 1981 | 07.08.80 | US | 5000 evacuated | mechanical failure |
| 1980 | --.01.70 | Canada | 0 | frozen relief valve |
| 1964 | 06.04.53 | US | 0 | external cause |
| 1647 | 16.06.78 | US | 7 injured | mechanical failure |
| 1395 | 11.08.82 | US | 2 injured | mechanical failure |
| 982 | 07.02.68 | US | 9 killed, 7 injured | mechanical failure |
| 903 | 10.11.67 | Great Britain | 0 | frozen relief valve |
| 843 | 31.03.44 | US | 5 killed, 21 injured | Incorrect design |
| 648 | 23.03.67 | France | 0 | mechanical failure |
| 560 | 04.08.78 | Italy | >1000 evacuated | mechanical failure |
| 358 | 30.03.72 | Brazil | 39 killed, 51 injured | mechanical failure + Bleve |
| 277 | 30.05.71 | Great Britain | 0 | incorrect set pressure |
| 211 | 19.01.71 | US | 21 injured | frozen relief valve |
| 182 | 05.06.70 | Canada | 0 | mechanical failure |
| 119 | 04.08.62 | Saudi Arabia | 1 killed, 6 injured | mechanical failure |
| 70 | 27.01.75 | Great Britain | 0 | external cause |
| 37 | 16.05.80 | Ireland | 0 | mechanical failure |

The author understood that under “mechanical failure” the valve failed to open on demand or opened without demand. Thus the leakages are included here. Moreover, included under this concept are valves that relieve at set pressure but release a flammable or toxic product to the atmosphere, even though this is not a relief valve failure. From Table 2-1, the following statistical parameters could be derived:

- Mechanical failure of the safety valve accounts for 73% of the cases
- Relief valve failure due to a release of a flammable or toxic material to atmosphere was 21 %
- Tanker had a transport accident and the valve was damaged as a consequence, giving release of product to atmosphere 4%
- Valve was undersized because of another scenarios (polymerization, runaway reaction) 4%
- The majority of the accidents occurred during tanker transport (road, rail, barge, etc.). Thus as already pointed out in the early study of Vilchez et al. (1995) the road transport of hazardous materials is the main cause of accidents.

Another literature survey of accidents attributable to safety valves was performed looking for a specific survey in the web of the Chemical Safety Board of US among others; the following accidents were found:

- A catastrophic vessel explosion occurred on March 4, 1998 in Pitkin, Louisiana, US, in the installations of Sonat Exploration Co., which killed four workers and resulted in more than $200,000 in damage. The vessel lacked a pressure relief system and ruptured due to overpressurization during start-up, releasing flammable material which ignited (CSB, 2000).

- On the morning of April 11, 2003, one worker was killed at the D.D. Williamson food additive plant in Louisville, Kentucky, US, when a process vessel became overpressurized and failed catastrophically. The failure caused a release of aqueous ammonia as well as causing extensive damage to the plant. The root cause was that the company staff did not consider the feed tanks to be pressure vessels, consequently they had no relief device for overpressure protection (CSB, 2004).

- On June 11, 2008, one worker was killed and approximately seven others were injured during a maintenance operation on a heat exchanger. Ammonia overpressured inside the exchanger, causing it to rupture. Goodyear operators closed an isolation valve between the heat exchanger shell (ammonia cooling side) and a relief valve in order to replace a burst rupture disc under the relief valve that provided overpressure protection. Maintenance workers replaced the rupture disk that day but the closed isolation valve was not reopened (CSB, 2011).

- Politz (1985) reported a case of chattering of safety valves, which produced severe vibration of the piping causing a failure in the inlet flange of the valve, spraying hot crude oil on nearby equipment. The root cause was the oversizing of the relieving capacity, i.e. two valves were installed in parallel with a very similar set pressure (470 and 475 psig) ignoring the fact that for the required flow in this blocked outlet scenario, only one valve would be necessary.

The author wants to remark here, as has been already pointed out by Smith and Burgess (2012), that OSHA (Department of Labor, 1992) had difficulties in finding examples of accidents directly attributable to the safety valves. Moreover, as it has been seen before, the information available in MHIDAS is not enough to find out if the incident was caused by a specific problem with the safety valve. More research is necessary especially in the refining/petrochemical/chemical industry private datafiles.

## 2.2 Statistical analysis of deficiencies in pressure relief systems. Literature survey

A few authors have developed surveys of deficiencies and failures in pressure relief systems, performing the corresponding statistical analysis of the collected data to obtain useful conclusions. The results of the most representative ones are commented on this section.

### Berwanger et al. report

One of the most complete statistical analysis of deficiencies in pressure relief systems was conducted by Berwanger and coworkers (Berwanger et al., 2000). Its own company, Berwanger Inc., performed a large number of government mandated per OSHA 1910.119 pressure relief systems design audits. Specifically in this analyses, 272 processing units were evaluated, which corresponded to 31,509 pieces of equipment (the equipment was categorized as: vessels, heat exchangers, air coolers, compressors, filters, pumps and others). As the centrifugal pumps normally do not require overpressure protection, they were excluded from the original 31,509 items, leaving a total of 24,303 pieces of equipment. All this equipment was protected by 14,873 different pressure relief devices (pressure relief valves, rupture discs and pressure-vacuum valves).

Approximately 10,000 deficiencies were identified and categorized in three series:

100 Series: no relief device present on equipment with one or more potential overpressure scenarios. The results are presented in Table 2.2.

**Table 2-2. Summary of statistical results for each 100 Series category.**

| 100 Series | Equipment with deficiency | Equipment in sample pool | Deficiency frequency (%) |
|---|---|---|---|
| Category 101. Blocked outlet | 440 | 24303 | 1.8 |
| Category 102. Inlet control valve failure | 36 | 22840 | 0.2 |
| Category 103. External fire | 1315 | 22840 | 5.8 |

| 100 Series | Equipment with deficiency | Equipment in sample pool | Deficiency frequency (%) |
|---|---|---|---|
| Category 104.Heat exchanger tube rupture | 226 | 7298 | 3.1 |
| Category 105. Thermal expansion | 627 | 23640 | 2.7 |
| Category 106. Multiple overpressure scenarios | 868 | 24303 | 3.6 |
| Category 107. Other | 151 | 24303 | 0.6 |
| TOTAL | 3663 | 24303 | 15.1 |

The percentages of the different categories in 100 Series based on the equipment with deficiencies is presented in Figure 2-1.

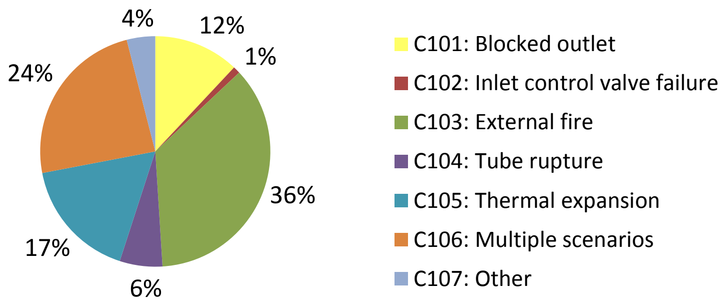



**Figure 2-1. Population of different categories of concerns in Group 100 equipment-based.**

200 Series: undersized relief device present on equipment with one or more potential overpressure scenarios. The results are presented in Table 2-3.

**Table 2-3. Summary of statistical results for each 200 Series category.**

| 200 Series | Equipment with deficiency | Equipment in sample pool | Deficiency frequency (%) |
|---|---|---|---|
| Category 201. Blocked outlet | 280 | 24303 | 1.2 |
| Category 202. Inlet control valve failure | 282 | 22840 | 1.2 |
| Category 203. Heat exchanger tube rupture | 334 | 7298 | 4.6 |
| Category 204. Loss of condensing or reflux failure | 34 | 9741 | 0.3 |
| Category 205. External fire | 854 | 22840 | 3.7 |
| Category 206. Multiple overpressure scenarios | 252 | 24303 | 1.0 |
| Category 207. Other | 58 | 24303 | 0.2 |
| TOTAL | 2094 | 24303 | 8.6 |

The percentages of different categories in 200 Series based on the equipment with deficiencies is presented in Figure 2-2.

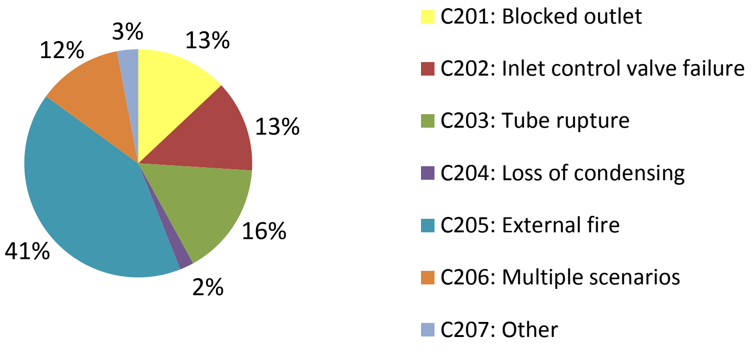



**Figure 2-2. Population of different categories of concerns in Group 200 equipment-based.**

300 Series: Improperly installed relief device. The results are presented in Table 2-4.

**Table 2-4. Summary of statistical results for each 300 Series category.**

| 300 Series | Relief devices with deficiency | Relief devices in sample pool | Deficiency frequency (%) |
|---|---|---|---|
| Category 301. Set pressure too high | 292 | 14873 | 2.0 |
| Category 302. Potential to block relief pathway | 277 | 14873 | 1.9 |
| Category 303. Inlet pressure drop too high | 1072 | 13049 | 8.2 |
| Category 304.Outlet pressure drop too high | 1606 | 13049 | 12.3 |
| Category 305. Other installation deficiencies | 30 | 14873 | 0.2 |
| TOTAL | 3277 | 14873 | 22.0 |

The percentages of different categories in 300 Series based on the relief devices with deficiencies is presented in Figure 2-3.

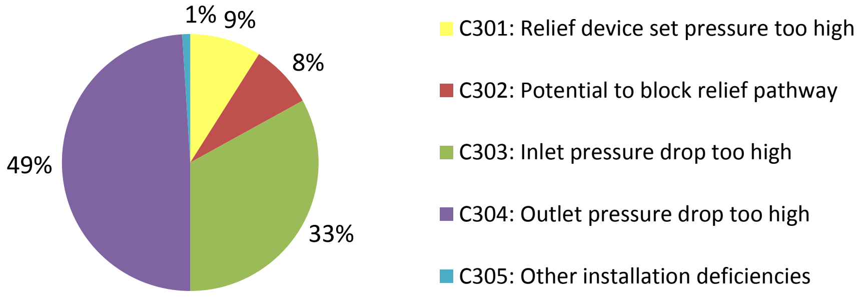



**Figure 2-3. Population of different categories of concern in Group 300 equipment-based.**

Berwanger et al. draw the following conclusions from their statistical study:

Conclusion No.1: approximately 40% of equipment surveyed had at least one pressure relief system deficiency.

The deficiencies encountered as a function of the number of equipment pieces examined can be seen in Figure 2-4.

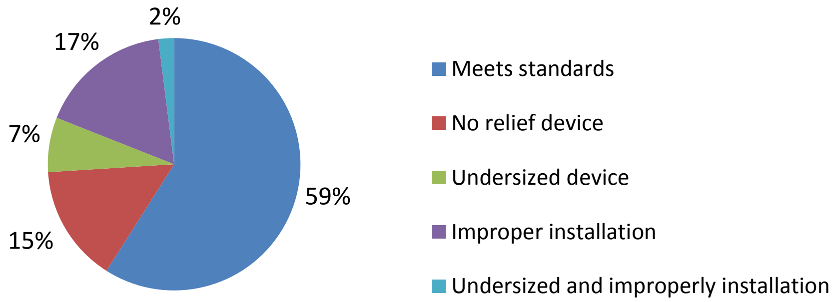



**Figure 2-4. Equipment overpressure protection status.**

Conclusion No. 2: current Process Hazard Analysis (PHA) methodologies do not capture most deficiencies. Berwanger et al. pointed out that most deficiencies were not identified because of time constraints and a general lack of pressure relief system expertise in PHA teams.

Conclusion No. 3: contractor design methods can be improved by adopting an equipment-based approach.

Although most of the 272 processing units had been designed by reputable design contractors, Berwanger and coworkers attributed the high number of deficiencies to the fact that they had not used the equipment-based approach.

Conclusion No. 4: current information management techniques have not worked.

According to these authors "In an effort to comply with the Process Safety Management regulation, many operators have recreated their pressure relief system design and design basis information essentially from scratch". Moreover, the only information available was the specification sheet and this implies only 25% of the information generated in the engineering phase (Kreder and Berwanger, 1995)

**Kumana and Aldeeb report**

Kumana and Aldeeb (2014), from Siemens Energy, Inc., Houston, performed a statistical analysis of the findings obtained in the pressure relief system design audits. The results of 1,197 audits performed in US between 2005 and 2014 were checked. The analysis included 174,943 pieces of equipment and 80,372 pressure relief devices. For this impressive data set, a total of 150,106 findings were reported, of which 77,679 belonged to the primary query criteria.

This primary query criteria consisted of:

- No pressure relief device present for equipment with an applicable overpressure scenario
- Relief device present on equipment but not adequately sized for at least one applicable overpressure scenario
- The calculated inlet non-recoverable pressure losses for at least one applicable overpressure is greater than 3%
- Outlet pressure losses calculated to be greater than the allowable overpressure for conventional valves, greater than 30 % for balanced valves, or greater than 50% for pilot operated valves. It should be noted that the outlet pressure loss findings are identified for individual discharges, not for accumulative discharges due to audit methodology (one audit is done for individual relief systems and the other one for the common discharge header)
- The pressure relief device has a set pressure above the MAWP of associated equipment
- Installation issues, including block valves administrative control violations, reduced bore valves in the inlet, outlet pocketing concerns, etc.

Although these authors worked in classifying the findings according to the refinery units (Alkylation, Crude/Vacuum, etc.) and company size, only the data for the completed audit year will be presented here in Table 2-5, so that they can be compared with Berwanger et al. results. The last column describes the ratio of the five findings presented in the left columns divided by the total number of findings. However, the methodology followed by the authors implied that if both inlet and outlet excessive losses were found for the same pressure relieve device, it was counted as two findings.

**Table 2-5. Pressure relief device related findings by audit completion year.**

| Year | Undersized % | Inlet pressure drop % | Outlet pressure drop % | Set pressure % | Installation % | Ratio % |
|---|---|---|---|---|---|---|
| 2005 | 13.1 | 13.8 | 1.3 | 16.9 | 49.4 | 61.9 |
| 2006 | 9.4 | 14.4 | 1.3 | 14.7 | 34.1 | 51.7 |
| 2007 | 8.8 | 15.8 | 1.9 | 11.4 | 25.6 | 46.9 |
| 2008 | 5.6 | 12.2 | 0.6 | 12.1 | 28.1 | 43.5 |
| 2009 | 10.3 | 16.2 | 0.8 | 11.1 | 23.6 | 46.1 |
| 2010 | 14.2 | 17.8 | 3.5 | 12.3 | 30.1 | 51.5 |
| 2011 | 16.3 | 16.4 | 6.9 | 10.3 | 27.1 | 51.5 |
| 2012 | 13.5 | 13.1 | 4.0 | 8.7 | 16.6 | 38.9 |
| 2013 | 13.8 | 14.1 | 2.8 | 9.4 | 21.3 | 43.6 |
| 2014 | 16.6 | 15.6 | 2.4 | 8.6 | 24.5 | 47.7 |
| Overall Result | 13.4 | 15.6 | 3.1 | 10.5 | 24.9 | 47.0 |

Conclusions

- The percentage of unprotected equipment is 13.4%. This figure agrees with the value of 15.1% of Berwanger et al.
- The number of undersized pressure relief devices is 13.4%
- The number of pressure relief devices with at least one major issue is 47.0%. Comparison with Berwanger et al. data is not possible, because those are expressed in terms of equipment and do not necessary map a 1:1 relationship
- The low numbers of outlet pressure drop issues are not representative, as the valves discharging to a flare network are not included
- The overall picture of the state of the industry has not changed significantly: 13.4 % unprotected equipment in the report vs 15% in the Berwanger et al. study.

Not all possible deficiencies were included in the Berwanger et al. statistical analysis, for example, issues not included were: excessive flare radiation levels, inadequate knockout drums, poorly designed quench systems, discharge of toxic or flammable fluids to the atmosphere and general process safety information upon which to base a safe pressure relief design. Therefore, one can think that the total deficiencies reported might even be underestimated.

### Westphal and Köper report

Westphal and Köper (2003), from the Process Safety Department of Siemens Axiva, performed a multi-year analysis of pressure relief devices within many process plants. This systematic analysis covered about 4,000 already existing safety valves and rupture discs.

The results of their investigation are summarized in Table 2-6.

**Table 2-6. Results of the investigation of about 4000 safety valves.**

| Pressure rise cause | % of total number | % of deficiencies per design case | Diagnosed deficiencies | | |
|---|---|---|---|---|---|
| | | | Safety valve relief diameter is not sufficient | Pressure loss in the inlet pipe > 3% | Back pressure in the vent line > 15% |
| Chemical reaction | 3% | 6% | 33% | 50% | 17% |
| External heating | 16% | 36% | 41% | 31% | 38% |
| Gas feed | 18% | 46% | 82% | 16% | 18% |
| Liquid feed | 19% | 22% | 70% | 18% | 17% |
| Thermal expansion | 44% | 2% | 81% | - | 23% |
| TOTAL | 100% | 17% | 60% | 18% | 22% |

The authors pointed out the following aspects:

a) Deficiencies were found in 17% of the safety valves. The approach was safety valves focused, and not pressure equipment like Berwanger et al.
b) Only 6% of deficiencies were found in reactors. This was due to the fact that systems with chemical reactions had been carefully analyzed and already provided with the necessary safety measures.
c) The very small number of deficiencies in the thermal expansion case was due to the fact that the use of a standard DN 25x25 proportional safety valve is generally over-dimensioned, and the pressure losses in the inlet pipes and tail pipes are negligible.
d) The most frequent deficiencies were found in the case of the gas feed, which corresponded to the scenario “control valve fails open”; in 46% of cases the safety valves were undersized.

Conclusions

According to these authors, the number of deficient safety valves (17%) was not surprising because some of these deficiencies were caused by changes in the calculation procedures, some were due to changes in process conditions without considering the consequence on the relief system, and some were due to changes in piping upstream and downstream the valve.

**Short report**

Short (2003, 2004, 2006) conducted a statistical study for his PhD thesis, based on his work at Pressure Systems Engineering, Inc. (Newark, Delaware), related to the survey of relief devices in 7 chemical process plants over 10 years (1993-2003). The plants had about 1000 pressure relief devices (pressure relief valves, rupture discs and a combination of them both). A sample of 120 relief devices (67 safety valves and 53 rupture discs) were taken as the most representative ones for the 7 scenarios studied for each item (external fire, exchanger tube failure, control valve or actuated valve failure, cooling failure, blocked outlet, hydraulic expansion and process upset -runaway reaction-). Among these 120 relief devices, 35% were new relief device installations and 65% were existing ones. However, from a total of 840 potential relief calculations only 213 were performed, because some contingencies from the gathered information clearly were not the governing case for PRV sizing or were not credible.

The Short results were as follows:

- 17.5% of the 120 relief devices (both existing and new ones) were undersized for the governing relief conditions
- The undersized range for existing relief devices was an astounding 26.9%
- For existing relief devices, the range of undersized ones ranged between 12.3% and 26.9% Taking into account these percentages, Short deduced that in the US there were between 558,000 and 1,220,000 undersized relief devices, when considering all the chemical plants and the combination of chemical/petroleum and related manufacturing plants

- Short also investigated whether fire was the most governing contingency in relation to the other six scenarios studied; he found that the ratio was about 50% for both.

**Smith report**

Smith (1995) published the results of different surveys undertaken in order to check the actual performance of installed safety valves with respect to their cold set pressures in the pre-pop tests. The original survey consisted of a sample of 5,073 valves from a variety of industries in the United Kingdom. The question that Smith tried to answer was “Would the valve have lifted at the correct pressure?” The results of this survey were as following:

- 6.3% of the total valves (321 units) failed to lift (when valves lifted at 1.9 times the cold set pressure or above this value, they were adjudged “failed to lift”)
- 14% of the valves lifted at a $\frac{\text{Prepop pressure}}{\text{Cold set pressure}} > 1.10$
- The general approach for several companies according to this author was “fit and forget”.

In 1994 Smith made a second survey, adding the results from plants in France, Germany, Holland and Belgium and those from one worldwide survey by a multinational petrochemical company. A total of 12,790 safety valves were analyzed. Figure 2-5 presents the results for the valves which $1.1 \leq \frac{\text{Prepop pressure}}{\text{Cold set pressure}} \leq 2.0$

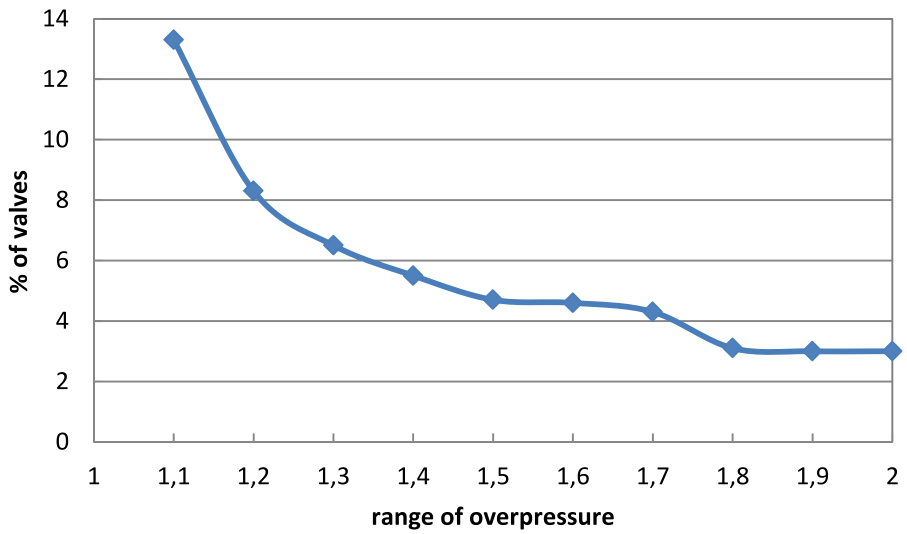



**Figure 2-5. Percentage of valves (sample of 12,790) opening above 10% of the set pressure.**

The Smith final conclusions were:

- 13% safety valves did not lift at 10% above set pressure
- 5% safety valves did not lift at 50% above set pressure
- 3% safety valves did not lift at twice set pressure.

**Aird report**

Aird (1982) applied the reliability concept to pressure relief valves. This author accepted the ICI company criteria that the failure of a safety valve is lifting at less or more pressure than 10% deviation of the cold set pressure. The database used consisted of the records provided by a large chemical company of 1,041 inspections performed during 2 years. As in many cases, the valves were so dirty that they had to be stripped down and cleaned before they could be tested (pre-pop), the number of available tests was 866.

Moreover, 120 tests had no value recorded for the period covered. Accordingly, the final number of tests analyzed was 746.

The results of Aird are summarized in Table 2-7.

**Table 2-7. Proportion of failures after different periods in service.**

| Period in service (weeks) | Mean (weeks) | Number of valves | Number of failures | Amount % |
|---|---|---|---|---|
| 1-39 | 17.1 | 104 | 46 | 44.2 |
| 40-57 | 48.6 | 104 | 36 | 34.6 |
| 58-90 | 70.6 | 107 | 52 | 48.6 |
| 91-112 | 103.0 | 103 | 44 | 42.7 |
| 113-147 | 130.3 | 103 | 43 | 41.7 |
| 148-182 | 160.9 | 106 | 51 | 48.1 |
| 183-364 | 261.0 | 103 | 55 | 53.4 |
| All | 119.1 | 746 | 332 | 44.5 |

Aird wrote "Thus, it is not possible to reject the hypothesis that all the results come from valves having the same probability of failure regardless of the period in service". This result is surprising because it implies that 44.5% of the valves in service lift outside the ± 10% of the set pressure.

The conclusions of this author were:

- A mean of 44.5% of all valves lifted outside the ± 10% range, independently of the period in service from 1 week to 1 year
- Besides dirt on the seats or product clogging the entrance of the safety valve, other less extreme mechanisms were working: spring relaxation, vibration and low temperatures
- If the spring was loaded and exposed to moderate temperatures for a few days, the stiffness of the spring changed, causing a decrease in set pressure of 5 to 10%
- A test with one valve under vibration caused the set point to fall to approximately 94% of its original value
- A test with a temperature soaked valve (80ºC) produced an almost linear increase in initial lift pressure with time, reaching 175% after 800 hours. However, the subsequent lifts (at least 10 readings were taken) were within a few per cent of the original cold set pressure.

**Hellemans report**

Hellemans (2009) gave a statistical analysis from the manufacturer's point of view. Based on a period of one year, he analyzed 750 complains coming from Europe, Middle East, Africa and Asia only. The results can be seen in Figure 2-6.

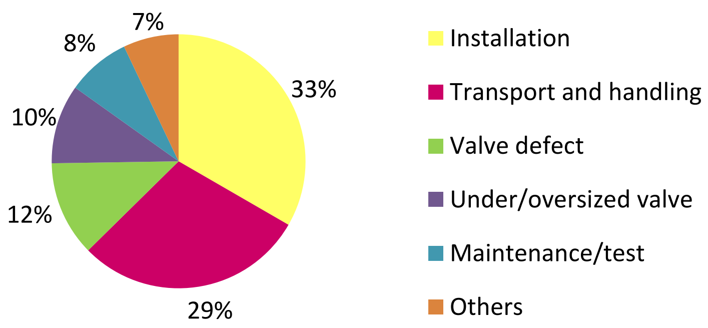



**Figure 2-6. Distribution of complaint causes.**

According to this author, 75% of the incoming complaints about the so-called "non-functioning SRV,s" were because of careless handling during and after transportation, installation, testing or maintenance.

Hellemans gives support to the Berwanger work, and writes that the results can be transposed to other parts of the world, perhaps taking a 5% on the safe side for Western Europe (that means that the 41% of equipment, which does not meet standards in US, will be 45 % in Western Europe because old codes still apply above PED 97/23/EC (Pressure Equipment Directive)).

### Riha and Streblow report

Riha and Streblow (2015) presented the results of a statistical survey of significant issues encountered in the analysis of the existing flare systems of five US on-shore gas plants. The results are exposed in the table 2-8.

**Table 2-8. Significant issues with specific relief system components from multiple flare studies.**

| Component design issues from US on-shore gas plants | Plant 1 | Plant 2 | Plant 3 | Plant 4 | Plant 5 | Percentage of occurrences in total number of PRVs |
|---|---|---|---|---|---|---|
| Number of PRVs | 27 | 10 | 11 | 14 | 28 | total: 90 PRVs |
| PRV inlet/outlet pipes needed to increase in size | | 8 | 11 | | 28 | 53% |
| PRV relieving exceeded data sheet back pressure | 25 | | 11 | 6 | | 48% |
| Relief flow exceeded 0.7 Mach number | 22 | | 11 | 9 | | 48% |
| Relief flow exceeded maximum piping design temperature | 27 | | 2 | 7 | | 41% |
| Relief flow at header exceeds maximum design temperature | 25 | 4 | | | | 33% |
| Relief flow exceeded PRV typical maximum back pressure | 10 | | 11 | 6 | | 31% |
| Relief flow exceeded maximum piping design pressure | 15 | | | 7 | | 25% |
| Slug/Chocked flow present in piping | 4 | | | 9 | | 15% |
| Excessive liquid to knock out drum | | 4 | | | 4 | 9% |

The results show that 53% of the valves not fulfill the pressure drop rules in inlet and outlet pipes. This demonstrates deficiencies in the detail engineering work conducted by the engineering and contruction companies. The figures of plant 5 are fully unacceptable.

### Other reports

a) Parry (1992) presents the reliability data from Pearce and the results from the Reliability Data Bank of the NCSR. A summary of the results are:
   - Pearce reported that from a sample of 1,062 valves, 10% lifted at a pressure < 90% of the set pressure and 7% lifted at a pressure > 110% of the set pressure
   - The results of the NCSR Reliability Data Bank, based on a sample of 4,289 valves, are reported in the following table:

**Table 2-9. Safety valves failure data according to the NCSR Reliability Data Bank.**

| | |
|---|---|
| No. of safety valves | 608 |
| Total experience, years | 2576 |
| No. of failures | 821 |
| Falure rate per year | 0.319 |
| Failure to open at the set pressure | 37(5%)* |
| Failure to open fully at the relieving pressure | 307(37%) |
| Premature opening below the set pressure | 240 (29%) |
| Failure to reseat after opening | 2 (0 %) |
| Valve chatter | 5 (1%) |
| Leakage through the valve seat | 224 (27%)** |

| | |
|---|---|
| Leakage through the valve body | 0 (0%) |
| Rupture of the valve body | 6 (1%) |

[*] Percentage of failures.
[**] The high value in this case indicates that failures in operation have been noted whereas failures under test have been ignored or not observed.

b) Smith, Burgess and Powers (2011) reported that in a mid-sized US-based refinery with approximately 550 relief devices, 64 of them (12%) did not satisfy the 3% inlet pressure rule.
c) CCPS (1998) reported that, amongst the 100 worst major accidents that occurred in the process industry between 1956 and 1986, twenty-five could be attributed, at least partly, to the inadequate design or maintenance of pressure relief systems.
d) Chien et al. (2009) treated statistically the results from a lubricant plant with 252 spring-loaded pressure relief valves. Roughly, 60% of the valves were used in liquid service, while about 4% and 36% were used in vapor and two-phase services, respectively. The final sample was 229, because some valves had process deposits and mechanical damages and were sticky. The results were that 4% of the valves opened at a pressure higher than 20% of the set pressure and 12 % opened a pressure less than 95 % of the set pressure.

# Chapter 3. Development of a new methodology to increase the reliability of pressure relief valves based on the Safety Life Cycle analysis

A lot of authors have worked on the concept of safety life cycle. Knegtering (2002) devoted his PhD thesis to the safety life cycle management in the process industries, focused on the Safety Instrumented Systems (SIS). Riyaz (2005) reported the benefits of the safety life cycle in SISs projects. Gruhn and Cheddie (2006) analyzed the evolution of these standards; they reported that the ANSI/ISA S84.01-1996 was the first standard to introduce the Safety Life Cycle concept. It was defined as a “sequence of activities involved in the implementation of the Safety Instrumented Systems from conception to decommissioning”.

IEC 61508-1/7 (1998-2000) gives, in part 4, another definition as “necessary activities involved in the implementation of safety-related systems, occurring during a period of time that starts at the concept phase of a project and finishes when all of the electrical/electronic/programmable electronic safety-related systems, other technology safety-related systems and external risk reduction facilities are no longer available for use”. A note is added: “The term “functional safety lifecycle” is strictly more accurate, but the adjective “functional” is not considered necessary in this case within the context of this standard”.

IEC 61508 is a generic standard that outlines key requirements for all phases of the Safety Instrumented System Life Cycle. From this standard, various sector specific standards have been developed, such as IEC 61511 for the process industry, IEC 61513 for the nuclear power industry, etc. They are commonly known as functional safety standards. The main feature that distinguishes them from other related standards is that functional safety standards are performance-based, whereas many other standards, e.g. API 521, are prescriptive (Gruhn and Cheddie, 2006); instead of simply requiring one pressure relief valve to be installed in the prescriptive standards, the functional safety standards provide work process procedures and tools for the engineer to help decide how many safety valves are needed and how the valves should be designed, installed, operated and revised.

IEC 61511-1/3 (2003) or ANSI/ISA-84.00.01-2004 part 1, also give the definition “necessary activities involved in the implementation of safety instrumented function(s), occurring during a period of time that starts at the concept phase of a project and finishes when all of the safety instrumented functions are no longer available for use”

Each author has used the concept in different variants. For example Riyaz (2005) defines Safety Life Cycle as “a good, common sense design process with the same fundamental steps that any good design process would follow. First a problem is identified and assessed - then a design is done to solve the problem. Finally the design is verified (checked and tested) to make sure that it actually solves the original problem that was identified”. Some people complain that performing all of the steps in the Safety Life Cycle, will increase overall costs and result in lower productivity. Gruhn (2006) referenced a study performed by Levenson in 1995, where it was demonstrated that production increased as safety increased.

The steps of the new Safety Life Cycle methodology proposed in this thesis have been adopted from IEC 61511-1 (2003) and can be seen in Figure 3-1.

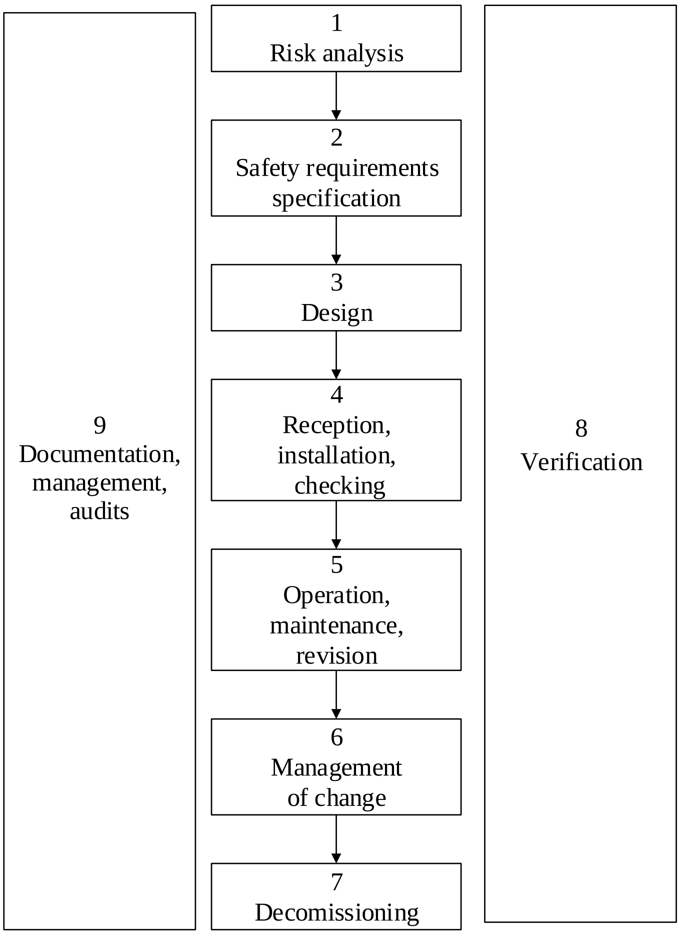



**Figure 3-1. The main steps of the new methodology based on the Safety Life Cycle analysis.**

## 3.1 Phase 1. Risk analysis

**Literature survey on problems detected**

As pointed out by Berwanger et al. (2000), the Process Hazard Analysis (PHA) methodologies (Hazop, What if, etc.) do not identify all overpressure scenarios in the equipment. This conclusion came from the fact that essentially all the 250 operating units analyzed by these authors had already undergone hazard analyses. They added that most deficiencies are not identified in PHAs because of time constraints and a general lack of pressure relief systems expertise in PHA teams. Another point that contributes to this fact is that the number of pieces of equipment in each node in the Hazop analysis is usually too big. These authors proposed using an equipment-based approach, i.e. using a screening of the 17 possible overpressure scenarios listed in API 521 (2014) for each piece of equipment (Berwanger and Kreder, 1998).

Marshall et al. (2011) stated that “Contrary to conventional thinking, the PHA process is not an effective tool for capturing relief system deficiencies”. They concluded that neither Hazop, What-If/Checklists, Fault-Tree, LOPA, etc. are appropriate. On the contrary, these methods take credit for proper relief system design as an inherent safeguard in mitigating loss of containment consequences.

Hellemans (2009) also remarks on this problem, adding that even though most of the plants have been designed by reputable engineering firms, the vast majority of deviations are not identified through PHAs but either via external audits or when accidents have already occurred. He adds that the problem in engineering companies is that the design, instrumentation, process and piping engineers have such a variety of components to cover, that there is little specialization in pressure relief systems alone. Moreover, he remarked that the standard internal audit may not catch the piece of equipment that does not have a valve but should have it. Hellemans also concluded that many specialists in pressure relief systems believe that the conventional PHA methods are not always the most effective tools for evaluating pressure relief systems.

Dunjó (2010), in his PhD thesis, wrote about the inefficiencies of the Hazop methodology, among them the node-selection methodology. He proposed defining nodes with no more than two major equipment for a time concept optimization. In our case, his recommendations would be taking nodes with very little equipment, especially with the guidewords "more pressure" and "more temperature".

The opinion of this author is that in minor projects performed in already existing process plants and according to the risk analysis demanded by Management of Change procedure, a less demanding method than Hazop is normally used, for instance What-if. Although the analysis is done by experienced people who work and have worked for a long time in the unit, the same problems, as explained by Hazop, could occur.

Wong (2001) adds to the list of 17 scenarios of overpressure the human error. Although human error is one of the most important issues in accidents (Anonymous, 2000), this risk factor is not enough implemented yet in Hazop studies, as demonstrated by Dunjó (2010).

**Procedure in the new methodology**

The risk assessment will be performed focusing on an "equipment by equipment" approach. In each piece of equipment the list of Table 1 of API 521 (2014) will be used as a check list slightly modified. This work will be done together with the formal PHA studies together with the review of past accidents and incidents with the same substances and processes used, as has been recommended by Directive 2012/18/EU (article 10, annex II) . A specific template will be used, which can be seen in Figures 3-2 and 3-3. The explanation of each contingency of Figure 3-2 can bee seen in section 4.1.2.

| **Contingency** | | **Comments** | **Justification** |
|---|---|---|---|
| 1 | Blocked outlets | | |
| 2 | Abnormal heat input | | |
| 3 | Exchanger tube breakage | | |
| 4 | Auto control failure | | |
| 5 | Reflux failure | | |
| 6 | Fire | | |
| 7 | Cooling water failure | | |
| 8 | Power failure | | |
| 9 | Instrument air failure | | |
| 10 | Inadvertent valve open/close | | |
| 11 | Mechanical equipment failure | | |
| 12 | Heat loss (series fractionating columns) | | |
| 13 | Thermal | | |
| 14 | Loss of quench/cold feed | | |
| 15 | Chemical reaction | | |
| 16 | Steam out | | |

**Figure 3-2. Contingency analysis data sheet.**

| TAG/EQUIP. NUMBER | UNIT /SERVICE: | P&ID: | PLANT: | COST CENTER: |
|---|---|---|---|---|

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| EQUIPMENT PROTECTED: | | | SET PRESS: BARG | | BASIS: | | |
| | | | DISCHARGE DISPOSITION: | | INLET PRESSURE DROP: BAR | | |
| | | | CONSTANT BACKPRESSURE: BARG | | VARIABLE BACK PRESS.: BARG | Kd = | |
| EQUIPMENT DESIGN CONDITIONS: ( ) MAWP | ( ) Design | ( ) Other | BUILT-UP BACKPRESSURE: BARG | | TOTAL BACKPRESSURE: BARG | Kb = | |
| NORMAL OPER. BARG | ºC | Rupture Disk ,Y/N | FIRE SUMMARY | | WETTED AREA: m² | ATTACH SKETCH FOR AREA CALCULATION: | |
| MAX OPER. BARG | ºC | Derating Factor= | INSULATION | | TYPE | THCKNS mm | Insul factr, 1=none |
| DESIGN BARG | ºC | (Use 0.9 if have rupture disk) | Q = KJ/h | | | | |
| CONN: RATING FACING : | PIPE SPEC, IN/OUT: / | | | | | | |

| **Causes of Relief**<br>Refer to API RP520, RP521 and ISO 4126 | | | **RELIEF LOAD** | | **RELIEF CONDITIONS** | | **FLUID PHYSICAL PROPERTIES AT RELIEF CONDITIONS:** | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **Contingency** | Comments NA, etc | % OV PR | VAPOR kg/h | LIQUID m3/h | PRESS BARG | TEMP °C | FLUID TYPE | VAPOR MOL WT | SP GRAVITY LIQUID | COMPR FACTOR Z | LATENT HEAT L, KJ/kg | SP HEAT RATIO k | LIQUID VISC cP | VAPOR VISC cP | VAPOR AREA V mm² | LIQUID AREA L mm² | TOTAL AREA T mm² |
| 1. BLOCKED OUTLETS | | | | | | | | | | | | | | | | | |
| 2. ABNORMAL HEAT INPUT | | | | | | | | | | | | | | | | | |
| 3. EXCHANGER TUBE BREAKAGE | | | | | | | | | | | | | | | | | |
| 4. AUTO CONTROL FAILURE | | | | | | | | | | | | | | | | | |
| 5. REFLUX FAILURE | | | | | | | | | | | | | | | | | |
| 6. FIRE | | | | | | | | | | | | | | | | | |
| 7. COOLING WATER FAILURE | | | | | | | | | | | | | | | | | |
| 8. POWER FAILURE | | | | | | | | | | | | | | | | | |
| 9. INSTR. AIR FAILURE | | | | | | | | | | | | | | | | | |
| 10. INADVERTENT VA. OPEN/CLOSE | | | | | | | | | | | | | | | | | |
| 11. MECH. EQUIP. FAILURE | | | | | | | | | | | | | | | | | |
| 12. HEAT LOSS (SERIES FRAC.) | | | | | | | | | | | | | | | | | |
| 13. THERMAL | | | | | | | | | | | | | | | | | |
| 14. LOSS OF QUENCH/COLD FEED | | | | | | | | | | | | | | | | | |
| 15. CHEMICAL REACTION | | | | | | | | | | | | | | | | | |
| 16. STEAM OUT | | | | | | | | | | | | | | | | | |
| 17. | | | | | | | | | | | | | | | | | |
| 18. | | | | | | | | | | | | | | | | | |
| 19. | | | | | | | | | | | | | | | | | |
| 20. | | | | | | | | | | | | | | | | | |

| NOTES: | | | |
|---|---|---|---|
| | GENERAL DATA | BY: | DATE: |
| | PROCESS DATA | BY: | DATE: |
| | VALVE SIZING | BY: | DATE: |
| | CHECKED/APPROVED | BY: | DATE: |

EXISTING RV DETAILS:

| SIZING CASE SELECTED: | | | RELIEF DEVICE TYPE: / | TOTAL ORIFICE AREA REQD: **mm²** | | |
|---|---|---|---|---|---|---|
| DEVICES SELECTED - | QTY: | INLET SIZE: mm | OUTLET SIZE: mm | ORIFICE/AREA (1): mm² | SET PRES: BARG | |
| | QTY: | INLET SIZE: mm | OUTLET SIZE: mm | ORIFICE/AREA (1): mm2 | SET PRES: BARG | **Relieving Loads Summary Data Sheet** |
| | QTY: | INLET SIZE: mm | OUTLET SIZE: mm | ORIFICE/AREA (1): mm2 | SET PRES: BARG | |
| | QTY: | INLET SIZE: mm | OUTLET SIZE: mm | ORIFICE/AREA (1): mm2 | SET PRES: BARG | |

**Figure 3-3. Relieving loads summary data sheet.**

Concerning the PHA studies, Hazop methodology integrated with LOPA analysis has been used in this thesis, with especial emphasis in the guidewords “more pressure” and “more temperature”.

Moreover, through the LOPA analysis, each pressure relief valve has been allocated as Independent Protection Layer (IPL) with assigned credits depending of its reliability. Figure 3-4 shows the example of the valve YS702-01, taken as study case, where it has 4 IPL credits.

Session: (5) 29/05/2012 Revision: (1) 08/01/2013
Node: (2) PROPYLENE PURIFICATION AND FEEDING. From H70001, through K700, K702 A/B pumping by P700 A/B, to F40101/40102/40103/40123/40125 and F40106 for R400 and equivalent valves for R410. Pumps P700A/B recirculation through F70201 and W700 is also included. Propylene line to compressor V410 included until valve H70204.
Intention: Feed purified propylene to reactors R400/410 at fixed pressure and flow. Psuction = 10 barg. Pdischarge = 37 barg. T = 40ºC. Q = 20-30 t/h.
Drawings: B401 - REV 19 (H); B411 - REV 13 (H); B700 - REV 9 (H); B701 - REV 7 (H); B702 - REV 12 (H)
Parameter: Temperature

| GW | DEVIATION | CAUSES | CONSEQUENCES | UL | UF | UR | SAFEGUARDS | CL | CF | CR | RECOMMENDATION | ML | MF | MR | BY | REMARKS |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| More | Higher Temperature | 65. Same As 62 | | | | | | | | | | | | | | |
| | | 66. No cooling water in W700. | 66.1. If this situation is maintained over time, increase of propylene temperature and pressure. Vaporization of propylene, cavitation of P700A/B. Mechanical damage to pump. | 1 | -1 | D | | | | | | | | | | |
| | | 67. No cooling water in PW701A/B. | 67.1. If the situation is maintained over time, damage to pump. | 1 | -1 | D | 67.1.1. T70207/70208 that stops P700A/B in case of high temperature. | 1 | -2 | D | | | | | | |
| | | 68. External fire. | 68.1. Increase of pressure. Mechanical integrity at risk for different equipment in the node. Risk of fire or explosion. | 3 | -2 | B | 68.1.1. YS700/01 and YS700/02 (1 in stand-by) for K700. | 3 | -6 | D | | | | | | - 4 IPL credits given to safety valves. Clean service, designed for external fire. |
| | | | | | | | 68.1.2. YS701/03 and YS701/04 (1 in stand-by) for K702A. | | | | | | | | | |
| | | | | | | | 68.1.3. YS701/01 and YS701/02 (1 in stand-by) for K702B. | | | | | | | | | |
| | | | | | | | 68.1.4. YS702/01 for W700. | | | | | | | | | |
| | | | | | | | 68.1.5. SV702/10 for F700. | | | | | | | | | |
| Less | Lower Temperature | 69. No significative situations that introduce additional hazards are identified. | | | | | | | | | | | | | | |

**Figure 3-4. Hazop integrated with LOPA, where each safety valve shows its allocated Independent Protection Layer. Example of YS702-01.**

A proposal has been presented to DIERS to stablish a standard education program for “Expert in pressure relief systems”, similar to the program for experts in Safety Instrumented Systems according to IEC 61508/61511 with the courses “Certified Functional Safety Expert” and “Certified Functional Safety Professional”.

## 3.2 Phase 2. Safety requirements specification

**Literature survey on problems detected**

Kreder and Berwanger (1995) pointed out that the steps of the traditional process of specifying a relief device are:

1. Identifying a piece of equipment or piping system that may be exposed to overpressure.
2. Determining all the possible causes of overpressure for this item.
3. Quantifying the required relief load for each scenario of overpressure.
4. Developing a relief strategy to protect the unit. Usually, one relief valve protects the item from several causes of overpressure.
5. Calculating the required size of the relief device.
6. Identifying any special considerations: need of a bellows valve, two-phase flow, etc.
7. Specifying the relief device based on the largest size requirement, taking into account any special requirements.

From all this information, usually generated by the engineering firm, the owner and operator of the plant received only a datasheet for each relief device. Also the information generated in steps 2 through 6 was not transferred to the data sheet. Although it is reasonable to assume that the designer of the original relief system did consider various causes of overpressure, the engineers of the plant could not demonstrate that all possible causes of overpressure had been addressed. The authors wrote that a typical pressure relief valve data sheet in use today, fails to capture about 75% of the design process, and often does not address the changes that take place over time in the plant. The authors also recommended the use of an intelligent electronic data base rather than an all-paper system.

Melhem (2010), from ioMosaic, listed the following risks in existing relief systems:

- Outdated, different formats, and/or non-existent design basis and supporting calculations
- Most data do not comply with new API 521(2008) documentation suggestions
- Missing or outdated material and energy balances
- Missing or outdated isometrics and vessel design data
- Missing or outdated vent containment design basis
- Atmospheric relief devices
- Overloaded flare systems
- Existing relief calculations ignore chemical reactivity and multiphase flow
- A large majority of existing systems using pressure relief valves suffer from excessive inlet pressure drop and excessive backpressure
- High pressure systems - cold temperatures downstream of the pressure relief valve
- Vibration risk.

The results from this author indicate the same problem as those from Kreder et al., the pressure relief valve data sheet does not give the information generated originally by the engineering firm.

Hellemans (2009) also wrote "Many specialists conclude that the pressure relief system design process could be improved. Working closely with a lot of the design firms, I concluded that they merely try to comply with the codes at a minimum cost and care very little about Life Cycle Cost of the components". He added that the consequences fall on the end users' maintenance departments.

My own experience, having worked in an engineering and construction firm, is that there are so many subjective decisions in the calculation of the relieving load of a pressure relief valve, that any modification made in the data sheet at the detail engineering phase could mean, for instance, a change in the diameter of the nozzle of the equipment protected, increasing the cost of the project and perhaps delaying its execution. That is why I had never been allowed to show such calculations to the client.

**Procedure in the new methodology**

A new data sheet has been generated. It has been developed based on the model presented at Annex D of API 520 (Part I, 2008); nevertheless, it has been significantly modified in order to incorporate all the relevant information required. Figure 3-5 presents the new safety valve requirement specification for the pressure relief valve YS 702/01 that corresponds to the valve of the case study 6.4.

The new data sheet has a stability section with all the necessary parameters, as will be discussed in section 4.6. Already in the detail engineering phase it is necessary to assign an inspection period for the valve according to company guidelines. A recommended proposal in this work is the proposal of API 581 (2008) as it is discussed in section 4.11.

| | | | |
|---|---|---|---|
| | Piping Engineering<br>Special Item<br>**Safety Valve**<br>**Requirement Specification** | | |

| No. | Group | Item | Value | Item | Value |
|---|---|---|---|---|---|
| 01 | | Ident. No YS702/01 | | Special Item No. | |
| 02 | | Designation | **Pressure relief valve** | | |
| 03 | | Component key No./Var. | | Ident. No. Customer | |
| 04 | | Piping Engineering | | Prepared: Basco | Date: June 30, 2014 |
| 05 | | Location | Petrochemical plant | P&I-diagram | |
| 06 | | Regulations | ☑ AD-Merkblatt ☐ API ☐ | Unit No. | |
| 07 | | Additional regulations | | Protected pressure chamber | $m^3$ |
| 08 | Safety | ☐ Lifting lever | ☐ Blocking gag | Allowable working pressure | 45 bar |
| 09 | Safety | ☐ Change-over valve with locking device at inlet | | Allowable working temperature | 120 ºC |
| 10 | Safety | ☐ Change-over valve with locking device at outlet | | | |
| 11 | Construction Valve | ☐ weight loaded | ☑ spring loaded | Connection criterion – Piping class – Inlet | ST860E-A |
| 12 | Construction Valve | ☐ supplement. loaded | ☐ gas-tight | Connection criterion – Piping class – Outlet | ST261C-E |
| 13 | Construction Valve | ☑ metallic seat | ☐ soft seat | Connection criterion – Facing | ☑ DIN ☐ ASME B 16.5 ☐ |
| 14 | Construction Valve | ☐ open bonnet | ☑ closed bonnet | Connection criterion – Facing – Inlet | Outlet |
| 15 | Construction Valve | ☐ | | Proposed material (body) | |
| 16 | | **Process Data** | | | |
| 17 | | Fluid | Propylene ( liquid ) | Operating pressure $P_A$ | 16.2 bar |
| 18 | | (mixed fluids to be | | Operating temperature $t_A$ | 42 ºC |
| 19 | | determined in weight %) | | Max. total back pressure $P_a$ – constant | 0.15 bar |
| 20 | Condition and properties of the fluid | ☑ gas | ☐ steam | Max. total back pressure $P_a$ – variable | 3.3 bar |
| 21 | Condition and properties of the fluid | ☑ liquid | ☐ vapour – liquid mixture | Set pressure (Start to lift press.) P | 45 bar |
| 22 | Condition and properties of the fluid | ☐ vapour | ☐ fat containing | Allowable accumulation | 10 % |
| 23 | Condition and properties of the fluid | ☐ tending to conglutinate | ☐ toxic | Temperature on discharge – Inlet | 100 ºC |
| 24 | Condition and properties of the fluid | ☐ solids containing | | Temperature on discharge – Outlet | ºC |
| 25 | Condition and properties of the fluid | Corrosive matter | No | Required discharge capacity | 10000 kg/h |
| 26 | | | | Isentropic exponent | 1.55 - |
| 27 | | Critical blow-off condition | Fire | Compressibility factor Z=pV/RT | 0.7 - |
| 28 | | | | Flow coefficient $\psi$ | |
| 29 | | | | Factor *x* for steam | $hmm^2 bar/kg$ |
| 30 | Valve heating | Temperature | ºC | Dynamic viscosity | mPas |
| 31 | Valve heating | ☐ Inlet | ☐ Body | Molecular mass *M* | 42 kg/kmol |
| 32 | Valve heating | Type | | Density at set press. and discharge temp. | kg/m3 |
| 33 | | | | Maximum discharge capacity | 15827 kg/h |
| 34 | | Completed: | Date: | Modified on Rev.: | Date: |
| 35 | | Approved: | Date: | Approved: | Date: |

Remarks:

| **Stability Calculations** | | | **Maintenance** | |
|---|---|---|---|---|
| Inlet pressure drop[2] | 1.4 | % (<3% SP) | First assigned revision interval | 5 years |
| Outlet pressure drop[2] | 7.6 | < 10% conventional<br>< 30-50 % bellows | Revision of intervals according to: | ☐ own guidelines<br>☑ API 581 |
| Max inlet line length | 0.3 | m | Noise level | 142 dB |
| Acoustic pressure losses | 25.7 | bar<br>(Blowdown > ΔP fricc. + ΔP acoustic) | | |
| Vortex shedding inlet line | 26 | m | | |
| Acoustic Induced vibration(AIV ) | ☑ check | ☐ No check | | |
| Body Bowl Chocking | ☑ check | ☐ No check | | |

| Rev. | Sheet | Name | Date | Name | Name | Date | Status | Remark, kind of revision |
|---|---|---|---|---|---|---|---|---|
| 0 | 1-2 | Basco | 7/1/14 | | | | | Revised as marked |
| | | | | | Approved | | | |

| Designation | | Basic document | DG | DL-Nr. | DCC |
|---|---|---|---|---|---|
| | Project Name<br>PhD Thesis | Project-No. | Document-No. | Sheet/of<br>1/2 | Revision<br>0 |

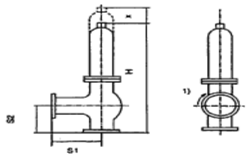
| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | | | Piping Engineering<br>Special Item<br>**Safety Valve**<br>**Requirement Specification** | | | | | |
| 01 | | Ident. No YS 702/01 | | | | Special Item No. | | |
| 02 | | Construction features | | | | | | |
| 03 | • | Manufacturer | LESER | | • | Type test approval by | □ TUV | □ UV-Stamp |
| 04 | • | Manufacturer's type | 4564.6052 | | • | No. of type test approval | | |
| 05 | • | Class – orifice letter | | | • | Material: Body | 1.4581 | Test certificate acc. to EN 10204 |
| 06 | • | Type □ Safety valve | ☑ Full lift | □ Relief | • | Inlet nozzle | 1.4581 | |
| 07 | • | Flow diameter | required | mm | • | Bonnet bolts | | |
| 08 | • | $d_0$ per valve in service | selected | 20 mm | • | Bonnet nuts | | |
| 09 | • | Flow Area $A_0$ | required | 220.55 $mm^2$ | • | Seat, disc | 1.4404 | |
| 10 | • | per valve in service | selected | 314.1593 $mm^2$ | • | Cooling spacer | | |
| 11 | • | a/1, 1-value (certified coefficient of discharge a | | | • | Balanced bellows | | |
| 12 | • | Connection – Inlet: DN | 25 | PN 100 | • | Spring, bonnet, yoke | 1.4310 | |
| 13 | • | NPS | | Class | • | | | |
| 14 | • | Facing | E DIN | ASME | • | Test of material | TRB 801 No. 45 | |
| 15 | • | Thread | | Standard | • | Spring, weight adjustment by | AD – A2 | |
| 16 | • | Connection – Outlet: DN | 50 | PN 40 | • | Constr. features: □ Bonnet open | □ Bonnet closed | |
| 17 | • | NPS | | Class | • | □ Bonnet gas tight | □ Blocking gag | |
| 18 | • | Facing | C DIN | ASME | • | □ Balanced bellows | □ Cooling spacer | |
| 19 | • | Thread | | Standard | • | □ Lifting lever | □ Lift stopper | |
| 20 | • | Test pressure $Pe = P\cdot Pa$ | bar | | • | Special features of construction | Lifting device has to be sealed | |
| 21 | • | Reaction force on discharge $F_R$ | 4082 | N | • | | | |
| 22 | • | Opening pressure 2) | 10 | % | • | | | |
| 23 | • | Reseating pressure 2) | 10 | % | • | | | |
| 24 | • | | | | • | Kind of interlocking | | |
| 25 | • | | | | • | Weight | 20 kg/piece | |
| 26 | • | Dimensions S 120 mm | X mm | $S_1$ mm | | No. required | In service 1 Piece | Spares Piece |
| 27 | • | mm | $S_2$ mm | H mm | | | Total quantity | Piece |

↑ The marked rows are to be completed by the manufacturer, when not specified by others Row revised under Rev. No ↑

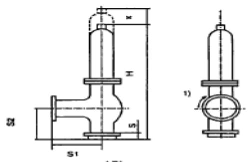
Remarks:

1) Marking durable on flange and on name-plate
2) Referred to set pressure
   All pressure data indicated are gauge pressures

□ Mark if applicable



DIN



API

**Remarks:**

Certified coefficient of discharge for steam and gases, $\propto_{d,DG} = 0.8$
Certified coefficient of discharge for liquid, $\propto_{d,F} = 0.6$

| | Project Name | Project No. | Document No. | Sheet /of | Rev. |
|---|---|---|---|---|---|
| | PhD Thesis | | | 2/2 | 0 |

**Figure 3-5. New safety valve requirement specification (data sheet).**

## 3.3 Phase 3. Design

**Literature survey on problems detected**

In Chapter 2 the results of statistical surveys about deficiencies in existing pressure relief valves due to undersizing showed similar results: 13.4 % (Kumana and Aldeeb, 2014), 9% (Berwanger et al., 2000), 11% (Hellemans, 2009) and 10% (Hellemans, 2009), although the last two results included undersized and oversized safety relief valves (note: the 10% is from Hellemans own data).

The design problems identified by Berwanger et al. are listed below by frequency and in descending order:

- External fire, 41%. Calculations were performed taking into account the equations of the API 521 for fire, considering the adequacy of the firefighting and drainage systems. The wetted area was based on the high liquid level in the vessel up to 7.6 m above grade. An environmental factor of 1.0 was generally used unless the insulation was fireproof
- Exchanger tube rupture, 16%. The “two-thirds rule” was applied; nevertheless, tube rupture scenarios were identified only for shell and tube exchangers that did not fulfill the rule. The calculations were performed considering two sharp-edged orifices. The maximum expected pressure upstream of the high pressure side and the relieving pressure of the low pressure side were used to calculate the pressure drop across the orifice
- Blocked outlet, 13%. The quantity of the material to be released was determined at relieving conditions based on the capacity of upstream pressure sources or on the heat load of process heaters
- Inlet control valve failure, 13%. Calculations were based on manufacturer’s valve flow calculations, assuming a fully open valve. The differential pressure across the control valve was the difference between the maximum expected upstream value and the downstream relieving pressure. Sometimes, the flow through the control valve was limited by the capacity of upstream equipment, such as a pump, or by the piping. The failure of level control valves that control the flow of liquid from a higher to a lower pressure system can result in a loss of liquid level and the flow of a gas or vapor through the valve. This is known as “gas blow-by”. In this case, the flow was the maximum allowed for the control valve. Consideration was also given to the possibility for the downstream vessel to fill above the normal level which could result in the vapor flow from the control valve entering the liquid space in the downstream vessel and subsequent two-phase relief
- Multiple scenarios, 12%. In this case, the piece of equipment had an undersized relief device for more than one potential overpressure scenario
- Other, 3%. This category includes the other scenarios not presented here but listed in the 16 overpressure scenarios of API 521
- Loss of condensation, 2%. This case corresponds to a loss of cooling water, for example in a distillation column. Moreover, in a typical design the loss of cooling can also result in a loss of reflux after 15 minutes. Because API 521 recommends the comparison of the relieving load before and after the loss of reflux, both cases were performed by Berwanger et al. The method followed is from Sengupta and Staats (1978).

According to the results of Berwanger et al. the question that arises is whether the engineers who developed the information worked properly. Bravo et al. (1995) pointed out the following pitfalls in doing the relief system design:

- A common mistake is assigning the engineering of relief systems to a junior engineer. Even though with the computer software available today it is easy to oversimplify the engineering process, experience is critical in the proper selection of the dimensioning scenario
- Although codes and regulations are available (API 520, API 521, ISO 4126, API 2000, AD Merkblatt, etc.), these codes are not precise and good judgment is necessary. As an example, the author presents the common mistake of taking credit for the environmental factor of the fire case when the insulation is not fireproof certified

- Consideration of the possibility of runaway reactions when sizing the pressure relief valve. The use of Fauske nomograph for design and not for preliminary sizing only, is one of the common mistakes that engineers make
- Considering two or more simultaneous failures
- Taking credit for the properly work of the instrumentation in an upset condition. Similarly, relying on the proper operation of check valves, even if two are installed in series
- Ignoring the need of a valve in case of thermal expansion of liquids
- Using incorrect or inaccurate physical-property data, for example, by assuming ideal gas behavior for a gas near its critical point. Another example is using the heat of vaporization of 115 kJ/kg for liquids near the critical point. A further example is using a heat capacity of 2510 J/kgºC as is commonly suggested for organic liquids, and this value can lead to undersized valves.

Smith and Burgess (2012) wrote about the article of Berwanger et al. (2000) “Since the publication of this article, many of these concerns have undergone a more detailed review showing that modifications to the facility were not required to address these concerns”. The authors emphasize the use of good judgement to see if the relieving scenario has realistic or unrealistic conservative assumptions, as for example:

- Pumps that can only pump the relief pressure if the upstream system is also upset (however, a simultaneous upset would be double jeopardy)
- Systems where overpressure comes from heat input, but the relief temperature of the process fluid exceeds the relief temperature of the utility fluid
- Control valve failure calculations that are based on the capacity of a control valve instead of on a long section of piping.

Other authors focused on specific pitfalls:

a) Basco et al. (2012) pointed out that one common mistake is not considering the expansion of the liquid inside the vessel in case of fire due to the decrease of density, before boiling starts. In this case, the wet area is bigger than the normal maximal level at the beginning of the fire. Or the lack of checking with the available methods if there are two-phases at the input of the pressure relief valve or not. Another common mistake the authors exposed is the lack of rigorous calculation of the latent heat of mixtures, considering the maximal of a function that involves the parameters related to the area of the pressure relief valve as presented by Wong (2000).
b) Shackelford (2003) presented a comparison of the errors of undersizing that the engineers do when using the real Cp/Cv value, normally obtained from commercial simulators, together with the compressibility factor in the common formulas for calculating the required relieving area in the case of gases.
c) Bradford and Durrett (1984) wrote about common mistakes in using the total gross overhead load for sizing the pressure relief valve in case of condensation failure in distillation columns. One of the most frequent ones is taking credit for the pinch of temperatures in the reboiler, without considering the composition of the feed and the holdup of the bottoms among other parameters.

**Procedure in the new methodology**

To tackle these problems, the following information must be generated for each pressure relief valve:

- A chronological record of all the files including all reviews, evaluations, design modifications, etc., which have been performed on the system
- Data sheets including the justification of the scenarios that have been considered (reference: Table 2 of API 521 (2007, 2014)
- A diagram of the safety valve, equipment protected and connecting pipes, including diameters and thickness. Alternatively, the P&ID may be sufficient
- Relief load analysis (contingency analysis), including the calculations corresponding to all the relief cases assumed for the valve that are both specific to the unit (blocked outlet, reflux failure,

rupture of a tube in a heat exchanger, etc.) and cumulative (fire, total/partial power failure, cooling water failure, etc.). Commercial simulators or specific software may in certain cases be required, such as in the isentropic expansion of supercritical fluids (Ouderkirk, 2002, Aspen-Hysys v8.6, 2015) or the relief in distillation columns due to cooling failure (Sengupta and Staats, 1978) or two-phase flow relieving (Leung, 1996; EN ISO 4126-10, 2010; API 520, 2008; Schmidt, 2012; etc.). The simplified process calculations done to demonstrate that a particular case is not the governing one for pressure relief valve sizing should also be included

- Calculation of the required relieving area of the safety valve, including the results of the software used: PS PPM software (Siemens), Aspen-Hysys v8.6, Valvestar (LESER), etc.
- Isometric diagrams of the valve inlet and outlet pipes, showing all details required for a proper installation
- Calculations of the pressure drop in the pipe that connects the protected unit to the safety valve. These calculations should identify the size and equivalent lengths or K coefficients used for each section of piping and the roughness factor employed. They should be performed for the maximum flow rate that the valve can release, rather than the required flow rate in case of gases/vapors
- Calculation of the pressure drop in the outlet pipe, taking into account the specific circumstances: superimposed backpressure, pressure drop in the flare system if fluids are discharged to a flare, the influence of two-phase flow (assuming isentropic behavior at the valve, especially when relief conditions are close to the critical point), possible sonic velocity at certain points (Mach number), etc. Tail pipes must be calculated for maximum flow rates, and headers and subheaders for the required flow rate (API 521, 2007). Pressure drop at the valve outlet must be limited to 10% of the set pressure for conventional valves, and 30%-50% for balanced valves
- Calculation of the forces and moments imposed on the valve and piping during the relief, especially in the case of discharges to the atmosphere
- Calculation of the stability of the safety valve. The time required by the pressure wave generated by the opening to reach the protected vessel and travel back to the valve should be less than the valve opening time. This is still a difficult point (Melhem, 2011). According to Smith et al. (2011), the methods proposed by Cremers et al. (2001) and Melhem (2011) are good approaches for establishing the maximum value of the characteristic constant K corresponding to the inlet pipe. Included in these calculations are the inlet frictional and acoustic pressure losses, the vortex shedding criteria and the oversized criteria (Smith et al., 2011). Recently, the API Simple Force Balance method developed by Melhem (2014, 2015) and presented in the API 520 (Part II, 2015) is gaining acceptance in the community, nevertheless the necessity of knowing the blowdown with precision is required. The dynamic models (Darby et al., 2013, 2014; Melhem, 2014, 2015; Hös et al., 2014, 2015) including the system (equipment protected) or not, are actually too much complicated without dedicated software and without the characteristic parameters of the valve being known, like spring constant, mass of the moving parts, damping factor and geometric parameters (Darby et al. 2013, 2014). However, they can be applied to a specific case. An example of the typical damage to the disc and the seating surface of a safety valve because of chattering is presented in Figure 4.4
- Calculation of Acoustic Induced Vibration (AIV). The vibration at certain points (connection of tail pipes to subheaders or headers) can give rise to releases. The methodology proposed by Eisinger (1997) or the Energie Institute (2008) for screening this is often used
- Noise calculation according to codes (for instance, section 7.3.4.3, API 521, 2007)
- Calculation of the body bowl chocking pressure, to determine the critical flow in the valve just before the outlet nozzle. This must be checked only in conventional valves. D'Alessandro (2011) proposed a methodology that assumes isentropic flow from the valve inlet to the outlet nozzle and adiabatic flow from the nozzle to the valve outlet. Izuchi (2015) presented a more complex procedure but this has been validated by experimental work with relative good results
- Calculation of the optimal revision time, according to pressurized equipment standards (RD2060, 2008) or to risk analysis methodologies, such as API 581 (section 7, 2008).

## 3.4 Phase 4. Reception, installation and checking

**Literature survey on problems detected**

Concerning the reception deficiencies, once more Hellemans (2009) gives his own data: 29 % of the 750 complaints during one year were due to transport and handling issues. In Hellemans words “it is unimaginable how carelessly safety relief valves are sometimes treated within the industry, and this is mostly due to a lack of knowledge”.

Scully (1981) reported that 2-3% of the valves shipped from the manufacturer had become misaligned due to vibration or rough handling during transportation.

The same graph reported by Hellmans shows that 12 % of the valves were defective due to issues with manufacturing. This is why, before installing a pressure relief valve in the plant, it is necessary to always test it to confirm its opening pressure setting (Malek, 2006).

In Chapter 2, the results of statistical surveys about deficiencies in existing pressure relief valves due to installation (based on the number of valves, not pieces of equipment), showed a result of 24.9 % (Kumana and Aldeeb, 2014), 19% (Berwanger et al., 2000), 11% (Hellemans, 2009) and 33% (Hellemans, 2009) (note: the 33% is Hellemans own data based on the 750 complaints mentioned before).

The design problems identified by Berwanger are listed below by frequency in a descending order:

- Outlet pressure drop too high, 49%. This concern is due to the limitation imposed on conventional pressure relief valves of a maximum built-up back pressure in the outlet of the valve of approximately 10%. The pressure drop should be based on the rated valve capacity consistent with the inlet piping pressure drop. This pressure drop should be calculated between the exit of the valve and the exit of the tail pipe (the atmosphere or the entrance to a main relief header) (note: although there is no mention in Berwanger’s paper about the treatment of the balanced and pilot operated valves, according to the Kumana and Aldeeb work, which includes the statistics of Berwanger et al., the pressure drops higher than 30% for conventional valves and 50% for pilot operated valves had also been considered)
- Inlet pressure drop too high, 33%. Inlet pressure drops were calculated for frictional losses only at the rated capacity of the valve
- Blocked relief pathway, 8%. The isolation valves installed in the inlet pipe should be full bore and should have the capacity of being locked open or carsealed open. Deficiencies were identified by checking the entire relief pathway for each protected piece of equipment, to ensure the existence of an open pathway
- Set pressure too high, 9%. Deficiencies were identified by comparing the current set or burst pressure to the Maximum Allowed Working Pressure of all protected equipment
- Others, 1%. This category represents all other installation deficiencies related to other installation requirements of codes API 520 (Part II, 2015) and ASME Section VIII Div 1.

Basco et al. (2012) also pointed out the problem that mechanical maintenance people in process plants usually associate pressure relief valves with gate valves rather than with control valves.

**Procedure in the new methodology**

The solution to these problems could be a pre-startup safety review which has been proposed, covering the recommendations included in API 520 Part II (2015), ASME Code B31.1 Appendix II (2010), ISO 4126-9 (2008), Hellemans (2009), Malek (2006) as well as the recommendations and information from the manufacturers: Leser, Bopp & Reuther, Pentair, Nacional Safety Valves (formerly Válvulas Nacional, SA), etc. and following the experience of the author.

Thus, a dedicated check list has been created according to previous considerations (Annex B). It follows the framework of CCPS (Appendix A, 2007).

The checking process consists of a validation in hardware and documentation for the installation that fulfills all the requirements as listed in the Safety Requirements Specification, in short, the pressure relief valve data sheet. This part is integrated into the above prestart-up safety review.

## 3.5 Phase 5. Operation, maintenance and revision

**Literature survey on problems detected**

The concept of operation could sound strange when applied to a device which is expected never to operate. There are lots of pressure relief valves which have never opened, except during a revision in the workshop. Thus, the only way to demonstrate that a pressure relief valve can open when an upset occurs is through testing it from time to time.

One of the problems in correct operation is avoiding leakages. Leakages can be discovered after a test on the manufacturing site due to damage in the seating face obtained during shipping, to mishandling, to contamination or to poor installation (Bright, 1972; Malek, 2006). The most common cause of leakage is when there is dirt or scale on the seating face (API 576, 2009). However, some leakage is permitted, according to API 527 (2007); for the air test at 90 % of the set pressure and for pressures up to 70 barg, a range of 20-40 bubbles per minute is allowed depending on the orifice diameter. Any value higher than that is considered a leakage. For water, a maximum of 10 ml/h per inch of nominal inlet diameter is accepted.

Scully (1981) and Nelson (1993) reported that the most common causes of leakage are:

- Operating pressure too close to the set pressure
- Corrosion or erosion of the nozzle and disk
- Solid particles between seat and disk (frequent in pump applications)
- Unsupported outlet piping
- Thermal stress on outlet piping
- Vibration of piping or vessel being protected
- Valve installed upright
- Incorrect assembly
- Incorrect lapping of seats
- Nature of the process media (f.e. hydrogen)
- Test errors in the workshop.

Smith (1995) reported that 14.5 % of a total of 5,073 valves opened at a pressure less than 90% of the set pressure. Depending on the difference between operating and set pressure, a lot were leaking. Parry (1992) reported that 27 % of a list of failures of valves were due to leakage through the valve seat, a surprisingly high figure.

Concerning the maintenance part of this phase, Hellemans (2009), reported that 8% of the 750 complaints were due to maintenance/test problems.

The usual inspection work consists of:

a) Pretest - the testing of a pressure relief valve prior to disassembly to determine the opening pressure, blowdown and seat tightness. It is very important to collect this data for optimization of the revision intervals. After the pre-pop test, an inspection is made to determine if corrosion, scaling or unusual conditions are present.
b) Disassembly - the valve is carefully dismantled following manufacturer’s recommendations. All the parts are inspected to decide the extent of repairs required.
c) Reparation - repair includes cleaning, reconditioning, replacement, lapping and minor machining of parts. All components are checked for wear, damage, roughness or corrosion. Parts that are damaged beyond tolerance are replaced or reconditioned. If evidence of wear is found on the disk or nozzle, their seating surfaces are lapped.
d) Assembly - once the necessary reparation has been completed, the valve is assembled.

e) Valve testing - testing is done to set the valve to its nameplate set pressure and blowdown, and to check seat tightness. Once the valve is on the test stand, the spring is adjusted for the last time to confirm that the valve will relieve at the required cold differential testing pressure. Once the valve has popped at the correct pressure, it is checked for seat tightness, i.e. leakage.

The reliability of pressure relief valves at a given moment can only be proved with a complete test in a testing arrangement. The probability of failure on demand of a valve increases with the time between consecutive revisions. The pre-test and internal and external inspections provide valuable information for modifying the inspection frequency. A literature survey of the recommended periods between revisions is presented in the next paragraphs.

*Revision period: methods based on experience*

Some guides and standards propose revision periods:

An essential condition has been established that in no case should the interval between pressure relief valves exceed the interval between inspections of the pressure vessels involved.

In Spain, the maximal periods are established by state law and the pressure relief valves are inspected at the same time as the elements to which they are fitted. These periods range between 6 and 12 years, depending on the vessel class (5 classes) and the level of inspection (3 levels) (RD2060, 2008). A pressure relief valve is assimilated to level B of inspection (requires the shutdown of the pressure vessel).

The UK's Institute of Petroleum, with the publication IP 12 Model Code of Safe Practice in the Petroleum Industry-Pressure Vessel Examination 1993 (Hare, 2012), established certain degrees for pressurized equipment. Degree 0 implies a revision every 24 months, Degree 1 every 36 months, and Degree 2 every 72 months. Degree 0 applies to new equipment that shows quick wear and a behavior that is difficult to predict. Degrees 1, 2 and 3 are assigned to vessels with more predictable behavior. The maximal period is 72 months.

API 510 (2006) indicates that the revision frequency should guarantee the reliability of equipment under overpressure. A typical process should not surpass 5 years, unless experience shows that this period can be extended. For clean, non-corrosive services, the period can be increased up to 10 years or more if a risk analysis is performed (for example, see the complete application of API 581 (2008)).

The Engineering Equipment Material Users Association of United Kingdom, in its publication No. 188-Guide for establishing operating period of safety valves (EEMUA, 2009), has proposed an empirical methodology based on a previous risk analysis; it classifies the materials as clean, dirty and corrosive, and establishes limiting values for deviations in opening pressure. It defines the unacceptable performance as when a valve lifts at a pressure greater than 110% of cold set pressure, lower than 90% or does not lift at 150% of cold set pressure.

The Safety Assessment Federation, in its Guideline on Periodicity of Examinations (SAFed, 2003), states that the revision period for clean fluids should not exceed 26 months. If corrosion or fouling may occur, the revision period should not exceed 14 months.

Woolfolk and Sanders (1987) recommend the following intervals: a) Storage vessels with no heat sources, every 24 months. b) Vessels processing corrosive chemicals, every 12 months.

Bravo et al. (1995) also recommended the following revision frequencies:

| Typical frequency, yr | Valve type/service |
|---|---|
| 1 maximum | Conventional valves in dirty or fouling forming services<br>Any type of relief valve with significant exposure consequences |
| 1-2 | Corrosive service<br>Relief valves protected by rupture disk with knife blade |
| 2-3 | Conventional relief valve on clean service<br>Bellows valves on dirty service |

| Typical frequency, yr | Valve type/service |
|---|---|
| | Conventional valves on water, condensate, and steam services<br>(except as regulated by local or state laws) |
| 3-5 | Relief valves for clean, dry, and noncorrosive gases<br>Relief valves protected by rupture disks without knife blade |

Nelson (1993) gives the following recommendations as a "starting point for testing frequencies based on general service conditions and past experiences with thousands of safety relief valves":

| Frequency | Service |
|---|---|
| 1-1.5 years | Dirty services such as pipe still, crude towers, cat debutanizers with nonbellows valves |
| 2-3 years | Above services equipped with bellows valves<br>Nonbellows valves in cleaner process services<br>Nonbellows valves combined with rupture discs |
| 3-4 years | Bellows type valves in cleaner process service |
| 4-5 years | Clean services such as air, natural gas, steam and some water services |

Hellemans (2009) also reports a procedure taking into account the importance of historical data for each individually installed valve. The author uses the Grade 1 to Grade 4; when the valve is installed being Grade 1 and it is revised after 1 year.

Finally, some companies apply an initial revision period of 2 years to all safety valves; this period is then suitably modified as a function of inspection results. The revisions involve partial or total programmed shutdowns of the plant.

*Revision period: methods based on risk analysis*

Some methods are based on the risk matrix (Hare, 2012) and consider the probability of failure on demand of the valve and its consequences. The recommended inspection periods (in months) can be seen in the following table:

| **Probability of failure of the valve** | **Consequence** | | |
|---|---|---|---|
| | High | Medium | Low |
| High | 24 | 36 | 60 |
| Medium | 36 | 48 | 72 |
| Low | 48 | 60 | 84 |

To estimate the probability of failure, the valve history, type of fluid, vibrations, etc. must be considered.

API 576 (2009) gives general qualitative recommendations but no specific intervals. This recommended practice adds that the manufacturers can give valuable information, especially in the case of special designs. It gives also the possibility of using risk-based techniques which recognize the fact that there are many different overpressures scenarios and that some pressure relief valve applications are more critical than others (for instance, API 581).

API RP 581 (2008) proposes a methodology for optimizing the revision period. It is based on the following steps:

1. Use of a Weibull distribution to establish the accumulated probability of failure of opening on demand. From this, the $\beta$ (shape factor) and $\eta$ (characteristic life) factors will be obtained.

2. An initial value is attributed to $\eta$, depending on the type of valve and service (Table 7.5 from API 581). This value is modified as a function of the data obtained from the pre-tests.

3. Obtaining the frequencies of failure on demand for each overpressure scenario. If own data are lacking, those from Table 7.2 in API 581 can be used.

4. Finally, the area risk thus obtained is compared with the established values, for example, 4.6 $m^2$/year for Level 2 consequence model and 0.92 $m^2$/year for Level 1. In each case, the assessment is finished with an ALARP analysis, to obtain the optimum inspection period.

Some authors have suggested that both procedures can be used additionally (Bond et al., 2011).

**Procedure in the new methodology**

Two methods have been proposed and tested. The first one is a modification of Hellemans procedure, allowing the quantification of the deficiency of the safety valve in order to correct the revision period in a more objective way (see section 6.9 for an application). The second one is a modification of the API 581 method, which is described in section 4.11and applied as case study in section 6.8.

## 3.6 Phase 6. Management of change

**Literature survey on problems detected**

Westphal and Köper (2003) stated that some deficiencies encountered in the 17% of pressure relief valves with problems found were due to changes in process related service conditions, while some others were due to piping changes upstream or downstream of the valve.

Many authors have cited the accident of Flixborough (1974) with 28 fatalities, as the best example of what can happen without a Mangement of Change (MOC) procedure implemented in a process plant. The concept of management of change comes from 1992, when OSHA promulgated the Process Safety Management (PSM) of Highly Hazardous Chemicals regulation. This PSM rule mandates a process hazard analysis, written operating procedures, employee training, pre startup safety reviews, evaluation of mechanical integrity of critical equipment and implementation of the management of change procedure, before performing certain changes. A change is defined in PSM as any alteration (except for replacement-in-kind) to process materials, processing conditions, equipment, maintenance materials, procedures, utilities, facilities, control systems, etc. In Europe, the Directive 2012/18/EU demands the application of the MOC procedure in the safety management system (point IV, annex III).

According to CCPS (2008), MOC is a process for evaluating and controlling modifications in facility design, operation, organization, or activities -prior to implementation- to make certain that no new hazards are introduced and that the risk of existing hazards to employees, the public, or the environment is not unknowingly increased.

As explained in Appendix G of the above publication, a lot of problems arose at the beginning of its implementation. However, the more forward in the learning curve the companies are, the better results they obtain.

Spearow et al. (2014) reported that there are two main problems when using MOC procedure in a relief system; the first one is identifying the impact of a MOC on relief systems being carried out by personnel without expertise, and the second one is updating all the process safety information affected by the change.

These authors give examples of common MOCs that affect relief systems: addition or removal of check valves, control valve modifications, change or re-rating the design pressure, new system pressure source or alteration of an existing one, relief piping modification, set pressure changes, etc. Concerning the complications encountered with relief systems documentation, the authors wrote: “The personnel involved do not have proper relief system training, the existing relief system documentation is inadequate or non-existent and involved personnel are unaware of the relief systems documentation”. They emphasized the need of providing proper training in relief systems to all the people involved in MOC processes.

**Procedure in the new methodology**

According to the model of Appendix C of CCPS (2008), the following document has been generated. See Figure 3-7.

## MANAGEMENT OF CHANGE RECORD

Process:___________________ Change #:___________________ Originator: ___________________

Title of Change:_____________________________________________ Date: ________________

**Section 1: Description of Change**

**Does the proposal result from a change in:**

| | Yes | No | | Yes | No |
|---|---|---|---|---|---|
| Process chemistry | | | Computer software or hardware | | |
| Raw materials or additives | | | Alarms, interlocks, or relief setpoints | | |
| Established safe operating limits (temperature, pressure, charge quantities, order of addition, or experimentation) | | | Materials of construction | | |
| Operating procedures | | | Design specifications | | |
| Additions, deletions or bypasses of equipment piping or instrumentation (including new equipment) | | | Operating state (i.e., decommissioning) | | |
| Area electrical classification | | | Other (specify) | | |

**If the answer to any question is yes, proceed with this form.**

*To consider the process HSE implications of a proposed change, the originator of the change should complete the following to ensure that the potential hazards associated with a change are identified and addressed. If a section does not apply, insert N/A in the appropriate blank. If additional comment space is required, attach pages.*

**Description and purpose of change: (attach P&ID or sketch with P&ID reference, if appropriate)**

**Technical Basis for Change:**

**Impact on product regulatory approval.**

**Assessment of environmental impact (i.e. air emissions, water discharges, solid wastes generation, etc.) due to the process change.**

**Duration of Change:**

**( ) Permanent** **( ) Temporary; will be removed by:** __________

**Yes/No**

**Is a design review (PHA) required?** ________

**Person assigned to lead review?** ____________________
**Methodology to be used (What if, Checklist, HAZOP, etc?)**____________________
**Is change in accordance with latest HSE Design Criteria?** ____________________

**Training for personnel:**

**Specify: Personnel** ___________________ **Type of Training** ___________________

**Personnel** ___________________ **Type of Training** ___________________

**Are the following required?**

| | Yes | No | | Yes | No |
|---|---|---|---|---|---|
| Changes to the SOP | | | Materials of construction | | |
| Updates to PSI | | | Relief System Design | | |
| P&IDs | | | Ventilation System Design | | |
| Process Flowsheets | | | Material and Energy Balances | | |
| Electrical Classification | | | Other | | |
| Design Standards | | | Changes to the emergency response plan | | |

**Is a Pre-Start-up Safety Review (PSSR) required?** **Yes/No** ________

**Person Assigned to Lead Review** ___________________

**Approval for Section 1 is required prior to implementation of the change.**

**Approved by** ______________________________ **Date:**______________
**Director of Area**

**Approved by** ______________________________ **Date:**______________
**HSE Manager**

**Section 2: Pre-startup Requirements**

| | Date | Verified by: | | Date | Verified by: |
|---|---|---|---|---|---|
| Design HSE review/PHA complete | | | Training in SOPs completed | | |
| Operating procedures (SOPs) revised | | | | | |

If a PSSR is required, forward this document to the person doing the PSSR once Section 1 is approved.

**PSSR completed by:** _________________ **Field inspection completed** __________

**Approval of Section 2 is required prior to start-up of change.**

**Approved for start-up by** ________________________ **Date:**______________
**Director of Area**

**Approved for start-up by** ________________________ **Date:**______________
**HSE Manager**

**Section 3: Review and Follow-up**

| | Date | Verified By: | | Date | Verified By: |
|---|---|---|---|---|---|
| Operating Procedures Updated | | | Emergency Response Plan (EPR) updated | | |
| Process HSE information updated | | | Training in above completed | | |
| Training manuals/modules updated | | | PSI Updated and PSI Form Completed | | |

**Section 4: Distribution**

Copies of this record should be distributed to Originator, Authorizing Personnel and Management of Change file.

**Figure 3-7. Management of Change procedure.**

## 3.7 Phase 7. Decommissioning

**Literature survey on problems detected**

Decommissioning in this context, means performing a review to make sure that removing a pressure relief valve from an active status will not have a negative impact on the plant or on any other surrounding units. No accidents have been reported for this phase. Operating instructions of chemical plants do not allow for removing a pressure relief valve for inspection leaving the pressure vessel with the possibility of an overpressure. The permit work management of process plants requires that the piece of equipment be isolated and purged.

An exception is the case of pressure vessels protected by a changeover valve. In this case, two valves are installed in parallel, each one designed by the required relieving load so that one of them can be removed without disturbing the process. However, one must be cautious in using these valves due to their high pressure drop.

Another possibility is the decommissioning of a pressure relief valve by way of its substitution by a High-Integrity Protection System (HIPS). As referenced in Annex E of API 521 (2008), in the large majority of cases a SIL 2 (99% availability) or SIL 3 (99.9% availability) is required for the HIPS.

Another common situation for removing a pressure relief valve in a process plant is the relief valve being installed to avoid thermal expansion in the cold side of a heat exchanger. By leaving, for example, one blocked valve carsealed open, decommissioning of the relief valve is possible.

**Procedure in the new methodology**

Prior to decommissioning of any pressure relief valve from active service, a formal analysis of the risk will be conducted, together with the required approvals of plant management through the Management of Change procedure (Phase 6).

## 3.8 Phase 8. Verification

**Literature survey on problems detected**

Grun and Cheddie (2006) referenced a report from the United Kingdom Health and Safety Executive in which 34 accidents related to control systems (not pressure relief systems) were analyzed. The majority of accidents (44%) were due to incorrect and incomplete specifications. Although this value is not related to pressure relief systems, it gives an idea of how important the verification process in a project is.

It is interesting to cite here what an expert in this field, Hellemans (2009), wrote: “ More than 25 years of experience in advising designers, end-users and maintenance people in the selection, handling and maintenance of safe safety relief systems, together with independent studies, have shown that more than half of the pressure-containing equipment installed in the process industry has a small to serious pressure relief system deficiency as compared to widely accepted engineering practices and even legal codes. The types of deficiencies are roughly split between absent and/or undersized pressure relief devices, wrongly selected valves and improperly installed valves”.

Verification, in this context, means the process of demonstrating by review, analysis, and/or testing that the outputs of each phase of the life cycle analysis satisfy the requirements. A Safety Verification Plan manages changes in the Safety Requirements Specification (Phase 2) during project implementation as well.

**Procedure in the new methodology**

A checklist will be used in each phase of the safety life cycle. The checklist is an attempt to list as many recommendations from procedures, codes, guidelines, etc. with the target that, by following a systematic review of an overall relief system project, nothing will be forgotten.

## Phase 1. Risk analysis

| Item No. | Item | Mark a choice | | | Comments |
|---|---|---|---|---|---|
| 1.1 | Have persons responsible for carrying out the risk analysis been identified and informed of their responsibilities? | Y | N | N/A | |
| 1.2 | Are persons competent to perform the risk analysis? | Y | N | N/A | |
| 1.3 | Is personnel competency documented in terms of knowledge, experience, and training? | Y | N | N/A | |
| 1.4 | Has a PHA been performed? | Y | N | N/A | |
| 1.5 | Has the PHA followed an approach equipment-based considering all the scenarios of the contingency analysis data sheet (figure 3-2)? | Y | N | N/A | |

## Phase 2. Safety Requirements Specification

| Item No. | Item | Mark a choice | | | Comments |
|---|---|---|---|---|---|
| 2.1 | Is there a clear and concise description of the scenarios that affect each pressure relief valve, according to the contingency analysis data sheet (figure 3-2)? | Y | N | N/A | |
| 2.2 | Is the relieving loads summary data sheet (figure 3-3) completed with the relieving loads (dimensioning and not dimensioning) for each safety valve? | Y | N | N/A | |
| 2.3 | Is the safety valve requirement specification completed (figure 3-5), with the information available at that moment, for each valve? | Y | N | N/A | |

## Phase 3. Design

| Item No. | Item | Mark a choice | | | Comments |
|---|---|---|---|---|---|
| 3.1 | Has a file been opened for each pressure relief valve (paper based or through dedicated software)? | Y | N | N/A | |
| 3.2 | Is the following documentation available for each valve?<br>- Relief load calculation for each scenario of the relieving loads summary data sheet<br>- Selection and design (required an rated area) of the safety valve<br>- Inlet pressure drop<br>- Outlet pressure drop<br>- Stability calculations<br>- Forces and moments imposed to the safety valve<br>- Fluid induced vibration (vortex shedding)<br>- Acoustic induced vibration<br>- Noise<br>- Body bowl chocking<br>- Assignation of the first revision interval. | Y | N | N/A | |

## Phase 4. Reception, installation and checking

| Item No. | Item | Mark a choice | | | Comments |
|---|---|---|---|---|---|
| 4.1 | Has the vendor supplied all the information requested in the safety valve requirement specification (figure 3-5) including the spare parts? | Y | N | N/A | |
| 4.2 | Has the pre start-up safety review checklist been applied (Annex II)? | Y | N | N/A | |
| 4.3 | Have montage personnel received appropriate training? | Y | N | N/A | |
| 4.4 | Have adequate precautions been taken for storage of items before installation? | Y | N | N/A | |
| 4.5 | Is there documentation showing the following?<br>-Identification of each safety valve in field and in the Piping and Instrument Diagram<br>-Confirmation that the reception of the valve has been | Y | N | N/A | |

| Item No. | Item | Mark a choice | | | Comments |
|---|---|---|---|---|---|
| | successfully completed, including set pressure test<br>-Authorized signatures indicating the safety valve was successfully installed.<br>-Training of operations personnel successfully carried out before start-up. | | | | |

Phase 5. Operation, maintenance and revision

| Item No. | Item | Mark a choice | | | Comments |
|---|---|---|---|---|---|
| 5.1 | Have operators received training in order to detect leakages in the safety valves? | Y | N | N/A | |
| 5.2 | Has an initial revision interval been assigned to each new pressure relief valve? | Y | N | N/A | |
| 5.3 | For already installed safety valves, is there a quantitative procedure of interval revision, depending on the results of the inspection and the pretest (like API 581) or more empirical methods? | Y | N | N/A | |
| 5.4 | Is there a tracking procedure for dismantling the safety valve, revision in the workshop and mounting it again, in order to avoid exchange of valves? | Y | N | N/A | |
| 5.5 | In the case that the pretest has given opening pressures higher than 10 % of the set pressure, has the Management of Change procedure immediately been applied? | Y | N | N/A | |

Phase 6. Management of change

| Item No. | Item | Mark a choice | | | Comments |
|---|---|---|---|---|---|
| 6.1 | Has the Management of Change procedure (figure 3-7) been used and recorded? | Y | N | N/A | |
| 6.2 | Has the process safety engineer followed the execution of the Management of Change recommendations and closed it when finished? | Y | N | N/A | |

Phase 7. Decommissioning

| Item No. | Item | Mark a choice | | | Comments |
|---|---|---|---|---|---|
| 7.1 | Has the Management of Change procedure always been applied when a pressure relief valve has been eliminated? | Y | N | N/A | |
| 7.2 | Are there procedures to maintain the safety of the process during decommissioning? | Y | N | N/A | |
| 7.3 | Are there procedures that define the level of authorization required for decommissioning? | Y | N | N/A | |

Phase 8. Verification

| Item No. | Item | Mark a choice | | | Comments |
|---|---|---|---|---|---|
| 8.1 | Has a pressure relief valve verification plan been created, following the recommendations of this section and including responsibilities and signature of the authorized personnel? | Y | N | N/A | |

Phase 9. Documentation and technical audits

| Item No. | Item | Mark a choice | | | Comments |
|---|---|---|---|---|---|
| 9.1 | Has each pressure relief valve got a dedicated file (paper based or specific software) with all the information listed in section 3-9? | Y | N | N/A | |
| 9.2 | Have all the findings of the last relief system audit already been solved? | Y | N | N/A | |

## 3.9 Phase 9. Documentation and technical audits

**Literature survey on problems detected**

One of the four conclusions of Berwanger et al. (2000) report was that “Current information management techniques have not worked”. They concluded that companies had focused on the pressure relief valve data sheet as the basis for all the required documentation for the relief system, a purpose for which they had never been intended.

Westphal and Köper (2003), also pointed out that one problem in their work of revision of 4000 valves was, particularly for old plants, that the design and the exact reasons for its need had not been adequately documented.

Marshall et al. (2011), described that the results of the 2007 and 2009 OSHA’s National Emphasis Program specific to refineries and chemical plants, respectively, demonstrated that a large number of citations involved missing, inaccurate and incomplete process information as well as outdated relief system studies.

Melhem (2010) remarked on the “Outdated, different formats, and/or non-existent design basis and supporting calculations” in his presentation in DIERS. This author also stated that “Relief system design is 30% calculation and 70% data life cycle management”, and introduced the features that the management of these data should include: Reliability, Availability, Auditability, and Maintainability (RAAM)$^{TM}$. Conditions that its own company software product ioXpress$^{TM}$ fulfills.

In order to keep all the relief system data updated, it is necessary to have regular audits. In fact, this is one of the recommendations in OSHA,s Process Safety Management Programs. These audits are a method of certifying that a plant is well protected against overpressure by existing pressure relief devices. Equally important, these evaluations identify deficiencies and propose required changes; with them, the company can meet the requirements of codes.

Concerning the audits, Marshall et al. (2011), recommended the Three-Tier auditing approach:

Tier I. These are audits performed each month, for example with the local process safety engineer. They consist in reviewing the MOC workflow, as well as the design basis for randomly selected pressure relief valves.

Tier II. These audits use relief systems specialists from other plants or engineering consultants with expertise in relief system design. They should be performed at least annually.

Tier III. These audits use relief systems specialists at the corporate level or engineering consultants experts in this area. They should be conducted at least every three years.

Wong (1998) insisted on the fact that the most important issue for a successful audit is the competence and experience of the audit team and the exact definition of the scope of the audit before performing it.

**Procedure in the new methodology**

API 521 (2014) and Prophet (2015) give recommendations on the required documentation for pressure relief system design, although stating that they are not mandatory.

In this work, the documentation has been collected according to the following schema.

A. Relief System Design Documentation

1) Guideline for the engineering of pressure relief systems. This guideline will give specific procedures for the following issues, not explicitly stated in API 521-2014:
    - Method for identification and definition of credible causes of overpressure
    - Guidance for evaluating control valve bypasses

- Credit for operator intervention, e.g. to stop a liquid overfilling situation
- Credit for check valves and sizing procedures for backflow through pumps
- Credit for vessel insulation and adequate drainage for the fire case
- Determination of physical properties (e.g. heat of vaporization of mixtures)
- Fire sizing for air coolers
- Required pump impeller size to use in blocked discharge scenario
- Control valve trim size
- Credit for sealed or locked open isolation valves in the pressure-relief path
- Guidance related to double jeopardy
- Credit for protective instrumentation
- Guidance on heat exchanger internal failure scenarios
- The use of rated flow vs. required flow for inlet pressure drop of modulating valves
- The use of 3% requirement for inlet pressure loss vs. use of blowdown minus 2% for new and old installations
- The correct usage of a two-phase discharge coefficient
- How to estimate two-phase density where slip is involved
- The use of fire dynamic simulation considering the decreased wetted area with time
- The total use of vessel surface area in case of fire for two-phase flow
- Treatment of fire exposure of gas filled vessels.

2) A chronological record of the file including all reviews, evaluations, design modifications, etc. which have been performed on the pressure relief valve.
3) A sketch of the pressure relief device and the system in which it is installed. This sketch should show all equipment protected by the pressure relief valve and the corresponding design conditions for the equipment. The P&ID could also be used.
4) Contingency analysis summary, with specific data sheet (see 4.1).
5) Relief load analysis and sizing calculations with specific data sheet (see 4.2).
6) Safety Requirements Specification (specification sheet) for the valve and sizing computations.
7) Isometrics of the inlet and outlet piping (always as-built isometrics).
8) Calculations performed to size the pressure relief valve inlet and outlet piping.
9) Forces and moments imposed on the valve.
10) Stability calculations to avoid chattering.
11) Calculations to check for Acoustic Induced Vibration (AIV).
12) Calculations of noise.
13) Checking of the Body Bowl Chocking.
14) Assignation of the revision interval.
15) Collection of all manufacturers data for spare parts purchasing, maintenance instructions, Purchase Order Requisition, etc.

B. Flare Header Design

1) Guideline for the engineering of flare header design.
    - For each flare header scenario, a description of the initiating event and the intermediate consequences that lead to relief flow is needed. For example, for total power failure the cascade effects should be described
    - The use of rated vs. required flow for calculating the backpressure and the sub-header/flare header pressure drops
    - Documentation of the basis used to define the flare system configuration for the network-flow simulation model
    - Schematic diagram of the flare system showing a pressure profile for each flare header scenario analyzed. The pressure profile will show calculated backpressure for each relief source discharging in the given scenario
    - Electronic copies of input files used for the network simulation
    - PRV size-selection datasheets showing valve manufacturer (for existing valves), type of valve, set pressure, size and inlet and outlet flange ratings

- List of disposal system loads including source name, temperature, molecular weight, composition and flow rate.
- List of all credit taken to reduce or eliminate disposal system peak loads, including instrumentation.
- List of instrumentation assumed not to work for each relieving scenario and the basis for selection of failure combinations
- Backpressure limit for each source and basis for limit (e.g. downstream piping design pressure).

The audit schema will follow the structure shown in the following table.

<table>
<tr><th>Pre-audit activities<br>-by plant management</th><th>Detailed audit activities<br>-by audit team</th><th>Post-audit activities</th></tr>
<tr><td>Management defines the units for auditing and its schedule.<br>Management selects audit team (internal or external).<br>Management defines audit subject and priorities.</td><td>Define audit scope:<br>1. Design basis, codes, regulations.<br>2. Responsibilities for field verification and inspection of existing PRV system.<br>Define audit activities:<br>1. Data gathering, generating.<br>2. Review previous audit reports and incident history.<br>3. Study the existing PRV maintenance log.<br>4. Overpressure cases analysis and PRV selection.<br>5. Evaluating the existing PRV system.<br>6. Report findings.<br>7. Conclusions, recommendations and options, etc.</td><td>Documentation including:<br>1. Report of overpressure case studies.<br>2. Proposal of implementation program.<br>3. Basic required maintenance schedule.<br>Develop a feasible action plan:<br>1. Resolve the differences between audit team and management.<br>2. Propose a feasible action plan.<br>3. Define a practical item schedule.<br>4. Elect a monitoring team.<br>Follow-up until the next schedule audit.</td></tr>
</table>

## 3.10 Commercial software available. Critical review

Kreder and Berwanger (1995) wrote “It is difficult not to visualize a day when essentially all process safety information will be maintained in an intelligent electronic database”. Certainly, the introduction of the hardware and software applications in companies has grown exponentially and will continue.

Melhem (2010), according to his classification of the relief systems data lifecycle (see section 3.9), classified the relief system calculation and documentation software tool as follows:

| Relief systems data lifecycle solution | Reliability | Availability | Auditability | Maintainability |
|---|---|---|---|---|
| Home grown tools such as spreadsheets and simple applications | Med | Med | Low | Low |
| Commercial applications mostly geared at calculations | High | Med | Low | Low |
| Commercial database systems incorporating simple computational tools | Low | Med | Med | Med |
| Commercial data management systems linked to external computational tools | High | Med | Med | Med |
| Knowledge management systems with integrated computational tools | High | High | High | Med |
| Knowledge management systems with integrated computational tools and workflow | High | High | High | High |

All these possibilities can be found in different companies. The choice depends, among other factors, on the number of pressure relief valves in the site. A large refinery can have more than 1,000 relief devices; a medium petrochemical company can have 200-400 relief devices.

The software codes available for relief system calculations and documentation are:

**Superchems™**

This software comes from ioMosaic Company. IoMosaic was founded by a pioneer in process safety, Arthur D. Little. Superchems is an umbrella-software for a lot of process safety issues, but it has specific modules for pressure relief valves, having also pioneered the possibility of calculating the dynamic stability of pressure relief valves. Today, this company is managed by Dr. Melhem, one of the most recognized worldwide experts in relief systems. The relief system documentation is managed with the module ioXpress.

**PS PPM™**

This software was developed by the formerly company Berwanger Inc. Today is part of the Siemens AG.

The Process Safety Pressure Protection Manager (PS PPM) was built as a relational database architecture for integration with other engineering and plant information. This software has been used in this thesis in the calculation of the latent heat of mixtures for the scenario of fire, relieving loads in case of heat exchanger tube rupture, and inlet pressure drops. However, it does not incorporate the DIN norm for diameters of the piping and fittings; this makes the work harder, because one must find the schedule from the ASME B31.3 norm that matches the real inside diameter.

**iPRSM™**

This software is offered by Farris Engineering Services. It is a web-enabled pressure relief systems management software, i.e. data accessibility from any web connection. This software has a lot of calculation possibilities and integrates with documentation as well.

**PSVPlus™**

This software was developed by Softbits Consultants Ltd from the UK. In 2012, this software was acquired by Aspen Technology. PSVPlus allows the preparation of the specification sheets of the safety valves performing the calculations of the relieving load and the orifice of the valve as well.

**Aspen-Hysys™**

The version 8.6 of the Aspen-Hysys incorporates new possibilities for sizing pressure relief valves, including supercritical fluids with the Ouderkirk (2002) method among other features. It allows integrating the documentation as well. The results are easily available for the Aspen Flare System Analyzer™.

**Aspen Flare System Analyzer™**

This software (formerly Flarenet) belongs to Aspen Technology. It performs the calculations of the total backpressures of the relief devices in a rigorous way. It has been used in this thesis to check the backpressures of the valves, together with an own developed spreadsheet.

**Valvestar™**

Valvestar is a free software to design Leser Valves. The version 7.1.4 has the possibility of designing for two-phase flow according the Omega method from API 520. It suffers from lack of flexibility when there is a restriction lift in the valve.

**PRV2SIZE™**

This software was generated by Pentair (the formerly Crosby and Anderson-Greenwood belong to this group). It has all the data of Crosby and Anderson-Greenwood valves.

# Chapter 4. Detailed development of the design phase

The existing criteria concerning the diverse aspects of the design of pressure relief valves (relief load, type of flow, type of valve, stability, etc.) are commented here, and the most convenient ones are selected for the calculations associated with the new proposed methodology.

## 4.1 Relief load analysis (contingency analysis)

A contingency analysis consists of the calculations, analysis and reasoning, which are performed to identify potential relief cases and determine the required relief loads for each case. The relief load calculations may consist of any or all of the following aspects:

a) Detailed process calculations and analysis done to fully develop required loads for sizing cases, or to analyze those cases which could not be eliminated as potential sizing ones by simpler analysis methods.
b) Simplified process calculations performed to demonstrate that a particular case is not the governing one for pressure relief valve sizing.
c) Qualitative analysis or reasoning demonstrating that a specific potential case is either not a valid case or that it is not a governing one.
d) Identification of specific operating procedures or equipment limitations agreed to in determining pressure relief system size.

Before explaining the framework developed to manage the contingency analysis, it is appropriate to describe the Basic Design Philosophy.

### 4.1.1 Basic design philosophy

- Because of the design margins inherent in the sizing and specification of process equipment, the process plants are often capable of operating at feed rates significantly above their original nameplate capacity. The guidelines used in this thesis for establishing equipment capacities are the following ones:
    - The relief requirements of distillation columns have been at 85% of theoretical tray flooding or at the limiting reboiler or condenser duty, if the latter is a more severe constraint
    - The centrifugal pumps have been evaluated using both installed and maximum impeller sizes. Anyway, if the maximum impeller resulted in a size increase for the required pressure relief valve, the decision has been studied on a case by case basis, for instance, establishing administrative controls avoiding the change of the impeller
    - The evaluation of the compressors has been made at the maximum molecular weight of the fluid, within the power capabilities of the motor. The compressor efficiencies are high estimates, not guaranteed manufacturers' efficiencies
    - For heat exchangers the maximum duties have been calculated under clean, unfouled conditions. However, duties for condensers and other heat removal equipment have been calculated for the fouled condition when applying credit to a particular scenario.
- The Double Jeopardy concept is very important and its application is "no double jeopardy in overpressure assessment" (Wong, 1999). Causes of overpressure, including external fire, have been considered as unrelated if no process, mechanical, or electrical linkages existed among them, or if the length of time between possible successive events was sufficient to regard them as unrelated

- The case of total power failure also means cooling water failure because there are no turbine-driven cooling water pumps. In this way, no credit has been given to the automatic start-up of parallel cooling water pumps, air compressors, etc. which are not already operating
- The variations in composition of the fluid relieved during the relieving event have also been considered. For instance, in a blocked in fire scenario in which the vessel contains a wide boiling liquid mixture, the initial relief vapors will consist primarily of the lower boiling components, but as material is removed, the molecular weight and temperature will increase, while the latent heat decreases. Another case that affects the equilibrium is when very large differences between normal and relieving pressures occur. In both examples, a sensitivity analysis is required to determine the worst case set of conditions for pressure relief valve sizing
- Every block valve has been considered subject to inadvertent opening or closure at any time. The only exceptions were the locked-open and locked-closed valves, which have been assumed to remain in the locked position
- Check valves have been considered to fail in the full open position due to mechanical debris or chemical materials preventing the valve from reseating on flow reversal. However, double check valves of different designs in a series may reduce flow reversal if installed in clean vapor upflow service. Nevertheless, the reduction in flow for double check valves cannot exceed half of the reverse flow if no check valves are present
- When bypass valves are provided for control valves, the potential for inadvertent operation of both valves must be considered. Only if the bypass valve is used exclusively for non-standard operation may it be excluded from relief load calculations. The bypass valve must be smaller (less than 25% of the control valve) (Cheremisinoff, 1998). Operational procedures must clearly indicate its use and ensure that the valve is never open during normal operation
- As the response of control systems is uncertain, no credit has been given to the favorable response of automatic controls. In any relieving situation the control valves which are not under consideration as the cause of overpressure, and whose regular automatic action would tend to reduce the relief load, remain in the position required for normal process operation. It has been accepted to give credit for continued flow through these valves, corrected to relieving conditions, but only to the extent permitted by their normal operating position. The downstream system must also be capable of handling any increase in flow
- API 521 (2014) allows for using the operator intervention procedure to avoid a relief incident if an operator actuates to cut feed, reduce the input, etc. Some authors pointed out that 30 minutes is an adequate time between operator notification through control room alarm, for example, and the time to reach the pressure relief valve set point. However, Wong (1999) warned about the underestimation of human error and, generally no credit should be given. In this thesis no credit has been given to operator intervention
- The administrative procedures are accepted by the codes (API 521, 2014); however, assuming that the possibility of human error cannot be completely eliminated, they should not be used in place of a pressure relief valve. An example is taking credit for the “car seals open, CSO” and “car seals locked, CSL” valves. Here, credit has been taken under the condition that administrative controls are in place.

Figure 4-1shows the contingency analysis data sheet used in this work and Figure 4-2 shows the relieving loads summary data sheet associated to each contingency; both figures concern the pressure relief valve YS702/01.

| Contingency | | Comments | Justification |
|---|---|---|---|
| 1 | Blocked outlets | Not applicable | If a valve is blocked downstream W700, the instruments PZ70220 and PZ70221 will shutdown P700A/B |
| 2 | Abnormal heat input | Not applicable | |
| 3 | Exchanger tube breakage | Not applicable | Propylene circuit working pressure (40 barg) is higher than cooling water working pressure (4barg). If an exchanger tube breakage occurs propylene will go into cooling water system and detected by means of hydrocarbon analyzers in cooling towers |
| 4 | Auto control failure | Not applicable | |
| 5 | Reflux failure | Not applicable | |
| 6 | Fire | See attachments | |
| 7 | Cooling water failure | See attachments | |
| 8 | Power failure | Not applicable | Shutdown of P700A/B. No consequences |
| 9 | Instrument air failure | Not applicable | Instrument air failure closes inlet valve from K702A/B (H70102 and H70104) and opens the by-pass valve FVK70201. Overpressure in this case is not possible because no ingress of material is produced |
| 10 | Inadvertent VA open/close | Not applicable | The situation that manual by-pass valve on W700 is opened and no propylene is cooled is similar to cooling water failure scenario |
| 11 | Mech. Equip. Failure | Not applicable | Design pressure = 45 barg. PZ70220/PZ70221 set pressure at 45 barg |
| 12 | Heat loss (series frac.) | Not applicable | |
| 13 | Thermal | See attachments | Thermal expansion will be considered taking into account a $\Delta T = 25$ ºC starting at 20 barg. ($\Delta T$ between night and day) |
| 14 | Loss of quench/cold feed | Not applicable | |
| 15 | Chemical reaction | Not applicable | |
| 16 | Steam out | Not applicable | |

**Figure 4-1. Contingency analysis data sheet for YS702/01.**

| | | | |
|---|---|---|---|
| TAG/EQUIP. NUMBER **SV 702/01** | UNIT /SERVICE: **Propylene purification unit** | P&ID: | PLANT: COST CENTER: |
| EQUIPMENT PROTECTED: | | SET PRESS: **45** BARG | BASIS: **List basis for set pressure** |
| | | DISCHARGE DISPOSITION: **Flare** | INLET PRESSURE DROP: **1,4% SP** |
| | | CONSTANT BACKPRESSURE: **0,150** BARG | VARIABLE BACK PRESS.: **0** BARG Kd = **0,800** |
| EQUIPMENT DESIGN CONDITIONS: ( ) MAWP ( X ) Design ( ) Other | | BUILT-UP BACKPRESSURE: **3,29** BARG | TOTAL BACKPRESSURE: **3,435** BARG Kb = **1,00** |
| NORMAL OPER. **16,2** BARG **42** ºC | Rupture Disk ,Y/N **No** | FIRE SUMMARY WETTED AREA: **14,41** m² | ATTACH SKETCH FOR AREA CALCULATION: |
| MAX OPER. **45** BARG **100** ºC | Derating Factor= **1,0** | INSULATION **No** TYPE **-** THCKNS **-** mm | Insul factr, 1=none **F =1.0 (4)** |
| DESIGN **45** BARG **100** ºC | (Use 0.9 if have rupture disk) | Q = **1.386,4** kw | (Note : see back up material) |
| CONN: RATING FACING : **PN100 F.E /PN40 F.C** | PIPE SPEC, IN/OUT: **St860E-A** / **St261C-E** | | |

| Causes of Relief<br>Refer to API RP520, & RP521 and corporate Relief Manual<br>Contingency | Comments NA, etc | % OV PR | RELIEF LOAD<br>VAPOR kg/h | LIQUID m3/h | RELIEF (2) CONDITIONS<br>PRESS BARG | TEMP °C | FLUID PHYSICAL PROPERTIES AT RELIEF CONDITIONS:<br>FLUID TYPE | VAPOR MOL WT | SP GRAVITY LIQUID | COMPR FACTOR Z | LATENT HEAT L, KJ/kg | SP HEAT RATIO k | LIQUID VISC cP | VAPOR VISC cP | VAPOR AREA V mm² | LIQUID AREA L mm² | TOTAL AREA T mm² |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1. BLOCKED OUTLETS | **Not applicable** | | | | | | | | | | | | | | | | |
| 2. ABNORMAL HEAT INPUT | **Not applicable** | | | | | | | | | | | | | | | | |
| 3. EXCHANGER TUBE BREAKAGE | **Not applicable** | | | | | | | | | | | | | | | | |
| 4. AUTO CONTROL FAILURE | **Not applicable** | | | | | | | | | | | | | | | | |
| 5. REFLUX FAILURE | **Not applicable** | | | | | | | | | | | | | | | | |
| 6. FIRE | **See attachments** | **10** | **10.000** | | **49,5** | **100** | **Vapor** | **42,00** | | **0,70** | | | | **0,080** | | | **221 (3)** |
| 7. COOLING WATER FAILURE | **See attachments** | | | **10,2** | **49,5** | **45-92** | **Liquid** | **42,00** | **0,327** | | | | | | | | **38,3 (5)** |
| 8. POWER FAILURE | **Not applicable** | | | | | | | | | | | | | | | | |
| 9. INSTR. AIR FAILURE | **Not applicable** | | | | | | | | | | | | | | | | |
| 10. INADVERTENT VA. OPEN/**CLOSE** | **Not applicable** | | | | | | | | | | | | | | | | |
| 11. MECH. EQUIP. FAILURE | **Not applicable** | | | | | | | | | | | | | | | | |
| 12. HEAT LOSS (SERIES FRAC.) | **Not applicable** | | | | | | | | | | | | | | | | |
| 13. THERMAL | **AT=25ºC 15 barg** | **10** | **Negligible** | | **49,5** | **27** | **Liquid** | **42,00** | **0,500** | | | | | | | | |
| 14. LOSS OF QUENCH/COLD FEED | **Not applicable** | | | | | | | | | | | | | | | | |
| 15. CHEMICAL REACTION | **Not applicable** | | | | | | | | | | | | | | | | |
| 16. STEAM OUT | **Not applicable** | | | | | | | | | | | | | | | | |
| 17. | | | | | | | | | | | | | | | | | |
| 18. | | | | | | | | | | | | | | | | | |
| 19. | | | | | | | | | | | | | | | | | |
| 20. | | | | | | | | | | | | | | | | | |

NOTES:
- -1 SEE RELIEF DEVICE DATA SHEET FOR ACTUAL ORIFICE, COLD DIFF TEST PRESSURE AND OTHER SPECIFICATIONS
- -2 RELIEF CONDITIONS ARE AT SET PRESSURE + OVER PRESSURE.
- -3 See rigorous calculation with the Ouderkirk method (back up material)
- -4 Insulation was not considered in the Fire Case calculation
- -5 Calculation performed by Direct Integration (API 520 Part I-2008, procedure C.2,1) (Back up material)

| | | |
|---|---|---|
| GENERAL DATA | BY: | DATE: |
| PROCESS DATA | BY: | DATE: |
| VALVE SIZING | BY: | DATE: |
| CHECKED/APPROVE | BY: | DATE: |

EXISTING RV DETAILS: **LESER 4564.6052**

| | | | | | |
|---|---|---|---|---|---|
| SIZING CASE SELECTED: **FIRE** | | RELIEF DEVICE TYPE: / | TOTAL ORIFICE AREA REQD: **221** mm² | | |
| DEVICES SELECTED - | QTY: **1** | INLET SIZE: **25,0** mm | OUTLET SIZ **50,0** mm | ORIFICE/AREA (1): **314,160** mm² | SET PRES: **45** BARG |
| | QTY: | INLET SIZE: mm | OUTLET SIZE: mm | ORIFICE/AREA (1): mm2 | SET PRES: BARG |
| | QTY: | INLET SIZE: mm | OUTLET SIZE: mm | ORIFICE/AREA (1): mm2 | SET PRES: BARG |
| | QTY: | INLET SIZE: mm | OUTLET SIZE: mm | ORIFICE/AREA (1): mm2 | SET PRES: BARG |

**Relieving Loads Summary Data Sheet**

**Figure 4-2. Relieving loads summary data sheet for YS702/01.**

API 521 (2014) gives an explanation of what is included in each contingency. Wong (2001) explained each scenario supplying more detailed information than API.

### 4.1.2 Specific design basis

The specific design basis used for each contingency in this thesis is detailed in the next paragraphs.

1) Blocked outlets
   The most obvious overpressure situation is the case of a vessel with blocked outlets, in which fluid source pressure exceeds design pressure. The source pressure may be the operating pressure in another vessel, it may be generated by a pump or compressor, or it may be an energy input, generating a relief requirement via thermal expansion.
   Generally, the pressure relief valve must release all fluid entering the vessel under relief conditions. If not all exits are blocked, credit may be taken for flow through the unblocked outlets, with the pressure relief valve handling the surplus. If the outlets contain control valves, it is assumed that they remain in the position required for normal process operations. It is acceptable to take credit for continued flow through these valves -corrected to relieving conditions- but only to the extent permitted by their normal operating position. The downstream system must also be capable of handling any increase in flow.
   If a vessel has a vapor space with at least 15 minutes residence time above the high level alarm, liquid relief for blocked outlets scenario has not been considered. This is the case with the bottom liquid in distillation columns. On the other hand, if vapor accumulation appears immediately as increasing pressure; once the set pressure is reached, all net vapor inlets must be relieved.

2) Abnormal heat input
   Abnormal heat input is a special case of control valve failure involving the flow of fuel or heating medium to process heat transfer equipment. The limitations on heat input are either those imposed by the maximum heat transfer capabilities of the equipment involved, or the hydraulic capacity of the system supplying the fuel or heating medium.
   For reboilers, the behavior of the condensate system may regulate the possible maximum duty. If the condensate system has less capacity than the steam supply system, reboiler flooding can result, which would reduce reboiler duty. In this thesis no credit has been given to this possibility, because it is believed that the operator can bypass the steam trap when the flooding is detected.

3) Exchanger tube breakage
   This case corresponds to the flow across a broken tube, which cannot be absorbed by the low pressure side.

4) Auto control failure
   Control valves are subjected to inadvertent full opening or closing during normal operation, regardless of the failure position of the valve on the loss of air signal or the loss of instrument power. Operator error or a partial failure in a control loop may cause a control valve to fully open even if it has a fail closed action. When bypass valves are provided for control valves, the possibility of inadvertent operation of both valves has been considered.
   In some cases, failure of a control valve to a full-open or full-closed condition results in an unreasonably sized pressure relief valve. In these cases, the installation of limit stops to restrict the capacity or to limit the minimum flow of the control valve may be considered. Some basic requirements relative to limit stops are:
   - Before limit stops are considered, reduction of valve capacity should be examined. The trim can be changed to reduce capacity
   - When installed, limit stops should be nonadjustable and permanently installed. They should either be installed in the valve trim or inside the valve actuator
   - Administrative procedures must be implemented to prevent removal of the limit stops or modification of the control valve capacity without first evaluating the effect of the change on pressure relief valve requirements.

Often the relief load resulting from an inadvertent control valve opening can be reduced by installing a restriction orifice or a section of small diameter in the piping which supplies material to the system. As with limit stops, any restriction orifices installed to reduce relief loads must have proper administrative procedures to ensure that the orifices are always present and are not modified without the effect on the pressure relief valves being evaluated.

5) Reflux failure
With few exceptions, overhead cooling and/or reflux failure is a major relief case, if not the controlling case. This "loss of reflux" may be caused by either a loss of coolant, a pump failure, or a control valve failure. Depending on the specific system configuration, each of those causes could produce different relieving flows, because of variations in secondary effects and available credits. Completion of the corresponding upset heat balance will determine the unbalanced heat flow on which the relief load is based. If the failure is such that the reflux drum and condenser are assumed to be flooded, the net vapor product, if any, will be unable to reach its normal exit and will have to be relieved by the valve.
Reduction or elimination of reflux failure relief is possible if there is clearly sufficient surge capacity in the reflux drum to cover the duration of the upset. The following criteria have been applied here based on the Cheremisinoff analysis (1998):
   - If the failure is due to loss of coolant and the reflux drum has at least 15 minutes surge time below low liquid level (based on total outflow), credit may be taken for the continued flow of reflux to the column in reducing the calculated relief load
   - If the reflux failure is due to malfunction of a pump or control valve, credit may be taken for continued heat removal via the condenser, provided the reflux drum has at least 15 minutes surge time above high liquid level based on total liquid inflow. If some liquid outflow continues, e.g., if product flow is maintained, this determination should be based on net inflow.

6) Fire
Fire represents an unexpected energy input into a system, which results in overpressure either by thermal expansion or vaporization of the contained fluid. The definition of the basic fire case assumes that the equipment is exposed while fully blocked, with normal maximal liquid contents intact. In the static, non-flowing fire situation, all heat input goes to the generation of relief vapor once the content has been heated sufficiently to reach the relieving pressure.
Fire occurring during normal operation should also be considered. In this dynamic case, the flowing fluid often absorbs the sensible heat pickup without resultant overpressures.

7) Cooling water failure
Loss of cooling water causes an energy imbalance, which can produce major relief loads. A relieving situation may not result if only sensible cooling is lost, although downstream equipment could be subjected to overpressure through unexpected flashing. If condensation is lost, however, the excess vapor will generally have to be relieved. Note also that any condition which leads to loss of circulation on the hot side of the exchanger, such as condenser flooding, is also equivalent to a total loss of cooling.

8) Power failure
In case of power failure the following equipment can be potentially affected: motors, pumps for circulating cooling water, reflux pump in distillation columns, air-cooled heat exchangers, cooling towers, instrument air, instrumentation, motor operated valves, etc.
API 521 (2008) discusses total and partial power failure, noting that credit for automatic start-up of parallel cooling water pumps, air compressors, etc., which are not already operating, should not be taken.
The interdependence of utility systems must be considered. In case of power failure, a plant which has two motor-driven and two turbine-driven cooling water pumps in service would also undergo a 50% loss of cooling water.

9) Instrument air failure
Instrument air failure may be caused by power failure or mechanical equipment failure, and can affect all the control valves of the plant. Usually the design of the plant allows 15 minutes of air reserve in case of an upset in the air compressor. Instrument air failure causes that control valves assume their fail safe position (fail-open, fail-closed or fail-last position). As pointed out by Wong (1999), the air fail position should not be taken as overpressure relief protection.
Be aware that this case is not the same as auto control failure as explained before. Auto control failure affects only one control valve (fails fully open or fully closed).

10) Inadvertent valve open/close
Every block valve is considered to be subject to inadvertent opening or closure at any time. The only exceptions to these requirements are locked-open or locked-closed valves, which are assumed to remain in the locked position.
Check valves may fail in the fully open position due to mechanical debris or chemical materials preventing the check from reseating on flow reversal. Tight shutoff can never be guaranteed. However, double check valves of different designs in a series may reduce flow reversal if installed in clean vapor upflow service. API 521 (2008) allows the reduction in flow for double check valves as the flow through a single orifice with a diameter equal to one-tenth of the largest valve's nominal flow diameter.

11) Mechanical equipment failure
Dynamic mechanical equipment consists of centrifugal pumps, centrifugal compressors, reciprocating pumps, reciprocating compressors, screw compressors, roots blowers, steam turbines, etc. All this equipment is frequently a source of overpressure, and it is necessary to understand their performance characteristics in order to predict the resultant relief load.
Static mechanical equipment consists of heat exchangers, pressure vessels, piping, etc.

12) Heat loss (series fractionating columns)
When multiple distillation columns are disposed in series, it is possible that an upstream failure be the cause of relief by altering the composition of process streams. For example, if reboiler heat were lost in an upstream column, which thus prevents the removal of light ends from the feed to a second column, the latter could be overpressurized by an inability to condense all the light material it was receiving.
For relief purposes, the first approximation would be to consider all material lighter than the normal feed as non-condensable, which must be vented or relieved. If a more detailed analysis is desired, the condenser performance may be reevaluated based on the revised composition at the relieving pressure.

13) Thermal
When a liquid filled system is blocked in, any heat input will result in overpressure due to thermal expansion (sometimes called hydraulic expansion). If the heat source is capable of vaporizing the trapped fluid, overpressure can result whether or not the system runs liquid full. The heat source may be process heat input, steam coils or tracing, solar radiation, radiant heat from nearby hot equipment or exposure to an external fire.

14) Loss of quench/cold feed
Loss of quench is analogous to loss of reflux, except that quench usually refers to a stream from an external source which is used in a tower for direct contact vapor cooling. The steady state relief in this case would be the hot incoming vapors. However, the short-term peak load must include any vaporization of liquid from the column trays, if this is possible.
The loss of cold feed in distillation columns can be a major case where a significant portion of the reboiler heat goes to sensible heating of the bottoms product. As with all cases involving disruptions to the column energy balance, the relief load will be determined by completing the upset heat balance, with all net heat input available for generating relief vapor.

15) Chemical reaction
    In the case of exothermic chemical reactions, the temperature could rise very quickly, especially if the reaction is a runaway one; the high volumes of gases produced can cause the internal pressure of the vessel to reach the set point of the valve. Pressure relief valves may not provide any protection due to their relatively slow response time. In this case, vapor depressurizing systems, rupture discs or emergency vents are better (Wong, 2001).

16) Steam out
    In case of loss of steam the following equipment may be affected: turbine drivers for pumps, compressors, blowers, reboilers, ejectors, etc.

## 4.2 Relief load calculation including vapor/liquid disengagement study

The following subchapters will describe in a detailed way the procedures followed in this research work for the calculation of the most important parameter in the sizing of a pressure relief valve: the required relief load.

### 4.2.1 External fire

There are a lot of references as API 521 (2008, 2014), Wong (1999, 2000), Katkar (2010), Rahimi Mofrad and Norouzi (2007), Hauser et al. (2001) and Cheremisinoff (1998) among others, discussing the basic nature of fire as it applies to pressure relief systems in pressure vessels and processing equipment. For low pressure, atmospheric and refrigerated storage tanks designed with standards like API 650 (2012) or DIN 4119 (1979, 1980), API 2000 (2009) applies. However, this case is not treated here. Fire is treated as a heat source which causes the temperature of the fluid contained in the vessel to increase. In most cases where the vessel contains a liquid, external fires causes the liquid to boil. However, not all process fluids behave in this way, and the actual fluid characteristics also need to be taken into account and will be analyzed here.

The specific basic assumptions taken in this thesis for sizing a fire pressure relief system are:

a) The process is assumed to be shut down and isolated from other vessels or sources of process fluid or from other potential paths of relief.
b) Liquid inventories are assumed to be at their normal maximum values. Thus, high liquid levels will be used for computing the wetted area.
c) Small diameter pipes are not included. Large diameter pipes (>DN500) are included in computing the wet surface area.
d) Heat input values are computed from empirical equations developed from the results of actual fire tests. API 521 (2007, 2014) equations for processing units have been used. Other equations developed for storage tanks coming from Compressed Gas Association or NPFA have not been used. Crozier (1985) gives a good summary of the various equations available.
e) Except for a few unusual applications, the time element of fire relief is not recognized. That is, the time required to heat the contents of a vessel to relieving conditions is not considered in sizing the pressure relief system.

The heat flux equations used here are:

For a pool fire heat input with adequate drainage and prompt firefighting:

$$Q = 43200 \cdot F \cdot A_w^{0.82} \tag{4.1}$$

For a pool fire heat input without adequate drainage and prompt firefighting:

$$Q = 70900 \cdot F \cdot A_w^{0.82} \tag{4.2}$$

where

Q is the total heat absorption by the external wetted surface, W; F is an environmental factor for fireproofing (F=1 for no fireproofing); $A_w$ is the total wetted surface, $m^2$.

This is an empirical method. The analytical one appears in the new edition of API 521 (2014) and is recommended for special cases and fires outside the scope of this empirical method. This analytical method has been reviewed recently by Zamejk (2014).

The 43200 factor includes credits for conditions normally encountered in refinery process units, which are:

a) The grading and drainage systems for most process units are sloped, so that flammable liquid will not pool directly under process vessels.
b) It can be expected that adequate firefighting activity will start very quickly after a fire starts.
c) It is very difficult to have a fire totally engulfing a process vessel, even under test conditions with a fairly small vessel. Drainage systems normally present in petrochemical units can be expected to make a fire large enough to engulf a vessel impossible to sustain.

Here, only this factor has been used.

Because API equation is derived from experimental data from tests performed in the 1940's including partially full vessels, the effect of any heat transfer through the vapor space of the vessel area is already taken into account.

API 521 recommends that only the first 7.6 m of height of a process vessel be considered as being "exposable" to fire. This measurement should be from grade, or a solid deck or a platform where a pool fire could occur.

The wetted area used here for different process equipment is as follows:

a) For vessels which are partly liquid full, such as flash drums, separators, reflux accumulators, reboilers, etc. the external area of vessel below the lower of the vessel high liquid level mark or 7.6 m above grade or major platform or deck; in cases where there is no high liquid level specified for the vessel, the highest of the top range of any level transmitter, gauge glass, indicator, level switch or controller installed on the vessel has been used.
b) For distillation columns, the liquid level is computed taking into account that all the liquid on the trays is allowed to drain into the column bottom, with the column bottom initially at its high liquid level mark. The wetted area is the vessel area up to the calculated liquid level or 7.6 m above grade. Here, the estimation of clear liquid holdup in each tray is 7.5 cm height and the downcomers are full of liquid and occupy 10% of the column cross sectional area. For packed columns, 10% of the packed volume as liquid holdup is used.
c) The wetted area can also be calculated based on the adjusted liquid level due to liquid swell. This approach, discussed by Egan (2011), has not been adopted here.

The environmental factor "F" is introduced in the fire equation to account for such factors as vessel insulation or special firefighting provisions which may be present. API 521 (2014) does not allow any credit for other mitigation systems as water spray. Credit is taken for F depending on the ability of the insulation to withstand both the temperature of the fire and the mechanical forces of firewater for a minimum time of 20 min to 1 h (Wong, 2000).

Here no credit has been given to insulation, because it is usually mineral wool but not held with stainless steel wire and not protected with stainless steel sheathing. In the process plants studied, the banding is always aluminum which melts at 660ºC. Thus, the value F = 1 has been applied.

In many process units it is likely that a single fire can affect more than one vessel. API 521 (2014) recommends an area of 2500 to 5000 $ft^2$. In this work an area of 5000 $ft^2$ has been used which corresponds a circle of 12.2 m of radius considering that an adequate drainage for spills exists.

In vessels partly full of liquid in which at the relieving pressure the fluid is below the critical pressure, a boiling process can occur depending on the duration of the fire and the boiling temperature.

In this case, the relief load can be calculated by

$$W = \frac{Q}{\lambda} \tag{4.3}$$

where

W is the relief load, kg/s; Q is the total heat absorbed from the fire, kW; λ is the fluid latent heat of vaporization, kJ/kg.

The following considerations apply in using this expression:

a) The latent heat should be evaluated at the fully accumulated increasing relieving pressure.
b) For multicomponent mixtures, the effective latent heat should be obtained in a rigorous way.
c) If the calculated latent heat is less than 115 kJ/kg, consider the use of the methods for supercritical fluids.

There are some methods available in the literature for calculating in a rigorous way the latent heat of multicomponent mixtures. Wong (2000) gives a method that requires a process simulator to calculate the physical properties depending on the percentage of vaporization, and recommends an interval between 5 and 30% mole with 2 mole % intervals. Rahimi Mofrad and Norouzi (2007) also give an example of the so called “dynamic approach” and “semi-dynamic approach” calculations. PS PPM software (Siemens) has also an option to calculate it with this dynamic approach. Katkar (2010) indicates that a conservative approach is the determination of the relief load assuming that the vessel is filled with a single component of the mixture each time, and selecting the maximum relief area among them.

Wong (1999) discusses thoroughly the case of a vapor filled vessel exposed to fire and points out that the safety valve will not protect the vessel, concluding that a gas filled vessel cannot be protected by a safety valve alone. The same indication is found in API 521 (section 5.15.4.1, 2007). Other proposed protections are:

- cooling the equipment surface by a water deluge system
- providing automatic vapor depressurizing systems
- installing external fire-roofing insulation
- using reliable fire-monitoring systems and a rapid-action fire-fighting team.

The equation used has been taken from API 521 (2007):

$$W = 0.1406 \cdot \sqrt{M \cdot P_1}\ \frac{(T_w - T_1)^{1.25}}{T_1^{1.1506}}\ A \tag{4.4}$$

where

W is the relief flow, lb/h; A is the exposed surface area of the vessel, $ft^2$; $P_1$ is the reliving pressure, psia; M is the molecular weight of vapor; $T_w$ is the vessel wall temperature, ºR (recommended as 1560 ºR for carbon steel); $T_1$ is the relieving temperature, determined by the following equation:

$$T_1 = T_n \cdot \frac{P_1}{P_n} \cdot \frac{z_n}{z_1} \tag{4.5}$$

where

$P_n$ is the normal operating pressure, psia; $T_n$ is the normal operating temperature, ºR; $z_n$ is the compressibility factor at normal operating conditions; $z_1$ is the compressibility factor at relieving conditions.

As pointed out in API 521, this simplified equation often predicts very conservative relief loads because of the assumptions involved in its derivation, which are:

- the vessel is uninsulated
- the vapor in the vessel is an ideal gas
- the time required to heat up the vessel and its contents is ignored
- the temperature of the vapor in the vessel is constant
- vessel failure will not occur.

Fluids which may exceed their critical point during the pressure rise to relieving conditions in a fire, require special consideration in computing the relief load. Fluids near or above their critical pressures have zero or very low latent heats and behave somewhat like vapors. The first rigorous method of calculation is that of Francis and Shackelton (1985). This method basically requires that the energy coming into the vessel from the fire be balanced by the energy leaving the vessel with the relief stream at a constant relieving pressure. In this analysis, the mass, temperature and density of the fluid in the vessel vary with time. Here, an improved method based on the Francis work and proposed by Ouderkirk (2002) has been used (see section 6.3).

Concerning the scenario of fire applied to liquid filled systems, the following phenomena should be taken into account:

- Initially there is no vapor space. Before boiling of the contents, hydraulic expansion will occur as the vessel heats up.
- Once boiling begins there is little or no vapor/liquid disengaging area. Depending upon the physical configuration, the relief stream may be vapor, liquid or two phase flow.
- A number of vessels, particularly heat exchangers, may be connected in series or parallel, and may have to be treated as a unit (Wong, 1992b).

Unfortunately, the duration of fire has not been discussed much in relation to the contingency in the diverse standards. But there is the possibility that the fighting team can extinguish the fire before the inside fluid reaches the relief valve set pressure. Wong (1999) points out that the duration should be no more than 20 minutes. In this way, the author recommends that if the liquid inventory of the vessel will be vaporized in less than 20 minutes the low liquid inventory vessel should be treated as a gas filled vessel. Rahimi Mofrad and Norouzi (2007) give a decision flowchart to determine the calculation of the relieving load for the fire case considering wetted or unwetted surface.

Here, the time required for vaporizing all the liquid in a vessel has been calculated as being the time to heat the liquid to the boiling point plus the time to vaporize the inventory (Wong, 1999; Rahimi Mofrad and Norouzi, part 1, 2007):

$$t = t_1 + t_2 = \frac{V_L \rho c_p (T_{bp} - T_n)}{Q} 60 + \frac{V_L \rho \lambda}{Q} 60 \tag{4.6}$$

where

$t$ time, minutes; $V_L$ initial liquid volume in the vessel, $ft^3$; $\rho$ liquid density, $lb/ft^3$; $c_p$ specific heat of liquid, Btu/lb°F; $T_{bp}$ average boiling point of liquid inventory at the relieving pressure, °F; $T_n$ normal liquid operating temperature, °F; $\lambda$ latent heat of vaporization of the liquid, Btu/lb; Q total heat absorption across wetted surface area, Btu/h.

If the total calculated time $t$ is less than 20 minutes, a gas filled vessel scenario has to be considered. If the time $t$ is more than 20 minutes, liquid filled vessel has to be considered.

Another consideration to take into account in equipment containing low liquid inventory is determining the final pressure in the vessel when the last drop of liquid is vaporized. Wong (1999) gives an approximate approach by considering the temperature inside the vessel at its boiling point; then the moles of vapor and liquid are calculated assuming an ideal gas and the final pressure in the vessel with the total moles in the gas phase is compared to the set pressure. If the calculated pressure is less than the set pressure, a gas filled vessel must be assumed.

### 4.2.2 Liquid filled systems

Liquid filled systems have their own characteristics and these are different from vapor systems. The differences are:

a) Liquid systems have very little capacitance, so system response to overpressure is very rapid and much more prone to instability.
b) Liquid relief valves have their own characteristics, which are very different from valves in vapor services.
c) Liquid relief systems present greater difficulties in disposing of relief streams.

The blocked discharge of centrifugal pump systems is one of the most frequent cases of overpressure. Here the evaluation of pressure relief systems for equipment which is pressurized by a centrifugal pump, has been based on these criteria:

a) Maximal shut-off head is evaluated for installed and potential maximum impeller sizes and maximum continuous speed (for turbines or frequency driven motors).
b) Blocked conditions should be evaluated at the maximal normal operating suction pressure. This pressure should be taken to be 90% of the pressure relief device set pressure of the source vessel plus the liquid head imposed by the liquid in the suction vessel at high liquid level. If no exact operating limits are known, this approach represents a practical upper operating limit.
c) Blocked conditions should be evaluated at maximal suction pressure and normal flow rates. In this case the source vessel operates at its full accumulated relieving pressure, plus static head equivalent to high liquid level. In evaluating potential relief requirements, the downstream system is not blocked and is assumed to be able to pass normal flow, handled by the control valve. Any excess flow generated under these conditions may have to be handled by a pressure relief valve.

The calculation procedure is described in Shell Guideline (Shell, 2004).

Simultaneous blocked discharge and maximal suction pressure due to unrelated events is considered to be double jeopardy. An exception is the case of a blocked control valve of the reflux in a distillation column, which causes a suction overpressure in the accumulator, because these are coincident events.

Thermal relief is another frequent case of overpressure in liquid filled systems. In this case, the overpressure is due to the hydraulic expansion of the process liquid when the system is blocked in and exposed to an outside heat source. Some typical examples are:

a) The cold side of an exchanger is blocked in while the hot side continues flowing.
b) Liquid filled piping or vessels are blocked in and are subsequently heated by heat coils or tracing.
c) Liquid filled piping or vessels are blocked in at near ambient temperature and are heated by solar radiation.

The calculation procedure used in this thesis for liquid expansion is based on the equations of the paragraph 5.14.3 of API 521 (2007).

In case of piping or vessels exposed to solar radiation, the maximum heat flux of 1.15 kW/$m^2$ has been applied here not only to the upper half of the external pipe surface, but to all the external area as a conservative basis.

Concerning the pressure relief valve sizing in liquid filled fire services, the following criteria has been adopted as recommended by API 521 (2014):

a) The pressure relief valve is sized for the larger of the hydraulic expansion case preceding boiling or during the boiling process.

b) If the pressure relief valve is located where the vapor produced by boiling cannot immediately reach the valve, it is sized for a liquid flowrate equal to the generated vapor which displaces it (liquid equivalent volume flow).

### 4.2.3 Distillation equipment

Evaluation of the potential relief loads which may occur in distillation columns is a very complex task. The relief causes fall into three main categories:

1) Operational failures: loss of feed, loss of reflux, loss of intermediate reflux, loss of quench, utility failure (power, steam, instrument air, cooling water), control failure, abnormal heat input, absorbent medium failure, blocked outlet.
2) Compositional changes: loss of heat (upstream), accumulation of non-condensables.
3) Other conditions: fire, heat exchanger tube breakage, thermal instability of bottom material, reaction (contamination).

Generally, the operational failures represent the most severe upsets to normal column operation and will govern pressure relief valve sizing.

The different methods available for designing the pressure relief valve which protects a distillation column and its related equipment are:

a) Gross overhead vapor: related to the maximum vapor rate that can support the top of the distillation column. Although it seems a conservative method, Bradford and Durrett (1984) indicated that it could undersize the valves.
b) Flash drum approach: used for the condenser failure scenario, because the column loses its liquid inventory and approaches the point where it would simply act as a flash drum. The feed stream is flashed at relieving pressure with additional heat coming from the reboiler.
c) Absorber operation: model developed for the scenario of loss of reflux; Rahimi Mofrad (2008b) explains this method but it requires a process simulator.
d) Unbalance heat load method: developed by Sengupta and Staats (1978) to be used with the enthalpy diagrams, and updated by Nezami (2008) considering the availability of commercial simulators; this method is largely accepted today in most new relieving calculations (Xie et al., 2013).
e) Dynamic simulation: this method is the best available, but it requires construction equipment details, hydraulic information of the column and ancillary equipment, and control system details. A lot of authors prefer this method in revamping or debottlenecking projects, but not in the design phase of a project because of the work load required (Rahimi Mofrad, 2008b; Nezami, 2008; Arbo et al., 2008; Depew and Dessing, 1999). Another concern is the validation of the model vs. the dynamic behavior of the real plant from an upset as plant data is required. Xie et al. (2013) pointed out that the optimum relief load for flare networks is a combination of the unbalanced heat load method and dynamic simulation.

The unbalanced heat load method has been used in this work (see section 6.5). The following basic assumptions are implicit in this method:

1) The column equilibrium temperatures are adjusted by estimating new bubble points at relieving pressure, assuming that liquid compositions and tray pressure drop are unchanged. The vapor relief temperature corresponds to the bubble point temperature at the accumulated relief pressure for the equilibrium stage nearest to the pressure relief valve (usually the top tray).
2) Unless significantly affected by system hydraulics at relief pressure, all feeds, stripping media and reflux streams will continue at normal flow rates with the exception that loss is under consideration in the relieving case.
3) All streams leaving and entering the column are at vapor/liquid equilibrium at relieving pressure, with the exception of the feeds.
4) Vapors may not accumulate in the column during relief, after the column reaches the relieving pressure.

5) Liquids can accumulate in the column, i.e. levels rise or fall, and can absorb heat whether or not they leave the system.
6) In calculating the upset heat balance, credit may be taken for product sensible heat absorption from the feed temperature after it enters the column at relief condition to the normal outlet temperature.
7) Unbalanced heat resulting from an operating failure will be relieved by vaporizing top tray liquid, at a temperature and latent heat corresponding to relief conditions. It is assumed that there is an ample supply of top tray liquid during the short time period over which peak relief rates occur.
8) Liquid entrainment from the top tray is not included in the scope of the relief load. However, if the column is well above flood point liquid entrainment should be considered.
9) The vaporized part of a feed has to be considered as a contribution to relief rate once flashed isentropically at relief conditions.
10) Credit may be taken for pinch temperatures in the reboiler, if and only if light material cannot reach the reboiler under normal operation. Because of many mistakes as pointed out by Bradford and Durrett (1984), if any doubt exists, no credit should be taken.
11) The safety margins used in the distillation tower design have to be considered, i.e. the ultimate capacity of the column has to be considered (85% of the theoretical tray flooding or at the limiting reboiler duty in unfouled conditions and condensers at fouled conditions).

The relief rate can be calculated by (Melhem, 2007):

$$W_R = \frac{\sum Q_I - \sum Q_O + \sum W_F H_F - \sum W_P H_P - W_B H_B - (\sum W_F - \sum W_P - W_B)(H_R - L_R)}{L_R} \tag{4.7}$$

where

$W_B$ bottoms accumulation rate, lb/hr; $H_B$ entalphy of bottoms accumulation, Btu/lb; $H_R$ enthalpy of the relief vapors, Btu/lb; $L_R$ latent heat of vaporization of the relief vapors, Btu/lb; $\Sigma Q_I$ summation of all heat inputs, Btu/hr; $\Sigma Q_O$ summation of all heat removed, Btu/hr; $\Sigma W_F H_F$ summation of products of each feed rate (lb/hr) times its enthalpy (Btu/lb), Btu/hr; $\Sigma W_P H_P$ summation of products of each product rate (lb/hr) times its enthalpy (Btu/lb), Btu/hr.

Concerning the calculation of $L_R$, it is straightforward when there is a pure component, but in the case of a multicomponent mixture there is no true latent heat. Wong (1999) pointed out that the orifice area is at maximum value when the following function is at its maximum:

$$f(T, Z, M, \lambda) = \frac{\sqrt{\frac{TZ}{M}}}{\lambda} \tag{4.8}$$

where

T relieving fluid temperature K; Z relieving fluid compressibility factor; M relieving fluid molecular weight g/mol; λ latent heat of vaporization, kJ/kg

To determine the effective latent heat, differential isenthalpic flashes of the bubble point liquid are performed at various vapor fractions over the entire boiling range. This procedure has been explained by Rahimi Mofrad and Norouzi (2007) and has been followed here:

a) Beginning with the liquid mixture at its bubble point, a simulation has been run (Aspen Hysys) which vaporizes 0.1% of the liquid. Then the liquid is cooled to its original temperature to offset its sensible heating. The effective latent heat is obtained by dividing the net rate of heat input by the vapor generation rate.
b) Taking the original liquid mixture, an isenthalpic flash is performed such that 5% is vaporized then the liquid is cooled again as above. This procedure is continued with 5% steps over the entire boiling range of the mixture. A plot is made of both the latent heat and the function $f(T, Z, M, \lambda)$ versus percent vaporized.

### 4.2.4 Heat exchanger tube rupture

Heat exchanger tube rupture is one of the possibilities that can overpressurize a system. However, minor leakages are not considered significant in terms of overpressure potential, but in cases involving a complete break, or separation of a tube from a tubesheet, they must be evaluated unless the design values of low pressure and high pressure sides of the exchanger meet the 10/13 rule for an exchanger designed by ASME Code VIII-I (2013). API 521 (2014) does provide some criteria for evaluating tube breakage, including some recommendations for the determination of the required size, by steady-state and by dynamic analysis. Recently, the Energy Institute (2015) has published a new edition of a specific guideline dedicated to this topic, which has had very good acceptance by the engineering community since the first edition in year 2000.

Considerable theoretical work has been done to predict the behavior of liquid filled systems in response to tube breakage (Perez Muñoz et al., 2011; Ennis et al., 2011; Urdaneta, Oude, 2015). These analyses have been based upon theoretical, instantaneous, complete ruptures of exchanger tubes and use of dynamic models based upon liquid transient equations. The conclusions are that the relief devices must open very fast. Pressure relief valves are not good candidates for such a fast increase of pressure and rupture discs also have limitations according to manufacturers (Schmidt, 2010).

Anyway, in industrial practice the following criteria are accepted:

a) The theoretical model of an instantaneous, full area, rupture of a tube has not been demonstrated in practice.
b) The response times required for pressure relief devices to protect against an instantaneous rupture, is often less than 10 milliseconds by fully opening, have not been demonstrated as routinely achievable by such industrial devices. Rupture discs are sometimes presumed to have this capability, but there is no experimental data to back this up with the exception of Energy Institute (2015) experiments. Most rupture disc manufacturers will not guarantee such a quick response time.
c) Many companies consider that the most prudent design practice for tube breakage is to perform all analyses of tube failures using steady state, volumetric displacement techniques (Wong, 1992a, 1992b).
d) Experience has demonstrated that a tube breakage needs 0.2 to 0.5 seconds to occur (as opposed to instantaneously), which results in little or no transient peak pressure.

Tube failure is a viable contingency if the hydrostatic test pressure of the low pressure side is exceeded by the design pressure of the high pressure side. Because, according to ASME, the hydrotest pressure is 1.3 x MAWP, the ratio of pressures is 10/13, known as the “10/13” rule. In Spain it is 1.43 or $1.25\frac{\sigma_{25º}}{\sigma_T}$ times the design pressure, the higher of these two values ($\sigma_{25º}$ is the allowable stress of the material at 25ºC and $\sigma_T$ is the allowable stress at the design temperature).

When calculating the flow across a broken tube, the following assumptions are made:

a) Tube breakage is regarded as a sharp break in a single tube, with the high-pressure fluid flowing through both sides of the break.
b) The fluid flowing through the broken tube suffers an isenthalpic expansion
c) For calculation purposes, each side of the break behaves as a sharp-edged orifice having the cross-sectional area of one tube.
d) The treatment is identical for flow into or out of the broken tube, i.e., either the shell or tube side may be the high-pressure side.
e) The driving force for flow is the difference between the maximum normal operating pressure of the high pressure side and the accumulated relieving pressure of the low pressure side (Wong, 1992b). Some authors use the set pressure in the low pressure side (Schmidt, 2010).

For vapor flow, the equation used based on an orifice coefficient of 0.6 and the total flow through both sides of the break is considered (API 521, 2007) to be:

$$W = 1605 \cdot d^2 \cdot \sqrt{\rho k P_1 \left(\frac{2}{k+1}\right)^{\left(\frac{k+1}{k-1}\right)}} \tag{4.9}$$

where

W vapor flow rate, lb/h; d inside diameter of tube , in; k specific heat ratio for vapor; $P_1$ high pressure side operating pressure, psia; $\rho$ vapor density at high pressure side operating conditions , lb/ft$^3$.

For liquid flow through both sides of the break, assuming a coefficient of discharge of 0.6:

$$Q = 35.9 \cdot d^2 \cdot \sqrt{\frac{\Delta P}{SG}} \tag{4.10}$$

where

Q liquid flow rate, gpm; d inside diameter of tube, in; ΔP normal high pressure side operating pressure minus allowable low pressure side accumulated overpressure, psi; SG liquid specific gravity at flowing conditions.

If multiple-phase flow is present or can result (flashing) from the pressure loss across the tube break, models for treating this are presented by Leung (1996), CCPS (1998) and Schmidt (2010) among others. The models can incorporate thermodynamic nonequilibrium (boiling delay) and/or the mechanical nonequilibrium (slip between phases). There are two possible locations for a tube break to occur: at the tubesheet or along the tube length. The location of the break will dictate the type of flow model that should be used. The concern is whether or not there is sufficient path length for equilibrium to be established between the phases. Here, it has been assumed that the break occurs along the tube length and, consequently, equilibrium between the phases is assumed (see section 6.6 for a comparison of the results using different models).

Concerning the capacity credits for the low pressure side, the influx of material from the high pressure side is assumed to back out the normal flow entering the low-pressure side. If the low pressure side is in a parallel network, such as in a cooling water system, the normal supply to adjacent exchangers on the subheader, both upstream and downstream, may also stop (Wong, 1992b). The calculation procedure presented by Wong has been used here.

### 4.2.5 Vapor/liquid disengagement

Considerable research related to the behavior of pressure relief systems in two-phase, reacting or flashing services has been done by the AIChE's Design Institute for Emergency Relief Systems (DIERS) (Fisher et al., 1992)

Multiphase flow in the inlet of the pressure relief valve may occur when relieving:

- a two phase stream
- liquids that may flash
- reactive systems
- supercritical fluids
- a vessel with high liquid level
- a vessel with internals or solids beds that obstruct disengagement
- foamy, bubbly or viscous liquids.

For some of these systems, the pressure relief valve inlet quality may be obvious. For others, it will be necessary to predict vapor-liquid disengagement characteristics in the vessel to be protected. Often, the inlet quality varies with time and requires a dynamic analysis.

There are varying degrees of vapor-liquid disengagement. Complete disengagement for a top-mounted relief device is characterized by all vapor venting. Zero disengagement is characterized by homogeneous vessel venting. When homogeneous venting is assumed, the inlet quality is equal to the vapor fraction of the bulk fluid within the vessel. Between these two extremes are the partial disengagement models, churn-turbulent and bubbly. The churn turbulent model assumes greater disengagement than the bubbly one. The governing equations and associated charts for this model are exposed in DIERS project manual among others (Fisher et al., 1992; ISO 4126-10, 2010; Fisher and Forrest, 1995). Christ et al. (2011), in a very interesting contribution, give some recommendations for the case of external heating. They point out that ISO 4126-10 is less conservative than DIERS. Egan (2011), after reviewing the literature and comparing the results for runaway reactions and external fire for some case studies, concludes that in most cases the thermal expansion from storage to the relief case is decisive for predicting the disengagement. Wong (1997) pointed out the traditional practice in the engineering community of ignoring the vapor/liquid disengagement phenomenon in designing liquid blockage in flash drums. Raman (2015) in a very interesting article suggests a practical procedure to model the sizing of pressure relief valves installed below a liquid level.

Summarizing, the most conservative model for top-mounted vent sizing is the homogeneous one, followed by bubbly, churn-turbulent and, lastly, all vapor.

In the methodology proposed here, the first criterion is screening the foamy or non-foamy behavior of the liquid that results in the possibility of producing multiphase flow during pressure relief. The conservative approach recommended by Katkar (2010) has been used. It is expected that a liquid is foamy if:

-its viscosity is greater than 100 cP
-liquid contains surfactants
-liquid is dirty and contains solids
-multicomponent liquid with wide range of boiling points
-chemical reacting liquids
-there is more than one liquid phase.

Another screening for multiphase flow during pressure relief is based on the initial fill level. If the liquid level when the relieving process begins is higher than the limit level, a mixed phase relief has to be considered. A conservative approach for this limit level is considering two-phase relief when the initial liquid level (before valve opens) is above 20% for foamy liquids and above 80% for non-foamy liquids.

The simple approach explained by Katkar has been used:

$$h = \sqrt{\left(\frac{W}{2\pi U_e \rho_v}\right)\frac{1}{3600}} \tag{4.11}$$

where $U_e$ is the entrainment velocity, which is calculated as:

$$U_e = 3\left(\frac{\sigma g \rho_L}{\rho_v^2}\right)^{1/4} \tag{4.12}$$

where

h free board height, m; W relief rate, kg/h; $\rho_v$ density of gas, kg/m$^3$; σ surface tension N/m; g gravitational acceleration 9,81 m/s$^2$; $\rho_L$ density of liquid, kg/m$^3$; $U_e$ Kutateladze entrainment velocity, m/s.

## 4.3 Selection and design (required area) of the pressure relief valve

API 520 (Part I, 2014) gives a classification of pressure relief valves in two categories (spring-loaded: conventional, liquid service; and pilot operated) and a detailed list of recommendations for their selection according to the fluid, the total backpressure, etc.

The design of pressure relief valves requires three components (Darby, 2005):

a) Flow model. The isentropic model has demonstrated to be the most adequate:

$$G_0 = \rho_n \left(-2\int_{P_0}^{P_n} \frac{dP}{\rho}\right)^{1/2} \tag{4.13}$$

where

$G_0$ is the theoretical mass flux through an isentropic nozzle; $P_0$ is the pressure at the entrance to the valve; $P_n$ is the pressure at the nozzle exit; ρ is the fluid (or mixture) density at pressure P; $\rho_n$ is the fluid density at pressure $P_n$, the nozzle exit or throat; $K_d$ is the dimensionless discharge coefficient that accounts for the difference between the predicted ideal mass flux and the actual.

b) Fluid property data or model (density versus pressure). The accuracy of the calculation depends on the accuracy of ρ vs. P information over the isentropic path from $P_0$ to $P_n$.

c) Test data to adjust for deviation from isentropic nozzle flow (the discharge coefficient $K_d$):

$$G_n = \rho_n K_d \left(-2\int_{P_0}^{P_n} \frac{dP}{\rho}\right)^{1/2} \tag{4.14}$$

where $G_n$ is the actual mass flux through nozzle $= K_d G_0$

For liquids assuming a constant density, the equation reduces to

$$G_0 = \sqrt{2\rho(P_0 - P_n)} \tag{4.15}$$

The equation is valid for fully turbulent flow ($R_e$ > 100,000). For high viscosity liquids the equation has to be multiplied by a correction factor available in API 520.

For ideal gases and choked flow, the equation reduces to

$$G_0 = \sqrt{kP_0\rho_0}\left(\frac{2}{k+1}\right)^{\frac{k+1}{2(k-1)}} \tag{4.16}$$

where k is the isentropic exponent for a gas.

When the gas (or mixture) behaves as non ideal, for instance when the compressibility factor (Z) is out of the range 0.8 < Z < 1.1 which corresponds to high pressures or critical-point regions, an equation of state is required to evaluate by an isentropic flash the variation of the density with the pressure. However, Kim et al. (2011) and Shackelford (2003) gave an equation which uses a "non-ideal k value" that reduces considerably the calculations. Anyway, a process simulator is required as well.

However, industry has looked for simple sizing formulas. Table 4-1 presents the main sizing expressions covered by some codes. Copigneaux (1980) compared different Codes and Standards from US, France, Germany, United Kingdom and Russia and found that they use implicitly the same formula.

**Table 4-1. Summary of sizing formulas in relevant codes.**

| MEDIUM | Units | ASME VIII/API 520 (1) | ISO 4126-1(2) | AD 2000-A2(3) |
|---|---|---|---|---|
| Gases and vapors (critical flow) | SI | $A = \frac{W}{CK_bK_cK_dP_1}\sqrt{\frac{TZ}{M}}$ | $A = \frac{Q_m}{p_0CK_{dr}}\sqrt{\frac{T_0Z}{M}}$ | $A_0 = 0.1791\frac{q_m}{\Psi\alpha_w p_0}\sqrt{\frac{TZ}{M}}$ |
| Gases and vapors (subcritical flow) | SI | $A = \frac{17.9W}{F_2K_dK_c}\sqrt{\frac{ZT}{MP_1(P_1-P_2)}}$ | $A = \frac{Q_m}{p_0CK_bK_{dr}}\sqrt{\frac{T_0Z}{M}}$ | |
| Steam | SI | $A = \frac{190.5W}{P_1K_dK_bK_cK_NK_{SH}}$ | $A = \frac{1}{0.2883}\frac{Q_m}{CK_{dr}}\sqrt{\frac{v}{p_0}}$ | $A_0 = \frac{xq_m}{\alpha_w p_0}$ |

| MEDIUM | Units | ASME VIII/API 520 (1) | ISO 4126-1(2) | AD 2000-A2(3) |
|---|---|---|---|---|
| Liquids | SI | $A = \frac{11.78Q}{K_d K_w K_c K_v}\sqrt{\frac{G1}{P_1 - P_2}}$ | $A = \frac{1}{1.61}\frac{Q_m}{K_{dr} K_v}\sqrt{\frac{v}{p_0 - p_b}}$ | $A_0 = 0.6211\frac{q_m}{\alpha_w\sqrt{\rho(p_0 - p_a)}}$ |

Notes

1. A orifice area, G specific gravity, Q volumetric flow, W mass flow, Z compressibility factor, T relieving temperature, M molecular weight, $F_2$ coefficient of subcritical flow, $K_b$ capacity correction factor due to back pressure (gas, vapor, steam), $K_c$ = 1 (safety valve without rupture disc) and 0.9 ( safety valve with rupture disc), $K_d$ discharge coefficient, $K_N$ correction factor for Napier equation, $K_{SH}$ superheat steam correction factor, $K_v$ correction factor due to viscosity, $K_w$ correction factor due to back pressure (liquids), $P_1$ relieving pressure, Coefficient $C = 0.03948\sqrt{k\left(\frac{2}{k+1}\right)^{\frac{(k+1)}{(k-1)}}}$
2. A orifice area, $Q_m$ mass flow, Z compressibility factor, $T_0$ relieving temperature, v specific volume, $K_b$ theoretical capacity correction factor for subcritical flow, $K_{dr}$ certified derated coefficient of discharge, $K_v$ viscosity correction factor, $p_0$ relieving pressure, $p_b$ back pressure, Coefficient $C = 3.948\sqrt{k\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}}$
3. $A_0$ orifice area, $q_m$ mass flow, T relieving temperature, Z compressibility factor, $p_0$ relieving pressure, $p_b$ back pressure, $\alpha_w$ certified discharge coefficient, Ψ outflow function (gas flow), x pressure medium coefficient (gas flows).

The ASME Code covers the certification of safety valves for the flow of saturated steam, water, air and natural gas (Section VIII UG-131). However, API 520 is a recommended practice for standardizing the pre-selection of pressure relief valves for gases, vapors, liquids and two-phase flow in the design phase of the plant. API 520 uses the same formulas as the ASME Code but extends them with correction factors, like back pressure and viscosity, to make them applicable to many practical applications. In API 520 the pre-selection of the valve requires the determination of an “effective relief area” and an “effective coefficient of discharge”, which are nominal values and therefore independent from the selection of either the design or the manufacturer. The effective relief areas are those listed in API 526 (2009) in increasing order from letter D to T.

Once the valve orifice is selected, it must be demonstrated that the certified capacity meets or exceeds that of preliminary sizing. For this calculation the “actual discharge coefficient “and the “actual discharge area” from the manufacturers catalog must be used.

The discharge coefficients, $K_d$, $K_{dr}$, and $\alpha_w$ take into account the deviation of the pressure relief valve from an ideal nozzle (isentropic). Thus, these values are the quotient of the actual measured flow divided by the theoretical flow in an ideal nozzle. Codes like ASME, ISO 4126 or AD 2000-A2 require actual flow tests to establish the flow efficiency of manufacturers valves. ISO 4126-1 requires 9 tests (three valves of three different sizes), which results are averaged if none of these 9 discharge coefficients vary more than plus or minus 5% of the average coefficient. An additional requirement is to reduce the flow tested discharge coefficient by 10%. This reduced coefficient provides an additional safety factor when calculating the required area of the valve. Thus $K_{dr}$ (r of rated) means that the factor $K_d$ has been multiplied by 0.9

Here the following design basis has been used:

### 4.3.1 Gases/Vapors

The pressure relief valves of the plants studied were designed by the AD 2000-A2 code. The majority of the valves are from two manufacturers: Leser and Sempell. The Leser valves were designed by Valvestar Software. This software follows the Leser Engineering Book. The Sempell valves were designed following the Sempell Catalogue (Sempell KS27585E).

Some codes recommend not to use in the design phase compressibility factors below 0.8 or greater than 1.1 (API 520, 2007), or less than 0.7 and greater than 1.4 (Wong, 2000), because the results are not adequate at very high pressure conditions or critical-point regions. Here credit has been given for the exact value of Z, calculated by software (Aspen- Hysys) and checked by specific compressibility charts when available (Ingersoll-Rand, 1981).

The ratio of ideal specific heats at inlet relieving temperature (API 520, Part I, 2014) or isentropic exponent of the medium in the pressure chamber (AD 2000-A2) was calculated by API Technical Data Book (1997) and Aspen Hysys.

According to API 520 (Part I, 2007, 2014) if the compressibility factor of the fluid is greater than 0.8, then the ideal gas ratio of specific heats may be used to determine the expansion coefficient. If the compressibility factor is less than 0.8, then the expansion coefficient should be based on the isentropic expansion coefficient. However, recently Smith and Burgess (2015) suggested that, when the fluid critical volume is 2 or lower, a direct integration method is required to accurately estimate relief-device capacities. If the critical volume is greater than 2 and the compressibility factor is less than 0.8, then using the nozzle equation with the isentropic expansion coefficient factor is acceptable (see comparison of different methods in section 6.2).

Following the excellent work of Shackelford (2003) and Kim et al. (2011), it is possible to design in the region $Z < 0.8$, without using the tedious homogeneous direct integration method described in Annex B API 520 (Part I, 2008). The best results are obtained when the inlet compressibility factor and the actual isentropic expansion coefficient for the fluid are at relieving conditions. This isentropic coefficient is calculated through an equation of state that describes the pressure-volume relationship along an isentropic expansion path. Peng-Robinson and Lee Kesler equations have been used.

Both authors and API 520 remark that the use of the real gas specific heat ratio at relieving conditions with the real compressibility factor should be avoided, because it can result in a significantly undersized relief valve.

To limit the discharge capacity of a pressure relief valve, a restricted lift is used. A restricted lift is used when:

a) The safety valve is oversized (gas and two-phase flow).

b) The inlet pressure drop is larger than 3% of set pressure or the built-up back pressure is too large because of excessive flow (gas, liquids and two-phase flow).

Leser catalog (the technical handbook) gives figures for obtaining the corrected value of $K_{dr}/\alpha_w$ due to lift restriction. AD 2000-A2 allows a lift restriction of below 30% of the maximum lift as long as it is more than 1 mm.

The discharge coefficient has to be corrected in case of higher back pressures, i.e. higher than 15% for conventional valves: Leser catalog has been used.

### 4.3.2 Steam

The pressure medium coefficient x, is calculated through AD 2000-A2:

$$x = 0{,}6211 \cdot \frac{\sqrt{p_0 \cdot v}}{\Psi} \quad \text{in h mm}^2\text{ bar/kg}$$

where

$p_0$ is the absolute pressure in pressure chamber, bara; $\nu$ is the specific volume of medium in pressure chamber $m^3$/kg; $\Psi$ is the outflow function.

### 4.3.3 Liquids

The equation for liquids applies when there is no change of phase (no flashing) when flowing through the valve. Although AD2000-A2 (2006) gives no correction factor for the viscosity of liquids, the sizing procedure shown in ISO 4126-1 is used. ISO 4126-1 and API 520 Part I give the viscosity correction factor for Newtonian liquids as a chart. The calculation of the required area of the valve is an iterative process beginning with the correction factor $K_v = 1$ and calculating the Reynolds number with the selected (standard) area. The method is valid for $Re > 34$. Darby and Molavi (1997) developed a correction factor that covers a greater range of Reynolds numbers. When using these formulas for non-newtonian liquids, the pressure relief valves are oversized according to Moncalvo and Friedel (2009).

### 4.3.4 Two phase-flow

An impressive quantity of literature has been produced in the last twenty years related to the release of fluid in pressure vessels that produces in the inlet of the pressure relief valve, a flashing, a condensing or a non-condensing flow (frozen). Flashing is the most frequent occurring process and happens when the entering fluid consists of a saturated liquid or a subcooled liquid that reaches the saturation point in the nozzle, or a two-phase vapor liquid mixture. Another case is the retrograde condensation that occurs when a supercritical fluid enters the valve and condenses at the exit of the nozzle, when the pressure decreases.

One of the most used nozzle models is the Homogeneous-Equilibrium Model. It is based on assuming that the fluid through the valve is a well mixed gas/vapor liquid that can be taken as a one phase, with properties which are a suitable combination of each one. Moreover, it is assumed that the two phases are in mechanical (no slip between phases) and thermodynamic (no delay in vaporization) equilibrium. The thermodynamic path through the valve is assumed to be isentropic, although in reality, it is a mixture between adiabatic and isentropic.

A literature survey on the methods available for designing pressure relief valves for two-phase flow has been made (Guinea, 2015); the results are presented in table 4-2.

**Table 4-2. Literature survey on methodologies for pressure relief valves with two-phase flow.**

| Author/source | Characteristics | Advantages | Disadvantages | References |
|---|---|---|---|---|
| API 520-1993<br>VdTÜV 100/2 | Addition of the area required for the vapor and the liquid | Easy calculations | It has been proven that can undersize | (API 520 -1993)<br>(VdTÜV-Merkblatt Sicherheitsventile 100/2 – 1973)<br>(Bozóki, 1996)<br>(Wagner, 1999) |
| Melhem | Generalized charts | Easy calculations | Only available for few substances | Melhem 2003 Flow maps for common chemicals |
| Leung | Omega method (Assumes homogeneous equilibrium) | Need to evaluate fluid properties at only one point of the isentropic transform | a)The linearized equation of state (Clasius-Clapeyron) may not give exact two-phase density values<br>b) The method is unreliable near the critical point or in case of retrograde condensation<br>c) Slip and non-equilibrium effects are not accounted for<br>d) Aplication only if $T_{red} < 0.9$ or $P_{red} < 0.5$<br>e) A modification of the basic model is required for slightly subcooled liquids | (Leung, 1996) |
| API 520-2000 (HEM model)<br>API 520-2008 | Omega method with one point or two-points (Assumes homogeneous equilibrium). Neither thermodynamic nor mechanical nonequilibrium is taken into account | The same as Leung model | a)Neither thermodynamic nor mechanical nonequilibrium is taken into consideration<br>b) For flashing mixtures with a boiling range > 150ºC two isentropic points are | (API 520 Part I, Appendix D, 2000)<br>(API 520 Part I, Annex C, 2008) |

| Author/source | Characteristics | Advantages | Disadvantages | References |
|---|---|---|---|---|
| | | | required | |
| Henry/Fauske | Homogeneous non-equilibrium model ,HNE) | Very precise results are obtained | Several equations have to be solved simultaneously and detailed property data is necessary including entropy | (CCPS,1998 ) (Fauske,1999) |
| HNE-DS Model (Diener/Schmidt) ISO 4126-10 | The Diener/Schmidt model is the Omega method improved with boiling delay of the liquid and slip between phases. The ISO 4126-10 is the Diener/Schmidt model without slip. | Requires physical properties only at the inlet conditions of the valve. The ISO 4126-10 includes the extension of Schmidt for initially subcooled liquids | | (Diener,Schmidt, 2004) (ISO 4126-10, 2010) (Schmidt, Westphal, 1997) |
| HNE-DS Model extended (Schmidt) | Omega method taking into account the mechanical and thermodynamic non-equilibrium. Schmidt extended the model for subcooling entering liquid | Requires physical properties only at the inlet conditions of the valve | The calculation of the critical pressure is an iterative process | (Schmidt, 2007, 2013) (Schmidt, Claramunt, 2014) |
| Simpson TPHEM | Two-phase homogeneous model. A universal equation | The method is simple and flexible | Requires good density data | (Simpson, 1991) (CCPS, 1998) |
| Darby API 520-2008 | Homogeneous direct integration (HDI) method for nozzles > 10 cm | a) It is applicable for all fluids (gas, vapor, liquids and two-phase flow) b) The procedure is independent of if the entering fluid is cold liquid, subcooled flashing liquid, condensing vapor or a two phase mixture | a) Requires multiple data points (Pressure, density, quality) over an isentropic range of pressures. Requires a process simulator b) Accurate thermodynamic and physical property data are required to obtain good results c) Nonequilibrium effects are not taken into account d) Can produce excessive conservative results when the fluid is in two-phase frozen flow (no flashing) or nozzle < 10 cm and there is flashing flow | (Darby, 2005) |
| Darby | Homogeneous nonequilibrium direct integration method (HNDI) for nozzles < 10 cm | The same as HDI | The same as HDI except c) | (Darby, Self, Edwards, 2002) |
| Leung Darby ISO 4126-10 | Calculation of the discharge coefficient, which depends on the geometry of the valve | Darby recommends using the discharge coefficient for gases if the flow is chocked. Leung uses the liquid coefficient and the ISO uses the liquid and gas coefficients from the manufacturer | All the models lack of experimental evaluation | (Darby, 2004) (Leung, 2004) |
| Melhem | High viscosity two-phase flow | For valves with an inlet diameter ≥ 4 inches use the omega method. | The calculation requires all system (valve and piping) to be considered. The | (Melhem, 2004) |

| Author/source | Characteristics | Advantages | Disadvantages | References |
|---|---|---|---|---|
| | | In case that the inlet diameter ≤ 4 inches use a homogeneous nonequilibrium model or Darby model with slip. | author recommends a rupture disc for high viscosity flows. | |

Several authors have compared the different methods described in Table 4-2. Schmidt (Schmidt and Egan, 2009; Schmidt, 2013) compared the homogeneous equilibrium model (omega methods) with different nonequilibrium models, i.e. those which have taken into account the boiling delay and/or the slip between phases, with more than 3000 points of experimental data. The best model is the Diener/Schmidt homogeneous non equilibrium, but these authors pointed out that near the critical point, the precision of data is very important and the direct integration method of Darby is recommended. Darby (2000, 2002, 2005, Darby et al., spring 2001) also made a comparison between different models for industrial products: ethylene, chlorine and ammonia; the results were that all the equilibrium models and the HNE model (Fauske, 1999) for nozzle lengths greater than 6 inches gave comparable results for conditions not near the critical point. Nevertheless, in the vicinity of the critical point area, Darby does not recommend omega and HNE methods. Another comparison made by Darby et al. (2001) recommends the HNE method (Fauske, 1999) if the slip is included over the TPHEM (Simpson, 1991) method and emphasizes the importance of knowing the physical state of the fluid entering the valve. Sorvari (2008) compared the methods of Leung (1996), API 520 (Part I, 2000), ISO 4126-10 and the Direct Integration method. His comparison showed that the Direct Integration method with a non-equilibrium factor related to vapor mass fraction and nozzle length (Darby, 2002) was the best one. Concerning the discharge coefficient Boccardi et al. (2005, 2008) found that, for two phase flow, the discharge coefficient is higher than for vapour only flow.

Other analytical integral methods have been proposed. Kim et al. (2013) proposed a universal mass-flux equation for sizing pressure relief valves for liquid flow, vapor flow and two-phase flow for critical and subcritical conditions; it is based on the work of Simpson (1991).

Siegal et al. (2013) recommended the Direct Integration method of Darby in all cases. They argue that with the availability of commercial simulators there are no concerns when doing the isentropic flashes through Aspen Hysys, for instance, and extrapolating the results to a spreadsheet which performs the integration with the Trapezoidal Rule, as recommended by API 520 (Part I, 2008). According to these authors, only in the case of two-phase frozen flow (no flashing) or if the valve has a short nozzle ( < 10 cm) with flashing flow, does the method not give precise results and must be avoided. They pointed out that the use of a slip factor can solve the problem but there are no guidelines available for calculating this factor.

Here the Direct Integration method of Darby has been used in all cases of supercritical fluid with retrograde condensation, together with Ouderkirk (2002) and Self and Do (2010) procedures.

Concerning the discharge coefficient for two-phases, the values recommended by API 520 (Part I, 2008) have been used in this work as a conservative basis. Leung (2004) gives a procedure in which the discharge coefficient in two phase compressible flow is related through the omega parameter to the liquid coefficient. Darby (2004) proposes a method which is independent of the model used to size the valve. He recommends taking the certified liquid coefficient when the flow in the valve is not chocked, and using the value of the gas discharge coefficient when the flow is chocked using the Direct Integration Method. ISO 4126-10 (2010) also gives a method which introduces a weight factor based on the volume fraction of the gas phase between the experimental liquid and gas coefficient.

### 4.3.5 Supercritical flow with retrograde condensation

A pressure relief valve protecting a vessel can, in case of an external fire, release a supercritical fluid if the relieving pressure of the valve is higher than the critical point. No boiling process occurs in this case.

Fluids near or above their critical pressures have zero latent heat and behave somewhat like vapors. Through an open literature review, the following methods are available:

a) Treating the vessel as unwetted and using an empirical formula of API 521(2008).
b) Using the value of 115 kJ/kg as a latent heat for hydrocarbons near the critical point.
c) Using the method of Francis and Shackelton (1985). This method requires that the energy coming into the vessel from the fire be balanced by the energy leaving the vessel with the released stream at a constant vessel pressure. The method considers the heat absorbed by the vessel and the fluid and heat transfer from the vessel to the fluid by natural convection.
d) Using the method of Ouderkirk (2002).
e) Using the method of Doane ( 2010).
f) Using the method of Self and Do (2010). This method can be used for both vapors and liquids if no phase change occurs. In this approach, the relieving rates are calculated with a time basis (15 seconds in the example). Nezami and Price (2012) worked with this procedure in some case studies using both temperature and the time increments. This method was adopted in the last edition of API 520 (Part I, 2014)

Grolmes (2013), in a detailed study, demonstrated that the method of Ouderkirk was better than Doane's.

The Ouderkirk and API 520 methods have been used here together with a commercial simulator (Aspen-Hysys) to obtain the isenthalpic/isentropic data.

### 4.3.6 Liquid thermal expansion

Liquid filled systems must be protected against overpressure due to the hydraulic expansion of the fluid, when the system is blocked in and exposed to an outside heat source, for instance when:

- the cold side of an exchanger is blocked in while the hot side continues to flow.
- liquid filling piping or vessels are blocked in and heated by heat coils or tracing.
- liquid filled piping or vessels are blocked in at a near ambient temperature and heated by solar radiation.

API 521 (2008) gives a formula for calculating the thermal expansion of liquids:

$$q = \frac{\alpha_v \cdot \emptyset}{1000\, d\, c} \tag{4.17}$$

where

q is the volume flow rate at the flowing temperature, $m^3/s$; $\alpha_v$ is the cubic expansion coefficient for the liquid at the expected temperature, 1/ºC; $\emptyset$ is the total heat transfer rate, expressed in watts; $d$ is the relative density referred to water at 15,6 ºC; $c$ is the specific heat capacity of the trapped fluid, J/kgK

If large duties are involved, it is recommended that a less conservative approach be considered, particularly with respect to the rate of heat input. Valdes and Svoboda (1985) indicated that the API method gives very conservative results in the case of heat exchangers. They recommended the formula for the case of blocked in piping. These authors proposed a method called "balanced-enthalpy method", which takes into account the variation of the duty in different segments of the heat exchanger as a function of time.

## 4.4 Inlet pressure drop

A safety valve can, under certain conditions, respond in an unstable manner. The flow rate changes as the pressure at the inlet and or outlet varies during the overpressure scenario. A safety valve may experience three types of dynamic response to different flow rates: cycling, fluttering and chattering.

Cycling refers to a valve that opens and closes multiple times during a relief at a frequency lower than 1 Hz and occurs normally when the system upstream does not generate enough relieving load and the safety valve has overcapacity. Fluttering refers to a high-frequency self-excited periodic oscillation of the valve at a frequency higher than 10 Hz that does not result in the valve completely closing off, i.e. the seat does not contact the disc. Chattering is the rapid cycling, higher than 1 Hz, of a valve opening and closing which may lead to a loss of containment because of the mechanical failure of the relief valve or related piping. Flutter and cycling are not considered significant safety hazards concerning the mechanical integrity part, but chattering causes relieving flow reduction because of insufficient valve opening.

API 520 (part II, 2011, 2015) and AD-Merkblatt-A2 (2006) limit the non-recoverable pressure losses at the inlet to 3% of the set pressure. This is to ensure that the pressure in the protected vessel will not increase above what is allowed by pressure vessel codes and to confirm that the valve will not chatter. This 3% rule assumes that the blowdown of the safety valve is higher than 5 % of the set pressure allowing a minimum of 2% of safety margin.

The origin of this 3% inlet line loss appears first in 1948, when the University of Michigan was commissioned by industry to address relief valve instability. They set an arbitrary guideline for maximal inlet line losses at 3% of relief valve set pressure based on relief valve blowdown which was at that time 4%. Blowdown for a relief valve is the difference in the set or lift pressure and the reseating pressure expressed as a percentage of set pressure. The attached diagram (Figure 4-3) for vapor relief in a vapor certified valve shows blowdown pictorially:

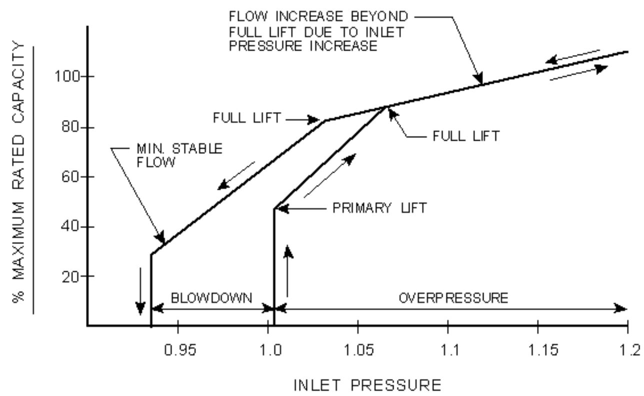



**Figure 4.3. Blowdown in a gas/vapor characteristic pressure relief valve (adapted from API 520, Part I, 2014)**

API 520 Part II included the 3% rule in 1963; mention of this rule was part of the non-mandatory appendix M for ASME Section VIII beginning in 1986 for vapor services only.

ISO 4126-9 adds a new condition to the 3 % rule. The inlet pressure drop should not exceed one third of the blowdown. The lower value has to be taken (this new condition assumes that the standard blowdown is 10%).

The calculation of inlet and outlet line losses for a relief valve may appear to be a simple exercise; however, several assumption errors have been noted in the bibliography (Berwanger et al., 2000; Bravo et al., 1995). Vapor release calculations through a vapor certified relief valve are to be analyzed for inlet and outlet line losses based on the relief valve's rated capacity at 10% overpressure (code allowed accumulation) of the valve. Overpressure is pressure above the set pressure of the valve; accumulation is pressure above the maximum allowable working pressure (MAWP). Overpressure equals accumulation only when the safety valve is set at MAWP. From the diagram above, full lift typically occurs at 5-7% overpressure in the valve.

However, in those cases where the required relief rate is < 50% of the safety valve's rated capacity at 10% overpressure and is > 25% of the rated capacity, a deviation from the codes has been allowed in this work i.e. using 50% of the valve's rated capacity for inlet and outlet analysis. The justification for this deviation is based on the behavior of the pop action vapor certified valve shown above. A vapor certified valve pops to ∼ 50% open and can operate there in a stable manner. At a required relief load < 25% of the valve's capacity, the valve will reseat.

There are two valves which do not require inlet pressure drop calculations: thermal relief valves exposed to ambient temperature (solar radiation) and remotely sensed pilot-operated relief valves.

The calculation of the inlet pressure drop (frictional) has been done here using the following steps:

### 4.4.1 Compressible gases/vapors

Even though there are some simplified methods for doing this calculation (i.e. Cox method (Cox and Weirick, 1980); ISO 4126-9 Annex C; Westman, 1997a; AD-Merkblätt-A2 paragraph 6.2 and API 521 (API 521, 2008, 2014)), there are also rigorous procedures considering isothermal and adiabatic flow for one and two phases, i.e. Aspen-Hysys, PS PPM, Flowmaster, etc.

Anyway, the main issue is finding the rigorous values of resistance coefficients of fittings. Here the values of Crane (1999) have been used.

The roughness of the inner wall of the pipe is very important in situations where the inlet pressure drop is almost 3%. The choice of one roughness or another can decide if a safety valve fulfills the 3% rule or not. Here a roughness of 0.3 mm has been adopted, which corresponds to a lightly corroded pipe (VDI Heat Atlas, 2010).

The calculation of the inlet pressure drop was performed with an own developed spreadsheet, applying the following equation from API 521 (API, Part I, 2008) and Mukherjee (2008):

$$\frac{f.l}{d} = \frac{1}{Ma_2^2}\left[\left(\frac{p_1}{p_2}\right)^2\right]\left[1-\left(\frac{p_2}{p_1}\right)^2\right] - \ln\left(\frac{p_1}{p_2}\right)^2 \tag{4.18}$$

the Mach number being calculated with:

$$Ma_2 = 3{,}23\ \cdot 10^{-5}\left(\frac{q_m}{p_2 \cdot d^2}\right)\left(\frac{Z\cdot T}{M}\right)^{0.5} \tag{4.19}$$

As the PS PPM program was available, the critical cases, i.e. the cases with a frictional pressure drop at the inlet near 3% of the set pressure or higher, were also recalculated with the program. The problem is that PS PPM does not account for piping with DIN diameters, but only with ANSI ones. This forces the process safety engineer to look for a schedule which match the real inside diameter of the piping and fitting.

### 4.4.2 Liquids and two-phase flow

For liquids, the equations of Crane (1999) have also been used. The same problems related to resistance coefficients of fittings and PS PPM program apply.

For two-phase flow the homogeneous equilibrium model of Leung, described in API 521 (2008, 2014) and available in the dedicated software PS PPM, has been used.

## 4.5 Outlet pressure drop

Pressure relief streams must be disposed of in a safe, economic and environmentally acceptable manner. The alternatives are (Shell, 2004; Cheremisinoff, 1998; section 7.3, API 521, 2008; Casal et al., 1999):

- Discharge to atmosphere, grade or open drainage. This option is usually applied for steam, water, air and nitrogen, among others
- Discharge to sewer. Used for liquid discharges: the sewer system must have capacity to contain the volume of the released discharge
- Discharge to a process vessel. This method returns the released fluid back into a process system operating at lower pressure than the system from which it is released

- Discharge to a closed system. This system may treat or cool the release stream and either recover some or all of the material or route it to a remote location, where it can be safely disposed of. This type of system can be a liquid blowdown one, a flare or a vapor recovery system.

Due to the fact that the products of the petrochemical plants studied are mainly hydrogen, ethylene, propylene, propane and n-heptane, in this case the best option is a flare system. A flare system consists of a suitably designed combustion assembly, liquid knock-out and the piping networks which connect them to the pressure relief valves.

The design of flare systems consists of the following major activities (API 521, 2007, 2014):

a) Identification and quantification of the maximum loads which may enter the system.
   This activity requires assessment of the causes of release into a flare system and determination of which flows are likely to occur simultaneously. Typical emergency situations to be considered are (Cheremisinoff, 1998):
   - Single elements failures (called individual loads), such as instrument device failure, mechanical device failure, etc.
   - Area covered by fire (defined in this thesis as a circle with a diameter of 30 m)
   - Operation of emergency depressurizing systems, for instance the plant reactors
   - Plant wide utility failures (called cumulative loads): total power failure, cooling water failure, instrument air failure, etc.
b) Sizing the flare header piping.
   This is typically an iterative process based on flow requirements, hydraulic back pressure requirements and economics.
c) Mechanical design of flare header piping and supports.
   The piping must be supported to resist forces of potential liquid slugs and thermal stresses caused by flow of hot or cold fluids. Low points and traps should be avoided to prevent collection of liquid or slug formation.
d) Sizing of liquid knock-out and removal systems ahead of the flare.
   The liquid separator vessel in a flare system prevents liquid from reaching the flare itself. Flare separator vessels must be sized to be large enough to collect all of the liquid which might be released to a flare during an emergency, and still have sufficient vapor space to allow incoming vapor and liquid to disengage.
e) Sizing and selection of the flare itself.
   This work is done by qualified flare manufacturers.

Here, due to the fact that the flare system was already built, the procedure for calculating the total backpressure of the safety valves relieving to the flare was the major concern. A rigorous procedure is required to calculate this backpressure, taking into account the possibility of two phase flow at the exit of the pressure relief valves, due to propylene liquid flashing and retrograde condensation as well.

An extensive literature survey was performed both for one phase flow, mainly vapor, and two phase flow.

### 4.5.1 Vapor phase pressure drop in flare networks

The following procedures are available in open literature:

a) API 521 (API, 2007, 2014) gives an equation for compressible and isothermal flow based on the work of Mak (1978). This author pointed out that the assumption of isothermal flow conditions is not a concern, because the real flow in relief systems usually takes place somewhat between adiabatic and isothermal. Mak found that for critical and subcritical flows, the isothermal flow equation gives more pressure drop than adiabatic flow and concluded that for K's greater than 10, the difference is less than 4% in both subcritical and critical flows. Mukherjee (2008) uses this method in a worked example, calculating the built-up back pressures of a flare network.
   API 521 also gives an alternative method based on the work of Lapple. It requires a stepwise calculation because the compressibility factor has to be taken at flow conditions, although a

simplification using an average compressibility factor is proposed. This method, however, assumes that there are no enlargements or contractions.

b) AD Merkblatt-A2 (2006) gives a formula and graphs for compressible flow but in terms of the maximum permissible coefficient of resistance (K), accepting maximal backpressures of 15% for conventional valves and 30% for balanced valves. A method is proposed as well for calculating the back pressure for a specific K, in an iterative procedure.
c) ISO 4126-9 Annex D (2008) gives in five graphs the maximal K values for built-up backpressures of 10%, 15%, 20%, 30% and 40%, assuming adiabatic compressible pipe flow of an ideal gas with constant specific heat ratio; this method is based in the work of Lapple.
d) Other authors have made comparisons between different methods. Bonilla (1978) compared adiabatic and isothermal flow. He concluded that both isothermal and adiabatic flow equations give safe design but this does not happen with low temperature gas flow. Kern (1975) presented a worked example of the rupture of a tube of a heat exchanger to show the calculations of adiabatic and isothermal flow, and concluded that in this example, with the L/D ratio of 390, the difference between the two flows was negligible, i.e. 0.2%. Westman (1997b, 1998) presented an adiabatic model for design of relief header systems based on the work of Lapple. He recommended the use of a computer to solve simultaneously all the equations derived for the ideal adiabatic flow. More recently, Kumar (2002) made a comparison for critical flow taking into account adiabatic flow with the Crane (1999) graphical approach for compressible flow, after modeling the equations through a spreadsheet. The results are similar, although the author pointed out that "reading a plot is an imprecise exercise". Walters (2000) presented complex equations for the case when the compressibility of the gas cannot be ignored, and pointed out that the error when treating a pipe as adiabatic when there is heat transfer, can be sizable.
e) Interesting commercial software codes for flare systems are: Aspen Tech's Flare System Analyzer (Aspen, 2011) and SimSci-Esscor Visual Flow (SimSci's, 2014) and for pressure drop calculations: Aspen Plus, Aspen Hysys, Superchems, Flowmaster, Honeywell Unisim Process Pipeline Modeler, Hydrosystem, PS Pressure Protection Manager, etc.

### 4.5.2 Two-phase pressure drop in flare networks

If the flare network includes two-phase flow, the pressure drop calculation is more complex. Special emphasis has been applied to the research of the best method, as this is the type of flow found in the case studies of this thesis.

A lot of petrochemical and engineering companies used over the last 30-40 years the method of Dukler's homogeneous flow, which can either be used for hand calculations with a pocket calculator, or with a scientific programming language and a personal computer. Dukler's method begins by checking the flow pattern for horizontal flow (through the Baker's two-phase flow map for horizontal pipes) and for vertical upward flow (through the Oshinowo-Charles two phase flow map for such a flow), in order to avoid slug and mist flow. The acceleration losses were calculated only if a) $\Delta P > 10\%$ of the known inlet absolute pressure, b) mixed phase velocity $V_l + V_g > 30.5$ m/s, and c) there was substantial flashing in the line. These conditions are commonly found in relief networks and this term ought to be always calculated.

It is commonly accepted that Duckler's procedure has an estimated accuracy of 20% for systems other than steam-water. For steam-water systems the Martinelli-Nelson pressure drop correlation (a modification of the Lockhart-Martinelli method) has an accuracy of 30%. These accuracy figures give an idea of the difficulty in finding a universal and exact method for two-phase pressure drop calculations. As a matter of fact, Barua et al. (1992) pointed out that in flare networks designs more than 50% of the pressure drop resulting from acceleration is not uncommon.

As stated by Brosius and Dial (1997), many of the pressure drop calculation methods for two-phase flow that have been developed over the past several years have their roots in the production and pipeline industries, where the most significant problem is the best calculation of liquid holdup; additionally, the fluid velocities are normally low to reduce compression and pumping costs. In these cases, the acceleration losses are almost always considered to be negligible, friction and elevation changes being the major components of the total pressure drop. However, when modeling relief systems, the fluid may reach very high velocities, sometimes higher than 0.5 Mach. At these velocities the liquid is often entrained in the vapor phase; thus, the overall fluid may be considered as an homogeneous mixture of both phases moving down the pipe.

When selecting a pressure drop method for use in modeling relief systems, the model must focus on the friction and acceleration terms, as opposed to the normal considerations of liquid holdup found in most multiphase flow correlations.

In piping systems, critical flow will occur at a point where there is an expansion or enlargement of the cross-section, such as a change in pipe diameter or the entrance of a feeder pipe into the main header. Once the line is operating in critical flow, the pressure must be increased dramatically for any additional flow to be discharged through the line.

It has to be noted that one of the key factors for critical flow in multiphase fluids is that, for certain ranges of liquid fraction, the velocity at which the flow will become critical can be substantially lower than the sonic condition for the gas phase alone.

Key design factors when considering critical flow are noise and vibration. A known expert in this field, Shackelford (2013), wrote “I have seen a few situations in which the momentum forces straighten elbows or cause pipe to jump off the rack”.

A Siemens research (Köper and Volbrecht, 2011) emphasizes that the selection of the fluid properties model has a big influence on the overall accuracy of the relief system model. Incorrect density or viscosity calculations can lead to extreme errors in the overall pressure drop. For instance, fluids that exhibit the behavior of retrograde condensation may move in and out of the two-phase region within the boundaries of the relief system. It is known that in lines that operate at or near critical flow, the addition of a small amount of liquid will significantly reduce the flow capacity of the line. Köper and Volbrecht work was performed in order to find the best correlation, taking into account the different models available in the commercial simulator Aspen Plus v7.1. Its pipe block is based on the well-known HTFS: the cases and results were checked with the experimental data from Prof. Friedel, from the Hamburg-Harburg University. The results were:

a) For horizontal flow, the best models are HTFS (optimistic) and Beggs-Brill (conservative). Beggs-Brill gives a correct physical description of the influence of surface roughness.
b) For vertical flow HTFS and Beggs-Brill were also the best ones (not recommended: Slack, Orkiszewski and Angel-Welchon-Ros).

Unfortunately, this research group did not compare Aspen Plus with other software codes such as: Aspen Hysys, Superchems, Flowmaster, Honeywell UniSim Process Pipeline Modeler, Hydrosystem, PS Pressure Protection Manager, etc.

API 521 (2014), part 5 “Disposal Systems”, gives general recommendations for two-phase flow in the inlet and outlet piping of pressure relieving devices. One is the use of Beggs-Brill model, but with an adjustment in the acceleration term, due to the velocity frequently found in flare headers. The other one is the “homogeneous equilibrium method” of Leung (1987, 1996); critical flow conditions are typically handled by assuming homogeneous flow and by applying basic thermodynamic relationships. An example of using this model is presented in the Leung paper (1996).

CCPS (1998) also gives free software for two-phase flow, i.e. TPHEM. Comparisons of case studies were made by Adair and Fisher (1999) with reasonably good results.

The conclusions of this research are:

1) In terms of a practical approach, it is better to prevent (or significantly reduce the likelihood of) the release of liquids or two-phase fluids into the flare header. Any situation involving two-phase flows in flare networks, particularly at high velocities and high liquid fractions, are problematic. The multiphase pressure drop calculations generally used are mostly empirical and have a strong dependence on the flow regime map, and the flow regime tends to be not only dependent on velocity and liquid fraction, but also on previous flow regime (hysteresis).

2) Commercial two-phase flow simulation programs generally use an equation of state to predict vapor/liquid equilibrium, and afterwards apply a two-phase pressure drop equation with an adjustment in the acceleration term, due to the high velocity, to calculate the line pressure drop.
3) There is no universal model available for allowing an exact calculation of two-phase pressure drop for the pressure relief valve inlet pipe or outlet pipe to the tip of the flare. This is because the high velocity and/or critical flow is combined with the whole quality range from 0 to 1, in all possible flow directions (horizontal and vertical pipes) and all flow regimes (from bubble to mist pattern) (Westphal, 2013). That is why API 521 recommends the Leung's "homogeneous equilibrium model".
4) There are still four big areas that have not been solved in the two-phase pressure drop calculations: a) pipe wall roughness influence; b) pipe fitting losses calculations (90º elbows, tees, etc.); c) thermodynamical non-equilibrium flow calculation and two-phase critical flow, and d) pipe networks with flow separation in T-junctions (Wallenhofer and Muschelknautz, 2010).
5) The recommended method is the Beggs-Brill one if a commercial simulator is available. Without this option, the Leung's homogeneous equilibrium method can be solved by hand if the thermodynamic properties are available.

## 4.6 Stability calculations: engineering analysis

Relief valve instability results from a combination of static non-recoverable inlet line frictional losses and acoustic wave phenomena. The static piece of this equation has been present in standards API 520, Part II and ASME, Appendix M, since 1963 and 1986 respectively. Only recently has the more complex phenomenon of acoustic wave interaction been studied.

Instability can be sorted into three discrete behaviors: chattering, fluttering and rapid cycling. Chattering is the most important one, because it can cause severe damage to upstream equipment, especially for liquid flow. It can damage valve components due to strong forces, for example in large valves and/or with high set pressures as well. It can also cause loss of containment leading to fire, explosion and/or toxicity risks. Figure 4-4 shows an example of a safety valve having suffered chattering.

**Figure 4-4. Typical damage in the disc and the seating surface of a safety valve because of chattering.**

Melhem (2013) considered chattering, fluttering and cycling probability as follows:

a) Chattering is most probably in vapor/gas service, especially for large valves and/or for valves in high pressure service.
b) Chattering is least likely in flashing two-phase flow and liquid flow.
c) Fluttering will always occur in liquid service with pop action valves (fast opening).
d) Cycling is most likely to occur in flashing two-phase flow.

He also points out that the piping damage is more probably in liquid services during chatter or flutter.

According to the work of Smith et al. (2011), API 520 (part II, 2015) and Melhem (2014), among others, related to causes that produce chattering, the following scenarios have to be checked:

Excessive inlet valve pressure loss (well in excess of blowdown): when a pressure relief valve opens, the pressure acting on the disc of the valve will be decreased due to the pressure drop generated by piping and fittings. If this pressure drop is large enough, the valve inlet pressure may fall below the reseating pressure, forcing the valve to close and reopen immediately since the static pressure will be above the set pressure.

Historically, the concept of limiting the inlet pressure drop was established in the API 520 Part I of the 1963 edition based on the report generated by Professor Katz and colleagues of the University of Michigan in 1948. They set an arbitrary guideline for maximum inlet line losses at 3% of the set pressure based on relief valve blowdown at that time of 4%. For the first time, in the 1994 edition of API 520 Part II appeared the concept of “Engineering Analysis” allowing an inlet pressure drop greater than 3%. Nevertheless, no indications are given about how to perform it. ASME Section VIII Appendix M (non-mandatory) introduced the 3% rule in 1986.

In Europe, before the European Code Directive was issued (PED 97/23/EC), each country had its local code: BS (United Kingdom), ISPESL (Italy), TÜV (Germany), Stoomwezen (The Netherlands), UDT (Poland), AFNOR (France), etc. Some of these codes had subcodes limiting the inlet pressure drop; for example, AD Merkblatt-A2 (Germany) limited the non-recoverable pressure losses in the inlet to 3% of the set pressure. However, the precondition established by the german code was that the blowdown had to be higher than 5% of the set pressure allowing a minimum of 2% of safety margin.

With the PED 97/23/EC obligatory in Europe since 2002, the ISO 4126-9 was adopted with an additional condition to the 3% rule: the inlet pressure drop should not exceed 1/3 of the blowdown. The lower value has to be taken (this new condition assumes that the standard blowdown is 10%) and added “in all cases, the difference between blowdown and pressure drop to the valve inlet shall be at least 2% of the set pressure”. However, there is not a concept similar to that of “Engineering Analysis” in the ISO 4126-9.

A new procedure has been developed here to perform an “Engineering Analysis” to pressure relief valves with an inlet pressure drop greater than 3%. It has been described in detail in section 6.7.

Excessive build-up back pressure: when a conventional pressure relief valve opens, the discharged fluid generates a built-up back pressure in the outlet piping and, as a result, a force acts on the valve disc tending to close it. If this return force is large enough, it may force the valve to close completely, only to reopen immediately when the flow has stopped. This is a common phenomenon of instability.

To prevent this instability, historical design practice for conventional PRV discharge has been limiting the built-up back pressure to the valve’s allowable overpressure: 10-15% for conventional safety valves. To solve this problem, a balanced valve is often used which allows a built-up back pressure of 30-50% of the set pressure.

Acoustic interaction (pressure surge rule): this phenomenon has been thoroughly studied by Friedel and coworkers (Frommann and Friedel, 1998, 2000; Cremers et al., 2001; Cremers and Friedel, 2003) among others.

The problem arises when the pressure of the vapor/gas in a vessel increases due an upset which generates the rapid opening (popping) of the safety valve and a longitudinal decompression wave travelling towards the vessel nozzle. Thus, the initially resting fluid accelerates against the wave expansion direction by the pressure gradient, i.e. towards the valve; because of this, the pressure in the valve inlet decreases due to wall friction and the expansion wave generated. This produces a deceleration of the open movement of the disc of the safety valve. If this pressure is higher than the blowdown (reseating pressure difference), the valve disc movement would be stopped or reversed. If stopped, a compression wave is in turn generated and emitted, similar to the expansion wave produced in the opening. In the meantime, the initial expansion wave is either partly reflected at changes in the pipe area of the inlet line and/or finally at the vessel nozzle, acting as an infinite cross section enlargement leading to a compression wave. The superimposition of the returning reflected wave and this new wave produces an oscillating pressure at the valve inlet. If the pressure rise at the valve inlet is sufficiently large, the valve opens again and another expansion wave starts travelling towards the vessel. This variation in disc motion direction can repeat at short time intervals because the acoustic pressure waves are recoverable, producing chattering.

The so called Pressure Surge Criterion/Rule, initially developed by Föllmer (1992), takes into account the concept that, in order to avoid chattering, the returning pressure wave should contact the disc of the valve before it begins to close. This can be expressed as:

$$t_0 > \frac{2.L}{c} \tag{4.20}$$

where

L is the inlet line acoustic length, m; c is the speed of sound in the fluid, m/s; $t_0$ is the opening time for the PRV, s.

The calculation of $t_0$, opening time of the valve, can be done according the method of Cremers et al. (2001) or as in Grolmes (2011). Here the model of Cremers has been used because of its simplicity, as the Grolmes model requires disk area, nozzle area, spring constant, weight in motion of the valve (disk mass and spring mass), etc. These parameters are not usually provided by the manufacturer and are difficult to estimate. A comparison of both methods has been made by Melhem (2014), showing a relatively good agreement.

The Cremers expression for the opening time is:

$$t_{open} \sim \left[0.015 + 0.02 \frac{\sqrt{2d_{PSVi}}}{\left(\frac{P_s}{P_{ATM}}\right)^{2/3} \left(1 - \frac{P_{ATM}}{P_s}\right)^2}\right] \left(\frac{h}{h_{max}}\right)^{0.7} \tag{4.21}$$

Through a bibliographical review, Smith et al. (2011) pointed out that the initial valve lift is in the range 40% - 100% of its full lift. The fixed value of 60 % has been adopted here for the stability calculation at the initial open for the gas/vapor relieving.

Another required parameter is the speed of sound in the fluid c; considering a perfect gas its value is:

$$c = 223 \sqrt{\frac{kT}{MW}} \tag{4.22}$$

where k is the isentropic expansion factor ($C_p/C_v$ for an ideal gas), dimensionless; T is the temperature of the gas, ºR; MW is the molecular weight of the gas; c is the speed of sound, ft/s.

This expression has been used here. A more precise expression would be:

$$c = \sqrt{\left(\frac{\partial P}{\partial \rho}\right)_s} \tag{4.23}$$

For liquids, the value of c is:

$$c = \left(\frac{K_s}{S}\right)^{0.5} \tag{4.24}$$

$K_s$ being the isentropic bulk modulus of elasticity for the fluid and $S$ the specific gravity of the fluid at relieving conditions. This value has been calculated when required through Aspen-Hysys.

Nevertheless, there is some uncertainty in the speed of sound when it is calculated in liquids and two-phase flow. The piping flexibility can lower the value of the speed of sound, and adding for instance a 0.1 % by volume of gas in a liquid can lower the speed of sound by a factor of two.

Combining equations 4.20 and 4.22

$$L < 111.5\, t_{open} \sqrt{\frac{kT}{MW}} \tag{4.25}$$

Smith et al. (2011) rearranged the original equation developed by Frommann and Friedel (1998) for the maximal length of the inlet pipe; they assumed a sudden reduction of the pressure as a consequence of a generated expansion wave of 20% of the set pressure (as assumed previously by Föllmer for gases) and that a blowdown of 10 % will not produce an inversion during the disc opening due to its inertia for gas/vapor flow.

$$L_i = 9078 \frac{d_i^2}{w_{\%O}} (P_s - P_B) t_0 \tag{4.26}$$

Smith et al. (2011) rearranged another expression of the same authors for the maximal length, assuming that the sudden reduction in pressure is the blowdown:

$$L_i < 45390 \frac{d_i^2}{w_{\%O}} \left(\frac{P_s - P_{rc}}{P_s}\right) (P_s - P_B)\, t_0 \tag{4.27}$$

Cremers et al. (2001, 2003) proposed another model by introducing the backpressure in calculations. This method has not been used here because, after a comparative analysis, we found that it is too much conservative.

Another constraint for the length of the inlet piping comes from the work of Singh (1982, 1983). In order to avoid chattering or fluttering, the following equation related to total pressure losses in the inlet pipe has to be fulfilled:

$$P_s - P_{RC} > \Delta P_{Total} = \Delta P_{Frictional} + \Delta P_{Acoustic} \tag{4.28}$$

where $\Delta P_{Acoustic}$ after rearrangement can be calculated as follows (Smith et al., 2011):

$$\Delta P_{Acoustic} = \frac{L w_{PSV}}{12.6 d_i^2 t_0} + \frac{1}{10.5\rho} \left(\frac{w_{PSV} L}{c d_i t_0}\right)^2 \tag{4.29}$$

This expression can be applied only if the reflected wave reaches the valve before it is fully open. Singh developed formulas for the case in which the valve fully opens before the reflected wave reaches it. Melhem (2014) used the equations proposed by Singh in his model; however, Singh multiplies the frictional pressure drop for incompressible flow by a factor that decreases the pressure drop due to the reflection wave (Darby, personal communication, 2014).

Recently, Melhem (2014, 2015) has introduced a simple force balance on the disk to avoid chattering. This equation can be used for relatively simple geometries; it requires an estimate of opening/closing time of the valve and blowdown. This “Simple force balance” methodology has been adopted by the last edition of API 520 (part II, 2015). The main problem for the practicing process safety engineer is to know the exact value of the blowdown.

For conventional safety valves:

$$P_{Source} - \Delta P_{friction,wave} - \Delta P_{wave} - \Delta P_{back} > P_{close} \tag{4.30}$$

For balanced valves (assuming that the bellows protect only 90% of de disk surface):

$$P_{Source} - \Delta P_{friction,wave} - \Delta P_{wave} - \Delta P_{back} \cdot 0.1 > P_{close} \tag{4.31}$$

This expression has been used here in the checking for chattering for critical safety valves.

For liquids the same phenomena explained for gases/vapor applies, with the difference that chattering occurs only due to the oscillations in pressure (incompressible fluid) caused by the decompression/ compression wave.

Smith et al. (2011) presented for this case the expression:

$$L_i < 0.55\, t_0 \left(\frac{K_s}{\rho}\right)^{0.5} \tag{4.32}$$

where $K_s$ is the isentropic bulk modulus of elasticity.

Nevertheless, the following equations have to be followed as well:

$$P_s - P_{RC} > \Delta P_{Total} = \Delta P_{Frictional} + \Delta P_{Wave} \tag{4.33}$$

$$\Delta P_{Wave} = \frac{c\rho}{4636.8}\,(V_0 - V_F) \tag{4.34}$$

As with gases, these equations have to be verified for opening, full flow and closing conditions.

Flow induced vibration

Baldwin and Simmons (1986) found that flow-induced vibration was the cause of chattering, premature pop off or vibrating of safety valves, causing fret, gall o fatigue in its internal parts. The mechanism is related to high velocity flow past a cavity, in the same way that the stub of a closed safety valve creates vortices which can couple with an acoustic resonance of the stub.

These authors point out that small vortex pulsations can be amplified up to the order of 1400 kPa peaks. Such pulsations add peak pressures to the disc of the valve that exceed the valve set pressure and tend to lift the valve (Figure 4-5 shows the mechanism of flow induced vibration).

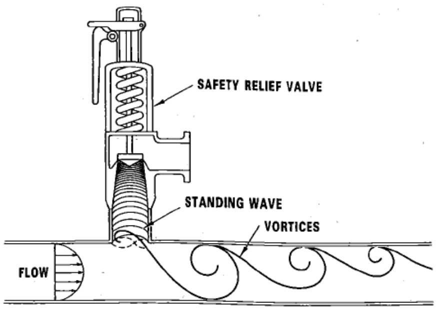



**Figure 4-5. Flow induced vibration in a pressure relief valve (gas phase) (taken from Melhem, 2012).**

They give the following correlation to avoid flow induced vibration in gas phase:

$$L_i < \frac{d_i c}{2.4\, U} \tag{4.35}$$

where $L_i$ is the length of the stub (inlet pipe), ft; $d_i$is the inlet diameter, inches; $c$ is the speed of sound, ft/s, and U is the process fluid velocity as it passes the PSV nozzle, ft/s.

Retrograde condensation at inlet: in the case that the fluid to be relieved in a process upset is at supercritical condition and the pressure increases up to the set pressure of the valve, the inlet pressure of the valve can decrease -as has been explained before- and retrograde condensation can occur. This condensation could originate a volumetric contraction that might force the valve to close. Once the valve closed, the condensate would flash and the cycle would repeat. This phenomenon can cause chattering. No cases have been found in this study. Figure 4-6 presents an example of retrograde condensation represented in a Mollier diagram.

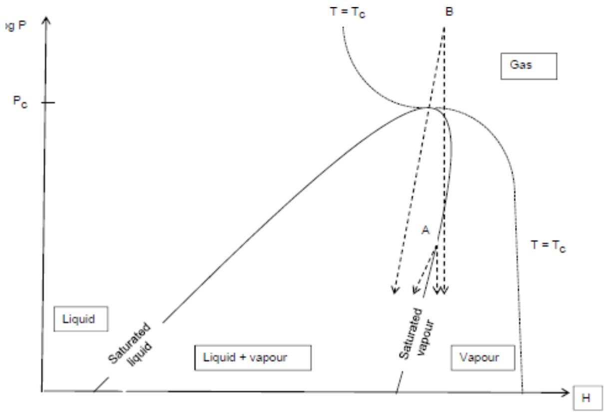



**Figure 4-6. Retrograde condensation process (taken from Egan, 2011).**

<u>Improper valve selection</u>: safety valve selection trim is a very important factor in designing relief systems, in order to avoid possible instability problems.

There are three trims: vapor certified, liquid certified and trims that are dual certified either in ASME Code or in PED Code.

It is known that gas and vapors have different relief characteristics. Until 1985, the ASME code allowed an overpressure of 25% for liquid applications and manufacturers provided the same trim for both gases and liquids resulting in an opening/closing curve as represented in Figure 4-7.

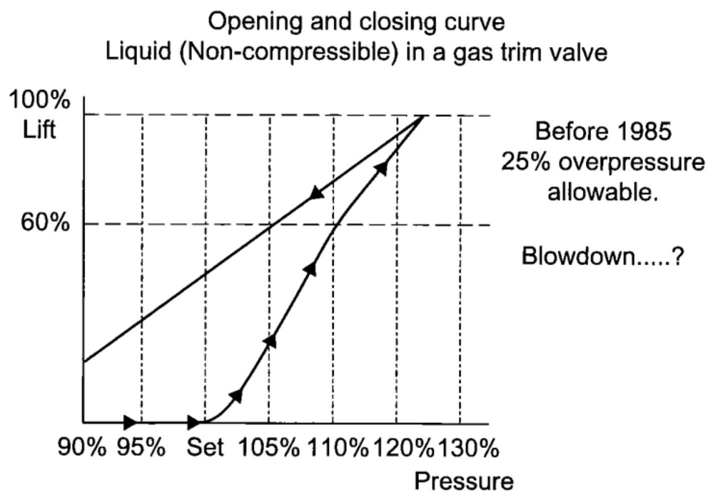



**Figure 4-7. Characteristic curve of a relieving liquid in a gas/liquid trim before 1985 (see how the allowable overpressure was 25% and the blowdown was not defined) (taken from Hellemans, 2009).**

However, since 1985 ASME has also required a maximum overpressure of 10 % on liquid valves. Actually most manufacturers have valves that fit for gases/vapors and liquids.

The problem concerning chattering is related essentially to the case when a valve with a gas trim releases a liquid (Figure 4-8).

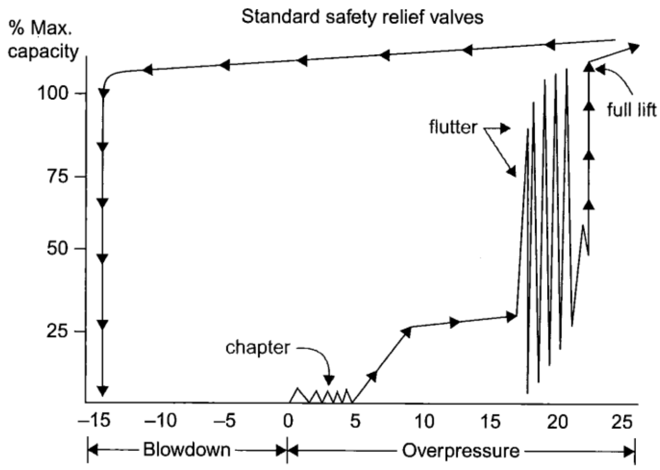



**Figure 4-8. Characteristic of a valve gas trim releasing liquid (taken from Hellemans, 2009).**

Liquid relief through a vapor certified valve should be analyzed for flows that imply 10% overpressure on the valve. Such relief does not achieve stability up to 10% overpressure and the valve does not achieve full lift up to 25% overpressure (see Figure 4-9). Operation below 10% overpressure has been demonstrated to be unstable. Liquid relief through vapor certified valves experiences little to no blowdown, and these scenarios have been noted as the cause for many of the incidents attributed to relief valve instability within the industry.

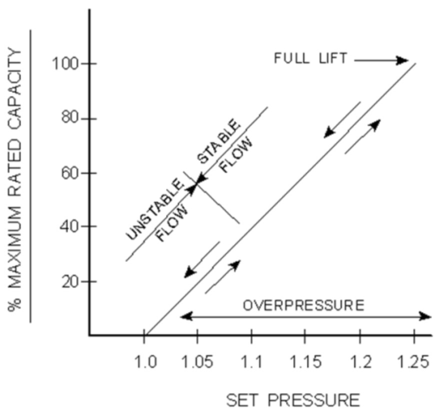



**Figure 4-9. Characteristic of a liquid relief through a vapor certified valve.**

Relief valves that must provide relief protection for both a vapor and liquid scenario should be selected with both in mind. The relief valve should be a dual certified valve for both services.

Liquid releases through liquid trim or dual certified relief valves have to be evaluated for inlet and outlet line losses at the required flow, due to the modulating action of this types of valve. Liquid trim relief valves entered into the industry in the 1980's to address the special characteristics of incompressible flow. They have little to no pop action, modulate flow through and up to full lift of the valve, and have an extended blowdown to account for the effects of liquid hammer (see Figure 4-10)

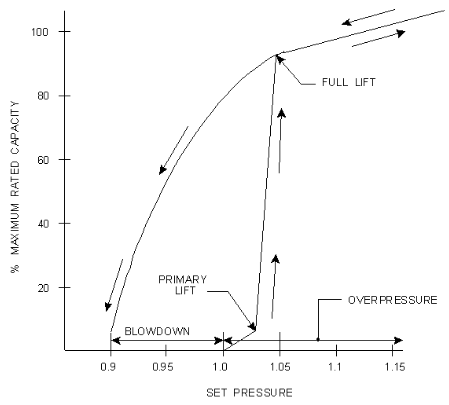



**Figure 4-10. Characteristic of a liquid trim releasing liquid for valves after 1985.**

Oversized pressure safety valve: safety valves relieving gas/vapor with a flowrate less than 25% of rated flow, are candidate to chattering. This fact comes from the characteristics of each safety valve. API 520 (part I, 2008) presents the typical behavior of a safety valve relieving gas (see Figure 4-11):

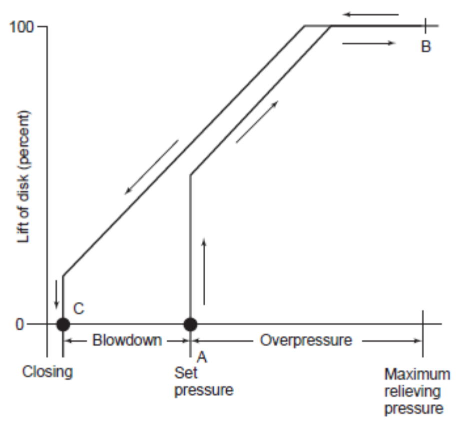



**Figure 4-11. Behavior of a gas valve when the required flow is less than 25% of its rated capacity.**

As shown in the figure, the valve closes at 20-25 % of its rated capacity; thus, another condition for stability in vapor filled systems is:

$$W_{PSV} > 4W_{required} \qquad (4.36)$$

In order to chatter, an oversized valve for gas/vapor service must have a system capable of increasing the pressure in a cycle time of 1 second or less. Assuming a safety factor of 500%, that means a system cycling time of 5 seconds instead of one second:

$$W_{PSV} > 0.2 \cdot V_{System}(\rho_{set} - \rho_{shut}) + W_{required} \qquad (4.37)$$

where

$W_{PSV}$ is the rated mass flow rate, lb/s; $V_{System}$ is the volume protected, $ft^3$; $\rho_{set}$ is the density at the set pressure; $\rho_{shut}$ is the density at set minus blowdown pressure and $W_{required}$ is the required mass flow rate, lb/s.

Outlet area/orifice area ratio: according to API 526 (2009), the ratio output area/orifice area for 4P6 and 6R8 are 4.3 and 3.0, respectively. Due to these small ratios, the build-up back pressure for conventional valves is higher than in other smaller safety valves, giving the same problem of chattering as "Excessive built-up back pressure" presented before. The problem arises because once the valve open, the build-up back pressure resulting from discharge flow results in a force upon the valve disc, forcing the valve to close if the force is sufficiently large, and it will reopen again when the discharge flow has stopped. Instability comes with the repetition of this cycle.

## 4.7 Forces and moments imposed on the pressure relief valve

Reaction forces are generated when a safety valve actuates. When it opens, the reaction forces at the relief valve exit act in a direction opposite to the direction of the discharging flow. To prevent damage to items connected to the relief valve such as pipes and equipment nozzles, it is necessary to calculate these forces (thrust). Knowing their magnitude, appropriate restraints can be installed to avoid accidents.

### 4.7.1 Discharge to atmosphere

API 520 (part II, 2015) gives mathematical expressions for discharge to atmosphere in case of compressible fluids at critical flow at the exit of the tail pipe. The safety valve discharges to the atmosphere through an elbow and a vertical discharge pipe, without any compensation of flow reaction forces (see Figure 4-11). The reaction force (F) includes the simultaneous effects of both momentum and static pressure. Thus:

$$F = 129W\sqrt{\frac{kT}{(k+1)M}} + \frac{(AP)}{1000} \tag{4.38}$$

where

F reaction force at the point of discharge to the atmosphere, N; W flow of any gas or vapor, kg/s; k ratio of specific heats (Cp/Cv) at the outlet conditions; T stagnation temperature at the pipe outlet, K; M molecular weight of the process fluid; A area of the outlet at the point of discharge, $mm^2$; P static pressure within the outlet pipe immediately before terminal expansion to atmosphere, kPag.

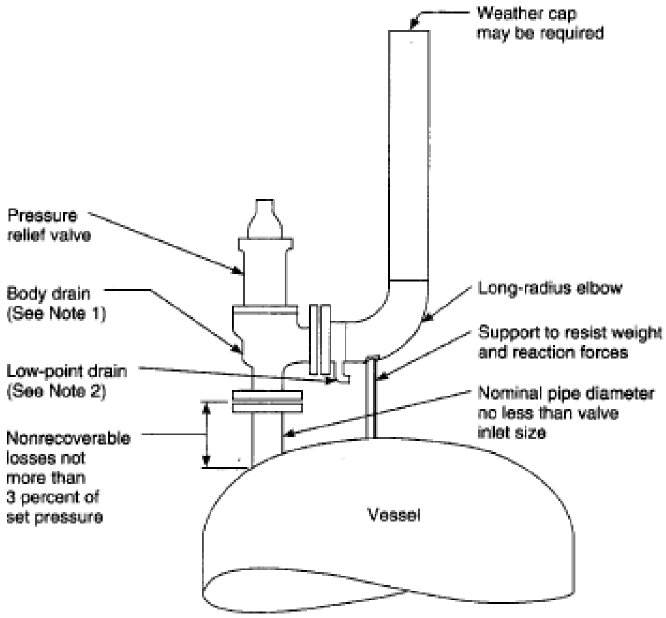



**Figure 4-12. Typical pressure relief valve installation with vent pipe (taken from API 520-II-2011).**

Other available equations for simplified layouts are the annex E of ISO 4126-9 (2009), AD Merkblatt-A2 (2006), CCPS (1998) and Fisher et al. (1992). D'Alessandro (2006) proposes as well equations and charts for the case of not chocked flow.

Muschelnautz and Wellendorfer (2003) developed a model for calculating the reaction forces during the transient phases of opening and closing a safety valve. The Appendix II of ASME Code B31.1 presents a calculation method for dynamic loads.

### 4.7.2 Discharge to flare network

Pressure relief valves which relieve into a flare network usually do not transfer large forces and bending moments to the valve, because changes in pressure and velocity within the closed system components are small. Because there is no simplified analytical techniques to solve this problem, complex calculations including time history analisis to obtain the reaction forces an moments, are required (API 520 Part II, 2015).

White et al. (2012) proposed a screening procedure to determine which system requires a detailed analysis of dynamic loads, with CAESAR II as an example of dedicated software. The restrictive hypotheses for their procedure have been adopted here:

a) All pressure relief valves discharging to flare are adequately supported for an individual release.
b) All liquid and two-phase relief contingencies require detailed analysis.
c) Pressure relief valves sized for external fire scenario only will not require reaction forces evaluation. The effect of the fire due to thermal expansion of the valve and the attached piping is higher than the stresses caused by the flow rate.
d) Pressure relief valves sized and installed for liquid hydraulic expansion only, will not require any reaction force evaluation.

## 4.8 Acoustic induced vibration

The phenomenon of Acoustic Induced Vibration (AIV) is caused by fluid turbulence, and is further enhanced by a flow restriction device such as a safety valve. Problems have been encountered with gaseous systems, since sound energy propagates most easily in compressible media. Liquid relieving systems tend to dampen vibrations and, as a result, have not had any failures to date (Melhem, 2012). The sound Power Level (PWL) quantifies the amount of acoustic energy emitted immediately downstream of the restriction and is calculated using process data such as flowrate, temperature, molecular weight and the pressure ratios across the valve. This energy is usually in the form of a standing wave, which causes vibrations when discontinuities in the piping system are encountered. The piping system response to these vibrations depends on the mechanical natural frequency of the system, which is a function of the material properties, pipe size, support fixity, etc. If the frequency of the vibrations in the system approaches the natural frequency, a resonant condition will cause severe amplification of the vibration. This vibration produces a cycling effect that may result in a fatigue failure. The areas most susceptible to it are branch connections, welded support attachments and other areas of stress intensification (geometric discontinuities). Acceptable PWL’s have been documented based on industry-wide failure data and operating experience. Design of a piping system within these acceptable limits will greatly reduce the risk of a fatigue failure from AIV.

Reviews of the state of the art have been made by Melhem (2012) and Swindell (2013). Melhem reported that “According to the UK Health and Safety Executive (HSE), 21% of all piping failures offshore are caused by fatigue/vibration”

Swindell shows an example of a failure occurred during blowdown after a few minutes (see Figure 4-13).

**Figure 4-13. A failure occurred during blowdown (taken from Swindell, 2013).**

The methods available to screening an Acoustic Induced Vibration problem are:

a) Experience based: D method (Carucci and Mueller, 1982) and D/t method (Eisinger,1997). The method of Eisinger has been adopted by the NORSOK standard (NORSOK, 2006).
b) Experience based: Mach number method. A common criteria used in the process industry is based on limiting the Mach number: 0.3 < Mach < 0.9, and limiting the kinetic energy of a gas: $\rho \cdot u^2 < 1 \cdot 10^5 \frac{Kg}{ms^2}$ and two-phase flow to $\rho \cdot u^2 < 5 \cdot 10^4 \frac{Kg}{ms^2}$
where
ρ is the density in kg/m$^3$ and $u$ is the velocity of the fluid m/s.
c) The Energy Institute method (2008).
d) Detailed structural dynamic methods.

Here the method of Eisinger (1997) has been used in critical safety valves relieving to the flare network. The mathematical expression used is:

$$PWL = 10 \log_{10}\left[\left(\frac{\Delta P}{P_1}\right)^{3.6} \cdot W^2 \cdot \left(\frac{T_1}{M}\right)^{1.2}\right] + 126.1 \qquad (4.39)$$

$$PWL_{allowable} = 173.6 - 0.125\frac{D_i}{t} \qquad (4.40)$$

where

PWL is the sound power level in dB (with reference $10^{-12}$ W); $P_1$ is the upstream pressure of the safety valve at relieving conditions, barg; ΔP is the reduction in pressure, bar; $T_1$ is the upstream pressure, K; W is the gas flow, kg/s; W is the molecular weight, kg/kmol; $D_i$ is the internal diameter of the exit pipe, mm; t is the pipe wall thickness, mm.

## 4.9 Noise

As fluid passes through the safety valve and tail and pipe header, significant noise could be generated and transmitted along the system: one of the common safety requirements is to limit this noise level to 115 dBA (noise level with A-weighted) during intermittent emergency relief scenarios.

ISO 4126-9 Annex F (2008) gives a procedure for calculating the noise level of safety valve discharging to atmosphere. The sound power level of the safety valve, $P_{WL}$, expressed in dB, can be estimated with the following equation:

$$P_{WL} = 20\log_{10}(10^{-3}d_A) - 10\log_{10}\nu + 80\log_{10}u - 53 \qquad (4.41)$$

where

$d_A$ is the internal diameter of outlet pipe, mm; $v$ is the specific volume of the stream at relieving pressure and temperature, $m^3/kg$; $u$ is the velocity of fluid in outlet pipe, m/s.

The sound pressure level, $P_{SLr}$ , expressed in dB, at a distance r from the point of discharge to the atmosphere can be estimated with the following equation:

$$P_{SLr} = P_{WL} - 10\log_{10} 2\pi r^2 \tag{4.42}$$

where r is the distance from noise source, m.

API 521 (2007) gives an expression for calculating the noise:

$$L_{30(100)} = L + 10\log_{10}(0.5q_m \cdot c^2) \tag{4.43}$$

where

$L_{30(100)}$ is the noise level at 30 m from the point of discharge to the atmosphere, dB; L is the noise level, dB; $q_m$ mass flow through the valve, kg/s; c is the speed of sound in the gas at the valve, m/s.

API 521 gives a diagram (see Figure 4-14) showing L as a function of pressure ratio (absolute static pressure upstream from the safety valve divided by the absolute pressure downstream of the valve while relieving). In this work, the noise emission has been calculated by both methods in the critical safety relief valves.

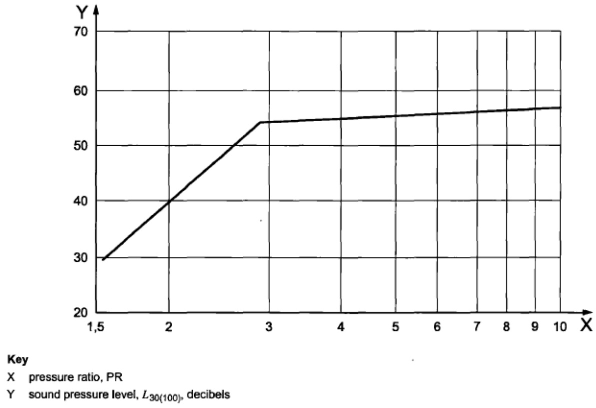



**Figure 4-14. Sound pressure level at 30 m from the source (taken from API 581, 2007).**

## 4.10 Body bowl choking

Body bowl choking occurs when the pressure in the body of an unbalanced safety valve causes a critical flow condition (sonic velocity) at the valve outlet (D'Alessandro, 2011)

In this case, the pressure in the valve body rises independently of the back-pressure at the safety valve outlet piping. This pressure rise in the valve body could result in insufficient opening and/or unstable motion for the conventional spring loaded pressure relief valve, in which the pressure rise in the valve body directly affects the valve disc motion. This phenomenon occurs in larger safety valves (8T10, for example) and higher set pressures.

For stable operation, the stagnation pressure (relieving pressure) should be less than the value given by the following equation (D'Alessandro, 2011):

$$P_0 < \frac{P_c}{(1+F_0)\frac{A_n}{A_e}\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} - F_0} \tag{4.44}$$

the ratio $F_0$ being defined by:

$$F_0 = \frac{P_0 - P_s}{P_s - P_c} \tag{4.45}$$

where

$A_n$ throat area, mm$^2$; $A_e$ outlet area, mm$^2$; $P_0$ relieving pressure, bara; $P_s$ set pressure, bara; $P_c$ superimposed backpressure, bara; k heat capacity ratio for ideal gas.

Izuchi (2015) proposes a new model based on the parameter "PRV relief valve body pressure to set pressure ratio". It is calculated as the pressure difference between those in the PRV body and the PRV outlet divided by the set pressure. The author gives two calculation methods for this parameter and compares them with experimental results, with good results. Izuchi concluded that a conventional spring loaded PRV becomes unstable if the calculated parameter becomes equal to or larger than 15%.

## 4.11 Assignation of the first turnaround interval

API 581 (chapter 7, 2008), is the risk-based inspection method chosen in this thesis, for calculating the first revision time of pressure relief valves. A study was performed applying this procedure to all the safety valves of the critical equipment of the three petrochemical plants studied (Albornà, 2014).

API 581 studies two scenarios in case of pressure relief valves: failure on demand and leakage. In the failure on demand case, which means that the valve does not open when required causing the rupture of the protected equipment, the consequences are much more severe than in leakage. For this reason, the scenario of leakage has been not analyzed here. The process of applying API 581 is resumed in Figure 4-15.

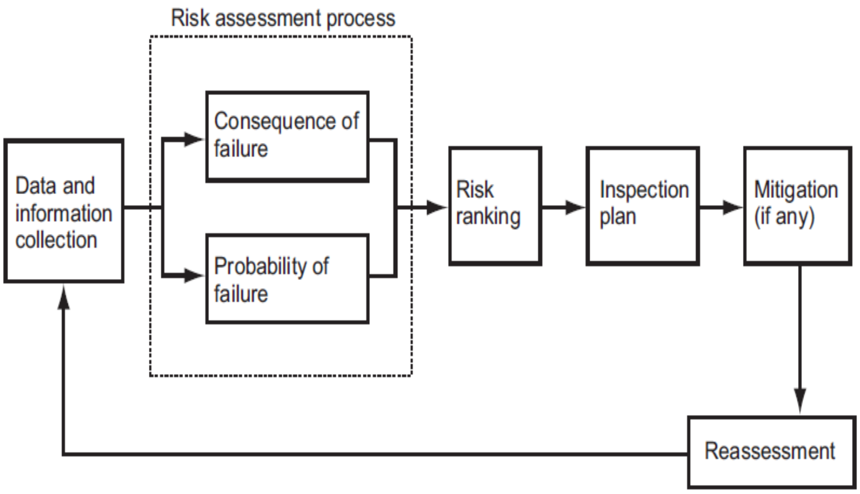



**Figure 4-15. Block Diagram of API 581 application process.**

The risk assessment process consists of obtaining a quantitative risk level, according to the following expression (Casal, 2008):

Risk = Probability of failure x Consequence of failure, which in API 581 notation is:

$$\text{Risk (m}^2\text{/year)} = P_{f,j}^{prd}(year^{-1}) \cdot CA(m^2) \tag{4.46}$$

(Note: Strictly speaking the equation should have frequency in place of probability of failure because the basis is one year. However, the name probability of failure will be used to follow API 581)

where

$P_{f,j}^{prd}$ is the probability of failure, which depends on: generic failure factor, damage factor and management system factor; CA is the consequence of failure, which depends on: storage and flash conditions, release rate and type, event tree and consequence area.

Figure 4-16 presents a block diagram of the steps required to calculate the risk.

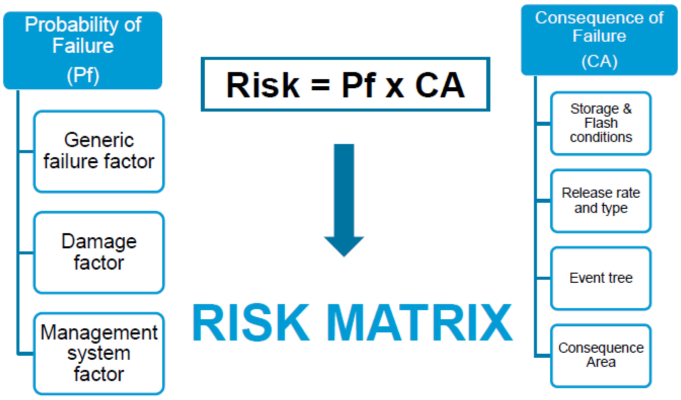



**Figure 4-16. Risk calculation process for pressure relief valves according to API 581 (2008).**

The approach followed in this thesis is the more rigorously level 2 consequence analysis, according to the API 581 (2008) classification.

### 4.11.1 Probability of failure

The steps followed in obtaining the parameters of figure 4-15, are summarized as follows:

1. Select an inspection interval, $t_{insp}$. The inspection interval is given by the periods established for the current inspection code used at the company (see section 4.11).
2. Determine the default values for the Weibull parameters, β and $\eta_{def}$ using Tables 7.4, 7.5 and 7.6 (part 1, API 581).
3. Determine the adjustment factor, Fc:
   Fc = 0.75 for conventional valves discharging to closed system or flare
   Fc = 1 for all other cases.
4. For each overpressure scenario, determine the adjustment factor, $F_{op}$ , using the equation:

$$F_{op} = \frac{1}{3.375} \cdot \left(\frac{P_o}{MAWP} - 1.3\right) \tag{4.47}$$

   To perform this step, an estimate of the maximal overpressure reached in the protected equipment in case the relief valve does not open is required. Table 7.3 (part 1, API 581) can be used.
5. Determine the environmental adjustment factor, $F_{env}$, using Table 7.6 (part 1, API 581).
6. Calculate the modified characteristic life, $\eta_{mod}$, using the following equation and the factors obtained from steps 3, 4, and 5:

$$\eta_{mod} = F_c \cdot F_{op} \cdot F_{env} \cdot \eta_{def} \tag{4.48}$$

7. Assemble the pressure relief valve inspection history. Grade each record using the inspection effectiveness table 7.7 (part 1, API 581). Record the results of each inspection record; pass/fail and no leak/leak and determine the confidence factors, $CF_i$, as applicable, for each inspection history based on the results of the test. Determine the time duration, $t_{dur,i}$, of each inspection cycle. Confidence factors may be seen in table 7.8 (part 1, API 581).
8. Update the modified characteristic life, $\eta_{mod}$, determined in step 6, as follows:
   a. Calculate the prior probability of failure using the following equation. The time period for use in this equation is the time duration of the inspection cycle, $t_{dur,i}$, as determined in step 7. Note that for the first inspection record, the modified characteristic life, $\eta_{mod}$, is used.

$$P_{f,prior}^{prd} = 1 - exp\left[-\left(\frac{t}{\eta_{mod}}\right)^{\beta}\right] \quad (4.49)$$

b. Calculate the prior probability of passing using the following equation:

$$P_{p,prior}^{prd} = 1 - P_{f,prior}^{prd} \quad (4.50)$$

c. Determine the conditional probability of failure and the conditional probability of pass using the following equations, respectively:

$$P_{f,cond}^{prd} = \left(1 - CF_{pass}\right)P_{p,prior}^{prd} \quad (4.51)$$

$$P_{f,cond}^{prd} = CF_{fail} \cdot P_{f,prior}^{prd} + \left(1 - CF_{pass}\right)P_{p,prior}^{prd} \quad (4.52)$$

d. Calculate the weighted probability of failure, $P_{f,wgt}^{prd}$, using the appropriate equation from table 7.9 (part 1, API 581).
e. Determine the updated characteristic life, $\eta_{upd}$, using the following equation and the weighted probability of failure, $P_{f,wgt}^{prd}$:

$$\eta_{upd} = \frac{t}{\left(-\ln\left[1 - P_{f.wgt}^{prd}\right]\right)^{\frac{1}{\beta}}} \quad (4.53)$$

9. For each overpressure demand case, determine the initiating event frequency, $EF_j$, using table 7.2 (part 1, API 581).
10. Determine the demand rate reduction factor, $DRRF_j$, which accounts for any layers of protection in the process that may reduce the probability of overpressuring the protected piece of equipment. Use table 7.2 (part 1, API 581).
11. For each overpressure demand case, determine the demand rate placed in the safety valve, $DR_j$ ,using the following equation:

$$DR_j = EF_j \cdot DRRF_j$$

12. Determine the MAWP of the protected equipment.
13. If an RBI study has been completed for the protected equipment, calculate its damage adjusted probability of failure, $P_f$. Since the damage factor for the protected equipment is a function of time, the damage factor must be determined at the PRD inspection interval, $t_{insp}$, specified in step 1.
14. Calculate the probability of failure of the protected equipment at the elevated overpressure, $P_{f,j}$ using the following equation. Use the overpressure determined in step 4, the MAWP of the protected equipment and the probability of failure determined in step 13:

$$P_{f,j} = P_f + \left(\frac{1 - gfft}{3}\right) \cdot \left(\frac{P_{o,j}}{MAWP} - 1\right) \quad (4.54)$$

15. Calculate the probability of failure, $P_{f,j}^{prd}$, using the following equation:

$$P_{f,j}^{prd} = P_{fod,j} \cdot DR_j \cdot P_{f,j} \quad (4.55)$$

### 4.11.2 Consequence of failure

Although the procedure explained here corresponds to an analysis done for small, medium and large hole sizes, in the case study selected to follow the new methodology (see section 6.8) only the rupture case has been considered (conservative basis).

*Fluid composition and properties*
The determination of fluid composition and properties is differentiated in two parts: at storage conditions and at flash conditions. The properties at storage conditions have been collected from the specification sheets and the process flow diagram of the analyzed installation. To ensure the feasibility of such properties, they have been verified by using the proper Mollier diagrams in case of pure substances and/or Aspen-Hysys. The properties at flash conditions are obtained realizing an isentropic flash calculation from storage pressure to atmospheric pressure. Aspen Hysys with Peng Robinson as EOS has been used.

*Release hole size selection*
Hole sizes have been taken from table 5.4M (part 3, API 581).

*Release phase and rate calculation*
Release phase is referred to the phase immediately downstream of the release point. When release phase is identified, the proper equation to calculate the release rate through the hole may be selected, and then, release rate may be obtained. Release rate also depends on the hole size selected. The procedure followed is described next.

1. Determine the stored fluid's saturation pressure, $Psat_s$, at the storage temperature:
$$Psat_s(T) = 10^{\left(A-\frac{B}{C+T_S}\right)} \tag{4.56}$$
   Determine the release phase:
   a. If $Psat_s \geq P_s > P \rightarrow$ release phase is vapor.
   b. If $P_s \geq Psat_s > Patm \rightarrow$ release phase is two-phase.
   c. If $P_s \geq Patm > Psat_s \rightarrow$ release phase is liquid.
2. For each release hole size, compute the release hole size area, $A_n$:
$$A_n = \frac{\pi \cdot d_n^2}{4} \tag{4.57}$$
3. For each release holes size, calculate the release rate, $W_n$:
   a. For liquids:
$$W_n = C_d \cdot K_{v,n} \cdot \rho_l \cdot \frac{A_n}{C_1}\sqrt{\frac{2 \cdot g_c \cdot (P_s - P_{atm})}{\rho_l}} \tag{4.58}$$
   b. For vapor:
      i. If $P_s > P_{trans}$:
$$W_n = \frac{C_d}{C_2} \cdot A_n \cdot P_s \sqrt{\left(\frac{k \cdot MW \cdot g_c}{R \cdot Ts}\right)\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}} \tag{4.59}$$
      ii. If $P_s \leq P_{trans}$:
$$W_n = \frac{C_d}{C_2} \cdot A_n \cdot P_s \sqrt{\left(\frac{MW \cdot g_c}{R \cdot Ts}\right)\left(\frac{2 \cdot k}{k-1}\right)\left(\frac{Patm}{Ps}\right)^{\frac{2}{k}}\left(1-\left(\frac{Patm}{Ps}\right)^{\frac{k-1}{k}}\right)} \tag{4.60}$$
$$P_{trans} = P_{atm}\left(\frac{k+1}{2}\right)^{\frac{k}{k-1}} \tag{4.61}$$
   c. For two-phase release the same equation as defined for liquids is used.

*Estimate the fluid inventory*
Fluid inventory (or holdup) is the amount of mass contained in the equipment.

*Release type*
Release type may be modeled in two ways:
i. Instantaneous release: this occurs so rapidly that the fluid disperses as a single large cloud or pool.

ii. Continuous release: this occurs over a longer period of time (>180 seconds) allowing the fluid to disperse in the shape of an elongated ellipse (depending of course on weather conditions).

The calculation procedure followed is as follows:

1. For each release holes size, calculate the time required to release 4536 Kg of the fluid:

$$t_n = \frac{C_3}{W_n} \tag{4.62}$$

2. For each release holes size, determine if the release type is continuous or instantaneous using the following criteria:

   a. If the release holes size is 6.35 mm or less, then the release type is continuous.
   b. If $t_n \leq 180$ s or the release mass is greater than 4536 kg, then the release is instantaneous; otherwise, the release is continuous.

*Estimate the impact of detection and isolation systems in release magnitude*

Detection, isolation and mitigation systems are designed to reduce the effects of a release of hazardous materials. Some of them reduce the release duration and some other the consequences. A release correction factor ($fact_{di}$) is obtained after analyzing the impact of such systems:

1. Determine the detection and isolation systems present in the unit.
2. Using table 5.5 (part 3, API 581) select the appropriate classification (A, B, C) for the detection system.
3. Using Table 5.5 (part 3, API 581) select the appropriate classification (A, B, C) for the isolation system.
4. Using Table 5.6 (part 3, API 581) and the classifications determined in steps 2 and 3, determine the release reduction factor, $fact_{di}$.
5. Using Table 5.7 (part 3, API 581) and the classifications determined before, determine the total leak durations for each of the selected release hole sizes, $ld_{max,n}$.

*Determine the release rate and mass for consequence analysis*

Depending on the type of release, the release rate will have different values:

1. For each release holes size, calculate the adjusted release rate, $rate_n$, where the theoretical release rate is $W_n$, from step 3 of the release phase and rate calculation:

$$rate_n = W_n \cdot (1 - fact_{di}) \tag{4.63}$$

2. For each release hole size, calculate the leak duration, $ld_n$, of the release. Note that the leak duration cannot exceed the maximum duration, $ld_{max}$:

$$ld_n = \min\left[\left\{\frac{mass_{avail,n}}{rate_n}\right\}, \left\{60 \cdot ld_{max,n}\right\}\right] \tag{4.64}$$

3. When a release is a two-phase one, there is an amount of liquid entrained in the jet (vapor) portion of the release (aerosol). The remaining liquid portion of the release is the rainout. Determine the rainout mass fraction from the released fluid:

$$frac_{ro} = 1 - 2 \cdot frac_{fsh} \quad \text{for} \quad frac_{fsh} < 0.5 \tag{4.65}$$

$$frac_{ro} = 0.0 \quad \text{for } frac_{fsh} \geq 0.5 \tag{4.66}$$

4. For each release hole size selected, calculate the release rate of liquid that settles to the ground for the pool calculations, $W_n^{pool}$:

$$W_n^{pool} = rate_n \cdot frac_{ro} \tag{4.67}$$

5. For each hole size selected, calculate the release rate of vapor including entrained liquid remaining in the jet, $W_n^{jet}$:

$$W_n^{jet} = rate_n \cdot (1 - frac_{ro}) \tag{4.68}$$

6. Calculate the mass fraction of entrained liquid, $frac_{entl}$, within the jet portion of the release:

$$frac_{entl} = \frac{(frac_l \cdot frac_{fsh})}{(1 - frac_{ro})} \tag{4.69}$$

Vapor sources from boiling or non-boiling pools:

If Tb < Tg, where Tg is the ground temperature, then liquid pool case must be considered. The assumptions made are the following ones:

- The liquid contained in the pool is assumed as the product contained in the equipment.
- The whole liquid contained in the pool is vaporized.
- The evaporation rate, $V_{p,n}$, is assumed as 1 $m^3/s$.
- The time required to evaporate, $t_{p,n}$, the whole liquid is $t_{p,n}$ = volume/evaporation rate.

The next equation will provide an approximation of the radius, $r_{p,n}$, of the evaporating pool.

$$r_{p,n} = \sqrt{\frac{2}{3}} \left(\frac{8g \cdot V_{p,n}}{\pi}\right)^{0.25} {t_{p,n}}^{0.75} \tag{4.70}$$

*Determining flammable and explosive consequences*

Flammable and explosive consequences are based on the event tree analysis. This analysis determines the probabilities of various outcomes as a result of release of hazardous fluids to the atmosphere. These probabilities are then used to weight the overall consequences of release.

The first step is to calculate the outcome probabilities using experimental expressions. All possible outcomes are: vapor cloud explosions, flash fire, jet fire, fireball, pool fire and safe dispersion. Depending on the release phase and type, some of the outcome may not be considered. For example, it is not possible to have pool fire if release phase is vapor.

After the outcome probabilities are obtained, outcome consequence area is calculated. When outcome probability and consequence area are obtained, average weighted consequence area for flammable and explosive consequences is calculated.

A full description of this calculation step is as follows:

1. Determine the mass fraction of the release rate that contains a flammable component, $mfrac^{flam}$. For instance, as propylene is flammable, $mfrac^{flam} = 1$.
2. For each hole size, calculate the flammable release rate, $rate_n^{flam}$. Also calculate the liquid portion, $rate_{l,n}^{flam}$, and the vapor portion, $rate_{v,n}^{flam}$, of the flammable release rate:

$$rate_n^{flam} = rate_n \cdot mfrac^{flam} \tag{4.71}$$

$$rate_{l,n}^{flam} = rate_n^{flam} \cdot (1 - frac_{fsh}) \tag{4.72}$$

$$rate_{v,n}^{flam} = rate_n^{flam} \cdot frac_{fsh} \tag{4.73}$$

3. For each hole size, select the appropriate event tree using figures 6.3 and 6.4 (part 3, API 581) and the phase of the fluid after flashing the atmosphere.
4. For each holes size, including the rupture case, calculate the probability of ignition of the release.
    a. Determine the probability of ignition at ambient temperature for the liquid portion of the release, $poi_{l,n}^{amb}$ with the figure 6.5 (part 3, API 581).
    b. Determine the probability of ignition at ambient temperature for the vapor portion of the release, $poi_{v,n}^{amb}$ with the figure 6.6 (part 3, API 581).
    c. Determine the maximum probability of ignition for the liquid, $poi_l^{ait}$, and the vapor, $poi_v^{ait}$ at the Auto Ignition Temperature (AIT):

$$poi_l^{ait} = 1.0 \tag{4.74}$$

$$poi_v^{ait} = \max\left[0.7, 0.7 + 0.2 \cdot \left(\frac{170-MW}{170-2}\right)\right] \tag{4.75}$$

    d. Calculate the probability of ignition for the liquid, $poi_{l,n}$, and the vapor, $poi_{v,n}$, at normal storage temperatures:

$$poi_{l,n} = poi_{l,n}^{amb} + \left(poi_l^{ait} - poi_{l,n}^{amb}\right)\left(\frac{T_s - C_{16}}{AIT - C_{16}}\right) \tag{4.76}$$

$$poi_{v,n} = poi_{v,n}^{amb} + \left(poi_v^{ait} - poi_{v,n}^{amb}\right)\left(\frac{T_s - C_{16}}{AIT - C_{16}}\right) \tag{4.77}$$

    e. For two phase releases, calculate the probability of ignition, $poi_{2,n}$, at normal storage temperatures:

$$poi_{2.n} = poi_{l,n} \cdot frac_{fsh} + poi_{v,n} \cdot \left(1 - frac_{fsh}\right) \tag{4.78}$$

5. For each hole size, determine the probability of immediate ignition given ignition.
    a. Obtain the probabilities of immediate ignition at ambient conditions for the liquid portion and the vapor portions of the release, $poii_{l,n}^{amb}$ and $poii_{v,n}^{amb}$ from table 6.3 (part 3, API 581) based on whether there is instantaneous or continuous liquid or vapor release.
    b. Calculate the probability of immediate ignition given ignition at storage condition for the liquid portion of the release, $poii_{l,n}$, and the vapor portion of the release, $poii_{v,n}$. Use $poii^{ait} = 1.0$.

$$poii_{l,n} = poii_{l,n}^{amb} + \left(\frac{T_s - C_{16}}{AIT - C_{16}}\right) \cdot \left(poii^{ait} - poii_{l,n}^{ait}\right) \tag{4.79}$$

$$poii_{v,n} = poii_{v,n}^{amb} + \left(\frac{T_s - C_{16}}{AIT - C_{16}}\right) \cdot \left(poii^{ait} - poii_{v,n}^{ait}\right) \tag{4.80}$$

    $poii^{ait}$ is obtained from the referenced table.
    c. For two-phase releases, calculate the probability of immediate ignition given ignition, $poii_{2,n}$, at normal storage temperatures and the flash fraction, $frac_{fsh}$:

$$poii_{2,n} = frac_{fsh} \cdot poii_{l,n} + (1 - frac_{fsh}) \cdot poii_{v,n} \tag{4.81}$$

6. Determine the probability of VCE given a delayed ignition.
    a. Determine the probability of VCE given delayed ignition, pvcedi from table 6.3 (part 3, API 581) as a function of the release type and phase for release. The probability of a VCE given delayed ignition for a liquid release is $pvcedi_{l,n}$; for a vapor it is $pvcedi_{v,n}$.
    b. For two phase releases, calculate the probability of VCE, given delayed ignition, $pvcedi_{2,n}$.

$$pvcedi_{2,n} = frac_{fsh} \cdot pvcedi_{l,n} + \left(1 - frac_{fsh}\right) \cdot pvcedi_{v,n} \tag{4.82}$$

7. Determine the probability of flash fire given delayed ignition.

a. Determine the probability of flash fire given delayed, pffdi from Table 6.3 (part 3, API 581) as a function of the release type and phase of release.
b. For two phase releases, calculate the probability of flash fire given delayed ignition $pffdi_{2,n}$:

$$pffdi_{2,n} = frac_{fsh} \cdot pffdi_{l,n} + \left(1 - frac_{fsh}\right) \cdot pffdi_{v,n} \tag{4.83}$$

8. Determine the probability of a fireball given an immediate release, pfbii:
$pfbii = 1.0$ for instantaneous vapor or two-phase releases.
$pfbii = 0.0$ for all other cases.

9. Select the appropriate event tree.
The event trees selected can be seen in figures 6.3 (part 3, API 581) for small, medium and large hole sizes, and figure 6.4 of the same reference for the rupture case.

10. For each hole size, determine the probability of each of the possible event outcomes on the event tree selected in step 9.

11. For each hole size, calculate the component damage consequence area, $CA_{cmd,n}^{vce}$, and the personnel injury consequence area, $CA_{inj,n}^{vce}$, of a vapor cloud explosion.

12. For each of the hole sizes, calculate the component damage consequence area, $CA_{cmd,n}^{flash}$, and the personnel injury consequence area, $CA_{inj,n}^{flash}$, of a flash fire.
13. For the rupture case only, calculate the component damage consequence area, $CA_{cmd,n}^{flam}$ and the personnel injury consequence area, $CA_{inj,n}^{fball}$, of a fireball. For example, in case of fireball the following procedure has to be followed:
    a. Determine the flammable mass of the fluid contained in the equipment using the following equation:

$$mass_{fb} = mfrac^{flam} \cdot mass_{avail,n} \tag{4.84}$$

    b. Calculate the maximum diameter, $Dmax_{fb}$, and the center height, $H_{fb}$, of the fireball using the following equations, respectively:

$$Dmax_{fb} = C_{22} \cdot mass_{fb}^{0.333} \tag{4.85}$$

$$H_{fb} = 0.75 \cdot Dmax_{fb} \tag{4.86}$$

    c. Calculate the duration of the fireball, $t_{fb}$, using the following equations based on the mass of the fireball:

$$t_{fb} = C_{23} \cdot mass_{fb}^{0.333} \qquad \text{for } mass_{fb} \leq 29937 \text{ kg} \tag{4.87}$$

$$t_{fb} = C_{24} \cdot mass_{fb}^{0.167} \qquad \text{for } mass_{fb} > 29937 \text{ kg} \tag{4.88}$$

    d. Calculate the amount of energy radiated by the fireball, $Q_{rad}^{fball}$, using:

$$Qrad^{fball} = \frac{C_{14} \cdot \beta_{fb} \cdot mass_{fb} \cdot HC_l}{\pi \cdot Dmax_{fb}^2 \cdot t_{fb}} \text{ with } \beta_{fb} = C_{25} \cdot P_B^{0.32} \tag{4.89}$$

    e. For the component damage consequence area, in this thesis a radiation limit of 37.8 $kW/m^2$ has been used and for personnel injury 12.6 $kW/m^2$ was used. These radiation limits are used to determine the safe distances, $xs_{cmd,n}^{fball}$ and $xs_{inj,n}^{fball}$ , from the fireball using the following 4-step iterative procedure:

i. Guess at an acceptable distance from the fireball, $xs^{fball}$.
ii. Calculate the atmospheric transmissivity, $\tau_{atm}$, and the spherical view factor, Fsph, using the following equations (both of these parameters are functions of the distance from the fireball chosen above, $xs^{fball}$):

$$\tau_{atm} = 0.819 \cdot (P_w \cdot xs_n)^{-0.09} \tag{4.90}$$

$$Fsph = \frac{(Dmax_{fb})^2}{4C_{fb}^2} \text{ and} \tag{4.91}$$

$$C_{fb} = \sqrt{\left(\frac{Dmax_{fb}}{2.0}\right)^2 + \left(\frac{xs^{fball}}{2.0}\right)^2} \tag{4.92}$$

iii. Calculate the received thermal heat flux, $Ith^{fball}$, at the distance chosen and compare it to the acceptable radiation limit (37.8 kW/m2 for component damage and 12.6 kW/m2 for personnel injury):

$$Ith^{fball} = \tau_{atm} \cdot Qrad^{fball} \cdot Fsph \tag{4.93}$$

iv. Adjust the distance, $xs^{fball}$, accordingly, and repeat the above steps until the calculated received thermal heat flux equals the allowable limit.

f. Calculate the component damage consequence area, $CA_{cmd}^{fball}$, and the personnel injury consequence area, $CA_{inj}^{fball}$ using the following equations:

$$CA_{cmd}^{fball} = \pi \cdot \left(xs_{cmd}^{fball}\right)^2 \text{ and } CA_{inj}^{fball} = \pi \cdot \left(xs_{inj}^{fball}\right)^2 \tag{4.93}$$

Finally, the risk for each contingency of the event tree is calculated and compared to the acceptable risk level specified. In this thesis, the normally accepted value for petrochemical plants and level 2 consequence analysis of 4.6 $m^2$/year, has been taken.

Case study 6.8 shows the application of this methodology to the pressure relief valve YS702/01.

# Chapter 5. Application of the new methodology to three existing petrochemical plants

## 5.1 Design basis of the relief system

The petrochemical plants studied in this thesis were built in 1978 (Plant I), 1992 (Plant II) and 2001 (Plant III). Plants I and II are of midsized capacity and plant III has a wordwide scale capacity comparable to plants constructed today. The engineering and construction companies, which performed the detail engineering of the plants, are reputable worldwide.

Because the plants handle light hydrocarbons, a flare is necessary (Casal et al., 1999). For plant I, a flare was built with enough overcapacity to handle the loads of plants I and II. For plant III, a new flare was built that could handle the capacity of all three plants and the old flare was shutdown. A summary of the design parameters of the flare system is given in Table 5-1. In all cases, the dimensioning scenario is fire in the plants, or in plants I and II because they are together or in plant III, but not simultaneously.

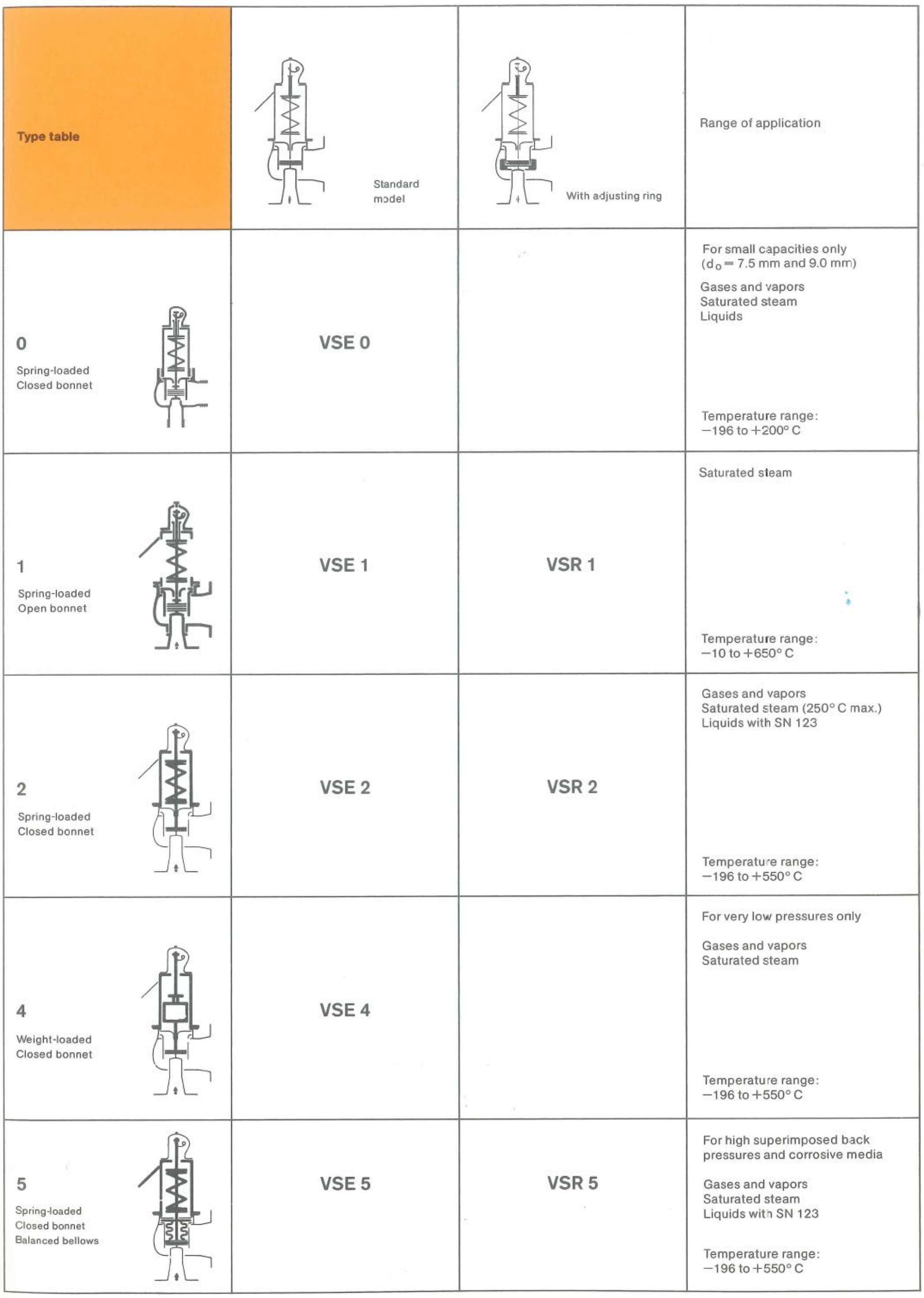
**Table 5-1. Characteristic data of the flare loads for plants I, II and III.**

| **Gas composition** | **Plants I and II, wt%** | **Plant III, wt%** | **Plants I, II and III, wt%** |
|---|---|---|---|
| Propylene | 94.5 | 68.4 | 74.9 |
| Propane | 5.2 | 7.8 | 7.2 |
| Ethylene | -- | 22.7 | 17.0 |
| Ethane | remainder | 0.7 | remainder |
| Heptane | remainder | remainder | remainder |
| Hydrogen | remainder | remainder | remainder |
| Total maximal load | 20 t/h | 60 t/h | 80 t/h smokeless<br>160 t/h non smokeless |

In the case of fire, it is always assumed that the reactors relieve instantaneously. This is the first mitigation action for large pressure vessels with high pressure gas inside (Johnson, 2007). This load is added to the load coming from the affected zone where the fire is produced.

The guidelines adopted for the design of the relief system were: AD Merkblatt-A2 and A4, together with API 520 and API 521. For plants I and II, the manufacturer of the pressure relief valves was Sempell (now Pentair) and for Plant III, it was Leser.

The Sempell valve type of choice was: VSE0 for thermal expansion (60%); VSE2 (26%) for gases/vapors and liquids; VSR2 (10%) for gases/vapors and liquids but with the possibility of adjusting the blowdown and the rest (4%) VSE4 type for very low set pressures (0.4 barg) and VSE5 with bellows for high total backpressures. A summary of the models and the application characteristics is given in Figure 5-1.



| Type table | Standard model | With adjusting ring | Range of application |
|---|---|---|---|
| 0<br>Spring-loaded<br>Closed bonnet | VSE 0 | | For small capacities only<br>($d_0$ = 7.5 mm and 9.0 mm)<br>Gases and vapors<br>Saturated steam<br>Liquids<br>Temperature range:<br>−196 to +200° C |
| 1<br>Spring-loaded<br>Open bonnet | VSE 1 | VSR 1 | Saturated steam<br>Temperature range:<br>−10 to +650° C |
| 2<br>Spring-loaded<br>Closed bonnet | VSE 2 | VSR 2 | Gases and vapors<br>Saturated steam (250° C max.)<br>Liquids with SN 123<br>Temperature range:<br>−196 to +550° C |
| 4<br>Weight-loaded<br>Closed bonnet | VSE 4 | | For very low pressures only<br>Gases and vapors<br>Saturated steam<br>Temperature range:<br>−196 to +550° C |
| 5<br>Spring-loaded<br>Closed bonnet<br>Balanced bellows | VSE 5 | VSR 5 | For high superimposed back pressures and corrosive media<br>Gases and vapors<br>Saturated steam<br>Liquids with SN 123<br>Temperature range:<br>−196 to +550° C |

**Figure 5-1. Models of Sempell safety valves installed in plant I and II (taken from catalogue KS27585E).**

The manufacturer, Sempell, does not give the characteristic operating curves of the valves. However, the following information about overpressures and blowdown are available in its catalog. See the following table:

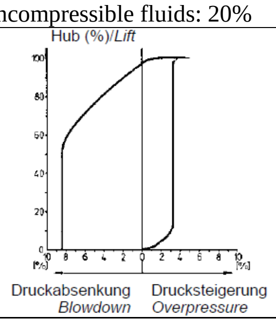
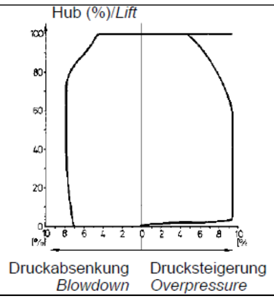
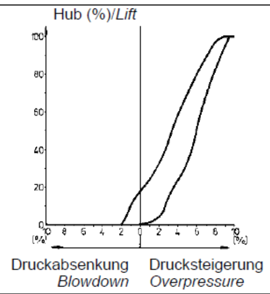
| Valve type | Gases and vapors | |
|---|---|---|
| | **Overpressure, % of SP** | **Blowdown, % of SP** |
| VSE | +5 | -10 |
| VSR | +3 | -8 |
| VSR1 with SN144* | +3 | -6 |

| Valve type | Gases and vapors | |
|---|---|---|
| | **Overpressure, % of SP** | **Blowdown, % of SP** |
| | **Liquids** | |
| VSR with SN123* | +10 | -10 |
| VSE with SN123* | +10 | -20 |

* SN144 means a valve with balanced piston (equivalent to bellows) and SN123 is a seating for liquids especial design.

The manufacturer, Leser, gives information about the operating characteristics of safety valves. This information is very useful in determining stability criteria for safety valves, as has been seen in section 6.7. A summary of the installed Leser valves and their characteristics is presented in Table 5-2.

**Table 5-2. Operating characteristics of the Laser valves used in plant III.**

| | **Full lift safety valves** | **Safety relief valves** | **Relief valves** |
|---|---|---|---|
| Opening characteristics | After responding within an overpressure of 5%, full lift safety valves open by a pop action up to the restricted lift. The portion of the lift before the valve opens suddenly must not exceed 20% of the total lift. | After responding within a maximum overpressure of 10%, these safety valves achieve the lift required for the mass flow to be discharged. | They open almost continuously as a function of the overpressure. They do not open with pop-action and without any further increase in pressure over a range of more than 10% of the lift. After responding within a maximum overpressure of 10%, these safety valves achieve the lift required for the mass flow to be discharged. |
| Use | Where large mass flow of gases/vapors is required. It reaches the full lift inclusive with small overpressure. | Are ideal for liquids, their wide proportional range leads to continuous action and depressurerizing of pressure peaks, without the immediate discharge of such a large quantity as a full lift safety valve. | Are used wherever only small mass flows are generally to be expected, for example, thermal expansion of liquids. |
| Leser types | 441, 442, 455, 456, 457 and 458 | 431, 433, 543 and 544 | 427, 429, 532 and 534 |
| Blowdown (for SP > 3 barg) | Compressible fluids: 10%<br>Incompressible fluids: 20% | | |
| Function |  |  |  |

Another characteristic of this relief system is the intensive use of changeover valves. The advantage of these valves is that there is the possibility of the removal of one of the two for inspection at the maintenance workshop without interruption of plant operation. The disadvantage is that they introduce an element with a high pressure drop in the inlet pipe that normally makes difficult the fulfilling of the 3% rule. The type of changeover valve installed is showed in Figures 5-2 and 6-11.

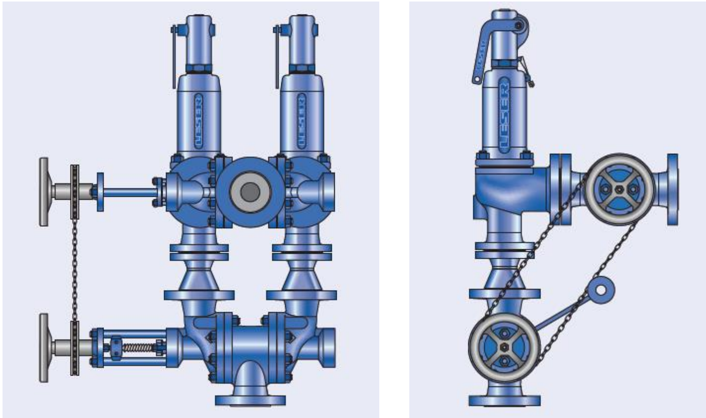


**Figure 5-2. Type of Leser changeover valve installed in plant III (taken from Leser catalog 487/01/E, 2009).**

## 5.2 Scope of the work. Data source

This first step in applying the new methodology to the three plants was the checking of all the information available on the relief system. The results were:

- Only a specification sheet was available for all the valves of plants I and III. For plant II a list of valves with their main characteristics was also available
- A database was available in the inspection department containing the technical parameters of the valve (manufacturer, type, set pressure, orifice diameters, construction materials, size, type of flanges, and the results of the periodical revision in the workshop)
- For plant III, the calculations, done by Leser, of the required area of the valve from an already specified relieving load were available, but not for plants I and II
- No calculations were available at all to justify the relieving load
- The following parameters were not available for the three plants: inlet pressure drop, outlet pressure drop, stability, reactions and forces and, of course there were no values for the new ones: body bowl choking, acoustic induced vibration, flow induced vibration, noise, etc.

The information available was the normal information that the plant operators have, as has been pointed out by companies which performed relief systems audits (Berwanger et al., 2000; Westphal and Köper, 2003)

In order to introduce the new methodology the following work was carried out (see table 5-3):

**Table 5-3. Scope of the work performed to implant the new methodology.**

| Task | Done by | Time | Work performed | Results |
|---|---|---|---|---|
| Contingency analysis for each safety valve | Plant manager, plant engineer and thesis author | February-June, 2012 | For each pressure equipment of the plants the template of figure 3-1 was applied. | Contingency analysis with the relieving scenarios that apply to each valve. |
| Hazop + LOPA analysis of the plants | Plants staff including the thesis author and an external specialized PHA company | November 2011-August 2012 | Integrated Hazop analysis and LOPA of the plants | Allocation of the safety valves with IPL credits and checking the necessity of the safety valves with especial emphasis on the guidewords: more pressure and more temperature. |
| Built-up backpressure calculations of the safety valves which | Zapico (2013) and thesis author | June-December, 2012 | Calculation of the equivalent lengths of the discharge pipe according to the existing isometrics.<br>Calculation of built-up backpressures for 6 cumulative cases: 3 for plant III and 3 | Built-up back pressures for all the safety valves, which discharge to the flare. |

| Task | Done by | Time | Work performed | Results |
|---|---|---|---|---|
| discharge to flare | | | for plant I and II.<br>Proposal of mitigation options for the findings encountered. | |
| Application of API 581 to find the optimal revision interval of the safety valves | Albornà (2014) and thesis author | June-December, 2013 | Gathering all the information about the revision periods of the critical safety valves.<br>Preforming API 581 analysis level 2 with the target to increase the revision periods.<br>Economic analysis. | According to the API the revision periods could be increased. However the methodology has not yet been accepted by spanish authorities. |
| Relieving loads summary data sheets | Thesis author | July 2012 to September 2015 | Calculation of the required and maximal loads for each scenario.<br>Checking the calculated required loads with the ones from the original datasheets.<br>Analysis of the deficiencies in order to get statistical data. | Relieving loads summary data sheets for each safety valve. |
| Fulfilling of 3% rule | Thesis author | September 2014-September 2015 | Gathering isometrics of the inlet piping.<br>Performing inlet pressure drop.<br>Statistical analysis and proposed mitigation measures. | List of valves which do not fulfill the 3% rule including mitigation measures. |
| Engineering analysis for cases with inlet pressure drop > 3% | Thesis author | September 2014-September 2015 | Development of a new methodology for performing the engineering analysis according to the latest state of art knowledge.<br>Performing the engineering analysis for valves which 3% < $\Delta P_{inlet}$ < 5%. | List of valves which passed the engineering analysis. |
| Inspection methodology applied to a turnaround | Workshop clerk and thesis author | February 2015 | Prepop analysis of the critical pressure relief valves.<br>Statistical analysis of the results. | Recommendations for new revision periods. |

The list of pressure relief valves of the plants with the work performed is shown in Annex C.

## 5.3 Actions taken and difficulties encountered in implementing the new methodology.

To better show the action taken and the difficulties encountered when trying to implement it, a table has been prepared following the phases of the new methodology, Table 5.4.

**Table 5-4. Work performed and difficulties encountered in implementing the new methodology.**

| Phase | Actions performed | Difficulties encountered | Comments |
|---|---|---|---|
| Risk analysis | Following the Hazop revision policy of the company, every 5 years the Hazop has to be revised. This was conducted in November 2011 until August 2012, with especial emphasis on the recommended guide works "more pressure" and "more temperature" and including an integrated LOPA analysis. Moreover, the nodes were selected taking into account the recommended procedures. | None | The Hazop team leader was a very experienced engineer from an external process safety company. This was the key factor of the success of the PHA. Another good point was selecting a scriber with high knowledge of the Primatech software (from PHAworks). |
| Safety valve requirements specification | The new safety requirements specification (figure 3-5) has been used for the valves of plants I and II. For the valves of plant III the original data sheet was supplemented with | Lack of information from the valves built in 1980s. It was necessary to contact the manufacturer Sempell to get information.<br>Sometimes the information | Some people found this new safety valve requirement specification as "more paperwork". |

| Phase | Actions performed | Difficulties encountered | Comments |
|---|---|---|---|
| | the necessary information to get an equivalent document. | was gathered during the inspection in the workshop in a turnaround. Convincing the maintenance engineer of the importance of having a dedicated specification for each valve and that it is not a "paperwork exercise". | |
| Design | The relief valves of the three plants have been revised according to the new design methodology as explained in section 3.3 including the recommended documentation (section 3-9). Annex C shows the list of valves. This work has been done by the thesis author during 2012-2015 with very few breaks. Some case studies were intensively worked with the latest knowledge. | Although the required relieving load was available for the dimensioning case; there were no calculations to demonstrate it. It has not been possible, until now, to apply the new methodology to changing the revision periods of the valves based on a quantitative approach (see section 4-11). It is a corporative decision of the company. | From the study cases worked (chapter 6) recommendations for the different calculations procedures are presented. Prepop test has been included as a standard procedure for all critical safety valves. |
| Reception, installation and checking | The MOC requires a pre-startup safety review. The checklist of Annex B has to be followed. Information given about how to handle the safety valves was given to contractor personnel. Moreover, during a turnaround the inspection engineer supervises the dismantling, inspection and montage of the valves. | Difficulties of changing people's attitudes. Majority believe that a safety valve is like a gate valve. | |
| Operation, maintenance and revision | Training has been given to plant operators about how to detect a leakage of a safety valve. The inspection engineer has to control the exact fulfillment of the inspection test template in the workshop (see section 6-9) according to the new procedure. A meeting with the plant staff is required if the prepop test shows that the valve opens higher than 10% of SP. | | |
| Management of change (MOC) | The template (figure 3-7) has been implemented as standard for each modification. See the "relief system design" in the template. | None | The implementation of the MOC is a very good tool for the process safety of the plants. |
| Decommissioning | The procedure to treat it is included in MOC. | None | |
| Verification | The checklist of section 3.8 is integrated in the engineering procedures of the plants. It will be followed in a revamping or in a new plant construction. | None | |
| Documentation and technical audits | Dedicated file (paper and electronic database) has been | Engineering management had difficulties in understanding | Dedicated software like PS PPM could be implemented |

| Phase | Actions performed | Difficulties encountered | Comments |
|---|---|---|---|
| | created for each PRV. The documentation will be audited in 2016. | that this procedure is not a “paper for the audit” exercise. | in the future. |

## 5.4 Deficiencies encountered in the design phase. Statistical analysis

The data set used to perform a statistical analysis was taken from three existing petrochemical plants. The characteristic of the sample is presented in table 5-5. The complete list of the pressure relief devices is showed in Annex C.

**Table 5-5. Charateristics of the pressure relief device sample.**

| | Plant I and II | Plant III | TOTAL |
|---|---|---|---|
| Total number of pressure relief devices | 226 | 317 | 543 |
| Rupture discs | 0 | 13 | 13 |
| Overpressure-vacuum valves | 7 | 15 | 22 |
| Explosion relief panel | 0 | 1 | 1 |
| Others | 2 | 2 | 4 |
| Total number of pressure relief valves | 217 | 286 | 503 |
| Safety valves excluded from analysis | 77 | 107 | 184 |
| Safety valves considered in the analysis | 140 | 179 | 319 |

The reason for excluding 184 pressure relief valves is due to the fact that they belong to dynamic equipment and the safety valve is part of the package unit, for example, the internal safety valve of reciprocating or diaphragm pumps or reciprocating or screw compressors, etc. Another example is the internal safety valve in the pressure reduction system of pressurized bottles of nitrogen, CO, etc.

Concerning the frequency of a dimensioning scenario, Table 5-6 shows the results:

**Table 5-6. Statistic of the dimensioning scenario for the valves.**

| Dimensioning scenario | Plant I and II | Plant III | TOTAL | % |
|---|---|---|---|---|
| Thermal expansion | 65 | 104 | 169 | 53 |
| Fire | 28 | 37 | 65 | 20 |
| Auto control failure | 20 | 16 | 36 | 11 |
| Blocked outlets | 20 | 10 | 30 | 9 |
| Chemical reaction | 4 | 4 | 8 | 3 |
| Exchanger tube breakage | 2 | 3 | 5 | 2 |
| Abnormal heat input | 0 | 3 | 3 | 1 |
| Mechanical equipment failure | 1 | 2 | 3 | 1 |
| Reflux failure | 0 | 0 | 0 | 0 |
| Cooling water failure | 0 | 0 | 0 | 0 |
| Power failure | 0 | 0 | 0 | 0 |
| Instrument air failure | 0 | 0 | 0 | 0 |
| Inadvertent VA open/close | 0 | 0 | 0 | 0 |
| Heat loss (series frac.) | 0 | 0 | 0 | 0 |
| Loss of quench/cold feed | 0 | 0 | 0 | 0 |
| Steam out | 0 | 0 | 0 | 0 |
| **TOTAL** | **140** | **179** | **319** | **100** |

The findings related to incorrectly installed relief valves has been classified following Kumana and Aldeeb (2014) and are presented in Table 5-7.

**Table 5-7. Finding categories of the pressure relief valves analyzed.**

| Finding category | PLANTS | | | No. of safety valves affected | % |
|---|---|---|---|---|---|
| | I and II | III | TOTAL | | |
| Equipment without protection | 0 | 0 | 0 | 458 (1) | 0 |
| Safety valve undersized | 2 | 2 | 4 | 319 | 1 |

| Finding category | PLANTS | | | No. of safety valves affected | % |
|---|---|---|---|---|---|
| | I and II | III | TOTAL | | |
| Inlet pressure drop > 3% | 7 | 15 | 22 | 150 (2) | 15 |
| Outlet pressure drop > 15 % | 20 | 6 | 26 | 150 (2) | 17 |
| Outlet pressure drop > 30 %, balanced | 0 | 0 | 0 | 150 (2) | 0 |
| Set pressure above the design pressure | 0 | 1 | 1 | 319 | 0 |
| Installation (reduced bore valves on inlet, outlet pocketing concerns, administrative deficiencies, etc.) | 4 | 1 | 5 | 319 | 1.5 |
| Missing data | 0 | 2 | 2 | 319 | 0.5 |
| Oversized | 1 | 1 | 2 | 319 | 0.5 |
| **TOTAL** | **34** | **28** | **62** | **319** | **19.5** |

(1) This figure is the number of pressure equipment.

(2) Excluding the thermal expansion valves.

The comparison of the ratio pressure equipment versus number of pressure relief devices is presented in Table 5-8.

**Table 5-8. Comparison of ratios: pressure equipment vs pressure relief devices.**

| | Plant I + II + III | Kumana et al. | Berwanger et al. |
|---|---|---|---|
| (1) Number of pressure relief devices | 543 | 80,372 | 14,873 |
| (2) Number of pressure equipment | 458 | 174,943 | 24,303 |
| Ratio (1)/(2) | 1.2 | 0.5 | 0.6 |
| Comments | The specific design of the plants studied requires more changeover valves than a typical process plant. | Here there are 5 PRDs for each 10 pressure equipment | Here compressors are included and pumps excluded from the pressure equipment sample. |

The comparison of the results of this study with that of Berwanger et al. (2000), Kumana and Aldeeb (2014) and Köper and Westphal (2000) is showed in Table 5-9.

**Table 5-9. Comparison of the results of this study vs other reports.**

| Deficiency | This study, % | Kumana report,% | Berwanger report,% | Köper report,% |
|---|---|---|---|---|
| Equipment without protection | 0 | 13.4 | 15.1 (1) | |
| Safety valve undersized (3) | 1.0 | 13.4 | 14.1 (2) | |
| Inlet pressure drop (3) | 15.0 | 15.6 | 8.2 | 18.0 |
| Outlet pressure drop (3) | 17.0 | 3.1 (4) | 12.3 | 22.0 |
| Installation (3) | 1.5 | 24.9 | 0.2 | |
| Set pressure above design pressure (3) | 0 | 10.5 | 2.0 | |

(1) Expresed as number of equipment unprotected/total number of equipment excluding pumps.

(2) Corrected from the original report taking into account the total number of PRDs

(3) Kumana and Berwanger reports used pressure relief device basis. In this study and in Köper report the basis is pressure relief valves. The differences should be minimal.

(4) Backpressures calculated for individual discharge scenarios only. The figure is not representative.

## Results

a) Deficiencies were revealed in 19.5% of the 319 pressure relief valves studied. This figure is similar that of Köper and Westphal (2003) report: 17%, but less than the Kumana report: 47%.
b) The case of fire is the dimensioning scenario for 20% of the valves. Köper and Westphal report pointed out a value of 16% but Short (2006) gives a value of 50%.
c) The result of the inlet pressure drop deficiencies obtained in this study is 15.0 % matching almost exactly with the values of Kumana and Köper 15.6%.

d) The outlet pressure drop percentage obtained: 17% is comparable with that of Berwanger and Köper: 12.3% and 22% respectively.
e) The set pressure above design pressure is normally not an issue as demonstrated by the results obtained in this study and that of Berwanger.
f) The thermal expansion scenario is the most usual, as demonstrated in this study: 50% very similar with Köper: 44%.
g) The chemical reaction scenario is the dimensioning one for the 3% of the valves and matches exactly with the result of Köper: 3% as well.

The author has found the following design patterns that the original engineering company followed, which could explain the deviations with other statistical sources:

- The calculation of the required load for the case of fire near the critical point has been performed taking 115 kJ/kg as latent heat, as recommended by API 521 (2008)
- Although in most cases the relieving condition of light hydrocarbons mixtures (ethylene, propylene, etc) have a compressibility factor < 0.8, the required area calculation of the valve has been performed with the ideal specific heat ratio, Cp/Cv, which could give undersized valves. However, because the valves were chosen with a greater area than required, this deficiency has had no relevance
- The relieving load of thermal expansion valves for piping was dimensioned in a standard way, i.e. taking 500 kg/h or 1000 kg/h of relieving liquid depending on the length. The rigorous calculations performed by the author considering the real length of the piping, a solar radiation of 1.15 $kW/m^2$ and sizing in a rigorous way, the two-phase flow through the valve in case of light hydrocarbons, demonstrated that the original engineering approach was very conservative because the valve was installed with an area of 10 times or bigger
- Fire scenario is not applied to piping as stated in the paragraph 5.14.4, API 521 (2008)
- Intensive use of a restriction orifice downstream of a pressure control valve to limit the relieving load of the pressure relief valve
- No consideration of the expansion phenomenon in case of fire for the liquid in a vessel, before the boiling point is reached. Consequently the wet area is less than in reality. Nevertheless, this was not an issue because the valves were oversized
- The required area calculations of the relief valves for thermal expansion of liquids which flash through it, were sized considering no flashing. Again, no problem arose because of oversized valves
- The relieving load for a fire scenario of a vessel full of gas/vapor has been calculated by the empirical formula 4.4, which has been demonstrated in section 6-4 that could undersize valves. However, again, no problems were found
- The exchanger tube breakage scenario is rarely applied because the design basis kept the same design pressure for tubes and shell parts independent of the operation pressures.

## 5.5 Evaluation of the deficiencies and mitigation actions

The deficiencies encountered were resolved as soon as they were evaluated in accordance with the Management of Change procedure.

Concerning the outlet pressure drop deficiency, the following items were affected:

| Item No. | Outlet pressure drop, %SP | Mitigation action | Recalculated outlet pressure drop, %SP |
|---|---|---|---|
| Y13A | 20.2 | Y15C/D out of service | 10.2 |
| Y13B | 20.2 | Y15C/D out of service | 10.2 |
| Y14A | 20.5 | Y15C/D out of service | 10.4 |
| Y14B | 20.5 | Y15C/D out of service | 10.4 |
| Y15A | 23.5 | Y15C/D out of service | 17.2 |
| Y15B | 23.5 | Y15C/D out of service | 17.2 |
| Y15C | 25.4 | Out of service | -- |
| Y15D | 25.4 | Out of service | -- |
| Y103 | 27.2 | Y104/105 out of service | -- |

| Item No. | Outlet pressure drop, %SP | Mitigation action | Recalculated outlet pressure drop, %SP |
|---|---|---|---|
| Y104 | 25.8 | Out of service | -- |
| Y105 | 25.8 | Out of service | -- |
| Y1123 | 31.5 | Administrative changes | -- |
| Y1124 | 31.5 | Administrative changes | -- |
| Y1312 | 16.9 | None | 16.9 |
| Y3034 | 28.0 | Modulating action | 12.9 |
| Y3925 | 51.6 | Modulating action | 13.1 |
| Y6510A | 28.9 | Restriction lift | 15.2 |
| Y6510B | 28.9 | Restriction lift | 15.2 |
| Y72B | 56.4 | Smaller valve | -- |
| YS702/10 | 37.8 | Increased diameter | 18.0 |
| YS720/1 | 33.4 | New valve installed with bellows | 10.0 |
| YS861/1 | 15.3 | None | 15.3 |
| YS861/4 | 16.7 | Increased diameter | 12.0 |
| YS861/5 | 16.3 | Modified layout | 11.4 |
| YS861/8 | 15.0 | None | 15.0 |

Concerning the inlet pressure drop deficiency, the following mitigation actions have been taken:

| Item No. | Inlet pressure drop, %SP | Mitigation action |
|---|---|---|
| Y13A | 15.9 | Increasing diameter |
| Y13B | 15.9 | Increasing diameter |
| Y14A | 6.9 | Increasing diameter |
| Y14B | 6.9 | Increasing diameter |
| Y106 | 5.6 | Engineering analysis |
| Y107 | 5.6 | Engineering analysis |
| Y3925 | 4.6 | Engineering analysis |
| YS399/2 | 5.0 | Engineering analysis |
| YS405/5 | 12.7 | Increasing diameter |
| YS420/1 | 3.0 | None |
| YS491/4 | 5.0 | Engineering analysis |
| YS700/1 | 4.2 | Engineering analysis |
| YS700/2 | 4.2 | Engineering analysis |
| YS701/1 | 3.1 | Engineering analysis |
| YS701/2 | 3.1 | Engineering analysis |
| YS701/3 | 3.1 | Engineering analysis |
| YS701/4 | 3.1 | Engineering analysis |
| YS702/15 | 6.5 | Modification inlet piping |
| YS720/1 | 16.5 | Modification inlet piping |
| YS860/1 | 5.8 | Modification inlet piping |
| YS861/4 | 3.7 | Engineering analysis |
| YS861/5 | 11.9 | Modification inlet piping |

# Chapter 6. Special case studies related to the design phase

This chapter will present nine case studies, which represent the most complicated scenarios for calculating the required relief loads, required areas of the valves, inspection interval optimization and results of a turnaround. Moreover each case study shows a comparison between the different models available in the open literature to perform the calculations. Recommendations for the correct use of each model/procedure are also given. It is believed they would be interesting for any and every process safety engineer.

## 6.1 Total backpressure calculations in the flare network of existing plants

As explained in section 4.5 the outlet pressure drop calculation of the safety valves is required in order to design them and to check for stability problems as well. In this work the three existing petrochemical plants discharge to one flare.

However, the total backpressures of the valves were not available. Thus, in order to check the design of each valve which relieves to the flare network, the back pressures had to be calculated.

The existing engineering documentation of the plants reported that the dimensioning case for the flare was fire. Total power failure and cooling water failure were checked by the author as well and, it was confirmed that they are not the dimensioning case.

According to API 521 (2014) the first step is checking if the fire scenario is a plausible one. Fire contingency can be omitted in the following circumstances:

- When equipment is not located in or adjacent to areas containing flammable chemicals
- When equipment is located above a certain level from the grade or platform where potential accumulation of flammable liquid may occur. According to API 521 this level is 7.6 m
- When heat input from fire is insufficient to vaporize the liquid in the equipment within the reasonable amount of time required for corrective action to be taken by operators. Usually this time is 20 minutes (Mofrad and Norouzi, 2007)
- When equipment can be emptied safely if such fire occurs.

All these assumptions have been accepted here with the exception of the 20 minutes for corrective operation as explained below. This time is considered too optimistic.

**Assumptions taken in this work for the fire relief case calculations**

- Each piece of equipment engulfed in the fire is assumed to be isolated, which means that all heat and material inputs and outputs are assumed to have stopped. This assumption is based on the general plant operational practice to shut down the plant whenever fire is detected.

- Potential external fire is assumed to occur in only one particular fire zone. This fire zone is assumed to be a circle with a diameter of 30 m as a conservative basis.

- All of the relief valves in one particular fire zone are assumed to relieve at the same time and at their maximum relieving flowrates (not required flowrates).

- The amount of heat absorbed by equipment exposed to fire will depend on many factors, such as type of fuel, equipment shape and size, fire proofing and so on, but heat input is determined by correlations of API 521(2014) (not NFPA 30).

- Since depressurizing systems and operating procedures can fail in the event of a fire, no credit for such systems is considered during sizing the relief device for fire contingency.

- Similarly, an effective water deluge system depends on many factors, such as freezing weather, high winds, clogged systems, reliable water supply and equipment surface conditions. Hence, no credit is recommended by API 521 for environmental factors used in the equations to determine the heat load due to fire.

- Credit for insulation can only be taken if the insulation is a fire-proofing insulation that meets specific criteria (Parry, 1992). In this work no credit has been taken for the insulation of the plants, i.e. always F=1.

- The normal liquid level has not been used to calculate heat input for vessels and equipment that have an automatic level controller as recommended for API 521 (2014). Following good engineering practice the normal high liquid level is used in this work.

- Although adding 10% or 15% to the wetted area to take into account the annexed piping of the vessels is done by some designers, this approach was not considered in this work. Only if the pipe had a diameter greater than DN500 was it considered.

- To determine vapor generation, only that portion of the vessel that is wetted by its internal liquid and is equal to or less than 7.6 m above the flame source has been considered.

- The vessel heads protected by support skirts (i.e. bottoms from distillation columns) have been included in determining the wetted area.

- Equipment containing liquids or gases will behave differently under the effect of fire. Equipment containing liquids with a “reasonable” boiling point has the benefit of a good heat-transfer rate between the equipment walls and the contained liquid, resulting in a slow temperature rise at the walls. On the other hand, for the equipment containing gases, vapors or supercritical fluids, there will be a poor heat-transfer rate between the equipment walls and the contained fluid, which results in a very rapid temperature rise of equipment walls. Therefore, separate procedures have been followed to determine the fire case relief loads for these different situations.

- Concerning the duration of the fire except for a few unusual applications, the time element of fire relief is not recognized. That is, the time required to heat the contents of a vessel to relieving conditions is not considered in sizing the pressure relief system. All fire heat input is therefore assumed to be available for vaporizing or heating vessel contents. In this thesis there are the following unusual applications: A vessel full of liquid propylene has a relieving pressure higher than its critical pressure (45.6 barg). As the fire starts the temperature of the liquid increases and an isochoric transformation occurs until the liquid reaches the set pressure of the safety valve. From this point, liquid is relieved through the valve with an isentropic flash giving two phases at the outlet. If fire continues, the liquid changes to gas phase (supercritical fluid) when the temperature reaches 92.4 ºC (critical temperature of propylene) without boiling. From this moment the situation is a vessel with gas and it is known that the time can be 15-20 minutes before the rupture (API 521, 2008). Moreover, through the valve occurs a retrograde condensation, i.e. in the inlet of the valve there is a supercritical fluid and at the exit there are two phases until approximately 270 ºF is reached. With higher temperatures there is no retrograde condensation. In this case, the exit temperature is < 0 ºF depending on the total backpressure. It is necessary to write that each phase requires different approaches for relieving load calculations: a) in the first there is a thermal expansion of the liquid propylene with flashing through the valve. The relieving load can be calculated with the relieving formulas of API 521 (paragraph 5.14.3, 2008), on a timely basis, because the thermal-expansion coefficient changes until the critical temperature is reached. In this work the values of this coefficient were calculated up to 73.37 ºC by the Yaws correlation (Yaws, 1995) as it is the maximal temperature allowed for the correlation. The difficulty of getting the value of this coefficient from 73.37 ºC to 92.4 ºC was solved using the NIST databook (http://webbook.nist.gov/chemistry/fluid/) together with the approximation of API 521 (paragraph 5.14.4, 2008). In the second part of the process only supercritical fluid is relieved, in this case the equations of API 521 (2008) for vapor filled vessels exposed to fire were used together with other more exact methods (Ouderkirk, 2002; API 521 (paragraph 4.4.13.2.4.4, 2014)).

- There are some cases where it is necessary to determine if a vessel should be considered wetted or unwetted. Different authors (Wong, 1999 and Mofrad and Norouzi, 2007) considered that 20 minutes is the fire team response time. After these 20 minutes from the start of the fire, the vessel's exposed surface is cooled by the fire team and the relief requirements approaches zero. They wrote that this is a good practice especially for local fires. As stated before no consideration has been made here for this concept, because this assumption demands an exceptionally trained team of operators and firefighters.

- Before determining the final relief rate, the designer has the important task of determining the latent heat of vaporization of the contained liquid. In this work most cases are a simple liquid component. Only a few cases have been calculated through a semidynamic model as explained in PS PPM software or in Mofrad publication (Mofrad and Norouzi, 2007) for the determination at different times of the latent heat of liquid mixtures.

- In case of relieving conditions near the critical region, the latent heat of vaporization approaches zero as sensible heat dominates. For such conditions API 521(2008) suggests using a minimum latent heat value of 115 kJ/kg as an approximation. In this work this approach has not been used.

- The heat transfer between equipment walls and the contained fluid is very poor when the contained fluid is a gas, vapor or supercritical fluid. This results in a very rapid temperature rise in the equipment walls causing equipment failure due to heat stress even before the internal pressure reaches the set pressure of the safety valve. These vessels have to be protected by:
  a) Cooling the equipment surface with a water deluge system
  b) Providing automatic vapor depressuring systems
  c) Locating equipment to either eliminate or reduce the effects of fire
  d) Installing external fire-proofing insulation
  e) Using reliable fire-monitoring systems and a rapid-action fire-fighting team.

- For the equipment containing a low liquid inventory, that is to say, equipment in which all the contained liquid could vaporize within 20 minutes, if the vessel pressure when the last drop of liquid vaporizes is less than the valve set pressure, then the relief load to be considered is based on the procedure for gas filled equipment. However, if the vessel pressure when the last drop of liquid vaporizes is more than relief valve set pressure, the relief load to be considered was the maximum of relief load calculated based on considering liquid filled or gas filled vessel.

- One of the significant problems in designing the relief systems is knowing exactly if there is a single phase (vapor or liquid) or a mixed phase flow in the inlet and outlet of the safety valve. Generally a mixed phase flow requires a larger relief area than vapor or liquid phase flow.

However, it has been found that relief can be two-phase relief depending upon the nature of liquid and the initial fill level (see section 4.2).

**Calculation of the required area of the pressure relief valves**

The procedure followed in this work to calculate the required area was:

- The required relief area of safety valves for gas and liquid phases only were calculated according to manufacturer's information as explained in section 4.3. For the newest plant (Plant III) almost all the valves are LESER, also the software VALVESTAR 7.2.1 has been used together with the Engineering Book available at their web: www.leser.de. For the two older plants (Plants I and II) all the safety valves are Sempell. The Sempell Catalogue KS 27585 E "Safety Relief valves with DIN- and ANSI- Flanges" has been used.

- The calculation of two phase flow has been done using some of the following methods:
  a) API 520 Part I Annex C, 2014
  b) TPHEM modified (private software based on Simpson procedure (Simpson,1991))
  c) Leung method (Leung, 1996)
  d) Direct integration method (Darby, 2002 and API 520 Part I, 2014)
  e) ISO 4126-10 (2010)
  f) HNE-DS (Diener and Schmidt, 2004; Schmidt, 2013)
  g) Ouderkirk method (Ouderkirk, 2002)
  h) PS PPM software (method based on CCPS, 1998).

- The vapor release calculations through a vapor certified relief valve has been analyzed for inlet and outlet line losses based on the pressure relief valve's rated capacity at 10% overpressure (code allowed accumulation) of the valve. This is a very important concept because a few years ago the required rate was used.

- In this thesis, good engineering practice of relief system engineering was used: where the required vapor relief rate is < 50% of the relief valve's rated capacity at 10% overpressure and is < 25% of the rated capacity, a deviation of the above procedure has been allowed, using to use 50% of the valve's rated capacity for inlet and outlet line analysis if and only if the characteristic of the valve is as shown in figure 4-3.

**Design basis for the calculation of total back pressures of the flare network**

The design basis used here to calculate the built-up backpressure of each pressure relief valve has been:

- Required flow vs. rated flow: according to the API 521 (table 12, 2007) the design basis for tail pipes of a spring-operated relief valves is the rated capacity and the required relieving rate for the main header.

- Calculation of the backpressures in the safety valves: cases where the backpressure is greater than the limits specified by the valve manufacturer may result in chattering. The installation of bellows relief valves was explicitly explained as the solution to the problem of excessive backpressure (see section 4.6). In a conventional spring loaded pressure relief valve, the maximum allowable back pressure is typically limited to 10% of the valve set pressure. In this thesis up to 17 % of set pressure has been allowed for LESER valves according to the manufacturer. For the balanced valves (with bellows), the maximum allowable back pressure without derating of the valve's capacity has been limited to 30% of the valve set pressure.

- It is common to perform a flare newtwork study by evaluating only the cumulative cases that overpressure the system (total power failure, cooling water failure, fire, etc.). While this consideration is correct for the header and sub-header sizing, this evaluation may undersize the individual PRV tail piping. The evaluation of PRVs relieving independently will give the most conservative size of the valve outlet piping especially if there is a maximal Mach number limit (Zamora, Streblow, 2015). API 521 (2014) also gives the following recommendation “If the user has established a velocity criterion for tail pipes, the maximum velocity in a tail pipe should be calculated with the single source (the relief or depressurization device) as the only source discharging into the disposal system”

- Forces and moments imposed on the safety valve including thermal effects: as it was pointed out in section 4.7, API 521 (2008) recommends the calculation of the allowable forces and moments which may be imposed on the pressure relief device. This should include allowable forces and moments for both flowing and non-flowing conditions, where applicable. This data must be obtained from the device manufacturer. The same standard requires calculations of non-flowing piping loads on the pressure relief device. This includes both dead loads and loads imposed by thermal effects. In this work only a very few cases of vapor flowing to atmosphere have been calculated. The valves relieving to flare have not been checked as this work had been done by the engineering and construction company.

- Calculations of Acoustic Induced Vibration (AIV): as stated in section 4.8 a pressure relief valve relieving to a flare network would cause high frequency acoustic excitation to downstream piping which is potentially damaging to it. If the Sound Power Level generated by the safety valve is below 155 dB, the piping downstream of the safety valve is considered safe from AIV fatigue failure.

- Mach number: the Mach number for the pressure relief valve tail pipe is commonly limited to 0.7 for the maximum flow. For the collection header the Mach number is limited to 0.5 for the maximum flow as well, although it may be better to use 0.3 (NORSOK, 2006). In this thesis, there are some cases with Mach = 1 in plants I and II, i.e. there is critical flow at this point, in accordance with the statistical results of Riha and Streblow (2015), which found 15% of the valves with chocked flow present in piping.

- Momentum (density · velocity$^2$): as explained in section 4.5 for tail pipes, maximum momentum may be limited to 150,000 Pa, whilst for collection headers, it should be limited to 100,000 Pa. In any case, one has to check that the piping supports and the vibration analysis are correct as well.

- Noise level: fluid passing through the safety valve and tail and pipe header can generate significant noise and it can be transmitted along the tail pipe and header. One of the common safety requirements is to limit the noise level to 115 dBA (Noise level with A-weighted) during intermittent emergency relief scenario. Here, some cases have been evaluated.

- Two-phase flow pattern: the consideration of isentropic transformation through the safety valve in this thesis gives two-phase flow at the outlet with temperatures near the minimum-design metal temperature (MDMT) which may result in brittle fracture. In any case the piping specification for flare systems accounts for this issue (design temperature -50 ºC). There are also some cases with retrograde condensation.

  On a conservative basis, the calculations are made taking into account not only isothermal pressure-drop equations but also isenthalpic transformations with specific software (Aspen Flare System Analyzer v7.3)

  The adopted roughness of the pipe in this case study is ε = 0.07 mm, which corresponds to a not corroded pipe. However, in some cases where it was estimated a light corrosion might exist due to the characteristic of the released substances, a roughness factor of 0.3 mm has been used.

- Polymerization, hydrate and/or ice formation: a MOC analysis was conducted (see section 3.6) to check for the possible formation of ice in some points where steam is discharged together (cumulative cases) with light hydrocarbons when there is the possibility of ice formation and plugging/blocking the relief tail pipe.

**Results**

Zapico (2013) carried out an extensive study to find the dimensioning scenario for the existing flare. For the case of fire in the plants, he used a circle of 30 m of diameter (conservative) and looked for the maximal relieving load taking into account the equipment included in the circle. Six cases were identified by the author:

1. Fire in distillation area Plant III. Elevation 0 m
2. Fire in reaction area Plant III. Elevation 0 m
3. Fire in reaction area Plant III. Elevation 9.5 m
4. Fire in purification area Plants I&II. Elevation 0 m
5. Fire in reaction area Plants I&II. Elevation 7.0 m
6. Fire in distillation area Plants I&II. Elevation 0 m

The results of this study showed that case 1 (Fire in distillation area Plant III. Elevation 0 m) is the dimensioning one for the flare and will be given, as an example, in this section.

In all cases, it has been considered that when a fire starts, the reactors are depressured immediately by operations personnel. This consideration was adopted in the original specification of the flare and it has been maintained as it is recommended by paragraph 5.20 “Vapour depressuring” of API 521 (2008).

Table 6-1 shows the relieving loads for the case 1, resulting:

Maximum required load: 145500 kg/h

Maximum relief load: 271369 kg/h

Molecular weight: 42.3 kg/kmol

Temperature: 78 ºC

Figure 6-1 presents a flowsheet of the flare network including the total backpressures as a percentage of the set pressures. Tables 6-2a, 6-2b and 6-2c show the results of the total backpressure of the affected safety valves. Table 6-3 presents the pressure drop in the inlet of the safety valves. Tables 6-4a, 6-4b and 6-4c give the resistance coefficients for pipe and fittings in the discharge lines. Table 6.5 shows the same resistance coefficients for the inlet piping. Figure 6-2 represents the flowsheet used with the software Aspen Tech’s Flare System Analyzer (Aspen, 2011).

**Table 6-1. Plant III flare design basis.**

CASE 1: FIRE IN DISTILLATION AREA ELEVATION 0.0 m OF PLANT III

| SV No. | EQUIPMENT PROTECTED | SET PRESSURE barg | REQUIRED RELIEF LOAD kg/h | REQUIRED RELIEF LOAD kmol/h | MAXIMUM RELIEF LOAD kg/h | MAXIMUM RELIEF LOAD kmol/h | MOLECULAR WEIGHT kg/kmol | RELIEVING TEMPERATURE ºC | INPUT PHASE V or L |
|---|---|---|---|---|---|---|---|---|---|
| YS 860/08 | K862 | 40 | 15000 | 356.3 | 17241 | 409.5 | 42.1 | 90 | V |
| YS861/01 | B860 | 31 | 21000 | 498.8 | 21421 | 508.8 | 42.1 | 76 | V |
| YS861/04 | K860 | 15.5 | 15000 | 340.1 | 42274 (4) | 958.6 | 44.1 | 52 | V |
| YS861/05 | B864 | 15.5 | 4000 | 95.0 | 10183 (4) | 241.9 | 42.1 | 44 | V |
| YS861/08 | W860 | 31 | 9000 | 213.8 | 13474 | 320.0 | 42.1 | 77 | V |
| YS860/12 | W866 | 40 | 6000 | 142.5 | 44147 | 1048.6 | 42.1 | 87 | V |
| YS860/01 | B862 | 40 | 15500 | 372.1 | 62629 (4) | 1503.7 | 41.65 (2) | 89 | V |
| YS865/02.1.2 | | | 7500 (1) | | | | 42.1 | 66 | V |
| YS865/01 | | | 3000 (1) | | | | 42.1 | 66 | V |
| YS865/05 | | | 1000 (1) | | | | 42.1 | 66 | V |
| YS865/03 | | | 500 (1) | | | | 42.1 | 66 | V |
| YS865/04 | | | 2000 (1) | | | | 42.1 | 66 | V |
| Vent (F40115) | R400 | 30 | 30000 | 712.6 | 30000 | 712.6 | 42.1 | 80 | V |
| Vent (F41115) | R410 | 30 | 30000 | 712.6 | 30000 | 712.6 | 42.1 | 80 | V |
| | | | | | | | | | |
| **TOTAL** | | | **145500 (3)** | **3443.8** | **271369** | **6416.3** | **42.3** | **78** | |

NOTES

1. This relief load is not considered in the fire case. The refrigeration unit is not exposed to fire.
2. Homogeneous case. It is the dimensioning case for the flare.
3. Without considering the refrigeration unit XV 860. The $\Delta P$ in the bottom of flare is 697 mbar (flare manufacturer Information).
4. The $\Delta p$ in the input and output of the safety valve are calculated at 50% of maximum load.

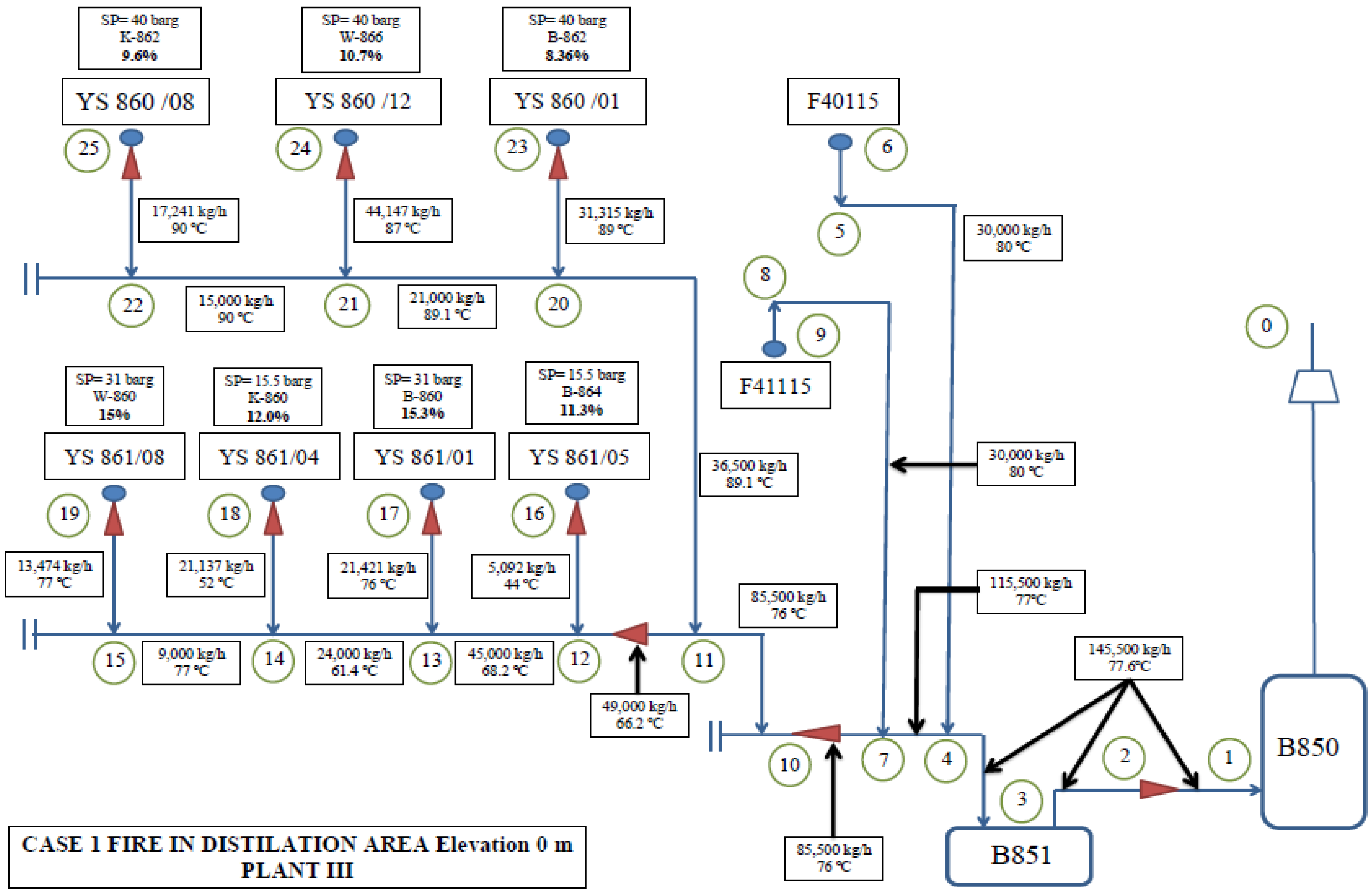



**Figure 6-1. Flare network flowsheet including calculated total backpressures for case 1.**

**Table 6-2a. Results of the total backpressure of the safety valves.**

CASE 1: FIRE IN DISTILLATION AREA ELEVATION 0.0 m OF PLANT III
PRESSURE IN THE BOTTOM OF THE FLARE: 697 mbar @ 145.5 t/h

| | | | | | F-40115 | | | F-41115 |
|---|---|---|---|---|---|---|---|---|
| **Section** | 1-2 | 2-3 | 3-4 | 4-5 | 5-6 | 4-7 | 7-8 | 8-9 |
| Nominal Diameter, mm | 500 | 600 | 600 | 400 | 200 | 600 | 400 | 200 |
| Inside Diameter, mm | 495.4 | 597.4 | 597.4 | 396.4 | 211.1 | 597.4 | 396.4 | 211.1 |
| Load, kg/h | 145,500 | 145,500 | 145,500 | 30,000 | 30,000 | 115,500 | 30,000 | 30,000 |
| Temperature, ºC | 77.6 | 77.6 | 77.6 | 80.0 | 80.0 | 77.0 | 80.0 | 80.0 |
| Viscosity, cP | 0.0100 | 0.0101 | 0.0101 | 0.0102 | 0.0102 | 0.01002 | 0.0107 | 0.0107 |
| Molecular Weight, kg/kmol | 42.44 | 42.44 | 42.44 | 42.10 | 42.10 | 42.53 | 42.10 | 42.10 |
| Downstream pressure, bar g | 0.697 | 0.810 | 0.877 | 0.960 | 0.984 | 0.960 | 0.970 | 0.992 |
| Mach Number | 0.320 | 0.206 | 0.199 | 0.090 | 0.313 | 0.151 | 0.089 | 0.311 |
| Reynolds Number x $10^6$ | 10.34 | 8.574 | 8.573 | 2.631 | 4.941 | 6.826 | 2.497 | 4.684 |
| Friction Factor (Moody) | 0.0129 | 0.0125 | 0.0125 | 0.0138 | 0.0155 | 0.0126 | 0.0139 | 0.0154 |
| Upstream pressure, bar g | 0.810 | 0.877 | 0.960 | 0.984 | 1.144 | 0.970 | 0.992 | 1.196 |
| **ΔP (bar)** | 0.113 | 0.067 | 0.083 | 0.024 | 0.160 | 0.010 | 0.022 | 0.204 |
| **Total backpressure, %** | - | - | - | - | **3.47** | - | - | **3.63** |

**Table 6-2b. Results of the total backpressure of the safety valves.**

CASE 1: FIRE IN DISTILLATION AREA ELEVATION 0.0 m OF PLANT III
PRESSURE IN THE BOTTOM OF THE FLARE: 697 mbar @ 145.5 t/h

| | | | | YS 861/05 | | YS 861/01 | | YS 861/04 |
|---|---|---|---|---|---|---|---|---|
| **Section** | 7-10 | 10-11 | 11-12 | 12-16 | 12-13 | 13-17 | 13-14 | 14-18 |
| Nominal Diameter, mm | 600 | 300 | 200 | 100 | 200 | 150 | 200 | 200 |
| Inside Diameter, mm | 597.4 | 315.9 | 211.1 | 107.1 | 211.1 | 160.3 | 211.1 | 211.1 |
| Load, kg/h | 85,500 | 85,500 | 49,000 | 10,183 | 45,000 | 21,421 | 24,000 | 42,274 |
| Temperature, ºC | 76.0 | 76.0 | 66.2 | 44.0 | 68.2 | 76.0 | 61.4 | 52.0 |
| Viscosity, cP | 0.0100 | 0.0100 | 0.0096 | 0.0091 | 0.0097 | 0.0101 | 0.00932 | 0.00887 |
| Molecular Weight, kg/kmol | 42.68 | 42.68 | 43.25 | 42.1 | 42.77 | 42.1 | 43.35 | 44.1 |
| Downstream pressure, bar g | 0.970 | 0.974 | 1.409 | 1.478 | 1.478 | 1.571 | 1.571 | 1.585 |
| Mach Number | 0.111 | 0.395 | 0.406 | 0.311 | 0.366 | 0.297 | 0.184 | 0.315 |
| Reynolds Number x $10^6$ | 5.081 | 9.608 | 8.532 | 3.684 | 7.801 | 4.695 | 4.313 | 7.980 |
| Friction Factor (Moody) | 0.0127 | 0.0142 | 0.0153 | 0.0179 | 0.0154 | 0.0163 | 0.0154 | 0.0154 |
| Upstream pressure, bar g | 0.974 | 1.409 | 1.478 | 2.504 | 1.571 | 4.734 | 1.585 | 2.588 |
| **ΔP (bar)** | 0.004 | 0.435 | 0.069 | 1.027 | 0.093 | 3.163 | 0.014 | 1.003 |
| **Total backpressure, %** | - | - | - | **16.3** | - | **15.3** | - | **16.7** |

**Table 6-2c. Results of the total backpressure of the safety valves.**

CASE 1: FIRE IN DISTILLATION AREA ELEVATION 0.0 m OF PLANT III
PRESSURE IN THE BOTTOM OF THE FLARE: 697 mbar @ 145.5 t/h

| | | YS 861/08 | | YS 860/01 | | YS 860/12 | | YS 860/08 |
|---|---|---|---|---|---|---|---|---|
| **Section** | 14-15 | 15-19 | 11-20 | 20-23 | 20-21 | 21-24 | 21-22 | 22-25 |
| Nominal Diameter, mm | 200 | 100 | 250 | 150 | 250 | 250 | 250 | 150 |
| Inside Diameter, mm | 211.1 | 107.1 | 265.0 | 160.3 | 265.0 | 265.0 | 265.0 | 160.3 |
| Load, kg/h | 9,000 | 13,474 | 36,500 | 62,629 | 21,000 | 44,147 | 15,000 | 17,241 |
| Temperature, ºC | 77.0 | 77.0 | 89.1 | 89.0 | 89.1 | 87.0 | 90.0 | 90.0 |
| Viscosity, cP | 0.0101 | 0.0101 | 0.0104 | 0.0104 | 0.0104 | 0.0104 | 0.0105 | 0.0105 |
| Molecular Weight, kg/kmol | 42.1 | 42.1 | 41.9 | 41.65 | 42.1 | 42.1 | 42.1 | 42.1 |
| Downstream pressure, bar g | 1.585 | 1.588 | 1.409 | 1.532 | 1.535 | 1.536 | 1.536 | 1.539 |
| Mach Number | 0.072 | 0.416 | 0.202 | 0.903 | 0.110 | 0.231 | 0.079 | 0.247 |
| Reynolds Number x $10^6$ | 1.494 | 4.408 | 4.667 | 13.23 | 2.684 | 5.675 | 1.913 | 3.634 |
| Friction Factor (Moody) | 0.0157 | 0.0179 | 0.0148 | 0.0162 | 0.0149 | 0.0147 | 0.0151 | 0.0164 |
| Upstream pressure, bar g | 1.588 | 4.660 | 1.532 | 6.916 | 1.536 | 4.263 | 1.539 | 3.841 |
| **ΔP (bar)** | 0.003 | 3.072 | 0.123 | 5.384 | 0.005 | 2.727 | 0.003 | 2.301 |
| **Total backpressure, %** | - | **15.0** | - | **17.3** | - | **10.7** | - | **9.60** |

**Table 6-3. ΔP in the input of the safety valves.**

CASE 1: FIRE IN DISTILLATION AREA ELEVATION 0.0 m OF PLANT III

| **SAFETY VALVE** | YS 861/05 | YS 861/01 | YS 861/04 | YS 861/08 | YS 860/01 | YS 860/12 | YS 860/08 |
|---|---|---|---|---|---|---|---|
| Set Pressure, bar g | 15.5 | 31.0 | 15.5 | 31.0 | 40.0 | 40.0 | 40.0 |
| Nominal Diameter, mm | 40 | 40 | 80 | 50 | 50 | 80 | 80 |
| Inside Diameter, mm | 43.1 | 43.1 | 82.5 | 54.5 | 54.5 | 81.7 | 83.1 |
| Load, kg/h | 10,183 | 21,421 | 42,274 | 13,474 | 62,629 | 44,147 | 17,241 |
| Temperature, ºC | 44.0 | 76.0 | 52.0 | 77.0 | 89.0 | 87.0 | 90.0 |
| Molecular Weight, kg/kmol | 42.1 | 42.1 | 44.1 | 42.1 | 41.65 | 42.1 | 42.1 |
| Viscosity, cP | 0.01 | 0.0126 | 0.01 | 0.0126 | 0.0146 | 0.01 | 0.0153 |
| Relieving Pressure, bar a | 18.06 | 35.11 | 18.06 | 35.11 | 45.01 | 45.01 | 45.01 |
| Mach Number | 0.233 | 0.2327 | 0.270 | 0.092 | 0.296 | 0.098 | 0.036 |
| Reynolds Number x $10^6$ | 8.356 | 13.95 | 18.12 | 6.940 | 26.95 | 19.11 | 4.796 |
| Friction Factor (Moody) | 0.022 | 0.022 | 0.019 | 0.021 | 0.021 | 0.019 | 0.019 |
| Pressure at the exit of equipment, bara | 19.81 | 35.38 | 18.98 | 35.59 | 47.11 | 45.71 | 45.62 |
| **ΔP (bar)** | 1.75 | 0.27 | 0.92 | 0.48 | 2.10 | 0.70 | 0.61 |
| **%** | **11.3** | **0.88** | **5.91** | **1.55** | **5.24** | **1.75** | **1.52** |

**Table 6-4a. Resistance coefficients K of pipe and fittings.**

CASE 1: FIRE IN DISTILLATION AREA ELEVATION 0.0 m OF PLANT III (DISCHARGE PIPING)

| | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | F-40115 | | | | | | F-41115 | |
| **Section** | 1-2 | | 2-3 | | 3-4 | | 4-5 | | 5-6 | | 4-7 | | 7-8 | | 8-9 | |
| Nominal diameter, mm | 500 | | 600 | | 600 | | 400 | | 200 | | 600 | | 400 | | 200 | |
| Inside Diameter, mm | 495.4 | | 597.4 | | 597.4 | | 396.4 | | 211.1 | | 597.4 | | 396.4 | | 211.1 | |
| Pipe length, m and K | 0.19 | $5\cdot10^{-3}$ | 14.9 | 0.31 | 18.2 | 0.38 | 19.6 | 0.68 | 3.97 | 0.29 | 8.91 | 0.19 | 23.4 | 0.82 | 4.2 | 0.30 |
| 90º elbows, nº and K | 1 | 0.144 | 2 | 0.29 | 4 | 0.58 | 3 | 0.47 | 1 | 0.20 | - | - | 3 | 0.47 | 1 | 0.20 |
| 45º elbows, nº and K | - | - | 3 | 0.58 | - | - | 1 | 0.21 | 1 | 0.22 | - | - | 1 | 0.21 | 3 | 0.67 |
| Straight T , nº and K | - | - | - | - | 1 | 0.24 | 3 | 0.78 | - | - | 1 | 0.24 | 2 | 0.52 | - | - |
| Branch T, nº and K | - | - | - | - | - | - | 1 | 0.78 | 1 | 0.84 | - | - | 1 | 0.78 | 1 | 0.84 |
| Enlargements, nº and K | - | - | - | - | - | - | - | - | - | - | - | - | - | - | - | - |
| Contractions, nº and K | 1 | 0.07 | - | - | - | - | - | - | - | - | - | - | - | - | - | - |
| Change-over valve, DN and K | - | - | - | - | - | - | - | - | - | - | - | - | - | - | - | - |
| Equipment entrance, K | 1 | 1.00 | - | - | 1 | 1.00 | - | - | - | - | - | - | - | - | - | - |
| Equipment exit, K | - | - | 1 | 0.50 | - | - | - | - | - | - | - | - | - | - | - | - |
| **Sum of K** | 1.21 | | 1.68 | | 2.20 | | 2.92 | | 1.55 | | 0.43 | | 2.79 | | 2.01 | |

**Table 6-4b. Resistance coefficients K of pipe and fittings.**

CASE 1: FIRE IN DISTILLATION AREA ELEVATION 0.0 m OF PLANT III (DISCHARGE PIPING)

| | | | | YS 861/05 | | YS 861/01 | | YS 861/04 |
|---|---|---|---|---|---|---|---|---|
| **Section** | 7-10 | 10-11 | 11-12 | 12-16 | 12-13 | 13-17 | 13-14 | 14-18 |
| Nominal Diameter, mm | 600 | 300 | 200 | 100 | 200 | 150 | 200 | 200 |
| Inside Diameter, mm | 597.4 | 315.9 | 211.1 | 107.1 | 211.1 | 160.3 | 211.1 | 211.1 |
| Pipe length, m and K | 4.22 0.09 | 14.0 0.63 | 0.48 0.03 | 9.3 1.55 | 2.96 0.22 | 14.1 1.44 | 0.32 0.02 | 12.5 0.91 |
| 90º elbows, nº and K | - - | 1 0.18 | - - | 2 0.48 | - - | 5 1.05 | - - | 5 0.98 |
| 45º elbows, nº and K | - - | - - | - - | - - | - - | 2 0.48 | - - | 1 0.22 |
| Straight T, nº and K | 1 0.24 | 3 0.78 | 1 0.05 | - - | 1 0.28 | - - | 1 0.28 | - - |
| Branch T, nº and K | - - | 1 0.78 | - - | 2 2.04 | - - | 1 0.90 | - - | 1 0.84 |
| Enlargements, nº and K | - - | 1 0.35 | 1 0.21 | 1 5.51 | - - | 1 39.4 | - - | 1 5.69 |
| Contractions, nº and K | - - | - - | - - | - - | - - | - - | - - | - - |
| Change-over valve, DN and K | - - | - - | - - | - - | - - | - - | - - | - - |
| Equipment entrance, K | - - | - - | - - | - - | - - | - - | - - | - - |
| Equipment exit, K | - - | - - | - - | - - | - - | - - | - - | - - |
| **Sum of K** | 0.33 | 2.72 | 0.29 | 9.58 | 0.50 | 43.3 | 0.30 | 8.64 |

**Table 6-4c. Resistance coefficients K of pipe and fittings.**

CASE 1: FIRE IN DISTILLATION AREA ELEVATION 0.0 m OF PLANT III (DISCHARGE PIPING)

| | | YS 861/08 | | YS 860/01 | | YS 860/12 | | YS 860/08 |
|---|---|---|---|---|---|---|---|---|
| **Section** | 14-15 | 15-19 | 11-20 | 20-23 | 20-21 | 21-24 | 21-22 | 22-25 |
| Nominal Diameter, mm | 200 | 100 | 250 | 150 | 250 | 250 | 250 | 150 |
| Inside Diameter, mm | 211.1 | 107.1 | 265 | 160.3 | 265 | 265 | 265 | 160.3 |
| Pipe lenght, m and K | 2.17 0.16 | 3.73 0.62 | 5.29 0.29 | 12.5 1.27 | 0.17 0.01 | 7.83 0.43 | 1.96 0.11 | 3.96 0.40 |
| 90º elbow, nº and K | - - | 1 0.24 | 1 0.20 | 5 1.05 | - - | 3 0.59 | - - | 3 0.63 |
| 45º elbow, nº and K | - - | 1 0.27 | - - | - - | - - | - - | - - | - - |
| Straight T, nº and K | 1 0.28 | - - | 4 1.12 | - - | 1 0.28 | - - | 1 0.28 | 1 0.30 |
| Branch T, nº and K | - - | 1 1.02 | 1 0.84 | 1 0.90 | - - | 1 0.84 | - - | 1 0.90 |
| Enlargements, nº and K | - - | 1 18.0 | - - | 1 5.18 | - - | 1 58.4 | - - | 1 39.4 |
| Contrations, nº and K | - - | - - | - - | - - | - - | - - | - - | - - |
| Change-over valve, DN and K | - - | - - | - - | - - | - - | - - | - - | - - |
| Equipment entrance, K | - - | - - | - - | - - | - - | - - | - - | - - |
| Equipment exit, K | - - | - - | - - | - - | - - | - - | - - | - - |
| **Summe of K** | 0.44 | 20.2 | 2.45 | 8.40 | 0.29 | 60.3 | 0.39 | 41.6 |

**Table 6-5. Resistance coefficients K of pipe and fittings.**

CASE 1: FIRE IN DISTILLATION AREA ELEVATION 0.0 m OF PLANT III (INLET PIPING)

| SAFETY VALVE | YS 861/05 | | YS 861/01 | | YS 861/04 | | YS 861/08 | | YS 860/01 | | YS 860/12 | | YS 860/08 | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Nominal Diameter, mm | 40 | | 40 | | 80 | | 50 | | 50 | | 80 | | 80 | |
| Inside Diameter, mm | 43.1 | | 43.1 | | 82.5 | | 54.5 | | 54.5 | | 81.7 | | 83.1 | |
| Pipe lenght, m and K | 2.58 | 1.29 | - | - | 43.1 | 0.12 | 0.5 | 0.09 | 0.32 | 0.12 | 7.81 | 1.04 | 24.6 | 5.60 |
| 90º elbow, nº and K | 3 | 0.59 | - | - | 5 | 0.01 | 1 | $1 \cdot 10^{-4}$ | - | - | 2 | 0.33 | 7 | 1.76 |
| 45º elbow, nº and K | - | - | - | - | 1 | $2 \cdot 10^{-3}$ | - | - | - | - | - | - | 1 | 0.29 |
| Straight T, nº and K | 1 | 0.42 | - | - | - | - | - | - | 1 | 0.38 | - | - | 3 | 1.08 |
| Branch T, nº and K | 1 | 1.26 | - | - | 1 | 1.08 | 1 | 1.14 | - | - | 1 | 1.08 | 1 | 1.08 |
| Enlargements, nº and K | - | - | - | - | - | - | - | - | - | - | - | - | - | - |
| Contractions, nº and K | - | - | 1 | 0.08 | 1 | 0.18 | 1 | 2.01 | - | - | 1 | 0.58 | 1 | 11.1 |
| Change-over valve, DN and K | - | - | - | - | - | - | - | - | - | - | - | - | - | - |
| Equipment entrance, K | - | - | - | - | - | - | - | - | - | - | - | - | - | - |
| Equipment input, K | 1 | $3.8 \cdot 10^{-4}$ | 1 | 0.20 | 1 | $5 \cdot 10^{-3}$ | 1 | $5 \cdot 10^{-4}$ | 1 | 0.50 | 1 | 0.17 | 1 | 0.50 |
| **Summe of K** | 3.55 | | 0.28 | | 1.33 | | 3.24 | | 1.00 | | 3.21 | | 21.4 | |

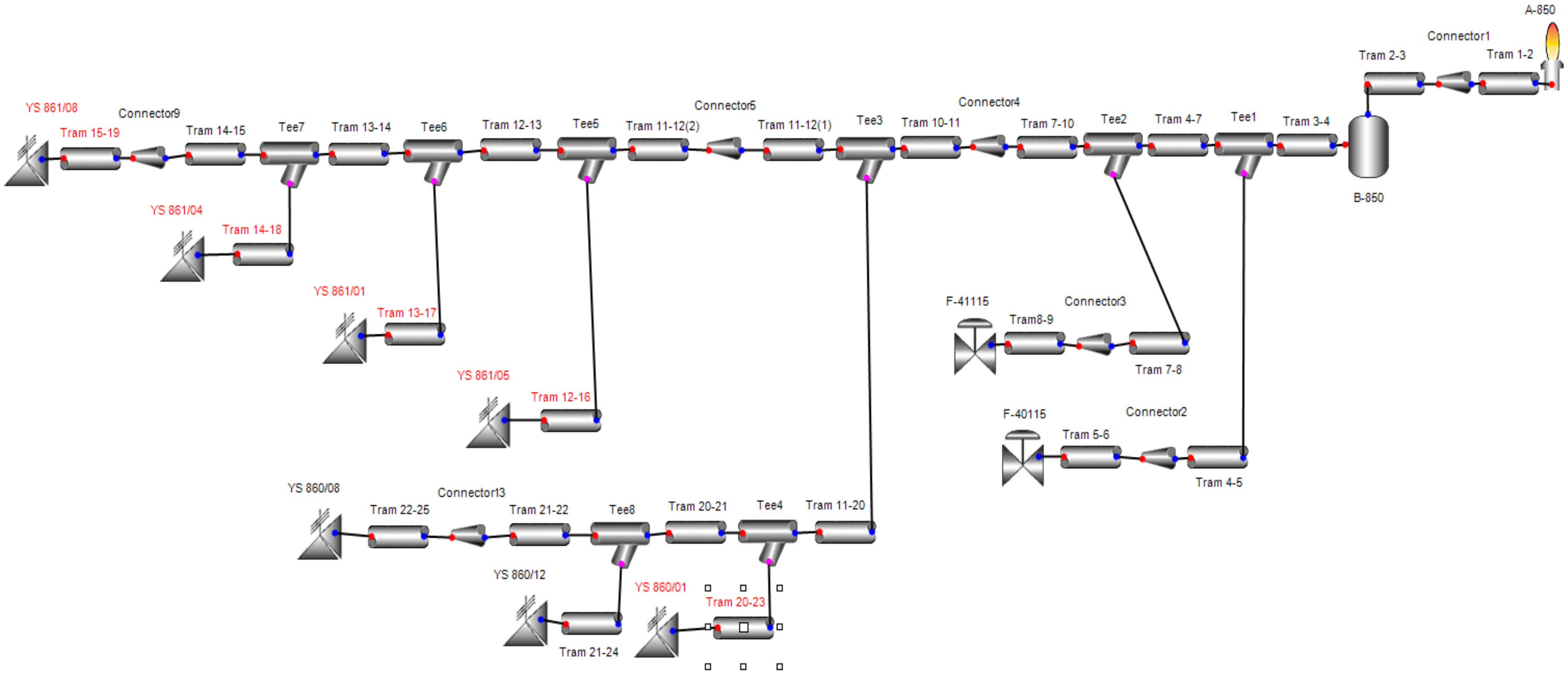



**Figure 6-2. Flowsheet used with the software Aspen Tech's Flare System Analyzer for case 1.**

The results of the different total backpressures for the pressure relief valves affected for the Case 1, are showed in Table 6-6.

**Table 6-6. Results of total backpressures of the PRV's Case 1 (% with respect to gauge set pressure) (Zapico, 2013).**

| PRV No | API 521(2007) isothermal (gas) (1) | AFSA v7.3 Isothermal (gas) (2) | API 520-2008 Adiabatic (gas) (3) | AFSA v7.3 Adiabatic (gas) (4) | AFSA v7.3 (two-phase flow) (5) |
|---|---|---|---|---|---|
| YS 860/01 | 17.3 | 17.6 | 17.3 | 17.3 | 14.7 |
| YS 860/08 | 9.6 | 10.8 | 9.8 | 10.5 | 7.9 |
| YS 860/12 | 10.7 | 11.4 | 10.8 | 11.0 | 8.0 |
| YS 861/01 | 15.3 | 16.7 | 15.6 | 16.5 | 11.4 |
| YS 861/04 | 16.7 | 19.8 | 16.8 | 19.5 | 13.5 |
| YS 861/05 | 16.3 | 17.7 | 16.3 | 17.4 | 11.9 |
| YS 861/08 | 15.0 | 17 | 15.2 | 16.7 | 14.6 |

Notes:

1- The equations 26 and 27 of API 521(2008) have been used. Roughness 0.07 mm

2- The Aspen Flare System Analyzer v7.3 - aspenONE was used. Roughness 0.07 mm

3- The equations of Annex E of API 520-2008 have been used. Roughness 0.07 mm

4- Peng Robinson as EOS was used. Roughness 0.07 mm

5- In this case AFSA is allowed to perform heat transfer calculations to ambient. Peng Robinson as EOS was used and Beggs Brill for the pressure drop. Atmospheric temperature 25 ºC and wind velocity 5m/s. Roughness 0.07 mm

**Considerations:**

a) The generally accepted idea that the isothermal flow is conservative with respect to pressure drop does not follow this example. The reason for this, as already pointed out by Bonilla (Bonilla, 1978), is because of the lower temperature reached by the fluid at the exit of the valve in comparison with the isothermal flow, that considers no temperature changes through the pressure relief valve. Some tail pipes have product at -30ºC.
b) The API 521 (2007) model for isothermal flow was implemented through an independently developed spreadsheet and the properties were previously calculated through API Technical Data Book and the NIST webbook (http://webbook.nist.gov/chemistry/fluid/).
c) Results are greater with the Aspen software probably due to the improved physical properties.
d) The difference between the adiabatic and isothermal results is not significant.
e) A practical problem with the Aspen software was that the DIN piping diameter had to match the inlet diameter of the ANSI norm (Schedule Number) because the software does not allow working in DIN dimensions. The diameters were elected through a schedule number always taking a diameter equal or smaller as a conservative basis.
f) Only the valve YS 861/04 not follows the 17 % maximum backpressure rule and has to be modified.

## 6.2 Comparison of different methods for the design of PRVs with Z < 0.8 or Z > 1.1

In this work, many cases exist in which the relieving fluid is a vapor or gas with a compressibility factor less than 0.8. In this case, API 520 (Part I, 2008, 2014) recommends a rigorous calculation using the direct integration method.

In reviewing the literature for this situation, some models are available for the engineer to perform the design of pressure relief valves. A case study is chosen consisting of the fire scenario for a vertical condenser protected by the pressure relief valve YS 414/1-2. Figure 6-3 shows a scheme of the valve situation and Figure 6-4 is a picture of the valve.

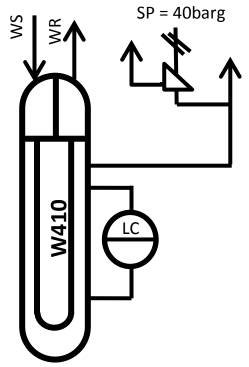



**Fig 6-3. Scheme of the installation of YS 414/1-2. Fig 6-4. Picture of YS 414/1-2 and equipment.**

### Design basis

Set pressure = 40 barg; $P_R$ = 45.013 bara; $T_R$ = 88.85 ºC; Relieving load: 30000 kg/h; $\alpha_{d,DG} = 0.56$; $\alpha_{d,F} = 0.42$; Initial temperature = 60 ºC; Initial pressure = 29 barg.

The range of boiling temperatures of the mixture at 45.013 bara is between 88.3ºC and 89.5 ºC according to the Aspen Hysys V7.3 with Peng Robinson as EOS simulation results. Taking a temperature of 89.5 ºC, the isentropic flashes have been performed. The initial point is (gas phase):

($P_R$ = 45.013 bara and $T_R$ = 89.5 ºC) with a composition (mass fraction): ethane 0.0019, propylene 0.9693, propane 0.0279, heptane 0.0003, methane 0.0002; hydrogen 0.0004; and the thermodynamic properties:

Entropy = 61.59 kJ/kmol ºC; Density = 156.6 kg/m$^3$; Z = 0.400; Cp/Cv ideal = 1.014; Cp/Cv real = 8.534; Viscosity = 0.01788 Cp; Molecular Weight = 41.87 kg/kmol; Molar volume = 0.2675 m$^3$/kmol; Critical temperature = 91.72 ºC; Critical pressure = 47.21 bara; Critical volume = 0.2009 m$^3$/kmol.

### Case 1. Direct integration

The formulas to be applied are (API 520, part I, 2014):

$$G^2 = \left[(\rho_t)^2 \cdot \left(-2\int_{P_0}^{P_t} \frac{dP}{\rho}\right)\right]_{max}$$

and

$$\int_{P_0}^{P_t} \frac{dP}{\rho} \cong \sum_{i=0}^{t} 2\frac{P_{i+1} - P_i}{\rho_{i+1} + \rho_i}$$

A step of 4% of $P_R$ has been taken as in the API 520 example. Thus,

| Pressure Pa | Temperature ºC | Density kg/m$^3$ | Integrand m$^2$/s$^2$ | Summe m$^2$/s | Mass Flux kg/sm$^2$ |
|---|---|---|---|---|---|
| 4501000 | 89.5 | 156.6 | -- | -- | -- |
| 4321000 | 87.2 | 147.5 | -1183.8 | -1183.8 | 7177 |
| 4141000 | 84.8 | 138.5 | -1258.7 | -2442.5 | 9680 |

| | | | | | |
|---|---|---|---|---|---|
| 3961000 | 82.4 | 129.8 | -1341.8 | -3784.3 | 11292 |
| 3781000 | 79.9 | 121.4 | -1433.1 | -5217.4 | 12401 |
| 3601000 | 77.3 | 113.3 | -1533.9 | -6751.3 | 13166 |
| 3421000 | 74.6 | 105.5 | -1645.3 | -8396.6 | 13672 |
| 3241000 | 71.8 | 97.96 | -1769.4 | -10166.0 | 13968 |
| 3061000 | 69.9 | 90.73 | -1907.9 | -12073.9 | **14099** |
| 2881000 | 65.8 | 83.79 | -2062.8 | -14136.7 | 14089 |
| 2701000 | 62.6 | 77.14 | -2237.0 | -16373.7 | 13959 |
| 2521000 | 59.2 | 70.75 | -2434.2 | -18807.9 | 13722 |
| 2341000 | 55.6 | 64.61 | -2659.6 | -21467.5 | 13388 |
| 2161000 | 51.9 | 58.70 | -2919.5 | -24386.9 | 12964 |
| 1981000 | 47.9 | 53.00 | -3222.9 | -27609.8 | 12454 |
| 1801000 | 43.6 | 47.51 | -3934.4 | -31544.2 | 11933 |

The maximal mass flux is obtained at 30.61 bara and 69.9 ºC.

According to equation C.10 of API 520 (2014):

$$A = \frac{277.8 \cdot W}{K_d \cdot K_b \cdot K_c \cdot K_v \cdot G}$$

where

A required effective discharge area, mm$^2$; W mass flow rate, kg/h; $K_d$ discharge coefficient; $K_b$ = 1 (no balanced valve); $K_c$ = 1 (no rupture disc installed); $K_v$ = 1 viscosity correction factor and $K_d$ = 0.56 from LESER datasheet (lift restricted to 8 mm), which corresponds to the gas phase one, as recommended by Darby (2003).

$$A = \frac{277.8 \cdot 30000}{0.56 \cdot 1 \cdot 1 \cdot 1 \cdot 14099} = 1055 \text{ mm}^2$$

**Case 2. Calculation using the isentropic expansion factor**

Smith and Burgess (2015) presented a method that will be followed here because of its simplicity. First the reduced volume of the mixture is calculated (data from process simulator):

$$V_R = \frac{V}{V_c} = \frac{0.2675 \frac{m^3}{kmol}}{0.2009 \frac{m^3}{kmol}} = 1.33$$

Although Smith does not recommend using an isentropic expansion factor for the case $V_R \leq 2$, it will be used here for comparison.

The isentropic expansion factor is by definition

$$n = \frac{\nu}{\rho}\left(\frac{\partial P}{\partial \nu}\right)_T \frac{C_p}{C_v}$$

According to Kim et al. (2011), the value of n for the Peng Robinson equation of state is:

$$n = \ln\frac{P_1}{P_2}\left[\ln\frac{\nu_2}{\nu_1}\right]^{-1}$$

According to the previous calculation by the direct integration method, the choking pressure is 30.61 bara. Thus

$\rho @\ 45.01\ bara = 156.6 \frac{kg}{m^3}$ → ν = 0.006385 m$^3$/kg

$\rho @ 30.61\ bara\ = 90.73 \frac{kg}{m^3}$ → ν = 0.011022 $m^3$/kg

Given values

$$n = \ln \frac{45.013}{30.61} \left[ \ln \frac{0.011022}{0.006385} \right]^{-1} = 0.706$$

Using the AD Merkblatt-A2 (2006) equations:

$$\psi_{max} = \sqrt{\frac{k}{k+1} \left( \frac{2}{k+1} \right)^{\frac{1}{k-1}}} = \sqrt{\frac{0.706}{1.706} \left( \frac{2}{1.706} \right)^{\frac{1}{0.706-1}}} = 0.3745$$

$$A_0 = 0.1791 \frac{30000}{0.3745 \cdot 0.56 \cdot 45.013} \sqrt{\frac{362.65 \cdot 0.4}{41.87}} = 1059\ mm^2$$

**Case 3. Calculation using ideal gas specific heat ratio**

In this case the Cp/Cv ideal at the relieving temperature is 1.014 and the compressibility factor is 0.4. Using the AD Merkblatt A2 equations:

$$\psi_{max} = \sqrt{\frac{k}{k+1} \left( \frac{2}{k+1} \right)^{\frac{1}{k-1}}} = \sqrt{\frac{1.014}{2.014} \left( \frac{2}{2.014} \right)^{\frac{1}{0.014}}} = 0.431$$

$$A_0 = 0.1791 \frac{30000}{0.431 \cdot 0.56 \cdot 45.013} \sqrt{\frac{362.65 \cdot 0.4}{41.87}} = 921\ mm^2$$

The chocked pressure is:

$$P_{chocke} = P_1 \left( \frac{2}{k+1} \right)^{\frac{k}{k-1}} = 45.013 \left( \frac{2}{2.014} \right)^{\frac{1.014}{0.014}} = 27.16\ bara$$

**Case 4. Calculation using real gas specific heat ratio**

Using the equation of Kim e al. (2011) with the design basis: Cp/Cv real = 8.534 and Z = 1

$$P_{chocke} = P_1 \left( \frac{2}{k+1} \right)^{\frac{k}{k-1}} = 45.013 \left( \frac{2}{9.534} \right)^{\frac{8.534}{7.534}} = 7.67\ bara$$

$$C_k = 0.03948 \sqrt{k \left( \frac{2}{k+1} \right)^{\frac{k+1}{k-1}}} = 0.03948 \sqrt{8.534 \left( \frac{2}{9.534} \right)^{\frac{9.534}{7.534}}} = 0.0429$$

$$A_0 = \frac{W}{C_k \cdot K_d \cdot P_1 \cdot K_b \cdot K_c} \sqrt{\frac{T_1}{M}} = \frac{30000}{0.0429 \cdot 0.56 \cdot 4501.3 \cdot 1 \cdot 1} \sqrt{\frac{362.65}{41.87}} = 816\ mm^2$$

**Case 5. Calculation using ideal gas specific heat ratio and Z = 1**

In this case Cp/Cv ideal at $T_R$ is 1.014 and Z = 1 and $\psi_{max} = 0.431$ as in case 3

The required area according to the AD Merkblatt-A2 (2006) is

$$A_0 = 0.1791 \frac{30000}{0.431 \cdot 0.56 \cdot 45.013} \sqrt{\frac{362.65 \cdot 1}{41.87}} = 1455\ mm^2$$

In Table 6-7 a summary of results is presented:

**Table 6-7. Comparison of sizing methods when the compressibility factor Z < 0.8**

| CASE | Ideal gas specific heat ratio used | Relieving temperature used, °C | Isentropic expansion coefficient used | Compressibility factor Z | Discharge area required $mm^2$ | Likely to be conservative |
|---|---|---|---|---|---|---|
| Original datasheet | 1.2 (1) | 85.0 | -- | 0.400 | 859 | No |
| 1. Direct integration | -- | 89.5 | -- | -- | 1055 | -- |
| 2. Using the isentropic expansion factor | -- | 89.5 | 0.706 | 0.400 | 1059 | -- |
| 3.Using ideal gas specific heat ratio at relieving conditions | 1.014 | 89.5 | -- | 0.400 | 921 | No |
| 4.Using real gas specific heat ratio | -- | 89.5 | 8.534 (2) | 1.0 | 816 | No |
| 5.Using ideal gas specific heat at relieving conditions and Z = 1 | 1.014 | 89.5 | -- | 1.0 | 1455 | Yes |

Notes:

(1) It is not clear how this value was obtained by the Engineering Company. It was common practice before using process simulators to take the predominant component in a stream and find the ideal gas specific heat ratio at standard conditions for that component (so, using table 7 of API 520, part I, 2014 would have yielded 1.15 for propylene, rounded to 1.2).
(2) The isentropic expansion coefficient in this case was estimated to be the real gas specific heat ratio.

### Considerations

- Accepting that the exact method is the direct integration one, the required area (1055 $mm^2$) is higher than the original value calculated by the original company (859 $mm^2$). Therefore the valve is undersized
- As pointed out by Shackelford (2003), Kim et al. (2011), Singh (2011) and Smith and Burgess (2015), the use of the isentropic expansion factor matches the results of the more tedious direct integration method. In this case study, the agreement is very good: 1055 $mm^2$ vs. 1059 $mm^2$. This result contradicts the recommendation of Smith and Burgess (2015) of avoiding the isentropic expansion factor when the reduced volume is less than 2. In this case it was 1.33
- The use of the ideal gas specific heat ratio and a compressibility factor equal to 1 is likely to produce a conservative result. However, in no case should the real gas specific heat ratio be used as the isentropic expansion factor, because this can lead to significant underprediction of the required discharge area
- The use of the ideal gas specific heat ratio and the actual compressibility factor can underpredict the required area for conditions below the critical point, as already reported by Shackelford (2003). This case corresponds to the original engineering company calculation, which reported a required area of 859 $mm^2$, undersizing the pressure relief valve YS 414/1-2 for this scenario. However, the installed area is 1256

mm$^2$, higher than the 1055 mm$^2$ required. Also, no mitigation actions are required for this case.

## 6.3 Comparison of different methods for the design of pressure relief valves with two-phase flow

A case study will be performed for the pressure relief valve YS860/10 which protects a heat exchanger in case of tube rupture. The required two-phase relief load has been calculated in section 6.6. At the inlet of the tube side there is a two-phase mixture of hydrocarbons that is cooled from 41.9 ºC to 18 ºC. At the inlet of the shell side there is propylene liquid at -9 ºC which is evaporated keeping a constant level in the shell with a level control which acts on the control valve at the inlet. On the dome of the shell there is an extra volume (horizontal vessel) to avoid any entrainment of propylene liquid flowing to the compressor suction.

A scheme of the installation showing the position of YS 860/10 on the shell of the heat exchanger is presented in figure 6-3.

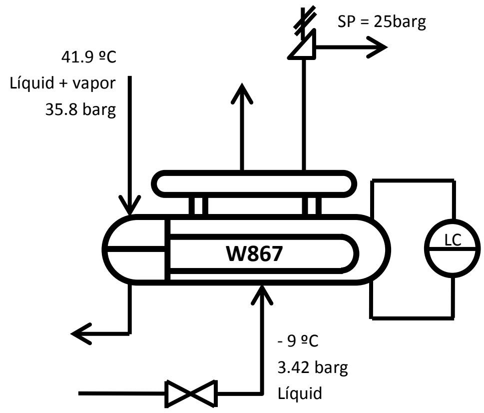



**Figure 6-3. Scheme of YS 860/10 installation.**

**Design basis**

All physical and thermodynamic properties are obtained through the Aspen Hysys v7.3 with Peng Robinson as EOS.

The relieving properties of the two phase hydrocarbon mixture at the inlet of the safety valve are:

Set pressure (SP): 25 barg; Relieving pressure ($P_R$): 25 · 1.1 + 1.013 = 28.513 bara; Total back pressure ($P_b$): 1 barg (estimated); $W_v$ = 6674 kg/h; $W_L$ = 19728 kg/h; $T_R$ = 33 ºC = 306.15 K; Quality (mass fraction of vapor): 0.2528; Entropy of the inlet mixture: 2.008 kJ/kgºC and k = 1.125 (ideal at 33 ºC).

According to the manufacturer of the valve, the model Leser 4564.6062 has a certified coefficient of discharge for gas of 0.8 and for liquid 0.54.

The composition at the inlet of the valve is:

| Element | Mass fraction | | |
|---|---|---|---|
| | Inlet mixture | Inlet vapor | Inlet liquid |
| Ethylene | 0.2155 | 0.3546 | 0.1684 |
| Ethane | 0.0314 | 0.0436 | 0.0273 |
| Propylene | 0.7084 | 0.5661 | 0.7565 |

| Element | Mass fraction | | |
|---|---|---|---|
| | Inlet mixture | Inlet vapor | Inlet liquid |
| Propane | 0.0418 | 0.0312 | 0.0454 |
| n-heptane | 0.0015 | 0.0001 | 0.0020 |
| Methane | 0.0003 | 0.0007 | 0.0001 |
| Hydrogen | 0.0011 | 0.0037 | 0.0002 |

**Calculation of the required area (API 520, part I, 1993)**

It was only in March 1993 that API published the first procedure for calculating the required area of a pressure relief valve for two-phase flow. API wrote “This is a reasonable and conservative method”. The method consists of adding the required areas of the vapor and the liquid at the exit of the valve but at the inlet conditions. In summary, the procedure is:

a) Determine the quantity of liquid that flashes by isenthalpic expansion from relieving condition to the critical downstream pressure for the flashed vapor or the back pressure, whichever is greater.

b) Calculate the orifice area to pass the flashed vapor.

c) Calculate the orifice area to pass the flashed liquid. The pressure drop is the inlet relieving pressure minus the backpressure. However, Wong (1992) wrote that it should be not the backpressure, but the critical pressure which is to be subtracted. Here the API criteria has been followed.

d) Add the individual areas.

Thus,

$$P_c = P_R\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} = 28.513\left(\frac{2}{1.125+1}\right)^{\frac{1.125}{1.125-1}} = 16.52\ bara$$

$P_b = 1 + 1.013 = 2.013\ bara < 16.52\ bara$ → critical flow

Performing an isenthalpic flash from 28.513 bara and 33 ºC to 16.52 bara:

$$\left.\begin{array}{c}28.513\ bara\\ 33\ ℃\end{array}\right\}\ isenthalpic\ flash\ \left\{\begin{array}{c}Vapor:\\ P = 16.52\ bara; T = 13.8\ ℃; Z = 0.804\\ quality = 0.3928;\ k_{ideal} = 1.125; MW = 33.75\,\frac{kg}{kmol}\\ Liquid:\\ P = 16.52\ bara; T = 13.8\ ℃;\ \rho = 497\ kg/m^3\end{array}\right.$$

Using the procedure provided in AD Merkblatt-A2 (2006) for single phases

Vapor phase

$W_v = 0.3928 \cdot 26400 = 10370$ kg/h

$$\psi_{max} = \sqrt{\frac{k}{k+1}\left(\frac{2}{k+1}\right)^{\frac{1}{k-1}}} = \sqrt{\frac{1.125}{2.125}\left(\frac{2}{2.125}\right)^{\frac{1}{0.125}}} = 0.448$$

$$A_0 = 0.1791\frac{q_m}{\psi\cdot\alpha_d\cdot P}\sqrt{\frac{TZ}{M}} = 0.1791\frac{10370}{0.448\cdot 0.8\cdot 28.513}\sqrt{\frac{(273.15+13.8)\cdot 0.804}{33.75}}$$

$$A_0 = 475\ mm^2$$

Liquid phase

$W_L$ = 0.6072 · 26400 = 16030 kg/h

$$A_0 = 0.6211\frac{q_m}{\alpha_{dF}\sqrt{\Delta P\cdot\rho}} = 0.6211\frac{16030}{0.54\cdot\sqrt{11.99\cdot 497}} = 239\ mm^2$$

$$A_{TOTAL} = 475\ mm^2 + 239\ mm^2 = 714\ mm^2$$

**Calculation of the required area (VdTÜV, 1973)**

According to the German publication VdTÜV- Merkblatt Sicherheitsventile 100/2 January 1973 edition, the same procedure applies but multiplied by a factor of 1.2

Thus,

$$A_{TOTAL} = 1.2\ \cdot\ 714 = 857\ mm^2$$

**Calculation of the required area (API 520, part I, 2008)**

Using the procedure described in paragraph C2.2 of API 520 (part I, 2008) known as "2 point-ω-method" the omega parameter is calculated by:

$$\omega = 9\left(\frac{\nu_9}{\nu_0} - 1\right)$$

Where $\nu_9$ is the specific volume evaluated at 90% of the PRV inlet pressure $P_0$ in $m^3$/kg and $\nu_0$ is the specific volume of the two phase system at the PRV inlet in $m^3$/kg.

Performing an isenthalpic flash with Aspen Hysys v7.3 with Peng Robisnson as EOS from 28.513 bara and 33 ºC to 25.66 bara, on get:

$\nu_9 = 0.007887\ m^3/kg$ and $\nu_0 = 0.006528\ m^3/kg$ thus,

$$\omega = 9\left(\frac{\nu_9}{\nu_0} - 1\right) = 9\left(\frac{0.007887}{0.006528} - 1\right) = 1.874$$

The critical pressure ratio $\eta_c$ is calculated by

$$\eta_c^2 + (\omega^2 - 2\omega)(1-\eta_c)^2 + 2\omega^2\ln\eta_c + 2\omega^2(1-\eta_c) = 0$$

$$\eta_c^2 + (1.874^2 - 2\cdot 1.874)(1-\eta_c)^2 + 21.874^2\ln\eta_c + 21.874^2(1-\eta_c) = 0$$

By trial and error

$$\eta_c = 0.685$$

Thus $P_c = \eta_c\cdot P_0 = 0.685\cdot 28.513 = 19.53\ bara$

$P_a = 1 + 1.013 = 2.013\ bara$

$19.53\ bara\ > 2.013\ bara$ → critical flow

Now, the mass flux and the required area can be calculated

$G = \eta_c \sqrt{\frac{P_0}{\nu_0 \omega}} = 0.685 \sqrt{\frac{2851300}{0.006528 \cdot 1.874}} = 10458\ kg/m^2 s$ and

$$A = \frac{277.8\ W}{K_d K_b K_c K_v G} = \frac{277.8 \cdot 26400}{0.85 \cdot 1 \cdot 1 \cdot 1 \cdot 10458} = 825\ mm^2$$

A = 825 mm$^2$

**Calculation of the required area (ISO 4126-10, 2010) with boiling delay**

The procedure of ISO 4126-10 will be followed using the same nomenclature as the original source. The critical pressure of the mixture is 56.53 bara and the critical temperature is 72.5 ºC, according to the results of Aspen Hysys v7.3 with Peng Robinson as EOS.

Application range of the method

$T_{red} = \frac{33+273.15}{72.5+273.15} = 0.89 < 0.9$ check

$P_{red} = \frac{28.513}{56.53} = 0.5\ < 0.5$ check (limit)

The $P_{red}$ is in the limit, however, as pointed out by Schmidt (Schmidt, Egan, 2009) the permissible range is wider as currently accepted by the omega models.

Calculation of the dischargeable mass flux through the safety valve

$$\omega_{eq} = \frac{\dot{x}_0 \cdot \nu_{g,0}}{k_0 \cdot \nu_0} + \frac{C_{pl,0} \cdot P_0 \cdot T_0}{\nu_0} \left( \frac{\nu_{g,0} - \nu_{l,0}}{\Delta h_{v,0}} \right)^2$$

Giving values using the results of previous isenthalpic flash simulation:

$\dot{x}_0 = 0.2528$; $\nu_{g,0}$ = 0.0192 m$^3$/kg; $k_0$ = 1.125 (ideal); $\nu_0$ = $\dot{x}_0 \cdot \nu_{g,0} + (1 - \dot{x}_0) \cdot \nu_{l,0} = 0.2528 \cdot$ 0.0192 + 0.7472 · 0.00224 = 0.00653 m$^3$/kg; $C_{pl,0}$ = 3372 J/kg K; $P_0$ = 2851300 Pa; $T_0$ = 306 K; $\nu_{l,0}$ = 0.002237 m$^3$/kg and $\Delta h_{v,0}$ = 446900 J/kg.

Thus,

$$\omega_{eq} = \frac{0.2528 \cdot 0.0192}{1.125 \cdot 0.00653} + \frac{3372 \cdot 2851300 \cdot 306}{0.00653} \left( \frac{0.0192 - 0.00224}{446900} \right)^2 = 0.661 + 0.649$$

$= 1.31$

Because $\omega_{eq} = 1.31$, the critical pressure ratio $\eta_{crit}$ has to be calculated by trial and error

$$0 = \eta_{crit}^2 + (\omega^2 - 2\omega) \cdot (1 - \eta_{crit})^2 + 2\omega^2 \ln \eta_{crit} + 2\omega^2 (1 - \eta_{crit})$$

$$0 = \eta_{crit}^2 + (1.31^2 - 2 \cdot 1.31) \cdot (1 - \eta_{crit})^2 + 2 \cdot 1.31^2 \ln \eta_{crit} + 2 \cdot 1.31^2 (1 - \eta_{crit})$$

$\eta_{crit} = 0.640$

$$\eta_b = \frac{P_b}{P_0} = \frac{1 + 1.013}{25 + 1.013} = 0.115$$

$\eta_b < \eta_{crit}$ therefore the flow is choked

The boiling delay factor is

$$N = \left[\dot{x}_0 + C_{pl,0} \cdot P_0 \cdot T_0 \frac{(\nu_{g,0} - \nu_{l,0})}{\Delta h_{\nu,0}^2} \ln\left(\frac{1}{\eta_{crit}}\right)\right]^{2/5}$$

$$N = \left[0.2528 + 3372 \cdot 2851300 \cdot 306 \frac{(0.0192 - 0.00224)}{446900^2} \ln\left(\frac{1}{0.640}\right)\right]^{2/5} = 0.6677$$

The compressibility coefficient is ω = 1.31 · 0.6677 = 0.875

To calculate the discharge coefficient the method needs the void fraction in the valve seat area $\varepsilon_{seat}$.

$$\varepsilon_{seat} = 1 - \frac{\nu_{l.0}}{\nu_0\left[\omega\left(\frac{1}{\eta_{crit}} - 1\right) + 1\right]} = 1 - \frac{0.002237}{0.00653\left[0.875\left(\frac{1}{0.64} - 1\right) + 1\right]} = 0.770$$

$$K_{dr,2ph} = K_{dr,g} \cdot \varepsilon_{seat} + (1 - \varepsilon_{seat}) \cdot K_{dr,l} = 0.8 \cdot 0.77 + 0.23 \cdot 0.54 = 0.740$$

The flow coefficient is

$$C = \frac{\sqrt{\omega \ln\left(\frac{1}{\eta_{crit}}\right) - (\omega - 1)(1 - \eta_{crit})}}{\omega\left(\frac{1}{\eta_{crit}} - 1\right) + 1} = \frac{\sqrt{0.875 \ln\left(\frac{1}{0.640}\right) - (0.875 - 1)(1 - 0.640)}}{0.875\left(\frac{1}{0.640} - 1\right) + 1} = 0.442$$

The mass flux is calculated by

$$\dot{m}_{SV} = K_{dr,2ph} \cdot C \cdot \sqrt{\frac{2 \cdot P_0}{\nu_0}} = 0.74 \cdot 0.442 \sqrt{\frac{2 \cdot 2851300}{0.00653}} = 9665.7 \frac{kg}{m^2 s}$$

The required area is

$$A_0 = \frac{Q_{m,out}}{\dot{m}_{SV}} = \frac{26400}{9665.7} \frac{1\,h}{3600\,s} = 760\,mm^2$$

**Calculation of the required area (ISO 4126-10, 2010) without boiling delay**

According to the ISO norm, the boiling delay factor N is recommended in case of saturated liquid or low quality mixtures. In this case the quality is 25.28 % and the liquid is saturated, but in reality it is a mixture in equilibrium. Also, the required area could be calculated with N = 1 as a conservative estimate as well

Thus, $\omega = 1.31$ and $\eta_{crit} = 0.640$

$$\varepsilon_{seat} = 1 - \frac{\nu_{l.0}}{\nu_0\left[\omega\left(\frac{1}{\eta_{crit}} - 1\right) + 1\right]} = 1 - \frac{0.002237}{0.00653\left[1.31\left(\frac{1}{0.64} - 1\right) + 1\right]} = 0.803$$

$$K_{dr,2ph} = K_{dr,g} \cdot \varepsilon_{seat} + (1 - \varepsilon_{seat}) \cdot K_{dr,l} = 0.8 \cdot 0.803 + 0.197 \cdot 0.54 = 0.749$$

$$C = \frac{\sqrt{\omega \ln\left(\frac{1}{\eta_{crit}}\right) - (\omega - 1)(1 - \eta_{crit})}}{\omega\left(\frac{1}{\eta_{crit}} - 1\right) + 1} = \frac{\sqrt{1.31 \ln\left(\frac{1}{0.640}\right) - (1.31 - 1)(1 - 0.640)}}{1.31\left(\frac{1}{0.640} - 1\right) + 1}$$

$$C = 0.396$$

$$\dot{m}_{SV} = K_{dr,2ph} \cdot C \cdot \sqrt{\frac{2 \cdot P_0}{\nu_0}} = 0.749 \cdot 0.396 \sqrt{\frac{2 \cdot 2851300}{0.00653}} = 8765.1\ \frac{kg}{m^2 s}$$

$$A_0 = \frac{Q_{m,out}}{\dot{m}_{SV}} = \frac{26400}{8765.1}\frac{1\ h}{3600\ s} = 837\ mm^2$$

**Calculation of the required area (Simpson Method)**

Simpson (1991) presented a generalized equation that requires an integration along an isentropic path from the inlet pressure toward the downstream pressure until a maximum mass velocity is found. Some simplifications are taken to facilitate the integration. The Simpson model with small variations is available on a compact disc that accompanies the CCPS Guideline book “Pressure relief and effluent handling systems” (CCPS, 1998). The program is TPHEM. This program has been converted to an excel spreadsheet because it was available for DOS operating systems only. The spreadsheet gave the following results working with the 2-point option model:

*Considering an isenthalpic flash*

Performing an isenthalpic flash with Aspen Hysys v7.3 from the relieving pressure to 80% of this pressure as recommended by CCPS: 0.80 · 28.513 = 22.810 bara with Peng Robinson as EOS:

$$\left.\begin{array}{r} P_R = 28.513\ bara \\ T_R = 33\ ^\circ C \\ \rho_V = 3.248\ \frac{lb}{ft^3} \\ \rho_L = 27.9\ \frac{lb}{ft^3} \\ Quality = 0.2528 \end{array}\right\} \text{Isenthalpic flash with } \Delta P = 5.703 \text{ bar} \left\{\begin{array}{l} P = 22.810\ bara \\ T = 25\ ^\circ C \\ \rho_V = 2.548\ \frac{lb}{ft^3} \\ \rho_L = 29.34\ \frac{lb}{ft^3} \\ Quality = 0.3192 \end{array}\right.$$

The discharge coefficient adopted is 0.84 as recommended by CCPS (1998) in a similar example.

Area required = 814 mm$^2$

*Considering an isentropic flash*

Again the two reference points would be the relieving pressure and 80% of this pressure but in this case considering an isentropic flash, performed with the same process simulator and the same parameters as before:

$$\left.\begin{array}{r} P_R = 28.513\ bara \\ T_R = 33\ ^\circ C \\ \rho_V = 3.248\ \frac{lb}{ft^3} \\ \rho_L = 27.9\ \frac{lb}{ft^3} \\ Quality = 0.2528 \end{array}\right\} \text{Isenthalpic flash with } \Delta P = 5.703 \text{ bar} \left\{\begin{array}{l} P = 22.810\ bara \\ T = 25\ ^\circ C \\ \rho_V = 2.539\ \frac{lb}{ft^3} \\ \rho_L = 29.36\ \frac{lb}{ft^3} \\ Quality = 0.3074 \end{array}\right.$$

Area required = 792 $mm^2$

**Calculation of the required area (Direct Integration method)**

The procedure explained in API 520 (part I, 2008) paragraph C.2.2 Annex C, will be used.

The equation to be solved is:

$$G^2 = \left[(\rho_t)^2 \cdot \left(-2\int_{P_0}^{P_t} \frac{dP}{\rho}\right)\right]_{max}$$

and

$$\int_{P_0}^{P_t} \frac{dP}{\rho} \cong \sum_{i=0}^{t} 2\frac{P_{i+1} - P_i}{\rho_{i+1} + \rho_i}$$

The isentropic flashes have been performed taking steps of 4% of the relieving pressure, i.e. 1.140 bar as it is used in the API example. Again Peng Robinson as EOS is used in the Aspen Hysys v7.3 process simulator

| Pressure Pa | Temperature ºC | Density kg/m$^3$ | Integrand m$^2$/s$^2$ | Summe m$^2$/s$^2$ | Mass flux kg/m$^2$s |
|---|---|---|---|---|---|
| 2851000 | 33.00 | 153.2 | 0 | 0 | 0 |
| 2737000 | 31.47 | 144.1 | -766.9 | -766.9 | 5643 |
| 2623000 | 29.87 | 135.3 | -816.0 | -1582.9 | 7613 |
| 2509000 | 28.21 | 126.8 | -869.9 | -2452.8 | 8881 |
| 2395000 | 26.48 | 118.6 | -929.1 | -3381.9 | 9754 |
| 2281000 | 24.66 | 110.7 | -994.3 | -4376.2 | 10356 |
| 2167000 | 22.76 | 103.1 | -1066.4 | -5442.6 | 10757 |
| 2053000 | 20.78 | 95.77 | -1146.5 | -6589.1 | 10994 |
| **1939000** | **18.69** | **88.70** | **-1235.9** | **-7825.1** | **11096** |
| 1825000 | 16.50 | 81.90 | -1336.4 | -9161.5 | 11086 |
| 1711000 | 14.19 | 75.34 | -1450.0 | -10611.5 | 10975 |
| 1597000 | 11.75 | 69.02 | -1579.4 | -12190.9 | 10777 |

The maximum mass flow (critical flow) occurs at 19.39 bara and 18.7 ºC

The relieving area is calculated with equation C.10 of API 520 (2014):

$$A = \frac{277.8 \cdot W}{K_d K_b K_c K_v G} = \frac{277.8 \cdot 26402}{0.8 \cdot 1 \cdot 1 \cdot 1 \cdot 11096} = 826\ mm^2$$

Note that the length between the inlet of the valve and the seating surface is 150 mm for a Leser model 4564.6062 valve (DN50xDN80). Because 150 mm > 100 mm, the thermodynamic equilibrium is reached and it is not necessary to apply the Homogeneous Non-equilibrium Direct Integration Method as proposed by Darby et al. (2002). Moreover, as stated by the same author (Darby, 2004) the recommended value for the discharge coefficient for two phase choked flow is the manufacturer value for the gas/vapor. In this case $K_d$ = 0.8.

**Summary of results for the required area**

In Table 6-8 the results of the required area for tube rupture scenario of YS 860/10 are presented:

**Table 6-8. Required area for YS 860/10 in case of tube rupture with two-phase flow.**

| Method | Required area mm$^2$ | $\frac{\textbf{Required area}}{\textbf{826 mm}^2}$ **x 100** |
|---|---|---|
| Installed area | 1256.6 | 152 |

| | | |
|---|---|---|
| API 520-1993 | 714 | 86 |
| VdTÜV-1973 | 857 | 104 |
| API 520-2008 (2 point-ω-method) | 825 | 100 |
| ISO 4126-10 (No boiling delay) | 837 | 101 |
| ISO 4126-10 (With boiling delay) | 760 | 92 |
| Simpson method (isenthalpic) | 814 | 99 |
| Simpson method (isentropic) | 792 | 96 |
| Direct integration | 826 | 100 |

**Considerations**

- The discharge coefficients for each model have been taken as recommended by the bibliographical source, i.e. no changes between models and discharge coefficients have been performed
- Accepting the value of the required area calculated by the Direct Integration method of 826 $mm^2$ as the most exact, it is concluded that the method of API 520-2008 (2 point-ω-method) gives the best results and requires only two points of the isentropic path
- Only the methods of VdTÜV-1973 and ISO 4126-10 (no boiling delay) overpredict the required area and are conservative, the other methods API 520-1993, ISO 4126-10 (with boiling delay), Simpson (isenthalpic) and Simpson (isentropic) underpredict the required area
- It seems that the API 520 (2008) and the ISO 4126-10 (No boiling delay) give the best results and in the case of ISO only the relieving data is required, i.e. no isenthalpic/isentropic calculations.
- As stated by some sources (API 520, ISO 4126-10, etc.), the isenthalpic calculation is more conservative than the isentropic
- In any case the safety valve has an oversizing of 152 %.That means a safety side design.

## 6.4 Relieving of supercritical fluids with retrograde condensation

In the three petrochemical plants studied here, there are some cases in which the relieving fluid is propylene at supercritical condition, which suffers condensation at the outlet of the valve. This case has been elected as a suitable case study due to its complexity. The pressure relief valve YS702/01 is a representative item of this phenomenon. Moreover, a comparison between different methods, both rigorous and simplified, will be performed. The theoretical framework of the process has been already explained in section 4-3.

**Design basis**

The pressure relief valve YS702-1 protects the shell of a heat exchanger which cools a propylene stream through cooling water in the tubes. There is no phase change in both sides of the heat exchanger. The scenario analyzed is external fire. Figure 6-4 gives a simplified diagram of the installation and Figure 6-5 is a picture of the heat exchanger and the pressure relief valve.

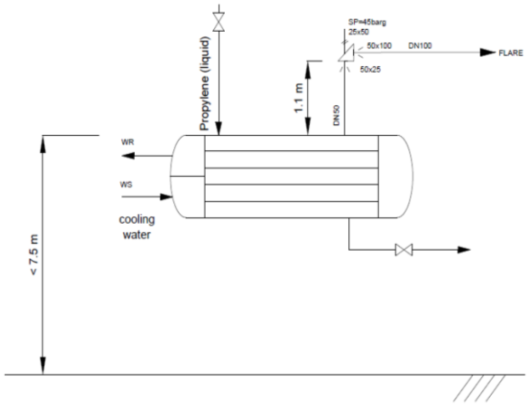



Figure 6-4. Scheme of YS702/01 installation


Figure 6-5. Picture of YS702/01 and equipment

From the specification sheet of YS702/01:

Set pressure: 45 barg; Relieving pressure: 1.1 x 45 + 1.013 = 50.513 bara = 732.6 psia; Overpressure: 10%; Insulation: No; Manufacturer/Model: LESER/4564.6052 without bellows; Area of the valve: 314.16 mm$^2$;$\alpha_{d,DG}$ = 0.8 certified and $\alpha_{d,F}$ = 0.6 certified.

Propylene critical pressure: 46.646 bara (source: http://webbook.nist.gov/chemistry/fluid/)

Propylene critical temperature: 92.42 ºC (source: http://webbook.nist.gov/chemistry/fluid/)

The shell side is full of propylene liquid at 16.2 barg and 42 ºC when the fire begins. As stated in API 521 (2014) the heat exchanger remains blocked during the fire contingency. The heat exchanger is installed at less than 7.6 m from the ground of the plant.

The process that the blocked propylene suffers when the fire starts is represented by the Mollier diagram showed in Figure 6-6.

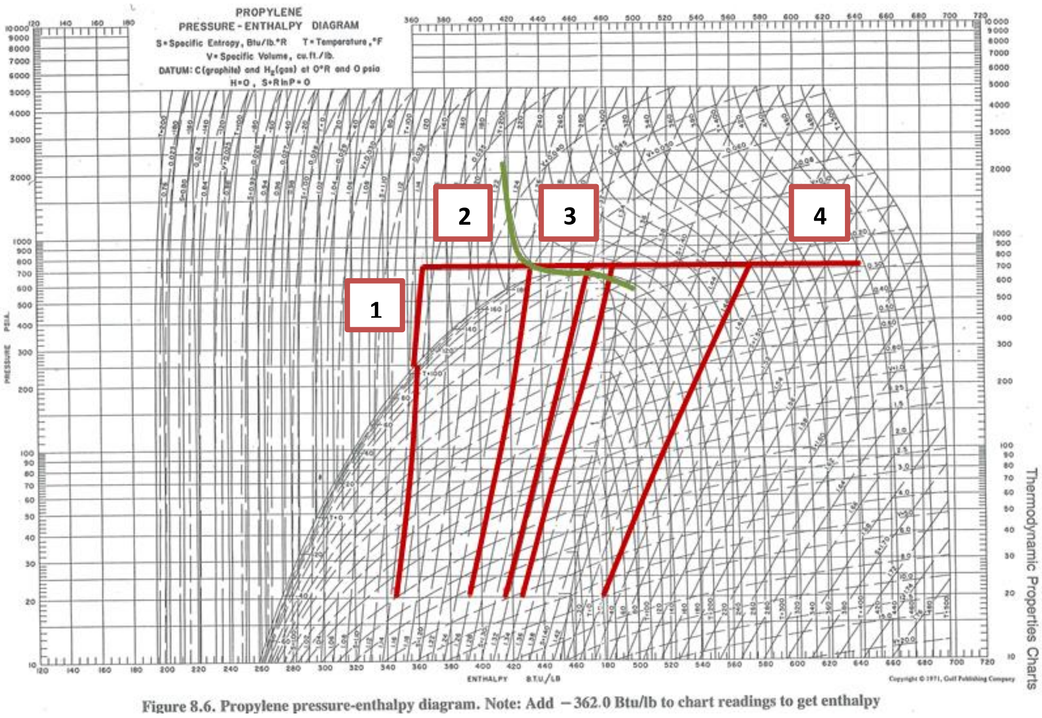



**Figure 6-6. Relieving path followed by YS702-1(supercritical fluid with retrograde condensation).**

Step 1. The liquid blocked in the shell of the heat exchanger at 16.2 barg and 42 ºC (the density of 472.67 kg/m$^3$ remains constant) has an isochoric transformation until it reaches the set pressure (45 barg) (curve 1 of the diagram). At this point, YS702/01 begins to open and depending on the relieving load the pressure increases up to the relieving pressure (50.5 barg), which corresponds to a relieving temperature of 45.5 ºC.

Step 2. The relieving of the subcooled liquid propylene through the valve suffers an isentropic transformation and flashing flow is produced at the outlet. (curve 2 of the diagram). The temperature of the liquid propylene increases until it reaches its critical temperature, i.e. 92.42 ºC and transforms to its gas phase without boiling.

Step 3. The relieving of the supercritical propylene gives two-phase flow at the outlet. (curve 3 of the diagram). This phenomenon is called retrograde condensation. The cricondentherm concept (Dole et al., 2014), is the maximum temperature above which liquid cannot be formed, regardless of pressure. In our case this is 132 ºC at a minimum discharge pressure of 1.4 bara (estimated).

Step 4. From 132 ºC no retrograde condensation occurs and the gas relieves until the vessel ruptures. (curve 4 of the diagram).

*Calculation of the wetted area*

Shell of the exchanger: $\pi DL = \pi \cdot 0.508 \cdot 3 = 4.79\ m^2$

Inlet piping to the exchanger, 25 m of pipe DN 80 (OD = 88.9 mm): $\pi DL = \pi \cdot 0.0889 \cdot 25 = 6.98\ m^2$

Outlet piping from the exchanger, 5 m of pipe DN 150 (OD = 168.3 mm): $\pi DL = \pi \cdot 0.1683 \cdot 5 = 2.64\ m^2$

Total wetted area = 4.79 + 6.98 + 2.64 = 14.41 m$^2$ (Note: In order to get more exact values of area, the exposed piping has been considered only in this case study)

Because of the characteristics of the installation, concerning the firefighting possibilities and drainage design, the following equation for the heat absorbed will be used, as explained in section 3.3

$$Q = 43200 \cdot F \cdot A_w^{0.82} = 43200 \cdot 1 \cdot 14.41^{0.82} = 385113\, W$$

Now, following Rahimi Mofrad and Norouzi (2007) a screening of the design basis (wetted or unwetted) is performed.

The liquid heating time until supercritical fluid is formed, is calculated according to:

$$\theta_h = \frac{\rho_L V_L CP_L (T_{critical} - T_0)}{Q}$$

where

$\rho_L$ liquid density; $V_L$ liquid volume; $CP_L$liquid specific heat capacity; $T_{critical}$ is 92.42 ºC in this case, $T_0$ operation temperature, Q total heat absorption across the wetted surface area.

The result, 2.0 minutes, is discussed at the end of this section.

Assuming an effective fire team response time of 20 minutes (section 4.3) from the beginning of the fire, 18 minutes is left for the exchanger wall being exposed to fire. As stated before, it is supposed that the relief requirement approaches zero after 20 minutes, because the surface of the vessel is cooled by the fire team. (Note: Only in this case study the 20 minutes fire duration concept has been applied).

**Step 1. Calculation of the relieving load**

According to the API 521 (paragraph 5.14.3, 2008)

$$q = \frac{\alpha_v \cdot \emptyset}{1000 \cdot d \cdot c}$$

First of all, the relieving temperature will be obtained through the Mollier Diagram for propylene:

$$\begin{Bmatrix} 16.2\, barg \\ 42\text{°C} \end{Bmatrix} \xrightarrow{\text{isochoric transformation}} \begin{Bmatrix} P = 50.513\, bara\ (relieving\ pressure) \\ T = 45.5\ \text{°C} \end{Bmatrix}$$

According to Yaws (1995)

$$\alpha_v\ liq\ at\ 25\text{°C} = 0.003503\ \frac{1}{\text{°C}}$$

$$\alpha_v\ at\ 45.5\ \text{°C} = 0.00107 \left(1 - \frac{273 + 45.5}{364.76}\right)^{-0.6975} = 0.0045\ \frac{1}{\text{°C}}$$

The specific gravity of propylene at 45.5 ºC = 0.481 (http://webbook.nist.gov/chemistry/fluid/) and $C_P$ propylene at 45.5 ºC = 0.6613 kcal/kg K = 2767 J/kg K

Giving values

$$q = \frac{\alpha_v \cdot \emptyset}{1000 \cdot d \cdot c} = \frac{0.0045 \cdot 385113}{1000 \cdot 0.481 \cdot 2767} \cdot \frac{3600\, s}{1\, h} = 4.69\ \frac{m^3}{h}$$

$$q = 4.69 \frac{m^3}{h} = 2256\ \frac{kg}{h}$$

Calculation of the required area

According to API 520 (table C.1, Annex C, part I, 2008) in case of a subcooled liquid the procedures C.2.1 or C.2.3 can be used.

*Using the procedure C.2.1 Direct Integration*

$$G^2 = \left[(\rho_t)^2 \cdot \left(-2\int_{P_0}^{P_t} \frac{dP}{\rho}\right)\right]_{max}$$

and

$$\int_{P_0}^{P_t} \frac{dP}{\rho} \cong \sum_{i=0}^{t} 2\frac{P_{i+1} - P_i}{\rho_{i+1} + \rho_i}$$

Data from isentropic flashes from 50.513 bara to 1.013 bara →ΔP = 49.5 bar is required

Taking steps of 4% as in the example in API 520: $49.5 \cdot 0.04 = 1.98\ bar = 198000\ Pa$

The isentropic flashes were obtained with Aspen Hysys v7.6 with Peng Robinson as EOS. The results were:

| **Pressure Pascals** | **Temperature ºC** | **Density kg/m$^3$** | **Entropy kJ/kmolºC** | **Mass quality** | **Integrand m$^2$/s$^2$** | **Summation m$^2$/s$^2$** | **Mass flux kg/sm$^2$** |
|---|---|---|---|---|---|---|---|
| 5051300 | 45.5 | 484.40 | 24.89 | 0 | 0 | 0 | 0 |
| 4853300 | 45.3 | 483.95 | 24.89 | 0 | -408.9 | -408.9 | 13840 |
| 4655300 | 45.0 | 483.50 | 24.89 | 0 | -409.3 | -818.2 | 19559 |
| 4457300 | 44.8 | 483.05 | 24.89 | 0 | -409.7 | -1227.9 | 23938 |
| 4259300 | 44.6 | 482.59 | 24.89 | 0 | -410.1 | -1638.0 | 27622 |
| 4061300 | 44.4 | 482.14 | 24.89 | 0 | -410.5 | -2048.5 | 30860 |
| 3863300 | 44.1 | 481.69 | 24.89 | 0 | -410.9 | -2459.4 | 33783 |
| 3665300 | 43.9 | 481.24 | 24.89 | 0 | -411.2 | -2870.6 | 36464 |
| 3467300 | 43.6 | 480.80 | 24.89 | 0 | -411.6 | -3282.2 | 38955 |
| 3269300 | 43.4 | 480.35 | 24.89 | 0 | -412.0 | -3694.2 | 41289 |
| 3071300 | 43.1 | 479.90 | 24.89 | 0 | -412.4 | -4106.6 | 43492 |
| 2873300 | 42.9 | 479.46 | 24.89 | 0 | -412.8 | -4519.4 | 45583 |
| 2675300 | 42.7 | 479.02 | 24.89 | 0 | -413.1 | -4932.6 | 47577 |
| 2477300 | 42.4 | 478.58 | 24.89 | 0 | -413.5 | -5346.1 | 49487 |
| 2279300 | 42.2 | 478.15 | 24.89 | 0 | -413.9 | -5760.0 | 51320 |
| 2081300 | 41.9 | 477.71 | 24.89 | 0 | -414.0 | -6174.0 | 53084 |
| **1883300** | **41.6** | **477.28** | **24.89** | **0** | **-414.7** | **-6588.7** | **54788** |
| 1685300 | 41.0 | 455.79 | 24.89 | 0.004 | -424.4 | -7013.1 | 53980 |
| 1487300 | 35.6 | 278.22 | 24.89 | 0.053 | -539.5 | -7552.6 | 34194 |
| 1289300 | 29.6 | 183.37 | 24.89 | 0.099 | -857.9 | -8410.5 | 23782 |
| 1091300 | 22.9 | 124.63 | 24.89 | 0.145 | | | |
| 893300 | 15.3 | 84.86 | 24.89 | 0.191 | | | |
| 695300 | 6.3 | 56.31 | 24.89 | 0.237 | | | |

The required area can be calculated according the equation C.9 of API 520 (2008)

$$A = \frac{277.8 \cdot W}{K_d \cdot K_b \cdot K_c \cdot K_v \cdot G}$$

where

A required effective discharge area, mm$^2$; W mass flow rate, kg/h; $K_d$= 0.65 because of subcooled liquid; $K_b$ = 1 no balanced valve; $K_c$ = 1 no rupture disc installed; $K_v$ = 1 viscosity correction factor.

Giving values

$$A = \frac{277.8 \cdot 2256}{0.65 \cdot 1 \cdot 1 \cdot 1 \cdot 54788} = 17.6\ mm^2$$

*Using the procedure C.2.3 (Omega method)*

In this case, the following mathematical expression will be used

$$w_s = 9\left(\frac{\rho_{lo}}{\rho_9} - 1\right)$$

where

$\rho_{lo}$ liquid density at the inlet of the valve, kg/m$^3$; $\rho_9$ density at 90% of the vapor pressure at the inlet relieving temperature.

Thus,

Vapor pressure at 45.5 ºC = 18.673 bara (saturated)

90% of 18.673 bara = 16.806 bara

Using Aspen Hysys V7.6, and Peng Robinson as EOS the following isentropic flash is performed

$$\left.\begin{matrix} 50.513\ bara \\ 45.5\ ^\circ\mathrm{C} \\ \rho = 484\ \dfrac{kg}{m^3} \end{matrix}\right\} \xrightarrow{isentropic\ flash} \left\{\begin{matrix} 16.806\ bara \\ \rho_9 = 453\ \dfrac{kg}{m^3} \end{matrix}\right.$$

Thus

$$w_s = 9\left(\frac{\rho_{lo}}{\rho_9} - 1\right) = 9\left(\frac{484}{453} - 1\right) = 0.616$$

The $n_{st}$ (transition saturation pressure ratio) is calculated

$$n_{st} = \frac{2w_s}{1 + 2w_s} = \frac{2 \cdot 0.616}{1 + 2 \cdot 0.616} = 0.552$$

Comparing $P_s$: $n_{st}P_0$

18.673 bara : 0.552 · 50.513 bara → 18.673 bara < 27.883 bara → the fluid is in the high subcooling region

Determination if the flow is critical/subcritical.

$P_s$ versus $P_a$ will be compared

$P_s$ = 18.673 bara

$P_a$ = 1.013 bara

$$\eta_a = \frac{P_a}{P_0} = \frac{1.013}{18.673} = 0.054 \rightarrow \text{Critical flow}$$

For the high subcooling region, the mass flux is (equation C.4.4 of API 520 (2008)):

$$G = 1.414\left(\rho_{lo}(P_0 - P_s)\right)^{0.5} = 1.414\left(484(5051300 - 1867300)\right)^{0.5} = 55508\ \frac{kg}{sm^2}$$

The required area is, according to equation C.46 of the same reference

$$A = 16.67\frac{Q\rho_{lo}}{K_d K_b K_v G}$$

where

Q = 4690 l/h = 78.2 l/min; $\rho_{lo}$= 484 kg/m$^3$;$K_d$ = 0.65 (subcooled liquid); $K_b$ = 1 (no bellows); $K_v$ = 1 (no viscosity liquid)

Giving values

$$A = 16.67\frac{78.2 \cdot 484}{0.65 \cdot 1 \cdot 1 \cdot 55508} = 17.5\ mm^2$$

*Using the procedure HNE-DS modified by Schmidt (2007)*

Schmidt (2007) has extended the original range of the application of the HNE-DS method including subcooled liquid. The same nomenclature will be followed here as in the original article.

The range of application of the method is $P_R \leq 0.5$ and $T_R \leq 0.9$. In this case:

$P_R = \frac{50.513}{46.65} = 1.08$ and $T_R = \frac{45.5+273.15}{92.42+273.15} = 0.87$

The condition for $P_R$ is not fulfilled. However, the procedure will be followed taking the results with caution.

Input data: $P_0$ = 50.513 bara; $T_0$ = 45.5 ºC = 318.65 K; q = 4.69 m$^3$/h = 2256 kg/h and $x_0$ = 0 (inlet mass flow quality)

Other conditions: $P_{sat}$ ($T_0$) = 18.6 bara; $\Delta H_{v0}$ at $P_{sat}$ = 290 kJ/kg; $Cp_{l0}$ at $P_0$ = 2979 J/kgºK; $\nu_{l0}$ at $P_{sat}$ = (465 kg/m$^3$)$^{-1}$ = 0.002148 m$^3$/kg; $\nu_{g0}$ at $P_{sat}$ = 0.02442 m$^3$/kg; $P_b$ = 1 bara. (Data from webbook.nist.gov)

Certified derated discharge coefficients: $K_{dg}$ = 0.8 and $K_{dl}$ = 0.6 (from Leser data sheet)

Calculation of the dischargeable mass flux through a safety valve:

$$\eta_s = \frac{P_{sat}(T_0)}{P_0} = \frac{18.6}{50.513} = 0.368$$

$$\eta_b = \frac{1.013}{50.513} = 0.02$$

Calculation of maximal flow coefficient and critical pressure ratio:

Iteration steps: 10

Interval = $\frac{1-n_b}{steps-1} = \frac{1-0.02}{10-1} = 0.10889$ and j = 0,1,…,9

$\eta_0$ = 1- interval·j = 1-0.10889·0 = 1; $\eta_1$ = 1-0.10889·1 = 0.8911; $\eta_2$ = 0.7822; $\eta_3$ = 0.6733…$\eta_9$ = 0.0199 and

$$N_j = \left(x_0 + Cp_{l0} \cdot P_0 \cdot \eta_s \cdot T_0 \frac{\nu_{g0} - \nu_{l0}}{\Delta h_{\nu 0}^2} \ln\left(\frac{\eta_s}{\eta_j}\right)\right)^{\eta_s^{-0.6}} \quad \text{for j = 0}$$

$$N_0 = \left(0 + 2979 \cdot 5051300 \cdot 0.368 \cdot 318.65 \frac{0.02442 - 0.002148}{290000^2} \ln\left(\frac{0.368}{\eta_0}\right)\right)^{0.368^{-0.6}} \quad \text{and}$$

$$\omega_j = \frac{1}{k}\frac{x_0 \cdot \nu_{g0}}{\nu_0} + \frac{Cp_{l0} \cdot T_0 \cdot P_0 \cdot \eta_s}{\nu_0}\left(\frac{\nu_{g0} - \nu_{l0}}{\Delta h_{\nu 0}}\right)^2 \cdot N_j$$

$$\omega_0 = 0 + \frac{2979 \cdot 318.65 \cdot 5051300 \cdot 0.368}{0.002148}\left(\frac{0.02442 - 0.002148}{290000}\right)^2 \cdot N_0 = 4.845 \cdot N_0 \quad \text{and}$$

$$C_j = \frac{\sqrt{(1 - \eta_s) + \left[\omega_j \cdot \eta_s \cdot \ln\left(\frac{\eta_s}{\eta_j}\right) - (\omega_j - 1)(\eta_s - \eta_j)\right]}}{\omega_j\left(\frac{\eta_s}{\eta_j} - 1\right) + 1}$$

$$C_0 = \frac{\sqrt{(1 - 0.368) + \left[\omega_0 \cdot 0.368 \cdot \ln\left(\frac{0.368}{\eta_0}\right) - (\omega_0 - 1)(0.368 - \eta_0)\right]}}{\omega_0\left(\frac{0.368}{\eta_0} - 1\right) + 1}$$

Thus

| **η** | **$N_j$** | **$\omega_j$** | **$C_j$** |
|---|---|---|---|
| $\eta_0 = 1$ | -- | -- | -- |
| $\eta_1 = 0.8911$ | -- | -- | -- |
| $\eta_2 = 0.7822$ | -- | -- | -- |
| $\eta_3 = 0.6733$ | -- | -- | -- |
| $\eta_4 = 0.5644$ | -- | -- | -- |
| $\eta_5 = 0.4555$ | -- | -- | -- |
| $\eta_6 = 0.3466$ | 0.0015 | 0.0073 | 0.808 |
| 0.34 | 0.00246 | 0.01192 | 0.812 |
| 0.33 | 0.00441 | 0.02136 | 0.817 |
| 0.32 | 0.00694 | 0.03361 | 0.821 |
| 0.31 | 0.01012 | 0.04904 | 0.823 |
| **0.30** | **0.01385** | **0.06711** | **0.824** |
| 0.29 | 0.01832 | 0.08877 | 0.823 |
| 0.28 | 0.02353 | 0.11403 | 0.820 |
| $\eta_7 = 0.2377$ | 0.0554 | 0.2684 | 0.765 |
| $\eta_8 = 0.1288$ | 0.2732 | 1.3236 | 0.298 |
| $\eta_9 = 0.0199$ | 0.1.7587 | 8.5209 | 0.018 |

According to the results the maximum value of $C_j$ is 0.824, and $\eta_{max} = 0.30$ pressure ratio; $N_{max} = 0.01385$ boiling delay factor; $\omega_{max} = 0.06711$ compressibility coefficient.

The two-phase discharge coefficient

$$\varepsilon = 1 - \frac{\nu_{l0}}{\nu_0\left[\omega\left(\frac{1}{\eta} - 1\right) + 1\right]} = 1 - \frac{0.002148}{0.002148\left[0.06711\left(\frac{1}{0.30} - 1\right) + 1\right]} = 0.13$$

And the derated two-phase discharge coefficient of the safety valve is:

$$K_{d2ph} = K_{dg} \cdot \varepsilon + (1 - \varepsilon) \cdot K_{dl} = 0.8 \cdot 0.13 + 0.87 \cdot 0.6 = 0.626$$

$$m_{sv} = K_{d2ph} \cdot C \cdot \sqrt{\frac{2P_0}{v_0}} = 0.626 \cdot 0.824 \cdot \sqrt{\frac{2 \cdot 5051300}{0.002148}} = 35375\ \frac{kg}{m^2 s}$$

And finally, the minimum cross sectional area is:

$$A_{SV} = \frac{Q_m}{m_{sv}} = \frac{2256}{35375} \cdot \frac{1\ h}{3600\ s} = 0.0000177\ m^2 = 17.7\ mm^2$$

**Step 2. Calculation of the relieving load**

At the critical temperature point, the relieving conditions are:

$$\left.\begin{matrix} Inlet\ of\ the\ PRV \\ 50.513\ bara \\ 92.42\ ℃ \end{matrix}\right\} \xrightarrow{isentropic\ transformation} \left\{\begin{matrix} Outlet\ of\ the\ PRV \\ 1.013\ bara\ (estimated) \end{matrix}\right.$$

Following the same procedure as in step 1

According to the Yaws (1995), the expansion coefficient $\alpha_v$ will be calculated by:

$$\alpha_v\ at\ T(℃) = 0.00107\left(1 - \frac{273 + T}{364.76}\right)^{-0.6975}$$

The experimental value at 25 ºC according the paper is

$$\alpha_v\ at\ 25\ ℃ = 0.003503\ \frac{1}{℃}$$

Because the maximal value that can be used in the equation presented by Yaws for the propylene is 346,52 K = 73.37 ºC too far from the critical temperature, a value of 90 ºC will be used as an approximation

$$\alpha_v\ at\ 90\ ºC = 0.00107\left(1 - \frac{273 + 90}{364.76}\right)^{-0.6975} = 0.044\ \frac{1}{℃}$$

Using the NIST webbook (http://webbook.nist.gov/chemistry/fluid/):

Specific gravity propylene at 92.42 ºC = 0.327; $C_p$ propylene at 92.42 ºC = 1.7009 kcal/kg K = 7116 J/kg K

Giving values

$$q = \frac{\alpha_v \cdot \emptyset}{1000 \cdot d \cdot c} = \frac{0.044 \cdot 385113}{1000 \cdot 0.327 \cdot 7116} \cdot \frac{3600\ s}{1\ h} = 26.2\ \frac{m^3}{h}$$

$$q = 26.2 \frac{m^3}{h} = 8572\ \frac{kg}{h}$$

Calculation of the required area

According to API 520 (table C.1, Annex C, 2008) in case of saturated liquid the procedures C.2.1 or C.2.3 can be used.

*Using the procedure C.2.1 Direct Integration*

Performing an isentropic flash like in step 1, and using the same decrements in pressure on get:

| Pressure | Temperature | Density | Entropy | Mass | Integrand | Summation | Mass flux |
|---|---|---|---|---|---|---|---|

| Pascals | ºC | kg/m$^3$ | kJ/kmolºC | quality | m$^2$/s$^2$ | m$^2$/s$^2$ | kg/s m$^2$ |
|---|---|---|---|---|---|---|---|
| 5051300 | 92.4 | 297.3 | 47.53 | 0 | 0 | 0 | 0 |
| 4853300 | 91.3 | 293.6 | 47.53 | 0 | -670.2 | -670.2 | 10749 |
| 4655300 | 90.1 | 291.8 | 47.53 | 0 | -676.5 | -1346.7 | 15144 |
| 4457300 | 88.8 | 291.5 | 47.53 | 0 | -678.9 | -2025.6 | 18554 |
| 4259300 | 87.2 | 286.4 | 47.53 | 0.035 | -685.2 | -2710.8 | 21088 |
| **4061300** | **84.6** | **257.7** | **47.53** | **0.16** | **-727.8** | **-3438.6** | **21370** |
| 3863300 | 81.8 | 231.5 | 47.53 | 0.24 | -809.5 | -4248.0 | 21338 |
| 3665300 | 78.9 | 207.7 | 47.53 | 0.30 | -901.6 | -5149.6 | 21078 |
| 3467300 | 76.0 | 186.2 | 47.53 | 0.34 | -1005.3 | -6154.9 | 20659 |
| 3269300 | 72.9 | 166.6 | 47.53 | 0.38 | | | |
| 3071300 | 69.6 | 148.7 | 47.53 | 0.41 | | | |
| 2873300 | 66.2 | 132.7 | 47.53 | 0.43 | | | |
| 2675300 | 62.6 | 118.2 | 47.53 | 0.46 | | | |
| 2477300 | 58.8 | 105.0 | 47.53 | 0.48 | | | |
| 2279300 | 54.8 | 92.8 | 47.53 | 0.50 | | | |
| 2081300 | 50.5 | 81.7 | 47.53 | 0.52 | | | |
| 1883300 | 45.9 | 71.3 | 47.53 | 0.54 | | | |
| 1685300 | 41.0 | 61.7 | 47.53 | 0.56 | | | |
| 1487300 | 35.6 | 52.8 | 47.53 | 0.575 | | | |
| 1289300 | 29.6 | 44.4 | 47.53 | 0.59 | | | |
| 1091300 | 22.9 | 36.6 | 47.53 | 0.61 | | | |
| 893300 | 15.3 | 29.3 | 47.53 | 0.62 | | | |
| 695300 | 6.3 | 22.3 | 47.53 | 0.64 | | | |

The required relieving area can be calculated according to equation C.9 of API 520 (2008)

$$A = \frac{277.8 \cdot W}{K_d \cdot K_b \cdot K_c \cdot K_v \cdot G}$$

where

A required effective discharge area, mm$^2$; W mass flow rate, kg/h; $K_d$ = 0.85 because of saturated liquid; $K_b$ = 1 no balanced valve; $K_c$ = 1 no rupture disc installed; $K_v$ = 1 viscosity correction factor.

Giving values

$$A = \frac{277.8 \cdot 8752}{0.85 \cdot 1 \cdot 1 \cdot 1 \cdot 21370} = 133.8\ mm^2$$

It is questionable if $K_d$ is 0.85 because of saturated liquid, is as a matter of fact a supercritical fluid and the proposed value of 0.975 could be used according to Annex B of API 521-2008 and is the value used by Ouderkirk (2002). However, as a conservative basis the value of 0.85, has been adopted in this work.

**Step 3. Calculation of the relieving load**

From 92,42 ºC till 132 ºC there is retrograde condensation. As stated in section 4.3 the Ouderkirk method will be followed.

*Relieving load calculation and required area according to Ouderkirk method*

The first part of the method deals with the search for the maximal volumetric flow and mass flow relieved at the relieving pressure of 50.513 bara.

The following design basis will be followed:

a) Increase of temperature will be ΔT = 5 ºC from the critical temperature.

b) Between the maximal volumetric flow and the maximal mass flow, the ΔT will be decreased to 1ºC
c) The thermodynamic properties are obtained from Aspen-Hysys v7.6 and Peng Robinson as EOS
d) The volumetric flow rate is calculated $\dot{V}\left(\frac{m^3}{s}\right) = \dot{Q}(kW)\frac{\Delta V}{\Delta H} = 385.113\frac{\Delta V\left(\frac{m^3}{kg}\right)}{\Delta H\left(\frac{kJ}{kg}\right)}$
e) The mass flow rate is calculated $\dot{m}\left(\frac{kg}{s}\right) = \frac{\dot{V}\left(\frac{m^3}{s}\right)}{V\left(\frac{m^3}{kg}\right)}$

The results are presented in the following table:

| **Pressure bara** | **Temperature ºC** | **Entropy kJ/kgºK** | **Enthalpy kJ/kg** | **V $m^3$/kg** | **$\dot{V}$ $m^3$/s** | **$\dot{m}$ kg/s** |
|---|---|---|---|---|---|---|
| 50.513 | 92.42 | 1.8549 | 485.10 | 0.0030548 | 0.008701 | 2.848 |
| 50.513 | 97.42 | 2.0580 | 559.96 | 0.0047462 | 0.013894 | **2.927** |
| 50.513 | 102.42 | 2.2122 | 617.43 | 0.0068196 | **0.014466** | 2.121 |
| 50.513 | 107.42 | 2.2835 | 644.38 | 0.0078319 | 0.014010 | 1.789 |
| 50.513 | 112.42 | 2.3379 | 665.20 | 0.0085893 | 0.013502 | 1.572 |
| 50.513 | 117.42 | 2.3845 | 683.28 | 0.0092232 | 0.013049 | 1.414 |
| 50.513 | 122.42 | 2.4265 | 699.76 | 0.0097816 | 0.012614 | 1.289 |
| 50.513 | 127.42 | 2.4653 | 715.22 | 0.010288 | 0.012227 | 1.195 |
| 50.513 | 132.42 | 2.5019 | 729.96 | 0.010756 | 0.011870 | 1.103 |
| 50.513 | 137.42 | 2.5367 | 744.17 | 0.011194 | 0.011564 | 1.033 |
| 50.513 | 142.42 | 2.5701 | 757.99 | 0.011609 | 0.011251 | 0.969 |
| 50.513 | 147.42 | 2.6025 | 771.51 | 0.012004 | --- | --- |

From the table the temperature for the maximal mass flow is 97.42 ºC and the temperature for the maximal volumetric flow is 102.42 ºC. New calculations will be made to get more precision. The ΔT is 1ºC

| **Pressure bara** | **Temperature ºC** | **Entropy kJ/kgºK** | **Enthalpy kJ/kg** | **V $m^3$/kg** | **$\dot{V}$ $m^3$/s** | **$\dot{m}$ kg/s** |
|---|---|---|---|---|---|---|
| 50.513 | 94.42 | 1.9018 | 502.27 | 0.0033138 | 0.007274 | 2.195 |
| 50.513 | 95.42 | 1.9365 | 515.04 | 0.003555 | 0.008960 | 2.520 |
| 50.513 | 96.42 | 1.9893 | 534.54 | 0.0040087 | 0.011173 | **2.787** |
| 50.513 | 97.42 | 2.0580 | 559.96 | 0.0047462 | 0.013040 | 2.747 |
| 50.513 | 98.42 | 2.1103 | 579.37 | 0.0054034 | 0.013996 | 2.590 |
| 50.513 | 99.42 | 2.1454 | 592.44 | 0.0058784 | 0.014394 | 2.4486 |
| 50.513 | 100.42 | 2.1719 | 602.31 | 0.0062473 | 0.014554 | 2.3296 |
| 50.513 | 101.42 | 2.1936 | 610.42 | 0.0065538 | **0.014602** | 2.228 |
| 50.513 | 102.42 | 2.2122 | 617.43 | 0.0068196 | 0.014586 | 2.139 |
| 50.513 | 103.42 | 2.2289 | 623.69 | 0.0070567 | --- | --- |

The results show that the maximal mass flow happens at 96.42 ºC and the maximal volumetric flow happens at 101.42 ºC.

The second part of the method deals with the calculation of the required area. The following design basis will be followed:

a) The maximal area would be between 96.42 ºC and 101.42 ºC.
b) Isentropic flashes with Aspen Hysys v7.6 with Peng Robinson as EOS will be used.
c) For the sake of simplicity only 4 points will be selected in the interval, with a ΔT of 2 ºC. Thus the starting points of the isentropic path are:

| **Point** | **Pressure bara** | **Temperature ºC** | **ΔP interval bar** | **Entropy kJ/kg ºC** | **Enthalpy, $H_0$ kJ/kg** |
|---|---|---|---|---|---|
| 1 | 50.513 | 96.42 | 1.98 | 1.259 | 400.5 |

| | | | | | |
|---|---|---|---|---|---|
| 2 | 50.513 | 98.42 | 1.98 | 1.355 | 436.4 |
| 3 | 50.513 | 100.42 | 1.98 | 1.409 | 456.3 |
| | 50.513 | 102.42 | 1.98 | 1.445 | 469.9 |

d) According to Ouderkirk (2002), the mass flux G, will be calculated by:

$$G = \frac{\sqrt{2(H_0 - H_b)\ \cdot 101.96 \cdot 9.81}}{V_b}$$

where

G is the mass flux, kg/m$^2$s; $H_0$ is the enthalpy at initial point, kJ/kg; $H_b$ is the enthalpy at each temperature selected on the isentropic path, kJ/kg; $V_b$ is the specific volume at each point, m$^3$/kg

Note that 1kJ = 101.96 kp·m and $g_c = 9.81 \frac{kg \cdot m}{kp \cdot s^2}$

Point 1. Isentropic flash from 50.513 bara and 96.42ºC

| **$T_b$ ºC** | **$P_b$ bara** | **$V_b$ m$^3$/kg** | **$H_b$ kJ/kg** | **G kg/m$^2$s** |
|---|---|---|---|---|
| 96.42 | 50.513 | 0.004340 | $H_0$ = 400.5 | --- |
| 94.41 | 48.533 | 0.004442 | 399.7 | 9001 |
| 92.24 | 46.553 | 0.004558 | 398.8 | 12788 |
| 89.80 | 44.573 | 0.004854 | 397.8 | 15133 |
| 87.23 | 42.593 | 0.005252 | 396.8 | 16373 |
| 84.57 | 40.613 | 0.005701 | 395.7 | 17179 |
| 81.81 | 38.633 | 0.006203 | 394.5 | 17653 |
| **78.95** | **36.653** | **0.006766** | **393.2** | **17851** |
| 75.97 | 34.673 | 0.007396 | 391.8 | 17828 |
| 72.86 | 32.693 | 0.008110 | 390.3 | 17167 |

Point 2. Isentropic flash from 50.513 bara and 98.42ºC

| **$T_b$ ºC** | **$P_b$ bara** | **$V_b$ m$^3$/kg** | **$H_b$ kJ/kg** | **G kg/m$^2$s** |
|---|---|---|---|---|
| 98.42 | 50.513 | 0.005455 | $H_0$ = 436.4 | --- |
| 95.74 | 48.533 | 0.005646 | 435.3 | 8304 |
| 92.85 | 46.553 | 0.005858 | 434.1 | 11573 |
| 89.80 | 44.573 | 0.006120 | 432.9 | 13665 |
| 87.23 | 42.593 | 0.006566 | 431.7 | 14760 |
| 84.57 | 40.613 | 0.007062 | 430.3 | 15634 |
| 81.81 | 38.633 | 0.007616 | 428.9 | 16075 |
| 78.95 | 36.653 | 0.008230 | 427.3 | 16385 |
| **75.97** | **34.673** | **0.008913** | **425.6** | **16482** |
| 72.86 | 32.693 | 0.009690 | 423.7 | 16441 |
| 69.61 | 30.713 | 0.010560 | 421.7 | 16230 |

Point 3. Isentropic flash from 50.513 bara and 100.42ºC

| **$T_b$ ºC** | **$P_b$ bara** | **$V_b$ m$^3$/kg** | **$H_b$ kJ/kg** | **G kg/m$^2$s** |
|---|---|---|---|---|
| 100.42 | 50.513 | 0.006184 | $H_0$ = 456.3 | --- |
| 97.54 | 48.533 | 0.006427 | 455.0 | 7930 |
| 94.48 | 46.553 | 0.006698 | 453.7 | 10762 |
| 91.21 | 44.573 | 0.007003 | 452.4 | 12606 |
| 87.81 | 42.593 | 0.007353 | 451.0 | 13996 |
| 84.57 | 40.613 | 0.007819 | 449.5 | 14909 |
| 81.81 | 38.633 | 0.008396 | 447.9 | 15431 |
| 78.95 | 36.653 | 0.009033 | 446.1 | 15806 |
| 75.97 | 34.673 | 0.009756 | 444.3 | 15873 |
| **72.86** | **32.693** | **0.010557** | **442.2** | **15900** |
| 69.62 | 30.713 | 0.011466 | 440.1 | 15692 |

| $T_b$ ºC | $P_b$ bara | $V_b$ $m^3/kg$ | $H_b$ kJ/kg | G $kg/m^2s$ |
|---|---|---|---|---|
| 66.21 | 28.733 | 0.012497 | 437.7 | 15427 |

Point 4. Isentropic flash from 50.513 bara and 102.42ºC

| $T_b$ ºC | $P_b$ bara | $V_b$ $m^3/kg$ | $H_b$ kJ/kg | G $kg/m^2s$ |
|---|---|---|---|---|
| 102.42 | 50.513 | 0.006702 | 469.9 | --- |
| 99.50 | 48.533 | 0.0069735 | 468.5 | 7585 |
| 96.40 | 46.553 | 0.007283 | 467.1 | 10271 |
| 93.12 | 44.573 | 0.007628 | 465.7 | 12010 |
| 89.65 | 42.593 | 0.008013 | 464.1 | 13435 |
| 85.99 | 40.613 | 0.008453 | 462.5 | 14386 |
| 82.11 | 38.633 | 0.008952 | 460.8 | 15064 |
| 78.95 | 36.653 | 0.009588 | 458.9 | 15463 |
| 75.97 | 34.673 | 0.010323 | 457.0 | 15554 |
| **72.86** | **32.693** | **0.011151** | **454.8** | **15578** |
| 69.61 | 30.713 | 0.012084 | 452.5 | 15431 |
| 66.21 | 28.733 | 0.013142 | 450.0 | 15174 |

For the 4 points with the higher mass flux, the required area will be calculated according to:

$$A = \frac{\dot{m}}{G_{chocked} \cdot K_b \cdot K_c \cdot K_d \cdot K_v} \cdot 10^6$$

where

A required area $mm^2$; $\dot{m}$ mass flow, kg/s; $G_{chocked}$is the choked mass flux, $kg/m^2s$; $K_b$ = 1 assuming a backpressure < 10% of the set pressure; $K_c$ = 1 no rupture disc; $K_v$ = 1 no viscosity correction; $K_d$ = 0.8 (from LESER manufacturer, assuming vapor)

Finally the results are:

| Pressure bara | Temperature ºC | Entropy kJ/kgºK | Enthalpy kJ/kg | V $m^3/kg$ | $\dot{V}$ m3/s | $\dot{m}$ kg/s | $(G)_{max}$ $kg/m^2s$ | A $mm^2$ |
|---|---|---|---|---|---|---|---|---|
| 50.513 | 96.42 | 1.9893 | 534.54 | 0.0040087 | 0.011173 | 2.787 | 17851 | 195 |
| **50.513** | **98.42** | **2.1103** | **579.37** | **0.0054034** | **0.013996** | **2.590** | **16482** | **196** |
| 50.513 | 100.42 | 2.1719 | 602.31 | 0.0062473 | 0.014554 | 2.3296 | 15900 | 183 |
| 50.513 | 102.42 | 2.2122 | 617.43 | 0.0068196 | 0.014586 | 2.139 | 15578 | 172 |

The results from the Ouderkirk methods are:

Relieving pressure: 50.513 bara; Relieving temperature: 98.4 ºC; Choked temperature: 76 ºC; Choked mass flux: 16482 $kg/m^2s$ and

Area required: 196 $mm^2$

The graphic representation of the Ouderkirk results is showed in Fig 6-7, where it can be appreciated that the maximum required orifice area stands between the maximum mass relief rate and the maximum volumetric relief rate, according the case studies of other authors (Nezami and Price, 2012; Self and Do, 2010)

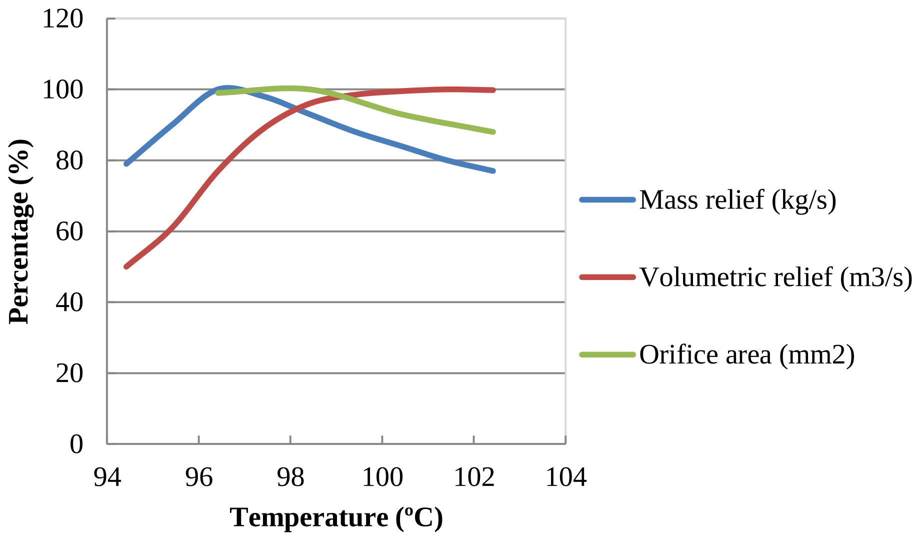



**Figure 6-7. Graphical representation of Ouderkirk (2002) results for YS702/01**

*Calculation of the time elapsed since the onset of the fire*

The volume of propylene is

Heat exchanger shell: $\frac{\pi D^2}{4} L - 120 \frac{\pi d^2}{4} L = \frac{\pi 0.5^2}{4} 3 - 120 \frac{\pi 0.0254^2}{4} 3 = 0.407\ m^3$ where 120 is the number of tubes.

Annexed piping: $\frac{\pi D^2}{4} L_{inlet} + \frac{\pi D^2}{4} L_{outlet} = \frac{\pi 0.08^2}{4} 25 + \frac{\pi 0.16^2}{4} 5 = 0.226\ m^3$

Total volume of propylene = 0.633 $m^3$

Heat absorbed = 385113 W = 331143 kcal/h

*1st Step: Liquid phase*

$\Delta T$ = 92.42 ºC – 42 ºC = 50.42 ºC

$$\bar{T} = \frac{92.42 + 42}{2} = 67.2\ ^{\circ}\mathrm{C}$$

$C_P$ at 67.2 ºC = 0.86 kcal/kg ºC (API, Technical Data Book)

ρ at 67.2 ºC = 410 kg/$m^3$ (API, Technical Data Book)

m = 0.633 $m^3$ · 410 kg/$m^3$ = 260 kg of propylene

$$t = \frac{mC_p\Delta T}{Q} = \frac{260 \cdot 0.86 \cdot 50.42}{331143} = 0.0340\ h = 2.04\ min.$$

*2nd Step: Supercritical phase*

According to Ouderkirk article, the equations to be used are

$$\bar{V} = \frac{V_{n+1} + V_n}{2}\ ; \quad m = \frac{0.633 m^3}{\bar{V}}\ ; \quad t_{n+1} = t_n + m \frac{\overline{\Delta H}}{60 \cdot 385.113}\ min$$

Thus,

| t<br>min | T<br>ºC | H<br>kJ/kg | V<br>$m^3/kg$ | M<br>kg |
|---|---|---|---|---|
| 0.0 | 92.42 | 485.10 | 0.0030548 | --- |
| 0.52 | 97.42 | 559.96 | 0.0047462 | 162 |
| 0.79 | 102.42 | 617.43 | 0.0068196 | 109 |
| 0.89 | 107.42 | 644.38 | 0.0078319 | 86 |
| 0.96 | 112.42 | 665.20 | 0.0085893 | 77 |
| 1.02 | 117.42 | 683.28 | 0.0092232 | 71 |
| 1.07 | 122.42 | 699.76 | 0.0097816 | 67 |
| 1.11 | 127.42 | 715.22 | 0.010288 | 63 |
| 1.15 | 132.42 | 729.96 | 0.010756 | 60 |
| 1.19 | 137.42 | 744.17 | 0.011194 | 58 |
| 1.22 | 142.42 | 757.99 | 0.011609 | 55 |
| 1.25 | 147.42 | 771.51 | 0.012004 | 54 |

As a conclusion the time required for the propylene to increase in temperature from 42 ºC to 147 ºC is 2.04 + 1.24 = 3.28 minutes

*Comparison of Ouderkirk results with the method of gas expansion of API 521 (equation 12, 2008)*

$$q_{m,relief} = 0.1406\sqrt{M \cdot P_1} \cdot \frac{(T_w - T_1)^{1.25}}{T_1^{1.1506}} A$$

where

M = Molecular weight of propylene; $P_1$ = 50.513 bara = 732.6 psia; $T_w$ = 1560 ºR for carbon steel; $T_1$ = 98.4 ºC = 669 ºR and A = 14.4 $m^2$ = 155 $ft^2$.

Giving values

$$q_{m,relief} = 0.1406\sqrt{42.08 \cdot 732.6} \cdot \frac{(1560 - 669)^{1.25}}{669^{1.1506}} \cdot 155 = 10452\frac{lb}{h} = 4741\frac{kg}{h}$$

Now, using the Valvestar software from Leser with $\frac{C_p}{C_v} = 1.14$, the required area is 139 $mm^2$

The result shows that the API equation is not always conservative. This result contradicts the results of the example developed by Doane (2010) where the author found that the difference between the required area calculated by the above equation is almost 3 times greater than with his rigorous model.

Self and Do (2010) presented a model for supercritical fluids, that can be used for all the steps mentioned before, if there is no phase change. This method has been adopted in the latest edition of API 520 (part I, 2015) and presumably gives the most exact values. A process simulator is required. The results are presented in table 6-8 but the calculations are in annex D.

### Step 4. Calculation of the relieving load

In step 4, the temperature is higher than 132 ºC and there is no retrograde condensation. No representative case.

The results are summarized in the Table 6-8.

**Table 6-8. Results of different methods for rigorous calculation of the required area for YS702/01.**

| Case | $P_r$<br>bara | $T_r$<br>ºC | Relieving load<br>kg/h | Required area<br>$mm^2$ |
|---|---|---|---|---|
| Original design (engineering company) | 50.513 | 100 | 10000 | 220.6 |

| Case | $P_r$ bara | $T_r$ ºC | Relieving load kg/h | Required area $mm^2$ |
|---|---|---|---|---|
| | | | | (installed: 314.16) |
| Taking λ = 115 kJ/kg (API 521-2008) | 50.513 | 100 | 12055 | 266 |
| Step 1 (liquid thermal expansion) + Direct Integration (DI) | 50.513 | 45.5 | 2256 | 17.6 |
| Step 1 (liquid thermal expansion) + Leung - Omega | 50.513 | 45.5 | 2256 | 17.5 |
| Step 1 (liquid thermal expansion) HNE-DS revised for subcooling (Schmidt, 2007) | 50.513 | 45.5 | 2256 | 17.7 |
| Step 2 Critical temperature Supercritical fluid expansión + DI | 50.513 | 92.42 | 8752 | 133.8 |
| Step 3 Supercritical fluid expansion + DI | 50.513 | 101.7 | 4656 | 97.1 |
| Step 3 Supercritical fluid expansion + Leung - Omega | 50.513 | 101.7 | 4656 | 91.5 |
| Ouderkirk (2002) method | 50.513 | 98.4 | 9324 | 196 |
| Self and Do (2010) method | 50.513 | 98.8 | 8080 | **172.1** |
| Gas expansion with empirical equation 12 of API 521 (2008) + Valvestar for gas (ignoring retrograde condensation) | 50.513 | 100 | 4695 | 139 |

## Considerations

- Once the fire has started, the critical temperature is reached very fast, i.e. 2.0 minutes. No boiling occurs during this time. The required relieving areas during this period are smaller than in the supercritical phase
- The Self and Do (2010) method is tedious in comparison with the others but is the most exact. A process simulator is required to perform multiples isentropic flashes at each relieving temperature elected and, the direct integration method of Darby (2002) is used at each point. In this case, any boiling delay factor has not been taken into account because the nozzle of the valve YS702-1 is greater than 10 cm (Darby, 2002). The Ouderkirk method also gives excellent results and matches the results of the first authors. In both cases, the effective coefficient of discharge $K_d$ is fixed by the author as 0.975 which corresponds to the value used by Ouderkirk in his method. The frequently used empirical formula in industry for supercritical gases (equation 12 of API 521, 2008) did not give conservative results as it is demonstrated: 139 $mm^2$ vs. 172.1 $mm^2$. This result contradicts Doane's (2010) results when he pointed out that the empirical formula in his case study, gives an area 300% greater than the area calculated by his proposed rigorous model
- The engineering company gave conservative values for the relieving load and calculated the required area, ignoring retrograde condensation, with the AD Merkblatt-A2 for gases. And after that the elected area was 314.16 $mm^2$ that means a safety factor of 314.16/172.1 → 82%
- It is demonstrated that the first phase of the relieving process before the critical temperature is reached is not relevant for the dimensioning of the valve
- The extensively used formula of taking a latent heat of 115 kJ/kg in the critical region, gave a very conservative result. This widely accepted solution in the industry has had a boomerang effect because the oversizing of the valves can produce stability problems as is explained in this thesis (see section 4.6).
- In the Self and Do method it is very important the selection of the time step elected for the calculations. In this case, 15 s were taken and the result was 172.1 $mm^2$. In case of 30 s, the required area would be 155 $mm^2$.
- Of course, there are other factors, which have big influence in the final results independently of the method selected, e.g. properties of the fluid near the thermodynamic critical point, exposed surface area, fuel composition, radiation and

convective heat transfers, environmental conditions, etc. that have to be considered as well.

## 6.5 Calculation of relieving loads in distillation columns

In section 4.2 a critical literature review of the calculation methods for the required load in distillation columns in case of an overpressure has been given. A case study has been selected in this thesis to show the complexity of the calculation process if a rigorous model is chosen, for example, the unbalance heat load method (Sengupta and Staats, 1978; Houston, 2014). The safety valve YS6510A/B protects a propane-propylene splitter (see figure 6-21) for different contingencies. Here, the dimensioning case will be evaluated: cooling water failure.

The procedure requires a process simulator, because the first step is doing a heat and mass balance of the column before the upset occurs. Figure 6-8 shows the flow diagram of the column represented as an Aspen Hysys screen.

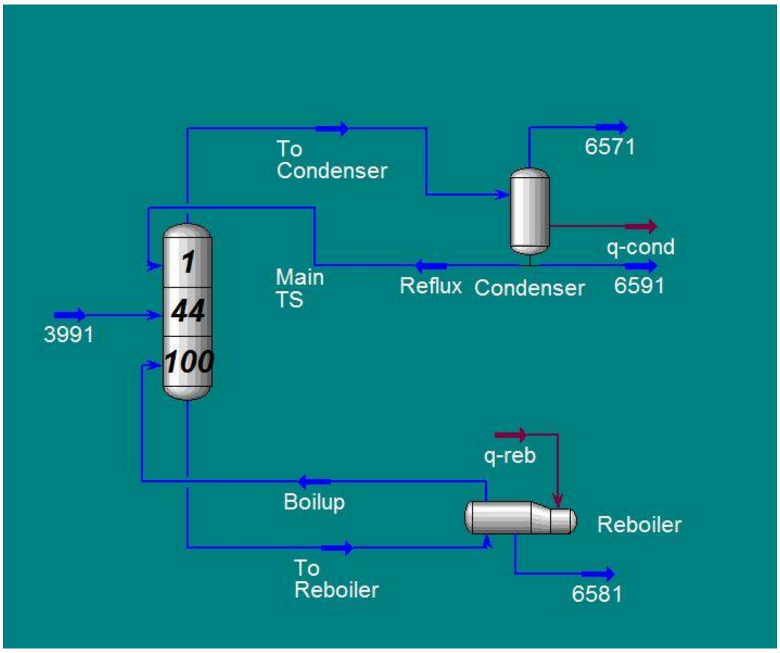



**Figure 6-8. Flow diagram of the propane-propylene splitter according to Aspen Hysys v7.3.**

Cooling water failure means the loss of condensation of the head vapors of the splitter. In this case the reflux drum liquid level drops causing the level control of the liquid distillate to close. Reflux continues at a constant rate due to the flow control valve until there is no more liquid in the drum. The feed continues at a constant rate since its pressure, upstream of the control valve, is higher than the relief pressure. On flow control, the steam control valve opens wide due to the increase of temperature of the bottoms of the column and the reboiler chest pressure equalizes with the steam header pressure (a zero pressure drop is assumed across the control valve).

The basic assumptions that must be made to enable determining the required relief load are exposed in section 4.2. The detailed procedure disclosed by Nezami (2008) has been followed. Aspen Hysys v7.3 with Peng Robinson as EOS has been used.

In table 6-9 the properties of the feed and products are identified.

**Table 6-9 System properties for YS6510A/B.**

| Column parameters | | | Value |
|---|---|---|---|
| Operating pressure (top), barg | | | 15.7 |
| Safety valve set pressure, barg | | | 20 |
| Relieving pressure, bara | | | 23.013 |
| Normal condenser duty, kJ/h | | | 4956000 |
| Condenser duty at relief, kJ/h | | | 0 |
| Normal reboiler duty, kJ/h | | | 4515000 |
| Reboiler duty at relief, kJ/h | | | 6104000 |
| Top tray latent heat at relieving conditions, kJ/kg | | | 321.1 |
| | | | |
| **Feed properties** | | | |
| Feed normal flow rate, kg/h | | | 1235 |
| Feed rate at relieving conditions, kg/h | | | 1235 |
| Specific enthalpy at normal conditions, kJ/kg | | | 320.0 |
| Specific enthalpy at relief conditions, kJ/kg | | | 320.0 |
| Bubble point specific enthalpy at relief, kJ/kg | | | -85.3 |
| | | | |
| **Products properties** | **Vapor distillate** | **Liquid distillate** | **Bottoms** |
| Normal flow rate, kg/h | 38 | 1126 | 71 |
| Flowrate at relieving conditions, kg/h | 45 | 0 | 71 |
| Specific enthalpy at normal conditions, kJ/kg | 211.9 | 74.57 | -1958 |
| Specific enthalpy at relief conditions | 283.8 | 121.5 | -1893 |
| Phase | vapor | liquid | liquid |
| Bubble point specific enthalpy at relief, kJ/kg | -563.2 | | |

In table 6-10 a liquid product adjusted rates and enthalpies has been performed.

**Table 6-10. liquid product adjusted rates and enthalpies for YS6510A/B.**

| Description | Vapor distillate | Liquid distillate | Bottoms | Total |
|---|---|---|---|---|
| (1)Product normal flowrates, kg/h | 38 | 1126 | 71 | 1235 |
| (2)Feed vapor phase flowrate at relieving, kg/h | 1235 | 0 | 0 | 1235 |
| (3)Adjusted rates (1) – (2) | -1197 | 1126 | 71 | 0 |
| (4)Bubble point specific enthalpies at relief, kJ/kg (4) | -563.2 | 121.5 | -1893 | |
| (5)Total enthalpies of products (3) x (4), kJ/h | 674150 | 136809 | -134403 | 676556 |

Load calculations

The load contribution from reboiler is calculated according

$$W_R = \frac{Q_R}{\lambda} = \frac{6104000}{321.1} = 19010\ \frac{kg}{h}$$

The feed streams vapor contribution is calculated as:

$$Q_{FV} = \sum(H_{FVD}i - H_{FVB}i)F_{FV}\,i = [320 - (-85.3)] \cdot 1235 = 500545\ \frac{kJ}{h}$$

$$W_F = \frac{Q_{FV}}{\lambda} = \frac{500545}{321.1} = 1559\frac{kg}{h}$$

Vapor product credits

$$Q_{PV} = \sum(H_{PVD}j - H_{PVB}j)F_{PV}j = [283.8 - (-563.2)] \cdot 45 = 38115\frac{kJ}{h}$$

$$W_P = \frac{Q_{PV}}{\lambda} = \frac{38115}{321.1} = 119\frac{kg}{h}$$

Condenser credits

$$W_C = \frac{Q_C}{\lambda} = 0\frac{kg}{h}$$

Liquids enthalpy imbalance

$$Q_H = \sum H_{PL}j \cdot F_{PL}j - \sum H_{FL}i \cdot F_{FL}i = 676556 - 320 \cdot 0 = 676556\frac{kJ}{h}$$

The liquid product specific enthalpies and adjusted rates as well as the sum of the product total enthalpies, $\sum H_{PL}j \cdot F_{PL}j$, are summarized in table 6-10

Thus

$$W_H = \frac{Q_H}{\lambda} = \frac{676556}{321.1} = 2107\frac{kg}{h}$$

Finally,

$$W = W_R + W_F - W_V - W_C - W_H = 19010 + 1559 - 119 - 0 - 2107 = 18343\frac{kg}{h}$$

It is remarked that with no liquid distillate product, the light component (propene) could get to the column bottom; that is why no credit has been given to the reboiler temperature pinch. Another point is that the vapor distillate pressure control valve keeps its position by relieving and the flow has been corrected taking this approximation: the flow rate through the control valve is proportional to $\sqrt{\text{inlet pressure}}$.

If the gross overhead method were used and taking into account that this value is usually increased 20% due to the safety factor in designing the condenser, the required relieving load for the splitter in cooling water failure would be:

Gross overhead vapor of the splitter: 13790 kg/h

Final gross overhead vapor: $1.2 \cdot 13790 = 16548\frac{kg}{h}$

**Considerations**

- As pointed out by Bradford and Durrett (1984) the gross overhead vapor method could undersize safety valves, as happens here: 16548 kg/h vs. 18343 kg/h. Also the unbalance heat load method is recommended.
- However, with the unbalance heat load method a process simulator is required and depending on the products and columns configuration, convergence problems could appear.

## 6.6 Tube rupture in heat exchangers

There are some cases in this work where the possibility of tube breakage in heat exchangers has to be considered because the mechanical design does not follow the 10/13 rule (see section 3.3). The case study presented here refers to the pressure relief valve YS860/10 which protects a heat exchanger in case of tube rupture. The figure 6-3 (section 6.3) offers a scheme for the installation and, figure 6-9 offers pictures of the safety valve and the protected equipment.


**Figure 6-9. Pictures of YS860/10 and protected equipment.**

The contingency analysis datasheet and the relieving loads summary data sheet of YS860/10 are presented in Figures 6-10 and 6-11 respectively.

| Contingency | | Comments | Justification |
|---|---|---|---|
| 1 | Blocked outlets | Not applicable | Confined chilling propylene refrigerant between valves is heated up by carrier gas stream from V490. Calculate increase of pressure in W867 considering normal operating level of 60%. Blocked outlets at XV860 compressor discharge side is protected by pressure switches P86414, P86416, P86417. |
| 2 | Abnormal heat input | Not applicable | Confined chilling propylene refrigerant between valves is heated up by carrier gas stream from V490. Calculate increase of pressure in W867 considering normal operating level of 60%. |
| 3 | Exchanger tube breakage | See attachments | Tube breakage in W867 (normal operation): carrier gas circuit working pressure (30barg) is higher than chilling propylene working pressure (3.4 barg). If a tube breaks, carrtier gas will ingress into chilling circuit. Shell side is protected by YS860/10. Tube breakage in W(867) (shutdown): Propylene circuit pressure at atmospheric conditions is the same as propylene of the chilling unit. |
| 4 | Auto control failure | Not applicable | Inlet liquid refrigerant valve L86008 opens 100%: Level W867 increases up to switch value of L86008, item L86021, which automatically closes inlet valve LV86008. |
| 5 | Reflux failure | Not applicable | |
| 6 | Fire | See attachments | W867 is located at elevation +9.00. Thus fire scenario is not applicable. |
| 7 | Cooling water failure | Not applicable | Cooling water failure at XV860 compressor discharge side is protected by pressure switches P86414, P86416, P86417. |
| 8 | Power failure | Not applicable | Shutdown of carrier gas compressor V490. No feeding stream. |
| 9 | Instrument air failure | Not applicable | L86008 and P86012 fail closed |
| 10 | Inadvertent VA open/close | Not applicable | See blocked outlet |
| 11 | Mechanical equip. failure | Not applicable | W867 design pressure = 25 barg. YS860/10 set pressure 25 barg. |
| 12 | Heat loss (series frac.) | Not applicable | |
| 13 | Thermal | See atachments | Calculate thermal expansion by sun radiation of confined propylene between valves (80% level) |
| 14 | Loss of quench/cold feed | Not applicable | |
| 15 | Chemical reaction | Not applicable | |
| 16 | Steam out | Not applicable | |

**Figure 6-10. Contingency analysis data sheet of YS860/10**

| | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| TAG/EQUIP. NUMBER | **SV 860/10** | UNIT /SERVICE: **Propylene recovery** | | | | | | P&ID: | | | | PLANT: | | | COST CENTER: | | |
| EQUIPMENT PROTECTED: | | | | | | | SET PRESS: | | **25** | BARG | BASIS: | **List basis for set pressure** | | | | | |
| | | | | | | | DISCHARGE DISPOSITION: | | **Flare** | | INLET PRESSURE DROP: | | **1,08% SP** | | | | |
| | | | | | | | CONSTANT BACKPRESSURE: | | **0,150** | BARG | VARIABLE BACK PRESS.: | | **xx** | BARG | | Kd = | **0,80** |
| EQUIPMENT DESIGN CONDITIONS: | ( ) MAWP | ( X ) Design | | ( ) Other | | | BUILT-UP BACKPRESSURE: | | **xx** | BARG | TOTAL BACKPRESSURE: | | **<10%SP** | BARG | | Kb = | **1,00** |
| NORMAL OPER. | **3,40** BARG | **-9** ºC | | Rupture Disk ,Y/N | **No** | | FIRE SUMMARY | | WETTED AREA: | | m² | ATTACH SKETCH FOR AREA CALCULATION: | | | | | |
| MAX OPER. | **25** BARG | **150** ºC | | Derating Factor= | **1,0** | | INSULATION | **No** | TYPE | **-** | THCKNS | **-** | mm | Insul factr, 1=none | | | |
| DESIGN | **25** BARG | **150** ºC | | (Use 0.9 if have rupture disk) | | | Q = | | **xx** | KJ/h | | | | | | | |
| CONN: RATING FACING : | **PN63 F.E /PN16 F.C** | PIPE SPEC, IN/OUT: | **RA548E** | / | **St261C-E** | | | | | | | | | | | | |

| **Causes of Relief**<br>Refer to API RP520, & RP521 and corporate Relief Manual | | | **RELIEF LOAD** | | **RELIEF (2) CONDITIONS** | | **FLUID PHYSICAL PROPERTIES AT RELIEF CONDITIONS:** | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **Contingency** | Comments NA, etc | % OV PR | VAPOR kg/h | LIQUID m3/h | PRESS BARG | TEMP °C | FLUID TYPE | VAPOR MOL WT | SP GRAVITY LIQUID | COMPR FACTOR Z | LATENT HEAT L, KJ/kg | SP HEAT RATIO k | LIQUID VISC cP | VAPOR VISC cP | VAPOR AREA V mm² | LIQUID AREA L mm² | TOTAL AREA T mm² |
| 1. BLOCKED OUTLETS | **Not applicable** | | | | | | | | | | | | | | | | |
| 2. ABNORMAL HEAT INPUT | **(3)** | **10,0** | **31.000** | | **27,5** | **61,0** | **Propylene** | **42,0** | | **0,700** | | | | | | | **1205,048** |
| 3. EXCHANGER TUBE BREAKAGE | **(4)** | **10,0** | **6.670** | **19.730,0** | **27,5** | **33,0** | **Propylene** | **VAPOR 33,2**<br>**LIQUID 38,4** | **0,450** | **0,710** | | | **0,052** | **0,0115** | | | **680,00** |
| 4. AUTO CONTROL FAILURE | **Not applicable** | | | | | | | | | | | | | | | | |
| 5. REFLUX FAILURE | **Not applicable** | | | | | | | | | | | | | | | | |
| 6. FIRE | **See attachments** | | | | | | | | | | | | | | | | |
| 7. COOLING WATER FAILURE | **Not applicable** | | | | | | | | | | | | | | | | |
| 8. POWER FAILURE | **Not applicable** | | | | | | | | | | | | | | | | |
| 9. INSTR. AIR FAILURE | **Not applicable** | | | | | | | | | | | | | | | | |
| 10. INADVERTENT VA. OPEN/**CLOSE** | **See attachments** | | | | | | | | | | | | | | | | |
| 11. MECH. EQUIP. FAILURE | **Not applicable** | | | | | | | | | | | | | | | | |
| 12. HEAT LOSS (SERIES FRAC.) | **Not applicable** | | | | | | | | | | | | | | | | |
| 13. THERMAL | **See attachments** | | | | | | | | | | | | | | | | |
| 14. LOSS OF QUENCH/COLD FEED | **Not applicable** | | | | | | | | | | | | | | | | |
| 15. CHEMICAL REACTION | **Not applicable** | | | | | | | | | | | | | | | | |
| 16. STEAM OUT | **Not applicable** | | | | | | | | | | | | | | | | |
| 17. | | | | | | | | | | | | | | | | | |
| 18. | | | | | | | | | | | | | | | | | |
| 19. | | | | | | | | | | | | | | | | | |
| 20. | | | | | | | | | | | | | | | | | |

| | | | | |
|---|---|---|---|---|
| NOTES: | -1 SEE RELIEF DEVICE DATA SHEET FOR ACTUAL ORIFICE, COLD DIFF TEST PRESSURE AND OTHER SPECIFICATIONS | GENERAL DATA | BY: | DATE: |
| | -2 RELIEF CONDITIONS ARE AT SET PRESSURE + OVER PRESSURE. | PROCESS DATA | BY: | DATE: |
| | -3 FIGURES FROM UHDE DATASHEET. IT HAS BEEN NOT POSSIBLE TO FIND ITS ORIGIN. | VALVE SIZING | BY: | DATE: |
| | -4 CALCULATION ACCORDING TO ISO 4126-10. | CHECKED/APPROVE | BY: | DATE: |

EXISTING RV DETAIILESER 4564.6062

| | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| SIZING CASE SELECTED: | ORIGINAL FROM UHDE | | | RELIEF DEVICE TYPE: | / | | TOTAL ORIFICE AREA REQD: | **1205,048** | **mm²** | | | |
| DEVICES SELECTED - | QTY: | **1** | INLET SIZE: **50,0** mm | OUTLET SIZ **80,0** | mm | | ORIFICE/AREA (1): | **1256,637** | mm² | SET PRES: **25** | BARG | |
| | QTY: | | INLET SIZE: mm | OUTLET SIZE: | mm | | ORIFICE/AREA (1): | | mm2 | SET PRES: | BARG | **Relieving Loads Summary Data Sheet** |
| | QTY: | | INLET SIZE: mm | OUTLET SIZE: | mm | | ORIFICE/AREA (1): | | mm2 | SET PRES: | BARG | |
| | QTY: | | INLET SIZE: mm | OUTLET SIZE: | mm | | ORIFICE/AREA (1): | | mm2 | SET PRES: | BARG | |

**Figure 6-11. Relief loads summary data sheet for YS860/10.**

A two-phase light hydrocarbon mixture with propylene as the major component at 41.9 ºC and 35.8 barg enters through the tubes and exits from the heat exchanger W867 at 18ºC. In the shell there is a vaporization process of chilling liquid propylene which belongs to the cooling unit of the plant. The chilling propylene enters at -9ºC and 3.42 barg and exits at the same pressure and temperature in vapor phase. The liquid level in the shell is controlled through a control valve in the inlet. The heat exchanger has a separator mounted above the shell to facilitate the liquid/vapor disengagement and to prevent the entrainment of liquid droplets (see figure 6-3).

In case of tube rupture the high pressure liquid/vapor mixture of the tubes will enter the shell and could pressurize the shell side up to 35.8 barg, but the design pressure of the downstream system of the shell is 25 barg. That is why the pressure relief valve YS860/10 has a set pressure of 25 barg.

The design pressure of the tube side of the heat exchanger is 40 barg and the hydrostatic test pressure of the shell side is 1.3 x 25 barg = 32.5 barg. The factor 1.3 comes from the Spanish "Reglamento de Aparatos a presion, RD1244/1979" in force in the year 2001 in Spain, when the heat exchanger was manufactured.

Thus as 40 barg > 32.5 barg the tube rupture scenario has to be considered.

Calculation hypothesis:

a) Tube failure is a sharp break in one tube and occurs at the back side of the tubesheet.
b) The high-pressure fluid flows both through the tube stub remaining in the tubesheet and through the other longer section of tube.
c) API 521 (paragraph 4.4.14.2.2, 2014) accepts the simplifying assumption of two orifices, since this produces a larger relief flow rate than the approach of a long open tube and tube stub. That means the rupture opening equals twice the cross-sectional area of one tube.
d) The fluid flowing through a sharp-edged orifice experiences an isenthalpic expansion.
e) The fluid from the break flows to the relief valve after initial low side mixing with the liquid boiling propylene.
f) In this case the low pressure side is not capable of absorbing the high pressure side flow across a tube rupture due to the specific design of the cooling system.

Flashing results

An isenthalpic flashing was done by Aspen Hysys v7.6 (Peng Robinson EOS used) and the results are presented in table 6.9.

**Table 6-9. Physical properties in the heat exchanger before and after the tube rupture.**

| **Physical Properties** | **Inlet liquid/vapor** | **Outlet vapor** | **Outlet liquid** |
|---|---|---|---|
| Pressure, barg | 35.8 | 27.5 | 27.5 |
| Temperature, ºC | 41.9 | 33 | 33 |
| % w/w vapor | 15.1 | 25.28 | |
| Density, kg/m3 | 236 | 52.04 | 447 |
| Cp/Cv ideal | | 1.125 | |
| **Flow Composition** | | | |
| Ethylene, kg/h | 933.23 | 388.29 | 544.94 |
| Ethane, kg/h | 136.12 | 47.74 | 88.38 |
| Propylene, kg/h | 3068.35 | 619.86 | 2448.49 |
| Propane, kg/h | 181.03 | 34.16 | 146.87 |
| Heptane, kg/h | 6.68 | 0.06 | 6.62 |
| Methane, kg/h | 1.28 | 0.82 | 0.46 |
| Hydrogen, kg/h | 4.78 | 4.07 | 0.71 |
| TOTAL | 4331.47 | 1095.00 | 3236.47 |

**Calculation of relieving load (Wong Method)**

The method of Wong (1992) will be followed keeping the same nomenclature of the original article.

Check for critical flow condition:

$$P_{CFR} = P_1 \left(\frac{2}{K+1}\right)^{\frac{K}{K-1}} = 36.813 \left(\frac{2}{1.125+1}\right)^{\frac{1.125}{1.125-1}} = 21.33 \text{ bara}$$

Comparing the critical flow pressure and the relieving pressure: 21.33 bara vs. 28.513 bara respectively; it is demonstrated that the relieving load is controlled by the relieving pressure of the valve, i.e. this is a case of subcritical flow.

Vapor ratio of the two-phase fluid crossing the tube rupture:

According to Aspen Hysys results: 25.28 %

Required relieving capacity for the vapor (The original equation of Wong's article has been modified taking into account that C is 0.74 for tube-into-shell flow and 0.6 for shell-into-tube flow (Aspen Hysys v8.6). Thus, here C = 0.74)

$$W_V = 1781.7\, A_V \left(1 - 0.317 \frac{dP}{P_1}\right) (dP \cdot \rho_V)^{0.5}$$

where

dP = 35.8 barg – 27.5 barg = 8.3 bar = 120.4 psi; $P_1$ = 35.8 barg = 533.9 psia; $\rho_v$ = 52.04 kg/m$^3$ = 3.25 lb/ft3.

Giving values

$$W_V = 1781.7\, A_V \left(1 - 0.317 \frac{120.4}{533.9}\right) (120.4 \cdot 3.25)^{0.5} = 32725\, A_V$$

Required relieving capacity for the liquid, considering C = 0.74

$$W_L = 1781.7\, A_L (dP \cdot \rho_L)^{0.5}$$

$\rho_L$ = 447 kg/m3 = 27.9 lb/ft3

Giving values $W_L = 1781.7\, A_L (120.4 \cdot 27.9)^{0.5} = 103264\, A_L$

The total area of the exchanger tube with OD = 25 mm, ID = 21 mm = 0.827 in

$$A_{TOTAL} = A_V + A_L = 2 \frac{\pi D^2}{4} = 2 \frac{\pi \cdot 0.827^2}{4} = 1.074\, in^2$$

Thus

$$\begin{cases} W_V = 32725\, A_V \\ W_L = 103264\, A_L \\ A_V + A_L = 1.074 \\ \frac{W_V}{W_V + W_L} = 0.2528 \end{cases}$$

and solving the system on get

$$\begin{cases} W_L = 53646 \frac{lb}{h} = 24333 \frac{kg}{h} \\ W_V = 18146 \frac{lb}{h} = 8231 \frac{kg}{h} \\ A_V = 0.5545\, in^2 \\ A_L = 0.5195\, in^2 \end{cases}$$

**Calculation of relieving load (ISO 4126-10)**

The procedure given in ISO 4126-10 (2010) will be followed with the same nomenclature.

The first step is checking for the applicability of the method. According to ISO 4126-10 the method is applicable if fulfills

$P_R \leq 0.5$ and $T_R \leq 0.9$

In this case the values of the critical pressure and critical temperature of the hydrocarbon mixture are obtained by Aspen Hysys v7.3 with Peng Robinson EOS. The results are: $P_C = 56.89\ bara$ and $T_C = 72.6$ °C

Thus

$P_R = \frac{P}{P_C} = \frac{36.813\ bara}{56.89\ bara} = 0.65$ and $T_R = \frac{T}{T_C} = \frac{41.9+273.15}{72.6+273.15} = 0.91$

Strictly speaking the method is out of the recommended range of application but following Schmidt (Schmidt and Egan, 2009) this range can be enlarged if the operating conditions are far from the critical region or it can be demonstrated that the property data is maintained substantially constant between the inlet and the nozzle of the valve and the heat of vaporization is not very small. Both last conditions have been confirmed in this case.

Compressibility coefficient ω

$$\omega = \frac{\dot{x}_0 \cdot v_{g,0}}{K_0 \cdot v_0} + \frac{C_{pl,0} \cdot p_0 \cdot T_0}{v_0} \left( \frac{v_{g,0} - v_{l,0}}{\Delta h_{v,0}} \right)^2$$

were (the values were obtained by Aspen Hysys as it was explained in the Wang procedure)

$\dot{x}_0 = 0.1508$ mass fraction of vapor at the inlet of the tube; $v_{g,0} = 0.0145\ \frac{m^3}{kg}$ specific volume of gas; K = 1.109 isentropic expansion (ideal); $v_0 = 0.0042\ \frac{m^3}{kg}$specific volume of the mixture; $C_{pl,0} = 3856\ \frac{J\ kg}{K}$ heat capacity of the liquid; $p_0 = 3681300$ Pa (abs); $T_0 = 315.05$ K; $v_{l,0} = 0.0024\ \frac{m^3}{kg}$ specific volume of liquid and $\Delta H_{v,0} = 320700\ \frac{J}{kg}$ latent heat of vaporization.

Giving values

$$\omega = \frac{0.1508 \cdot 0.0145}{1.109 \cdot 0.0042} + \frac{3856 \cdot 3681300 \cdot 315.05}{0.0042} \left( \frac{0.0145 - 0.0024}{320700} \right)^2 = 1.985$$

The critical pressure ratio must be calculated by the equation

$$0 = \eta_{crit}^2 + (\omega^2 - 2\omega)(1 - \eta_{crit})^2 + 2 \cdot \omega^2 \cdot \ln \eta_{crit} + 2 \cdot \omega^2 (1 - \eta_{crit})$$

By trial and error

$\eta_{crit} = 0.6916$

The boiling delay is 1.0 because Schmidt (2010) recommends this value for pipes

There is no critical flow because:

$$\eta_b = \frac{p_b}{p_0} = \frac{28.513}{36.813} = 0.775$$

$\eta_b = 0.775 > 0.6916 = \eta_{crit}$ → no critical flow

The flow coefficient is calculated by

$$C = \frac{\sqrt{\omega \ln\left(\frac{1}{\eta_b}\right) - (\omega - 1)(1 - \eta_b)}}{\omega\left(\frac{1}{\eta_b}\right) + 1}$$

Given values

$$C = \frac{\sqrt{1.985\ \ln\left(\frac{1}{0.775}\right) - (1.985 - 1)(1 - 0.775)}}{1.985\left(\frac{1}{0.775}\right) + 1} = 0.3383$$

And by the mass flux

$$\dot{m}_{SV} = K_{dr,2ph} \cdot C \cdot \sqrt{\frac{2 \cdot p_0}{\nu_0}} = 0.9 \cdot 0.3383 \cdot \sqrt{\frac{2 \cdot 3681300}{0.0042}} = 12747.8\ \frac{kg}{m^2 s}$$

The value of $K_{dr,2ph} = 0.9$ comes from the recommendation of Leung (1996) in a similar example. The ISO 4126-10 gives a generalized value of 0.85 for two-phase flow at the inlet.

The area of the both sides of the tube is

$$A = 2\pi \frac{D^2}{4} = 2\pi \frac{0.021^2}{4} = 0.0006927\ m^2$$

The flow through the tubes is

$$Q_{m,out} = A \cdot \dot{m}_{SV} = 0.0006927 \cdot 12747.8\ = 8.8307\ \frac{kg}{s} = 31790\ \frac{kg}{h}$$

From this quantity 8037 kg/h are vapor and 23753 kg/h are liquid.

**Calculation of relieving load (Schmidt)**

Schmidt (2010) in a chapter of the VDI Heat Atlas dedicated to heat exchangers tube breakage presents the HNE-DS model. The difference with the ISO 4126-10 is the possibility of taking into account the thermodynamic nonequilibrium (boiling delay) and the mechanical nonequilibrium (slip between phases). This model allows for the possibility of simulating the tube rupture at the tubesheet and the consideration of boiling delay if the thickness of the tubesheet is ≤ 10 cm. Thus the relieving load will be calculated independently for the two parts of the ruptured tube.

*Relieving through the short nozzle of the tubesheet*

The boiling delay factor is

$$N = \left[\dot{x}_0 +\ C_{pl,0} \cdot p_0 \cdot T_0 \frac{\nu_{g,0} - \nu_{l,0}}{\Delta h_{v,0}{}^2} \ln\frac{1}{\eta_{crit}}\right]^{3/5}$$

The value of the exponent 3/5 comes from the table 1 of Schmidt and Diener (2005). Thus giving values

$$N = \left[0.1508 + 3856 \cdot 3681300 \cdot 315.05 \frac{0.0145 \cdot 0.0024}{320700^2} \ln\left(\frac{1}{0.6916}\right)\right]^{3/5} = 0.528$$

$\omega = 1.985 \cdot 0.528 = 1.048$

The flow coefficient, will be calculated by

$$C = \frac{\sqrt{\omega \ln\left(\frac{1}{\eta_{crit}}\right) - (\omega - 1)(1 - \eta_{crit})}}{\omega\left(\frac{1}{\eta_{crit}} - 1\right) + 1} = \frac{\sqrt{1.048 \ln\left(\frac{1}{0.6916}\right) - (1.048 - 1)(1 - 0.6916)}}{1.048\left(\frac{1}{0.6916} - 1\right) + 1}$$

$C = 0.4155$

Thus

$$\dot{m}_{SV} = K_{dr,2ph} \cdot C \sqrt{\frac{2 \cdot p_0}{v_0}} = 0.9 \cdot 0.4155 \sqrt{\frac{2 \cdot 3681300}{0.0042}} = 15657 \ \frac{kg}{m^2 s}$$

The area is $A = \pi \frac{D^2}{4} = \pi \frac{0.021^2}{4} = 0.0003464 \ m^2$

And finally

$$Q_{m,tubesheet} = A \cdot \dot{m}_{SV} = 0.0003464 \cdot 15657 = 5.4230 \frac{kg}{s} = 19523 \frac{kg}{h}$$

*Relieving through the long tube of the tubesheet*

In this case as it is recommended by Schmidt (2010): $a = 0 \rightarrow N = 1$ thus

$$Q_{m,tube} = A \cdot \dot{m}_{SV} = 0.0003464 \cdot 12747.8 = 4.4158 \frac{kg}{s} = 15897 \frac{kg}{h}$$

The pressure drop in the pipe has been ignored (conservative basis)

Thus finally the flow to be relieved is

Short nozzle (tubesheet): 19523 kg/h

Long tube: 15897 kg/h

Total: 34420 kg/h

**Calculation of relieving load (Leung)**

The same nomenclature of the original paper of Leung (1996) will be followed. The first step is calculating the omega parameter

$\omega = \frac{x_0 \cdot v_{g0}}{v_0} + \frac{C_p \cdot T_0 \cdot P_0}{v_0}\left(\frac{v_{fg0}}{h_{fg0}}\right)^2$ given values

$$\omega = \frac{0.1508 \cdot 0.0145}{0.0042} + \frac{3856 \cdot 315.05 \cdot 3681300}{0.0042}\left(\frac{0.0121}{320700}\right)^2 = 2.036$$

From Fig 3 $G_0^* = 0.495$ and $\frac{P_C}{P_0} = 0.7$

$$G_0 = G_0^* \cdot \left(\frac{P_0}{\nu_0}\right)^{\frac{1}{2}} = 0.495 \cdot \left(\frac{3681300}{0.0042}\right)^{0.5} = 14655\ \frac{kg}{m^2 s}$$

$P_C = 0.7 \cdot 36.813 = 25.77$ bara

25.77 bara < 28.513 bara → Flow no choked !

Thus $\frac{P_b}{P_0} = \frac{28.513}{36.813} = 0.775$

The $G_0$ corrected by Fig 4 → 0.475

Finally

$$G_0 = 0.475 \cdot \left(\frac{3681300}{0.0042}\right)^{0.5} = 14063\ \frac{kg}{m^2 s}$$

Case a) tubesheet

$$W = K_d G_0 A_N = 0.9 \cdot 14063 \cdot 0.0003464 = 4.384\ \frac{kg}{s} = 15783\ \frac{kg}{h}$$

Case b) tube long

$$4f\frac{L}{D} = 4 \cdot 0.005 \cdot \frac{7}{0.021} = 6.7$$

$f$ is the Fanning factor and according to Leung a value of 0.005 for two-phase flow is proposed. L is the length of the tubes of the heat exchanger. In this case a U tube has 7 m approximately.

From Fig 6 → $\frac{G_c}{G_{oc}} = 0.515$

$$W = G_{oc}\left(\frac{G_c}{G_{oc}}\right) A_p = 14655 \cdot 0.515 \cdot 0.0003464 = 2.614\ \frac{kg}{s} = 9412\ \frac{kg}{h}$$

From Fig 9 → $\frac{P_{2c}}{P_0} = 0.36$ → $P_{2c} = 0.36 \cdot 36.813 = 13.3$ bara

13.3 bara < 28.513 bara → No critical flow !

As no choked flow at the exit of the pipe the Fig 10 will be used

$\frac{P_0 - P_b}{P_0 - P_{2c}} = \frac{36.813 - 28.513}{36.813 - 13.3} = 0.35$ → $\frac{G}{G_c} = 0.80$ thus

$$G = 0.8 \cdot G_c = 0.8 \cdot 14655 \cdot 0.515 \cdot 0.0003464 \cdot \frac{3600\ s}{1h} = 7529\ \frac{kg}{h}$$

The summary of relieving loads is

Short nozzle (tubesheet): 15783 kg/h

Long tube: 7529 kg/h

Total: 23312 Kg/h

**Calculation of relieving load (TPHEM modified)**

The TPHEM modified is an independently developed software based on the CCPS (1998). It follows the methodology of Simpson (1991).

The program has been run with the following calculation hypothesis:

a) Two points model has been adopted. The first point is the pressure relieving point and the second one corresponds to 70% of the first one because it is already known that the critical pressure is near this pressure. Even though the original CCPS book recommends 80%.
b) The obtained values of the program have been corrected because no critical flow is present.
c) The value of Kd has been fixed at 0.84 as recommended by the program instructions. For instance the ISO 4126-10 recommended 0.85 for two-phase flow.
d) Isenthalpic flashes have been performed with Aspen Hysys v7.3 with Peng Robinson EOS. They are conservative with respect to isentropic and in this case we are not in the critical region.

Summary of the results of TPHEM modified:

| Model | Application | Gchocke, lb/ft$^2$s | G @ 413.55 psia | Wrelief, lb/h |
|---|---|---|---|---|
| Slip Equilibrium Model, SEM | Tube | 3661.8 | 3525 | 39741 |
| Homogeneous Non Equilibrium Model, HNEM | Tubesheet < 10 cm | 3479.9 | 3319 | 37419 |

Thus the quantity to be relieved is 39741 + 37419 = 77160 lb/h = 35000 kg/h

**Calculation of relieving load (PS PPM software, Siemens)**

This program has a wide acceptance in the industry and it is based on different models of the CCPS Book (1998). Three points are required for the option of a two-phase flow at the inlet of the valve/tube. The points should be: the relieving point, the critical point (estimated) and the downstream point, i.e. 35.8 barg, 24.75 barg and 27.5 barg in this case.

Previously isenthalpic flashes with Aspen Hysys v7.3 and Peng Robinson as EOS were obtained.

Other parameters: ε (roughness) = 0.00152 mm; orifice discharge coefficient = 0.6; ID tube = 21 mm, tube length = 7 m; nozzle discharge coefficient = 0.85

Summary of results

| PS PPM model | Flow, kg/h | Total, kg/h |
|---|---|---|
| Nozzle/tube | Tube (7 m): 10676<br>Nozzle (tubesheet): 17789 | 28465 |
| 2 orifice | 20736/each | 41472 |
| Orifice/tube | Orifice: 20736<br>Tube: 10676 | 31412 |
| 2 nozzle | 17789/each | 35578 |

In table 6-10 the summary of the results is presented.

**Table 6-10. Results of the relieving load in case of tube breakage for YS860/10.**

| Method | Model characteristics | Discharge coefficient, Kd | Calculated required flow, kg/h | Comments |
|---|---|---|---|---|
| **Basic Engineering**<br>**(Original datasheet)** | Phase vapor only. | Kd: 0.8 | 31000 Kg/ hr of light hydrocarbon vapor. | It is not clear if this scenario was studied. In any case it was usual for the engineering company taking a very conservative scenario including not related events. |
| **Wong (1992)** | Two sharp edged orifices and isenthalpic expansion. No taking into account the tubesheet and tube part.<br>Critical flow calculated for the gas only. Crane formulas for pressure drop are used. | C = 0.6 (shell-into-tube flow).<br>C = 0.74 (tube-into-shell flow) | 32564 kg/h of a mixture vapor/liquid with a quality of 25.28%. No choked flow. | Although the method is easy to implement, the concept of adding the areas after treating the vapor and the liquid separately lacks consistency. |
| **ISO 4126-10 (2010)** | Based on the Homogeneous Equilibrium Model developed by Leung and coworkers. Improved because of boiling delay factor possibility. Requires tedious iterations in the case of subcooled liquid. | The value of Kd in the case of tube rupture is not addressed. It has been taken as 0.9 based on the Leung example No 1 (1996). | 31790 kg/h of a mixture with a quality of 25.28% (isenthalpic flash). No boiling delay, a=0 → N=1 | As in all the methods explored, this one has no precise instructions for the calculation of the discharge coefficient in tube rupture. |
| **Schmidt (2010)** | Similar to ISO 4126-10<br>In this case, the tubesheet was treated as a nozzle with boiling delay and with the exponent a= 3/5 and the other part as a tube with a=0 → N=1 | The value of Kd in case of tube rupture is not addressed. It has been taken as 0.9 based on the Leung example No. 1 | 35420 kg/h of a mixture with a quality of 25.28%. The incremental flashing across the tube part is ignored. | No instructions about the calculation of the discharge coefficient. |
| **Leung (1996)** | Homogeneous Equilibrium Model is thoroughly developed and charts for a pencil-and-paper approach are presented. The tubesheet is treated as a nozzle and the pipe is treated with its equivalent length including fitting. | Leung uses in the example 1, pp 38, a discharge coefficient for the nozzle of 0.9 but gives no justification why. | 23312 kg/h of a mixture with a quality of 25.28% | The Leung method is a very rigorous method and includes not only the tubesheet as a nozzle, but the pipe part of the rupture.<br>However no instructions about the calculation of the discharge coefficient are given. |
| **TPHEM modified (1998)** | Model based in the papers of Simpson considering isentropic transformation and performing an integral. Only 2 or three isentropic points are required.<br>Treatment of the | The coefficients elected in the examples are very correct for the cases studied. | 35000 kg/h of a mixture with a quality of 25.28% | Recommended Method. But using the discharge coefficients of the examples. |

| Method | Model characteristics | Discharge coefficient, Kd | Calculated required flow, kg/h | Comments |
|---|---|---|---|---|
| | tubesheet as a nozzle with the Homogeneous Non-Equilibrum Model (HNEM) and Slip Equilibrium Model (SEM) for the tube side. | | | |
| **PS PPM (Siemens)** | Methods based on the CCPS Book Guidelines for pressure relief and effluent handling systems, 1998 | Treatment of the tubesheet as a nozzle with a discharge coefficient of 0.85 | 28465 Kg /h of a mixture with a quality of 25.28% | Recommended method (conservative side) but using the correct discharge coefficient. |

### Considerations

- The Schmidt and the TPHEM methods, which model the tubesheet as a nozzle and the tube part as an orifice with thermodynamic and mechanical equilibrium, showed higher relieving loads. Both methods are recommended when an exact value of the required load is necessary
- The analysis of different options of modeling the tube breakage performed with PS PPM software confirms that the option 2 orifices gives the greater required load, as it is pointed out by API 521 (2014).This option is followed by the engineering community in the design phase because it is conservative
- The method of Leung allows a paper and pencil approach and can be solved quickly with a spreadsheet
- The Wong method was the standard in the petrochemical industry in the last century. As shown here, it is a relatively conservative method based on the model of two sharp edged orifices.

## 6.7 Application of the engineering analysis methodology according to API 520 Part II-2015

As explained in section 4.6 the Engineering Analysis concept appeared in the 1994 version of API 520 Part II for pressure relief valves with an inlet pressure drop greater than 3%. However, no guidance was given on how to perform this analysis.

Only in 2015, the 6$^{th}$ Edition of API 520 Part II, are some recommendations given about the topics the users may consider for their engineering analysis and adds “Because the relationship between inlet pressure loss and PRV chatter is not definitively understood, detailed requirements for an engineering analysis are the responsibility of the user”.

The recommendations of API have been adapted to form an original screening process developed for this thesis. The process consists of answering the questions of table 6-11.

**Table 6-11. Screening test to detect chattering possibilities.**

| Question | Answer to avoid chattering | Comments |
|---|---|---|
| According to the inspection records is there any evidence of past chattering? | No | |
| Is the pressure relief valve well installed according to API 520, ISO 4126-9, etc.? | Yes | Consider the manufacturers recommendations as |

| | | |
|---|---|---|
| | | well. |
| Is the inlet piping and fittings at least as large as the PRV inlet? | Yes | |
| Is there at least a 2% Set Pressure (SP) margin between PRV blowdown and the inlet pressure loss? | Yes | |
| Does excessive built-up backpressure occur according to the specific PRV? | No | Conventional 10% of SP<br>Balanced 30-50% of SP. |
| Is the time that the decompression wave goes back to the protected equipment and returns to the valve, less than the time required for the full opening of the valve? | Yes | |
| Does the PRV fulfill API 520 II-2015 Simple Force Balance? | Yes | |
| Is the risk of relieving of the existing pressure relief valve quantified? | Yes (acceptable) | Reference is made to the relieving to flare or atmosphere. |
| Is a risk analysis done comparing an unsuccessful mechanical change in the field with the risk of chattering? | Yes (acceptable) | Consider the risk of removal an existing inlet piping and installation of a new one, for example. |

If only one question does not conform to those in the table, chattering has to be considered and mitigation measures have to be implemented.

The typical mitigation actions for the case that a pressure relief valve has an inlet pressure drop greater than 3% and failed the engineering analysis, are:

- Increasing the diameter of the inlet pipe of the valve.
- Reduction of the distance between pressure relief valve and protected equipment.
- Restriction of the lift of the pressure relief valve considering the required flow.
- Changing the valve for another with the same size but with less area.
- Changing to remote sensing pop-action pilot PRV.
- Installation of a vibration damper.
- Others.

A case study will be presented applying the screening process developed for this thesis. It corresponds to the valve YS700/01-02 with an inlet pressure drop of 4.18% of the set pressure (not fulfill the 3% rule).

A scheme for the installation of the valve is presented in figure 6-12 and a picture of the valve and the protected equipment is presented in figure 6-13.

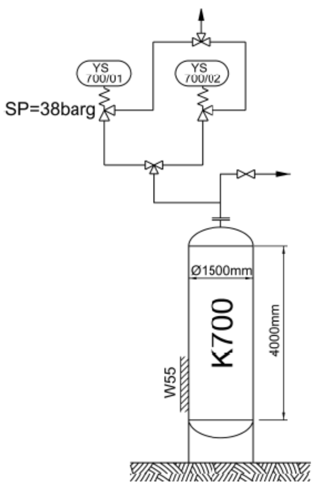



**Figure 6-12. Scheme of YS700/01-02 installation. Figure 6-13. Picture of YS700/01-02 and protected equipment.**

YS700/01-02 protects the liquid propylene dryer from two relieving scenarios: fire and thermal expansion according to the contingency analysis done. See figure 6-15.The dryer is full of propylene liquid at 23 ºC and 16 bara in operating conditions, which corresponds to a lightly subcooled propylene.

| Contingency | | Comments | Justification |
|---|---|---|---|
| 1 | Blocked outlets | Not applicable | Pumps P12A/B/C are protected against overpressure by SV399/01 and SV399/02 at 23.5 barg set pressure and SV151.2 at 30 barg set pressure. |
| 2 | Abnormal heat input | Not aplicable | Heat provided by internals regeneration with hot nitrogen is protected by SV700/03. |
| 3 | Exchanger tube breakage | Not applicable | |
| 4 | Auto control failure | Not applicable | |
| 5 | Reflux failure | Not applicable | |
| 6 | Fire | See attachments | |
| 7 | Cooling water failure | Not applicable | No cooling water system in place. |
| 8 | Power failure | Not applicable | |
| 9 | Instrument air failure | Not applicable | Instrument air failure closes inlet valve PV70001 and closes outlet valves H70102 (to K702A) and H20104 (to K702B). |
| 10 | Inadvertent VA open/close | Not applicable | Nitrogen 35 barg inlet is blocked by a spectacle blind. It is not possible to enter $N_2$ by human error. |
| 11 | Mechanical equip. failure | Not applicable | |
| 12 | Heat loss (series frac.) | Not applicable | |
| 13 | Thermal | Applicable | Calculate wit a $\Delta T$ = 25 ºC (difference between night and day) |
| 14 | Loss of quench/cold feed | Not applicable | |
| 15 | Chemical reaction | Not applicable | Reaction heat produced by internal catalyst could lead to an increase of temperature. Temperature inside column is redundant measured (T70006 and T70007). High temperature closes inlet valve PV70001. |
| 16 | Steam out | Not applicable | |

**Figure 6-14. Results of the contingency analysis for YS700/01-02.**

The relieving loads for each scenario have been calculated and are presented in figure 6-15.

| | | | |
|---|---|---|---|
| TAG/EQUIP. NUMBER **SV 700/01 & 02** | UNIT /SERVICE: **Propylene purification unit** | P&ID: | PLANT: / COST CENTER: |
| EQUIPMENT PROTECTED: | | SET PRESS: **38** BARG | BASIS: **Relief system update** |
| | | DISCHARGE DISPOSITION: **Flare** | INLET PRESSURE DROP: **4,2% SP** |
| | | CONSTANT BACKPRESSURE: **0,150** BARG | VARIABLE BACK PRESS.: BARG Kd = **0,28 (3)** |
| EQUIPMENT DESIGN CONDITIONS: ( ) MAWP ( **X** ) Design ( ) Other | | BUILT-UP BACKPRESSURE: **2,98** BARG | TOTAL BACKPRESSURE: **3,13** PSIG Kb = **1** |
| NORMAL OPER. **30** BARG **20** ºC | Rupture Disk ,Y/N **No** | FIRE SUMMARY | WETTED AREA: **21** m² ATTACH SKETCH FOR AREA CALCULATI **See backup material** |
| MAX OPER. **40** BARG **120** ºC | Derating Factor= **1,0** | INSULATION **Yes** | TYPE **Rockwool** THCKNS **70** mm Insul factr, 1=none **F =1.0 (4)** |
| DESIGN **40** BARG **120** ºC | (Use 0.9 if have rupture disk) | Q = **523,8** KJ/h | (See backup material) |
| CONN: RATING FACING : **PN63 F.E /PN16 F.C** | PIPE SPEC, IN/OUT: **St860E-A** / **St261C-E** | | |

| Causes of Relief (Refer to API RP520, & RP521 and corporate Relief Manual) Contingency | Comments NA, etc | % OV PR | RELIEF LOAD VAPOR kg/h | RELIEF LOAD LIQUID kg/h | RELIEF (2) CONDITIONS PRESS BARG | RELIEF (2) CONDITIONS TEMP °C | FLUID TYPE | VAPOR MOL WT | SP GRAVITY LIQUID | COMPR FACTOR Z | LATENT HEAT L, KJ/kg | SP HEAT RATIO k | LIQUID VISC cP | VAPOR VISC cP | VAPOR AREA V mm² | LIQUID AREA L mm² | TOTAL AREA T mm² |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1. BLOCKED OUTLETS | **Not applicable** | | | | | | | | | | | | | | | | |
| 2. ABNORMAL HEAT INPUT | **Not applicable** | | | | | | | | | | | | | | | | |
| 3. EXCHANGER TUBE BREAKAGE | **Not applicable** | | | | | | | | | | | | | | | | |
| 4. AUTO CONTROL FAILURE | **Not applicable** | | | | | | | | | | | | | | | | |
| 5. REFLUX FAILURE | **Not applicable** | | | | | | | | | | | | | | | | |
| 6. FIRE | **Applicable** | **10** | **15.000** | | **41,8** | **86** | **vapor** | **42,00** | | **0,50** | **125** | **1,230** | | **0,012** | | | **349 (7)** |
| 7. COOLING WATER FAILURE | **Not applicable** | | | | | | | | | | | | | | | | |
| 8. POWER FAILURE | **Not applicable** | | | | | | | | | | | | | | | | |
| 9. INSTR. AIR FAILURE | **Not applicable** | | | | | | | | | | | | | | | | |
| 10. INADVERTENT VA. OPEN/**CLOSE** | **Not applicable** | | | | | | | | | | | | | | | | |
| 11. MECH. EQUIP. FAILURE | **Not applicable** | | | | | | | | | | | | | | | | |
| 12. HEAT LOSS (SERIES FRAC.) | **Not applicable** | | | | | | | | | | | | | | | | |
| 13. THERMAL | **Applicable** | **10,0** | | **243 (6)** | **41,8** | **70,0** | **Liquid** | **42,00** | | | | | | | | | **4,1 (5)** |
| 14. LOSS OF QUENCH/COLD FEED | **Not applicable** | | | | | | | | | | | | | | | | |
| 15. CHEMICAL REACTION | **Not applicable** | | | | | | | | | | | | | | | | |
| 16. STEAM OUT | **Not applicable** | | | | | | | | | | | | | | | | |
| 17. | | | | | | | | | | | | | | | | | |
| 18. | | | | | | | | | | | | | | | | | |
| 19. | | | | | | | | | | | | | | | | | |
| 20. | | | | | | | | | | | | | | | | | |

NOTES:
- -1 SEE RELIEF DEVICE DATA SHEET FOR ACTUAL ORIFICE, COLD DIFF TEST PRESSURE AND OTHER SPECIFICATIONS
- -2 RELIEF CONDITIONS ARE AT SET PRESSURE + OVER PRESSURE.
- -3 The lift is restricted to 4,5 mm. That is why Kd=0,28
- -4 Although there is 70 mm insulation, it is not considered-> conservative design.
- -5 Calculation performed acc. C2.3 Annex C, API 520-I-2008
- -6 Relief load due to solar radiation 1,15 kw/m2
- -7 This value is obtained ignoring retrograde condensation. However using method C2.2. of API 520-I-2008 the required area=385 mm2
- -8 An engineering analysis according to API 520-II-2014 due to unstability was performed

GENERAL DATA BY: DATE:
PROCESS DATA BY: DATE:
VALVE SIZING BY: DATE:
CHECKED/APPROVE BY: DATE:

EXISTING RV DETAILS: **LESER 4564.6062**

| | | | | | |
|---|---|---|---|---|---|
| SIZING CASE SELECTED: **Fire** | | RELIEF DEVICE TYP **PSV /** | TOTAL ORIFICE AREA REQD: **349,000** mm² | | |
| DEVICES SELECTED - | QTY: **2** | INLET SIZE: **50,0** mm | OUTLET SIZ **80,0** mm | ORIFICE/AREA (1): **1256,637** mm² | SET PRES: **38** BARG |
| | QTY: | INLET SIZE: mm | OUTLET SIZE: mm | ORIFICE/AREA (1): mm2 | SET PRES: BARG |
| | QTY: | INLET SIZE: mm | OUTLET SIZE: mm | ORIFICE/AREA (1): mm2 | SET PRES: BARG |
| | QTY: | INLET SIZE: mm | OUTLET SIZE: mm | ORIFICE/AREA (1): mm2 | SET PRES: BARG |

**Relieving Loads Summary Data Sheet**

**Figure 6-15. Relieving loads for each scenario of YS700/01-02.**

The design conditions for the pressure relief valve according to the original specification sheet are:

Scenario: fire, Relieving fluid: propylene gas; Set pressure: 38 barg; Relieving pressure: 42.8 bara; Relieving temperature: 82.5 ºC; Relieving load: 15,000 kg/h; Overpressure: 10%; Molecular weight: 42 kg/kmol; Z (compressibility factor) = 0.5; Cp/Cv = 1.23.

However, the relieving load, in case of fire, depends on the moment of the process. Once the fire begins, the phenomenon that happens is:

- Isochoric transformation (thermal expansion) of the trapped liquid propylene from 23 ºC and 16 bara (normal operating conditions) to 42.8 bara (relieving pressure).
- When the valve opens, it relieves propylene liquid which vaporizes following an isentropic transformation until the total backpressure is 4.14 bara (this total backpressure was calculated according to the methodology of section 6.1).
- When the liquid reaches 88 ºC, it begins to boil with formation of bubbles at the wall and the valve releases two-phase flow at the inlet and at the outlet, and only after the disengagement of the vapor/liquid phases, begins the release of vapor.
- After the boiling phase an expansion of the gas begins with the possibility of retrograde condensation.

The relieving process can be represented by the following Mollier Diagram, figure 6-16. Although it is a dynamic process some singular points are representative of the phenomenon and have been represented on it.

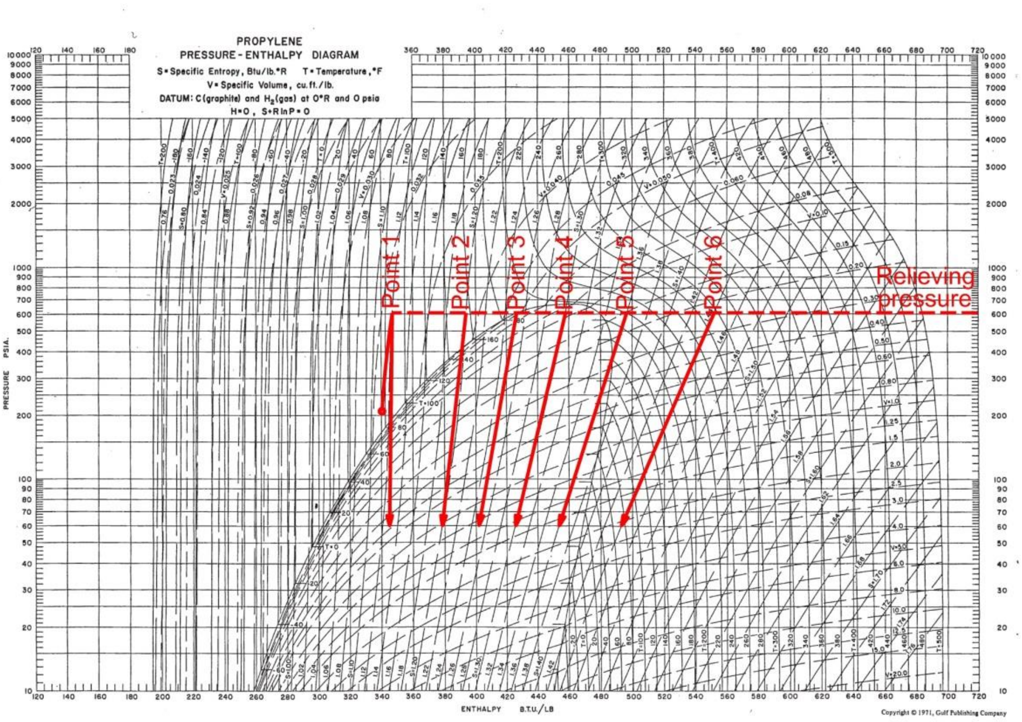



**Figure 6-16. Representation of the relieving process of YS700/01-02 in case of fire.**

**1st Point (42.8 bara & 27ºC), thermal expansion**

The 1st point of the diagram arises from an isochoric transformation from the operating conditions (23 ºC and 16 bara) and the result is 42.8 bara and 27 ºC. The relieving load can be calculated according to API 521 (paragraph 4.4.12.3, 2014). It is considered the hydraulic expansion of the liquid propylene.

$$q = \frac{\alpha_v \cdot \emptyset}{1000 \cdot d \cdot c} = \frac{0.00358 \cdot 523835}{1000 \cdot 0.5 \cdot 2889} \cdot \frac{3600\, s}{1\, h} = 4.67\ \frac{m^3}{h}$$

where

$q$ is the volume flow rate, $m^3/s$; $\alpha_v$ is the cubic expansion coefficient, 1/ºC; $\emptyset$ is the total heat transfer rate, watts; $d$ is the specific gravity; $c$ is the specific heat capacity, J/kg K.

The cubic expansion coefficient for propylene liquid is calculated with the Yaws (1995) correlation. Thus,

$$\alpha_{v\ @\ 27\ ℃} = 0.00107\left(1 - \frac{273.15 + 27}{364.76}\right)^{-0.6975} = 0.00358\ 1/℃$$

Propylene specific gravity @ 27 ºC is 0.500 and its specific heat capacity is 2889 J/kg K (API Technical Data Book, 1997)

The heat transfer rate has been calculated as (Refer to point 4 to find the wetted area):

$$Q = 43200 \cdot F \cdot {A_w}^{0.82} = 43200 \cdot 1 \cdot 20.97^{0.82} = 523835\ watts$$

**2nd Point (42.8 bara & 56 ºC), thermal expansion**

The 2nd point has been selected as the middle point between 27 ºC and the boiling point at relieving conditions (86ºC), thus, 56 ºC.

$$q = \frac{\alpha_v \cdot \emptyset}{1000 \cdot d \cdot c} = \frac{0.00542 \cdot 523835}{1000 \cdot 0.44\ \cdot 3349} \cdot \frac{3600\, s}{1\, h} = 6.94\ \frac{m^3}{h}$$

Using the Yaws correlation for the cubic expansion coefficient

$$\alpha_{v\ @\ 56\ ℃} = 0.00107\left(1 - \frac{273.15 + 56}{364.76}\right)^{-0.6975} = 0.00542\ 1/℃$$

And the following parameters are obtained from the API Technical Data Book, (1997):

$d_{@\ 56\ ℃} = 0.44$ and $c_{@\ 56\ ℃} = 3349\ \frac{J}{kg\ K}$

**3rd Point (42.8 bara & 86ºC-bubble point), thermal expansion**

$$q = \frac{\alpha_v \cdot \emptyset}{1000 \cdot d \cdot c} = \frac{0.0197 \cdot 523835}{1000 \cdot 0.343 \cdot 6833} \cdot \frac{3600\, s}{1\, h} = 15.85\ \frac{m^3}{h}$$

Using the Yaws correlation for the cubic expansion coefficient

$$\alpha_{v\ @\ 86\ ℃} = 0.00107\left(1 - \frac{273.15 + 86}{364.76}\right)^{-0.6975} = 0.0197\ 1/℃$$

The maximal temperature to be used in the equation is 73 ºC according to Yaws (1995). Due to the lack of experimental data, an extrapolation of the Yaws equation has been performed.

The following physical properties are obtained from the API Technical Data Book (1997):

$d_{@\ 86\ °C} = 0.343$ and $c_{@\ 86\ °C} = 6833\ \frac{J}{kg\ K}$

**4th Point (42.8 bara & 86 ºC), boiling phase with vapor/liquid disengagement**

There is a period of time when the first bubbles of vapor arise during which it is necessary to relieve the mixtures of both phases simultaneously, either as flashing, bubble, slug, froth or mist flow until sufficient vapor space is generated inside the vessel to allow phase separation. API 521 (2014) points out that this period is usually neglected during sizing and selecting of the pressure relief device, with the exception of foamy fluids, reactive systems and narrow flow passages (for example, vessel jackets).

Thus, in this case the relieving load at full disengagement during the boiling phase is:

Fluid: propylene; Set pressure: 38 barg; Overpressure: 10%; Relieving pressure: 42.8 bara; Relieving temperature: 86 ºC, Latent heat at 86 ºC: 125 kJ/kg (API Technical Data Book, 1997)

$$Wet\ area(A_w) = \pi DL + 1.2\pi\frac{D^2}{4} = \pi \cdot 1.5 \cdot 4 + 1.2 \cdot \pi \cdot \frac{1.5^2}{4} = 20.97\ m^2$$

$$Q = 43200 \cdot F \cdot A_w^{0.82} = 43200 \cdot 1 \cdot 20.97^{0.82} = 523835 \text{ watts}$$

$$W = \frac{Q}{\lambda} = \frac{523835}{125000} = 4.19\frac{kg}{s} = 15086\frac{kg}{h}$$

The original specification sheet from the Engineering and Construction company which made the detail engineering gives the value of 15000 kg/h. These results match.

Notes

-No credit is given for the insulation because it is not fireproofing. Thus, F (environmental factor) = 1.0

-All the vessel is exposed to fire because it is within 7.6 m of the ground.

-The wetted area of the attached piping is not considered.

-The upper head of the vessel is not considered because there is no contact with the flames.

**5th Point (42.8 bara & 95ºC), supercritical propylene with retrograde condensation**

Once the boiling period is finished, a process begins consisting of the expansion of a vapor because of heating due to external fire. One particular and interesting point is the case in which there is supercritical propylene at the input of the valve and two phases at the outlet, due to retrograde condensation. For instance, the point 42.8 bara and 95 ºC, fulfils this conditions (see figure 6-14). The relieving load has been calculated with the methods in section 6.4, but will not be presented here as it has no relevance for the engineering analysis.

**6th Point (42.8 bara & 137ºC), supercritical propylene without retrograde condensation**

The relieving load has to be calculated with the methods presented in section 6.2 because the compressibility factor, Z = 0.73, is out of the range $0.8 < Z < 1.1$, but again, as before, it has no relevance for the engineering analysis.

**Inlet pressure drop calculation**

A scheme of the inlet piping of YS700/01-02 showing the configuration of the changeover valve and the nominal diameters of each section is presented in figure 6-17.

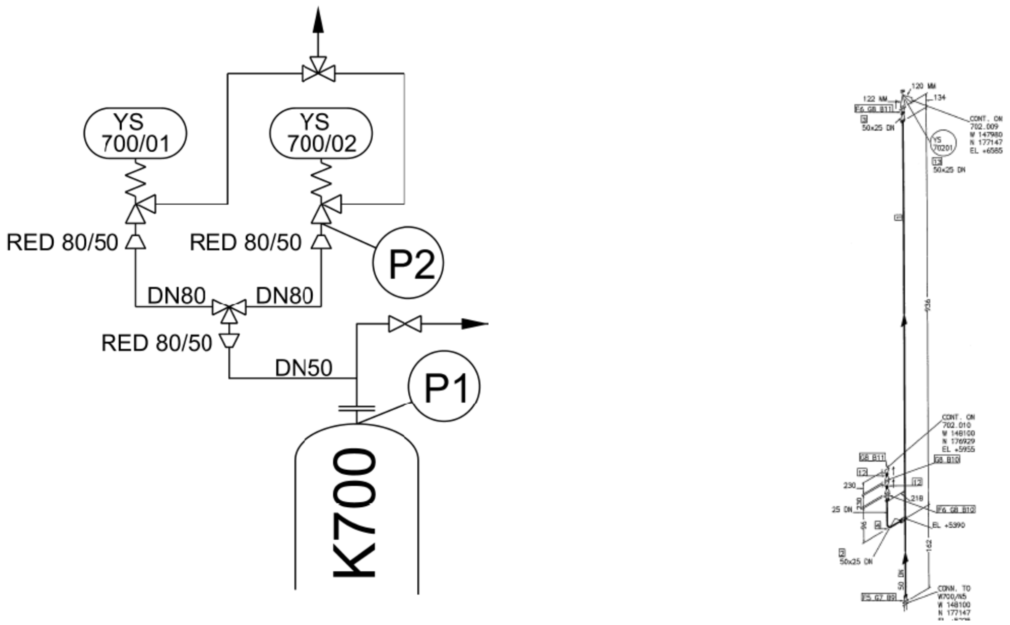



**Figure 6-17. Left side: scheme of the inlet piping configuration of YS700/01-02. Right side: isometric of the inlet pipe of the same valve.**

Design data: Set pressure: 38 barg, Relieving pressure: 42.813 bara; Relieving temperature: 86 ºC; Z = 0.5 (from Ingersoll-Rand, 1981); Gas viscosity = 0.0000103 kg/ms (from API Technical Data Book, 1997), Maximal flow = 21007 kg/h (from Leser datasheet).

*Calculation hypothesis*

- The inlet pressure drop will be calculated from the exit of K700 up to the inlet flange of the safety valve
- The reference diameter is DN 50 (inlet of the safety valve), which corresponds to an internal diameter of 54.5 mm.

*Equivalent length*

| **Pipe/Fitting** | $\boldsymbol{K_i}\left(=\frac{\boldsymbol{fL}}{\boldsymbol{D}}\right)$ | $\boldsymbol{\beta^4}=\left(\frac{\boldsymbol{d_1}}{\boldsymbol{d_2}}\right)^4$ | $\sum \boldsymbol{K_i}$ |
|---|---|---|---|
| Tower exit | 0.5 | $\left(\frac{54.5}{157.1}\right)^4 = 0.0145$ | 0.0072 |
| 0.37 m pipe ID 157.1 mm | 0.073 | 0.0145 | 0.0011 |
| Reduced T (Branch T + sudden contraction) | 0.75 (T ID 157.1 mm)<br>0.5 (157.1 x 54.5) | 0.0145<br>1 | 0.0109<br>0.5000 |
| 4.2 m pipe ID 54.5 mm | 2.3890 | 1 | 2.3890 |
| 7 elbows 90º | 0.28/each | 1 | 1.960 |
| 1 expansion 50x80 (54.5x 81.7 mm) θ = 30º | 0.6726 | 1 | 0.6726 |
| 1 change-over valve DN80 | 2 (from LESER) | $\left(\frac{54.5}{81.7}\right)^4 = 0.198$ | 0.396 |
| 1 reduction 80x50 (81.7 x 54.5 mm) θ = 30º | 0.199 | 1 | 0.199 |
| TOTAL K | | | 6.136 |

Calculation of the Reynolds Number

- Vapor density = $\frac{P \cdot MW}{ZRT} = \frac{42.24 \cdot 42}{0.5 \cdot 0.082 \cdot 359.15} = 120.5\ \frac{kg}{m^3}$

- Volumetric flow = $\frac{W}{\rho} = \frac{21007}{120.5 \cdot 3600} = 0.0484\ \frac{m^3}{s}$

- Velocity $= \frac{Q}{\pi\frac{D^2}{4}} = \frac{0.0484}{\pi\frac{0.0545^2}{4}} = 20.74\ \frac{m}{s}$

- Reynolds No. $= \frac{Dv\rho}{\mu} = \frac{0.0545 \cdot 20.74 \cdot 120.5}{0.0000103} = 13.2 \cdot 10^6$

The friction factor (Moody) for lightly corroded pipe

$\frac{\epsilon}{D} = \frac{0.3\ mm}{54.5\ mm} = 0.0055$ → $f = 0.031$ (from Crane, 1999)

Assuming isothermal flow and using the equation 31, pp 88 of API 521(2014)

$$\frac{fL}{D} = \frac{1}{Ma_2^2}\left[\left(\frac{P_1}{P_2}\right)^2 - 1\right] - \ln\left(\frac{P_1}{P_2}\right)^2$$

The Mach No. in the inlet of the safety valve DN50 is

$$Ma_2 = 3.23 \cdot 10^{-5}\left(\frac{q_m}{P_2 \cdot d^2}\right)\left(\frac{zT}{M}\right)^{0.5}$$

$$Ma_2 = 3.23 \cdot 10^{-5}\left(\frac{21007}{4281 \cdot 0.0545^2}\right)\left(\frac{0.5 \cdot 359.15}{42}\right)^{0.5} = 0.1103$$

Thus,

$$6.136 = \frac{1}{0.1103^2}\left[\left(\frac{P_1}{P_2}\right)^2 - 1\right] - \ln\left(\frac{P_1}{P_2}\right)^2$$

By trial and error $\frac{P_1}{P_2} = 1.0371$

$P_1 = 1.0371 \cdot 42.813 = 44.401$ bara

ΔP = 44.401 – 42.813 = 1.588 bar

$$\frac{\Delta P}{SP} = \frac{1.588}{38} \cdot 100 = 4.18\%$$

4.18 % > 3 % → Does not follow the 3% rule !!!

**Total backpressure calculation**

The safety valve YS 700/01-02 relieves to the flare network. The backpressure is calculated according to the design scenario for the flare network, which gives higher backpressure at the safety valve. In this case, as it has been shown in section 6.1 the design scenario is: fire in the reaction area.

Two calculations were performed, and the results are:

a) Isothermal calculation was performed with an independently developed spreadsheet in which the API 521 equations have been used (equation 31, 2014) considering maximal flow in the tail pipe (i.e. 21007 kg/h) and required in the header (i.e. 15000 kg/h), according to the recommendations of API 521 (paragraph 5.5.3, 2014) already explained in section 4.5. The result was:

   $P_{back} = 3.13\ barg$ that represents 8.2 % of the set pressure.

b) Adiabatic calculation considering two-phase flow and using the Beggs & Brill correlation available in the Aspen Flare System Analyzer v7.3. The result was:

$P_{back} = 3.49\ barg$ that represents 9.2 % of the set pressure. Other results were:

- Flow pattern : annular
- Mach No. (exit tail pipe): 0.244 which fulfills the condition < 0.7
- Momentum (exit tail pipe): $\rho \cdot v^2 = 15288\ \frac{kg}{ms^2}$

The flowsheet of the Aspen is presented in figure 6-18.

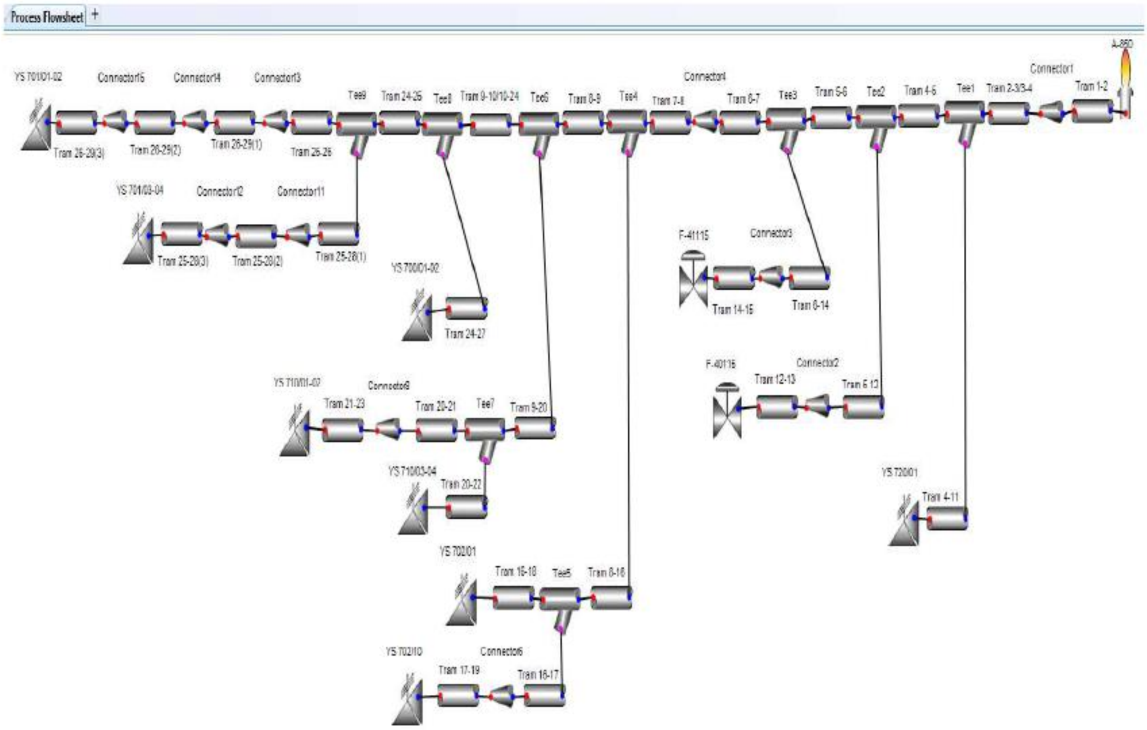



**Figure 6-18. Built-up backpressure calculation of YS700/01-02 using ASFA software.**

Both results were acceptable as the Leser valve can accept a total backpressure of 15%. However, the general concept that the isothermal calculation of the backpressure is conservative with respect to the adiabatic one (Bonilla, 1978), failed in this case.

### Acoustic Induced Vibration

Using Eisinger's (1997) mathematical expression (see section 4.8):

$$PWL = 10\log\left[\left(\frac{\Delta P}{P_1}\right)^{3.6} \cdot W^2 \cdot \left(\frac{T_1}{M}\right)^{1.2}\right] + 126.1$$

PWL is the sound power level in dB (with reference $10^{-12}$ W); $P_1$ is the upstream pressure of the source bara; ΔP is the reduction on pressure, bar; $T_1$ is the upstream temperature, K; W is the gas flow, kg/s; M molecular weight, kg/kmol

Given values

ΔP = 42.8 bara – 2.5 bara = 40.3 bar (Total backpressure assumed 2.5 bara as a conservative basis)

$P_1 = 41.8\ barg$

$$PWL = 10\log\left[\left(\frac{40.3}{41.8}\right)^{3.6} \cdot \left(\frac{21007}{3600}\right)^{2} \cdot \left(\frac{273.15+86}{42}\right)^{1.2}\right] + 126.1 = 152\ dB$$

The allowable PWL is calculated according to the equation:

$$PWL_{allow} = 173.6 - 0.125\frac{D_i}{t}$$

According to the piping specification

DN80 → $D_i$ = 82.5 mm, t = 3.2 mm

DN150 →$D_i$ = 160.3 mm, t = 4 mm

$$PWL_{allow\ DN80} = 173.6 - 0.125\ \frac{82.5}{3.2} = 170\ dB$$

$$PWL_{allow\ DN150} = 173.6 - 0.125\ \frac{160.3}{4} = 168\ dB$$

Thus, 168 dB > 152 dB. It is therefore concluded that the piping downstream of the safety valve (tail pipe) is considered safe from AIV fatigue failure.

**Stability of the valve**

The trim characteristic for the Leser valve (model 4564.6062) installed is (figure 6-19):

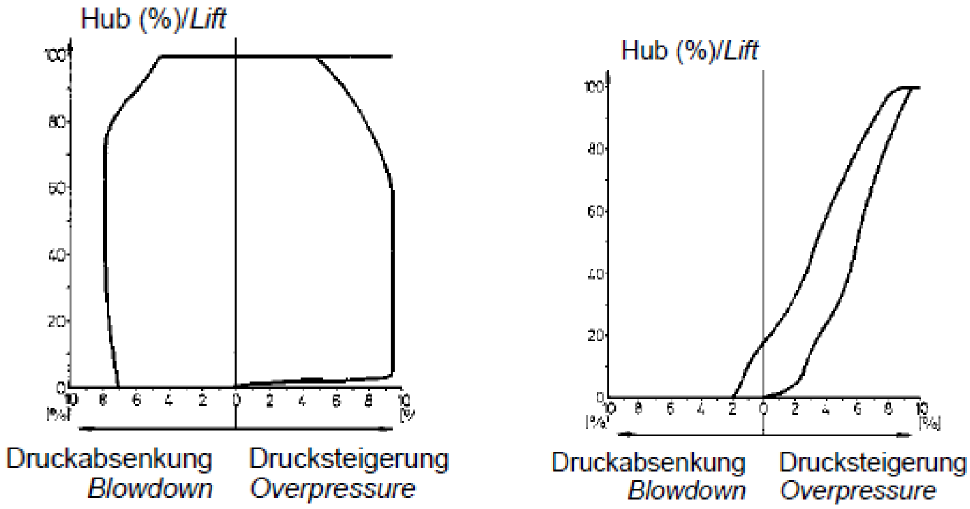



**Figure 6-19.Trim characteristic of the Leser valve model 4564.6062.The left figure corresponds to gas relieving and the figure of the right to a liquid.**

The lift is restricted at 4.5 mm. The total lift is 12.5 mm, which means a partial lift of 36%. The behavior of the pressure relief valve for gases is as represented for figure 6.19 left, i.e. after the overpressure increases until 9.5% the lift pops to the imposed limit of 36%. That is why the stability analysis is done at 100% of flow capacity, which corresponds to a maximal rated flow of 21007 kg/h of propylene gas.

At the beginning, when the valve relieves liquid propylene, the behavior is like figure 6.19 right.

*Inlet line length criteria (Smith et al., 2011)*

The mathematical expressions presented in section 4.6 have been used. First, Cremers et al. (2001, 2003):

$L < 111.5 \cdot t_0 \cdot \sqrt{\frac{kT}{MW}}$ ; $t_0 > \frac{2L}{c}$ and $c = 223\sqrt{\frac{kT}{MW}}$

where

L is the length of the inlet piping, ft (here 5.66 m); $t_0$ is the opening time of the valve, s; k is the isentropic expansion factor (for an ideal gas corresponds to Cp/Cv); T, is the temperature ºR; MW is the molecular weight of the fluid; c is the speed of sound, ft/s.

And $t_0 = \left[0.015 + 0.02\frac{\sqrt{2d_{PSVi}}}{\left(\frac{P_S}{P_{ATM}}\right)^{2/3}\left(1-\frac{P_{ATM}}{P_S}\right)^2}\right]\left(\frac{h}{h_{max}}\right)^{0.7}$

In this case $d_{PSVi} = 40\ mm\ = 1.575\ inches$, thus

$$t_0 = \left[0.015 + 0.02\frac{\sqrt{2 \cdot 1.575}}{\left(\frac{565.8}{14.696}\right)^{2/3}\left(1 - \frac{14.696}{565.8}\right)^2}\right]\left(\frac{4.5}{12.52}\right)^{0.7} = 0.009\ s$$

The full lift has a length = 0.313 · 40 = 12.52 mm, because has been restricted by the manufacturer to 4.5 mm

Thus $\frac{h}{h_{max}} = \frac{4.5}{12.52} = 0.359$

The speed of sound $c = 223\sqrt{\frac{1.13 \cdot 647}{42}} = 930\frac{ft}{s} = 283\ \frac{m}{s}$ and $k = \frac{C_p}{C_v} = \frac{17.5}{17.5-1.986} =$ 1.13 (from API Technical Data Book, 1997)

$$t_{wave(go\ \&\ return)} = \frac{2L}{c} = \frac{2 \cdot 5.66}{283} = 0.04\ s$$

According to API 520 (part II, Annex C, 2015) there is not an acoustic reflection point in the intersection of the branch (DN 150 and DN 50) because:

Area DN 50 = $\frac{\pi \cdot 0.0545^2}{4} = 0.00233\ m^2$

Area DN 150 = $\frac{\pi \cdot 0.1571^2}{4} = 0.0194\ m^2$, the conditions to be fulfilled for considering the intersection an acoustic reflection point are:

0.0194 < 10 ·0.00233 No check !

$L_{upstream}$= 0.4 m > 20 ·0.0545 No check !

Then, no acoustic reflection in the intersection of the branch of DN 50 with the Pipe DN 150 is expected.

Finally

$L < 111.5 \cdot t_0 \cdot \sqrt{\frac{kT}{MW}} = 111.5 \cdot 0.009 \cdot \sqrt{\frac{1.13 \cdot 647}{42}} = 4.19\ ft = 1.28\ m$ → Chattering !!!

*Inlet line length criteria (Frommann and Friedel, 1998, 2000) ΔP: 20%*

$$L_{100\%} < 9078 \frac{d_i^2}{w_{100\%}} (P_S - P_B) t_0$$

$w_{100\%} = 21007 \; \frac{kg}{h} = 46312 \; \frac{lb}{h}; \, P_S = 38 \cdot 14.5038 \; = 551 \, psig;$

$P_B = 3.13 \cdot 14.5038 \; = 45 \, psig$

$L_{100\%} < 9078 \frac{2.146^2}{46312} (551 - 45) \; 0.009 = 4.11 \, ft \; = 1.25 \, m$ → Chattering !!!

*Inlet line length criteria (Frommann and Friedel, 1998, 2000) ΔP: blowdown*

$$L < 45390 \; \frac{{d_i}^2}{w_{\%}} \left( \frac{P_S - P_{RC}}{P_S} \right) (P_S - P_B) t_0$$

The blowdown is 10% according to the Leser Manufaturer for this model. The built-up backpressure is 45 psig according to the backpressure calculations.

$L < 45390 \; \frac{2.146^2}{46312} (0.1)(551 - 45) \; 0.009 = 2.06 \, ft = 0.63 \, m$ → Chattering !!!

*Oversizing restrictions*

According to Smith et al. (2011) there are two constraints to be fulfilled to have chattering due to oversized pressure relief valves:

a) $w_{PSV} > 0.2 \cdot V_{system}(\rho_{set} - \rho_{shut}) + w_{required}$ where $w_{PSV}$ is the rated flow in lb/s; $V_{system}$ is the volume of the protected equipment available for gas expansion in ft$^3$; $\rho_{set}$ is the density of the fluid at set pressure conditions in lb/ft$^3$; $\rho_{shut}$ is the density of the fluid at blowdown conditions; $w_{required}$ is the required flow in lb/s.

   Thus,

   $\rho_{set}$ @ 38 barg and 86 ºC = 6.353 lb/ft$^3$ (http://webbook.nist.gov/chemistry/fluid/)
   $\rho_{shut}$ @ 34.2 barg and 86 ºC = 4.993 lb/ft$^3$ (http://webbook.nist.gov/chemisty/fluid)
   $V_{system}$ = Volume K700 – Volume internal catalyst = 476 ft$^3$ – 205 ft$^3$ = 271 ft$^3$
   Giving values
   $12.9 \; \frac{lb}{s} > 0.2 \cdot 271 \; \cdot (6.353 - 4.993) + 9.2 \; \frac{lb}{s} = 83 \; \frac{lb}{s}$ → Chattering !!!

b) $w_{required} > 0.25 \cdot w_{rated}$
   15000 kg/h > 0.25 ·21007 = 5252 kg/h OK ! → No chattering !!!

*Acoustic pressure losses*

According to Smith et al.(2011) the equation to be fulfilled to avoid chattering for this phenomenon is:

$$P_S - P_{RC} = P_S \cdot BD > \Delta P_{total} = \Delta P_{frictional} + \Delta P_{acoustic}$$

Where $P_S$ is the set pressure; $P_{RC}$ is the reseating pressure; BD is the blowdown.

In this case $L > \frac{ct}{2}$ because 5.66 m > $\frac{283 \cdot 0.009}{2} = 1.27 \, m$ and the following equation has to be used according to Darby (2013, 2014):

$$\Delta P_{acoustic} = \frac{c \cdot w}{A_i \cdot g_c} + \frac{w^2}{2 \cdot \rho_0 \cdot A_i{}^2 \cdot g_c} = \frac{930 \cdot 12.86}{\pi \frac{0.1788^2}{4} 32.2} + \frac{12.86^2}{2 \cdot 6.85 \left(\pi \frac{0.1788^2}{4}\right)^2 32.2}$$

$$\Delta P_{acoustic} = 14792 \frac{lb}{ft^2}$$

$\Delta P_{acoustic} = 14792 \frac{lb}{ft^2}$ =107 psi = 7.4 bar

$\Delta P_{friction} = 1.6\ bar$ (see inlet pressure drop in this section)

$\Delta P_{total}$ = 7.4 bar + 1.6 bar = 9.0 bar

Thus,

$P_S \cdot BD = 38 \cdot 0.10 = 3.8\ bar$ < 9 bar → Chattering !!!

*Body bowl choking*

The equation presented by D'Alessandro (2011) will be used:

$$P_0 < \frac{P_c}{(1+F_o)\frac{A_n}{A_e}\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} - F_o}$$

here

$P_0$ = relieving pressure = 42.813 bara = 620.9 psia; $A_n$ = throat area $= \frac{\pi d^2}{4} = \frac{\pi \cdot 40^2}{4} =$ $1256.6$ mm²; $A_e$ = outlet area = $5242.4$ mm² (DN 80, NP 16, ID 81.7 mm); k = 1.13 (ideal gas); $P_c$ = superimposed backpressure = 0.15 barg = 16.9 psia; $F_o$ = overpressure = 10%.

Giving values

$$620.9\ psia < \frac{16.9}{(1+0.1)\frac{1256.6}{5242.4}\left(\frac{2}{1.13+1}\right)^{\frac{1.13}{1.13-1}} - 0.1} = 322\ psia$$

620.9 psia < 322 psia No check ! → Possibility of secondary backpressure !!!

*API Simple Force Balance Method (Melhem, 2015)*

The last version of API 520 (part II, 2015) presents a method developed by Melhem. It is based on the following equation for conventional valves:

$$P_{source} - \Delta P_{f,wave} - \Delta P_{wave} - \Delta P_{back} - P_{close} > 0$$

$P_{source}$ = 38 ·1.1 = 41.8 barg = 606 psig

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

$$\tau = min\left(\frac{t_{wave}}{t_{valve}}, 1\right) = \left(\frac{0.04}{0.009}, 1\right) = 1$$

$\Delta P_{f,wave} = \tau^2 \Delta P_f = 1^2 \cdot 1.6\ bar = 1.6\ bar = 23\ psi$

$$\Delta P_{wave} = \frac{\tau c_0 \dot{M}}{A_p} + \frac{\tau^2 \dot{M}^2}{2\rho_0 {A_p}^2}$$

giving values

$$\Delta P_{wave} = \frac{1 \cdot 283 \cdot 5.835}{\pi \frac{0.0545^2}{4}} + \frac{1^2 \cdot 5.835^2}{2 \cdot 109.7 \cdot \left(\pi \frac{0.0545^2}{4}\right)^2} = 707855\ Pa + 28515\ Pa$$

$$\Delta P_{wave} = 736370\ Pa$$

$$\Delta P_{wave} = 736370\ Pa = 7.36\ bar = 107\ psi$$

$\Delta P_{back}$ = 13 bar = 45 psi

According to Leser documentation for 4564 model, the blowdown for this valve is 10% of the set pressure:

$$P_{close} = 38 - 0.1 \cdot 38 = 34.2\ bar = 496\ psi$$

finally,

606 psig - 23 psi - 107 psi - 45 psi - 496 psi > 0

- 65 psi > 0 No check → Chattering !!!

**Engineering Analysis Summary**

According to the engineering analysis procedure described in the first part of this section (see table 6.11), the following questions will be treated:

1) According to the inspection records is there any evidence of past chattering?
   **No**

2) Is the pressure relief valve well installed according to API 520, ISO 4126-9, etc.?
   **No, it does not follow the recommendation that the inlet pipe must be as short as possible and with a diameter larger than the inlet flange of the PRV.**

3) Is the inlet piping and fittings at least as large as the PRV inlet?
   **Yes**

4) Is there at least a 2% Set Pressure (SP) margin between PRV blowdown and the inlet pressure loss? **Yes**

   **SP = 38 barg**
   **Blowdown: 10% → 34.2 barg**
   **2% of 38 barg = 0.76 barg**
   **ΔP allowable for inlet pipe = 38 – (34.2 + 0.76) = 3.04 bar**
   **ΔP friction inlet pipe = 1.59 bar**
   **3.04 bar > 1.59 bar OK !**

5) Does excessive built-up backpressure occur according to the specific PRV?
   **No, the built-up back backpressure is 2.98 bar, thus**
   $\frac{2.98}{38} 100 = 7.8\ \% < 10\%$ **OK for a conventional valve!!!**

6) Is the time that the decompression wave goes back to the protected equipment and returns to the valve, less than the time required for the full opening of the valve? **See point 7.**

7) Does the PRV fulfill API 520 II-2015 Simple Force Balance?

   The results of the stability analysis are as following:

| **Parameter evaluated** | **Inlet line length, m** | **Inlet line length to avoid chatter, m** | **Fulfils the condition?** | **Will chatter?** |
|---|---|---|---|---|
| Inlet line length (Cremers et al., 2001, 2003) | 5.7 | 1.3 | No | Yes |
| Inlet line length (Frommann and Friedel, 1998) ΔP 20% | 5.7 | 1.3 | No | Yes |
| Inlet line length (Frommann and Friedel, 1998) ΔP blowdown | 5.7 | 0.6 | No | Yes |
| Required flow > 25% rated flow (oversizing) | | | Yes | No |
| Compressible vapors criteria (oversizing) | | | No | Yes |
| Total backpressure for a conventional valve < 10% SP | | | Yes | No |
| Body bowl choking | | | No | Unknown |
| Acoustic pressure losses | | | No | Yes |
| API Simple Force Balance (Melhem, 2014, 2015) | | | No | Yes |

8) Is the risk of relieving of the existing pressure relief valve quantified?
   **Yes, very low risk. It discharges to flare.**

9) Is a risk analysis done comparing an unsuccessful mechanical change in the field with the risk of chattering?
   **One solution is increasing the diameter of the inlet pipe. The risk is reduced to the removal of the existing inlet piping and installation of a new one. Acceptable risk.**

**Considerations**

- A new methodology for performing an engineering analysis for the pressure relief valves which have an inlet pressure drop greater than 3%, has been presented in this thesis, in an integrated way. It has been incorporated in a spreadsheet and can be used easily, especially the stability part
- Concerning the stability issue we have to continue knowing that there are many models but little experimental data to validate them. It is the author's opinion that the model of Melhem (2014, 2015) will be used in future, if and only if, the manufacturers can provide the exact value of the blowdown. As it was pointed out in section 3.3, the dynamic models (Hös et al., 2014, 2015, etc.) can only be applied if specific software is available and the detailed mechanical characteristics of the valve are known
- Another concern for the practicing safety engineer is getting the exact value of the blowdown, as mentioned before. This value is not guaranteed by US manufacturers and the European ones give a generic value for each series. Moreover, the shop test for safety valves does not usually have a system with capacity enough to perform a correct blowdown test.

## 6.8 Optimization of the revision interval of pressure relief valves

The procedure based on the API 580 (2009) and API 581 (2008) and described in section 4.11 will be applied here to the pressure relief valve YS702/01. This valve protects the shell of a heat exchanger for the case of fire. All the design parameters are already described in section 6.4.

### Design basis

The accepted level of risk for the petrochemical plants with level 2 contingency analysis, studied here is 4.6 $m^2$/year (see section 4-11).

### Probability of failure on demand

To find the initial inspection interval for the shell part of the heat exchanger, the different recommendations available in the open literature and given in section 3.5 will be presented in table 6-12. To remark that the conditions for evaluating the risk of shell rupture are: the fluid is clean propylene and the holdup of the shell of the heat exchanger is small.

**Table 6-12. Initial revision interval of YS 702-01 according to open literature.**

| Norm/guideline | Initial interval | Comments |
|---|---|---|
| Spanish RD 2060/2008 (ITC EP-3) | 12 years (Class 4) | Corresponds to level B of pressure vessels inspection |
| Institute of Petroleum IP-12 (1993) | 2 years | Grade 0; consequence: marginal; probability: low. |
| API 510 (2006) | 5 years | If the service is clean, the period could be increased to 10 years. |
| EEMUA-188 (2009) | 5 years | According to the risk evaluation |
| SAFed (2003) | 26 months | Clean service |
| One worldwide petrochemical company | 4 years | Clean service |

Because of this wide dispersion of recommended intervals, the actual period will be used: 5 years.

The results to obtain the modified characteristic life are as following:

| $\beta$ | $\eta_{def}$ | $F_c$ | MAWP (kPa) | $P_0$(kPa) | $F_{op,j}$ | $F_{env}$ | $\eta_{mod}$ |
|---|---|---|---|---|---|---|---|
| 1.8 | 37.875 | 0.75 | 4500 | 18000 | 0.8 | 1 | 22.725 |

The results to obtain the time duration of each inspection cycle is presented here taking into account the Confidence Factors (CF):

| Inspection effect | Inspection result | $CF_{pass}$ | $CF_{fail}$ | $T_{dur,i}$ (yr) |
|---|---|---|---|---|
| A | Pass | 0.9 | 0 | 5 |

And the updated characteristic life is:

| $P_{f,prior}$ | $P_{p,prior}$ | $P_{f,cond}$ | $P_{f,wgt}$ | $\eta_{upd}$ | $P_{fod}$ |
|---|---|---|---|---|---|
| 0.063 | 0.937 | 0.094 | 0.065 | 22.465 | 0.065 |

And finally, the damage factor and the probability of failure are:

| EF (event/yr) | DRRF | DR | MAWP (kPa) | $D_f$ | gfft | FMS | $P_f$ |
|---|---|---|---|---|---|---|---|
| 0.004 | 0.1 | 0.0004 | 4500 | 1 | 6.00E-07 | 0.11 | 6.61E-08 |

$P_{f,j}$ = 0.999 and finally

$$P_{f,j}^{prd} = P_{fod,j} \cdot DR_j \cdot P_{f,j} = 2.59 \cdot 10^{-5}$$

The probability of failure on demand is $P_{f,j} = 2.59 \cdot 10^{-5}$ $yr^{-1}$

**Consequence of failure on demand**

The event tree for rupture of the protected vessel, releasing propylene is showed in figure 6-20.

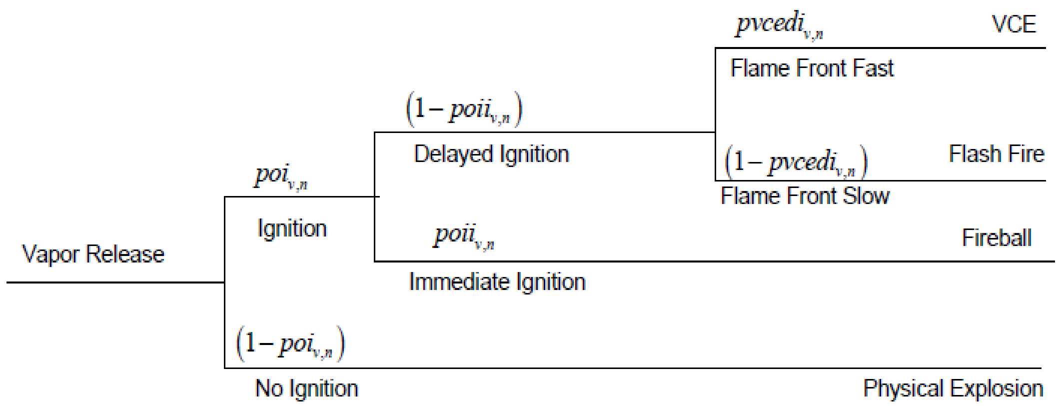



**Fig 6-20. Consequence analysis event tree for YS702-1 (vapor release case, from part 3, API 581).**

The results from probability outcome equations, are:

| pvce | pflash | pfball | pphys | total |
|---|---|---|---|---|
| 0.0321 | 0.0965 | 0.626 | 0.245 | 1 |

The consequence area results for each event outcome is:

| | VCE | Flash fire | Fireball | Physical explosion |
|---|---|---|---|---|
| Component damage, $m^2$ (<37.8 $kW/m^2$) | 229 | 1.2 | 10343 | 1292 |
| Personnel injury, $m^2$ (<12.6 $kW/m^2$) | 2973 | 4.6 | 37415 | 2650 |

The flammable consequence results are:

| $CA_{cmd,n}$ $m^2$ | $CA_{inj,n}$ $m^2$ |
|---|---|
| 6803 | 24184 |

and the final consequence area results are:

| gff,n | $CA_{cmd,n}$ $m^2$ | $CA_{inj,n}$ $m^2$ | CA $m^2$ |
|---|---|---|---|
| 6.00E-07 | 6803 | 24184 | 24184 |

**Risk calculation**

$Risk = P_{f,j} \cdot CA = 2.6 \cdot 10^{-5} \cdot 24184 = 0.626$ m$^2$/year

Thus

0.63 m$^2$/year < 4.6 m$^2$/year → it is possible to increase the revision period of 5 years

**Considerations**

- The API procedure gives in an objective way to establish the revision periods allowable for a pressure relief valve, according to an acceptable level of risk
- It has been demonstrated in this case that the valve can increase the revision period to more than the 5 years fixed at the beginning. This implies that in some countries the authorities may allow increasing the revision periods to 10 or more years (Holland, US, etc.), if the residual risks are acceptable.

## 6.9 Inspection methodology applied to the planned turnarounds

Concerning the inspection of safety valves, the new methodology developed in this thesis was applied to a programmed turnaround of the plants and conducted over two weeks in February 2015.

A total of 45 pressure relief valves were analyzed following the standard procedure explained here. A representative case study was elected: the pressure relief valve YS6510B, which protects the overpressure upset of a propane-propylene splitter. Figure 6-21 shows the valve YS6510B before removal.

**Figure 6-21. Pictures of YS6510B in the top of the column before removal.**

The steps to be followed according to the new methodology are:

**1st step: inspection of the valve and system piping during removal**

a) The valve tag must be inspected and verified. Equipment number, installation date, etc. must be included.
b) Immediately after removing the valve, the inlet and outlet nozzles and piping must be inspected for plugging and fouling. A specific template has to be used as a field check sheet, see figure 6-22. A Management of change procedure (section 3.6) has to be conducted in the case that the inlet or outlet nozzles or lines were more than 50% plugged. For valves that vent to atmosphere, the vent stack should have a 10 mm weep hole at a low point to prevent water accumulation. No fouling was present.

☐ 8 Hour
☐ 1 Day
☐ 3 Day
☐ Return to Stock

Required Return Date: ____________

# PSV Traveler Tag

Order #__________ REQ # __________

Company PO# ______________

PSV Serial # ______________

## PSV Removal Inspection

SAP Technical ID: ______________

Unit________ Location ____________

| | 0-10% | 10-25% | 25-50% | 50-75% | 75-100% |
|---|---|---|---|---|---|
| Inlet Nozzle Plugging & Fouling | ☐ | ☐ | ☐ | ☐ | ☐ |
| Inlet Pipe Plugging & Fouling | ☐ | ☐ | ☐ | ☐ | ☐ |

| | 0-10% | 10-25% | 25-50% | 50-75% | 75-100% |
|---|---|---|---|---|---|
| Outlet Nozzle Plugging & Fouling | ☐ | ☐ | ☐ | ☐ | ☐ |
| Outlet Pipe Plugging & Fouling | ☐ | ☐ | ☐ | ☐ | ☐ |

Service______________________

Decontaminated?______________

Unit Inspector ______________

Removal Date_________________

## PSV Installation Inspection

SAP Technical ID: ______________

Unit ________ Location ____________

| | |
|---|---|
| ☐ Set Pressure | ☐ Block Valves |
| ☐ Size | ☐ Tags |
| ☐ Weep Hole | ☐ Seals |
| ☐ Pipe Strain | ☐ Pipe Supports |
| ☐ Bolts | ☐ Gaskets |

Installation Date ______________

Unit Inspector ______________

Prod. Technician ______________

## PSV Traveler Tag Instructions:

### *PSV Removal*

This tag shall be attached to each PSV (Relief Valve) prior to removal or installation. This tag will remain with the PSV until reinstalled and/or tag is removed at Unit Inspector's discretion.

**Production** fills in required return date and shop turnaround time on tag prior to PSV removal. For PSVs replaced by a spare, mark the 'Return to stock' field.
Fill in the 'Order #' and either the Requisition # or the company PO#' field as well as the SAP Technical ID, Unit, Location, and Service fields.
Provide Permit & MSDS to Technician removing PSV

**Technician removing PSV (maint. or prod.)** notifies the unit inspector that a PSV is about to be removed, fills in thecompany PSV Serial # field, attaches tag & MSDS to the PSV & removes the PSV noting any abnormalities such as pipe strain, plugging, fouling, corrosion, etc. on the comments lines. Percent pluggage and/or fouling in the inlet and outlet of the nozzle and piping shall be noted where possible.

**The unit inspector (or designee)** inspects the PSV prior to decontamination, documents abnormalities in the comments section and signs and dates the tag in the space provided.

**Production** decontaminates the PSV and marks tag.

**Technician removing PSV** notifies shipping that the PSV is ready for pickup or takes the PSV to stores shipping area.

### *PSV Installation*

**Production** provides permit & MSDS to technician installing PSV and fills in 'SAP Technical ID, Unit, & Location fields.

**Technician installing PSV (maint. or prod.)** notifies the unit inspector that a PSV is about to be installed, inspects inlet and outlet lines for pluggage, installs the PSV, and documents any abnormalities such as pipe strain, on the Installation comments lines.

**The unit inspector (or designee)** inspects the PSV installation, documents results on the tag and signs and dates the tag.

**Production** verifies the PSV installation is correct and signs the tag when acceptable.

**The unit inspector (or designee)** removes the tag and updates the PSV records.

**Figure 6-22. Field check sheet used during removal of a safety valve for inspection.**

c) The valve must be decontaminated and prepared for shipping to the repair facility. If a valve cannot be verified as being decontaminated, a pre-test should not be performed unless the repair shop is equipped accordingly.

### 2nd step: pre-tests and inspection

Before a pressure relief valve is cleaned, disassembled or repaired, an "as received" inspection of the device condition and functionality must be performed. The steps are:

d) All the valves are visually inspected in the "as received" condition including deposits or corrosion which could block the inlet or outlet of the valve. Checking the bellows, noting any leakage from the bonnet vent. Verification that set pressure and blowdown adjustment seals are present. Condition of inlet and outlet flanges for pitting and other anomalies in the seating surfaces. Condition of the spring for corrosion or cracking and for correct pressure range at the valve's operating pressure and temperature. Position of set screws and openings in the bonnet. Condition of the external surfaces for any indication of a corrosive atmosphere or of mechanical damage. Verification of valve components and materials to match the information on the identification tag and registration card.

e) All valves, whether spring or pilot operated, receive a pre-test also known as a POP test. The following parameters are documented: Point at which simmering or an audible warning occurs; the inlet pressure where the valve pops open, noting also if the action was an acceptable pop or if the opening behavior was deficient in any manner; point at which the valve closes by reducing pressure noting if proper closure occurred and if any leakage is present after closure.

f) The pressure at which a valve pops open during the pre-test results in different actions as follows:
   - If the valve lifts inside the set point tolerance as defined by AD Merkblatt-A2 (2006) for instance, the test result is acceptable.
   - If the valve lifts outside the test point tolerance as defined by AD Merkblatt-A2, but within ± 10% of the set pressure, the test is repeated and the inspection engineer decides if the interval should be decreased
   - If the valve lifts at least than 90% of the set pressure the test is repeated and the inspection engineer decides if further evaluation is required
   - If the valve lifts a pressure greater than 110% of the set pressure, but less than 150% of set pressure, or if the valve inlet and/or outlet has more than 50% plugging or fouling, the Management of Change exercise must be done including a failure analysis
   - If the valve lifts at or greater than 150% of set pressure, or if the valve does not lift at 100% of the test stand's pressure the investigation and the MOC procedure shall begin within 48 hours of the pre-test failure.

g) A pre-test waiver may be obtained for valves which leak excessively in excess of test stand volume capability or if the valve inlet is completely plugged or the valve is installed in non-critical services such as thermal relief valve in cooling water services.

### 3rd step: disassembly and inspection

Following pre-test and inspection, relief valves are disassembled and the parts inspected, repaired and/or replaced as necessary.

The work performed is documented according to the valve inspection and minimum work requirements sheet (figure 6.23).

| ITEM | CONDITION RECEIVED | WORK PERFORMED |
|---|---|---|
| 1. Inlet | Check<br>• Flange surface or<br>• Inlet threads | • Remachine as needed. If dimensional tolerances are exceeded, replace valve<br>• Retap or remachine |
| 2. Outlet | Check<br>• Flange surface or<br>• Inlet threads | • Remachine as needed. If dimensional tolerances are exceeded, replace valve<br>• Retap or remachine |
| 3. Body | Check<br>• Wall thickness<br>• Bonnet or yoke gasket surfaces | • Clean<br>• Machine faces to remove any damage |
| 4. Lever | Check<br>• Cotter pin | Replace if necessary |
| 5. Cap | Check<br>• Threads<br>• Bent<br>• Corroded | Replace if damaged |
| 6. Floating Washers | | Replace |
| 7. Set Screws | Check<br>• Bent, broken, corroded<br>• Threads damaged | • Replace<br>• Replace |
| 8. Adjusting Screw | Check<br>• Upper washer bearing surface<br>• Threads<br>• Inside diameter | • Clean with fine emery cloth if dirty or slightly marked<br>• Replace if pitted or worn<br>• If worn, check mfg. allowable tolerance; replace if needed. |
| 9. Spindle | Check<br>• Cleanliness<br>• Straightness<br>• Pitting, Corrosion<br>• Integrity of threads<br>• Wear at contact with adjusting bolt | • Clean and polish<br>• Straighten within mfg tolerances or replace<br>• Lightly machine or replace<br>• Remove imperfections |
| 10. Spring | Check<br>• Cleanliness<br>• Overall height<br>• Parallelism<br>• Perpendicularity<br>• For Corrosion and Cracks<br>• Spring number verification | • Clean DO NOT BLAST CLEAN<br>• Replace if<br>• corroded<br>• cracked<br>• below free height tolerance<br>• out of square |
| 11. Spring Washers | Check<br>• Cleanliness<br>• Pitted or worn<br>• Plating | • Clean and polish<br>• Replace<br>• Replace |
| 12. Guide | Check<br>• Bore diameter<br>• Bore surface finish<br>• For scoring and/or galling of inside bore<br>• Thread conditions<br>• Bonnet / Body fit<br>• Gasket surfaces | • Clean and polish all surfaces<br>• Polish where needed to remove burrs<br>• Clean adjusting ring threads<br>• Replace if out of mfg tolerance.<br>• Replace if pitted or worn |
| 13. Disc Holder | Check<br>• Outside diameter for galling, scoring, pitting and the dimensions<br>• Spindle bearing surface for corrosion and pitting<br>• Face for steam cutting and damage<br>• Depth of counter bore<br>• Gasket surface (for bellows valves) | • Replace<br>• Replace<br>• Replace |

| ITEM | CONDITION RECEIVED | WORK PERFORMED |
|---|---|---|
| | | • Clean or remachine if necessary or replace |
| 14. Disc | Check<br>• For seat damage<br>• Overall height<br>• Seat step<br>• Bearing surface | • Machine surface if needed and if allowed by manufacturer otherwise replace |
| 15. Nozzle | Check<br>• For seat damage<br>• Seat dimensions<br>• OD<br>• ID<br>• Seat step<br>• Overall height<br>• Nozzle ring threads and guiding diameter | • Machine out damage if this does not exceed mfg. tolerances, otherwise replace |
| 16. Ring pins | Check<br>• Bent, broken or corroded<br>• Threads damaged | Replace if damaged in any way |
| 17. Guide ring | Check<br>• ID for galling<br>• Thread integrity<br>• Face for damage such as steam cutting<br>• Angles and corners on top surfaces<br>• Broken or missing notches | Replace if any damage is found |
| 18. Nozzle ring | Check<br>• ID for galling<br>• Thread integrity<br>• Face for damage such as steam cutting<br>• Angles and corners on top surfaces<br>• Broken or missing notches | Replace if any damage is found |
| 19. Soft parts | Check<br>• Condition | • Replace all soft goods |

**Figure 6-23. Valve inspection and minimum work requirements sheet.**

### 4th step: final inspection and testing

Following reassembly of the valve, it is set and tested prior to shipment. A certified test stand shall be used with pressure gauges which have been calibrated. The valves in vapor service may be tested on either air, nitrogen or steam. Valves in liquid service are tested on water. For stainless steel valves, the maximum allowable chloride content is 50 ppm. The steps to be followed are:

h) Leakage tests will be performed at no less than 90% of the set pressure and no more than 95% of the set pressure according to API 527 criteria.
i) A bellows integrity test is performed at a pressure not less than 1.4 barg on the outlet side to check for leakage from the bonnet vent.
j) The blowdown ring of the valve will be set to meet the requirements of the manufacturer.
k) All adjustments access covers are sealed with stainless steel wire looped through holes in the valve body and sealed with a lead seal.
l) A tag indicating the date of testing and inspection should be permanently attached next to the manufacturer’s code nameplate.

### 5th step: installation inspection

An installation inspection is performed prior to the valve being approved for return to service: The work consists of:

m) The location which the valve has been installed is verified by operators.
n) Presence of all required accessories are verified, including lifting levers, rupture discs, vent piping, weep hole on atmospheric vents, pilot operated valves connected properly, etc.

*Inspection interval for new valves*

The initial interval is determined according to the chemical severity of the product (see table 6-13). In this case the propylene was assigned as “clean”.

**Table 6-13. Chemical severity of products.**

| CHEMICAL SEVERITY | CHARACTERISTICS | EXAMPLES |
|---|---|---|
| Very Clean | A service where the process stream has zero chance of polymerizing causing pluggage on the inlet or outlet of the relief device | • Nitrogen<br>• Dry breathing air<br>• Dry instrument air<br>• Carbon monoxide |
| Clean | A service where the process stream is highly unlikely to polymerize causing pluggage on the inlet or outlet of the relief device | • Fuel gas<br>• Process air<br>• Non-treated air (not dry air)<br>• Various hydrocarbons such as ethylene or other clean, filtered hydrocarbon products at moderate temperatures |
| Average | A service which is not considered “Very Clean” or “Clean” but exhibits little chance of pluggage. Some minor fouling of surfaces might occur inside the relief device. | • Some hydrocarbons with small amounts of particulate matter<br>• Potential presence of separate aqueous phase<br>• Service temperatures up to 260° C |
| Dirty | A service which has exhibited polymerization under certain circumstances which are not part of normal operation, or a service which is somewhat corrosive in nature. | • 10% or less acid or caustic concentrations.<br>• Some hydrocarbons with particulates in varying amounts<br>• Service temperatures over 260° C |
| Very Dirty | A service which has exhibited polymerization causing pluggage during normal operations, or a service which is highly corrosive in nature. | • Butadiene<br>• Over 10% acid or caustic concentrations.<br>• High content of sulfur and/or chlorides |

And the interval is fixed by table 6-14. In this case, a “clean” product gives 48 months (4 years).

**Table 6-14. Inspection interval according to the severity of the products.**

| Chemical Severity | Months |
|---|---|
| Very Clean | 60 |
| Clean | 48 |
| Average | 38 |
| Dirty | 26 |
| Very Dirty | 14 |

*Inspection and test interval of existing valves*

The existing interval is compared to the interval assigned in the case of when the valve is new. When the existing interval is longer and there is no evidence of mechanical damage, the existing interval is adopted as a basis for analysis. But when the existing interval is shorter than the assigned, the shorter will be adopted for the analysis if and only if: a) adequate documented inspection data for a minimum of two test cycles exist b) there is no evidence of mechanical damage to the valve c) management approval is obtained through the Management of Change process.

### Case study SV6510B

The following sheet (figure 6-24) was prepared for the safety valves shop

| | | | |
|---|---|---|---|
| Inspection Date: | February 6, 2015 | Equipment: | Distillation column |
| SAP Order No.: | | Service: | Splitter C3/C3= |
| Unit: | | Valve Tag No.: | SV6510B |
| Date of Last Inspection: | February 28, 2012 | Requested by: | Inspection Department |
| Name of Repair/Testing Shop: | | Static mechanical shop | |
| Testing and Repairs performed by (print name): | | | |
| Company Inspector (print name): | | | |
| Valve Specification No.: | | | |
| Size: | 100 x 150 | Model: | VSE-2 |
| MFG: | Sempell | Serial No.: | |
| Spring No.: | | Valve Type: | Conventional |
| Valve Capacity: | | Capacity Units: | |
| Operating Temp (°C): | | Set Pressure (barg): | 20 |
| Back Pressure (barg): | | Vacuum Set (barg): | -- |
| Vacuum Set: | | Inlet Size: | 100 |
| Inlet Rating: | PN40 | Inlet Type: | Flanges |
| Outlet Size: | 150 | Outlet Rating: | PN25 |
| Outlet Type: | Flanges | Installed with Rupture Disk? (Y/N) | No |
| Rupture Disk Lot No.: | -- | Rupture Disk Set Pressure (psi) | -- |
| Rupture Disk Set Temperature (°C): | -- | | |
| Comments: | | | |
| The valve had from its installation (July 26, 1991) an assigned revision interval of 60 months. However, due to leakage problems on February 28, 2012, the interval was decreased to 3 years | | | |

**Figure 6-24. Relief valve shop inspection testing and repair report for YS6510B.**

And the shop prepared the following documentation (figure 6-26)

| | | | |
|---|---|---|---|
| Was valve upright and on skid when received? (Y/N) | Yes | Were flanges protected with flange covers? (Y/N) | No |
| **AS-RECEIVED PRE-TEST** | | | |
| Pressure of first simmer (psi) | Not available | POP Pressure (barg) | 20 |
| Pressure at valve closure (barg) | Not available | Set Pressure on TAG (barg) | 20 |
| If Pre-Test Not Performed, Explain: | | | |
| The Pre-test was performed. No deposits or corrosion present in the inlet and outlet.<br>The disc and the seating surfaces have some deposits. Cleaning, machine and lap had been applied. | | | |

| **SEATS** | | **STEMS & GUIDES** | | **SPRING** | | **INLET** | | **OUTLET** | |
|---|---|---|---|---|---|---|---|---|---|
| | GOOD | X | GOOD | | GOOD | X | GOOD | X | GOOD |
| X | FOULED | | FOULED | | FOULED | | FOULED | | FOULED |
| X | CORRODED | | CORRODED | X | CORRODED | | CORRODED | | CORRODED |
| | ERODED | | ERODED | | ERODED | | ERODED | | ERODED |
| | CRACKED | | CRACKED | | CRACKED | | CRACKED | | CRACKED |

**Figure 6-25. As-received valve inspection for YS6510B.**

And finally the shop reported the results of the test using the final test sheet (figure 6-26).

| | | |
|---|---|---|
| Set Pressure (barg): | 20 | |
| Vacuum Pressure: | | |
| Cold Differential Set (barg): | 20 | |
| Bubble Tight at (barg): | | |
| Blowdown: | No blowdown ring available | |
| Tightness B/P Test: | | |
| Nozzle Ring Set | | |
| Notches _________________ | UP | DOWN |
| Guide Ring Set | | |
| Notches _________________ | UP | DOWN |
| Additional Comments | | |
| The test was performed successfully. | | |
| TESTED BY | WITNESSED BY | |
| | Josep Basco | |

**Figure 6-26. Final test sheet for YS6510B.**

Some pictures of the process are presented here:


**Figure 6-27. Picture of 6510B disassembled (light corrosion on the disc and on the spring can be appreciated).**


**Figure 6-28. Picture of the disc of 6510B. Detail of deposits and corrosion on the disc.**

**Figure 6-29. Picture of the seating surface of 6510B. It is in good condition after machine and leap.**


**Figure 6-30. Picture of the disc of 6510B after lapping process.**

**Figure 6-31. Picture of the assembling process of YS6510.**


**Figure 6-32. Picture of YS3925 ready to do the test (see the bottle with water to count the bubbles).**

The results of the inspection of the 45 pressure relief valves can be seen in Table 6-15. However, the valves which had a prepop test were 32.

**Table 6-15. Results of inspection (Turnaround of February, 2015)**

| Item | SP, barg | Size | Product | Pre-test barg | Deviation % | Comments |
|---|---|---|---|---|---|---|
| 3085 | 8 | 20x25 | propylene | 13 | 163 | A management of change was open. Disc and seating surface were lap |
| 3027 | 40 | 20x25 | propylene | 50 | 125 | Disc and seating surface were lap |
| 3028 | 40 | 20x25 | propylene | 42 | 105 | Disc and seating surface were lap |
| 3082 | 10 | 15x25 | cooling water | 30 | 300 | A management of change was open Disc and seating surface were lap |
| 3083 | 52 | 15x25 | nitrogen | 50 | 96 | Disc and seating surface were lap |
| 6516 | 20 | 20x25 | propylene | 18 | 90 | Disc and seating surface were lap |
| 6517 | 20 | 20x25 | propylene | 22 | 110 | Disc and seating surface were lap |
| 6518 | 20 | 20x25 | propylene | 21 | 105 | Disc and seating surface were lap |
| 6519 | 20 | 20x25 | propylene | 19.1 | 95.5 | Disc and seating surface were lap |
| 860-07 | 7 | | propylene | 2.8 | 40 | Disc and seating surface were lap |
| 493-01 | 8 | 25x25 | water | 8.8 | 110 | Disc and seating surface were lap |
| 412-08 | 8 | 25x40 | mineral oil | 8.8 | 110 | Disc and seating surface were lap |
| 413-03 | 7.2 | 25x25 | water | 5.4 | 75 | Disc and seating surface were lap |
| 870-15 | 1 | | water | 0.9 | 90 | Disc and seating surface were lap |
| 882-01 | 8 | 40x50 | steam | 4 | 50 | Disc and seating surface were lap |
| 420-04 | 6 | 25x25 | propylene | 3 | 50 | Disc and seating surface were lap |
| 42-04 | 30 | 25x25 | propylene | -- | -- | Disc and seating surface were lap |
| 115 | 55 | 25x50 | mineral oil | 57 | 104 | Disc and seating surface were lap |
| 117 | 10 | 15x25 | cooling water | 10.5 | 105 | Disc and seating surface were lap |
| 657 | 20 | 25x25 | propylene | 26.5 | 133 | A management of change was open Disc and seating surface were lap |
| 3023 | 10 | 150x250 | water | 10 | 100 | Disc and seating surface were lap |
| 3913 | 6 | 15x25 | cooling water | 6 | 100 | Disc and seating surface were lap |
| 3925 | 24 | 25x40 | propylene | 24 | 100 | Disc and seating surface were lap |
| 6510B | 20 | 100x150 | propylene | 20 | 100 | Disc and seating surface were lap |
| 6511 | 8 | 20x25 | cooling water | 8 | 100 | Disc and seating surface were lap |
| 551 | 6 | 15x25 | cooling water | 6 | 100 | Disc and seating surface were lap |
| 821 | 4 | 15x25 | air | -- | -- | Disc and seating surface were lap |
| 822 | 4 | 15x25 | air | -- | -- | Disc and seating surface were lap |
| 845 | 4 | 25x25 | air | -- | -- | Disc and seating surface were lap |
| 810 | 4 | 50x80 | nitrogen | -- | -- | Disc and seating surface were lap |
| 491-01 | 8 | 80x125 | propylene | 8 | 100 | Disc and seating surface were lap |

| Item | SP, barg | Size | Product | Pre-test barg | Deviation % | Comments |
|---|---|---|---|---|---|---|
| 491-02 | 19 | 50x80 | propylene | 19 | 100 | Disc and seating surface were lap |
| 42-26 | 34 | 25x25 | propylene | -- | -- | Disc and seating surface were lap |
| 131 | 6 | 25x25 | nitrogen | -- | -- | Disc and seating surface were lap |
| 149 | 9.5 | 25x25 | mineral oil | -- | -- | Disc and seating surface were lap |
| 3236 | 0.75 | 200x200 | nitrogen | | | Disc and seating surface were lap |
| 490-02 | 6 | 50x250 | propylene | 6 | 100 | Disc and seating surface were lap |
| 861-14 | 30 | 15 | propylene | 30 | 100 | Disc and seating surface were lap |
| 549 | 6 | 15x25 | water | 1 | 17 | A management of change was open<br>Disc and seating surface were lap |
| 07 | 45 | 15x25 | propylene | -- | -- | Disc and seating surface were lap |
| 1125 | 59.5 | 15x25 | propylene | -- | -- | Disc and seating surface were lap |
| 1126 | 59.5 | 15x25 | propylene | -- | -- | Disc and seating surface were lap |
| 3230 | 5 | 15x25 | nitrogen | 5 | 100 | Disc and seating surface were lap |
| 3232 | 6 | 15x25 | cooling water | 2 | 33 | Disc and seating surface were lap |
| 548 | 6 | 15x25 | cooling water | 0 | 0 | A management of change was open<br>Disc and seating surface were lap |

## Considerations

- A new procedure has been presented to do the inspection process in a methodological way. The presentation here has been paper based but it is already being implemented in a relational database. The information collected during this process is very important in order to improve the revision periods in a more objective way
- The author knows that during a turnaround time pressure is hard and the maintenance engineer does not want waste time completing data sheets. However, the documentation has to be prepared <u>before</u> the shutdown of the plant, not during it, and the use of forms or checklists should facilitate documentation of inspection results
- A major concern is the handling of valves. Usually this work is done by temporary contractors, who do not know how to handle a pressure relief valve. The inspection engineer must control this process right from the beginning
- Some statistical data:
    1. 3 valves opened at a pressure higher than 110%, that means 9% (Smith (1995), reported 14%)
    2. 6 valves opened before 90% of the set pressure, that means 19 %
    3. The valve 3082 in a cooling water circuit opened at 30 barg, when the set pressure was 10 barg. The inspection period has been modified
    4. The valve 3232 opened at 2 barg, when the set pressure is 6 barg and the operating pressure of the cooling water system is 4 barg, that means a leakage. The inspection period has been modified as well
- Lapping is performed in the disc and in the surface seat as standard treatment for all valves. But according to the figure 6-23 it should be done only if necessary. However, this proposal was not accepted by the inspection department in this turnaround
- A holistic view of table 6-15 suggests there are some bigger issues related to selection and installation (presumably materials) in relief valves used in cooling water services.

# Chapter 7. Conclusions and future research

The work performed in this thesis has allowed drawing of the following summarized conclusions:

**1**. A new methodology has been developed to improve the reliability of pressure relief valves, which uses a step by step approach based on the nine phases of valves engineering process and applied to their whole life cycle. By using it, the probability of errors or latent failures decreases significantly.

**2**. In the framework of this methodology, a series of systematic tools have been designed:

- a list of parameters to be evaluated in the basic and detail phase of valve design through a new safety valve requirement specification
- a list of documentation to be provided for each PRV concerning all its life cycle
- a pre-startup safety review checklist
- a checklist for the verification during the engineering process
- a new procedure for the technical audit phase.

These tools improve significantly the whole process of pressure relief valve engineering, increasing both its reliability and traceability.

**3**. The analysis of the diverse methods proposed for calculating the required relief load and the valve area has allowed the identification of the most adequate ones:

Required relief load:

- fire on a vapor/gas filled vessels: Oudekirk (2002) and Self and Do (2010)
- fire on liquid filled vessels: a dynamic method carefully analyzing the involved phenomena (liquid expansion, boiling or formation of a supercritical fluid, vapor/gas thermal expansion)
- heat exchanger tube breakage: models in PS PPM software, Schmidt (2010) methodology
- distillation columns: Sengupta and Staats (1978) unbalanced heat load model.

Required area:

- gas/vapor: assume isentropic flash from upstream pressure to choke pressure/back pressure. However, as this is a tedious process, the recommendation is to use the conventional API equation with specific considerations according to the value of Z
- two-phase flow: the direct integration method if a process simulator is available; otherwise, use the Schmidt (2013) method
- supercritical fluids: the methods of Self and Do (2010) or API 520 (part I, 2014) are recommended.
- liquids: API 520 (part I, 2014) or DIN EN ISO 4126-1 (2004) sizing equations.

**4**. A new method of performing an “engineering analysis” according to API 520 (part II, 2015) for the pressure relief valves which have an inlet pressure drop > 3% of the set pressure has been developed.

**5**. A specific risk matrix has been designed to improve the methodology for assigning the revision intervals of PRVs though a quantitative approach. This matrix allows the classification of each safety valve according to the risk generated in case of failure to open on demand.

**6.** A new procedure has been developed to improve the revision of a PRV in a turnaround. It follows all the steps from removal, transporting it to the workshop, prepop, revision and reinstallation (this procedure has not been completely adopted in the studied plants, as it is a decision at a corporative level).

**7**. The new methodology has been applied to three petrochemical plants that have 503 PRVs, with no incidents related to these valves so far. The following conclusions were reached:

Some management people from these plants found that the paperwork increased too much and the author had to cope with some initial resistance. Two historical prejudices were detected concerning PRVs: lack of complexity, except for pilot operated relief valves, and the fact that they rarely give problems to the maintenance crew. Thus, it has been perceived that operators associate the function of a PRV more to a gate valve rather than to a control valve.

There is a lack of knowledge of relief systems among the process engineers. The author found very few experts on relief systems in European companies.

Even though there are very good references for studying this chemical process safety area, the design of relief systems is still more an art than a science; a significant example is the calculation of the required load for each contingency.

The author firmly believes that the methodology presented in this thesis will contribute to a better reliability of PRVs during their whole life cycle. However, the success of implementing it in the refining/petrochemical/chemical companies needs, as a first step, the strong support of the management and a good knowledge (preferably by official certification) of relief system by the specialist engineers.

**Future research**

The author had to cope with knowledge barriers in developing this thesis. Therefore, the research and study of the following topics would be of high interest:

- the correct value of the discharge coefficient for two-phase flow, not only in pressure relief valves but also in restriction orifices and nozzles, be identified on the basis of experimental testing and comparison of actual mass flux to theoretical mass flux beyond the limited experimentation to-date with water.
- a robust and reliable model for the prediction of the instability of pressure relief valves for gases/vapors and liquids
- identification of the influence of the body bowl choking in the stability of the safety valves
- estimation of the two-phase density where slip between phases is involved. The local density where slip is involved depends on the slip model selected to calculate the slip ratio.

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# Annexes

## Annex A: Standards and guidelines

The publications addressing design, installation and operation of pressure relieving systems fall into four major categories: Codes, Laws, Standards and Practices. There is a considerable number of such publications around the world, and many are sections of other codes related to boilers or pressure vessels. The definitions are:

Codes are a set of rules developed by various private or non-governmental organizations to define basic design requirements for pressurized and non-pressurized equipment or for the handling and storage of hazardous material. Examples are Section I and VIII of the ASME Code or the European Pressure Equipment Directive (PED 97/23/EC). These Codes are not mandatory in themselves, but are often adopted by industry as standards which represent prudent and responsible levels of practice and, as such, may become de facto requirements. For instance, in US many local jurisdictions may make adherence to part or all of a recognized Code mandatory by law.

Laws governing the design of pressure relief systems are enacted by national or local governments to maintain minimum standards of safety. These laws may adopt recognized Codes and practices as mandatory requirements for that jurisdiction.

Standards are compilations of rules and requirements associated with the design or purchase of equipment, materials, services, etc. Standards are produced by individual companies to define minimum requirements, or by professional or trade organizations to aid their members in specifying requirements for various items. As with Codes, Standards are generally voluntary, but laws may adopt an industry or trade association standard.

Practices are compilations of recommended design procedures, typically prepared and published by members of a given industry or used within that industry. While published practices are not usually adopted as legal requirements by any jurisdiction, they can often supplement mandatory regulations by offering guidance in areas not specifically addressed by Codes and laws. Many API publications are recommended practices.

Since 2002, the two major worldwide Codes are ASME and the PED 97/23/EC. The PED supersedes local codes in all European member states, codes such as BS (United Kingdom), ISPESL (Italy), TÜV (Germany), Stoomwezen (The Netherlands), UDT (Poland), AFNOR (France), etc. Compliance with PED allows the manufacturer to CE mark their product as required by the European Union and is an assurance that the product is manufactured in accordance with the law.

According to Hellemans (2006), the ASME and PED codes cover about 80% of all worldwide requirements. Exceptions are, however, China and India because they have their own guidelines, although these are more focused on boiler applications.

**USA and influence areas**

<u>API (American Petroleum Institute)</u>

API Standard 520 Sizing, election and Installation of Pressure-relieving Devices in Refineries, Part I-Sizing and Selection, 9th Edition, July 2014.

API Standard 520 Sizing, election and Installation of Pressure-relieving Devices in Refineries, Part II-Installation, 6th Edition, March 2015.

API Standard 521 Pressure-relieving and Depressuring Systems, 6th Edition, January 2014.

API Standard 526 Flanged Steel Pressure-relief Valves, 6th Edition, May 2009.

API Standard 527 Seat Tightness of Pressure Relief Valves, 3rd Edition, July 2007.

API Recommended Practice 576 Inspection of Pressure-relieving Devices, 3rd Edition, November 2009.

API Recommended Practice 580 Risk-Based Inspection, 2nd Edition, November 2009.

API Recommended Practice 581 Risk-Based Inspection Technology, 2nd Edition, September 2008.

API 510 Pressure Vessel Inspection Code: In-Service Inspection, Rating, Repair, and Alteration, 9th Edition, June 2006.

API Standard 2000 Venting Atmospheric and Low-pressure Storage Tanks, 6th Edition, November 2009.

ASME International

ASME Boiler and Pressure Vessel Code, Section I, Power Boilers and Section VIII, Unfired Pressure Vessels, American Society of Mechanical Engineers, New York.

ASME PTC 25.3-1976 Performance Test Code - Safety and Relief Valves, American Society of Mechanical Engineers, New York.

ASME Code for Pressure Piping B31.1-2010, Appendix II: Rules for the design of safety valve installations

Others

NFPA 30 - Flammable and Combustible Liquids Code, National Fire Protection Association, Quincy, MA.

NFPA 58 – Liquefied Petroleum Gases – Storage and Use, National Fire Protection Association, Quincy, MA.

ANSI/ISA-S84.01-1996 - Application of Safety Instrumented Systems for the Process Industries, ISA Research Triangle Park, North Carolina.

**Europe**

Directive 97/23/EC (PED)

EN ISO 4126-1: 2004, Safety devices for protection against excessive pressure – Part 1: Safety valves.

EN ISO 4126-2: 2003, Safety devices for protection against excessive pressure – Part 2: Bursting disc safety devices.

EN ISO 4126-3: 2006, Safety devices for protection against excessive pressure – Part 3: Safety valves and bursting disc safety devices in combination.

EN ISO 4126-4: 2004, Safety devices for protection against excessive pressure – Part 4: Pilot-operated safety valves.

EN ISO 4126-5: 2004, Safety devices for protection against excessive pressure – Part 5: Controlled safety pressure relief systems (CSPRS).

EN ISO 4126-6: 2004, Safety devices for protection against excessive pressure – Part 6: Application, selection and installation of bursting disc safety devices.

EN ISO 4126-7: 2004, Safety devices for protection against excessive pressure – Part 7: Common data.

EN ISO 4126-9: 2008, Safety devices for protection against excessive pressure – Part 9: Application and installation of safety devices excluding stand-alone bursting disc safety devices.

EN ISO 4126-10: 2010, Safety devices for protection against excessive pressure – Part 10: Sizing of safety valves for gas/liquid two-phase flow.

Others

IEC 61508 Functional safety of electrical/electronic/programmable electronic safety-related systems, Part 1-7, Geneva, Switzerland.

IEC 61511Functional safety: Safety Instrumented Systems for the process industry sector, Part 1-3, Geneva, Switzerland.

AD 2000- Merkblatt A2, Safety devices against excess pressure – Safety Valves

# Annex B: Pre start-up safety review checklist for the new methodology

| RELIEF SYSTEMS<br>Pre Start-up Safety Review Checklist | Prio. = 1: Elevated Risk<br>Prio. = 2: Increased Risk<br>Prio. = 3: Medium Risk |
|---|---|

| | | Check | | Status | Ok | | Action | Prio. | Resp. |
|---|---|---|---|---|---|---|---|---|---|
| | | on-site | docu | | yes | no | | | |
| | **Preinstallation, handling and testing** | | | | | | | | |
| 1 | Is the PRV located where there are pressure fluctuations (e.g. close to control valves, orifice plates, elbows, discharge of displacement pump or compressor)? | X | | | | | | | |
| 2 | Is the PRV located at least 10 or more pipe diameters from any device that causes turbulence? | X | | | | | | | |
| 3 | Has the join between inlet piping and main piping a well-rounded entry branch connection? | X | | | | | | | |
| 4 | Is the PRV located in a vibration area? | X | | | | | | | |
| 5 | Is the PRV well supported? | X | | | | | | | |
| 6 | Is the PRV located in the cleaner portion of the process? | X | X | | | | | | |
| 7 | Is the inlet and discharge piping free-draining (no pockets)? | X | | | | | | | |
| 8 | Is the PRV installed in a location that facilitates access and maintenance? | X | | | | | | | |
| 9 | In case of a Balanced PRV, is the bonnet vented? | X | | | | | | | |
| 10 | Is the PRV stored indoors and are the inlet and outlet flanges closed off? | X | | | | | | | |
| 11 | Is the PRV tested before installation to confirm its opening pressure setting? | X | X | | | | | | |
| 12 | Is the workshop, where testing will be performed, clean? | X | | | | | | | |
| 13 | In case of equipment hydrotesting, is the PRV removed or isolated? | X | | | | | | | |
| 14 | Is the PRV mounted in a vertical, upright position? | X | | | | | | | |
| 15 | Is the bonnet shipping plug removed before installation? | X | | | | | | | |
| 16 | Is the PRV transported fixed in a wooden box and in a vertical position? | X | X | | | | | | |
| 17 | Is manufacturer's operation, maintenance and start-up manual available for each type of valve? | | X | | | | | | |
| 18 | Before installation, are the flanges surfaces cleaned and without damages or scratches? | X | | | | | | | |
| 19 | Are the correct flange gaskets used? | X | X | | | | | | |
| 20 | Do the gaskets obstruct the inlet or outlet passage? | X | | | | | | | |
| 21 | Has the installation been purged before installing the PRVs? | X | | | | | | | |

# RELIEF SYSTEMS

## Pre Start-up Safety Review Checklist

Prio. = 1: Elevated Risk
Prio. = 2: Increased Risk
Prio. = 3: Medium Risk

| | | Check | | Status | Ok | | Action | Prio. | Resp. |
|---|---|---|---|---|---|---|---|---|---|
| | | on-site | docu | | yes | no | | | |
| | | | | | | | | | |
| | **Inlet piping** | | | | | | | | |
| 22 | Is the nominal size of the inlet piping and fittings the same as or larger than the nominal size of the PRV inlet connection? | x | x | | | | | | |
| 23 | Is the equivalent L/D ratio of the inlet piping < 5? | x | x | | | | | | |
| 24 | In case L/D>5, has the diameter of the piping been increased with respect to PRV inlet connection? | x | x | | | | | | |
| 25 | In case of an inlet piping to multiple relief valves, is the flow area of the common piping at least equal to the combined inlet areas? | | x | | | | | | |
| 26 | Have long radius elbows been used? | x | x | | | | | | |
| 27 | Is the PRV installed at the end of a long horizontal inlet pipe, where rust or scale may accumulate? | x | | | | | | | |
| 28 | The nonrecoverable pressure loss is less than 3% of the set pressure, using the rated flow if the PRV has no modulating characteristics? (with the exception of thermal expansion and remotely sensed pilot-operated relief valves) | | x | | | | | | |
| 29 | In case of a rupture disc installation in the inlet piping, does it have the same diameter as the inlet pipe? | x | x | | | | | | |
| 30 | Is there a process lateral piping connected to the inlet piping of the PRV? | x | x | | | | | | |
| 31 | Is the inlet piping stress-free (mechanical and thermal) both static and discharging? | x | x | | | | | | |
| 32 | The reaction forces and the bending moments are low enough not to produce excessive stresses on any of the components of the inlet piping? | | x | | | | | | |
| 33 | In case of open discharge, have the reaction forces been calculated by simplified methods? | | x | | | | | | |
| 34 | In case of closed discharge, have the reaction forces been calculated by specific software? | | x | | | | | | |
| | | | | | | | | | |
| | **Discharge piping** | | | | | | | | |
| 35 | In case of an open discharge, is there a long radius elbow? | x | x | | | | | | |
| 36 | In case of an open discharge, is there a low-point drain away from relief valve, structural steel and operating area? | x | | | | | | | |
| 37 | Is there any possibility of piping experiencing brittle fracture due to auto-refrigeration? | | x | | | | | | |
| 38 | Is the rated capacity of the conventional or balanced spring-loaded or pop-action pilot-operated PRV used in sizing the discharge piping? | | x | | | | | | |
| 39 | Is the common relief header piping in a closed system sized using the required flow? | | x | | | | | | |
| 40 | In case of a modulating pilot-operated PRV, is the discharge piping sized using the required flow? | | x | | | | | | |
| 41 | In case of thermal expansion of long pipeline or large liquid-filled vessels (e.g. LPG or LNG) is there any possibility of evaporation due to solar radiation taken | | x | | | | | | |

| **RELIEF SYSTEMS**<br><br>**Pre Start-up Safety Review Checklist** | | | | | | | | | Prio. = 1: Elevated Risk<br>Prio. = 2: Increased Risk<br>Prio. = 3: Medium Risk |
|---|---|---|---|---|---|---|---|---|---|
| | | Check | | Status | Ok | | Action | Prio. | Resp. |
| | | on-site | docu | | yes | no | | | |
| | into account? | | | | | | | | |
| 42 | In case of a pilot-operated PRV, can the superimposed backpressure be higher than the set pressure allowing reverse flow? | | X | | | | | | |
| 43 | In case of an isolated valve installed, does it have the capability of being locked or car-sealed in the appropriate position? | X | | | | | | | |
| 44 | Has draining been provided to prevent the accumulation of liquids on the downstream side of the PRV? | X | X | | | | | | |
| | | | | | | | | | |
| | **Stability** | | | | | | | | |
| 45 | In case of multiple PRVs with staggered settings, is the set pressure determined based on the maximum allowable pressure accumulation for multiple valve installations? | | X | | | | | | |
| 46 | Has stability analysis been done to avoid chattering?<br>- Excessive PRV inlet pressure loss<br>- Excessive built-up backpressure<br>- Acoustic interaction<br>- Retrograde condensation in the inlet<br>- Improper valve selection<br>- Oversized PRV | | X | | | | | | |
| 47 | In case the inlet pressure drop > 3% of set pressure, has the PRV capacity been corrected? | | X | | | | | | |
| 48 | For liquid filled systems, has the liquid static head taken into account to adjust the set pressure? | | X | | | | | | |
| 49 | In the cases where the inlet pressure drop is higher than 3%, has an “engineering analysis” been performed? | | X | | | | | | |
| 50 | In the calculation of the inlet pressure losses, has the entrance friction loss from the protected equipment been considered? | | X | | | | | | |
| 51 | If the PRV is installed on a normally flowing process line, is the 3% rule applied to the sum of the loss in the normally nonflowing PRV inlet piping and, the incremental pressure loss in the process line caused by the flow through the PRV? | | X | | | | | | |
| 52 | Are the pressure loss calculations made not only for the sizing case, but for the others? | | X | | | | | | |
| 53 | Is the trim of the PRV appropriate? | | X | | | | | | |
| | | | | | | | | | |
| | **Isolation (block) valves** | | | | | | | | |
| 54 | In case of an isolation valve located in the inlet or outlet piping of the PRV, have administrative controls been put in place? | | X | | | | | | |
| 55 | Has the isolation valve at least the same diameter as the inlet or outlet piping? | X | X | | | | | | |
| 56 | Have the isolation valves the capacity of being locked or car-sealed? | X | | | | | | | |
| 57 | Is a bleed valve installed between the isolation valve and the PRV? | X | X | | | | | | |
| 58 | Is the isolation valve painted in a special color? | X | | | | | | | |

| **RELIEF SYSTEMS**<br><br>**Pre Start-up Safety Review Checklist** | Prio. = 1: Elevated Risk<br>Prio. = 2: Increased Risk<br>Prio. = 3: Medium Risk |
|---|---|

| | | Check | | Status | Ok | | Action | Prio. | Resp. |
|---|---|---|---|---|---|---|---|---|---|
| | | on-site | docu | | yes | no | | | |
| 59 | If three-way changeover valves are installed, have the opening and closing consequences of the isolation valves, been well understood by the operators? | x | x | | | | | | |
| | | | | | | | | | |
| | **Bonnet** | | | | | | | | |
| 60 | In case of a balanced valve, is the bonnet vented to atmosphere and designed to avoid plugging caused by insects or freezing? | x | | | | | | | |
| 61 | In case of a bellows rupture, has the leakage from the downstream side of the PRV to the atmosphere been considered? | | x | | | | | | |
| 62 | In the case where the vent line of the bonnet is routed away from personnel due to hazardous vapors, has a dispersion analysis been performed? | | x | | | | | | |
| | | | | | | | | | |
| | **Bolting and gasketing** | | | | | | | | |
| 63 | Are the gaskets dimensionally correct for the specific flanges? | x | | | | | | | |
| 64 | According to the design pressure and temperature, are the gaskets, flange facings, and bolting adequate? | | x | | | | | | |
| | | | | | | | | | |
| | **Documentation and training** | | | | | | | | |
| 65 | Is all the documentation specific for each safety valve available (vendor calculations, drawings, spare parts recommended, material certificates, maintenance instructions, etc.) | | x | | | | | | |
| 66 | Are all the pressure relief valves included in the Piping and Instrument Diagram of the plant? | | x | | | | | | |
| 67 | Has the inspection department all the relevant information, and has a dedicated file for each valve been opened? | | x | | | | | | |
| 68 | Have all the operations personnel received training about the safety valves, especially for the balanced valves and the changeover valves? | | x | | | | | | |
| | | | | | | | | | |

# Annex C: List of pressure relief valves studied

## PLANT III

| | | | | | | | | | Size | | | | | | | | | | Calculations | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Item | P&ID | Fluid | Design case | Equipment protected | Type | Set pressure (barg) | Capacity (kg/h) | Seat area (mm2) | Inlet | Outlet | Discharge | Revision sheet | P&ID | Contingency analysis | Spec. Sheet | Isometrics (Inlet/Outlet) | Equipment documentation (Purchase orders + Inspection records) | Relief load analysis and sizing + "vapor disengagement ISO 4126-10" | Inlet pressure loss, %SP | Outlet pressure loss, %SP | Stability (Chattering) | Engineering Analysis ΔPinlet>3%SP | Body Bowl Choking | Noise Level | Load/Momentum (Static/Dynamic) | Acoustic Induced Vibration | Revision period RV IFT v2.0 years |
| YS399/01A | B399A | Propylene | Thermal expansion | 888.001 | Sv | 23,5 | 1.000 | 113,1 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS399/01B | B399A | Propylene | Thermal expansion | 888.001 | Sv | 23,5 | 1.000 | 113,1 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS399/02 | B399B | Nitrogen | PCV39920 failure | 885.016 | Sv | 5,5 | 6.500 | 1738,5 | 50 | 50 | Atm | X | X | X | X | X | X | X | 5.0 | <15 | | | | | | | 5 |
| YS401/02 | B401 | Propylene | Reaction, fire | R400 | Sv | 40,0 | 60.000 | 2827,4 | 80 | 100 | Flare | X | X | X | X | X | X | X | 1.2 | 3.1 | | | | | | | 3 |
| YS401/03 | B401 | Propylene | Reaction, fire | R400 | Sv | 40,0 | 60.000 | 2827,4 | 80 | 100 | Flare | X | X | X | X | X | X | | 1.2 | 3.1 | | | | | | | 3 |
| H40120 | B401 | Propylene | Overpressure | R400 | Ball valve | 0-40 | 30.000 | | 100 | 100 | Flare | Not considered in the study | | | | | | | | | | | | | | | |
| YS401/05 | B401 | Heptane | Overpressure | P409 | Sv | 160,0 | | | 9 | 12 | Internal | Not considered in the study (internal relief valve for pump P709). Out of service | | | | | | | | | | | | | | | |
| YS401/06 | B401 | Propylene | Thermal expansion | 404.003 | Sv | 43,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS401/07 | B401 | Propylene | Thermal expansion | 702.003 | Sv | 46,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS402/01 | B402 | Lube oil | Overpressure | P405 | Sv | 55,0 | | | 25 | 25 | Internal | Not considered in the study (Discharge to P405 suction side) | | | | | | | | | | | | | | | |
| YS402/02 | B402 | Lube oil | Thermal expansion | B404 | Sv | 52,0 | | 44.2 | 25 | 25 | Flare | X | X | X | X | X | X | missing data | | | | | | | | | 5 |
| YS402/03 | B402 | Lube oil | Overpressure | P404 | Sv | 60,0 | | | 15 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS402/04 | B402 | Cooling water | Thermal expansion | W404 | Sv | 8,0 | 1.000 | 26,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS402/05 | B402 | Cooling water | Thermal expansion | W406 | Sv | 8,0 | 1.000 | 26,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS402/06 | B402 | Lube oil | Overpressure | P406A | Sv | 8,0 | | | Internal | Internal | Internal | Not considered in the study (internal relief valve for pump P406A). Discharge to suction side | | | | | | | | | | | | | | | |
| YS402/07 | B402 | Lube oil | Overpressure | P406B | Sv | 8,0 | | | Internal | Internal | Internal | Not considered in the study (internal relief valve for pump P406B). Discharge to suction side | | | | | | | | | | | | | | | 5 |
| YS402/08 | B402 | Lube oil | Overpressure | P406A/B | Sv | 8,0 | | | 50 | 50 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS403/01 | B403 | Propylene | Auto control failure | F490A/B | Sv | 5,0 | 555 | 254,5 | 25 | 25 | Flare | X | X | X | X | X | X | X | 0.9 | | | | | | | | 3 |
| YS403/02 | B403 | Propylene | Overpressure | Cromatograph | Sv | 2,0 | | | 12 | 12 | Flare | Not considered in the study | | | | | | | | | | | | | | | |
| YS404/1 | B404 | Propylene | Fire case, thermal expansion | W400 | Sv | 40,0 | 30.000 | 1256,6 | 50 | 80 | Flare | X | X | X | X | X | X | X | 2.4 | 14.1 | | | | | | | 3 |
| YS404/2 | B404 | Propylene | Fire case, thermal expansion | W400 | Sv | 40,0 | 30.000 | 1256,6 | 50 | 80 | Flare | X | X | X | X | X | X | X | 2.4 | 14.1 | | | | | | | 3 |
| YS404/3 | B404 | Propylene | Thermal expansion | 404.012 | Sv | 43,0 | 1.000 | 113,1 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS404/4 | B404 | Propylene | Thermal expansion | 404.014 | Sv | 43,0 | 1.000 | 113,1 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS404/5 | B404 | Propylene | Thermal expansion | 404.048 | Sv | 49,0 | 1.000 | 113,1 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS404/6 | B404 | Winog oil | Overpressure | XP400-1 | Sv | 60,0 | | | 12 | 12 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS404/7 | B404 | Winog oil | Overpressure | XP400-2 | Sv | 60,0 | | | 12 | 12 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS404/8 | B404 | Winog oil | Overpressure | XP400-3 | Sv | 60,0 | | | 12 | 12 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS404/9 | B404 | Propylene | Thermal expansion | 404.070 | Sv | 43,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS404/10 | B404 | Propylene | Thermal expansion | 404.001 | Sv | 46,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS404/11 | B404 | Propylene | Thermal expansion | 404.056 | Sv | 43,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS404/12 | B404 | Propylene | Thermal expansion | 404.056 | Sv | 43,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS404/13 | B404 | Propylene | Thermal expansion | 404.002 | Sv | 43,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 4 |
| YS404/14 | B404 | Propylene | Thermal expansion | 702.020 | Sv | 46,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | |
| YS405/1 | B405 | Catalyst | Overpressure | R401A | Rupture disk | 11,5 | | | 25 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS405/2 | B405 | Catalyst | Overpressure | R401B | Rupture disk | 11,5 | | | 25 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS405/3 | B405 | Winog oil | Overpressure | 406.006 | Rupture disk | 10,0 | | | 25 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS405/5 | B405 | Nitrogen | Auto control failure | Failure of PCV40503 | Sv | 0,7 | 300 | 415,5 | 25 | 40 | Atm | X | X | X | X | X | X | X | 12.7 | | | | | | | | 3 |
| YS405/7 | B405 | Lube oil | Thermal expansion | RB401A | Sv | 63,0 | | | 12 | 12 | Atm | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS405/8 | B405 | Lube oil | Thermal expansion | RB401B | Sv | 63,0 | | | 12 | 12 | Atm | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS405/9 | B405 | Catalyst | Overpressure | P407A | Sv | 9,0 | | | 15 | 25 | Internal | Not considered in the study (internal relief valve for pump P407A). Discharge to suction side | | | | | | | | | | | | | | | |
| YS405/10 | B405 | Catalyst | Overpressure | P407B | Sv | 9,0 | | | 15 | 25 | Internal | Not considered in the study (internal relief valve for pump P407B). Discharge to suction side | | | | | | | | | | | | | | | |
| YS405/11 | B405 | Winog oil | Thermal expansion | 406.006 | Rupture disk | 8,5 | | | 20 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS406/1 | B406 | Winog oil | Overpressure | P402 | Sv | 55,0 | | | 6 | 9 | Internal | Not considered in the study (internal relief valve for pump P402). Discharge to suction side | | | | | | | | | | | | | | | |

| | | | | | | | | | Size | | | | | | | | | Calculations | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Item | P&ID | Fluid | Design case | Equipment protected | Type | Set pressure (barg) | Capacity (kg/h) | Seat area (mm2) | Inlet | Outlet | Discharge | Revision sheet | P&ID | Contingency analysis | Spec. Sheet | Isometrics (Inlet/Outlet) | Equipment documentation (Purchase orders + Inspection records) | Relief load analysis and sizing + "vapor disengagement ISO 4126-10" | Inlet pressure loss, %SP | Outlet pressure loss, %SP | Stability (Chattering) | Engineering Analysis ΔPinlet>3%SP | Body Bowl Choking | Noise Level | Load/Momentum (Static/Dynamic) | Acoustic Induced Vibration | Revision period RV IFT v2.0 years |
| YS406/2 | B406 | Lube oil | Thermal expansion | RB402 | Sv | 20,0 | | | 12 | 12 | Atm | Not considered in tyhe study. By agitator vendor | | | | | | | | | | | | | | | |
| YS411/02 | B411 | Propylene | Reaction, fire | R410 | Sv | 40,0 | 60.000 | 2827,4 | 80 | 100 | Flare | X | X | X | X | X | X | X | 2.1 | 3.0 | | | | | | | 3 |
| YS411/03 | B411 | Propylene | Reaction, fire | R410 | Sv | 40,0 | 60.000 | 2827,4 | 80 | 100 | Flare | X | X | X | X | X | X | X | 2.1 | 3.0 | | | | | | | 3 |
| H41120 | B411 | Propylene | Overpressure | R410 | Ball valve | 0-40 | 30.000 | | 100 | 100 | Flare | Not considered in the study | | | | | | | | | | | | | | | |
| YS411/06 | B411 | Propylene | Thermal expansion | 414.003 | Sv | 43,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS411/07 | B411 | Propylene | Thermal expansion | 702.019 | Sv | 46,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS412/02 | B412 | Lube oil | Thermal expansion | B414 | Sv | 52,0 | | | 25 | 25 | Flare | X | X | X | X | X | X | missing data | | | | | | | | | 5 |
| YS412/03 | B412 | Lube oil | Blocked outlet | P414 | Sv | 60,0 | | | 15 | 25 | Atm | X | X | X | X | X | X | | | | | | | | | | 5 |
| YS412/04 | B412 | Cooling water | Thermal expansion | W414 | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS412/05 | B412 | Cooling water | Thermal expansion | W416 | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS412/06 | B412 | Lube oil | Blocked outlet | P416A | Sv | 8,0 | | | Internal | Internal | Internal | Not considered in the study (internal relief valve for pump P416A) | | | | | | | | | | | | | | | |
| YS412/07 | B412 | Lube oil | Blocked outlet | P416B | Sv | 8,0 | | | Internal | Internal | Internal | Not considered in the study (internal relief valve for pump P416B) | | | | | | | | | | | | | | | |
| YS412/08 | B412 | Lube oil | Blocked outlet | P416A/B | Sv | 8,0 | | | 50 | 50 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS413/01 | B413 | Propylene | Auto control failure PCV41318 | F490A/B | Sv | 5,0 | 670 | 415,5 | 25 | 40 | Atm | X | X | X | X | X | X | X | 2.9 | | | | | | | | 3 |
| YS413/02 | B413 | Propylene | Abnormal heat input | 412.043 | Sv | 40,0 | 10.000 | 314,2 | 25 | 50 | Flare | X | X | X | X | X | X | oversized | | | | | | | | | 3 |
| YS413/03 | B413 | Condensates | Fire case | B413 | Sv | 8,0 | | | 25 | 25 | Atm | X | X | X | X | X | X | X | 2.2 | | | | | | | | 5 |
| YS413/04 | B413 | Propylene | Thermal expansion | Cromatograph | Sv | 2,0 | | | 12 | 12 | Flare | Not considered in the study | | | | | | | | | | | | | | | |
| YS414/1 | B414 | Propylene | Fire case, thermal expansion | W410 | Sv | 40,0 | 30.000 | 1256,6 | 50 | 80 | Flare | X | X | X | X | X | X | X | 2.7 | 14.1 | | | | | | | 3 |
| YS414/2 | B414 | Propylene | Fire case, thermal expansion | W410 | Sv | 40,0 | 30.000 | 1256,6 | 50 | 80 | Flare | X | X | X | X | X | X | X | 2.7 | 14.1 | | | | | | | 3 |
| YS414/3 | B414 | Propylene | Thermal expansion | 414.012 | Sv | 43,0 | 1.000 | 113,1 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS414/4 | B414 | Propylene | Thermal expansion | 414.014 | Sv | 43,0 | 1.000 | 113,1 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS414/5 | B414 | Propylene | Thermal expansion | 414.048 | Sv | 49,0 | 1.000 | 113,1 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS414/6 | B414 | Winog oil | Blocked outlet | XP400-4 | Sv | 60,0 | | | 12 | 12 | Atm | Not considered in the study. By vendor | | | | | | | | | | | | | | | |
| YS414/7 | B414 | Winog oil | Blocked outlet | XP400-5 | Sv | 60,0 | | | 12 | 12 | Atm | Not considered in the study. By vendor | | | | | | | | | | | | | | | |
| YS414/8 | B414 | Winog oil | Blocked outlet | XP400-6 | Sv | 60,0 | | | 12 | 12 | Atm | Not considered in the study. By vendor | | | | | | | | | | | | | | | |
| YS414/9 | B414 | Winog oil | Blocked outlet | XP400-7 | Sv | 60,0 | | | 12 | 12 | Atm | Not considered in the study. By vendor | | | | | | | | | | | | | | | |
| YS414/10 | B414 | Winog oil | Blocked outlet | XP400-8 | Sv | 60,0 | | | 12 | 12 | Atm | Not considered in the study. By vendor | | | | | | | | | | | | | | | |
| YS414/11 | B414 | Propylene | Thermal expansion | 414.056 | Sv | 43,0 | 1.000 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS414/12 | B414 | Propylene | Thermal expansion | 414.056 | Sv | 43,0 | 1.000 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS414/13 | B414 | Propylene | Thermal expansion | 414.056 | Sv | 43,0 | 1.000 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS414/14 | B414 | Propylene | Thermal expansion | 702.026 | Sv | 46,0 | 1.000 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS414/16 | B414 | Propylene | Thermal expansion | 414.001 | Sv | 46,0 | 1.000 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS415/1 | B415 | Catalyst | Overpressure | R411A | Rupture disk | 11,5 | | | 25 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS415/2 | B415 | Catalyst | Overpressure | R411B | Rupture disk | 11,5 | | | 25 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS415/3 | B415 | Winog oil | Overpressure | 406.016 | Sv | 10,0 | | | 25 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS415/7 | B415 | Lube oil | Thermal expansion | RB411A | Sv | 63,0 | | | 12 | 12 | Atm | Not considered in the study. By vendor | | | | | | | | | | | | | | | |
| YS415/8 | B415 | Lube oil | Thermal expansion | RB411B | Sv | 63,0 | | | 12 | 12 | Atm | Not considered in the study. By vendor | | | | | | | | | | | | | | | |
| YS415/9 | B415 | Catalyst | Overpressure | P417A | Sv | 9,0 | | | 15 | 25 | Internal | Not considered in the study (internal relief valve for pump P417A) | | | | | | | | | | | | | | | |
| YS415/10 | B415 | Catalyst | Overpressure | P417B | Sv | 9,0 | | | 15 | 25 | Internal | Not considered in the study (internal relief valve for pump P417B) | | | | | | | | | | | | | | | |
| YS415/11 | B415 | Winog oil | Thermal expansion | 406.016 | Rupture disk | 8,5 | | | 20 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS416/1 | B416 | Cooling water | Thermal expansion | 416.004 | Sv | 10,0 | | | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS416/2 | B416 | Cooling water | Thermal expansion | VW410-1 | Sv | 50,0 | | | 15 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS416/3 | B416 | Propylene | Fire, thermal expansion, cooling water failure | W415 | Sv | 45,0 | 10.000 | 314,1 | 25 | 50 | Flare | X | X | X | X | X | X | X | 0.9 | 8.4 | | | | X | X | | 5 |

| | | | | | | | | | Size | | | | | | | | | | Calculations | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Item | P&ID | Fluid | Design case | Equipment protected | Type | Set pressure (barg) | Capacity (kg)h) | Seat area (mm2) | Inlet | Outlet | Discharge | Revision sheet | P&ID | Contingency analysis | Spec. Sheet | Isometrics (Inlet/Outlet) | Equipment documentation (Purchase orders + Inspection records) | Relief load analysis and sizing + "vapor disengagement ISO 4126-10" | Inlet pressure loss, %SP | Outlet pressure loss, %SP | Stability (Chattering) | Engineering Analysis ΔPinlet>3%SP | Body Bowl Choking | Noise Level | Load/Momentum (Static/Dynamic) | Acoustic Induced Vibration | Revision period RV IFT v2.0 years |
| YS416/4 | B416 | Propylene | Thermal expansion | 414.021 | Sv | 46,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS416/5 | B416 | Propylene | Thermal expansion | 414.016 | Sv | 46,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS416/6 | B416 | Cooling water | Thermal expansion | VM410 | Sv | 8,0 | | | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS416/7 | B416 | Cooling water | Thermal expansion | X410 | Sv | 8,0 | | | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS418/01 | B418 | Lube oil | Overpressure | VP410-1 | Sv | 5,0 | | | 50 | 50 | Atm | Not considered in the study (internal safety valve for pump VP410-1) | | | | | | | | | | | | | | | |
| YS418/02 | B418 | Lube oil | Overpressure | VP410-2 | Sv | 5,0 | | | 50 | 50 | Atm | Not considered in the study (internal safety valve for pump VP410-2) | | | | | | | | | | | | | | | |
| YS418/03 | B418 | Cooling water | Thermal expansion | XV410 | Sv | 8,0 | | | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS420/1 | B420 | Nitrogen | Blocked outlet | 420.038 | Sv | 6,0 | | | 25 | 25 | Atm | X | X | X | X | X | X | X | 3.0 | | | | | | | | 5 |
| YS420/2 | B420 | Cooling water | Thermal expansion | 420.069 | Sv | 8,0 | | | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS420/4 | B420 | Nitrogen | Failure of PCV42021 | 420.058 | Sv | 6,0 | | | 25 | 25 | Atm | X | X | X | X | X | X | X | 0.03 | | | | | | | | 5 |
| YS420/5 | B420 | Cooling water | Thermal expansion | 420.071 | Sv | 8,0 | | | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS420/6 | B420 | Cooling water | Thermal expansion | 420.052 | Sv | 8,0 | | | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS420/7 | B420 | Lube oil | Thermal expansion | RB422-1 | Sv | 16,0 | | | 15 | 15 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS420/8 | B420 | Lube oil | Thermal expansion | RB420-1 | Sv | 16,0 | | | 15 | 15 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS430/01 | B430 | Nitrogen | Overpressure/Vacuum | B430 | Kito | -0,01 / 0,135 | | | 250 | 250 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS430/02 | B430 | Nitrogen | Overpressure/Vacuum | B431 | Kito | -0,01 / 0,135 | | | 250 | 250 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS430/03 | B430 | Nitrogen | Overpressure/Vacuum | B432 | Kito | -0,01 / 0,135 | | | 250 | 250 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS430/04 | B430 | Nitrogen | Overpressure/Vacuum | B433 | Kito | -0,01 / 0,135 | | | 250 | 250 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS430/05 | B430 | Nitrogen | Overpressure | F443 | Kito | 0,200 | | | 80 | 80 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS430/06 | B431 | Air | Overpressure | H449 | Sv | 3,5 | | | 100 | 150 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| KT432-01 | B432 | Air | Vacuum/Overpressure | B190 | Kito | -0,005/0,045 | | 7.088/31.416 | 200 | 200 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| KT432-02 | B432 | Air | Vacuum/Overpressure | B191 | Kito | -0,005/0,045 | | 7.088/31.416 | 200 | 200 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| KT432-03 | B432 | Air | Vacuum/Overpressure | B192 | Kito | -0,005/0,045 | | 7.088/31.416 | 200 | 200 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y7416 | B433 | Air | Overpressure | V375A | Sv | 3,0 | | | 50 | 80 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y7417 | B433 | Air | Overpressure | V375A | Sv | 3,0 | | | 50 | 80 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y7418 | B433 | Air | Overpressure | V375B | Sv | 3,0 | | | 50 | 80 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y7419 | B433 | Air | Overpressure | V375B | Sv | 3,0 | | | 50 | 80 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y7420 | B433 | Air | Overpressure | V375A | Sv | 8,0 | | | 50 | 80 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y7421 | B433 | Air | Overpressure | V375B | Sv | 8,0 | | | 50 | 80 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y7422 | B433 | Cooling water | Thermal expansion | V375A | Sv | 6,0 | | | 50 | 80 | Atm | Not considered inthe study | | | | | | | | | | | | | | | |
| Y7423 | B433 | Cooling water | Thermal expansion | V375B | Sv | 6,0 | | | 50 | 80 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y7424 | B433 | Air | Overpressure | B375 | Sv | 8,0 | | | 50 | 80 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS440/01 | B440 | Nitrogen | Overpressure | F442 | Kito | 0,200 | | | 80 | 80 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS441/01 | B441 | Steam | Thermal expansion | 444.028 | Sv | 39,0 | | | 15 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS441/02 | B441 | Steam | Thermal expansion | 444.032 | Sv | 39,0 | | | 15 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS441/03 | B441 | Steam | Thermal expansion | 444.047 | Sv | 39,0 | | | 15 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS441/04 | B441 | Air | Failure of steam trap or valve. Calculated for FZ44108 | 444.012 | Sv | 8,0 | 2.000 | 660,0 | 40 | 50 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS441/05 | B441 | Cooling water | Thermal expansion | 444.022 | Sv | 8,0 | 1.000 | 254.5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS441/06 | B441 | Demi water | Failure of steam trap or valve. Calculated for FZ44108 | 444.010 | Sv | 12,0 | 2.000 | 415.5 | 25 | 40 | Atm | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS441/08 | B441 | Cooling water | Thermal expansion | 441.072 | Sv | 8,0 | 1.000 | 254.5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS441/09 | B441 | Cooling water | Thermal expansion | 441.007 | Sv | 8,0 | 1.000 | 254.5 | 25 | 25 | Atm | | | | | | | | | | | | | | | | |
| YS441/10 | B441 | Lube oil | Overpressure | XPM441A | Sv | 6,0 | | | | | Internal | Not considered in the study (internal safety valve for pump XPM441A). Discharge to suction side | | | | | | | | | | | | | | | |
| YS441/11 | B441 | Lube oil | Overpressure | XPM441B | Sv | 6,0 | | | | | Internal | Not considered in the study (internal safety valve for pump XPM441B). Discharge to suction side | | | | | | | | | | | | | | | |

| | | | | | | | | | Size | | | | | | | | | Calculations | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Item | P&ID | Fluid | Design case | Equipment protected | Type | Set pressure (barg) | Capacity (kg)h) | Seat area (mm2) | Inlet | Outlet | Discharge | Revision sheet | P&ID | Contingency analysis | Spec. Sheet | Isometrics (Inlet/Outlet) | Equipment documentation (Purchase orders + Inspection records) | Relief load analysis and sizing + "vapor disengagement ISO 4126-10" | Inlet pressure loss, %SP | Outlet pressure loss, %SP | Stability (Chattering) | Engineering Analysis ΔPinlet>3%SP | Body Bowl Choking | Noise Level | Load/Momentum (Static/Dynamic) | Acoustic Induced Vibration | Revision period RV IFT v2.0 years |
| YS441/12 | B441 | Cooling water | Thermal expansion | 441.063 | Sv | 8,0 | 1.000 | 254.5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YB442/01 | B441 | Hydraulic oil | Overpressure | XP442-2 | Sv | 600,0 | | | 25 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YB442/02 | B441 | Hydraulic oil | Overpressure | XP442-1 | Sv | 600,0 | | | 25 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS442/01 | B442 | Thermal oil | Thermal expansion | XP443-1 | Sv | 10,0 | | | 25 | 25 | Internal | Not considered in the study (internal safety valve to tank XB443. Nitrogen blanket 100 mbar) | | | | | | | | | | | | | | | |
| YS442/02 | B442 | Thermal oil | Thermal expansion | XP443-2 | Sv | 10,0 | | | 25 | 25 | Internal | Not considered in the study (internal safety valve to tank XB443. Nitrogen blanket 100 mbar) | | | | | | | | | | | | | | | |
| YS442/03 | B442 | Thermal oil | Overpressure. Calculated for FZ44213. Nitrogen 4,5 barg | XB443 | Sv | 1,0 | | | 25 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS442/04 | B442 | Cooling water | Thermal expansion | 442.001 | Sv | 8,0 | 1.000 | 254.5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS443/01 | B443 | Cooling water | Thermal expansion | 443.006 | Sv | 8,0 | 1.000 | 254.5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS443/02 | B443 | Cooling water | Thermal expansion | 443.003 | Sv | 8,0 | 1.000 | 254.5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS443/03 | B443 | Lube oil | Thermal expansion | XPM440A | Sv | 6,0 | | | Internal | Internal | Internal | Not considered in the study (internal safety valve for pump XPM440A). Discharge to suction side | | | | | | | | | | | | | | | |
| YS443/04 | B443 | Lube oil | Thermal expansion | XPM440B | Sv | 6,0 | | | Internal | Internal | Internal | Not considered in the study (internal safety valve for pump XPM440B). Discharge to suction side | | | | | | | | | | | | | | | |
| YS450/01 | B450 | Cooling water | Thermal expansion | 450.003 | Sv | 8,0 | 1.000 | 254.5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | |
| YS450/02 | B450 | Cooling water | Thermal expansion | 450.021 | Sv | 8,0 | 1.000 | 254.5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS450/03 | B450 | Cooling water | Thermal expansion | 450.013 | Sv | 8,0 | 1.000 | 254.5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS450/04 | B450 | Nitrogen | Fire case | B451 | Sv | 6,0 | 500 | 415.5 | 25 | 40 | Atm | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS460/01 | B460 | Cooling water | Thermal expansion | 460.004 | Sv | 8,0 | 1.000 | 254.5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 2 |
| YS460/02 | B460 | Air | Overpressure | H469 | Sv | 3.5 | | | 80 | 125 | Atm | Not considered in the study. By vendor | | | | | | | | | | | | | | | |
| YS470/01 | B470 | Steam | Autocontrol failure. Maximum flowrate of FZ47009 | B470 | Sv | 0,3 | 12.000 | 314.1 | 250 | 350 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS470/02.1 | B470 | Nitrogen | Pressure/vacuum valve | B470 | Kito | -0,09 | | | 100 | 100 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS470/02.2 | B470 | Nitrogen | Pressure/vacuum valve | B470 | Kito | 0,27 | | | 50 | 50 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS471/01 | B471 | Air | Overpressure | H474 | Sv | 6,0 | | | 100 | 150 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS480/01 | B480 | Air | Overpressure | H480A/B/C | Sv | 3,5 | | | 80 | 125 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS480/02 | B480 | Air | Overpressure | H480A | Sv | 3,5 | | | 100 | 150 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS480/03 | B480 | Air | Overpressure | H480B | Sv | 3,5 | | | 100 | 150 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS480/04 | B480 | Air | Overpressure | H480C | Sv | 3,5 | | | 100 | 150 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS490/01 | B490 | Cooling water | Thermal expansion | 490.001 | Sv | 8,0 | 1.000 | 254.5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS490/02 | B490 | Propylene | Autocontrol failure, fire, thermal expansion | 490.003 | Sv | 6,0 | 50.000 | 1227.2 | 150 | 250 | Flare | X | X | X | X | X | X | X | 1.0 | | | | | | | | 3 |
| YS491/01 | B491 | Propylene | Blocked outlet, fire, thermal expansion | VB490-2 | Sv | 8,0 | 20.400 | 4300,0 | 80 | 125 | Flare | X | X | X | X | X | X | X | 2.9 | | | | | | | | 5 |
| YS491/02 | B491 | Propylene | Blocked outlet, fire, thermal expansion | VB490-4 | Sv | 19,0 | 20.400 | 1456,0 | 50 | 80 | Flare | X | X | X | X | X | X | X | 1.9 | | | | | | | | 5 |
| YS491/03 | B491 | Propylene | Blocked outlet, fire, thermal expansion | VB490-6 | Sv | 40,0 | 20.400 | 661,0 | 50 | 80 | Flare | X | X | X | X | X | X | X | 0.6 | | | | | | | | 3 |
| YS491/04 | B491 | Propylene | Overpressure | V490 | Sv | 2,0 | | | 125 | 200 | Flare | X | X | X | X | X | X | X | 5.0 | | | X | | | | | 5 |
| YS491/05 | B491 | Cooling water | Thermal expansion | VW490-2 | Sv | 8,0 | | | 32 | 50 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS492/01 | B492 | Lube oil | Overpressure | VP490-1 | Sv | 5,0 | | | Internal | Internal | Internal | Not considered in the study (internal safety valve for pump VP490-1). Discharge to suction side | | | | | | | | | | | | | | | |
| YS492/02 | B492 | Lube oil | Overpressure | VP490-2 | Sv | 5,0 | | | Internal | Internal | Internal | Not considered in the study (internal safety valve for pump VP490-2). Discharge to suction side | | | | | | | | | | | | | | | |
| YS493/01 | B493 | Cooling water | Thermal expansion | 493.001 | Sv | 8,0 | | | 25 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS700/01 | B700 | Propylene | Fire, thermal expansion | K700 | Sv | 38,0 | 15.000 | 1256,6 | 50 | 80 | Flare | X | X | X | X | X | X | X | 4.2 | 8.2 | X | X | X | | | X | 5 |
| YS700/02 | B700 | Propylene | Fire, thermal expansion | K700 | Sv | 38,0 | 15.000 | 1256,6 | 50 | 80 | Flare | X | X | X | X | X | X | X | 4.2 | 8.2 | X | X | X | | | X | 5 |
| YS700/03 | B700 | Nitrogen | Fire, thermal expansion | W705 | Sv | 5,5 | 500 | 415,5 | 25 | 40 | Flare | X | X | X | X | X | X | X | (pending) | | | | | | | | 5 |
| YS700/04 | B700 | Propylene | Thermal expansion | 700.018 | Sv | 38,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS700/05 | B700 | Propylene | Thermal expansion | 700.018 | Sv | 38,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS701/01 | B701 | Propylene | Fire, thermal expansion | K702A | Sv | 38,0 | 20.000 | 1256,6 | 50 | 80 | Flare | X | X | X | X | X | X | X | 3.1 | 10.8 | | | | | | | 5 |
| YS701/02 | B701 | Propylene | Fire, thermal expansion | K702A | Sv | 38,0 | 20.000 | 1256,6 | 50 | 80 | Flare | X | X | X | X | X | X | X | 3.1 | 10.8 | | | | | | | 5 |
| YS701/03 | B701 | Propylene | Fire,thermal expansion | K702B | Sv | 38,0 | 20.000 | 1256,6 | 50 | 80 | Flare | X | X | X | X | X | X | X | 3.1 | 9.9 | | | | | | | 5 |

| | | | | | | | | | Size | | | | | | | | | Calculations | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Item | P&ID | Fluid | Design case | Equipment protected | Type | Set pressure (barg) | Capacity (kg)h) | Seat area (mm2) | Inlet | Outlet | Discharge | Revision sheet | P&ID | Contingency analysis | Spec. Sheet | Isometrics (Inlet/Outlet) | Equipment documentation (Purchase orders + Inspection records) | Relief load analysis and sizing + "vapor disengagement ISO 4126-10" | Inlet pressure loss, %SP | Outlet pressure loss, %SP | Stability (Chattering) | Engineering Analysis ΔPinlet>3%SP | Body Bowl Choking | Noise Level | Load/Momentum (Static/Dynamic) | Acoustic Induced Vibration | Revision period RV IFT v2.0 years |
| YS701/04 | B701 | Propylene | Fire, thermal expansion | K702B | Sv | 38,0 | 20.000 | 1256,6 | 50 | 80 | Flare | X | X | X | X | X | X | X | 3.1 | 9.9 | | | | | | | 5 |
| YS701/05 | B701 | Propylene | Thermal expansion | 701.032 | Sv | 43,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS702/01 | B702 | Propylene | Fire, cooling water failure, thermal expansion | W700 | Sv | 45,0 | 10.000 | 314,2 | 25 | 50 | Flare | X | X | X | X | X | X | X | 1.4 | 7.6 | X | | X | X | X | X | 5 |
| YS702/03 | B702 | Propylene | Thermal expansion | 702.054 | Sv | 46,0 | 1.000 | 113,1 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS702/04 | B702 | Propylene | Thermal expansion | 702.012 | Sv | 46,0 | 1.000 | 113,1 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS702/05 | B702 | Cooling water | Thermal expansion | W700 | Sv | 45,0 | 1.000 | 113,1 | 15 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS702/06 | B702 | Lube oil | Overpressure | P701A | Sv | 45,0 | | | Internal | Internal | Internal | Not considered in the study (internal safety valve for lubricating pump P701A) | | | | | | | | | | | | | | | |
| YS702/07 | B702 | Lube oil | Overpressure | P701B | Sv | 45,0 | | | Internal | Internal | Internal | Not considered in the study (internal safety valve for lubricating pump P701B) | | | | | | | | | | | | | | | |
| YS702/08 | B702 | Cooling water | Thermal expansion | PW701A | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS702/09 | B702 | Cooling water | Thermal expansion | PW701B | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS702/10 | B702 | Propylene | Fire, thermal expansion | F700 | Sv | 45,0 | 1.000 | 113,1 | 15 | 25 | Flare | X | X | X | X | X | X | X | 0.8 | 37.8 | X | | | | | | 5 |
| YS702/11 | B702 | Propylene | Thermal expansion | 702.037 | Sv | 46,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS702/12 | B702 | Propylene | Thermal expansion | 702.006 | Sv | 46,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS702/13 | B702 | Propylene | Thermal expansion | 702.016 | Sv | 46,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS702/14 | B702 | Propylene | Thermal expansion | 702.003 | Sv | 46,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS702/15 | B702 | Propylene | Fire case | W702 | Sv | 39,0 | 4.400 | 198,0 | 40 | 65 | Flare | X | X | X | X | X | X | X | 6.5 | | | | | | | | 5 |
| YS702/16 | B702 | Propylene | Thermal expansion | 702.056 | Sv | 46,0 | | | 25 | 25 | Flare | X | X | X | X | X | X | X | (pending) | | | | | | | | 5 |
| YS710/1 | B710 | Propylene | Fire, thermal expansion | K710B | Sv | 38,0 | 3.000 | 314,2 | 25 | 50 | Flare | X | X | X | X | X | X | X | 0.47 | 6.8 | | | | | | | 5 |
| YS710/2 | B710 | Propylene | Fire, thermal expansion | K710B | Sv | 38,0 | 3.000 | 314,2 | 25 | 50 | Flare | X | X | X | X | X | X | X | 0.47 | 6.8 | | | | | | | 5 |
| YS710/3 | B710 | Propylene | Fire, thermal expansion | K710A | Sv | 38,0 | 3.000 | 314,2 | 25 | 50 | Flare | X | X | X | X | X | X | X | 0.47 | 6.8 | | | | | | | 5 |
| YS710/4 | B710 | Propylene | Fire, thermal expansion | K710A | Sv | 38,0 | 3.000 | 314,2 | 25 | 50 | Flare | X | X | X | X | X | X | X | 0.47 | 6.8 | | | | | | | 5 |
| YS720/1 | B720 | IPA | Fire, thermal expansion | B720 | Sv | 6,0 | 3.000 | 660,5 | 40 | 50 | Flare | X | X | X | X | X | X | X | 16.5 | 33.4 | X | | | | | X | 5 |
| YS720/2 | B720 | IPA | Overpressure | P723A | Sv | 55,0 | | | 6 | 9 | Internal | Not considered in the study (internal safety valve for pump P723A). Discharge to suction side | | | | | | | | | | | | | | | |
| YS720/3 | B720 | IPA | Overpressure | P723B | Sv | 55,0 | | | 6 | 9 | Internal | Not considered in the study (internal safety valve for pump P723B). Discharge to suction side | | | | | | | | | | | | | | | |
| YS734/1 | B730 | Silane | Overpressure | P734A | Sv | 44,0 | | | Internal | Internal | Internal | Not considered in the study (internal safety valve for pump P734A). Discharge to suction side | | | | | | | | | | | | | | | |
| YS734/2 | B730 | Silane | Overpressure | P734B | Sv | 44,0 | | | Internal | Internal | Internal | Not considered in the study (internal safety valve for pump P734B). Discharge to suction side | | | | | | | | | | | | | | | |
| YS734/3 | B730 | Silane | Overpressure | P734C | Sv | 44,0 | | | Internal | Internal | Internal | Not considered in the study (internal safety valve for pump P734C). Discharge to suction side | | | | | | | | | | | | | | | |
| YB823/1 | B823 | Teal | Fire case | B740 | Rupture disk | 4,9 | | | 25 | 25 | Burning pit | Not considered in the study. Discharge to burning pit | | | | | | | | | | | | | | | |
| YB823/2 | B823 | Teal | Fire case | B740 | Rupture disk | 4,9 | | | 25 | 25 | Burning pit | Not considered in the study. Discharge to burning pit | | | | | | | | | | | | | | | |
| YS823/3 | B823 | Teal | Fire case | B740 | Sv | 4,0 | | | 25 | 25 | Burning pit | Not considered in the study. Discharge to burning pit | | | | | | | | | | | | | | | |
| YS823/4 | B823 | Teal | Fire case | B740 | Sv | 4,0 | | | 25 | 25 | Burning pit | Not considered in the study. Discharge to burning pit | | | | | | | | | | | | | | | |
| YS823/5 | B823 | Teal | Overpressure | 824.802 | Sv | 25,0 | | | 25 | 25 | Burning pit | Not considered in the study. Discharge to burning pit | | | | | | | | | | | | | | | |
| YS740/1 | D824 | Teal | Overpressure | P740A | Sv | 44,0 | | | Internal | Internal | Internal | Not considered in the study (Internal safety valve for pump P740A). Discharge to suction side | | | | | | | | | | | | | | | |
| YS740/2 | D824 | Teal | Overpressure | P740B | Sv | 44,0 | | | Internal | Internal | Internal | Not considered in the study (internal safety valve for pump P740B). Discharge to suction side | | | | | | | | | | | | | | | |
| YS740/3 | D824 | Teal | Overpressure | P740C | Sv | 44,0 | | | Internal | Internal | Internal | Not considered in the study (internal safety valve for pump P740C). Discharge to suction side | | | | | | | | | | | | | | | |
| YS750/1 | B750 | Cooling water | Thermal expansion | 750.014 | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS750/2 | B750 | Cooling water | Thermal expansion | 750.021 | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YB750/01 | B750 | Peroxide | Overpressure | P750 | Rupture disk | 4,0 | | | 25 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YB750/02 | B750 | Peroxide | Overpressure | P750 | Rupture disk | 4,5 | | | 25 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YB750/03 | B750 | Air | Overpressure | Peroxide room | Explosion relief panel | - | | | 1 m2 | 1 m2 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YB750/04 | B750 | Peroxide | Overpressure | P751 | Rupture disk | 4,0 | | | 25 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YB750/05 | B750 | Peroxide | Overpressure | P751 | Rupture disk | 4,5 | | | 25 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS760/01 | B760 | Nitrogen | Pressure/vacuum valve | B760 | Kito | 0,09/-0,05 | | | 50 | 50 | Atm | Not considered in the study | | | | | | | | | | | | | | | |

| | | | | | | | | | Size | | | | | | | | | Calculations | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Item | P&ID | Fluid | Design case | Equipment protected | Type | Set pressure (barg) | Capacity (kg)h) | Seat area (mm2) | Inlet | Outlet | Discharge | Revision sheet | P&ID | Contingency analysis | Spec. Sheet | Isometrics (Inlet/Outlet) | Equipment documentation (Purchase orders + Inspection records) | Relief load analysis and sizing + "vapor disengagement ISO 4126-10" | Inlet pressure loss, %SP | Outlet pressure loss, %SP | Stability (Chattering) | Engineering Analysis ΔPinlet>3%SP | Body Bowl Choking | Noise Level | Load/Momentum (Static/Dynamic) | Acoustic Induced Vibration | Revision period RV IFT v2.0 years |
| YS760/02 | B760 | Nitrogen | Pressure/vacuum valve | B762 | Kito | 0,09/-0,06 | | | 50 | 50 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS760/03 | B760 | Nitrogen | Pressure/vacuum valve | B764 | Kito | 0,09/-0,07 | | | 50 | 50 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS760/04 | B760 | Nitrogen | Pressure/vacuum valve | B765 | Kito | 0,09/-0,08 | | | 100 | 100 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS760/05 | B760 | Nitrogen | Autocontrol valve failure. PCV 76017 | 760.055 | Sv | 2,0 | | | 50 | 80 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS760/06 | B760 | Nitrogen | Autocontrol valve failure. Failure of PCV76027 | H760/H762/ H764/H767 | Sv | 0,7 | 50 | 254,469 | 25 | 25 | Atm | X | X | X | X | X | X | X | 0.3% | | | | | | | | 5 |
| YS762/01 | B762 | Demi water | Thermal expansion, fire | W785 | Sv | 6,0 | 500 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS762/02 | B762 | Antistaticum (loxiol, atmer) | Overpressure | P780 | Sv | 155 | | | 25 | 25 | Internal | Not considered in the study (internal safety valve for pump P780). Discharge to suction side | | | | | | | | | | | | | | | |
| YS762/03 | B762 | Antistaticum (loxiol, atmer) | Overpressure | P781 | Sv | 155 | | | 9 | 12 | Internal | Not considered in the study (internal safety valve for pump P780). Discharge to suction side | | | | | | | | | | | | | | | |
| YS762/04 | B762 | Demi water | Thermal expansion | 762.025 | Sv | 2,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS851/02 | B851 | Organic waste | Fire case | B856 | Sv | 6,0 | 4.000 | 660,5 | 40 | 50 | Sewerage | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS852/01 | B852 | Nitrogen | Autocontrol valve failure | 852.008 | Sv | 6,0 | | | 12 | 12 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS852/02 | B852 | Nitrogen | Autocontrol valve failure | 852.009 | Sv | 6,0 | | | 12 | 12 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS852/03 | B852 | Propane | Autocontrol valve failure | 852.022 | Sv | 6 | | | 12 | 12 | Natural gas subheader | Not considered in the study | | | | | | | | | | | | | | | |
| YS854/01 | D1275 | Propylene/ethane | Thermal expansion | 860.019 | Sv | | | | 15 | 25 | Flare | Dismantled | | | | | | | | | | | | | | | |
| YS860/01 | B860 | Propylene/ethane | Fire, thermal expansion | B862 | Sv | 40,0 | 15.500 | 1256,6 | 50 | 80 | Flare | X | X | X | X | X | X | X | 5.8% | 17.3% | | | | | | | 3 |
| YS860/02 | B860 | Propylene/ethane | Thermal expansion | 860.001 | Sv | 40,0 | 1.000 | 113,1 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS860/03 | B860 | Cooling water | Thermal expansion | W861 | Sv | 10,0 | 1.000 | 63,6 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS860/04 | B860 | Propylene/ethane | Autocontrol valve failure FV41601, thermal expansion | W861 | Sv | 40,0 | 12.000 | 1256,7 | 50 | 80 | Flare | X | X | X | X | X | X | X | 1.03% | | | | | | | | 5 |
| YS860/05 | B860 | Propylene/ethane | Blocked outlet, thermal expansion | P862A | Sv | 45,0 | 1.000 | 113,1 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS860/06 | B860 | Propylene/ethane | Blocked outlet, thermal expansion | P862B | Sv | 45,0 | 1.000 | 113,1 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS860/07 | B860 | Propylene/ethane | Abnormal heat input, control valve fails open | B866 | Sv | 7 | 2.000 | 415,5 | 25 | 40 | Flare | X | X | X | X | X | X | X | 1.4% | | | | | | | | 3 |
| YS860/08 | B860 | Propylene/ethane | Fire, thermal expansion | K862 | Sv | 40,0 | 15.000 | 314,2 | 25 | 50 | Flare | X | X | X | X | X | X | X | 1.7% | 9.6% | | | | | | | 5 |
| YS860/09 | B860 | Propylene | Exchanger tube breakage, thermal expansion | W865 | Sv | 25 | 15.500 | 706,9 | 50 | 80 | Flare | X | X | X | X | X | X | X | 0.03% | | | | | | | | 5 |
| YS860/10 | B860 | Propylene | Exchanger tube breakage | W867 | Sv | 25 | 31.000 | 1256,6 | 50 | 80 | Flare | X | X | X | X | X | X | X | 1.08% | | | | | | | | 5 |
| YS860/11 | B860 | Propylene/ethane | Thermal expansion | 860.060 | Sv | 40,0 | 1.000 | 113,1 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS860/12 | B860 | Propylene | Blocked outlet, fire | W866 | Sv | 40 | 40.000 | 1256,6 | 50 | 80 | Flare | X | X | X | X | X | X | X | 1.5% | 10.7% | | | | | | | 5 |
| YS860/13 | B860 | Propylene/ethane | Thermal expansion | 860.019 | Sv | 40,0 | 1.000 | 113,1 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS860/14 | B860 | Propylene | Exchanger tube breakage | W866 | Sv | 7,5 | 7360 (L) / 12640 (V) | 2827,4 | 65 | 100 | Flare | X | X | X | X | X | X | X | 2.8% | | | | | | | | 5 |
| YS860/15 | B860 | Propylene/ethane | Thermal expansion | 860.002 | Sv | 40 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS860/16 | B860 | Propylene/ethane | Thermal expansion | Cromatograph | Sv | 2,0 | | | 12 | 12 | Flare | Not considered in the study | | | | | | | | | | | | | | | |
| YS860/17 | B860 | Propylene/ethane | Thermal expansion | Cromatograph | Sv | 2,0 | | | 12 | 12 | Flare | Not considered in the study | | | | | | | | | | | | | | | |
| YS860/18 | B860 | Propylene/ethane | Thermal expansion | Cromatograph | Sv | 2,0 | | | 12 | 12 | Flare | Not considered in the study | | | | | | | | | | | | | | | |
| YS860/19 | D1275 | Propylene/ethane | Thermal expansion | 24.442 | Sv | 8,0 | | | 15 | 25 | Flare | Not considered in the study | | | | | | | | | | | | | | | |
| YS861/01 | B860 | Propylene | Fire, abnormal heat input | B860 | Sv | 31,0 | 14.000 | 660,5 | 40 | 50 | Flare | X | X | X | X | X | X | X | 0.9% | 15.3% | | | | | | | 3 |
| YS861/02 | B860 | Propylene | Thermal expansion | 861.016 | Sv | 40,0 | 1.000 | 113,1 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS861/03 | B861 | Propylene | Thermal expansion | 861.023 | Sv | 40,0 | 1.000 | 113,1 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS861/04 | B861 | Propylene | Blocked outlet, exchanger tube breakage, fire | K860 | Sv | 15,5 | 30.800 | 2827,4 | 65 | 100 | Flare | X | X | X | X | X | X | X | 3.66% | 16.7% | | | | | | | 5 |
| YS861/05 | B861 | Propylene | Abnormal heat input, blocked outlet, fire | B864 | Sv | 15,5 | 8.000 | 660,5 | 40 | 50 | Flare | X | X | X | X | X | X | X | 11.9% | 16.3% | X | | X | | | X | 5 |
| YS861/06 | B861 | Cooling water | Thermal expansion | V860 | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS861/07 | B861 | Cooling water | Thermal expansion | W862 | Sv | 10,0 | 1.000 | 63,6 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS861/08 | B861 | Propylene | Fire case | W860 | Sv | 31,0 | 9.000 | 415,5 | 25 | 40 | Flare | X | X | X | X | X | X | X | 1.4% | 15.0% | | | | | | | 5 |
| YS861/09 | B861 | Propylene | Thermal expansion | 860.019 | Sv | 40,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS861/10 | B861 | Propylene | Thermal expansion | 861.019 | Sv | 45,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |

| | | | | | | | | | Size | | | | | | | | | Calculations | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Item | P&ID | Fluid | Design case | Equipment protected | Type | Set pressure (barg) | Capacity (kg)h) | Seat area (mm2) | Inlet | Outlet | Discharge | Revision sheet | P&ID | Contingency analysis | Spec. Sheet | Isometrics (Inlet/Outlet) | Equipment documentation (Purchase orders + Inspection records) | Relief load analysis and sizing + "vapor disengagement ISO 4126-10" | Inlet pressure loss, %SP | Outlet pressure loss, %SP | Stability (Chattering) | Engineering Analysis ΔPinlet>3%SP | Body Bowl Choking | Noise Level | Load/Momentum (Static/Dynamic) | Acoustic Induced Vibration | Revision period RV IFT v2.0 years |
| YS861/11 | B861 | Propylene | Thermal expansion | 861.019 | Sv | 45,0 | 500 | 63,6 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS861/12 | B861 | Propylene | Thermal expansion | Cromatograph | Sv | 2,0 | | | 12 | 12 | Flare | Not considered in the study | | | | | | | | | | | | | | | |
| YS861/13 | B861 | Propylene | Thermal expansion | Cromatograph | Sv | 2,0 | | | 12 | 12 | Flare | Not considered in the study | | | | | | | | | | | | | | | |
| YS861/14 | B861 | Propylene | Mechanical equipment failure | P860A | Sv | 30 | 500 | 78,5 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS861/15 | B861 | Propylene | Mechanical equipment failure | P860B | Sv | 30 | 500 | 78,5 | 15 | 25 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS862/01 | B862 | Lube oil | Overpressure | VP860-1 | Sv | 5,0 | | | 25 | 25 | Internal | Not considered in the study | | | | | | | | | | | | | | | |
| YS862/02 | B862 | Lube oil | Overpressure | VP860-2 | Sv | 5,0 | | | 25 | 25 | Internal | Not considered in the study | | | | | | | | | | | | | | | |
| YS863/01 | B863 | Cooling water | Thermal expansion | XV860-3 | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS863/02 | B863 | Cooling water | Thermal expansion | XV860-1 | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS864/01 | B864 | Propylene | Overpressure | XV860 | Sv Pilot operated | 21,0 | | | 50,0 | 50,0 | Internal | Not considered in the study (discharge to XV860 suction side) | | | | | | | | | | | | | | | |
| YS865/01 | B865 | Propylene | Fire, thermal expansion | XW860-1 | Sv | 25,0 | | | 25,0 | 25,0 | Flare | X | X | X | X | X | X | X | 1.7% | | | | | | | | 5 |
| YS865/2.1 | B865 | Propylene | Fire case | XB860-2 | Sv | 25,0 | | | 40,0 | 100,0 | Flare | X | X | X | X | X | X | X | 1.0% | | | | | | | | 5 |
| YS865/2.2 | B865 | Propylene | Fire case | XB860-2 | Sv | 25,0 | | | 40,0 | 100,0 | Flare | X | X | X | X | X | X | X | 1.4% | | | | | | | | 5 |
| YS865/03 | B865 | Propylene | Thermal expansion | XT860-1 | Sv | 25,0 | | | 15,0 | 25,0 | Flare | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS865/04 | B865 | Propylene | Fire case | XW860-4 | Sv | 25,0 | | | 25,0 | 40,0 | Flare | X | X | X | X | X | X | X | 1.5% | | | | | | | | 5 |
| YS865/05 | B865 | Propylene | Fire case | XW860-2 | Sv | 25,0 | | | 15,0 | 25,0 | Flare | X | X | X | X | X | X | X | 1.9% | | | | | | | | 5 |
| YS870/03 | B870 | Cooling water | Thermal expansion | W871 | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS870/04 | B870 | Lube oil | Overpressure | VP870A | Sv | 1,5 | | | Internal | Internal | Internal | Not considered in the study | | | | | | | | | | | | | | | |
| YS870/05 | B870 | Nitrogen | Overpressure | V870A | Sv | 3,5 | | | 50 | 80 | Nitrogen subheader | | Not considered in the study | | | | | | | | | | | | | | |
| YS870/06 | B870 | Lube oil | Overpressure | VP870B | Sv | 1,5 | | | Internal | Internal | Internal | | Not considered in the study | | | | | | | | | | | | | | |
| YS870/07 | B870 | Nitrogen | Overpressure | V870B | Sv | 3,5 | | | 50 | 80 | Nitrogen subheader | | Not considered in the study | | | | | | | | | | | | | | |
| YS870/08 | B870 | Lube oil | Overpressure | VP870C | Sv | 1,5 | | | Internal | Internal | Internal | | Not considered in the study | | | | | | | | | | | | | | |
| YS870/09 | B870 | Nitrogen | Overpressure | V870C | Sv | 3,5 | | | 50 | 80 | Nitrogen subheader | | Not considered in the study | | | | | | | | | | | | | | |
| YS870/10 | B870 | Cooling water | Thermal expansion | VW870A | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS870/11 | B870 | Cooling water | Thermal expansion | VW870B | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS870/12 | B870 | Cooling water | Thermal expansion | VW870C | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS870/13 | B870 | Cooling water | Thermal expansion | W872 | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS870/15 | B870 | Nitrogen | Overpressure | V873A | Sv | 1,5 | | | 50 | 80 | Nitrogen subheader | | Not considered in the study | | | | | | | | | | | | | | |
| YS870/16 | B870 | Cooling water | Thermal expansion | W873A | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS870/17 | B870 | Nitrogen | Overpressure | V873B | Sv | 1,5 | | | 50 | 80 | Nitrogen subheader | | Not considered in the study | | | | | | | | | | | | | | |
| YS870/18 | B870 | Cooling water | Thermal expansion | W873B | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS870/19 | B870 | Nitrogen | Autocontrol valve failure CV87001 | 870.001 | Sv | 0,5 | 500 | 660,5 | 40 | 50 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS881/09 | B881 | Cooling water | Thermal expansion | F880B | Sv | 6,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS881/10 | B881 | Cooling water | Thermal expansion | F880A | Sv | 6,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS881/11 | B881 | Cooling water | Thermal expansion | P880A | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS881/12 | B881 | Cooling water | Thermal expansion | P880B | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS881/13 | B881 | Cooling water | Thermal expansion | P880C | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS881/14 | B881 | Cooling water | Thermal expansion | P880D | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS881/15 | B881 | Cooling water | Thermal expansion | P880E | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS882/01 | B882 | Steam | Steam trap or valve failure. Calculated for FZ44402 | B881 | Sv | 8,0 | 2.000 | 660,5 | 40 | 50 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS882/02 | B882 | Steam | Steam trap or valve failure | B882 | Sv | 8,0 | | | 25 | 40 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS882/04 | B882 | Cooling water | Thermal expansion | 882.020 | Sv | 8.0 | 1.000 | 254.5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 3 |
| YS884/03 | B884 | Nitrogen | Fire case | B800 | Sv | 50,0 | 7.000 | 314,2 | 25 | 50 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS884/04 | B884 | Nitrogen | Autocontrol valve failure PCV88418 | 884.002 | Sv | 40,0 | 1.000 | 113,1 | 15 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS884/05 | B884 | Nitrogen | Fire case | VB800-1 | Sv | 14,0 | | | 12 | 20 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS884/06 | B884 | Nitrogen | Overpressure | VW800-1 | Sv | 50,0 | | | 12 | 20 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS884/07 | B884 | Cooling water | Thermal expansion | VW800-1 | Sv | 8,0 | 1.000 | 254,5 | 25 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| YS884/08 | B884 | Nitrogen | Overpressure | 884.009 | Sv | 50,0 | | | 12 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS884/09 | B884 | Nitrogen | Overpressure | 884.009 | Sv | 50,0 | | | 12 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| SV151.1A | D559 | Propylene | Fire case | PB12A | SV | 20,0 | | | 50,0 | 80,0 | Flare | X | X | X | X | X | X | X | 2.7% | | | | | | | | 5 |
| SV151.1B | D559 | Propylene | Fire case | PB12B | SV | 20,0 | | | 50,0 | 80,0 | Flare | X | X | X | X | X | X | X | 2.7% | | | | | | | | 5 |
| SV151.1C | D559 | Propylene | Fire case | PB12C | SV | 20,0 | | | 50,0 | 80,0 | Flare | X | X | X | X | X | X | X | 2.7% | | | | | | | | 5 |
| SV151.2 | D560 | Propylene | Fire case | 151.816 | SV | 30,0 | | | 15,0 | 25,0 | Flare | X | X | X | X | X | X | X | 1.9% | | | | | | | | 5 |

# PLANTS I&II

| | | | | | | | | | Size | | | | | | | | | Calculations | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Item | P&ID | Fluid | Design case | Equipment protected | Type | Set pressure (barg) | Capacity (kg/h) | Seat area (mm2) | Inlet | Outlet | Discharge | Revision sheet | P&ID | Contingency analysis | Spec. Sheet | Isometrics (Inlet/Outlet) | Equipment documentation (Purchase orders + Inspection records) | Relief load analysis and sizing + "vapor disengagement ISO 4126-10" | Inlet pressure loss, %SP | Outlet pressure loss, %SP | Stability (Chattering) | Engineering Analysis ΔPinlet>3% | Body Bowl Choking | Noise Level | Load/Momentum (Static/Dynamic) | Acoustic Induced Vibration | Revision period RV IFT v2.0 years |
| SV42.01 | D451 | Propylene | Thermal expansion | P10B | Safety valve | 59,5 | 150 | 254,0 | 25 | 50 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| SV42.02 | D451 | Propylene | Thermal expansion | P10A | Safety valve | 59,5 | 150 | 254,0 | 25 | 50 | Flare | X | X | X | X | | X | X | | | | | | | | | 3 |
| SV42.03 | D451 | Propylene | Thermal expansion | 06.404 | Safety valve | 59,5 | 150 | 254,0 | 15 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| SV42.04 | D691 | Propylene | Thermal expansion | 54.044 | Safety valve | 30,0 | | | 25 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| SV42.05 | D691 | Propylene | Fire case | W10 | Safety valve | 50,0 | | | 25 | 25 | Flare | X | X | X | X | | X | X | 1.6% | | | | | | | | 5 |
| SV42.24 | D691 | Propylene | Fire case | P11A | Safety valve | 20,0 | | | 50 | 50 | Flare | X | X | X | X | | X | X | 0.2% | | | | | | | | 5 |
| SV42.25 | D691 | Propylene | Fire case | P11B | Safety valve | 20,0 | | | 50 | 50 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| SV42.26 | D691 | Propylene | Thermal expansion | 06.405 | Safety valve | 34,0 | 150 | 254,0 | 25 | 25 | Flare | Not considered in the study | | | | | | | | | | | | | | | |
| SV42.27 | D691 | Nitrogen | Mechanical seal rupture/CVFO | PB11A/B | Safety valve | 17,0 | | | 10 | 10 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y06 | D451 | Propylene | Fire case / Thermal | F10 | Safety valve | 45,0 | | | 15 | 25 | Flare | x | | | | | | | | | | | | | | | |
| Y07 | D452 | Propylene | Thermal expansion | 48.826 | Safety valve | 45,0 | 150 | 254,0 | 15 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y10 | D452 | Propylene | Thermal expansion | 07.801 | Safety valve | 59,5 | 150 | 254,0 | 15 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y11 | D452 | Nitrogen | PC-0140 failure, fire | F15 | Safety valve | 1,0 | 100 | 254,0 | 25 | 40 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y12 | D452 | Nitrogen | FIC-0140 failure | V15 | Safety valve | 1,0 | 100 | 254,0 | 25 | 40 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y13A | D452 | Propylene | Fire, abnormal heat in., thermal exp. | K10A | Safety valve | 45,0 | 5.530 | 254,0 | 25 | 50 | Flare | X | X | X | X | X | X | X | 15.9% | 20.2% | X | X | X | X | X | X | 5 |
| Y13B | D452 | Propylene | Fire, abnormal heat in., thermal exp. | K10A | Safety valve | 45,0 | 5.530 | 254,0 | 25 | 50 | Flare | X | X | X | X | X | X | X | 15.9% | 20.2% | X | X | X | X | X | X | 5 |
| Y14A | D452 | Propylene | Fire, abnormal heat in., thermal exp. | K10B | Safety valve | 45,0 | 5.530 | 254,0 | 25 | 50 | Flare | X | X | X | X | X | X | X | 6.9% | 20.5% | X | X | X | X | X | X | 5 |
| Y14B | D452 | Propylene | Fire, abnormal heat in., thermal exp. | K10B | Safety valve | 45,0 | 5.530 | 254,0 | 25 | 50 | Flare | X | X | X | X | X | X | X | 6.9% | 20.5% | X | X | X | X | X | X | 5 |
| Y15a | D448 | Propylene | Fire, thermal expansion | K11A | Safety valve | 45,0 | 8.782 | 254,0 | 25 | 40 | Flare | X | X | X | X | X | X | X | 1.8% | 23.5% | | | | X | X | X | 5 |
| Y15b | D448 | Propylene | Fire, thermal expansion | K11A | Safety valve | 45,0 | 8.782 | 254,0 | 25 | 40 | Flare | X | X | X | X | X | X | X | 1.8% | 23.5% | | | | X | X | X | 5 |
| Y15c | D448 | Propylene | Fire, thermal expansion | K11B | Safety valve | 45,0 | 8.782 | 254,0 | 25 | 40 | Flare | X | X | X | X | X | X | X | 1.7% | 25.4% | | | | X | X | X | 5 |
| Y15d | D448 | Propylene | Fire, thermal expansion | K11B | Safety valve | 45,0 | 8.782 | 254,0 | 25 | 40 | Flare | X | X | | X | X | X | X | 1.7% | 25.4% | | | | X | X | X | 5 |
| Y30 | D560 | Mineral oil | Fire case | K30 | Safety valve | 5,5 | | 471,0 | 50 | 80 | Flare | Heavy component needs more than 30 min to start boiling. Not considered fire case. This valve was designed originally for containing heptane in protected vessel. | | | | | | | | | | | | | | | |
| Y31 | D454 | Heptane | Fire case | B32 | Safety valve | 6,0 | 7.804 | 64 | 15 | 25 | Flare | X | X | X | X | X | X | X | 0.35% | > 100% | | | | X | X | X | 5 |
| Y34A | D454 | Mineral oil | Failure LS0310/ Fire case | B30 | Safety valve | 5,5 | | 855 | 80 | 100 | Flare | Heavy component needs more than 30 min to start boiling. Not considered fire case. This valve was designed originally for containing heptane in protected vessel. | | | | | | | | | | | | | | | |
| Y34B | D454 | Mineral oil | Fire case | B710 | Safety valve | 6,0 | | 855 | 80 | 100 | Flare | Heavy component needs more than 30 min to start boiling. Not considered fire case. This valve was designed originally for containing heptane in protected vessel. | | | | | | | | | | | | | | | |
| Y70 | D561 | Nitrogen | Failure of PIC0710 | B231 | Safety valve | 1,1 | 2,5 | 254,0 | 25 | 40 | Burning pit | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y71 | D561 | TEAL | Fire, autocontrol valve failure | B230 | Safety valve | 1,1 | 2,5 | 254,0 | 25 | 40 | Burning pit | Not considered in the study | | | | | | | | | | | | | | | |
| Y72 | D561 | TEAL | Fire case (dismantled equipment) | Out of service (Not dismantled due to TEAL handling risk) | | | | | | | | | | | | | | | | | | | | | | | 5 |
| Y80 | D459 | Atmer | Overpressure | P80/1 | Safety valve | 100,0 | | | 15 | 15 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y82 | D459 | Heating Water | Thermal expansion | W85 | Safety valve | 6,0 | | | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y83 | D459 | Heating Water | Thermal expansion | P85 | Safety valve | 5,0 | | 64 | 25 | 25 | Atm | | | | | | | | | | | | | | | | 5 |
| Y84 | D459 | Steam | TIC806 fails open | W85 | Safety valve | 4,0 | | 491 | 25 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y85 | D459 | Atmer | Blocked outlet | P281 | Safety valve | 100,0 | | | 15 | 20 | Atm | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y86 | D459 | Atmer | Blocked outlet | P86 | Safety valve | 160,0 | | 95 | 15 | 20 | Atm | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y100A | D462 | Propylene | Thermal expansion | P103A | Safety valve | 54,0 | | 64 | 25 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y100B | D462 | Propylene | Thermal expansion | P103B | Safety valve | 54,5 | | 64 | 25 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y101 | D461 | Propylene | Reaction | R100 | Safety valve | 40,0 | 5.000 | 471 | 50 | 80 | Flare | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y102 | D461 | Propylene | Reaction | R100 | Safety valve | 40,0 | 5.000 | 471 | 50 | 80 | Flare | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y103 | D462 | Propylene | Fire, exchanger tube breakage | W101 | Safety valve | 40,0 | | | 40 | 50 | Flare | X | X | X | X | | X | | 1.2% | 27.2% | | | | X | X | X | 5 |
| Y104 | D462 | Propylene | Fire, thermal expansion | W100 | Safety valve | 39,5 | 160 | 64 | 25 | 25 | Flare | X | X | X | X | | X | X | | 25.8% | | | | X | X | X | 2 |
| Y105 | D462 | Propylene | Fire, thermal expansion | W100 | Safety valve | 40,0 | 160 | 64 | 25 | 25 | Flare | X | X | X | X | | X | X | | 25.8% | | | | X | X | X | 5 |
| Y106 | D462 | Propylene | Fire, blocked outlet | P100A | Safety valve | 44,5 | 10.800 | 254 | 25 | 50 | Flare | X | X | X | X | | X | X | 11.8% | 13.1% | X | X | X | X | X | X | 2 |
| Y107 | D462 | Propylene | Fire, blocked outlet | P100B | Safety valve | 44,5 | 10.800 | 254 | 25 | 50 | Flare | X | X | X | X | | X | X | 11.8% | 13.1% | X | X | X | X | X | X | 3 |

| | | | | | | | | | Size | | | | | | | | | Calculations | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Item | P&ID | Fluid | Design case | Equipment protected | Type | Set pressure (barg) | Capacity (kg/h) | Seat area (mm2) | Inlet | Outlet | Discharge | Revision sheet | P&ID | Contingency analysis | Spec. Sheet | Isometrics (Inlet/Outlet) | Equipment documentation (Purchase orders + Inspection records) | Relief load analysis and sizing + "vapor disengagement ISO 4126-10" | Inlet pressure loss, %SP | Outlet pressure loss, %SP | Stability (Chattering) | Engineering Analysis ΔPinlet>3% | Body Bowl Choking | Noise Level | Load/Momentum (Static/Dynamic) | Acoustic Induced Vibration | Revision period RV IFT v2.0 years |
| Y108 | D462 | Cooling water | Thermal expansion | W100 | Safety valve | 6,0 | | 64 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y109 | D462 | Cooling water | Thermal expansion | W101 | Safety valve | 6,0 | | | 32 | 32 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y111 | D449 | Catalyst suspension | Overpressure | R111 | Safety valve | 6,0 | | | 25 | 40 | Atm | Dismantled | | | | | | | | | | | | | | | |
| Y112 | D450 | Catalyst suspension | Thermal expansion | R115 | Safety valve | 3,0 | | | 25 | 40 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y114 | D465 | Nitrogen | Autocontrol valve failure PC1063 | B108 | Safety valve | 50,0 | 100 | 254,0 | 25 | 50 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y115 | D465 | Mineral oil | Blocked outlet | P108 | Safety valve | 50,0 | 180 | 254,0 | 25 | 50 | Flare | Not considered in the study. Included in pump P108 | | | | | | | | | | | | | | | |
| Y116 | D465 | Cooling water | Thermal expansion | W108 | Safety valve | 8,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y117 | D465 | Cooling water | Thermal expansion | W107 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y119 | D465 | Mineral oil | Blocked outlet | P109 | Safety valve | 55,0 | | 64,0 | 25 | 25 | Atm | Not considered in the study. Included in pump P109 | | | | | | | | | | | | | | | |
| Y121 | D466 | Propylene | Failure of PC-1205 | B121 | Safety valve | 5,5 | 20 | 64 | 15 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y123 | D469 | Nitrogen | Blocked outlet | V120A | Safety valve | 1,0 | | | 25 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y124 | D469 | Nitrogen | Blocked outlet | V120B | Safety valve | 1,0 | | | 25 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y125 | D469 | Cooling water | Thermal expansion | W120 | Safety valve | 6,0 | | 64 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y126 | D469 | Nitrogen | Autocontrol valve failure PIC1301 | F123A/B | Safety valve | 1,0 | | 64 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y127 | D467 | Nitrogen | Autocontrol valve failure PC1235 | FB122 | Safety valve | 5,0 | | | 25 | 80 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y130 | D471 | Nitrogen | Autocontrol valve failure PC1330 | FB130 | Safety valve | 6,0 | | | 20 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| KT130 | D471 | Nitrogen | Overpressure | B130 | Kito | -0,02/+0,05 | | 300 | 300 | 300 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y131 | D471 | Nitrogen | Autocontrol failure PC1331 | FB131 | Safety valve | 6,0 | | | 20 | 25 | Atm | X | X | X | X | | | X | | | | | | | | | 5 |
| KT131 | D471 | Nitrogen | Overpressure | B131 | Kito | -0,02/+0,05 | | 300 | 300 | 300 | Atm | | | | | | | | | | | | | | | | |
| YS143 | D472 | Conveying air | Overpressure | B26 | Kito | -0,02/+0,05 | | | | | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| YS144 | D472 | Conveying air | Overpressure | B26 | Kito | -0,02/+0,05 | | | | | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y148 | D474 | Heating oil | Blocked outlet | W140 | Safety valve | 9,5 | | | 25 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y149 | D474 | Heating oil | Thermal expansion | W140 | Safety valve | 9,5 | | | 25 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y156 | D486 | Cooling water | Thermal expansion | W2422 | Safety valve | 6,0 | | 64,0 | 50 | 50 | Atm | X | X | X | X | | X | X | | | | | | | | | 2 |
| Y157 | D480 | Cooling water | Thermal expansion | W157 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y158 | D480 | Cooling water | Thermal expansion | W158 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y160 | D481 | Steam | Failure of FC-1607 | T160 | Safety valve | 0,4 | 400 | 1521,0 | 80 | 150 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| KT160 | D481 | Nitrogen | Overpressure | T160 | Kito | -0,03/+0,08 | | 300 | 300 | 300 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y164 | D481 | Air | Failure of PC-1640 | FB164 | Safety valve | 6,0 | | | 25 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| SV170 | D485 | Air | Autocontrol valve failure | H170A/B/C | Safety valve | 3,0 | | | | | Atm | X | X | X | X | | X | X | | | | | | | | | |
| SV171 | D725 | Air | Autocontrol valve failure | H171 | Safety valve | 3,0 | | | | | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y185 | D477 | Cooling water | Thermal expansion | W185 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y186 | D477 | Organic vapour | Fire, blocked outlet | B185 | Safety valve | 0,5 | | | 50 | 80 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y286 | D459 | Atmer | Overpressure | P286 | Safety valve | 160,0 | | | 15 | 20 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| KV330 | D575 | Nitrogen | Overpressure | B330 | Kito | -0,025/0,05 | | 300 | 300 | 300 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| KV331 | D575 | Nitrogen | Overpressure | B331 | Kito | -0,025/0,05 | | 300 | 300 | 300 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y400 | D0792 | Nitrogen | Failure of PC79201, fire, | B400 | Safety valve | 0,3 | | | 15 | 15 | Flare/atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y401/04 | D461 | Nitrogen | Ovepressure | N2 bottles | Safety valve | 160,0 | | | 12 | 12 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y411/04 | D565 | Nitrogen | Ovepressure | N2 bottles | Safety valve | 160,0 | | | 12 | 12 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y521 | D483 | Water | Thermal expansion | W521 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y522 | D483 | Liquid organic waste | Fire | B521A | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y523 | D483 | Liquid organic waste | Fire | B521B | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y540 | D468 | Propylene | Autocontrol valve failure of PC-1904 | W540 | Safety valve | 1,5 | | 1.134 | 80 | 125 | Flare | Dismantled | | | | | | | | | | | | | | | |
| Y541 | D468 | Propylene | Autocontrol valve failure of PC-1904 | W540 | Safety valve | 1,5 | | 1.134 | 80 | 125 | Flare | Dismantled | | | | | | | | | | | | | | | |

| | | | | | | | | | Size | | | | | | | | | Calculations | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Item | P&ID | Fluid | Design case | Equipment protected | Type | Set pressure (barg) | Capacity (kg/h) | Seat area (mm2) | Inlet | Outlet | Discharge | Revision sheet | P&ID | Contingency analysis | Spec. Sheet | Isometrics (Inlet/Outlet) | Equipment documentation (Purchase orders + Inspection records) | Relief load analysis and sizing + "vapor disengagement ISO 4126-10" | Inlet pressure loss, %SP | Outlet pressure loss, %SP | Stability (Chattering) | Engineering Analysis ΔPinlet>3% | Body Bowl Choking | Noise Level | Load/Momentum (Static/Dynamic) | Acoustic Induced Vibration | Revision period RV IFT v2.0 years |
| Y542A | D468 | Propylene | Overpressure | W541A | Safety valve | 3,5 | | | 32 | 40 | Flare | Package unit | | | | | | | | | | | | | | | |
| Y543A | D468 | Propylene | Overpressure | W541A | Safety valve | 3,5 | | | 32 | 40 | Flare | Package unit | | | | | | | | | | | | | | | |
| Y542B | D468 | Propylene | Overpressure | W541B | Safety valve | 3,5 | | | 32 | 40 | Flare | Package unit | | | | | | | | | | | | | | | |
| Y543B | D468 | Propylene | Overpressure | W541B | Safety valve | 3,5 | | | 32 | 40 | Flare | Package unit | | | | | | | | | | | | | | | |
| Y544 | D468 | Propylene | Overpressure | B542 | Safety valve | 6,0 | | | 25 | 40 | Flare | Package unit | | | | | | | | | | | | | | | |
| Y545 | D468 | Propylene | Overpressure | B542 | Safety valve | 6,0 | | | 25 | 40 | Flare | Package unit | | | | | | | | | | | | | | | |
| Y546 | D468 | Propylene | Overpressure | F545 | Safety valve | 6,0 | | | 25 | 25 | Flare | Package unit | | | | | | | | | | | | | | | |
| Y548 | D468 | Cooling water | Thermal expansion | W540 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | Dismantled | | | | | | | | | | | | | | | |
| Y549 | D468 | Cooling water | Thermal expansion | W541 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y551 | D468 | Cooling water | Thermal expansion | W542 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y553 | nsacion gas | Propylene | Thermal expansion | 45.450 | Safety valve | 40,0 | | | | | Flare | Not considered in the study | | | | | | | | | | | | | | | |
| Y657 | D591 | Propylene | Thermal expansion | W657 | Safety valve | 20,0 | | | 25 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y810 | D489 | Nitrogen | Autocontrol failure of PIRCA-8101 | 81.502 | Safety valve | 4,0 | 375 | 471,0 | 50 | 80 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y811 | D489 | Cooling water | Thermal expansion | V811 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y812 | D489 | Nitrogen | Blocked outlet | B811 | Safety valve | 46,0 | | | | | Atm | Package unit V811 | | | | | | | | | | | | | | | |
| Y813 | D489 | Nitrogen | Overpressure | W811 | Safety valve | 46,0 | | | | | Atm | Package unit V811 | | | | | | | | | | | | | | | |
| Y814 | D489 | Nitrogen | Overpressure | B812 | Safety valve | 15,0 | | | | | Atm | Package unit V811 | | | | | | | | | | | | | | | |
| Y815 | D489 | Cooling water | Thermal expansion | V812 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y816 | D489 | Nitrogen | Blocked outlet | VB812-1 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | Package unit | | | | | | | | | | | | | | | |
| Y817 | D489 | Nitrogen | Blocked outlet | VW812-1 | Safety valve | 15,0 | | 64,0 | 15 | 25 | Atm | Package unit | | | | | | | | | | | | | | | |
| Y818 | D489 | Nitrogen | Blocked outlet | VW812-2 | Safety valve | 49,0 | | 64,0 | 15 | 25 | Atm | Package unit | | | | | | | | | | | | | | | |
| Y821 | D490 | Compressed air | Autocontrol failure of PIRCA-8200 | 82.330 | Safety valve | 4,0 | 130 | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y822 | D490 | Compressed air | Autocontrol failure of PIRCA-8201 | 82.332 | Safety valve | 4,0 | 26 | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y840 | D492 | Steam | Failure of PV-8101 | 84.209 | Safety valve | 20,0 | 100 | 254,0 | 25 | 40 | Atm | Dismantled | | | | | | | | | | | | | | | |
| Y841 | D492 | Steam | Overpressure | B841 | Safety valve | 7,0 | 300 | 254,0 | 25 | 40 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y842 | D492 | Steam | Overpressure | B842 | Safety valve | 7,0 | 500 | 254,0 | 25 | 40 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y843 | D492 | Cooling water | Thermal expansion | W543 | Safety valve | 6,0 | | 64,0 | 20 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y844 | D489 | Nitrogen | Overpressure | N2 bottles | Safety valve | 45,0 | | | 10/12 | 10/12 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y845 | | Compressed air | Overpressure | Inst. air header | Safety valve | 4,0 | | | 25 | 25 | Atm | Package unit | | | | | | | | | | | | | | | |
| Y846 | D490 | Nitrogen | Failure of PC | B250A/B25A | Safety valve | 6,0 | | | 25 | 25 | Atm | Dismantled | | | | | | | | | | | | | | | |
| HS1001 | D461 | Propylene | Overpressure | R100 | Ball valve | 0-45 | 10.000 | KV | 100 | 100 | Flare | Not considered in the study | | | | | | | | | | | | | | | |
| Y1123 | D452 | Propylene | Fire case | K10C | Safety valve | 45,0 | 8.782 | 254,0 | 25 | 40 | Flare | X | X | X | X | X | X | X | 2.3 | 31.5 | | | | X | X | X | 5 |
| Y1124 | D452 | Propylene | Fire case | K10C | Safety valve | 45,0 | 8.782 | 254,0 | 25 | 40 | Flare | X | X | X | X | X | X | X | 2.3 | 31.5 | | | | X | X | X | 5 |
| Y1125 | D452 | Propylene | Thermal expansion | 112.810 | Safety valve | 60,0 | | | 15 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y1126 | D452 | Propylene | Thermal expansion | 07.804 | Safety valve | 60,0 | | | 15 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y1311 | D558 | Cooling water | Thermal expansion | W20 | Safety valve | 6,0 | | 64,0 | 20 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y1312 | D558 | Ethylene | Fire, abnormal heat input | K20 | Safety valve | 45,00 | 360 | 44 | 15 | 25 | Flare | X | X | X | X | X | X | X | 2.0 | 16.9 | | | | X | X | X | 5 |
| Y1313 | D558 | Ethylene | Overpressure | V20 | Safety valve | 45,00 | | | 25 | 25 | Flare | Package unit V20 | | | | | | | | | | | | | | | |
| Y1314 | D558 | Ethylene | Overpressure | V20 | Safety valve | 30,00 | | | 25 | 25 | Flare | Package unit V20 | | | | | | | | | | | | | | | |
| Y1315 | D558 | Cooling water | Thermal expansion | V20 | Safety valve | 6,0 | | 64,0 | 20 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y1521 | D560 | Mineral oil | Thermal expansion | K25 | Safety valve | 45,0 | 234 | 44,0 | 15 | 25 | Flare | Heavy component needs more than 30 min to start boiling. Not considered fire case. This valve was designed originally for containing heptane in protected vessel. | | | | | | | | | | | | | | | |
| Y1522 | D560 | Mineral oil | Overpressure | P26 | Safety valve | 9,0 | | | 25 | 40 | Suction | Heavy component needs more than 30 min to start boiling. Not considered fire case. This valve was designed originally for containing heptane in protected vessel. | | | | | | | | | | | | | | | |
| Y1576 | D576 | Nitrogen | Blocked outlet | F342 | Safety valve | 0,2 | | | 25 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 3 |
| SV2501 | D564 | Nitrogen | Blocked outlet | F250A | Safety valve | 5,0 | | | 15 | 25 | Flare | Not considered in the study | | | | | | | | | | | | | | | |

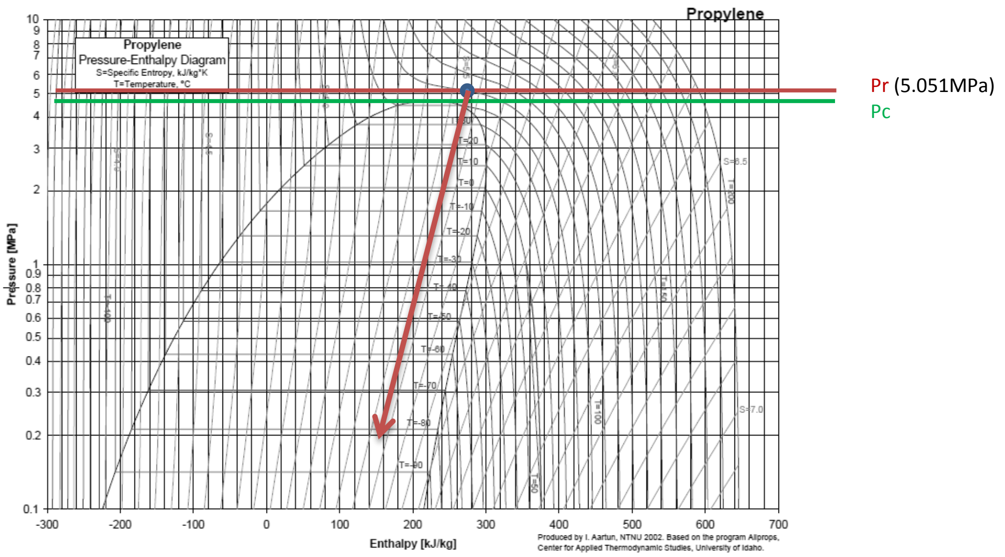
| | | | | | | | | | Size | | | | | | | | | Calculations | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Item | P&ID | Fluid | Design case | Equipment protected | Type | Set pressure (barg) | Capacity (kg/h) | Seat area (mm2) | Inlet | Outlet | Discharge | Revision sheet | P&ID | Contingency analysis | Spec. Sheet | Isometrics (Inlet/Outlet) | Equipment documentation (Purchase orders + Inspection records) | Relief load analysis and sizing + "vapor disengagement ISO 4126-10" | Inlet pressure loss, %SP | Outlet pressure loss, %SP | Stability (Chattering) | Engineering Analysis ΔPinlet>3% | Body Bowl Choking | Noise Level | Load/Momentum (Static/Dynamic) | Acoustic Induced Vibration | Revision period RV IFT v2.0 years |
| SV2502 | D564 | Nitrogen | Blocked outlet | F250B | Safety valve | 5,0 | | | 15 | 25 | Flare | Not considered in the study | | | | | | | | | | | | | | | |
| SV2503 | D564 | Nitrogen | Blocked outlet | F250C | Safety valve | 5,0 | | | 15 | 25 | Flare | Not considered in the study | | | | | | | | | | | | | | | |
| Y3011 | D565 | Propylene | Reaction | R300 | Safety valve | 40,0 | 5.000 | 254 | 25 | 40 | Flare | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y3012 | D565 | Propylene | Reaction | R300 | Safety valve | 40,0 | 5.000 | 254 | 25 | 40 | Flare | X | X | X | X | | X | X | | | | | | | | | 2 |
| HS30103 | D565 | Propylene | Overpressure | R300 | Ball valve | 0-40 | 10.000 | KV | 100 | 100 | Flare | Not considered in the study | | | | | | | | | | | | | | | |
| Y3021 | D0832 | Propylene | Fire, blocked outlet | P300A | Safety valve | 45,0 | 38.000 | 471 | 40 | 65 | Flare | X | X | X | X | X | X | X | 4.1 | | X | X | X | X | X | X | 3 |
| Y3022 | D0832 | Propylene | Fire, blocked outlet | P300B | Safety valve | 45,0 | 38.000 | 471 | 40 | 65 | Flare | X | X | X | X | X | X | X | 4.1 | | X | X | X | X | X | X | 3 |
| Y3024 | D566 | Propylene | Thermal expansion | W300A/B | Safety valve | 40,0 | | 64,0 | 25 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y3025 | D566 | Propylene | Thermal expansion | W300A/B | Safety valve | 40,0 | | 64,0 | 25 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y3026A | D0832 | Propylene | Blocked outlet | P301A | Safety valve | 45,0 | | 64,0 | 25 | 40 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3026B | D0832 | Propylene | Blocked outlet | P301B | Safety valve | 45,0 | | 64,0 | 25 | 40 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3027 | D566 | Propylene | Thermal expansion | W300A/B | Safety valve | 40,0 | | 64,0 | 20 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3028 | D0832 | Propylene | Thermal expansion | 302.819 | Safety valve | 40,0 | | 64,0 | 20 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3029A | D0832 | Propylene | Mechanical failure | PB301A | Safety valve | 2,5 | | | 15 | 15 | Flare | Package unit P300A | | | | | | | | | | | | | | | |
| Y3029B | D0832 | Propylene | Mechanical failure | PB301B | Safety valve | 2,5 | | | 15 | 15 | Flare | Package unit P300B | | | | | | | | | | | | | | | |
| Y3031 | D567 | Mineral oil | Thermal expansion | 303.808 | Safety valve | 7,0 | | | 20 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 2 |
| Y3032 | D567 | Catalyst suspension | Blocked outlet | R301 | Safety valve | 6,0 | | | 25 | 40 | Atm | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y3033 | D567 | Cooling water | Thermal expansion | R301 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3034 | D566 | Propylene | Fire case | W301 | Safety valve | 40,0 | | | 40 | 40 | Flare | X | X | X | X | X | X | X | 0.7 | 28 | | | | X | X | X | 5 |
| Y3035 | D566 | Cooling water | Thermal expansion | W301 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3044 | D568 | Nitrogen | Overpressure | R301/R305 | Safety valve | 3,0 | | | 25 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y3081 | D570 | Cooling water | Thermal expansion | W307 | Safety valve | 10,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y3082 | D570 | Cooling water | Thermal expansion | W308 | Safety valve | 10,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3083 | D570 | Nitrogen | Overpressure | B308 | Safety valve | 52,0 | | 64,0 | 15 | 25 | Flare | Package unit RGD300 | | | | | | | | | | | | | | | |
| Y3084 | D588 | Propylene | Fire case | F393 | Safety valve | 6 | | | 20 | 25 | Flare | Dismantled | | | | | | | | | | | | | | | |
| Y3209A | D462 | Propylene | Mechanical seal failure | PB103A | Safety valve | 2,5 | | | 15 | 15 | Flare | Package unit P103A | | | | | | | | | | | | | | | |
| Y3209B | D462 | Propylene | Mechanical seal failure | PB103B | Safety valve | 2,5 | | | 15 | 15 | Flare | Package unit P103B | | | | | | | | | | | | | | | |
| Y3211 | D571 | Propylene | Autocontrol failure of PIC32105 | B321 | Safety valve | 6,0 | 14 | 44,0 | 15 | 15 | Flare | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y3220 | D572 | Nitrogen | Autocontrol failure of PIC32209 | FB322 | Safety valve | 5,0 | | | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3225 | D573 | Nitrogen | Blocked outlet | F323 | Safety valve | 1,0 | | | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 4 |
| Y3230 | D573 | Nitrogen | Overpressure | FB323 | Safety valve | 5,0 | | | 15 | 25 | Atm | Dismantled | | | | | | | | | | | | | | | |
| Y3231 | D573 | Nitrogen | Autocontrol failure PCV32301 | FB323 | Safety valve | 1,0 | 25 | 44,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3232 | D573 | Cooling water | Thermal expansion | W320 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3233 | D573 | Cooling water | Thermal expansion | W325A | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3234 | D573 | Cooling water | Thermal expansion | W325B | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y3301 | D575 | Nitrogen | Overpressure | FB330 | Safety valve | 5,0 | | | 15 | 25 | Atm | Package unit F330 | | | | | | | | | | | | | | | |
| Y3310 | D575 | Nitrogen | Overpressure | FB331 | Safety valve | 5,0 | | | 15 | 25 | Atm | Package unit F331 | | | | | | | | | | | | | | | |
| Y3421 | D577 | Cooling water | Thermal expansion | S340 Barrel#1 | Safety valve | 10,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3430 | D577 | Lube oil | Overpressure | P347 | Safety valve | 10,0 | | | 15 | 25 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y3511 | D581 | Cooling water | Thermal expansion | W357 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3611 | D582 | Steam | Blocked outlet | T360 | Safety valve | 0,4 | 2.500 | 6793,0 | 150 | 250 | Atm | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y3640 | D582 | Compressed air | Blocked outlet | FB364 | Safety valve | 5,0 | | | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3811 | D587 | Cooling water | Thermal expansion | W380 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3812 | D587 | Cooling water | Thermal expansion | W385 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 3 |

| | | | | | | | | | Size | | | | | | | | | Calculations | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Item | P&ID | Fluid | Design case | Equipment protected | Type | Set pressure (barg) | Capacity (kg/h) | Seat area (mm2) | Inlet | Outlet | Discharge | Revision sheet | P&ID | Contingency analysis | Spec. Sheet | Isometrics (Inlet/Outlet) | Equipment documentation (Purchase orders + Inspection records) | Relief load analysis and sizing + "vapor disengagement ISO 4126-10" | Inlet pressure loss, %SP | Outlet pressure loss, %SP | Stability (Chattering) | Engineering Analysis ΔPinlet>3% | Body Bowl Choking | Noise Level | Load/Momentum (Static/Dynamic) | Acoustic Induced Vibration | Revision period RV IFT v2.0 years |
| Y3911 | D588 | Propylene | Failure of PC-39104 | F390A/B | Safety valve | 1,5 | | 1.520 | 80 | 150 | Flare | Dismantled | | | | | | | | | | | | | | | |
| Y3912 | D588 | Propylene | Failure of PC-39104 | F390A/B | Safety valve | 1,5 | | 1.520 | 80 | 150 | Flare | Dismantled | | | | | | | | | | | | | | | |
| Y3913 | D588 | Cooling water | Thermal expansion | W390 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | Dismantled | | | | | | | | | | | | | | | |
| Y3915 | D588 | Propylene | Overpressure | VB390-1 | Safety valve | 5,0 | | | 40 | 80 | Flare | Package unit V390 | | | | | | | | | | | | | | | |
| Y3916 | D588 | Propylene | Overpressure | VB390-1 | Safety valve | 5,0 | | | 40 | 80 | Flare | Package unit V390 | | | | | | | | | | | | | | | |
| Y3917 | D588 | Propylene | Overpressure | B392 | Safety valve | 22,0 | | | 40 | 40 | Flare | Package unit V390 | | | | | | | | | | | | | | | |
| Y3918 | D588 | Propylene | Overpressure | B392 | Safety valve | 22,0 | | | 40 | 40 | Flare | Package unit V390 | | | | | | | | | | | | | | | |
| Y3919 | D588 | Cooling water | Thermal expansion | W392 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y3920 | D588 | Propylene | Heat exchanger tube breakage | W393 | Safety valve | 3,0 | 1.380 | 1.520 | 80 | 150 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3921 | D588 | Propylene | heat exchanger tube breakage | W393 | Safety valve | 3,0 | 1.380 | 1.520 | 80 | 150 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3922 | D588 | Cooling water | Thermal expansion | W391 | Safety valve | 4,5 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3923 | D588 | Cooling water | Fire | V390 suction | Safety valve | 6,0 | | | 25 | 40 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3924 | D588 | Propylene | Fire, thermal expansion | 391.801 | Safety valve | 22,0 | | 64 | 20 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3925 | D588 | Propylene | Fire case | B393 | Safety valve | 24,0 | | | 25 | 40 | Flare | X | X | X | X | X | X | X | 5.3 | 51.6 | X | X | X | X | X | X | 5 |
| Y3926 | D730 | Cooling water | Thermal expansion | W395 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y3927 | D730 | Cooling water | Thermal expansion | W394 | Safety valve | 6,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| Y3928 | D730 | Propylene | Blocked outlet, fire | B398 | Safety valve | 22,0 | | | 25 | 40 | Flare | Package unit V391 | | | | | | | | | | | | | | | |
| Y3929 | D730 | Propylene | Blocked outlet, fire | B398 | Safety valve | 22,0 | | | 25 | 40 | Flare | Package unit V391 | | | | | | | | | | | | | | | |
| Y3930 | D730 | Propylene | Blocked outlet, fire | B396 | Safety valve | 5,0 | | | 50 | 65 | Flare | Package unit V391 | | | | | | | | | | | | | | | |
| Y3931 | D730 | Propylene | Blocked outlet, fire | B396 | Safety valve | 5,0 | | | 50 | 65 | Flare | Package unit V391 | | | | | | | | | | | | | | | |
| Y3932 | D730 | Cooling water | Thermal expansion | V391 | Safety valve | 4,0 | | 64,0 | 15 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y3952 | D730 | Propylene | Fire, thermal expansion | W395 | Safety valve | 20,0 | | | 20 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y6211 | D590 | Steam | Blocked outlet | W620 | Safety valve | 6,0 | 400 | 254,0 | 25 | 40 | Atm | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y6214 | D590 | Cooling water | Thermal expansion | W621 | Safety valve | 6,0 | 7.500 | 64,0 | 25 | 40 | Atm | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y6510A | D591 | Propylene | Fire, cooling water failure | K655A/B | Safety valve | 20,0 | 35.750 | 2734,0 | 100 | 150 | Flare | X | X | X | X | X | X | X | 2.9 | 28.9 | | X | | X | X | X | 5 |
| Y6510B | D591 | Propylene | Fire, cooling water failure | K655A/B | Safety valve | 20,0 | 35.750 | 2734,0 | 100 | 150 | Flare | X | X | X | X | X | X | X | 2.9 | 28.9 | | X | | X | X | X | 5 |
| Y6511 | D591 | Cooling water | Thermal expansion | W658 | Safety valve | 8,0 | | 64,0 | 20 | 25 | Atm | X | X | X | X | X | X | X | | | | | | | | | 5 |
| Y6512A | D591 | Cooling water | Thermal expansion | W655A | Safety valve | 8,0 | | 50,0 | 20 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y6512B | D591 | Cooling water | Thermal expansion | W655B | Safety valve | 8,0 | | 50,0 | 20 | 25 | Atm | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y6513A | D591 | Propylene | Thermal expansion | P655A | Safety valve | 25,0 | | | 20 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y6513B | D591 | Propylene | Thermal expansion | P655B | Safety valve | 25,0 | | | 20 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y6514A | D591 | Propylene | Thermal expansion | P656A | Safety valve | 47,0 | | | 20 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y6514B | D591 | Propylene | Thermal expansion | P656B | Safety valve | 47,0 | | | 20 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y6515A | D591 | Propylene | Thermal expansion | P657A | Safety valve | 25,0 | | | 20 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y6515B | D591 | Propylene | Thermal expansion | P657B | Safety valve | 25,0 | | | 20 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 3 |
| Y6516A | D591 | Propylene | Thermal expansion | Mech seal P656A | Safety valve | 48,0 | | | 20 | 25 | Flare | Package unit P656A | | | | | | | | | | | | | | | |
| Y6516B | D591 | Propylene | Thermal expansion | Mech seal P656B | Safety valve | 48,0 | | | 20 | 25 | Flare | Package unit P656B | | | | | | | | | | | | | | | |
| Y6516 | D591 | Propylene | Thermal expansion | 391.802 | Safety valve | 22,0 | | | 20 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y6517 | D591 | Propylene | Thermal expansion | 651.806 | Safety valve | 20,0 | | | 20 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y6518 | D591 | Propylene | Thermal expansion | 651.808 | Safety valve | 20,0 | | | 20 | 25 | Flare | X | X | X | X | | X | X | | | | | | | | | 5 |
| Y6519 | D591 | Propylene | Fire, thermal expansion | W656 | Safety valve | 20,0 | | | 50 | 80 | Flare | X | X | X | X | | X | X | | | | | | | | | |
| Y7416 | B433 | Compressed air | Ovepressure | VW375A-1 | Safety valve | 3,0 | | | 15 | 25 | Atm | Package unit | | | | | | | | | | | | | | | |
| Y7417 | B433 | Compressed air | Ovepressure | VW375A-1 | Safety valve | 3,0 | | | 15 | 25 | Atm | Package unit | | | | | | | | | | | | | | | |
| Y7418 | B433 | Compressed air | Ovepressure | VB375B-2 | Safety valve | 3,0 | | | 15 | 25 | Atm | Package unit | | | | | | | | | | | | | | | |
| Y7419 | B433 | Compressed air | Ovepressure | VB375B-1 | Safety valve | 3,0 | | | 15 | 25 | Atm | Package unit | | | | | | | | | | | | | | | |
| Y7420 | B433 | Compressed air | Ovepressure | VW375A-3 | Safety valve | 8,0 | | | 15 | 25 | Atm | Package unit | | | | | | | | | | | | | | | |
| Y7421 | B433 | Compressed air | Ovepressure | VW375B-3 | Safety valve | 8,0 | | | 15 | 25 | Atm | Package unit | | | | | | | | | | | | | | | |
| Y7422 | B433 | Cooling water | Thermal expansion | V375A | Safety valve | 7,0 | | 64,0 | 15 | 25 | Atm | Package unit | | | | | | | | | | | | | | | |
| Y7423 | B433 | Cooling water | Thermal expansion | V375B | Safety valve | 7,0 | | 64,0 | 15 | 25 | Atm | Package unit | | | | | | | | | | | | | | | |
| Y7424 | B433 | Compressed air | Ovepressure | B375 | Safety valve | 8,0 | | | 80 | 125 | Atm | Package unit | | | | | | | | | | | | | | | |
| Y7425 | D723 | Compressed air | Ovepressure | B376 | Safety valve | 8,0 | | | 80 | 125 | Atm | Not considered in the study | | | | | | | | | | | | | | | |
| Y72B | | Propylene | Fire | B72 | Safety valve | 45,0 | 3.000 | 1256.6 | 50 | 80 | Flare | X | X | X | X | | X | X | | 56.4 | X | | | X | X | X | 5 |
| Y12101 | D568A | Catalyst suspension | Ovepressure | P309 | Safety valve | 9,0 | | | 15 | 25 | Suction | Not considered in the study | | | | | | | | | | | | | | | |

# Annex D: Calculation of relief rate due to fluid expansion and external heat for YS 702/01

The design basis for calculating the relief load and required area is exposed in section 6.4.

The following Mollier Diagram shows the relieving pressure and the critical pressure, to confirm that the relieving of supercritical propylene could give retrograde condensation.




Calculation methodology to be followed: Self and Do (2010)[1] and API 520, part I, 2014[2]

Initial parameters:

$$Total\ wetted\ area, A = shell\ area + inlet\ piping\ area + outlet\ piping\ area =$$

$$\pi \cdot h \cdot D + \pi \cdot h \cdot D + \pi \cdot h \cdot D = \pi \cdot 0{,}508 \cdot 3 + \pi \cdot 0{,}0889 \cdot 25 + \pi \cdot 0{,}1683 \cdot 5 = 14{,}41\ m^2$$

$$Q = 43200 \cdot F \cdot A^{0{,}82} = 43200 \cdot 1 \cdot 14{,}41^{0{,}82} = 385113\ W$$

Volume of liquid propylene $= \pi \frac{0.5^2}{4} \cdot 3 + \pi \frac{0.08^2}{4} \cdot 25 + \pi \frac{0.16^2}{4} \cdot 5 = 0.815\ m^3$

The volume occupied by the tubes is neglected as a conservative basis.

With the density at 45,5 ºC, the initial mass is obtained.

$$M\ initial = \rho_{45{,}5\ ^\circ\mathrm{C}} \cdot V_{initial} = 484\ \frac{kg}{m^3} \cdot 0.815\ m^3 = 394\ kg$$

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[1] Freeman Self and Huyen Do (2010) Calculation of relief rate due to fluid expansion and external heat. Revision 4 prepared for the API 2010 summer meeting, August 18, 2010.

[2] API Standard 520 (2014) Sizing, Selection, and Installation of Pressure-relieving Devices, Part I- Sizing and Selection, 9th edition.

Finally, in order to know the duty in each interval it has been calculated:

$$385113\,\frac{J}{s}\,15s = 5776695\,J$$

Summary table:

| Parameter | Value | Units |
|---|---|---|
| A, wetted area | 14,41 | $m^2$ |
| F | 1 | - |
| Q total | 385113 | W |
| V initial | 0.815 | $m^3$ |
| M initial | 394 | kg |
| Q interval | 5776695 | J |

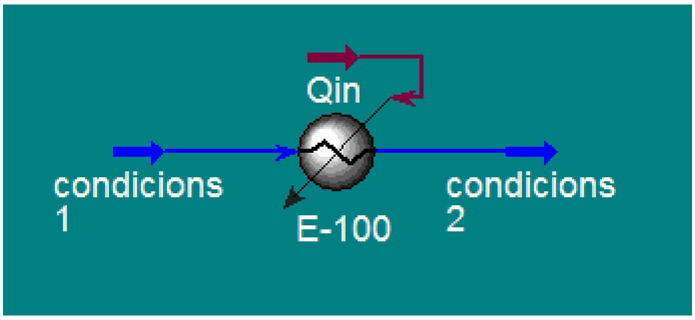
Equations used:

$$Relief\ rate\ \left(\frac{m^3}{h}\right) = V_2 - V_1 = \frac{\left({}^1/_{\rho_2} - {}^1/_{\rho_1}\right)\cdot Q}{H_2 - H_1}$$

$$Relief\ rate\ \left(\frac{kg}{h}\right) = M_1 - M_2 = V_1(\rho_1 - \rho_2)$$

where the vessel volume is V, $m^3$; mass in the volume M, kg; fluid density ρ, $kg/m^3$; enthalpy H, kJ/kg and vessel heat input Q, kJ. Note that time is included in the heat input term.

In order to obtain the properties in each point, the commercial software Aspen Hysys v7.3 is used with the following flowsheet:



Mass and Volumetric relief for each 30 seconds with a constant heat duty:

| **t**<br>**s** | **P**<br>**kPa** | **T**<br>**ºC** | **d**<br>**kg/m³** | **H**<br>**kJ/kg** | **ΔV**<br>**m³** | **ΔM**<br>**kg** | **M**<br>**kg/h** | **Volume flow**<br>**m³/h** | **Mass flow**<br>**kg/h** |
|---|---|---|---|---|---|---|---|---|---|
| 0 | 5051 | 45,50 | 484,40 | 167,42 | 0 | 0 | 394,00 | | |
| 15 | 5051 | 50,49 | 475,03 | 182,08 | 0,01604 | 7,632 | 386,37 | 3,848 | 1831,7847 |
| 30 | 5051 | 55,40 | 465,21 | 197,03 | 0,01717 | 8,0044 | 378,36 | 4,121 | 1921,0451 |
| 45 | 5051 | 60,22 | 454,80 | 212,30 | 0,01862 | 8,485 | 369,88 | 4,468 | 2036,3159 |
| 60 | 5051 | 64,94 | 443,61 | 227,92 | 0,02052 | 9,122 | 360,76 | 4,925 | 2189,3628 |
| 75 | 5051 | 69,54 | 431,36 | 243,94 | 0,02310 | 9,986 | 350,77 | 5,543 | 2396,6743 |
| 90 | 5051 | 73,99 | 417,48 | 260,40 | 0,02704 | 11,312 | 339,46 | 6,489 | 2714,9817 |
| 105 | 5051 | 78,27 | 399,13 | 277,42 | 0,03738 | 14,954 | 324,50 | 8,970 | 3588,8752 |
| 120 | 5051 | 82,37 | 376,98 | 295,22 | 0,04777 | 18,052 | 306,45 | 11,465 | 4332,5181 |
| 135 | 5051 | 86,24 | 351,20 | 314,06 | 0,05973 | 21,012 | 285,44 | 14,335 | 5042,8753 |
| 150 | 5051 | 89,78 | 322,34 | 334,30 | 0,07275 | 23,513 | 261,93 | 17,459 | 5643,2346 |
| 165 | 5051 | 92,83 | 292,59 | 356,38 | 0,08252 | 24,246 | 237,68 | 19,804 | 5819,1348 |
| 180 | 5051 | 95,18 | 258,51 | 380,68 | 0,10709 | 27,777 | 209,90 | 25,703 | 6666,3736 |
| 195 | 5051 | 96,80 | 219,60 | 408,27 | 0,14355 | 31,715 | 178,19 | 34,452 | 7611,7096 |
| 210 | 5051 | 98,77 | 178,29 | 440,69 | 0,18801 | 33,668 | 144,52 | 45,123 | **8080,2891** |
| 225 | 5051 | 104,40 | 140,57 | 480,71 | 0,21723 | 30,739 | 113,78 | **52,135** | 7377,3948 |
| 240 | 5051 | 117,60 | 111,25 | 531,48 | 0,21329 | 23,893 | 89,89 | 51,188 | 5734,2894 |

Once the relief rate has been calculated, the next step is to calculate the required area of the valve. The direct integration method of API 520 (2014) is used.

The maximal mass flux is (in USC units):

$$G^2 = \rho_t^2 \cdot \left[ -9266.1 \cdot \int_{P_0}^{P_t} \frac{dP}{\rho} \right]$$

When this value has been determinated, the required orifice area can be calculated using the following equation (in USC units):

$$A = \frac{0.04 \cdot W}{K_d \cdot K_b \cdot K_c \cdot K_v \cdot G}$$

Where A is the effective discharge area (in$^2$), W the mass flow rate (lb/h), $K_d$ the discharge coefficient, $K_b$ the backpressure correction factor for vapor that should be obtained from the valve manufacturer, $K_c$ the combination correction factor for installations with a rupture disk upstream of the PRV and $K_v$ is the viscosity correction factor.

| **Parameter** | **Value** |
|---|---|
| $K_d$ | 0.8 |
| $K_b$ | 1 |
| $K_c$ | 1 |
| $K_v$ | 1 |

The design pressure is 652.67 psig, being the maximum allowable overpressure 10%, and the relieving pressure is 732.63 psia. The step used, as in the API 520 example, is 4% of the relieving pressure 29.30 psi.

The isentropic flashes calculated are:

Temperature 122.9 ºF (50.49 ºC), mass flow rate 4038.39 lb/h and constant entropy of 26.81 kJ/(kg·C).

| | **T ºF** | **P psia** | **Vapour fraction** | **ρ lb/ft3** | **Integrand $ft^2/s^2$** | **Summation $ft^2/s^2$** | **G $lb/s{\cdot}ft^2$** |
|---|---|---|---|---|---|---|---|
| State 1 | 122,9 | 732,59 | 0,00 | 29,66 | 0 | 0 | 0 |
| State 2 | 122,4 | 703,28 | 0,00 | 29,62 | -4580,9 | -4580,9 | 2835,45 |
| State 3 | 122,0 | 673,97 | 0,00 | 29,59 | -4585,8 | -9166,7 | 4006,68 |
| State 4 | 121,5 | 644,67 | 0,00 | 29,56 | -4590,8 | -13757,5 | 4903,17 |
| State 5 | 121,0 | 615,36 | 0,00 | 29,53 | -4595,8 | -18353,2 | 5657,10 |
| State 6 | 120,6 | 586,06 | 0,00 | 29,50 | -4600,7 | -22954,0 | 6319,70 |
| State 7 | 120,1 | 556,75 | 0,00 | 29,46 | -4605,7 | -27559,7 | 6917,29 |
| State 8 | 119,6 | 527,45 | 0,00 | 29,43 | -4610,7 | -32170,4 | 7465,49 |
| State 9 | 119,1 | 498,14 | 0,00 | 29,40 | -4615,7 | -36786,0 | 7974,53 |
| State 10 | 118,6 | 468,84 | 0,00 | 29,37 | -4620,6 | -41406,7 | 8451,49 |
| State 11 | 118,1 | 439,53 | 0,00 | 29,34 | -4625,6 | -46032,2 | 8901,55 |
| State 12 | 117,6 | 410,23 | 0,00 | 29,31 | -4630,5 | -50662,7 | 9328,63 |
| State 13 | 117,1 | 380,92 | 0,00 | 29,28 | -4635,4 | -55298,1 | 9735,77 |
| State 14 | 116,6 | 351,61 | 0,00 | 29,24 | -4640,3 | -59938,3 | 10125,41 |
| State 15 | 116,1 | 322,31 | 0,00 | 29,21 | -4645,1 | -64583,5 | 10499,51 |
| **State 16** | **115,5** | **293,00** | **0,00** | **29,18** | **-4649,9** | **-69233,4** | **10859,71** |
| State 17 | 111,9 | 263,70 | 0,02 | 24,39 | -5068,4 | -74301,8 | 9403,20 |
| State 18 | 102,5 | 234,39 | 0,07 | 16,14 | -6700,0 | -81001,8 | 6494,99 |
| State 19 | 92,2 | 205,09 | 0,11 | 11,20 | -9931,7 | -90933,5 | 4778,31 |
| State 20 | 80,8 | 175,78 | 0,16 | 7,93 | -14188,7 | -105122,2 | 3637,79 |
| State 21 | 67,9 | 146,48 | 0,20 | 5,61 | -20043,1 | -125165,3 | 2809,11 |
| State 22 | 52,9 | 117,17 | 0,25 | 3,89 | -28565,9 | -153731,2 | 2157,80 |
| State 23 | 34,8 | 87,87 | 0,29 | 2,57 | -42033,2 | -195764,3 | 1607,38 |
| State 24 | 11,4 | 58,56 | 0,34 | 1,53 | -66273,6 | -262037,9 | 1106,55 |
| State 25 | -23,7 | 29,25 | 0,40 | 0,69 | -122142,4 | -384180,4 | 608,92 |

| | | |
|---|---|---|
| Gmax | 10859,7 | $lb/s{\cdot}ft^2$ |
| Area | 0,018593471 | $in^2$ |

Temperature 148.9 ºF (64.94 ºC), mass flow rate 4826.72 lb/h and constant entropy of 32.64 kJ/(kg·C).

| | T ºF | P psia | Vapour fraction | ρ lb/ft3 | Integrand $ft^2/s^2$ | Summation $ft^2/s^2$ | G $lb/s{\cdot}ft^2$ |
|---|---|---|---|---|---|---|---|
| State 1 | 148,9 | 732,59 | 0,00 | 27,69 | 0 | 0 | 0 |
| State 2 | 148,3 | 703,28 | 0,00 | 27,65 | -4906,5 | -4906,5 | 2739,15 |
| State 3 | 147,6 | 673,97 | 0,00 | 27,61 | -4913,9 | -9820,4 | 3869,31 |
| State 4 | 147,0 | 644,67 | 0,00 | 27,57 | -4921,5 | -14741,9 | 4733,49 |
| State 5 | 146,3 | 615,36 | 0,00 | 27,52 | -4929,0 | -19670,8 | 5459,51 |
| State 6 | 145,6 | 586,06 | 0,00 | 27,48 | -4936,5 | -24607,4 | 6096,94 |
| State 7 | 145,0 | 556,75 | 0,00 | 27,44 | -4944,0 | -29551,4 | 6671,26 |
| State 8 | 144,3 | 527,45 | 0,00 | 27,40 | -4951,5 | -34502,9 | 7197,64 |
| State 9 | 143,6 | 498,14 | 0,00 | 27,36 | -4959,0 | -39461,9 | 7685,99 |
| State 10 | 142,9 | 468,84 | 0,00 | 27,32 | -4966,4 | -44428,3 | 8143,22 |
| State 11 | 142,1 | 439,53 | 0,00 | 27,28 | -4973,7 | -49402,1 | 8574,37 |
| State 12 | 141,4 | 410,23 | 0,00 | 27,24 | -4981,0 | -54383,0 | 8983,30 |
| **State 13** | **140,6** | **380,92** | **0,00** | **27,20** | **-4988,1** | **-59371,1** | **9373,00** |
| State 14 | 136,0 | 351,61 | 0,03 | 22,63 | -5449,0 | -64820,2 | 8149,33 |
| State 15 | 128,5 | 322,31 | 0,08 | 17,18 | -6820,1 | -71640,2 | 6503,96 |
| State 16 | 120,5 | 293,00 | 0,13 | 13,33 | -8899,8 | -80540,0 | 5349,70 |
| State 17 | 111,9 | 263,70 | 0,17 | 10,45 | -11417,3 | -91957,3 | 4483,41 |
| State 18 | 102,5 | 234,39 | 0,21 | 8,23 | -14535,2 | -106492,5 | 3797,09 |
| State 19 | 92,2 | 205,09 | 0,25 | 6,45 | -18496,4 | -124988,9 | 3226,55 |
| State 20 | 80,8 | 175,78 | 0,28 | 5,01 | -23691,1 | -148680,0 | 2731,23 |
| State 21 | 67,9 | 146,48 | 0,32 | 3,81 | -30787,1 | -179467,1 | 2283,56 |
| State 22 | 52,9 | 117,17 | 0,35 | 2,81 | -41036,2 | -220503,3 | 1863,22 |
| State 23 | 34,8 | 87,87 | 0,39 | 1,95 | -57087,2 | -277590,5 | 1453,71 |
| State 24 | 11,4 | 58,56 | 0,43 | 1,22 | -85708,1 | -363298,6 | 1037,60 |
| State 25 | -23,7 | 29,25 | 0,47 | 0,58 | -151075,4 | -514374,0 | 588,45 |

| | | |
|---|---|---|
| Gmax | 9373,0 | $lb/s{\cdot}ft^2$ |
| Area | 0,025747982 | $in^2$ |

Temperature 180.3 ºF (82.37 ºC), mass flow rate 9551.57 lb/h and constant entropy of 40.80 kJ/(kg·C).

| | **T**<br>**ºF** | **P**<br>**psia** | **Vapour fraction** | **ρ**<br>**lb/ft3** | **Integrand**<br>**$ft^2/s^2$** | **Summation**<br>**$ft^2/s^2$** | **G**<br>**$lb/s{\cdot}ft^2$** |
|---|---|---|---|---|---|---|---|
| State 1 | 180,3 | 732,59 | 0,00 | 23,54 | 0 | 0 | 0 |
| State 2 | 179,1 | 703,28 | 0,00 | 23,51 | -5772,6 | -5772,6 | 2525,64 |
| State 3 | 178,0 | 673,97 | 0,00 | 23,48 | -5779,1 | -11551,7 | 3569,28 |
| State 4 | 176,8 | 644,67 | 0,00 | 23,47 | -5783,9 | -17335,5 | 4369,55 |
| State 5 | 175,6 | 615,36 | 0,00 | 23,46 | -5786,6 | -23122,1 | 5044,95 |
| State 6 | 174,3 | 586,06 | 0,00 | 23,46 | -5787,0 | -28909,1 | 5641,97 |
| State 7 | 173,0 | 556,75 | 0,00 | 23,48 | -5784,6 | -34693,7 | 6184,91 |
| **State 8** | **171,7** | **527,45** | **0,00** | **23,51** | **-5778,7** | **-40472,4** | **6689,21** |
| State 9 | 167,8 | 498,14 | 0,04 | 21,21 | -6071,5 | -46543,9 | 6472,31 |
| State 10 | 162,1 | 468,84 | 0,11 | 17,83 | -6954,6 | -53498,5 | 5832,98 |
| State 11 | 156,1 | 439,53 | 0,16 | 15,08 | -8249,9 | -61748,4 | 5300,45 |
| State 12 | 149,7 | 410,23 | 0,20 | 12,85 | -9721,4 | -71469,8 | 4858,30 |
| State 13 | 143,1 | 380,92 | 0,24 | 10,99 | -11391,5 | -82861,2 | 4472,95 |
| State 14 | 136,0 | 351,61 | 0,28 | 9,41 | -13314,7 | -96175,9 | 4125,70 |
| State 15 | 128,5 | 322,31 | 0,31 | 8,04 | -15560,5 | -111736,4 | 3802,69 |
| State 16 | 120,5 | 293,00 | 0,35 | 6,86 | -18224,7 | -129961,1 | 3495,31 |
| State 17 | 111,9 | 263,70 | 0,38 | 5,81 | -21440,7 | -151401,8 | 3196,64 |
| State 18 | 102,5 | 234,39 | 0,40 | 4,88 | -25403,7 | -176805,5 | 2901,98 |
| State 19 | 92,2 | 205,09 | 0,43 | 4,05 | -30407,9 | -207213,4 | 2607,25 |
| State 20 | 80,8 | 175,78 | 0,46 | 3,30 | -36925,6 | -244139,0 | 2308,64 |
| State 21 | 67,9 | 146,48 | 0,48 | 2,63 | -45764,0 | -289903,1 | 2002,44 |
| State 22 | 52,9 | 117,17 | 0,51 | 2,02 | -58426,7 | -348329,8 | 1684,26 |
| State 23 | 34,8 | 87,87 | 0,53 | 1,46 | -78084,2 | -426414,0 | 1348,04 |
| State 24 | 11,4 | 58,56 | 0,56 | 0,95 | -112814,3 | -539228,3 | 983,76 |
| State 25 | -23,7 | 29,25 | 0,59 | 0,47 | -191402,8 | -730631,1 | 569,87 |

| | | |
|---|---|---|
| Gmax | 6689,2 | $lb/s{\cdot}ft^2$ |
| Area | 0,071395377 | $in^2$ |

Temperature 199.1 ºF (92.83 ºC), mass flow rate 12829.00 lb/h and constant entropy of 47.92 kJ/(kg·C).

| | **T** **ºF** | **P** **psia** | **Vapour fraction** | **ρ** **lb/ft3** | **Integrand** **$ft^2/s^2$** | **Summation** **$ft^2/s^2$** | **G** **$lb/s{\cdot}ft^2$** |
|---|---|---|---|---|---|---|---|
| State 1 | 199,1 | 732,59 | 0,00 | 18,27 | 0 | 0 | 0 |
| State 2 | 196,9 | 703,28 | 0,00 | 18,03 | -7481,8 | -7481,8 | 2205,22 |
| State 3 | 194,6 | 673,97 | 0,00 | 17,89 | -7560,1 | -15042,0 | 3103,14 |
| State 4 | 192,1 | 644,67 | 0,00 | 17,87 | -7593,5 | -22635,4 | 3802,13 |
| State 5 | 188,6 | 615,36 | 0,08 | 17,11 | -7763,5 | -30399,0 | 4218,24 |
| State 6 | 183,7 | 586,06 | 0,20 | 15,40 | -8354,1 | -38753,0 | 4286,61 |
| **State 7** | **178,6** | **556,75** | **0,27** | **13,83** | **-9290,9** | **-48043,9** | **4286,96** |
| State 8 | 173,3 | 527,45 | 0,32 | 12,41 | -10349,8 | -58393,8 | 4240,03 |
| State 9 | 167,8 | 498,14 | 0,36 | 11,12 | -11543,1 | -69936,9 | 4157,95 |
| State 10 | 162,1 | 468,84 | 0,40 | 9,94 | -12896,0 | -82832,9 | 4045,43 |
| State 11 | 156,1 | 439,53 | 0,43 | 8,87 | -14438,4 | -97271,3 | 3911,47 |
| State 12 | 149,7 | 410,23 | 0,45 | 7,90 | -16190,5 | -113461,8 | 3765,14 |
| State 13 | 143,1 | 380,92 | 0,48 | 7,03 | -18185,4 | -131647,2 | 3606,36 |
| State 14 | 136,0 | 351,61 | 0,50 | 6,23 | -20484,9 | -152132,2 | 3435,21 |
| State 15 | 128,5 | 322,31 | 0,52 | 5,49 | -23169,4 | -175301,5 | 3252,15 |
| State 16 | 120,5 | 293,00 | 0,54 | 4,81 | -26348,6 | -201650,1 | 3056,90 |
| State 17 | 111,9 | 263,70 | 0,56 | 4,18 | -30177,2 | -231827,3 | 2849,57 |
| State 18 | 102,5 | 234,39 | 0,57 | 3,60 | -34879,1 | -266706,4 | 2629,64 |
| State 19 | 92,2 | 205,09 | 0,59 | 3,06 | -40795,2 | -307501,6 | 2396,46 |
| State 20 | 80,8 | 175,78 | 0,61 | 2,55 | -48468,9 | -355970,5 | 2148,79 |
| State 21 | 67,9 | 146,48 | 0,62 | 2,07 | -58827,1 | -414797,6 | 1884,81 |
| State 22 | 52,9 | 117,17 | 0,64 | 1,62 | -73594,2 | -488391,8 | 1601,52 |
| State 23 | 34,8 | 87,87 | 0,65 | 1,20 | -96397,3 | -584789,1 | 1294,00 |
| State 24 | 11,4 | 58,56 | 0,67 | 0,79 | -136450,2 | -721239,3 | 953,09 |
| State 25 | -23,7 | 29,25 | 0,68 | 0,41 | -226543,2 | -947782,5 | 557,73 |

| | | |
|---|---|---|
| Gmax | 4287,0 | $lb/s{\cdot}ft^2$ |
| Area | 0,149628102 | $in^2$ |

Temperature 203.03 ºF (95.18 ºC), mass flow rate 14696.84 lb/h and constant entropy of 50.71 kJ/(kg·C).

| | **T**<br>**ºF** | **P**<br>**psia** | **Vapour fraction** | **ρ**<br>**lb/ft3** | **Integrand**<br>**$ft^2/s^2$** | **Summation**<br>**$ft^2/s^2$** | **G**<br>**$lb/s{\cdot}ft^2$** |
|---|---|---|---|---|---|---|---|
| State 1 | 203,3 | 732,59 | 0,00 | 16,13 | 0 | 0 | 0 |
| State 2 | 200,4 | 703,28 | 0,00 | 15,85 | -8491,7 | -8491,7 | 2064,94 |
| State 3 | 197,2 | 673,97 | 0,00 | 15,54 | -8651,7 | -17143,5 | 2877,76 |
| State 4 | 193,3 | 644,67 | 0,17 | 14,96 | -8901,8 | -26045,3 | 3415,15 |
| State 5 | 188,6 | 615,36 | 0,32 | 13,71 | -9471,8 | -35517,1 | 3652,88 |
| State 6 | 183,7 | 586,06 | 0,38 | 12,50 | -10360,6 | -45877,6 | 3787,59 |
| State 7 | 178,6 | 556,75 | 0,43 | 11,38 | -11369,5 | -57247,1 | 3850,62 |
| **State 8** | **173,3** | **527,45** | **0,46** | **10,34** | **-12504,6** | **-69751,8** | **3860,45** |
| State 9 | 167,8 | 498,14 | 0,49 | 9,37 | -13780,6 | -83532,4 | 3829,53 |
| State 10 | 162,1 | 468,84 | 0,51 | 8,47 | -15222,8 | -98755,2 | 3763,78 |
| State 11 | 156,1 | 439,53 | 0,53 | 7,63 | -16862,3 | -115617,5 | 3671,37 |
| State 12 | 149,7 | 410,23 | 0,55 | 6,87 | -18724,4 | -134341,9 | 3559,75 |
| State 13 | 143,1 | 380,92 | 0,57 | 6,16 | -20847,4 | -155189,3 | 3430,72 |
| State 14 | 136,0 | 351,61 | 0,59 | 5,50 | -23294,7 | -178483,9 | 3285,52 |
| State 15 | 128,5 | 322,31 | 0,60 | 4,88 | -26151,1 | -204635,0 | 3124,97 |
| State 16 | 120,5 | 293,00 | 0,61 | 4,31 | -29532,4 | -234167,4 | 2949,65 |
| State 17 | 111,9 | 263,70 | 0,63 | 3,77 | -33601,0 | -267768,5 | 2759,91 |
| State 18 | 102,5 | 234,39 | 0,64 | 3,26 | -38592,7 | -306361,1 | 2555,63 |
| State 19 | 92,2 | 205,09 | 0,65 | 2,79 | -44866,0 | -351227,1 | 2336,32 |
| State 20 | 80,8 | 175,78 | 0,67 | 2,34 | -52992,5 | -404219,5 | 2101,02 |
| State 21 | 67,9 | 146,48 | 0,68 | 1,91 | -63946,1 | -468165,7 | 1847,99 |
| State 22 | 52,9 | 117,17 | 0,69 | 1,50 | -79537,9 | -547703,6 | 1574,41 |
| State 23 | 34,8 | 87,87 | 0,70 | 1,12 | -103573,1 | -651276,6 | 1275,41 |
| State 24 | 11,4 | 58,56 | 0,71 | 0,75 | -145710,8 | -796987,4 | 941,96 |
| State 25 | -23,7 | 29,25 | 0,72 | 0,38 | -240306,1 | -1037293,5 | 552,97 |

| | | |
|---|---|---|
| Gmax | 3860,4 | $lb/s{\cdot}ft^2$ |
| Area | 0,190351373 | $in^2$ |

Temperature 206.2 ºF (96.80 ºC), mass flow rate 16780.95 lb/h and constant entropy of 53.84 kJ/(kg·C).

| | **T** **ºF** | **P** **psia** | **Vapour fraction** | **ρ** **lb/ft3** | **Integrand** **$ft^2/s^2$** | **Summation** **$ft^2/s^2$** | **G** **$lb/s·ft^2$** |
|---|---|---|---|---|---|---|---|
| State 1 | 206,2 | 732,59 | 1,00 | 13,72 | 0 | 0 | 0 |
| State 2 | 202,2 | 703,28 | 1,00 | 13,36 | -10028,5 | -10028,5 | 1892,24 |
| State 3 | 197,9 | 673,97 | 1,00 | 12,98 | -10308,4 | -20337,0 | 2618,01 |
| State 4 | 193,4 | 644,67 | 0,56 | 12,12 | -10816,1 | -31153,1 | 3026,48 |
| State 5 | 188,6 | 615,36 | 0,58 | 11,21 | -11637,5 | -42790,6 | 3279,14 |
| State 6 | 183,7 | 586,06 | 0,59 | 10,33 | -12607,1 | -55397,7 | 3438,50 |
| State 7 | 178,6 | 556,75 | 0,61 | 9,50 | -13696,3 | -69094,0 | 3530,07 |
| **State 8** | **173,3** | **527,45** | **0,62** | **8,71** | **-14916,3** | **-84010,2** | **3569,69** |
| State 9 | 167,8 | 498,14 | 0,63 | 7,97 | -16284,1 | -100294,3 | 3568,21 |
| State 10 | 162,1 | 468,84 | 0,64 | 7,27 | -17826,1 | -118120,4 | 3531,65 |
| State 11 | 156,1 | 439,53 | 0,65 | 6,61 | -19574,8 | -137695,2 | 3466,79 |
| State 12 | 149,7 | 410,23 | 0,66 | 5,99 | -21560,4 | -159255,6 | 3379,74 |
| State 13 | 143,1 | 380,92 | 0,67 | 5,41 | -23826,1 | -183081,8 | 3272,76 |
| State 14 | 136,0 | 351,61 | 0,68 | 4,86 | -26439,1 | -209520,8 | 3147,47 |
| State 15 | 128,5 | 322,31 | 0,69 | 4,35 | -29488,3 | -239009,1 | 3005,09 |
| State 16 | 120,5 | 293,00 | 0,70 | 3,86 | -33095,7 | -272104,8 | 2846,41 |
| State 17 | 111,9 | 263,70 | 0,71 | 3,40 | -37432,8 | -309537,6 | 2671,87 |
| State 18 | 102,5 | 234,39 | 0,72 | 2,96 | -42749,0 | -352286,6 | 2481,51 |
| State 19 | 92,2 | 205,09 | 0,72 | 2,54 | -49422,2 | -401708,8 | 2275,00 |
| State 20 | 80,8 | 175,78 | 0,73 | 2,14 | -58055,6 | -459764,3 | 2051,38 |
| State 21 | 67,9 | 146,48 | 0,74 | 1,76 | -69676,2 | -529440,5 | 1809,04 |
| State 22 | 52,9 | 117,17 | 0,75 | 1,39 | -86190,8 | -615631,3 | 1545,17 |
| State 23 | 34,8 | 87,87 | 0,76 | 1,04 | -111604,6 | -727235,9 | 1254,98 |
| State 24 | 11,4 | 58,56 | 0,76 | 0,70 | -156074,3 | -883310,2 | 929,41 |
| State 25 | -23,7 | 29,25 | 0,76 | 0,36 | -255707,0 | -1139017,2 | 547,41 |

| | | |
|---|---|---|
| Gmax | 3569,7 | $lb/s·ft^2$ |
| Area | 0,235048002 | $in^2$ |

Temperature 209.8 ºF (98.77 ºC), mass flow rate 17813.99 lb/h and constant entropy of 57.54 kJ/(kg·C).

| | **T** **ºF** | **P** **psia** | **Vapour fraction** | **ρ** **lb/ft3** | **Integrand** **$ft^2/s^2$** | **Summation** **$ft^2/s^2$** | **G** **$lb/s·ft^2$** |
|---|---|---|---|---|---|---|---|
| State 1 | 209,8 | 732,59 | 1,00 | 11,13 | 0 | 0 | 0 |
| State 2 | 204,8 | 703,28 | 1,00 | 10,73 | -12422,4 | -12422,4 | 1691,72 |
| State 3 | 199,3 | 673,97 | 1,00 | 10,32 | -12899,2 | -25321,6 | 2322,13 |
| State 4 | 193,5 | 644,67 | 1,00 | 9,88 | -13441,6 | -38763,2 | 2751,87 |
| State 5 | 188,6 | 615,36 | 0,89 | 9,22 | -14211,5 | -52974,7 | 3002,49 |
| State 6 | 183,7 | 586,06 | 0,84 | 8,57 | -15260,2 | -68234,9 | 3166,01 |
| State 7 | 178,6 | 556,75 | 0,82 | 7,94 | -16444,4 | -84679,2 | 3268,73 |
| State 8 | 173,3 | 527,45 | 0,81 | 7,34 | -17764,8 | -102444,0 | 3323,71 |
| **State 9** | **167,8** | **498,14** | **0,80** | **6,77** | **-19241,2** | **-121685,3** | **3339,79** |
| State 10 | 162,1 | 468,84 | 0,79 | 6,22 | -20901,1 | -142586,4 | 3322,68 |
| State 11 | 156,1 | 439,53 | 0,79 | 5,70 | -22778,4 | -165364,8 | 3277,56 |
| State 12 | 149,7 | 410,23 | 0,79 | 5,20 | -24909,8 | -190274,6 | 3209,07 |
| State 13 | 143,1 | 380,92 | 0,79 | 4,73 | -27344,4 | -217619,0 | 3119,58 |
| State 14 | 136,0 | 351,61 | 0,79 | 4,28 | -30152,9 | -247771,8 | 3010,85 |
| State 15 | 128,5 | 322,31 | 0,80 | 3,85 | -33429,7 | -281201,5 | 2884,15 |
| State 16 | 120,5 | 293,00 | 0,80 | 3,43 | -37304,2 | -318505,7 | 2740,31 |
| State 17 | 111,9 | 263,70 | 0,80 | 3,04 | -41958,6 | -360464,2 | 2579,82 |
| State 18 | 102,5 | 234,39 | 0,80 | 2,66 | -47657,8 | -408122,1 | 2402,73 |
| State 19 | 92,2 | 205,09 | 0,81 | 2,30 | -54803,5 | -462925,5 | 2208,72 |
| State 20 | 80,8 | 175,78 | 0,81 | 1,95 | -64035,6 | -526961,1 | 1996,86 |
| State 21 | 67,9 | 146,48 | 0,81 | 1,61 | -76443,4 | -603404,5 | 1765,54 |
| State 22 | 52,9 | 117,17 | 0,82 | 1,28 | -94047,9 | -697452,4 | 1511,97 |
| State 23 | 34,8 | 87,87 | 0,82 | 0,96 | -121089,4 | -818541,9 | 1231,33 |
| State 24 | 11,4 | 58,56 | 0,82 | 0,65 | -168312,5 | -986854,4 | 914,57 |
| State 25 | -23,7 | 29,25 | 0,81 | 0,34 | -273890,0 | -1260744,4 | 540,62 |

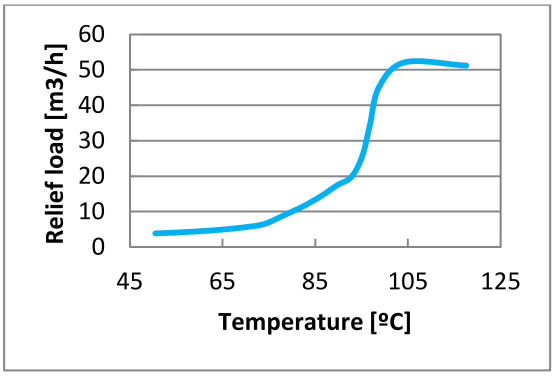
| | | |
|---|---|---|
| Gmax | 3339,8 | $lb/s·ft^2$ |
| Area | 0,2666933 | $in^2$ |

Temperature 243.7 ºF (117.60 ºC), mass flow rate 12641.94 lb/h and constant entropy of 67.60 kJ/(kg·C).

| | T ºF | P psia | Vapour fraction | ρ lb/ft3 | Integrand $ft^2/s^2$ | Summation $ft^2/s^2$ | G $lb/s·ft^2$ |
|---|---|---|---|---|---|---|---|
| State 1 | 243,7 | 732,59 | 1,00 | 6,95 | 0 | 0 | 0 |
| State 2 | 238,6 | 703,28 | 1,00 | 6,66 | -19965,0 | -19965,0 | 1329,98 |
| State 3 | 233,2 | 673,97 | 1,00 | 6,36 | -20857,2 | -40822,1 | 1818,31 |
| State 4 | 227,7 | 644,67 | 1,00 | 6,07 | -21840,4 | -62662,5 | 2148,73 |
| State 5 | 221,9 | 615,36 | 1,00 | 5,77 | -22927,1 | -85589,5 | 2389,06 |
| State 6 | 215,9 | 586,06 | 1,00 | 5,48 | -24132,0 | -109721,6 | 2566,27 |
| State 7 | 209,6 | 556,75 | 1,00 | 5,18 | -25472,9 | -135194,4 | 2694,59 |
| State 8 | 203,1 | 527,45 | 1,00 | 4,89 | -26970,6 | -162165,1 | 2782,72 |
| State 9 | 196,3 | 498,14 | 1,00 | 4,59 | -28651,0 | -190816,1 | 2836,48 |
| **State 10** | **189,2** | **468,84** | **1,00** | **4,30** | **-30545,2** | **-221361,3** | **2860,09** |
| State 11 | 181,7 | 439,53 | 1,00 | 4,01 | -32692,6 | -254053,9 | 2856,70 |
| State 12 | 173,9 | 410,23 | 1,00 | 3,72 | -35142,3 | -289196,2 | 2828,73 |
| State 13 | 165,6 | 380,92 | 1,00 | 3,43 | -37957,7 | -327153,8 | 2778,13 |
| State 14 | 156,9 | 351,61 | 1,00 | 3,15 | -41221,3 | -368375,1 | 2706,41 |
| State 15 | 147,7 | 322,31 | 1,00 | 2,88 | -45043,3 | -413418,4 | 2614,73 |
| State 16 | 137,9 | 293,00 | 1,00 | 2,60 | -49574,0 | -462992,4 | 2503,95 |
| State 17 | 127,3 | 263,70 | 1,00 | 2,33 | -55024,5 | -518016,8 | 2374,60 |
| State 18 | 115,8 | 234,39 | 1,00 | 2,07 | -61700,8 | -579717,7 | 2226,87 |
| State 19 | 103,2 | 205,09 | 1,00 | 1,81 | -70065,4 | -649783,0 | 2060,56 |
| State 20 | 89,2 | 175,78 | 1,00 | 1,55 | -80851,9 | -730635,0 | 1874,95 |
| State 21 | 73,2 | 146,48 | 1,00 | 1,30 | -95303,0 | -825937,9 | 1668,60 |
| State 22 | 54,3 | 117,17 | 1,00 | 1,05 | -115708,0 | -941646,0 | 1438,99 |
| State 23 | 34,8 | 87,87 | 0,99 | 0,80 | -146994,2 | -1088640,1 | 1178,62 |
| State 24 | 11,4 | 58,56 | 0,98 | 0,55 | -201635,6 | -1290275,7 | 880,25 |
| State 25 | -23,7 | 29,25 | 0,95 | 0,29 | -323388,0 | -1613663,7 | 524,10 |

| | | |
|---|---|---|
| Gmax | 2860,1 | $lb/s·ft^2$ |
| Area | 0,221005839 | $in^2$ |

The results for the relief rates and orifice area are plotted in the next figures:



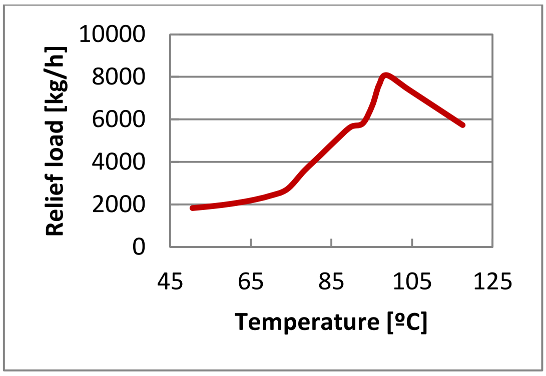




Figure 1. Volumetric relief rate of YS702/01. Figure 2. Mass relief rate of YS702/01.

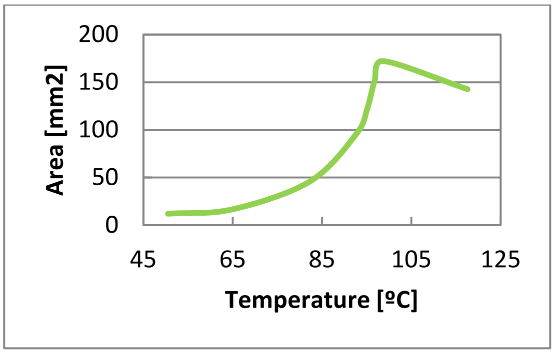



Figure 3. Orifice area of YS702/01.

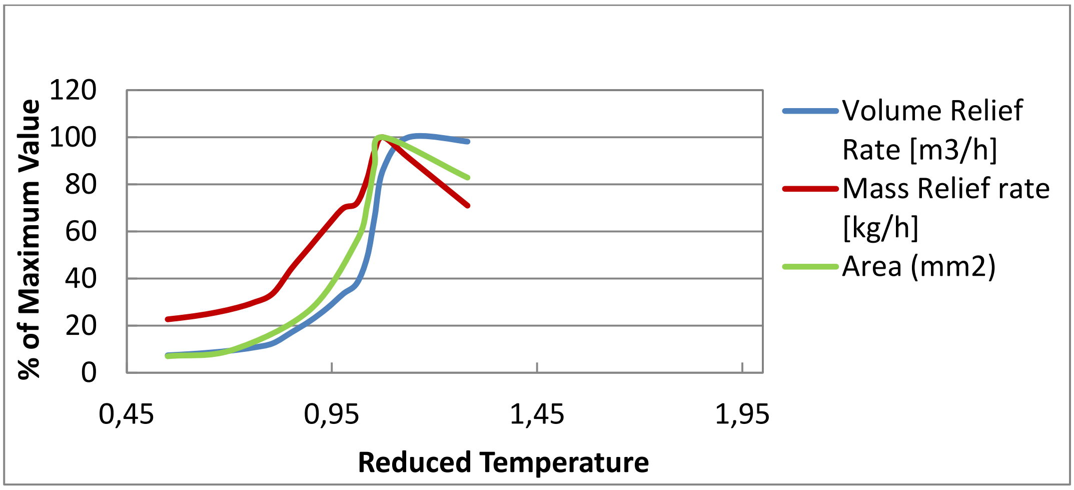



Figure 4. % Maximum value vs. reduced temperature of YS702/01.

Summary table:

| | | | |
|---|---|---|---|
| V ,max | 52,13 | $m^3/h$ | at 104,4ºC |
| M ,max | 8080,29 | kg/h | at 98,77ºC |
| A ,max | 172,06 | $mm^2$ | at 98,77ºC |