Sizing, Selection, and Installation of Pressure-relieving Devices ~~in Refineries~~

Part I—Sizing and Selection

API STANDARD 520

~~EIGHTH~~ NINTH EDITION, DECEMBER ~~2008~~2013

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~~**Downstream Segment**~~

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Foreword

API Standard 520, *Sizing, Selection, and Installation of Pressure-relieving Devices ~~in Refineries~~*, is the result of several years' work by engineers in the petroleum industry. The information in this standard is intended to supplement the information contained in Section VIII—*Pressure Vessels*, of the ASME *Boiler and Pressure Vessel Code*. The recommendations presented in this publication are not intended to supersede applicable laws and regulations.

Users of this standard are reminded that no publication of this type can be complete, nor can any written document be substituted for qualified engineering analysis.

Shall: As used in a standard, “shall” denotes a minimum requirement in order to conform to the specification.

Should: As used in a standard, “should” denotes a recommendation or that which is advised but not required in order to conform to the specification.

The current edition of this standard, published in two parts, has been updated with respect to the practices generally used in the installation of all devices covered in the previous editions; the current edition also contains additional information based on revisions suggested by many individuals and several organizations.

The 1st Edition of this standard was initially released as a recommended practice in 1955. The 2nd Edition was published in two parts: Part I, *Design*, in 1960 and Part II, *Installation*, in 1963. The 3rd Edition of Part I was issued in November 1967 and reaffirmed in 1973. The 4th edition was issued in December 1976, the 5th Edition was issued in July 1990, the 6th Edition was issued in March 1993, ~~and~~ the 7th Edition was issued in January 2000 and the 8th Edition was issued in December 2008.

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Suggested revisions are invited and should be submitted to the Downstream Segment, API, 1220 L Street, NW, Washington, D.C. 20005, standards@api.org.

Sizing, Selection, and Installation of Pressure-relieving Devices in Refineries Part I—Sizing and Selection

1 Scope

This standard applies to the sizing and selection of pressure relief devices used in refineries, chemical facilities and related industries for equipment that has a maximum allowable working pressure of 15 psig (103 kPag) or greater. The pressure relief devices covered in this standard are intended to protect ~~unfired~~ pressure vessels and related equipment against overpressure from operating and fire contingencies.

This standard includes basic definitions and information about the operational characteristics and applications of various pressure relief devices. It also includes sizing procedures and methods based on steady state flow of Newtonian fluids.

Pressure relief devices protect a vessel against overpressure only; they do not protect against structural failure when the vessel is exposed to extremely high temperatures such as during a fire. See API Std 521 for information about appropriate ways of reducing pressure and restricting heat input.

Atmospheric and low-pressure storage tanks covered in API Std 2000 and pressure vessels used for the transportation of products in bulk or shipping containers are not within the scope of this standard.

The rules for overpressure protection of fired vessels are provided in ASME Section I~~ and ASME B31.1~~, and are not within the scope of this standard.

2 Normative References

The following referenced documents are cited in this document for informational purposes. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

API Std~~RP~~ 520, *Sizing, Selection, and Installation of Pressure-relieving Devices~~ in Refineries~~, Part II—Installation*

API Std 521~~/ISO 23251~~, *Guide for Pressure-relieving and Depressuring Systems*

API Std 526, *Flanged Steel Pressure Relief Valves*

API Std 527, *Seat Tightness of Pressure Relief Valves*

API Std 2000, *Venting Atmospheric and Low-pressure Storage Tanks: Nonrefrigerated and Refrigerated*

ASME *Boiler and Pressure Vessel Code* [1], *Section I—Power Boilers*

ASME *Boiler and Pressure Vessel Code, Section VIII—Pressure Vessels, Division 1*

~~ASME *BPVC* Code Case 2091-3~~ ~~[2]~~, ~~*Nonreclosing Pin Pressure Relief Devices*~~

ASME *BPVC* Code Case 2203 [3], *Omission of Lifting Device Requirements for Pressure Relief Valves on Air, Water Over 140°F, or Steam Service*

~~ASME *BPVC* Code Case 2487, *Breaking Pin Pressure Relief Devices*~~

[1] ASME International, 3 Park Avenue, New York, New York 10016, www.asme.org.

~~[2] Code Cases are temporary in nature and may not be acceptable in all jurisdictions. The user should verify the current applicability of the referenced Code Cases.~~

[3] Code Cases are temporary in nature and may not be acceptable in all jurisdictions. The user should verify the current applicability of the referenced Code Cases.

ASME B31.1, *Power Piping*

ASME B31.~~1~~3, *Process Piping*

ASME PTC 25, *Pressure Relief Devices*

3 Terms and Definitions

For the purposes of this document, the following definitions apply. Many of the terms and definitions are taken from ASME PTC 25.

3.1
accumulation
The pressure increase over the maximum allowable working pressure of the vessel, expressed in pressure units or as a percentage of maximum allowable working pressure (MAWP) or design pressure. Maximum allowable accumulations are established by applicable codes for emergency operating and fire contingencies.

3.2
actual discharge area
actual orifice area
The area of a pressure relief valve (PRV) is the minimum net area that determines the flow through a valve.

3.3
backpressure
The pressure that exists at the outlet of a pressure relief device as a result of the pressure in the discharge system. Backpressure is the sum of the superimposed and built-up backpressures.

3.4
balanced pressure relief valve
A spring-loaded pressure relief valve that incorporates a bellows or other means for minimizing the effect of backpressure on the operational characteristics of the valve.

3.5
blowdown
The difference between the set pressure and the closing pressure of a pressure relief valve, expressed as a percentage of the set pressure or in pressure units.

3.6
bore area
nozzle area
nozzle throat area
throat area
The minimum cross-sectional flow area of a nozzle in a pressure relief valve.

3.7
built-up backpressure
The increase in pressure at the outlet of a pressure relief device that develops as a result of flow after the pressure relief device opens.

3.8
burst pressure
The value of the upstream static pressure minus the value of the downstream static pressure just prior to when the disk bursts. When the downstream pressure is atmospheric, the burst pressure is the upstream static gauge pressure.

3.9
burst pressure tolerance
The variation around the marked burst pressure at the specified disk temperature in which a rupture disk shall burst.

3.10
capacity
The rated capacity of steam, air, gas or water as required by the applicable code.

3.11
chatter
Chattering is where the PRV opens and closes at a very high frequency (on the order of the natural frequency of the valve's spring mass system). Spring loaded PRVs are mass-spring devices and consequently are susceptible to dynamic interaction with the system.

3.11
closing pressure
The value of decreasing inlet static pressure at which the valve disc reestablishes contact with the seat or at which lift becomes zero as determined by seeing, feeling or hearing.

3.12
coefficient of discharge
The ratio of the mass flow rate in a valve to that of an ideal nozzle. The coefficient of discharge is used for calculating flow through a pressure relief device.

3.13
cold differential test pressure
The pressure at which a pressure relief valve is adjusted to open on the test stand. The cold differential test pressure includes corrections for the service conditions of backpressure or temperature or both.

3.14
conventional pressure relief valve
A spring-loaded pressure relief valve whose operational characteristics are directly affected by changes in the backpressure.

3.15
curtain area
The area of the cylindrical or conical discharge opening between the seating surfaces above the nozzle seat created by the lift of the disc.

3.16
Cycling
Cycling is the relatively low frequency (a few cycles per second) opening and closing of a relief valve.

3.16
design pressure
Pressure, together with the design temperature, used to determine the minimum permissible thickness or physical characteristic of each vessel component as determined by the vessel design rules. The design pressure is selected by the user to provide a suitable margin above the most severe pressure expected during normal operation at a coincident temperature. It is the pressure specified on the purchase order. This pressure may be used in place of the maximum allowable working pressure (MAWP) in all cases where the MAWP has not been established. The design pressure is equal to or less than the MAWP.

3.17
effective coefficient of discharge
A nominal value used with an effective discharge area to calculate the relieving capacity of a pressure relief valve per the preliminary sizing equations.

3.18
effective discharge area
effective orifice area
A nominal area used with an effective discharge coefficient to calculate the relieving capacity of a pressure relief valve per the preliminary sizing equations. API 526 provides effective discharge areas for a range of sizes in terms of letter designations, "D" through "T."

3.19
Flutter
Fluttering is where the PRV is open but the dynamics of the system cause abnormal, rapid reciprocating motion of the moveable parts of the PRV. During the fluttering, the disk does not contact the seat but reciprocates at the frequency of the flutter.

3.19
huddling chamber
An annular chamber located downstream of the seat of a pressure relief valve for the purpose of assisting the valve to achieve lift.

3.20
inlet size
The nominal pipe size (NPS) of the device at the inlet connection, unless otherwise designated.

3.21
leak-test pressure
The specified inlet static pressure at which a seat leak test is performed.

3.22
lift
The actual travel of the disc from the closed position when a valve is relieving.

3.23
lot of rupture disks
Disks manufactured at the same time and of the same size, material, thickness, type, heat and manufacturing process, including heat treatment.

3.24
manufacturing design range
The pressure range in which the rupture disk shall be marked. Manufacturing design ranges are usually catalogued by the manufacturer as a percentage of the specified burst pressure. Catalogued manufacturing design ranges may be modified by agreement between the user and the manufacturer.

3.25
marked burst pressure
rated burst pressure
The burst pressure established by tests for the specified temperature and marked on the disk tag by the manufacturer. The marked burst pressure may be any pressure within the manufacturing design range unless otherwise specified by the customer. The marked burst pressure is applied to all of the rupture disks of the same lot.

3.26
maximum allowable working pressure
MAWP
The maximum gauge pressure permissible at the top of a completed vessel in its normal operating position at the designated coincident temperature specified for that pressure. The pressure is the least of the values for the internal or external pressure as determined by the vessel design rules for each element of the vessel using actual nominal thickness, exclusive of additional metal thickness allowed for corrosion and loadings other than pressure. The maximum allowable working pressure is the basis for the pressure setting of the pressure relief devices that protect the vessel. The MAWP is normally greater than the design pressure but can be equal to the design pressure when the

design rules are used only to calculate the minimum thickness for each element and calculations are not made to determine the value of the MAWP.

3.27
maximum operating pressure
The maximum pressure expected during normal system operation.

3.28
minimum net flow area
The calculated net area after a complete burst of a rupture disk with appropriate allowance for any structural members which may reduce the net flow area through the rupture disk device.

3.29
modulating pressure relief device
a pressure relieving valve that its lift adjusts to flow a varying relief load within the range of pressure from closing pressure to the specified overpressure with the minimum relief load that can be flowed specified by the Manufacturer of the pressure relief valve

3.29
non-fragmenting rupture disk
A rupture disk designed and manufactured to be installed upstream of other piping components. Non-fragmenting rupture disks do not impair the function of pressure relief valves when the disk ruptures.

3.30
non-reclosing pressure relief device
A pressure relief device which remains open after operation. A manual resetting means may be provided.

3.31
opening pressure
The value of increasing inlet static pressure at which there is a measurable lift of the disc or at which discharge of the fluid becomes continuous, as determined by seeing, feeling, or hearing.

3.32
operating ratio of a pressure relief valve
The ratio of maximum system operating pressure to the set pressure.

3.33
operating ratio of a rupture disk
The ratio of the maximum system operating pressure to a pressure associated with a rupture disk (see Figure 26 and Figure 28). For marked burst pressures above 40 psi, the operating ratio is the ratio of maximum system operating pressure to the disk marked burst pressure. For marked burst pressures between 15 psi and 40 psi, the operating ratio is the ratio of maximum system operating pressure to the marked burst pressure minus 2 psi. For marked burst pressures less than 15 psi, the operating ratio should be determined by consulting the manufacturer.

3.34
outlet size
The nominal pipe size (NPS) of the device at the discharge connection, unless otherwise designated.

3.35
overpressure
The pressure increase over the set pressure of the relieving device. Overpressure is expressed in pressure units or as a percentage of set pressure. Overpressure is the same as accumulation only when the relieving device is set to open at the maximum allowable working pressure of the vessel.

3.36
pilot-operated pressure relief valve
A pressure relief valve in which the major relieving device or main valve is combined with and controlled by a self actuated auxiliary pressure relief valve (pilot).

3.37
pin-actuated device
A non-reclosing pressure relief device actuated by static pressure and designed to function by buckling or breaking a pin which holds a piston or a plug in place. Upon buckling or breaking of the pin, the piston or plug instantly moves to the full open position.

3.38
pressure relief device
PRD
A device actuated by inlet static pressure and designed to open during emergency or abnormal conditions to prevent a rise of internal fluid pressure in excess of a specified design value. The device also may be designed to prevent excessive internal vacuum. The device may be a pressure relief valve, a non-reclosing pressure relief device, or a vacuum relief valve.

3.39
pressure relief valve
PRV
A pressure relief device designed to open and relieve excess pressure and to reclose and prevent the further flow of fluid after normal conditions have been restored.

3.40
rated coefficient of discharge
A value used with the actual discharge area to calculate the rated flow capacity of a pressure relief valve. The rated coefficient of discharge of a pressure relief valve is determined in accordance with the applicable code or regulation.

3.41
rated relieving capacity
The basis for the application of a pressure relief device. This capacity is determined in accordance with the applicable code or regulation and is provided by the manufacturer.

3.42
relief valve
A spring-loaded pressure relief valve actuated by the static pressure upstream of the valve. The valve opens normally in proportion to the pressure increase over the opening pressure. A relief valve is used primarily with incompressible fluids.

3.43
relieving conditions
The inlet pressure and temperature on a pressure relief device during an overpressure condition. The relieving pressure is equal to the valve set pressure (or rupture disk burst pressure) plus the overpressure. The temperature of the flowing fluid at relieving conditions may be higher or lower than the operating temperature.

3.44
rupture disk
A pressure containing, pressure and temperature sensitive element of a rupture disk device.

3.45
rupture disk device
A non-reclosing pressure relief device actuated by static differential pressure between the inlet and outlet of the device and designed to function by the bursting of a rupture disk. A rupture disk device includes a rupture disk and a rupture disk holder.

3.46
rupture disk holder
The structure which encloses and clamps the rupture disk in position. Some disks are designed to be installed between standard flanges without holders.

3.47
safety relief valve
A spring-loaded pressure relief valve that may be used as either a safety or relief valve depending on the application.

3.48
safety valve
A spring-loaded pressure relief valve actuated by the static pressure upstream of the valve and characterized by rapid opening or pop action. A safety valve is normally used with compressible fluids.

3.49
set pressure
The inlet gauge pressure at which the pressure relief device is set to open under service conditions.

3.50
simmer
The audible or visible escape of compressible fluid between the seat and disc of a pressure relief valve which may occur at an inlet static pressure below the set pressure prior to opening.

3.51
specified burst pressure
The burst pressure specified by the user. The marked burst pressure may be greater than or less than the specified burst pressure but shall be within the manufacturing design range. The user is cautioned to consider manufacturing design range, superimposed backpressure and specified temperature when determining a specified burst pressure.

3.52
specified disk temperature
The temperature of the disk when the disk is expected to burst. The specified disk temperature is the temperature the manufacturer uses to establish the marked burst pressure. The specified disk temperature is rarely ever the design temperature of the vessel and may not even be the operating temperature or relief temperature, depending on the relief system configuration.

3.53
superimposed backpressure
The static pressure that exists at the outlet of a pressure relief device at the time the device is required to operate. Superimposed backpressure is the result of pressure in the discharge system coming from other sources and may be constant or variable.

4 Pressure Relief Devices

4 (PRDs)

4.1 General

This section describes the basic principles, operational characteristics, applications, and selection of pressure relief devices used independently or in combination. These devices include spring-loaded and pilot-operated Pressure Relief Valves (PRVs), rupture disk devices, and other pressure relief devices. These devices are described in the text and illustrated in Figure 1 through Figure 29.

4.2 Pressure Relief Valves (PRVs)

4.2.1 Spring-loaded PRVs

4.2.1.1 Conventional PRVs

4.2.1.1.1 A conventional PRV (see Figure 1 and Figure 4) is a self-actuated spring-loaded PRV which is designed to open at a predetermined pressure and protect a vessel or system from excess pressure by removing or relieving fluid from that vessel or system. The valve shown in Figure 4 is available in small sizes, commonly used for thermal relief valve applications. The basic elements of a spring-loaded PRV include an inlet nozzle connected to the vessel or system to be protected, a movable disc which controls flow through the nozzle, and a spring which controls the position of the disc. Under normal system operating conditions, the pressure at the inlet is below the set pressure and the disc is seated on the nozzle preventing flow through the nozzle.

Figure 1—Conventional PRV with a Single Adjusting Ring for Blowdown Control

Figure 2—Balanced-bellows PRV

Figure 3—Balanced-bellows PRV with an Auxiliary Balanced Piston

Figure 4—Conventional PRV with Threaded Connections

4.2.1.1.2 Spring-loaded PRVs are referred to by a variety of terms, such as safety valves, relief valves and safety relief valves. These terms have been traditionally applied to valves for gas/vapor service, liquid service, or multi-service applications, respectively. The more generic term, PRV, is used in the text and is applicable to all three.

4.2.1.1.3 The operation of a conventional spring-loaded PRV is based on a force balance (see Figure 5). The spring load is preset to equal the force exerted on the closed disc by the inlet fluid when the system pressure is at the set pressure of the valve. When the inlet pressure is below the set pressure, the disc remains seated on the nozzle in the closed position. When the inlet pressure exceeds set pressure, the pressure force on the disc overcomes the spring force and the valve opens. When the inlet pressure is reduced to the closing pressure, the valve re-closes.

4.2.1.1.4 When the valve is closed during normal operation (see Figure 5, Item A) the system or vessel pressure acting against the disc surface (area A) is resisted by the spring force. As the system pressure approaches the set pressure of the valve, the seating force between the disc and the nozzle approaches zero.

4.2.1.1.5 In vapor or gas service, the valve may “simmer” before it will “pop.” When the vessel pressure closely approaches the set pressure, fluid will audibly move past the seating surfaces into the huddling chamber B. As a result of the restriction of flow between the disc holder and the adjusting ring, pressure builds up in the huddling chamber B (see Figure 5, Item B). Since pressure now acts over a larger area, an additional force, commonly referred to as the expansive force, is available to overcome the spring force. By adjusting the adjusting ring, the opening in the annular orifice can be altered, thus controlling the pressure build-up in the huddling chamber B. This controlled pressure build-up in the huddling chamber will overcome the spring force causing the disc to move away from the nozzle seat, and the valve will pop open.

4.2.1.1.6 Once the valve has opened, an additional pressure build-up at C occurs (see Figure 5, Item C). This is due to the sudden flow increase and the restriction to flow through another annular orifice formed between the inner edge of the disc holder skirt and the outside diameter of the adjusting ring. These additional forces at C cause the disc to lift substantially at pop.

Figure 5—PRV Operation—Vapor/Gas Service

4.2.1.1.7 Flow is restricted by the opening between the nozzle and the disc until the disc has been lifted from the nozzle seat approximately one quarter of the nozzle diameter. After the disc has attained this degree of lift, flow is then controlled by the bore area rather than by the area between the seating surfaces.

4.2.1.1.8 The valve closes when the inlet pressure has dropped sufficiently below the set pressure to allow the spring force to overcome the summation of forces at A, B, and C. The pressure at which the valve re-seats is the closing pressure. The difference between the set pressure and the closing pressure is blowdown.

4.2.1.1.9 Figure 6 shows the disc travel from the set pressure, A, to the maximum relieving pressure, B, during the overpressure incident and to the closing pressure, C, during the blowdown.

4.2.1.2 Spring-loaded PRVs Designed for Liquid Service Applications

4.2.1.2.1 Liquid service valves do not pop in the same manner as vapor service valves (see Figure 7), since the expansive forces produced by vapor are not present in liquid flow. Liquid service valves shall necessarily rely on reactive forces to achieve lift.

4.2.1.2.2 When the valve is closed, the forces acting on the valve disc are the same as those applied by vapor until a force balance is reached and the net force holding the seat closed approaches zero. From this point on, the force relationship is totally different.

4.2.1.2.3 At initial opening, the escaping liquid forms a very thin sheet of fluid, as seen in Figure 7, Item A, expanding radially between the seating surfaces. The liquid strikes the reaction surface of the disc holder and is deflected downward, creating a reactive (turbine) force tending to move the disc and holder upward. These forces typically build very slowly during the first 2 % to 4 % of overpressure.

4.2.1.2.4 As the flow gradually increases, the velocity head of the liquid moving through the nozzle also increases. These momentum forces, combined with the reactive forces of the radially discharging liquid as it is deflected downward from the reaction surface (see Figure 7, Item B), are substantial enough to cause the valve to go into lift. Typically, the valve will suddenly surge to 50 % to 100 % lift at 2 % to 6 % overpressure. As the overpressure increases, these forces continue to grow, driving the valve into full lift. Liquid service valves, capacity certified in accordance with the ASME Code, are required to reach full rated capacity at 10 % or less overpressure.

4.2.1.2.5 In the closing cycle, as the overpressure decreases, momentum and reactive forces decrease, allowing the spring force to move the disc back into contact with the seat.

Figure 6—Typical Relationship Between Lift of Disk in a PRV and Vessel Pressure

Figure 7—PRV Operation—Liquid Service

4.2.1.2.6 Historically, many PRVs used in liquid applications were safety relief or relief valves designed for compressible (vapor) service. Many of these valves, when used in liquid service, required high overpressure (25 %) to achieve full lift and stable operation, since liquids do not provide the expansive forces that vapors do. Where liquid PRVs were required to operate within the accumulation limit of 10 %, a conservative factor of 0.6 was applied to the valve capacity when sizing the valves. Consequently, many installations were oversized and instability often resulted. The criteria used for sizing this type of valve may be found in 5.9.

4.2.1.2.7 Rules have been incorporated into the ASME Code, as well as other international standards which address performance of liquid service valves at 10 % overpressure and which require a capacity certification. PRVs designed for liquid service have been developed which achieve full lift, stable operation, and rated capacity at 10 % overpressure in compliance with the requirements. Blowdown is adjustable in some designs. Some valves are designed so that they operate on liquid and gas. Such valves may, however, exhibit different operational characteristics, depending on whether the flow stream is liquid, gas, or a combination of the two. Many PRVs designed for liquid service, for example, will have a much longer blowdown (typically 20 %) on gas than on liquid service. Additionally, some variation in set pressure may occur if the valve is set on liquid and required to operate on gas or vice versa.

4.2.1.2.8 The rules for sizing PRVs designed for liquid service are given in 5.8. If a capacity on gas service is required, 5.6.2 or 5.6.3 should be used for the preliminary sizing calculation. Capacity certification data for sizing on liquid and gas service should be obtained from the manufacturer for use in final sizing and application of the valve.

4.2.1.2.9 Spring-loaded PRVs designed for liquid (or liquid and gas) applications and which are balanced to minimize the effects of backpressure are recommended for two phase applications when the fluid being relieved may be liquid, gas, or a multi-phase mixture. Many manufacturers recommend that valves designed for liquid or liquid-and-gas service be used if the mass percentage of the two phase mixture at the valve inlet is 50 % vapor or less. In addition, if the ratio of liquid to gas in the flow stream is not certain, a valve specifically designed for liquid service or for service on liquid and gas should be used.

4.2.1.2.10 PRVs designed for liquid and gas service should be specified for the fluid to which the valve is normally exposed. For example, if a liquid and gas service valve is located in the vapor region of a vessel containing a liquid level, the valve should be specified for gas service. The valve capacity stamped on the nameplate will be in SCFM of air. If a liquid and gas service valve is located on the waterside of a heat exchanger, then the valve should be specified in liquid service. This valve will have a capacity stamped in gallons per minute of water.

4.2.1.2.11 In some applications, the valve may be required to relieve a liquid or a gas depending on the condition causing the overpressure (e.g. heat exchanger tube rupture). In this application, a valve designed for liquid service or one designed for liquid and gas service is recommended.

4.2.1.2.11 The user is cautioned that vapor certified relief valves relieving liquid are more prone to chatter than liquid relief valves relieving liquid. It has been observed on PRV test stands that vapor certified PRVs relieving liquid will at very low overpressures (about 5% of PRV capacity) will flow without much flutter but at higher overpressures, the PRV disk may lift abruptly resulting in unstable valve operation. Vapor certified valves relieving liquid may exhibit stable flow where overpressure is high (20% or higher) or where the valve lift is mechanically limited. Liquid certified valves were designed to solve the observed stability problems with liquid relief.

4.2.1.3 Balanced PRVs

4.2.1.3.1 A balanced PRV is a spring-loaded PRV which incorporates a bellows or other means of balancing the valve disc to minimize the effects of backpressure on the performance characteristics of the valve (see Figure 2 and Figure 3).

4.2.1.3.2 When a superimposed backpressure is applied to the outlet of a spring-loaded PRV, a pressure force is applied to the valve disc which is additive to the spring force. This added force increases the pressure at which an unbalanced PRV will open. If the superimposed backpressure is variable then the pressure at which the valve will open will vary (see Figure 8).

Figure 8—Typical Effects of Superimposed Backpressure on the Opening Pressure of Conventional PRVs

Figure 9—Typical Effects of Backpressure on the Set Pressure of Balanced PRVs

4.2.1.3.3 In a balanced-bellows PRV, a bellows is attached to the disc holder with an effective bellows area, A_B, approximately equal to or greater than the seating area of the disc, A_N (see Figure 9). This isolates an area on the disc, approximately equal to the disc seat area, from the backpressure. If the bellows area, A_B were identical to the area of the disc, A_N, the variable backpressure would not effect the PRV opening pressure. However, considerations concerning the bellows manufacturing tolerances and the possible variations of opening pressure with backpressure, lead to the use of bellows with minimum effective bellows area A_B greater than or equal to A_N. This ensures that backpressure does not increase the opening pressure of the balanced PRV. With the addition of a bellows, therefore, the opening pressure of the PRV will remain constant or decrease with increases in backpressure. This change in opening pressure corresponds to the variation within the bellows manufacturing tolerance. The magnitude of this effect is normally acceptable. Where opening pressures are reduced due to backpressures, the closing pressures will also be reduced resulting in longer blowdown. Consult the manufacturer when there are concerns about backpressure effects on opening pressure.

4.2.1.3.4 The internal area of the bellows in a balanced-bellows spring-loaded PRV is referenced to atmospheric pressure in the valve bonnet. It is important that the bonnet of a balanced PRV be vented to the atmosphere for the bellows to perform properly. If the valve is located where atmospheric venting would present a hazard or is not permitted by regulations, the vent shall be piped to a safe location that is free of backpressure that may affect the PRV opening pressure.

4.2.1.3.5 Other means of balancing a spring-loaded PRV such as a sealed piston are used in some valve designs. These designs perform in a manner similar to the balanced bellows design.

4.2.1.3.6 When the superimposed backpressure is constant, the spring load can be reduced to compensate for the effect of backpressure on set pressure, and a balanced valve is not required. There are cases where superimposed backpressure is not always constant and such cases shall be evaluated carefully.

4.2.1.3.7 Balanced PRVs should be considered where the built-up backpressure (backpressure caused by flow through the downstream piping after the PRV lifts) is too high for a conventional pressure relief (see 5.3.3.1). A detailed discussion of backpressure and its effects on PRV performance and flow capacity can be found in 5.3.

4.2.1.3.8 Balanced PRVs may also be used as a means to isolate the guide, spring, bonnet and other top works parts within the valve from the relieving fluid. This may be important if there is concern that the fluid will cause corrosive damage to these parts.

4.2.1.3.9 It is important to remember that the bonnet of a balanced PRV shall be vented to atmosphere at all times. The user should be cautioned of the potential for freezing of atmospheric moisture inside the bonnet in cold service due to auto-refrigeration or cold ambient temperatures.

4.2.2 Pilot-operated PRVs

4.2.2.1 General

4.2.2.1.1 A pilot-operated PRV consists of the main valve, which normally encloses a floating unbalanced piston assembly, and an external pilot (see Figure 10 through Figure 14). The piston is designed to have a larger area on the top than on the bottom. Up to the set pressure, the top and bottom areas are exposed to the same inlet operating pressure. Because of the larger area on the top of the piston, the net force holds the piston tightly against the main valve nozzle. As the operating pressure increases, the net seating force increases and tends to make the valve tighter. This feature allows most pilot-operated valves to be used where the maximum expected operating pressure is higher than the percentage shown in Figure 15. At the set pressure, the pilot vents the pressure from the top of the piston; the resulting net force is now upward causing the piston to lift, and process flow is established through the main valve. After the overpressure incident, the pilot will close the vent from the top of the piston, thereby re-establishing pressure, and the net force will cause the piston to reseat.

Figure 10—Pop-action Pilot-operated Valve (Flowing-type)

Figure 11—Pop-action Pilot-operated Valve (Non-flowing-type)

Figure 12—Modulating Pilot-operated Valve (Flowing-type)

Figure 13—Pilot-operated Relief Valve with a Non-flowing Modulating Pilot Valve

Figure 14—Low-pressure Pilot-operated Valve (Diaphragm-type)

Figure 15—Pressure Level Relationships for PRVs

4.2.2.1.2 The main valve of the pilot-operated PRV can use a diaphragm in lieu of a piston to provide the unbalanced moving component of the valve. A disc, which normally closes the main valve inlet, is integral with a flexible diaphragm (see Figure 14). The external pilot serves the same function to sense process pressure, vent the top of the diaphragm at set pressure, and reload the diaphragm once the process pressure is reduced. As with the piston valve, the seating force increases proportionally with the operating pressure because of the differential exposed area of the diaphragm.

4.2.2.1.3 The lift of the main valve piston or diaphragm, unlike a conventional or balanced spring-loaded valve, is not affected by built-up backpressure. This allows for even higher pressures in the relief discharge manifolds.

4.2.2.1.4 The pilot vent can be either directly exhausted to atmosphere or to the main valve outlet depending upon the pilot's design and user's requirement. Only a balanced-type of pilot, where set pressure in unaffected by backpressure, should be installed with its exhaust connected to a location with varying pressure (such as to the main valve outlet). Slight variations in backpressure may be acceptable for unbalanced pilots (see 5.3.3.1).

4.2.2.2 Backflow Prevention

A backflow preventer is required when the possibility exists of developing a pressure on the discharge side of the valve that exceeds the inlet pressure of the valve. The higher discharge pressure can cause sufficient upward force on the diaphragm or piston to open the valve and cause flow reversal. The backflow preventer allows the discharge pressure to provide a net downward force on the diaphragm or piston to keep the valve closed (see Figure 11). The proper operation of the backflow preventer is critical to further insuring no flow reversal occurs in the valve. The selection of the material and seals in the backflow preventer should be consistent with the pilot-operated PRV.

4.2.2.3 Pilot Types

4.2.2.3.1 The pilot that operates the main valve can be either a pop-action or modulating-action pilot. The pop-action pilot, as shown in Figure 16, causes the main valve to lift fully at set pressure without overpressure. This immediate release of pressure provides extremely high opening and closing forces on the main valve seat. This opening action is typically not recommended for liquid services to avoid valve instability.

Figure 16—Typical Relationship Between Lift of Disk or Piston and Vessel Pressure in a Pop-action Pilot-operated PRV

4.2.2.3.2 The modulating pilot, as shown in Figure 17, opens the main valve only enough to satisfy the required relieving capacity and can be used in gas, liquid or two-phase flow applications. A modulating pilot-operated valve, in contrast to a pop action valve, limits the amount of relieving fluid to only the amount required to prevent the pressure from exceeding the allowable accumulation. Since a modulating pilot only releases the required relieving rate, the calculation of built-up backpressure may be based on the required relieving rate instead of the rated relieving capacity of the valve. The modulating pilot valve also can reduce interaction with other pressure control equipment in the system during an upset condition, reduce unwanted atmospheric emissions and reduce the noise level associated with discharge to the atmosphere.

Figure 17—Typical Relationship Between Lift of Disk or Piston and Vessel Pressure in a Modulating-action Pilot-operated PRV

4.2.2.3.3 The pilots may be either a flowing or non-flowing type. The flowing type allows process fluid to continuously flow through the pilot when the main valve is open; the non-flowing type does not. The non-flowing pilot type is generally recommended for most services to reduce the possibility of hydrate formation (icing) or solids in the lading fluid affecting the pilot's performance.

4.2.2.4 Application and Limitations of Pilot-operated PRVs

4.2.2.4.1 Pilot-operated PRVs are available for use in liquid and vapor services. Operating characteristics of some pilot-operated PRVs are unaffected by the state of fluid (liquid or gas) and these types are recommended for two-phase flow applications.

4.2.2.4.2 Similar to soft-seated spring-loaded valves, most main valves and their pilots contain non-metallic components and process temperature and fluid compatibility can limit their use. In addition, as with all pressure relief devices, fluid characteristics such as susceptibility to polymerization or fouling, viscosity, the presence of solids, and corrosiveness should be considered. The manufacturer should be consulted to ensure that the proposed application is compatible with available valves.

4.2.2.5 Pilot-operated PRV Accessories

4.2.2.5.1 General

A variety of accessories and options are available for pilot-operated PRVs to provide additional functions. The following is a list of some of the more common accessories. The manufacturer should be contacted to determine availability.

4.2.2.5.2 Field Test Connection

Pilot-operated PRVs may be readily tested for verification of set pressure during normal system operation with this accessory. This field test is typically done via pressure from an independent source such as a nitrogen bottle, where the source gas is slowly admitted through a metering valve. The pilot and main valve dome are pressurized simulating an increased system pressure. The field test pressure will actuate the pilot and may or may not actuate the main valve. The valve manufacturer should be contacted for details.

4.2.2.5.3 Backflow Preventer

This accessory, sometimes called a "vacuum block," prevents a pilot-operated valve from reverse flow when the pressure at the outlet flange (superimposed backpressure) is greater than the current system pressure. Reverse flow can occur with any standard type or design of pilot-operated PRV when sufficient reverse differential pressure exists. A backflow preventer permits the introduction of outlet pressure into the dome of the main valve, thereby holding the piston firmly on the nozzle, overcoming the effect of a reverse differential pressure. A backflow preventer should be specified whenever any of the following situations apply:

— the protected equipment can be depressured and isolated (e.g. to prepare for maintenance) while lined up to an active flare header;

— a vacuum may be present at the inlet connection due to unusual operating conditions or during startup;

— the valve is connected to a downstream pressure vessel where pressure may vary from time to time in excess of the pressure in the upstream system;

— the discharge of multiple PRVs is combined into a single manifold or vent system, creating superimposed backpressure in excess of the current upstream system pressure.

4.2.2.5.4 Pilot Supply Filter

A pilot supply filter protects the pilot from particulate matter in the flow stream. This accessory, installed in the pilot supply line has expanded the service applications for pilot-operated PRVs. The user is cautioned that in services prone to plugging, frequent maintenance of the filter may be required to achieve the benefits. For applications with excessive particulate matter, other methods such as the addition of a purge system may be required.

4.2.2.5.5 Pressure Spike Snubber

A pressure spike snubber is recommended for use on compressible or incompressible lading fluid installations where instantaneous pressure spikes or pulsations approach or exceed the simmer or set pressure and may cause inadvertent valve wear and actuation (e.g. downstream of positive displacement rotating equipment). The device dampens these transient pressure rises before they reach the sensing chambers of the pilot without affecting the valve's set pressure.

4.2.2.5.6 Remote Pressure Sense Connection

This optional feature permits the pilot to sense system pressure at a location that most accurately reflect the actual operating pressure of the protected system. It can also be used to eliminate the false system pressure indication that will occur during relieving conditions due to pressure losses in the inlet piping. The addition of a remote pilot sense line allows the pilot to correctly sense system pressure and to keep the valve from rapid cycling or chattering due to high inlet piping pressure losses. Relieving capacity will be proportionately reduced whenever there is inlet pressure loss to the valve.

4.2.2.5.7 Manual or Remote Unloader

An unloader permits the main valve to be opened either manually or remotely to depressurize the system. Its use has no effect on the sealed pressure settings. Either a manual, pneumatic or solenoid operated valve is connected to the main valve dome. Opening this valve vents the dome pressure faster than it can be recharged by the pilot supply allowing the piston to lift. When permitted by code, the manual unloader may be substituted for a mechanical lift lever.

4.2.2.5.8 Pilot Lift Lever

This accessory is provided for those applications where the mechanical lifting of the pilot is required for verification of valve operation. Lifting of the pilot spindle will permit the main valve to lift when the system pressure is at least equal to or greater than 75 % of the set pressure. The ASME Code requires the use of this device or a manual unloader for air, hot water over 140 °F (60 °C) and steam applications, unless ASME Code Case 2203 has been used to eliminate the need for a lifting lever.

4.2.2.5.9 Pilot Valve Isolation

There are several methods that use an inert fluid to isolate all or part of the pilot valve portion of the pilot-operated PRV from the service conditions. This may allow the pilot-operated PRV to be considered in applications where, for example, there is polymerization, high viscosity, presence of solids, or corrosiveness. These isolation systems should be designed to provide the inert fluid to protect the critical parts of the pilot valve during normal operating conditions and also when the pilot valve operates during an over pressure event. The user should contact the manufacturer to discuss the various isolation options available with a particular pilot design and what isolation options would be suitable for the specific service condition. The user and manufacturer should also discuss any local Codes or regulations that may be required for the application and the suitability of a particular pilot valve isolation system in meeting such a requirement.

4.2.3 Cold Differential Test Pressure (CDTP)

4.2.3.1 The actual service conditions under which a PRV is required to open may be different from the conditions at which the PRV is set to operate on a test stand. To compensate for this effect, a CDTP is specified for adjusting the set pressure of the valve on the test stand. The CDTP may include a correction for actual service conditions of backpressure and/or temperature.

4.2.3.2 A temperature correction factor may be applied when the pressure relief valve temperature exceeds the ambient temperature. The magnitude of the correction factor and the minimum temperature at which it may be applied is dependent on the pressure relief valve model. The correction factor shall be obtained from the pressure relief valve manufacturer and is often available in their Installation, Operation and Maintenance manuals. The correction factor is normally provided as a multiplier to the set pressure. If a CDTP is applied due to back pressure and temperature, the multiplier shall be applied after the correction for back pressure is made.

~~**4.2.3.2**~~ The temperature used for the correction factor should be based on the temperature at the inlet to the relief valve at its normal service (non-relieving) conditions. The temperature at the valve may not be equal to the operating temperature of the process due to valve's physical location, collection of non-condensable vapors below the valve inlet, isolation from the process by a rupture disk, or heat tracing of the valve. The factor compensates for variations in spring load due to thermal growth in valve components as well as changes in the spring material properties. Compensation may also be required for low temperature service below -75 °F (-60 °C).~~A temperature correction factor (multiplier) may be required when the relief device inlet relieving temperature is expected to exceed 250 °F (120 °C) at the time the relief device is called upon to open. The factor compensates for variations in spring load due to thermal growth in valve components as well as changes in the spring material properties. Compensation may also be required for low temperature service below –75 °F (–60 °C). When such temperature compensation is required, the correction factor should be obtained from the PRV manufacturer.~~

4.2.3.3 A conventional PRV, operating with a constant superimposed backpressure, normally requires a correction factor to compensate for the backpressure. In this case the required set pressure minus the superimposed backpressure is equal to the CDTP. This change accounts for the additional closing force exerted on the valve disk by the backpressure. In the case of a balanced spring-loaded PRV, the change in closing force due to the superimposed backpressure is negligible and no correction is required.

4.2.3.4 When the CDTP is to include a correction for backpressure and temperature, the differential pressure is calculated and then multiplied by the temperature correction to determine the CDTP.

4.2.3.5 Pilot-operated PRVs (see 4.2.2) may require a CDTP when used in high temperature or backpressure service. The valve manufacturer should be consulted regarding backpressure and temperature limits, and required correction factor.

4.3 Rupture Disk Devices

4.3.1 General

4.3.1.1 Rupture disk devices are non-reclosing pressure relief devices used to protect vessels, piping and other pressure containing components from excessive pressure and/or vacuum. Rupture disks are used in single and multiple relief device installations. They are also used as redundant pressure relief devices.

4.3.1.2 With no moving parts, rupture disks are simple, reliable, and faster acting than other pressure relief devices. Rupture disks react quickly enough to relieve some types of pressure spikes. Because of their light weight, rupture disks can be made from high alloy and corrosion-resistant materials that are not practical in PRVs.

4.3.1.3 Rupture disks can be specified for systems with vapor (gas) or liquid pressure relief requirements. Also, rupture disk designs are available for highly viscous fluids. The use of rupture disk devices in liquid service should be carefully evaluated to ensure that the design of the disk is suitable for liquid service. The user should consult the manufacturer for information regarding liquid service applications.

4.3.1.4 The rupture disk is also a temperature sensitive device. Burst pressures can vary significantly with the temperature of the rupture disk device. This temperature may be different from the normal fluid operating temperature. As the temperature at the disk increases, the burst pressure usually decreases. Since the effect of temperature depends on the rupture disk design and material, the manufacturer should be consulted for specific applications. For these reasons, the rupture disk shall be specified at the pressure and temperature the disk is expected to burst.

4.3.1.5 Care shall be taken during installation to avoid damaging the disk and to ensure that the disk and holder are properly oriented relative to the flow. A damaged or improperly oriented disk may burst considerably higher than its marked burst pressure, depending on the style of the disk. Contact the manufacturer for information about the effects of damage or improper orientation for a specific style of disk.

4.3.1.6 Care shall also be taken to follow the manufacturer's bolt torque and tightening procedures during installation. Improper torque can also affect the disk's burst pressure.

4.3.2 Application of Rupture Disks

4.3.2.1 Single, Multiple, and Fire Applications

Rupture disks can be used in any application requiring overpressure protection where a non-reclosing device is suitable. This includes single, multiple, and fire applications as specified in UG-134 of the ASME Code. Figure 18 provides the pressure level relationships between rupture disks and the protected equipment in accordance with the ASME Code.

Figure 18—Pressure Level Relationships for Rupture Disk Devices

4.3.2.2 Rupture Disk Device at the Inlet of a PRV

4.3.2.2.1 The ASME Code also allows for the use of rupture disks in combination with PRVs (see Figure 19). Rupture disks are used upstream of PRVs to seal the system to meet emissions standards, to provide corrosion protection for the valve, and to reduce valve maintenance.

Figure 19—Rupture Disk Device in Combination with a PRV

4.3.2.2.2 When a rupture disk device is installed at the inlet of a PRV, the devices are considered to be close coupled, and the specified burst pressure and set pressure should be the same nominal value. When installed in liquid service it is especially important for the disk and valve to be close coupled to reduce shock loading on the valve.

4.3.2.2.3 The space between the rupture disk and the PRV shall have a free vent, pressure gauge, trycock, or suitable telltale indicator as required in UG-127 of the ASME Code. A non-vented space with a pressure gauge without alarms or other devices is not recommended as a suitable telltale indicator. Users are warned that a rupture disk will not burst in tolerance if backpressure builds up in a non-vented space between the disk and the PRV, which will occur should leakage develop in the rupture disk due to corrosion or other cause.

4.3.2.3 Rupture Disk Device at the Outlet of a PRV

A rupture disk device may be installed on the outlet of a PRV to protect the valve from atmospheric or downstream fluids. Consideration should be given to the valve design so that it will open at its proper pressure setting regardless of any backpressure that may accumulate between the valve and rupture disk. See UG-127 of the ASME Code for other requirements and considerations.

4.3.2.4 Highly Corrosive Applications

In highly corrosive applications, two rupture disks are often used together. A double disk assembly consists of two rupture disks mounted in a special holder with a vapor space between them. If the first disk develops a leak due to corrosion the second disk will contain the fluid. The vapor space between the disks should have a free vent, pressure gauge, trycock or suitable telltale indicator for monitoring of pressure build-up. This gives the user an indication that replacement of the rupture disk is required.

4.3.2.5 Highly Viscous Applications

Rupture disk designs are available for processes with high viscosity fluid, including nonabrasive slurries, where fluid flow is directed across the rupture disk inlet to prevent product build-up which may otherwise adversely affect rupture disk performance. The disk manufacturer should be consulted for details in these applications.

4.3.3 Types of Rupture Disks

There are three major rupture disk types:

— forward-acting, tension loaded;

— reverse-acting, compression loaded;

— graphite, shear loaded.

4.3.3.1 Forward-acting Solid Metal Rupture Disks

A forward-acting rupture disk is a formed (domed), solid metal disk designed to burst at a rated pressure applied to the concave side (see Figure 20). This rupture disk typically has an angular seat design and provides a satisfactory service life when operating pressures are up to 70 % of the marked burst pressure of the disk (70 % operating ratio). Consult the manufacturer for the actual recommended operating ratio for the specific disk under consideration. If vacuum or backpressure conditions are present, the disk can be furnished with a support to prevent reverse flexing. These disks have a random opening pattern and are considered fragmenting designs that are not suitable for installation upstream of a PRV.

Figure 20—Forward-acting Solid Metal Rupture Disk

4.3.3.2 Forward-acting Scored Rupture Disks

The scored forward-acting rupture disk is a formed (domed) disk designed to burst along scored lines at a rated pressure applied to the concave side (see Figure 21). Some designs provide satisfactory service life when operating pressures are up to 85 % to 90 % of the marked burst pressure of the disk (85 % to 90 % operating ratio). Consult the manufacturer for the actual recommended operating ratio for the specific disk under consideration. Most designs withstand vacuum conditions without a vacuum support. If backpressure conditions are present, the disk can be furnished with a support to prevent reverse flexing. Because the score lines control the opening pattern, this type of disk can be manufactured to be non-fragmenting and is acceptable for installation upstream of a PRV. The scored, forward-acting rupture disk is manufactured from thicker material than non-scored designs with the same burst pressure, and provides additional resistance to mechanical damage.

Figure 21—Forward-acting Scored Rupture Disk

4.3.3.3 Forward-acting Composite Rupture Disks

4.3.3.3.1 A forward-acting composite rupture disk is a flat or domed multi-piece construction disk (see Figure 22). The domed composite rupture disk is designed to burst at a rated pressure applied to the concave side. The flat composite rupture disk may be designed to burst at a rated pressure in either or both directions. Some designs are non-fragmenting and acceptable for use upstream of a PRV.

Figure 22—Forward-acting Composite Rupture Disk

4.3.3.3.2 The domed composite rupture disk is available in flat seat or angular seat design. The burst pressure is controlled by the combination of slits and tabs in the top section and a metallic or nonmetallic seal member under the top section. Composite rupture disks are generally available in burst pressures lower than those of forward acting, non-scored rupture disks. Composite rupture disks may offer a longer service life as a result of the corrosion resistant properties of the seal material selected.

4.3.3.3.3 The slits and tabs in the top section provide a predetermined opening pattern for the rupture disk. If vacuum or backpressure conditions are present, composite disks can be furnished with a support to prevent reverse flexing (see Figure 22). A domed, composite rupture disk generally provides satisfactory service life when the operating pressure is 80 % or less of the marked burst pressure (80 % operating ratio). Consult the manufacturer for the actual recommended operating ratio for the specific disk under consideration.

4.3.3.3.4 A flat composite rupture disk is available for the protection of low-pressure vessels or the isolation of equipment such as exhaust headers or the outlet side of a PRV. This disk usually comes complete with gaskets and is designed to be installed between companion flanges rather than within a specific rupture disk holder. Flat composite rupture disks generally provide satisfactory service life when operating pressures are 50 % or less of the marked burst pressure (50 % operating ratio). Consult the manufacturer for the actual recommended operating ratio for the specific disk under consideration.

4.3.3.4 Reverse-acting Rupture Disks

4.3.3.4.1 A reverse-acting rupture disk typically is a formed (domed) solid metal disk designed to reverse and burst at a rated pressure applied on the convex side. Reverse-acting rupture disks are designed to open by such methods as shear knife blades, tooth rings, or scored lines (see Figure 23 and Figure 24).

Figure 23—Reverse-acting Rupture Disk with Knife Blades

Figure 24—Reverse-acting Scored Rupture Disk

4.3.3.4.2 Reverse-acting rupture disks may be manufactured as non-fragmenting and are suitable for installation upstream of PRVs. These disks provide satisfactory service life when operating pressures are 90 % or less of marked burst pressure (90 % operating ratio). Some types of reverse-buckling disks are designed to be exposed to pressures up to 95 % of the marked burst pressure. Consult the manufacturer for the actual recommended operating ratio for the specific disk under consideration. Because a reverse-acting rupture disk is operated with pressure applied on the convex side, thicker disk materials may be used, thereby lessening the effects of corrosion, eliminating the need for vacuum support, and providing longer service life under pressure/vacuum cycling conditions and pressure fluctuations.

4.3.3.4.3 Knife blades installed in holders should be constructed of corrosion-resistant material and should be inspected periodically to ensure sufficient sharpness to open the disk. Dull or damaged knife blades may prevent proper opening of the disk.

4.3.3.5 Graphite Rupture Disks

4.3.3.5.1 Graphite rupture disks are typically machined from a bar of fine graphite that has been impregnated with a sealing compound to seal the porosity of the graphite matrix (see Figure 25). The disk operates on a pressure differential across the center diaphragm or web portion of the disk. Graphite rupture disks provide a satisfactory service life when operating pressures are up to 80 % of the marked burst pressure (80 % operating ratio) and can be used in both liquid and vapor service. Consult the manufacturer for the actual recommended operating ratio for the specific disk under consideration.

Figure 25—Graphite Rupture Disk

4.3.3.5.2 If vacuum or backpressure conditions are present, the disk can be furnished with a support to prevent reverse flexing. These disks have a random opening pattern and are considered fragmenting designs that are not suitable for installation upstream of a PRV. A metallic ring called armoring is often added to the outside diameter of the disk to help support uneven piping loads and minimize the potential for cracking of the outer graphite ring and blowout of process fluid.

4.3.4 Rupture Disk Holders

Rupture disk holders are used to clamp the rupture disk in place and effect a leak-tight, metal-to-metal seal. The seating area of the holders is typically unique to specific manufacturers and styles of rupture disks. Rupture disk holders are available in a variety of configurations including full bolting, weldneck, threaded, etc. The most common configuration is the insert type which fits between standard pipe flanges, and the outside diameter of the holder fits inside the flange studs. Rupture disk holders are available in a variety of materials and coatings.

4.3.5 Rupture Disk Accessories

A variety of accessories are available for use with rupture disks in various applications. The following provides a brief description of some of these components and their application.

a) *Rupture indicators and sensors.* These devices typically provide an electrical or mechanical signal which can indicate the opening and/or leakage of a rupture disk or PRV.

b) *Alarm monitors.* Alarm monitors are available to monitor rupture disk indicators or sensors. Alarm monitors are available with intrinsically safe circuits.

c) *Heat shields*. Heat shields are generally installed upstream of the rupture disk in high temperature processes to reduce the temperature at the rupture disk.

d) *Baffle plates*. When venting to atmosphere, baffle plates can be used to deflect process discharge away from personnel and equipment. Baffle plates are also commonly used to assist in absorbing the recoil forces of an activating rupture disk.

4.3.6 Rupture Disk Selection and Specification

4.3.6.1 General

4.3.6.1.1 Rupture disk selection is based on the operating parameters of the system in which it is installed. These parameters should be specified by the purchaser when purchasing rupture disks. These parameters include, but are not limited to:

— MAWP of vessel or piping;

— fluid state (vapor, liquid, or multiphase);

— range of operating pressures and operating temperature;

— cyclic or pulsating service;

— required relieving capacity or maximum resistance coefficient;

— corrosiveness of upstream and downstream environment;

— vacuum or backpressure conditions;

— location upstream or downstream of a PRV;

— single or multiple devices.

4.3.6.1.2 The following rupture disk parameters are selected or determined based on the above system operating parameters:

— burst pressure and temperature (see Figure 26);

— operating ratio, manufacturing design range and burst tolerance (see Figure 27);

— disk type, material and construction;

— disk and holder size (based on required flow per 5.11).

4.3.6.2 Rupture Disk Selection

4.3.6.2.1 Rupture disk types and basic performance characteristics are described in 4.3.3 and may be used as a basis for selection. The relationship between system pressures and the operating characteristics of a rupture disk device are shown in Figure 26. Since the marked burst pressure of a rupture disk can be anywhere within its manufacturing design range, the user is cautioned to make sure that the upper limit of the manufacturing design range does not exceed the MAWP of the equipment being protected. As shown in Figure 28, when the disk has a positive manufacturing design range, the marked burst pressure of the disk can actually be greater than the specified pressure.

Figure 26—Rupture Disk Application Parameters Assuming No Superimposed Backpressure

Figure 27—Common Types of Manufacturing Ranges and Corresponding Burst Pressure Marking

4.3.6.2.2 The maximum pressure at which a rupture disk may be marked to burst is the upper limit of its manufacturing design range. The minimum pressure at which a rupture disk may be marked to burst is the lower limit of its manufacturing design range. Figure 27 provides graphical examples of common relationships between burst pressure, manufacturing design range, burst tolerance, and operating pressure.

4.3.6.2.3 Rupture disk selection is an iterative and sometimes complex process. The procedure given below should be used for rupture disk selection where there is no superimposed backpressure. Consult the manufacturer for assistance if needed.

a) Select the upper limit of the manufacturing design range. This is typically based on the MAWP of the protected equipment as determined by the ASME Code or process requirements. In some applications, such as in multiple or supplemental device installation (see 5.4.2 and 5.4.3), the upper limit of the manufacturing design range may exceed the MAWP of the protected equipment.

b) Determine the specified burst pressure by subtracting the positive portion of the manufacturing design range, as listed in the manufacturer's catalog, from the upper limit of the manufacturing design range.

c) Determine the lower limit of the manufacturing design range by subtracting the negative portion of the manufacturing design range, as listed in the manufacturer's catalog, from the specified burst pressure.

d) Determine the operating ratio by dividing the maximum operating pressure by the lower limit of the manufacturing design range.

e) When calculating the operating ratio for disks with specified burst pressures between 15 psig and 40 psig (103 kPag and 276 kPag), subtract the burst tolerance of 2 psi (14 kPa) from the lower limit of the manufacturing design range prior to calculating the operating ratio. Since some rupture disks are offered in extremely low pressures well below 15 psig (103 kPag), users are advised to verify burst tolerances and manufacturing design ranges applicable to those pressures

f) Select a rupture disk based on the specified burst pressure and the manufacturing design range, and compare the operating ratio with the manufacturer's maximum recommended operating ratio as listed in the product catalog. If the operating ratio exceeds the manufacturer's maximum recommended operating ratio, select a smaller manufacturing design range, if available, for that disk style or change disk style and repeat Steps a) through e).

4.3.6.2.4 Superimposed backpressure significantly complicates the design and selection process of the rupture disk device. Figure 28 provides an example of a rupture disk with superimposed backpressure. The impact of the superimposed backpressure shall be considered when selecting the specified burst pressure and determining the operating ratio. Consideration shall also be given in the event the superimposed backpressure is inadvertently reduced below that which was used to specify the disk, since this could result in undesired disk activation.

Figure 28—Rupture Disk Application Parameters with Superimposed Backpressure

4.3.6.2.5 For most closed systems, the superimposed backpressure normally varies between some minimum and maximum pressure. For the particular rupture disk device being designed, the superimposed backpressure does not normally include the pressure caused by other relief devices venting into the closed system unless that pressure would cause the relief pressure to exceed the code allowed accumulated pressure. However, the backpressure, caused by the venting of other relief devices, still needs to be considered when specifying the disk and may result in additions such as a vacuum or backpressure support to protect the disk.

4.3.6.3 Rupture Disk Device Specification

Accurately and completely documenting the process conditions and rupture disk device specifications is a key element in selecting the proper rupture disk. Annex A provides a rupture disk device specification sheet and step-by-step guidance for completing the specification sheet.

4.4 Pin-actuated Devices

4.4.1 General

Pin-actuated pressure relief devices are non-reclosing devices consisting of a moving disc exposed to the pressure system, and an external mechanism housing a pin which is mechanically linked to the disc. Pins may be loaded in tension (breaking pins) or in compression (buckling pins, see Figure 29). The pin restrains the movement of the disc until the specified set pressure is reached. At this point the pin fails and the disc opens.

4.4.2 Buckling Pin Devices

4.4.2.1 General

4.4.2.1.1 Buckling pin devices, as shown in Figure 29, are compression-loaded, pin-actuated devices and are the most extensively used type of pin-actuated device. Compression-loaded buckling pin devices are very stable and well suited to applications that have both cyclic operating conditions, and an operating pressure to set pressure ratio greater than or equal to 90 %.

Figure 29—Buckling Pin Valve

4.4.2.1.2 Buckling pin devices may be sensitive to differential pressures. Operating conditions on both sides of the device need to be reviewed between the user and the manufacturer.

4.4.2.1.3 See UG-127(b) and UG-138 of ~~ASME Code Case 2091-3 defines~~ the ASME Code for sizing, application and minimum requirements for buckling pin devices.

4.4.2.2 Set Pressure and Temperature

4.4.2.2.1 The set pressure of the pin-actuated device should be determined by the user, and an agreed set pressure tolerance on either side of the nominal set pressure should be established with the manufacturer.

4.4.2.2.2 The wetted parts of the device shall be designed to meet the process temperature to ensure that acceptable materials are selected. However, since the pin is external to the process, the pin is not exposed to the process temperature conditions but rather to the external ambient conditions. Therefore, the pin shall be designed based on the external ambient temperature to ensure that the set pressure of the device is correctly established.

4.4.2.2.3 Compression-loaded buckling pins have a low sensitivity to temperature. If a pin device will see service over a wide range of ambient temperatures, or outside of an ambient temperature range, then advice concerning change in set pressure should be sought from the manufacturer. In some cases it may be recommended to conduct specific temperature testing of pins before delivery of the device.

4.4.2.3 Leak Tightness

4.4.2.3.1 The buckling pin device typically uses elastomer seals. The seal material should be carefully chosen to satisfy both the chemical conditions and the anticipated service temperatures. It is recommended that the leak tightness of the device be tested per API 527 before shipment by the manufacturer.

4.4.2.3.2 If the application is vacuum service and/or backpressure exists, the manufacturer needs to be notified to ensure proper sealing under such conditions.

4.4.2.4 Marking and Tagging

The buckling pin device should be clearly marked to indicate the direction of flow, set pressure, nominal size, serial number and model or type designation, materials of construction, and the manufacturer. Each pin, or tag attached to each pin, should be marked with manufacturer, lot number, device model or type, set pressure and pin/device identifier. The lot number should appear on the manufacturer's certification report, together with the serial number of the device or the device identifier for which the pins have been calibrated.

4.4.3 Breaking Pin Devices

4.4.3.1 A breaking pin device is a non-reclosing pressure relief device with a movable disc held in the closed position by a pin loaded in tension. When pressure reaches the set pressure of the device, the pin breaks and the disc opens. Breaking pin devices are generally used in combination with a PRV where valve tightness is of concern, for example, in corrosive or vibrating environments such as on fluid transport vessels.

4.4.3.2 Breaking pin devices are designed to operate at a specified differential pressure. If pressure is allowed to build up on the downstream side of the breaking pin device, the opening pressure will be increased. Therefore, the space between a breaking pin device and a PRV shall be provided with a gauge, trycock, free vent or suitable telltale indicator to detect any build-up of pressure in that cavity.

4.4.3.3 See UG-127(b) and UG-138 of t~~T~~he ASME Code for sizing, application, and minimum requirements for breaking pin devices. ~~allows breaking pin devices to be used only in combination with PRVs. ASME Code Case 2487 allows the use of breaking pin devices in single relief device installations and provides additional requirements.~~

4.5 Open Flow Paths or Vents

4.4.1 A flow path or vent that is open directly or indirectly to the atmosphere may be used as the sole relief device to protect vessels, piping, and other equipment from excessive pressure and/or vacuum. With proper analysis to determine the calculated capacity (see 5.12), open flow paths or vents can be specified for systems with vapor (gas) and/or liquid relief requirements.

4.5 Other Types of Devices

Other pressure relief devices not described in this section are occasionally specified in refineries and related industries. Users should consult the manufacturer for information about designs and special applications.

5 Procedures for Sizing

5.1 Determination of Relief Requirements

5.1.1 To establish the size and design of a pressure relief device for any application, the designer shall first determine the conditions for which overpressure protection may be required. Care should be exercised in establishing the various contingencies that could result in overpressure.

5.1.2 The contingencies that can cause overpressure shall be evaluated in terms of the pressures generated and the rates at which fluids are required to be relieved. The process flow diagram, material balance, piping and instrument diagrams, equipment specification sheets, and design basis for the facility are needed to calculate the individual relieving rates for each pressure relief device. Process equipment vendor data is also helpful if available.

5.1.3 API 521 lists a number of common operational conditions for which overpressure protection may be required. This list is by no means complete; each plant may have unique features that shall be considered in addition to those listed. API 521 provides a detailed discussion of relief requirements for these emergency operating conditions. API 521 also provides a detailed discussion of the relief requirements for the special case of fire.

5.2 API Effective Area and Effective Coefficient of Discharge

5.2.1 PRVs may be initially sized using the equations presented in 5.6 through 5.10 as appropriate for vapors, gases, liquids, or two phase fluids. These equations utilize effective coefficients of discharge and effective areas which are independent of any specific valve design. In this way, the designer can determine a preliminary PRV size.

5.2.2 The designer can use API 526 to select a PRV. API 526 is a purchase specification for steel flanged valves. This standard lists specific valve configurations specified by inlet/outlet size and flange configuration, materials of construction, pressure/temperature limits, inlet and outlet center to face dimensions, and effective orifice designation. When a valve is specified per this standard, the orifice size is expressed in terms of a letter designation ranging from the smallest, “D,” to the largest, “T.” An effective area is specified for each letter orifice.

5.2.3 Sections 5.6 through 5.10 provide sizing information which may be used for the initial selection of a PRV from the incremental D through T orifice sizes specified in API 526. The effective orifice areas listed in API 526 and the effective coefficient of discharge used for the initial selection are nominal values not directly related to a specific valve design.

5.2.4 The rated coefficient of discharge for a PRV, as determined per the applicable certification standards, is generally less than the effective coefficient of discharge used in API 520 (particularly for vapor service valves where the effective coefficient of discharge is 0.975). This is true of valves certified per the rules of the ASME Code, where the average coefficient from a series of valve test results is multiplied by 0.9 to establish a rated coefficient of discharge. For this reason, the actual discharge or orifice area for most valve designs is greater than the effective discharge area specified for that valve size per API 526. This allows the rated capacity of most valve designs to meet or exceed the estimated capacity for preliminary sizing determined per the API 520 calculations.

5.2.5 When a specific valve design is selected for an application, the rated capacity of that valve can be determined using the actual orifice area, the rated coefficient of discharge, and the equations presented in this document. This rated relieving capacity is then used to verify that the selected valve has sufficient capacity to satisfy the application. It is important to remember that the effective area and the effective coefficient of discharge are used only for the initial selection. The actual orifice area and the rated coefficient of discharge shall always be used to verify the actual capacity of the PRV. In no case should an effective area be used with a rated coefficient of discharge for calculating the capacity of a PRV. Similarly, an actual area should not be used in conjunction with an effective coefficient of discharge.

5.2.6 In summary, the effective orifice size and effective coefficient of discharge specified in API standards are assumed values used for initial selection of a PRV size from configurations specified in API 526, independent of an individual valve manufacturer’s design. In most cases, the actual area and the rated coefficient of discharge for an API letter orifice valve are designed so that the actual certified capacity meets or exceeds the capacity calculated using the methods presented in API 520. There are, however, a number of valve designs where this is not so. When the PRV is selected, therefore, the actual area and rated coefficient of discharge for that valve shall be used to verify the rated capacity of the selected valve and to verify that the valve has sufficient capacity to satisfy the application.

5.3 Backpressure

5.3.1 General

5.3.1.1 Pressure existing at the outlet of a PRV is defined as backpressure. Regardless of whether the valve is vented directly to atmosphere or the discharge is piped to a collection system, the backpressure may affect the operation of the PRV. Effects due to backpressure may include variations in opening pressure, reduction in flow capacity, instability, or a combination of all three.

5.3.1.2 Backpressure which is present at the outlet of a PRV when it is required to operate is defined as superimposed backpressure. This backpressure can be constant if the valve outlet is connected to a process vessel or system that is held at a constant pressure. In most cases, however, the superimposed backpressure will be variable as a result of changing conditions existing in the discharge system.

5.3.1.3 Backpressure which develops in the discharge system after the PRV opens is defined as built-up backpressure. Built-up backpressure occurs due to pressure drop in the discharge system as a result of flow from the PRV. Short tailpipes that vent directly to the atmosphere typically result in lower built-up backpressures than long discharge systems. However, choked flow can occur at the outlet of even short tailpipes vented directly to atmosphere, resulting in a high built-up backpressure. For this reason, the magnitude of the built-up backpressure should be evaluated for all systems, regardless of the outlet piping configuration.

5.3.1.4 The magnitude of the backpressure which exists at the outlet of a PRV, after it has opened, is the total of the superimposed and the built-up backpressure.

5.3.2 Effects of Superimposed Backpressure on PRV Opening

5.3.2.1 Superimposed backpressure at the outlet of a conventional spring-loaded PRV acts to hold the valve disc closed with a force additive to the spring force. The actual spring setting can be reduced by an amount equal to the superimposed backpressure to compensate for this [see 4.2.3 for a discussion of cold differential test pressure (CDTP)].

5.3.2.2 Balanced PRVs (see 4.2.1.3) utilize a bellows or piston to minimize or eliminate the effect of superimposed backpressure on set pressure. Many pilot-operated PRVs have pilots that are vented to atmosphere or are balanced to maintain set pressure in the presence of variable superimposed backpressure. Balanced spring-loaded or pilot-operated PRVs should be considered if the superimposed backpressure is variable. However, if the amount of variable superimposed backpressure is small, a conventional valve could be used provided

5.3.2.2 :

a) the bench set pressure (CDTP) has been appropriately compensated for superimposed backpressure; and

b) the maximum pressure during relief does not exceed the code-allowed limits for accumulation in the equipment being protected.

5.3.2.3 For example, conventional valves are often used when the outlet is piped into a relief header without compensating the set pressures for the superimposed backpressure caused by other relieving devices. This approach can be used provided the allowable accumulation is not exceeded during the release.

5.3.3 Effects of Backpressure on PRV Operation and Flow Capacity

5.3.3.1 Conventional PRVs

5.3.3.1.1 Conventional PRVs show unsatisfactory performance when excessive backpressure develops during a relief incident, due to the flow through the valve and outlet piping. The built-up backpressure opposes the lifting force which is holding the valve open.

5.3.3.1.2 Excessive built-up backpressure can cause the valve to operate in an unstable manner. This instability may occur as flutter or chatter. Chatter refers to the abnormally rapid reciprocating motion of the PRV disc where the disc contacts the PRV seat during cycling. This type of operation may cause damage to the valve and interconnecting piping. Flutter is similar to chatter except that the disc does not come into contact with the seat during cycling.

5.3.3.1.3 In a conventional PRV application, built-up backpressure should not exceed 10 % of the set pressure at 10 % allowable overpressure. A higher maximum allowable built-up backpressure may be used for allowable overpressures greater than 10 % provided the built-up backpressure does not exceed the allowable overpressure. For example, for the fire case, pressure (built-up plus constant) should not exceed 21%; similarly for multiple relief valves installation with staggered setting total back pressure should not exceed 16%. When the superimposed backpressure is constant, the spring load may be reduced to compensate for the superimposed backpressure. When the downstream piping is designed within the above backpressure criteria, no backpressure capacity correction (K_b = 1.0) is required in the valve sizing equations, for gases at critical flow or for liquids. When the backpressure is expected to exceed these specified limits, a balanced or pilot-operated PRV should be specified.

Note that the built-up back pressure limitations discussed in paragraph 5.3.3.1.3 do not apply to valves with open bonnet design due to the nature of their design. Consult the manufacturer for guidance.

5.3.3.2 Balanced PRVs

5.3.3.2.1 A balanced PRV should be used where the built-up backpressure is too high for conventional PRVs or where the superimposed backpressure varies widely compared to the set pressure. Balanced valves can typically be applied where the total backpressure (superimposed plus built-up) does not exceed approximately 50 % of the set pressure. The specific manufacturer should be consulted concerning the backpressure limitation of a particular valve design. With a balanced valve, high backpressure will tend to produce a closing force on the unbalanced portion of the disc. This force may result in a reduction in lift and an associated reduction in flow capacity. Capacity correction factors, called backpressure correction factors, are provided by manufacturers to account for this reduction in flow. Typical backpressure correction factors may be found for compressible fluid service in Figure 30 and for incompressible fluid (liquid) service in Figure 31. For liquid service applications, the factor shown in Figure 31 is applicable for all overpressures. For compressible fluid service, however, the factor may vary depending on whether the allowable overpressure is 10 %, 16 %, or 21 %.

NOTE The backpressure correction factors from Figure 30 and Figure 31 are suitable for the preliminary sizing procedures found in this document. Final sizing calculations should always be completed using the manufacturer's actual charts.

Figure 30—Backpressure Correction Factor, K_b, for Balanced-bellows PRV (Vapors and Gases)

Figure 31—Capacity Correction Factor, K_w, due to Backpressure on Balanced-bellows PRVs in Liquid Service

In some applications, set pressure may be significantly less than MAWP allowing for overpressures in excess of those specified above. In such cases, the manufacturer should be consulted for guidance.

5.3.3.2.2 In most applications, the allowable overpressure is 10 % and the backpressure correction factor for 10 % overpressure shall be used. In the special case of multiple valve installations, the low set valve may operate at overpressures up to 16 %. A backpressure correction factor for 16 % overpressure may be used for that low set valve. The high set valve is actually operating at a maximum overpressure of 10 % (assuming the high set valve is set at 105 % of the MAWP), however, and the backpressure correction factor for 10 % overpressure shall be used for that high set valve. A supplemental valve used for an additional hazard created by exposure to fire (see 5.4.3.4), may be set to open at 10 % above MAWP. In this case, the backpressure correction factor for 10 % overpressure shall be used because the valve is actually operating at 10 % overpressure, even though the accumulation is at 21 %. When calculating the additional capacity for the first (non-fire) valve at 21 % overpressure, a backpressure correction factor of 1.0 may be used (for backpressures up to 50 % of set pressure, see Figure 30, Note 3).

5.3.3.2.3 The backpressure correction factors specified in Figure 30 and Figure 31 are applicable to balanced spring-loaded PRVs with backpressures up to 50 % of set pressure.

5.3.3.2.4 When backpressures in compressible fluid applications (does not include multiphase applications) exceed approximately 50 % of set pressure, the flow is subcritical. Nonetheless, the critical flow formulas found in 5.6.2 should be used. The PRV manufacturer should be consulted when backpressures exceed approximately 50 % of set pressure to obtain backpressure correction factors or any special limitations on valve operation.

5.3.3.3 Pilot-operated PRVs

For pilot-operated PRVs, the valve lift is not affected by backpressure. For compressible fluids at critical flow conditions, a backpressure correction factor of 1.0 should be used for pilot-operated PRVs.

5.3.4 Effects of Backpressure and Header Design on PRV Sizing and Selection

5.3.4.1 For conventional PRVs connected to a flare header, there are several considerations that affect PRV sizing and selection. The PRV discharge line and flare header shall be designed so that the built-up backpressure does not exceed the allowable limits as specified in 5.3.3. In addition, the flare header system shall be designed in order to ensure that the superimposed backpressure, caused by venting or relief from another source, will not prevent PRVs from opening at a pressure adequate to protect equipment per the ASME or applicable code. Once the superimposed, built-up, and total backpressures are calculated based on a pressure drop analysis of the discharge system, they should be specified on the data sheet for the PRV under consideration.

5.3.4.2 Total backpressure may affect the capacity of the PRV. Sizing a balanced PRV is a two-step process. The PRV is sized using a preliminary backpressure correction factor, K_b. The correction factor could either be set initially equal to 1.0 or can be based on an assumed total backpressure. Once a preliminary valve size and capacity is determined, the discharge line and header size can be determined based on pressure drop calculations. The final size, capacity, backpressure, and backpressure correction factor, K_b, can then be calculated. The backpressure should be included on the data sheet for the PRV under consideration.

5.3.4.3 For a pilot-operated PRV, neither the set pressure nor the capacity is typically affected by backpressure, for compressible fluids at critical flow conditions. Tail pipe and flare header sizing are typically based on other considerations.

5.3.4.4 Outlet pipe sizing and flare header sizing are discussed in more detail in API 520, Part 2 and API 521.

5.4 Relieving Pressure

5.4.1 General

5.4.1.1 Relieving pressure, shown as P_1 in the various sizing equations, is the inlet pressure of the pressure relief device at relieving conditions. The relieving pressure is the total of set pressure plus overpressure. The examples cited in this section for the determination of relieving pressure refer to PRVs, however, they are also applicable to non-reclosing pressure relief devices (see Figure 15 and Figure 18 for pressure level relationships for these types of devices).

5.4.1.2 The allowable overpressure is established from the accumulation permitted by the applicable code. The allowable overpressure may vary for different applications depending on the relationship of the set pressure to the maximum allowable working pressure of the vessel or system that is protected. Although the allowable overpressure differs from the allowable accumulation by the pressure drop between the protected system and the pressure relief device (when the set pressure is equal to the maximum allowable working pressure), this difference is neglected in PRV sizing and selection when the inlet pressure drop doesn't exceed that allowed by API 520 Part II.~~Allowable overpressure is the same as allowable accumulation only when the set pressure is equal to the maximum allowable working pressure.~~

5.4.1.3 The discussion in this section generally cites the ASME Code as the applicable code. Unless stated otherwise, citations refer only to Section VIII of the ASME Code. The designer should be aware of revisions to the ASME Code. If pertinent revisions occur, the discussion in this section should be adjusted accordingly by the designer. Adjustments may also be required by the designer if other (non-ASME) codes apply.

5.4.1.4 Sections 5.4.2 through 5.4.3 discuss methods of determining the relieving pressure for pressure relief devices. In applications where these paragraphs do not apply, alternate accumulations are sometimes specified, as required by other codes or the equipment manufacturer.

5.4.1.5 Table 1 summarizes the maximum accumulation and set pressure for pressure relief devices specified in accordance with the ASME Code.

Table 1—Set Pressure and Accumulation Limits for Pressure Relief Devices

Contingency	Single Device Installations		Multiple Device Installations	
	Maximum Set Pressure %	Maximum Accumulated Pressure %	Maximum Set Pressure %	Maximum Accumulated Pressure %
Non-fire Case				
First relief device	100	110	100	116
Additional device(s)	—	—	105	116
Fire Case				
First relief device	100	121	100	121
Additional device(s)	—	—	105	121
Supplemental device	—	—	110	121
NOTE All values are percentages of the maximum allowable working pressure.				

5.4.2 Operating Contingencies

5.4.2.1 Single-device Installation

5.4.2.1.1 In accordance with the requirements of the ASME Code, accumulated pressure shall be limited to 110 % of the maximum allowable working pressure, MAWP, in vessels that are protected by a single pressure relief device sized for operating (non-fire) contingencies. The set pressure of the device shall not exceed the MAWP.

5.4.2.1.2 The allowable accumulation is 3 psi (21 kPa) when the MAWP is between 15 psig and 30 psig (103 kPag and 207 kPag) in accordance with the ASME Code.

5.4.2.1.3 Table 2 shows an example determination of relieving pressure for a single device whose set pressure is less than or equal to the vessel's MAWP.

Table 2—Example Determination of Relieving Pressure for Operating Contingencies for a Single Relief Device Installation

Characteristic	Value
Relief Device Set Pressure Equal to MAWP	
Protected vessel MAWP, psig (kPag)	100.0 (689)
Maximum accumulated pressure, psig (kPag)	110.0 (758)
Relief device set pressure, psig (kPag)	100.0 (689)
Allowable overpressure, psi (kPa)	10.0 (69)

Barometric pressure, psia (kPa)	14.7 (101)
Relieving pressure, P_1, psia (kPa)	124.7 (860)
Relief Device Set Pressure Less Than MAWP	
Protected vessel MAWP, psig (kPag)	100.0 (689)
Maximum accumulated pressure, psig (kPag)	110.0 (758)
Relief device set pressure, psig (kPag)	90.0 (621)
Allowable overpressure, psi (kPa)	20.0 (138)
Barometric pressure, psia (kPa)	14.7 (101)
Relieving pressure, P_1, psia (kPa)	124.7 (860)
NOTE The above examples assume a barometric pressure of 14.7 psia (101.3 kPa). The barometric pressure corresponding to site elevation should be used.	

5.4.2.2 Multiple-device Installation

5.4.2.2.1 A multiple-device installation requires the combined capacity of two or more pressure relief devices to alleviate a given overpressure contingency.

5.4.2.2.2 In accordance with the requirements of the ASME Code accumulated pressure shall be limited to 116 % of the maximum allowable working pressure, MAWP in vessels that are protected by multiple pressure relief devices sized for operating (non-fire) contingencies. The set pressure of the first device shall not exceed the MAWP. The set pressure of the additional device or devices shall not exceed 105 % of the MAWP.

5.4.2.2.3 The allowable accumulation is 4 psi (28 kPa) when the MAWP is between 15 psig and 30 psig (103 kPag and 207 kPag).

5.4.2.2.4 Table 3 shows an example determination of the relieving pressure for a multiple-device installation in which the set pressure of the first device is equal to the MAWP of the vessel, and the set pressure of the additional device is 105 % of the vessel's MAWP.

Table 3—Example Determination of Relieving Pressure for Operating Contingencies for a Multiple Relief Device Installation

Characteristic	Value
First Relief Device (Set Pressure Equal to MAWP)	
Protected vessel MAWP, psig (kPag)	100.0 (689)
Maximum accumulated pressure, psig (kPag)	116.0 (800)
Relief device set pressure, psig (kPag)	100.0 (689)
Allowable overpressure, psi (kPa)	16.0 (110)
Barometric pressure, psia (kPa)	14.7 (101)
Relieving pressure, P_1, psia (kPa)	130.7 (901)
Additional Relief Device (Set Pressure Equal to 105 % of MAWP)	
Protected vessel MAWP, psig (kPag)	100.0 (689)
Maximum accumulated pressure, psig (kPag)	116.0 (800)

Relief device set pressure, psig (kPag)	105.0 (724)
Allowable overpressure, psi (kPa)	11.0 (76)
Barometric pressure, psia (kPa)	14.7 (101)
Relieving pressure, P_1, psia (kPa)	130.7 (901)
NOTE The above examples assume a barometric pressure of 14.7 psia (101.3 kPa). The barometric pressure corresponding to site elevation should be used.	

5.4.3 Fire Contingencies

5.4.3.1 General

5.4.3.1.1 In accordance with the requirements of the ASME Code accumulated pressure shall be limited to 121 % of the maximum allowable working pressure, MAWP, in vessels that are protected by pressure relief devices sized for fire contingencies. This applies to single-, multiple-, and supplemental-device installations.

5.4.3.1.2 Single or multiple devices sized for fire may also be utilized for relieving requirements attributed to operating (non-fire) contingencies, provided that the constraint of 110 % and 116 % (of the MAWP) accumulated pressure for the non-fire contingencies is observed.

5.4.3.2 Single-device Installation

5.4.3.2.1 Where a vessel is protected by a single device sized for fire, the set pressure shall not exceed the maximum allowable working pressure, MAWP.

5.4.3.2.2 Table 4 shows an example determination of relieving pressure for a single device whose set pressure is less than or equal to the vessel's MAWP.

Table 4—Example Determination of Relieving Pressure for Fire Contingencies for a Single Relief Device Installation

Characteristic	Value
Relief Device Set Pressure Equal to MAWP	
Protected vessel MAWP, psig (kPag)	100.0 (689)
Maximum accumulated pressure, psig (kPag)	121.0 (834)
Relief device set pressure, psig (kPag)	100.0 (689)
Allowable overpressure, psi (kPa)	21.0 (145)
Barometric pressure, psia (kPa)	14.7 (101)
Relieving pressure, P_1, psia (kPa)	135.7 (936)
Relief Device Set Pressure Less Than MAWP	
Protected vessel MAWP, psig (kPag)	100.0 (689)
Maximum accumulated pressure, psig (kPag)	121.0 (834)
Relief device set pressure, psig (kPag)	90.0 (621)
Allowable overpressure, psi (kPa)	31.0 (214)
Barometric pressure, psia (kPa)	14.7 (101)
Relieving pressure, P_1, psia (kPa)	135.7 (936)

NOTE The above examples assume a barometric pressure of 14.7 psia (101.3 kPa). The barometric pressure corresponding to site elevation should be used.

5.4.3.3 Multiple-device Installation

5.4.3.3.1 A multiple-device installation requires the combined capacity of two or more devices to alleviate overpressure. The set pressure of the first device to open shall not exceed the maximum allowable working pressure, MAWP. The set pressure of the last device to open shall not exceed 105 % of the MAWP.

5.4.3.3.2 Table 5 shows an example determination of relieving pressure for a multiple-device installation in which the set pressure of the first device is equal to the vessel's MAWP, and the set pressure of the additional device is 105 % of the vessel's MAWP.

Table 5—Example Determination of Relieving Pressure for Fire Contingencies for a Multiple Relief Device Installation

Characteristic	Value
First Relief Device (Set Pressure Equal to MAWP)	
Protected vessel MAWP, psig (kPag)	100.0 (689)
Maximum accumulated pressure, psig (kPag)	121.0 (834)
Relief device set pressure, psig (kPag)	100.0 (689)
Allowable overpressure, psi (kPa)	21.0 (145)
Barometric pressure, psia (kPa)	14.7 (101)
Relieving pressure, P_1, psia (kPa)	135.7 (936)
Additional Relief Device (Set Pressure Equal to 105 % of MAWP)	
Protected vessel MAWP, psig (kPag)	100.0 (689)
Maximum accumulated pressure, psig (kPag)	121.0 (834)
Relief device set pressure, psig (kPag)	105.0 (724)
Allowable overpressure, psi (kPa)	16.0 (110)
Barometric pressure, psia (kPa)	14.7 (101)
Relieving pressure, P_1, psia (kPa)	135.7 (936)

NOTE The above examples assume a barometric pressure of 14.7 psia (101.3 kPa). The barometric pressure corresponding to site elevation should be used.

Table 6—Example Determination of Relieving Pressure for Fire Contingencies for a Supplemental Valve Installation

Characteristic	Value
First Relief Device (Set Pressure Equal to MAWP)	
Protected vessel MAWP, psig (kPag)	100.0 (689)
Maximum accumulated pressure, psig (kPag)	121.0 (834)
Relief Device set pressure, psig (kPag)	100.0 (689)
Allowable overpressure, psi (kPa)	21.0 (145)
Barometric Pressure, psia (kPa)	14.7 (101)
Relieving pressure, P_1, psia (kPa)	135.7 (936)
Supplemental Relief Device (Set Pressure Equal to 110 % MAWP)	
Protected vessel MAWP, psig (kPag)	100.0 (689)
Maximum accumulated pressure, psig (kPag)	121.0 (834)
Relief Device set pressure, psig (kPag)	110.0 (758)
Allowable overpressure, psi (kPa)	11.0 (76)
Barometric Pressure, psia (kPa)	14.7 (101)
Relieving pressure, P_1, psia (kPa)	135.7 (936)
NOTE The above examples assume a barometric pressure of 14.7 psia (101.3 kPa). The barometric pressure corresponding to site elevation should be used.	

5.4.3.4 Supplemental-device Installation

5.4.3.4.1 A supplemental-device installation provides relieving capacity for an additional hazard created by exposure to fire or other unexpected sources of external heat. The set pressure of a supplemental device for fire shall not exceed 110 % of the maximum allowable working pressure, MAWP.

5.4.3.4.2 Supplemental devices are used only in addition to devices sized for operating (non-fire) contingencies.

5.4.3.4.2

5.4.3.4.3 Table 6 shows an example determination of relieving pressure for a supplemental device installation in which the set pressure of the first (non-fire) device does not exceed the vessel's MAWP (see 5.4.1 for determination of relieving pressure), and the set pressure of the supplemental device is 110 % of the vessel's MAWP.

5.5 Development of Sizing Equations

5.5.1 The assumption of isentropic nozzle flow for a homogeneous fluid provides a standard theoretical framework for PRV sizing equations. See Annex B for more information regarding the assumptions and/or simplifications that are made to the isentropic nozzle flow equation which have resulted in the analytical equations presented in 5.6 through 5.10.

5.5.2 Annex C provides additional information on the development of the sizing equations for two-phase flow.

5.6 Sizing for Gas or Vapor Relief

5.6.1 Applicability

The sizing equations for pressure relief devices in vapor or gas service provided in this section assume that the pressure-specific volume relationship along an isentropic path is well described by the expansion relation,

PV^k = constant

where

k is the ideal gas specific heat ratio at the relieving temperature.

Years of experience with this basis indicates that this approach has provided satisfactory results over a wide range of conditions. However, the validity of the assumption may diminish at very high pressures or as the vapor or gas approaches the thermodynamic critical locus. One indicator that the vapor or gas may be in one of these regions is a compressibility factor, Z, less than approximately 0.8, or greater than approximately 1.1. To ensure the most appropriate sizing results, users should establish the limits of applicability for their own systems. See Annex B for guidance on sizing when this section is not applicable.

5.6.2 Critical Flow Behavior

5.6.2.1 If a compressible gas is expanded across a nozzle, an orifice, or the end of a pipe, its velocity and specific volume increase with decreasing downstream pressure. For a given set of upstream conditions (using the example of a nozzle), the mass rate of flow through the nozzle will increase until a limiting velocity is reached in the nozzle. It can be shown that the limiting velocity is the velocity of sound in the flowing fluid at that location. The flow rate that corresponds to the limiting velocity is known as the critical flow rate.

5.6.2.2 The absolute pressure ratio of the pressure at the nozzle exit at sonic velocity (P_{cf}) to the inlet pressure (P_1) is called the critical pressure ratio. P_{cf} is known as the critical flow pressure.

5.6.2.3 Under critical flow conditions, the actual pressure at the nozzle exit of the pressure relief device cannot fall below the critical flow pressure even if a much lower pressure exists downstream. At critical flow, the expansion from nozzle pressure to downstream pressure takes place irreversibly with the energy dissipated in turbulence into the surrounding fluid.

5.6.2.4 The critical flow pressure ratio in absolute units may be estimated using the ideal gas relationship in Equation (1), provided the expansion law, PV^k = constant, is a good approximation of the pressure/specific volume relationship:

$$\frac{P_{cf}}{P_1} = \left[\frac{2}{k+1}\right]^{\frac{k}{k-1}} \tag{1}$$

where

P_{cf} is the critical flow nozzle pressure;

P_1 is the upstream relieving pressure;

k is the ratio of specific heats (C_p/C_v) for an ideal gas at relieving temperature.

The ideal gas specific heat ratio is independent of pressure. Most process simulators will provide real gas specific heats which should not be used in the above equation because the real gas specific heat ratio does not provide a good representation of the isentropic expansion coefficient (see Annex B).

5.6.2.5 The sizing equations for pressure relief devices in vapor or gas service fall into two general categories depending on whether the flow is critical or subcritical. If the pressure downstream of the nozzle is less than, or equal to, the critical flow pressure, P_{cf}, then critical flow will occur, and the procedures in 5.6.3 should be applied. If the downstream pressure exceeds the critical flow pressure, P_{cf}, then subcritical flow will occur, and the procedures in 5.6.4 or 5.6.5 should be applied. See Table 7 for typical critical flow pressure ratio values.

Table 7—Properties of Gases

Gas	Molecular Weight	Ideal Gas Specific Heat Ratio ($k = C_P/C_v$) at 60 °F and One Atmosphere	Ideal Gas Critical Flow Pressure Ratio at 60 °F and One Atmosphere	Ideal Gas Specific Gravity at 60 °F and One Atmosphere	Critical Constants		Condensation Temperature One Atmosphere °F (°C)	Flammability Limits (volume % in air mixture)
					Pressure psia (kPa)	Temperature °F (°C)		
Methane a	16.04	1.31	0.54	0.554	673 (4640)	–116 (–82)	–259 (–162)	5.0 – 15.0
Ethane [4]a	30.07	1.19	0.57	1.058	718 (4950)	90 (32)	–128 (–89)	2.9 – 13.8
Ethylene a	28.03	1.24	0.57	0.969	742 (5116)	50 (10)	–155 (–104)	2.7 – 34.8
Propane a	44.09	1.13	0.58	1.522	617 (4254)	206 (97)	–44 (–42)	2.1 – 9.5
Propylene	42.08	1.15	0.58	1.453	667 (4599)	197 (92)	–54 (–48)	2.8 – 10.8
Isobutane a	58.12	1.10	0.59	2.007	529 (3647)	273 (134)	11 (–12)	1.8 – 8.4
n-Butane a	58.12	1.09	0.59	2.007	551 (3799)	304 (151)	31 (–1)	1.9 – 8.4
1-Butene	56.10	1.11	0.59	1.937	586 (4040)	296 (147)	21 (–6)	1.4 – 9.3
Isopentane a	72.15	1.08	0.59	2.491	483 (3330)	369 (187)	82 (28)	1.4 – 8.3
n-Pentane a	72.15	1.08	0.59	2.491	490 (3378)	386 (197)	97 (36)	1.4 – 7.8
1-Pentene a	70.13	1.08	0.59	2.421	510 (3930)	377 (192)	86 (30)	1.4 – 8.7
n-Hexane a	86.18	1.06	0.59	2.973	437 (3013)	454 (234)	156 (69)	1.2 – 7.7
Benzene	78.11	1.12	0.58	2.697	714 (4923)	552 (289)	176 (80)	1.3 – 7.9
n-Heptane a	100.20	1.05	0.60	3.459	397 (2737)	513 (267)	209 (98)	1.0 – 7.0

[a] Estimated

[4]

Toluene	92.13	1.09	0.59	3.181	590 (4068)	604 (318)	231 (111)	1.2 – 7.1
n-Octane a	114.22	1.05	0.60	3.944	362 (2496)	564 (296)	258 (126)	0.96 – 6.7
n-Nonane	128.23	1.04	0.60	4.428	332 (2289)	610 (321)	303 (151)	0.87 – 2.9
n-Decane	142.28	1.03	0.60	4.912	304 (2096)	632 (333)	345 (174)	0.78 – 2.6
Air	28.96	1.40	0.53	1.000	547 (3771)	–221 (–141)	–313 (–192)	—
Ammonia	17.03	1.30	0.53	0.588	1636 (11280)	270 (132)	–28 (–33)	15.5 – 27.0
Carbon Dioxide	44.01	1.29	0.55	1.519	1071 (7384)	88 (31)	–109 (–78)	—
Hydrogen	2.02	1.41	0.52	0.0696	188 (1296)	–400 (–240)	–423 (–253)	4.0 – 74.2
Hydrogen Sulfide	34.08	1.32	0.53	1.176	1306 (9005)	213 (101)	–77 (–61)	4.3 – 45.5
Sulfur Dioxide	64.04	1.27	0.55	2.212	1143 (7881)	316 (158)	14 (–10)	—
Steam	18.01	1.33	0.54	0.622	3206 (22104)	706 (374)	212 (100)	—

5.6.3 Sizing for Critical Flow

5.6.3.1 General

5.6.3.1.1 Pressure relief devices in gas or vapor service that operate at critical flow conditions (see 5.6.1) may be sized using Equation (2) through Equation (7). Each of the equations may be used to calculate the effective discharge area, A, required to achieve a required flow rate through a pressure relief device. A PRV that has an effective discharge area equal to or greater than the calculated value of A is then chosen for the application.

In USC units:

$$A = \frac{W}{CK_dP_1K_bK_c}\sqrt{\frac{TZ}{M}} \quad (2)$$

$$A = \frac{V\sqrt{TZM}}{6.32CK_dP_1K_bK_c} \quad (3)$$

$$A = \frac{V\sqrt{TZG_v}}{1.175CK_dP_1K_bK_c} \quad (4)$$

In SI units:

$$A = \frac{W}{CK_dP_1K_bK_c}\sqrt{\frac{TZ}{M}} \quad (5)$$

$$A = \frac{2.676 \times V\sqrt{TZM}}{CK_dP_1K_bK_c} \quad (6)$$

$$A = \frac{14.41 \times V\sqrt{TZG_v}}{CK_dP_1K_bK_c} \quad (7)$$

where

A is the required effective discharge area of the device, in.^{2} (mm^2) (see 3.18);

W is the required flow through the device, lb/h (kg/h);

C is a function of the ratio of the ideal gas specific heats ($k = C_p/C_v$) of the gas or vapor at inlet relieving temperature.

The coefficient, C, is determined as follows.

In USC units [for use in Equation (2) through Equation (4) only]:

$$C = 520\sqrt{k\left(\frac{2}{k+1}\right)^{\frac{(k+1)}{(k-1)}}} \quad (8)$$

In SI units [for use in Equation (5) through Equation (7) only]:

$$C = 0.03948\sqrt{k\left(\frac{2}{k+1}\right)^{\frac{(k+1)}{(k-1)}}} \quad (9)$$

The ideal gas specific heat ratio is independent of pressure. Most process simulators will provide real gas specific heats which should not be used in the above equation; otherwise the pressure relief device may be undersized. The value of C can be obtained from Figure 32 or Table 8. For ideal gases, where k cannot be established, it is suggested that a conservative value of C equal to 315 (0.0239) be used. The units for C are as follows.

Table 8—Values of Coefficient *C*

k	*C*		*k*	*C*		*k*	*C*		*k*	*C*	
	USC	**SI**		**USC**	**SI**		**USC**	**SI**		**USC**	**SI**
1.00	315	0.0239	1.26	343	0.0261	1.51	365	0.0277	1.76	384	0.0292
1.01	317	0.0240	1.27	344	0.0261	1.52	366	0.0278	1.77	385	0.0292
1.02	318	0.0241	1.28	345	0.0262	1.53	367	0.0279	1.78	386	0.0293
1.03	319	0.0242	1.29	346	0.0263	1.54	368	0.0279	1.79	386	0.0293
1.04	320	0.0243	1.30	347	0.0263	1.55	369	0.0280	1.80	387	0.0294
1.05	321	0.0244	1.31	348	0.0264	1.56	369	0.0280	1.81	388	0.0294
1.06	322	0.0245	1.32	349	0.0265	1.57	370	0.0281	1.82	389	0.0295
1.07	323	0.0246	1.33	350	0.0266	1.58	371	0.0282	1.83	389	0.0296
1.08	325	0.0246	1.34	351	0.0266	1.59	372	0.0282	1.84	390	0.0296
1.09	326	0.0247	1.35	352	0.0267	1.60	373	0.0283	1.85	391	0.0297
1.10	327	0.0248	1.36	353	0.0268	1.61	373	0.0283	1.86	391	0.0297
1.11	328	0.0249	1.37	353	0.0268	1.62	374	0.0284	1.87	392	0.0298
1.12	329	0.0250	1.38	354	0.0269	1.63	375	0.0285	1.88	393	0.0298
1.13	330	0.0251	1.39	355	0.0270	1.64	376	0.0285	1.89	393	0.0299
1.14	331	0.0251	1.40	356	0.0270	1.65	376	0.0286	1.90	394	0.0299
1.15	332	0.0252	1.41	357	0.0271	1.66	377	0.0286	1.91	395	0.0300
1.16	333	0.0253	1.42	358	0.0272	1.67	378	0.0287	1.92	395	0.0300

1.17	334	0.0254	1.43	359	0.0272	1.68	379	0.0287	1.93	396	0.0301
1.18	335	0.0254	1.44	360	0.0273	1.69	379	0.0288	1.94	397	0.0301
1.19	336	0.0255	1.45	360	0.0274	1.70	380	0.0289	1.95	397	0.0302
1.20	337	0.0256	1.46	361	0.0274	1.71	381	0.0289	1.96	398	0.0302
1.21	338	0.0257	1.47	362	0.0275	1.72	382	0.0290	1.97	398	0.0302
1.22	339	0.0258	1.48	363	0.0276	1.73	382	0.0290	1.98	399	0.0303
1.23	340	0.0258	1.49	364	0.0276	1.74	383	0.0291	1.99	400	0.0303
1.24	341	0.0259	1.50	365	0.0277	1.75	384	0.0291	2.00	400	0.0304
1.25	342	0.0260	—	—	—	—	—	—	—	—	—

NOTE 1 Values of C in USC units apply to Equation (2), Equation (3), and Equation (4) only.

NOTE 2 Values of C in SI units apply to Equation (5), Equation (6), and Equation (7) only.

Figure 32—Curve for Evaluating Coefficient C in the Flow Equation from the Specific Heat Ratio, Assuming Ideal Gas Behavior (USC Units)

Figure 33—Curve for Evaluating Coefficient C in the Flow Equation from the Specific Heat Ratio, Assuming Ideal Gas Behavior (SI Units)

In USC units:

$$\frac{\sqrt{\mathrm{lb}_m \times \mathrm{lb-mole} \times {}^\circ\mathrm{R}}}{\mathrm{lb}_f \times \mathrm{hr}}$$

In SI units:

$$\frac{\sqrt{\mathrm{kg} \times \mathrm{kg-mole} \times K}}{\mathrm{mm}^2 \times \mathrm{hr} \times \mathrm{kPa}}$$

K_d is the effective coefficient of discharge;

for preliminary sizing, use the following values:

— 0.975, when a PRV is installed with or without a rupture disk in combination,

— 0.62, when a PRV is not installed and sizing is for a rupture disk in accordance with 5.11.1.2.1.

P_1 is the upstream relieving pressure, psia (kPa);

this is the set pressure plus the allowable overpressure (see 5.4) plus atmospheric pressure.

K_b is the capacity correction factor due to backpressure;

this can be obtained from the manufacturer's literature or estimated for preliminary sizing from Figure 30. The backpressure correction factor applies to balanced bellows valves only. For conventional and pilot-operated valves, use a value for K_b equal to 1.0 (see 5.3). See 5.6.4 for conventional valve applications with backpressure of a magnitude that will cause subcritical flow.

K_c is the combination correction factor for installations with a rupture disk upstream of the PRV (see 5.11.2);

K_c equals 1.0 when a rupture disk is not installed.

K_c equals 0.9 when a rupture disk is installed in combination with a PRV and the combination does not have a certified value.

T is the relieving temperature of the inlet gas or vapor, $°R$ $(°F + 460)$ $[K$ $(°C + 273)]$;

Z is the compressibility factor for the deviation of the actual gas from a perfect gas, a ratio evaluated at inlet relieving conditions;

M is the molecular weight of the gas or vapor at inlet relieving conditions;

various handbooks carry tables of molecular weights of materials, but the composition of the flowing gas or vapor is seldom the same as that listed in tables. This value should be obtained from the process data. Table 7 lists values for some common fluids, lbm/lb-mole (kg/kg-mole).

V is the required flow through the device, scfm at 14.7 psia and 60 °F (Nm^3/min at 0 °C and 101.325 kPa);

G_v is the specific gravity of gas at standard conditions referred to air at standard conditions (normal conditions). In other words, G_v = 1.00 for air at 14.7 psia and 60 °F (101.325 kPa and 0 °C).

5.6.3.1.2 While ideal gas law behavior (with compressibility factor, Z, included) is generally acceptable for the majority of refinery applications, Annex B should be referred to for unusual situations in which deviation from ideal behavior is significant.

5.6.3.2 Example 1

5.6.3.2.1 In this example, the following relief requirements are given:

a) required hydrocarbon vapor flow, W, caused by an operational upset, of 53,500 lb/h (24,270 kg/h);

b) the hydrocarbon vapor is a 50/50 (by mole) mixture of n-butane (C_4) and propane (C_3). The molecular weight of the vapor, M, is 51;

c) relieving temperature, T, of 627 °R (167 °F) (348 K);

d) PRV set at 75 psig (517 kPa), which is the design pressure of the equipment;

e) backpressure of 14.7 psia (0 psig) [101.325 kPa (0 kPag)];

f) overpressure of 10 %.

5.6.3.2.2 In this example, the following data are derived:

a) relieving pressure, P_1, of 75 × 1.1 + 14.7 = 97.2 psia (670 kPa);

b) calculated compressibility, Z, of 0.90 (If a calculated compressibility is not available, a Z value of 1.0 should be used);

c) critical flow pressure (from Table 7) of 97.2 × 0.585 = 56.9 psia (42.2 psig) (392 kPa);

NOTE Since the backpressure [0 psig (0 kPag)] is less than the critical flow pressure [42.2 psig (291 kPag)], the PRV sizing is based on the critical flow equation [see Equation (2) and 5.6.1 and 5.6.2].

d) $C_p/C_v = k$ (from Table 7) of 1.11. From Table 8, C = 328 (0.0249);

NOTE For this example problem, k was obtained from Table 7 at standard conditions, which should result in a conservative answer. If the k value is known at the relieving temperature, use this value instead. In this case, the ideal gas specific heat ratio is a good approximation of the isentropic exponent for the purposes of this calculation. See Appendix B for further discussion.

e) capacity correction due to backpressure, Kb, of 1.0;

f) capacity correction for rupture disk, Kc = 1.0

5.6.3.2.3 The size of a single PRV is calculated using Equation (2) or Equation (5) as follows.

In USC units:

$$A = \frac{53{,}500}{328 \times 0.975 \times 97.2 \times 1.0 \times 1.0}\sqrt{\frac{627 \times 0.90}{51}} = 5.73 \text{ in.}^2 \tag{10}$$

In SI units:

$$A = \frac{24{,}270}{0.0249 \times 0.975 \times 670 \times 1.0 \times 1.0}\sqrt{\frac{348 \times 0.90}{51}} = 3698 \text{ mm}^2 \tag{11}$$

5.6.3.2.4 See API 526 for selection of the proper orifice size. API 526 provides standard effective orifice areas in terms of letter designations. For this example, a "P" size orifice should be selected since it has an effective orifice area of 6.38 in.^{2} (4116 mm^2).

5.6.3.2.5 A completed PRV specification sheet for this example is provided in Figure 34 (a blank specification sheet is provided in Annex D).

Figure 34—Sample of Completed PRV Specification Sheet

5.6.4 Sizing for Subcritical Flow: Gas or Vapor

5.6.4.1 Conventional and Pilot-operated PRVs

When the ratio of backpressure to inlet pressure exceeds the critical pressure ratio P_{cf}/P_1, the flow through the pressure relief device is subcritical (see 5.6.1). Equation (12) through Equation (17) may be used to calculate the required effective discharge area for a conventional PRV that has its spring setting adjusted to compensate for superimposed backpressure. Equation (12) through Equation (17) may also be used for sizing a pilot-operated PRV.

In USC units:

$$A = \frac{W}{735 \times F_2 K_d K_c}\sqrt{\frac{ZT}{M \times P_1(P_1 - P_2)}} \tag{12}$$

$$A = \frac{V}{4645 \times F_2 K_d K_c}\sqrt{\frac{ZTM}{P_1(P_1 - P_2)}} \tag{13}$$

$$A = \frac{V}{864 \times F_2 K_d K_c}\sqrt{\frac{ZTG_v}{P_1(P_1 - P_2)}} \tag{14}$$

In SI units:

$$A = \frac{17.9 \times W}{F_2 K_d K_c}\sqrt{\frac{ZT}{M \times P_1(P_1 - P_2)}} \tag{15}$$

$$A = \frac{47.95 \times V}{F_2 K_d K_c}\sqrt{\frac{ZTM}{P_1(P_1 - P_2)}} \tag{16}$$

$$A = \frac{258 \times V}{F_2 K_d K_c}\sqrt{\frac{ZTG_v}{P_1(P_1 - P_2)}} \tag{17}$$

where

A is the required effective discharge area of the device, in.^{2} (mm^2) (see 3.18);

W is the required flow through the device, lb/h (kg/h);

F_2 is the coefficient of subcritical flow, see Figure 35 for values, or use Equation (18).

Figure 35—Values for F_2 for Subcritical Flow

$$F_2 = \sqrt{\left(\frac{k}{k-1}\right) r^{\left(\frac{2}{k}\right)} \left[\frac{1 - r^{\left(\frac{k-1}{k}\right)}}{1 - r}\right]} \tag{18}$$

where

k is the ratio of the specific heats (C_p/C_v) for an ideal gas at relieving temperature;

the ideal gas specific heat ratio is independent of pressure. Most process simulators can provide real gas-specific heats, which should not be used in Equation (18) because the real gas specific heat ratio does not provide a good representation of the isentropic expansion coefficient (see Annex B).;

r is the ratio of backpressure to upstream relieving pressure, P_2/P_1;

K_d is the effective coefficient of discharge; for preliminary sizing, use the following values:

— 0.975, when a PRV is installed with or without a rupture disk in combination,

— 0.62, when a PRV is not installed and sizing is for a rupture disk in accordance with 5.11.1.2.

K_c is the combination correction factor for installations with a rupture disk upstream of the PRV (see 5.11.2);

use the following values for the combination correction:

— 1.0, when a rupture disk is not installed;

— 0.9, when a rupture disk is installed in combination with a PRV and the combination does not have a certified value.

Z is the compressibility factor for the deviation of the actual gas from a perfect gas, evaluated at relieving inlet conditions;

T is the relieving temperature of the inlet gas or vapor, °R (°F + 460) [K (°C + 273)];

M is the molecular weight of the gas or vapor;

various handbooks carry tables of molecular weights of materials, but the composition of the flowing gas or vapor is seldom the same as that listed in the tables. This value should be obtained from the process data. Table 7 lists values for some common fluids, lbm/lb-mole (kg/kg-mole);

P_1 is the upstream relieving pressure, psia (kPa);

this is the set pressure plus the allowable overpressure (see 5.4) plus atmospheric pressure.

P_2 is the backpressure, psia (kPa);

V is the required flow through the device, SCFM at 14.7 psia and 60 °F (Nm^3/min at 101.325 kPa and 0 °C);

G_v is the specific gravity of gas at standard conditions referred to air at standard conditions (normal conditions) i.e. G_v = 1.00 for air at 14.7 psia and 60 °F (101.325 kPa and 0 °C).

5.6.4.2 Example 2

5.6.4.2.1 In this example, the following relief requirements are given.

a) required hydrocarbon vapor flow, W, caused by an operational upset, of 53,500 lb/h (24270 kg/h).

b) the hydrocarbon vapor is a 50/50 (by mole) mixture of n-butane (C_4) and propane (C_3). The molecular weight of the vapor, M, is 51.

c) relieving temperature, T, of 627 °R (167 °F) (348 K).

d) PRV set at 75 psig (517 kPag), which is the design pressure of the equipment.

e) constant backpressure of 55 psig (379 kPa);

e) for a conventional valve, the spring setting of the valve should be adjusted according to the amount of constant backpressure obtained. In this example, the cold differential test pressure, CDTP, would be 20 psig (138 kPa).

f) overpressure of 10 %.

5.6.4.2.2 In this example, the following data are derived.

a) relieving pressure, P_1, of 75 × 1.1 + 14.7 = 97.2 psia (670 kPa).

b) calculated compressibility, Z, of 0.90 (If a calculated compressibility is not available, a value for Z of 1.0 should be used).

c) critical backpressure (from Table 7) of 97.2 × 0.585 = 56.9 psia (42.2 psig) [392 kPa (291 kPag)].

NOTE Since the backpressure [55 psig (379 kPag)] is greater than the critical backpressure [42.2 psig (291 kPag)], the PRV sizing is based on the subcritical flow equation [see Equation (12) and 5.6.2 and 5.6.4].

d) permitted built-up backpressure of 0.10 × 75 = 7.5 psi (51.7 kPa).

d) The actual built-up backpressure should be used if known.

e) total backpressure of 55 + 7.5 + 14.7 = 77.2 psia (532 kPa).

f) $C_p/C_v = k$ (from Table 7) of 1.11;

f) for this example problem, k was obtained from Table 7 at standard conditions, which should result in a conservative answer. If the k value is known at the relieving temperature, use this value instead.

g) P_2/P_1 = 77.2/97.2 = 0.794.

h) coefficient of subcritical flow, F_2, of 0.86 (from Figure 35).

i) capacity correction for rupture disk, K_c = 1.0.

5.6.4.2.3 The size of a single PRV is derived from Equation (12) as follows:

$$A = \frac{53{,}500}{735 \times 0.86 \times 0.975 \times 1.0}\sqrt{\frac{0.90 \times 627}{51 \times 97.2 \times (97.2 - 77.2)}} = 6.55 \text{ in.}^2 (4226 \text{ mm}^2) \tag{19}$$

5.6.4.2.4 For selection of the proper orifice size, see API 526. For this example, a "Q" size orifice should be selected since it has an effective orifice area of 11.05 in.^{2} (7129 mm^2).

5.6.4.3 Balanced PRVs

Balanced PRVs should be sized using Equation (2) through Equation (7) in 5.6.3.1.1. The backpressure correction factor in this application accounts for flow velocities that are subcritical as well as the tendency for the disc to drop below full lift (the use of subcritical flow equations are appropriate only where full lift is maintained). The backpressure correction factor, K_b, for this application should be obtained from the manufacturer.

5.6.5 Alternate Sizing Procedure for Conventional and Pilot-operated Valves in Subcritical Flow

5.6.5.1 General

As an alternative to using the subcritical flow equations given in 5.6.3, the critical flow Equation (2) through Equation (7), presented in 5.6.3 may be used to calculate the required effective discharge area of a conventional or pilot-operated PRV used in subcritical service. The area obtained using this alternate sizing procedure is identical to the area obtained using the subcritical flow equations. In this alternate method, the capacity correction factor due to backpressure, K_b, is derived by setting the subcritical flow equation (see 5.6.4) equal to the critical flow equation (see 5.6.3) and algebraically solving for K_b. A graphical presentation of the capacity correction factor, K_b, is given in Figure 36. This alternate sizing procedure allows the designer to use the critical flow equation to calculate the same area obtained with the subcritical flow equation provided K_b is obtained from Figure 36 (instead of a K_b value of 1.0 when the critical flow equations of 5.6.3 are used).

Figure 36—Constant Backpressure Correction Factor, K_b, for Conventional PRVs (Vapors and Gases Only)

It should be noted that this method is used only for the sizing of pilot-operated PRVs and conventional (non-balanced) PRVs that have their spring settings adjusted to compensate for the superimposed backpressure. This method should not be used to size balanced-type valves.

5.6.5.2 Example 3

5.6.5.2.1 In this example, the following relief requirements are given.

a) required hydrocarbon vapor flow, W, caused by an operational upset, of 53,500 lb/h (24,270 kg/h).

b) the hydrocarbon vapor is a mixture of n-butane (C_4) and propane (C_3). The molecular weight of the mixture, M, is 51.

c) relieving temperature, T, of 627 °R (167 °F) [348 K (75 °C)].

d) PRV set at 75 psig (517 kPag), which is the design pressure of the equipment.

e) Constant backpressure of 55 psig (379 kPa).

e) For a conventional valve, the spring setting of the valve should be adjusted according to the amount of constant backpressure obtained. In this case, the valve spring should be adjusted to open in the shop at a CDTP of 20 psig (138 kPag).

5.6.5.2.2 In this example, the following data are derived:

a) permitted accumulation of 10 %;

b) relieving pressure, P_1, of 75 × 1.1 + 14.7 = 97.2 psia (670 kPa);

c) calculated compressibility, Z, of 0.90 (If a calculated compressibility is not available, a value for Z of 1.0 should be used);

d) critical backpressure (from Table 7) of 97.2 × 0.585 = 56.9 psia (42.2 psig) [392 kPa (291 kPag)];

NOTE Since the backpressure [55 psig (379 kPag)] is greater than the critical backpressure [42.2 psig (291 kPag)], the sizing of the PRV is based on subcritical flow. The backpressure correction factor, K_b, should be determined using Figure 36 when the critical flow formulas are used [see Equation (2) through Equation (7)].

e) built-up backpressure of 0.10 × 75 = 7.5 psi (51.7 kPa);

f) total backpressure of 55 + 7.5 + 14.7 = 77.2 psia (532 kPa);

g) $Cp/Cv = k$ of 1.11;

NOTE For this example problem, k was obtained from Table 7 at standard conditions, which should result in a conservative answer. If the k value is known at the relieving temperature, use this value instead).

h) P_2/P_1 = 77.2 /97.2 = 0.794;

i) backpressure correction factor, K_b, of 0.88 (from Figure 36);

j) coefficient determined from an expression of the ratio of the specific heats of the gas or vapor at inlet relieving conditions, C, of 328 (0.0249) (from Table 8);

k) capacity correction for rupture disk, Kc = 1.0.

5.6.5.2.3 The size of the PRV is derived from Equation (2) as follows:

$$A = \frac{53{,}500}{328 \times 0.975 \times 97.2 \times 0.88 \times 1.0}\sqrt{\frac{0.90 \times 627}{51}} = 6.51 \text{ in.}^2 (4197 \text{ mm}^2) \tag{20}$$

NOTE This area requirement is roughly the same as that obtained using the subcritical flow Equation (12). See the example in 5.6.4.2.

5.7 Sizing for Steam Relief

5.7.1 General

Pressure relief devices in steam service that operate at critical flow conditions may be sized using Equation (21) and Equation (22).

In USC units:

$$A = \frac{W}{51.5 \times P_1 K_d K_b K_c K_N K_{SH}} \tag{21}$$

In SI units:

$$A = \frac{190.5 \times W}{P_1 K_d K_b K_c K_N K_{SH}} \tag{22}$$

where

A is the required effective discharge area, in.^{2} (mm^2) (see 5.2.2);

W required flow rate, lb/h (kg/h);

P_1 is the upstream relieving pressure, psia (kPa);

this is the set pressure plus the allowable overpressure (see 5.4) plus the atmospheric pressure.

K_d is the effective coefficient of discharge. For preliminary sizing, use the following values:

— 0.975, when a PRV is installed with or without a rupture disk in combination,

— 0.62, when a PRV is not installed and sizing is for a rupture disk in accordance with 5.11.1.2.1.

K_b is the capacity correction factor due to backpressure;

this can be obtained from the manufacturer's literature or estimated from Figure 30. The backpressure correction factor applies to balanced bellows valves only. For conventional valves, use a value for K_b equal to 1.0 (see 5.3). See 5.6.4 for conventional valve applications that involve superimposed backpressure of a magnitude that will cause subcritical flow;

K_c is the combination correction factor for installations with a rupture disk upstream of the PRV (see 5.11.2);

— the combination correction factor is 1.0, when a rupture disk is not installed,

— the combination correction factor is 0.9, when a rupture disk is installed in combination with a PRV and the combination does not have a certified value.

K_N is the correction factor for the Napier Equation [2] as shown in Equation (23), Equation (24), and Equation (25);

$$K_N = 1.0 \tag{23}$$

where

$P_1 \leq$ 1500 psia (10,339 kPa)

In USC units:

$$K_N = \frac{0.1906 \times P_1 - 1000}{0.2292 \times P_1 - 1061} \tag{24}$$

where

P_1 > 1500 psia (10,339 kPa) and ≤ 3200 psia (22,057 kPa)

In SI units:

$$K_N = \frac{0.02764 \times P_1 - 1000}{0.03324 \times P_1 - 1061} \tag{25}$$

where

P_1 > 1500 psia (10,339 kPa) and ≤ 3200 psia (22,057 kPa);

K_{SH} is the superheat correction factor.

This can be obtained from Table 9. For saturated steam at any pressure, K_{SH} = 1.0. For temperatures above 1200°F, use the critical vapor sizing Equation (2) through Equation (7).

Table 9—Superheat Correction Factors, K_{SH}

	Temperature °F (°C)									
Set Pressure psig (kPag)	**300 (149)**	**400 (204)**	**500 (260)**	**600 (316)**	**700 (371)**	**800 (427)**	**900 (482)**	**1000 (538)**	**1100 (593)**	**1200 (649)**
15 (103)	1.00	0.98	0.93	0.88	0.84	0.80	0.77	0.74	0.72	0.70
20 (138)	1.00	0.98	0.93	0.88	0.84	0.80	0.77	0.74	0.72	0.70
40 (276)	1.00	0.99	0.93	0.88	0.84	0.81	0.77	0.74	0.72	0.70
60 (414)	1.00	0.99	0.93	0.88	0.84	0.81	0.77	0.75	0.72	0.70
80 (551)	1.00	0.99	0.93	0.88	0.84	0.81	0.77	0.75	0.72	0.70

100 (689)	1.00	0.99	0.94	0.89	0.84	0.81	0.77	0.75	0.72	0.70
120 (827)	1.00	0.99	0.94	0.89	0.84	0.81	0.78	0.75	0.72	0.70
140 (965)	1.00	0.99	0.94	0.89	0.85	0.81	0.78	0.75	0.72	0.70
160 (1103)	1.00	0.99	0.94	0.89	0.85	0.81	0.78	0.75	0.72	0.70
180 (1241)	1.00	0.99	0.94	0.89	0.85	0.81	0.78	0.75	0.72	0.70
200 (1379)	1.00	0.99	0.95	0.89	0.85	0.81	0.78	0.75	0.72	0.70
220 (1516)	1.00	0.99	0.95	0.89	0.85	0.81	0.78	0.75	0.72	0.70
240 (1654)	—	1.00	0.95	0.90	0.85	0.81	0.78	0.75	0.72	0.70
260 (1792)	—	1.00	0.95	0.90	0.85	0.81	0.78	0.75	0.72	0.70
280 (1930)	—	1.00	0.96	0.90	0.85	0.81	0.78	0.75	0.72	0.70
300 (2068)	—	1.00	0.96	0.90	0.85	0.81	0.78	0.75	0.72	0.70
350 (2413)	—	1.00	0.96	0.90	0.86	0.82	0.78	0.75	0.72	0.70
400 (2757)	—	1.00	0.96	0.91	0.86	0.82	0.78	0.75	0.72	0.70
500 (3446)	—	1.00	0.96	0.92	0.86	0.82	0.78	0.75	0.73	0.70
600 (4136)	—	1.00	0.97	0.92	0.87	0.82	0.79	0.75	0.73	0.70
800 (5514)	—	—	1.00	0.95	0.88	0.83	0.79	0.76	0.73	0.70
1000 (6893)	—	—	1.00	0.96	0.89	0.84	0.78	0.76	0.73	0.71
1250 (8616)	—	—	1.00	0.97	0.91	0.85	0.80	0.77	0.74	0.71
1500 (10339)	—	—	—	1.00	0.93	0.86	0.81	0.77	0.74	0.71
1750 (12063)	—	—	—	1.00	0.94	0.86	0.81	0.77	0.73	0.70
2000 (13786)	—	—	—	1.00	0.95	0.86	0.80	0.76	0.72	0.69

2500 (17232)	—	—	—	1.00	0.95	0.85	0.78	0.73	0.69	0.66
3000 (20679)	—	—	—	—	1.00	0.82	0.74	0.69	0.65	0.62

5.7.2 Example 4

5.7.2.1 In this example, the relief requirement, W, is the saturated steam at 153,500 lb/h (69,615 kg/h) at 1600 psig (11,032 kPag) set pressure with 10 % accumulation.

NOTE The set pressure is equal to the design pressure in this example.

5.7.2.2 In this example, the following data are derived:

a) relieving pressure, P_1, of 1600 × 1.1 + 14.7 = 1774.7 psia (12,236 kPa);

b) effective coefficient of discharge, K_d, of 0.975;

c) backpressure correction factor, K_b, of 1.0 for conventional valve discharging to atmosphere;

d) capacity Correction for rupture disk, K_c = 1.0, since there is no rupture disk;

e) correction factor for the Napier equation, K_N, of 1.01, calculated using Equation (24);

$$K_N = \frac{0.1906 \times P_1 - 1000}{0.2292 \times P_1 - 1061} = \frac{0.1906 \times (1774.7) - 1000}{0.2292 \times (1774.7) - 1061} = 1.01 \tag{26}$$

f) superheat steam correction factor, K_{SH}, of 1.0.

5.7.2.3 The size of the PRV is derived from Equation (21) as follows:

$$A = \frac{153{,}500}{51.5 \times 1774.7 \times 0.975 \times (1.0) \times (1.0) \times (1.01) \times (1.0)} = 1.705 \text{ in.}^2 (1100 \text{ mm}^2) \tag{27}$$

5.7.2.4 For selection of the proper orifice size, see API 526. For this example, a "K" size orifice should be selected since it has an effective orifice area of 1.838 in.^{2} (1186 mm^2).

5.8 Sizing for Liquid Relief: PRVs Requiring Capacity Certification

5.8.1 General

5.8.1.1 The ASME Code requires that capacity certification be obtained for PRVs designed for liquid service. The procedure for obtaining capacity certification includes testing to determine the rated coefficient of discharge for the liquid PRVs at 10 % overpressure.

5.8.1.2 The sizing equations for pressure relief devices in liquid service provided in this section assume that the liquid is incompressible (i.e. the density of the liquid does not change as the pressure decreases from the relieving pressure to the total backpressure).

5.8.1.3 Valves in liquid service that are designed in accordance with the ASME Code which require a capacity certification may be initially sized using Equation (28) or Equation (29).

In USC units:

$$A = \frac{Q}{38 \times K_d K_w K_c K_v} \sqrt{\frac{G_l}{P_1 - P_2}} \tag{28}$$

In SI units:

$$A = \frac{11.78 \times Q}{K_d K_w K_c K_v} \sqrt{\frac{G_l}{P_1 - P_2}} \tag{29}$$

where

A is the required effective discharge area, in.^{2} (mm^2);

Q is the flow rate, U.S. gal/min (L/min);

K_d is the rated coefficient of discharge that should be obtained from the valve manufacturer;

for preliminary sizing, an effective discharge coefficient can be used as follows:

— 0.65, when a PRV is installed with or without a rupture disk in combination,

— 0.62, when a PRV is not installed and sizing is for a rupture disk in accordance with 5.11.1.2.1.

K_w is the correction factor due to backpressure;

if the backpressure is atmospheric, use a value for K_w of 1.0. Balanced bellows valves in backpressure service will require the correction factor determined from Figure 31. Conventional and pilot-operated valves require no special correction (see 5.3);

K_c is the combination correction factor for installations with a rupture disk upstream of the PRV (see 5.11.2);

use the following values for the combination correction factor:

— 1.0, when a rupture disk is not installed,

— 0.9, when a rupture disk is installed in combination with a PRV and the combination does not have a certified value.

K_v is the correction factor due to viscosity, as determined from Figure 37 or from Equation (30);

$$K_v = \left(0.9935 + \frac{2.878}{Re^{0.5}} + \frac{342.75}{Re^{1.5}}\right)^{-1.0} \tag{30}$$

where

G_l is the specific gravity of the liquid at the flowing temperature referred to water at standard conditions;

P_1 is the upstream relieving pressure, psig (kPag);

this is the set pressure plus allowable overpressure.

P_2 is the total backpressure, psig (kPag).

Figure 37—Capacity Correction Factor, K_v, Due to Viscosity

5.8.1.4 When a PRV is sized for viscous liquid service, it should first be sized as if it were for a non-viscous type application (i.e. K_v = 1.0) so that a preliminary required discharge area, A, can be obtained from Equation (28) or Equation (29). From API 526 standard orifice sizes, the next orifice size larger than A should be used in determining the Reynold's Number, Re, from either of the following relationships.

In USC units:

$$Re = \frac{Q(2800 \times G_l)}{\mu\sqrt{A}} \tag{31}$$

or

$$Re = \frac{12{,}700 \times Q}{U\sqrt{A}} \tag{32}$$

In SI units:

$$Re = \frac{Q(18{,}800 \times G_l)}{\mu\sqrt{A}} \tag{33}$$

or

$$Re = \frac{85{,}220 \times Q}{U\sqrt{A}} \tag{34}$$

where

Re is the Reynold's Number;

Q is the flow rate at the flowing temperature in U.S. gal/min (L/min);

G_l is the specific gravity of the liquid at the flowing temperature referred to water at standard conditions;

μ is the absolute viscosity at the flowing temperature, centipoise;

A is the effective discharge area in in.^{2} (mm^2);

(from API 526 standard orifice areas);

U is the viscosity at the flowing temperature in Saybolt universal seconds (SSU).

Equation (32) and Equation (34) are not recommended for viscosities less than 100 Saybolt universal seconds.

5.8.1.5 After the Reynold's Number, Re, is determined, the factor K_v is obtained from Figure 37. K_v is then applied in Equation (28) or Equation (29) to correct the preliminary required discharge area. If the corrected area exceeds the chosen standard orifice area, the above calculations should be repeated using the next larger standard orifice size.

5.8.2 Example 5

5.8.2.1 In this example, the following relief requirements are given:

a) required crude oil flow caused by blocked discharge, Q, of 1800 gal/min (6814 L/min);

b) the specific gravity, G_l, of the crude oil is 0.90. The viscosity of the crude oil at the flowing temperature is 2000 Saybolt universal seconds;

c) PRV set at 250 psig (1724 kPag), which is the design pressure of the equipment;

d) backpressure is variable from 0 to 50 psig (345 kPag);

e) overpressure of 10 %.

5.8.2.2 In this example, the following data are derived:

a) relieving pressure, P_1, of 1.10 × 250 = 275 psig (1896 kPag);

b) backpressure of (50/250) × 100 = 20 %;

c) a balanced bellows valve should be selected, since the backpressure is variable. From Figure 31, the backpressure capacity correction factor, K_w = 0.97;

d) the effective coefficient of discharge for preliminary sizing, K_d = 0.65.

5.8.2.3 Sizing first for no viscosity correction (K_v = 1.0), the size of the PRV is derived from Equation (28) as follows:

$$A_R = \frac{1800}{38.0 \times 0.65 \times 0.97 \times 1.0 \times 1.0}\sqrt{\frac{0.90}{275-50}} = 4.752 \text{ in.}^2 (3066 \text{ mm}^2) \tag{35}$$

where

A_R is the required area of the PRV without any viscosity correction.

An area of 6.38 in.^{2} (4116 mm^2) using a P orifice should be selected from API 526.

5.8.2.4 The Reynold's Number, Re, is then calculated using Equation (32).

$$Re = \frac{12{,}700 \times 1800}{2000\sqrt{6.38}} = 4525 \tag{36}$$

5.8.2.5 From Figure 37, the viscosity correction factor is determined, K_v = 0.964,

therefore:

$$A = \frac{A_R}{K_V} = \frac{4.752}{0.964} = 4.93 \text{ in.}^2 (3180 \text{ mm}^2) \tag{37}$$

5.9 Sizing for Liquid Relief: PRVs Not Requiring Capacity Certification

5.9.1 Before the ASME Code incorporated requirements for capacity certification, valves were generally sized for liquid service using Equation (38) and Equation (39). This method assumes an effective coefficient of discharge, K_d = 0.62, and 25 % overpressure. An additional capacity correction factor, K_p, is needed for relieving pressures other than 25 % overpressure (see Figure 38). This sizing method may be used where capacity certification is not required or was never established.

Figure 38—Capacity Correction Factors Due to Overpressure for Noncertified PRVs in Liquid Service

5.9.2 This method will typically result in an oversized design where a liquid valve is used for an application with 10 % overpressure (see 4.2.1.2). A K_p correction factor of 0.6 is used (see Figure 38) for this situation.

In USC units:

$$A = \frac{Q}{38 \times K_d K_w K_c K_v K_p} \sqrt{\frac{G_l}{1.25P_s - P_2}} \tag{38}$$

In SI units:

$$A = \frac{11.78 \times Q}{K_d K_w K_c K_v K_p} \sqrt{\frac{G}{1.25P_s - P_2}} \tag{39}$$

where

A is the required effective discharge area, in in.^{2} (mm^2);

Q is the flow rate, in U.S. gal/min (L/min);

K_d is the rated coefficient of discharge that should be obtained from the valve manufacturer;

for a preliminary sizing estimation, an effective discharge coefficient of 0.62 can be used;

K_w is the correction factor due to backpressure. If backpressure is atmospheric, K_w is equal to 1.

Balanced bellows valves in backpressure service will require the correction factor determined from Figure 31. Conventional and pilot-operated valves require no special correction (see 5.3);

K_c is the combination correction factor for installations with a rupture disk upstream of the PRV (see 5.11.2); use a value for K_c equal to 1.0 when a rupture disk does not exist;

K_v is the correction factor due to viscosity as determined from Figure 37 or Equation (30);

K_p is the correction factor due to overpressure;

at 25 % overpressure, K_p is equal to 1.0. For overpressures other than 25 %, K_p is determined from Figure 38;

G_l is the specific gravity of the liquid at the flowing temperature referred to water at standard conditions;

P_s is the set pressure in psig (kPag);

P_2 is the total backpressure in psig (kPag).

5.10 Sizing for Two-phase Liquid/Vapor Relief

5.10.1 A pressure relief device handling a liquid at vapor liquid equilibrium or a mixed phase fluid will produce flashing with vapor generation as the fluid moves through the device. The vapor generation shall be taken into account, since it may reduce the effective mass flow capacity of the device.

5.10.2 Recommended methods for sizing pressure relief devices in two-phase service are presented in Annex C. The user should be aware that there are currently no pressure relief devices with certified capacities for two-phase flow since there are no testing methods for certification.

5.10.3 A balanced or pilot-operated PRV may be necessary when the increase in backpressure due to flashing or two-phase flow conditions is excessive or cannot be adequately predicted. The actual flow rate through a device can be many times higher if equilibrium is not achieved in the nozzle.

5.10.4 For information about saturated water, see ASME Section VIII, Appendix 11.

5.10.5 The designer should also investigate the effect of any auto-refrigeration that may arise from the flashing of liquid. Materials of construction shall be adequate for the outlet temperatures involved; in addition, the installation shall preclude the possibility of flow blockage occurring from hydrate or possibly solid formation.

5.11 Sizing for Rupture Disk Devices

5.11.1 Rupture Disk Devices Used Independently

5.11.1.1 General

Rupture disk devices may be used as the primary relief device for gas, vapor, liquid or multiphase service. The rupture disk size, when used as the sole relieving device shall be determined as specified in 5.11.1.2 or 5.11.1.3. Section 5.11.1.2 may only be used when a rupture disk device discharges directly to the atmosphere, is installed within eight pipe diameters from the vessel nozzle entry, has a length of discharge not greater than five pipe diameters, and has nominal diameters of the inlet and outlet discharge piping equal to or greater than the nominal pipe size of the device. Section 5.11.1.3 applies in all other cases.

5.11.1.2 Rupture Disk Sizing Using Coefficient of Discharge Method (K_d = 0.62)

5.11.1.2.1 The required discharge area, A, in in.^{2} (mm^2), can be calculated using the appropriate equation for the flowing fluid. See Equation (2) through Equation (7) for critical gas or vapor flow; Equation (12) through Equation (17) for subcritical gas or vapor flow; Equation (21) and Equation (22) for steam; Equation (28) or Equation (29) for liquid; and Annex C for two-phase flow.

5.11.1.2.1 When using these equations, a coefficient of discharge, K_d, of 0.62 should be used (see 5.11.1.1 for limitations on using this method).

5.11.1.2.2 The nominal size of the rupture disk device selected shall have a minimum net flow area, MNFA, equal to or greater than the required calculated discharge area. Consult the manufacturer for the minimum net flow area of the rupture disk device.

5.11.1.3 Rupture Disk Sizing Using Flow Resistance Method

The calculated size of a pressure relief system containing a rupture disk device may also be determined by analyzing the total system resistance to flow. This analysis shall take into consideration the flow resistance of the rupture disk device, piping and other piping components, entrance and exit losses (see 5.13), elbows, tees, reducers, and valves. The calculation shall be made using accepted engineering practices for determining fluid flow through piping systems. The calculated relieving capacity shall be multiplied by a factor of 0.90 or less to allow for uncertainties inherent with this method. In these calculations, flow resistance for rupture disk devices can be obtained from the manufacturer. The flow resistance is expressed in terms of velocity head loss (K_R). ASME Code certified values should be used where available.

An example of the flow resistance method is provided in Annex E.

5.11.2 Rupture Disk Devices Used in Combination with PRVs

The capacity of a rupture disk device in combination with a PRV, where the rupture disk device is located at the valve inlet may be determined by multiplying the ASME stamped valve capacity by the combination capacity factor, K_c. K_c values are certified and published by the National Board of Boiler and Pressure Vessel Inspectors for specific disk/valve combinations. When a disk/valve combination does not have a certified K_c then a K_c value of 0.90 shall be used provided the flow area is equal to or greater than the inlet of the PRV.

5.12 Sizing for Open Flow Paths or Vents

The calculated capacity of a pressure relief system containing a flow path or vent open directly or indirectly to the atmosphere may be determined by analyzing the total system resistance to flow. This analysis shall take into consideration the flow resistance of the piping and other piping components, entrance and exit losses (see 5.13), elbows, tees, reducers, and valves. The calculation shall be made using accepted engineering practices for determining fluid flow through piping systems. For pressure vessels within the scope of the ASME Code, this calculated relieving capacity shall be multiplied by a factor of 0.90 or less to allow for uncertainties inherent with this method (See UG-127(d)). Other international codes and standards may have different requirements and should be followed accordingly.

5.13 Use of Exit Loss Resistance Coefficent in Pressure Drop Calculations

For all practical purposes the exit loss does not need to be included when calculating flow through piping exiting to atmosphere [14]. The value of a pipe exit loss has a friction loss term and a term for the kinetic energy change. The friction loss is commonly determined from a K value or velocity head. The kinetic energy change (in pressure units) is proportional to the difference in the densities times velocities to the second power. When both the friction loss term and the kinetic energy change are evaluated, the exit loss may be considered essentially zero for all but the most precise applications. This is because the two terms either are close to zero or the terms cancel each other. This generalization holds for all physical situations including when a gas exits into a gas or liquid, or a liquid exits into a gas or other liquid.

For example, when a gas exits into a liquid reservoir and the gas velocity decelerates to zero, the kinetic energy change is equal to the friction loss. Thus the exit loss is zero. Similarly, when a liquid exits into a gas reservoir, the liquid velocity changes only slightly and there is minimal friction loss. Both the friction and kinetic energy terms are zero with a resulting zero exit loss.

The user is cautioned that when the fluid chokes at the pipe discharge, the pressure at the exit is the pressure required to discharge the choked flow rate through the pipe exit.

Annex A
(informative)

Rupture Disk Device Specification Sheet

A line-by-line description of the information to be provided on the rupture disk device specification sheet is provided in Table A.1, followed by a typical blank specification sheet shown in Figure A.1.

Table A.1—Rupture Disk Device Specification Sheet Instructions (Continued)

Line No.	Instructions
1	*Item Number*: Sequential number from requisition.
2	*Tag Number*: Number assigned to rupture disk which identifies rupture disk location.
3	*Service, Line, or Equipment No.*: Number identifying the service, line, or equipment in which the rupture disk is installed.
4	*Applicable Codes or Standards*: Specify applicable codes or standards (e.g. ASME, API, ISO, etc.) for sizing, marking, burst tolerance, testing, etc.
5	*Vessel or Piping Maximum Allowable Working Pressure*: This pressure is defined in the ASME Code and is specified by the user for the vessel or piping to be protected. This pressure may also be used to evaluate proper sizing and marking.
6	*Fluid*: The process media is used by the user to define compatible materials for rupture disks and holders.
7	*Fluid State (initiating rupture)*: Gas (vapor) or liquid. Some disks are designed to burst with vapor only. The user should consult the rupture disk manufacturer for information about liquid service applications.
8	*Fluid State (relieving conditions)*: Gas (vapor), liquid, or multiphase flow. Users need this information to calculate flow rates and size the rupture disk device.
9	*Required Relieving Capacity*: User to document the required relieving capacity and units for the disk specified. See 5.11 for sizing of rupture disk devices.
10	*Molecular Weight or Specific Gravity (at relieving temperature)*: Needed to size relieving system components.
11	*Viscosity (at relieving temperature)*: Needed to size relieving system components if viscous fluid. User to specify units. Viscosity is required for the liquid sizing case only.
12	*Compressibility Factor (Z)*: This factor is used as a constant in disk sizing using the coefficient of discharge method. The compressibility factor is required for vapor sizing cases only.
13	*Specific Heat Ratio*: This constant is used in disk sizing calculations. The specific heat ratio is required for vapor sizing cases only. This is the ideal gas specific heat ratio. It is independent of pressure. Most process simulators will provide real gas specific heats which should not be used here; otherwise the rupture disk may be undersized.
14	*Normal Maximum Operating Pressure*: The maximum pressure at which the system normally operates. This pressure is used to calculate the operating ratio.
15	*Normal Maximum Operating Temperature*: The maximum temperature at which the system normally operates. This temperature is used to evaluate disk type, material, and performance.
16	*Pressure Fluctuations (static, cyclic, pulsating)*: Specify cyclic or pulsating service when applicable. Cyclic service is considered as a large amplitude and low frequency. Cyclic service with vacuum cycles shall be indicated. Pulsating service is considered as small amplitude and high frequency. For certain types of rupture disks, the operating ratio affects the service life in cyclic applications.
17	*Superimposed Backpressure*: See 3.53 for definition. A rupture disk is a differential pressure device, therefore, this pressure needs to be considered when specifying burst pressure. Additionally, superimposed backpressure is used to determine disk type and construction (e.g. vacuum/backpressure supports). For disks vented to atmosphere, the superimposed backpressure is atmospheric pressure and it is constant. See 4.3.6.2.4 for a discussion on the effects of superimposed backpressure on rupture disk selection.

Users of instructions should not rely exclusively on the information contained in this document. Sound business, scientific, engineering, and safety judgment should be used in employing the information contained herein.

18	*Built-up Backpressure*: See 3.7 for definition. This pressure is used to determine system backpressure.
19	*Backpressure*: See 3.3 for definition.
20	*Inlet Vacuum Conditions*: Inlet vacuum conditions are used to determine rupture disk type and construction (e.g. vacuum supports). Select and document vacuum units carefully, absolute units have positive values and gage units have negative values.
21	*Outlet Vacuum Conditions*: A rupture disk is a differential pressure device, therefore, outlet vacuum needs to be considered when specifying the burst pressure.
22	*Disk Located Upstream of Pressure Relief Valve (yes/no)*: This information is needed to verify proper selection (e.g. 3 % rule) of non-fragmenting disks.
23	*Disk Located Downstream of Pressure Relief Valve (yes/no)*: This information is needed by the user to verify installation and sizing requirements for this application.
24	*Non-fragmenting Design (yes/no)*: See 3.29 for definition. User shall specify non-fragmenting requirement to the manufacturer.
25	*Nominal Pipe Size*: This information is used to identify the nominal size of the mating fittings.
26	*Applicable Flange Standard & Class*: This information is used to identify pressure ratings and dimensions of holders.
27	*Flange Facing (inlet/outlet)*: Used to identify the mating flange facing (e.g. RF, FF).
28	*Piping Connection (schedule/bore)*: This information is used to evaluate flow area and proper selection of holder-less disks.
29	*Holder Tag No.*: Number assigned to rupture disk holder which identifies holder location.
30	*Nominal Holder Size*: Specify nominal holder size. In some cases nominal holder size may be larger than the relief piping to obtain lower burst pressures.
31	*Design Type*: Specify holder type, such as insert or full bolting. Holder selection may be based on ease of installation and maintenance or mating connections. Full bolting holders may reduce the heat flow to flange studs in a fire.
32	*Model Designator*: When known, specify the applicable manufacturer's model number, name, or designator.
33	*Quantity Required*: Specify quantity of holders required. Preventive maintenance and spares should be considered.
34	*Holder Material & Coatings (inlet)*: User should select an inlet material compatible with process fluids. Coatings and linings are sometimes used to enhance corrosion resistance or reduce product build-up.
35	*Holder Material & Coatings (outlet)*: Outlet holder material may be different from inlet holder material and should be selected based on frequency and duration of exposure to process and downstream fluids.
36	*Gauge Tap (yes/no) and Size (NPT) (outlet)*: Gauge taps in holder outlets are primarily used to vent and/or monitor the cavity between a rupture disk and a downstream pressure relief valve (PRV). See 4.3.2.2.3.
37	*Studs and Nuts (yes/no) and Material*: Specify if studs and nuts are to be supplied with the rupture disk holder and if so what materials (e.g. alloy or stainless steel).
38	*Jackscrews (yes/no)*: Indicate if jackscrews are required. Jackscrews are used to separate mating flanges to facilitate installation and maintenance of holders.
39	*Telltale Assembly (yes/no) and Material*: Telltale assemblies typically consist of a pressure gauge, excess flow valve, and connecting fittings. These assemblies are installed in holder outlets that are located upstream of PRVs. These devices provide venting and monitoring of the cavity between the disk and valve as specified in 4.3.2.2.3. If other monitoring devices are required, indicate here.
40	*Other*: Space provided for specifying other accessories.
41	*Nominal Disk Size*: Specify nominal disk size. In some cases the nominal disk size may be larger than the relief piping to obtain lower burst pressures.

Users of instructions should not rely exclusively on the information contained in this document. Sound business, scientific, engineering, and safety judgment should be used in employing the information contained herein.

42	*Required Discharge Area:* Specify the required discharge area. This is only required when the disk is sized using the coefficient of discharge method, see 5.11.1.2. The minimum net flow area of the disk, MNFA, shall be greater than or equal to this value.
43	*Disk Type*: Identify preference, if any, for forward-acting, reverse-acting, or graphite.
44	*Model Designator*: When known, specify the applicable manufacturer's model number, name, or designator.
45	*Quantity Required*: Specify quantity of disks required. Startup, preventive maintenance, and spares should be considered.
46	*Manufacturing Design Range*: User to specify the desired manufacturing design range. The manufacturing design range shall always be evaluated before the specified burst pressure is determined to ensure that the marked burst pressure is within applicable ASME Code pressure limits. Manufacturing design ranges generally depend on (a) the specified burst pressure level, (b) the rupture disk design type, and (c) the rupture disk manufacturer. Manufacturing design ranges are expressed as (a) plus or minus a percentage of the specified pressure, (b) plus or minus pressure units, or (c) zero percent or no manufacturing design range. See 3.24 for definition.
47	*Specified Burst Temperature*: User to specify the temperature at which the disk is to be rated and marked.
48	*Specified Burst Pressure*: A pressure specified by the user taking into consideration manufacturing design range, burst tolerance, superimposed backpressure and operating pressure.
49	*Maximum Marked Burst Pressure*: This pressure is calculated by adding the positive manufacturing design range to the specified burst pressure. The maximum marked burst pressure is then verified to meet the vessel or piping protection requirements for single, multiple, fire, or redundant applications.
50	*Minimum Marked Burst Pressure*: This pressure is calculated by subtracting the negative manufacturing design range from the specified burst pressure. The minimum marked burst pressure is used to calculate the operating ratio.
51	*Operating Ratio*: See 3.33 for definition. The operating ratio is used to evaluate the proper selection of the rupture disk and is calculated as follows: (a) for marked pressures above 40 psig the operating ratio is equal to the maximum normal operating pressure divided by the minimum marked burst pressure, (b) for marked pressures between 15 psig and 40 psig, the operating ratio is equal to the maximum normal operating pressure divided by the minimum marked burst pressure, less 2 psig. For marked pressures less than 15 psig, consult the manufacturer.
52	*Maximum Flow Resistance Factor* (K_R): When using the total flow resistance method to size relief piping components, specify the maximum flow resistance factor required for the rupture disk. The maximum flow resistance factor is expressed as a velocity head loss.
53	*Rupture Disk Materials*: The user is responsible for selecting and specifying rupture disk materials that are compatible with system fluids. Verify the selected materials are available for the rupture disk type, pressure, and temperature specified above.
54	*Manufacturer's Data*: When available specify the manufacturer's name and lot number. If the rupture disk has been previously ordered, the manufacturer will have lot number traceability to the previous order rupture disk specifications.

Users of instructions should not rely exclusively on the information contained in this document. Sound business, scientific, engineering, and safety judgment should be used in employing the information contained herein.

Figure A.2—Rupture Disk Device Specification Sheet

Annex B
(informative)

Review of Flow Equations Used in Sizing Pressure Relief Devices

B.1 Development of Flow Equations

B.1.1 General

B.1.1.1 The objectives of this annex are:

a) to provide the theoretical foundation that was used in the development of the sizing equations provided in the main body, including background on the use of the ideal gas specific heat ratio as an estimate for the isentropic expansion coefficient in the vapor sizing equations; and

b) to provide sizing techniques that could be used in situations where the assumptions made to develop the sizing equations are not appropriate.

B.1.1.2 The development of the flow equations used in sizing PRVs is based on the following assumptions:

a) that the flow-limiting element in a fully opened PRV is the nozzle in the body of the PRV between the inlet opening and the seating surface,

b) that the actual flow through a PRV can be adequately estimated by determining the theoretical maximum flow through the nozzle and then adjusting this theoretical flow to account for deviations from ideality,

c) that the appropriate thermodynamic path for determining the theoretical maximum flow through the nozzle is adiabatic and reversible (i.e. isentropic), a common assumption that has been validated experimentally for well-formed nozzles,

d) that the flow is one-dimensional,

e) that the fluid is homogeneous, i.e. it is in thermal (no heat transfer between phases) and mechanical (phases traveling at the same velocity) equilibrium, and its density is radially uniform normal to the direction of flow.

B.1.1.3 The one-dimensional isentropic nozzle flow assumption provides a standard theoretical framework for the PRV sizing equations. The general volumetric energy balance for isentropic nozzle flow of a homogeneous fluid forms the basis for the mass flux calculation [3], [4], [5], which is shown in Equation (B.1) and Equation (B.2).

In USC units

:

$$G^2 = \left[\frac{-2 \times \int_{P_1}^{P} 32.174 \times 144 \times v \times dP}{v_t^2}\right]_{\max} = \left[(\rho_t^2) \times \left(-2 \times \int_{P_1}^{P} \frac{4633 \times dP}{\rho}\right)\right]_{\max} \tag{B.1}$$

In SI units

:

$$G^2 = \left[\frac{-2 \times \int_{P_1}^{P} v \times dP}{v_t^2}\right]_{\max} = \left[(\rho_t^2) \times \left(-2 \times \int_{P_1}^{P} \frac{dP}{\rho}\right)\right]_{\max} \tag{B.2}$$

where

max is the maximization of this calculation with respect to pressure;

G is the mass flux (mass flow per unit area) through the nozzle, lb/s·ft^2 (kg/s·m^2);

v is the specific volume of the fluid, ft^3/lb (m^3/kg);

ρ is the mass density of the fluid, lb/ft^3 (kg/m^3);

P is the stagnation pressure of the fluid, psia (Pa);

1 is the fluid condition at the inlet to the nozzle;

t is the fluid condition at the throat of the nozzle where the cross-sectional area is minimized.

B.1.1.4 It is important to note that this energy balance is valid irrespective of the non-ideality or compressibility of the fluid. As a result, this general expression for isentropic nozzle flux can be used for any homogeneous fluid provided the relationship of fluid density to pressure at constant entropy is known.

B.1.1.5 Where the integral cannot be evaluated analytically, it can be evaluated numerically for any fluid by direct summation over small pressure intervals using an appropriate quadrature technique (e.g. Trapezoidal Rule or Simpson's Rule). The error associated with the numerical integration technique is related to the size of the increment, with the smaller increment resulting in less error. The user is cautioned that fluid property data may not always be available or the thermodynamic property model may not represent the true behavior of the specific fluid at or near the relieving conditions. In this case, additional work, possibly including bench scale testing, may be required to determine a suitable set of fluid properties. Using the Trapezoidal Rule, the isentropic mass flux integration can be estimated as shown in Equation (B.3).

$$\int_{P_1}^{P_n} v \times dP \approx \frac{1}{2} \times \sum_{j=1}^{n-1} (P_{j+1} - P_j) \times (v_{j+1} + v_j) \tag{B.3}$$

where

v is the specific volume of the fluid, ft^3/lb (m^3/kg);

P is the pressure of the fluid, psia (Pa);

1 is the fluid condition at the inlet to the nozzle;

n is the fluid condition at the assumed endpoint pressure;

j is the increment counter used for summation purposes.

B.1.1.6 If the size of the pressure increment is constant for each step, the integration estimated by the Trapezoidal Rule simplifies as shown in Equation (B.4).

$$\cancel{\int_{P_1}^{P_n} v \times dP \approx \frac{h}{2} \times \left(v_1 + v_n + 2 \times \sum_{j=1}^{n-1} v_j \right)}$$

$$\int_{P_1}^{P_n} v \times dP \approx \frac{h}{2} \times \left(v_1 + v_n + 2 \times \sum_{j=2}^{n-1} v_j \right) \tag{B.4}$$

where

- v is the specific volume of the fluid, ft^3/lb (m^3/kg);
- P is the pressure of the fluid, psia (Pa);
- h is the constant pressure step size chosen for summation purposes, psi (Pa);
- 1 represents fluid conditions at the inlet to the nozzle;
- n is the index for fluid conditions at the assumed throat pressure (i.e. the assumed endpoint pressure for the integral);
- j is the increment counter used for summation purposes.

B.1.1.7 The isentropic nozzle mass flux expression is evaluated at each pressure interval until a maximum mass flux is obtained or the total backpressure at the nozzle is reached, whichever occurs first. If the throat pressure corresponding to the maximum mass flux is greater than the total backpressure, this indicates the flow is choked in the nozzle. It is important to note that the specific volume for each pressure increment is the specific volume at the end of that pressure increment for each step in the summation.

B.1.1.8 A convenient basis for defining the constant pressure step size, h, is to subtract the minimum backpressure on the PRV (typically atmospheric pressure) from the inlet relief pressure, and to use a pressure interval that is 1 % of the total pressure range. Note that for compressible fluids, the entire pressure range may not need to be evaluated due to the potential for reaching the maximum mass flux before the backpressure. To ensure an appropriate step size was chosen, the calculation may be rerun with a smaller step size, and the results compared to determine whether or not the value changes significantly from the previous calculation.

B.1.2 General Applicability

The applicability of the isentropic nozzle mass flux expression [Equation (B.1) and Equation (B.2)] for all homogeneous fluid regimes provides not only a generic calculation technique covering all common sizing equations, but also a useful technique for situations not represented by the common sizing equations. Such situations would include compressible supercritical fluids that may enter the two-phase region as the pressure is decreased along an isentropic path.

B.1.3 Numerical Integration Example

B.1.3.1 In this example calculation of the theoretical nozzle mass flux, a supercritical fluid is relieved and enters the two-phase region, the following data are given:

— the relief fluid consists of supercritical ethylene at stagnation conditions;

— the pressure, P_1, of the relief fluid entering the nozzle is 797.7 psia (5500 kPa);

— the temperature, T_1, of the relief fluid entering the nozzle is 80 °F (300 K);

— the backpressure on the PRV is atmospheric pressure [14.7 psia (101.325 kPa)].

B.1.3.2 In this example, the following data are derived:

— total pressure range of 797.7 – 14.7 = 783 psi (5500 – 101.3 = 5399 kPa);

— pressure interval for summation purposes 0.01 × 783 = 7.83 psi (54 kPa);

— the table of pressure (P), temperature (T), and specific volume (v) data at constant entropy shown in Table B.1.

NOTE Fluid property packages are commercially available to perform constant entropy flashes to get the necessary properties to perform this analysis.

B.1.3.3 Results of the integration of Equation (B.1) or (B.2) using Equation (B.4) are shown in Table B.1.

B.1.3.4 The maximum theoretical mass flux of 3201 lb/s·ft^2 (15,630 kg/s·m^2) was encountered at a pressure of 468.8 psia (3232 kPa), well before the minimum expected backpressure of 14.7 psia (101.325 kPa) was reached.

B.1.4 Capacity Corrections

B.1.4.1 Once the theoretical mass flux through the nozzle has been determined, various correction factors are employed to derive an expression for the actual PRV capacity. These correction factors may include discharge coefficients (K_d), backpressure correction factors (K_b and K_w), viscosity correction factors (K_v), and combination capacity correction factors (K_c) depending on the applicability of those correction factors. A general expression for the PRV sizing equation is shown in Equation (B.5).

$$W = G\,A\,\Pi[K] \tag{B.5}$$

where

G is the theoretical mass flux through the nozzle, lb/s·ft^2 (kg/s·m^2);

W is the mass flow through the PRV, lb/s (kg/s);

A is the effective discharge area of the PRV, ft^2 (m^2);

$\Pi[K]$ is the product of all applicable correction factors (no units).

B.1.4.2 Note that when using the direct numerical integration technique, the choice of the applicable correction factors should be made based on the nature of the relieving fluid. For fluids that behave as incompressible fluids or those that do not choke within the PRV itself, correction factors pertaining to liquid and/or subcooled flashing liquid service have been used. For fluids that behave as compressible fluids or those that do choke within the PRV itself, correction factors pertaining to vapor and/or two-phase service have been used [5] [6] [7].

B.2 Flow Equations for Subcooled Liquids

B.2.1 General

B.2.1.1 For highly subcooled liquids that have a vapor pressure lower than the lowest pressure on the outlet of the PRV and are incompressible, the isentropic nozzle flux expression [Equation (B.1) and Equation (B.2)] is readily simplified. The incompressible constraint indicates the density is constant regardless of the pressure of the liquid. In addition, the highly subcooled liquid with a vapor pressure below the lowest pressure on the outlet of the PRV indicates that the liquid will not flash as it flows through the valve and thus will not choke. As a result, the pressure at the throat of the nozzle is the same as the total backpressure on the PRV, and the theoretical mass flux equation simplifies to Equation (B.6) and Equation (B.7).

Table B.1—Results for Supercritical Fluid Example Problem B.1.3 (Continued)

P psia (kPa)	***T*** °F (K)	***v*** ft^3/lb (m^3/kg)	**Integral term of (B.1) or (B.2)** ft^2/s^2 (m^2/s^2)	**Mass Flux** lb/s-ft^2 (kg/s-m^2)
797.7 (5500)	80.33 (300.0000)	0.152 (0.009477)	—	—

789.9 (5446)	79.06 (299.2919)	0.153 (0.009552)	5530 (514)	687 (3356)
782.0 (5392)	77.77 (298.5773)	0.154 (0.009628)	11105 (1032)	966 (4718)
774.2 (5338)	76.47 (297.8560)	0.155 (0.009706)	16724 (1554)	1176 (5743)
766.4 (5284)	75.16 (297.1279)	0.157 (0.009785)	22388 (2080)	1350 (6591)
758.6 (5230)	73.84 (296.3929)	0.158 (0.009866)	28100 (2611)	1500 (7324)
750.7 (5176)	72.50 (295.6508)	0.159 (0.009949)	33858 (3146)	1633 (7972)
742.9 (5122)	71.15 (294.9016)	0.161 (0.010033)	39666 (3685)	1753 (8557)
735.1 (5068)	69.79 (294.1451)	0.162 (0.010119)	45523 (4229)	1861 (9088)
727.2 (5014)	68.42 (293.3813)	0.164 (0.010207)	51430 (4778)	1962 (9577)
719.4 (4960)	67.03 (292.6099)	0.165 (0.010297)	57389 (5332)	2054 (10028)
711.6 (4906)	65.63 (291.8308)	0.166 (0.010389)	63401 (5890)	2140 (10447)
703.7 (4852)	64.21 (291.0440)	0.168 (0.010483)	69467 (6454)	2220 (10838)
695.9 (4798)	62.78 (290.2492)	0.169 (0.010579)	75588 (7022)	2295 (11203)
688.1 (4744)	61.33 (289.4463)	0.171 (0.010677)	81765 (7596)	2365 (11545)
680.2 (4690)	59.87 (288.6352)	0.173 (0.010777)	88000 (8176)	2430 (11865)
672.4 (4636)	58.40 (287.8157)	0.174 (0.010879)	94293 (8760)	2492 (12167)
664.6 (4582)	56.91 (286.9877)	0.176 (0.010984)	100647 (9351)	2550 (12450)
656.7 (4528)	55.40 (286.1509)	0.178 (0.011091)	107063 (9947)	2605 (12716)
648.9 (4474)	53.88 (285.3052)	0.179 (0.011201)	113542 (10548)	2656 (12967)
641.1 (4420)	52.34 (284.4505)	0.181 (0.011314)	120085 (11156)	2704 (13203)
633.2 (4366)	50.79 (283.5866)	0.183 (0.011429)	126695 (11770)	2750 (13425)
625.4 (4312)	49.21 (282.7132)	0.185 (0.011547)	133372 (12391)	2792 (13633)
617.6 (4258)	47.62 (281.8302)	0.187 (0.011668)	140119 (13018)	2832 (13829)

609.7 (4204)	46.02 (280.9373)	0.189 (0.011791)	146936 (13651)	2870 (14013)
601.9 (4150)	44.39 (280.0345)	0.191 (0.011918)	153827 (14291)	2905 (14185)
594.1 (4096)	42.75 (279.1214)	0.193 (0.012049)	160792 (14938)	2938 (14346)
586.2 (4042)	41.09 (278.1978)	0.195 (0.012182)	167834 (15592)	2969 (14496)
578.4 (3988)	39.40 (277.2635)	0.197 (0.012319)	174955 (16254)	2998 (14636)
570.6 (3934)	37.70 (276.3184)	0.200 (0.012460)	182156 (16923)	3024 (14766)
562.7 (3880)	35.98 (275.3620)	0.202 (0.012604)	189440 (17600)	3049 (14885)
554.9 (3826)	34.24 (274.3942)	0.204 (0.012752)	196809 (18284)	3071 (14996)
547.1 (3772)	32.48 (273.4147)	0.207 (0.012905)	204266 (18977)	3092 (15097)
539.3 (3718)	30.69 (272.4233)	0.209 (0.013061)	211812 (19678)	3111 (15189)
531.4 (3664)	28.89 (271.4196)	0.212 (0.013222)	219450 (20388)	3128 (15272)
523.6 (3610)	27.06 (270.4034)	0.214 (0.013388)	227184 (21106)	3143 (15347)
515.8 (3556)	25.20 (269.3744)	0.217 (0.013558)	235015 (21834)	3157 (15413)
507.9 (3502)	23.33 (268.3321)	0.220 (0.013733)	242946 (22571)	3169 (15471)
500.1 (3448)	21.43 (267.2764)	0.223 (0.013914)	250981 (23317)	3179 (15521)
492.3 (3394)	19.50 (266.2069)	0.226 (0.014100)	259122 (24073)	3187 (15562)
484.4 (3340)	17.55 (265.1231)	0.229 (0.014291)	267373 (24840)	3194 (15596)
476.6 (3286)	15.57 (264.0248)	0.232 (0.014489)	275737 (25617)	3200 (15622)
468.8 (3232)	13.73 (262.9991)	0.236 (0.014703)	284221 (26405)	3201 (15630)
460.9 (3178)	12.47 (262.3016)	0.240 (0.014964)	292843 (27206)	3193 (15588)
453.1 (3124)	11.20 (261.5952)	0.244 (0.015234)	301619 (28022)	3183 (15540)
445.3 (3070)	9.91 (260.8797)	0.248 (0.015512)	310555 (28852)	3172 (15485)

In USC units:

$$G = \sqrt{\frac{2 \times 4633 \times (P_1 - P_b)}{v}} = \sqrt{2 \times 4633 \times \rho \times (P_1 - P_b)} \quad \text{(B.6)}$$

In SI units:

$$G = \sqrt{\frac{2 \times (P_1 - P_b)}{v}} = \sqrt{2 \times \rho \times (P_1 - P_b)} \quad \text{(B.7)}$$

where

G is the theoretical mass flux through the nozzle, lb/s·ft^2 (kg/s·m^2);

v is the specific volume of the fluid, ft^3/lb (m^3/kg);

ρ is the mass density of the fluid, lb/ft^3 (kg/m^3);

P is the pressure of the fluid, psia (Pa);

1 represents fluid conditions at the inlet to the nozzle;

b represents fluid conditions at the outlet of the nozzle.

B.2.1.2 The final expressions for liquid PRV sizing presented in this standard (see 5.8) are obtained by algebraically rearranging these equations, substituting what is believed to be readily available variables (e.g. specific gravity instead of density), and applying the appropriate correction factors (see B.1.4).

B.2.2 Subcooled Liquid Example

B.2.2.1 In this example, the following data are given:

— the relief fluid consists of subcooled liquid water at stagnation conditions;

— the pressure, P_1, of the relief fluid entering the nozzle is 114.7 psia (791 kPa);

— the temperature, T_1, of the relief fluid entering the nozzle is 80 °F (300 K);

— the backpressure on the PRV is atmospheric pressure or 14.7 psia (101.325 kPa).

B.2.2.2 In this example, the following data are derived:

— total pressure range of 114.7 – 14.7 = 100 psi (689.5 kPa);

— pressure interval for summation purposes 0.1 × 100 = 10 psi (69 kPa);

— the table of pressure (P), temperature (T), and specific volume (v) data at constant entropy shown in Table B.2.

NOTE Fluid property packages are commercially available to perform constant entropy flashes to get the necessary properties to perform this analysis.

B.2.2.3 Results of the integration of Equation (B.1) or (B.2) using Equation (B.4) are shown in Table B.2.

B.2.2.4 The maximum mass flux of 7,592 lb/s·ft^2 (37,068 kg/s·m^2) was encountered at a pressure of 14.7 psia (101.325 kPa), which is the minimum backpressure on the PRV.

B.2.3 Comparison of Results to Liquid Sizing Equations

B.2.3.1 For comparison purposes, the required orifice area of a PRV can be determined using the integration approach of the sample problem in B.2.2 and the liquid sizing formula presented in 5.8 [Equation (28) and Equation (29)]. The following data are given:

— required flow rate of 528 gal/min (2,000 L/min).

B.2.3.2 For this comparison, the following data are derived:

— specific gravity of the fluid at inlet conditions = 0.9969;

— effective discharge coefficient for preliminary sizing, K_d = 0.65;

— all other correction factors (K_w, K_c, and K_v) = 1.0.

Table B.2—Results for Subcooled Liquid Example Problem B.2.2

P psia (kPa)	***T*** °F (K)	**v** ft^3/lb (m^3/kg)	**Integral term of (B.1) or (B.2)** ft^2/s^2 (m^2/s^2)	**Mass Flux** lb/s-ft^2 (kg/s-m^2)
114.7 (790.800)	80.33 (300.0000)	0.016069 (0.0010031)	—	—
104.7 (721.900)	80.33 (299.9986)	0.016069 (0.0010032)	745 (69.2)	2401 (11724)
94.7 (652.900)	80.33 (299.9973)	0.016070 (0.0010032)	1489 (138.0)	3396 (16580)
84.7 (584.000)	80.32 (299.9959)	0.016070 (0.0010032)	2234 (208.0)	4159 (20306)
74.7 (515.000)	80.32 (299.9945)	0.016071 (0.0010033)	2978 (277.0)	4802 (23447)
64.7 (446.100)	80.32 (299.9932)	0.016071 (0.0010033)	3723 (346.0)	5369 (26214)
54.7 (377.100)	80.32 (299.9918)	0.016072 (0.0010033)	4467 (415.0)	5881 (28716)
44.7 (308.200)	80.31 (299.9904)	0.016072 (0.0010034)	5212 (484.0)	6352 (31016)
34.7 (239.200)	80.31 (299.9891)	0.016073 (0.0010034)	5957 (553.0)	6791 (33156)
24.7 (170.300)	80.31 (299.9877)	0.016073 (0.0010034)	6701 (623.0)	7203 (35167)
14.7 (101.325)	80.31 (299.9863)	0.016074 (0.0010035)	7446 (692.0)	7592 (37068)

B.2.3.3 Using the theoretical mass flux obtained from numerical integration above, one may determine the required effective discharge area:

In USC units:

$$A = \frac{W}{G \times K_d} = \frac{Q \times \rho}{60\frac{\text{sec}}{\text{min}} \times 7.4805\frac{\text{gal}}{\text{ft}^3}} \times \frac{1}{G \times K_d}$$

$$A = \frac{528 \times 62.2}{60 \times 7.4805} \times \frac{1}{7592.14 \times 0.65} = 0.0148 \text{ ft}^2 = 2.135 \text{ in.}^2 \tag{B.8}$$

In SI units:

$$A = \frac{W}{G \times K_d} = \frac{Q \times \rho}{60\frac{\text{sec}}{\text{min}} \times 1000\frac{\text{liter}}{\text{m}^3}} \times \frac{1}{G \times K}$$

$$A = \frac{2000 \times 996.9}{60 \times 1000} \times \frac{1}{37{,}068 \times 0.65} = 1.379 \times 10^{-3} \text{ m}^2 = 1379 \text{ mm}^2 \tag{B.9}$$

where

G is the theoretical mass flux through the nozzle, lb/s·ft^2 (kg/s·m^2);

W is the required relief rate, lb/s (kg/s);

Q is the required relief rate, gal/min (L/min);

$\rho = {}^{1}/_{v}$ is the fluid density, lb/ft^3 (kg/m^3);

K_d is the effective coefficient of discharge (no units);

A is the required effective discharge area, ft^2 (m^2).

B.2.3.4 Using the liquid sizing [Equation (28) and Equation (29)], one may also determine the required effective discharge area:

In USC units:

$$A = \frac{Q}{38 \times K_d \times K_w \times K_c \times K_v}\sqrt{\frac{G_l}{P_1 - P_2}}$$

$$A = \frac{528}{38 \times 0.65 \times 1.0 \times 1.0 \times 1.0}\sqrt{\frac{0.997}{114.7 - 14.7}} = 2.134 \text{ in.}^2 \tag{B.10}$$

In SI units:

$$A = \frac{11.78 \times Q}{K_d \times K_w \times K_c \times K_v}\sqrt{\frac{G_l}{P_1 - P_2}}$$

$$A = \frac{11.78 \times 2000}{0.65 \times 1.0 \times 1.0 \times 1.0}\sqrt{\frac{0.997}{790.830 - 101.325}} = 1378 \text{ mm}^2 \tag{B.11}$$

where

Q is the required relief rate, gal/min (L/min);

G_l is the specific gravity of the fluid at flowing temperature (no units);

P_1 is the pressure at the inlet to the PRV, psia (kPa);

P_2 is the backpressure on the PRV, psia (kPa);

K_d is the effective coefficient of discharge (no units);

A is the required effective discharge area, in.^{2} (mm^2).

B.3 Flow Equations for Gases and Vapors

B.3.1 Real Gases

B.3.1.1 For vapors and gases with a constant isentropic expansion coefficient, the expression for the specific volume to pressure relationship along an isentropic path is shown in Equation (B.12).

$$P \times v^n = P_1 \times v_1^n \tag{B.12}$$

where

v is the specific volume of the fluid, ft^3/lb (m^3/kg);

P is the pressure of the fluid, psia (Pa);

n is the isentropic expansion coefficient.

B.3.1.2 Determining the isentropic expansion coefficient for a real gas is somewhat complicated because it is a function of both pressure and temperature and, while in most cases, it is relatively constant, it may vary throughout the expansion process. The coefficient can generally be obtained from an equation of state that describes the pressure-volume relationship along an isentropic expansion path. In the event the isentropic expansion coefficient is constant, an expression for the isentropic expansion coefficient in terms of thermodynamic state variables can be derived. This expression is shown in Equation (B.13).

$$\cancel{n = \frac{v}{P} \times \left(\frac{\partial P}{\partial v}\right)_T \times \frac{C_p}{C_v}}$$

$$n = -\frac{v}{P} \times \left(\frac{\partial P}{\partial v}\right)_T \times \frac{C_p}{C_v} \tag{B.13}$$

where

n is the isentropic expansion coefficient;

v is the specific volume of the fluid, ft^3/lb (m^3/kg);

P is the pressure of the fluid, psia (Pa);

T represents the partial derivative taken at constant temperature;

C_p is the specific heat capacity of the fluid at constant pressure, Btu/lb°F (J/kg°C);

C_v is the specific heat capacity of the fluid at constant volume, Btu/lb°F (J/kg°C).

B.3.1.3 Note that these variables can be evaluated at any point along the isentropic path; however, the inlet conditions are most convenient as the relief temperature is known at this point and the specific heat capacities can be readily obtained.

B.3.1.4 Alternatively, an isentropic expansion coefficient can be used based on an average value between the upstream pressure and the pressure in the throat of the nozzle which, in the case of maximum flow, is the critical-flow pressure.

B.3.1.5 For vapors and gases that follow the constant isentropic expansion expression, the isentropic nozzle flux equation [Equation (B.1) and Equation (B.2)] can be solved analytically to give the expression shown in Equation (B.14) and Equation (B.15).

In USC units:

$$G^2 = \left(\frac{2 \times 4633}{v_1 \times P_1^{1/n}}\right) \times (P_t^{2/n}) \times \left(\frac{n}{n-1}\right) \times (P_1^{n-1/n} - P_t^{n-1/n}) \tag{B.14}$$

In SI units:

$$G^2 = \left(\frac{2}{v_1 \times P_1^{1/n}}\right) \times (P_t^{2/n}) \times \left(\frac{n}{n-1}\right) \times (P_1^{n-1/n} - P_t^{n-1/n}) \tag{B.15}$$

where

- n is the isentropic expansion coefficient;
- G is the mass flux through the nozzle, lb/s·ft^2 (kg/s·m^2);
- v is the specific volume of the fluid, ft^3/lb (m^3/kg);
- P is the pressure of the fluid, psia (Pa);
- 1 represents conditions at the inlet of the nozzle;
- t represents conditions at the throat of the nozzle.

B.3.1.6 For the special case of an isentropic expansion coefficient equal to 1.0:

In USC units:

$$G^2 = \left(\frac{2 \times 4633}{v_1 \times P_1}\right) \times (P_t^2) \times (\ln P_1 - \ln P_t) \tag{B.16}$$

In SI units:

$$G^2 = \left(\frac{2}{v_1 \times P_1}\right) \times (P_t^2) \times (\ln P_1 - \ln P_t) \tag{B.17}$$

B.3.1.7 For vapors and gases that follow the constant isentropic expansion expression, the pressure at which maximum mass flux is obtained in the nozzle (i.e. the choking pressure), can be determined as a function of the inlet pressure and the isentropic expansion coefficient, as shown in Equation (B.18) for values of n not equal to one.

$$P_{\text{choke}} = P_1 \times \left(\frac{2}{n+1}\right)^{n/_{n-1}} \tag{B.18}$$

where

n is the isentropic expansion coefficient;

P is the pressure of the fluid, psia (Pa);

1 is the condition at the inlet of the nozzle;

choke is the choking condition.

B.3.1.8 For the special case of an isentropic expansion coefficient equal to one:

$$P_{\text{choke}} = P_1 \times \frac{1}{\sqrt{e}} \tag{B.19}$$

where

e is the base of the natural logarithm $\approx$ 2.7183;

1 represents conditions at the inlet of the nozzle.

B.3.1.9 The backpressure on the PRV outlet can then be compared to the choking pressure of the fluid in order to determine whether or not the flow is choked through the nozzle. If the flow is choked (i.e. the backpressure is less than the choking pressure), then the pressure at the throat of the nozzle is the choking pressure; otherwise, the flow is not choked (i.e. the backpressure is greater than the choking pressure), and the pressure at the throat of the nozzle is the backpressure. This effective throat pressure can then be used in the expression shown in Equation (B.14) or Equation (B.15) above. In the event the flow is choked, the analytical expressions can be simplified even further by substituting Equation (B.18) into Equation (B.14) or Equation (B.15), as shown in Equation (B.20) and Equation (B.21).

In USC units:

$$G^2 = 4633 \times \left(\frac{P_1}{v_1}\right) \times (n) \times \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}} \tag{B.20}$$

In SI units:

$$G^2 = \left(\frac{P_1}{v_1}\right) \times (n) \times \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}} \tag{B.21}$$

where

n is the isentropic expansion coefficient;

v is the specific volume of the fluid, ft^3/lb (m^3/kg);

P is the pressure of the fluid, psia (Pa);

1 is the condition at the inlet of the nozzle;

t is the condition at the throat of the nozzle.

B.3.1.10 For the special case of an isentropic expansion coefficient equal to one:

In USC units:

$$G^2 = 4633 \times \left(\frac{P_1}{v_1}\right) \times \frac{1}{e} \tag{B.22}$$

In SI units:

$$G^2 = \left(\frac{P_1}{v_1}\right) \times \frac{1}{e} \tag{B.23}$$

where

v is the specific volume of the fluid, ft^3/lb (m^3/kg);

P is the pressure of the fluid, psia (Pa);

e is the base of the natural logarithm $\approx$ 2.7183;

1 is the condition at the inlet of the nozzle.

B.3.1.11 As the temperature and compressibility factor for vapors and gases may be more readily available than the specific volume, the real gas law as shown in Equation (B.24) may be used to substitute these variables for the specific volume, as shown in Equation (B.25) and Equation (B.26).

$$P \times v = Z \times \frac{R_u}{M} \times T \tag{B.24}$$

where

P is the pressure of the fluid, psia (Pa);

v is the specific volume of the fluid, ft^3/lb (m^3/kg);

Z is the compressibility factor of the fluid (no units);

R_u is the universal gas constant = 10.73 psia·ft^3/lb-mole·°R [8.314 Pa·m^3/kg-mole·K)];

M is the molecular weight, lb/lb-mole (kg/kg-mole);

T is the temperature of the fluid, °R (K).

In USC units:

$$G^2 = 4633 \times \left(\frac{P_1^2 M}{Z_1 \times R_u \times T_1}\right) \times (n) \times \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}} \tag{B.25}$$

In SI units:

$$G^2 = \left(\frac{P_1^2 M}{Z_1 \times R_u \times T_1}\right) \times (n) \times \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}} \tag{B.26}$$

where

n is the isentropic expansion coefficient (no units);

P is the pressure of the fluid, psia (Pa);

Z is the compressibility factor of the fluid, no units;

R_u is the universal gas constant = 10.73 psia·ft^3/lb-mole·°R (8.314 Pa·m^3/kg-mole·K);

M is the molecular weight, lb/lb-mole (kg/kg-mole);

T is the temperature of the fluid, °R (K);

1 is the condition at the inlet of the nozzle.

B.3.2 Ideal Gas Assumption

B.3.2.1 For vapors and gases that can be considered ideal gases, which follow the ideal gas law as shown in Equation (B.27), the expression for the constant isentropic expansion coefficient [Equation (B.13)] can be further reduced by deriving the expression for the partial derivative of pressure with respect to specific volume at constant temperature for the ideal gas. The isentropic expansion coefficient for an ideal gas is constant and is the ratio of the ideal gas specific heat capacity at constant pressure to the ideal gas specific heat capacity at constant volume (i.e. the ideal gas specific heat ratio, k), as shown in Equation (B.28).

$$P \times v = \frac{R_u}{M} \times T \tag{B.27}$$

where

P is the pressure of the fluid, psia (Pa);

v is the specific volume of the fluid, ft^3/lb (m^3/kg);

R_u is the universal gas constant = 10.73 psia·ft^3/lb-mole·°R (8.314 Pa·m^3/kg-mole·K);

M is the molecular weight, lb/lb-mole (kg/kg-mole);

T is the temperature of the fluid, °R (K).

B.3.2.2 It is important to note that the derivation of the ideal gas specific heat ratio, k, as an approximation for the isentropic expansion coefficient, n, is based on the ideal gas assumption, where the compressibility factor, Z, is equal to 1.

$$k = -\frac{v}{P} \times \left(-\frac{P}{v}\right) \times \frac{C_p^*}{C_v^*} = \frac{C_p^*}{C_v^*} \tag{B.28}$$

where

k is the isentropic expansion coefficient for an ideal gas;

P is the pressure of the fluid, psia (Pa);

v is the specific volume of the fluid, ft^2/lb (m^2/kg);

C_p is the specific heat capacity at constant pressure, Btu/lb·°F (J/kg·°C);

C_v is the specific heat capacity at constant volume, Btu/lb·°F (J/kg·°C);

$*$ is the ideal gas constraint.

B.3.2.3 This expression can be evaluated at any point along the isentropic path; however, the inlet conditions are most convenient as the relief temperature is known at this point and the specific heat capacities can be readily obtained. It is useful to note that the ideal gas specific heat ratio is typically not significantly dependent on temperature (and is not dependent at all on pressure); hence, the ideal gas specific heat ratio at standard conditions may be a good estimate in the absence of other information.

B.3.2.4 The final expressions for PRV sizing in vapor service presented in 5.6 are obtained by algebraically rearranging these equations, substituting readily available variables (e.g. using the expression for specific volume as a function of temperature, pressure, compressibility, and the ideal gas constant), and applying the appropriate correction factors. It is important to note that some of the algebraic manipulation involves the introduction of additional variables that represent parts of the equation and/or unit conversions, such as the variable C in the choked flow calculations, and the variable F_2 in the non-choked flow calculations.

B.3.3 Gas Example

B.3.3.1 In this example, the following data are given:

— the relief fluid consists of superheated air at stagnation conditions;

— the pressure, P_1, of the relief fluid entering the nozzle is 114.7 psia (790.8 kPa);

— the temperature, T_1, of the relief fluid entering the nozzle is 80 °F (300 K);

— the backpressure on the PRV is atmospheric pressure of 14.7 psia (101.325 kPa).

B.3.3.2 In this example, the following data are derived:

— total pressure range of 114.7 – 14.7 = 100 psi (790.8 – 101.3 = 689.5 kPa);

— pressure interval for summation purposes 0.01 × 100 = 1 psi (6.895 kPa);

— the table of pressure (P), temperature (T), and specific volume (v) data at constant entropy shown in Table B.3.

NOTE Fluid property packages are commercially available to perform constant entropy flashes to get the necessary properties to perform this analysis.

B.3.3.3 Results of the integration of Equation (B.1) or (B.2) using Equation (B.4) are shown in Table B.3.

B.3.3.4 The maximum mass flux of 379.1 lb/s·ft^2 (1,851 kg/s·m^2) was encountered at a pressure of 60.7 psia (418.5 kPa).

Table B.3—Results for Gas Example Problem B.3.3 (Continued)

P psia (kPa)	T °F (K)	v ft^3/lb (m^3/kg)	Integral term of (B.1) or (B.2) ft^2/s^2 (m^2/s^2)	Mass Flux lb/s-ft^2 (kg/s-m^2)
114.7 (790.8)	80.33 (300.0000)	1.741 (0.108670)	—	—
113.7 (783.9)	78.97 (299.2462)	1.752 (0.109347)	8090 (752)	72.6 (354.6)

112.7 (777.0)	77.61 (298.4876)	1.763 (0.110034)	16231 (1508)	102.2 (499.1)
111.7 (770.1)	76.23 (297.7242)	1.774 (0.110731)	24423 (2269)	124.6 (608.4)
110.7 (763.2)	74.85 (296.9559)	1.785 (0.111440)	32668 (3035)	143.2 (699.1)
109.7 (756.4)	73.46 (296.1827)	1.797 (0.112159)	40965 (3806)	159.3 (777.9)
108.7 (749.5)	72.06 (295.4043)	1.808 (0.112890)	49316 (4582)	173.7 (848.0)
107.7 (742.6)	70.65 (294.6209)	1.820 (0.113632)	57722 (5363)	186.7 (911.4)
106.7 (735.7)	69.23 (293.8322)	1.832 (0.114386)	66184 (6149)	198.6 (969.5)
105.7 (728.8)	67.80 (293.0382)	1.845 (0.115153)	74702 (6940)	209.6 (1023.1)
104.7 (721.9)	66.36 (292.2388)	1.857 (0.115931)	83277 (7737)	219.8 (1073.0)
103.7 (715.0)	64.91 (291.4340)	1.870 (0.116723)	91910 (8539)	229.3 (1119.6)
102.7 (708.1)	63.45 (290.6230)	1.883 (0.117528)	100603 (9346)	238.3 (1163.3)
101.7 (701.2)	61.98 (289.8075)	1.896 (0.118347)	109356 (10160)	246.7 (1204.5)
100.7 (694.3)	60.50 (288.9857)	1.909 (0.119179)	118170 (10978)	254.7 (1243.3)
99.7 (687.4)	59.01 (288.1580)	1.923 (0.120026)	127047 (11803)	262.2 (1280.1)
98.7 (680.5)	57.51 (287.3243)	1.936 (0.120888)	135987 (12634)	269.3 (1314.9)
97.7 (673.6)	56.00 (286.4847)	1.950 (0.121764)	144991 (13470)	276.1 (1348.0)
96.7 (666.7)	54.48 (285.6388)	1.965 (0.122656)	154061 (14313)	282.5 (1379.4)
95.7 (659.8)	52.95 (284.7867)	1.979 (0.123564)	163198 (15162)	288.6 (1409.3)
94.7 (652.9)	51.40 (283.9282)	1.994 (0.124489)	172403 (16017)	294.5 (1437.7)
93.7 (646.0)	49.84 (283.0632)	2.009 (0.125430)	181677 (16879)	300.0 (1464.8)
92.7 (639.1)	48.27 (282.1916)	2.025 (0.126389)	191022 (17747)	305.3 (1490.6)
91.7 (632.2)	46.69 (281.3132)	2.040 (0.127365)	200438 (18621)	310.3 (1515.2)
90.7 (625.3)	45.10 (280.4280)	2.056 (0.128360)	209928 (19503)	315.1 (1538.6)
89.7 (618.5)	43.49 (279.5358)	2.072 (0.129374)	219492 (20392)	319.7 (1561.0)
88.7 (611.6)	41.88 (278.6364)	2.089 (0.130408)	229132 (21287)	324.1 (1582.2)
87.7 (604.7)	40.24 (277.7298)	2.106 (0.131462)	238849 (22190)	328.2 (1602.5)
86.7 (597.8)	38.60 (276.8158)	2.123 (0.132536)	248646 (23100)	332.2 (1621.8)
85.7 (590.9)	36.94 (275.8942)	2.141 (0.133632)	258523 (24018)	335.9 (1640.1)
84.7 (584.0)	35.27 (274.9649)	2.158 (0.134750)	268482 (24943)	339.5 (1657.5)
83.7 (577.1)	33.58 (274.0277)	2.177 (0.135891)	278525 (25876)	342.9 (1674.1)
82.7 (570.2)	31.88 (273.0825)	2.195 (0.137056)	288654 (26817)	346.1 (1689.8)

81.7 (563.3)	30.16 (272.1292)	2.214 (0.138245)	298870 (27766)	349.1 (1704.6)
80.7 (556.4)	28.43 (271.1674)	2.234 (0.139459)	309175 (28724)	352.0 (1718.7)
79.7 (549.5)	26.68 (270.1972)	2.254 (0.140700)	319571 (29689)	354.7 (1731.9)
78.7 (542.6)	24.92 (269.2182)	2.274 (0.141967)	330061 (30664)	357.3 (1744.4)
77.7 (535.7)	23.14 (268.2303)	2.295 (0.143262)	340645 (31647)	359.7 (1756.1)
76.7 (528.8)	21.35 (267.2332)	2.316 (0.144586)	351327 (32640)	361.9 (1767.1)
75.7 (521.9)	19.54 (266.2260)	2.338 (0.145940)	362108 (33641)	364.0 (1777.4)
74.7 (515.0)	17.71 (265.2110)	2.360 (0.147325)	372990 (34652)	366.0 (1786.9)
73.7 (508.1)	15.86 (264.1854)	2.383 (0.148743)	383977 (35673)	367.8 (1795.8)
72.7 (501.2)	14.00 (263.1498)	2.406 (0.150193)	395070 (36704)	369.5 (1803.9)
71.7 (494.3)	12.12 (262.1040)	2.430 (0.151678)	406272 (37744)	371.0 (1811.4)
70.7 (487.4)	10.22 (261.0478)	2.454 (0.153200)	417586 (38795)	372.4 (1818.2)
69.7 (480.6)	8.30 (259.9808)	2.479 (0.154758)	429014 (39857)	373.7 (1824.4)
68.7 (473.7)	6.36 (258.9029)	2.505 (0.156355)	440558 (40930)	374.8 (1829.9)
67.7 (466.8)	4.39 (257.8137)	2.531 (0.157993)	452223 (42013)	375.8 (1834.7)
66.7 (459.9)	2.41 (256.7130)	2.558 (0.159673)	464012 (43109)	376.6 (1838.9)
65.7 (453.0)	0.41 (255.6005)	2.585 (0.161396)	475926 (44215)	377.4 (1842.5)
64.7 (446.1)	–1.61 (254.4759)	2.614 (0.163165)	487970 (45334)	378.0 (1845.4)
63.7 (439.2)	–3.66 (253.3387)	2.643 (0.164982)	500147 (46466)	378.5 (1847.8)
62.7 (432.3)	–5.73 (252.1888)	2.673 (0.166848)	512461 (47610)	378.8 (1849.5)
61.7 (425.4)	–7.82 (251.0258)	2.703 (0.168766)	524915 (48767)	379.0 (1850.5)
60.7 (418.5)	–9.94 (249.8492)	2.735 (0.170737)	537513 (49937)	379.1 (1851.0)
59.7 (411.6)	–12.08 (248.6587)	2.767 (0.172766)	550260 (51121)	379.1 (1850.8)
58.7 (404.7)	–14.25 (247.4539)	2.801 (0.174853)	563160 (52320)	378.9 (1850.0)
57.7 (397.8)	–16.45 (246.2345)	2.835 (0.177003)	576217 (53533)	378.6 (1848.6)
56.7 (390.9)	–18.67 (244.9998)	2.871 (0.179217)	589435 (54761)	378.2 (1846.6)
55.7 (384.0)	–20.92 (243.7496)	2.907 (0.181499)	602821 (56004)	377.7 (1844.0)

B.3.4 Comparison of Results to Gas/Vapor Sizing Equations

B.3.4.1 For comparison purposes, the required orifice area of a PRV can be determined using the integration approach of the sample problem in B.3.3 and the vapor sizing formula presented in 5.6 [Equation (2) through Equation (7)]. The following data are given:

— required flow rate of 158,700 lb/h = 44.09 lb/s (72,000 kg/h = 20 kg/s);

— assume air is an ideal gas for use with the sizing equations.

B.3.4.2 For this comparison, the following data are derived:

— fluid molecular weight, M = 28.96;

— fluid compressibility factor, Z = 1.0;

— fluid ideal gas specific heat ratio at standard conditions, k = 1.4;

— ratio of specific heats coefficient, C = 356.06 (0.027);

— effective discharge coefficient for preliminary sizing, K_d = 0.975;

— all other correction factors (K_b and K_c) = 1.0.

B.3.4.3 Using the theoretical mass flux obtained from the numerical integration in example problem in B.3.3, one may determine the required effective discharge area.

In USC units:

$$A = \frac{W}{G \times K_d}$$

$$A = \frac{44.09}{379.1 \times 0.975} = 0.11928 \text{ ft}^2 = 17.176 \text{ in.}^2 \tag{B.29}$$

In SI units:

$$A = \frac{W}{G \times K_d}$$

$$A = \frac{20}{1851 \times 0.975} = 11082 \times 10^{-6} \text{ m}^2 = 11082 \text{ mm}^2 \tag{B.30}$$

where

G is the theoretical mass flux through the nozzle, lb/s·ft^2 (kg/s·m^2);

W is the required relief rate, lb/s (kg/s);

K_d is the effective coefficient of discharge (no units);

A is the required effective discharge area, ft^2 (m^2)

.

B.3.4.4 Using the critical vapor sizing equation [Equation (2) and Equation (5)], one may also determine the required effective discharge area:

In USC units:

$$A = \frac{W}{C \times K_d \times P_1 \times K_b \times K_c} \times \sqrt{\frac{T \times Z}{M}}$$

$$A = \frac{158{,}700}{356.06 \times 0.975 \times 114.7 \times 1.0 \times 1.0} \times \sqrt{\frac{540 \times 1.0}{28.96}} = 17.21 \text{ in.}^2 \qquad \text{(B.31)}$$

In SI units:

$$A = \frac{W}{C \times K_d \times P_1 \times K_b \times K_c} \times \sqrt{\frac{T \times Z}{M}}$$

$$A = \frac{72{,}000}{0.0270 \times 0.975 \times 790.8 \times 1.0 \times 1.0} \times \sqrt{\frac{300 \times 1.0}{28.96}} = 11{,}131 \text{ mm}^2 \qquad \text{(B.32)}$$

where

W is the required relief rate, lb/h (kg/h);

C is the specific heat ratio coefficient, a function of the ideal gas specific heat ratio;

K_d is the effective coefficient of discharge (no units);

P_1 is the pressure of the fluid entering the nozzle, psia (kPa);

T is the temperature of the fluid entering the nozzle, °R (K);

Z is the compressibility factor of the fluid entering the nozzle (no units);

M is the molecular weight of the fluid;

A is the required effective discharge area, in.^{2} (mm^2).

B.3.4.5 As an additional comparison, the theoretical flow for air as provided by UG-131 of the ASME Code is provided:

$$W_T = 356 \times A_T \times P \times \sqrt{\frac{M}{T}}$$

$$G_T = \frac{W_T}{A_T} = 356 \times P \times \sqrt{\frac{M}{T}}$$

$$G_T = 356 \times 114.7 \times \sqrt{\frac{28.96}{540.33}} = 9453.3 \text{ lb/hr} \times \text{in.}^2 \; (1846.2 \text{ kg/s} \times \text{m}^2) \qquad \text{(B.33)}$$

where

W_T is the theoretical flow rate, lb/h;

A_T is the actual discharge area, in.^{2};

P is the pressure of the fluid entering the nozzle, psia;

T is the temperature of the fluid entering the nozzle, °R;

M is the molecular weight of the fluid;

G_T is the theoretical mass flux, lb/h $\times$ in.^{2} .

B.3.5 Typical Refinery Gases and Vapors

B.3.5.1 Although many vapors encountered in refinery service do not follow the ideal gas law, in most cases, a PRV is adequately sized based on this assumption. This assumption is valid providing the *ideal* gas specific heat ratio is used as an estimate for the isentropic expansion coefficient and the ideal gas specific volume is calculated based on Equation (B.25) and Equation (B.26) (specifically, using a compressibility factor equal to one for the ideal gas).

B.3.5.2 A gas or vapor is close to ideal when the compressibility factor is close to one; nonetheless, the use of the ideal gas assumption for relief device sizing has been found not to introduce significant error for low molecular weight hydrocarbons with compressibility factors greater than 0.8.

B.4 Flow Equations for Two-phase Flow

The isentropic nozzle flux equation is also used in the development of the two-phase flow sizing equations. Please refer to Annex C for more information.

Annex C
(informative)

Sizing for Two-phase Liquid/Vapor Relief

C.1 Sizing for Two-phase Liquid/Vapor Relief

C.1.1 General

The methods for two-phase sizing, presented in this Annex, are among several techniques currently in use and newer methods are continuing to evolve as time goes on. It is recommended that the particular method to be used for a two-phase application be fully understood. It should be noted that the methods presented in this Annex have not been validated by test, nor is there any recognized procedure for certifying the capacity of PRVs in two-phase flow service.

C.1.2 Application of Equations

C.1.2.1 Many different scenarios are possible under the general category of two-phase liquid/vapor relief. In all of these scenarios either a two-phase mixture enters the PRV or a two-phase mixture is produced as the fluid moves through the valve. Vapor generation as a result of flashing shall be taken into account, since it may reduce the effective mass flow capacity of the valve. The methods presented in paragraphs C.2.1 through C.2.3 can be used for sizing PRVs in two-phase liquid/vapor scenarios. In addition, C.2.1 can be used for supercritical fluids in condensing two-phase flow. Use Table C.1 to determine which section to consult for a particular two-phase relief scenario.

C.1.2.2 The equations presented in C.2.1 are based on the Homogeneous Equilibrium Method [4], which assumes the fluid mixture behaves as a "pseudo-single phase fluid," with a density that is the volume-averaged density of the two phases. This method is based on the assumption that thermal and mechanical equilibrium exist as the two-phase fluid passes through the PRV (other specific assumptions or limitations are presented in the appropriate section). For high momentum discharges of two-phase systems in nozzles longer than 4 in. (10 cm), both thermal and mechanical equilibrium can be assumed. These assumptions correspond to the homogeneous equilibrium flow model (HEM).

Table C.1—Two-phase Liquid/Vapor Relief Scenarios for PRVs

Two-phase Liquid/Vapor Relief Scenario	Example	Section
Two-phase system (liquid vapor mixtures, including saturated liquid) enters PRV and flashes. No non-condensable [a] gas present. Also includes fluids both above and below the thermodynamic critical point in condensing two-phase flow.	Saturated liquid/vapor propane system enters PRV and the liquid propane flashes.	C.2.1 or C.2.2
Two-phase system (highly subcooled [b] liquid and either non-condensable gas, condensable vapor or both) enters PRV and does not flash.	Highly subcooled propane and nitrogen enters PRV and the propane does not flash.	C.2.1 or C.2.2
Two-phase system (the vapor at the inlet contains some non-condensable gas and the liquid is either saturated or subcooled) enters PRV and flashes. Non-condensable gas enters PRV.	Saturated liquid/vapor propane system and nitrogen enters PRV and the liquid propane flashes.	C.2.1 or C.2.2
Subcooled liquid (including saturated liquid) enters PRV and flashes. No condensable vapor or non-condensable gas enters PRV.	Subcooled propane enters PRV and flashes.	C.2.1 or C.2.3

[a] A noncondensable gas is a gas that is not easily condensed under normal process conditions. Common noncondensable gases include air, oxygen, nitrogen, hydrogen, carbon dioxide, carbon monoxide and hydrogen sulfide.

[b] The term highly subcooled is used to reinforce that the liquid does not flash passing through the PRV.

C.1.2.3 The equations presented in C.2.2 and C.2.3 are based on the Leung Omega Method [6], which is a version of the Homogeneous Equilibrium Method. In the procedures presented in C.2.2 and C.2.3, the Omega parameter is calculated based on specific volume data obtained from a flash calculation. This is often referred to as a two-point method since fluid properties are determined at the inlet relieving conditions and at flashed conditions at a lower pressure. The omega parameter itself is a correlation between the density of the two-phase fluid and the pressure, using the following relationship:

$$\omega = \frac{\frac{\rho_o}{\rho_x} - 1}{\frac{P_o}{P_x} - 1} = \frac{\frac{v_x}{v_o} - 1}{\frac{P_o}{P_x} - 1} \quad \text{(C.1)}$$

where

P is the pressure from the flash calculation (absolute);

ρ is the overall two-phase density from the flash calculation;

v is the overall two-phase specific volume from the flash calculation;

o is the initial condition (e.g. PRV inlet condition) for the flash;

x is the flash result at one lower pressure.

C.1.2.4 In most cases, a flash pressure at 90 % of the initial pressure provides a reasonable correlation parameter; however, lower flash pressures may be more appropriate under some conditions (e.g. near the thermodynamic critical point [12]). In some instances, it is possible to estimate the omega parameter using only the fluid properties at the relieving conditions (one-point method) [11]. Based on the assumptions used to develop the one-point omega parameter estimation technique, the use of this technique is generally not valid for any of the following situations:

— the nominal boiling range for a multi-component system is greater than 150 °F (the nominal boiling range is the difference in atmospheric boiling points of the lightest and heaviest components in the system);

— the fluid is close to its thermodynamic critical point ($T_r \geq 0.9$ or $P_r \geq 0.5$);

— the solubility of a non-condensable gas, if present, in the liquid is appreciable;

— the composition of a multi-component system contains more than 0.1 weight percent hydrogen;

— the gas fraction of a multi-component system with non-condensable gas is low;

$$\left(\frac{P_{vo}}{P_o} \geq 0.9 \text{ or } \frac{P_{go}}{P_o} \leq 0.1\right)$$

where

T_r is the reduced temperature at the PRV inlet;

P_r is the reduced pressure at the PRV inlet;

T_o is the temperature at the PRV inlet (°R);

P_o is the pressure at the PRV inlet (psia).

This is the PRV set pressure (psig) plus the allowable overpressure (psi) plus atmospheric pressure;

P_{vo} is the saturation (vapor) pressure corresponding to the inlet relieving temperature T_o (psia). For a multi-component system, use the bubble point pressure corresponding to T_o;

P_{go} is the non-condensable gas partial pressure at the PRV inlet (psia).

C.1.2.5 If any of these situations apply, the methods presented in sections C.2.1 through C.2.3 should be used.

C.1.2.6 A more rigorous approach using a fluid property database or vapor/liquid equilibrium (VLE) thermodynamic models incorporated into analytical or numerical methods based on HEM can be considered. See C.2.1 for more information.

C.1.3 Saturated Water Capacity for ASME Certified Safety Valves

For information about saturated water, see ASME Section VIII, Appendix 11.

C.1.4 Coefficient of Discharge

The value for the effective coefficient of discharge for two-phase flow is a subject of current debate [10], [13], [14], and is not likely to be resolved without actual testing of PRV behavior with two-phase fluids. As a result, a conservative recommendation regarding an estimate of the effective coefficient of discharge has been provided within the guidance on sizing methods, see paragraphs C.2.1.1, C.2.2.1, and C.2.3.1.

C.2 Sizing Methods

C.2.1 Sizing by Direct Integration of the Isentropic Nozzle Flow

C.2.1.1 General

C.2.1.1.1 The inlet nozzle of a relief device is commonly assumed to be the limiting flow element of a fully opened relief device and thus provides the model on which to determine the flow capacity of that relief device. To determine the maximum mass flux through a converging nozzle, the nozzle is assumed to be adiabatic and reversible (both constraints are needed for the isentropic assumption), a common assumption that has been borne through various experimental evidence for well-formed nozzles. The general energy balance for isentropic nozzle flow forms the basis for the mass flux calculation, as shown in Equation (C.2) and Equation (C.3).

In USC units:

$$G^2 = \left[\frac{-2 \times \int_{P_o}^{P} 4633 \times v \times dP}{v_t^2}\right]_{\max} = \left[(\rho_t^2) \times \left(-2 \times \int_{P_o}^{P} \frac{4633 \times dP}{\rho}\right)\right]_{\max} \tag{C.2}$$

In SI units:

$$G^2 = \left[\frac{-2 \times \int_{P_o}^{P} v \times dP}{v_t^2}\right]_{\max} = \left[(\rho_t^2) \times \left(-2 \times \int_{P_o}^{P} \frac{dP}{\rho}\right)\right]_{\max} \tag{C.3}$$

where

max is the maximization of this calculation, which accounts for potential choking of the fluid;

G is the mass flux, lb/s·ft^2 (kg/s·m^2);

v is the specific volume of the fluid, ft^3/lb (m^3/kg);

ρ is the mass density of the fluid, lb/ft^3 (kg/m^3);

P is the stagnation pressure of the fluid, psia (Pa);

o is the fluid condition at the inlet to the nozzle;

t is the fluid condition at the throat of the nozzle where the cross-sectional area is minimized.

C.2.1.1.2 It is important to note that this energy balance is irrespective of the non-ideality or compressibility of the fluid. As a result, this equation forms the basis for the two-phase flow calculations in the homogeneous equilibrium model employed in PRV calculations.

C.2.1.1.3 With the generic nozzle flow equation, the density of the fluid at various stagnation pressures from the inlet of the nozzle to the throat of the nozzle is needed. Given these values, the numerical integration calculation can be performed to determine the maximum mass flux through the nozzle. Using a suitable physical property database or an appropriate thermodynamic model, one can generate the values for the density at various pressures by starting with the fluid at the inlet stagnation conditions to determine the inlet stagnation entropy, S_o, and then performing successive isentropic flashes at lower pressures (P, S_o). The path followed for the fluid over this range is normally assumed to be isentropic; however, an isenthalpic path may be acceptable for many conditions (specifically, for low-quality mixtures that are far from the thermodynamic critical point). The pressure and density data points are generated for successively lower pressures until either the mass flux correlation reaches a maximum (representing choked conditions) or the actual backpressure on the nozzle is reached, whichever occurs first. It is at this point that the pressure at the minimal cross-sectional area at the nozzle (the throat pressure) is taken. For a known throat pressure, P_t, the energy balance can be written as shown in Equation (C.4) and Equation (C.5).

In USC units:

$$G^2 = (\rho_t^2) \times \left(-9266.1 \times \int_{P_o}^{P_t} \frac{dP}{\rho} \right) \tag{C.4}$$

In SI units:

$$G^2 = (\rho_t^2) \times \left(-2 \times \int_{P_o}^{P_t} \frac{dP}{\rho} \right) \tag{C.5}$$

C.2.1.1.4 The integral can be readily evaluated numerically for any fluid by direct summation over small pressure intervals:

$$\int_{P_0}^{P_t} \frac{dP}{\rho} \approx \sum_{i=0}^{t} 2 \times \left(\frac{P_{i+1} - P_i}{\rho_{i+1} + \rho_i} \right) \tag{C.6}$$

where

ρ_i is the overall mass density of the fluid at the stagnation pressure, P_i.

C.2.1.1.5 The overall mass density for a mixture in thermal and mechanical equilibrium can be calculated based on the density of each phase and the volume fraction of the vapor phase in the mixture:

$$\rho = (\alpha \times \rho_v) + (1 - \alpha)\rho_l \tag{C.7}$$

where

- ρ is the density of the two-phase mixture;
- ρ_v is the density of the vapor;
- ρ_l is the density of the liquid;
- α is the volume fraction of vapor phase in the mixture.

C.2.1.1.6 The volume fraction of the vapor phase in the mixture is related to the mass quality of the mixture (mass fraction of the vapor phase) by the following:

$$\frac{\alpha}{1-\alpha} = \frac{x}{1-x} \times \frac{\rho_l}{\rho_v} \tag{C.8}$$

C.2.1.1.7 Once the value for the mass flux has been determined, the required orifice area can be calculated using Equation (C.9) and Equation (C.10).

In USC units:

$$A = \frac{0.04W}{K_d K_b K_c K_v G} \tag{C.9}$$

In SI units:

$$A = \frac{277.8 \times W}{K_d K_b K_c K_v G} \tag{C.10}$$

where

- A is the required effective discharge area, in.^{2} (mm^2);
- W is the mass flow rate, lb/h (kg/h);
- K_d is the discharge coefficient. For a preliminary sizing estimation, a discharge coefficient of 0.85 can be used for a two-phase mixture or saturated liquid entering the PRV inlet. For the case of a liquid entering the PRV inlet, a discharge coefficient equal to 0.65 is consistent with the single-phase method in Equation (28) and Equation (29). Note that a value of 0.65 may result in a conservative valve size for liquids that are only slightly sub-cooled. The user may select other methods for determining a discharge coefficient [13], [14], [16];
- K_b is the backpressure correction factor for vapor that should be obtained from the valve manufacturer.

  For a preliminary sizing estimation, use Figure 30. The backpressure correction factor applies to balanced-bellows valves only;
- K_c is the combination correction factor for installations with a rupture disk upstream of the PRV (see 5.11.2);
  - — the combination correction factor is 1.0, when a rupture disk is not installed;

— the combination correction factor is 0.9, when a rupture disk is installed in combination with a PRV and the combination does not have a certified value.

K_v is the viscosity correction factor.

C.2.1.2 Example

C.2.1.2.1 In this example, the following data are given:

- — required two-phase flow rate caused by an operational upset of 300,000 lb/h (136,000 kg/h); the relief fluid is from a hydro-desulfurization vessel and contains a significant amount of hydrogen at a high pressure;
- — temperature at the PRV inlet of 80.4 °F (26.9 °C);
- — PRV set at 1,958 psi (13,500 kPag), the design pressure of the equipment;
- — downstream total backpressure of 29 psig (200 kPag) (superimposed backpressure = 0 psig, built-up backpressure = 29 psi);
- — allowable overpressure (accumulation) of 10 %;
- — a viscosity correction factor, K_v, of 1.0 is assumed.

C.2.1.2.2 In this example, the following values are calculated:

- — relieving pressure of 1.10 × 1958 = 2,153.8 psig (14,850 kPag);
- — percent of gauge backpressure = (29/1958) × 100 = 1.5 %, thus, the backpressure correction factor K_b = 1.0 (from Figure 30).

C.2.1.2.3 The following is a step-by-step procedure.

a) Step 1—Evaluate the stagnation entropy, S_o. Given the relief fluid composition, relief pressure, and relief temperature, the inlet stagnation entropy was determined by a thermodynamic engine to be –8.178 Btu/lb-mol-°R. Note that the entropy reference state may be different for various thermodynamic engines; therefore, this value may not be the same value obtained from other thermodynamic engines. As long as the same thermodynamic engine and models are used, the reference state is consistent and the flash calculations will be appropriate.

b) Step 2—Perform successive isentropic flashes to generate fluid properties and numerically evaluate nozzle flow integral. Without a known throat pressure, a number of successive isentropic flashes were performed to generate the fluid properties. A step size of 4 % of the absolute relief pressure was arbitrarily chosen to generate the flash pressures. Note that the smaller the step size, the closer the summation will be to the actual integration. With the fluid properties obtained from the isentropic flashes, Equation (C.6) was used to numerically evaluate the nozzle flow integral. The isentropic flashes were performed until the mass flux reached a maximum value. Results are presented in Table C.2.

c) Step 3—Find the maximum mass flux. The mass flux is maximized at a throat pressure of 1,214.4 psia (8.373 kPa) for a value of 4,830.8 lb/s-ft^2 (23,586 kg/s-m^2). This throat pressure is greater than the total backpressure indicated above; therefore, the flow is critical (choked) through the nozzle.

d) Step 4—Calculate the required orifice area. The required area A of the PRV is calculated from Equation (C.9) as follows:

$$A = \frac{0.04 \times 300{,}000}{0.85 \times 1.0 \times 1.0 \times 1.0 \times 4830.8} = 2.922 \text{ in.}^2 \qquad \text{(C.11)}$$

e) Step 5—Select an M-orifice PRV (3.600 in.^{2}).

Table C.2—Results for Direct Integration Example C.2.1.2

Pressure psia (Pa)	**Temperature** °F (K)	**Mass** Quality	**Density** lb/ft^3 (kg/m^3)	**Integrand** ft^2/s^2 (m^2/s^2)	**Summation** ft^2/s^2 (m^2/s^2)	**Mass Flux** lb/s-ft^2 (kg/s-m^2)
2168.5 (14951325)	80.3 (300.0)	1.000	5.18 (83.04)	0 (0)	0 (0)	0.0 (0.0)
2081.8 (14353272)	75.0 (297.0)	1.000	5.05 (80.88)	–78546 (–7297.2)	–78546 (–7297.2)	2001.1 (9770.4)
1995.0 (13755219)	69.4 (293.9)	1.000	4.91 (78.67)	–80696 (–7496.9)	–159243 (–14794.1)	2771.6 (13532.1)
1908.3 (13157166)	63.6 (290.7)	1.000	4.77 (76.41)	–83019 (–7712.7)	–242261 (–22506.8)	3320.5 (16212.3)
1821.5 (12559113)	57.6 (287.4)	1.000	4.63 (74.11)	–85535 (–7946.4)	–327796 (–30453.2)	3745.9 (18289.1)
1734.8 (11961060)	52.6 (284.6)	0.995	4.46 (71.46)	–88443 (–8216.6)	–416239 (–38669.9)	4070.5 (19874.2)
1648.1 (11363007)	48.2 (282.1)	0.988	4.28 (68.63)	–91899 (–8537.7)	–508138 (–47207.6)	4319.4 (21089.0)
1561.3 (10764954)	43.5 (279.5)	0.982	4.11 (65.76)	–95801 (–8900.2)	–603939 (–56107.7)	4511.7 (22027.9)
1474.6 (10166901)	38.6 (276.8)	0.975	3.92 (62.84)	–100120 (–9301.5)	–704059 (–65409.2)	4654.8 (22726.8)
1387.8 (9568848)	33.5 (274.0)	0.969	3.74 (59.86)	–104930 (–9748.3)	–808989 (–75157.5)	4753.7 (23209.5)
1301.1 (8970795)	28.1 (271.0)	0.963	3.55 (56.84)	–110319 (–10249.0)	–919307 (–85406.5)	4811.6 (23492.3)
1214.4 (8372742)	22.5 (267.9)	0.958	3.36 (53.77)	–116401 (–10814.0)	–1035709 (–96220.5)	4830.8 (23585.8)
1127.6 (7774689)	16.5 (264.6)	0.952	3.16 (50.63)	–123323 (–11457.1)	–1159032 (–107677.6)	4812.6 (23497.1)
1040.9 (7176636)	10.2 (261.1)	0.946	2.96 (47.44)	–131274 (–12195.8)	–1290306 (–119873.4)	4757.8 (23229.5)

C.2.2 Sizing for Two-phase Flashing or Non-flashing Flow Through a PRV Using the Omega Method

C.2.2.1 General

The method presented in this section can be used for sizing PRVs handling either flashing or non-flashing flow. These methods are also appropriate for fluids both above and below the thermodynamic critical point in condensing two-phase flow. Finally, the methods presented in this section can be used for liquids that are saturated as they enter the relief device.

Note that for saturated liquids, the methods presented in C.2.3 are equivalent to the methods presented in C.2.2.

In all cases, the Omega parameter is determined using the specific volume data obtained using fluid property data or flash calculations for a mixture at the stagnation conditions and one additional pressure (two-point method). The following procedure can be used.

a) Step 1—Calculate the Omega parameter, ω. To calculate the omega parameter using two pressure-specific volume data points, use Equation (C.12).

$$\omega = 9\left(\frac{v_9}{v_o} - 1\right) \tag{C.12}$$

where

v_9 is the specific volume evaluated at 90 % of the PRV inlet pressure, P_o in ft^3/lb;

when determining v_9, the flash calculation should be carried out isentropically, but an isenthalpic (adiabatic) flash is sufficient for low-quality mixtures far from the thermodynamic critical point.

v_o is the specific volume of the two-phase system at the PRV inlet, ft^3/lb.

b) Step 2—Determine if the flow is critical or sub-critical

b) .

$$\begin{aligned} P_c \geq P_a &\Rightarrow \text{critical flow} \\ P_c < P_a &\Rightarrow \text{subcritical flow} \end{aligned} \tag{C.13}$$

where

P_c is the critical pressure, psia (Pa);

$P_c = \eta_c P_o$

where

η_c is the critical pressure ratio from Figure C.1.

Figure C.1—Correlation for Nozzle Critical Flow of Flashing and Nonflashing Systems

NOTE This ratio can also be obtained from the following expression:

$$\eta_c^2 + (\omega^2 - 2\omega)(1 - \eta_c)^2 + 2\omega^2 \ln \eta_c + 2\omega^2(1 - \eta_c) = 0 \tag{C.14}$$

or from the following approximation:

$$\eta_c = [1 + (1.0446 - 0.0093431 \times \omega^{0.5}) \times \omega^{-0.56261}]^{(-0.70356 + 0.014685 \times \ln\omega)} \tag{C.15}$$

where

P_o is the pressure at the PRV inlet (psia or Pa);

this is the PRV set pressure (psig or Pag) plus the allowable overpressure (psi or Pa) plus atmospheric pressure.;

P_a is the downstream backpressure (psia or Pa).

c) Step 3—Calculate the mass flux. For critical flow, use Equation (C.16) or Equation (C.18). For sub-critical flow, use Equation (C.17) or Equation (C.19).

In USC units:

$$G = 68.09 \times \eta_c \sqrt{\frac{P_o}{v_o \omega}} \tag{C.16}$$

$$G = \frac{68.09 \times \{-2[\omega \ln \eta_a + (\omega - 1)(1 - \eta_a)]\}^{1/2}}{\omega\left(\frac{1}{\eta_a} - 1\right) + 1} \sqrt{P_o / v_o} \tag{C.17}$$

In SI units:

$$G = \eta_c \sqrt{\frac{P_o}{v_o \omega}} \tag{C.18}$$

$$G = \frac{\{-2 \times [\omega \ln \eta_a + (\omega - 1)(1 - \eta_a)]\}^{1/2}}{\omega\left(\frac{1}{\eta_a} - 1\right) + 1} \sqrt{P_o / v_o} \tag{C.19}$$

where

G is the mass flux, lb/s×ft^2 (kg/s·m^2);

P_o is the pressure at the PRV inlet in psia (Pa);

v_o is the specific volume of the two-phase system at the PRV inlet in ft^3/lb (m^3/kg);

η_a is the backpressure ratio, $\eta_a = \frac{P_a}{P_o}$.

d) Step 4—Calculate the required area of the PRV.

In USC units:

$$A = \frac{0.04W}{K_d K_b K_c K_v G} \tag{C.20}$$

In SI units:

$$A = \frac{277.8W}{K_d K_b K_c K_v G} \tag{C.21}$$

where

A is the required effective discharge area, in.^{2} (mm^2);

W is the mass flow rate, lb/h (kg/h);

K_d is the discharge coefficient. For a preliminary sizing estimation, a discharge coefficient of 0.85 can be used;

K_b is the backpressure correction factor for vapor that should be obtained from the valve manufacturer; for a preliminary sizing estimation, use Figure 30. The backpressure correction factor applies to balanced-bellows valves only;

K_c is the combination correction factor for installations with a rupture disk upstream of the PRV (see 5.11.2);

— the combination correction factor is 1.0, when a rupture disk is not installed;

— the combination correction factor is 0.9, when a rupture disk is installed in combination with a PRV and the combination does not have a certified value.

K_v is the viscosity correction factor.

C.2.2.2 Example

C.2.2.2.1 In this example, the following relief requirements are given:

— required crude column overhead two-phase flow rate caused by an operational upset of 477,430 lb/h (216,560 kg/h). This flow is downstream of the condenser;

— temperature at the PRV inlet of 200 °F (659.7 °R = 366.5 K);

— PRV set at 60 psig (413.7 kPag), the design pressure of the equipment;

— downstream total backpressure of 15 psig (29.7 psia) (204.7 kPa) (superimposed backpressure = 0 psig, built-up backpressure = 15 psi);

— two-phase specific volume at the PRV inlet of 0.3116 ft^3/lb (0.01945 m^3/kg);

— allowable overpressure (accumulation) of 10 %;

— for this example problem, a viscosity correction factor, K_v, of 1.0 is assumed.

C.2.2.2.2 In this example, the following values are calculated:

— relieving pressure of 1.10 × 60 = 66 psig (80.7 psia) (556.4 kPa);

— percent of gauge backpressure = (15/60) × 100 = 25 %, thus, the backpressure correction factor, K_b = 1.0 (from Figure 30).

— Since the downstream backpressure is greater than 10 % of the set pressure, a balanced-bellows PRV should be used.

C.2.2.2.3 The following is a step by step procedure.

a) Step 1—Calculate the omega parameter, ω. Equation (C.12) is used to calculate the omega parameter, ω. The specific volume evaluated at 0.9 × 80.7 = 72.63 psia (500.8 kPa) using the results of an isenthalpic (adiabatic) flash calculation from a process simulator is 0.3629 ft^3/lb (0.02265 m^3/kg). The omega parameter is calculated from Equation (C.12) as follows.

In USC units:

$$\omega = 9\left(\frac{0.3629}{0.3116} - 1\right) = 1.482 \tag{C.22}$$

In SI units:

$$\omega = 9\left(\frac{0.02265}{0.01945} - 1\right) = 1.482 \tag{C.23}$$

b) Step 2—Determine if the flow is critical or sub-critical. The critical pressure ratio η_c is 0.66 (from Figure C.1). The critical pressure, P_c, is calculated as follows.

In USC units:

$$P_c = 0.66 \times 80.7 = 53.26 \text{ psia} \tag{C.24}$$

In SI units:

$$P_c = 0.66 \times 556{,}379 = 367{,}210 \text{ Pa} \tag{C.25}$$

The flow is determined to be critical since $P_c > P_a$.

$$53.26 > 29.7$$

c) Step 3—Calculate the mass flux. The mass flux G is calculated from Equation (C.12) as follows.

In USC units:

$$G = 68.09 \times 0.66 \times \sqrt{\frac{80.7}{0.3116 \times 1.482}} = 594.1 \text{ lb/s} \cdot \text{ft}^2 \tag{C.26}$$

In SI units

:

$$G = 0.66 \times \sqrt{\frac{556{,}379}{0.01945 \times 1.482}} = 2900 \text{ kg/s} \cdot \text{m}^2 \tag{C.27}$$

d) Step 4—Calculate the required area of the PRV. The required area of the PRV, A, is calculated from Equation (C.20) or Equation (C.21) as follows.

In USC units:

$$A = \frac{0.04 \times 477{,}430}{0.85 \times 1.0 \times 1.0 \times 1.0 \times 594.1} = 37.8 \text{ in.}^2 \tag{C.28}$$

In SI units:

$$A = \frac{277.8 \times 216{,}560}{0.85 \times 1.0 \times 1.0 \times 1.0 \times 2900} = 24{,}400 \text{ mm}^2 \tag{C.29}$$

e) Step 5—Select the orifice area. This area requirement may be met by selecting two (2) Q orifice and (1) R orifice PRVs ($2 \times 11.05 + 1 \times 16.00 = 38.1$ in.^{2}). Since this example resulted in multiple valves, the required area could be re-calculated at 16 % overpressure.

C.2.3 Sizing for Subcooled Liquid at the PRV Inlet Using the Omega Method

C.2.3.1 PRVs Requiring Capacity Certification

The method presented in this section can be used for sizing PRVs handling a subcooled (including saturated) liquid at the inlet. No condensable vapor or non-condensable gas should be present at the inlet. The subcooled liquid either flashes upstream or downstream of the PRV throat depending on which subcooling region the flow falls into. The equations in this section also apply to all-liquid scenarios. The following procedure can be used.

a) Step 1—Calculate the saturated omega parameter, ω_s. To calculate the saturated omega parameter using two pressure-specific volume data points, use Equation (C.15).

$$\omega_s = 9\left(\frac{\rho_{lo}}{\rho_9} - 1\right) \tag{C.30}$$

where

ρ_{lo} is liquid density at the PRV inlet, lb/ft^3 (kg/m^3);

ρ_9 is density, lb/ft^3 (kg/m^3) evaluated at 90 % of the saturation (vapor) pressure, P_s, corresponding to the PRV inlet relieving temperature, T_o, lb/ft^3 (kg/m^3).

For a multi-component system, use the bubble point pressure corresponding to T_o for P_s. When determining ρ_9, the flash calculation should be carried out isentropically, but an isenthalpic (adiabatic) flash is sufficient for low-quality mixtures far from the thermodynamic critical point.

b) Step 2—Determine the subcooling region.

$$P_s \geq \eta_{st} P_o \Rightarrow \text{low subcooling region (flashing occurs upstream of throat)}$$

$$P_s < \eta_{st} P_o \Rightarrow \text{high subcooling region (flashing occurs at the throat)} \tag{C.31}$$

where

η_{st} is transition saturation pressure ratio,

$$\eta_{st} = \frac{2\omega_s}{1 + 2\omega_s} \tag{C.32}$$

where

P_o is pressure at the PRV inlet, psia (Pa).

This is the PRV set pressure, psig (Pag) plus the allowable overpressure (psi or Pa) plus atmospheric pressure.

c) Step 3—Determine if the flow is critical or sub-critical. For the low subcooling region, use the following comparisons:

$$P_c \geq P_a \Rightarrow \text{critical flow}$$

$$P_c < P_a \Rightarrow \text{subcritical flow} \tag{C.33}$$

For the high subcooling region, use the following comparisons:

$$P_s \geq P_a \Rightarrow \text{critical flow}$$

$$P_s < P_a \Rightarrow \text{subcritical flow (all-liquid flow)} \tag{C.34}$$

where

P_c is the critical pressure in psia (Pa).

$$P_c = \eta_c P_o \tag{C.35}$$

where

η_c is the critical pressure ratio from Figure C.2 using the value of η_s.

Figure C.2—Correlation for Nozzle Critical Flow of Inlet Subcooled Liquid

For $\eta_s \leq \eta_{st}$:

$$\eta_c = \eta_s \tag{C.36}$$

For $\eta_s > \eta_{st}$, the critical pressure ratio, η_c, can be calculated implicitly using Equation (C.37) or approximated using Equation (C.38).

$$\frac{\left(\omega_s + \frac{1}{\omega_s} - 2\right)}{2 \times \eta_s} \times \eta_c^2 - 2 \times (\omega_s - 1) \times \eta_c + \omega_s \times \eta_s \times \ln\left(\frac{\eta_c}{\eta_s}\right) + \frac{3}{2} \times \omega_s \times \eta_s - 1 = 0 \tag{C.37}$$

$$\eta_c = \eta_s \times \left(\frac{2 \times \omega}{2 \times \omega - 1}\right) \times \left[1 - \sqrt{1 - \frac{1}{\eta_s} \times \left(\frac{2 \times \omega - 1}{2 \times \omega}\right)}\right] \tag{C.38}$$

where

η_s is the saturation pressure ratio as calculated in Equation (C.39).

$$\eta_s = \frac{P_s}{P_o} \tag{C.39}$$

where

P_a is the downstream backpressure in psia (Pa);

η_a is the subcritical pressure ratio per Equation (C.40).

$$\eta_a = \frac{P_a}{P_o} \tag{C.40}$$

d) Step 4—Calculate the mass flux. In the low subcooling region, use Equation (C.41) or Equation (C.43). If the flow is critical, use η_c for η and if the flow is sub-critical, use η_a for η. In the high subcooling region, use Equation (C.42) or Equation (C.44). If the flow is critical, use P_s for P and if the flow is sub-critical (all-liquid flow), use P_a for P.

In USC units:

$$G = \frac{68.09 \times \left\{ 2(1-\eta_s) + 2\left[\omega_s \eta_s \ln\left(\frac{\eta_s}{\eta}\right) - (\omega_s - 1)(\eta_s - \eta) \right] \right\}^{1/2}}{\omega_s \left(\frac{\eta_s}{\eta} - 1 \right) + 1} \sqrt{P_o \cdot \rho_{lo}}$$

$$\cancel{G = \frac{68.09 \times \left\{ 2(1-\eta_s) + 2\left[\omega_s \eta_s \ln\left(\frac{\eta_s}{\eta}\right) - (\omega_s - 1)(\eta_s - \eta) \right] \right\}^{1/2}}{\omega_s \left(\frac{\eta_s}{\eta} - 1 \right) + 1} \sqrt{P \cdot \rho_{lo}}} \tag{C.41}$$

$$G = 96.3 \times [\rho_{lo}(P_o - P)]^{1/2} \tag{C.42}$$

In SI units:

$$\cancel{G = \frac{\left(2(1-\eta_s) + 2\left[\omega_s \eta_s \ln\left(\frac{\eta_s}{\eta}\right) - (\omega_s - 1)(\eta_s - \eta) \right] \right)^{1/2}}{\omega_s \left(\frac{\eta_s}{\eta} - 1 \right) + 1} \sqrt{P \cdot \rho_{lo}}}$$

$$G = \frac{\left(2(1-\eta_s) + 2\left[\omega_s \eta_s \ln\left(\frac{\eta_s}{\eta}\right) - (\omega_s - 1)(\eta_s - \eta) \right] \right)^{1/2}}{\omega_s \left(\frac{\eta_s}{\eta} - 1 \right) + 1} \sqrt{P_o \cdot \rho_{lo}} \tag{C.43}$$

$$G = 1.414[\rho_{lo}(P_o - P)]^{1/2} \tag{C.44}$$

where

G is the mass flux, lb/s×ft^2 (kg/s·m^2);

η is the backpressure ratio.

e) Step 5—Calculate the required area of the PRV. Equation (C.45) and Equation (C.46) are only applicable to turbulent flow systems. Most two-phase relief scenarios will be within the turbulent flow regime.

In USC units:

$$A = 0.3208 \frac{Q \rho_{lo}}{K_d K_b K_v G} \quad A = 0.3208 \frac{Q \times \rho_{lo}}{K_d K_b K_c K_v G} \tag{C.45}$$

In SI units:

$$A = 16.67 \frac{Q \times \rho_{lo}}{K_d K_b K_v G} \quad A = 16.67 \frac{Q \times \rho_{lo}}{K_d K_b K_c K_v G} \tag{C.46}$$

where

A is the required effective discharge area, in.^{2} (mm^2);

G is the mass flux, lb/s×ft^2 (kg/s·m^2);

Q is the volumetric flow rate, gal/min (L/min);

K_d is the discharge coefficient.

For a preliminary sizing estimation, a discharge coefficient 0.65 for subcooled liquids and 0.85 for saturated liquids can be used. A value of 0.65 for slightly subcooled liquids may result in a conservative valve size. The user may select other methods for determining a discharge coefficient [13], [14], [16];

K_b is the backpressure correction factor for liquid that should be obtained from the valve manufacturer.

For a preliminary sizing estimation, use Figure C.3. The backpressure correction factor applies to balanced-bellows valves only;

K_c is the combination correction factor for installations with a rupture disk upstream of the PRV (see 5.11.2);

— the combination correction factor is 1.0, when a rupture disk is not installed,

— the combination correction factor is 0.9, when a rupture disk is installed in combination with a PRV and the combination does not have a certified value.

K_v is the viscosity correction factor.

f) Step 6—Select the orifice size.

Figure C.3—Backpressure Correction Factor, K_b, for Balanced-bellows PRVs (Liquids)

C.2.3.2 Example

C.2.3.2.1 In this example, the following relief requirements are given:

— required propane volumetric flow rate caused by blocked in pump of 100 gal/min (378.5 L/min);

— PRV set at 260 psig (1,792.6 kPag), the design pressure of the equipment;

— downstream total backpressure of 10 psig (24.7 psia) (170.3 kPa) (superimposed backpressure = 0 psig, built-up backpressure = 10 psi);

— temperature at the PRV inlet of 60 °F (519.67 °R) (288.7 K);

— liquid propane density at the PRV inlet of 31.920 lb/ft^3 (511.3 kg/m^3);

— liquid propane specific heat at constant pressure at the PRV inlet of 0.6365 Btu/lb×°R (2.665 kJ/kg·K);

— saturation pressure of propane corresponding to 60 °F of 107.6 psia (741.9 kPa);

— specific volume of propane liquid at the saturation pressure of 0.03160 ft^3/lb (0.00197 m^3/kg);

— specific volume of propane vapor at the saturation pressure of 1.001 ft^3/lb (0.0625 m^3/kg);

— latent heat of vaporization for propane at the saturation pressure of 152.3 Btu/lb (354.2 kJ/kg);

— for this example problem, a viscosity correction factor, K_v, of 1.0 is assumed.

C.2.3.2.2 In this example, the following values are calculated:

— overpressure of 10 %;

— relieving pressure of 1.10 × 260 = 286 psig (300.7 psia) (2,073.3 kPa);

— percent of gauge backpressure = (10/260) × 100 = 3.8 %.

— Since the downstream backpressure is less than 10 % of the set pressure, a conventional PRV may be used. Thus, the backpressure correction factor K_b = 1.0.

— since the propane is subcooled, a discharge coefficient, K_d, of 0.65 can be used.

C.2.3.2.3 The following is a step-by-step procedure:

a) Step 1—Calculate the saturated omega parameter, ω_s. Since the propane system is a single-component system far from its thermodynamic critical point, the saturated omega parameter, ω_s, could be calculated using the one-point projection technique; however, in this example, Equation (C.30) is chosen to calculate the omega parameter, ω. The specific volume evaluated at 0.9 × 107.6 = 96.84 psia (667.7 kPa) using the results of an isenthalpic (adiabatic) flash calculation from a process simulator is 0.06097 ft^3/lb (0.00381 m^3/kg). This gives a fluid density of 16.40 lb/ft^3 (262.7 kg/m^3). The omega parameter is calculated from Equation (C.30) as follows.

In USC units:

$$\omega_s = 9 \times \left(\frac{31.920}{16.402} - 1\right) = 8.515 \tag{C.47}$$

In SI units:

$$\omega_s = 9 \times \left(\frac{511.3}{262.7} - 1\right) = 8.515 \tag{C.48}$$

b) Step 2—Determine the subcooling region. The transition saturation pressure ratio, η_{st}, is calculated as follows:

$$\eta_{st} = \frac{2 \times 8.515}{1 + 2 \times 8.515} = 0.9445 \tag{C.49}$$

The liquid is determined to fall into the high subcooling region since:

$$P_s < \eta_{st} P_o$$
$$107.6 < 0.9445 \times 300.7 = 284.0 \tag{C.50}$$

c) Step 3—Determine if the flow is critical or sub-critical. The flow is determined to be critical since:

$$P_s > P_a$$
$$107.6 > 24.7 \tag{C.51}$$

d) Step 4—Calculate the mass flux. The mass flux, G, is calculated from Equation (C.42) as follows.

In USC units:

$$G = 96.3 \times [31.92 \times (300.7 - 107.6)]^{1/2} = 7560 \text{ lb/s} \times \text{ft}^2 \tag{C.52}$$

In SI units:

$$G = 1.414 \times [511.3 \times (2{,}073{,}250 - 741{,}875)]^{1/2} = 36{,}890 \text{ kg/s} \times \text{m}^2 \tag{C.53}$$

e) Step 5—Calculate the required area of the PRV. The required area, A, of the PRV is calculated from Equation (C.45) and Equation (C.46) as follows.

In USC units:

$$A = 0.3208 \times \frac{100 \times 31.92}{0.65 \times 1.0 \times 1.0 \times 7560} = 0.208 \text{ in.}^2 \tag{C.54}$$

In SI units:

$$A = 16.67 \times \frac{378.5 \times 511.3}{0.65 \times 1.0 \times 1.0 \times 36{,}890} = 134.5 \text{ mm}^2 \tag{C.55}$$

f) Step 6—Select an F orifice PRV (0.307 in.^{2}).

C.2.3.3 PRVs Not Requiring Capacity Certification

If the PRV is one that was never certified in liquid service (see 5.9 for a discussion on non-certified PRVs), then the mass flux calculated using Equation (C.45) and Equation (C.46) needs to be adjusted to account for the higher overpressures required to get the valve to go to full lift. Equation (C.45) and Equation (C.46) are modified as shown in Equation (C.56) and Equation (C.57) to handle liquid PRVs that have never been certified:

In USC units:

$$A = 0.3208 \frac{Q \rho_{lo}}{K_d K_b K_v G} \times \frac{\sqrt{P_1 - P_2}}{K_p \sqrt{1.25 \times P_s - P_2}} \tag{C.56}$$

In SI units:

$$A = 16.67 \frac{Q \rho_{lo}}{K_d K_b K_v G} \times \frac{\sqrt{P_1 - P_2}}{K_p \sqrt{1.25 \times P_s - P_2}} \tag{C.57}$$

where

K_p is the correction factor due to overpressure. At 25 % overpressure, K_p = 1.0. For overpressures other than 25 %, K_p is determined from Figure 38;

P_s is the set pressure, psig (Pag);

P_1 is the upstream relieving pressure, psig (Pag). This is the set pressure plus allowable overpressure;

P_2 is the total backpressure, psig (Pag).

Annex D
(informative)

Pressure Relief Valve Specification Sheets

Table D.1—Instructions for Spring-loaded PRV Specification Sheet (Continued)

Line No.	Instruction
1	Fill in item number.
2	Fill in user's pressure relief valve (PRV) identification or tag number.
3	Specify service, line, or equipment to be protected. An equipment tag number and description should be included
4	Specify number of valves required. A Specification Sheet for each PRV of a multiple valve installation should be completed
5	Specify the applicable code(s) and whether Code Symbol nameplate stamping is required.
6	Valve should comply with API 526.
7	Check fire or specify other basis of selection. Consideration should be given to supplying all applicable overpressure scenarios. As a minimum, provide the governing sizing scenario for the vapor sizing case as well as the liquid sizing case, if both are applicable.
8	Specify whether a rupture disk is being used under the PRV inlet.
9	Specify whether the PRV is conventional, balanced-bellows, and/or balanced piston.
10	Give description of PRV inlet design (full nozzle, semi-nozzle, or other type). Full nozzles are integral wetted components which offer the advantage of removal for maintenance and replacement. The biggest advantage of a full nozzle is that the nozzle material can be upgraded without upgrading the metallurgy of the base. Semi-nozzles have the potential for leakage and require threaded connection that may require gaskets or welding. The use of a semi-nozzle also requires that the material of the base be compatible with the process fluid.
11	Specify open or closed bonnet. Open bonnets are limited to services that are non-hazardous, such as steam, air and water, since the process fluid will escape through the open bonnet upon actuation of the PRV. Open bonnets allow the spring to be cooled by ambient conditions in high temperature applications.
12	Specify metal-to-metal or resilient seat. Metal-to-metal seats are typical. Soft-seat designs are available that will minimize leakage. Operating temperature, pressure and fluid corrosivity may limit the applicability of soft goods.
13	If other than API 527, specify seat test requirements. The ASME Code requires PRVs to be tested in accordance with API 527. Other codes may have alternate requirements.
14	Specify pipe size of inlet, flange rating, and type of facing. Inlet flange rating and facing should be chosen to meet the requirements of the upstream-protected equipment specification.
15	Specify pipe size of outlet, flange rating, and type of facing. Outlet flange rating and facing should be chosen to meet the requirements of the downstream discharge system. Typically, the discharge system will be designed to ANSI Class 150. However, higher flange class ratings may be required when the PRV discharge is to a closed system.
16	Specify type of connection if other than flanged (e.g. threaded, socket weld, etc.).
17	Specify material of body. API 526 provides general construction materials that are appropriate for selection based on pressure and temperature only. Where added corrosion resistance is needed to meet the severity of the process fluid, the user should provide this information here.

Users of instructions should not rely exclusively on the information contained in this document. Sound business, scientific, engineering, and safety judgment should be used in employing the information contained herein.

18	Specify material of bonnet (see guidance for Line 17).
19	Specify material of seat or nozzle and disk. These materials are chosen to meet process conditions.
20	If a resilient seat is required, specify material. Resilient, soft-seated designs provide tighter sealing and can reduce the amount of leakage through the PRV. Operating temperature, pressure, and fluid corrosivity may limit the applicability of soft goods. Typical resilient seat materials include fluoroelastomers, nitrile, and ethylene propylene (EPR). Other materials are available to meet process conditions.
21	Specify material of guide (see guidance for Line 17).
22	Specify material of adjusting ring or rings (see guidance for Line 17).
23	Specify material of spring and spring washer (see guidance for Line 17).
24	Specify material of bellows (see guidance for Line 17).
25	Specify material of balanced piston (see guidance for Line 17).
26	Specify the NACE document, if any, with which the materials should comply (see guidance for Line 17). The NACE requirement is typically needed for processes containing H_2S. Special attention is given to spring or bellows materials since they are subject to stress corrosion cracking.
27	Specify material of internal gaskets. Internal gasket material should be compatible with the process fluid on the discharge of the PRV.
28	Specify any other special material requirements (see guidance for Line 17).
29	Specify screwed or bolted cap. Screwed caps are more economical. Bolted caps offer higher protection against leakage.
30	Specify if the valve is to have a plain or packed lifting lever, or none. Lifting levers are required by the ASME Code in air, steam, and hot water services. Note that ASME Code Case 2203 should be specified on this datasheet, if the purchaser wants to waive this requirement. A packed lifting lever is used when there is backpressure due to closed discharge systems or when there is an overpressure scenario that might relieve hot liquid.
31	Specify whether a test gag is required. A test gag shall only be used during hydrostatic testing of the protected equipment and shall be removed and replaced with a plug during operation.
32	Specify whether a bug screen in the bonnet vent of a bellows, or balanced piston valve is required.
33	Specify other accessories that are required (e.g. limit switch).
34	Indicate flowing fluid and state (liquid, gas, vapor, or two-phase). Note that in two-phase applications, the purchaser is required to provide the manufacturer with the relieving mass flux (see Line 36).
35	Specify quantity of fluid that the valve is required to relieve at relieving conditions and unit of measure (such as lb/h, gal/min, or ft^2/min).
36	Specify the mass flux and the basis for the calculations. This is required for two-phase applications only. The basis for the calculations could be the HEM Omega method or the direct integration method (HDI) as discussed in of API 520, Part I, Annex C. Other calculations methods are acceptable.
37	Specify the molecular weight or specific gravity of the fluid at the flowing temperature. Molecular weight is required for vapor sizing only, specific gravity is required for liquid sizing only. Nothing is required here for two-phase applications provided the mass flux is provided in Line 36.
38	Specify viscosity and unit of measure at the flowing temperature. Viscosity is required for the liquid sizing case only.

39	Specify maximum operating pressure and unit of measure. Note that this should typically be the same as the maximum operating pressure of the protected equipment.
40	Specify set pressure and unit of measure.
41	Specify maximum blowdown as a percent of set pressure if different than manufacturer's standard.
42	Specify the latent heat of vaporization. This is only required for vapor sizing for the Fire Case only.
43	Specify the operating temperature and unit of measure. Note that this may not be the same as the operating temperature of the protected equipment due to heat loss from the inlet line. Insulated (or insulated and traced) inlet piping and PRVs may have operating temperatures close to the operating temperature of the protected equipment.
44	Specify the actual temperature at relieving conditions and unit of measure.
45	Specify the increase in pressure in the discharge header as a result of flow. Conventional PRVs are limited in the amount of built-up backpressure they can tolerate. In some cases, a balanced type of valve may be required.
46	Specify the amount of superimposed backpressure that normally exists on the valve outlet and unit of measure. If backpressure is variable, specify the minimum and maximum.
47	Manufacturer to specify the set pressure at which the valve is adjusted to open on the test stand. This value is the cold differential test pressure (CDTP). The CDTP includes backpressure and/or temperature corrections to the set pressure for the service conditions.
48	Specify the overpressure allowed, as a percent of set pressure or as a unit of measure.
49	Specify the compressibility factor, if used. The compressibility factor is required for vapor sizing cases only.
50	Specify the ideal gas specific heat ratio as $k = C_p / C_v$. The specific heat ratio is required for vapor sizing cases only.
51	2 Specify the calculated orifice area, in.^{2} (mm). Note that the purchaser should supply this value and it should be validated by the manufacturer, see Line 58.
52	2 Specify the selected effective discharge area, in.^{2} (mm). The effective discharge areas are provided in API 526. Special discharge areas exist that exceed the API 526 effective areas. Consult the manufacturer for additional information.
53	Specify the letter designation of the selected orifice. The effective orifice letter designations are provided in API 526. Consult the manufacturer for additional information.
54	Fill in the name of the manufacturer, if desired.
55	Fill in the manufacturer's model or type numbers, if desired.
56	Confirmation of orifice sizing calculations required from vendor. It is recommended to always have the manufacturer verify the sizing using actual orifice areas and rated coefficients of discharge.

Users of instructions should not rely exclusively on the information contained in this document. Sound business, scientific, engineering, and safety judgment should be used in employing the information contained herein.

Table D.2—Instructions for Pilot-operated PRV Specification Sheet (Continued)

Line No.	Instructions
1	Fill in item number.
2	Fill in user's pressure relief valve (PRV) identification number.
3	Specify service, line, or equipment to be protected. An equipment tag number and description should be included.
4	Specify number of valves required. A Specification Sheet for each PRV of a multiple valve installation should be completed.
5	Specify the applicable code(s) and whether Code Symbol nameplate stamping is required.
6	Valve should comply with API 526.
7	Check fire or specify other basis of selection. Consideration should be given to supplying all applicable overpressure scenarios. As a minimum, provide the governing sizing scenario for the vapor sizing case as well as the liquid sizing case, if both are applicable.
8	Specify whether a rupture disk is being used under the main valve inlet.
9	Specify type of main valve operation, piston, diaphragm, or bellows.
10	Specify number of pilots per main valve. Multiple pilots are sometimes advantageous in critical applications. The multiple pilots allow switching, inspecting, repairing and testing of pilots in applications when higher inspection frequencies are warranted.
11	Specify if pilot is flow or non-flowing type. Typically, a non-flowing pilot is preferred.
12	Specify type of action, pop or modulating. Modulating pilots minimize the amount of fluid discharged during actuation and also have advantages in high inlet pressure drop applications by reducing the amount of flow to the valve.
13	Specify sensing point as integral at main valve inlet or at a remote location. If remote sensing, the purchaser should supply the distance between the pilot and its sense point.
14	Specify metal-to-metal or resilient seat. Resilient seats are typical. Operating temperature, pressure and fluid corrosivity or compatibility may limit the applicability of soft goods.
15	If other than API 527, specify seat tightness test requirements. The ASME Code requires PRVs to be tested in accordance with API 527. Other codes may have alternate requirements.
16	Specify if pilot venting is to atmosphere, valve outlet, or other closed system. If pilot vent discharges to a closed system that is different from the main valve discharge, the purchaser needs to supply the pilot vent discharge system pressure.
17	Specify pipe size of inlet, flange rating, and type of facing. Inlet flange rating and facing should be chosen to meet the requirements of the upstream-protected equipment specification.
18	Specify pipe size of outlet, flange rating, and type of facing. Outlet flange rating and facing should be chosen to meet the requirements of the downstream discharge system. Typically, the discharge system will be designed to ANSI Class 150. However, higher flange class ratings may be required when the PRV discharge is to a closed system.
19	Specify type of connection if other than flanges (e.g. threaded, socket weld, etc.).
20	Specify material of body. API 526 provides general construction materials that are appropriate for selection based on pressure and temperature only. Where added corrosion resistance is needed to meet the severity of the process fluid, the user should provide this information here.
21	Specify material of seat or nozzle and piston (see guidance for Line 20).

22	Specify material of resilient seat (if required) and seals. Typical resilient seat materials include fluoroelastomers, nitrile, and ethylene propylene (EPR). Other materials are available to meet process conditions.
23	Specify material of piston seal. Typical seal materials include fluoroelastomers, nitrile, and ethylene propylene (EPR). Other materials are available to meet process conditions.
24	Specify material of piston liner or guide. Typically, this information is completed by the manufacturer.
25	Specify material of main valve diaphragm or bellows. Diaphragm materials are chosen for compatibility with the process fluid. Typical diaphragm materials include fluoroelastomers, nitrile, and ethylene propylene (EPR).
26	Specify material of pilot body and bonnet (see guidance for Line 20).
27	Specify material of pilot internals (see guidance for Line 20).
28	Specify material of seat and seals of the pilot. Typically, this information is completed by the manufacturer.
29	Specify material of pilot diaphragm. Diaphragm materials are chosen for compatibility with the process fluid. Typical diaphragm materials include fluoroelastomers, nitrile, and ethylene propylene (EPR).
30	Specify material of tubing and fittings. Typically, stainless steel, other materials are available based on compatibility with the process fluid.
31	Specify material of filter body and cartridge. Typically, stainless steel, other materials are available based on compatibility with the process fluid.
32	Specify material of pilot spring. Generic pilot spring materials are not provided in API 526 as a function of pressure and temperature. Since the pilot spring is not exposed to the process fluid atmosphere, material selection is generally based on the ambient temperature and ambient environment. Typically, this information is completed by the manufacturer.
33	Specify the NACE document, if any, with which the materials should comply (see guidance for Line 20). The NACE requirement is typically needed for processes containing H_2S. Special attention is given to spring or bellows materials since they are subject to stress corrosion cracking.
34	Specify any other special material requirements.
35	Specify if external filter is required. External filters are used in the pilot sense line when there are contaminants or high concentrations of solids in the process fluid that may cause plugging in the pilot.
36	Specify if the pilot is to have a plain or packed lifting lever, or none. Lifting levers are required by the ASME Code in air, steam, and hot water services. Note that ASME Code Case 2203 should be specified on this datasheet, if the purchaser wants to waive this requirement.
37	Specify if field test connection is required. Test connections can be used to verify the set pressure of the pilot while in operation without removing the valve from the installation.
38	Specify if field test indicator is required. The field test indicator indicates that the valve has opened. This device indicates any opening during operation as well as opening during field tests (see Line 37).
39	Specify if backflow preventer is required. Back flow preventers cause the main valve to remain closed when the outlet pressure exceeds the inlet pressure. This is of particular importance during shutdowns when the protected equipment is isolated from a live discharge system.
40	Specify if manual blowdown valve is required. Manual blowdown valves allow opening of the main valve without actuation of the pilot.
41	Specify if test gag is required. A test gag shall only be used during hydrostatic testing of the protected equipment and shall be removed and replaced with a plug during operation.

Users of instructions should not rely exclusively on the information contained in this document. Sound business, scientific, engineering, and safety judgment should be used in employing the information contained herein.

42	Specify other accessories that are required.
43	Indicate flowing fluid and state (liquid, gas, vapor or two-phase). Note that in two-phase applications, the purchaser is required to provide the manufacturer with the relieving mass flux (see Line 45).
44	Specify quantity of fluid that the valve is required to relieve at relieving conditions and unit of measure (such as lb/h, gal/min, or ft2/min). If the fire case is the governing vapor overpressure scenario, this relieving rate is not required provided the latent heat of the fluid is provided in Line 51.
45	Specify the mass flux and the basis for the calculations. This is required for two-phase applications only. The basis for the calculation could be the HEM Omega method or the direct integration method (HDI) as discussed in of API 520, Part I, Annex C. Other calculations methods are acceptable.
46	Specify the molecular weight or specific gravity of the fluid at the flowing temperature. Molecular weight is required for vapor sizing only; specific gravity is required for liquid sizing only.
47	Specify viscosity and unit of measure at the flowing temperature. Viscosity is required for liquid sizing cases.
48	Specify maximum operating pressure and unit of measure. Note that this should typically be the same as the maximum operating pressure of the protected equipment.
49	Specify set pressure and unit of measure.
50	Specify the maximum blowdown as a percent of set pressure if different than manufacturer's standard.
51	Specify the latent heat of vaporization and unit of measure. This is required for vapor sizing for the fire case only.
52	Specify the operating temperature and unit of measure. Note that this should typically be the same as the operating temperature of the protected equipment.
53	Specify the actual temperature at relieving conditions and unit of measure.
54	Specify the increase in pressure in the discharge header as a result of flow.
55	Specify the amount of superimposed backpressure that normally exists on the valve outlet and unit of measure. If backpressure is variable, specify the minimum and maximum.
56	Specify the set pressure at which the valve is adjusted to open on the test stand. This value is the cold differential test pressure (CDTP). The CDTP includes corrections to the set pressure for the service conditions of backpressure or temperature or both.
57	Specify the overpressure allowed, as a percent of set pressure or as a unit of measure.
58	Specify the compressibility factor, if used. The compressibility factor is required for vapor sizing cases only.
59	Specify the ideal gas specific heat ratio as $k = C_p/C_v$. The specific heat ratio is required for vapor sizing cases only (see Annex B).
60	Specify the calculated discharge area, in.^{2} (mm2). Note that the purchaser should supply this value and it should be validated by the manufacturer, see Line 66.
61	Specify the selected effective discharge area, in.^{2} (mm2). The effective discharge areas are provided in API 526. Special discharge areas exist that exceed the API 526 effective areas. Consult the manufacturer for additional information.
62	Specify the letter designation of the selected discharge area. The effective discharge area letter designations are provided in API 526. Consult the manufacturer for additional information.
63	Fill in the name of the manufacturer, if desired.
64	Fill in the manufacturer's model or type numbers, if desired.
65	Confirmation of orifice sizing calculations required from vendor. It is recommended to always have the vendor verify the sizing using actual discharge areas and rated coefficients of discharge.

Users of instructions should not rely exclusively on the information contained in this document. Sound business, scientific, engineering, and safety judgment should be used in employing the information contained herein.

SPRING-LOADED PRESSURE RELIEF VALVE SPECIFICATION SHEET	Sheet No. Page of
	Requisition No.
	Job No.
	Date
	Revised
	By

GENERAL		BASIS OF SELECTION	
1.	Item Number:	5.	Code: ASME VIII [ ] Stamp Required: Yes [ ] No [ ]
2.	Tag Number:		Other [ ] Specify:
3.	Service, Line, or Equipment Number:	6.	Comply with API 526: Yes [ ] No [ ]
4.	Number Required:	7.	Fire [ ] Other [ ] Specify:
		8.	Rupture Disk: Yes [ ] No [ ]
VALVE DESIGN		**MATERIALS**	
9.	Design Type:	17.	Body:
	Conventional [ ] Bellows [ ] Balanced Piston [ ]	18.	Bonnet:
10.	Nozzle Type: Full [ ] Semi [ ]	19.	Seat (Nozzle): Disk:
	Other [ ] Specify:	20.	Resilient Seat:
11.	Bonnet Type: Open [ ] Closed [ ]	21.	Guide:
12.	Seat Type: Metal-to-Metal [ ] Resilient [ ]	22.	Adjusting Ring(s):
13.	Seat Tightness: API 527 [ ]	23.	Spring: Washer:
	Other [ ] Specify:	24.	Bellows:
		25.	Balanced Piston:
CONNECTIONS		26.	Comply with NACE: Yes [ ] No [ ]
14.	Inlet Size: Rating: Facing:	27.	Internal Gasket Materials:
15.	Outlet Size: Rating: Facing:	28.	Other [ ] Specify:
16.	Other [ ] Specify:		
SERVICE CONDITIONS		**ACCESSORIES**	
34.	Fluid and State:	29.	Cap: Screwed [ ] Bolted [ ]
35.	Required Capacity per Valve and Units:	30.	Lifting Lever: Plain [ ] Packed [ ] None [ ]
36.	Mass Flux and Basis:	31.	Test Gag: Yes [ ] No [ ]
37.	Molecular Weight or Specific Gravity:	32.	Bug Screen: Yes [ ] No [ ]
38.	Viscosity at Flowing Temperature and Units:	33.	Other [ ] Specify:
39.	Operating Pressure and Units:		
40.	Set Pressure and Units:		
41.	Blowdown: Standard [ ] Other [ ]		
42.	Latent Heat of Vaporization and Units:		
43.	Operating Temperature and Units:	**SIZING AND SELECTION**	
44.	Relieving Temperature and Units:	51.	Calculated Orifice Area (in square inches):
45.	Built-up Back Pressure and Units:	52.	Selected Effective Orifice Area (in square inches):
46.	Superimposed Back Pressure and Units:	53.	Orifice Designation (letter):
47.	Cold Differential Test Pressure and Units:	54.	Manufacturer:
48.	Allowable Overpressure in Percent or Units:	55.	Model Number:
49.	Compressibility Factor, *Z*:	56.	Vendor Calculations Required: Yes [ ] No [ ]
50.	Ratio of Specific Heats:		

Note: Indicate items to be filled in by the manufacturer with an asterisk (*).

Figure D.1—Spring-loaded PRV Specification Sheet

PILOT-OPERATED PRESSURE RELIEF VALVE SPECIFICATION SHEET		Sheet No. Page of	
		Requisition No.	
		Job No.	
		Date	
		Revised	
		By	
	GENERAL		BASIS OF SELECTION
1.	Item Number:	5.	Code: ASME VIII [ ] Stamp Required: Yes [ ] No [ ]
2.	Tag Number:		Other [ ] Specify:
3.	Service, Line, or Equipment Number:	6.	Comply with API 526: Yes [ ] No [ ]
4.	Number Required:	7.	Fire [ ] Other [ ] Specify:
		8.	Rupture Disk: Yes [ ] No [ ]
	VALVE DESIGN		MATERIALS, MAIN VALVE
9.	Design Type: Piston [ ] Diaphragm [ ] Bellows [ ]	20.	Body:
10.	Number of Pilots:	21.	Seat (Nozzle): Piston:
11.	Pilot Type: Flowing [ ] Nonflowing [ ]	22.	Resilient Seat: Seals:
12.	Pilot Action: Pop [ ] Modulating [ ]	23.	Piston Seal:
13.	Pilot Sense: Internal [ ] Remote [ ]	24.	Piston Liner/Guide:
14.	Seat Type: Metal-to-Metal [ ] Resilient [ ]	25.	Diaphragm/Bellows:
15.	Seat Tightness: API 527 [ ]		
	Other [ ] Specify:		MATERIALS, PILOT
16.	Pilot Vent: Atmosphere [ ] Outlet [ ]	26.	Body/Bonnet:
	Other [ ] Specify:	27.	Internals:
		28.	Seat: Seals:
		29.	Diaphragm:
	CONNECTIONS	30.	Tubing/Fittings:
17.	Inlet Size: Rating: Facing:	31.	Filter Body: Cartridge:
18.	Outlet Size: Rating: Facing:	32.	Spring:
19.	Other [ ] Specify:	33.	Comply with NACE: Yes [ ] No [ ]
		34.	Other [ ] Specify:
	SERVICE CONDITIONS		ACCESSORIES
43.	Fluid and State:	35.	External Filter: Yes [ ] No [ ]
44.	Required Capacity per Valve and Units:	36.	Lifting Lever: Plain [ ] Packed [ ] None [ ]
45.	Mass Flux and Basis:	37.	Field Test Connection: Yes [ ] No [ ]
46.	Molecular Weight or Specific Gravity:	38.	Field Test Indicator: Yes [ ] No [ ]
47.	Viscosity at Flowing Temperature and Units:	39.	Backflow Preventer: Yes [ ] No [ ]
48.	Operating Pressure and Units:	40.	Manual Blowdown Valve: Yes [ ] No [ ]
49.	Set Pressure and Units:	41.	Test Gauge: Yes [ ] No [ ]
50.	Blowdown: Standard [ ] Other [ ]	42.	Other [ ] Specify:
51.	Latent Heat of Vaporization and Units:		
52.	Operating Temperature and Units:		SIZING AND SELECTION
53.	Relieving Temperature and Units:		
54.	Built-up Back Pressure and Units:	60.	Calculated Orifice Area (in square inches):
55.	Superimposed Back Pressure and Units:	61.	Selected Effective Orifice Area (in square inches):
56.	Cold Differential Test Pressure and Units:	62.	Orifice Designation (letter):
57.	Allowable Overpressure in Percent or Units:	63.	Manufacturer:
28.	Compressibility Factor, Z :	64.	Model Number:
59.	Ratio of Specific Heats:	65.	Vendor Calculations Required: Yes [ ] No [ ]

Note: Indicate items to be filled in by the manufacturer with an asterisk (*).

Figure D.2—Pilot-operated PRV Specification Sheet

Annex E
(informative)

Capacity Evaluation of Rupture Disk and Piping System 100 % Vapor Flow and Constant Pipe Diameter

E.1 General

E.1.1 The following method can be used to estimate the vapor capacity of a rupture disk/piping system of constant diameter. The method is based on compressible pipe flow equations contained in Crane Technical Paper 410 [17] and the application of standard resistance factors (K values) from API 521.

E.1.2 The method assumes that C_p/C_v is equal to 1.4. This assumption provides conservative results. The method can be applied to a piping system with varying diameters by treating each section of constant diameter separately.

E.1.3 The method presented in Crane 410 is based on graphical evaluation of several parameters. Curve fits of the graphical data are also presented below to allow direct solution without the graphical data. The use of the curve fitting equations introduces negligible error relative to the accuracy of the K factors.

E.2 Example Problem

Figure E.1 shows the arrangement of the vessel and rupture disk/piping system for the example problem.

a) Step 1—Determine required information

MAWP = 100 psig;

P_1 = relieving pressure = 110 % = 124.7 psia;

T_1 = relieving temperature = 200 °F + 460 °F = 660 °R;

Z_1 = relieving compressibility = 1.0;

M = molecular weight = 20;

P_2 = backpressure = 14.7 psia.

b) Step 2—Determine overall piping resistance factor, K, from Table E.1.

Table E.1—Determination of Overall Piping Resistance Factor, K

Description	K Value	Source of K Value Data
Sharp-edged Entrance	0.50	Crane [17], Page A29
Rupture Disk	1.50	Consult the rupture disk manufacturer
15 ft NPS-3 Schedule 40 Pipe	1.04	$K = \frac{fL}{D}$ f = 0.0178 L = 15 ft $D = \frac{3.068}{12}$ = 0.2557 ft
Sudden Expansion (Exit Loss)	1.00	API 521, Table 12 (See Note)
Total System K	4.04	

Note:
The exit loss coefficient should be neglected when discharging compressible fluids into a large reservoir such as the atmosphere if the pressure drop equation accounts for kinetic energy changes (e.g., the Isothermal method in API 521 clause 5.5.5 [14]). The User is cautioned that some piping pressure drop calculation methods do not account for kinetic energy in which case the exit loss should be included in the pressure drop equation.

c) Step 3—Determine Y_{sonic} and $\frac{dP_{\text{sonic}}}{P_1}$ based on total system K.

This step is based on the Crane 410 Chart A-22 Method [17] for obtaining Y_{sonic} and $\frac{dP_{\text{sonic}}}{P_1}$. From the chart and table on A-22, k (C_p/C_v) = 1.4.

$$Y_{\text{sonic}} = 0.653$$

$$\frac{dP_{\text{sonic}}}{P_1} = 0.70$$

As an alternate to the chart method, a curve fit of Crane 410, Chart A-22, for obtaining Y_{sonic} and $\frac{dP_{\text{sonic}}}{P_1}$ has been provided as Equation (E.1) through Equation (E.4).

For $\frac{dP_{\text{sonic}}}{P_1}$:

$$\text{If } 1.2 < K \le 10, \text{ then } \frac{dP_{\text{sonic}}}{P_1} = 0.1107\ln(K) + 0.5352 \tag{E.1}$$

If $10 < K \leq 100$, then $\frac{dP_{\text{sonic}}}{P_1} = 0.0609\ln(K) + 0.6513$ (E.2)

For Y_{sonic}:

If $1.2 < K \leq 20$, then $Y_{\text{sonic}} = 0.0434\ \ln(K) + 0.5889$ (E.3)

If $20 < K \leq 100$, then $Y_{\text{sonic}} = 0.710$ (E.4)

Based on K = 4.04:

$$\frac{dP_{\text{sonic}}}{P_1} = 0.69$$

$$Y_{\text{sonic}} = 0.65$$

d) Step 4—Compare $\frac{dP_{\text{sonic}}}{P_1}$ to $\frac{dP_{\text{actual}}}{P_1}$

d) .

$$\frac{dP_{\text{actual}}}{P_1} = \frac{(124.7 - 14.7)}{124.7} = 0.88$$

Since $\frac{dP_{\text{sonic}}}{P_1} < \frac{dP_{\text{actual}}}{P_1}$, the flow will be sonic (critical).

Use Y_{sonic} and $\frac{dP_{\text{sonic}}}{P_1}$ and skip to Step 6 (if subsonic, proceed to Step 5).

e) Step 5—evaluate Y_{actual} (subsonic cases only):

Using the Crane 410 Chart A-22 Method to obtain Y_{actual}:

1) at $\frac{dP_{\text{actual}}}{P_1}$ and K determine Y_{actual} from Chart A-22;

2) use $\frac{dP_{\text{actual}}}{P_1}$ and Y_{actual} in Step 6.

Using the Curve Fit Method for obtaining Y_{actual}:

1) calculate Y_{actual} from the Equation (E.5);

$$Y_{\text{actual}} = 1 - \frac{(1 - Y_{\text{sonic}})}{dP_{\text{sonic}}/P_1}\left(\frac{dP_{\text{actual}}}{P_1}\right) \tag{E.5}$$

2) Use $\frac{dP_{\text{actual}}}{P_1}$ and Y_{actual} in Step 6 in place of $\frac{dP_{\text{sonic}}}{P_1}$ and Y_{sonic}.

f) Step 6—Calculate capacity based on Crane 410, Equation (3-20):

$$W = 0.9\left(1891 \times Y \times d^2 \sqrt{\frac{dP}{K \times V_1}}\right) \tag{E.6}$$

g) Step 7—Using the Chart Method values and Equation (E.6):

1) $Y = Y_{\text{sonic}} = 0.65$;

2) d = pipe ID (in.) = 3.068 in.;

3) $dP = \left(\frac{dP_{\text{sonic}}}{P_1}\right)(P_1) = 87.3 \text{ psi}$;

4) K = overall resistance = 4.04;

5) V_1 = vapor specific volume = 2.84 ft^3/lb (Obtained using ideal gas law and compressibility).

$$W = 0.9\left(1891 \times 0.65 \times 3.068^2 \sqrt{\frac{87.3}{4.04 \times 2.84}}\right) = 28{,}700 \text{ lb/hr}.$$

h) Step 8—Using the Curve Fit Method values from Figure E.2 and Equation (E.6):

1) $Y = Y_{\text{sonic}} = 0.65$;

2) d = Pipe ID (in.) = 3.068 in.;

3) $dP = (dP_{\text{sonic}}/P_1)(P_1) = 86.0$ psi;

4) K = Overall resistance = 4.04;

5) V_1 = vapor specific volume = 2.84 ft^3/lb (obtained using ideal gas law and compressibility).

$$W = 0.9\left(1891 \times 0.65 \times 3.068^2 \sqrt{\frac{86.0}{4.04 \times 2.84}}\right) = 28{,}508 \text{ lb/hr}$$

Figure E.1—Pressure Relief System for Example Problem

Figure E.2—Curve Fit for *Cp/Cv* = 1.4 (Crane 410, Chart A-22)

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