# Technical Seminar Manual

## Contents



 

# Section I

## Terminology

Reference on definitions is API RP 520, Part I, Sixth Edition (March 1993).

## 1.0 Pressure Relief Devices

A. **Pressure Relief Device:** A device actuated by inlet static pressure and designed to open during an emergency or abnormal conditions to prevent a rise of internal fluid pressure in excess of a specified value. The device may also 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.

B. **Spring Loaded Pressure Relief Valve:** A pressure relief device designed to automatically reclose and prevent the further flow of fluid.

C. **Relief Valve:** A 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.

D. **Safety Valve:** A 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.

E. **Safety Relief Valve:** A pressure relief valve that may be used as either a safety or relief valve, depending on the application.

F. **Conventional Pressure Relief Valve:** A spring-loaded pressure relief valve whose performance characteristics are directly affected by changes in the back pressure on the valve.

G. **Balanced Pressure Relief Valve:** A spring-loaded pressure relief valve that incorporates a means for minimizing the effect of back pressure on the performance characteristics.

H. **Pilot Operated Pressure Relief Valve:** A pressure relief valve in which the main valve is combined with and controlled by an auxiliary pressure relief valve.

I. **Rupture Disc:** A non-reclosing differential pressure relief device actuated by inlet static pressure and designed to function by bursting the pressure-containing rupture disc. A rupture disc device includes a rupture disc and a rupture disc holder.

## 2.0 Dimensional Characteristics of Pressure Relief Devices

A. **Actual Discharge Area:** The measured minimum net area that determines the flow through a valve.

B. **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.

C. **Equivalent Flow Area:** A computed area of a pressure relief valve, based on recognized flow formulas, equal to the effective discharge area.

D. **Nozzle Area:** The cross-sectional flow area of a nozzle at the minimum nozzle diameter.

E. **Huddling Chamber:** An annular pressure chamber in a pressure relief valve located beyond the seat for the purpose of generating a rapid opening.

F. **Inlet Size:** The nominal pipe size (NPS) of the valve at the inlet connection, unless otherwise designated.

G. **Outlet Size:** The nominal pipe size (NPS) of the valve at the discharge connection, unless otherwise designated.

H. **Lift:** The actual travel of the disc away from the closed position when a valve is relieving.

## 3.0 Operational Characteristics System Pressures

A. **Maximum Operating Pressure:** The maximum pressure expected during system operation.

B. **Maximum Allowable Working Pressure (MAWP):** The maximum gauge pressure permissible at the top of a completed vessel in its operating position for a designated temperature. The pressure is based on calculations for each element in a vessel using nominal thicknesses, exclusive of additional metal thicknesses 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.

C. **Design Gauge Pressure:** The most severe conditions of coincident temperature and pressure expected during operation. 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.

D. **Accumulation:** The pressure increase over the MAWP of the vessel during discharge through the pressure relief device, expressed in pressure units or as a %. Maximum allowable accumulations are established by applicable codes for operating and fire contingencies.

E. **Overpressure:** The pressure increase over the set pressure of the relieving device, expressed in pressure units or as a %. It is the same as accumulation when the relieving device is set at the maximum allowable working pressure of the vessel and there are no inlet pipe losses to the relieving device.

 

# Section I

**F. Rated Relieving Capacity:** That portion of the measured relieving capacity permitted by the applicable code or regulation to be used as a basis for the application of a pressure relief device.

**G. Stamped Capacity:** The rated relieving capacity that appears on the device nameplate. The stamped capacity is based on the set pressure or burst pressure plus the allowable overpressure for compressible fluids and the differential pressure for incompressible fluids.

## Device Pressures

**H. Set Pressure:** The inlet gauge pressure at which the pressure relief valve is set to open under service conditions.

**I. Cold Differential Test Pressure:** The pressure at which the pressure relief valve is adjusted to open on the test stand. The cold differential test pressure includes corrections for the service conditions of back pressure or temperature or both.

**J. Back Pressure:** The pressure that exists at the outlet of a pressure relief device as a result of the pressure in the discharge system. It is the sum of the superimposed and built-up back pressures.

**K. Built-Up Back Pressure:** The increase in pressure in the discharge header that develops as a result of flow after the pressure relief device opens.

**L. Superimposed Back Pressure:** The static pressure that exists at the outlet of a pressure relief device at the time the device is required to operate. It is the result of pressure in the discharge system coming from other sources and may be constant or variable.

**M. 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.

**N. 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.

**O. 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.

**P. Simmer:** The audible or visible escape of compressible fluid between the seat and disc at an inlet static pressure below the set pressure and at no measurable capacity.

**Q. Leak-Test Pressure:** The specified inlet static pressure at which a seat leak test is performed.

**R. Relieving Conditions:** The inlet pressure and temperature of a pressure relief device at a specific overpressure. The relieving pressure is equal to the valve set pressure (or rupture disc burst pressure) plus the overpressure. (The temperature of the flowing fluid at relieving conditions may be higher or lower than the operating temperature).

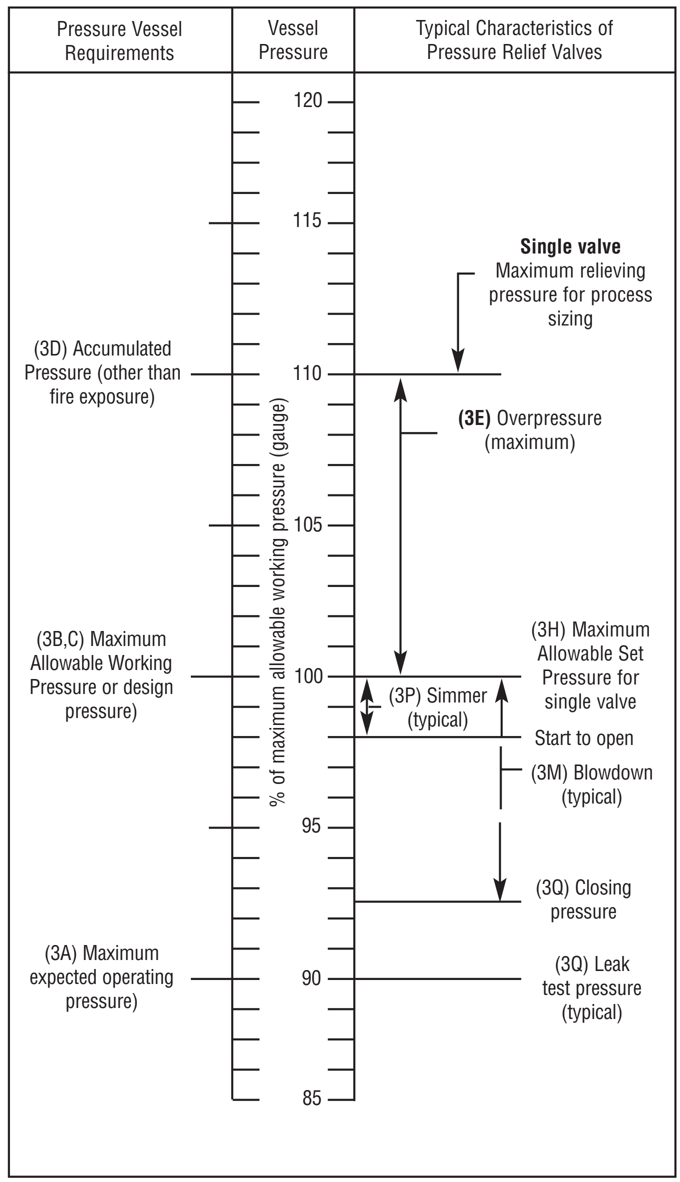


**Figure 1-1. Terminology**
**(Examples of terms 3A, 3B, 3C, 3D, 3E, 3H, 3M, 3O, 3Q)**

 

# Section II

## 2.0 Pressure Relief Device Design

Section 2 describes pressure relief device standards, types, and operation and performance. This information is provided to assist you in efficiently specifying, using, and servicing pressure relief valves.

## 2.1 Standards and Codes

The following documents govern the design of PRVs:

– American Petroleum Institute (API) Standard 526, Flanged Steel Safety Relief Valves
– American Society of Mechanical Engineers (ASME) Section VIII, Division I, Pressure Vessel Code, Paragraphs UG-125 through UG-136.
– ASME Section I, Boiler Code, paragraphs PG 67.1-PG.
– ASME Section III for Nuclear Pressure Vessels.
– ASME Section IV for Hot Water Boilers.

API 526 lists inlet/outlet flange sizes and ratings for different set pressure ranges, orifice sizes, materials and center to face dimensional standards. API 526 is a standard, not a code. Therefore, compliance is not mandatory to meet jurisdictional requirements.

ASME codes specify the following criteria:

– performance requirements
– material requirements
– set pressure spring design
– acceptable failure modes
– nameplate data
– capacity certification procedures

ASME code-compliance is mandatory in applicable jurisdictional areas. A few states have no law for unfired pressure vessels. However, for insurance purposes, most users require code-stamped valves. All states require compliance with the Boiler Codes.

## 2.2 Design Considerations

A PRV is a safety device, intended to protect life and property if all other safety measures fail. Its main purpose then, and the function it must successfully accomplish when it operates, is to prevent pressure in the system being protected from increasing beyond safe design limits. A secondary intent of a PRV is to minimize damage to other system components due to operation of the PRV itself. From a user's perspective then, these design features should be considered in a valve design:

– leakage (if any) at system operating pressure is within acceptable standards of performance
– opens at specified set pressure, within tolerance
– relieves the process product in a controlled manner
– closes at specified reseat pressure
– easy to maintain, adjust, and verify settings
– cost effective maintenance with minimal downtime and spare parts investment

## 2.3 Valve Types

The two general types of PRVs, direct-acting and pilot operated, are explained in the Sections 2.3.1 and 2.3.2 respectively. PRV operation is detailed in Section 2.4.

### 2.3.1 Direct-Acting PRVs

The oldest and most commonly used type of PRV is the direct-acting type. They are designated as direct acting because the force element keeping the valve closed is either a weight or a spring or a combination of both. The process to be relieved acts directly on a seat pallet or disc, which is held closed by the weight of a spring opposing the lifting force of the process pressure. When the lifting forces and opposing forces are equal, the valve is on the threshold of opening.

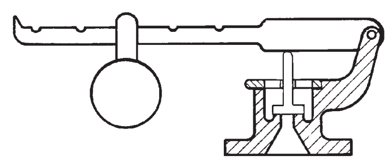

**Figure 2-1.**
**Early Design PRV (Circa 1900)**

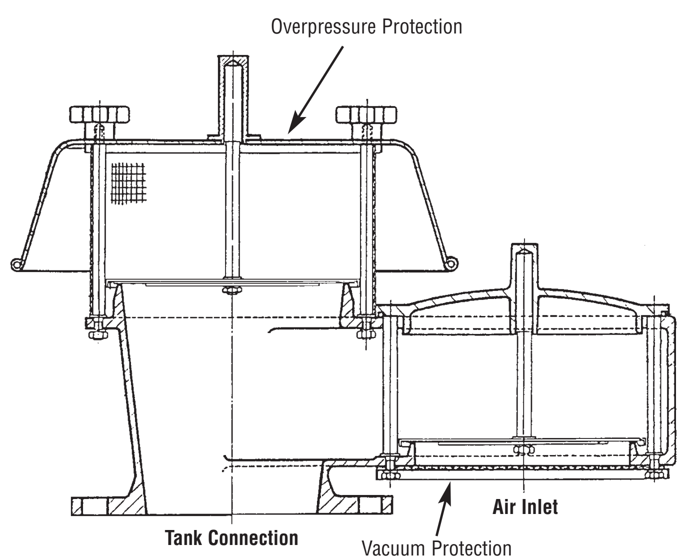


**Figure 2-2. Weight-Loaded Vacuum PRV**

Note

1. The ASME code applies only to pressure relief valves set at or above 15 psig.

 

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There are two kinds of direct-acting type PRVs, weight-loaded and spring-loaded.

### Weight-Loaded

The direct-acting, weight-loaded PRV was the first type of PRV to be used. They were employed to protect steam boilers from overpressure. The design of these early PRVs (see Figure 2-1) made them easy to adjust to a higher relieving pressure by adding more weight. Unfortunately, such ease of adjustment frequently resulted in boiler explosions and loss of life.

The current, weight-loaded, PRV design (see Figure 2-2) is commonly referred to as a weighted pallet valve, breather vent, conservation vent, or just vent.

### Spring-Loaded

The direct-acting, spring-loaded PRV (see Figure 2-3) is commonly referred to as a conventional PRV. A variation of the conventional PRV is the balanced PRV (see Figure 2-4). The balanced PRV is similar to the conventional PRV except that it has an additional part, either a metal bellows assembly, around the spindle/disc holder to balance the valve against the effect of back pressure, or a balanced spindle design. For higher temperature application, an open yoke design (see Figure 2-5) exposes the spring to allow ambient cooling. Spring loaded PRVs operate in pressure ranges from 5-6000 psig safely and temperatures from -400°F to +1000°F.

## 2.3.2 Pilot Operated PRVs

Pilot operated PRVs are not as commonly used as direct acting PRVs, but they have been applied in a wide variety of applications for 53 years. The primary difference between a pilot operated PRV and a direct acting PRV is that process pressure is used to keep the valve closed instead of a spring or weight. A pilot is used to sense process pressure and to pressurize or vent the dome pressure chamber which controls the valve opening or closing.

There are two general styles of pilot operated PRVs, piston and diaphragm. Both valve types consist of a main valve and a pilot. The pilot controls the pressure on the top side of the main valve unbalanced moving member. A resilient seat is normally attached to the lower end of this member, although some pilot valve designs for high temperature incorporate metal seating surfaces only.

– At pressures below set, the pressure on opposite sides of the moving member is equal.

– When set pressure is reached, the pilot opens, depressurizes the cavity on the top side and the unbalanced moving member moves upward, causing the main valve to relieve.

– When the process pressure decreases to a predetermined pressure, the pilot closes, the cavity above the piston is repressurized, and the main valve closes.

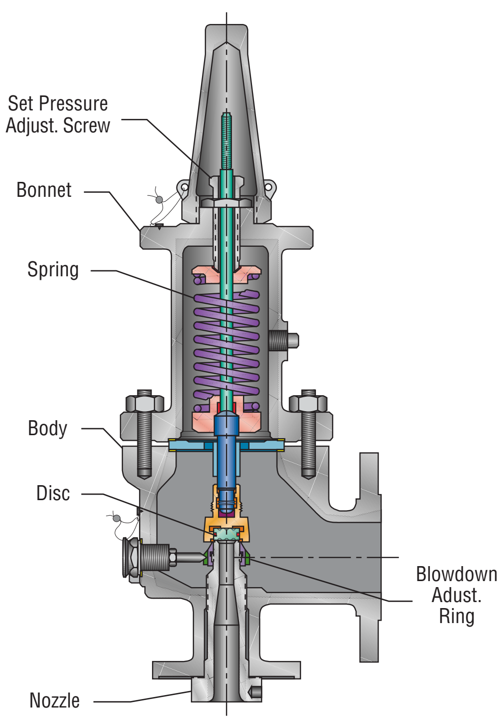


**Figure 2-3. Conventional Spring-Loaded PRV**

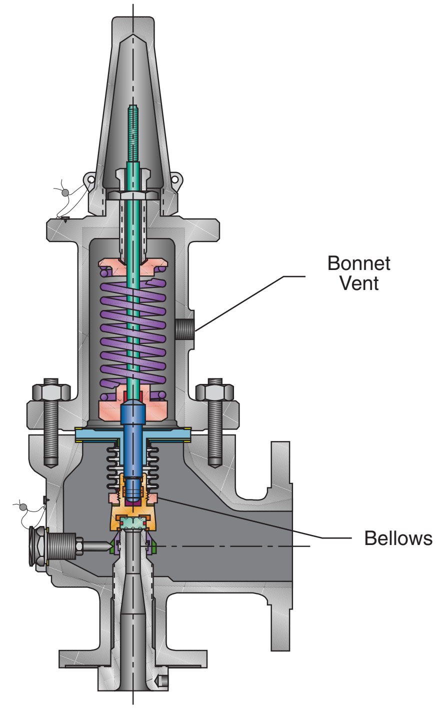


**Figure 2-4. Balanced Bellows, Spring-Loaded PRV**

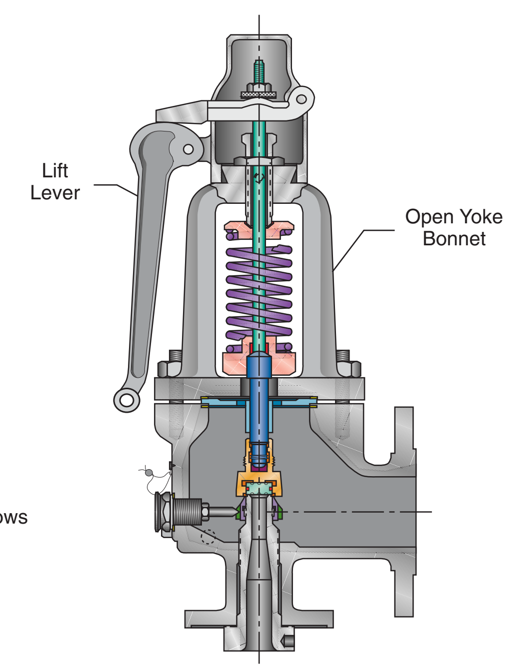


**Figure 2-5. Open Yoke Conventional PRV**

 

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A piston type pilot operated PRV (see Figure 2-6) uses a piston for the unbalanced moving member. A sliding O-ring or spring-loaded plastic seal is used to obtain a pressure seal for the dome cavity. The piston type valve has been used for pressures 5 psig to 10,000 psig, and could be applied to even higher pressures.

### Diaphragm Type, Pilot Operated PRV

The diaphragm type, pilot operated PRV (see Figure 2-7) is similar to the piston type except a flexible diaphragm is used to obtain a pressure seal for the dome volume instead of a piston and sliding piston seal. This is done to eliminate sliding friction and permit valve operation at much lower pressures than would be possible with a sliding seal. The diaphragm type valve can be used for pressures 3-inch water column (0.108 psig) to 50 psig.

### Metal-Seated Type, Pilot Operated PRV

The metal-seated type, pilot operated valve is similar to usual pilot operated valves except for the orientation of the piston. The metal-seated type piston is reversed with respect to the inlet and outlet flanges. The piston and nozzle face the outlet flange instead of the inlet flange. A pilot senses inlet pressure and is used to pressurize or vent the dome behind the piston. Inlet pressure also surrounds the piston pressurizing the body cavity. Two metal piston rings provide the seal between the piston and the liner.

– At pressures below set, inlet pressure in the piston cavity forces the piston against the nozzle providing a tight seal.

– When set pressure is reached, the pilot opens the unloader which rapidly depressurizes the piston cavity, and pressure in the body forces the piston away from the nozzle causing the main valve to relieve.

– When the process pressure decreases, the pilot pressurizes and closes the unloader, the dome is repressurized, and the main valve closes.

The metal-to-metal seated pilot operated valve (see Figure 2-8) was developed to be used for process ladings or temperatures where soft-seated pilot operated valves are not appropriate. A primary service is high pressure steam.

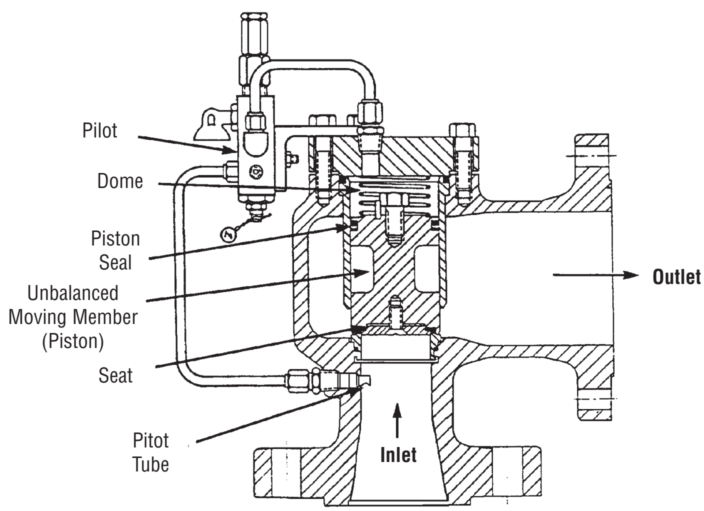


**Figure 2-6. Piston Type Pilot Operated PRV**

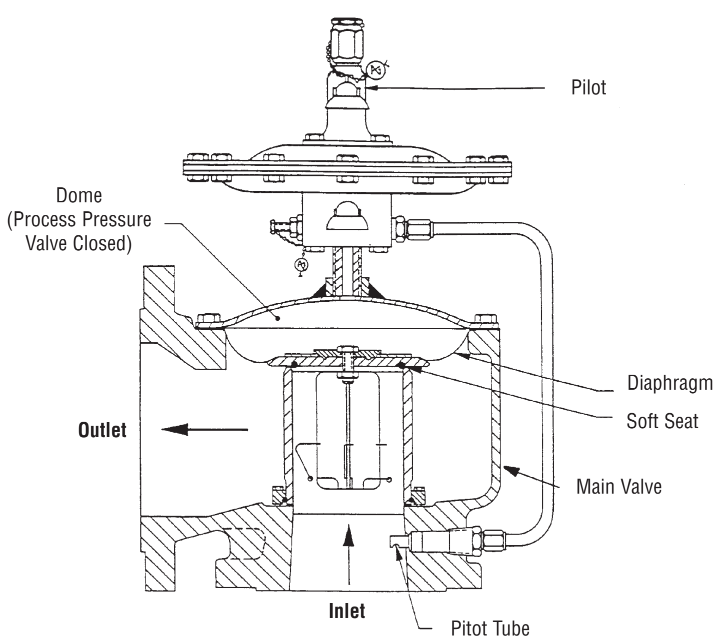


**Figure 2-7. Diaphragm Type Pilot Operated PRV**

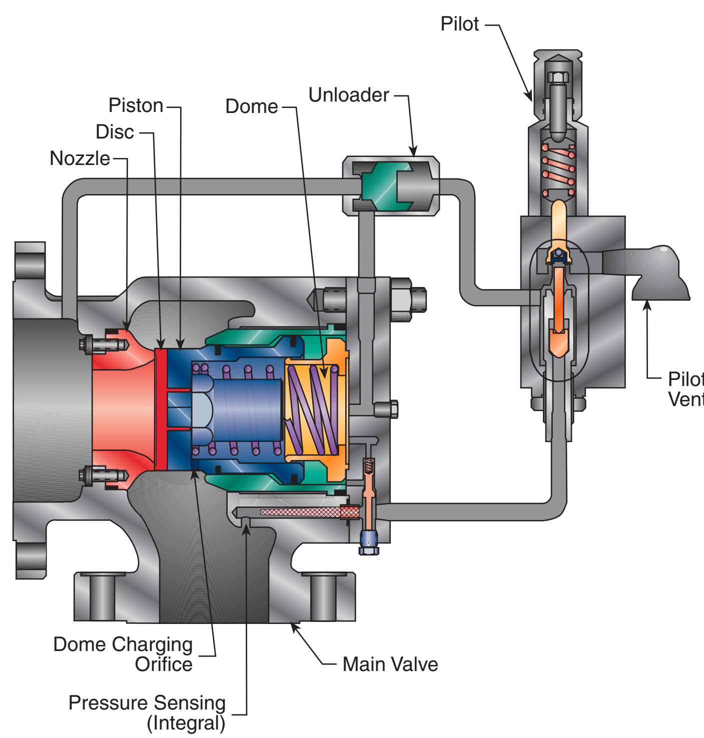


**Figure 2-8. Metal Seated Pilot Operated PRV**

# Section II

## 2.4 Valve Operation and Performance

All pressure relief valves are designed to provide a certain relieving capacity at a specified pressure. However, how this capacity is achieved from the valve closed to the valve open position may be quite varied. A discussion of the different valve types will show this.

### 2.4.1 Weighted Pallet PRV (Vent) Operation

The weighted pallet PRV is the simplest, least complex type of PRV. It is a direct acting valve because the weight of the seat assembly or pallet keeps the valve closed until the pressure acting on the underside equals this weight. Figure 2-9 is an illustration of such a weighted pallet PRV.

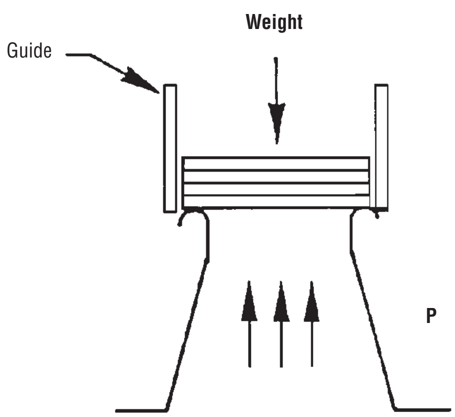


**Figure 2-9. Weighted Pallet PRV**

Because of the weight required to keep these valves closed, they are normally designed for set pressures less than 2 psig and, therefore, are not ASME Code valves. For example, a 12-inch valve, one commonly used on large storage tanks, would typically have a nozzle area of approximately 90-inch$^2$. To obtain a set pressure of 1.0 psig, a 90-pound weight would be required. The weights required for much higher set pressures become prohibitive and damaging should oscillation of the seat plate occur at valve opening.

The phenomenon of oscillation is similar to that which occurs with the cover on a pot of boiling water. The cover will oscillate on the pot, permitting water vapor to escape. When a weighted pallet valve with a heavy seat plate does this, the valve pallet guidance and seating surface can be damaged. There is little that can be done to prevent such oscillations, and nearly all weighted pallet valves exhibit this characteristic. As a rule, large valves like those described above are usually limited in set pressure to 0.5 psig because of the excessive weight required to obtain set pressure.

Another characteristic of such valves that limit their application is the amount of overpressure required at the valve inlet to obtain full lift and, therefore, rated capacity. For valves of this type, a required overpressure of 100% is not uncommon. For example, rated capacity of a valve set at 1.0 psig might not occur until the pressure in the vessel accumulated to 2.0 psig. Figure 2-10 shows the capacity characteristic of a typical weighted pallet valve. Set pressure is defined where the first measurable flow occurs through the valve.

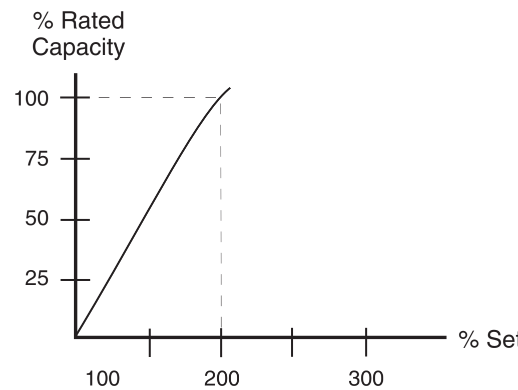


**Figure 2-10. Typical Weighted Pallet PRV Capacity - Set Pressure Characteristics**

The large pressure difference between where the valve opens and where rated capacity is achieved requires the tank to be built stronger or the maximum allowable operating pressure to be reduced. Either way, the efficient use of the tank is compromised.

Rated capacity is usually controlled by the nozzle diameter or bore in the valve. The minimum lift of the seat plate to achieve rated capacity must, therefore, be that lift where the annular curtain area around the periphery of the nozzle equals or exceeds the nozzle area. Expressed mathematically, this would be as follows:

$$\text{(Curtain Area)} \quad \pi DL = \frac{\pi D^2}{4} \quad \text{(Nozzle Area)}$$

Where: D = Nozzle bore diameter

L = Lift of seat plate

Dividing both sides by $\pi D$ gives:

$$L = \frac{\pi D^2}{4}$$

Theoretically, rated capacity is achieved when the lift equals 25% of the nozzle diameter. In actual practice, a lift of 40% is usually required for pressures below 15 psig because of flow losses.

 

## Section II

### 2.4.2 Conventional (Direct Spring Operated) PRV Operation

In a weighted pallet valve, the pressure force required to lift the pallet is only the weight of the pallet. This weight remains constant, regardless of the lift. In a spring loaded valve, the pressure force required to lift the seat disc is the pre-load of the spring, which is equal to the pressure under the disc times the seat sealing area, plus the force required to compress the spring as the valve opens. This compression force is equal to the spring rate times the lift of the seat disc, and must be generated during the allowable overpressure (see Figure 2-11).

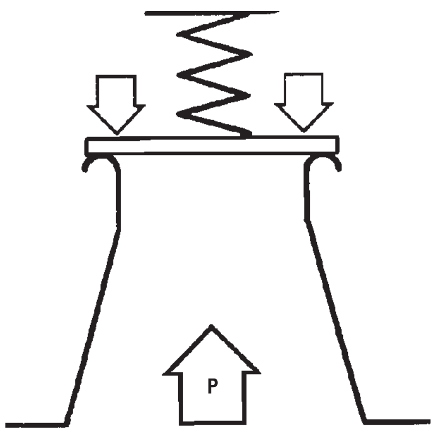


**Figure 2-11. Spring-Loaded PRV**

For ASME Section VIII pressure relief valves, the permissible overpressure to obtain full lift of the seat disc is normally 10%. For Section I PRVs, only 3% overpressure is allowed. This is a difficult requirement considering a pressure increase of 0 to 95% is available to balance the spring preload and only a 15% (95 to 110%) pressure increase is available to achieve full lift. A design feature commonly used to further compress the spring and achieve lift is the addition of a “skirt” to the seat disc as shown in Figure 2-12. The skirt redirects the flow downward as it discharges through the nozzle, resulting in a change of momentum. The gas or vapor also expands and acts over a larger area. Both the momentum change and expansion significantly increase the force available to compress the spring. The angle of the skirt can vary, but for most valves it is around 45°. The larger the angle, the greater the lifting force. A large lifting force, however, can prevent the valve from closing within the ASME specified 7% blowdown. This ASME requirement is only for capacity certification by the PRV manufacturers and does not apply to production valves.

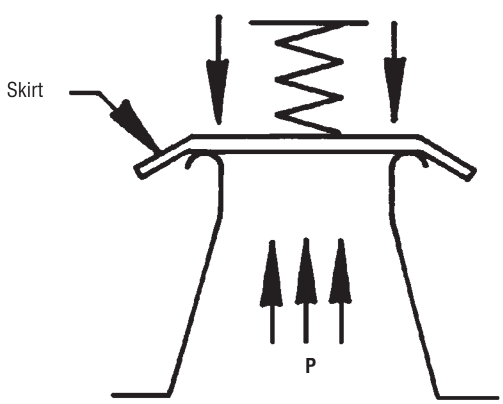


**Figure 2-12. Spring-Loaded PRV with Skirt on Seat Disc**

In order to achieve a significant lifting force without an extremely long blowdown, a ring is threaded around the valve nozzle and positioned to form a huddling chamber with the disc skirt (see Figure 2-13). Although the ring shown is commonly called a blowdown ring, its function is also very important for controlling the valve opening.

Pressure is generated in the huddling chamber when gas or vapor flows past the seat. The pressure in the huddling chamber, acting over a larger area than the seat sealing area, increases creating an instantaneous amplification of the upward force, and the seat disc rapidly lifts off the nozzle. This initial lift of the seat disc is enough to establish 60 - 75% full rated flow, driving the seat disc up to the change in momentum and the expansion of the gas can sustain lift.

When the blowdown ring is adjusted up, the forces required to lift the seat disc off the nozzle occur at a pressure very close to set pressure. The reason for this is that the huddling chamber is restricted and gas flowing into the chamber quickly pressurizes it. However, with the ring in the up position, the blowdown is long because the pressure between the seat disc skirt and the ring remains high, preventing the seat disc from losing lift until the pressure under the disc reduces to a much lower value.

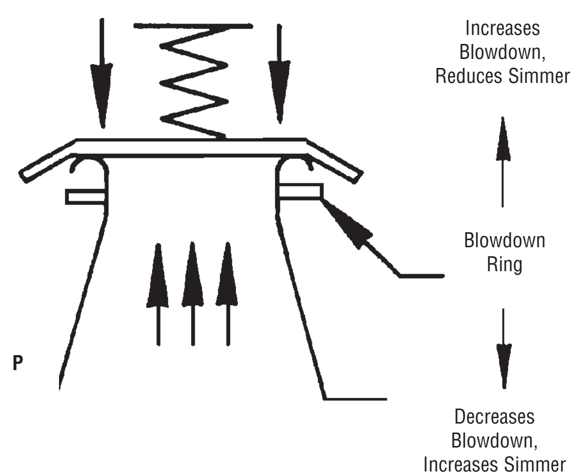


**Figure 2-13. Spring-Loaded PRV with Blowdown Ring**

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When the ring is adjusted down, the forces required to lift the seat disc off the nozzle do not occur until the pressure under the seat disc is considerably higher. The huddling chamber exit area is less restricted and considerably more gas must flow into the chamber to pressurize it. The blowdown with the ring in this position is short since the pressure between disc holder skirt and ring quickly decreases when the lift of the seat disc is decreased.

For protection of the working internals and safe disposal of the discharge through a valve, an enclosure or body encloses the nozzle and seat disc as shown in Figure 2-14. Body pressure which is generated during flow conditions must be controlled to ensure reliable and safe operations of the PRD, since it acts on the back side of the disc, in a direction that can prevent the disc from going into full lift. The pressure is additive to the spring load when the valve opens since built-up back pressure does not occur until then. If standard application installation recommendations are not adhered to, this pressure may prevent the valve from going into full lift, and may cause it to reclose prematurely and be unstable. Once closed, the flow stops, the back pressures diminishes, and the valve opens again, only to reclose. This type of opening and closing is called rapid-cycling or chatter. The only way to eliminate it is to reduce the built-up back pressure and inlet pressure loss, increase the lifting force, or change to another type of valve.

Figure 2-15 is an illustration of a typical, commercially-available, conventional, direct spring operated PRV.

## Seat Disc Lift

As noted earlier, the valve is on the threshold of opening when the upward force produced by the product of the process pressure (pounds per square inch) acting on the seat disc sealing area (square inch) equals the downward force of the spring. To obtain rated capacity, the seat disc must lift an amount equal to at least 30% of the nozzle bore diameter. The seat disc lift versus set pressure of a typical conventional valve is shown in Figure 2-16.

## Back Pressure

The balance of forces in a conventional valve is critical. Any change in pressure within the valve body downstream of the seat disc holder and huddling chamber can disturb the lifting forces. Figure 2-17 shows the relationship between back pressure and capacity of a typical conventional valve. Most manufacturers and both API RP 520, Part I, Section 2.2.4.1 and ASME Section VIII, Division 1, Appendix M-8 (c) recommend built-up back pressure for a conventional PRV not exceed 10% of the pressure at the valve inlet during relief. For a more detailed discussion of back pressure and its effect on pressure relief valve performance, refer to Gary Emerson's paper, "Handling Back Pressure On PRVs", included under the "Back Pressure" tab of this book.

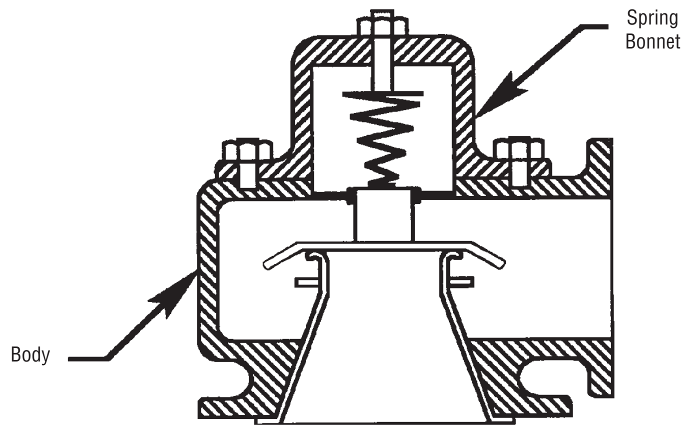



**Figure 2-14. Spring Loaded PRV with Body**

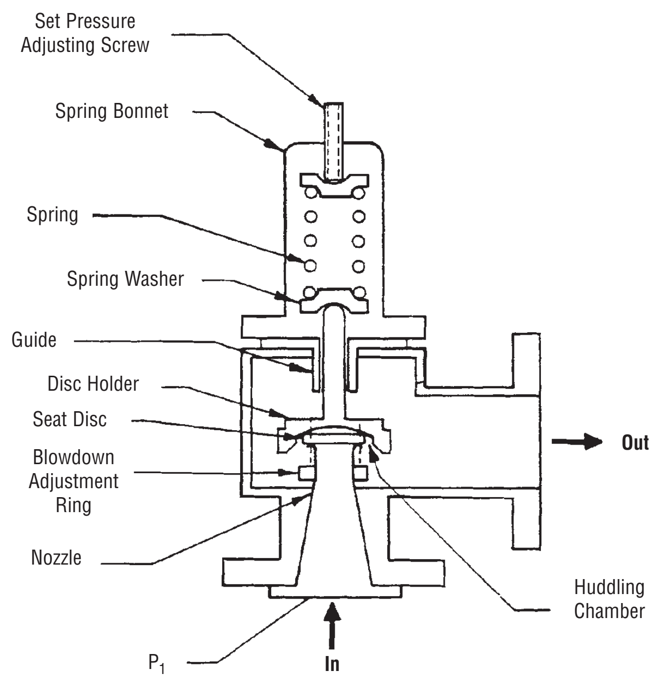



**Figure 2-15. Conventional Direct Spring Operated PRV**

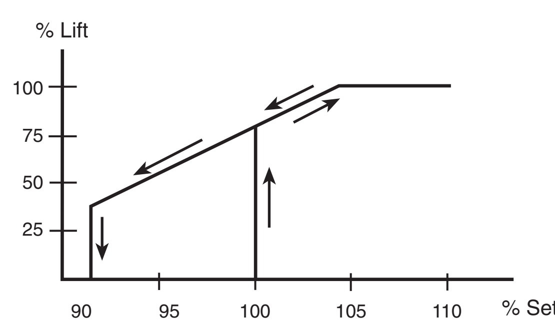
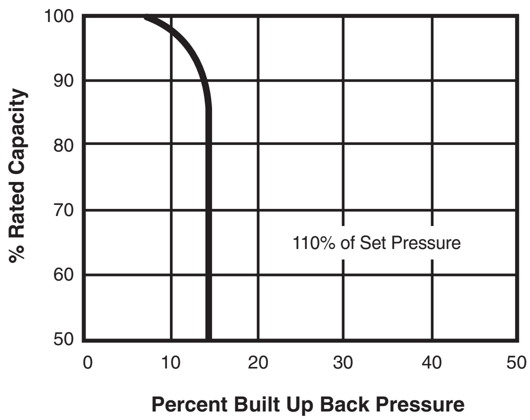



**Figure 2-16. Typical Conventional Valve Seat Disc Lift Characteristics**

# Section II




$$\frac{\text{Pressure at Valve Outlet, psig}}{\text{Pressure at Valve Inlet, psig}} \times 100$$

**Figure 2-17. Typical Back Pressure Characteristics of Conventional PRV**

Another important consideration of back pressure on a conventional valve is its effect on set pressure. Superimposed back pressure, that is, pressure that exists at the outlet before the PRV opens, will increase the set pressure on a one for one basis. For example, if the set pressure is 100 psig and a back pressure of 10 psig is superimposed on the valve outlet, the set pressure will increase to 110 psig.

Superimposed back pressure on a conventional valve produces a downward force on the seat disc that is additive to the spring force.

## Spring Requirements

The spring in all pressure relief valves must meet certain requirements to comply with the ASME code. One requirement is that the maximum compression of the spring be equal to or less than 80% of the nominal solid spring height. Another requirement is that springs have a reserve capacity sufficient to change the set pressure ±5% from the nameplate set. For example, if a valve is purchased with a set pressure of 300 psig, it should be capable of being reset in the field by the user to 285-315 psig without changing the spring or without degrading the valve performance.

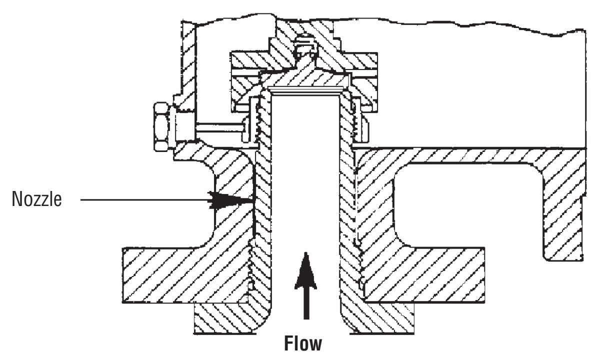
Care must be used in the design of springs for valves used where the process gas contains hydrogen sulfide. Hydrogen embrittlement of some hard metals can occur, resulting in fracture and failure of the part. CS and Series 300 SS spring materials are susceptible to such failures when exposed to hydrogen sulfide, if the material hardness exceeds Rockwell C-22. Inconel® or Monel® are good but expensive alternate materials. A less expensive alternate and one that works reasonably well is to coat the spring surface to shield it from contact with the hydrogen sulfide. However, NACE MR0175 (for sour service) does not recognize coatings as being acceptable to prevent stress corrosion. Another method which is often used is isolating the spring from the process environment. This may be accomplished by the use of a metal bellows.

## Construction Materials

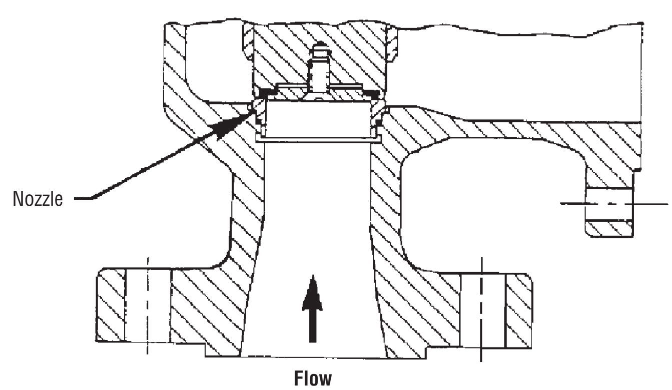
Other considerations for materials of construction are the ASME code requirements and the service conditions. To meet the requirements of the ASME Section I and VIII codes, the pressure-containing parts defined as the body, bonnet and yoke must be made of materials that are listed in Section II and Division 1 of Section VIII. For code requirements, the other parts need only be made of materials listed in ASTM specifications or materials controlled by a manufacturer specification.

## Valve Nozzle

Many valves are "full nozzle" designs where the nozzle prevents the lading fluid from contacting the body casting in the valve closed position. A semi-nozzle valve is one where the inlet side of the body is exposed to the lading fluid in the valve closed position. See Figure 2-18 for diagrams of full and semi-nozzle valves.




**Full Nozzle**




**Semi-Nozzle**

**Figure 2-18. Full and Semi-Nozzle Valve Design**

# Section II

## 2.4.3 Balanced (Direct Spring Operated) PRV Operation

A balanced bellows valve is similar to a conventional valve except the area downstream of the seat disc is enclosed within a protective pressure barrier to balance against back pressure. Figure 2-19 is an illustration of this type of valve.

The balanced valve was also designed to protect the disc holder and guide from the corrosive effects of the lading fluid when the valve opens and relieves. In the process of evaluating this type of design, it was observed that by enclosing an area the same as the nozzle/disc seating area, the set pressure would not be affected by back pressure. A sliding spindle seal can also be used to balance a spindle.

The term "balanced" means the set pressure of the valve is not affected by back pressure. Because the enclosed area on the back side of the seat disc is equal to the area on the process fluid side, back pressure is prevented from exerting down force to keep the seat disc closed. The lift characteristics of a balanced valve can still be affected by back pressure but to a much lesser degree, compared to a conventional valve. Figure 2-20 is a curve showing the change in lift with back pressure of a typical balanced spring valve. As the overpressure at the valve inlet increases, the loss in lift diminishes. As the inlet pressure increases over set, the valve will tolerate a higher back pressure before the lift begins to decrease.

### Design Considerations

An important design consideration for a balanced valve using a bellows is the collapse pressure rating of the bellows. The bellows must be strong enough to withstand the back pressure without collapsing, yet flexible enough not to affect the valve lift characteristics. The bellows must also be designed to resist flutter caused by turbulent flow during a relieving cycle or designed to be shielded from it. Such turbulence can cause premature failure due to metal fatigue. The bellows must also be corrosion resistant. Because of the thin metal used in bellows, pin holes due to corrosion can occur in a material considered suitable for the same service in a thicker gauge.

The spring bonnet on all balanced valves must be vented to atmosphere to ensure safe operation of the valve, in case a leak or failure occurs in the bellows or spindle seal. Any pressure accumulation within the bonnet will increase the set pressure by an equal amount. The valve then responds the same as a conventional valve subjected to back pressure, but with the bonnet closed, there is no way to detect the problem of the "balanced" valve becoming unbalanced due to a leaking bellows.

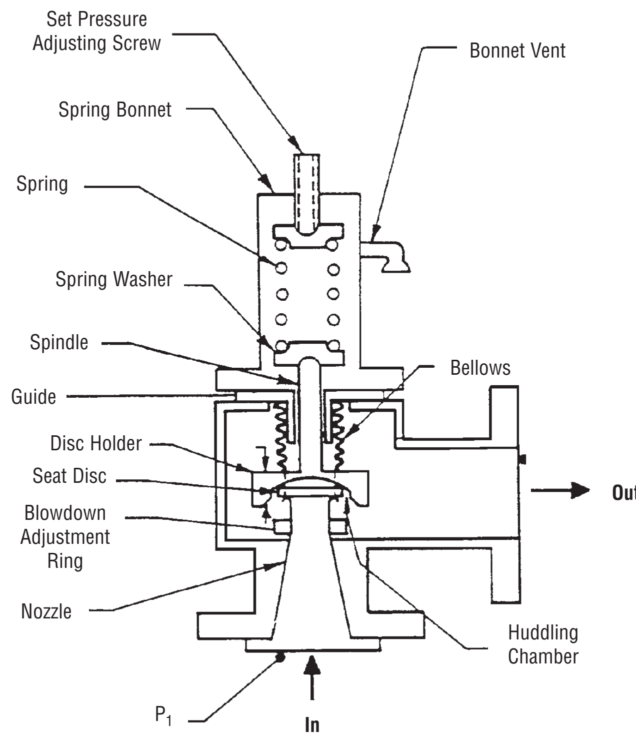



**Figure 2-19. Balanced Direct Spring PRV**

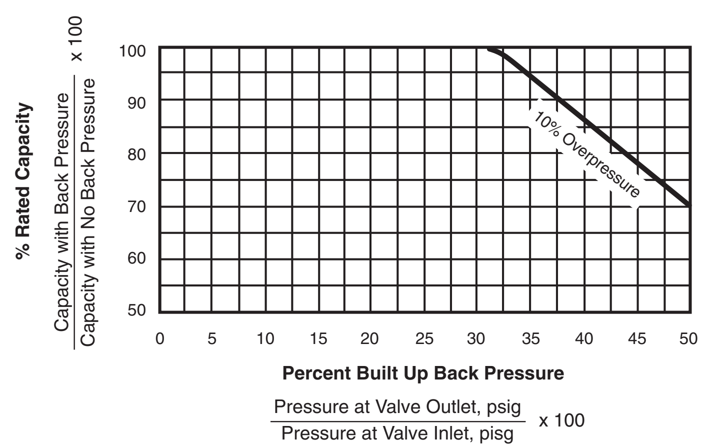



**Figure 2-20. Back Pressure Characteristic of a Balanced PRV**

Note

1. Inconel® and Monel® are registered trademarks of the International Nickel Company.

# Section II

## 2.4.4 Pilot Operated Valves

A piston type, pilot operated valve is illustrated in Figure 2-21.

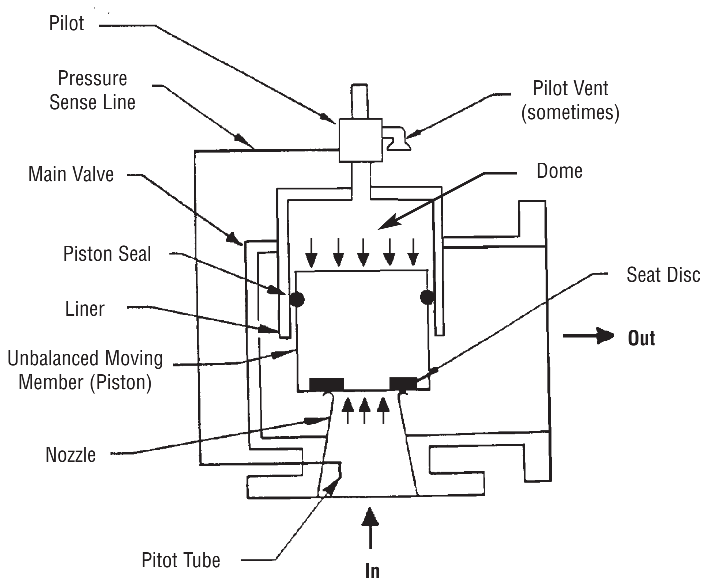



**Figure 2-21. Pilot Operated PRV**

For pilot operated valves, process pressure, instead of a spring, is used to keep the seat disc closed at pressures below set. This feature permits much higher set pressures with larger orifices than could be obtained with conventional or balanced valves. Larger spring forces do not exist and the physical spring size required in direct spring valves is unnecessary.

A pilot operated PRV design consists of a main valve and a pilot. The pilot controls the pressure on the top side of the unbalanced moving member, but it may also be a metal seat. A soft seat is usually attached to the opposite end of this member.

– At pressures below set, the pressure on opposite sides of the moving member is equal.

– When set pressure is reached, the pilot opens, depressurizes the volume on the top side of the piston, and the unbalanced moving member moves upward, opening the main valve to relieve pressure.

– When the process pressure decreases to the desired level, the pilot closes, the dome volume is repressurized, and the main valve closes.

### Unbalance of the Moving Member

The unbalance of the moving member usually ranges from 1.2:1 to 3.0:1. This unbalance ratio means the area of the dome side of the moving member is larger than the seat sealing area. The net force holding the seat closed is equal to the downward force minus the upward force.

Seating Force = Downward Force – Upward Force
= P1ASR – P1AS
= P1AS (R – 1)

Where: $P_1$ = System Pressure
AS = Seat Sealing Area
R = Ratio, Area Unbalance

For a valve with a seating area of 4 in$^2$, an area unbalance of 1.25 and a set pressure of 500 psig, the seat sealing force at 95% of set would be 475 pounds. For a direct spring valve, the seating force at 95% of set would be only approximately 20 - 30 pounds.

Seating Force = .95 (500 psig) (4-inch$^2$) (1.25 – 1)
Seating Force = 475 pounds

For the valve to open, the pilot must depressurize the dome to a pressure equal to 70% of the inlet pressure. When that occurs, the forces are in balance and the valve is on the threshold of opening. The piston will then move upward off the nozzle. When the valve closes, the reverse process occurs. The pilot closes, the dome is repressurized, and the piston closes against the nozzle.

The unbalance area ratio of the moving member is determined to a large extent by the pressure range for which the valve is designed. For low pressure, 0.10 psig to 15 psig, an area unbalance of 2.0:1 to 3.0:1 is common. In this pressure range the unbalanced member is usually a diaphragm and seat assembly. Because of the low pressures, the force available to hold the seat closed is small. Increasing the unbalance of the moving member increases these forces to ensure the seating member is held closed with sufficient force to obtain a tight seal.

# Section II

## Seat Disc Lift

The seat disc lift versus set pressure and overpressure at the valve inlet for a pilot operated valve is shown in Figure 2-22. Two curves are shown, one for a pop action pilot, and one for a modulating action pilot. For pop action, full lift occurs at set pressure and is maintained until reseat occurs. For modulating action, lift begins at set pressure and then proportionally increases with overpressure until full lift is obtained at some overpressure. Reseat occurs very close to set, with minimal blowdown.

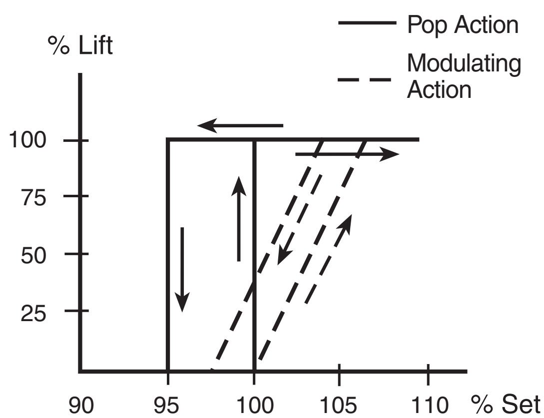



**Figure 2-22. Pilot Valve Seat Disc Lift Characteristics**

## Seats and Seals

Most pilot operated valves use elastomer or plastic seats and/or seals. The dome volume on top of the unbalanced moving member must be pressure-sealed from the downstream side of the valve. The easiest, most reliable and effective way to accomplish this sealing is with a sliding seal such as an O-ring, a spring-loaded plastic seal, or a flexible membrane such as a diaphragm. For low pressure valves, the flexible membrane is preferred because of its low resistance to movement. However, such a membrane is pressure limited to around 50 psig. Above those pressures, a sliding O-ring or spring-loaded plastic seal is much more durable and effective.

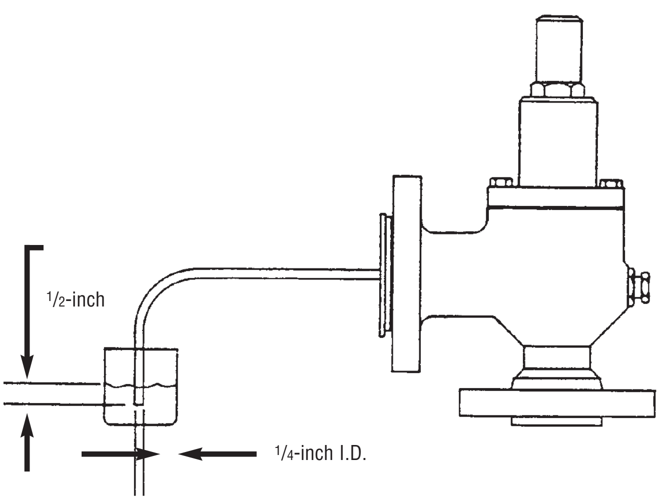



**Figure 2-23. Standard Bubble Tester per API 527**

Since the sliding seal is an elastomer or plastic, the seat sealing member is often of the same material. This configuration is commonly referred to as a “soft”-seated valve, as opposed to a “metal”-seated valve. A soft-seated valve is much easier to design for tight sealing. The method commonly used for measuring seat tightness is that given in API Standard 527, “Seat Tightness of Pressure Relief Valves”. A bubble tester, similar to that shown in Figure 2-23, is necessary for measuring seat tightness.

Seat tightness is specified in bubbles per minute. For soft-seated valves, a performance standard of zero bubbles per minute at 90% of set pressure is common. For pilot operated valves, this leakage rate applies to the pilot only. The main valve remains tight until set pressure is reached. For metal-seated valves, a leak rate of 20 to 100 bubbles per minute is allowed. Figure 2-24 shows the leakage rates permitted by API 527 and ASME Section VIII for both metal-seated and soft-seated PRVs.

| Orifice Size | Maximum Allowed Leakage Rate (Bubbles per Minute) 15 - 1000 | 1500 | 2000 | 2500 | 3000 | 4000 | 6000 |
|---|---|---|---|---|---|---|---|
| D, E, F | 40 | 60 | 80 | 100 | 100 | 100 | 100 |
| G & Larger | 20 | 30 | 40 | 50 | 60 | 80 | 100 |

**Figure 2-24. Pilot Valve Seat Disc Lift Characteristics**

A larger orifice valve usually has a lower leak rate than a smaller orifice one, even though the perimeter of the seal is larger. This is because the unit force per inch of circumference is directly proportional to the sealing diameter. Also a larger disc tends to be better self-aligning with the nozzle seating surface.

$$\text{Circumference} = \pi D$$

$$\text{Area} = \frac{\pi D^2}{4}$$

$$\text{Force} = PA$$

$$\text{Unit Force} = \frac{P\pi D^2}{4} \times \frac{1}{\pi D}$$

$$\text{Unit Force} = \frac{PD}{4}$$

# Section II

To improve the sealing characteristics of small orifice valves, the seat sealing area is sometimes made larger than the flow orifice. For soft seated valves, a softer seat material is also used. However, extremely soft materials usually cannot be used at pressures much above 300 - 500 psig. At higher pressures, distortion can also occur due to aerodynamic forces, causing blowout of a very soft seat. Harder seat materials, such as 90 durometer elastomers or Urethane are then used. Even with these limitations, soft seats are much more tolerant than metal seats to particulates in the process.

For service temperatures above 550°F, or for chemically harsh service where the soft seal material would be attacked, metal-seated valves must be used. With the availability of Teflon® and Kalrez®, a perfluorelastomer manufactured by DuPont®, or PEEK (polyetheretherketone), chemical compatibility is becoming much less of a limitation.

## Effect of Back Pressure

Because there are no heavy spring loads to overcome, lift of the seat disc in a pilot operated PRV is not affected by back pressure. For reference, Figure 2-25 is a curve of typical flow versus back pressure for a perfect nozzle. The curve shows the flow characteristic transitioning from sonic flow to subsonic flow.

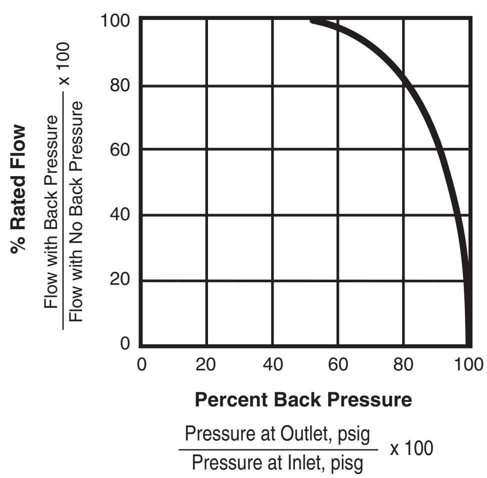



**Figure 2-25. Flow Characteristics of a Perfect Nozzle**

Back pressure on a standard pilot operated valve can cause the main valve to open and flow backwards if it is greater than the inlet process pressure. The back pressure, acting on the unbalanced area of the moving member downstream of the nozzle, produces an upward lifting force. Such a condition might occur if the valve is piped into a pressurized header and process pressure at the valve inlet decreased below the header pressure, such as in a process shut down. Figure 2-26 shows the forces acting on the unbalanced moving member that cause it to lift.

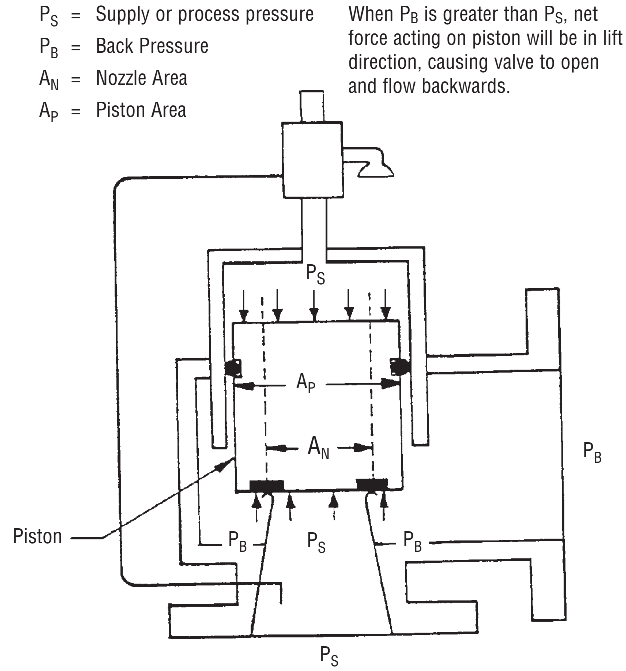



**Figure 2-26. Effect of Back Pressure on a Pilot Operated PRV**

## Backflow Preventer

An accessory called a backflow preventer is available to prevent a pilot operated valve from opening when back pressure exceeds inlet pressure. The most common type in use today consists of a shuttle valve. Figure 2-27 shows this type.

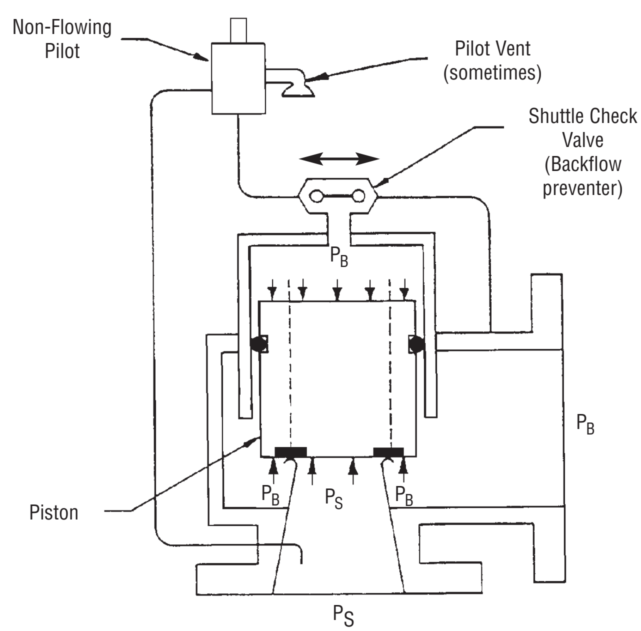



**Figure 2-27. Backflow Preventer – No-Flow Pilot Operated PRV**

# Section II

When $P_B$ is greater than $P_S$ and $P_S$ is below set, the shuttle check transfers to the left, blocking the flow of back pressure to the pilot. The volume on top of the piston is then quickly pressurized with downstream pressure. When $P_S$ exceeds $P_B$ but is below set, the shuttle transfers to the right, permitting pressurization of the volume on top of the piston with supply pressure. When $P_S$ exceeds set pressure, the pilot opens, the cavity depressurizes, the main valve opens, and the shuttle transfers to the left, thus blocking the flow of back pressure to the pilot out through the pilot vent.

The purpose of the second check valve function in the double check is to prevent back pressure from discharging through the pilot when the main valve is open and relieving. In a non-flowing pilot, the volume above the unbalanced moving member is ported directly to the pilot vent in the main-valve-open position. Back pressure acting on the pilot in this manner would also impose additional forces on the internal pressure seating members, causing erratic closure or blowdown of the main valve.

## Pilot Design

The design of the pilot for a pilot operated valve must be self-actuated; that is, it must be actuated by the process pressure. It must also be deemed to be fail-safe-open to comply with the ASME Section VIII code. Pilot valves are currently not permitted by ASME Section I code.

The most common type of pilot design is the no-flow, “pop” action type. A no-flow pilot is one designed to have no flow of the process gas when the main valve is open and relieving. A “pop” action is one in which the main valve rapidly opens at set pressure to full lift and re-closes at some pressure below set. The difference between opening pressure and reseat pressure is called blowdown and is usually expressed in percent of set pressure. For example, a valve that opens at 100 psig and closes at 95 psig would have a 5% blowdown. Conventional valves are also pop action valves, but they only go into partial lift at set, usually about 70% of full lift. As noted in the discussion on conventional valves, overpressure is required to obtain rated lift, to further compress the spring.

The second most common type of pilot design is the no-flow “modulating” action, type. This pilot type produces a main valve opening characteristic that is proportional to the relieving capacity required to maintain set pressure. In this sense, its performance is similar to a back pressure regulator. The pressure at which it opens and closes are nearly the same. Figures 2-28 and 2-29 are illustrations of the two types of pilots.

Pilots should have significant seat areas for consistent operation. Any adhesion of the seating members will cause an increase in set pressure. Pilot valves and conventional valves should have minimal internal friction and good guidance of the moving parts to minimize side loads. High friction and eccentric loads cause erratic operation. This is the primary reason the moving parts within a pilot should be in the vertical orientation.

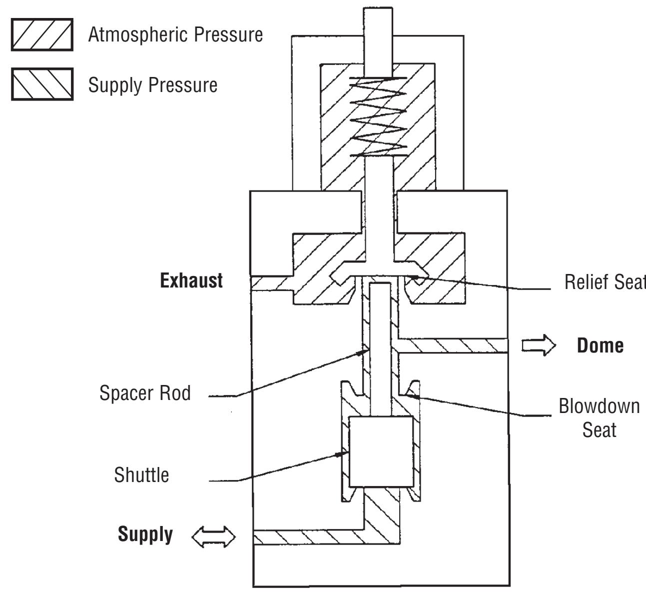



**Figure 2-28a. No-Flow Pop Action Pilot, Dome Supply Position**

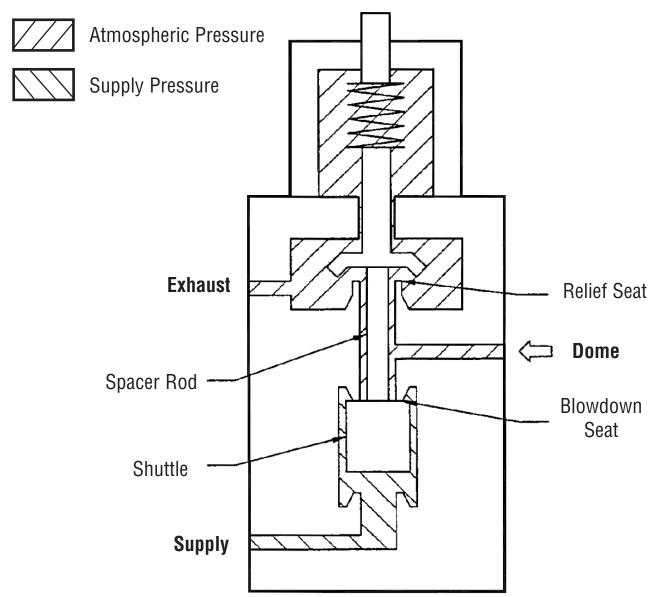



**Figure 2-28b. No-Flow Pop Action Pilot, Dome Exhaust Position**

# Section II

Flowing type pilots are those pilots which flow a small amount of process gas or liquid while the main valve is open and relieving. For some applications the flow through this type of pilot must be vented to a closed header or tailpipe if the process fluid is toxic or flammable or both. If vented to the outlet, the pilot must be internally balanced against back pressure. This means its operating characteristics and set point should not change with back pressure.

Incorporation of a reasonably sized internal filter screen is also desirable to minimize particulate contamination. A remote pilot sense with an auxiliary supply filter will further minimize particulate contamination. Remote sensing accomplishes this by sensing pressure at a location where the velocity of the process fluid to the pilot is extremely low. The velocity at the valve inlet is the highest in a system. Particulates are much more likely to be entrained in a high velocity fluid stream than a lower one.

Externally adjustable set pressure and blowdown adjustments (for pop action only) are also desirable. All external adjustments are required by code to be sealed to prevent unauthorized adjustments.

Other requirements of the code, which apply to all pressure relief valves, are wrenching flats on threaded valve bodies and self-draining outlets. The valve body must be designed to prevent liquids from accumulating in the valve body, downstream of the nozzle. These liquids could solidify and prevent the valve from opening at the correct set pressure.

A final design consideration, one applicable to the design of all pressure relief valves, is ease of maintenance. The optimum design is one that can be serviced in the field, that is, disassembled and assembled, using only an adjustable spanner wrench. Another desirable design feature is to be able to perform the necessary maintenance with the valve installed, without having to remove it from the inlet or outlet piping.

## 2.4.5 Rupture Discs

Rupture discs are non-reclosing safety relief devices designed to provide virtually instantaneous unrestricted pressure relief to a closed system at a predetermined pressure and coincident temperature. Their purpose is to provide overpressure protection to a system which may be subject to excessive pressure by malfunction of mechanical equipment, runaway chemical reaction, and external or internal fires.

Prior to the early 1930's, a rupture disc consisted of a flat metal membrane. Since these devices did not have predictable bursting pressures and since their service life was limited, they were not widely used. Today's rupture disc assembly comprises two parts:

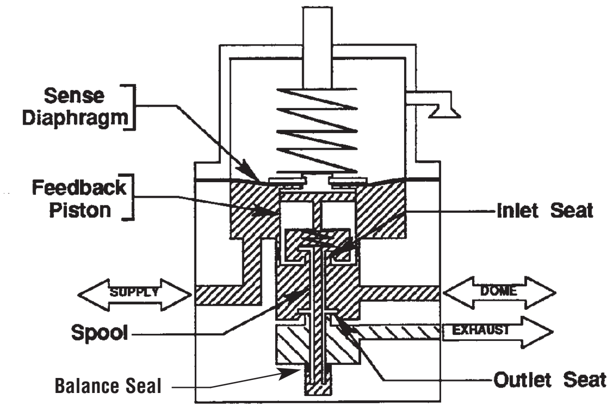



**Figure 2-29a. No-Flow Modulating Action Pilot, Dome Supply Position**

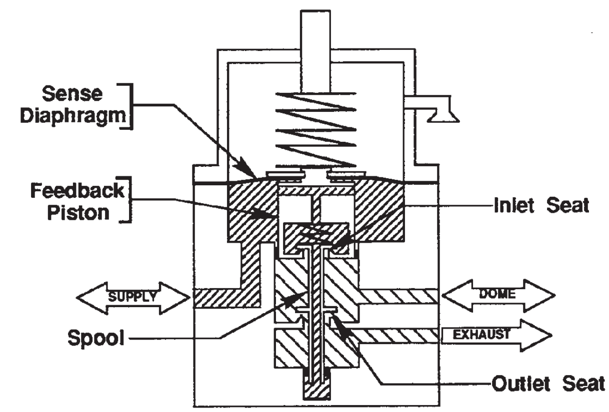



**Figure 2-29b. No-Flow Modulating Action Pilot, Dome Vent Position**

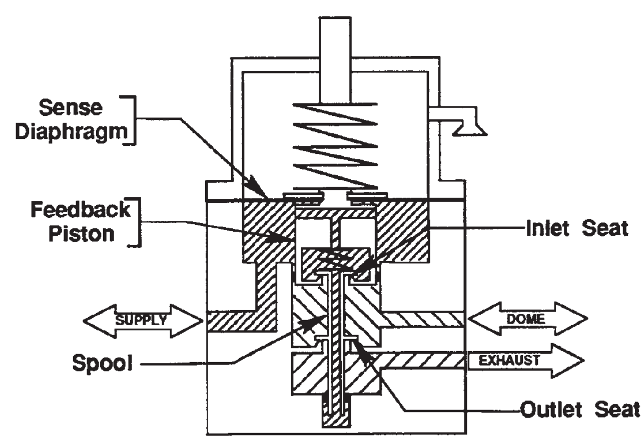



**Figure 2-29c. No-Flow Modulating Action Pilot, Null Position**

# Section II

1. A rupture disc, which is a thin metal diaphragm bulged to a spherical shape providing both a consistent burst pressure within a predictable tolerance and an extended service life; and
2. A rupture disc holder, which is a flange-type structure designed to properly hold the rupture disc in position.

## Configurations - Two Basic Designs

Forward-acting rupture discs are designed to fail in tension. When pressure applied to the concave side reaches the point where severe localized thinning of the metal occurs, the disc will rupture. This type of rupture disc is produced in conventional, composite, and scored designs (See Figure 2-30).

Reverse-acting rupture discs are designed to fail when the disc is in compression. With the convex side of the disc facing the system, pressure is applied until the disc “reverse buckles”. Once reversal pressure is reached, the crown of the disc will snap through the center of the holder and either be cut open by a knife blade or other cutting device, or open along score lines, allowing the pressure to be relieved. This type of rupture disc is classified as either reverse-acting with knife blades or reverse-acting scored (See Figures 2-31a and 2-31b).

## Operating Ratios

Operating ratios are defined as the relationship between operating pressure and the stamped burst pressure of the rupture disc, and are usually expressed as a percentage, i.e. $P_o/P_b$ x 100. In general, good service life can be expected when operating pressures do not exceed the following:

- 70% of stamped burst pressure for conventional prebulged rupture disc designs;
- 80% of stamped burst pressure for composite-design rupture discs;
- 80 to 90% of stamped burst pressure for forward-acting scored design rupture discs (depending upon material thickness);
- Up to 90% of stamped burst pressure for reverse-acting design rupture discs.

Regardless of their design, all rupture discs will exhibit greater service life when the operating pressure is considerable less than the burst pressure. Therefore, there is no advantage in specifying a 90% operating ratio mix when, for example, the process maximum operating pressure is 60 psig and the rated burst pressure is 100 psig. In this application, a rupture disc with a 70 or 80% operating ratio would be suitable for the application.

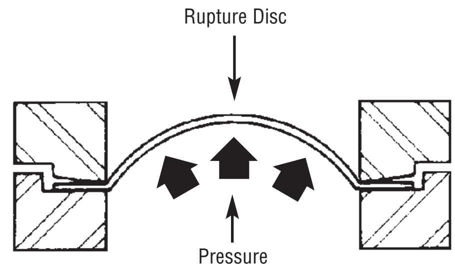



**Figure 2-30. Forward-acting flat seat arrangement (insert bolted-type holder)**

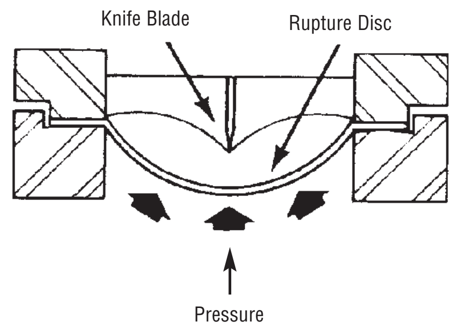



**Figure 2-31a. Reverse-acting, knife blade design**

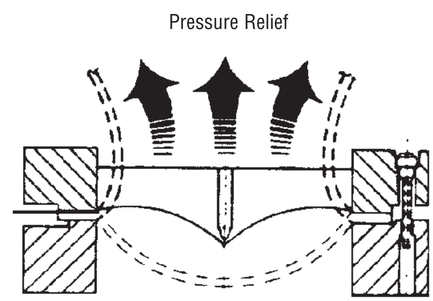



**Figure 2-31b. Pressure relief (disc cut open by knife blades)**

# Section II

## Reverse Acting Rupture Disc

In the mid-1960's the first reverse-acting rupture disc was introduced; the reverse acting rupture disc with knife blades. Its advantages were a 90% operating ratio, predictable opening pattern, and generally non-fragmenting characteristics. Since the disc is compression-loaded, it is able to withstand full vacuum without vacuum supports and will withstand pressures in excess of the burst pressure on the outlet side. When used to isolate pressure relief valves, an ability to withstand pressure on the outlet side seems to allow the testing of valve settings in place (see Figure 2-32). However, for metal-seated, spring type PRVs, there is inadequate volume between the rupture disc and PRV seat to achieve an **accurate** test result.

However, while reverse-acting discs with knife blades offer some advantages, there are also certain disadvantages if such a disc is not properly applied or installed or if it is damaged:

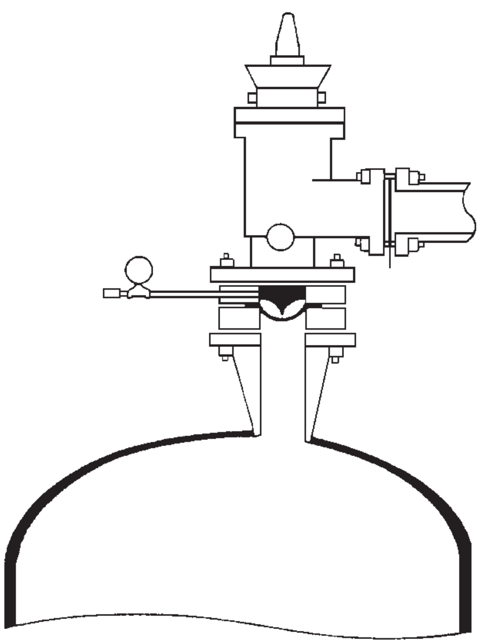


**Figure 2-32. PRV Isolation**

- If the rupture disc assembly is installed upside down or with the knife blades removed, the rupture disc may not fail until the pressure builds up to several times higher than the intended reversing pressure.
- Rupture discs that are damaged or improperly installed may provide lower reversal pressures, removing the "snap back" action required to move the disc through the knife blade, which causes the rupture disc to open. Consequently, the disc may fail to open or only partially open, depending on the particular knife-blade design.
- The proper operation of reversing discs requires the "snap back" action (inherent when operated by compressed gases) to drive them through the knife blades. Thus, in most liquid service applications, reverse-acting discs will not function reliably.

The original knife blade design consisted of a four-blade, straight-edge configuration which required relatively high burst pressures for full opening. As mentioned above, if not properly installed or applied, it was possible to experience incomplete opening of the rupture disc.

A subsequent three-blade design configuration provided improved partial opening characteristics, but the blades were still straight-edge design and did not totally alleviate the problem. In the mid-to-late 1970's, a modified, reverse knife blade design was introduced in the industry. This blade configuration has a "swooped" edge which provides enhanced performance characteristics. The configuration at the blade intersection, along with the radius of the swooped edge, minimizes the possibility for the disc to come to rest against the knife blade without opening (see Figures 2-31a and 2-31b).

Industry is now also using reverse-acting rupture disc designs which do not incorporate knife blades, most notably the scored reverse-acting design. This design simply replaces the knife blades with lines of weakness or scoring. However, if damaged or if improperly installed or applied, there still exists the potential problem of the disc reversing but failing to open until the pressure significantly exceeds the stamped burst pressure. Also, since all reverse-acting rupture discs require the "snap back" action to burst properly, careful consideration must be given to any use of a reverse-scored rupture disc in a liquid application. There are reverse-scored rupture discs that, when improper reversal occurs, will not burst until the pressure exceeds 200% of the stamped burst pressure.

# Section III

## 3.0 Requirements of ASME Section VIII Unfired Pressure Vessel Code

Pressure-relief devices for vessels within the scope of ASME Section VIII, Division 1, Unfired Pressure Vessels, are covered in Section UG-125 through UG-137 of the code. Section U-1 of the code gives vessel classifications outside the code jurisdiction. Pressure vessels in this area include:

– those under federal control

– those having internal or external operating pressure less than 15 psig

– those having an outside diameter of six inches or less

## 3.1 Conditions for Use

Pressure relief devices for vessels within the scope of ASME Section VIII, Division 1, are used under the following conditions.

1. UG-125(c). All pressure vessels other than unfired steam boilers should be protected by a pressure-relieving device that prevents the pressure from rising more than 10% or 3 psi (whichever is greater) above the maximum allowable working pressure (MAWP), except as permitted in UG-125(c)(1) and (c)(2). See Figure 3-1.
2. UG-134(a). If a single valve is used, it must be set at a pressure not higher than the MAWP. When more than one valve is used to meet the required relieving capacity, then only one valve need be set at or below the MAWP. The additional valves can be set at higher pressures, but in no case can they be set greater than 105% of MAWP, except as permitted in UG-134(b). See Figure 3-2.
3. UG-125(c)(1). When multiple pressure-relieving devices are provided and set in accordance with UG-134(a), they must prevent the pressure from rising more than 16% or 4 psi (whichever is greater) above the MAWP. See Figure 3-2.
4. UG-125(c)(2). When an additional hazard can be created by exposure of a pressure vessel to fire or other unexpected sources of external heat, supplemental pressure relieving devices should be installed to protect against excessive pressure. Such supplemental pressure relieving devices shall be capable of preventing the pressure from rising more than 21% above the MAWP. The same pressure relieving devices can be used to satisfy the capacity requirements of (c) or (c)(1) and (c)(2), provided that the pressure setting requirements of UG-134(a) are met. See Figure 3-1.

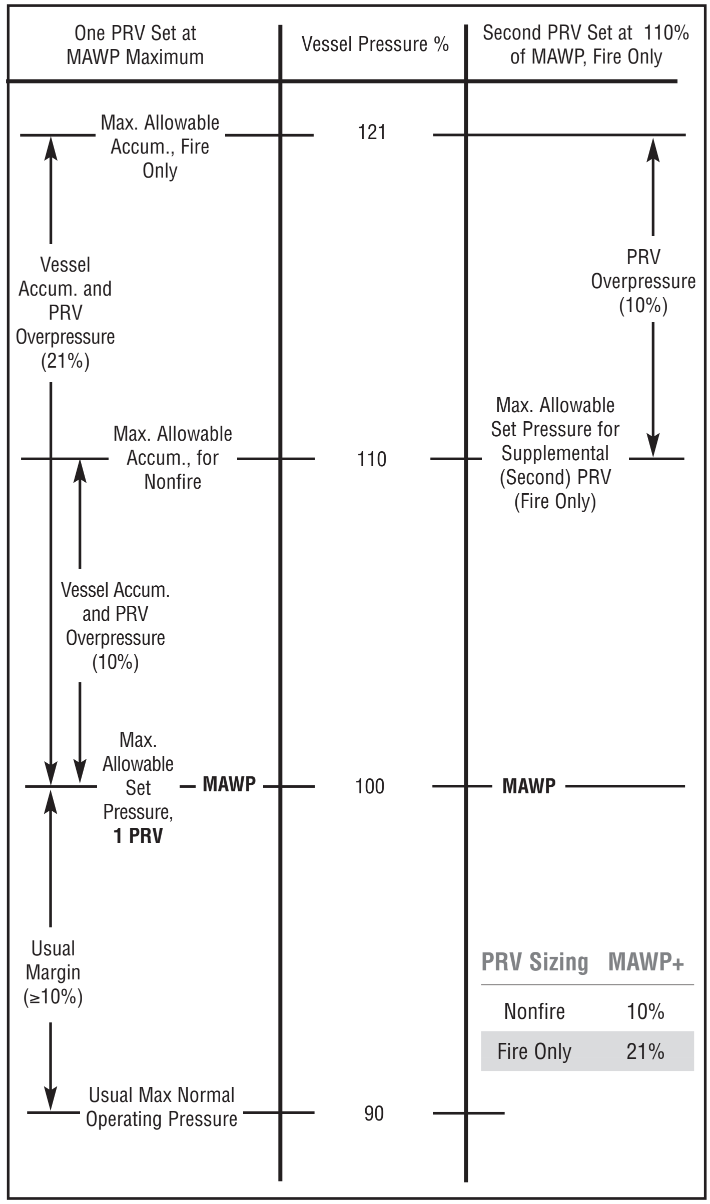



**Figure 3-1. One PRV Used For Nonfire and Fire Case with Supplemental PRV for Fire Case Only**

5. UG-134(b). Protective devices permitted by UG-125(c)(2) (such as fire case) can be adjusted to operate at a pressure not greater than 110% of MAWP. However, if such a device is used to meet the requirements of both UG-125(c) and UG-125(c)(2), it should be set to operate at not more than MAWP. See Figure 3-1.

# Section III

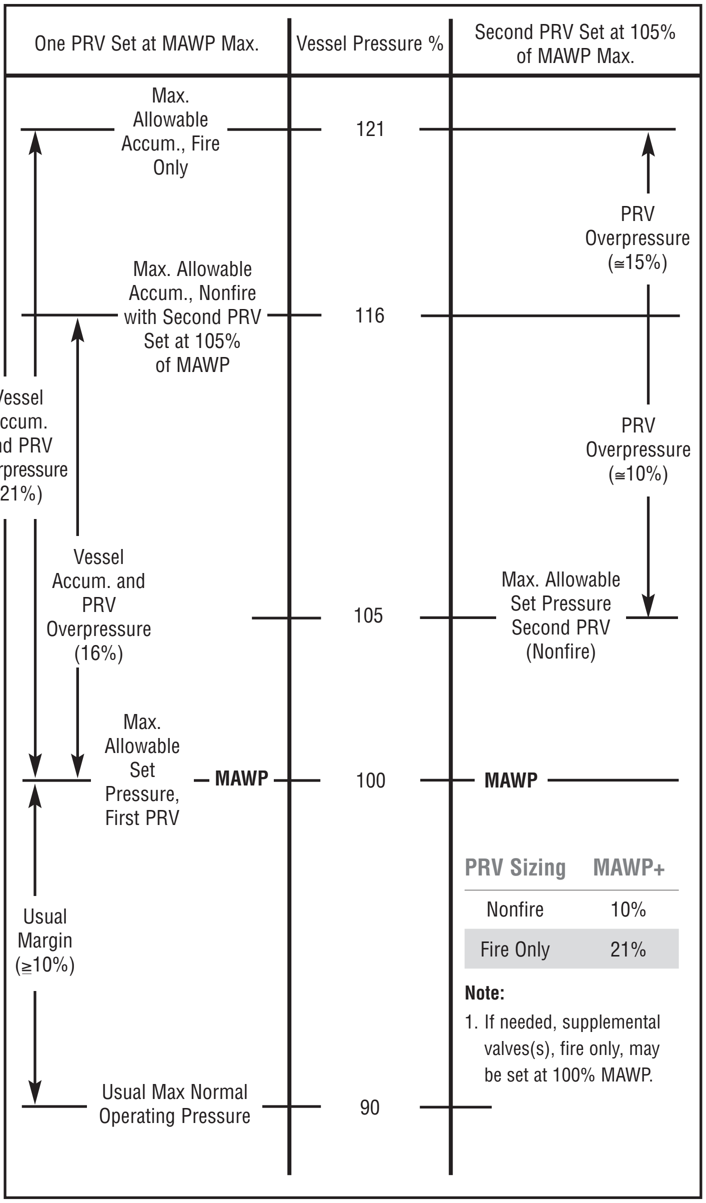



**Figure 3-2. Two Or More PRVs For Nonfire Case But Also Sized for Fire Case**

## 3.2 ASME and the National Board of Boiler and Pressure

### Vessel Inspectors

Concerning pressure relief valves, the ASME and the National Board of Boiler and Pressure Vessel Inspectors play the following roles:

1. The ASME does not certify or approve any device. The National Board of Boiler and Pressure Vessel Inspectors is responsible for surveying the following:
   a. Manufacturer's facilities
   b. Code compliance of the valve design
   c. Quality control systems
   d. Flow test facilities used to establish valve capacity in accordance with ASME Code.
2. Capacity tests must be conducted at a certified flow test facility in the presence of an authorized observer. The actual capacities of all valves tested must fall within specified limits to the average capacity. A "$K_D$" (or Coefficient of Discharge) is calculated for each test valve (total of 9 valves) when a family of valves of the same design is tested. The calculation is shown below:

$$K_D = \frac{\text{Actual Flow}}{\text{Theoretical Flow}}$$

The $K_D$ values for all the test valves are then averaged and multiplied by 0.90. (None of the individual coefficients can exceed plus or minus 5% of the average of the nine tests.) This becomes the ASME "K" (Nozzle Coefficient) that the manufacturer is permitted to use in publishing capacities and in stamping the capacity of the valve.

Capacity test data reports for each valve model, type, and size should be signed by the manufacturer and the authorized observer witnessing the tests shall be submitted to the National Board of Boiler and Pressure Vessel Inspectors for certification. Where changes are made in the design, capacity certification tests must be repeated. Detailed drawings showing the valve construction are also submitted.

3. A Certificate of Authorization is granted by ASME to the manufacturer of the pressure relief valve to apply the "UV" stamp to nameplates bearing the capacity that has been certified by the National Board. The nameplate also bears the "NB" symbol, indicating that the capacities have been certified. Reapplication must be made every 3 years for Certificate of Authorization renewal.
4. Valves certified by The National Board of Boiler and Pressure Vessel Inspectors are published in the publication "Relieving Capacities of Safety Valves and Relief Valves approved by The National Board."

# Section III

## 3.3 Pressure Relief Valve Requirements

ASME Section VIII, Division 1, Sections UG-130 and UG-131 list the specific code requirements for pressure-relief valves. Note the following information about these requirements:

1. UG-126(a). Safety relief and relief valves should be the direct spring loaded type.
2. UG-126(b). Pilot operated pressure relief valves can be used under the following conditions:
   a. The pilot is self-actuated.
   b. The main valve will open automatically at not over the set pressure.
   c. The main valve will discharge its full rated capacity if some essential part of the pilot fails.
3. UG-126(c). The spring in a pressure relief valve should not be reset for any pressure more than ±5% different from that for which the valve is marked without first contacting the PRV manufacturer to determine the proper spring for the new set pressure.
4. UG-136(a)(3). Lifting devices are required for steam, air, or water over 140°F. Lifting devices can be either the open or totally enclosed (packed) design. In lieu of a lift lever, pilot operated valves can be equipped with a field test connection which, when used, will actuate both the pilot and main valve. A lift lever must not be used when inlet pressure is less than 75% of set pressure.

   **EXCEPTION**
   ASME Code Case 2203 (March 7, 1996) allows the omission of lifting devices on PRVs for these services if:

   A. The purchase order for the new PRV states omission of the lifting device in accordance with Code Case 2203.
   B. The PRV user has a documented program for periodic testing and repair, if necessary.
   C. Permission must be obtained from the local jurisdictional authorities.
5. UG-136(a)(8). If the design of a pressure-relief valve is such that liquid can collect on the discharge side of the disc (or seat), the valve must be equipped with a drain at the lowest point where liquid can collect.

## 3.4 Set Pressure Tolerances of Pressure Relief Valves

Section UG-126(d) provides the following set pressure tolerances:

1. For set pressures up to and including 70 psig, the tolerance for PRVs is plus or minus 2 psig.
2. For set pressures above 70 psig, the tolerance for PRVs is plus or minus 3%.

## 3.5 Inlet Pressure Drop

1. UG-135(b). The pressure drop through the inlet piping should not reduce the valve relieving capacity below that required or adversely affect pressure relief valve operation. This allows the effective use of remotely sensed pop action. Note that a 3% maximum inlet pressure loss is <u>not</u> mandatory.
2. Non-mandatory Appendix M-7(a). Inlet piping pressure losses to the pressure relief valve should not exceed 3%. This is primarily for the benefit of direct spring operated PRVs.

## 3.6 Block (Stop) Valves

1. UG-135(e)(1) and (e)(2). The use of intervening block valves between the vessel and the protective device is permitted only under the following conditions:
   a. Block valves are constructed or positively controlled so that when the maximum number of valves are closed at one time, the relieving capacity provided by the unaffected relieving devices is not reduced below the required relieving capacity or
   b. Block valves are set up in accordance with conditions described in Appendix M.
2. Block (stop) valve placement can be as follows:
   a. Block (Stop) Valves on the Inlet to a Pressure Relief Device

According to ASME Section VIII, Division 1, Appendix M-5, block valves are allowed for the purpose of inspection and repair only. This valve should be a full-area design. ASME also advises that it should have a full round port area and be equal to or greater than the pressure relief valve inlet. The valve should be locked in the full open position, and should not be closed except by an authorized person. When the valve is closed during any period of the vessels operation, an authorized person should remain stationed at that block valve.

   b. Block (Stop) Valves on the Discharge Side of a Pressure Relief Device

## Section III

According to Appendix M-6, block valves are allowed when discharge through a pressure relief device is to a common header with other lines from other pressure relieving devices. This valve must also be of a full-area design and include a provision for locking in the open position or closed position. When the valve is to be closed during operation of the vessel, an authorized person should remain stationed there. Under no conditions should this valve be closed while the vessel is in operation, except when a block (stop) valve on the inlet side of the safety relieving device is installed and first closed. This is especially true of metal seated, direct spring valves which have poor seat tightness. The danger is in opening the inlet block valve, allowing the PRV seat leak to overpressure the discharge side of the PRV as well as the discharge piping upstream of the discharge block valve.

### 3.7 Valve Marking

UG-129(a) requires the manufacturer or assembler to mark all safety, safety relief, liquid relief, and pilot operated pressure relief valves with the following data in such a way that the marking will not be obliterated in service.

1. The name or an acceptable abbreviation of the manufacturer
2. Manufacturer's design or type number
3. Inlet size
4. Set pressure (psig)
5. Certified capacity
    a. SCFM or lb./min. air at an overpressure of 10% or 3 psi, whichever is greater.
    b. For valves certified on steam, lb./hr. saturated steam at an overpressure of 10% or 3 psi, whichever is greater.
    c. For valves certified on water, gal./min. water at 70°F at an overpressure of 10% or 3 psi, whichever is greater.
6. Year built, or a coding such that the manufacturer can identify the year built.

# Section III - ASME Section VIII and ASME Section I

| ASME Section VIII - Unfired Pressure Vessels | ASME Section I - Power Boilers |
|---|---|
| 1. UG-125 General:<br>(c) must prevent overpressure from rising more than 10% or 3 psi whichever is greater<br>multiple pressure relief devices – allowed to go to 16% or 4 psi whichever is greater | PG-67.2 Safety Valve or Safety Relief Valves prevent boiler pressure from rising more than 6% above maximum allowable working pressure. (Individual valves must reach their full rated capacity at 2 psi or 3% overpressure depending on the set pressure range.) |
| fire sizing - allowed to go 21% overpressure | No Provision for Fire Sizing |
| (e) Non-reclosing pressure relief devices (rupture discs) are permitted either alone or in combination with a pressure relief valves. | Rupture discs are not permitted by Section I – Except for organic fluid vaporizers as covered in PVG 12.3.4 |
| 2. UG-126 Pressure Relief Valves<br>(b) Permits pilot operated valves | Pilot valves are not permitted |
| 3. UG-131 through UG-136 Operational Requirements | PG-72 Operation |
| Set pressure tolerance<br>2 psi . . . . . from 15 psi through 70 psi<br>3% . . . . . . over 70 psi | Set pressure tolerance<br>2 psi . . . . . 15 psi through 70 psi<br>3% . . . . . . 71 psi through 300 psi<br>10 psi . . . . 301 psi through 1000 psi<br>1% . . . . . . Over 1000 psi |
| Blowdown:<br>Only required during product certification testing. Not a requirement for production valves; most manufacturers meet 10%. | Blowdown:<br>4 psi . . . . . less than 67 psi<br>6% . . . . . . greater than 67 psi to 250 psi<br>15 psi . . . . greater than 250 psi to 374 psi<br>(Except PG-72.1)<br>Safety valves on forced-flow steam generators with no fixed steam and waterline, and safety relief valves used on high-temperature water boilers may be set and adjusted to close after blowing down not more than 10% of the set pressure. |
| Overpressure:<br>3 psi or 10% whichever is greater | Overpressure:<br>No greater than 3% over the set pressure. |
| 4. UG-133 Determination of Pressure Relieving Requirements<br>(g) Address prorated capacity about 1.10 p | Prorated capacities are not considered – only consider combined capacity of each valve at 3% accumulation. |
| 5. UG-135 Installation:<br>Permits the use of intervening stop valves between the vessel and the pressure relief valve. | Section I does not allow intervening stop valves. |

# Section IV

## 4.0 Department of Transportation Code Requirements - Gas Transmission and Distribution Piping Systems

This review of overpressure protection devices is based on the ASME Guide for Gas Transmission and Piping Systems.

In November 1970, the U.S. Department of Transportation (DOT) issued minimum safety standards for the transmission and distribution of natural gas. These standards are contained in Part 192, Title 49 of the Code of Federal Regulations, titled "Transportation of Natural and Other Gas by Pipeline: Minimum Federal Standards".

Prior to November 1970, the ANSI B31.8 Code was used. Since the sponsoring organization of the ANSI B31.8 committee was the ASME, they formed the ASME Gas Piping Standards Committee in cooperation with the DOT Office of Pipeline Safety Operations (OPSO). This committee published the referenced ASME guide. The guide includes the DOT Federal Gas Pipeline Safety Standards and practices recommended by the committee. The guide provides "how to" advisory material to meet the DOT Code.

Unlike the ASME Code, the DOT Code permits use of pressure limiting devices as well as pressure relief devices in most applications to protect against an overpressure condition. However, bottle-tight facilities, such as storage vessels, are not allowed to use these devices.

## 4.1 Valve Requirements

Section 192.199 of the DOT Code lists the following valve requirements (rupture discs are excluded):

1. Valves must be constructed of corrosion resistant materials that will not impair device operation.
2. Valves must be designed with non-sticking seats that support continuous device operation.
3. Valves must be designed and installed to readily support the following requirements:
   a. Determination if the valve is free
   b. Testing to determine the operating pressure
   c. Testing for leaks when in the closed position
4. Valves must have support made of noncombustible material.
5. Valves must have discharge stacks, vents, or outlet ports designed to prevent accumulation of water, ice, and snow. These stacks, vents, and outlet ports must be located where gas can be discharged into the atmosphere without undue hazard.
6. Valves must be designed and installed to prevent valve hammering and impairment of relief capacity. The sizes of the following items must be adequate:
   a. Size of the openings, pipe, and fittings located between the system to be protected and the pressure relieving device.
   b. Size of the vent line.
7. Valves must be designed and installed to prevent any single incident from affecting the operation of both the overpressure protective device and the district regulator. (In this case, the district regulator is installed at a district regulator station to protect a pipeline system from overpressure.) Possible incidents include explosion in a vault or damage by a vehicle.
8. Valves must be designed to prevent unauthorized operation of any stop valve that will make the pressure relief valve or pressure limiting device inoperative. Valves that will isolate the system under protection from its source of pressure do not have to be designed to prevent unauthorized operation.

## 4.2 Overpressure Protection Requirements

The DOT Code addresses the following two areas that require overpressure protection:

– Compressor Stations (Section 192.169)

– Pipelines and Distribution Systems (Sections 192.195 and 192.197)

Overpressure protection is required in all cases, except on distribution systems operating under 60 psig with the following qualified service regulator:

– size: 2-inches or less

– single port valve with resilient seats that resist abrasion

–- self-contained with no external static or control lines

– will regulate accurately under normal conditions

Section 192.201 does require that an overpressure protection device be installed at or near each regulator station. This will limit the maximum pressure in the main to a value that will not exceed the safe operating pressure for any connected equipment.

Requirements for the following areas are discussed on the following pages:

– compressor stations

– bottle-tight facilities

– high-pressure distribution systems

– low-pressure distribution systems

# Section IV

## 4.2.1 Compressor Stations (192.169)

Note the following information about compressor stations:

1. A pressure relief device or automatic compressor-shutdown device must be installed in the discharge line of each compressor between the gas compressor and the first discharge block valve. From a practical standpoint, a pressure relief device is necessary on positive displacement compressors whether or not a shutdown device is used. The relieving capacity should be equal to or greater than the compressor capacity.
2. Additional relieving devices should be installed on the piping to prevent it from being overpressured if the devices under (1) do not suffice.
3. The maximum allowable operating pressure (MAOP) of the station should be protected from a pressure rise of more than 10%. (Note that no set pressure is specified. This allows PRVs to be set over MAOP if the valve achieves full lift at set pressure. This permits operating pressures over MAOP during peak-load times.) If a code stamped PRV is used, the certified nameplate capacity is at 110% of set pressure; therefore, if the PRV is set over MAOP, the valve nameplate capacity will still be shown at 110% of set pressure.

## 4.2.2 Bottle-Tight Facilities (192.195)

Bottle-tight facilities require the following equipment:

1. Spring-loaded or pilot operated PRVs that meet the requirements of the ASME Pressure Vessel Code, Section VIII, Division I.
2. Pilot operated, back pressure regulators designed to open should a control line fail.
3. Rupture discs that meet the requirements of the ASME Pressure Vessel Code, Section VIII, Division I.

Where a liquid separator is used in the suction line of a compressor (192.165), it must be designed in accordance with Section VIII of the ASME, Boiler and Pressure Vessel Code. In this case, only a device meeting (1) or (3) above can be used.

## 4.2.3 High-Pressure Distribution Systems (192.195, 192.197)

A high pressure distribution system is one in which the gas pressure in the main is higher than the pressure provided to the customer. High pressure distribution systems require the following equipment:

1. Spring-loaded PRVs as in Section 4.2.2, Number 1
2. Weight-loaded PRVs
3. A monitoring regulator installed in series with the primary regulator
4. A series regulator installed upstream from the primary regulator. This regulator must be set to continuously limit the pressure on the inlet of the primary regulator to the MAOP of the distribution system or less.
5. An automatic shutoff device installed in series with the primary regulator. This device must be set to shut off when the pressure in the distribution system reaches the MAOP or less.
6. Pilot operated back pressure regulators designed to open should a control line fail.
7. Spring-loaded diaphragm-type PRVs

## 4.2.4 Low-Pressure Distribution Systems (192.195, 192.197)

A low pressure distribution system in one in which the gas pressure in the main is substantially the same as the pressure provided to the customer. Low pressure distribution systems require the following equipment:

1. Liquid seal relief valve
2. Weight-loaded relief valve
3. Monitoring regulator as in Section 4.2.3, Number 3
4. Series regulator as in Section 4.2.3, Number 4
5. Automatic shutoff device as in Section 4.2.3, Number 5 (above)
6. Pilot operated back pressure regulator designed to open should a control line fail.

# Section IV

## 4.3 Capacity Requirements of Pressure Relieving and Limiting Stations (192.201)

Pressure relief stations must have enough capacity and must be set to operate to provide the following protection:

1. Low Pressure Distribution Systems - The device must prevent a pressure that would cause the unsafe operation of any connected and properly adjusted equipment using gas.
2. Pipelines - The device must prevent the pressure from exceeding the MAOP by the following criteria:
   a. 10% or the pressure that produces a hoop stress of 75% or the specified minimum yield strength, whichever is lower when the MAOP is 60 psig or greater
   b. 6 psig when the MAOP is 12 psig or more, but less than 60 psig
   c. 50% when the MAOP is less than 12 psig

## 4.4 Selection of an Overpressure Device

The first consideration in selecting a safety device from the long list of those permitted is whether pressure relief can be used or whether the installation should be designed around pressure limiting. This will likely be governed by the location of the installation and whether gas venting to the atmosphere or noise during pressure relief would be objectionable. Often a combination of pressure limiting and back-up pressure relief is used.

If it is possible to use a pressure relief valve, there are certain advantages over pressure limiting devices, as follows:

1. Pressure relief valves may be more economical.
2. Testing is much simpler.
3. A seat leak represents a safe rather than an unsafe condition.
4. They require less maintenance because they are not operating continuously.

## 4.5 Inspection and Testing of Overpressure Protection Devices (192.739)

Each installation must be inspected at intervals at least once each calendar year, but not to exceed 15 months, to determine that the devices used meet the following criteria:

– Good mechanical condition
– Adequate capacity
– Operationally stable
– Operational at the correct pressure
– Properly installed
– Protected from dirt, liquid, or other conditions that might prevent proper operation

Rupture discs are excluded from these requirements.

Use of a pilot operated PRV makes it easier to verify set pressure and that the moving parts are free. An optional field test accessory makes it possible to check the set pressure of the valve without removing it from service by using an external gas supply and a pressure gauge.

## 4.6 Capacity Testing of Relief Devices (192-743)

Verification of adequate capacity of all relief devices must be made at intervals at least once each calendar year, but not to exceed 15 months. If feasible, the device must be tested in place to determine if it has sufficient capacity to limit the pressure on the connected facilities to the desired maximum. Since this would involve blocking in a compressor and running it at full capacity or readjusting a pressure reducing valve to the wide open position against a closed downstream block valve, pipeline operators are reluctant to verify capacity in this manner.

In lieu of verification testing, the Code permits a mathematical review of required capacity as compared to available capacity. The review must be performed once a year. If the analysis reveals insufficient capacity, the situation must be corrected by adding new or additional devices. Use of an ASME “UV” code stamp relief valve is helpful in performing the review, because the relief capacity of the valve is certified by the National Board of Boiler and Pressure Vessel Inspectors and the valve can be used without further justification or testing.

## 4.7 Conclusion

There is no single device among the seven permissible types which can be called the best for all services. Selection must be governed by the particular installation. However, it is easier to comply with maintenance and testing requirements of the DOT Code by using ASME “UV” code stamp pressure relief valves that are designed for in-place field testing of set pressure and have National Board Certified capacities.

# Section V

## 5.0 Pressure Relief Valve Sizing

Sizing pressure relief valves involves determining the correct orifice for the specific valve type to be used to support a required relieving capacity. The typical method used for sizing pressure relief valves is as follows:

1. Establish a set pressure at which the PRV is to operate. This pressure is based on the pressure limits of the system and the applicable code.
2. Determine the relieving capacity required.
3. Select a valve size that will flow the required relieving capacity when set at the pressure determined in Item 1 above.

Pressure relief valves are sized either by calculation or by selection from a capacity chart according to the valve type and process fluid. Sizing from a capacity chart is self-explanatory. Sizing by calculation of the orifice area from a known required capacity is explained in the following sections. All of the formulas used were obtained or derived from APR RP 520, Part 1.

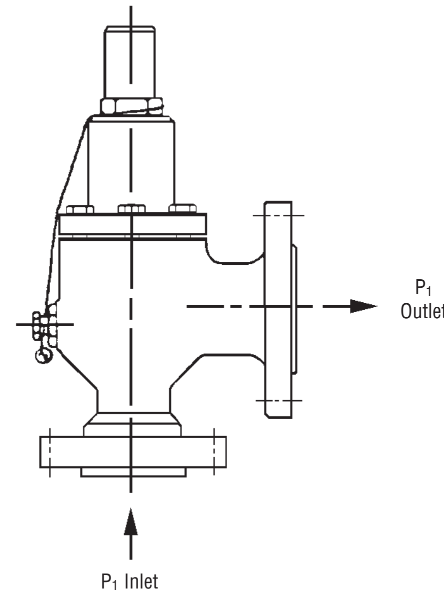



**Figure 5-1. Inlet/Outlet PRV Pressure**

## 5.1 Sizing for Vapors and Gases

Sizing for vapors or gases can be calculated either by capacity weight or volume. The formulas used are based on the perfect gas laws. These laws assume that the gas neither gains nor loses heat (adiabatic), and that the energy of expansion is converted into kinetic energy. However, few gases behave this way and the deviation from the perfect gas laws becomes greater as the gas approaches saturation. To correct for these deviations, various correction factors are used, such as the gas constant ("C") and compressibility factor ("Z").

### 5.1.1 Sonic Flow

The sizing formulas for vapor or gas fall into two general categories based on the flowing pressure with respect to the discharge pressure. These two categories are sonic and subsonic. Sonic flow generally occurs when the absolute pressure at the valve inlet is approximately two times the pressure at the valve outlet (see Figure 5-1) and is 15 psig [1.03 barg] or higher. At that ratio, the flow through the valve orifice becomes sonic. This means that the flow reaches the speed of sound for the particular flowing medium. Once the flow becomes sonic, the velocity remains constant. No decrease of $P_2$ will increase the flow. Flow under these conditions is sometimes referred to as "choked" flow.

The formulas used for calculating orifice areas for sonic flow are:

$$A = \frac{V\sqrt{MTZ}}{6.32\ C\ K\ P_1\ K_b} \qquad \text{(Equation 1)}$$

$$A = \frac{W\sqrt{TZ}}{C\ K\ P_1\sqrt{M}\ K_b} \qquad \text{(Equation 2)}$$

Where:

A = Valve orifice area (inch$^2$)

V = Flow capacity (SCFM)

W = Flow capacity (lb./hr.)

M = Molecular weight of flowing medium

T = Inlet temperature, absolute (°F + 460)

Z = Compressibility factor

C = Gas constant based on ratio of specific heats at standard conditions

K = ASME coefficient of discharge

$P_1$ = Inlet pressure (psia) during flow
Set pressure (psig) – inlet pressure drop (psig) + overpressure (psig) + local atmospheric

$K_b$ = Capacity correction factor due to back pressure

# Section V

To convert flow capacity from SCFM to lb/hr use:

$$W = \frac{MV}{6.32} \qquad \text{(Equation 3)}$$

The molecular weight ("M") of the flowing medium can also be determined from the specific gravity.

Where: M = 29G

and G = Specific gravity of medium referenced to 1.00 for air at 60°F and 14.7 psig

The compressibility factor "Z" is required for the deviation of the actual gas from the perfect gas at higher pressures and is a ratio evaluated at inlet conditions. For natural gas, AGA (American Gas Association) Committee Report No. 3, Table 16, gives full tables for the supercompressibility factor, $F_{pv}$.

The compressibility factor is equal to $\left(\frac{1}{F_{PV}}\right)^2$.

A plot of "Z" for a hydrocarbon gas is shown in Figure 5-2.

This figure shows the compressibility factor for the lower pressure ranges is usually less than 1.00. If "Z" is unknown, 1.00 can be used. However, the calculated relief area might be understated if the relieving pressure is high. If a correction factor is used, it is important to know the gas temperature at the valve during relief, since "Z" can be quite variable with pressure at the lower temperatures. A chart for evaluating "Z" for hydrocarbons can be found in API RP 520, Part 1.

The gas constant "C" is based on the ratio of specific heats $k = C_P/C_V$ at standard conditions and is usually given in most manufacturer's catalogs. Table 5-1 on the next page lists some typical gas properties.

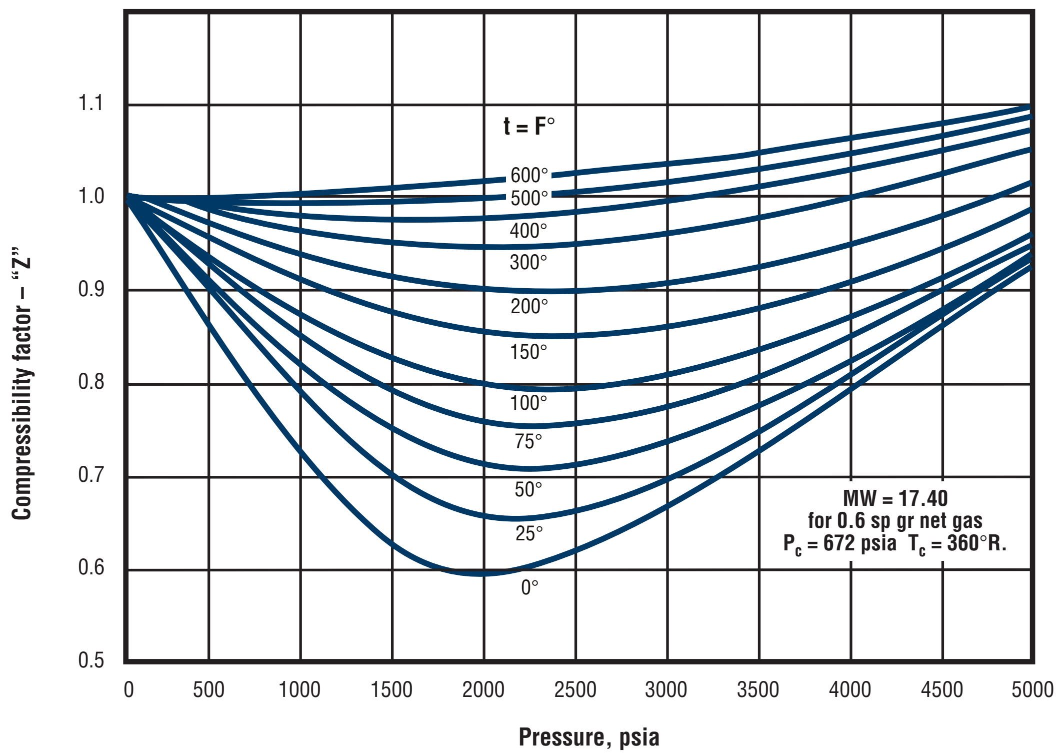



**Figure 5-2. Compressibility of Hydrocarbon Gas**

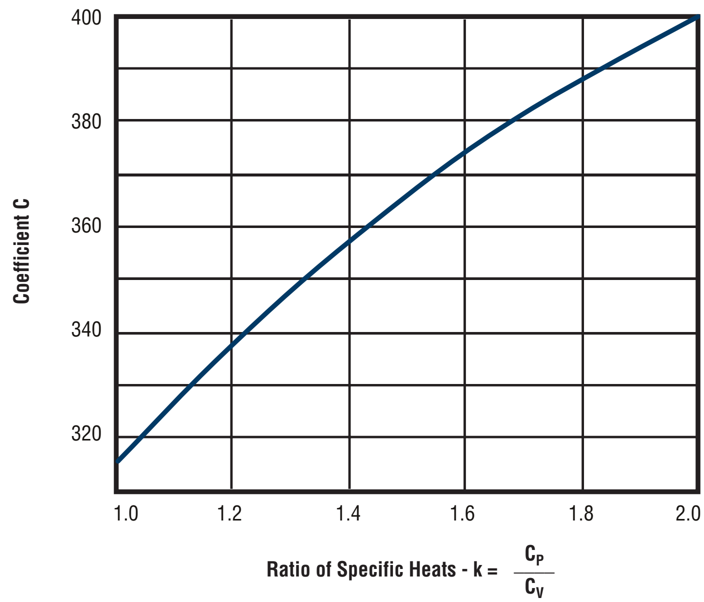
## Section V

**Table 5-1. Properties of Gases**

| Gas | Molecular Weight | "C" Factor | "k" Specific Heat Ratio |
|---|---|---|---|
| Acetylene | 26 | 343 | 1.26 |
| Air | 29 | 356 | 1.40 |
| Ammonia | 17 | 348 | 1.31 |
| Argon | 40 | 378 | 1.67 |
| Benzene | 78 | 329 | 1.12 |
| Butadiene | 54 | 329 | 1.12 |
| Carbon Dioxide | 44 | 345 | 1.28 |
| Carbon Monoxide | 28 | 356 | 1.40 |
| Ethane | 30 | 336 | 1.19 |
| Ethylene | 28 | 341 | 1.24 |
| Freon 22 | 86 | 335 | 1.18 |
| Helium | 4 | 377 | 1.66 |
| Hexane | 86 | 322 | 1.06 |
| Hydrogen | 2 | 357 | 1.41 |
| Hydrogen Sulfide | 34 | 349 | 1.32 |
| Methane | 16 | 348 | 1.31 |
| Methyl Mercapton | 48 | 337 | 1.20 |
| N-Butane | 58 | 326 | 1.09 |
| Natural Gas (0.65) | 18.9 | 344 | 1.27 |
| Nitrogen | 28 | 356 | 1.40 |
| Oxygen | 32 | 356 | 1.40 |
| Pentane | 72 | 323 | 1.07 |
| Propane | 44 | 330 | 1.13 |
| Propylene | 42 | 332 | 1.15 |
| Steam | 18 | 348 | 1.31 |
| Sulphur Dioxide | 64 | 346 | 1.29 |
| VCM | 62 | 335 | 1.18 |

If the gas or vapor is unknown, use C = 315, which will be conservative.

The gas constant "C" can also be calculated using the equation shown in Figure 5-3. The ratio of specific heat varies with pressure and temperature. The value of k used to determine the gas constant "C" that is published in most tables, including Table 5-1 above, is evaluated at 60°F and at atmospheric pressure. Evaluating "C" at the actual conditions that exist at the valve inlet is difficult because of the limited data available. Appendix B of API RP 520, Part 1 contains a definition and discussion of an isentropic coefficient "n" to account for the change in the coefficient k that occurs throughout gas expansion.




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

**Figure 5-3. Gas Constant, C**

The valve coefficient of discharge, "K" and actual PRV orifice areas that *can* be used for ASME Code applications is the one published in the National Board of Boiler and Pressure Vessel Inspector book of Safety Valve and Safety Relief Valve Relieving Capacities. This coefficient is equal to 90% of the actual coefficient of discharge of the valve, "$K_D$". For non-code applications or for applications where the code does not apply, such as those below 15 psig, the actual coefficient of discharge, "$K_D$" can be used. However, the effective coefficient for some pressure relief valves decreases in the subsonic flow region. The manufacturer should be consulted to determine if a deviation exists.

All of the parameters used to calculate the required orifice area are those based on inlet conditions at the pressure relief valve inlet. The inlet pressure used in equations (1) and (2) should be the pressure at the valve inlet when the valve is open and flowing. **Any inlet piping losses that occur between the process and valve inlet during relief should be taken into account.** Many times these losses can be significant, resulting in a greatly reduced relieving capacity.

The pressure at the inlet is the stagnation or total pressure. It is the sum of the static pressure and the velocity head. The pressure used for sizing at the valve outlet, discussed under Subsonic Flow, is static pressure.

# Section V

The formula for calculating orifice areas for sonic flow steam vapor is:

$$A = \frac{W}{51.5\, K\, K_{SH}\, K_P\, P_1}$$

**Where:** A = Orifice area, in²

W = Flow capacity, lb./hr.

K = ASME coefficient of discharge

$K_{SH}$ = Superheat correction factor

$K_P$ = Correction factor for pressures above 1500 psig

$P_1$ = Inlet pressure during flow (psia) [Set – Inlet Pressure Loss + Overpressure + Local Atmosphere]

The superheat factor $K_{SH}$ corrects the flow rate for steam above the saturated temperature. For saturated steam temperatures, $K_{SH}$ = 1.00. For temperatures higher than saturation temperature, $K_{SH}$ is less than 1.00. Table 5-2 on pages 30 and 31 is a list of superheat correction factors.

The high pressure correction factor $K_P$ corrects for the increase in flow rate above 1500 psig. It is dependent only on the absolute inlet pressure. Figure 5-4 is a curve showing this correction factor.

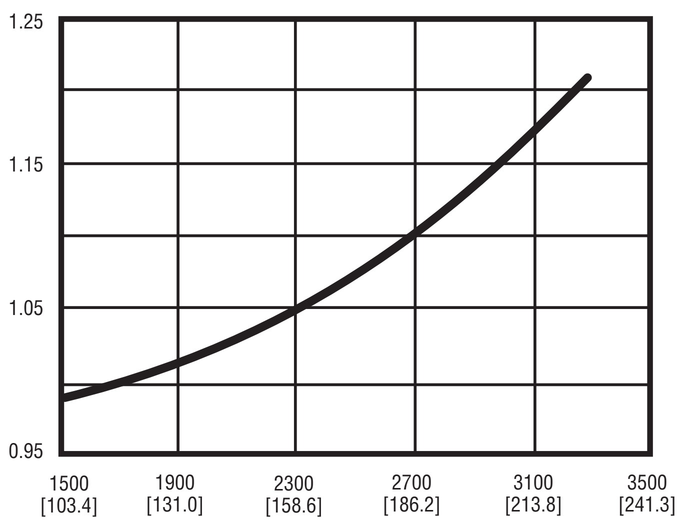



**Figure 5-4. High Pressure Correction Factor**

### Back Pressure Considerations for Vapor or Gas Relief

Increases in back pressure beyond the critical flow pressure of a particular PRV reduces flow and hence capacity. With a conventional valve, the capacity can be reduced quickly with back pressure due to reduced lift. As noted in the Design Session, the balance of forces acting on a valve seat disc in the flowing position is critical. Full lift of the disc must be able to be achieved by 110% of set. Any disturbance of the  pressure differentials across the disc/disc holder will upset these forces, permitting the disc lift to be suppressed. A typical capacity versus back pressure curve for a conventional valve is shown in Figure 5-5. This curve clearly illustrates why it is common, good engineering practice to limit built-up back pressure of a conventional PRV to 10%

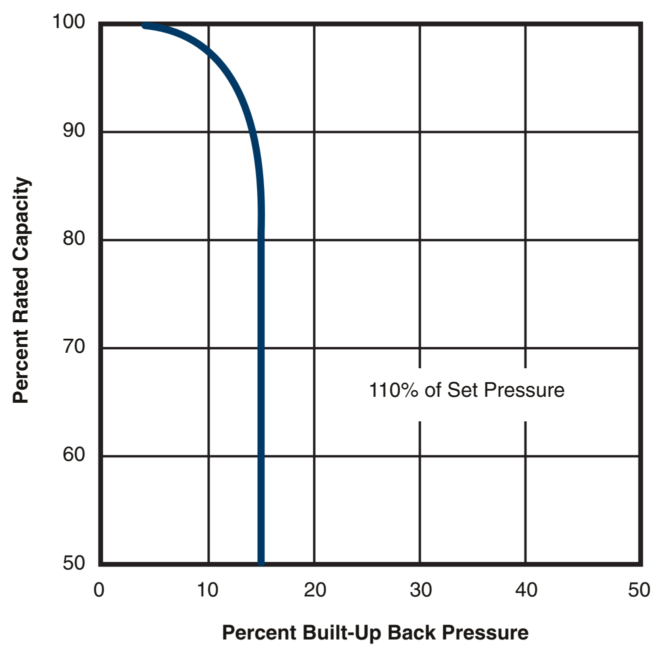



**Figure 5-5. Typical Effects of Built-Up Back Pressure on Relief Capacity of Conventional PRV**

# Section V

## Table 5-2. Steam Superheat Correction Factor

| Relieving Pressure psia | [bara] | Total Steam Temperature °F [°C] 400 [205] | 450 [232] | 500 [260] | 550 [288] | 600 [316] | 650 [343] | 700 [371] | 750 [399] | 800 [427] | 850 [455] | 900 [482] | 950 [510] | 1000 [538] | 1050 [566] | 1100 [593] | 1150 [621] | 1200 [649] |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 50 | [3.5] | .987 | .957 | .930 | .905 | .882 | .861 | .841 | .823 | .805 | .789 | .774 | .759 | .745 | .732 | .719 | .708 | .696 |
| 100 | [6.9] | .998 | .963 | .935 | .909 | .885 | .864 | .843 | .825 | .807 | .790 | .775 | .760 | .746 | .733 | .720 | .708 | .697 |
| 150 | [10.3] | .984 | .970 | .940 | .913 | .888 | .866 | .846 | .826 | .808 | .792 | .776 | .761 | .747 | .733 | .721 | .709 | .697 |
| 200 | [13.8] | .979 | .977 | .945 | .917 | .892 | .869 | .848 | .828 | .810 | .793 | .777 | .762 | .748 | .734 | .721 | .709 | .698 |
| 250 | [17.2] | | .972 | .951 | .921 | .895 | .871 | .850 | .830 | .812 | .794 | .778 | .763 | .749 | .735 | .722 | .710 | .698 |
| 300 | [20.7] | | .968 | .957 | .926 | .898 | .874 | .852 | .832 | .813 | .796 | .780 | .764 | .750 | .736 | .723 | .710 | .699 |
| 350 | [24.1] | | .968 | .963 | .930 | .902 | .877 | .854 | .834 | .815 | .797 | .781 | .765 | .750 | .736 | .723 | .711 | .699 |
| 400 | [27.6] | | | .963 | .935 | .906 | .880 | .857 | .836 | .816 | .798 | .782 | .766 | .751 | .737 | .724 | .712 | .700 |
| 450 | [31.0] | | | .961 | .940 | .909 | .883 | .859 | .838 | .818 | .800 | .783 | .767 | .752 | .738 | .725 | .712 | .700 |
| 500 | [34.5] | | | .961 | .946 | .914 | .886 | .862 | .840 | .820 | .801 | .784 | .768 | .753 | .739 | .725 | .713 | .701 |
| 550 | [37.9] | | | .962 | .952 | .918 | .889 | .864 | .842 | .822 | .803 | .785 | .769 | .754 | .740 | .726 | .713 | .701 |
| 600 | [41.4] | | | .964 | .958 | .922 | .892 | .867 | .844 | .823 | .804 | .787 | .770 | .755 | .740 | .727 | .714 | .702 |
| 650 | [44.8] | | | .968 | .958 | .927 | .896 | .869 | .846 | .825 | .806 | .788 | .771 | .756 | .741 | .728 | .715 | .702 |
| 700 | [48.2] | | | | .958 | .931 | .899 | .872 | .848 | .827 | .807 | .789 | .772 | .757 | .742 | .728 | .715 | .703 |
| 750 | [51.7] | | | | .958 | .936 | .903 | .875 | .850 | .828 | .809 | .790 | .774 | .758 | .743 | .729 | .716 | .703 |
| 800 | [55.1] | | | | .960 | .942 | .906 | .878 | .852 | .830 | .810 | .792 | .774 | .759 | .744 | .730 | .716 | .704 |
| 850 | [58.6] | | | | .962 | .947 | .910 | .880 | .855 | .832 | .812 | .793 | .776 | .760 | .744 | .730 | .717 | .704 |
| 900 | [62.1] | | | | .965 | .953 | .914 | .883 | .857 | .834 | .813 | .794 | .777 | .760 | .745 | .731 | .718 | .705 |
| 950 | [65.5] | | | | .969 | .958 | .918 | .886 | .860 | .836 | .815 | .796 | .778 | .761 | .746 | .732 | .718 | .705 |
| 1000 | [69.0] | | | | .974 | .959 | .923 | .890 | .862 | .838 | .816 | .797 | .779 | .762 | .747 | .732 | .719 | .706 |
| 1050 | [72.4] | | | | | .960 | .927 | .893 | .864 | .840 | .818 | .798 | .780 | .763 | .748 | .733 | .719 | .707 |
| 1100 | [75.9] | | | | | .962 | .931 | .896 | .867 | .842 | .820 | .800 | .781 | .764 | .749 | .734 | .720 | .707 |
| 1150 | [79.3] | | | | | .964 | .936 | .899 | .870 | .844 | .821 | .801 | .782 | .765 | .749 | .735 | .721 | .708 |
| 1200 | [82.7] | | | | | .966 | .941 | .903 | .872 | .846 | .823 | .802 | .784 | .766 | .750 | .735 | .721 | .708 |
| 1250 | [86.2] | | | | | .969 | .946 | .906 | .875 | .848 | .825 | .804 | .785 | .767 | .751 | .736 | .722 | .709 |
| 1300 | [89.6] | | | | | .973 | .952 | .910 | .878 | .850 | .826 | .805 | .786 | .768 | .752 | .737 | .723 | .709 |
| 1350 | [93.1] | | | | | .977 | .958 | .914 | .880 | .852 | .828 | .807 | .787 | .769 | .753 | .737 | .723 | .710 |
| 1400 | [96.5] | | | | | .982 | .963 | .918 | .883 | .854 | .830 | .808 | .788 | .770 | .754 | .738 | .724 | .710 |
| 1450 | [100.0] | | | | | .987 | .968 | .922 | .886 | .857 | .832 | .809 | .790 | .771 | .754 | .739 | .724 | .711 |
| 1500 | [103.4] | | | | | .993 | .970 | .926 | .889 | .859 | .833 | .811 | .791 | .772 | .755 | .740 | .725 | .711 |
| 1550 | [106.9] | | | | | | .972 | .930 | .892 | .861 | .835 | .812 | .792 | .773 | .756 | .740 | .726 | .712 |
| 1600 | [110.3] | | | | | | .973 | .934 | .894 | .863 | .836 | .813 | .792 | .774 | .756 | .740 | .726 | .712 |

# Section V

## Superheat Correction Factor ($K_{SH}$) for Superheated Steam

| Relieving Pressure psia | [bara] | Total Steam Temperature °F [°C] 400 [205] | 450 [232] | 500 [260] | 550 [288] | 600 [316] | 650 [343] | 700 [371] | 750 [399] | 800 [427] | 850 [455] | 900 [482] | 950 [510] | 1000 [538] | 1050 [566] | 1100 [593] | 1150 [621] | 1200 [649] |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1650 | [113.8] | | | | | | .973 | .936 | .895 | .863 | .836 | .812 | .791 | .772 | .755 | .739 | .724 | .710 |
| 1700 | [117.2] | | | | | | .973 | .938 | .895 | .863 | .835 | .811 | .790 | .771 | .754 | .738 | .723 | .709 |
| 1750 | [120.7] | | | | | | .974 | .940 | .896 | .862 | .835 | .810 | .789 | .770 | .752 | .736 | .721 | .707 |
| 1800 | [124.1] | | | | | | .975 | .942 | .897 | .862 | .834 | .810 | .788 | .768 | .751 | .735 | .720 | .705 |
| 1850 | [127.6] | | | | | | .976 | .944 | .897 | .862 | .833 | .809 | .787 | .767 | .749 | .733 | .718 | .704 |
| 1900 | [131.0] | | | | | | .977 | .946 | .898 | .862 | .832 | .807 | .785 | .766 | .748 | .731 | .716 | .702 |
| 1950 | [134.5] | | | | | | .979 | .949 | .898 | .861 | .832 | .806 | .784 | .764 | .746 | .729 | .714 | .700 |
| 2000 | [137.9] | | | | | | .982 | .952 | .899 | .861 | .831 | .805 | .782 | .762 | .744 | .728 | .712 | .698 |
| 2050 | [141.3] | | | | | | .985 | .954 | .899 | .860 | .830 | .804 | .781 | .761 | .742 | .726 | .710 | .696 |
| 2100 | [144.8] | | | | | | .988 | .956 | .900 | .860 | .828 | .802 | .779 | .759 | .740 | .724 | .708 | .694 |
| 2150 | [148.2] | | | | | | | .956 | .900 | .859 | .827 | .801 | .778 | .757 | .738 | .722 | .706 | .692 |
| 2200 | [151.7] | | | | | | | .955 | .901 | .859 | .826 | .799 | .776 | .755 | .736 | .720 | .704 | .690 |
| 2250 | [155.1] | | | | | | | .954 | .901 | .858 | .825 | .797 | .774 | .753 | .734 | .717 | .702 | .687 |
| 2300 | [158.6] | | | | | | | .953 | .901 | .857 | .823 | .795 | .772 | .751 | .732 | .715 | .699 | .685 |
| 2350 | [160.0] | | | | | | | .952 | .902 | .856 | .822 | .794 | .769 | .748 | .729 | .712 | .697 | .682 |
| 2400 | [165.5] | | | | | | | .952 | .902 | .855 | .820 | .791 | .767 | .746 | .727 | .710 | .694 | .679 |
| 2450 | [168.9] | | | | | | | .951 | .902 | .854 | .818 | .789 | .765 | .743 | .724 | .707 | .691 | .677 |
| 2500 | [172.4] | | | | | | | .951 | .902 | .852 | .816 | .787 | .762 | .740 | .721 | .704 | .688 | .674 |
| 2550 | [175.8] | | | | | | | .951 | .902 | .851 | .814 | .784 | .759 | .738 | .718 | .701 | .685 | .671 |
| 2600 | [179.3] | | | | | | | .951 | .903 | .849 | .812 | .782 | .756 | .735 | .715 | .698 | .682 | .664 |
| 2650 | [182.7] | | | | | | | .952 | .903 | .848 | .809 | .779 | .754 | .731 | .712 | .695 | .679 | .664 |
| 2700 | [186.2] | | | | | | | .952 | .903 | .846 | .807 | .776 | .750 | .728 | .708 | .691 | .675 | .661 |
| 2750 | [189.6] | | | | | | | .953 | .903 | .844 | .804 | .773 | .747 | .724 | .705 | .687 | .671 | .657 |
| 2800 | [193.1] | | | | | | | .956 | .903 | .842 | .801 | .769 | .743 | .721 | .701 | .684 | .668 | .653 |
| 2850 | [196.5] | | | | | | | .959 | .902 | .839 | .798 | .766 | .739 | .717 | .697 | .679 | .663 | .649 |
| 2900 | [200.0] | | | | | | | .963 | .902 | .836 | .794 | .762 | .735 | .713 | .693 | .675 | .659 | .645 |
| 2950 | [203.4] | | | | | | | | .902 | .834 | .790 | .758 | .731 | .708 | .688 | .671 | .655 | .640 |
| 3000 | [206.9] | | | | | | | | .901 | .831 | .786 | .753 | .726 | .704 | .684 | .666 | .650 | .635 |
| 3050 | [210.3] | | | | | | | | .899 | .827 | .782 | .749 | .722 | .699 | .679 | .661 | .645 | .630 |
| 3100 | [213.7] | | | | | | | | .896 | .823 | .777 | .744 | .716 | .693 | .673 | .656 | .640 | .625 |
| 3150 | [217.2] | | | | | | | | .894 | .819 | .772 | .738 | .711 | .688 | .668 | .650 | .634 | .620 |
| 3200 | [220.6] | | | | | | | | .889 | .815 | .767 | .733 | .705 | .682 | .662 | .644 | .628 | .614 |

# Section V

Figure 5-6 shows the typical capacity versus back pressure curve for a balanced direct spring valve. The loss of capacity is not as sudden nor as great as for a conventional valve, but loss still occurs at the higher levels of back pressure. The nomenclature "balance" refers primarily to set pressure. Unlike conventional direct spring valves, where the set pressure varies one-for-one with superimposed back pressure, the set pressure of a balanced valve does not change with back pressure.

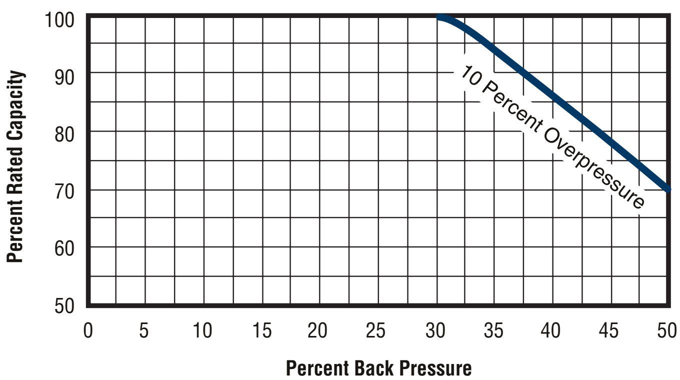



**Figure 5-6. Effects of Back Pressure on Relief Capacity of Typical Balanced Direct Spring PRV on Gas**

## 5.1.2 Sizing Example - Sonic Flow

What orifice area is required to protect a process vessel from overpressure due to an upstream control valve failure, if the maximum capacity of the control valve is 126,000 SCFM. The maximum allowable working pressure of the vessel is 1000 psig.

**Known Parameters**

| | |
|---|---|
| Required capacity | 126,000 scfm |
| MAWP | 1000 psig |
| Molecular weight of gas | 18.9 |
| Gas temperature | 60°F |
| Compressibility factor (assumed) | 1.00 |
| Gas constant | 344 |
| Cataloged PRV coefficient | 0.975 |
| Inlet piping pressure loss | 15% |
| Built-up back pressure | 150 psig |

**Solution**

We will use the MAWP as PRV set pressure.

The appropriate equation is:

$$A = \frac{V\sqrt{MTZ}}{6.32\ CKP_1\ K_b}$$

$$A = \frac{126{,}000\ \sqrt{(18.9)\ (460{+}60)\ (1.00)}}{6.32\ \ (344)\ (0.975)\ [1000 + 100)\ (.85) + 14.7]\ \text{-}\ 1.0}$$

$$A = 6.205 \text{ square inch}$$

The next larger orifice area is an API "P" orifice. Therefore, either a balanced bellows spring PRV or a pilot operated PRV in a 4P6 size would be the proper choices. Note the back pressure >10%, precluding the choice of a conventional PRV.

### Subsonic Flow

The second general category for vapor or gas sizing is when the pressure downstream of the valve exceeds the critical flowing pressure and the flow becomes subsonic. Under these conditions, the flow decreases with increasing back pressure even though the upstream pressure remains constant. The back pressure at which subsonic flow occurs varies with the flowing medium and can be calculated by the following formula:

$$P_2\ (\text{critical}) = P_1 \left[\frac{2}{k+1}\right]^{\frac{k}{k-1}} \qquad \text{(Equation 5)}$$

$$\text{where } k = \frac{C_P}{C_V}$$

The formulas used for calculating orifice areas for subsonic gas and vapor flow are:

$$A = \frac{V\sqrt{MTZ}}{4645\ K_{VC}\ P_1\ P} \qquad \text{(Equation 6)}$$

**or**

$$A = \frac{W\ \sqrt{TZ}}{735\ K_{VC}\ P_1\ F\ \sqrt{M}} \qquad \text{(Equation 7)}$$

## Section V

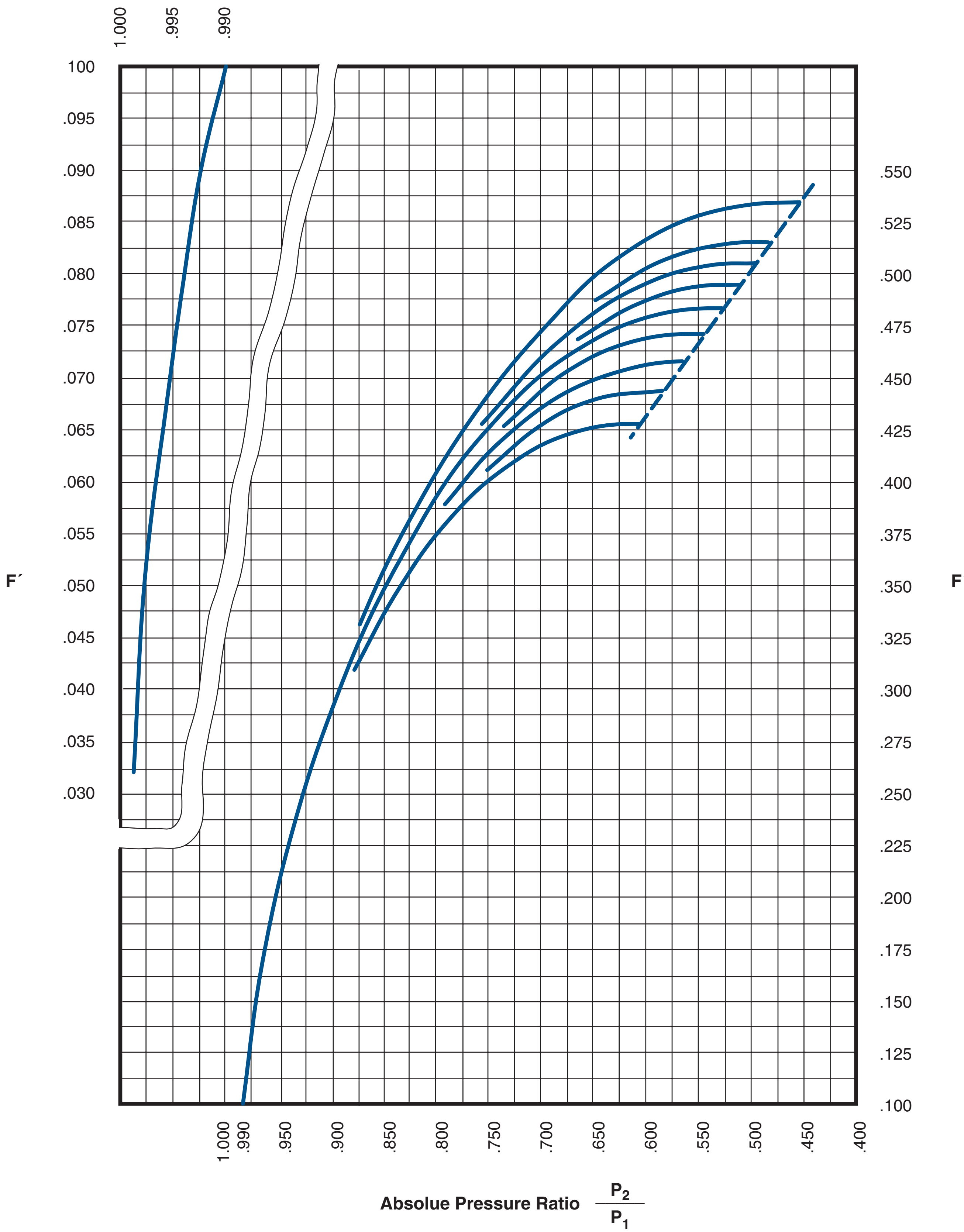



**Figure 5-7. Flow Correction Factor F**

# Section V

where F =

$$\sqrt{\frac{k}{k-1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{k}} - \left(\frac{P_2}{P_1}\right)^{\frac{k+1}{k}}\right]} \quad \text{(Equation 8)}$$

Refer to Figure 5-7 on page 36 for a plot of F' versus $P_2/P_1$.

All other terms and units are the same as previously noted except $K_{VC}$. In sonic flow $K_{VC}$ = K and K was defined as the coefficient of discharge of the nozzle and flow was independent of the pressure downstream of the nozzle. In subsonic flow, flow is dependent on downstream pressure; however, the controlling downstream pressure, $P_2'$ at the nozzle exit plane is not known (see Figure 5-8).

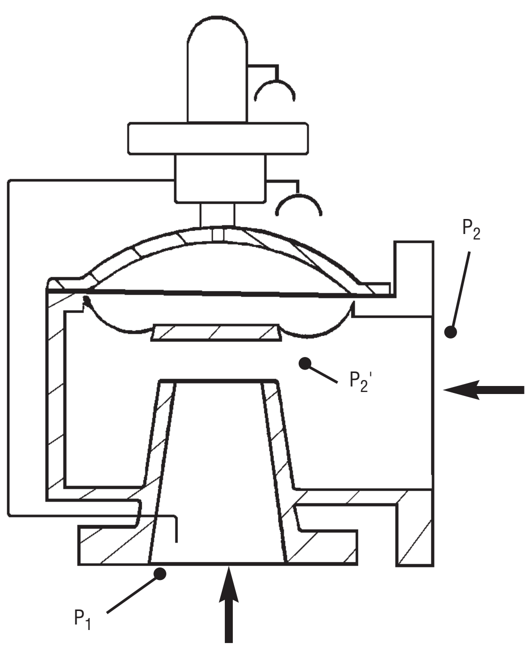



Figure 5-8. Subsonic Flow

For subsonic flow, $K_{VC}$ is defined at the valve coefficient. Unlike the sonic nozzle coefficient K which is constant regardless of the pressure ratio $P_2/P_1$ across the valve, $K_{VC}$ for subsonic flow is variable and dependent upon $P_2/P_1$. If $P_2'$ could be measured, the same constant value K for sonic flow could be used.

$K_{VC}$ is dependent on the valve body geometry as well as the nozzle geometry. It is experimentally determined by flow test. The valve manufacturer should be consulted for the value of $K_{VC}$. For the Anderson Greenwood Crosby Series 9300 PRV, $K_{VC}$ is shown in Figure 5-9.

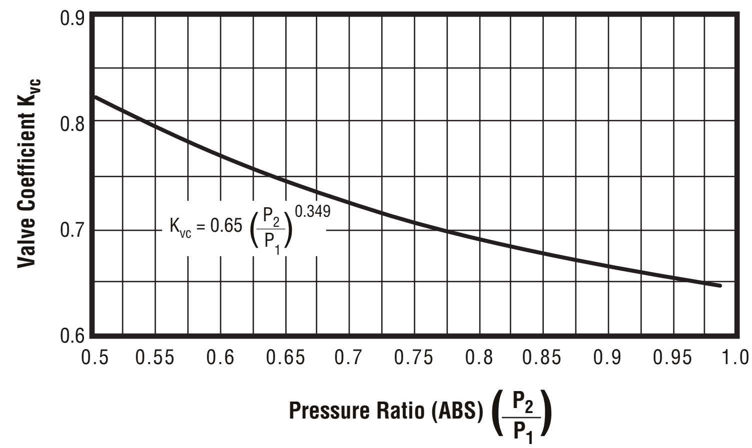



**Figure 5-9. $K_{VC}$ for Anderson Greenwood Crosby Series 9300 PRV**

The flow formulas given above may also be used for vacuum relief sizing using the same nomenclature as above but interchanging the locations of $P_1$ and $P_2$ with respect to the physical valve. Refer to Figure 5-10. The flow formula used will depend upon Pcf, the critical flow pressure. In vacuum relief from atmospheric pressure, critical flow occurs when the tank vacuum is 7.77 psia or less, based on a tank at sea level. The valve coefficient $K_{VC}$ may be different for vacuum relief and may be constant, independent of $P_2/P_1$. For the Anderson Greenwood Crosby Series 9300 pressure relief valve, $K_{VC}$ is a constant equal to 0.55.

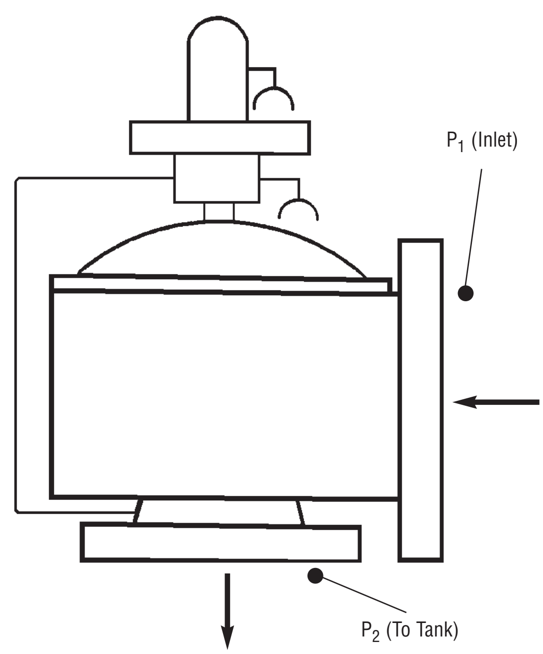



**Figure 5-10. Vacuum Relief**

# Section V

## 5.1.4 Sizing Example - Subsonic Flow

What orifice area would be required to protect a refrigerated LNG storage tank from overpressure due to vapor generation caused by failure of the boil off compressor? The calculated blow off rate is 21,000 SCFM. The maximum allowable working pressure of the vessel is 1.50 psig. Assume an Anderson Greenwood Type 9390 PRV will be used.

**Known Parameters**

| | |
|---|---|
| MAWP | 1.5 psig |
| Molecular weight of gas | 18.9 |
| Gas temperature | -260°F |
| Compressibility factor (assumed) | 1.0 |
| Ratio of specific heats | 1.27 |
| Pressure relief valve coefficient | 0.65 $[P_2/P_1]$ - 0.349 |
| Inlet piping pressure loss | 0% |
| Discharge piping | None |

**Solution:**

The appropriate equation is:

$$A = \frac{\sqrt{MTZ}}{4645\ K_{VC}\ P_1\ F}$$

$V = 21{,}000$ SCFM

$M = 18.9$

$T = (-260 + 460°F) = 200°R$

$Z = 1.00$

$P_1 = (1.50 + 0.15 + 14.7) = 16.35$ psia

$P_2 = 14.7$ psia

$K_{vc} = 0.676$ (from Figure 5-10 for $P_2/P_1 = 0.899$)

$k = 1.27$

$$F = \sqrt{\frac{k}{k-1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{k}} - \left(\frac{P_2}{P_1}\right)^{\frac{k+1}{k}}\right]}$$

$$F = \sqrt{\frac{1.27}{.27}\left[\left(\frac{14.7}{26.35}\right)^{\frac{2}{1.27}} - \left(\frac{14.7}{16.35}\right)^{\frac{2.27}{1.27}}\right]}$$

$$A = \frac{21{,}000\sqrt{(18.9)\ (200)\ (1.0)}}{4645\ (0.676)\ [16.35)\ (0.2984)}$$

$A = 84.28\ in^2$

The set pressure was selected to be the same as the MAWP since this would allow maximum utilization of the storage tank. The overpressure used was 10%. This value was selected since the storage tank was probably designed to meet the requirements of API Standard 620. Section 6.0 of this API standard specifies the maximum pressure should be limited to 110% of MAWP. This requirement is more stringent than the 3 psi overpressure permitted by the ASME code.

## 5.2 Sizing For Liquids

Sizing for liquid relief requires using a different formula than for vapors or gases. The formula used is as follows:

$$A = \frac{Q\sqrt{G}}{38\ K\ K_W\ K_V\ \sqrt{P_1 - P_2}} \qquad \text{(Equation 9)}$$

**Where:**

$A$ = Valve orifice area ($in^2$)

$Q$ = Flow rate (U.S. gal./min.)

$G$ = Specific gravity of liquid at flowing temperature referred to water = 1.00 at 70°F

$K$ = ASME coefficient of discharge on liquid

$K_w$ = Back pressure correction factor for direct spring operated valves due to reduced lift (All other valves $K_w$ = 1.00)

$K_V$ = Viscosity correction factor

$P_1$ = Inlet pressure during flow, set pressure – inlet pressure loss + allowable overpressure (psig)

$P_2$ = Back pressure during flow (psig)

The $K_w$ correction factor can be obtained from the valve manufacturer. The curve in Figure 5-11 is typical for a balanced direct spring valve in liquid service.

## Section V

In unbalanced valves, the set pressure will vary with back pressure. With balanced valves, the set pressure is not affected by back pressure. However, the valve lift is reduced at higher back pressures. The $K_w$ correction factor shown in Figure 5-11 should be used. In unbalanced direct spring operated valves, $K_w$ usually equals 1.00. This assumes that the superimposed back pressure is constant and the set pressure adjustment spring has been selected to compensate for this superimposed back pressure. For a variable superimposed back pressure, a conventional valve will also have a variable set pressure --- undesirable.

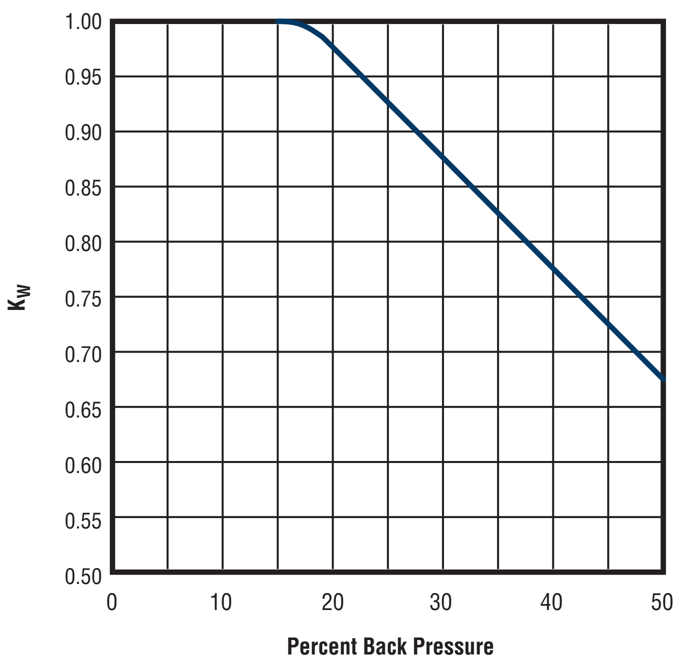



**Figure 5-11. $K_W$ for Balanced Bellows Spring Valves on Liquid**

For pilot operated safety relief valves, $K_w$ is always equal to 1.00 since lift is not affected by back pressure. Some pilot operated PRVs will display undesirable characteristics when used on a completely liquid filled system. The blowdown might change compared with gas service, the operating times might be too rapid (producing water hammer) or too slow, or the pilot could be unstable. It is important to check with the manufacturer of the particular valve to be used on liquid service to confirm its suitability for that service.

The $K_v$ correction factor is 1.00 for most applications. However, for service on viscous liquids, (above 1000 Saybolt Universal sec.) a preliminary valve orifice area must be calculated using $K_v = 1.00$. Then, using the next larger orifice area for the type of valve being sized, the Reynolds number must be calculated using one of the following formulas:

$$R = \frac{2{,}800\ G\ Q}{\mu \sqrt{A'}} \quad \text{(Equation 10)}$$

**Where:**

- R = Reynold's Number
- A´ = Next larger valve orifice area, in²
- G = Specific gravity of liquid
- Q = Required capacity U.S. GPM
- U = Viscosity at the flowing temperature, in Saybolt Universal seconds.
- μ = Absolute viscosity at flowing temperature, in centipoises

Knowing the Reynolds number, the viscosity correction factor $K_v$ can be determined from Figure 5-12. This curve is reproduced from API RP 520. Apply the $K_v$ factor to the original calculated preliminary orifice area. If the corrected area is less than the next larger orifice area "chosen" to calculate the Reynold's number, the "chosen" orifice is adequate.

$P_1$ is the inlet pressure during flow. The allowable overpressure for code applications is 10% of set. $P_2$ is the outlet pressure. If the valve discharges into a header or tailpipe, the back pressure developed during flow should be determined.

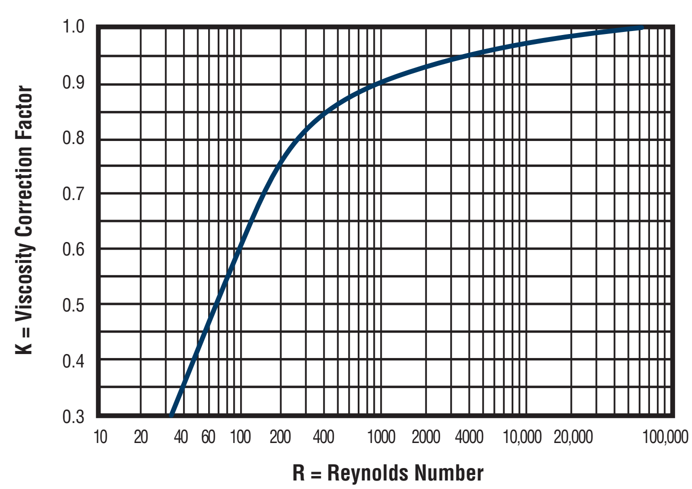



**Figure 5-12. Viscosity Correction Factor**

# Section V

## 5.2.2 Sizing Example - Liquid Flow

What orifice area is required to protect a lubrication oil system from overpressure if the pump capacity is 150 GPM? The maximum allowable working pressure of the system is 3400 psig. The pressure relief valve discharges into a closed header. Assume an ASME UV code stamped valve is used.

**Known Parameters:**

| | |
|---|---|
| MAWP | 1440 psig |
| Specific gravity of oil | 0.75 |
| PRV coefficient | 0.74 |
| Required flow rate | 150 US GPM |
| Built-up back pressure | 100 psig |
| Viscosity of oil | 2000 SSU |
| Inlet pressure losses | 3% |

*A full nozzle, spring PRV is requested.*

**Solution:**

The appropriate equation is:

$$A = \frac{Q\sqrt{G}}{38\ K\ K_W\ K_V\ \sqrt{P_1 - P_2}}$$

$$Q = 150$$

$$G = 0.75$$

$$K = 0.74$$

$$K_w = 1.00$$

$$P_1 = 1440 - 43 + 144 = 1541 \text{ psig}$$

$$P_2 = 100$$

$$\text{Assume } K_v = 1.00$$

$$A = \frac{150\sqrt{0.75}}{38\ (0.74)\ (1.00)\ (1.00)\ \sqrt{1541 - 100}}$$

$$A = 0.122 \text{ in}^2$$

To correct for viscosity, the next larger orifice available for the valve type chosen is used to calculate the Reynold's number. Assume the next larger orifice is 0.196 in$^2$. Therefore:

$$R = \frac{12{,}700\ Q}{U\sqrt{A'}}$$

$$R = \frac{12{,}700\ (150)}{2000\sqrt{0.196}} = 2151$$

$R = 2151$; therefore, $K_v = 0.94$ (from Figure 5-13)

$$\text{Corrected area } A = \frac{0.122}{0.94} = 0.130 \text{ in}^2$$

Since the corrected area of 0.130 in$^2$ is smaller than the next larger available orifice, the 0.196 in$^2$ orifice is adequate to handle the flow. Choose JLT - JOS PRV with "E" orifice.

## 5.3 Fire Sizing

The procedures used in fire sizing depend on the codes and engineering practices applied at each installation. Some of the engineering practices recommended for fire sizing are listed below:

- API RP 520, Part 1, Recommended Practices for the Design and Installation of Pressure-Relieving Systems in Refineries
- API Standard 2000, Venting Atmospheric and Low Pressure Storage Tanks
- API Standard 2510, Design of LP Gas Installations
- NFPA (National Fire Protection Association) Number 58, Storage and Handling Liquefied Petroleum Gases
- CGA (Compressed Gas Association), CGA S-1.3

Two types of tank conditions must be considered for fire sizing. The two types are liquid filled and gas filled tanks. The relieving capacity required for a liquid filled tank is always much greater than for a gas filled tank because of the liquid vaporization that occurs. Much of the liquid in liquid-filled tanks exposed to the direct or radiated heat of a fire will flash into vapor. The heat required to accomplish this will prevent the shell temperature of the tank from increasing rapidly. For gas or vapor filled tanks, the shell temperature increases rapidly. These temperatures can easily rise to a level where stress rupture can occur even though the pressure inside the tank does not exceed the maximum limit for fire conditions.

## 5.3.2 Fire Sizing per API RP520, Part I, Appendix D

For purposes of illustration, examples of vapor/gas-filled and of liquid-filled vessels will be reviewed.

## Section V

### 5.3.2.1. Fire Sizing of Vapor/Gas-Filled Vessels

The required orifice area for a PRV on vapor/gas-filled vessels exposed to external flames can be determined using the following formula:

$$A = \frac{F'\,A'}{\sqrt{P_1}}$$

**Where:**

A = Calculated PRV orifice area (in$^2$)

F' = Operating factor
**Note:** F' can be determined by a rather indeterminate relationship in API RP520, Appendix D.5.2.1. The recommended minimum value of F' is 0.01; when the value is unknown, F' = 0.045 should be used

A' = Exposed external surface area of the vessel, ft$^2$
**Note:** Any portion of the vessel higher than 25 ft. above grade is normally excluded, per Figure 5-13, except for a sphere for which it is to the mid-point.

$P_1$ = Relieving pressure at PRV inlet, psia = $P_{set} - P_{loss} + P_{O.P.} + P_{atmosphere}$

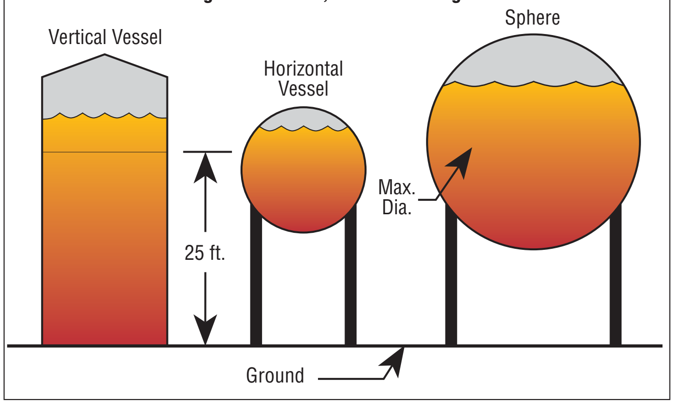



**Figure 5-13.**

Example: A vertical, hydrocarbon-filled vessel with 12 ft outside diameter, 63 ft height from grade, 3% inlet pressure loss to PRV, and MAOP of 285 psig

F' = 0.042 (conservation assumption)

A' = $\pi DL = \pi\,(12)\,25 = 942.5$ ft$^2$

$P_1$ = 285 – 9 + 60 + 14.7 = 350.7 psia

$$A = \frac{F'\,A'}{\sqrt{P_1}} = \frac{0.042\,(942.5)}{\sqrt{350.7}} = 2.11 \text{ in}^2$$

Select 3L4 (2.853 in$^2$ orifice) in either direct spring or pilot operated PRV configuration … or a 2FB3 (2.554 in$^2$) POPRV.

Notice that nowhere in the above fire sizing formula was the specific vapor/gas' characteristics considered.

### 5.3.2.2 Fire Sizing of Liquid-Filled Vessels

Where adequate drainage and fire-fighting equipment cannot be counted on, liquid-containing vessels exposed to external flames will require a PRV(s) sized in accordance with the following:

Determine the amount of heat absorbed through the wetted vessel inner wall and into the liquid.

$$Q = 34{,}500\ FA^{0.82}$$

**Where:**

Q = Total heat absorbed by the wetted surface of the vessels interior, BTU/hr.

F = Environmental factor (refer to following table 5-3)
**Note:** Though a vessel may be externally insulated, it is not uncommon to consider the vessel as bare, as the insulation may have burned off and/or been dislodged by firefighting water streams.

A = Total wetted surface, ft$^2$

**Table 5-3. Environment Factor, F**

| Type of Equipment | Factor F |
|---|---|
| Bare vessel | 1.0 |
| Insulated vessel (These arbitrary insulation conductance values are shown as examples and are in British thermal units per hour per square foot per degree Fahrenheit): | |
| 4 | 0.3 |
| 2 | 0.15 |
| 1 | 0.075 |
| 0.67 | 0.05 |
| 0.5 | 0.0376 |
| 0.4 | 0.03 |
| 0.33 | 0.026 |

# Section V

Next, the latent heat of vaporization of the contained liquid must be known. This value is normally stated in BTU/lb., with some typical examples following:

Methane . . . . . . 219
Propane . . . . . . 183
Ethylene . . . . . . 208
Benzene . . . . . . 169
Ammonia . . . . . 587
Oxygen . . . . . . . 92
Water . . . . . . . . 970
Butane . . . . . . 166
LPG . . . . . . . . . 167

To determine the mass flow rate required through the PRV in fire conditions:

$$W = \frac{Q}{V}$$

**Where:**

W = Mass flow, lbs./hr.

Q = Total heat absorbed, BTU/hr.

V = Latent heat of vaporization, BTU/lb.

Sizing for the PRV orifice area, the usual mass flow equation is used:

$$A = \frac{W\sqrt{TZ}}{C\,K\,P_1\sqrt{M}\,K_b}$$

"T" is the temperature at which liquid will flash into a vapor at set pressure. Lacking this information, 200°F is normally a conservative temperature to use. The other variables are taken from the usual vapor state.

## 5.4 Mixed-Phase Sizing per API RP 520, Part I, Appendix D

Until the past several years, API suggested treating each phase separately, with the total calculated orifice area being the total for all phases, calculated conventionally. Recognizing various pertinent factors, such as the amount of liquid "engaged" in the gas flow stream, alternative methodologies have been developed.

The Design Institute for Emergency Relief Systems (DIERS) has been involved in an extensive research program to develop methods for more accurately determining the PRD orifice area for multi-phase situations. The API 520, Part I, Appendix D now shows several sizing methods developed by DIERS (Design Institute for Emergency Relief Systems). Though the theory is essentially not yet proven, these complex methods are available. The DIERS Group is sponsored by the American Institute of Chemical Engineers (AIChE).

Worldwide the older "phase additive" method of sizing PRDs remains dominant.

## 5.5 Liquid Thermal Expansion Relief per API RP 521, Section 3

Where liquid-full equipment can be blocked in and continued heat input cannot be avoided --- as from the sun's radiation, a PRD must be provided. The rate of expansion is a function of the rate of heat input and the liquid's properties.

$$GPM = \frac{BH}{500\,G\,C}$$

**Where:**

GPM = Flow rate in U.S. gallons per minute

B = Cubical expansion coefficient per °F for the liquid at the expected temperature. Table 5-4 shows typical values for hydrocarbon liquids and water at 60°F.

H = Total heat transfer rate in BTU/hr. For heat exchangers, this can be taken as the maximum exchanger duty during operation.

G = Specific gravity

C = Specific heat of the trapped fluid, BTU/lb./°F Examples: Table 5-5

**Table 5-4. Typical Values of Cubical Expansion Coefficient for Hydrocarbon Liquids and Water at 60°F**

| Gravity of Liquid (°API) | B |
|---|---|
| 3 - 34.9 | 0.0004 |
| 35 - 50.9 | 0.0005 |
| 51 - 63.9 | 0.0006 |
| 64 - 78.9 | 0.0007 |
| 79 - 88.9 | 0.0008 |
| 89 - 93.9 | 0.00085 |
| 94 - 100 and lighter | 0.0009 |

# Section V

**Table 5-5. Typical Values of Specific heats @ 100°F**

| Liquid | C |
|---|---|
| Water | 4.18 |
| Ammonia | 2.18 |
| Methane | 2.27 |
| Propane | 1.75 |

## 5.6 Relieving Requirements for Sealed Storage Tanks Up to 15 psig per API 2000

$$CFH = 1107\ FA^{0.82}$$

**Where:**

CFH = $ft^3$ of free air at 60°F per hour

F = Environmental factor. Refer to Table 5-7 (Usually assume F = 1.0)

A = Exposed wetted surface, $ft^2$

Generally, Anderson Greenwood Crosby considers "F" = 1.0, assuming that any external insulation on an above-ground tank has either been burned off by the flames and/or has been blasted off by the fire control, water jet.

**Table 5-6. Wetted Tank Surface 2800 $ft^2$ or Less**

| A, $ft^2$ | SCFM Air | A, $ft^2$ | SCFM Air |
|---|---|---|---|
| 20 | 352 | 350 | 4800 |
| 30 | 527 | 400 | 5200 |
| 40 | 702 | 500 | 5900 |
| 50 | 878 | 600 | 6533 |
| 60 | 1053 | 700 | 7133 |
| 70 | 1228 | 800 | 7700 |
| 80 | 1403 | 900 | 8217 |
| 90 | 1580 | 1000 | 8733 |
| 100 | 1780 | 1200 | 9283 |
| 120 | 2100 | 1400 | 9783 |
| 140 | 2450 | 1600 | 10.233 |
| 160 | 2800 | 1800 | 10,650 |
| 180 | 3167 | 2000 | 11,033 |
| 200 | 3517 | 2400 | 11,733 |
| 250 | 2983 | 2800 | 12,367 |
| 300 | 4417 | over 2800 | use formula |

Use Air Capacity Tables in PRD Catalogs to select orifice area and/or device size.

# Section V

**Table 5-7. Environmental Factors**

| Tank Design/Configuration | Insulation Conductance (BTU/hr. ft² °F) | Insulation Thickness (inches) | *F* Factor |
|---|---|---|---|
| Bare metal tank | — | 0 | 1.0 |
| Insulated tanks[a] | 4.0 | 1 | 0.3[b] |
| " " | 2.0 | 2 | 0.15[b] |
| " " | 1.0 | 4 | 0.075[b] |
| " " | 0.67 | 6 | 0.05 |
| " " | 0.5 | 8 | 0.0375[b] |
| " " | 0.4 | 10 | 0.03[b] |
| " " | 0.33 | 12 | 0.025[b] |
| Concrete tank or fireproofing | — | — | (see note c) |
| Water-application facilities[d] | — | — | 1.0 |
| Depressuring and emptying facilities[e] | — | — | 1.0 |
| Underground storage | — | — | 0 |
| Earth-covered storage above grade | — | — | 0.03 |
| Impoundment away from tank[f] | — | — | 0.5 |

Notes:

a. The insulation shall resist dislodgment by fire-fighting equipment, shall be noncombustible, and shall not decompose at temperatures up to 1000°F [537.8°C). The user is responsible to determine if the insulation will resist dislodgment by the available fire-fighting equipment. If the insulation does not meet these criteria, no credit for insulation shall be taken. The conductance values are based on insulation with a thermal conductivity of 4 BTU per hour per square foot per °F per inch of thickness (9 Watts per square meter per °C per centimeter of thickness). The user is responsible for determining the actual conductance value of the insulation used. The conservative value of 4 BTU per hour per square foot per °F per inch of thickness (9 Watts per square meter per °C per centimeter of thickness) for the thermal conductivity is used.

b. These *F* factors are based on the thermal conductance values shown and a temperature differential of 1600°F [888.9°C] when using a heat input value of 21,000 BTU per hour per square foot (66,200 Watts per square meter) in accordance with the conditions assumed in API Recommended Practice 521. When these conditions do not exist, engineering judgement should be used to select a different *F* factor or to provide other means for protecting the tank from fire exposure.

c. Use the *F* factor for an equivalent conductance value of insulation.

d. Under ideal conditions, water films covering the metal surfaces can absorb most incident radiation. The reliability of water application depends on many factors. Freezing weather, high winds, clogged systems, undependable water supply, and tank surface conditions can prevent uniform water coverage. Because of these uncertainties, no reduction in environmental factors is recommended; however, as stated previously, properly applied water can be very effective.

e. Depressuring devices may be used, but no credit shall be allowed in sizing the venting device for fire exposure.

f. The following conditions must be met: A slope of not less than 1% away from the tank shall be provided for at least 50 feet [15 meters] toward the impounding area; the impounding area shall have a capacity that is not less than the capacity of the largest tank that can drain into it; the drainage system routes from other tanks to their impounding areas shall not seriously expose the tank; and the impounding area for the tank as well as the impounding areas for the other tanks (whether remote or with dikes around the other tanks) shall be located so that when the area is filled to capacity, its liquid level is no closer than 50 feet [15 meters] to the tank.

# Section VI

## 6.0 Pressure Relief Valve Installation

Pressure relief valve installation requires careful consideration of the following areas:

– inlet piping

– discharge piping

– preinstallation handling and testing

Marginal installation efforts in any of these areas can render the pressure relief valve inoperable or at least severely restrict its ability to perform properly and cause high maintenance costs.

## 6.1 Inlet Piping

The proper design of inlet piping to pressure relief valves is extremely important. Too often, PRVs are added to an installation at the most physically convenient location with little regard to flow considerations.

Pressure loss during flow in a pipe always occurs. Depending upon the size, geometry, and inside surface condition of the pipe, the pressure loss may be large (10, 20 or even 30%) or small (less than 3%). API RP 520, Part 2, and ASME Section VIII (non-mandatory) recommend a maximum inlet pressure loss to a PRV of 3%. This pressure loss is the sum total of the loss due to penetration configuration at the vessel or pipe, inlet pipe loss and, when a block valve is used, the loss through it. The losses should be calculated using the maximum actual rated flow through the pressure relief valve (using "$K_D$", not "K").

A maximum inlet loss of 3% is a commendable recommendation, but sometimes difficult to achieve, and is not mandatory in ASME Section VIII. If it cannot be achieved, then the effects of excessive inlet pressure loss, such as rapid cycling or chatter, should be known. In addition, on pilot operated valves, rapid or "short" cycling can occur when the pilot valve pressure sensing line is connected to the main valve inlet (integral pressure sensing). Each of these conditions results in a loss of valve capacity and premature valve wear or failure due to valve damage.

Pilot operated valves can tolerate higher inlet losses when the pilot senses the system pressure at a point not affected by inlet pipe pressure drop (remote pressure sensing) or is of the modulating type. However, even though the valve operates satisfactorily, reduced capacity will occur because of inlet pipe pressure losses. It is important that the sizing procedure consider the reduced, flowing inlet pressure ($P_1$) when the required orifice area "A" is calculated. A properly modulating pilot operated valve, remotely or integrally sensed, will also ensure PRV stability even with high inlet pressure losses.

### 6.1.2 Chatter and Rapid Cycling

Rapid valve cycling or chatter with direct spring operated valves occurs when the pressure at the valve inlet decreases at the start of relief valve flow. In Figure 6-1, a schematic of system pressures is shown before and during flow. Before flow begins, the pressure is the same in the tank and at the valve inlet. During flow, the pressure at the valve inlet is reduced due to the pressure loss in the inlet piping to the valve. Under these conditions the valve will cycle at a rapid rate rather than stay open until the system pressure is reduced to the desired blowdown pressure.

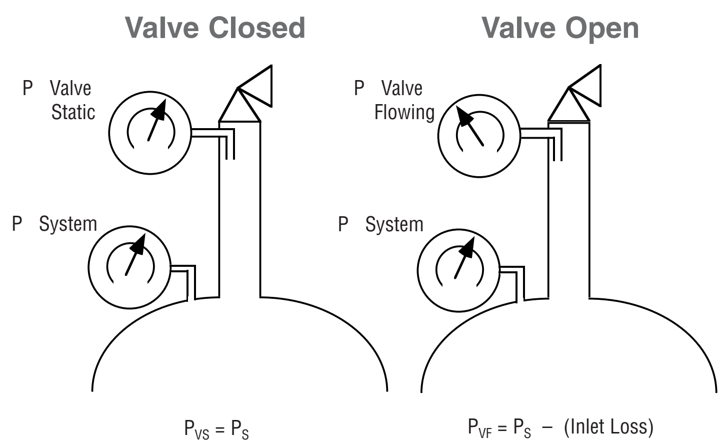



**Figure 6-1. System Pressure Before and During Flow**

The valve responds only to the pressure at its inlet. When that pressure decreases during flow to a value below the valve reseat point, the valve will close. However, as soon as the flow stops, the inlet pipe pressure loss becomes zero and the pressure at the valve inlet rises to tank pressure once again. If the tank pressure is still equal to or greater than the relief valve set pressure, the valve will open and close again. Rapid cycling or chatter reduces capacity and is destructive to the valve internals. In addition, all moving parts in the valve are subjected to excessive wear and possible seizure due to galling.

### 6.1.3 Remote Sensing Lines

On pilot operated pressure relief valves with the pilot pressure sensing line connected to a pitot tube at the main valve inlet, rapid cycling or chatter can also occur for the same reasons described above. The pressure at the valve inlet is substantially reduced during flow. Refer to Figure 6-2. With the pilot sense line located at the main valve inlet, the pilot responds to the pressure at the location and therefore closes the main valve --- sometimes prematurely. If the pressure loss during flow is small (less than the reseat pressure setting of the pilot valve), the main valve might or might not partially close depending on type of pilot design used.

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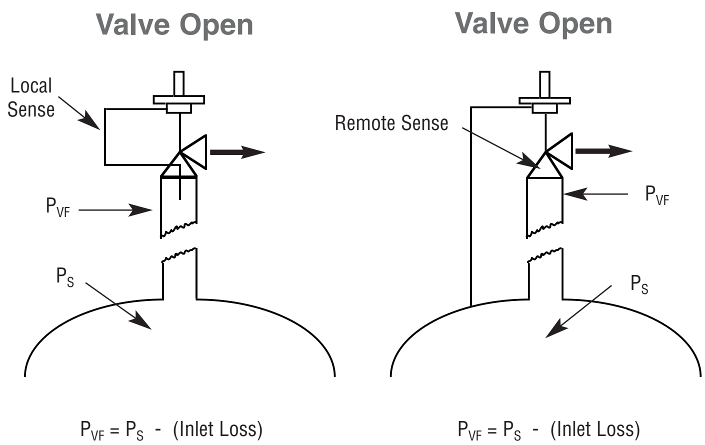



**Figure 6-1. System Pressure Before and During Flow**

In actual application, because of the longer response time of a pilot operated valve (compared with a direct spring operated valve) the main valve might only go into partial lift. The inlet pipe pressure loss occurs before the pilot has sufficiently vented the dome volume for full piston travel. As with direct spring operated valves, capacity is reduced.

## 6.1.4 Resonant Chatter

Resonant chatter can occur with pressure relief valves when the natural acoustical frequency of the inlet riser approximates the natural mechanical frequency of the PRV's basic moving parts (the piston assembly in pilot operated valves; disc holder, disc, and lower spring washer in direct spring valves).

Resonant chatter is more likely to occur when the following conditions are present:

– higher set pressure

– larger valve size

Unlike the rapid cycling noted in the previous section, resonant chatter is uncontrolled. Once started, resonant chatter cannot be stopped unless the pressure is removed from the valve inlet.

In actual application, the valve can self-destruct before a shutdown can take place. This is because of the magnitude of the forces involved in a resonant mode. Resonant chatter is extremely destructive and can result in massive PRV failure and personnel and equipment danger.

## 6.1.5 Preferred Piping Design

On new installations in the design stage, try to keep the equivalent L/D (pipeline length to pipeline diameter) ratio of the inlet piping to the relief valve inlet 5 or less (see Section 6.1.6). The significant word with respect to the L/D ratio is "equivalent". Various pipe fittings and tank penetrations have rather large L/D ratios. Figure 6-3 shows some common fittings and tank penetrations and their equivalent L/D ratios. As can be observed, only the straight inlet pipe with a concentric reducer produces the recommended L/D ratio of 5 or less. If these guidelines are not followed, rapid cycling or chatter can occur.

## 6.1.6 Minimizing Inlet Pressure Losses

The inlet piping design to PRVs is very important. Pressure losses occur in all piping during flow. If these pressure losses are high enough, PRV rapid cycling or chatter may occur, substantially reducing the relieving capacity of the valve. If the valve does not rapid cycle or chatter, the relief capacity will still be reduced since relief capacity is proportional to inlet pressure.

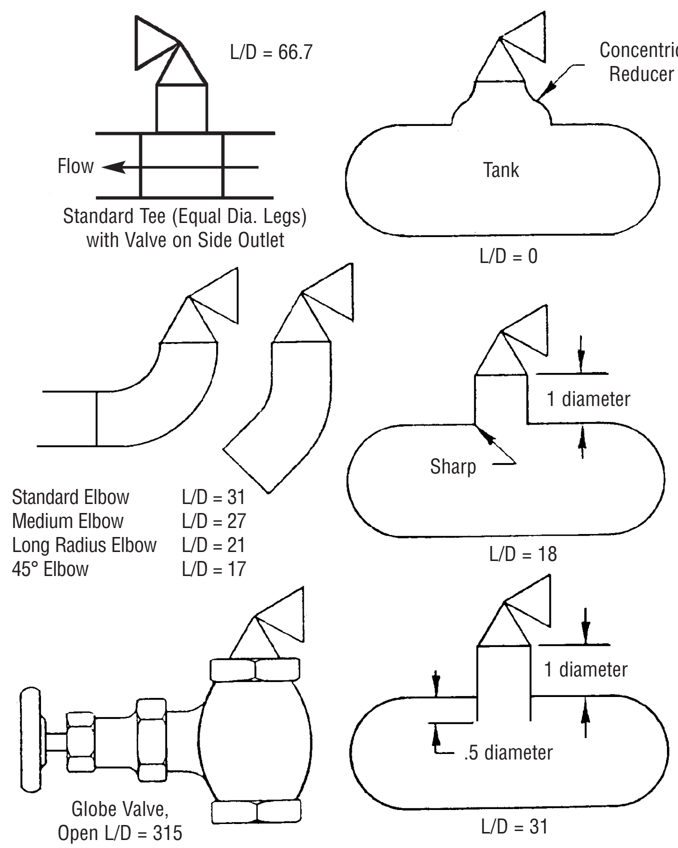



**Figure 6-3. Equivalent Lengths of Various Fittings (Crane "Resistance of Valves and Fittings to the Flow of Fluids")**

# Section VI

To minimize inlet pressure losses, the equivalent L/D ratio (pipe length to pipe diameter) of the inlet piping to the relief valve should not be greater than 5. If this ratio cannot be obtained because of the piping geometry or fittings, then piping and fittings one pipe size larger than the relief valve inlet should be used.

The above guidelines are conservative. To analytically determine the actual piping losses, refer to Anderson Greenwood Report No. 02-0175-128 “Determination of Flow Losses in Inlet and Discharge Headers Associated with Safety Relief Valves”.

Some recommended tank penetrations and valve inlet piping designs are shown in Figure 6-4. Tee fittings and multiple elbows should be avoided.

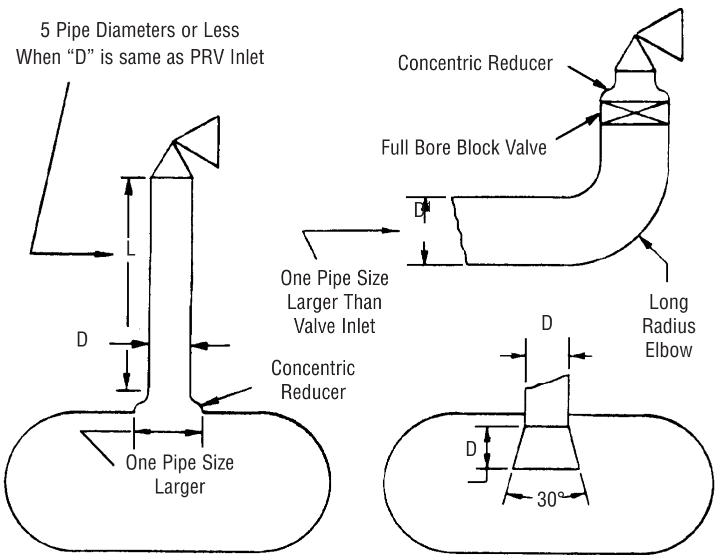



**Figure 6-4. Recommended Tank Penetrations and Inlet Piping Designs**

All Anderson Greenwood Crosby pilot operated pressure relief valves that could be subject to resonant chatter have design provisions built in to prevent such chatter from occurring. However, even though a valve does not chatter, there is no assurance that the valve is relieving at its design capacity. Flow capacity is proportional to inlet pressure, and inlet piping pressure loss results in reduced pressure at the valve inlet. To analytically determine the pressure loss in inlet piping, refer to Anderson Greenwood Report No. 2-0175-128 under the “Flow Losses” tab.

On existing installations, the possibility of corrective action is somewhat limited. For direct spring operated valves, increasing the blowdown setting can minimize or eliminate rapid cycling, if the blowdown pressure can be adjusted to a value below the inlet flowing pressure. Essentially, the blowdown setting (percentage) must exceed the inlet pressure losses (percentage). Example: 5% inlet pressure loss with 8% blowdown setting = 3% installed valve blowdown.

Unfortunately, it is not economically feasible to precisely adjust the blowdown of many spring-loaded pressure relief valves beyond 10% or so. Most manufacturers’ test set-ups are insufficient to accurately set blowdown, so they usually use empirical methods.

On pilot operated valves, remote pressure sensing can be used. Remote pilot sensing, as shown in Figure 6-2, can prevent rapid cycling or false blowdown, as can pilot operated PRVs having proper modulating action.

## 6.1.7 Typical Inlet Piping Arrangements

The most desirable inlet pipe arrangement is as follows:

– the inlet pipe is the same size or larger than the pressure relief device inlet

– the inlet pipe length does not exceed the face-to-face dimension of a standard tee of the proper pressure class.

Three other inlet piping arrangements are illustrated in Figure 6-5a-c.

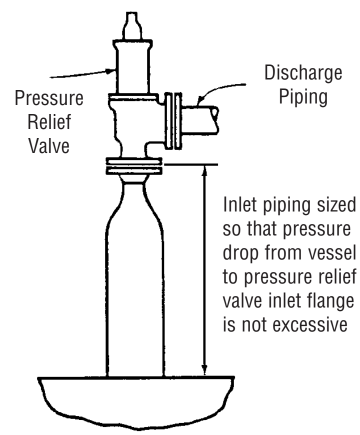



**Figure 6-5a**

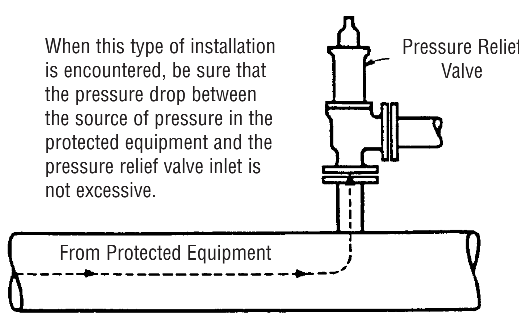
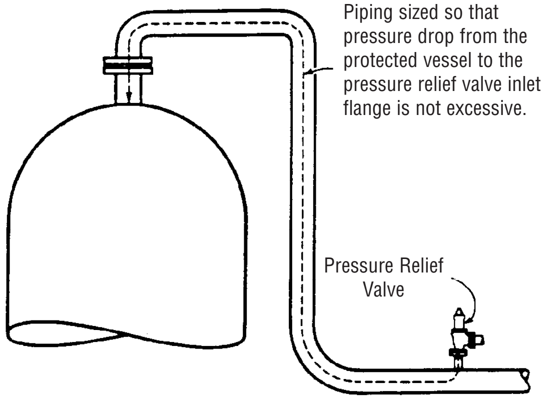



**Figure 6-5b**

# Section VI




**Figures 6-5c.**

## 6.2 Discharge Piping

Discharge piping is much more critical for direct spring operated valves than for pilot operated valves. As with inlet piping, pressure losses occur in discharge headers with large equivalent L/D ratios. As noted in Section V , excessive back pressure can reduce the lift of a direct spring operated valve, and enough back pressure can cause the valve to reclose and/or chatter. Refer to Figure 6-6.

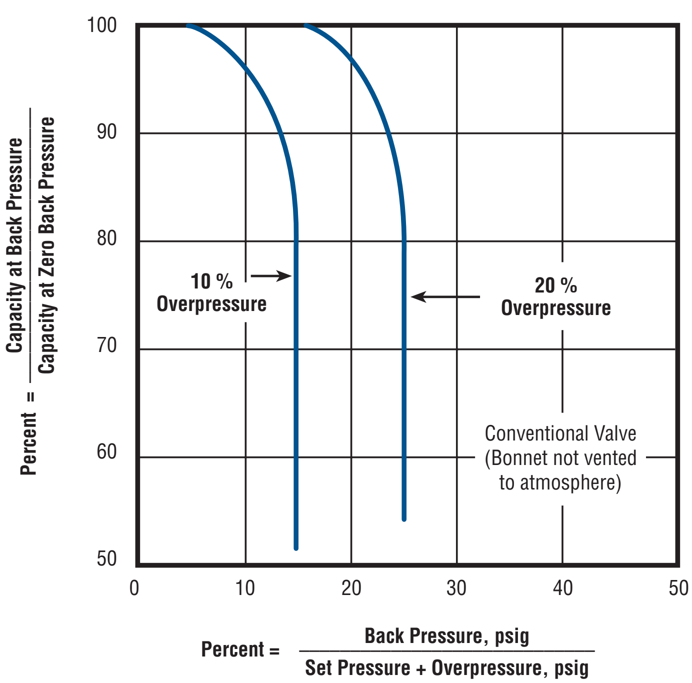



**Figure 6-6. Typical Effects of Variable Back Pressure on Capacity of Conventional Pressure Relief Valves**

As soon as the valve closes, the back pressure in the discharge header decreases and the valve opens again. Rapid cycling or chatter can then occur.

The operation of pilot operated PRVs is not affected by back pressure if the pilot is either vented to the atmosphere or is internally balanced for back pressure. However, if the discharge pressure can ever exceed the inlet pressure (such as could occur when multiple valves discharge into a common header), a backflow preventer should be used.

The valve relieving capacity for either direct or pilot operated PRVs can be affected by back pressure. This can happen if the flowing pressure, with respect to the discharge pressure, is critical (subsonic flow). To analytically determine the pressure loss in a discharge header, refer to Anderson Greenwood Report No. 02-0175-128, under “Flow Losses” tab.

Balanced bellows valves (direct spring operated) have limitations on maximum permissible back pressure due to the collapse pressure rating of the bellows element. Manufacturer literature should be consulted in every case. Keep in mind that if the bellows valve is used for systems with superimposed back pressure, the additional built-up back pressure under relieving conditions must be considered to calculate maximum back pressure.

Balanced bellows pressure relief valves have an open bonnet design. There is a vent port that must be left open during service. Otherwise a bellows failure or leak of any kind will pressurize the bonnet and the set pressure will be equal to the spring setting (cold differential test pressure) plus back pressure. In other words, the bellows valve becomes unbalanced.

As mentioned in the sizing portion of this seminar, the back pressure effect on balanced bellows must be taken into consideration (just as for conventional valves). This is because increasing back pressure tends to make the bellows longer due to its action on the convolutions. Being restrained at the upper end, the stiffened bellows can restrict lift and thereby effectively reduce lift and capacity. Consult the manufacturer in each case.

Good operating and flow capacity performance of pressure relief valves can be achieved by using the following discharge piping practices:

1. Discharge piping must be at least the same size as the valve outlet connection and may have to be increased to a larger size.
2. Flow direction changes should be minimized. When necessary, use long radius elbows and gradual transitions.
3. If the valve has a drain port on the outlet side, it should be vented to a safe area. Avoid low spots in discharge piping or drain them. It is preferable to pitch piping away from the valve outlet to avoid a liquid trap at the valve outlet.

## Section VI

4. Proper pipe supports must be used to overcome the following problems:
   a. Thermal effects
   b. Static loads due to pipe weight
   c. Stresses due to discharge reactive thrust forces.

### 6.3.1 Reactive Force for GASES

On larger orifice, higher pressure valves, the reactive forces during valve relief can be substantial. External bracing might be required. Refer to Figure 6-7.

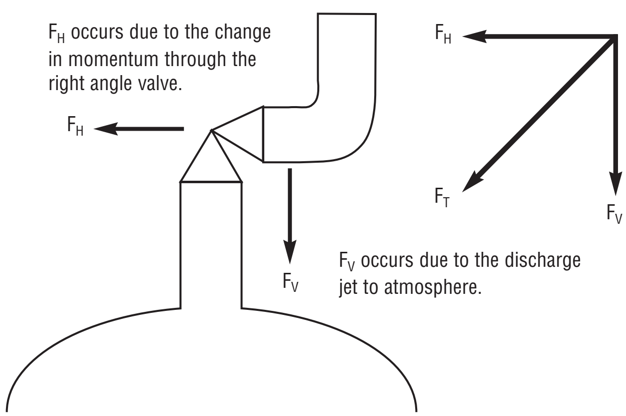



**Figure 6-7. Reactive Force**

API RP 520, Part 2 gives the following formula for calculating this force.

$$F_T = \frac{W\sqrt{\frac{kT}{(k+1)\,M}}}{366} + (A_o \times P_2) = F_H + F_V \qquad (1)$$

Where:

- $F_T$ = Reactive force at the point of discharge to the atmosphere (lbs.)
- $W$ = Flow of any gas or vapor (lb./hr.)
- $k$ = Ratio of specific heats ($C_p/C_v$)
- $T$ = Inlet temperature, absolute (°F + 460)
- $M$ = Molecular weight of flowing media
- $A_o$ = Area of the outlet at the point of discharge (in²)
- $P_2$ = Static pressure at the point of discharge (psig)

Example: Anderson Greenwood Crosby 24312P46/S1 (pop action)

"P" orifice having an API orifice area of 6.38 in² area

Set pressure 1000 psig

10% overpressure allowed

Fluid: Hydrocarbon vapor @ 75°F

$M = 17.4$ (SG = 0.60), $Z = 0.86$

$C = 344$ ($k = 1.27$)

$K = 0.975$

PRV discharge is to atmosphere through an elbow and vertical, Schedule 40 tail-pipe at the valve outlet.

Solve for discharge reaction force.

$$W = \frac{A\,C\,K_D\,P\sqrt{M}}{\sqrt{TZ}}$$

$$W = \frac{K}{0.90} = \frac{0.975}{0.90} = 1.083$$

(note equivalent, not ASME)

$$W = \frac{6.38\,(344)\,1.083\,[(1000 \times 1.10)+14.7]\sqrt{17.4}}{\sqrt{(460+75)\,0.86}}$$

$= 515{,}244$ lb./hr.

$A_0$ of 6-inch Schedule 40 discharge pipe = 28.89 in²

$$P_2 = \left[\frac{0.00245W}{(d_2)^2}\sqrt{\frac{T_1}{kM}}\right]$$

$d_2$ = I.D. of 6-inch discharge tail-pipe = 6.065-inch

$$P_2 = \left[\frac{0.00245\,(515{,}244)}{(6.065)^2}\sqrt{\frac{(460+75)}{1.27\,(17.4)}}\right]$$

$- 14.7 = 154$ psig

$$F_T = \frac{W\sqrt{\frac{kT}{(k+1)\,M}}}{366} + A_o\,P_2$$

## Section VI

$$F_T = \frac{515{,}244 \sqrt{\dfrac{1.27\ (460 + 75)}{(1.27 + 1)\ 17.4}}}{366}$$

$$+ 28.89\ (154) = 5839 + 4449$$

$$= 10{,}288 \text{ lbs. force total}$$

If bracing is not feasible, a dual outlet valve (available in some pilot operated pressure relief valve sizes) can be used. The reactive forces from the two outlets are equal and opposite. If the outlets are redirected, these forces can still impose bending loads that must be dealt with in some manner.

### 6.3.2 Reactive Force for Liquids

The reactive force for a pressure relief valve flowing liquid is as follows, as sourced from Mark's "Standard Handbook for Mechanical Engineers":

$$F_T = \frac{SG \times (Q)^2}{722A}$$

Where:

F = Reactive force, lb.

SG = Specific gravity

Q = Flow rate, U.S. gallons/minute

A = Area of outlet pipe, inches

Example: Anderson Greenwood Crosby Model 44312P46/S1 (modulating action)

Set pressure 1000 psig

10% overpressure allowed

Actual flow 6054 U.S. GPM water (as specified on PRV data sheet)

A = 28.89 in$^2$ (6-inch scheduled 40 discharge pipe)

$$F = \frac{1.0 \times (6054)^2}{722\ (28.89)} = 1757 \text{ lbs. force}$$

## 6.4 Pre-Installation Handling and Testing

Proper pre-installation handling and testing can help ensure pressure relief valves and their associated piping remain clean, free of damage, and operational. API RP 520, Part II, Section 10, contains sound recommendations.

The sub-sections below discuss the following pre-installation handling and testing areas:

– storage and handling of pressure relief valves

– inspection of valves before installation

– inspection and cleaning of systems before installation

– additional installation considerations

– proximity to other equipment

– pre-start-up testing

– hydrostatic testing

### 6.4.1 Storage and Handling of Pressure Relief Valves

Cleanliness is essential to the satisfactory operation and tightness of a pressure relief valve. Valve contamination can lead to internal damage, misalignment, and/or poor seat tightness.

The following precautions should be taken to ensure that no dirt or other foreign material contaminates the valve:

1. Handle valves with extreme care at all times.
2. Close off valves at the inlet and outlet flanges if they are not installed immediately after receipt from the manufacturer or maintenance repair.
3. Keep the valve inlet absolutely clean.
4. If valves must be stored, store them indoors where dirt and other foreign material are minimal.
5. Do not throw valves in a pile or place them on the bare ground while they are waiting to be installed.
6. Do not subject the valves to heavy shocks.

### 6.4.2 Inspection of Valves Before Installation

All pressure relief valves should have a thorough visual inspection for condition before installation. The manufacturers' maintenance manuals should be consulted for details relative to the specific valve. Be sure to remove all protective material on the valve flanges and any extraneous materials inside the valve body or nozzle. Foreign materials clinging to the inlet side of the PRV will be blown across the seating surfaces when the valve is operated. Some of these materials may damage the seats or can be trapped between the seats and cause leakage.

# Section VI

### 6.4.3 Inspection and Cleaning of Systems Before Installation

Because foreign materials passing into and through a pressure relief valve are damaging, the systems on which the valve is tested and finally installed must also be inspected and cleaned. New systems are especially prone to contain weld beads, weld rod stubs, pipe scale, and other foreign objects inadvertently trapped during construction. These contaminants can badly damage the seating surfaces the first few times the valve opens. The system should be purged thoroughly before the PRV is installed.

The valve should be isolated during hydrostatic pressure testing of the system, either by blanking or closing a stop valve. If gagging is used, extreme caution must be exercised to avoid damaging the valve with an overtightened gag and to ensure that the gag is removed after use.

### 6.4.4 Additional Installation Considerations

Note the following additional installation considerations:

1. Do not install a pressure relief valve at the end of a long horizontal line that does not normally have flow. The horizontal line becomes an ideal trash collector or liquid trap. Upon relieving, this foreign matter can interfere with operation or increase the maintenance requirements of the valve.
2. The pressure relief valve and pilot should always be installed vertically upright, as recommended in ASME Section VIII, UA359 and API RP 520, Paragraph 6.4. Any other orientation can adversely affect proper PRV operation.
3. Improper piping design can set up stresses due to thermal or mechanical reaction effects. These stresses can impair the function of a pressure relief valve, causing leakage or binding of moving parts. This is important on inlet and outlet piping, and is especially true of large direct spring operated valves.

### 6.4.5 Proximity To Other Equipment

Pressure relief valves should be located a sufficient distance from devices that can create turbulence in inlet piping. The turbulence can induce chatter, particularly in direct spring operated valves. The increased inlet pressure loss, due to separating the relieving device from the pressure source, must also be considered.

Note the following proximity considerations:

1. Pressure reducing stations - Turbulence can be expected downstream of these stations due to the regulating device and associated valves and fittings. Though the turbulence cannot be readily evaluated, care should be exercised in locating pressure relief valves.
2. Orifice plates and flow nozzles - Proximity to these devices can cause similarly adverse operation.
3. Pulsating compressor discharge - Although pulsation dampers will protect other types of equipment, “pressure spikes” can cause very severe problems with direct spring operated PRVs. If the valves are metal seated, this can cause progressively increased seat leakage, nuisance relief cycles, and premature wear-out. Pilot operated valves are not as susceptible to the effects of “pressure spikes” because of high seat loading. Pilot supply passages tend to dampen the spikes to the pilot. Pressure Spike Snubbers are uniquely available for Anderson Greenwood Crosby pilot operated valves.

### 6.4.6 Pre-Start Up Testing

On pilot operated PRVs with field test connections, the set pressure of the installed valve can be checked easily and accurately. The valve manufacturer should be consulted for this procedure.

On direct spring operated valves, the “pop” pressure can be checked with a suitable pressure source. However, if the volumetric capacity immediately upstream is insufficient, an accurate set pressure test may be impossible, particularly with metal-seated, spring valves.

### 6.4.7 Hydrostatic Testing

Pressure relief valves should be isolated during hydrostatic testing to keep water and particulate debris out of the valve. On pilot operated valves, it is important to keep water out of the pressure sensing lines and pilot. This is especially important where ambient temperatures may drop below freezing or where the set pressures of the valve are very low (below 2 to 3 psig). Water in pressure sense lines can effectively increase the pressure setting of low pressure, pilot operated valves.

If the valve is not isolated and a test gag is used, again, ensure the gag is removed prior to system start up.

# Section VI

## 6.5 Seat Tightness and Leakage

Seat tightness and leakage standards for direct spring and pilot operated pressure relief valves with metal-to-metal and soft seats are specified in API RP 527 and now mandatory in ASME Section VIII.

Leakage rates for these type valves are defined in these documents for set pressures up to 6,000 psig. Leakage rates can be supplied for valves requiring set pressures above 6,000 psig, if they are specified on your order or inquiry.

API/ASME seat tightness and leakage standards for soft-seated valves are no bubbles for one minute at 90% of set pressure. However, expect 0 bubbles in 1 minute at 95% of set pressure in higher pressure valves.

### 6.5.1 Seat Tightness Test Apparatus

Figure 6-8 illustrates a typical test arrangement for determining the seat tightness of a pressure relief valve. Leakage measurement should be made using 5/16-inch OD tubing with a 0.035-inch wall. The tube end should be cut square and smooth, and be set parallel to and 1/2-inch below the surface of the water, as specified in API RP527.

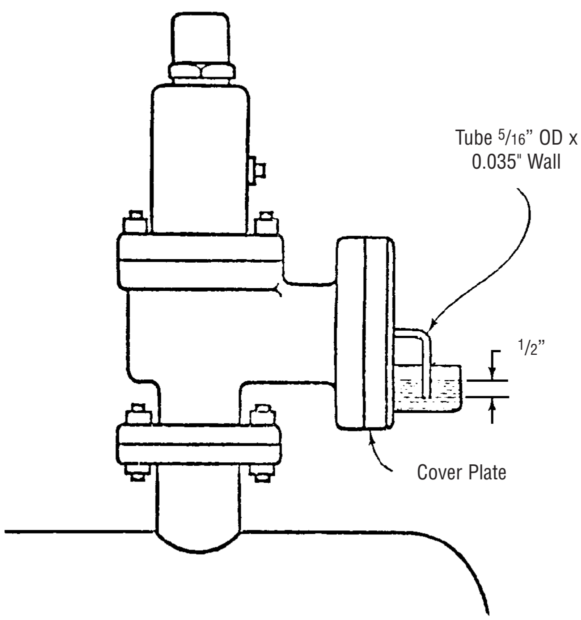



**Figure 6-8. Air Receiver**

### 6.5.2 Leakage Rate Determination

The following steps should be performed to determine the leakage rate of safety relief valves:

1. Mount the valve vertically, as shown in Figure 6-8.
2. Hold the pressure at the pressure relief valve inlet at 90% of set pressure immediately after popping.
3. Use air at approximately ambient temperature as the pressure medium for gas/vapor valves.

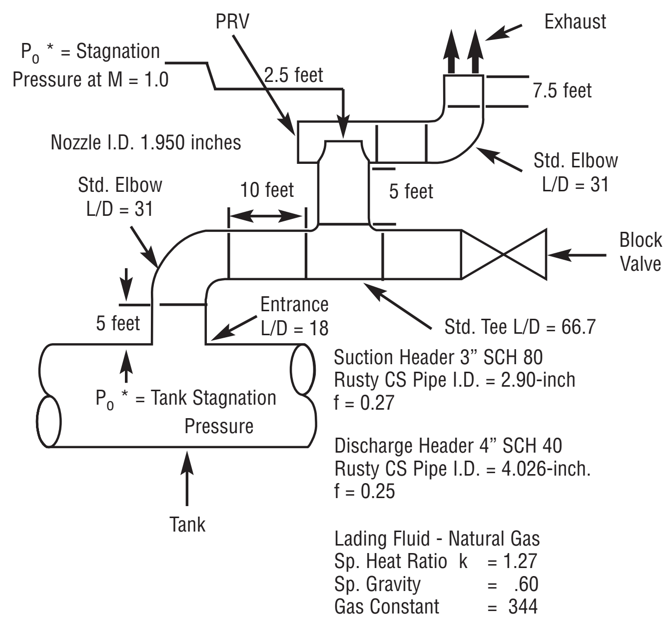
For estimating actual leakage, 20 bubbles per minute totals up to approximately 0.30 standard cubic feet per 24 hours.

For current API/ASME permissible seat leakage rates, please refer to Figure 2-23 under "PRV Design".

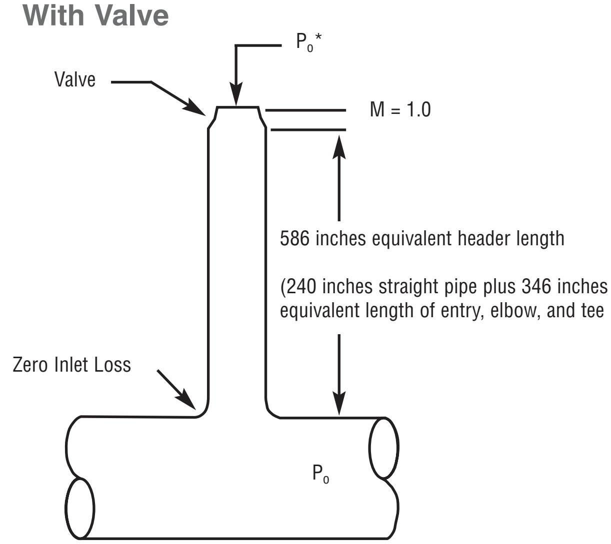
**Fanno Line Approach to Safety Relief Valve Suction and Discharge Header Pressure Distribution**

(For Detailed Calculations See Anderson Greenwood Report 2-0175-128)

#### 1. The Overall System




#### 2. Equivalent Straight Pipe Suction Header With Valve

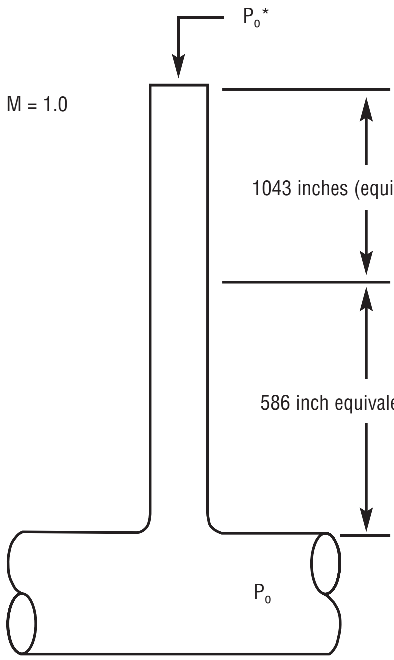


## Section VI

### 3. Calculate Equivalent Pipe to Replace Valve Using Adiabatic Area Ratio

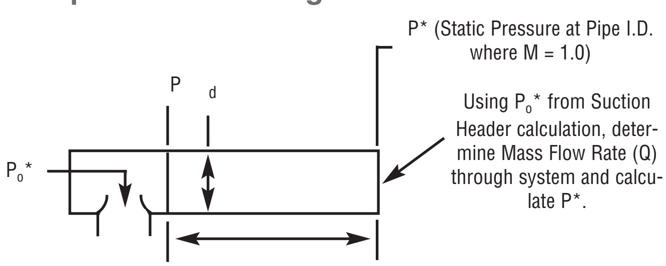
### 4. Equivalent Straight Pipe Suction Header Without Valve




### 5. Read Stagnation Pressure Ratio ($P_o / P_o$*) = Y

Therefore $P_o^* = \dfrac{P_o}{Y}$ and

### 6. Equivalent Discharge Header




Read Static Pressure Ratio $\dfrac{P}{P^*}$ = Z, Calculate P = P*Z

P is the Static Back Pressure which affects the PRV

## 6.6 Set Pressure Test

In addition to nameplate set pressure, there are set pressure tolerances to consider. For unfired pressure vessels and systems, ASME Section VIII, Division I allows a set pressure deviation of ±2 psig from 15 to 70 psig and ±3% over 70 psig. In contrast, ASME Section I, for power boilers, allows ±2 psig deviation for 15 to 70 psig, ±3% deviation for 71 to 300 psig, ±10 psig for 301 to 1000 psig, and ±1% over 1000 psig.

### 6.6.1 Test Set-Up

Regarding the accurate setting and set pressure verification of set pressure for valves in the field, it is not uncommon for the volume directly beneath the PRV to be inadequate. As a reference to this, a very excellent "real world" technical paper was written and presented to the American Institute of Chemical Engineers by two people from a major chemical plant near Lake Charles, LA.

Their very comprehensive test results indicated that a 3 $ft^3$ test tank would give representative set pressure performance with a metal-seated, single ring, safety valve having up to and including an "L" orifice. The range of safety valves with an "M" orifice and up to a "T" orifice required a test tank with a minimum of 15 $ft^3$.

In an attempt to fulfill their company's periodic set pressure verification requirements, some are under the impression that injecting a test pressure through a test insert sandwiched between a closed block valve and PRV inlet will give an accurate test. **Unless a volume-adding, length of pipe is interposed between the block valve and PRV to achieve the necessary test volume, the test result will be inaccurate.** The more deteriorated the PRV metal-seated seating surfaces are, the worse the problem.

An exception to the above situation is a pilot operated PRV with or without a field test connection. A field test connection normally allows the set pressure verification procedure to be accurately performed, while the PRV remains in service, protecting the system from overpressure.

Other possible exceptions are some soft-seated, direct spring SVs --- according to the seat material and the strength of the valve's huddling chamber.

# Section VII

## 7.0 Advantages and Limitations of Valve Types

The following summary of PRV type advantages and limitations is offered to provide relative information. The summary is not intended to be an absolute list of valve pros and cons.

Otherwise unacceptable valve types might be used if the following circumstances dictate:

– specific application

– prior experience

– available commercial or special valve configurations

– various optional accessories for pilot operated valves

– rupture disc in series with the PRV

– special valve location

### Weighted Pallet Type

| Advantages | Limitations |
|---|---|
| Low Cost | Set pressure not readily adjustable |
| Very low set pressures available (down to 0.5 ounce/in$^2$) | Extremely long simmer and poor tightness |
| Simple | High overpressure necessary for full lift (100% or more in some cases) |
| | Seat easily frozen closed at cryogenic temperatures |

### Conventional Metal-Seated

| Advantages | Limitations |
|---|---|
| Lowest cost (in smaller sizes and lower pressures) | Seat leakage, often resulting in lost product and unacceptable emissions, causing environmental pollution |
| Wide chemical compatibility | Simmer and blowdown adjustment is a compromise, which may result in intolerable leakage, product loss and high maintenance costs |
| High temperature compatibility | Vulnerable to effects of inlet pressure losses |
| Standardized center to face dimensions (API 526) | Sensitive to effects of back pressure (set pressure and capacity) |
| Modulating action during small pressure relief excursions may result in reduced product loss | Not normally able to obtain accurate, in-place set pressure verification |
| General acceptance for most applications | |

### Balanced Bellows, Metal-Seated

| Advantages | Limitations |
|---|---|
| Protected guiding surfaces and spring | Seat leakage, often resulting in unacceptable emissions, causing loss of product and environmental pollution |
| Set pressure stability with superimposed back pressure | Simmer or blowdown may be unacceptable |
| Capacity reduced only with higher levels back pressure | Bellows life limitations |
| Good chemical and high temperature capabilities | High maintenance costs |
| | Vulnerable to effects of inlet pressure losses |
| | Not normally able to obtain accurate, in-place set pressure verification |

### Conventional or Balanced Soft-Seated

| Advantages | Limitations |
|---|---|
| Good seat tightness before relieving | Temperature limited to seat material used |
| Good reseat tightness after relieving | Chemically limited according to soft goods used |
| Good cycle life and maintained tightness | Vulnerable to effects of inlet pressure losses |
| Low maintenance costs | Limited back pressure capability |

### Soft-Seated, Pilot Operated - Piston Type

| Advantages | Limitations |
|---|---|
| Smaller, lighter valves at higher pressure and/or with larger orifice sizes | Not recommended for polymerizing type services without pilot purge |
| Excellent seat tightness before relieving | Vital to match soft goods with process conditions |
| Excellent reseat tightness after relieving | Limited low pressure setting (about 15 psig) |
| Ease of setting and adjusting set pressure and blowdown | Not generally used in dirty services without options to eliminate introduction of particles into the pilot |
| Pop or modulating action available | Code restricted by ASME Section I |
| In-line maintenance of main valve | More wetted parts exposed to fluids. Exotic materials can result in an expensive valve |
| Adaptable for remote pressure sensing | |
| Short blowdown obtainable | |
| Set pressure can be field tested while in service | |
| Remote unloading available | |
| Lift not effected by back pressure (when pilot discharges to atmosphere or is balanced) | |

# Section VII

## Soft-Seated, Pilot Operated-Low Pressure (Diaphragm or Metal Bellows Type

| Advantages | Limitations |
|---|---|
| Good operation at very low set pressure (3-inch wc) | Not recommended for polymerizing type services without pilot purge |
| Excellent seat tightness before relieving | Vital to match soft goods with process conditions |
| Excellent reseat tightness after relieving | Limited high pressure setting (about 50 psig) |
| Ease of setting and adjusting set pressure and blowdown | Liquid service limitations |
| Pop or modulating action available | Not generally used in dirty services without options to eliminate introduction of particles into the pilot |
| Adaptable for remote pressure sensing | More wetted parts exposed to fluids. Exotic materials can result in an expensive valve |
| Short blowdown obtainable | |
| Set pressure can be field tested while in service | |
| Remote unloading available | |
| Lift not effected by back pressure (when pilot discharges to atmosphere or is balanced) | |
| Fully open at set pressure with no overpressure | |
| In-line maintenance of main valve | |

## Rupture Discs

| Advantages | Limitations |
|---|---|
| Absolute tightness when disc is intact | Relatively wide burst pressure tolerances |
| Available in exotic materials | Non-reclosing |
| Minimum space required | Can prematurely burst with presence of pressure pulsations. |

## Metal-to-Metal Seated, Pilot Operated - Pressure Relief Valves

| Advantages | Limitations |
|---|---|
| Excellent seat tightness before relieving | Only pop action available |
| Excellent seat tightness after reclosing | Pressure limited to 1200 psig |
| Ease of setting and adjusting set pressure and blowdown | Temperature limited to 1000°F |
| Adaptable for remote pressure sensing | |
| Short blowdown obtainable | |
| Set pressure can be field-tested while in service | |
| Excellent chemical and temperature compatibility | |
| Dual pilot option allows in-service pilot replacement | |

# Section VIII

## Document Related To Pressure Relief Valves

**1. ASME Boiler and Pressure Vessel Code Section VIII**

a. Paragraph UG-125 through UG-137

b. In the Scope section, certain vessels are excluded from ASME requirements, including all vessels with PRDs under set 15 psig.

**2. ASME Fired Boiler Code Section I**

a. Paragraph PG-67 through PG-77

b. In the Scope section, certain vessels are excluded from ASME requirements, including all vessels under 15 psig operating pressure.

**3. API RP 520 Part 1 - Design**

This API design manual is widely used for fire sizing of PRVs on both liquid and gas filled vessels. The recommended practice covers vessels at and above 15 psig.

a. Liquid vessels - Section 5 and 6

b. Gas filled vessels - Appendix C.3

c. Liquid relief - Appendix C.4

**4. API RP 520 Part II - Installation**

a. Recommended piping practices

b. Calculation formula for reactive force on valve (2.4)

c. Precautions for pre-installation handling and testing

**5. API RP 521 - Guide for Pressure-Relieving and Depressuring Systems**

This document discusses the following areas:

a. causes and prevention of overpressure

b. determination of individual relieving rates

c. selection and design of disposal systems

**6. API Guide for Inspection of Refinery Equipment - Chapter XVI Pressure-Relieving Devices**

This document provides the following information:

a. Guide for inspection and record keeping

b. Frequency of inspection paragraph 1602.03

**7. API STD. 526 - Flanged Steel Pressure Relief Valves**

This document provides industry standards for dimensions, pressure/temperature ratings, maximum set pressures, and body materials of direct spring and for pilot operated PRVs.

**8. API STD. 527 - Seat Tightness of Pressure Relief Valves**

This document describes the permissible leakage rate of conventional, bellows, and pilot operated valves and the related test procedure.

**9. API STD. 528 - Standard for Safety Relief Valve Nameplate Nomenclature**

This document provides standard covering information that should go on the nameplate of a safety relief valve.

**10. API RP576 - Inspection of Pressure-Relieving Devices**

**11. API STD. 620 - Design and Construction of Large, Welded, Low-Pressure Storage Tanks**

This document covers standards for tanks at less than 15 psig. Section 6 of this document gives recommendations on relief valve types.

## Section VIII

**12. API STD. 2000 - Venting Atmospheric and Low-Pressure Storage Tanks (non-refrigerated and refrigerated)**

This document covers tanks at less than 15 psig, capacity requirements calculations for both pressure and vacuum, and the sizing method for low pressure tanks.

**13. API 2028 - Flame Arresters in Piping Systems**

**14. API 2210 - Flame Arresters for Vents of Tanks Storing Petroleum Products**

**15. API BULLETIN 2521 - Use of Pressure-Vacuum Vent Valves for Atmospheric Pressure Tanks to Reduce Evaporation Loss**

This document describes the use of PV valves on very low pressure tanks, usually atmosphere to 12-inches wc pressure.

**16. API STD. 2510 - Design and Construction of LPG Installations at Marine and Piping Terminals, Natural Gas Processing Plants, Refineries, and Tank Farms**

Section 7 of this document covers pressure relief valves and covers refrigerated and non-refrigerated LPG vessels.

**17. ASME Guide for Gas Transmission and Distribution Piping Systems**

This document contains all of Title 49, Part 192 DOT (Department of Transportation) Federal Safety Standards and material describing how to use PRVs in natural gas transmission and distribution piping systems.

**18. OSHA - Title 29, Part 1910**

Part 1910 of this document includes handling, storage and safety requirements for LPG and Ammonia.

**19. NFPA - National Fire Protection Association**

This organization provides a series of standards, including:

a. NFPA #58: LP - Gas, Storage and Use

b. NFPA #59: LP - Gas, Utility Plants

c. NFPA #59A: LN - Gas, Storage and Handling

**20. CGA - Compressed Gas Association**

This organization provides a series of standards covering transportation, handling, and storage of compressed gases, including:

a. Pamphlet S - 1.2: Safety Relief Device Standards, Part 2: Cargo and Portable Tanks for Compressed Gases

b. Pamphlet S - 1.3: Safety Relief Service Standards, Part 3: Compressed Gas Storage Containers

# Section IX

## 9.0 Handling Back Pressure on PRVs

Back pressure can severely compromise the performance of some types of pressure relief valves (PRVs), therefore, also the safety of personnel and system equipment. The PRV set pressure, operational stability, and/or relieving capacity can be adversely affected beyond the limits of prevailing codes, standards, and good engineering practice.

## 9.1 Back pressure

There are two types of back pressure. Super- imposed back pressure is the static pressure existing at a closed PRV's outlet. Typically, this occurs when multiple pressure sources discharge into a common header system or perhaps a PRV discharges into the suction side of an active pump or compressor. Built-up back pressure is the PRV's outlet pressure when the PRV is open and flowing and can be a function of many factors: the ratio of PRV orifice area to the PRV's outlet area (the higher the ratio, the more severe); the size, length, and configuration of the discharge piping; whether or not other pressure sources are flowing into the same discharge header; perhaps a discharge header purge; mixed phase flow; flashing fluid flow; etc.

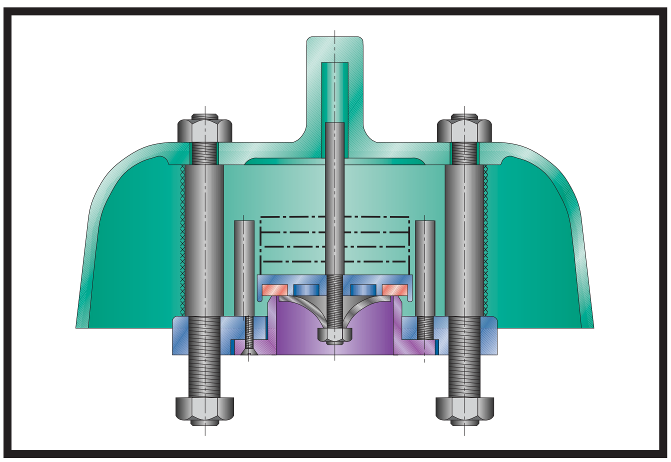


**Figure 1. Weight loaded PRV (breather vent).**

A typical weight-loaded PRV (breather vent, weighted-pallet valve, etc.), as illustrated in Figure 1, practically always discharges directly to the atmosphere at the valve and, therefore, normally has no discharge piping to cause back pressure. The very few weight-loaded valves that have a pipe-away discharge flange must be applied with great caution, because every unit of superimposed back pressure would add to the device's set pressure by the same amount. Deserving an equal amount of caution and knowledge of the device applied, built-up back pressure can severely reduce the discharge capacity of this style of PRV. On very low pressure, storage vessels, where weight-loaded PRVs are normally applied, this reduced PRV capacity can severely affect the mechanical integrity of the vessel.

A conventional PRV, shown in Figure 2, is unbalanced and it, too, is directly affected by superimposed back pressure, unit for unit, on an additive basis. For example, with a factory set pressure of 90 barg, a 5 barg superimposed back pressure will result in an actual, installed set pressure of 95 barg, outside normal, acceptable, set pressure tolerances. However, if the 5 barg superimposed back pressure is always constant, the PRV may be intentionally set low at the factory or in a field workshop by the amount of the superimposed back pressure to achieve the desired, installed set pressure. If all possible relief conditions are considered regarding the discharge header, constant superimposed back pressures are quite unusual.

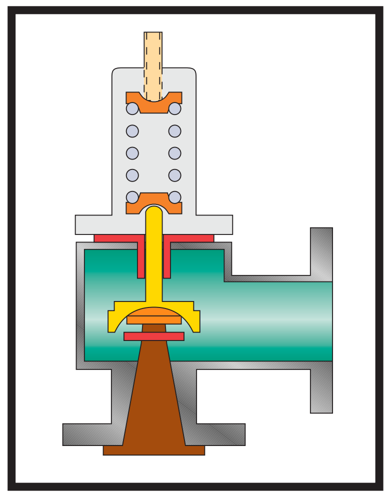


**Figure 2. Conventional direct spring PRV (unbalanced).**

The ability of a typical conventional PRV to tolerate the effects of built-up back pressure with 10% allowable overpressure is illustrated in Figure 3. The shape of the curve readily shows why API RP520, ASME Section VIII, and most (if not all) other codes of good practice recommend a maximum built-up back pressure on a conventional PRV of 10%. There is a slight loss of lift and capacity beginning at about 8% back pressure, rapidly increasing with higher back pressures until the PRV shuts prematurely, usually re-opening immediately and repeating the cycle very rapidly, normally referred to as “rapid cycling” or “chatter”. Any PRV inlet piping pressures losses will aggravate the possibility of PRV chatter, when combined with the effects of built-up back pressure

## Section IX

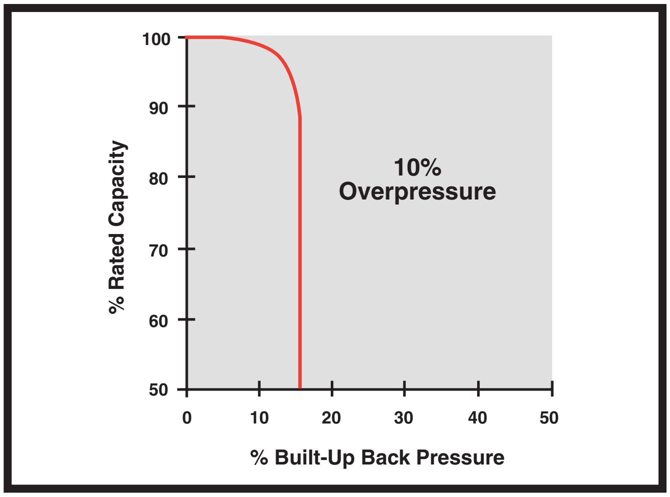



**Figure 3. Effect of built-up back pressure on conventional direct spring PRV.**

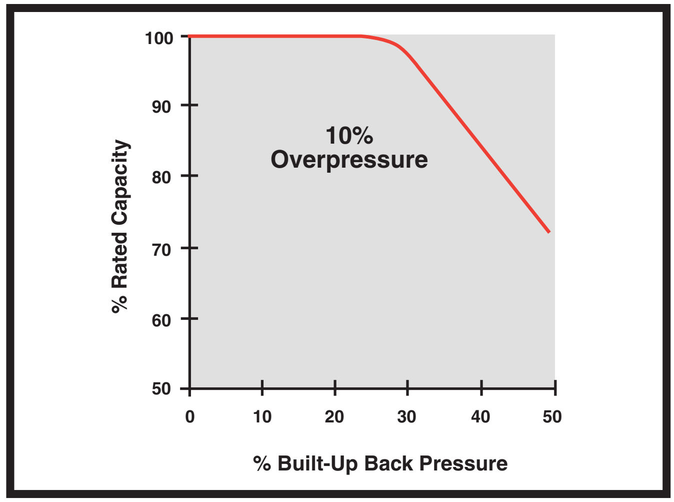



**Figure 5. Effect of built-up back pressure on balanced bellows direct spring PRV.**

Balanced bellows PRVs, as illustrated in Figure 4, are normally applied to spring valve applications when there is variable, superimposed back pressure or built-up back pressure in excess of 10%. The average area of the bellows convolutions is the same as the Seat/Disc sealing area on the top of the Nozzle.

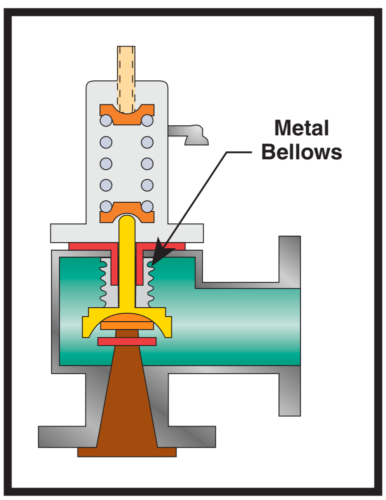



**Figure 4. Balanced bellows direct spring PRV.**

Therefore, the back pressure cannot get behind the Disc/Seat and increase the set pressure, as it does in a conventional PRV, resulting in a constant set pressure. Figure 5 shows the effects of built-up back pressure on a typical balanced bellows valve. The loss of lift, and therefore of capacity, at the higher levels of back pressure is caused by the back pressure acting on the external surface of the bellows, attempting to make it longer. Being restrained at the upper end, the bellows lengthens at the lower end, thereby restricting lift of the Disc/Seat. The PRV manufacturers generally limit built-up back pressure on balanced bellows PRVs to 50% to maintain the bellows' structural integrity (atmospheric pressure inside the bellows) and to avoid the probability of PRV instability.

Regarding the matter of direct spring PRV instability due to excessive inlet pressure losses and/or high back pressures, a PRV's operational stability can often be enhanced with a long blowdown setting.

A PRV that is similar in balancing effect is the Balanced Spindle type, as depicted in Figure 6. The sealing area of the Spindle Seal is the same as that of the Disc/Seat on the Nozzle. Back pressure again cannot get behind the Disc/Seat sealing area, maintaining a constant set pressure with variable superimposed back pressure. Beyond 10 to 20% built-up back pressure according to specific PRV design, the increasing Spindle Seal frictional force, caused by higher back pressures, restricts the Spindle lift and, therefore PRV capacity, as depicted in Figure 7. However, the Balanced Spindle design is considerably more rugged than that of a Balanced Bellows PRV and can withstand higher back pressure levels. The Balanced Spindle PRV design is available in valve sizes up to 2J3.

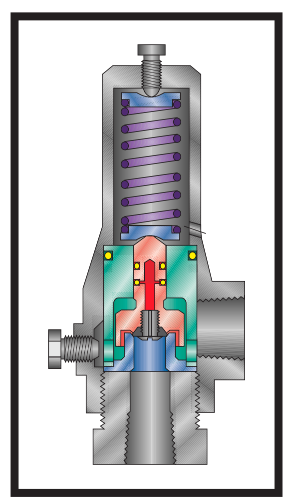


**Figure 6. Balanced spindle PRV.**

# Section IX

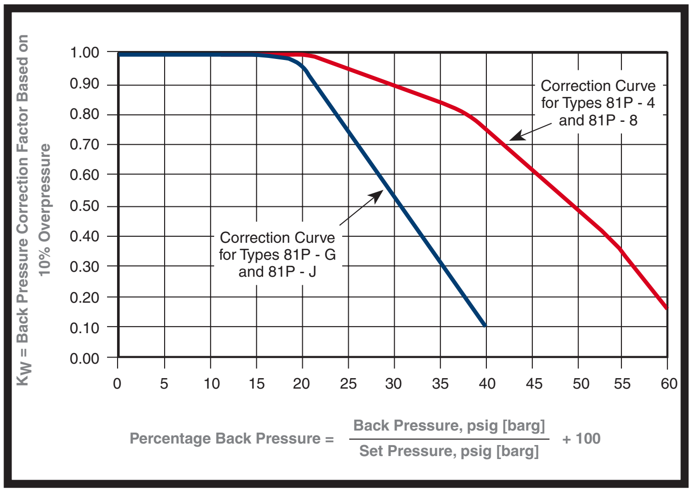



Figure 7. Balanced spindle PRV.

A standard, pilot operated PRV, shown in Figure 8, is inherently balanced against the effect of superimposed back pressure on set pressure if the pilot discharge is at atmospheric pressure or the pilot is internally balanced, in the event the pilot discharge is connected to a main valve outlet port. In terms of the minimized effect of built-up back pressure, a pilot operated PRV is superb compared to all other types of PRVs and is generally best suited to handle operating conditions when the PRV discharges into a common header system, which is becoming so commonplace nowadays due to increased environmental considerations, toxic processes, and product recovery systems.

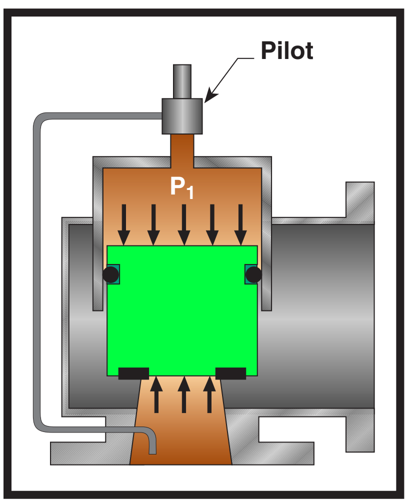



Figure 8. Pilot operated PRV.

Regarding a pilot operated PRV's ability to handle very high levels of built-up back pressure satisfactorily, please refer to Figure 9 for a graph of one manufacturer's PRV's capabilities, established by actual testing under controlled laboratory conditions. Please remember that the curves, following established principles of physics, are also a result of the main valve internal design; therefore, there is no standard set of curves for all manufacturers' pilot operated PRVs. When the Back

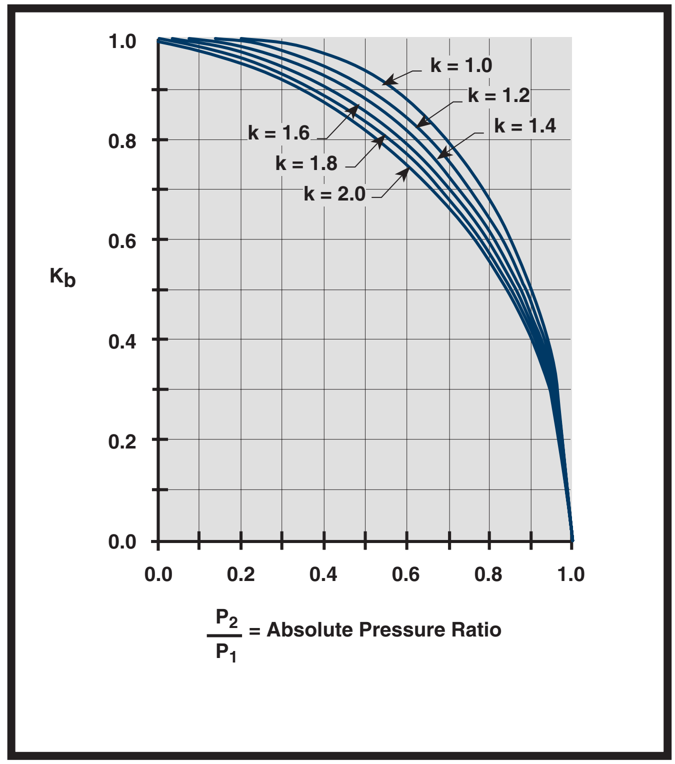



Figure 9. Back pressure correction factor for Anderson Greenwood Crosby piston POPRV.

Pressure Correction Factor drops down from 1.0, the flow of gas through the valve's Nozzle has changed from sonic to sub-sonic velocity, even though the main valve may be fully open. Notice that built-up back pressure levels can be extremely high as long as the pressure-containing components have suitable pressure ratings. It is not unusual for a manufacturer to furnish pilot operated PRVs with 600# or even 900# flanges on the main valve inlet and outlet and both inlet and outlet sections of the valve be fully rated.

## 9.2 Comparative performance

As a basis of comparison, let's consider the loss of gas capacity with flow through an ideal, convergent/divergent, straight-through nozzle as shown in Figure 10. When the downstream back pressure reaches the critical pressure for that medium (for example, 53% for air) the flow through the nozzle becomes sub-sonic, with the flow decreasing. There is a family of curves for perfect nozzles and PRVs, according to the ratio of specific heats, "k" for the particular flowing gas.

Considering a gas having a "k" of 1.3, we can compare the "Kb" (Back Pressure Correction Factor) of a perfect nozzle (Actual Coefficient of Discharge, Kd, of 1.0) and

# Section IX

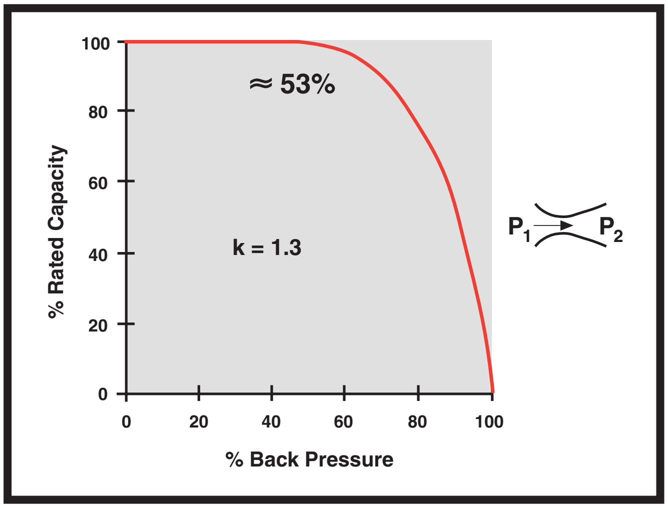



**Figure 10. A perfect nozzle ($K_D$ = 1.0).**

of a specific manufacturer's pilot operated PRV, each having, for example, 70% built-up back pressure. The perfect nozzle has a "Kb" of about 0.92 whereas this specific pilot operated PRV has a "Kb" of about 0.78. Neither a Balanced Bellows nor Balanced Spindle PRV should even be applied at these back pressure levels.

When applying a pilot operated PRV in applications where the PRV discharge will be piped into a common header, along with other PRVs and pressure sources, the pilot operated PRV should be equipped with a Backflow Preventer option. This is a simple, shuttle-type check valve that will prevent the main valve from opening with resultant backflow if the the superimposed back pressure is ever higher than the PRV's inlet pressure, such as could occur during system start-up. It simply loads the volume above the main valve Piston with the higher of the two pressures and also isolates the pilot from the utilized discharge pressure. This option is available on new pilot operated PRVs and is also easily field-retrofittable to existing valves in the field.

In the past, the most popular discharge arrangement for gas PRVs was a simple, short tail-pipe upwards to atmosphere. Except for steam, air, and a few processes, this is fast becoming a thing of the past. Most PRVs now discharge into closed header systems and older, existing PRVs are often converted to such a discharge arrangement, particularly as environmental and safety concerns increase. There are several situations to be aware of which, if overlooked, could cause serious problems.

When sizing discharge headers, do not use the rated capacity of the PRVs discharging into it. Due to a 1962 change in ASME Section VIII, most rated capacities are 10% less than their actual capacities. To determine a PRV's actual capacity, the rated/nameplate capacity should normally be divided by 0.90. I am aware of more than several customers who designed their discharge headers using rated PRV capacities and now have a PRV on their discharge header!

Considering the consequences of temporarily exceeding the pressure ratings of PRV outlet sections or closed header systems, the designer and/or end user may wish to apply ASME/ANSI B31.3-1999 Edition for Process Piping, which allows a continuous pressure rating to be exceeded by up to 33% for no more than 10 hours during any one excursion and no more than 100 total hours per year. A relieving PRV's discharge system would certainly seem to fit this code.

As additional PRV discharges are piped into existing headers, ensure that the header's pressure rating is sufficient for the worst-case PRV loads added after the header was built. This often involves multiple PRVs discharging into the same header at the same time.

Lastly, when converting PRVs from a simple tailpipe-to-atmosphere discharge arrangement to a closed discharge header situation, consider if the existing PRVs will be suitable for the back pressures they may now be exposed to from both a constant set pressure standpoint as well as from a stability and required, rated, PRV capacity standpoint.

## 9.3 Conclusion

Back pressure levels, when not properly and totally considered, can be a significant safety hazard as it relates to PRV performance, not to mention possible non-compliance with applicable codes, regulations, and accepted good engineering practice. The capability comparisons shown in Figure 11 should prove quite interesting!

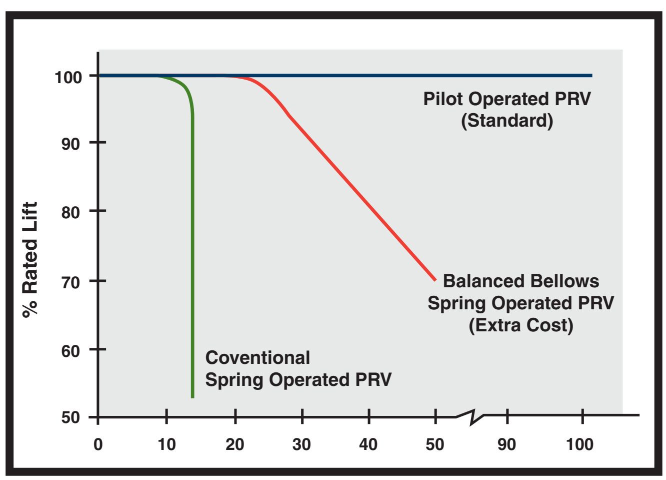



**Figure 11. Effect of back pressure on lift of pressure relief valve types.**

# Section X

## A New Parameter For Selecting A Pressure Relief Valve Size

Donald M. Papa, P.E., Anderson, Greenwood & Co., Houston, Texas October 1, 1990

Selecting the correct pressure relief valve size for over-pressure protection is usually done based on an API orifice area and inlet pipe size. The National Board certified relief capacity of valves selected on that basis can vary widely, depending on what nozzle coefficient and orifice area was used in calculating the valve capacity.

To make valve selection easier and to know what relief capacity is being selected, a new valve parameter called a "flow factor" is proposed. This factor would be equal to KA where K is the ASME valve nozzle coefficient and A is the actual valve nozzle area. Use of such a factor would make it easier to compare valves and would make standards such as API 5261 more meaningful since a better measure of a valve's relief capacity would be indicated.

### Criteria for Sizing

The procedure used in sizing and selecting safety relief valves is to establish the set pressure where the valve is to open, determine the required relieving capacity, and calculate the required orifice area of the valve. The set pressure is usually determined by the applicable Code or the particular operating conditions. The ASME Section VIII Code for unfired pressure primary pressure relief valve be set no higher than the maximum allowable operating pressure (MAOP) of the vessel. Secondary valves in multiple valve installations can be set higher than MAOP, but not more than 1.05% higher of MAOP2.

Determination of the required relieving capacity is based on a worst case analysis of the system being protected. Correctly determining worst case is based on an engineering judgment. Generally any equipment failure, operator error or external condition, such as fire, that would result in an over-pressure condition should be considered. After the set pressure and required relieving capacity are determined, the relieving area of the pressure relief valve can be calculated.

### Sizing Equations

Three basic equations are used for calculating the relieving area of a pressure relief valve. Two are for gas or vapor and one is for liquid. Refer to Figures 1, 2 and 3.

Notes

1. American Petroleum Institute Standard 526, Third Edition
2. ASME Section VIII, Div. 1, UG-134 (a)

$$A = \frac{V\sqrt{MTZ}}{6.32\ C\ K\ P_1} \quad \text{or} \quad A = \frac{W\sqrt{TZ}}{C\ K\ P_1\sqrt{M}}$$

**Where:**

A = Valve orifice area (inch²)

V = Flow rate (SCFM)

W = Flow rate (lb./hr.)

M = Molecular weight

T = Inlet temperature (°F + 460)

Z = Compressibility factor

C = Gas constant based on ratio of specific heats at standard conditions

K = Nozzle coefficient (ASME)

$P_1$ = Inlet pressure (psia) during flow. Set pressure (psig) + overpressure (psig) + local atmospheric.

**Figure 10-1. Sonic Flow**

$$A = \frac{V\sqrt{MTZ}}{4645\ F\ K\ P_1} \quad \text{or} \quad A = \frac{W\sqrt{TZ}}{735\ F\ K\ P_1\sqrt{M}}$$

**Where:**

A = Valve orifice area (inch²)

V = Flow rate (SCFM)

W = Flow rate (lb./hr.)

M = Molecular weight

T = Inlet temperature, (°F + 460)

Z = Compressibility factor

$$F = \sqrt{\left(\frac{k}{k-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{k}} - \left(\frac{P_2}{P_1}\right)^{\frac{k+1}{k}}\right]}$$

k = $C_p/C_v$

K = Nozzle coefficient (ASME)

$P_1$ = Inlet pressure (psia) during flow. Set pressure (psig) + overpressure (psig) + local atmospheric.

**Figure 10-2. Subsonic Flow**

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The two equations given for gas or vapor are for sonic flow and subsonic flow. Sonic flow occurs when the velocity at the exit of the valve nozzle is the velocity of sound for that gas or vapor at the pressure and temperature conditions in the nozzle. A distinguishing characteristic of sonic flow is that it is not dependent on downstream pressure. Subsonic gas or vapor flow and all liquid flow is dependent on upstream and downstream pressure. The transition from sonic to subsonic gas or vapor flow occurs when the absolute pressure at the nozzle exit is approximately 50% of the absolute pressure at the nozzle inlet. Figure 4 is the equation used to more precisely calculate this pressure at the nozzle exit.

$$A = \frac{GPM\sqrt{G}}{38\,K\,K_W\,K_V\,\sqrt{P_1 - P_2}}$$

**Where:**

| | | |
|---|---|---|
| A | = | Valve orifice area (inch²) |
| GPM | = | Flow rate (gallons/minute) |
| G | = | Specific gravity |
| K | = | Nozzle coefficient (ASME) |
| $K_W$ | = | Back pressure correction factor (balanced spring operated SRVs) |
| $K_V$ | = | Viscosity correction factor |
| $P_1$ | = | Inlet pressure (psia) |
| $P_2$ | = | Outlet pressure (psig) |

**Figure 10-3. Liquid Flow**

$$P_2 \geqq P_1 \left[\frac{2}{k+1}\right]^{\frac{k}{k-1}}$$

**Where:** $k = C_p/C_v$

**Figure 10-3. Subsonic Flow**

Two terms common to all three equations are K and A. K is the nozzle coefficient of discharge and A is the nozzle orifice area. The nozzle coefficient of discharge is the actual flow divided by the theoretical flow for the same nozzle with no flow losses. Refer to Figure 5.

$$K = 0.90\,K_d \text{ (ASME)}$$

$$K_d = \frac{\text{Actual Flow}}{\text{Theoretical Flow}}$$

$K_d$ is based on lift of seat disc being great enough so nozzle area controls the flow.

**Figure 10-5. Subsonic Flow**

The relieving capacity of a valve is directly related to KA. The other variables in the flow equations are dependent on the gas, vapor or liquid properties and conditions. One exception is the derating factor $K_w$ for balanced direct spring valves for liquid relief. However, this factor is dependent on back pressure.

## Valve Selection

Valve size is usually selected on the basis of orifice area. The areas referred to are frequently those listed in API Standard 526. Refer to Figure 6. Valve manufacturers usually list their valves by inlet size and the API letter designation for nozzle area. However that area can vary from manufacturer to manufacturer. Further inequities occur with the valve coefficient of discharge.

**Figure 10-6**

**API Valve Orifice Areas (inch square)**

| | | |
|---|---|---|
| D | = | 0.110 |
| E | = | 0.196 |
| F | = | 0.307 |
| G | = | 0.503 |
| H | = | 0.785 |
| J | = | 1.287 |
| K | = | 1.838 |
| L | = | 2.853 |
| M | = | 3.600 |
| N | = | 4.340 |
| P | = | 6.380 |
| Q | = | 11.050 |
| R | = | 16.000 |
| T | = | 26.000 |

Figure 7 (see page 60) is a list of areas and nozzle coefficients for some commonly used API "J" orifice valves. The J orifice area is 1.287 in². The actual areas available from the different manufacturers vary from 1.427-inch square to 1.635-inch square. The ASME and advertised nozzle coefficients vary from 0.788 to 0.975.

Figure 8 is a list of KA's for the same valves and how they compare to the API J orifice multiplied by an assumed K of 0.90. The deviation from the API KA varies from 106% to 112%. A K of 0.90 was used since this is theoretically the largest possible derated nozzle coefficient. The ASME Code 3 requires all valve coefficients be derated 10%. This requirement was added to the Code in 1962. The coefficients of all valves sold prior to that date were not derated.

 

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**Figure 10-7. API "J" Orifice (1.287 inch$^2$) - Air/Gas/Steam Service**

| Manufacturer | Valve Series | Advertised K | Advertised A (inch$^2$) | National Board K | National Board A (inch$^2$) |
|---|---|---|---|---|---|
| AG/Crosby | 727* | .975 | 1.287 | .788 | 1.635 |
| AG/Crosby | JOS | .967 | 1.287 | .865 | 1.453 |
| Dresser | 1900 | .950 | 1.287 | .855 | 1.496 |
| Farris | 2600 | .953 | 1.287 | .858 | 1.430 |
| Lonergan | D | .971 | 1.287 | .878 | 1.427 |

*Pilot Operated

**Figure 10-8. API "J" Orifice (1.287 inch$^2$) - Air/Gas/Steam Service**

| Manufacturer | Valve Series | KA Advertised | KA National Board | KA API* |
|---|---|---|---|---|
| AG/Crosby | 727* | .1.255 | 1.288 | 1.158 |
| AG/Crosby | JOS | 1.245 | 1.257 | 1.158 |
| Dresser | 1900 | 1.223 | 1.279 | 1.158 |
| Farris | 2600 | 1.227 | 1.227 | 1.158 |
| Lonergan | D | 1.250 | 1.253 | 1.158 |

*K = 0.900

In comparing the K and A of valves, additional confusion occurs because of a difference between the advertised values and the ones listed in the National Board of Boiler and Pressure Vessel Inspectors Pressure Relief Device Certifications Book. Frequently, neither the K nor the A listed in the National Board Book agree with the advertised K and A, however the advertised product of KA is always equal to or less than the National Board listing.

## Background for Difference in K and A

The reason for the difference between the advertised K and A and the National Board listed K and A dates back to 1962 when the ASME Section VIII Code was changed to derate all certified relieving capacities 10%. Most manufacturers elected to comply with this Code revision by not derating their advertised capacity or their nozzle coefficients, but by increasing the nozzle area of the safety valve 10% so the product of KA remained unchanged. However, the API orifice areas were still advertised. Therefore, valves were advertised as having API orifice areas with nozzle coefficients greater than 0.90.

## Proposed Valve Flow Parameter

To simplify valve orifice area calculations and selection, a new valve flow parameter called "flow factor" is proposed. This flow factor would be KA where K is the ASME valve coefficient and A is the actual valve nozzle area listed with the National Board. All valves would have a KA factor. When sizing and selecting a valve, KA would be calculated and a valve selected with a KA equal to or greater than that calculated. The sizing equations would take the form shown in Figure 9. Manufacturers would publish KA where K is the ASME K published in the National Board's certification book and is the actual orifice area.

## Sonic Flow

$$KA = \frac{V\sqrt{MTZ}}{6.32\ C\ P_1} = \frac{W\sqrt{TZ}}{C\ P_1\sqrt{M}}$$

## Subsonic Flow

$$KA = \frac{V\sqrt{MTZ}}{4645\ F\ P_1} = \frac{W\sqrt{TZ}}{735\ F\ P_1\sqrt{M}}$$

## Liquid Flow

$$KA = \frac{GPM\sqrt{G}}{38\ K\ K_W\ K_V\sqrt{P_1 - P_2}}$$

**Figure 10-9. Sonic, Subsonic and Liquid Flow**

**Figure 10-10**

| API Flow Factor "KA" | | |
|---|---|---|
| D | = | 0.099 |
| E | = | 0.176 |
| F | = | 0.276 |
| G | = | 0.453 |
| H | = | 0.707 |
| J | = | 1.158 |
| K | = | 1.654 |
| L | = | 2.568 |
| M | = | 3.240 |
| N | = | 3.906 |
| P | = | 5.742 |
| Q | = | 9.945 |
| R | = | 14.400 |
| T | = | 23.400 |

# Section X

A further proposal would be to use this flow factor in the planned revision of API 526. KA would replace the orifice areas now published. The K in the KA factor would be .90. Figure 10 is a list of the proposed flow factors.

## Need for KA Flow Factor

The discrepancy between actual and advertised K and A has caused confusion among users, inspectors, and manufacturers. Some users and inspectors have a difficult time trying to reconcile the difference between the advertised and National Board listed K and A of the different valve manufacturers' products. Manufacturers and their representatives are sometimes confused when asked to explain these differences.

Sizing errors can be made if the advertised and the National Board listed K's and A's are mixed. For example, using an advertised K with a National Board listed A would overstate the certified capacity about 10%. Conversely, using the National Board listed K with the advertised A would understate the capacity about 10%.

A KA factor would simplify the calculation of valve size since the sizing parameter is dependent only on the fluid properties and, where applicable, on the valve back pressure derating factor. Using a computer or calculator to determine valve size would be easier and the KA parameter calculated would be a more accurate measure of the valve size required.

## Conclusion

Standardization is an attempt to make problem solving easier. The proposed KA flow factor is a step in that direction. This factor would be a direct measure of a valve's relieving capacity, so when an API size valve was specified, the user would know what its relieving capacity would be with respect to a standard.

### Note

1. ASME Section VIII, Div. 1, UG-131 (d)(1), (d)(2)(a), (e)(2).

 