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The response of large pressure vessels in the process industries to excess pressure: literature review

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Large pressure vessels are used in a wide range of chemical process industries. They contain significant amounts of energy. In the event of a vessel failure, this energy could be released. This gives significant potential for harm to people and other plant. As part of effective risk control, pressure vessels are usually designed in accordance with established codes that define their minimum strength.

HSE is aware that there may be pressure vessel systems in Great Britain where there is potential for exposure to pressures greater than those allowed under relevant current design codes. This can result from: errors during the original vessel design phase; a change in the design code due to improved understanding of the reaction chemistry; or poor change management of plant or process. These situations may lead to insufficient capacity in pressure relief systems for vessels and the potential to generate excess pressure leading to vessel failure.

This report describes a literature review on the impact of overpressure on the likelihood of vessel failure. It includes an assessment of impact of different design standards. It captures sources of relevant good practice and supplementary information that may help in the assessment of risk. The report covers:

- **Likelihood of overpressures and failures;**
- **Comparison of different construction standards;**
- **Predictions of the effects of overpressures;**
- **Calculations of burst pressures.**

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The response of large pressure vessels in the process industries to excess pressure: literature review

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KEY MESSAGES

Pressure vessels are used in a wide range of process industries and can hold liquids and gases at high pressures. Therefore, they can contain significant amounts of energy that could be released in the event of a vessel failure, with significant potential for harm to people and other plant.

The Chemicals, Explosives and Microbiological Hazards Division (CEMHD) of the UK Health and Safety Executive (HSE) are aware that there may be systems where there is a potential for pressure vessels to be exposed to pressures greater than those allowable under the relevant design code. This report captures sources of relevant good practice and supplementary information that may help in the assessment of risk.

A literature review was undertaken, and design and assessment standards evaluated, to identify relevant published work that considers the likelihood of failure of pressure vessels due to overpressure events to determine, if possible, what level of overpressure may cause damage or failure of pressure vessels. This identified the following:

- There is considerable variability in pressure vessel failure rates calculated in literature. This may either be due to the quality of data collection, or an indication that failure rates may vary depending on factors such as the fluid service and the broader operating environment.
- For intact vessels, burst pressure can be estimated as the pressure at which the hoop stress reaches the ultimate tensile strength of the material. There are many formulae available for the calculation of burst pressure, but using the simple Barlow formula is likely to give a sufficiently accurate estimate.
- A number of other factors are likely to affect the burst pressure, including:
    - additional thickness due to corrosion allowance;
    - material properties exceeding the minimum specified for the grade; and
    - the duration of the overpressure.
- The ratio of the burst pressure to the maximum allowable pressure for the vessel will depend to the largest degree on the maximum allowable stress defined by the design code. The allowable stress could vary by a factor of two. Therefore, any system predicting the effects of overpressure, or setting limits for allowable overpressure, must take into account the code to which the vessel was designed.
- The lowest ratio for burst pressure to MAWP should be 2.35, for an intact vessel, or 2.0 for a cracked vessel, assuming the cracks have been assessed and the vessel re-rated if necessary.
- The estimating level of overpressure at which damage may occur is more difficult, but it should not be below the hydrotest pressure (at least 30 % higher than MAWP).
- These conclusions have been based on set safety factors for design stresses and calculations predicting actual burst pressure for intact and corroded cylindrical vessels. The assumption has been made that the findings will be applicable to other simple geometries where the internal pressure is the main source of stress, such as for heads and spherical vessels.
- The complex nature of nozzle and flange designs means that this approach has not been possible for these components. However, from the lack of information on these components it should not be assumed that they would not fail during an overpressure event.

EXECUTIVE SUMMARY

Background

Pressure vessels are used in a wide range of process industries. Many are large and can hold liquids and gases at high pressures, and therefore contain significant amounts of energy that could be released in the event of a vessel failure, with significant potential for harm to people and other plant. To control the risk, vessels are designed and operated to strict codes that define both the minimum strength of the vessels and the relief systems, which aim to prevent overpressure situations occurring.

The Chemicals, Explosives and Microbiological Hazards Division (CEMHD) of the UK Health and Safety Executive (HSE) are aware that there may be systems where there is a potential for pressure vessels to be exposed to pressures greater than those allowable under the relevant design code. These situations can come about as a result of specific overpressure scenarios having been either missed or discounted during the original design, an improved understanding of the reaction chemistry, or poor change management of plant or process, all leading to insufficient capacity in relief systems and the potential to generate excess pressure. Therefore, a literature review was conducted; to identify any relevant previously published work on the impact of overpressure on the likelihood of vessel failure. This report captures sources of relevant good practice and supplementary information that may help in the assessment of risk.

Aims

- Identify relevant published work that deals with the impact of various levels of overpressure on the likelihood of failure of pressure vessels.
- Determine, if possible, what level of overpressure may cause damage or failure of pressure vessels.

Methods (see Methods section)

- A literature review was conducted, with the initial search terms being overpressure, burst, fracture, flange leakage, leakage, pressure vessel, probability of failure, likelihood of failure, frequency of failure, relief discharge capacity and failure mode.
- Further literature was identified from sources known to the report authors, and citations in papers obtained in the literature search.
- The relevant literature that was identified, together with design standards for pressure vessels and fitness-for-service procedures were assessed for information in the following areas:
    - Likelihood of overpressures and failures,
    - Comparison of different construction standards,
    - Predictions of the effects of overpressures, and
    - Calculations of burst pressures.

Main findings

1. There is considerable variability in pressure vessel failure rates calculated in literature. This may either be due to the quality of data collection, or an indication that failure rates may vary depending on factors such as the fluid service and the broader operating environment.
2. For intact vessels, burst pressure can be estimated as the pressure at which the hoop stress reaches the ultimate tensile strength of the material. There are many formulae available for the calculation of burst pressure, but using the simple Barlow formula is likely to give a sufficiently accurate estimate.
3. A number of other factors are likely to affect the burst pressure, including:
    a. additional thickness due to corrosion allowance;
    b. material properties exceeding the minimum specified for the grade; and
    c. the duration of the overpressure.
4. The ratio of the burst pressure to the maximum allowable pressure for the vessel will depend to the largest degree on the maximum allowable stress defined by the design code. The allowable stress could vary by a factor of two. Therefore, any system predicting the effects of overpressure, or setting limits for allowable overpressure, must take into account the code to which the vessel was designed.
5. Both the American Society of Mechanical Engineers' ASME Boiler and Pressure Vessel Code, Section VIII and the Pressure Equipment Directive allow accumulation during pressure relief above the maximum allowable working pressure, but differ in the permissions they apply.
6. The likely effect of various levels of overpressure on pressure vessels has been presented in tables in a number of publications (Centre for Chemical Process Safety (CCPS), Williams and Prophet), but they are merely examples and are mainly for ASME VIII carbon steel vessels. The publications also caution against using the tables where there is existing damage to the vessel.
7. The American Petroleum Institute API 581 document assumes that vessels will certainly fail at an overpressure factor of 400 % design pressure and lists potential consequences at various degrees of overpressure up to this point. However, it is not clear on what experimental evidence these assumptions are based.
8. While API 581 assumes that vessels will certainly fail at an overpressure factor of 400 % design pressure, there is some incident evidence that vessels sometimes survive to higher degrees of overpressure before failure.
9. There is also corresponding evidence to suggest that vessels sometime fail at lower degrees of overpressure, possibly because these vessels are built to different design codes.
10. In the special case of fire-caused boiling liquid expanding vapour explosion (BLEVE) failure, weakening of the wall due to overheating can be sufficient to cause vessel failure within the design pressure.
11. Davenport and Warwick found that while leakage failures can occur on very thick walled vessels, there is some evidence to suggest that catastrophic failure becomes an increasingly likely failure mode as the vessel wall thickness increases.
12. The lowest ratio for burst pressure to maximum allowable working pressure should be 2.35, which would be for a vessel constructed to British design codes BS 5500, or PD 5500.

13. A corroded vessel found to be acceptable to the API 579 Fitness-for-Service procedure, may have a remaining strength factor as low as 0.9, reducing the ratio of minimum burst pressure to maximum allowable working pressure (MAWP) to 2.1.
14. Cracks may have an effect on the burst strength if they have propagated since hydrotesting, or at pressures above the hydrotest pressure. Brittle failure of vessels would be more likely at cold temperatures, or under shock loading. Assessing a cracked vessel using a fitness for service assessment should result in a ratio of burst pressure to MAWP of at least 2.
15. The estimating level of overpressure at which damage may occur is more difficult, but it should not be below the hydrotest pressure (at least 30 % higher than MAWP).
16. The ratios of burst pressure to MAWP given assume that the condition of the vessel is known, and has been rerated if necessary. Undetected corrosion or cracks could significantly decrease the burst pressure.
17. These conclusions have been based on set safety factors for design stresses and calculations predicting actual burst pressure for intact and corroded cylindrical vessels. The assumption has been made that the findings will be applicable to other simple geometries where the internal pressure is the main source of stress, such as for heads and spherical vessels.
18. The complex nature of nozzle and flange designs means that this approach has not been possible for these components. For example, the bolting force for a flanged joint is dependent on a number of factors, such as flange diameter, gasket type and location and internal pressure. Therefore it would not be possible to predict an overpressure ratio likely to cause failure. However, from the lack of information on these components it should not be assumed that they would not fail during an overpressure event.

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1 INTRODUCTION

Pressure vessels are used in a wide range of process industries. Many are large and can hold liquids and gases at high pressures, and therefore hold large energies that could be released in the event of a vessel failure, with significant potential for harm to people and other plant. To control the risk, vessels are designed and operated to strict codes, which define the minimum strength of the vessels and the relief systems which aim to prevent overpressure situations occurring.

The Chemicals, Explosives and Microbiological Hazards Division (CEMHD) of the UK Health and Safety Executive (HSE) are aware that there may be systems where there is a potential for pressure vessels to be exposed to pressures greater than those allowable under the relevant design code. These situations can come about as a result of specific overpressure scenarios having been either missed or discounted during the original design, an improved understanding of the reaction chemistry, or poor change management of plant or process, all leading to insufficient capacity in relief systems and the potential to generate excess pressure. Therefore, a literature review was conducted; to identify any relevant previously published work on the impact of overpressure and likelihood of vessel failure.

This report details the findings of the literature review into the effect of overpressure on pressure vessels. This includes past pressure vessel failures, equations for burst pressures for intact and corroded vessels, and examination of tables predicting the consequences of different levels of overpressure. The sources of relevant good practice and supplementary information captured in this report may help in the assessment of risk.

Due to the simple geometry of pressure vessels and the tight correlation between internal pressure and stress it has been possible to compare burst pressures predicted by equations and experimental work to tables predicting the effects of overpressure. For more complex components, such as nozzles and flanges, this approach has not been possible. For these, the stresses may not be directly linked to internal pressure (for example, for bolted flanges, where many factors like gasket type and flange design affect stresses).

2 METHODS

This investigation into the effects of overpressure on pressure vessels has taken the form of a literature review. A number of potentially useful documents were initially identified from the authors' experience and through discussions. These included documents such as relevant standards and publications detailing past failures and estimated frequencies of failures.

A literature search was also conducted, through HSE's Information Centre. The following search terms were requested:

Overpressure, burst, fracture, flange leakage, leakage, pressure vessel, probability of failure, likelihood of failure, frequency of failure, relief discharge capacity, failure mode

The date range for the search was not limited.

A total of 66 references were identified by the Information Centre. The list of references was examined for those that may be of interest when looking at the response of vessels to excessive pressure and the process safety or probability of failure. A total of 40 papers were selected from the references, however, the majority of papers were found to be of little relevance once obtained.

Further references were obtained through general internet searches and from following cited papers in the most relevant sources.

The assessment of the literature and writing of the report was split into the following topic areas:

- Likelihood of overpressures and failures,
- Comparison of different construction standards,
- Predictions of the effects of overpressures, and
- Calculations of burst pressures.

3 LIKELIHOOD OF OVERPRESSURES AND FAILURES

3.1 GENERAL FAILURE FREQUENCIES OF PRESSURE VESSELS

There are a large number of sources of information giving failure rates for pressure vessels. These have been reviewed by various regulatory authorities with a view to producing guidance on failure rates for use in quantitative risk assessments. These regulatory authorities include:

- The Netherlands 'National Institute for Public Health and the Environment' (RIVM);
- The Flemish 'Safety Reporting Division'; and
- The UK's Health & Safety Executive (HSE).

3.1.1 RIVM and Purple book

RIVM's '*Reference Manual BEVI Risk Assessments*' (2009) [1] contains various generic failure frequencies for use in standard risk assessments of pressurised vessels, but has no explanation of the origin of these numbers. These failure frequencies are the same as those quoted in the Purple Book [2], which references the 1981 COVO study [3] as its data source.

RIVM defines the extent of a pressurised vessel and gives failure frequencies for three scenarios:

1) Instantaneous release of entire contents;
2) Release of entire contents in 10 minutes in a continuous and constant stream; and
3) Continuous release of contents from a hole with an effective diameter of 10 mm.

These failure frequencies are given for above ground storage tanks and underground or mounded tanks, although the actual failure frequencies are the same for both types of tank. RIVM gives different, higher failure frequencies for the category of vessel described as 'reactor vessels and process vessels'. There are also separate tables of failure frequencies for distillation columns.

Details of the failure frequencies used and the parts included in the failure frequencies are given in Appendix A.

3.1.2 Flemish government '*Background Information Appendix to Handbook failure frequencies*' (2009)

The *Background Information Appendix to Handbook failure frequencies* [4] published by the Flemish government discusses the data from which failure frequencies are derived in some detail. The data comes from a study by Smith and Warwick [8] based on 20 000 pressure vessels with 310 000 vessel years, described as 'mainly steam pressure vessels'. The original data included 229 events, fitted into three categories:

1) 'Potential failures';
2) Failures with leakage; and
3) Catastrophic failure.

This data was screened by Technica to exclude pipework associated with vessels and the 'potential failures'. (Pipework up to the first flange, manholes and instrumentation connections are considered

as part of the vessel). Following the screening, the number of events associated with the vessel itself was reduced, to produce the data given in Table 1 below.

Table 1 Cause of failure for pressure vessels – copy of Table 5 in *Background Information Appendix to Handbook Failure Frequencies*

	Leak size (mm)					
Cause of failure	***< 25***	***25 - 50***	***50 - 150***	***Instantaneous***	***Total***	***Share* (%)***
Design error	2	1	0	0	3	6.5
Material and/or construction defect, probably including brittleness	5	3	1	0	9	19.5
Wear	1	0	0	0	1	2
Metal fatigue	6	3	0	0	9	19.5
Corrosion	1	3	0	0	4	9
Thermal fatigue	1	0	0	0	1	2
Creep	2	0	1	0	3	4
Overload	1	2	0	2	5	11
Unknown	6	4	1	0	11	24
Number of failures according to leak size (%)	25	16	3	2	46	100
Share in terms of percentage (%)	54	35	6.5	4		100

*The number of events corresponds to 310 000 vessel years

Assuming that the 'overload' term would be better translated as 'overpressure', the table indicates failure frequencies due to overpressure of:

(i) 6.5×10^{-6}/vessel year for catastrophic ('instantaneous') failure of a process pressure vessel; and

(ii) 9.7×10^{-6}/vessel year leakage of a process pressure vessel due to overpressurisation.

The catastrophic failure rate given above, which has been adopted as the recommended value by the Handbook, is broadly consistent with the value of 5×10^{-6}/vessel year recommended by RIVM for catastrophic failure of process vessels and reactors. The Handbook adopts a failure rate for all leaks, excluding catastrophic failures, of 1.4×10^{-4}/vessel year, based on Table 1 above, which is also broadly comparable with the value of 1.05×10^{-4}/vessel year recommended by RIVM for the two different leak scenarios.

3.1.3 HSE 'Failure Rate and Event Data for Use Within Risk Assessments' (FRED) (2012)

The HSE publication, '*Failure Rate and Event Data for Use Within Risk Assessments*' [5] (FRED) contains a bibliography quoting varying catastrophic failure rates from a large range of sources. The bibliography is reproduced in Appendix B. These sources are dated from 1972 to 1996. The majority indicate a catastrophic failure frequency for pressurised vessels in the range 1×10^{-6}/ vessel year and

1×10^{-5}/ vessel year, thus supporting the HSE guidance for using catastrophic failure rates in the range 2×10^{-6}/ vessel year to 6×10^{-6}/ vessel year for general pressure vessels.

FRED also gives failure rates for leaks from a range of hole diameter sizes. These are shown Table 2.

Table 2 Item failure rates

Type of release	*Failure rate (per vessel year)*	*Notes*
Catastrophic	6×10^{-6}	Upper failures
Catastrophic	4×10^{-6}	Median
Catastrophic	2×10^{-6}	Lower
50 mm diameter hole	5×10^{-6}	
25 mm diameter hole	5×10^{-6}	
13 mm diameter hole	1×10^{-5}	
6 mm diameter hole	4×10^{-5}	

The failure rates show that small leakages are considered substantially more likely than larger leakages.

3.1.4 DNV 'Failure frequency Guidance'

The DNV '*Failure frequency Guidance*' [6] makes use of the HSE's Hydrocarbon Release Database (HCRD) (dated 2012 in the references) to derive leak failure frequencies for a number of different vessel types, using a smoothing function to eliminate 'noise' in the data. The source compares these failure frequencies with those recommended by RIVM [1] and the Flemish '*Background Information Handbook on Failure Frequencies*' [4].

The report concludes that the failure frequencies used by RIVM and the Flemish '*Background Information Handbook on Failure Frequencies*' underestimates failure frequencies to a substantial degree. The estimated failure frequency for leakages greater than 150 mm in pressurised vessels given by DNV is 5.9×10^{-5} per vessel year, i.e. an order of magnitude higher than the values adopted by RIVM and the Flemish '*Background Information Handbook on Failure Frequencies*'.

The DNV source suggests a number of factors that could cause this effect:

1) The HCRD data is for offshore installations. It is possible that the nature of the offshore environment creates more onerous operating conditions, leading to higher failure rates.
2) THE HCRD data is a large and recent data set, whereas the onshore data sources for leak frequencies are generally based on smaller datasets, where the origin and reliability of the data are sometimes uncertain.
3) The onshore data tends to apply to leaks where the process fluid was at full operating pressure, whereas the HCRD data includes leaks that occurred from depressurised equipment and also leaks that were quickly isolated.
4) The offshore safety case requirements are more onerous than those for onshore facilities. This may result in more accurate reporting of releases.

Details of failure frequencies for leaks of different sizes are given in Appendix C.

3.1.5 Davenport and Warwick (1997)

The report by Davenport [7] describes a vessel survey intended to update and improve on the findings of Smith and Warwick [8]. The survey covered approximately 360 000 pressure vessels over five years of operation, i.e. 1.8 million vessel years.

One area of improvement in the survey was to separate data for boilers and steam receivers from that of other pressure vessels. The majority of the other pressure vessels were air receivers.

Significant conclusions drawn by Davenport and Warwick were that:

1) The failure rates for boilers and steam receivers showed good agreement with the previous findings of Smith and Warwick [8]; and
2) The failure rates for air receivers were approximately an order of magnitude lower than for the steam service vessels.

Davenport and Warwick suggested a number of possible explanations for conclusion 2, including:

I. Less corrosive service medium in air receivers;
II. Relative ease of construction of thinner walled vessels;
III. Over-design of thinner vessels due to minimum thickness of available plate; and
IV. Lower effective design stresses in specific air receiver design codes as compared with those for Class 1 codes.

The survey found four vessels that experienced 'disruptive' failure defined as 'catastrophic resulting in forcible release of contents'. Of these four failures, two were attributed to 'corrosion', one to 'maloperation' and one to 'not ascertained'. The survey report states that 'The vast majority of failures occurred in vessels under 18 mm thick, and although slightly more of these failures were in vessels thinner than 9 mm, they included only one of the four reported disruptive failures. This observation implies that the proportion of disruptive failures actually increases with vessel thickness.'

The survey report also states that 'It is important to note that there were two leakage failures of vessels over 27 mm thick, one of which was 2.5 inches (63 mm) thick. This is significant because the previous surveys had not identified any leakage failures on vessels of this thickness and had not given support to leak-before-break arguments for thicker walled vessels.'

3.2 HYDROCARBON RELEASE DATABASE (HCRD)

The HCRD [9] for the period 1992 to 2015 lists a total of 4656 incidents. The number of incidents related to vessels primarily described as 'Pressure Vessels' was 101. Of these 101 incidents, 50 had consequences described as 'Significant' and two had consequences described as 'Major'.

The two major incidents listed were caused by:

(A) Deficient procedure during cleaning at a pressure well below the maximum allowable operating pressure.

(B) A design fault becoming apparent at low pressure (less than 1 bar) during a restart after a process trip.

There are seven records primarily described as 'Pressure Vessels', where the pressure exceeded the design pressure. For four of these incidents, the design pressure is given as 0.03 bar, and described as 'failure related to design'. Of these seven incidents, six are classed as 'minor' and one is classed as 'significant'. The significant release involved the failure of the door gasket on a fuel gas vessel, resulting in a substantial release of gas.

The implication is that vessel failures as a result of overpressure are rare in the offshore sector.

3.3 INCIDENTS OF OVERPRESSURE CAUSING CATASTROPHIC VESSEL FAILURE

1) Union Carbide Seadrift, Texas – 12 March 1991 [10]

An ethylene oxide distillation column, with a maximum allowable working pressure (MAWP) of 6 barg, experienced autodecomposition of ethylene oxide, initiating a flame front. Pressure in the column reached 4 times MAWP, causing a ductile fracture and breakage of the column.

2) Shell, Stanlow – 20 March 1990 [11]

A halogen exchange reactor on the Fluoroaromatics plant was ruptured by the pressure generated by a runaway reaction. The pressure in the vessel reached a value of about 60 barg to 80 barg compared with the relief valve set pressure of 5 barg. Unfortunately, the maximum allowable working pressure of the vessel is not known.

3) Procter and Gamble, Worms, Germany – 21 November 1988 [10]

A CO_2 liquid storage tank failed, due to various contributing factors. Analysis after the event produced a failure pressure estimate of 35 bar to 51 bar. This range was 1.75 to 2.5 times the original design pressure.

4) Liquified Petroleum Gas (LPG) tank experiment [10]

Droste and Schoen (1988) describe an experiment in which an LPG tank failed at 39 bar or 2.5 times the opening pressure of its safety valve. [10]

5) BP Oil Refinery, Grangemouth, UK – 22 March 1987 [12]

An explosion of a hydrocracker vessel took place at the BP Oil refinery at Grangemouth on 22 March 1987. The explosion occurred due to overpressurisation of a low pressure separator labelled V306, which disintegrated. The design pressure of V306 was 10.7 barg, which was also the set pressure of the single relief valve. The HSE investigation concluded from the distribution of the fragments from V306 that the vessel failed at a pressure of approximately 50 bar, i.e. a factor of nearly five times the design pressure of the vessel.

A tabular summary of these incidents is given in Table 3. It should be noted that the maximum pressure that occurred during an event may not necessarily be the pressure required to cause vessel failure; it is possible vessel failure would have occurred at lower pressures. This would be most likely

where the pressure rise was rapid, or where the recorded pressure was not at the immediate vessel location.

Table 3 Overpressure ratio for various catastrophic pressure vessel failures

Date	Details	Failure pressure (bar)	Design pressure (bar)	Overpressure Factor
12 Mar 1991	Ethylene Oxide distillation column, Union Carbide, USA	24	6	4
20 Mar 1990	Halogen reactor, Fluoroaromatics plant, Ellesmere Port ,UK	60 – 80	5 (set pressure of PSV†)	12 – 16 (on set pressure of PSV† – Design pressure not known)
21 Nov 1988	Liquid CO_2 storage tank, Procter & Gamble, Germany	35 – 51	20	1.75 – 2.5
1988	Droste & Schoen report of LPG tank failure	39	15.6	2.5
22 Mar 1987	Hydrocracker explosion, BP Grangemouth, UK	50	10.7	5

†Pressure safety valve

3.4 NON-CATASTROPHIC FAILURE OF PRESSURE VESSELS

By their nature, non-catastrophic failures of pressure vessels are less likely to attract public attention or be subject to inspections by regulatory authorities. Finding specific examples or generic data on non-catastrophic failure due to overpressure is therefore more problematic.

One such example is a case study given by BG Group (BG). In August 2008, a temporary hydro-cyclone was exposed to an overpressure. The emergency shutdown valve (ESDV) failed to close and the pressure relief valves on the hydro-cyclone failed to open. The vessel was exposed to an overpressure of over two and a half times the unit design limit. An operator noticed the excessive reading on the pressure gauge and manually closed the ESDV. There was no release or rupture of equipment, but the vessel became distorted (barrelled) due to the overpressure and was assessed to have been very close to rupture.

3.5 BLEVES

Boiling Liquid Expanding Vapour Explosions (BLEVEs) are a special case of vessels failing due to overpressure. The failure occurs because of the twin effects of pressure increase in the vessel due to overheating of the liquid, and the weakening of the pressure vessel wall due to overheating. The result is a failure of the vessel at a lower pressure than would normally be expected.

Various experimental studies into BLEVE failures have been conducted. Birk, Poirier and & Davison [13] describe such an experiment using 500 gallon ASME propane tanks, intended to be approximately 1/3 linear scale for rail tank cars.

The tanks had a design pressure of 1.72 MPa at 46 °C, with a calculated burst pressure of 7.1 MPa at 46 °C, giving an as-built factor of safety of 4.1. Tank cylinders were made from SA 455 carbon steel, with a minimum Ultimate Tensile Strength (UTS) of 480 MPa to 515 MPa. The mill test report showed that the actual UTS was nearly 610 MPa, i.e. 27 % greater than required.

A pressure relief valve (PRV) was attached to each tank, with a set pressure of 2.63 MPa and a reseat pressure of 2.39 MPa.

The intention of the experiment was to partially engulf the tanks with flames, mimicking a pool fire. The expectation was that the vessel would fail within eight to ten minutes. However the first attempt at the experiment saw a sudden and extreme change in wind direction, resulting in poor contact between the flames and the vessel for approximately 36 minutes, during which there was some discharge of propane through the relief valve. A second change of the wind resulted in the flames engulfing the vessel again, rapidly resulting in a catastrophic failure in the form of 'a powerful BLEVE', with a blast overpressure breaking windows at 170 m, exceeding the predicted window breaking range of 140 m. The vessel failed with a wall temperature over 650 °C, and a pressure within the range of the PRV operation.

For the second, identical experiment, the wind conditions were more favourable for the intended flame engulfment of the vessel. The vessel failed 8 minutes after the start of the experiment, but with a different mode of failure, namely a finite rupture with a jet release. The second vessel failed with a failure tear length of 0.38 m, with a maximum opening width of 45 mm and a vertical deformation from the top of the tank of 89 mm. The measured wall temperature at the failure was in excess of 700 °C. Tank pressure was approximately 2.4 MPa at the time of failure, i.e. not high enough to lift the PRV.

A second, theoretical study was based on a series of BLEVEs that occurred on the Cosmo Oil LPG tank farm in Chiba and presents a model for predicting when a BLEVE will occur, validated against the events following the Tohoku earthquakes [14]. The modelling assumed a vessel operating pressure of 285 psi and nominal (unheated) burst strength of the vessels of 850 psi, i.e. three times the operating pressure.

The failure pressures predicted due to the weakening of the vessel walls with increasing temperature are in the range of 257 psi to 328 psi, i.e. approximately the assumed normal operating pressure of the vessels.

What these studies indicate is that the course of events for a BLEVE can involve the engulfment of the pressure vessel by fire, followed by effective operation of the pressure relief valve. However, increasing vessel wall temperature can result in the failure of the vessel within the design pressure of the vessel.

3.6 API RP 581

API Recommended Practice 581 [15] provides quantitative procedures for establishing an inspection program using risk-based methods for pressurised fixed equipment.

The basic equation for the probability of failure, $P_f(t)$, used for a Risk Based Inspection (RBI) program is:

$P_f(t) = gff.D_f(t).F_{MS}$ **Equation 1** (API RP 581 equation 1.1)

where *gff* is the generic failure frequency, $D_f(t)$ is a damage factor, and F_{MS} is a management systems factor. A table is provided with suggested generic failure frequencies to be used in the assessments for various types of components. The suggested values for tanks and vessels are shown in Table 4.

Table 4 Suggested component generic failure frequencies (*gff*) from API RP 581 (Part 2 - Table 3.1) **[15]**

Type of failure	*Tank*	*Vessel*
Small hole	7.0×10^{-5}	8.0×10^{-6}
Medium hole	2.5×10^{-5}	2.0×10^{-5}
Large hole	5.0×10^{-6}	2.0×10^{-6}
Rupture	1.0×10^{-7}	6.0×10^{-7}
Overall	1.0×10^{-4}	3.06×10^{-5}

The rupture failure frequencies suggested here for vessels are approximately an order of magnitude lower than those suggested by FRED, the Flemish Government Handbook and RIVM. However, in most cases, the generic failure frequency values used would have little effect on the final probability of failure in an overpressure situation.

Damage factors arising from a number of different mechanisms can be calculated. The procedure for assessing the damage factor for thinning has had a major revision in the latest version of the API 581 code (2016). The damage factor is based on the number and efficacy of inspections, the likely corrosion since the last inspection and future corrosion, and the strength ratio. Two equations are given for calculation of the strength ratio:

$$\mathrm{SR} = \frac{\mathrm{P.D}}{\mathrm{a.FS.t_{rd}}} \qquad \textbf{Equation 2}$$

Or

$$\mathrm{SR} = \frac{\mathrm{S.E}}{\mathrm{FS}} . \frac{\mathrm{Max}\ (\mathrm{t_{min}}, \mathrm{t_c})}{\mathrm{t_{rd}}} \qquad \textbf{Equation 3}$$

Where *P* is the pressure, *D* is the diameter, *FS* is the flow stress, *E* is weld joint efficiency, *S* is the allowable stress, t_{min} is the minimum required thickness, t_c is the minimum structural thickness and t_{rd} is the minimum measured thickness.

The flow stress is the average of the yield stress and the tensile strength of the material, which would be likely to be approximately twice the allowable stress for a modern vessel designed to a Pressure Equipment Directive based code (ASME Division 2, or BS EN 13445). Therefore, with no thinning, the value of *SR* would be approximately 0.5, and the damage factor would be negligible. With a thinning to half the minimum thickness, the SR value would increase to 1, with a significant increase in the damage factor (*DF* = 3200 assuming no further corrosion is expected). The calculations do not take into account the extent of the thinning, so highly localised thinning would be treated the same as general corrosion.

Part 1, section 7 describes calculations relating to the maintenance of pressure relief devices (PRDs). As part of this, it describes the need to consider and calculate the consequences of PRDs failing to open on demand, especially the probability of loss of containment (LOC) in the equipment being protected by the PRD, i.e. vessel failure due to overpressure.

API 581 contains a modification of equation 1.1 for calculating the probability of LOC, $P_{f,j}$, as a function of overpressure. The equation is:

$$P_{f,j} = \left(0.0312881 \cdot gfft \cdot D_f \cdot F_{MS}\right) \cdot e^{\left(3.464837 \cdot \frac{P_{o,j}}{MAWP}\right)}$$

Equation 4 (API RP 581 equation 1.32)

where $P_{o,j}$ is the pressure in the vessel.

The effects of this equation are seen in Figure 1: (taken from API 581 Part 1)

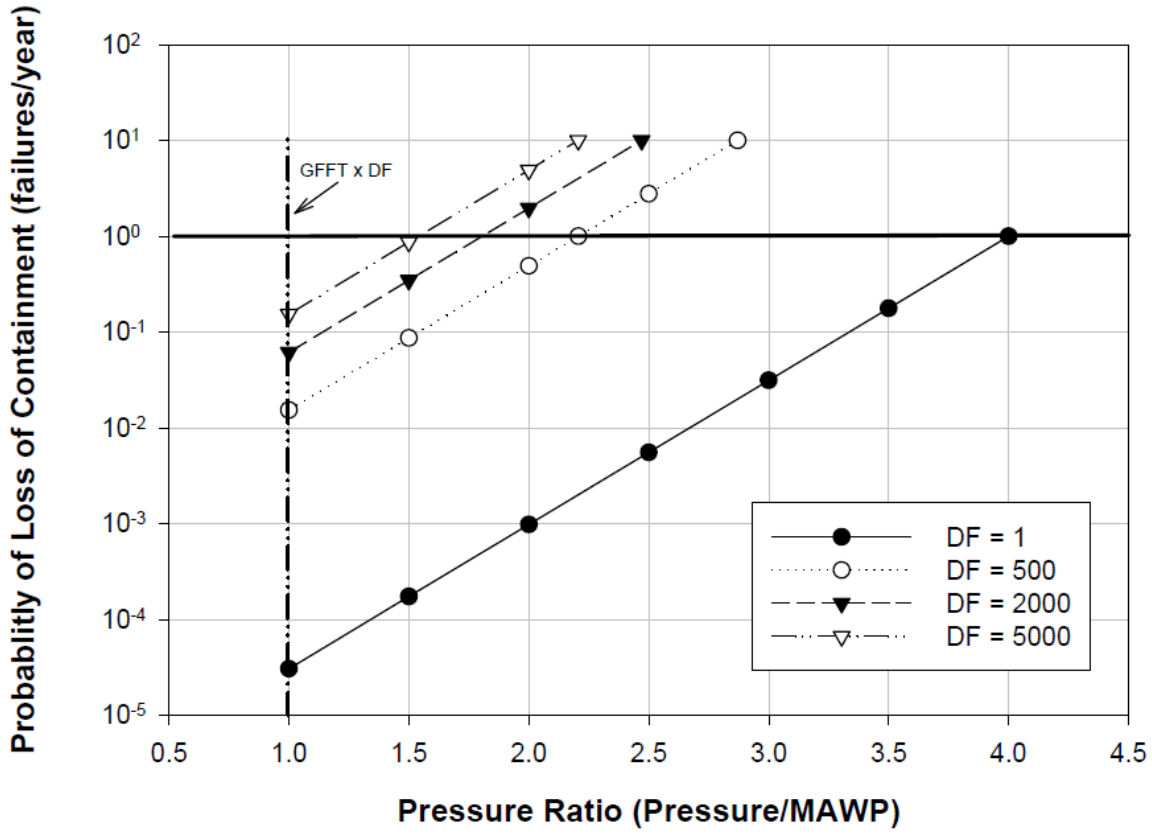

Figure 1 Probability of loss of containment, taken from API RP 581 Part 1 Figure 7.6

Where the pressure is equal to the *MAWP*, the probability of loss of containment is equal to the damage adjusted failure frequency (Equation 1). At elevated pressures, for an undamaged component, the probability of loss of containment will equal 1.0 when the pressure is four times the *MAWP*.

Section 7.4 of API 581 (Part 1) states that the burst pressure is assumed to be 4 times the *MAWP*. This is in keeping with generally expressed expectations that a mild steel pressure vessel built to the ASME code will have an ultimate tensile strength of approximately 4 times the design pressure, e.g. '*Guidelines for Vapour Cloud Explosion, Pressure Vessel Burst & Flash Fire Hazards*' [10] states:

'ASME code-compliant vessels should not exceed yield stress of the vessel at MAWP. As a result, ultimate failure pressure for an ASME code-compliant vessel is typically 3 to 4 times design MAWP, depending on the version of section of the code in use. Corrosion and fabrication allowances may further increase the ultimate failure pressure, in some instances up to 5 times MAWP. European codes design margins are typically less than US [codes], such as 2.4 MAWP for carbon steels without corrosion and fabrication allowances.'

However, ASME VIII Division 2 vessels have always had allowable stresses of at least UTS/3, and since 2007 have been in line with the European codes (i.e. UTS/2.4). There is no consideration in API 581 of the code to which vessels are constructed. Therefore, the validity of this approach for vessels constructed to different codes is questionable. There is further discussion of the differences between codes in Section 4 of this report.

As can be seen from Figure 1, the probability of loss of containment is sensitive to the damage factor (much more so than the previous version of API 581, Edition 2). An overpressure of 1.7 times *MAWP* would be required for a probability of loss of containment of 1 for a strength ratio, *SR* of 1, which would equate to a measured thickness of approximately half the required thickness, with no further corrosion considered.

4 COMPARISON OF VESSELS CONSTRUCTED TO DIFFERENT STANDARDS

There are a number of different standards that can be used for the design of pressure vessels. These standards have evolved over time as materials, processes, inspection methods and knowledge have improved. Each of the documents is extensive, extending to hundreds of pages, and therefore a full evaluation of the different standards is beyond the scope of this report.

The American Society of Mechanical Engineers (ASME) publishes a number of standards, including the Boiler and Pressure Vessel Code (BPVC). The BPVC has a number of sections covering different topics; Section VIII [16] covers the rules for the construction of pressure vessels.

The main European code is BS EN 13445: 2014 *Unfired Pressure Vessels* [17]. Part 3 covers the design rules. In the UK, this standard replaced BS 5500 *Specification for unfired, fusion welded pressure vessels*. BS 5500 took on the form of a published document (PD 5500) when BS EN 13445 was published. The transition from BS 5500 to BS EN 13445 was prompted by the adoption of the *Pressure Equipment Directive* [18], which being a harmonised standard, BS EN 13445 readily complies with.

4.1 DIFFERENCE IN ALLOWABLE STRESS

One of the most significant variations between the standards is the limits placed on the allowable stress. Typically, the maximum allowable stress has limits in terms of tensile, yield, and creep stresses. As the requirements for yield and creep vary little both between codes and over time, and as the failure criteria are more likely to be based on the ultimate tensile strength, this section will focus on changes to the tensile strength ratio.

If failure of a vessel is considered to occur when stresses in the vessel reach a certain level, then the allowable stresses will directly affect the level of overpressure that the vessel can withstand.

The current European standard, BS EN 13445 [17], and Division 2 of the American standard ASME BPVC Section VIII [16], follow the allowable stress limits set in the European Pressure Equipment Directive [18]. These are listed in Table 5. The yield stress, *Re/t*, indicates the value at the calculation temperature of:

— the upper flow limit for a material presenting upper and lower flow limits,

— the 1.0 % proof strength of austenitic steel and non-alloyed aluminium,

— the 0.2 % proof strength in other cases.

The strength limits, $R_{m/20}$ and $R_{m/t}$ represent the ultimate tensile strength at 20 °C and the calculation temperature respectively.

Table 5 Allowable stress criteria in the Pressure Equipment Directive **[18]**

Material	*Yield Criterion*	*Strength Criterion*
Ferritic steel including normalised steel, excluding fine-grained steel and specially heat-treated steel	2/3 $R_{e/t}$	5/12 $R_{m/20}$
Austenitic steel, rupture > 30 %	2/3 $R_{e/t}$	
Austenitic steel, rupture > 35 %	5/6 $R_{e/t}$	1/3 $R_{m/t}$
Non-alloy/low-alloy cast steel	10/19 $R_{e/t}$	1/3 $R_{m/20}$
Aluminium	2/3 $R_{e/t}$	
Aluminium alloys excluding precipitation hardening alloys	2/3 $R_{e/t}$	5/12 $R_{m/20}$

In the ASME codes, there are three divisions; Division 1 sets out the requirements for design, fabrication, inspection, testing, and certification of pressure vessels operating at either internal or external pressures exceeding 15 psig. The alternative rules in Division 2 are more rigorous than for Division 1, but higher design stresses are allowed. Division 3 sets out requirements for vessels with internal or external pressures in excess of 10 000 psi (690 bar), but is not discussed further here.

The allowable stress criteria based on ultimate tensile strength for the different codes are listed in Table 6. For some codes, these have changed over time.

Table 6 Changes to the allowable stress criteria based on ultimate tensile stress

Code	*Allowable stress (strength criterion)*
ASME VIII Div 1: pre 1951*	UTS/5
ASME VIII Div 1: 1951 - 1999	UTS/4
ASME VIII Div 1: post 1999	UTS/3.5
ASME VIII Div 2: pre 2007	UTS/3
ASME VIII Div 2: post 2007	UTS/2.4
BS 5500	UTS/2.35
PD 5500	UTS/2.35
BS EN 13445	UTS/2.4
Pressure Equipment Directive	UTS/2.4

*Changed to UTS/4 in 1943 to conserve material during the war; reverted to UTS/5 immediately after the war.

Large differences between more modern codes could result in significant differences. For example, a vessel constructed in the late 1990s to ASME VIII Division 1 would be likely to have a much higher failure pressure than a newer vessel constructed to Division 2, or one of the European codes.

4.2 LEVELS OF OVERPRESSURE ALLOWED IN CODES

Although pressure vessels are designed with a maximum allowable working pressure (*MAWP*), it is recognised that in certain circumstances, events may occur that generate pressures above this level. In such situations, pressure relief systems should be in place to limit the degree of overpressure occurring. The system requirements, and maximum overpressures allowed vary between codes.

4.2.1 ASME VIII

The ASME VIII [16] discusses the permitted accumulation during pressure relief. Accumulation is normally defined as the pressure increase over *MAWP* of the vessel (pressure units or % *MAWP*). ASME VIII Division 1 gives accumulation limits which are scenario dependent and given in Table 7. The wording of ASME VIII Division 1 is that the accumulation 'shall not exceed'.

Table 7 ASME VIII Scenarios and % *MAWP*

Scenario	***% MAWP***
Single valve other than fire	110
Single rupture disc other than fire	110
Multiple valve other than fire	116
Multiple rupture disc other than fire	116
Fire (all combinations)	121

4.2.2 Pressure Equipment Directive

The Pressure Equipment Directive (PED) – 97/23/EC [18] and the UK's associated Pressure Equipment Regulations 1999 applied to equipment with a maximum allowable gauge pressure greater than 0.5 bar, on the market in the European Union (EU) from 30 May 2002. The original directive was superseded by Directive 2014/68/EU, which was enacted in the UK as the Pressure Equipment (Safety) Regulations 2016, on 8 December 2016. The approach to accumulation has remained consistent.

Overpressure devices are termed 'Safety accessories'. Momentary surges are generally permitted up to 10 % of the design pressure. Thus the pressure limit is 1.1× maximum allowable pressure. The exception is for fire cases where detailed thermal response evaluation is required. Thus for fire cases, momentary surge limits are set by the vessel designer.

4.2.3 National Fire Protection Association

The National Fire Protection Association has produced two standards that may be relevant: NFPA 68 *Explosion Protection by Deflagration Venting* and NFPA 69 *Explosion Prevention Systems*. The purpose of NFPA 68 is to provide the user with criteria for design, installation, and maintenance of deflagration vents and associated components. As such, it does not apply to pressure vessels themselves, although they are to be designed to the ASME VIII pressure vessel code.

Both documents contain equations for the design pressure based on maximum deflagration pressure and whether permanent deformation can be tolerated. Where permanent deformation cannot be

tolerated, the systems have a design pressure based on two-thirds of the yield stress. This is effectively no overpressure.

For systems where permanent deformations (but not rupture) can be tolerated, the design pressure is based on two-thirds of the ultimate strength of the material. Depending on the relationship between yield and ultimate strengths, this could equate to a maximum overpressure of 160 % of maximum allowable pressure.

5 PREDICTIONS OF EFFECTS OF OVERPRESSURE

There are a number of tables in publications that aim to predict the likely consequences of different levels of overpressure.

5.1 CENTRE FOR CHEMICAL PROCESS SAFETY (CCPS)

The CCPS book [19] on Initiating events and independent protection layers in layers of protection analysis (LOPA) has Appendix E, which discusses Pressure Vessels and Piping Overpressure Considerations. It gives two options for dealing with the effect of excessive pressure in the pressure vessels:

> Option 1: Assume any pressure rise that exceeds code criteria will result in a rupture with major direct and indirect consequences in terms of human fatalities and injuries, capital and operating losses, and environmental impact.
>
> Option 2: Assume that there is a hierarchy of consequences, based upon the overpressure that occurs.

An example hierarchy of possible consequences as a function of vessel overpressure is given in Table 8. The example relates to carbon steel vessels designed as per ASME Boiler and Pressure Vessel Code (BPVC) Section VIII, Division 1 (2013). CCPS note that for other design codes and other materials and grades, the consequence versus percentage accumulation may be more severe. They also note that catastrophic failures will occur at lower overpressures if the vessel: is beyond its corrosion allowance, has experienced over temperature or overpressure excursions in the past, is operated below its ductile/brittle transition temperature, or exhibits pitting or cracking.

Table 8 Conceptual consequence vs. pressure vessel overpressure

Accumulation (% MAWP)*	***Significance***	***Potential Consequence***
110	Allowable accumulation for process upset cases (non-fire) protected by a single relief device	No expected consequence at this accumulation level
116	Allowable accumulation for process upset cases protected by multiple relief devices	No expected consequence at this accumulation level
121	Allowable accumulation for external fire relief cases, regardless of the number of relief devices	No expected consequence at this accumulation level
>121 to 130	Standard hydrostatic test pressure	Increased likelihood of leaks in associated flanges, piping, equipment, etc.
>130	Minimum yield and ultimate strength varies with material and grade	Catastrophic failure becomes increasingly likely. Since this level of overpressure goes beyond code allowance, an analysis and supporting documentation by the organisation will be necessary to evaluate the severity of the consequence of overpressure

Expressed as percentage over MAWP in source document, i.e. 110 % MAWP = 10 % **over MAWP*

5.2 J P WILLIAMS

J P Williams [20] discusses reliability for safety instrumented systems and provides Table 9 which summarizes the effect of overpressure on a pressure vessel where there are no mechanical integrity issues and the design temperature is not exceeded. The table is again for carbon steel vessels, but refers to various ASME VIII division codes. It also provides target event frequencies for two pressure ranges given in the table:

- 135 % to 165 % overpressure events have a threshold frequency of 1 x 10^{-4}/year;
- 165 % to 300 % overpressure events have a threshold frequency of 1 x 10^{-5}/year.

Table 9 Effect of pressure accumulation in carbon steel vessels

Accumulation (% MAWP)	***Effects***	***Remarks***
< 135	None expected	None
135 – 165	Potential for slight permanent deformation	This range of pressure corresponds to the tensile limit of the vessel and is both material and code dependent. The lower and upper limits correspond to the ASME VIII, Div.2 and ASME VIII, Div. 1 (1998 edition and earlier) vessels, respectively. ASME VIII, Div. 1 (1998 edition with 1999 addenda) vessels fall in between these values. Therefore, a representative value for this range is 150 %.
165 – 300	Permanent deformation, possible small leak	Valid for remote contingencies, as more frequent overpressure could weaken the vessel by fatigue.
300 – 400	Same as above, but with a higher likelihood of a large leak or burst	Dangerous overpressuring.
400 – 500	Burst	Typical for healthy ASME VIII code vessels.

5.3 API 581

API RP 581 [15] contains a table showing the potential consequences of degrees of overpressure, reproduced in Table 10.

Table 10 Expected effects of overpressure on pressure vessels, as given in API RP 581 **[15]**

Accumulation (% MAWP)*	***Significance***	***Potential Consequence***
110	ASME code allowable accumulation for process upset cases (non-fire) protected by a single relief device	No expected consequence at this accumulation level.
116	ASME code allowable accumulation for process upset cases (non-fire) protected by multiple relief devices	No expected consequence at this accumulation level.
121	ASME code allowable accumulation for external fire relief cases regardless of the number of relief devices	No expected consequence at this accumulation level.
150	ASME standard hydrostatic test pressure (may be 30 % on new designs)	Possible leaks in associated instrumentation, etc. Medium consequence.
190	Minimum yield strength (dependent on the materials of construction)	Catastrophic vessel rupture, remote possibility. Significant leaks probable. Failure of damaged vessel areas (corrosion, cracks, blisters, etc.) likely. High consequence.
400	Ultimate tensile strength (dependent on materials of construction)	Catastrophic vessel rupture predicted. Highest consequence.

Expressed as percentage over MAWP in source document, i.e. 110 % MAWP = 10 % **over MAWP*

As with Equation 1.31, it is not clear what information or data supports the statements in the table above.

5.4 N PROPHET

N Prophet *et al* [21] claim that the effects of pressure accumulation on steel vessels designed to ASME VIII pressure vessel code are well documented. Prophet presents the same data as in Table 9.

Prophet establishes a set of risk criteria using these overpressure effect characteristics:

Decide what level of overpressure is not acceptable and assign a very low event frequency such as 10^{-6}/yr. The probability of vessel failure becomes significant for any overpressure event that subjects a vessel to a pressure of 300 % of the *MAWP*. Such an event should however not knowingly be designed for. Hence, accumulations greater than this value are not considered.

The frequency for the 165 % to 300 % *MAWP* event should be set at 10^{-5}/yr, which is an order of magnitude more than the event frequency value for unacceptable pressure accumulations of greater than 300 %. However, this pressure range is barely above hydro-test at one extreme to a level above the yield point at the other. Thus a frequency of 10^{-5}/yr would be right for the upper end of the range; it is quite conservative at the lower end.

A better risk-consequence characterization is obtained by further dividing the 165 % to 300 % range into two ranges: 165 % to 200 % and 200 % to 300 %; with frequencies of 10^{-4}/yr and 10^{-5}/yr respectively.

Prophet then presents a target frequency for an overpressure event matrix, which is shown in Table 11.

Table 11 Arbitrary target event frequencies

Accumulation (% MAWP)	***Frequency***
<135	1 in 100 years (10^{-2}/yr)
135 – 165	1 in 1000 years (10^{-3}/yr)
165 – 200	1 in 10 000 years (10^{-4}/yr)
200 – 300	1 in 100 000 years (10^{-5}/yr)
>300	Not allowed

6 CALCULATIONS OF BURST PRESSURE

Calculations for burst pressure have been made, based on assessments of membrane stresses in the shell wall. For these cases, the geometry is simple and the stresses in the vessel are directly related to the internal pressure in the vessel, enabling a failure pressure to be predicted. For corroded vessels, predictions based on finite element analysis with experimental verification have been used.

For more complex geometry, such as nozzles and flanges, obtaining predictions of burst or leak pressures would be more difficult. The stresses in these components can vary according to a wide range of factors, including size, design (PD 5500 lists 16 different designs of bolted flange), gasket type and location, external forces from connections, bolting and internal pressure. Therefore, a direct relationship between internal pressure and stress is unlikely to exist.

From the absence of predictions for failure of other components, it should not be assumed that they would not fail in an overpressure event.

6.1 BURST PRESSURE FOR INTACT VESSELS

6.1.1 Simple tubes

There are many formulae for estimating the burst pressure of simple cylinders. The most straightforward is Barlow's formula, which simply states that failure occurs when the hoop stress reaches the ultimate tensile strength of the material:

$$\mathrm{P_b} = \frac{2\boldsymbol{\sigma}_\mathrm{u}\mathrm{t}}{\mathrm{d_o}} \qquad \textbf{Equation 5}$$

where P_b is the burst pressure, σ_u is the ultimate tensile strength, t is the wall thickness and d_o is the outer diameter of the cylinder. This equation has the advantage of requiring only readily available material data and has a form similar to those used in the design calculation of the vessel. If the maximum allowable stress has been defined in terms of a proportion of the tensile strength (e.g. maximum design stress = tensile strength/2.4) then the likely failure pressure could be expressed as a multiple of maximum allowable working pressure (in this case, burst pressure = 240 % *MAWP*). If other factors had been used when calculating the *MAWP*, such as a joint factor for a longitudinal weld, the ratio would be higher. The effect of welds and imperfections will be discussed later.

While Barlow's formula is appropriate for relatively thin-walled cylinders, Lamé's formula is more general and appropriate for thick-walled cylinders:

$$\boldsymbol{P_b = 2\sigma_u\left(\frac{d_o - d_i}{d_o + d_i}\right)} \qquad \textbf{Equation 6}$$

where, in addition to the variables already described, d_i is the inner diameter of the cylinder. A number of variations have been proposed, which are listed in Table 12. The effect of the variations is shown for an example cylinder, based on an example from the API 579 Example Manual [22]. This was for a cylinder with an outer diameter of 2320 mm, 17 mm wall thickness, yield stress of 260 MPa and UTS of 485 MPa.

Table 12 Burst pressure calculations based on ultimate tensile strength

Source	*Equation*	*Example Burst Pressure (MPa)*
Barlow	$\frac{2\sigma_u t}{d_o}$	7.11
Lamé	$2\sigma_u\left(\frac{d_o - d_i}{d_o + d_i}\right)$	7.16
Turner [23]	$\sigma_u \ln R$	7.16
Nadia [24]	$\frac{2}{\sqrt{3}}\sigma_u \ln R$	8.27
Soderberg [25]	$\frac{4}{\sqrt{3}}\sigma_u\left(\frac{R - 1}{R + 1}\right)$	8.27

In the above equations, $R = d_o/d_i$

While the Lamé and Turner equations, and the Nadia and Soderberg equations give the same burst pressure in this thin-walled example, differences emerge for thicker-walled vessels.

A number of equations have been proposed that take into account the yield stress and strain hardening, rather than relying solely on the ultimate tensile strength. Rajan *et al* [26] list a number of burst pressure equations taking into account further material properties, including those shown in Table 13.

Table 13 Burst pressure calculations taking into account yield stress and strain hardening

Source	*Equation*	*Notes*
Faupel and Furbeck	$\left(\frac{2\sigma_0}{\sqrt{3}}\right)\left(2 - \frac{\sigma_0}{\sigma_u}\right)\ln R$	Thick wall
Svensson	$\sigma_u\left(\frac{0.252}{n + 0.277}\right)\left(\frac{e}{n}\right)^n \ln R$	Thick wall
Svensson	$\frac{\sigma_0 t}{a}\left(\frac{n}{e}\right)^n \frac{2}{(\sqrt{3})^{n+1}}$	Thin wall

σ_0 is yield stress, σ_u is the ultimate tensile strength, e is the base of the natural logarithm, n is the strain hardening exponent, a is the cylinder radius and t is the thickness

Rajan *et al* found that the Svensson thick wall equation was a close match with experimental thick-walled pressure vessels, but the Svensson thin wall equation overpredicted the burst pressure for thin-walled vessels by as much as 11 %. They proposed an additional factor to allow for the anisotropic effect; the strength of the flow formed vessels they were investigating was generally higher in the longitudinal direction than the more important hoop direction. With the use of the factor, the error reduced to an average of 2.85 %

Zhu and Leis [27] also developed an expression for burst pressure taking into account strain hardening laws:

$$P_b = \left(\frac{C}{2}\right)^{n+1} \frac{4t}{d_o} \sigma_u \qquad \textbf{Equation 7}$$

where *C* is a yield criterion-dependent constant, and *n* is the strain hardening exponent of the material, which is dependent on the ratio of yield stress to ultimate tensile strength. The original paper presented values of *C* for two different yield criteria; Tresca (*C* = 1) and for von Mises (*C* = 2/√3). In a subsequent paper [28], a new yield criterion was proposed, named the Average Shear Stress Criterion (ASSC), with a *C* value of ½ + 1/√3.

Predicted burst pressures using the three yield criteria, normalised against the Barlow burst pressure, were plotted with experimental failure data from a number of previous studies. The von Mises criteria provided the highest predicted burst pressures, with the Tresca giving the lowest. The ASSC predictions were midway between the von Mises and Tresca, and for an *n* value of approximately 0.12, equates to the Barlow solution. The experimental data showed good agreement with the ASSC predictions. The von Mises provided an upper bound to the experimental data, and the Tresca, a lower bound. One point lay above the von Mises predictions, and four points below the Tresca. Therefore, the over-optimistic burst pressures obtained using the von Mises criterion are non-conservative; many of the experimental bursts occurred at pressures in excess of 10 % below the von Mises prediction.

6.1.2 Cylinders and pressure vessels

The failure of actual vessels (as opposed to idealised cylinders) was investigated by Kaptan and Kisioglu [29] who tested over 170 liquefied petroleum gas (LPG) cylinders and compared the burst pressures obtained to those predicted by finite element analysis. Three different sizes of cylinder were tested; 35 l, 60 l and 80 l, all with the same nominal diameter and wall thickness. The range of burst pressures over all the cylinders was 7.40 MPa to 9.36 MPa, with the smaller capacity cylinders generally showing slightly higher burst pressures. The finite element analysis predicted the burst pressure well, with the largest difference between finite element analysis (FEA) prediction and mean burst pressure being 6.7 % (35 l cylinder). The Barlow formula would predict a burst pressure of 7.6 MPa, based on the ultimate tensile strength of the cylinder in the hoop direction. This could represent a lower bound to the burst pressures, with just four cylinders failing at pressures below this.

The ability of FEA to predict the burst pressure and location for a cylindrical shell intersection (nozzle) was investigated by Xue, Widera and Sang [30]. While the model correctly predicted the location of the failure (at the nozzle/shell weld approximately 15° from the top centre line of the vessel), the predicted burst pressure was 14.5 % lower than the actual burst pressure of 21.4 MPa. It should be noted that the original weld leaked at a pressure of 16 MPa and had to be repaired before retesting. Using the Barlow formula to predict burst pressure of the shell would give 19.1 MPa, which is closer to the actual failure pressure than the FEA prediction. Therefore, it would appear that the stress concentration effects of the nozzle did not significantly affect the burst pressure, although a vessel without the nozzle was not tested.

6.1.3 Relation of burst pressure to maximum allowable pressure

Assuming that the burst pressure of a vessel can be estimated, the actual burst pressure in relation to the maximum allowable pressure will vary due to a number of other factors including:

- The allowable stress according to the design code (discussed in Section 4);
- The actual thickness of the vessel, including any additional thickness due to using standard plate thickness or corrosion allowances;
- Temperature, with material properties changing with elevated temperature;
- Duration of the overpressure, with very short pressure peaks being less severe than longer durations; and
- The degree to which the vessel material mechanical properties exceeds the minimum specified for the grade.

6.2 EFFECT OF IMPERFECTIONS AND DEGRADATION

6.2.1 Cracks and manufacturing defects

Cracks can significantly weaken a vessel and can cause catastrophic failure without warning. The risks posed by brittle fracture are higher for low toughness materials, or when working at low temperatures. Cracks may be present in a vessel for a number of reasons, such as:

- Manufacturing defects, including defects in welds;
- Cracks occurring due to fatigue crack growth, usually from a small initial crack, stress concentration or corrosion pit; or
- Stress corrosion cracking, where cracks form due to a combination of high stress, environmental conditions and a susceptible material.

Cracks due to manufacturing would be present during the commissioning stage, and may be identified during inspection or hydrotesting if large enough. However, the other mechanisms can cause in-service crack growth which may be difficult to detect.

In a study of thick-walled steel pressure vessels [31] it was found that there were typically 12 defects per cubic metre of weld metal. The majority of these (more than 90 %) were in the weld or associated heat affected zone. The number of defects decreased exponentially with increasing defect size. In a paper estimating the likelihood of spontaneous failure of LPG storage vessels, O'Donnell *et al* [32] claim that there is more recent work that suggests that the defect density may be underestimated for smaller flaws by as much as an order of magnitude. Unfortunately, they do not quote a reference for this.

Using a probabilistic fracture mechanics approach, O'Donnell *et al* estimate that failure frequencies are no more than 10^{-7} per vessel year in normal operating conditions for large LPG storage vessels, dropping to 10^{-9} for smaller vessels. Their work assumes that the vessels have been proof-tested, which has a number of benefits, such as screening out combinations of low toughness and large defects. However, they calculate an increase in failure probability of approximately two orders of magnitude when increasing the pressure from normal to the safety valve set pressure. This would appear to be a large increase in failure likelihood for a relatively small increase in pressure; an increase still well within the likely hydrotest pressure.

In a paper discussing the benefits and disadvantages of hydrotesting, Kirkwood and Cosham [33] define three classes of defect sizes;

- Class 1; small cracks that survive the hydrotest
- Class 2; defects that fail the hydrotest, but not at design pressure
- Class 3; defects that would fail at design pressure

There may also be other cracks that would be small enough to not cause failure under any loading conditions. It is likely that the number of Class 1 defects would greatly outnumber those in Class 3. By definition, a hydrotest over the maximum operating pressure would cause failure from Class 2 and Class 3 defects, leaving only Class 1 defects in the vessel. Since only the large Class 3 defects would fail at design pressure, a safety margin is established.

Any pressure over the hydrotest pressure may result in an increase in stress that causes Class 1 defects (small cracks that survive a hydrotest) to become critical and initiate failure. As the number of cracks increases as crack size decreases, there are likely to be many more small cracks, which could significantly increase the likelihood of failure. However, such failures occurring well below the plastic burst pressures have not been reported in experimental tests. This may be due to the relatively small number of experimental tests. It may also be likely that as the stresses in the vessel approach yield (which may occur near the hydrotest pressure), the plastic failure mode would dominate and the effect of small cracks would not be significant.

If cracked vessels are evaluated using a fitness for service procedure, such as API 579 [22] or BS 7910 [34], the calculations for stress intensity and reference stress are best estimates and the conservatism arises from the use of partial safety factors. For the API procedure, the partial safety factors apply to the stress, fracture toughness and crack size. The factors used depend on four variables; the target probability of failure, the crack depth (smaller or larger than 5 mm), coefficient of variation (how well the stresses are known) and the ratio of fracture toughness to yield stress. Where the fracture toughness to yield stress ratio is high, the partial safety factors only apply to the stress (the safety factors for toughness and crack size are 1.0).

Assuming that the consequences of a pressure vessel failure would be severe, and the lowest target failure probability is used, the partial safety factor on stress would be at least 1.7, and for toughness, 2.0, assuming a low toughness to yield ratio. This would give a minimum safety factor of 3.4. For a high toughness to yield ratio, the stress partial safety factor would be at least 2.0.

The British standard, BS 7910, also uses the approach of partial safety factors, but with slightly lower factors. The combined stress and toughness safety factors for the lowest failure probability would be 2.2, for the lowest coefficient of variation.

6.2.2 Corrosion

Uniform corrosion, resulting in a general thinning of the wall over the full surface area, would result in a reduction in strength that can be estimated by simply using the new wall thickness. For large areas of corrosion, this assessment approach may provide a useful estimate of the strength, at least as a lower bound estimate of the burst pressure.

Chiodo and Ruggieri [35] show burst pressure is predictable using the Zhu and Leis [27] equation for the remaining ligament for long defects. The width and profile of the defects were shown to have little effect, except for very narrow defects where the stress concentration effects may become significant.

However, for less extensive or localised corrosion, this approach would be overly conservative. There have been a large number of studies investigating the effect of corrosion on burst pressure. Although the majority of these have focused on pipes, the same principles should be transferable to cylindrical pressure vessels. Some have been used to formulate guidance documents such as ASME B31G [36] and DNV-RP-F101 [37]. These are generally considered to offer conservative assessments.

Most assessment methods for localised corrosion are based on a length factor:

$$\frac{l}{\sqrt{Rt}}$$

where l is the axial length of the defect, R is the radius and t is the wall thickness, although some investigators have used the original thickness, and some have used the corroded thickness. A typical plot of burst pressure variation with the length factor is shown in Figure 2. Kiefner [38] uses plots such as these, which shows the different failure mechanisms likely for various combinations of length and depth; short, deep areas of corrosion are more likely to result in leaks, whereas longer areas of corrosion are likely to result in rupture.

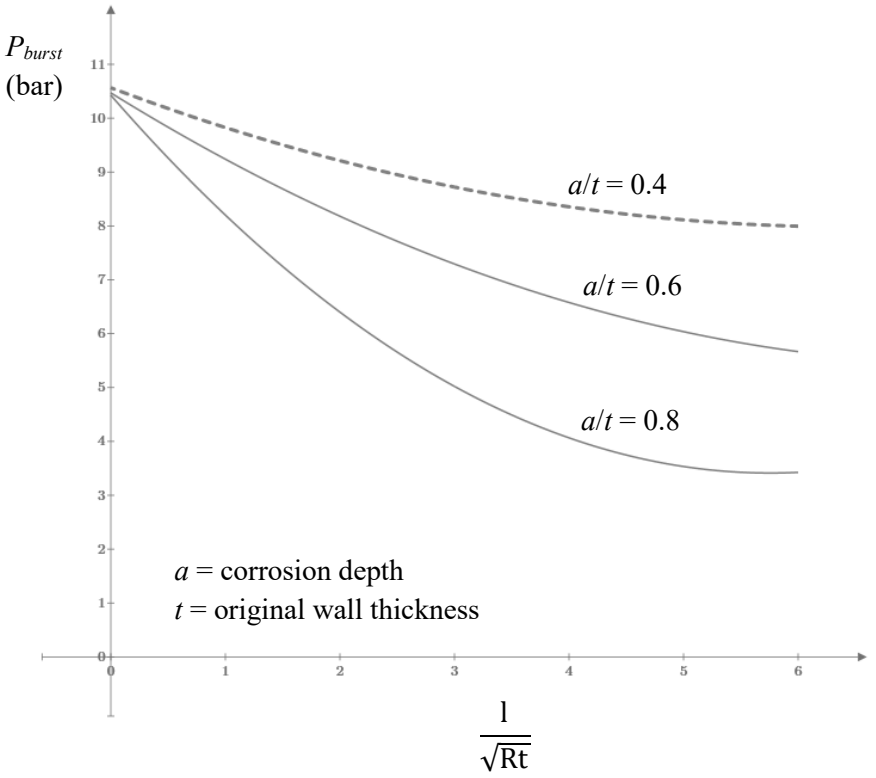

Figure 2 Typical plot relating length of corrosion defect to burst pressure

There are a number of different equations proposed for the evaluation of localised corrosion using similar approaches. The main focus of these investigations has been on obtaining a good agreement with the material properties, rather than the effect of the corrosion geometry. These are extensions of the uncorroded pipe burst pressure equations, where different terms relating to strain hardening and yield stress have been investigated.

Choi *et al* [39] used finite element analysis to investigate the most appropriate factor for strength. They compared FEA predictions based on yield, flow, various proportions of ultimate strength to experimental burst pressures and found that a simple factor of 0.9 × ultimate tensile strength fitted the data best. By applying regression analysis to the FEA results, a limit load solution was proposed:

$$P_{max} = 0.9 \times \frac{2t}{D_i} \boldsymbol{\sigma}_u \left[C_2 \left(\frac{l}{\sqrt{Rt}} \right)^2 + C_1 \left(\frac{l}{\sqrt{Rt}} \right) + C_0 \right] \qquad \textbf{Equation 8}$$

$$C_2 = 0.1163 \left(\frac{a}{t} \right)^2 - 0.1053 \left(\frac{a}{t} \right) + 0.0292$$

$$C_1 = -0.6913 \left(\frac{a}{t} \right)^2 + 0.4548 \left(\frac{a}{t} \right) - 0.1447$$

$$C_0 = 0.06 \left(\frac{a}{t} \right)^2 - 0.1035 \left(\frac{a}{t} \right) + 1.0$$

Where P_{max} is the predicted burst pressure, a is the defect depth, t is the original wall thickness, l in the flaw axial length, D_i is the inner diameter, and R is the radius.

7 WORKED EXAMPLES

In order to assess the implications of the different methods, two worked examples will be presented here, one for an intact vessel, and one for a corroded vessel. In these examples, an estimate of the actual burst pressure has been made, and compared to the maximum allowable working pressure and predictions from API 581. Both examples are based on a vessel featured in the API 579-2/ASME FFS-2 2009 Fitness-For-Service Example Problem Manual [22]. The details of the vessel are listed in Table 14. It is assumed that there is no corrosion allowance, and that the weld joint efficiency factor is 1.

Table 14 Details of vessel used in worked examples

Diameter	60 in	1524 mm
Wall thickness	1 in	25.4 mm
Material	516 Grade 70	
Yield strength	38 ksi	260 MPa
Ultimate tensile strength	70 ksi	485 MPa

7.1 INTACT VESSELS

For an intact cylindrical vessel, the burst pressure can be approximated by assuming failure occurs when the hoop stress reaches the ultimate tensile stress, as discussed in Section 6.1. Therefore, the ratio of burst pressure to *MAWP* depends on the maximum allowable stress, which depends on the design code used.

Table 15 shows the *MAWP* for the assessed vessel, assuming construction to various codes. As the material used was relatively ductile, with a low yield to strength ratio, the yield stress/1.5 criterion resulted in a lower allowable stress than the ultimate tensile strength criterion for ASME VIII Division 2 and European codes.

Table 15 Maximum allowable working pressures (*MAWP*) for vessel designed to different codes

	Pressure (bar)	
Estimated Burst Pressure (Lamé)	163	
Code	***MAWP* (bar)**	***Burst/MAWP***
ASME VIII Div 1: 1951 - 1999	40.8	4.0
ASME VIII Div 1: post 1999	46.6	3.5
ASME VIII Div 2: pre 2007	54.4	3.0
ASME VIII Div 2: post 2007	59.0	2.8
BS 5500/PD 5500	59.0	2.8
BS EN 13445	59.0	2.8

The codes with a burst/*MAWP* ratio of 2.8 would have a lower ratio of 2.4, or 2.35 for the BS/PD 5500 code, if the allowable stress was based on the ultimate tensile strength criterion (i.e. for steels with a higher yield stress). This would occur for steels with a yield to strength ratio above approximately 0.63.

The burst pressure calculation has assumed that the actual wall thickness is the minimum wall thickness; any unused corrosion allowance would increase the burst pressure. Also, the calculation has assumed that the actual tensile and yield strengths of the material are the minimum specified for the grade of material. In reality, the strengths are likely to be somewhat higher.

However, the assumptions made in some published tables of overpressure consequence appear to be optimistic; claiming that 4 times the *MAWP* would be required to burst a pressure vessel (as in the tables contained in Williams [20], Prophet [21] (Table 9) and API 581 [15] (Table 10)) would not be appropriate for modern vessels constructed to ASME VIII Division 2 or European standards.

7.2 LOCAL CORROSION

Under the API 579 *Fitness-for-Service* assessment procedures, a certain level of corrosion is acceptable before a reduction in the *MAWP* is required. This corresponds to an area of corrosion for which the Remaining Strength Factor, *RSF*, is equal to the allowable Remaining Strength Factor, RSF_a. For long areas of corrosion, the *RSF* approximates to the remaining wall thickness ratio. For short areas of corrosion, corrosion depths can be significantly larger.

The assessment procedure contains a table for the recommended values for the RSF_a, depending on which code the vessel has been designed to; however, the recommended value is 0.9 for all codes listed, which include ASME VIII Division 1 (pre and post 1999), Division 2 and BS/PD 5500. The document claims that these have been shown to be conservative.

For a range of corrosion lengths, the depths which gave *RSF* values of 0.9 were calculated using the API 579 procedures. For each combination of depth and length evaluated, burst pressures were calculated using the formula of Choi *et al* [39].

The results of the assessment of the vessel with a *RSF* of 0.9 are shown in Table 16. A number of different length/corrosion depth combinations were modelled. Burst pressures were predicted using the formula by Choi *et al* [39].

Table 16 Results for low level of corrosion, not requiring reduction in *MAWP*

Description of assessment	***Results***	***Notes***
MAWP Division 1	47 bar	Assuming UTS/3.5 as for Div 1 post 1999
MAWP Division 2	59 bar	Allowable stress limit based on yield/1.5 (lower than UTS/2.4 for this material)
Min Burst Pressure (Choi)	129 bar	Corrosion area 458 mm long, 4.5 mm deep
Max Burst Pressure (Choi)	139 bar	Corrosion area 674 mm long, 3.8 mm deep
Min burst pressure/*MAWP* (Division 1)	2.8	
Min burst pressure/*MAWP* (Division 2)	2.2	
API 581: Pressure/*MAWP* at failure frequency of 1/yr	3.3	Based on two inspections of level C effectiveness, deepest corrosion assumed (4.5mm)

The ratios of burst pressure/*MAWP* are lower than for the uncorroded example. This is to be expected, as the remaining strength factor is 0.9 compared to the original vessel. However, the reduction in the ratio is slightly higher than would be accounted for by this factor alone. This is mainly due to the additional 0.9 factor on the tensile strength in the burst pressure calculation of Choi *et al*. Other solutions use other factors, most of which are higher. Therefore, the burst pressure, and therefore the burst pressure to *MAWP* ratio, may be considered as a lower bound. There are also small differences in the calculation of the effect of thinning.

The lowest acceptable *RSF* for a corrosion area is 0.7, although in this case, the *MAWP* would have to be reduced to regain an effective remaining strength factor of 0.9. A second set of calculations were performed to model deeper corrosion with a *RSF* of 0.7 and the results are listed in Table 17. In these cases, the *MAWP* has been reduced by a factor of 0.7/0.9 (i.e. a factor of 0.78).

Table 17 Results for higher level of corrosion, requiring reduction in *MAWP* with RSF = 0.7

Description of assessment	***Results***	***Notes***
$MAWP_r$ Division 1	36 bar	Assuming UTS/3.5 as for Div 1 post 1999
$MAWP_r$ Division 2	46 bar	Allowable stress limit based on yield/1.5 (lower than UTS/2.4 for this material)
Min Burst Pressure (Choi)	112 bar	Corrosion area 674 mm long, 10.2 mm deep
Max Burst Pressure (Choi)	117 bar	Corrosion area 231 mm long, 15.2 mm deep
Min burst pressure/$MAWP_r$ (Division 1)	3.1	
Min burst pressure/$MAWP_r$ (Division 2)	2.4	
API 581: Pressure/$MAWP_r$ at failure frequency of 1/yr	1.9	Based on two inspections of level C effectiveness, deepest corrosion assumed (15.2 mm)

The burst pressure/*MAWP* ratios in this case are slightly higher than for the cases with the lower corrosion depth. This is due to the solutions for the effect of the corrosion; the API solution predicts a strength reduction of 0.7 for all depth/length combinations evaluated, whereas the Choi equation predicted strength reductions of between 0.74 and 0.78.

If the corrosion solutions were the same and no factor was applied to the strength, a vessel acceptable to the API 579 procedure would have a reduction in burst pressure/*MAWP* of 0.9. The lowest original (intact vessel) ratio would be 2.35, assuming a vessel constructed to BS/PD 5500 and a yield to tensile strength ratio above 0.63. Therefore, such a vessel with corrosion should have a burst pressure/*MAWP* ratio of 2.1. Some variation may occur due to different factors on tensile strength and strength reduction in the different burst pressure predictions.

The pressure ratio to give annual failure probability of 1.0 calculated using API 581 [15] are also given in the two tables.

API RP 581 states that:

> *"[Damage factors] do not provide a definitive Fitness-for-Service (FFS) assessment of the component. Fitness-for-Service analyses for pressurized components are covered by API 579-1/ASME FFS-1. The basic function of the DF is to statistically evaluate the amount of damage that may be present as a function of time in service and the effectiveness of an inspection activity to quantify that damage"*

The procedure for assessing thinning in API 581 is applicable to both general and localised thinning. The damage factor for thinning is dependent on the degree of thinning and the effectiveness of the inspection. For local thinning, a highly effective, category A inspection would involve 100 % visual examination and thickness measurements (intrusive inspection), or 50 % to 100 % coverage using automated ultrasonic scanning, or profile radiography in areas specified by a corrosion engineer (non-intrusive inspection). The least effect category (E) would involve spot ultrasonic thickness

measurements without the areas being specified by a corrosion engineer. Credit is given for more inspections for all but the least effective inspection category.

For the purposes of these worked examples, the calculations for pressure ratios to cause loss of containment in API 581 have been based on two inspections of level C effectiveness. API 581 does not distinguish between construction codes so the ratio given in Table 16 and Table 17 would be applicable to vessels constructed to both Division 1 and Division 2.

The calculated ratios of burst pressure to *MAWP* are sensitive to the degree of corrosion (in contrast with the previous version, Edition 2), with a reduction from 4.0 (no corrosion) to 1.9 (highest corrosion) for the failure probability of 1 for the larger amount of corrosion. The value of 1.9 was based on the largest depth of corrosion, which was only acceptable in the API 579 fitness-for-service assessment due to the short length of the corrosion. If the other combination of depth and length to give the same fitness-for-service results (674 mm length and 10.2 mm depth) were used, a pressure ratio of 2.7 would be obtained using API 581. The low value of 1.9 (the lowest ratio using any method) is therefore artificially low, and due to the inability of the API 581 procedure to take into account the effect of length of the corroded area.

8 CONCLUSIONS

1. There is considerable variability in pressure vessel failure rates calculated in literature. This may either be due to the quality of data collection, or an indication that failure rates may vary depending on factors such as the fluid service and the broader operating environment.
2. For intact vessels, burst pressure can be estimated as the pressure at which the hoop stress reaches the ultimate tensile strength of the material. There are many formulae available for the calculation of burst pressure, but using the simple Barlow formula is likely to give a sufficiently accurate estimate.
3. A number of other factors are likely to affect the burst pressure, including:
    a. additional thickness due to corrosion allowance;
    b. material properties exceeding the minimum specified for the grade; and
    c. duration of the overpressure.
4. The ratio of the burst pressure to the maximum allowable pressure for the vessel will depend to the largest degree on the maximum allowable stress defined by the design code. The allowable stress could vary by a factor of two. Therefore, any system predicting the effects of overpressure, or setting limits for allowable overpressure, must take into account the code to which the vessel was designed.
5. Both ASME Boiler and Pressure Vessel Code, Section VIII and the Pressure Equipment Directive allow accumulation during pressure relief above the maximum allowable working pressure but differ in the permissions they apply.
6. The likely effect of various levels of overpressure on pressure vessels has been presented in tables in a number of publications (Centre for Chemical Process Safety (CCPS), Williams and Prophet), but they are merely examples and are mainly for ASME VIII carbon steel vessels. The publications also caution against using the tables where there is existing damage to the vessel.
7. API 581 assumes that vessels will certainly fail at an overpressure factor of 400 % design pressure and lists potential consequences at various degrees of overpressure up to this point. However, it is not clear on what experimental evidence these assumptions are based.
8. While API 581 assumes that vessels will certainly fail at an overpressure factor of 400 % design pressure, there is some incident evidence that vessels sometimes survive to higher degrees of overpressure before failure.
9. There is also corresponding evidence to suggest that vessels sometimes fail at lower degrees of overpressure, possibly because these vessels are built to different design codes.
10. In the special case of fire-caused BLEVE failure, weakening of the wall due to overheating can be sufficient to cause vessel failure within the design pressure.
11. Davenport and Warwick found that while leakage failures can occur on very thick walled vessels, there is some evidence to suggest that catastrophic failure becomes an increasingly likely failure mode as the vessel wall thickness increases.
12. The lowest ratio for burst pressure to maximum allowable working pressure should be 2.35, which would be for a vessel constructed to British design codes BS 5500 or PD 5500.
13. A corroded vessel found to be acceptable to the API 579 Fitness-for-Service procedure, may have a remaining strength factor as low as 0.9, reducing the ratio of minimum burst pressure to maximum allowable working pressure (MAWP) to 2.1.

14. Cracks may have an effect on the burst strength if they have propagated since hydrotesting, or at pressures above the hydrotest pressure. Brittle failure of vessels would be more likely at cold temperatures, or under shock loading. Assessing a cracked vessel using a fitness for service assessment should result in a ratio of burst pressure to MAWP of at least 2.
15. The estimating level of overpressure at which damage may occur is more difficult, but it should not be below the hydrotest pressure (at least 30 % higher than MAWP).
16. The ratios of burst pressure to MAWP given assume that the condition of the vessel is known, and has been rerated if necessary. Undetected corrosion or cracks could significantly decrease the burst pressure.
17. These conclusions have been based on set safety factors for design stresses and calculations predicting actual burst pressure for intact and corroded cylindrical vessels. The assumption has been made that the findings will be applicable to other simple geometries where the internal pressure is the main source of stress, such as for heads and spherical vessels.
18. The complex nature of nozzle and flange designs means that this approach has not been possible for these components. For example, the bolting force for a flanged joint is dependent on a number of factors, such as flange diameter, gasket type and location and internal pressure. Therefore it would not be possible to predict an overpressure ratio likely to cause failure. However, from the lack of information on these components it should not be assumed that they would not fail during an overpressure event.

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10 APPENDIX A: SELECTED FAILURE FREQUENCIES TABLES PUBLISHED BY RIVM

Table 13 Scenarios for a pressurised storage tank aboveground

	Frequency (per annum)
1. Instantaneous release of entire contents	5×10^{-7}
2. Release of entire contents in 10 min. in a continuous and constant stream	5×10^{-7}
3. Continuous release of contents from a hole with an effective diameter of 10 mm	1×10^{-5}

Table 14 Parts included in the scenario for a pressurised storage tank aboveground

Included in the default scenarios	Not included in the default scenarios
- Welded stumps - Mounting plates - Instrumentation pipes - Pipe connections up to the first flange	- Transport pipelines from the (quick-closing) valve - Vapour return pipe - Pressure relief device - Pipeline system

3.9.3 Scenarios

The scenarios for reactor vessels and process vessels are set out in Table 31, the parts included in Table 32.

Table 31 Scenarios for reactor vessels and process vessels

	Frequency (per annum)
1. Instantaneous release of entire contents of the reactor vessel/processor vessel	5×10^{-6}
2. Release of entire contents in 10 min. in a continuous and constant stream	5×10^{-6}
3. Continuous release from a hole with an effective diameter of 10 mm	1×10^{-4}

Table 32 Parts included in the scenarios for a reactor vessel/process vessel

Included in the default scenarios	Not included in the default scenarios
- Instrumentation pipes - Pipe connections up to the first flange	- Pipelines - Pressure relief device

Points for consideration:

1. The composition of the contents of the (batch) reactor vessel / process vessel and the conditions in the reactor vessel/process vessel vary with time. If relevant a few representative situations and the associated time-adjusting frequency must therefore be calculated.

Tables taken from RIVM reference manual for risk assessment [1]

11 APPENDIX B: BIBLIOGRAPHY OF FAILURE FREQUENCY SOURCES GIVEN IN FRED

Title	Author	Date	Comments
A literature review of generic failure rates and comparison with the failure rates used in RISKAT. Confidential, not in the public domain.	R Hankin	Dec 91	6×10^{-6} per yr, catastrophic failure, average values for failure rate data.
Review of failure rate data used in risk assessment.	G Simpson	Sep 93	7.4×10^{-6}
Guidelines for process equipment reliability data.	American Institute of Chemical Engineers	1989	9.5×10^{-5}, catastrophic failure of pressure vessels page 205.
"Covo" report.	Rijnmond public authority	Nov 81	6×10^{-6}, catastrophic failures. Table IX.I.
Loss prevention in the process industries.	F P Lees	1980	1×10^{-5} per yr, catastrophic failure based on Canvey data. Page 1018.
CIMAH safety case. Confidential, not in the public domain.	W S Atkins	Jun 94	2×10^{-6} per yr, catastrophic failure.
CIMAH safety case support. Confidential, not in the public domain.	Technica (USA)	May 1989	6.5×10^{-6} per yr, catastrophic data (Smith and Warwick data).
A survey of defects in pressure vessels in the UK during the period 1962-1978 and its relevance to nuclear primary circuits.	Smith and Warwick	Dec 81	4.2×10^{-5} per vessel yr, catastrophic data (includes boilers).
Reliability Technology.	Green & Bourne	1972	Two values are given for pressure vessels: General – 3.0×10^{-8} per hr High standard – 0.3×10^{-6} per hr
The predicted BLEVE frequency of a selected 2000 m^3 butane sphere on a refinery site.	M Selway	August 1988	Determines BLEVE frequency of an LPG tank to be 9×10^{-7} per yr (p 24).
An initial prediction of the BLEVE frequency of a 100 te butane storage vessel.	K W Blything & A B Reeves	1988	Uses fault tree analysis (FTA) to determine BLEVE frequency of a butane tank to be 10^{-8} to 10^{-6} per vessel yr.
Failure rates – LPG tanks. Confidential, not in the public domain.		1994	9.4×10^{-7} per year, catastrophic failure.
A further survey of pressure vessel failures in the UK (1983 – 1988) – public domain version.	T J Davenport	1991	Value of 5.1×10^{-5} per yr derived for all pressure vessels. Also individual values derived for air receivers, steam receivers and boilers.
Proposed gas terminal. Confidential, not in the public domain.	Technica	Aug 1991	6.5×10^{-6}per yr, catastrophic failure.
CIMAH safety report for gas terminal. Confidential, not in the public domain.	Technica	Jun 94	6.5×10^{-6}per yr, rupture. 5, 25 and 100 mm diameter hole size failure rates also given.
Gas terminal study. SRD review of Cremer and Warner failure rates. Confidential, not in the public domain.	P L Holden	Sep 81	1×10^{-6} per yr, catastrophic, not including nozzle failures. Process vessels rise to 3 cpm.
QRA data. Confidential, not in the public domain.	Technica	May 89	6.5×10^{-6} per yr catastrophic failure rate, also contains failure rates for partial failures.

Title	Author	Date	Comments
Risk assessment. Confidential, not in the public domain.	A D Little	Sep 94	6 x 10^{-6} per yr catastrophic, 50 mm diameter hole size also given.
Safety report R2000 reactor rupture fault tree analysis. Confidential, not in the public domain.	Not given	1994	3.65 x 10^{-5} per yr, reactor bursts.
Safety report. Confidential, not in the public domain.	Technica	1994	Split into various causes.
Estimation of cold failure frequency of LPG tanks in Europe. Confidential, not in the public domain.	W Sooby & J M Tolchard	1994	A value of 2.7 x 10^{-8} per vessel yr is derived for the cold catastrophic failure of LPG pressure vessels.
Calculation of release frequencies. Confidential, not in the public domain.	WS Atkins	Jul 95	1.2 x 10^{-8} per yr for catastrophic rupture of pressure relief vessel (intermittent use only).
Chlorine safety report. Confidential, not in the public domain.	WS Atkins	Oct 95	5 x 10^{-6} per yr, for bulk storage tank.
Loss prevention in the process industries.	F P Lees	1986	General pressure vessel: 3 High standard: 0.3 (units of failures x10^{-6} per yr)
SR module. Confidential, not in the public domain.	Unknown	1978	1 x 10^{-6} per yr catastrophic failure.
Guidelines for the preparation and review of a report under the CIMAH regulations. Confidential, not in the public domain.	BP CIMAH Liaisons Group	May 93	Cold failure 6.5 x 10^{-6} per yr Hot failure (BLEVE) 26 x 10^{-6} per yr
Handbook of risk analysis. Confidential, not in the public domain.	Hydro	Not given	2 x 10^{-6}per yr, rupture.
Generic land use planning consultation zones - chlorine. Confidential, not in the public domain.	Not given	Oct 94	Catastrophic failure of chlorine storage vessel 2 x 10^{-8} per yr (lower bound).
Some data on the reliability of pressure equipment in the chemical plant environment.	D C Arulanantham & F P Lees	Oct 80	Various vessels; pressure vessels, boiler drums etc. (p 328).
Safety cases within the Control of Industrial Major Accident (CIMAH) Regulations 1984.	M L Ang & F P Lees	1989	Value given for chlorine pressure vessel.
The likelihood of accidental release events. Confidential, not in the public domain.	Rhône Poulenc Chemicals	Not dated	Various tank failures considered.
Quantified risk assessment. Confidential, not in the public domain.	AEA Technology	1996	Small leaks (0 – 25 mm): 2 x 10^{-4} per vessel yr Medium leaks (25 – 100 mm): 2 x 10^{-5} per vessel yr Large leaks (> 100 mm): 2 x 10^{-6} per vessel yr
A method for estimating the off-site risk from bulk storage of liquid oxygen (LOX). Confidential, not in the public domain.	BCGA/HSE/SRD Working Group	Not given	Estimates the failure rate of pressure vessels for LOX storage to be in the order of 10^{-5}per yr.
Risks associated with the storage of and use of chlorine at a water treatment plant (2nd draft). Confidential, not in the public domain.	SRD	Nov 81	This report derives a value for the failure rate for chlorine pressure vessels. Failure rates are thought to be over conservative.

Reproduced from FRED [5], Item 50, page 21 and 22 of 96

12 APPENDIX C: DNV LEAK FAILURE FREQUENCIES

Process Equipment Leak Frequencies		Rev.:	1
		Date:	26/9/2012
Equipment Type:	**Process Vessel**	Source:	HCRD 10/92 – 03/10

Definition:

Includes all types of pressure vessel (horizontal/vertical absorber, knock-out drum, reboiler, scrubber, stabiliser, separator and stabiliser), but not the HCRD category "other", which are mainly hydrocyclones. The scope includes the vessel itself and any nozzles or inspection openings, but excludes all attached valves, piping, flanges, instruments and fittings beyond the first flange. The first flange itself is also excluded.

Frequency Data:

Equipment Size	Category	Total	Full Pressure	Zero Pressure
0.5 in	1 - 3 mm	8.884E-04	7.859E-04	1.600E-04
	3 - 10 mm	5.946E-04	4.093E-04	1.393E-04
	10 - 50 mm	8.768E-04	3.448E-04	5.117E-04
	50 - 150 mm	0.000E+00	0.000E+00	0.000E+00
	> 150 mm	0.000E+00	0.000E+00	0.000E+00
	Total	2.360E-03	1.540E-03	8.110E-04
1 in	1 - 3 mm	8.884E-04	7.859E-04	1.600E-04
	3 - 10 mm	5.946E-04	4.093E-04	1.393E-04
	10 - 50 mm	8.768E-04	3.448E-04	5.117E-04
	50 - 150 mm	0.000E+00	0.000E+00	0.000E+00
	> 150 mm	0.000E+00	0.000E+00	0.000E+00
	Total	2.360E-03	1.540E-03	8.110E-04
2 in	1 - 3 mm	8.884E-04	7.859E-04	1.600E-04
	3 - 10 mm	5.946E-04	4.093E-04	1.393E-04
	10 - 50 mm	4.379E-04	2.236E-04	1.408E-04
	50 - 150 mm	4.389E-04	1.211E-04	3.709E-04
	> 150 mm	0.000E+00	0.000E+00	0.000E+00
	Total	2.360E-03	1.540E-03	8.110E-04
4 in	1 - 3 mm	8.884E-04	7.859E-04	1.600E-04
	3 - 10 mm	5.946E-04	4.093E-04	1.393E-04
	10 - 50 mm	4.379E-04	2.236E-04	1.408E-04
	50 - 150 mm	4.389E-04	1.211E-04	3.709E-04
	> 150 mm	0.000E+00	0.000E+00	0.000E+00
	Total	2.360E-03	1.540E-03	8.110E-04
6 in	1 - 3 mm	8.884E-04	7.859E-04	1.600E-04
	3 - 10 mm	5.946E-04	4.093E-04	1.393E-04
	10 - 50 mm	4.379E-04	2.236E-04	1.408E-04
	50 - 150 mm	1.652E-04	6.181E-05	7.316E-05
	> 150 mm	2.736E-04	5.930E-05	2.977E-04
	Total	2.360E-03	1.540E-03	8.110E-04

Reproduced from DNV "Failure Frequency Guidance" [6]

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The response of large pressure vessels in the process industries to excess pressure: literature review

Large pressure vessels are used in a wide range of chemical process industries. They contain significant amounts of energy. In the event of a vessel failure, this energy could be released. This gives significant potential for harm to people and other plant. As part of effective risk control, pressure vessels are usually designed in accordance with established codes that define their minimum strength.

HSE is aware that there may be pressure vessel systems in Great Britain where there is potential for exposure to pressures greater than those allowed under relevant current design codes. This can result from: errors during the original vessel design phase; a change in the design code due to improved understanding of the reaction chemistry; or poor change management of plant or process. These situations may lead to insufficient capacity in pressure relief systems for vessels and the potential to generate excess pressure leading to vessel failure.

This report describes a literature review on the impact of overpressure on the likelihood of vessel failure. It includes an assessment of impact of different design standards. It captures sources of relevant good practice and supplementary information that may help in the assessment of risk. The report covers:

- **Likelihood of overpressures and failures;**
- **Comparison of different construction standards;**
- **Predictions of the effects of overpressures;**
- **Calculations of burst pressures.**

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