Introduction to Safety Valves
Introduction

As soon as mankind was able to boil water to create steam, the necessity of the safety device became evident. As long as 2000 years ago, the Chinese were using cauldrons with hinged lids to allow (relatively) safer production of steam. At the beginning of the 14th century, chemists used conical plugs and later, compressed springs to act as safety devices on pressurised vessels.

Early in the 19th century, boiler explosions on ships and locomotives frequently resulted from faulty safety devices, which led to the development of the first safety relief valves.

In 1848, Charles Retchie invented the accumulation chamber, which increases the compression surface within the safety valve allowing it to open rapidly within a narrow overpressure margin. Today, most steam users are compelled by local health and safety regulations to ensure that their plant and processes incorporate safety devices and precautions, which ensure that dangerous conditions are prevented.

The primary function of a safety valve is therefore to protect life and property.

The principle type of device used to prevent overpressure in plant is the safety or safety relief valve. The safety valve operates by releasing a volume of fluid from within the plant when a predetermined maximum pressure is reached, thereby reducing the excess pressure in a safe manner. As the safety valve may be the only remaining device to prevent catastrophic failure under overpressure conditions, it is important that any such device is capable of operating at all times and under all possible conditions.

Safety valves should be installed wherever the maximum allowable working pressure (MAWP) of a system or pressure-containing vessel is likely to be exceeded. In steam systems, safety valves are typically used for boiler overpressure protection and other applications such as downstream of pressure reducing controls. Although their primary role is for safety, safety valves are also used in process operations to prevent product damage due to excess pressure. Pressure excess can be generated in a number of different situations, including:

- An imbalance of fluid flowrate caused by inadvertently closed or opened isolation valves on a process vessel.
- Failure of a cooling system, which allows vapour or fluid to expand.
- Compressed air or electrical power failure to control instrumentation.
- Transient pressure surges.
- Exposure to plant fires.
- Heat exchanger tube failure.
- Uncontrollable exothermic reactions in chemical plants.
- Ambient temperature changes.

The terms ‘safety valve’ and ‘safety relief valve’ are generic terms to describe many varieties of pressure relief devices that are designed to prevent excessive internal fluid pressure build-up. A wide range of different valves is available for many different applications and performance criteria. Furthermore, different designs are required to meet the numerous national standards that govern the use of safety valves.

A listing of the relevant national standards can be found at the end of this module.

In most national standards, specific definitions are given for the terms associated with safety and safety relief valves. There are several notable differences between the terminology used in the USA and Europe. One of the most important differences is that a valve referred to as a ‘safety valve’ in Europe is referred to as a ‘safety relief valve’ or ‘pressure relief valve’ in the USA. In addition, the term ‘safety valve’ in the USA generally refers specifically to the full-lift type of safety valve used in Europe.
The ASME/ANSI PTC25.3 standards applicable to the USA define the following generic terms:

- **Pressure relief valve** - A spring-loaded pressure relief valve which is designed to open to relieve excess pressure and to reclose and prevent the further flow of fluid after normal conditions have been restored. It is characterised by a rapid-opening ‘pop’ action or by opening in a manner generally proportional to the increase in pressure over the opening pressure. It may be used for either compressible or incompressible fluids, depending on design, adjustment, or application.

This is a general term, which includes safety valves, relief valves and safety relief valves.

- **Safety valve** - A pressure relief valve actuated by inlet static pressure and characterised by rapid opening or pop action.

Safety valves are primarily used with compressible gases and in particular for steam and air services. However, they can also be used for process type applications where they may be needed to protect the plant or to prevent spoilage of the product being processed.

- **Relief valve** - A pressure relief device actuated by inlet static pressure having a gradual lift generally proportional to the increase in pressure over opening pressure.

Relief valves are commonly used in liquid systems, especially for lower capacities and thermal expansion duty. They can also be used on pumped systems as pressure overspill devices.

- **Safety relief valve** - A pressure relief valve characterised by rapid opening or pop action, or by opening in proportion to the increase in pressure over the opening pressure, depending on the application, and which may be used either for liquid or compressible fluid.

In general, the safety relief valve will perform as a safety valve when used in a compressible gas system, but it will open in proportion to the overpressure when used in liquid systems, as would a relief valve.

The European standard EN ISO 4126-1 provides the following definition:

- **Safety valve** - A valve which automatically, without the assistance of any energy other than that of the fluid concerned, discharges a quantity of the fluid so as to prevent a predetermined safe pressure being exceeded, and which is designed to re-close and prevent further flow of fluid after normal pressure conditions of service have been restored.

Typical examples of safety valves used on steam systems are shown in Figure 9.1.1.

![Fig. 9.1.1 Typical safety valves](image-url)
Safety valve design

The basic spring loaded safety valve, referred to as ‘standard’ or ‘conventional’ is a simple, reliable self-acting device that provides overpressure protection.

The basic elements of the design consist of a right angle pattern valve body with the valve inlet connection, or nozzle, mounted on the pressure-containing system. The outlet connection may be screwed or flanged for connection to a piped discharge system. However, in some applications, such as compressed air systems, the safety valve will not have an outlet connection, and the fluid is vented directly to the atmosphere.

![Typical ASME valve](image1)

![Typical DIN valve](image2)

**Fig. 9.1.2 Typical safety valve designs**

The valve inlet (or approach channel) design can be either a full-nozzle or a semi-nozzle type. A full-nozzle design has the entire ‘wetted’ inlet tract formed from one piece. The approach channel is the only part of the safety valve that is exposed to the process fluid during normal operation, other than the disc, unless the valve is discharging.

**Full-nozzles** are usually incorporated in safety valves designed for process and high pressure applications, especially when the fluid is corrosive.

Conversely, the semi-nozzle design consists of a seating ring fitted into the body, the top of which forms the seat of the valve. The advantage of this arrangement is that the seat can easily be replaced, without replacing the whole inlet.

The disc is held against the nozzle seat (under normal operating conditions) by the spring, which is housed in an open or closed spring housing arrangement (or bonnet) mounted on top of the body. The discs used in rapid opening (pop type) safety valves are surrounded by a shroud, disc holder or huddling chamber which helps to produce the rapid opening characteristic.
The closing force on the disc is provided by a spring, typically made from carbon steel. The amount of compression on the spring is usually adjustable, using the spring adjuster, to alter the pressure at which the disc is lifted off its seat.

Standards that govern the design and use of safety valves generally only define the three dimensions that relate to the discharge capacity of the safety valve, namely the flow (or bore) area, the curtain area and the discharge (or orifice) area (see Figure 9.1.4).

1. Flow area - The minimum cross-sectional area between the inlet and the seat, at its narrowest point. The diameter of the flow area is represented by dimension ‘d’ in Figure 9.1.4.

\[
\text{Flow area} = \frac{\pi d^2}{4}
\]

2. Curtain area - The area of the cylindrical or conical discharge opening between the seating surfaces created by the lift of the disk above the seat. The diameter of the curtain area is represented by dimension ‘\(d_1\)’ in Figure 9.1.4.

\[
\text{Curtain area} = \pi d_1 L
\]

3. Discharge area - This is the lesser of the curtain and flow areas, which determines the flow through the valve.
Valves in which the flow area and not the curtain area determines the capacity are known as full lift valves. These valves will have a greater capacity than low lift or high lift valves. This issue will be discussed in greater depth in Module 9.2.

Although the principal elements of a conventional safety valve are similar, the design details can vary considerably. In general, the DIN style valves (commonly used throughout Europe) tend to use a simpler construction with a fixed skirt (or hood) arrangement whereas the ASME style valves have a more complex design that includes one or two adjustable blowdown rings. The position of these rings can be used to fine-tune the overpressure and blowdown values of the valve.

For a given orifice area, there may be a number of different inlet and outlet connection sizes, as well as body dimensions such as centreline to face dimensions. Furthermore, many competing products, particularly of European origin have differing dimensions and capacities for the same nominal size.

An exception to this situation is found with steel ASME specification valves, which invariably follow the recommendations of the API Recommended Practice 526, where centreline to face dimensions, and orifice sizes are listed. The orifice area series are referred to by a letter. It is common for valves with the same orifice letter to have several different sizes of inlet and outlet connection.

For example, 2" x J x 3" and 3" x J x 4" are both valves which have the same size (‘J’) orifice, but they have differing inlet and outlet sizes as shown before and after the orifice letter respectively. A 2" x J x 3" valve would have a 2" inlet, a ‘J’ size orifice and a 3" outlet.

### Basic operation of a safety valve

#### Lifting

When the inlet static pressure rises above the set pressure of the safety valve, the disc will begin to lift off its seat. However, as soon as the spring starts to compress, the spring force will increase; this means that the pressure would have to continue to rise before any further lift can occur, and for there to be any significant flow through the valve.

The additional pressure rise required before the safety valve will discharge at its rated capacity is called the overpressure. The allowable overpressure depends on the standards being followed and the particular application. For compressible fluids, this is normally between 3% and 10%, and for liquids between 10% and 25%.

In order to achieve full opening from this small overpressure, the disc arrangement has to be specially designed to provide rapid opening. This is usually done by placing a shroud, skirt or hood around the disc. The volume contained within this shroud is known as the control or huddling chamber.
As lift begins (Figure 9.1.6b), and fluid enters the chamber, a larger area of the shroud is exposed to the fluid pressure. Since the magnitude of the lifting force \((F)\) is proportional to the product of the pressure \((P)\) and the area exposed to the fluid \((A)\); \((F = P \times A)\), the opening force is increased. This incremental increase in opening force overcompensates for the increase in spring force, causing rapid opening. At the same time, the shroud reverses the direction of the flow, which provides a reaction force, further enhancing the lift.

These combined effects allow the valve to achieve its designed lift within a relatively small percentage overpressure. For compressible fluids, an additional contributory factor is the rapid expansion as the fluid volume increases from a higher to a lower pressure area. This plays a major role in ensuring that the valve opens fully within the small overpressure limit. For liquids, this effect is more proportional and subsequently, the overpressure is typically greater; 25% is common.

![Fig. 9.1.6 Operation of a conventional safety valve](image)

**Reseating**

Once normal operating conditions have been restored, the valve is required to close again, but since the larger area of the disc is still exposed to the fluid, the valve will not close until the pressure has dropped below the original set pressure. The difference between the set pressure and this reseating pressure is known as the ‘blowdown’, and it is usually specified as a percentage of the set pressure. For compressible fluids, the blowdown is usually less than 10%, and for liquids, it can be up to 20%.

![Fig. 9.1.7 Relationship between pressure and lift for a typical safety valve](image)
The design of the shroud must be such that it offers both rapid opening and relatively small blowdown, so that as soon as a potentially hazardous situation is reached, any overpressure is relieved, but excessive quantities of the fluid are prevented from being discharged. At the same time, it is necessary to ensure that the system pressure is reduced sufficiently to prevent immediate reopening.

The blowdown rings found on most ASME type safety valves are used to make fine adjustments to the overpressure and blowdown values of the valves (see Figure 9.1.8). The lower blowdown (nozzle) ring is a common feature on many valves where the tighter overpressure and blowdown requirements require a more sophisticated designed solution. The upper blowdown ring is usually factory set and essentially takes out the manufacturing tolerances which affect the geometry of the huddling chamber.

The lower blowdown ring is also factory set to achieve the appropriate code performance requirements but under certain circumstances can be altered. When the lower blowdown ring is adjusted to its top position the huddling chamber volume is such that the valve will pop rapidly, minimising the overpressure value but correspondingly requiring a greater blowdown before the valve re-seats. When the lower blowdown ring is adjusted to its lower position there is minimal restriction in the huddling chamber and a greater overpressure will be required before the valve is fully open but the blowdown value will be reduced.

Fig. 9.1.8 The blowdown rings on an ASME type safety valve
Approval authorities

For most countries, there are independent bodies who will examine the design and performance of a product range to confirm conformity with the relevant code or standard. This system of third party approval is very common for any safety related products and is often a customer requirement before purchase, or a requirement of their insurance company.

The actual requirements for approval will vary depending on the particular code or standard. In some cases, revalidation is necessary every few years, in others approval is indefinite as long as no significant design changes are made, in which case the approval authority must be notified, and re-approval sought. In the USA, the National Board of Boiler and Pressure Vessel Inspectors represents the US and Canadian government agencies empowered to assure adherence to code construction and repair of boilers and pressure vessels.

Some of the more commonly encountered bodies are listed in Table 9.1.1.

Table 9.1.1 Approval authorities

<table>
<thead>
<tr>
<th>Country</th>
<th>Abbreviation</th>
<th>Approval body</th>
</tr>
</thead>
<tbody>
<tr>
<td>Belgium</td>
<td></td>
<td>Bureau Veritas</td>
</tr>
<tr>
<td>Canada</td>
<td></td>
<td>Ministry of Labour Canada</td>
</tr>
<tr>
<td>France</td>
<td></td>
<td>CODAP</td>
</tr>
<tr>
<td></td>
<td></td>
<td>APAVE</td>
</tr>
<tr>
<td>Germany</td>
<td>TÜV</td>
<td>Association of Technical Supervision</td>
</tr>
<tr>
<td></td>
<td>DSRK</td>
<td>Deutsche Schiffs-Revision und Klassifikation</td>
</tr>
<tr>
<td>Italy</td>
<td>ISPESL RINA</td>
<td>Institution of Prevention and Security Italian Register of Shipping</td>
</tr>
<tr>
<td>Korea</td>
<td></td>
<td>Ministry of Power and Resources Korean Register of Shipping</td>
</tr>
<tr>
<td>Netherlands</td>
<td></td>
<td>Dienst voor het Stoomwezen</td>
</tr>
<tr>
<td>Norway</td>
<td>DNV</td>
<td>Det Norske Veritas</td>
</tr>
<tr>
<td>UK</td>
<td>SAFed</td>
<td>Safety Assessment Federation Type Approval Service (STAS) formerly Associated Offices Technical Committee AOTC and British Engine</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Lloyds Register of Shipping</td>
</tr>
<tr>
<td>United States</td>
<td>NB</td>
<td>National Board of Boiler and Pressure Vessel Inspectors</td>
</tr>
</tbody>
</table>
Codes and Standards

Standards relevant to safety valves vary quite considerably in format around the world, and many are sections within codes relevant to Boilers or Pressure Containing Vessels. Some will only outline performance requirements, tolerances and essential constructional detail, but give no guidance on dimensions, orifice sizes etc. Others will be related to installation and application. It is quite common within many markets to use several in conjunction with each other.

Table 9.1.2 Standards relating to safety valves

<table>
<thead>
<tr>
<th>Country</th>
<th>Standard No.</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Australia</td>
<td>SAA AS1271</td>
<td>Safety valves, other valves, liquid level gauges and other fittings for boilers and unfired pressure vessels</td>
</tr>
<tr>
<td>European Economic Area</td>
<td>EN ISO 4126</td>
<td>Safety devices for protection against excessive pressure</td>
</tr>
<tr>
<td></td>
<td>EN ISO 4126</td>
<td>EN ISO 4126 is a harmonised European Standard and has replaced many National Standards of which British Standard BS 6759 and the French Standard AFNOR NFE-E 29-411 to 416 and 421 are examples.</td>
</tr>
<tr>
<td>Germany</td>
<td>AD-Merkblatt A2</td>
<td>Pressure Vessel Equipment safety devices against excess pressure - safety valves</td>
</tr>
<tr>
<td></td>
<td>TRD 421</td>
<td>Technical Equipment for Steam Boilers Safeguards against excessive pressure - safety valves for steam boilers of groups I, III &amp; IV</td>
</tr>
<tr>
<td></td>
<td>TRD 721</td>
<td>Technical Equipment for Steam Boilers Safeguards against excessive pressure - safety valves for steam boilers of group II</td>
</tr>
<tr>
<td>Japan</td>
<td>JIS B 8210</td>
<td>Steam boilers and pressure vessels - spring loaded safety valves</td>
</tr>
<tr>
<td>Korea</td>
<td>KS B 6216</td>
<td>Spring loaded safety valves for steam boilers and pressure vessels</td>
</tr>
<tr>
<td>USA</td>
<td>ASME I</td>
<td>Boiler Applications</td>
</tr>
<tr>
<td></td>
<td>ASME III</td>
<td>Nuclear Applications</td>
</tr>
<tr>
<td></td>
<td>ASME VIII</td>
<td>Unfired Pressure Vessel Applications</td>
</tr>
<tr>
<td></td>
<td>ANSI/ASME</td>
<td>Safety and Relief Valves - performance test codes</td>
</tr>
<tr>
<td></td>
<td>PTC 25.3</td>
<td>Safety and Relief Valves - performance test codes</td>
</tr>
<tr>
<td></td>
<td>API RP 520</td>
<td>Sizing selection and installation of pressure-relieving devices in refineries</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Part 1 Design</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Part 2 Installation</td>
</tr>
<tr>
<td></td>
<td>API RP 521</td>
<td>Guide for pressure relieving and depressurising systems</td>
</tr>
<tr>
<td></td>
<td>API STD 526</td>
<td>Flanged steel pressure relief valves</td>
</tr>
<tr>
<td></td>
<td>API STD 527</td>
<td>Seat tightness of pressure relief valves</td>
</tr>
</tbody>
</table>

For steam boiler applications there are very specific requirements for safety valve performance, demanded by national standards and often, insurance companies. Approval by an independent authority is often necessary, such as British Engine, TÜV or Lloyd’s Register.

Safety valves used in Europe are also subject to the standards associated with the Pressure Equipment Directive (PED). Being classified as ‘Safety accessories’, safety valves are considered as ‘Category 4’ equipment, which require the most demanding level of assessment within the PED regime. This can usually be met by the manufacturer having an ISO 9000 quality system and the safety valve design and performance certified by an officially recognised approval authority referred to as a ‘Notified Body’.
Types of Safety Valves
Types of Safety Valves

There is a wide range of safety valves available to meet the many different applications and performance criteria demanded by different industries. Furthermore, national standards define many varying types of safety valve.

The ASME standard I and ASME standard VIII for boiler and pressure vessel applications and the ASME/ANSI PTC 25.3 standard for safety valves and relief valves provide the following definition. These standards set performance characteristics as well as defining the different types of safety valves that are used:

- **ASME I valve** - A safety relief valve conforming to the requirements of Section I of the ASME pressure vessel code for boiler applications which will open within 3% overpressure and close within 4%. It will usually feature two blowdown rings, and is identified by a National Board ‘V’ stamp.

- **ASME VIII valve** - A safety relief valve conforming to the requirements of Section VIII of the ASME pressure vessel code for pressure vessel applications which will open within 10% overpressure and close within 7%. Identified by a National Board ‘UV’ stamp.

- **Low lift safety valve** - The actual position of the disc determines the discharge area of the valve.

- **Full lift safety valve** - The discharge area is not determined by the position of the disc.

- **Full bore safety valve** - A safety valve having no protrusions in the bore, and wherein the valve lifts to an extent sufficient for the minimum area at any section, at or below the seat, to become the controlling orifice.

- **Conventional safety relief valve** - The spring housing is vented to the discharge side, hence operational characteristics are directly affected by changes in the backpressure to the valve.

- **Balanced safety relief valve** - A balanced valve incorporates a means of minimising the effect of backpressure on the operational characteristics of the valve.

- **Pilot operated pressure relief valve** - The major relieving device is combined with, and is controlled by, a self-actuated auxiliary pressure relief device.

- **Power-actuated safety relief valve** - A pressure relief valve in which the major pressure relieving device is combined with, and controlled by, a device requiring an external source of energy.

The following types of safety valve are defined in the DIN 3320 standard, which relates to safety valves sold in Germany and other parts of Europe:

- **Standard safety valve** - A valve which, following opening, reaches the degree of lift necessary for the mass flowrate to be discharged within a pressure rise of not more than 10%. (The valve is characterised by a pop type action and is sometimes known as high lift).

- **Full lift (Vollhub) safety valve** - A safety valve which, after commencement of lift, opens rapidly within a 5% pressure rise up to the full lift as limited by the design. The amount of lift up to the rapid opening (proportional range) shall not be more than 20%.

- **Direct loaded safety valve** - A safety valve in which the opening force underneath the valve disc is opposed by a closing force such as a spring or a weight.

- **Proportional safety valve** - A safety valve which opens more or less steadily in relation to the increase in pressure. Sudden opening within a 10% lift range will not occur without pressure increase. Following opening within a pressure of not more than 10%, these safety valves achieve the lift necessary for the mass flow to be discharged.

- **Diaphragm safety valve** - A direct loaded safety valve wherein linear moving and rotating elements are protected against the effects of the fluid by a diaphragm.

- **Bellows safety valve** - A direct loaded safety valve wherein sliding and (partially or fully) rotating elements and springs are protected against the effects of the fluids by a bellows. The bellows may be of such a design that it compensates for influences of backpressure.

- **Controlled safety valve** - Consists of a main valve and a control device. It also includes direct acting safety valves with supplementary loading in which, until the set pressure is reached, an additional force increases the closing force.
EN ISO 4126 lists the following definitions of types of safety valve:

- **Safety valve** - A safety valve which automatically, without the assistance of any energy other than that of the fluid concerned, discharges a quantity of the fluid so as to prevent a predetermined safe pressure being exceeded, and which is designed to re-close and prevent further flow of fluid after normal pressure conditions of service have been restored. Note; the valve can be characterised either by pop action (rapid opening) or by opening in proportion (not necessarily linear) to the increase in pressure over the set pressure.

- **Direct loaded safety valve** - A safety valve in which the loading due to the fluid pressure underneath the valve disc is opposed only by a direct mechanical loading device such as a weight, lever and weight, or a spring.

- **Assisted safety valve** - A safety valve which by means of a powered assistance mechanism, may additionally be lifted at a pressure lower than the set pressure and will, even in the event of a failure of the assistance mechanism, comply with all the requirements for safety valves given in the standard.

- **Supplementary loaded safety valve** - A safety valve that has, until the pressure at the inlet to the safety valve reaches the set pressure, an additional force, which increases the sealing force. Note; this additional force (supplementary load), which may be provided by means of an extraneous power source, is reliably released when the pressure at the inlet of the safety valve reaches the set pressure. The amount of supplementary loading is so arranged that if such supplementary loading is not released, the safety valve will attain its certified discharge capacity at a pressure not greater than 1.1 times the maximum allowable pressure of the equipment to be protected.

- **Pilot operated safety valve** - A safety valve, the operation of which is initiated and controlled by the fluid discharged from a pilot valve, which is itself, a direct loaded safety valve subject to the requirement of the standard.

The following table summarises the performance of different types of safety valve set out by the various standards.

**Table 9.2.1 Safety valve performance summary**

<table>
<thead>
<tr>
<th>Standard</th>
<th>Overpressure</th>
<th>Fluid</th>
<th>Blowdown</th>
</tr>
</thead>
<tbody>
<tr>
<td>A.D. Merkblatt A2</td>
<td>Standard 10% full lift 5%</td>
<td>Steam</td>
<td>10%</td>
</tr>
<tr>
<td></td>
<td>Standard 10% full lift 5%</td>
<td>Air or gas</td>
<td>10%</td>
</tr>
<tr>
<td></td>
<td>10%</td>
<td>Liquid</td>
<td>20%</td>
</tr>
<tr>
<td>ASME I</td>
<td>3%</td>
<td>Steam</td>
<td>2-6%</td>
</tr>
<tr>
<td></td>
<td>10%</td>
<td>Steam</td>
<td>7%</td>
</tr>
<tr>
<td></td>
<td>10%</td>
<td>Air or gas</td>
<td>7%</td>
</tr>
<tr>
<td></td>
<td>10% (see Note 2 below)</td>
<td>Liquid</td>
<td></td>
</tr>
<tr>
<td>EN ISO 4126</td>
<td>Value stated by manufacturer but not exceeding 10% of set pressure or 0.1 bar whichever is greater.</td>
<td>Compressible</td>
<td>Minimum 2% Maximum 15% or 0.3 bar whichever is greater.</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Incompressible</td>
<td>Minimum 2.5% Maximum 20% or 0.6 bar whichever is greater.</td>
</tr>
</tbody>
</table>

Notes:
1. ASME blowdown values shown are for valves with adjustable blowdown.
2. 25% is often used for non-certified sizing calculations and 20% can be used for fire protection of storage vessels.
Conventional safety valves

The common characteristic shared between the definitions of conventional safety valves in the different standards, is that their operational characteristics are affected by any backpressure in the discharge system. It is important to note that the total backpressure is generated from two components; superimposed backpressure and the built-up backpressure:

- **Superimposed backpressure** - The static pressure that exists on the outlet side of a closed valve.
- **Built-up backpressure** - The additional pressure generated on the outlet side when the valve is discharging.

Subsequently, in a conventional safety valve, only the superimposed backpressure will affect the opening characteristic and set value, but the combined backpressure will alter the blowdown characteristic and re-seat value.

The ASME/ANSI standard makes the further classification that conventional valves have a spring housing that is vented to the discharge side of the valve. If the spring housing is vented to the atmosphere, any superimposed backpressure will still affect the operational characteristics. This can be seen from Figure 9.2.1, which shows schematic diagrams of valves whose spring housings are vented to the discharge side of the valve and to the atmosphere.

By considering the forces acting on the disc (with area $A_D$), it can be seen that the required opening force (equivalent to the product of inlet pressure ($P_V$) and the nozzle area ($A_N$)) is the sum of the spring force ($F_S$) and the force due to the backpressure ($P_B$) acting on the top and bottom of the disc. In the case of a spring housing vented to the discharge side of the valve (an ASME conventional safety relief valve, see Figure 9.2.1 (a)), the required opening force is:

$$P_V A_N = F_S + P_B A_D - P_B (A_D - A_N)$$

which simplifies to Equation 9.2.1

<table>
<thead>
<tr>
<th>Equation 9.2.1</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_V A_N = F_S + P_B A_N$</td>
</tr>
</tbody>
</table>

Where:
- $P_V$ = Fluid inlet pressure
- $A_N$ = Nozzle area
- $F_S$ = Spring force
- $P_B$ = Backpressure
Therefore, any superimposed backpressure will tend to increase the closing force and the inlet pressure required to lift the disc is greater.

In the case of a valve whose spring housing is vented to the atmosphere (Figure 9.2.1b), the required opening force is:

\[
P_v A_N = F_s - P_b (A_d - A_n)
\]

Equation 9.2.2

Where:
- \(P_v\) = Fluid inlet pressure
- \(A_N\) = Nozzle area
- \(F_s\) = Spring force
- \(P_b\) = Backpressure
- \(A_d\) = Disc area

Thus, the superimposed backpressure acts with the vessel pressure to overcome the spring force, and the opening pressure will be less than expected.

In both cases, if a significant superimposed backpressure exists, its effects on the set pressure need to be considered when designing a safety valve system.

Once the valve starts to open, the effects of built-up backpressure also have to be taken into account. For a conventional safety valve with the spring housing vented to the discharge side of the valve, see Figure 9.2.1 (a), the effect of built-up backpressure can be determined by considering Equation 9.2.1 and by noting that once the valve starts to open, the inlet pressure is the sum of the set pressure, \(P_s\), and the overpressure, \(P_o\).

\[
(P_s + P_o) A_N = F_s + P_b A_N \quad \text{which simplifies to Equation 9.2.3}
\]

Equation 9.2.3

Where:
- \(P_s\) = Set pressure of safety valve
- \(A_N\) = Nozzle area
- \(F_s\) = Spring force
- \(P_b\) = Backpressure
- \(P_o\) = Overpressure

Therefore, if the backpressure is greater than the overpressure, the valve will tend to close, reducing the flow. This can lead to instability within the system and can result in flutter or chatter of the valve.

In general, if conventional safety valves are used in applications, where there is an excessive built-up backpressure, they will not perform as expected. According to the API 520 Recommended Practice Guidelines:

- A conventional pressure relief valve should typically not be used when the built-up backpressure is greater than 10% of the set pressure at 10% overpressure. A higher maximum allowable built-up backpressure may be used for overpressure greater than 10%.

The European Standard EN ISO 4126, however, states that the built-up backpressure should be limited to 10% of the set pressure when the valve is discharging at the certified capacity.

For the majority of steam applications, the backpressure can be maintained within these limits by carefully sizing any discharge pipes. This will be discussed in Module 9.4. If, however, it is not feasible to reduce the backpressure, then it may be necessary to use a balanced safety valve.
Balanced safety valves

Balanced safety valves are those that incorporate a means of eliminating the effects of backpressure. There are two basic designs that can be used to achieve this:

**Piston type balanced safety valve.**

Although there are several variations of the piston valve, they generally consist of a piston type disc whose movement is constrained by a vented guide. The area of the top face of the piston, \( A_P \), and the nozzle seat area, \( A_N \), are designed to be equal. This means that the effective area of both the top and bottom surfaces of the disc exposed to the backpressure are equal, and therefore any additional forces are balanced. In addition, the spring bonnet is vented such that the top face of the piston is subjected to atmospheric pressure, as shown in Figure 9.2.2.

![Fig. 9.2.2 Schematic diagram of a piston type balanced safety valve](image)

By considering the forces acting on the piston, it is evident that this type of valve is no longer affected by any backpressure:

\[
P_V A_N = F_S + P_B A_D - P_B (A_D - A_N)
\]

Where:
- \( P_V \) = Fluid inlet pressure
- \( A_N \) = Nozzle area
- \( F_S \) = Spring force
- \( P_B \) = Backpressure
- \( A_D \) = Disc area
- \( A_P \) = Piston area

Since \( A_P \) equals \( A_N \), the last two terms of the equation are equal in magnitude and cancel out of the equation. Therefore, this simplifies to Equation 9.2.4.

**Equation 9.2.4**

\[
P_V A_N = F_S
\]

Where:
- \( P_V \) = Fluid inlet pressure
- \( A_N \) = Nozzle area
- \( F_S \) = Spring force
Bellows type balanced safety valve.

A bellows with an effective area ($A_B$) equivalent to the nozzle seat area ($A_N$) is attached to the upper surface of the disc and to the spindle guide.

The bellows arrangement prevents backpressure acting on the upper side of the disc within the area of the bellows. The disc area extending beyond the bellows and the opposing disc area are equal, and so the forces acting on the disc are balanced, and the backpressure has little effect on the valve opening pressure.

The bellows vent allows air to flow freely in and out of the bellows as they expand or contract. Bellows failure is an important concern when using a bellows balanced safety valve, as this may affect the set pressure and capacity of the valve. It is important, therefore, that there is some mechanism for detecting any uncharacteristic fluid flow through the bellows vents. In addition, some bellows balanced safety valves include an auxiliary piston that is used to overcome the effects of backpressure in the case of bellows failure. This type of safety valve is usually only used on critical applications in the oil and petrochemical industries.

In addition to reducing the effects of backpressure, the bellows also serve to isolate the spindle guide and the spring from the process fluid, this is important when the fluid is corrosive.

Since balanced pressure relief valves are typically more expensive than their unbalanced counterparts, they are commonly only used where high pressure manifolds are unavoidable, or in critical applications where a very precise set pressure or blowdown is required.

![Schematic diagram of the bellows balanced safety valve](image)

**Fig. 9.2.3** Schematic diagram of the bellows balanced safety valve
Pilot operated safety valve

This type of safety valve uses the flowing medium itself, through a pilot valve, to apply the closing force on the safety valve disc. The pilot valve is itself a small safety valve.

There are two basic types of pilot operated safety valve, namely, the diaphragm and piston type.

The diaphragm type is typically only available for low pressure applications and it produces a proportional type action, characteristic of relief valves used in liquid systems. They are therefore of little use in steam systems, consequently, they will not be considered in this text.

The piston type valve consists of a main valve, which uses a piston shaped closing device (or obturator), and an external pilot valve. Figure 9.2.4 shows a diagram of a typical piston type, pilot operated safety valve.

The piston and seating arrangement incorporated in the main valve is designed so that the bottom area of the piston, exposed to the inlet fluid, is less than the area of the top of the piston. As both ends of the piston are exposed to the fluid at the same pressure, this means that under normal system operating conditions, the closing force, resulting from the larger top area, is greater than the inlet force. The resultant downward force therefore holds the piston firmly on its seat.

If the inlet pressure were to rise, the net closing force on the piston also increases, ensuring that a tight shut-off is continually maintained. However, when the inlet pressure reaches the set pressure, the pilot valve will pop open to release the fluid pressure above the piston. With much less fluid pressure acting on the upper surface of the piston, the inlet pressure generates a net upwards force and the piston will leave its seat. This causes the main valve to pop open, allowing the process fluid to be discharged.
When the inlet pressure has been sufficiently reduced, the pilot valve will reclose, preventing the further release of fluid from the top of the piston, thereby re-establishing the net downward force, and causing the piston to reseat.

Pilot operated safety valves offer good overpressure and blowdown performance (a blowdown of 2% is attainable). For this reason, they are used where a narrow margin is required between the set pressure and the system operating pressure. Pilot operated valves are also available in much larger sizes, making them the preferred type of safety valve for larger capacities.

One of the main concerns with pilot operated safety valves is that the small bore, pilot connecting pipes are susceptible to blockage by foreign matter, or due to the collection of condensate in these pipes. This can lead to the failure of the valve, either in the open or closed position, depending on where the blockage occurs.

**Full lift, high lift and low lift safety valves**

The terms full lift, high lift and low lift refer to the amount of travel the disc undergoes as it moves from its closed position to the position required to produce the certified discharge capacity, and how this affects the discharge capacity of the valve.

A full lift safety valve is one in which the disc lifts sufficiently, so that the curtain area no longer influences the discharge area. The discharge area, and therefore the capacity of the valve are subsequently determined by the bore area. This occurs when the disc lifts a distance of at least a quarter of the bore diameter. A full lift conventional safety valve is often the best choice for general steam applications.

The disc of a high lift safety valve lifts a distance of at least \( \frac{1}{12} \)th of the bore diameter. This means that the curtain area, and ultimately the position of the disc, determines the discharge area. The discharge capacities of high lift valves tend to be significantly lower than those of full lift valves, and for a given discharge capacity, it is usually possible to select a full lift valve that has a nominal size several times smaller than a corresponding high lift valve, which usually incurs cost advantages. Furthermore, high lift valves tend to be used on compressible fluids where their action is more proportional.

In low lift valves, the disc only lifts a distance of \( \frac{1}{24} \)th of the bore diameter. The discharge area is determined entirely by the position of the disc, and since the disc only lifts a small amount, the capacities tend to be much lower than those of full or high lift valves.
Materials of construction

Except when safety valves are discharging, the only parts that are wetted by the process fluid are the inlet tract (nozzle) and the disc. Since safety valves operate infrequently under normal conditions, all other components can be manufactured from standard materials for most applications. There are however several exceptions, in which case, special materials have to be used, these include:

- Cryogenic applications.
- Corrosive fluids.
- Where contamination of discharged fluid is not permitted.
- When the valve discharges into a manifold that contains corrosive media discharged by another valve.

The principal pressure-containing components of safety valves are normally constructed from one of the following materials:

- **Bronze** - Commonly used for small screwed valves for general duty on steam, air and hot water applications (up to 15 bar).
- **Cast iron** - Used extensively for ASME type valves. Its use is typically limited to 17 bar g.
- **SG iron** - Commonly used in European valves and to replace cast iron in higher pressure valves (up to 25 bar g).
- **Cast steel** - Commonly used on higher pressure valves (up to 40 bar g). Process type valves are usually made from a cast steel body with an austenitic full nozzle type construction.
- **Austenitic stainless steel** - Used in food, pharmaceutical or clean steam applications.

For extremely high pressure applications, pressure containing components may be forged or machined from solid.

For all safety valves, it is important that moving parts, particularly the spindle and guides are made from materials that will not easily degrade or corrode. As seats and discs are constantly in contact with the process fluid, they must be able to resist the effects of erosion and corrosion. For process applications, austenitic stainless steel is commonly used for seats and discs; sometimes they are ‘stellite faced’ for increased durability. For extremely corrosive fluids, nozzles, discs and seats are made from special alloys such as ‘monel’ or ‘hastelloy’.

The spring is a critical element of the safety valve and must provide reliable performance within the required parameters. Standard safety valves will typically use carbon steel for moderate temperatures. Tungsten steel is used for higher temperature, non-corrosive applications, and stainless steel is used for corrosive or clean steam duty. For sour gas and high temperature applications, often special materials such as monel, hastelloy and ‘inconel’ are used.

Safety valve options and accessories

Due to the wide range of applications in which safety valves are used, there are a number of different options available:

**Seating material**

A key option is the type of seating material used. Metal-to-metal seats, commonly made from stainless steel, are normally used for high temperature applications such as steam. Alternatively, resilient discs can be fixed to either or both of the seating surfaces where tighter shut-off is required, typically for gas or liquid applications. These inserts can be made from a number of different materials, but Viton, nitrile or EPDM are the most common. Soft seal inserts are not generally recommended for steam use.
<table>
<thead>
<tr>
<th>Seal material</th>
<th>Applications</th>
</tr>
</thead>
<tbody>
<tr>
<td>EPDM</td>
<td>Water</td>
</tr>
<tr>
<td>Viton</td>
<td>High temperature gas applications</td>
</tr>
<tr>
<td>Nitrile</td>
<td>Air and oil applications</td>
</tr>
<tr>
<td>Stainless steel</td>
<td>Standard material, best for steam</td>
</tr>
<tr>
<td>Stellite</td>
<td>Wear resistant for tough applications</td>
</tr>
</tbody>
</table>

**Levers**

Standard safety valves are generally fitted with an easing lever, which enables the valve to be lifted manually in order to ensure that it is operational at pressures in excess of 75% of set pressure. This is usually done as part of routine safety checks, or during maintenance to prevent seizing. The fitting of a lever is usually a requirement of national standards and insurance companies for steam and hot water applications. For example, the ASME Boiler and Pressure Vessel Code states that pressure relief valves must be fitted with a lever if they are to be used on air, water over 60°C, and steam.

A standard or open lever is the simplest type of lever available. It is typically used on applications where a small amount of leakage of the fluid to the atmosphere is acceptable, such as on steam and air systems, (see Figure 9.2.5 (a)).

Where it is not acceptable for the media to escape, a packed lever must be used. This uses a packed gland seal to ensure that the fluid is contained within the cap, (see Figure 9.2.5 (b)).

For service where a lever is not required, a cap can be used to simply protect the adjustment screw. If used in conjunction with a gasket, it can be used to prevent emissions to the atmosphere, (see Figure 9.2.6).

A test gag (Figure 9.2.7) may be used to prevent the valve from opening at the set pressure during hydraulic testing when commissioning a system. Once tested, the gag screw is removed and replaced with a short blanking plug before the valve is placed in service.
Open and closed bonnets

Unless bellows or diaphragm sealing is used, process fluid will enter the spring housing (or bonnet). The amount of fluid depends on the particular design of safety valve. If emission of this fluid into the atmosphere is acceptable, the spring housing may be vented to the atmosphere – an open bonnet. This is usually advantageous when the safety valve is used on high temperature fluids or for boiler applications as, otherwise, high temperatures can relax the spring, altering the set pressure of the valve. However, using an open bonnet exposes the valve spring and internals to environmental conditions, which can lead to damage and corrosion of the spring.

When the fluid must be completely contained by the safety valve (and the discharge system), it is necessary to use a closed bonnet, which is not vented to the atmosphere. This type of spring enclosure is almost universally used for small screwed valves and, it is becoming increasingly common on many valve ranges since, particularly on steam, discharge of the fluid could be hazardous to personnel.

![Open and closed bonnets](image)

**Fig. 9.2.8 Spring housings**

Bellows and diaphragm sealing

Some safety valves, most commonly those used for water applications, incorporate a flexible diaphragm or bellows to isolate the safety valve spring and upper chamber from the process fluid, (see Figure 9.2.9).

![Bellows and diaphragm sealing](image)

**Fig. 9.2.9 A diaphragm sealed safety valve**

An elastomer bellows or diaphragm is commonly used in hot water or heating applications, whereas a stainless steel one would be used on process applications employing hazardous fluids.
Safety Valves Selection
Safety Valve Selection

As there is such a wide range of safety valves, there is no difficulty in selecting a safety valve that meets the specific requirements of a given application. Once a suitable type has been selected, it is imperative that the correct relieving pressure and discharge capacity are established, and a suitably sized valve and set pressure is specified.

The selection of a specific type of safety valve is governed by several factors:

• **Cost** - This is the most obvious consideration when selecting a safety valve for a non-critical application. When making cost comparisons, it is imperative to consider the capacity of the valve as well as the nominal size. As mentioned in the previous module, there can be large variations between models with the same inlet connection but with varying lift characteristics.

• **Type of disposal system** - Valves with an open bonnet can be used on steam, air or non-toxic gas, if discharge to the atmosphere, other than through the discharge system, is acceptable. A lifting lever is often specified in these applications.

For gas or liquid applications, where escape to the atmosphere is not permitted, a closed bonnet must be specified. In such applications, it is also necessary to use either a closed/gas tight cap or packed lever.

For applications with a significant superimposed backpressure (common in manifolds, typically seen in the process industry) a balancing bellows or piston construction is required.

• **Valve construction** - A semi-nozzle type construction should be used for non-toxic, non-corrosive type media at moderate pressures, whereas valves with the full nozzle type construction are typically used in the process industry for corrosive media or for extremely high pressures. For corrosive fluids or high temperatures, special materials of construction may also be required.

• **Operating characteristics** - Performance requirements vary according to application and the valve must be selected accordingly. For steam boilers, a small overpressure is required, usually 3% or 5%. For most other applications, 10% overpressure is required, but according to API 520, for special applications such as fire protection, larger valves with overpressures of 20% are allowed. For liquids, overpressures of 10% or 25% are common, and blowdown values tend to be up to 20%.

• **Approval** - For many valve applications, the end user will state the required code or standard for the construction and performance of the valve. This is usually accompanied by a requirement for approval by an independent authority, to guarantee conformance with the required standard.
Setting and sealing

In order to establish the set pressure correctly, the following terms require careful consideration:

- **Normal working pressure (NWP)** - The operating pressure of the system under full-load conditions.

- **Maximum allowable working pressure (MAWP)** - Sometimes called the safe working pressure (SWP) or design pressure of the system. This is the maximum pressure existing at normal operating conditions (relative to the maximum operating temperature) of the system.

- **Maximum allowable accumulation pressure (MAAP)** - The maximum pressure the system is allowed to reach in accordance with the specification of the design standards of the system. The MAAP is often expressed as a percentage of the MAWP.

For steam using apparatus, the MAAP will often be 10% higher than the MAWP, but this is not always the case. If the MAWP is not readily available, the authority responsible for insuring the apparatus should be contacted. If the MAAP cannot be established, it must not be considered to be higher than the MAWP.

- **Set Pressure (P)** - The pressure at which the safety valve starts to lift.

- **Relieving pressure (PR)** - This is the pressure at which the full capacity of the safety valve is achieved. It is the sum of the set pressure (PS) and the overpressure (PO).

- **Overpressure (PO)** - The overpressure is the percentage of the set pressure at which the safety valve is designed to operate.

There are two fundamental constraints, which must be taken into account when establishing a safety valve set pressure:

1. The set pressure must be low enough to ensure that the relieving pressure never exceeds the maximum allowable accumulation pressure (MAAP) of the system.

2. The set pressure must be high enough to ensure that there is sufficient margin above the normal working pressure (NWP) to allow the safety valve to close. However, the set pressure must never be greater than the maximum allowable working pressure (MAWP).

In order to meet the first constraint, it is necessary to consider the relative magnitudes of the percentage overpressure and the percentage MAAP (expressed as a percentage of the MAWP). There are two possible cases:

- **The percentage overpressure of the safety valve is less than or equal to the percentage MAAP of the system** - This means that the set pressure can be made to equal the MAWP, as the relieving pressure will always be less than the actual MAAP.

  For example, if the safety valve overpressure was 5%, and the MAAP was 10% of the MAWP, the set pressure would be chosen to equal the MAWP. In this case, the relieving pressure (equal to the set pressure + 5% overpressure) would be less than the MAAP, which is acceptable.

  Note that if the percentage MAAP were higher than the percentage overpressure, the set pressure will still be made to equal the MAWP, as increasing it above the MAWP would violate the second constraint.

- **The percentage overpressure of the safety valve is greater than the percentage MAAP of the system** - In this case, making the set pressure equal to the MAWP will mean that the relieving pressure would be greater than the MAAP, so the set pressure must be lower than the MAWP.

  For example, if the safety valve overpressure was 25% and the percentage MAAP was only 10%, making the set pressure equal to the MAWP means that the relieving pressure would be 15% greater than the MAAP. In this instance, the correct set pressure should be 15% below the MAWP.
The following table summarises the determination of the set point based on the first constraint.

### Table 9.3.1
**Determination of the set pressure using safety valve overpressure and apparatus MAAP**

<table>
<thead>
<tr>
<th>Apparatus</th>
<th>Safety valve overpressure</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>5%</td>
</tr>
<tr>
<td>MAAP</td>
<td></td>
</tr>
<tr>
<td>20%</td>
<td>MAWP</td>
</tr>
<tr>
<td>15%</td>
<td>MAWP</td>
</tr>
<tr>
<td>10%</td>
<td>MAWP</td>
</tr>
<tr>
<td>5%</td>
<td>MAWP</td>
</tr>
</tbody>
</table>

Unless operational considerations dictate otherwise, in order to meet the second constraint, the safety valve set pressure should always be somewhat above the normal working pressure with a margin allowed for the blowdown. A safety valve set just above the normal working pressure can lead to a poor shut-off after any discharge.

When the system operating pressure and safety valve set pressure have to be as close as possible to one another, a 0.1 bar minimum margin between reseat pressure and normal operating pressure is recommended to ensure a tight shut-off. This is called the ‘shut-off margin’. In this case, it is important to take into account any variations in the system operating pressure before adding the 0.1 bar margin. Such variations can occur where a safety valve is installed after pressure reducing valves (PRVs) and other control valves, with relatively large proportional bands.

In practically all control systems, there is a certain amount of proportional offset associated with the proportional band (see Block 5, Control Theory, for more information regarding proportional offset). If a self-acting PRV is set under full-load conditions, the control pressure at no-load conditions can be significantly greater than its set pressure. Conversely, if the valve is set under no-load conditions, the full-load pressure will be less than its set pressure.

For example, consider a pilot operated PRV with a maximum proportional band of only 0.2 bar. With a control pressure of 5.0 bar set under full-load conditions, it would give 5.2 bar under no-load conditions. Alternatively, if the control pressure of 5.0 bar is set under no-load conditions, the same valve would exhibit a control pressure of 4.8 bar under full-load conditions.

When determining the set pressure of the safety valve, if the PRV control pressure is set under no-load conditions, then the proportional offset does not have to be taken into account. However, if the PRV control pressure is set under full-load conditions, it is necessary to consider the increase in downstream pressure as a result of the proportional offset of the PRV (see Example 9.3.1).

The amount of pressure control offset depends on the type of control valve and the pressure controller being used. It is therefore important to determine the proportional band of the upstream control valve as well as how this valve was commissioned.
Example 9.3.1
A safety valve, which is to be installed after a PRV, is required to be set as close as possible to the PRV working pressure. Given the parameters below, determine the most suitable safety valve set pressure:

PRV set pressure: 6.0 bar (set under full-load conditions)
PRV proportional band: 0.3 bar operating above the PRV working pressure
Safety valve blowdown: 10%

Answer:
Since it is necessary to ensure that the safety valve set pressure is as close to the PRV working pressure as possible, the safety valve is chosen so that its blowdown pressure is greater than the PRV working pressure (taking into account the proportional offset), and a 0.1 bar shut-off margin.

Firstly, the effect of the PRV proportional offset needs to be considered as the PRV is being set under load conditions; the normal maximum working pressure that will be encountered is:

6.0 bar + 0.3 bar = 6.3 bar (NWP)

By adding the 0.1 bar shut-off margin, the safety valve set pressure has to be 10% greater than 6.4 bar. For this example, this means that the safety valve’s set pressure has to be:

$$\left(6.3 + 0.1\right) \times \frac{100}{100 - 10} = 7.11 \text{ bar}$$

The set pressure would therefore be chosen as 7.11 bar, provided that this does not exceed the MAWP of the protected system.

Note that if the PRV were set at 6.0 bar under no-load conditions, and with a safety valve 10% blowdown, the safety valve set pressure would be:

$$\left(6.0 + 0.1\right) \times \frac{100}{100 - 10} = 6.78 \text{ bar}$$

Effects of backpressure on set pressure
For a conventional safety valve subject to a constant superimposed backpressure, the set pressure is effectively reduced by an amount equal to the backpressure. In order to compensate for this, the required set pressure must be increased by an amount equal to the backpressure. The cold differential set pressure (the pressure set on the test stand) will therefore be:

\[\text{Equation 9.3.1} \quad \text{CDSP} = \text{RISP} - \text{CBP}\]

Where:
CDSP = Cold differential set pressure
RISP = Required installed set pressure
CBP = Constant backpressure

For variable superimposed backpressure, the effective set pressure could change as the backpressure varies, and a conventional valve could not be used if the variation were more than 10% to 15% of the set pressure. Instead, a balanced valve would have to be used.

The pressure level relationships for pressure relief valves as shown in the API Recommended Practice 520 is illustrated in Figure 9.3.1.
Fig. 9.3.1 Pressure level relationships for pressure relief valves (from API 520)
**Setting a safety valve**

For most types of safety valve, air or gas setting is permissible. A specially constructed test stand is usually employed, allowing easy and quick mounting of the safety valve, for adjustment, and subsequent locking and sealing of the valve at the required set pressure.

The most important requirement, in addition to the usual safety considerations is that instrument quality gauges are used and a regular calibration system is in place. All safety valve standards will specify a particular tolerance for the set pressure (which is typically around 3%) and this must be observed. It is also important that the environment is clean, dust free and relatively quiet.

The source of the setting fluid can vary from a compressed air cylinder to an intensifier and accumulator vessel running off an industrial compressed air main. In the latter case, the air must be clean, oil, and water free.

It is worth noting that there is no requirement for any sort of capacity test. The test stand simply enables the required set pressure to be ascertained. Usually this point is established by listening for an audible ‘hiss’ as the set point is reached. When making adjustments it is imperative for both metal seated and soft seated valves that the disc is not allowed to turn on the seat or nozzle, since this can easily cause damage and prevent a good shut-off being achieved. The stem should therefore be gripped whilst the adjuster is turned.

There is a fundamental difference in the allowable setting procedures for ASME I steam boiler valves. In order to maintain the National Board approval and to apply the ‘V’ stamp to the valve body, these valves must be set using steam on a rig capable not only of achieving the desired set pressure but also with sufficient capacity to demonstrate the popping point and reseat point. This must be done in accordance with an approved, and controlled, quality procedure. For ASME VIII valves (stamped on the body with ‘UV’), if the setter has a steam setting facility, then these valves must also be set on steam. If not, then gas or air setting is permissible. For liquid applications with ASME VIII valves, the appropriate liquid, usually water, must be used for setting purposes.

In the case of valves equipped with blowdown rings, the set positions will need to be established and the locking pins sealed in accordance with the relevant manufacturer’s recommendations.

**Sealing**

For valves not claiming any particular standard and with no reference to a standard on the name-plate or supporting literature there is no restriction on who can set the valve. Such valves are normally used to indicate that a certain pressure has been reached, and do not act as a safety device.

For valves that are independently approved by a notified body, to a specific standard, the setting and sealing of the valve is a part of the approval. In this case, the valve must be set by the manufacturer or an approved agent of the manufacturer working in accordance with agreed quality procedures and using equipment approved by the manufacturer or the notified body.

To prevent unauthorised alteration or tampering, most standards require provision to be made for sealing the valve after setting.

The most common method is to use sealing wire to secure the cap to the spring housing and the housing to the body. It may also be used to lock any blowdown adjuster ring pins into position.

The wire is subsequently sealed with a lead seal, which may bear the imprint of the setter’s trademark.

---

*Fig. 9.3.2 Sealed cap showing a lead seal*
Safety valve positioning

In order to ensure that the maximum allowable accumulation pressure of any system or apparatus protected by a safety valve is never exceeded, careful consideration of the safety valve’s position in the system has to be made. As there is such a wide range of applications, there is no absolute rule as to where the valve should be positioned and therefore, every application needs to be treated separately.

A common steam application for a safety valve is to protect process equipment supplied from a pressure reducing station. Two possible arrangements are shown in Figure 9.3.3.

![Diagram of safety valve positioning](image)

**Fig. 9.3.3 Possible positioning of a safety valve in a pressure reducing station**

The safety valve can be fitted within the pressure reducing station itself, that is, before the downstream stop valve, as in Figure 9.3.3 (a), or further downstream, nearer the apparatus as in Figure 9.3.3 (b). Fitting the safety valve before the downstream stop valve has the following advantages:

- The safety valve can be tested in-line by shutting down the downstream stop valve without the chance of downstream apparatus being over pressurised, should the safety valve fail under test.
- When the testing is carried out in-line, the safety valve does not have to be removed and bench tested, which is more costly and time consuming.
- When setting the PRV under no-load conditions, the operation of the safety valve can be observed, as this condition is most likely to cause ‘simmer’. If this should occur, the PRV pressure can be adjusted to below the safety valve reseat pressure.
- Any additional take-offs downstream are inherently protected. Only apparatus with a lower MAWP requires additional protection. This can have significant cost benefits.
It is however sometimes practical to fit the safety valve closer to the steam inlet of any apparatus. Indeed, a separate safety valve may have to be fitted on the inlet to each downstream piece of apparatus, when the PRV supplies several such pieces of apparatus.

The following points can be used as a guide:

- If supplying one piece of apparatus, which has a MAWP pressure less than the PRV supply pressure, the apparatus must be fitted with a safety valve, preferably close-coupled to its steam inlet connection.

- If a PRV is supplying more than one apparatus and the MAWP of any item is less than the PRV supply pressure, either the PRV station must be fitted with a safety valve set at the lowest possible MAWP of the connected apparatus, or each item of affected apparatus must be fitted with a safety valve.

- The safety valve must be located so that the pressure cannot accumulate in the apparatus via another route, for example, from a separate steam line or a bypass line.

It could be argued that every installation deserves special consideration when it comes to safety, but the following applications and situations are a little unusual and worth considering:

- **Fire** - Any pressure vessel should be protected from overpressure in the event of fire. Although a safety valve mounted for operational protection may also offer protection under fire conditions, such cases require special consideration, which is beyond the scope of this text.

- **Exothermic applications** - These must be fitted with a safety valve close-coupled to the apparatus steam inlet or the body direct. No alternative applies.

- **Safety valves used as warning devices** - Sometimes, safety valves are fitted to systems as warning devices. They are not required to relieve fault loads but to warn of pressures increasing above normal working pressures for operational reasons only. In these instances, safety valves are set at the warning pressure and only need to be of minimum size. If there is any danger of systems fitted with such a safety valve exceeding their maximum allowable working pressure, they must be protected by additional safety valves in the usual way.

**Example 9.3.2**

In order to illustrate the importance of the positioning of a safety valve, consider an automatic pump trap (see Block 14) used to remove condensate from a heating vessel. The automatic pump trap (APT), incorporates a mechanical type pump, which uses the motive force of steam to pump the condensate through the return system. The position of the safety valve will depend on the MAWP of the APT and its required motive inlet pressure.

If the MAWP of the APT is more than or equal to that of the vessel, the arrangement shown in Figure 9.3.4 could be used.

**Fig. 9.3.4**

Pressure reducing station arrangement for automatic pump trap and process vessel system
This arrangement is suitable if the pump-trap motive pressure is less than 1.6 bar g (safety valve set pressure of 2 bar g less 0.3 bar blowdown and a 0.1 bar shut-off margin). Since the MAWP of both the APT and the vessel are greater than the safety valve set pressure, a single safety valve would provide suitable protection for the system.

However, if the pump-trap motive pressure had to be greater than 1.6 bar g, the APT supply would have to be taken from the high pressure side of the PRV, and reduced to a more appropriate pressure, but still less than the 4.5 bar g MAWP of the APT. The arrangement shown in Figure 9.3.5 would be suitable in this situation.

Here, two separate PRV stations are used each with its own safety valve. If the APT internals failed and steam at 4 bar g passed through the APT and into the vessel, safety valve ‘A’ would relieve this pressure and protect the vessel. Safety valve ‘B’ would not lift as the pressure in the APT is still acceptable and below its set pressure.

It should be noted that safety valve ‘A’ is positioned on the downstream side of the temperature control valve; this is done for both safety and operational reasons:

- **Safety** - If the internals of the APT failed, the safety valve would still relieve the pressure in the vessel even if the control valve were shut.

- **Operation** - There is less chance of safety valve ‘A’ simmering during operation in this position, as the pressure is typically lower after the control valve than before it.

Also, note that if the MAWP of the pump-trap were greater than the pressure upstream of PRV ‘A’, it would be permissible to omit safety valve ‘B’ from the system, but safety valve ‘A’ must be sized to take into account the total fault flow through PRV ‘B’ as well as through PRV ‘A’.

**Fig. 9.3.5** The automatic pump trap and vessel system using two PRV stations
**Example 9.3.3**
A pharmaceutical factory has twelve jacketed pans on the same production floor, all rated with the same MAWP. Where would the safety valve be positioned?

One solution would be to install a safety valve on the inlet to each pan (Figure 9.3.6). In this instance, each safety valve would have to be sized to pass the entire load, in case the PRV failed open whilst the other eleven pans were shut down.

![Fig. 9.3.6 Protection of the heating pans using individual safety valves](image)

As all the pans are rated to the same MAWP, it is possible to install a single safety valve after the PRV.

![Fig. 9.3.7 Protection of heating pans using a single safety valve](image)

If additional apparatus with a lower MAWP than the pans (for example, a shell and tube heat exchanger) were to be included in the system, it would be necessary to fit an additional safety valve. This safety valve would be set to an appropriate lower set pressure and sized to pass the fault flow through the temperature control valve (see Figure 9.3.8).

![Fig. 9.3.8 Possible safety valve arrangement if additional apparatus was included in the system](image)
Safety Valves Sizing
Safety Valve Sizing

A safety valve must always be sized and able to vent any source of steam so that the pressure within the protected apparatus cannot exceed the maximum allowable accumulated pressure (MAAP). This not only means that the valve has to be positioned correctly, but that it is also correctly set. The safety valve must then also be sized correctly, enabling it to pass the required amount of steam at the required pressure under all possible fault conditions.

Once the type of safety valve has been established, along with its set pressure and its position in the system, it is necessary to calculate the required discharge capacity of the valve. Once this is known, the required orifice area and nominal size can be determined using the manufacturer’s specifications.

In order to establish the maximum capacity required, the potential flow through all the relevant branches, upstream of the valve, need to be considered.

In applications where there is more than one possible flow path, the sizing of the safety valve becomes more complicated, as there may be a number of alternative methods of determining its size. Where more than one potential flow path exists, the following alternatives should be considered:

- The safety valve can be sized on the maximum flow experienced in the flow path with the greatest amount of flow.
- The safety valve can be sized to discharge the flow from the combined flow paths.

This choice is determined by the risk of two or more devices failing simultaneously. If there is the slightest chance that this may occur, the valve must be sized to allow the combined flows of the failed devices to be discharged. However, where the risk is negligible, cost advantages may dictate that the valve should only be sized on the highest fault flow. The choice of method ultimately lies with the company responsible for insuring the plant.

For example, consider the pressure vessel and automatic pump-trap (APT) system as shown in Figure 9.4.1. The unlikely situation is that both the APT and pressure reducing valve (PRV ‘A’) could fail simultaneously. The discharge capacity of safety valve ‘A’ would either be the fault load of the largest PRV, or alternatively, the combined fault load of both the APT and PRV ‘A’.

This document recommends that where multiple flow paths exist, any relevant safety valve should, at all times, be sized on the possibility that relevant upstream pressure control valves may fail simultaneously.

Fig. 9.4.1 An automatic pump-trap and pressure vessel system
Finding the fault flow

In order to determine the fault flow through a PRV or indeed any valve or orifice, the following need to be considered:

- The potential fault pressure - this should be taken as the set pressure of the appropriate upstream safety valve
- The relieving pressure of the safety valve being sized
- The full open capacity \( (K_{VS}) \) of the upstream control valve, see Equation 3.21.2

**Example 9.4.1**

Consider the PRV arrangement in Figure 9.4.2.

Where:
- NWP = Normal working pressure
- MAAP = Maximum allowable accumulated pressure
- \( P_s \) = Safety valve set pressure
- \( P_o \) = Safety valve overpressure
- \( P_r \) = Safety valve relieving pressure

\[ P_s = 4.0 \text{ bar g} \]
\[ P_o = 5\% \text{ of } P_s \]
\[ P_r = 4 \times 1.05 \]
\[ P_r = 4.2 \text{ bar g} \]

The supply pressure of this system (Figure 9.4.2) is limited by an upstream safety valve with a set pressure of 11.6 bar g. The fault flow through the PRV can be determined using the steam mass flow equation (Equation 3.21.2):

**Equation 3.21.2**

\[ \dot{m}_s = 12 K_v P_1 \sqrt{1 - 5.67 (0.42 - \chi)^2} \]

Where:
- \( \dot{m}_s \) = Fault load (kg/h)
- \( K_v \) = PRV full open capacity index \( (K_{VS} = 6.3) \)
- \( \chi \) = Pressure drop \[ \chi = \frac{P_1 - P_2}{P_1} \]
- \( P_1 \) = Fault pressure (taken as the set pressure of the upstream safety valve) (bar a)
- \( P_2 \) = Relieving pressure of the apparatus safety valve (bar a)

Equation 3.21.2 is used when the pressure drop ratio is less than 0.42. If the pressure drop ratio is 0.42 or greater, the mass flow is calculated using Equation 6.4.3

**Equation 6.4.3**

\[ \dot{m}_s = 12 K_v P_1 \]
In this example:

\[ P_1 = 11.6 \text{ bar g} = 12.6 \text{ bar a} \]
\[ P_2 = 4.2 \text{ bar g} = 5.2 \text{ bar a} \]

Therefore: \( \chi = \frac{12.6 - 5.2}{12.6} = 0.59 \)

Since \( \chi \) is greater than 0.42, critical pressure drop occurs across the control valve, and the fault flow is calculated as follows using the formula in Equation 6.4.3:

\[ \dot{m}_v = 12 K_v P_1 \]
\[ \dot{m}_v = 12 \times 6.3 \times 12.6 \]

Therefore: \( \dot{m}_v = 953 \text{ kg/h} \)

Consequently, the safety valve would be sized to pass at least 953 kg/h when set at 4 bar g.

Once the fault load has been determined, it is usually sufficient to size the safety valve using the manufacturer’s capacity charts. A typical example of a capacity chart is shown in Figure 9.4.3. By knowing the required set pressure and discharge capacity, it is possible to select a suitable nominal size. In this example, the set pressure is 4 bar g and the fault flow is 953 kg/h. A DN32/50 safety valve is required with a capacity of 1284 kg/h.

---

**SV615 flow capacity for saturated steam in kilogrammes per hour (kg/h)**
(calculated in accordance with EN ISO 4126 at 5% overpressure)

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<tr>
<th>Valve size DN</th>
<th>15/20</th>
<th>20/32</th>
<th>25/40</th>
<th>32/50</th>
<th>40/65</th>
<th>50/80</th>
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<td>314</td>
<td>452</td>
<td>661</td>
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<td>1 662</td>
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<td>Set pressure (bar g)</td>
<td>Flow capacity for saturated steam kg/h</td>
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<td></td>
<td></td>
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<tr>
<td>0.5</td>
<td>65</td>
<td>180</td>
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<td>5 831</td>
</tr>
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Fig. 9.4.3 A typical safety valve capacity chart
Where sizing charts are not available or do not cater for particular fluids or conditions, such as backpressure, high viscosity or two-phase flow, it may be necessary to calculate the minimum required orifice area. Methods for doing this are outlined in the appropriate governing standards, such as:

- AD-Merkblatt A2, DIN 3320, TRD 421
- ASME/API RP 520
- EN ISO 4126

The methods outlined in these standards are based on the coefficient of discharge, which is the ratio of the measured capacity to the theoretical capacity of a nozzle with an equivalent flow area.

\[
K_d = \frac{\text{Actual flowing capacity}}{\text{Theoretical flowing capacity}}
\]

Where:
\(K_d\) = Coefficient of discharge

**Coefficient of discharge**

Coefficients of discharge are specific to any particular safety valve range and will be approved by the manufacturer. If the valve is independently approved, it is given a ‘certified coefficient of discharge’.

This figure is often derated by further multiplying it by a safety factor 0.9, to give a derated coefficient of discharge. Derated coefficient of discharge is termed \(K_{dr} = K_d \times 0.9\).

When using standard methods of calculating the required orifice area, the following points may need to be considered:

- **Critical and sub-critical flow** - the flow of gas or vapour through an orifice, such as the flow area of a safety valve, increases as the downstream pressure is decreased. This holds true until the critical pressure is reached, and critical flow is achieved. At this point, any further decrease in the downstream pressure will not result in any further increase in flow.

A relationship (called the critical pressure ratio) exists between the critical pressure and the actual relieving pressure, and, for gases flowing through safety valves, is shown by Equation 9.4.2.

\[
\frac{P_B}{P_1} = \left( \frac{2}{k + 1} \right)^{k/(k-1)}
\]

Where:
\(P_B\) = Critical backpressure (bar a)
\(P_1\) = Actual relieving pressure (bar a)
\(k\) = Isentropic coefficient of the gas or vapour at the relieving conditions

For gases, with similar properties to an ideal gas, ‘\(k\)’ is the ratio of specific heat of constant pressure \(c_p\) to constant volume \(c_v\), i.e. \(c_p : c_v\). ‘\(k\)’ is always greater than unity, and typically between 1 and 1.4 (see Table 9.4.8).

For steam, although ‘\(k\)’ is an isentropic coefficient, it is not actually the ratio of \(c_p : c\). As an approximation for saturated steam, ‘\(k\)’ can be taken as 1.135, and superheated steam, as 1.3.

As a guide, for saturated steam, critical pressure is taken as 58% of accumulated inlet pressure in absolute terms.

- **Overpressure** - Before sizing, the design overpressure of the valve must be established. It is not permitted to calculate the capacity of the valve at a lower overpressure than that at which the coefficient of discharge was established. It is however, permitted to use a higher overpressure (see Table 9.2.1, Module 9.2, for typical overpressure values). For DIN type full lift (Vollhub) valves, the design lift must be achieved at 5% overpressure, but for sizing purposes, an overpressure value of 10% may be used.

For liquid applications, the overpressure is 10% according to AD-Merkblatt A2, DIN 3320, TRD 421 and ASME, but for non-certified ASME valves, it is quite common for a figure of 25% to be used.
• **Backpressure** - The sizing calculations in the AD-Merkblatt A2, DIN 3320 and TRD 421 standards account for backpressure in the outflow function,$(\Psi)$, which includes a backpressure correction. The ASME/API RP 520 and EN ISO 4126 standards, however, require an additional backpressure correction factor to be determined and then incorporated in the relevant equation.

• **Two-phase flow** - When sizing safety valves for boiling liquids (e.g. hot water) consideration must be given to vaporisation (flashing) during discharge. It is assumed that the medium is in liquid state when the safety valve is closed and that, when the safety valve opens, part of the liquid vaporises due to the drop in pressure through the safety valve. The resulting flow is referred to as two-phase flow.

The required flow area has to be calculated for the liquid and vapour components of the discharged fluid. The sum of these two areas is then used to select the appropriate orifice size from the chosen valve range. (see Example 9.4.3)

Many standards do not actually specify sizing formula for two-phase flow and recommend that the manufacturer be contacted directly for advice in these instances.

### Sizing equations for safety valves designed to the following standards

The following methods are used to calculate the minimum required orifice area for a safety valve, as mentioned in the most commonly used national standards.

**Standard - AD-Merkblatt A2, DIN 3320, TRD 421**

Use Equation 9.4.3 to calculate the minimum required orifice area for a safety valve used on steam applications:

\[
A_o = \frac{\chi \dot{m}}{\alpha_w P_R \Psi}
\]

Use Equation 9.4.4 to calculate the minimum required orifice area for a safety valve used on air and gas applications:

\[
A_o = \frac{0.1791 \dot{m}}{\Psi \alpha_w P_R^{0.1791}} \sqrt{\frac{T Z}{M}}
\]

Use Equation 9.4.5 to calculate the minimum required orifice area for a safety valve used on liquid applications:

\[
A_o = \frac{0.6211 \dot{m}}{\alpha_w \sqrt{\rho \Delta P}}
\]

Where:

- $A_o$ = Minimum cross sectional flow area (mm²)
- $\dot{m}$ = Mass flow to be discharged (kg/h)
- $P_R$ = Absolute relieving pressure (bar a)
- $\Delta P = P_R - P_B$
- $P_B$ = Absolute backpressure (bar a)
- $T$ = Inlet temperature (K)
- $\rho$ = Density (kg/m³) (see Appendix A at the back of this module)
- $M$ = Molar mass (kg/kmol) (see Appendix A at the back of this module)
- $Z$ = Compressibility factor (see Equation 9.4.6)
- $\alpha_w$ = Outflow coefficient (specified by the manufacturer)
- $\Psi$ = Outflow function (see Figure 9.4.4)
- $\chi$ = Pressure medium coefficient (see Figure 9.4.5)
The outflow function ($\psi$) for air and gas applications

Fig. 9.4.4 The outflow function ($\psi$) as used in AD-Merkblatt A2, DIN 3320 and TRD 421

Outflow function $\psi$

Pressure ratio ($P_B/P_R$)

$P_B = $ Absolute backpressure

$P_R = $ Absolute relieving pressure
Pressure medium coefficient ($\chi$) for steam applications

Fig. 9.4.5  Pressure medium coefficient ($\chi$) for steam as used in AD-Merkblatt A2, DIN 3320, TRD 421
Compressibility factor (Z)
For gases, the compressibility factor, Z, also needs to be determined. This factor accounts for the deviation of the actual gas from the characteristics of an ideal gas. It is often recommended that Z = 1 is used where insufficient data is available. Z can be calculated by using the formula in Equation 9.4.6:

\[
Z = \frac{10^5 P_R M \nu}{R_u T}
\]

Where:
- Z = Compressibility factor
- \( P_R \) = Safety valve relieving pressure (bar a)
- \( \nu \) = Specific volume of the gas at the actual relieving pressure and temperature (m\(^3\)/kg) (see Appendix A at the back of this module).
- Note: The specific volume of a gas will change with temperature and pressure, and therefore it must be determined for the operating conditions.
- M = Molar mass (kg/kmol) (see Appendix A at the back of this module)
- \( R_u \) = Universal gas constant (8 314 Nm/kmol K)
- T = Actual relieving temperature (K)
**Example 9.4.2**

Determine the minimum required safety valve orifice area under the following conditions:

- **Medium:** Saturated steam
- **Discharge quantity (ṁ):** 2500 kg/h
- **Set pressure (P_s):** 4 bar a
- **Backpressure:** Atmospheric pressure 1 bar a
- **Stated outflow coefficient (α_w):** 0.7

It is first necessary to determine the pressure medium coefficient using Figure 9.4.5.

- **Pressure medium coefficient (χ):** 1.88

Using Equation 9.4.3:

\[ A_0 = \frac{\chi \times \dot{m}}{\alpha_w \times P_s} \]

Therefore:

\[ A_0 = \frac{1.88 \times 2500}{0.7 \times 4} = 1678 \text{ mm}^2 \]

Consequently, the chosen safety valve would need an orifice area of at least 1678 mm².

**Two-phase flow**

In order to determine the minimum orifice area for a two-phase flow system (e.g. hot water), it is first necessary to establish what proportion of the discharge will be vapour (n). This is done using the Equation 9.4.7:

**Equation 9.4.7**

\[ n = \frac{h_{f1} - h_{f2}}{h_{fg2}} \]

Where:

- \( h_{f1} \): Enthalpy of liquid before the valve (kJ/kg)
- \( h_{f2} \): Enthalpy of liquid after the valve (kJ/kg)
- \( h_{fg2} \): Enthalpy of evaporation after the valve (kJ/kg)

For hot water, the enthalpy values can be obtained from steam tables.

In order to determine the proportion of flow, which is vapour, the discharge capacity is multiplied by n. The remainder of the flow will therefore be in the liquid state.

The area sizing calculation from Equations 9.4.3, 9.4.4 and 9.4.5 can then be used to calculate the required area to discharge the vapour portion and then the liquid portion. The sum of these areas is then used to establish the minimum required orifice area.
Example 9.4.3
Consider hot water under the following conditions:

- **Temperature**: 160°C
- **Discharge quantity (ṁ)**: 3900 kg/h
- **Set pressure (Pₛ)**: 10 bar g = 11 bar a
- **Backpressure (Pᴮ)**: Atmospheric
- **Density of water at 160°C (ρ)**: 908 kg/m³
- **ΔP = Pₛ - Pᴮ**: 10 bar
- **Stated outflow coefficient (αₜₚ)**: 0.7

Using steam tables, the proportion of vapour is first calculated:

\[ h_{f1} = 675 \text{ kJ/kg (at 160°C)} \]
\[ h_{f2} = 417 \text{ kJ/kg (at 1 bar a, atmospheric pressure)} \]
\[ h_{fg2} = 2258 \text{ kJ/kg (at 1 bar a, atmospheric pressure)} \]

Using Equation 9.4.7:

\[ n = \frac{h_{f1} - h_{f2}}{h_{fg2}} \]

Therefore:

\[ n = \frac{675 - 417}{2258} = 0.1143 \]

**Capacity discharge as vapour (steam)** = 0.1143 × 3900 kg/h = 446 kg/h

**Capacity discharge as liquid (water)** = 3900 kg/h - 446 kg/h = 3454 kg/h

Calculated area for vapour portion:

Using Equation 9.4.3:

\[ A₀ = \frac{χ \dot{m}}{αₜₚ PR} \]

Therefore:

\[ A₀_{\text{Steam}} = \frac{1.92 \times 446}{0.7 \times 11} = 111 \text{ mm}² \]

Calculated area for liquid portion:

Using Equation 9.4.5:

\[ A₀ = \frac{0.6211 \times \dot{m}}{αₜₚ \sqrt{ρ \Delta P}} \]

Therefore:

\[ A₀_{\text{liquid}} = \frac{0.6211 \times 3454}{0.7 \sqrt{908 \times 10}} = 33 \text{ mm}² \]

**Total required discharge area** = 111 + 33 = 144 mm²

Therefore, a valve must be selected with a discharge area greater than 144 mm².
**Standard - ASME/API RP 520**

The following formulae are used for calculating the minimum required orifice area for a safety valve according to ASME standards and the API RP 520 guidelines.

Use Equation 9.4.8 to calculate the minimum required orifice area for a safety valve used on steam applications:

**Equation 9.4.8**

\[ A_o = \frac{\dot{m}}{51.5 \ Pa \ K_d \ K_{SH}} \]

Use Equation 9.4.9 to calculate the minimum required orifice area for a safety valve used on air and gas applications:

**Equation 9.4.9**

\[ A_o = \frac{\dot{V} \sqrt{T} \ Z \ Q}{1.175 \ C_g \ K_d \ P_R \ K_B} \]

Use Equation 9.4.10 to calculate the minimum required orifice area for a safety valve used on liquid applications:

**Equation 9.4.10**

\[ A_o = \frac{\dot{V}^1}{38 \ K_d \ K_{\mu} \ K_{W}} \sqrt{\frac{G}{P_R - P_B}} \]

Where:

- \( A_o \) = Required effective discharge area (in²)
- \( \dot{m} \) = Required mass flow through the valve (lb/h)
- \( \dot{V} \) = Required volume flow through the valve (ft³/min)
- \( \dot{V}^1 \) = Required volume flow through the valve (U.S. gal/min)
- \( P_R \) = Upstream relieving pressure (psi a)
- \( P_B \) = Absolute backpressure (psi a)
- \( C_g \) = Nozzle gas constant (see Table 9.4.1)
- \( T \) = Relieving temperature (°R = °F + 460)
- \( G \) = Specific gravity (ratio of molar mass of the fluid to the molar mass of air (28.96 kg/kmol)) (see Appendix A at the back of this module)
- \( Z \) = Compressibility factor (see Equation 9.4.6)
- \( K_d \) = Effective coefficient of discharge (specified by the manufacturer)
- \( K_{SH} \) = Superheat correction factor (see Table 9.4.2)
- \( K_B \) = Backpressure correction factor for gas and vapour (see Figures 9.4.6 and 9.4.7)
- \( K_W \) = Backpressure correction factor for liquids (bellows balanced valves only) (see Figure 9.4.8)
- \( K_{\mu} \) = Viscosity factor (see Figure 9.4.9)
## Nozzle gas constant for ASME/API RP 520

### Table 9.4.1  Nozzle gas constant ($C_g$) relative to isentropic constant ($k$) for air and gases

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<th>$C_g$</th>
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<th>$C_g$</th>
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The nozzle gas constant $C_g$ is calculated using Equation 9.4.11, for air and gas applications and applied to Equation 9.4.9.

**Equation 9.4.11**

\[
C_g = \frac{520}{k} \sqrt{\frac{2}{k+1}} \left(\frac{k}{k+1}\right)^{\frac{k}{k+1}} \text{ for } k > 1
\]

\[
C_g = 315 \text{ for } k = 1
\]
Superheat correction factors for ASME/API RP 520

Table 9.4.2  Superheat correction factors (K\text{SH}) as used in ASME / API RP 520 (Imperial units)

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Gas and vapour constant backpressure correction factor for ASME/API 520

The backpressure correction factor (K\text{B}) is the ratio of the capacity with backpressure, C\text{1}, to the capacity when discharging to atmosphere, C\text{2}, see Equation 9.4.12.

Equation 9.4.12  \[ K_{B} = \frac{C_{1}}{C_{2}} \]

The value of K\text{B} can be established using the curves shown in Figure 9.4.6 to Figure 9.4.8. These are applicable to set pressures of 50 psi g and above. For a given set pressure, these values are limited to a backpressure less than the critical pressure, namely, critical flow conditions. For sub-critical flow and backpressures below 50 psi g, the manufacturer should be consulted for values of K\text{B}.
Balanced bellows valves

Equation 9.4.13  
\[
\% \text{ of gauge backpressure} = \frac{P_B}{P_S} \times 100
\]

Where:
\( P_B \) = Backpressure (psi g)
\( P_S \) = Set pressure (psi g)

\( K_B = \frac{C_1}{C_2} \)

Fig. 9.4.6  Constant backpressure correction factor \((K_B)\) for gas and vapour as used in ASME/API RP 520 for balanced bellows valves

Conventional valves

Equation 9.4.14  
\[
\% \text{ of gauge backpressure} = \frac{P_B}{P_R} \times 100
\]

Where:
\( P_B \) = Backpressure (psi g)
\( P_S \) = Relieving pressure (psi g)

\( K_B = \frac{C_1}{C_2} \)

\( k = \) isentropic coefficient (see Table 9.4.6)

Fig. 9.4.7  Constant backpressure correction factor \((K_B)\) for gas and vapour as used in ASME / API RP 520 for conventional valves
Liquid constant backpressure correction factor for ASME/API RP 520

Balanced bellows valves

Fig. 9.4.8 Constant backpressure correction factor ($K_W$) for liquids as used in ASME / API RP 520 for balanced bellows valves

Percent of gauge backpressure = \frac{\text{Back pressure}}{\text{Set pressure}} \times 100

Viscosity correction factor for ASME/API RP 520 and EN ISO 4126

This is used to make allowances for high viscosity fluids. In order to account for this, the valve size must first be established, assuming the fluid is non-viscous. Once the size has been selected, the Reynolds number for the valve is calculated and used to establish the correction factor from Figure 9.4.9.

The valve size should then be checked to ensure that the original size chosen would accommodate the flow after the viscous correction factor has been applied. If not this process should be repeated with the next largest valve size.

Fig. 9.4.9 Viscosity correction factor ($K_\mu$) as used in ASME / API RP 520 and BS 6759

The Reynolds number can be calculated using Equations 9.4.15 and 9.4.16:

<table>
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<tr>
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<td>$R_s = 0.3414 \frac{\dot{m}}{\mu \sqrt{A_o}}$</td>
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<table>
<thead>
<tr>
<th>Imperial units</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_s = \frac{2800 G \dot{V}}{\mu \sqrt{A_o}}$</td>
</tr>
</tbody>
</table>

Where:
- $R_s$ = Reynolds number
- $\dot{m}$ = Mass flow to be discharged (kg/h)
- $G$ = Relative density-specific gravity of the liquid (dimensionless)
- $\dot{V}$ = Volume flow to be discharged (U.S. gal/min)
- $\mu$ = Dynamic viscosity (Imperial – cP, Metric – Pa s)
- $A_o$ = Discharge area (Imperial – in², Metric – mm²)
Use Equation 9.4.17 to calculate the minimum required orifice area for a safety valve used on dry saturated steam, superheated steam, and air and gas applications at critical flow:

\[
A = \frac{\dot{m}}{0.288 \times 3 \times C \times K_{dr} \times \sqrt{\frac{P_o}{\varphi g}}}
\]

Use Equation 9.4.18 to calculate the minimum required orifice area for a safety valve used on wet steam applications at critical flow; Note: wet steam must have a dryness fraction greater than 0.9:

\[
A = \frac{\dot{m}}{0.288 \times 3 \times C \times K_{dr} \times \sqrt{\frac{P_o}{\varphi g} \times x}}
\]

Use Equation 9.4.19 to calculate the minimum required orifice area for a safety valve used on air and gas applications at sub-critical flow:

\[
A = \frac{\dot{m}}{0.288 \times 3 \times K_{dr} \times \sqrt{\frac{P_o}{\varphi g}}}
\]

Use Equation 9.4.20 to calculate the minimum required orifice area for a safety valve used on liquid applications:

\[
A = \frac{\dot{m}}{1.61 \times K_{dr} \times K_v \times \sqrt{\frac{P_o - P_b}{\varphi g}}}
\]

Where:
- A = Flow area (not curtain area) mm²
- \( \dot{m} \) = Mass flowrate (kg/h)
- C = Function of the isentropic exponent (see Table 9.4.3)
- K_{dr} = Certified derated coefficient of discharge (from manufacturer)
- P_o = Relieving pressure (bar a)
- P_b = Backpressure (bar a)
- \( \varphi_g \) = Specific volume at relieving pressure and temperature (m³/kg)
- x = Dryness fraction of wet steam
- K_v = Theoretical correction factor for sub-critical flow (see Table 9.4.4)
- K_v = Viscosity correction factor (see Figure 9.4.10)
Table 9.4.3
Value of C as a function of ‘k’ for steam, air and gas applications to the EN ISO 4126 standard. ‘k’ values are incorporated into the ISO 4126 standard: (Part 7). Alternatively, ‘k’ values can be obtained from the Spirax Sarco website steam tables.

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Table 9.4.4
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<td>0.633</td>
<td>0.584</td>
<td>0.527</td>
<td>0.461</td>
<td>0.380</td>
<td>0.271</td>
</tr>
<tr>
<td>1.8</td>
<td>-</td>
<td>0.998</td>
<td>0.987</td>
<td>0.967</td>
<td>0.936</td>
<td>0.895</td>
<td>0.841</td>
<td>0.773</td>
<td>0.741</td>
<td>0.706</td>
<td>0.677</td>
<td>0.624</td>
<td>0.575</td>
<td>0.519</td>
<td>0.453</td>
<td>0.373</td>
<td>0.266</td>
</tr>
<tr>
<td>1.9</td>
<td>-</td>
<td>0.996</td>
<td>0.983</td>
<td>0.961</td>
<td>0.929</td>
<td>0.886</td>
<td>0.832</td>
<td>0.764</td>
<td>0.732</td>
<td>0.697</td>
<td>0.658</td>
<td>0.615</td>
<td>0.566</td>
<td>0.511</td>
<td>0.446</td>
<td>0.367</td>
<td>0.262</td>
</tr>
<tr>
<td>2.0</td>
<td>1.000</td>
<td>0.994</td>
<td>0.979</td>
<td>0.955</td>
<td>0.922</td>
<td>0.879</td>
<td>0.824</td>
<td>0.755</td>
<td>0.723</td>
<td>0.688</td>
<td>0.649</td>
<td>0.606</td>
<td>0.558</td>
<td>0.504</td>
<td>0.440</td>
<td>0.362</td>
<td>0.258</td>
</tr>
<tr>
<td>2.1</td>
<td>0.999</td>
<td>0.992</td>
<td>0.975</td>
<td>0.950</td>
<td>0.915</td>
<td>0.871</td>
<td>0.815</td>
<td>0.747</td>
<td>0.715</td>
<td>0.680</td>
<td>0.641</td>
<td>0.599</td>
<td>0.551</td>
<td>0.497</td>
<td>0.434</td>
<td>0.357</td>
<td>0.254</td>
</tr>
<tr>
<td>2.2</td>
<td>0.999</td>
<td>0.989</td>
<td>0.971</td>
<td>0.945</td>
<td>0.909</td>
<td>0.864</td>
<td>0.808</td>
<td>0.739</td>
<td>0.707</td>
<td>0.672</td>
<td>0.634</td>
<td>0.592</td>
<td>0.544</td>
<td>0.490</td>
<td>0.428</td>
<td>0.352</td>
<td>0.251</td>
</tr>
</tbody>
</table>
**Example 9.4.4**

Size the minimum flow area required for a safety valve designed to EN ISO 4126 to relieve a superheated steam system of overpressure.

**Steam system conditions**

- Relieving pressure: 20 bar g
- \( P_o \): 21 bar a
- Steam temperature: 280°C
- Flowrate to pass (\( m \)): 2500 kg/h

It is necessary to obtain the following: \( C \), \( K_{dr} \) and \( \nu \)

From EN ISO 4126:7 \( C = 2.628 \)

From the manufacturer \( K_{dr} = 0.71 \)

From steam tables \( \nu_g = 0.1138 \text{ m}^3/\text{kg} \)

From Equation 9.4.23

\[
A = \frac{\dot{m}}{0.2883 \cdot C \cdot K_{dr} \cdot \sqrt{\frac{P_o}{\nu_g}}}
\]

\[
A = \frac{2500}{0.2883 \cdot 2.628 \cdot 0.71 \cdot \sqrt{\frac{21}{0.1138}}}
\]

\[
A = \frac{2500}{0.2883 \cdot 2.628 \cdot 0.71 \cdot 13.58}
\]

\[
A = 342 \text{ mm}^2
\]
Appendix A - Properties of industrial liquids

Table 9.4.5 Properties of some common industrial liquids
For specific gravity (G) used in ASME liquid sizing calculations, divide density by 998 (density of water).

<table>
<thead>
<tr>
<th>Liquid</th>
<th>Chemical formula</th>
<th>Boiling point (°C) at 1.013 bar a</th>
<th>Density (kg/m³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Acetone</td>
<td>CH₂.CO.CH₃</td>
<td>56.0</td>
<td>791</td>
</tr>
<tr>
<td>Ammonia</td>
<td>NH₃</td>
<td>-33.4</td>
<td>609</td>
</tr>
<tr>
<td>Benzene</td>
<td>C₆H₆</td>
<td>80.0</td>
<td>879</td>
</tr>
<tr>
<td>Butalene</td>
<td>C₄H₈</td>
<td>-6.3</td>
<td>600</td>
</tr>
<tr>
<td>Butane</td>
<td>C₄H₁₀</td>
<td>-0.5</td>
<td>580</td>
</tr>
<tr>
<td>Carbon disulphide</td>
<td>CS₂</td>
<td>46.0</td>
<td>1260</td>
</tr>
<tr>
<td>Carbon tetrachloride</td>
<td>CCl₄</td>
<td>76.7</td>
<td>1594</td>
</tr>
<tr>
<td>20% caustic soda</td>
<td>NaOH</td>
<td></td>
<td>1220</td>
</tr>
<tr>
<td>Crude oil</td>
<td></td>
<td>700 to 1040</td>
<td></td>
</tr>
<tr>
<td>Diesel oil</td>
<td></td>
<td>175.0</td>
<td>880</td>
</tr>
<tr>
<td>Ethanol</td>
<td>C₂H₅OH</td>
<td>78.0</td>
<td>789</td>
</tr>
<tr>
<td>Freon 12</td>
<td>CF₂Cl₂</td>
<td>-29.8</td>
<td>1330</td>
</tr>
<tr>
<td>Glycol</td>
<td>C₂H₄(OH)₂</td>
<td>197.5</td>
<td>1140</td>
</tr>
<tr>
<td>Light fuel oil</td>
<td></td>
<td>175.0</td>
<td>850</td>
</tr>
<tr>
<td>Heavy fuel oil</td>
<td></td>
<td>220.0 to 350.0</td>
<td>950</td>
</tr>
<tr>
<td>Kerosene</td>
<td></td>
<td>150.0 to 300.0</td>
<td>740</td>
</tr>
<tr>
<td>Methanol</td>
<td>C₃OH</td>
<td>65.0</td>
<td>792</td>
</tr>
<tr>
<td>Naphthalene</td>
<td>C₁₀H₈</td>
<td>218.0</td>
<td>1145</td>
</tr>
<tr>
<td>Nitric acid</td>
<td>HNO₃</td>
<td>86.0</td>
<td>1560</td>
</tr>
<tr>
<td>Propane</td>
<td>C₃H₈</td>
<td>-42.0</td>
<td>500</td>
</tr>
<tr>
<td>Sulphurous acid</td>
<td>H₂SO₃</td>
<td>338.0</td>
<td>1400</td>
</tr>
<tr>
<td>Toluene</td>
<td>C₆H₅.CH₃</td>
<td>111.0</td>
<td>867</td>
</tr>
<tr>
<td>Trichlorethylene</td>
<td>CHCl₂.CCl₂</td>
<td>87.0</td>
<td>1464</td>
</tr>
<tr>
<td>Water</td>
<td>H₂O</td>
<td>100.0</td>
<td>998</td>
</tr>
</tbody>
</table>
Properties of industrial gases

Table 9.4.6  Properties of some common industrial gases
For specific gravity (G) used in ASME gas sizing calculations, divide molar mass by 28.96 (molar mass of air).

<table>
<thead>
<tr>
<th>Gas</th>
<th>Chemical formula</th>
<th>Molar mass (M) kg kmol</th>
<th>Isentropic coefficient (k) at 1.013 bar a and 0°C †</th>
<th>Specific volume (V) m³/kg at 1.013 bar a and 0°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Acetylene</td>
<td>C₂H₂</td>
<td>26.02</td>
<td>1.26</td>
<td>0.853</td>
</tr>
<tr>
<td>Air</td>
<td></td>
<td>28.96</td>
<td>1.40</td>
<td>0.773</td>
</tr>
<tr>
<td>Ammonia</td>
<td>NH₃</td>
<td>17.03</td>
<td>1.31</td>
<td>1.297</td>
</tr>
<tr>
<td>Argon</td>
<td>Ar</td>
<td>39.91</td>
<td>1.66</td>
<td>0.561</td>
</tr>
<tr>
<td>Benzene</td>
<td>C₆H₆</td>
<td>78.00</td>
<td>1.10</td>
<td></td>
</tr>
<tr>
<td>Biphenyl oxide</td>
<td>C₁₂H₁₀</td>
<td>166.00</td>
<td>1.05*</td>
<td>0.0094*</td>
</tr>
<tr>
<td>Butane - n</td>
<td>C₄H₁₀</td>
<td>58.08</td>
<td>1.11</td>
<td>0.370</td>
</tr>
<tr>
<td>Butylene</td>
<td>C₄H₈</td>
<td>56.10</td>
<td>1.20</td>
<td></td>
</tr>
<tr>
<td>Carbon disulphide</td>
<td></td>
<td>76.00</td>
<td>1.21</td>
<td></td>
</tr>
<tr>
<td>Carbon dioxide</td>
<td>CO₂</td>
<td>44.00</td>
<td>1.30</td>
<td>0.506</td>
</tr>
<tr>
<td>Carbon monoxide</td>
<td>CO</td>
<td>28.00</td>
<td>1.40</td>
<td>0.800</td>
</tr>
<tr>
<td>Chlorine</td>
<td>Cl₂</td>
<td>70.91</td>
<td>1.35</td>
<td>0.311</td>
</tr>
<tr>
<td>Cyclohexane</td>
<td></td>
<td>84.00</td>
<td>1.08</td>
<td></td>
</tr>
<tr>
<td>Ethane</td>
<td>C₂H₆</td>
<td>30.05</td>
<td>1.22</td>
<td>0.737</td>
</tr>
<tr>
<td>Ethylene</td>
<td>C₂H₄</td>
<td>28.03</td>
<td>1.25</td>
<td>0.794</td>
</tr>
<tr>
<td>Freon 12</td>
<td>Cl₂F₂</td>
<td>121.00</td>
<td>1.14</td>
<td></td>
</tr>
<tr>
<td>Helium</td>
<td>He</td>
<td>4.00</td>
<td>1.66</td>
<td></td>
</tr>
<tr>
<td>Hexane</td>
<td>C₆H₁₄</td>
<td>86.00</td>
<td>1.08</td>
<td></td>
</tr>
<tr>
<td>Hydrogen</td>
<td>H₂</td>
<td>2.02</td>
<td>1.41</td>
<td>11.124</td>
</tr>
<tr>
<td>Hydrogen chloride</td>
<td>HCl</td>
<td>36.46</td>
<td>1.40</td>
<td>0.610</td>
</tr>
<tr>
<td>Hydrogen sulphide</td>
<td>H₂S</td>
<td>34.08</td>
<td>1.32</td>
<td>0.651</td>
</tr>
<tr>
<td>Isobutane</td>
<td>CH(CH₃)₃</td>
<td>58.05</td>
<td>1.11</td>
<td>0.375</td>
</tr>
<tr>
<td>Methane</td>
<td>CH₄</td>
<td>16.03</td>
<td>1.31</td>
<td>1.395</td>
</tr>
<tr>
<td>Methyl chloride</td>
<td>CH₃Cl</td>
<td>50.48</td>
<td>1.28</td>
<td>0.434</td>
</tr>
<tr>
<td>Natural gas</td>
<td></td>
<td>19.00</td>
<td>1.27</td>
<td></td>
</tr>
<tr>
<td>Nitrogen</td>
<td>N₂</td>
<td>28.02</td>
<td>1.40</td>
<td>0.799</td>
</tr>
<tr>
<td>Nitrous oxide</td>
<td>N₂O</td>
<td>44.02</td>
<td>1.30</td>
<td>0.746</td>
</tr>
<tr>
<td>Oxygen</td>
<td>O₂</td>
<td>32.00</td>
<td>1.40</td>
<td>0.700</td>
</tr>
<tr>
<td>Pentane</td>
<td>C₅H₁₂</td>
<td>72.00</td>
<td>1.09</td>
<td>0.451</td>
</tr>
<tr>
<td>Propane</td>
<td>C₃H₈</td>
<td>44.06</td>
<td>1.13</td>
<td>0.498</td>
</tr>
<tr>
<td>Sulphur dioxide</td>
<td>SO₂</td>
<td>64.07</td>
<td>1.29</td>
<td>0.342</td>
</tr>
<tr>
<td>Dry saturated steam</td>
<td>H₂O</td>
<td>18.015</td>
<td>1.135 †</td>
<td>†</td>
</tr>
<tr>
<td>Superheated steam</td>
<td>H₂O</td>
<td>18.015</td>
<td>1.30 †</td>
<td>†</td>
</tr>
</tbody>
</table>

† These are typical values, not values at 1.013 bar and 0°C
* At 15°C
Safety Valve Installation
Safety Valve Installation

Seat tightness
Seat tightness is an important consideration when selecting and installing a safety valve, as not only can it lead to a continuous loss of system fluid, but leakage can also cause deterioration of the sealing faces, which can lead to premature lifting of the valve.

The seat tightness is affected by three main factors; firstly by the characteristics of the safety valve, secondly by the installation of the safety valve and thirdly, by the operation of the safety valve.

Characteristics of the safety valve
For a metal-seated valve to provide an acceptable shut-off, the sealing surfaces need to have a high degree of flatness with a very good surface finish. The disc must articulate on the stem and the stem guide must not cause any undue frictional effects. Typical figures required for an acceptable shut-off for a metal seated valve are 0.5 µm for surface finish and two optical light bands for flatness. In addition, for a reasonable service life, the mating and sealing surfaces must have a high wear resistance.

Unlike ordinary isolation valves, the net closing force acting on the disc is relatively small, due to there being only a small difference between the system pressure acting on the disc and the spring force opposing it.

Resilient or elastomer seals incorporated into the valve discs are often used to improve shut-off, where system conditions permit. It should be noted, however, that a soft seal is often more susceptible to damage than a metal seat.

Safety valve installation
Seat damage can often occur when a valve is first lifted as part of the general plant commissioning procedure, because very often, dirt and debris are present in the system. To ensure that foreign matter does not pass through the valve, the system should be flushed out before the safety valve is installed and the valve must be mounted where dirt, scale and debris cannot collect.

It is also important on steam applications to reduce the propensity for leakage by installing the valve so that condensate cannot collect on the upstream side of the disc. This can be achieved by installing the safety valve above the steam pipe as shown in Figure 9.5.1.

Fig. 9.5.1 Correct position of a safety valve on a steam system
Where safety valves are installed below the pipe, steam will condense, fill the pipe and wet the upstream side of the safety valve seat. This type of installation is not recommended but is shown in Figure 9.5.2 for reference purposes.

![Incorrect position of a safety valve on a steam system](image1)

**Fig. 9.5.2**
**Incorrect position of a safety valve on a steam system**

Also, it is essential at all times to ensure that the downstream pipework is well drained so that downstream flooding (which can also encourage corrosion and leakage) cannot occur, as shown in Figure 9.5.3.

![Correct installation of a safety valve on a steam system](image2)

**Fig. 9.5.3**
**Correct installation of a safety valve on a steam system**

**Operation of the safety valve**
Leakage can also be experienced when there is dirt or scale sitting on the seating face. This usually occurs during the periodic lifting demanded by insurance companies and routine maintenance programs. Further lifting of the lever will generally clear any dirt that may be on the seating face.

The vast majority of safety valve seat leakage problems occur after initial manufacture and test. These problems typically result from damage during transit, and sometimes as a result of misuse and contamination, or because of poor installation.

Most safety valve standards do not include detailed shut-off parameters. For those that do, the requirements and recommended test procedures are usually based on the API 527 standard, which is commonly used throughout the safety valve industry.
The procedure for testing valves that have been set on air involves blocking all secondary leakage paths, whilst maintaining the valve at 90% of the set pressure on air (see Figure 9.5.4). The outlet of the safety valve is connected to a 6 mm internal diameter pipe, the end of which is held 12.7 mm below the surface of water contained in a suitable, transparent vessel. The number of bubbles discharged from this tube per minute is measured. For the majority of valves set below 70 bar g, the acceptance criteria is 20 bubbles per minute.

![Fig. 9.5.4 Apparatus to test seat tightness with air](image)

For valves set on steam or water, the leakage rate should be assessed using the corresponding setting media. For steam, there must be no visible leakage observed against a black background for one minute after a three-minute stabilisation period. In the case of water, there is a small leakage allowance, dependent on the orifice area, of 10 ml per hour per inch of the nominal inlet diameter.

The above procedure can be time consuming, so it is quite common for manufacturers to employ a test using alternative methods, for example, using accurate flow measuring equipment that is calibrated against the parameters set in API 527.

Under no circumstances should any additional load be applied to the easing lever nor should the valve be gagged in order to increase the seat tightness. This will affect the operating characteristics and can result in the safety valve failing to lift in overpressure conditions. If there is an unacceptable level of seat leakage, the valve can be refurbished or repaired, but only by authorised personnel, working with the approval of the manufacturer, and using information supplied by the manufacturer.

Commonly supplied spare parts typically include springs, discs and nozzles, resilient seals and gaskets. Many valves have seat rings which are not removable and these can sometimes be re-profiled and re-lapped in the body. However, it is important that the size of seat orifice is maintained exactly in line with the original drawings since this can alter the effective area and, subsequently affect the set pressure.

It is unacceptable for the disc to be lapped directly onto the seat in the body, since a groove will be created on the disc preventing a consistent shut-off after lifting.

In the case of resilient seal valves usually the seal (which is normally an ‘O’ ring or disc) can be changed in the disc assembly.

If Independent Authority Approval is to be maintained then it is mandatory that the repairer is acting as the manufacturer’s approved agent. For ASME approved valves, the repairer must be independently approved by the National Board and is subsequently allowed to apply a ‘VR’ stamp, which indicates a valve has been repaired.
Marking

Safety valve standards are normally very specific about the information which must be carried on the valve. Marking is mandatory on both the shell, usually cast or stamped, and the nameplate, which must be securely attached to the valve. A general summary of the information required is listed below:

On the shell:
• Size designation.
• Material designation of the shell.
• Manufacturer’s name or trademark.
• Direction of flow arrow.

On the identification plate:
• Set pressure (in bar g for European valves and psi g for ASME valves).
• Number of the relevant standard (or relevant ASME stamp).
• Manufacturer’s model type reference.
• Derated coefficient of discharge or certified capacity.
• Flow area.
• Lift and overpressure.
• Date of manufacture or reference number.

National Board approved ASME stamps are applied as follows:
V ASME I approved safety relief valves.
UV ASME VIII approved safety relief valves.
UD ASME VIII approved rupture disc devices.
NV ASME III approved pressure relief valves.
VR Authorised repairer of pressure relief valves.

Table 9.5.1 details the marking system required by TÜV and Table 9.5.2 details the fluid reference letters.

Table 9.5.1
Marking system used for valves approved by TÜV to AD-Merkblatt A2, DIN 3320 and TRD 421

<table>
<thead>
<tr>
<th>Marking system</th>
<th>TÜV</th>
<th>SV</th>
<th>98</th>
<th>XXX</th>
<th>XX</th>
<th>DGF</th>
<th>0.XX</th>
<th>X</th>
</tr>
</thead>
<tbody>
<tr>
<td>TÜV</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Safety valve</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Year of test</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Test number</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Minimum flow diameter (do)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fluid identification character (see Table 9.5.2, below)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Flow coefficient or flow</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Set pressure (bar g for European valves and psi g for ASME valves)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The $K_{dr}$ or $\alpha_w$ value can vary according to the relevant fluid and is either suffixed or prefixed by the identification letter shown in Table 9.5.2.

Table 9.5.2 Fluid types defined as steam, gas or liquid

<table>
<thead>
<tr>
<th>For $\alpha_w$</th>
<th>For $K_{dr}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>D (dampf) for steam</td>
<td>S for steam</td>
</tr>
<tr>
<td>G (gase) for gas</td>
<td>G for gas</td>
</tr>
<tr>
<td>F (flüssigkeiten) for liquids</td>
<td>L for liquids</td>
</tr>
</tbody>
</table>
**Installation**

Safety valves are precision items of safety equipment; they are set to close tolerances and have accurately machined internal parts. They are susceptible to misalignment and damage if mishandled or incorrectly installed.

Valves should be transported upright if possible and they should never be carried or lifted by the easing lever. In addition, the protective plugs and flange protectors should not be removed until actual installation. Care should also be taken during movement of the valve to avoid subjecting it to excessive shock as this can result in considerable internal damage or misalignment.

**Inlet pipework**

When designing the inlet pipework, one of the main considerations is to ensure that the pressure drop in this pipework is minimised. EN ISO 4126 recommends that the pressure drop be kept below 3% of the set pressure when discharging. Where safety valves are connected using short ‘stub’ connections, inlet pipework must be at least the same size as the safety valve inlet connection. For larger lines or any line incorporating bends or elbows, the branch connection should be at least two pipe sizes larger than the safety valve inlet connection, at which point it is reduced in size to the safety valve inlet size (see Figure 9.5.5a). Excessive pressure loss can lead to ‘chatter’, which may result in reduced capacity and damage to the seating faces and other parts of the valve. In order to reduce the pressure loss in the inlet, the following methods can be adopted:

- Increase the diameter of the pipe. (see Figure 9.5.5 (a)).
- Ensure that any corners are suitably rounded. The standard EN ISO 4126: Part 1 recommends that corners should have a radius of not less than one quarter of the bore (see Figure 9.5.5 (b)).
- Reduce the inlet pipe length.
- Install the valve at least 8 to 10 pipe diameters downstream from any converging or diverging ‘Y’ fitting, or any bend (see Figure 9.5.5 (c)).
- Never install the safety valve branch directly opposite a branch on the lower side of the steam line.
- Avoid take-off branches (such as for other processes) in the inlet piping, as this will increase the pressure drop.

![Fig. 9.5.5 Correct installations of safety valves](image)

Safety valves should always be installed with the bonnet vertically upwards. Installing the valve in any other orientation can affect the performance characteristics.

The API Recommended Practice 520 guidelines also state that the safety valve should not be installed at the end of a long horizontal pipe that does not normally have flow through it. This can lead to the accumulation of foreign material or condensate in the pipe, which may cause unnecessary damage to the valve, or interfere with its operation.
Outlet pipework

There are two possible types of discharge system – open and closed. An open system discharges directly into the atmosphere whereas a closed system discharges into a manifold along with other safety valves.

It is recommended that discharge pipework should rise for steam and gas systems, whereas for liquids, it should fall. Horizontal pipework should have a downward gradient of at least 1 in 100 away from the valve ensuring that any discharge will be self-draining. It is important to drain any rising discharge pipework. Vertical rises will require separate drainage. Note: all points of system drainage are subject to the same precautions, notably that valve performance must not be affected, and any fluid must be discharged to a safe location.

It is essential to ensure that fluid cannot collect on the downstream side of a safety valve, as this will impair its performance and cause corrosion of the spring and internal parts. Many safety valves are provided with a body drain connection, if this is not used or not provided, then a small bore drain should be fitted in close proximity to the valve outlet (see Figure 9.5.3).

One of the main concerns in closed systems is the pressure drop or built-up backpressure in the discharge system. As mentioned in Module 9.2, this can drastically affect the performance of a safety valve. The EN ISO 4126: Part 1 standard states that the pressure drop should be maintained below 10% of the set pressure. In order to achieve this, the discharge pipe can be sized using Equation 9.5.1.

\[
d = \sqrt[5]{\frac{L_e \cdot \dot{m}^2 \left( \frac{v_g + 2}{2} \right)}{0.08 \cdot P}}
\]

Where:
- \(d\) = Pipe diameter (mm)
- \(L_e\) = Equivalent length of pipe (m)
- \(\dot{m}\) = Discharge capacity (kg/h)
- \(P\) = Safety valve set pressure (bar g) x 0.1
- \(v_g\) = Specific volume of steam at the pressure (P) (m³/kg)

The pressure (P) should be taken as the maximum allowable pressure drop according to the relevant standard. In the case of EN ISO 4126: Part 1, this would be 10% of the set pressure and it is at this pressure \(v_g\) is taken.

Example 9.5.1

Calculate the nominal diameter of the discharge pipework for a safety valve required to discharge 1,000 kg/h of saturated steam; given that the steam is to be discharged into a vented tank via the pipework, which has an equivalent length of 25 m. The set pressure of the safety valve is 10 bar g and the acceptable backpressure is 10% of the set pressure. (Assume zero pressure drop along the tank vent).

Answer: If the maximum 10% backpressure is allowed, then the gauge pressure at the safety valve outlet will be:

\[
\frac{10}{100} \times 10 \text{ bar g} = 1.0 \text{ bar g}
\]

Using saturated steam tables, the specific volume at this pressure is, \(v_g = 0.88 \text{ m}^3/\text{kg}\).

Applying Equation 9.5.1:

\[
d = \sqrt[5]{\frac{25 \times 1000^2 \times \left( \frac{0.88 + 2}{2} \right)}{0.08 \times 1.0}} = 54 \text{ mm}
\]

Therefore, the pipework connected to the outlet of the safety valve should have an internal diameter of at least 54 mm. With schedule 40 pipe, this would require a DN65 pipe.
If it is not possible to reduce the backpressure to below 10% of the set pressure, a balanced safety valve should be used.

Balanced safety valves require that their bonnets be vented to atmosphere. In the case of the balanced bellows type, there will be no discharge of the process fluid, so they can be vented directly to the atmosphere. The main design consideration is to ensure that this vent will not become blocked, for example, by foreign material or ice. With the balanced piston type, consideration must be given to the fact that process fluid may be discharged through the bonnet vent. If discharging to a pressurised system, the vent has to be suitably sized, so that no backpressure exists above the piston.

Safety valves that are installed outside of a building for discharge directly into the atmosphere should be covered using a hood. The hood allows the discharge of the fluid, but prevents the build up of dirt and other debris in the discharge pipework, which could affect the backpressure. The hood should also be designed so that it too does not affect the backpressure.

**Manifolds**

Manifolds must be sized so that in the worst case (i.e. when all the manifold valves are discharging), the pipework is large enough to cope without generating unacceptable levels of backpressure. The volume of the manifold should ideally be increased as each valve outlet enters it, and these connections should enter the manifold at an angle of no greater than 45° to the direction of flow (see Figure 9.5.6). The manifold must also be properly secured and drained where necessary.

For steam applications, it is generally not recommended to use manifolds, but they can be utilised if proper consideration is given to all aspects of the design and installation.

![Fig. 9.5.6 A typical manifold discharge system](image)

**Reaction forces when discharging**

In open systems, careful consideration must be given to the effects of the reaction forces generated in the discharge system when the valve lifts. In these systems, there will be significant resultant force acting in the opposite direction to that of discharge. It is important to prevent excessive loads being imposed on the valve or the inlet connection by these reaction forces, as they can cause damage to the inlet pipework. The magnitude of the reaction forces can be calculated using the formula in Equation 9.5.2:

**Equation 9.5.2**

\[
F = 129 \dot{m} \sqrt{\frac{kT}{(k + 1)M}} + 0.1AP
\]

Where:

- \( F \) = Reaction force at the point of discharge to atmosphere (newtons)
- \( \dot{m} \) = Discharge mass flowrate (kg/s)
- \( k \) = Isentropic coefficient of the fluid - From steam tables
- \( T \) = Fluid temperature (K)
- \( M \) = Molar mass of the fluid (kg/kmol)

**Note:** For saturated and superheated steam, \( M = 18.015 \)
- \( A \) = Area of the outlet at the point of discharge (mm²)
- \( P \) = Static pressure at the outlet at the point of discharge (bar g)
The reaction forces are typically small for safety valves with a nominal diameter of less than 75 mm, but safety valves larger than this usually have mounting flanges for a reaction bar on the body to allow the valve to be secured.

These reaction forces are typically negligible in closed systems, and they can therefore be ignored.

Regardless of the magnitude of the reaction forces, the safety valve itself should never be relied upon to support the discharge pipework itself and a support should be provided to resist the weight of the discharge pipework. This support should be located as close as possible to the centreline of the vent pipe (see Figure 9.5.7).

Figures 9.5.8 and 9.5.9 show typical safety valve installations for both open and closed systems.

**Fig. 9.5.8**
A typical safety valve installation with open discharge system

- Pressure relief valve
- Body drain
- Low point small bore drain
- Non-recoverable losses not more than 3% of the set pressure
- Long radius elbow
- Support to resist weight and reaction forces
- Nominal pipe diameter no less than valve inlet size
- Non-recoverable losses along the discharge pipe not more than 12% of the set pressure
- Note: A weather cap may be required

**Fig. 9.5.9**
A typical safety valve installation with closed discharge system

- Bonnet vent piping for bellows type pressure relief valves, if required
- Flanged spool piece, if required to elevate PRV
- Nominal pipe diameter no less than valve inlet size
- To closed system (self-draining)
- Non-recoverable pressure losses not more than 3% of the set pressure
**Changeover valves**

Changeover valves (see Figure 9.5.10) permit two valves to be mounted side by side, with one in service and one isolated. This means regular maintenance can be carried out without interruption of service or the vessel being protected. Changeover valves are designed in such a way that when they are operated, the pass area is never restricted.

Changeover valves can also be used to connect safety valve outlets so that the discharge pipework does not have to be duplicated. The action of both inlet and outlet changeover valves has to be limited and synchronised for safety reasons. This is usually by means of a chain drive system linking both handwheels.

Consideration must be made to pressure loss caused by the changeover valve when establishing the safety valve inlet pressure drop, which should be limited to 3% of the set pressure.

![Fig. 9.5.10 Changeover valve](image)

**Noise emission**

Although discharge from a safety valve should not occur frequently, should it occur, the noise generated can often be significant. It is therefore necessary to determine the sound level of safety valves to ensure that relevant health and safety regulation levels are not exceeded.

Assuming a sonic flow nozzle discharge, an approximate value of the sound level, LP, in decibels at a flange outlet can be calculated using the formula given in Equation 9.5.3 (Source API 521).

**Equation 9.5.3**

\[
L_{30} = L_p + 10 \log \left( \frac{1}{2} m C^2 \right)
\]

Where:
- \(L_{30}\) = Sound pressure level at 30 m (dB)
- \(L_p\) = Sound pressure level, obtained from API 521, Figure 18
- \(m\) = Mass flowrate (kg/s)
- \(C\) = Speed of sound in outlet (m/s)

**Equation 9.5.4**

\[
L_r = L_{30} - 2 \log (\frac{r}{30})
\]

Where:
- \(L_r\) = Sound pressure level radius \(r\) metres (dB)
- \(L_{30}\) = Sound pressure level radius 30 metres (dB)
- \(r\) = Distance from vent outlet (m)

There are several ways to reduce noise level, the simplest being to use larger diameter discharge pipes, or to lag the discharge pipe (however, the valve must not be lagged). It is also permissible for a silencer to be used in extreme cases, in which case any backpressure generated must then be taken into account.
Alternative Plant Protection Devices and Terminology
Alternative Methods of Plant Protection

Although safety valves are by far the most common devices used for plant protection in steam systems, there are several other devices available to protect plant from overpressure conditions. Whilst some of them can be used in place of a safety valve, most have their own unique applications and indeed some devices, such as the bursting disc, may be used to complement the safety valve.

- **Weighted pallet** - This is the simplest type of overpressure protection device, and it is on low-pressure tanks and condensers, for pressure relief, vacuum relief or both. A weight is applied to the top of a disc, keeping it closed until the pressure acting on the underside of the pallet equals the weight. Due to the large weights required to keep a pallet closed, this type of valve is designed for low pressure applications below 0.1 bar. For higher set pressures, the weight required would be prohibitive and dangerous if oscillation of the pallet occurred at valve opening.

- **Counterweight safety valve** - Although these have been largely superseded by spring-loaded safety valves, they are still sometimes used for low-pressure applications. The closing force of the safety valve is provided by a weight rather than a spring. As the closing force is provided by a weight, it will remain constant and once the set pressure is reached, the safety valve will open fully.

![Fig. 9.6.1  A counterweight safety valve](image)

- **Supplementary loaded safety valve** - A supplementary loaded safety valve consists of a conventional safety valve provided with an additional sealing force that is released once the set pressure is reached. One of the main concerns with this type of device is ensuring that the load is suitably released when the set pressure is reached. The EN ISO 4126 standard states that even in the event of the release mechanism failing, the valve must attain its certified discharge capacity within 115% of the set pressure. Supplementary loaded safety valves tend only to be used where any leakage of the fluid below set pressure is unacceptable, or on very high pressure systems where maintaining a tight shut-off is otherwise difficult.

- **Controlled safety pressure relief systems (CSPRS)** - These are electric or electropneumatic systems, which are not self-acting. When an overpressure situation is detected, a control device acts to correct the situation.

![Fig. 9.6.2  Typical supplementary loaded safety valves](image)
Non-reclosing pressure relief devices

Non-reclosing devices are those which are designed to remain open after operation. A manual means of resetting is usually provided.

- **Bursting or rupture discs** - This consists of an elastomeric membrane or thin metal disk that will burst at a set pressure, relieving any overpressure. Although they can be used by themselves, on many applications, they are used in conjunction with a safety valve.

A rupture disc can be installed either on the inlet or outlet side of the safety valve. If installed on the inlet, it isolates the contained media from the safety valve. When there is an overpressure situation; the rupture disc bursts allowing the fluid to flow into the safety valve, which will then subsequently lift. This arrangement is used to protect the internals of the safety valve from corrosive fluids.

Alternatively, if the safety valve discharges into a manifold containing corrosive media, a rupture disc can be installed on the safety valve outlet, preventing any of the fluid from the manifold contacting the internals of the safety valve in normal use.

Rupture discs can also be installed alongside a safety valve as a secondary relief device.

Rupture discs are leak tight and low cost, but they require replacing after each operation. Most rupture disc installations contain a mechanism to indicate when the disc has ruptured and that it needs to be replaced. Typically, a pressure gauge is used (see Figure 9.6.3b).

Explosion panels or explosion rupture discs are similar to rupture discs but are designed for use at higher rates of pressure rise, and for larger capacities.

- **Fusible plug devices** - These consist of a plug with a lower melting point than the maximum operating temperature of the system that it is to protect. In old steam locomotives, this type of device was used to dump the boiler water onto the fire if overtemperature occurred.

- **Breaking or shear pin devices** - A breaking pin device is a non-reclosing pressure relief device actuated by inlet static pressure and designed to function by the breakage of a load carrying section of a pin, which supports a pressure-containing member. The force of overpressure forces the pin to buckle and the valve to open. The valve can then be reseated after the pressure is removed and a new pin can be installed. These devices are usually installed on low-pressure applications and large gas distribution systems. They have limited process applications.

![Fig. 9.6.3](image)
A rupture/bursting disc device (a) and a rupture disk installed on the inlet of a safety valve (b)

![Fig. 9.6.4](image)
An example of a fusible plug device
**Terminology**

The following definitions are taken from DIN 3320 but it should be noted that many of the terms and associated definitions used are universal and appear in many other standards. Where commonly used terms are not defined in DIN 3320 then ASME/ANSI PTC25.3 has been used as the source of reference. This list is not exhaustive and is intended as a guide only; it should not be used in place of the relevant current issue standard:

**Operating pressure (working pressure)** is the gauge pressure existing at normal operating conditions within the system to be protected.

**Set pressure** is the gauge pressure at which under operating conditions direct loaded safety valves commence to lift.

**Test pressure** is the gauge pressure at which under test stand conditions (atmospheric backpressure) direct loaded safety valves commence to lift.

**Opening pressure** is the gauge pressure at which the lift is sufficient to discharge the predetermined flowing capacity. It is equal to the set pressure plus opening pressure difference.

**Reseating pressure** is the gauge pressure at which the direct loaded safety valve is re-closed.

**Built-up backpressure** is the gauge pressure built up at the outlet side by blowing.

**Superimposed backpressure** is the gauge pressure on the outlet side of the closed valve.

**Backpressure** is the gauge pressure built up on the outlet side during blowing (built-up backpressure + superimposed backpressure).

**Accumulation** is the increase in pressure over the maximum allowable working gauge pressure of the system to be protected.

**Opening pressure difference** is the pressure rise over the set pressure necessary for a lift suitable to permit the predetermined flowing capacity.

**Reseating pressure difference** is the difference between set pressure and reseating pressure.

**Functional pressure difference** is the sum of opening pressure difference and reseating pressure difference.

**Operating pressure difference** is the pressure difference between set pressure and operating pressure.

**Lift** is the travel of the disc away from the closed position.

**Commencement of lift (opening)** is the first measurable movement of the disc or the perception of discharge noise.

**Flow area** is the cross sectional area upstream or downstream of the body seat calculated from the minimum diameter which is used to calculate the flow capacity without any deduction for obstructions.

**Flow diameter** is the minimum geometrical diameter upstream or downstream of the body seat.

**Nominal size designation** of a safety valve is the nominal size of the inlet.

**Theoretical flowing capacity** is the calculated mass flow from an orifice having a cross sectional area equal to the flow area of the safety valve without regard to flow losses of the valve.

**Actual flowing capacity** is the flowing capacity determined by measurement.

**Certified flowing capacity** is actual flowing capacity reduced by 10%.

**Coefficient of discharge** is the ratio of actual to the theoretical discharge capacity.

**Certified coefficient of discharge** is the coefficient of discharge reduced by 10% (also known as derated coefficient of discharge).
The following terms are not defined in DIN 3320 and are taken from ASME/ANSI PTC25.3:

**Blowdown (reseating pressure difference)** - difference between actual popping pressure and actual reseating pressure, usually expressed as a percentage of set pressure or in pressure units.

**Cold differential test pressure** - the pressure at which a valve is set on a test rig using a test fluid at ambient temperature. This test pressure includes corrections for service conditions e.g. backpressure or high temperatures.

**Flow rating pressure** - the inlet static pressure at which the relieving capacity of a pressure relief device is measured.

**Leak test pressure** - the specified inlet static pressure at which a quantitative seat leakage test is performed in accordance with a standard procedure.

**Measured relieving capacity** - the relieving capacity of a pressure relief device measured at the flow rating pressure.

**Rated relieving capacity** - that portion of the measured relieving capacity permitted by the applicable code or regulation to be used as a basis for the application of a pressure relieving device.

**Overpressure** - a pressure increase over the set pressure of a pressure relief valve, usually expressed as a percentage of set pressure.

**Popping pressure** - the value of increasing static inlet pressure of a pressure relief valve at which there is a measurable lift, or at which the discharge becomes continuous as determined by seeing, feeling or hearing.

**Relieving pressure** - set pressure plus overpressure.

**Simmer** - the pressure zone between the set pressure and popping pressure.

**Maximum operating pressure** - the maximum pressure expected during system operation.

**Maximum allowable working pressure (MAWP)** - the maximum gauge pressure permissible at the top of a completed vessel in its operating position for a designated temperature.

**Maximum allowable accumulated pressure (MAAP)** - the maximum allowable working pressure plus the accumulation as established by reference to the applicable codes for operating or fire contingencies.