

RELIEF VALVES

MODE: OVHD Receiver

Deviation

High press.

Cause

1. PRC fails closed
2. External fire
3. Operator Error: open block valve on H₂ line
4. Steam out

Consequence

- increase press fracture main sys.
- lose condenser
- over pressure
- vessel rupture
- vapor cloud explode
- fire
- injury people
- condensate duty → 0
- Vaporize liquid.
- pressure up system.
- PRC open wide
- copy down & flood.

Safeguards

- LFA on not vapor
- PSV on receiver
- PSV on OVHD
- FATH on vapors
- PSV 2

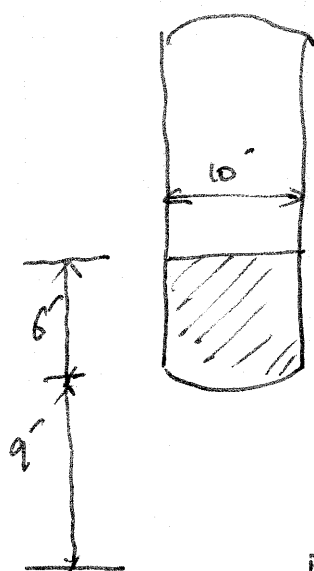
Recommendation

- Consider adding a TAT to TI-106 outlet of OVHD condenser.

Design Intent:

FV $\longleftrightarrow$ 200 psi

Problem



$$DP = 150 \text{ psig}$$

Not insulate.

$$MW = 95$$

$$Q_{\text{fire}} = 21000 \text{ F} \cdot SA^{0.82}$$

$$SA =$$

$$P_{\text{acc}} = 1.21 (P_{\text{design}}) + 14.7 =$$

ASME CODE

$$= 1.21 (150) + 14.7$$

$$= 186.2 \text{ psia}$$

$$MW = 95$$

}

$$T_{\text{rel}} = 4.5 \text{ 'F}$$

$$\Delta H_v = 88 \text{ Btu/lb}$$

Wetted Surface Area

$$SA = 1.089 d^2 + \pi D h$$

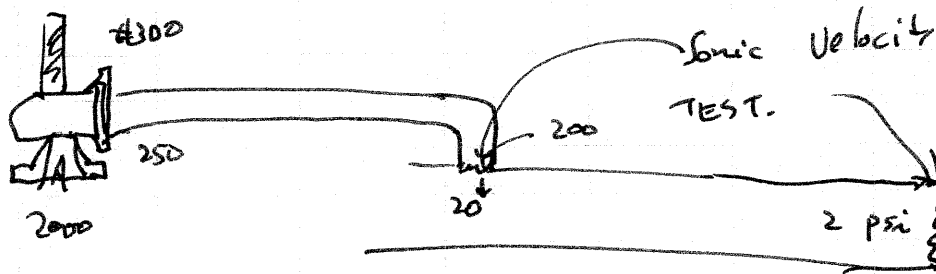
$$= \frac{215.72}{297.4} \text{ ft}^2$$

$$Q_{\text{fire}} = 21000 \cdot (1) (215.72)^{0.82}$$

$$= \frac{2240544}{1721912.5} \text{ Btu/hr}$$

$$W = \frac{Q_{\text{fire}}}{A} = \frac{25460}{18600} \text{ lb/hr}$$

BP & NO Sonic Velocity

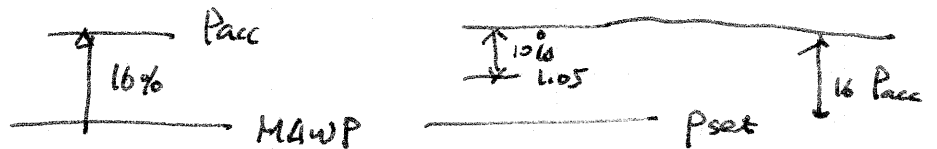


Accumulation (Code) → Press above MAWP vessel referenced to vessel.

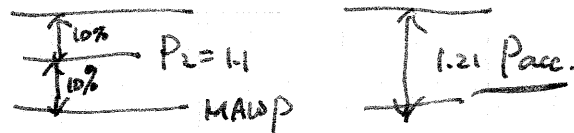
Overpressure → pressure above PSV set pressure referenced to PSV

16% accum

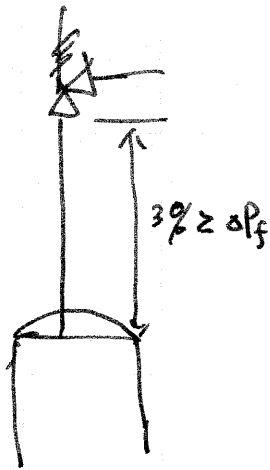
2 on more PSV



Fire
2 PSV



Chattering - destructive - piping, PSV



$$\begin{array}{c} \text{---} 100 \\ \text{---} 95 \end{array} > \Delta P_g > \text{blowdown } P$$

10 cps
10 cycle per second.

RELIEF SYSTEM DESIGN

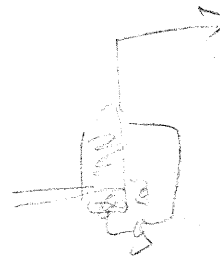
ENGINEERING DESIGN SEMINAR

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Disposal Systems

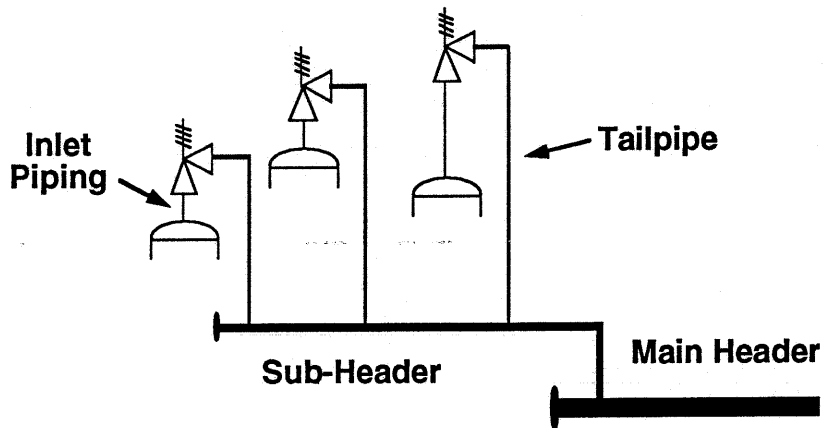
A pressure relief valves may discharge to:

- ☐ atmosphere
- ☐ another part of the process
- ☐ vent or blowdown vessel
- ☐ quench vessel
- ☐ absorber or scrubber
- ☐ incinerator or furnace
- ☐ elevated flare or a ground level flare



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Piping Components



PSV Inlet Piping

Reasons for adding length to PSV inlet piping:

- ✦ high (or low) temperature service
- ✦ improve access to PSV
- ✦ isolating valve provision
- ✦ avoid pressure fluctuations
- ✦ locating device outside of insulation
- + *free draining discharge piping (elevation)*

Design Considerations:

- ✦ piping diameter equal to or larger than PSV inlet flange
- ✦ predominantly vertical (free draining back to vessel)
- ✦ *short as possible*
- ✦ frictional drop less than 3% of PSV set pressure *(rated max. capacity based)*
- ✦ pipework and supports to transmit minimal stress *tail pipe*

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Relief Header System

- ✦ Determine peak flow rate to relief header
- ✦ PSV performance constrains the maximum allowable back pressure 4x 15 psi
- ✦ Economic analysis required to allocate pressure drop through out the relief header system

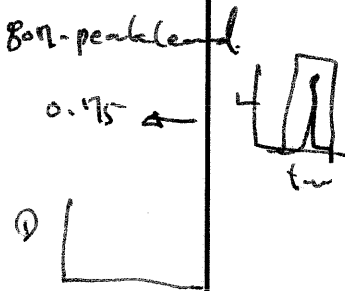
50 psi
15 psi

NOTE: Different sections of the relief header have different maximum flow rates and pressure drops available.

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Peak Flow Rate to Header

- ✦ Simultaneous Relieving Rates
 - power failure
 - cooling water failure
 - instrument air failure
 - fire case
- ✦ Large Individual Relief Rates
- ✦ Relieving Liquids



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Sub-Header Loads

Cooling Water Failure:

Location	Set Press	Type	Max BP	Flow	Adj	Adj Flow
Debutanizer	200	Bal	60	150,000	0.75	112,500
Stripper Col	150	Bal	45	75,000	0.75	56,250
Splitter Col	100	Bal	30	180,000	0.75	135,000
Total						<u>303,750</u>

for sizing tail pipe

MW(ave) 82
T(ave) 332 F
k(ave) 1.12

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Tabulation

From the tabulation, determine:

- Limiting back pressure
- Locations where a change in valve type would loosen the back pressure constraint
- Location where a special shutdown device would significantly reduce the flare load
- Total flow to the flare

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Adjustment Factors

- * No industry wide factors
- * 0.75 factor used by UOP to account for the improbability of simultaneous discharge at maximum rate
- * Shutdown devices may be used to reduce flare loads if a failure policy is applied to the devices

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*Case by company own policy.
Uop : take credit with agreement
of client.*

Types of Shutdown Devices

- ☐ Low flow shutdown for a fired heater
- ☐ High pressure trip on heating medium to the reboiler

CAUTIONS:

- ◆ shutdown devices can fail to operate
- ◆ shutdown devices can cause nuisance shutdowns
- ◆ install only when economic incentive

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Allocation of Pressure Drop

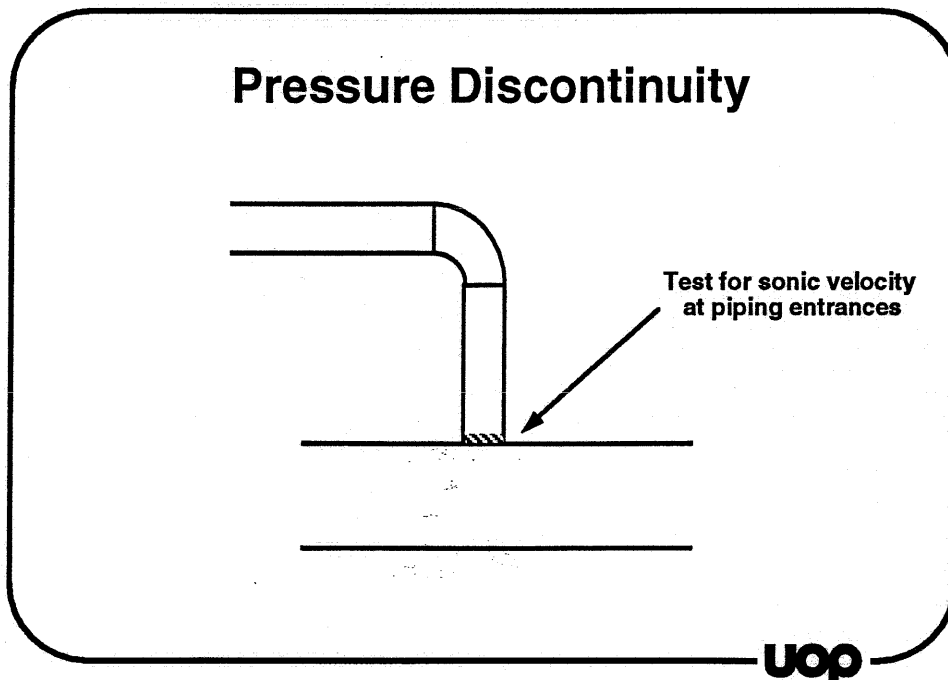
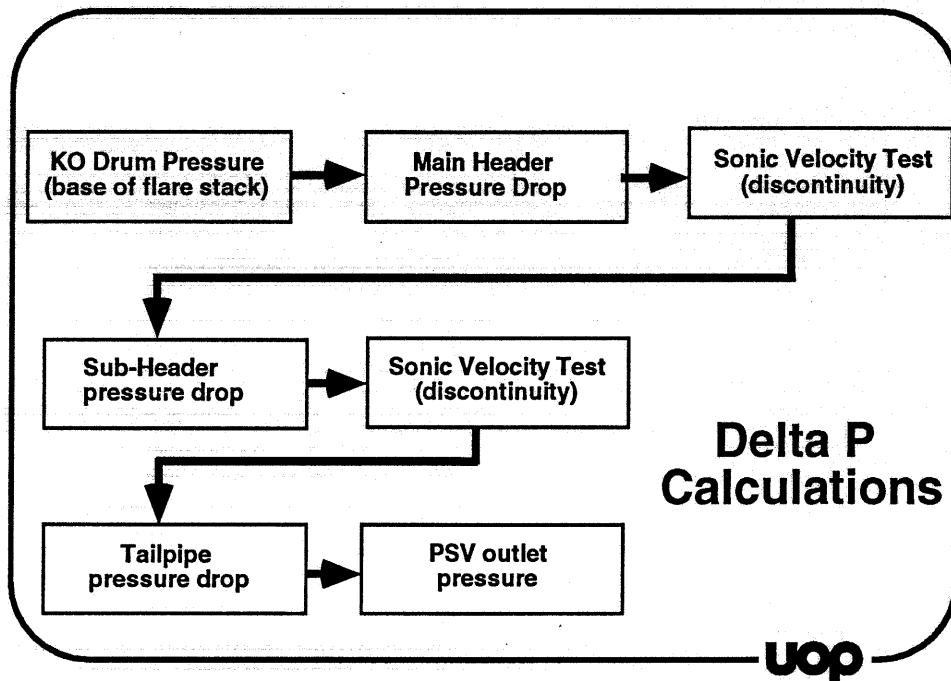
- Based on economics
- Typical allocation:
 - 70% for main flare header
 - 30% for sub-headers and tailpipe
- Typical pressure drops:
 - 15 psi for main header
 - 7 psi for sub-header and tailpipe
 - 3 psi for flare stack

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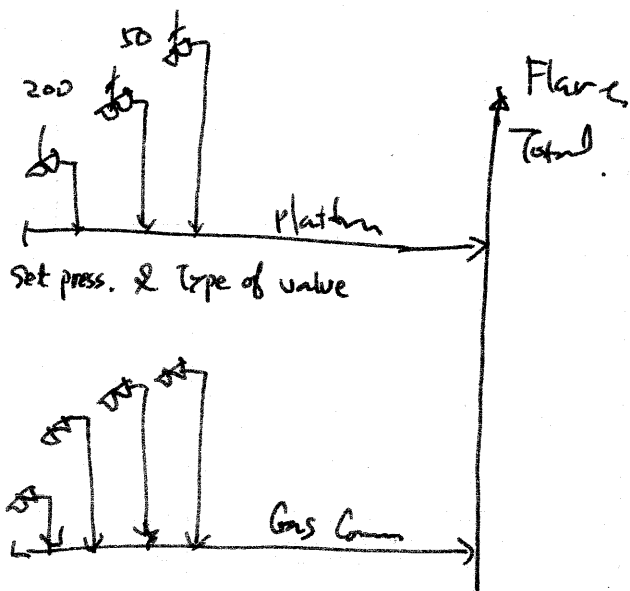
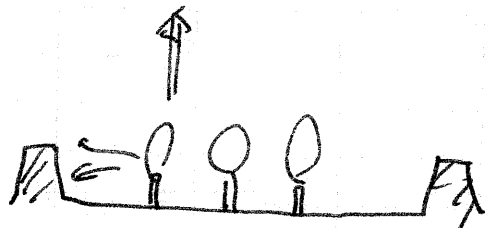
Additional Considerations

- ▼ Potential critical flow in header system
- ▼ Density of vapor changes significantly
- ▼ Material is accelerated in the system

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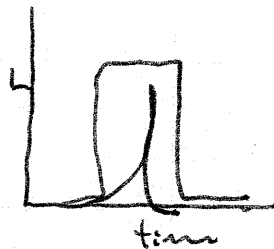


Ground flare



8.07 - Peak load

0.75
64
←



Sizing, Selection, and Installation Of Pressure-Relieving Devices in Refineries

Part I—Sizing and Selection

API RECOMMENDED PRACTICE 520
SIXTH EDITION, MARCH 1993



American National Standard

ANSI/API RP 520, Pt. I-1992

American Petroleum Institute
1220 L Street, Northwest
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H.O. Lee
Dec.

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Manufacturing, Distribution and Marketing Department

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FOREWORD

API Recommended Practice 520, *Sizing, Selection, and Installation of Pressure-Relieving Devices in Refineries*, is the result of several years' work by engineers in the petroleum industry.

The information in this recommended practice is intended to supplement the information contained in Section VIII, "Pressure Vessels," of the ASME *Boiler and Pressure Vessel Code*. The recommendations presented in this publication are not intended to supersede applicable laws and regulations.

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

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

The first edition of this recommended practice was issued in 1955. The second edition was published in two parts: Part I, "Design," in 1960 and Part II, "Installation," in 1963. The third edition of Part I was issued in November 1967 and reaffirmed in 1973.

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Suggested revisions are invited and should be submitted to the director of the Manufacturing, Distribution and Marketing Department, American Petroleum Institute, 1220 L Street, N.W., Washington, D.C. 20005.

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Sizing, Selection, and Installation of Pressure-Relieving Devices in Refineries

Part I—Sizing and Selection

SECTION 1—INTRODUCTION

1.1 Scope

This recommended practice applies to the sizing and selection of pressure relief devices used in refineries and related industries for equipment that has a maximum allowable working pressure of 15 pounds per square inch gauge or greater. The pressure relief devices covered in this recommended practice are intended to protect unfired pressure vessels and related equipment against overpressure from operating and fire contingencies.

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

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

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

Fired vessels in accordance with Section I of the ASME *Boiler and Pressure Vessel Code* and ASME B31.1 are also not within the scope of this recommended practice.

1.2 Definition of Terms

Terms used in this recommended practice relating to pressure relief devices and their dimensional and operational characteristics are defined in 1.2.1 through 1.2.3. The terms are covered more specifically in the applicable sections of text and accompanying illustrations.

1.2.1 PRESSURE RELIEF DEVICES

1.2.1.1 A *pressure relief device* is 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 also may be designed to prevent excessive internal vacuum. The device may be a pressure relief valve, a nonreclosing pressure relief device, or a vacuum relief valve.

1.2.1.2 A *spring-loaded pressure relief valve* is a pressure

relief device designed to automatically reclose and prevent the further flow of fluid.

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

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

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

1.2.1.2.4 A *conventional pressure relief valve* is a spring-loaded pressure relief valve whose performance characteristics are directly affected by changes in the back pressure on the valve.

1.2.1.2.5 A *balanced pressure relief valve* is a spring-loaded pressure relief valve that incorporates a means for minimizing the effect of back pressure on the performance characteristics.

1.2.1.3 A *pilot-operated pressure relief valve* is a pressure relief valve in which the main valve is combined with and controlled by an auxiliary pressure relief valve.

1.2.1.4 A *rupture disk device* is a nonreclosing differential pressure relief device actuated by inlet static pressure and designed to function by bursting the pressure-containing rupture disk. A rupture disk device includes a rupture disk and a rupture disk holder.

1.2.2 DIMENSIONAL CHARACTERISTICS OF PRESSURE RELIEF DEVICES

1.2.2.1 The *actual discharge area* is the measured minimum net area that determines the flow through a valve.

1.2.2.2 The *curtain area* is the area of the cylindrical or conical discharge opening between the seating surfaces above the nozzle seat created by the lift of the disk.

1.2.2.3 The *effective discharge area*, or equivalent flow area, is a nominal or computed area of a pressure relief valve

used in recognized flow formulas to determine the size of the valve. It will be less than the actual discharge area.

1.2.2.4 The *nozzle area* is the cross-sectional flow area of a nozzle at the minimum nozzle diameter.

1.2.2.5 A *huddling chamber* is an annular pressure chamber in a pressure relief valve located beyond the seat for the purpose of generating a rapid opening.

1.2.2.6 The *inlet size* is the nominal pipe size (NPS) of the valve at the inlet connection, unless otherwise designated.

1.2.2.7 The *outlet size* is the nominal pipe size (NPS) of the valve at the discharge connection, unless otherwise designated.

1.2.2.8 *Lift* is the actual travel of the disk away from the closed position when a valve is relieving.

1.2.3 OPERATIONAL CHARACTERISTICS

1.2.3.1 System Pressures

1.2.3.1.1 The *maximum operating pressure* is the maximum pressure expected during system operation.

1.2.3.1.2 The *maximum allowable working pressure (MAWP)* is 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.

1.2.3.1.3 The *design gauge pressure* refers to at least 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 in all cases where the MAWP has not been established. The design pressure is equal to or less than the MAWP.

1.2.3.1.4 *Accumulation* is the pressure increase over the maximum allowable working pressure of the vessel during discharge through the pressure relief device, expressed in pressure units or as a percent. Maximum allowable accumulations are established by applicable codes for operating and fire contingencies.

1.2.3.1.5 *Overpressure* is the pressure increase over the set pressure of the relieving device, expressed in pressure units or as a percent. It is the same as accumulation when the relieving device is set at the maximum allowable working pressure of the vessel.

1.2.3.1.6 *Rated relieving capacity* is 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.

1.2.3.1.7 *Stamped capacity* is 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.

1.2.3.2 Device Pressures

1.2.3.2.1 The *set pressure* is the inlet gauge pressure at which the pressure relief valve is set to open under service conditions.

1.2.3.2.2 The *cold differential test pressure* is 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.

1.2.3.2.3 *Back pressure* is 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.

1.2.3.2.4 *Built-up back pressure* is the increase in pressure in the discharge header that develops as a result of flow after the pressure relief device opens.

1.2.3.2.5 *Superimposed back pressure* is 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.

1.2.3.2.6 *Blowdown* is the difference between the set pressure and the closing pressure of a pressure relief valve, expressed as a percent of the set pressure or in pressure units.

1.2.3.2.7 *Opening pressure* is the value of increasing inlet static pressure at which there is a measurable lift of the disk or at which discharge of the fluid becomes continuous.

1.2.3.2.8 *Closing pressure* is the value of decreasing inlet static pressure at which the valve disk reestablishes contact with the seat or at which lift becomes zero.

1.2.3.2.9 *Simmer* is the audible or visible escape of compressible fluid between the seat and disk at an inlet static pressure below the set pressure and at no measurable capacity.

1.2.3.2.10 *Leak-test pressure* is the specified inlet static pressure at which a seat leak test is performed.

1.2.3.2.11 The term *relieving conditions* is used to indicate the inlet pressure and temperature on a pressure relief device at a specific overpressure. The relieving pressure is equal to the valve set pressure (or rupture disk burst pressure) plus the overpressure. (The temperature of the flowing

fluid at relieving conditions may be higher or lower than the operating temperature.)

Note: Figure 1 shows the relationships between various terms used in connection with single and multiple pressure relief valve installations.

1.3 Referenced Publications

The most recent editions of the following standards, codes, and specifications are cited in this recommended practice:

API

- Std 510 *Pressure Vessel Inspection Code—Maintenance Inspection, Rating, Repair, and Alteration*
 RP 520 *Sizing, Selection, and Installation of Pres-*

sure-Relieving Devices in Refineries, Part II, "Installation"

RP 521 *Guide for Pressure-Relieving and Depressurizing Systems*

Std 526 *Flanged Steel Safety Relief Valves*

Std 2000 *Venting Atmospheric and Low-Pressure Storage Tanks (Nonrefrigerated and Refrigerated)*

RP 2001 *Fire Protection in Refineries*

ASME¹

Boiler and Pressure Vessel Code, Section I, "Power Boilers"; and Section VIII, "Pressure Vessels," Division 1

B31.3 *Chemical Plant and Petroleum Refinery Piping*

SECTION 2—PRESSURE RELIEF DEVICES

2.1 General

This section describes the basic principles, operational characteristics, applications, and selection of pressure relief devices used independently or in combination. These devices include spring-loaded and pilot-operated pressure relief valves, rupture disk devices, and other pressure relief devices. These devices are described in the text and illustrated in Figures 2-17.

2.2 Spring-Loaded Pressure Relief Valves

2.2.1 SAFETY VALVES

Safety valves are spring-loaded pressure relief devices designed to provide full opening with minimum overpressure. Static pressure retained in the huddling chamber and the kinetic energy of the gas or vapor are utilized to overcome the spring force on the disk as it lifts, resulting in pop action. The closing pressure will be at a point below the set pressure and will be reached after the blowdown phase is completed.

2.2.2 RELIEF VALVES

Relief valves are spring-loaded pressure relief devices designed for use in liquid service. At set pressure, the inlet pressure force overcomes the spring force and the disk begins to lift off the seat. As inlet pressure increases, the disk lift increases to allow an increase in the flow. The closing pressure will be at a point below the set pressure and will be reached after the blowdown phase is completed. Relief valve capacities are usually rated at 10 or 25 percent overpressure, depending on the application.

2.2.3 SAFETY RELIEF VALVES

Safety relief valves are spring-loaded pressure relief devices that provide the characteristics of a safety valve when used in gas or vapor service and the characteristics of a relief valve when used in liquid service. Safety relief valves are generally provided with bonnets that enclose the spring and provide a pressure-tight housing for use in conventional or balanced types, depending on the effect of back pressure on their performance.

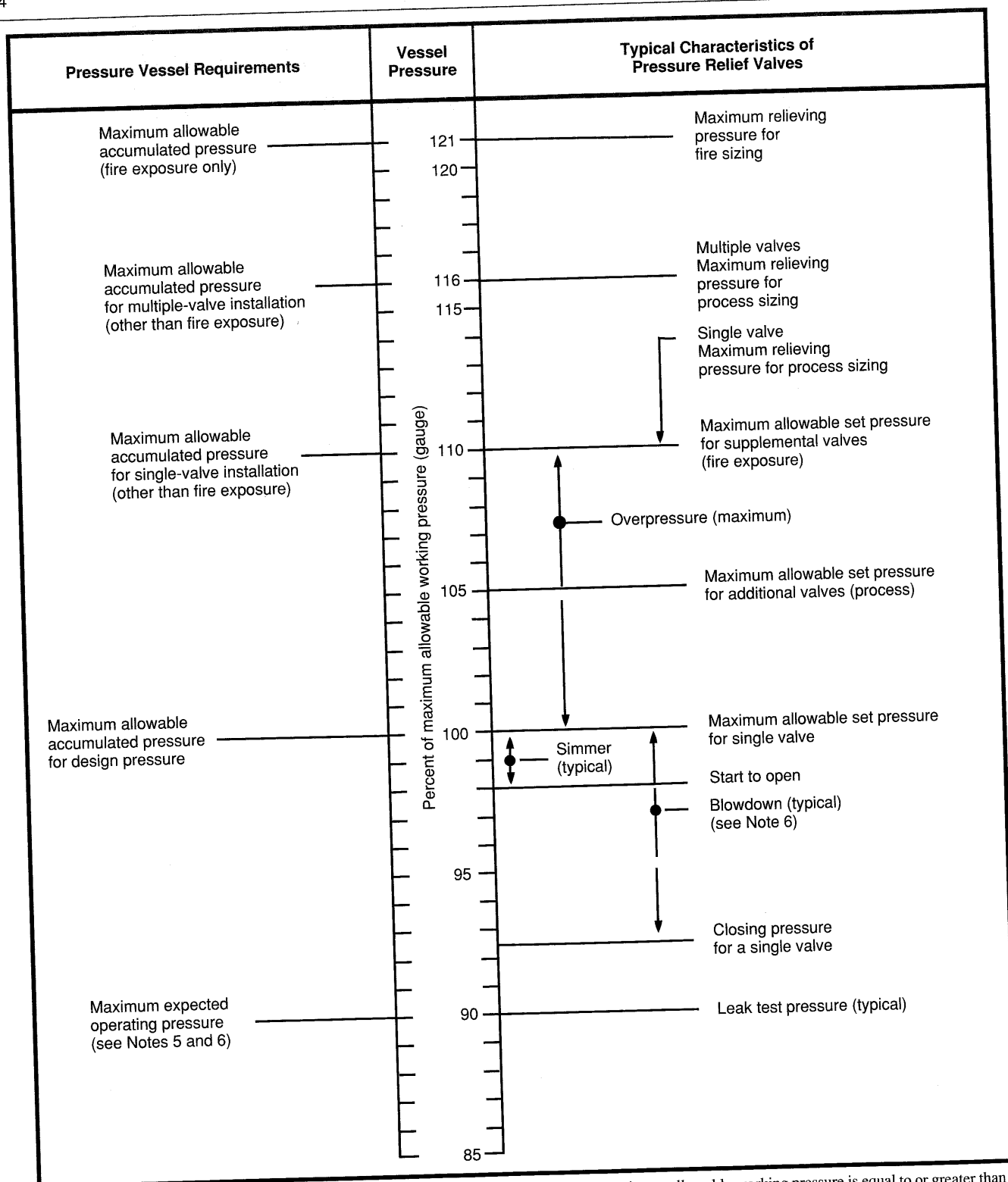
2.2.4 PRESSURE RELIEF VALVES

2.2.4.1 Conventional Pressure Relief Valves

Conventional pressure relief valves are used when the discharge is through a short tail pipe that vents to the atmosphere or through a low-pressure manifold system that carries the discharged fluid of one or more valves to a remote location for disposal. Normally, the spring force is the differential between the set pressure and atmospheric pressure. The set pressure will therefore be increased by superimposed back pressure unless the spring force is adjusted accordingly. Built-up back pressure may also affect valve performance; therefore, the impact of downstream pressure when one or more valves discharge into a common manifold should be determined by referring to the appropriate manufacturer's catalog.

The interaction of the forces within the valve and the effects of back pressure on the opening pressure are illustrated in Figure 18. Available conventional pressure relief valves have disks that have a greater disk area, A_D , than the nozzle seat area, A_N . If the spring bonnet is vented to the atmo-

¹American Society of Mechanical Engineers, 345 East 47th Street, New York, New York 10017.



Notes:

1. This figure conforms with the requirements of Section VIII of the ASME Boiler and Pressure Vessel Code.
2. The pressure conditions shown are for pressure relief valves installed on a pressure vessel.
3. Allowable set-pressure tolerances will be in accordance with the applicable codes.

4. The maximum allowable working pressure is equal to or greater than the design pressure for a coincident design temperature.
5. The operating pressure may be higher or lower than 90.
6. Section VIII, Division 1, Appendix M, of the ASME Code should be referred to for guidance on blowdown and pressure differentials.

Figure 1—Pressure-Level Relationships for Pressure Relief Valves

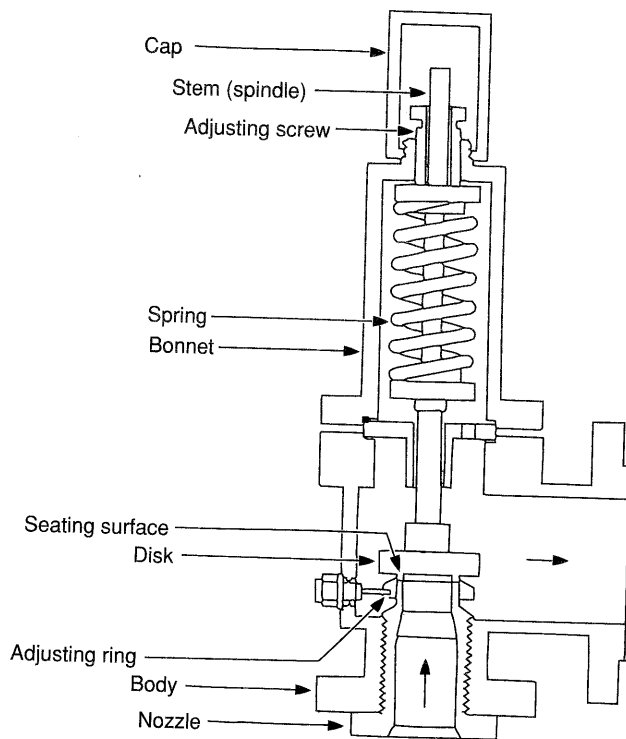
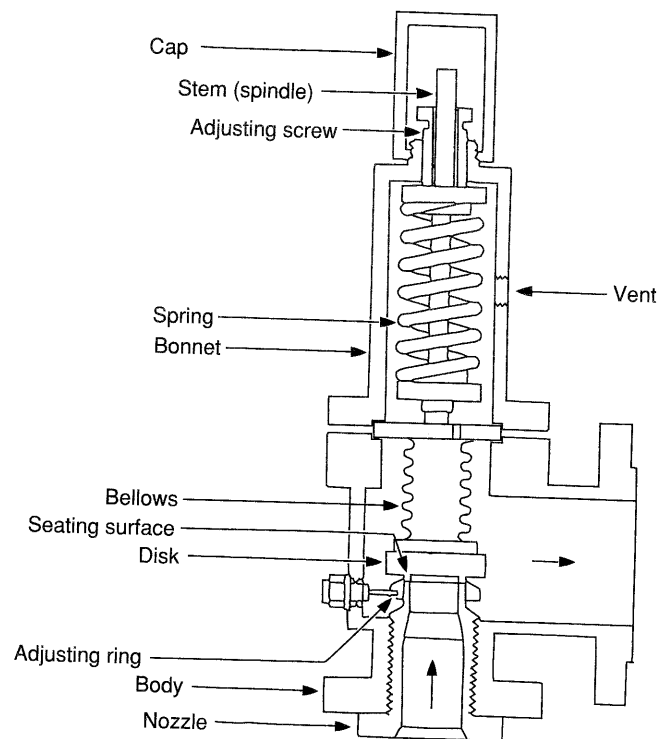


Figure 2—Conventional Safety Relief Valve With a Single Adjusting Ring for Blowdown Control



Note: For corrosion isolation, an unbalanced-bellows safety relief valve is available.

Figure 3—Balanced-Bellows Safety Relief Valve

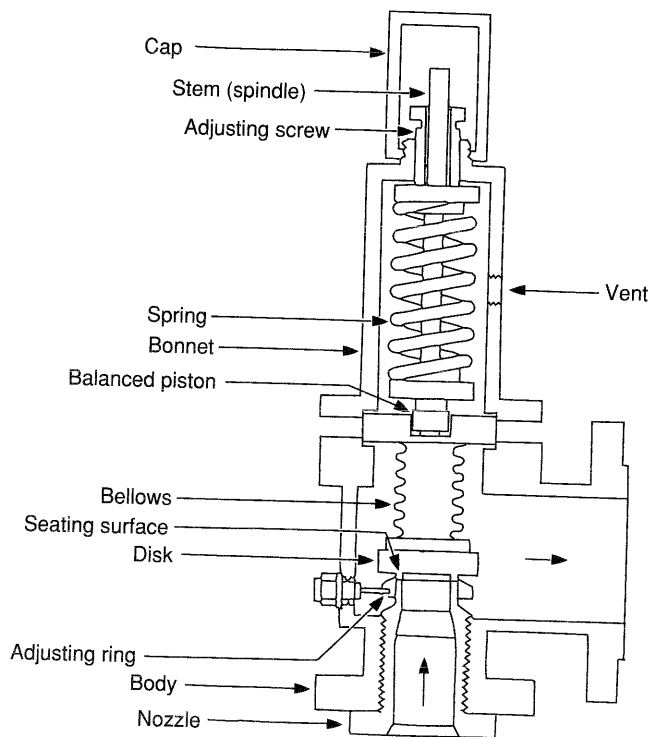


Figure 4—Balanced-Bellows Safety Relief Valve With an Auxiliary Balanced Piston

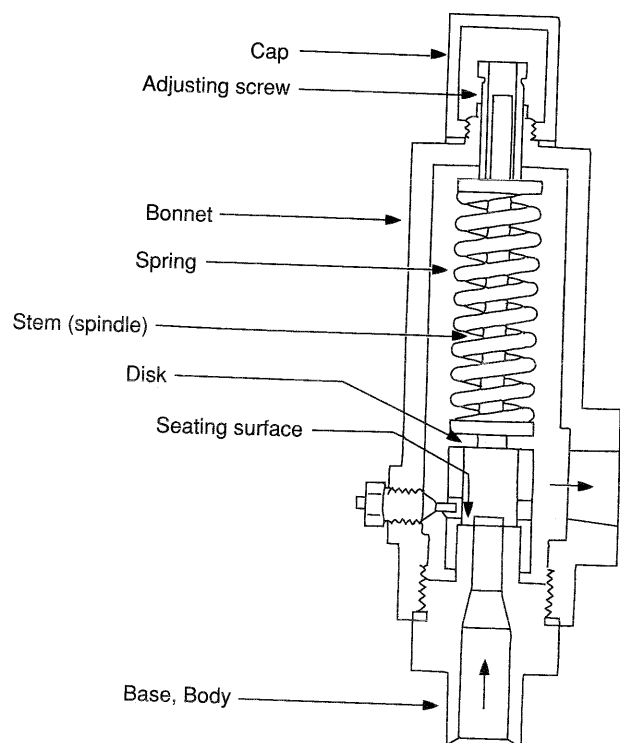


Figure 5—Thermal Relief (Liquid Relief Valve)

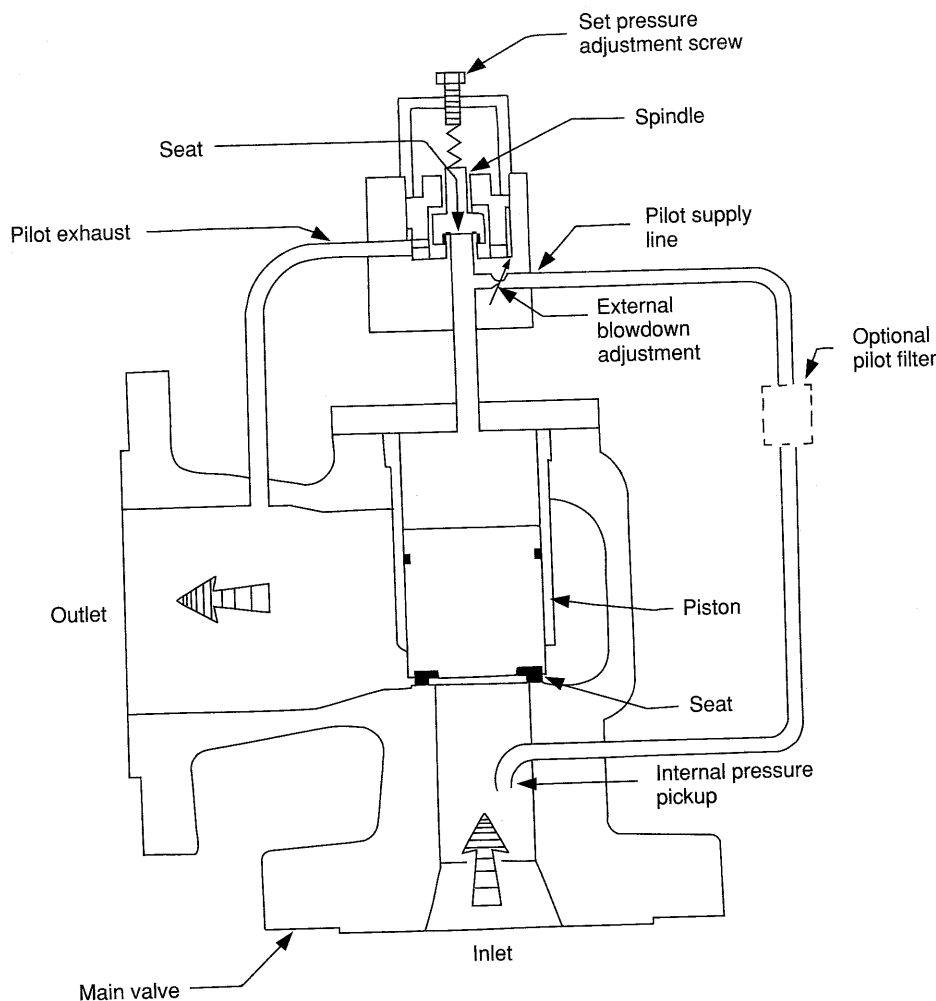


Figure 6—Pop-Action Pilot-Operated Valve (Flowing Type)

sphere, the back pressure acts with the vessel pressure to overcome the spring force. This condition makes the opening pressure less than it is when the valve is set with atmospheric pressure on its discharge; however, if the spring bonnet is vented to the valve discharge instead of to the atmosphere, the back pressure acts with the spring force to increase the opening pressure. Variation in the superimposed back pressure will directly affect the opening pressure and should be evaluated in system design.

Conventional pressure relief valves, as normally installed, show unsatisfactory performance when excessive built-up back pressure develops from the flow through the valve and piping as a result of the same unbalanced forces that affect the set pressure. Performance data observed during the investigation of the built-up back-pressure problem are shown in Figure 19. The information is plotted as the ratio of the valve capacity at any given built-up back pressure to the valve capacity without built-up back pressure versus the ratio of the built-up back pressure to the valve set pressure. The capacity curve is the result of the balance of forces acting on the disk. As long as the built-up back pressure is less

than the overpressure after the valve opens, the valve will remain open and perform satisfactorily under flowing conditions, and it will have flow characteristics that are basically similar to those in a theoretical nozzle performance. If, however, the built-up back pressure is increased at a greater rate than the overpressure, the balance of forces will tend to close the valve, which can become unstable and cause the flow to fall off rapidly. This instability is caused by a dynamic pressure imbalance or a harmonic resonance. The valve may start to flutter or chatter.

Flutter refers to the abnormally rapid reciprocating motion of the movable parts of a pressure relief valve in which the disk does not contact the seat. *Chatter* refers to the motion that causes the disk to contact the seat and damage the valve and associated piping. The allowable built-up back pressure must therefore be considered for each amount of overpressure used.

Conventional pressure relief valves should typically not be used when the built-up back pressure is greater than 10 percent of the set pressure at 10 percent overpressure. A higher maximum allowable built-up back pressure may be

used for overpressure greater than 10 percent. The combined effect of the superimposed and built-up back pressures on the performance characteristics of the valves must be considered when more than one pressure relief valve discharges into a common manifold at the same time.

The theoretical performance for a nozzle is plotted in Figure 20. The curve represents the maximum theoretical flow attainable for any ideal gas that has the specific heat ratio of $K = 1.3$. The theoretical nozzle maintains flow capacity up to the critical flow pressure and then gradually diminishes to zero.

The theoretical rate of flow through the nozzle depends on the absolute upstream pressure and is independent of the downstream pressure when the back pressure is less than the critical flow pressure; however, when the back pressure is increased beyond the critical flow pressure, the flow is reduced. (See 4.3 for information about the sizing of pressure relief valves for critical and subcritical flow.)

2.2.4.2 Balanced Pressure Relief Valves

The design of balanced pressure relief valves incorporates means for reducing the effect of back pressure on the set

pressure and for minimizing the effect of built-up back pressure on performance characteristics such as opening and closing pressure, lift, and relieving capacity (see Section 4). Balanced valves are of two basic types: the piston and the bellows (see Figure 21).

Several variations of the piston valve are manufactured. The guide on the piston is vented so that the back pressure on opposing faces of the valve disk is cancelled. The top face of the piston has an area, A_P , the same as the nozzle seat area, A_N , and is subjected to atmospheric pressure by venting the spring bonnet. The vented gas from the bonnets of balanced piston valves should be disposed of safely and with minimum restrictions.

The effective bellows area, A_B , of bellows valves is the same as the nozzle seat area, A_N . The arrangement of the bellows in the valve prevents the back pressure from acting on the top side of the disk within the effective bellows area, A_B . The disk area, A_D , extending beyond the bellows and the opposing nozzle seat area cancel the effect of the back pressure on the valve disk so that there are no unbalanced forces under any downstream pressure variations. The bellows additionally serves to isolate the disk guide, spring, and other top works parts from the lading fluid. This feature may be im-

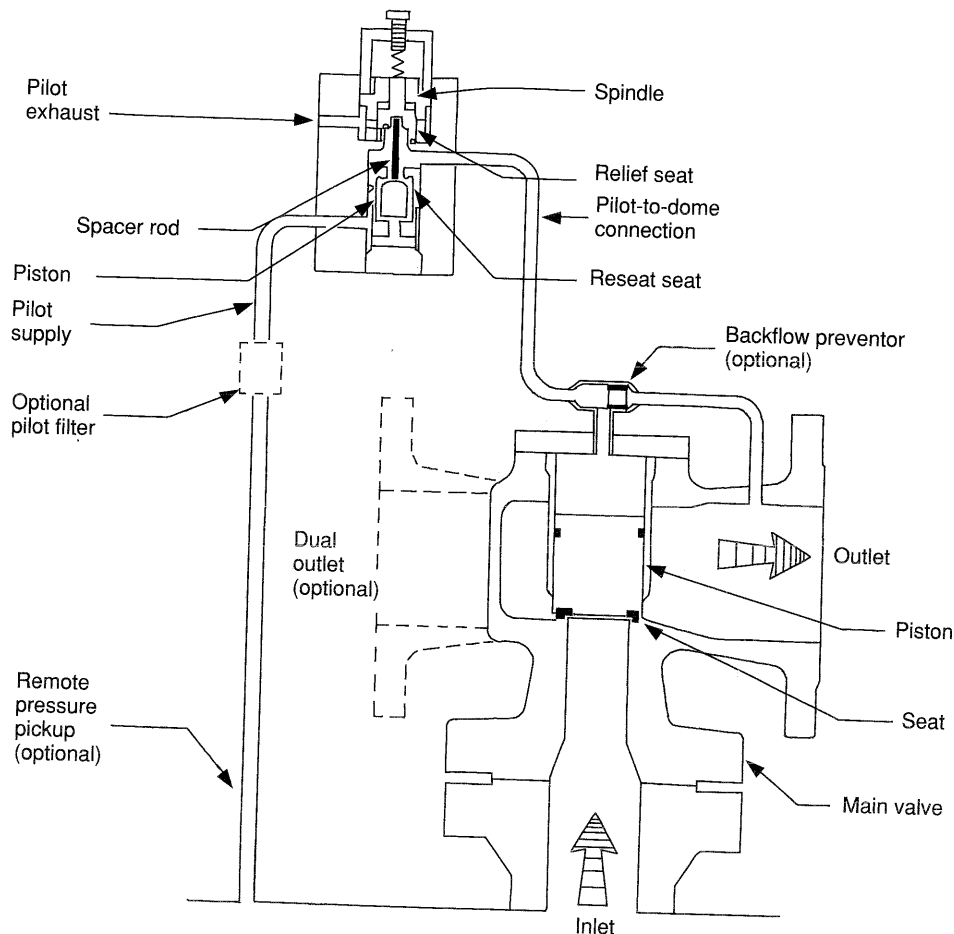


Figure 7—Pop-Action Pilot-Operated Valve (Nonflowing Type)

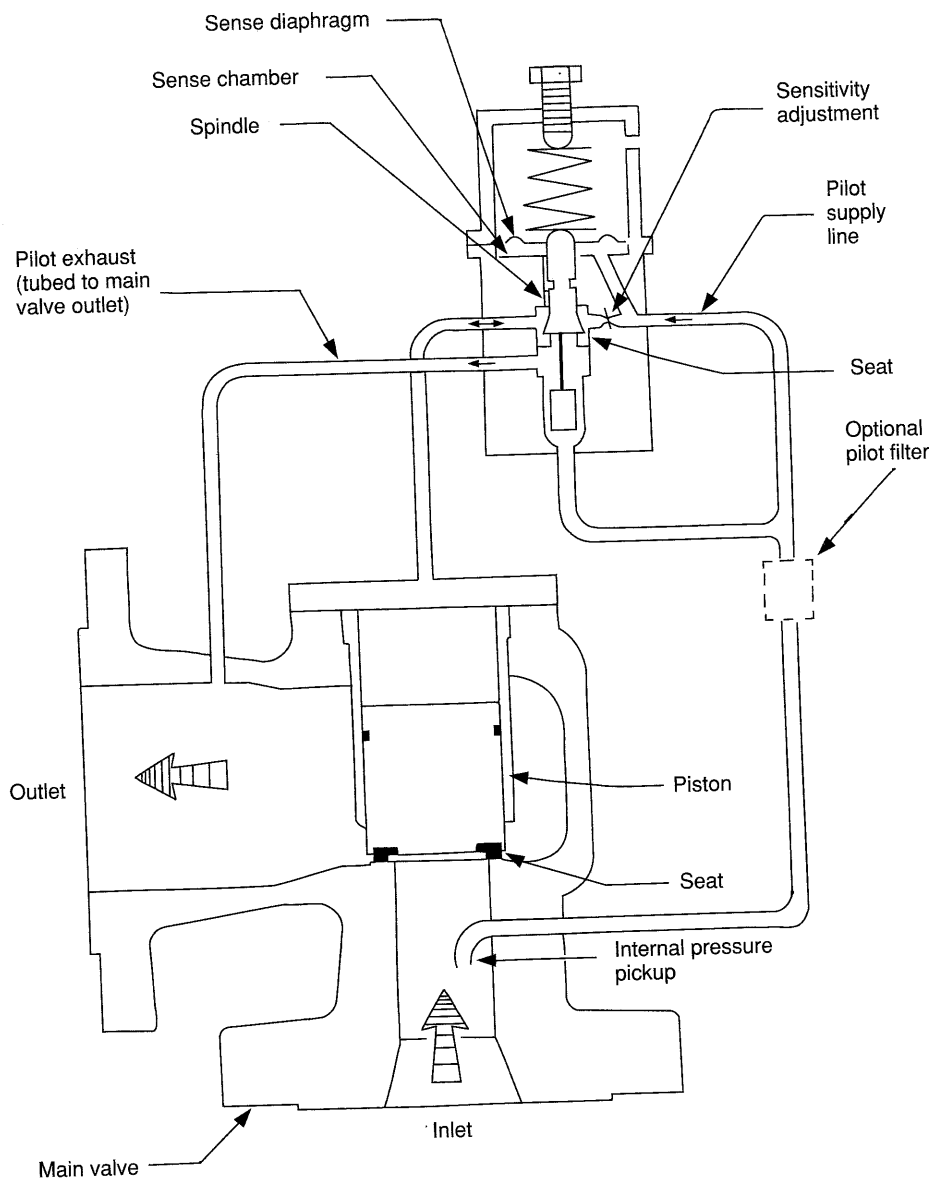


Figure 8—Modulating Pilot-Operated Valve (Flowing Type)

portant if the lading fluid is corrosive or may foul the pressure relief valve. Because of physical size limitations, balanced bellows are not available in certain valve designs and sizes. If balanced bellows are not available, unbalanced bellows valves may be specified when corrosion isolation alone is intended.

The balanced pressure relief valve makes possible higher pressures in the relief discharge manifolds. Both balanced valves shown in Figure 21 should have bonnet vents large enough to ensure that no appreciable back pressure exists during flow conditions. If the valve is located where atmospheric venting (usually not a large amount) would present a hazard, the vent should be piped to a safe location that is independent of the valve discharge system.

2.2.4.3 Valve Characteristics

Figure 22 shows the disk travel from the set pressure, *A*, to the maximum relieving pressure, *B*, during the overpressure incident and to the closing pressure, *C*, during the blow-down.

2.3 Pilot-Operated Pressure Relief Valves

The two basic types of pilot-operated pressure relief valves are the piston type and the diaphragm type.

The piston type valve consists of the main valve, which encloses a floating piston, and an external pilot valve (see Figures 6-9). The piston is designed to have a larger effective

area on the top than on the bottom. Up to the set pressure, the top and bottom areas are exposed to the same inlet operating pressure. Because of the larger effective area on the top of the piston, the net force holds the piston tightly to the main valve seat. As the operating pressure increases, the net seating force increases and tends to make the valve tighter. At the set point, the pilot vents the pressure from the top of the piston; the resulting net force unseats the piston, and process flow is established through the main valve. After the overpressure incident, the pilot will close the vent from the top of the piston, thereby reestablishing pressure, and the net force will cause the piston to reseat.

The diaphragm type pilot-operated relief valve is similar to the piston type except that the piston is replaced by a flexible diaphragm and disk. The diaphragm provides the unbalance function of the piston. The disk, which normally closes the main valve inlet, is integral with a flexible diaphragm (see Figure 10). The external pilot valve serves the same function

to sense process pressure, vent the top of the diaphragm at set pressure, and reload the diaphragm once the process pressure returns to normal. As with the piston valve, the seating force increases proportionally with the operating pressure because of the differential exposed area of the diaphragm.

The pilot valve that operates the main valve can be either a pop action or modulating action pilot. Figure 23 shows the action of the pop pilot; it shows that pilot operation causes the main valve to lift fully. Figure 24 shows the action of the modulating pilot; it shows that pilot operation opens the main valve only enough to satisfy the required relieving capacity.

The pilots may be either a flowing or nonflowing type. The flowing type allows process fluid to flow through the pilot when the main valve is open; the nonflowing type does not. The user should consult the manufacturer to determine the advantages and disadvantages of either type.

A backflow preventer is required when the possibility ex-

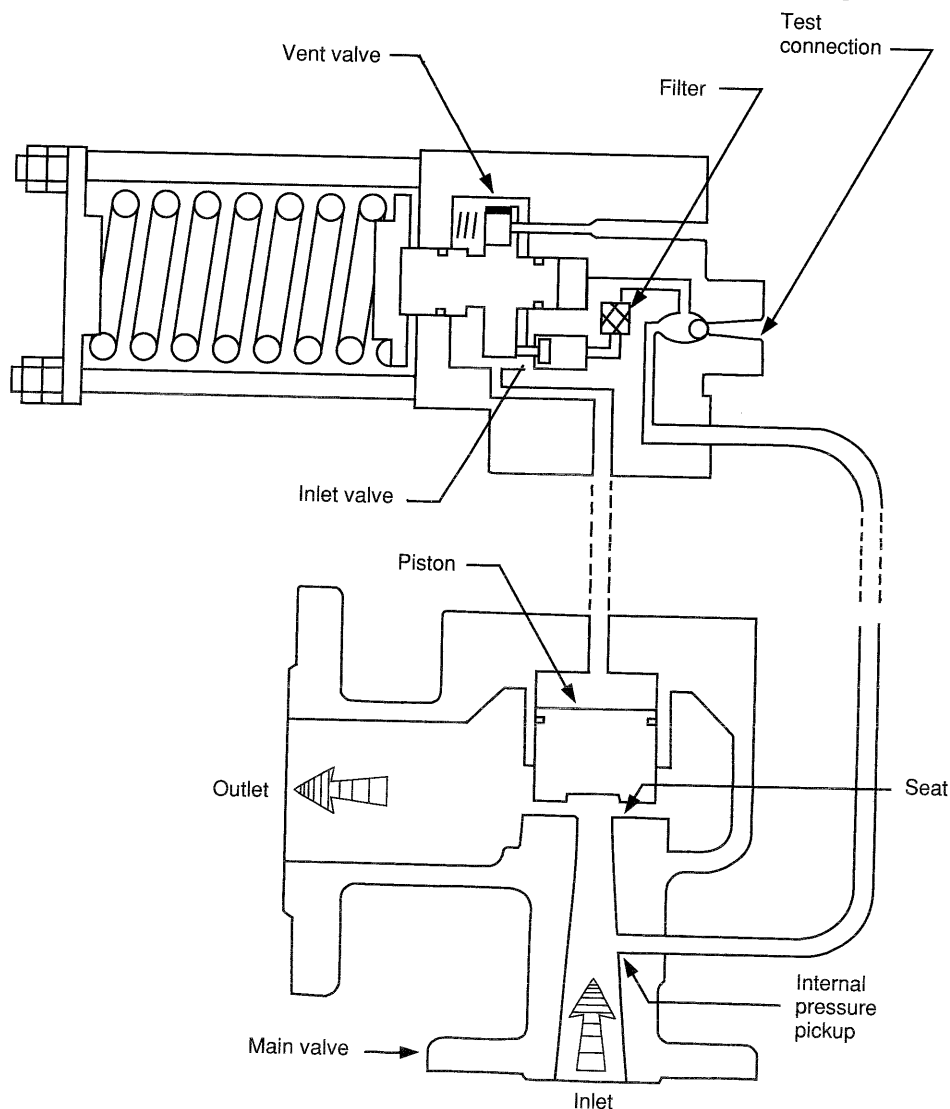


Figure 9—Pilot-Operated Relief Valve With a Nonflowing Modulating Pilot Valve

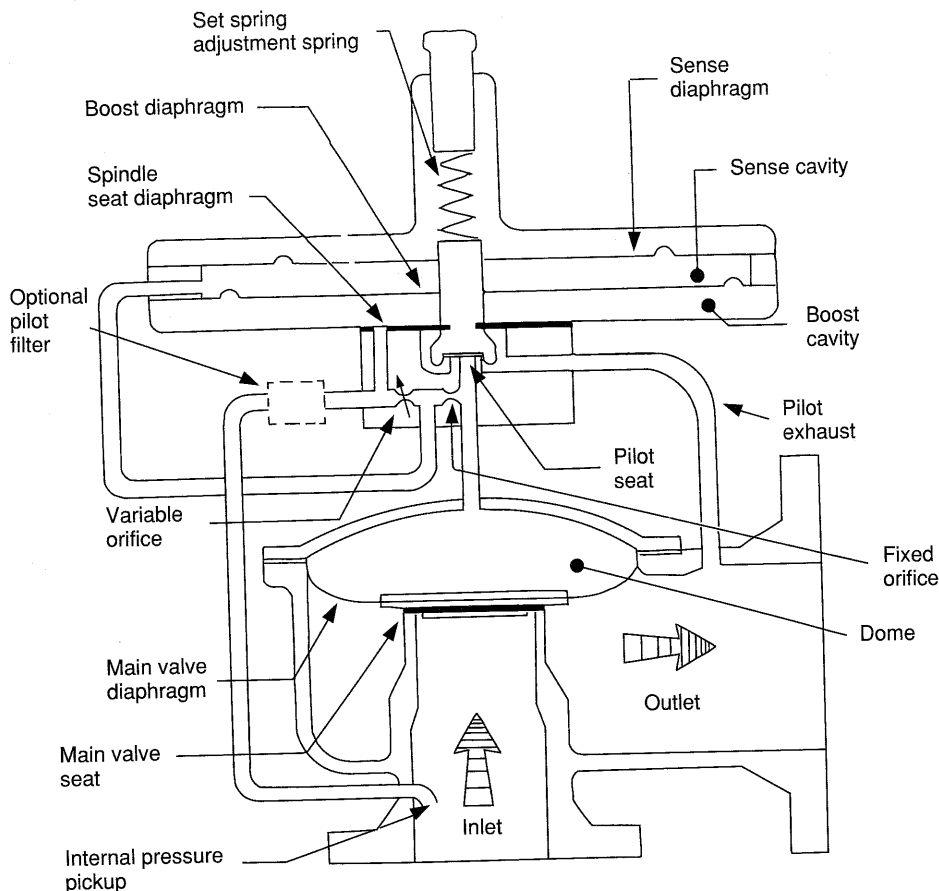


Figure 10—Low-Pressure Pilot-Operated Valve (Diaphragm Type)

ists of developing a pressure on the discharge side of the valve that exceeds the inlet pressure of the valve. The differential area will cause the piston to lift, and flow in the valve will be reversed (see Figure 7).

Pilot-operated relief valves are available for use in liquid and vapor services. Since the main valve and the pilot contain nonmetallic components, process temperature and fluid compatibility can limit their use. In addition, fluid characteristics such as susceptibility to polymerization or fouling, viscosity, the presence of solids, and corrosiveness may affect pilot reliability. The manufacturer should be consulted to ensure that the proposed application is compatible with available valves.

2.4 Rupture Disk Devices

The remainder of Section 2 provides definitions, descriptions, and operational characteristics of rupture disk devices. Specific terms and uses are covered within the applicable sections of text and accompanying illustrations (see Figures 11-17).

2.5 Rupture Disks

2.5.1 GENERAL

A rupture disk is the pressure-containing and pressure-sensitive element of a rupture disk device. Rupture disks may be designed in several configurations, such as flat, domed (prebulged), or reverse acting.

Rupture disks may require a rupture disk holder to enclose, seal, and clamp the rupture disk in position. The combination of the disk and holder is called a rupture disk device.

2.5.2 TYPES OF RUPTURE DISKS

2.5.2.1 Conventional Rupture Disk

A conventional domed rupture disk is a prebulged solid metal disk designed to burst when it is overpressured on the concave side (see Figure 11).

The conventional domed rupture disk of flat-seat or angular-seat design generally provides a satisfactory service life

when operating conditions are 70 percent or less of the rated burst pressure of the disk and when limited pressure cycling and temperature variations are present. If vacuum or back pressure conditions are present, the disk must be furnished with a support to prevent reverse flexing or implosion. Vacuum supports are designed for continuous service at full vacuum or less. Special designs are available for back pressure conditions that exceed 15 pounds per square inch gauge. The conventional domed rupture disk will fragment upon burst.

2.5.2.2 Scored Tension-Loaded Rupture Disk

The scored tension-loaded rupture disk is designed to open along scored lines (see Figure 12). This type of disk is designed to allow a closer ratio (generally 85 percent) of system operating pressure to burst pressure. Because the score lines control the opening pattern, this type of disk is normally nonfragmenting. The scored tension-loaded rupture disk is manufactured from thicker material than is used for

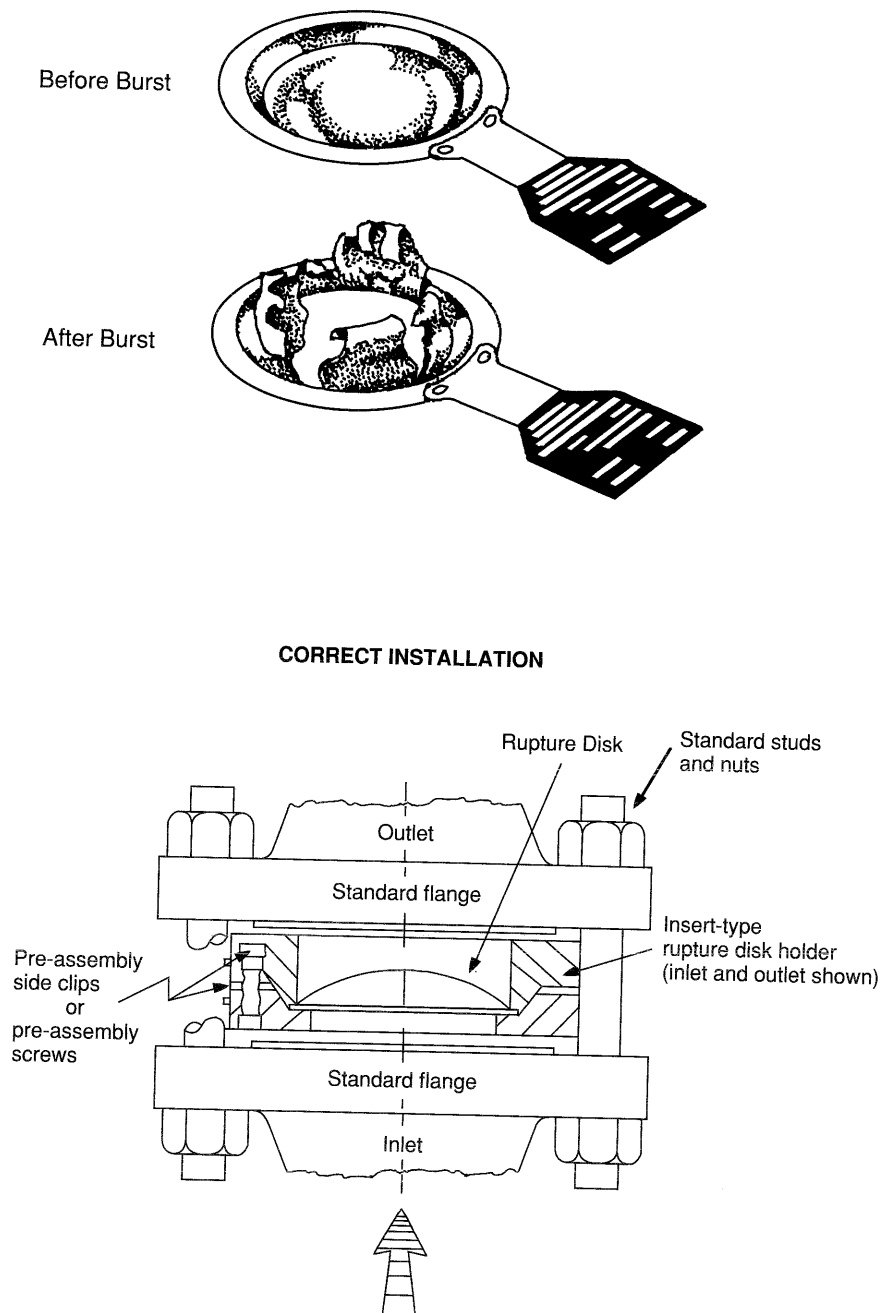
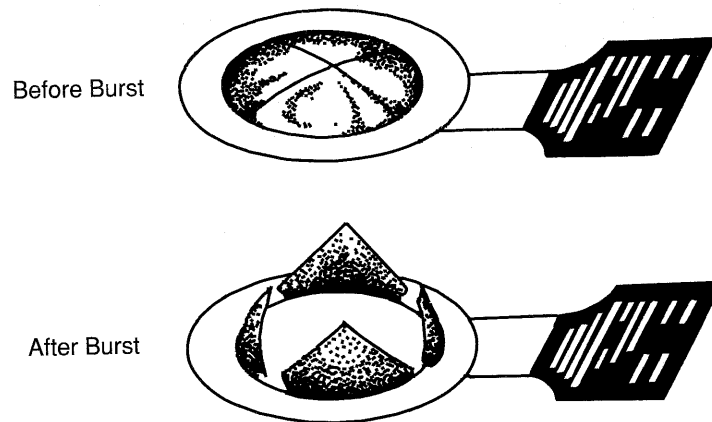


Figure 11—Conventional Rupture Disk



CORRECT INSTALLATION

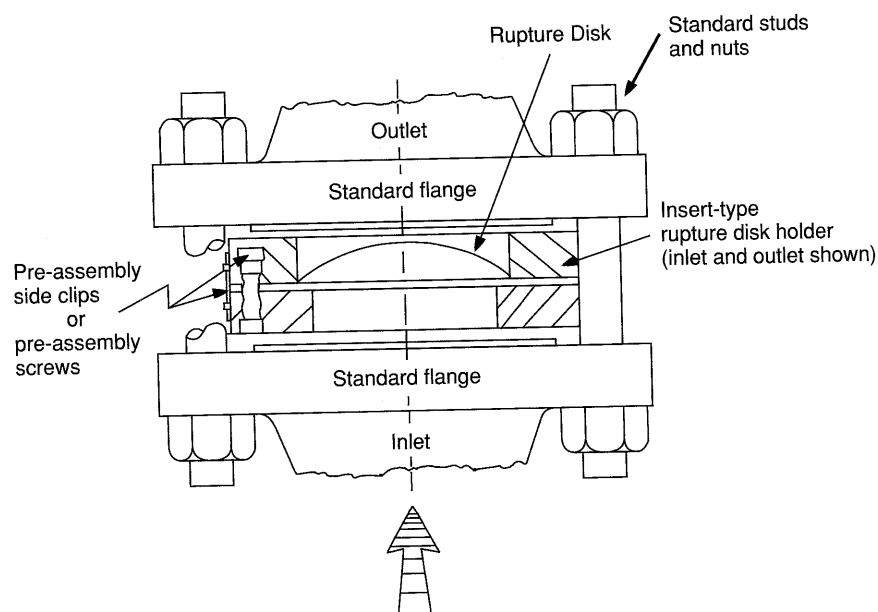


Figure 12—Scored Tension-Loaded Rupture Disk

unscored disks for the same burst pressure. The disk is mechanically scored to control the burst pressure and burst pattern. The increased material thickness of the disk provides additional resistance to mechanical damage; in most cases, the disk will withstand full vacuum without the addition of a separate vacuum support.

2.5.2.3 Composite Rupture Disk

A composite rupture disk is a flat or domed metallic or nonmetallic multipiece construction disk. The domed com-

posite rupture disk is designed to burst when it is overpressured on the concave side. The flat composite rupture disk is designed to burst when it is overpressured on the side designated by the manufacturer.

The domed composite rupture disk, available in flat-seat or angular-seat design, permits typical operation to 80 percent of the rated burst pressure under limited pressure cycling and temperature variations. The burst pressure is controlled by the combination of a slit top section and a metallic or nonmetallic seal member under the top section. Composite rupture disks are generally available in burst

pressures lower than those of the conventional domed rupture disks and may offer a longer service life as a result of the corrosion resistant properties of the seal material selected. The slit top section provides a predetermined opening pattern for the rupture disk and is designed to minimize fragmentation when the top section is supplied with a nonmetallic seal. As with the conventional domed rupture disk, a support is generally required when vacuum or back pressure conditions are present (see Figure 13).

A flat composite rupture disk is available for the protection of low pressure vessels or the isolation of equipment such as exhaust headers or the outlet sides of pressure relief valves. Intended as corrosion barriers only, flat composite rupture disks may typically operate at 50 percent of the rated burst pressure and are generally installed between compan-

ion flanges rather than specific rupture disk holders. Rupture disks that act in either direction are available; these provide positive pressure and vacuum protection.

2.5.2.4 Reverse-Acting Rupture Disk

A reverse-acting rupture disk is a domed solid metal disk designed to burst when it is overpressured on the convex side. Reverse-acting rupture disks are designed to open by such methods as shear, knife blades, knife rings, or scored lines (see Figures 14 and 15).

The reverse-acting rupture disk is designed to allow a closer ratio of system operating pressure to burst pressure, up to 90 percent of the rated burst pressure. This type of disk is usually nonfragmenting. Because the reverse-acting rupture

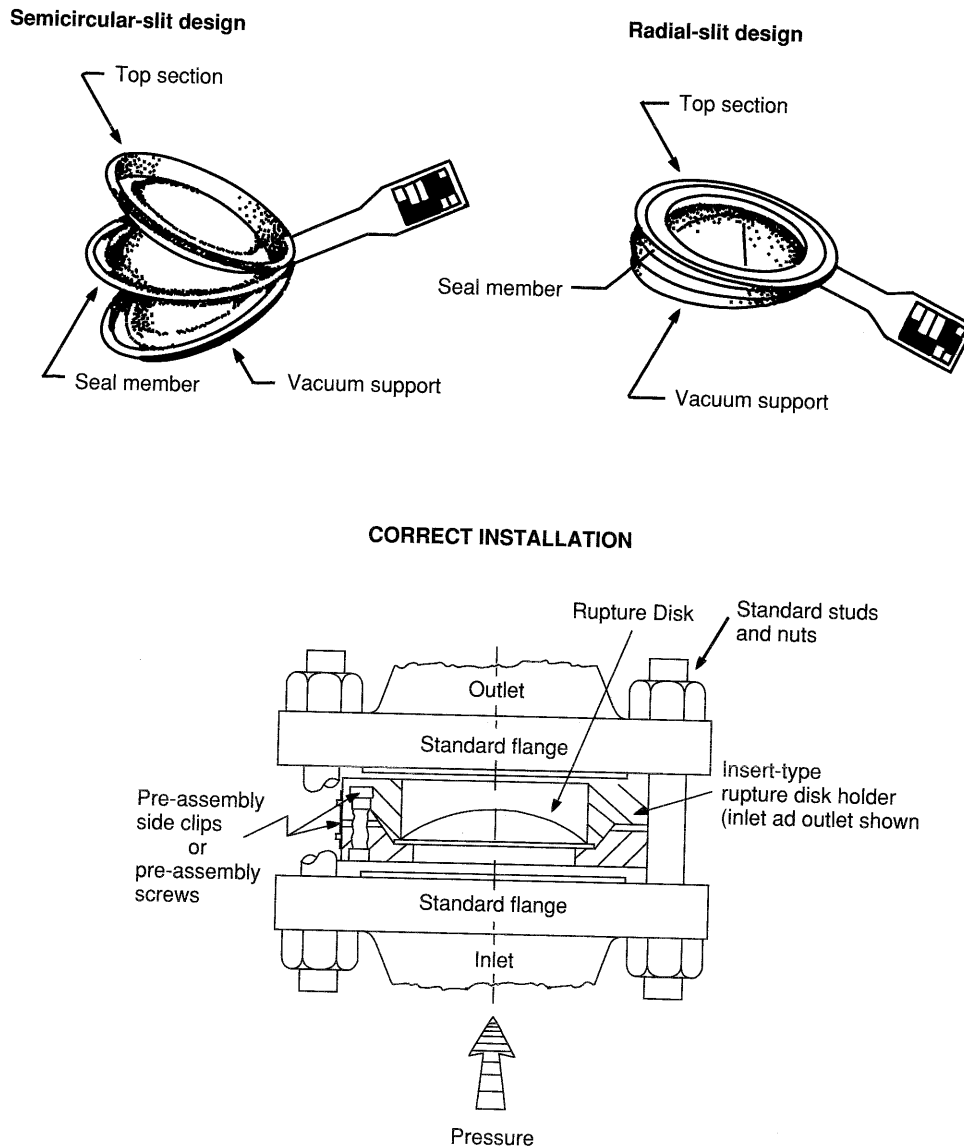


Figure 13—Domed Composite Rupture Disk

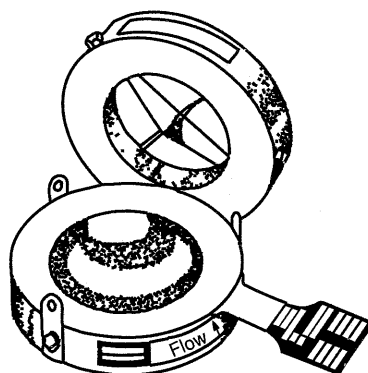
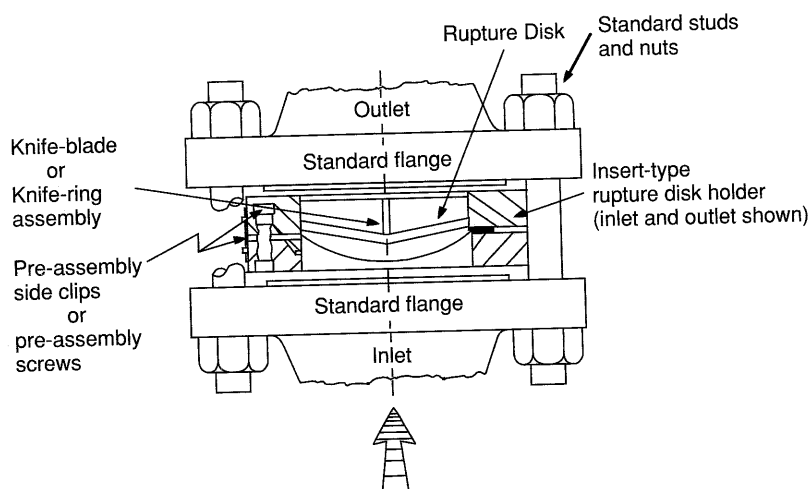
**CORRECT INSTALLATION**

Figure 14—Reverse-Acting Rupture Disk With Knife Blades

disk is actuated by overpressure on the convex side, thicker disk materials may be used, thereby improving corrosion resistance, eliminating the need for vacuum support, and providing longer service life under pressure/vacuum cycling conditions and temperature fluctuations.

2.5.2.5 Graphite Rupture Disk

A graphite rupture disk is manufactured from graphite impregnated with a binder material and designed to burst by bending or shearing (see Figure 16).

Graphite rupture disks are resistant to most acids, alkalis, and organic solvents. Operation to 70 percent of the rated burst pressure is generally permissible. A support may be required for disks that are rated 15 pounds per square inch gauge or less; a support also may be required under conditions of higher back pressure. Graphite rupture disks fragment upon rupture; provisions for capturing fragments may be required in certain applications.

2.5.3 APPLICATION OF RUPTURE DISKS

The ratio of the maximum operating pressure to the actual burst pressure rating is a primary factor in selecting a rupture disk and its associated manufacturing range.

The maximum expected operating pressure should be sufficiently below the design pressure of the vessel to prevent premature failure of the rupture disk due to fatigue or creep.

A rupture disk is a pressure differential device. The designer must consider pressures on both sides of the disk when the disk burst pressure is specified. In addition, the maximum operating percentage and other service conditions should be documented on the specification sheet in Appendix A.

The rupture disk is a temperature-sensitive device. Burst pressures can vary significantly with the temperature at the location of the rupture disk device. This temperature may be different from the normal fluid operating temperature. As the temperature at the disk increases, the burst pressure usually

decreases. Since the effect of temperature depends on the rupture disk and material, the manufacturer should be consulted. The rupture disk must be specified at the temperature and pressure at which the disk is expected to burst.

The application of rupture disk devices to liquid service

should be carefully evaluated to ensure that the design of the rupture disk device and the dynamic energy of the system on which it is installed will result in sufficient opening of the rupture disk. The user should consult the rupture disk manufacturer for information about liquid service applications.

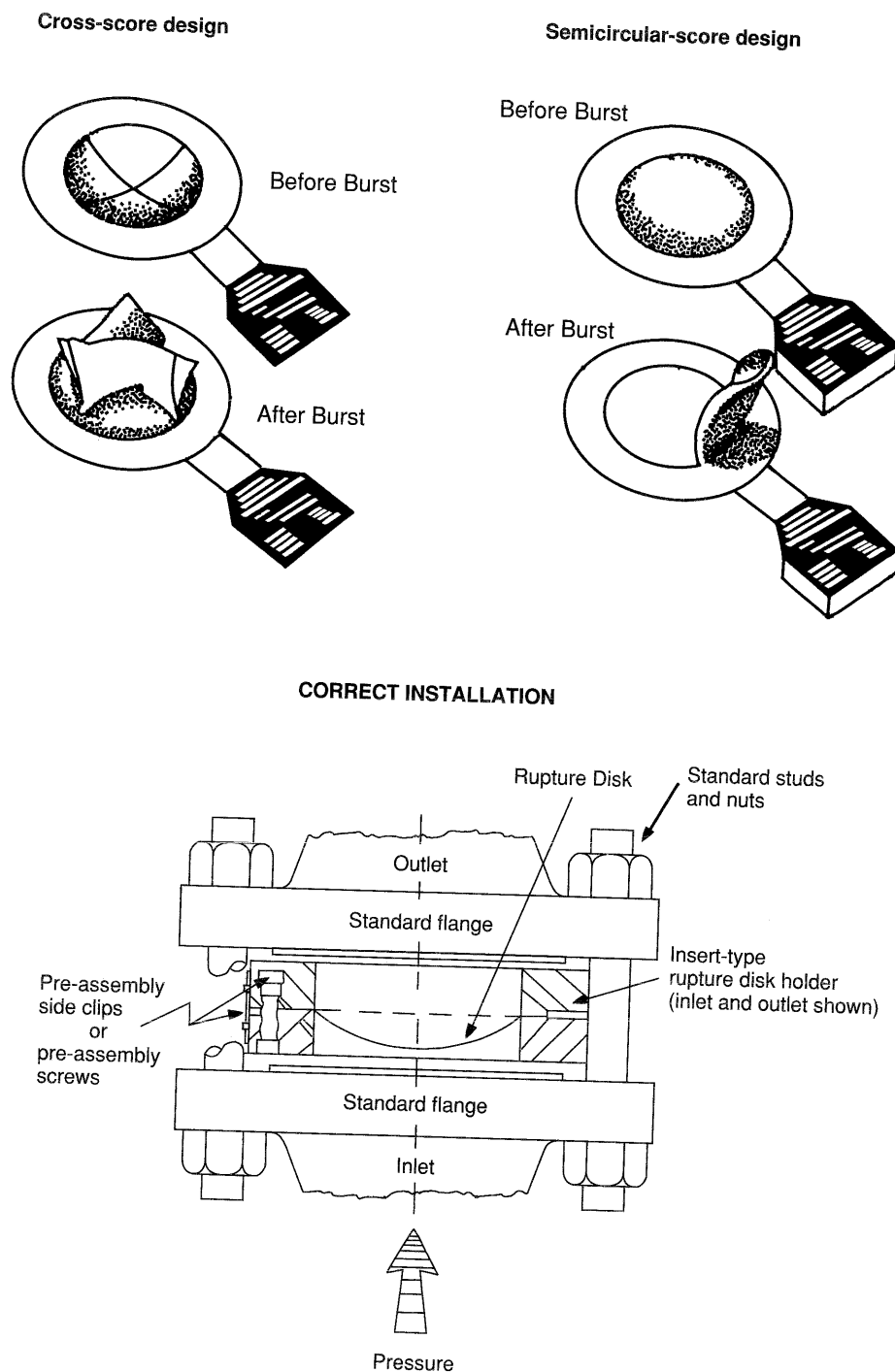


Figure 15—Reverse-Acting Rupture Disk With Scored Lines

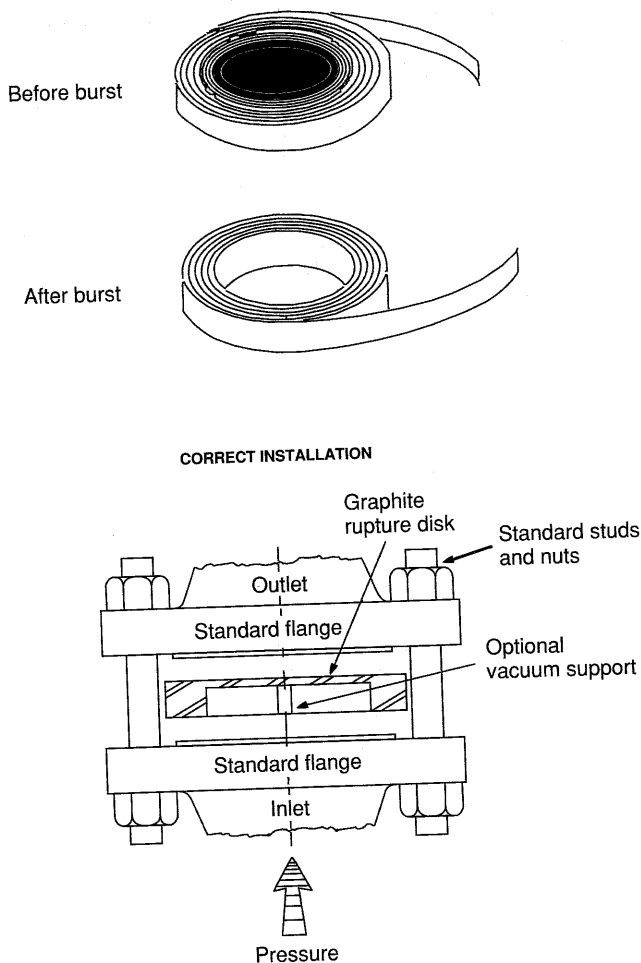


Figure 16—Graphite Rupture Disk

A rupture disk device may be used as a primary or supplemental pressure relief device. It may be used independently or in conjunction with pressure relief valves.

2.5.4 TERMINOLOGY FOR RUPTURE DISK DEVICES

Terms used specifically for rupture disk devices are defined in 2.5.4.1 through 2.5.4.4.

2.5.4.1 *Specified burst pressure* is the pressure requested at a coincident temperature when the burst requirements are specified.

2.5.4.2 The *stamped burst pressure*, or *rated burst pressure*, is the value of the pressure differential across the rupture disk at a coincident temperature at which the rupture disk is designed to burst. This type of pressure is derived from destructive tests performed from each rupture disk lot at the time of manufacture. The stamped burst pressure is marked on the rupture disk. Rupture disks that are manufactured at zero manufacturing range will typically be stamped at the specified burst pressure.

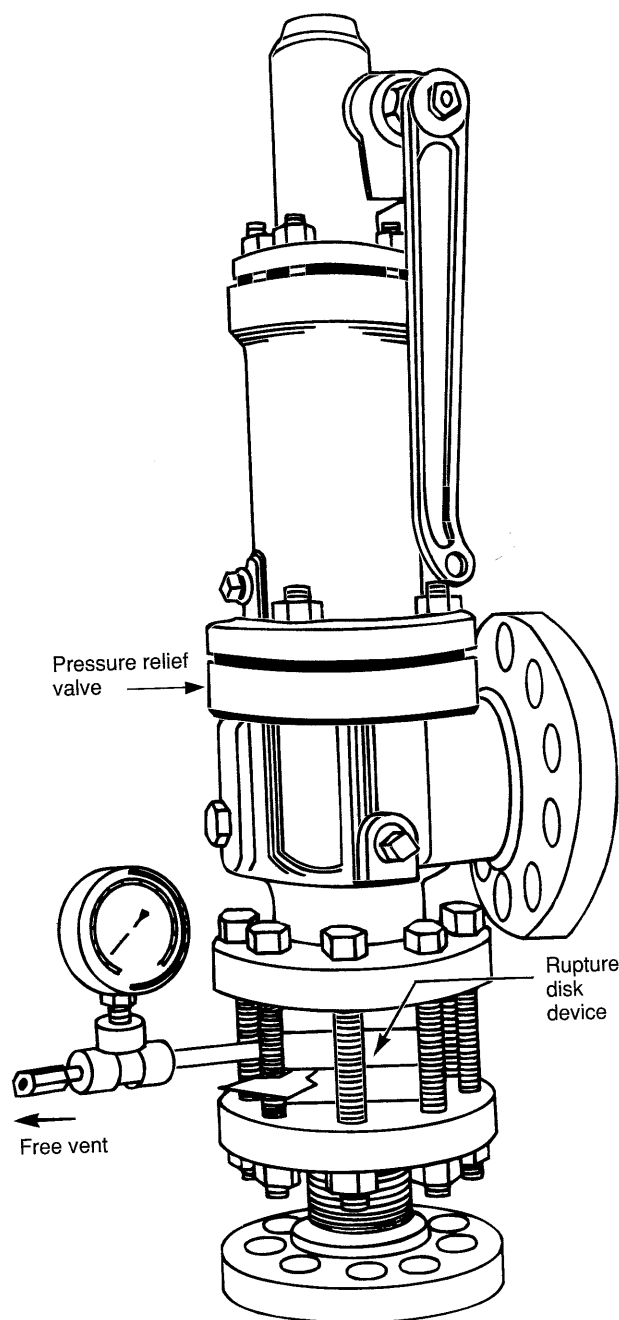


Figure 17—Rupture Disk Device in Combination With a Pressure Relief Valve

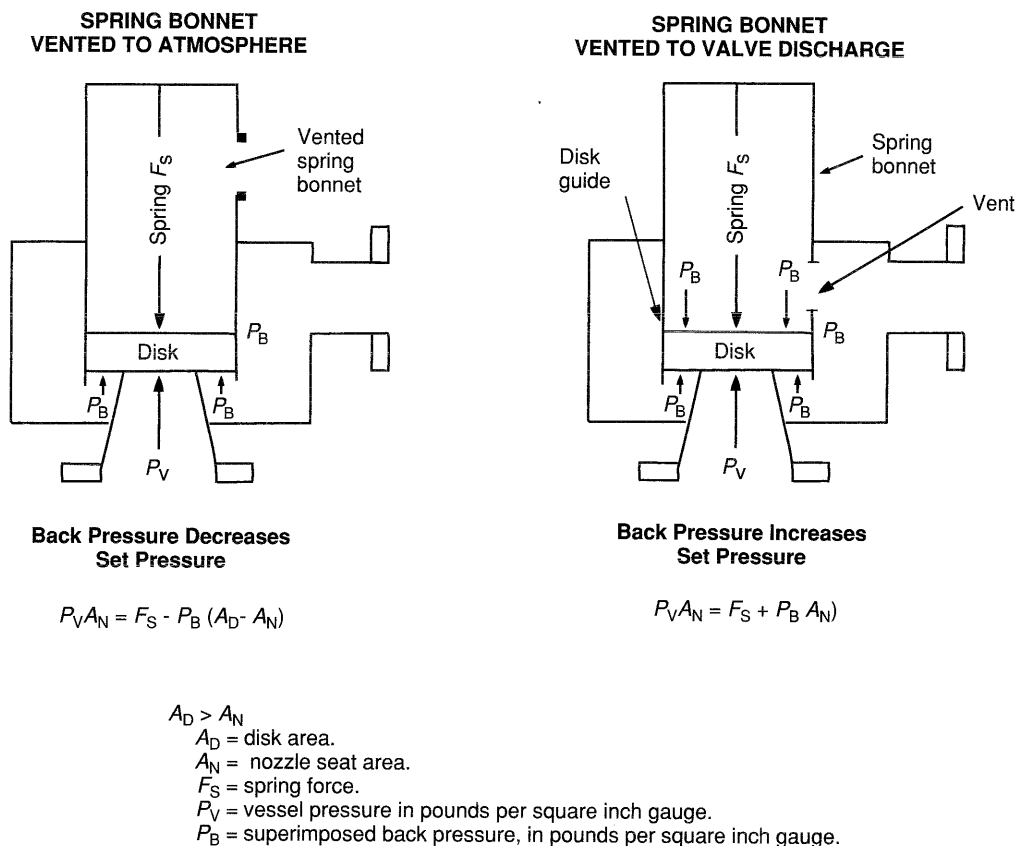


Figure 18—Typical Effects of Superimposed Back Pressure on the Opening Pressure of Conventional Pressure Relief Valves

2.5.4.3 The *manufacturing range*, agreed upon between the customer and the rupture disk manufacturer, is the pressure range in which the specified burst pressure must lie. The manufacturing range must always be evaluated before the specified burst pressure is selected to ensure that the stamped burst pressure will be within applicable ASME Code pressure limits. Manufacturing ranges generally depend on (a) the specified burst pressure level, (b) the rupture disk design type, and (c) the rupture disk manufacturer. Manufacturing ranges are expressed as (a) plus or minus a percentage of the specified pressure, (b) plus or minus pressure units, or (c) 0 percent or zero pressure units.

2.5.4.4 *Burst-pressure tolerance* is the variation around the stamped burst pressure in which the individual rupture disk burst pressures in a lot are required by the ASME Code to fall. For example, if the stamped burst pressure is 100 pounds per square inch gauge, all disks within the lot are required to burst 100 pounds per square inch gauge plus or minus the burst pressure tolerance.

Note: Figure 25 shows the relationships between various terms used in connection with rupture disk devices.

2.6 Rupture Disk Devices in Combination With Pressure Relief Valves

2.6.1 GENERAL

Using rupture disk devices in combination with pressure relief valves is permitted and may be advisable for systems containing substances that may corrode the pressure relief valve or render it inoperative. Using rupture disk devices in combination with pressure relief valves may also be advisable for minimizing the loss of valuable, noxious, or hazardous materials that may leak through the pressure relief valve (see Figure 17).

2.6.2 RUPTURE DISK DEVICES AT THE INLET OF A PRESSURE RELIEF VALVE

2.6.2.1 The capacity of a rupture disk device in combination with a pressure relief valve, where the rupture disk device is located at the valve inlet, may be determined by multiplying the ASME stamped valve capacity by the combination capacity factor, K_C . The value of K_C is 0.90 unless

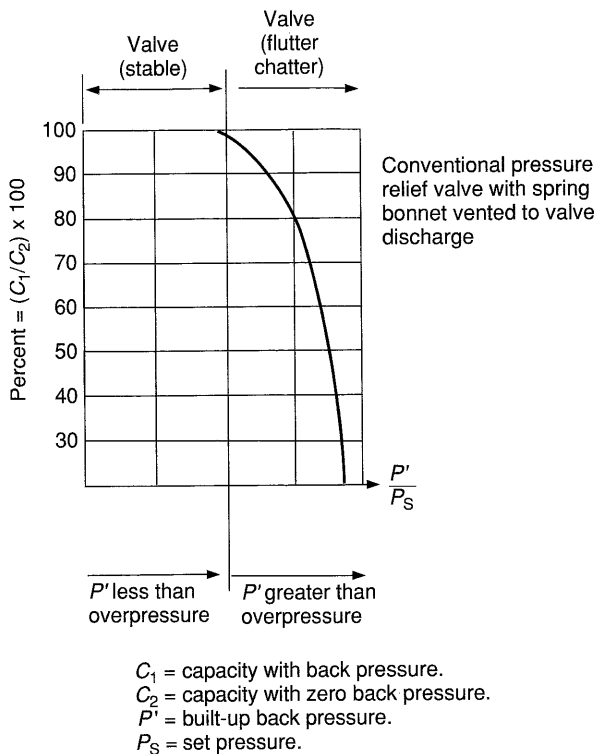
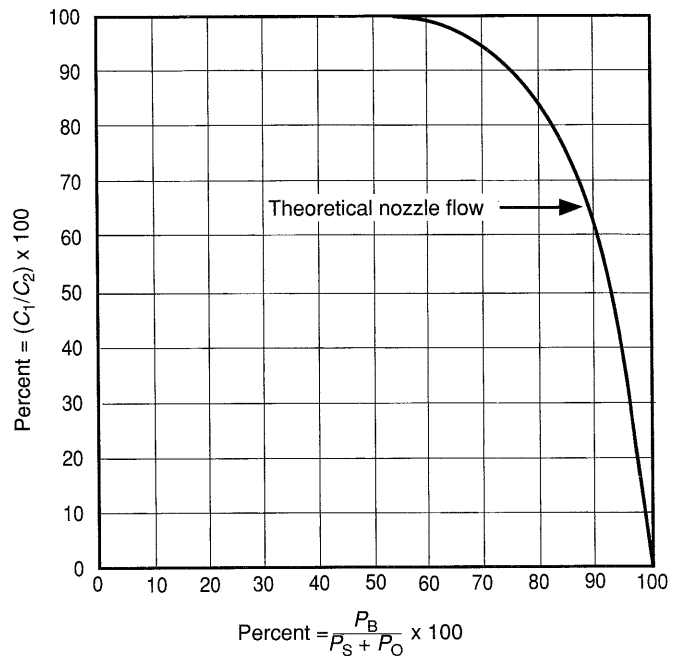
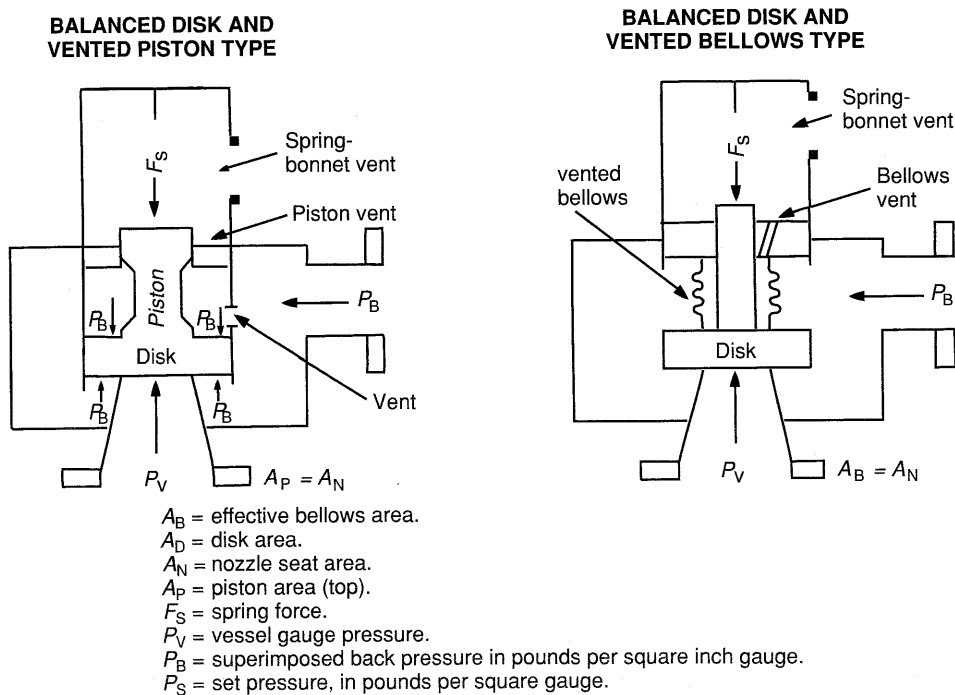


Figure 19—Typical Effects of Built-Up Back Pressure on the Capacity of Conventional Pressure Relief Valves



C_1 = capacity with back pressure.
 C_2 = capacity with zero back pressure.
 P_B = back pressure, in pounds per square inch absolute.
 P_S = set pressure, in pounds per square inch absolute.
 P_O = overpressure, in pounds per square inch.

Figure 20—Theoretical Performance for a Nozzle Discharging a Gas That Has a Specific Heat Ratio of 1.3



Note: In this figure, $P_V = P_S$; $(P_V)(A_N) = F_S$ (typical); and $P_S = F_S/A_N$.

Figure 21—Typical Effects of Back Pressure on the Set Pressure of Balanced Pressure Relief Valves

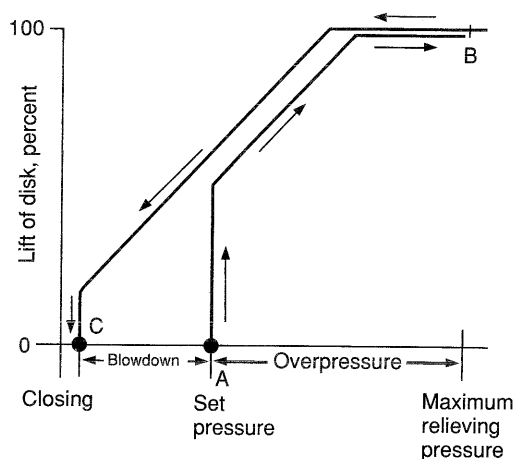


Figure 22—Typical Relationship Between Lift of Disk in a Pressure Relief Valve and Vessel Pressure

the manufacturer has established a certified combination capacity factor in accordance with Section VIII of the ASME Code.

2.6.2.2 When a rupture disk device is used in combination with a pressure relief valve and the devices are close coupled, the specified burst pressure and set pressure should be the same nominal value. The appropriate manufacturers

should be consulted for other arrangements and settings.

2.6.2.3 The space between the rupture disk and the pressure relief valve should have a free vent, pressure gauge, trycock, or suitable pressure indicator as required in Section VIII, Division I, of the ASME Code. If the process involves a hazardous or flammable material, a free vent to the atmosphere should be disposed of safely.

2.6.3 RUPTURE DISK DEVICES AT THE OUTLET OF A PRESSURE RELIEF VALVE

A rupture disk device may be installed on the outlet of a pressure relief valve to protect the valve from atmospheric or downstream fluids or to prevent hazardous or flammable materials from leaking through the outlet of the pressure relief valve to the atmosphere. The installation, capacity, and effects of back pressure on the pressure relief valve due to this rupture disk device shall be in accordance with the manufacturer's recommendations and the applicable ASME Code requirements.

2.7 Other Types of Devices

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

SECTION 3—CAUSES OF OVERPRESSURE

3.1 General

Causes of overpressure can range from a single contingent event to a complex combination of events. Table 1 summarizes the principal operational difficulties that give rise to the indicated relief capacities. Not all causes will occur simulta-

neously; therefore, the pressure relief devices should be sized for the condition or conditions that require the largest relief area. Pressure relief devices are not required on every vessel but are required on each vessel or group of vessels that may be isolated by control valves or other valves (see Appendix C).

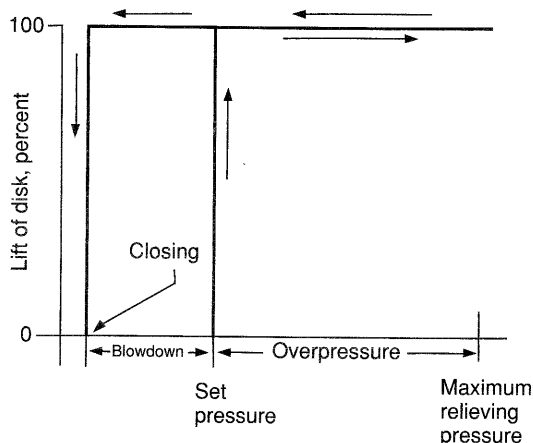


Figure 23—Typical Relationship Between Lift of Disk in a Pop-Action Pilot-Operated Relief Valve and Vessel Pressure

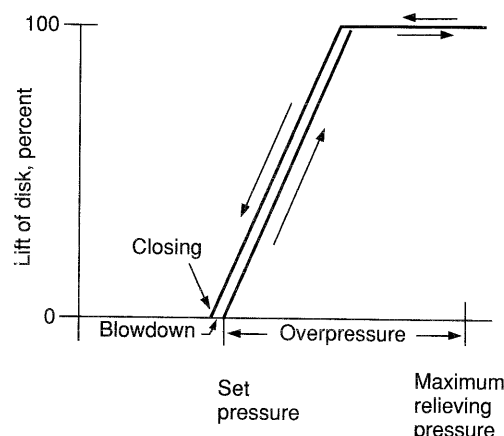
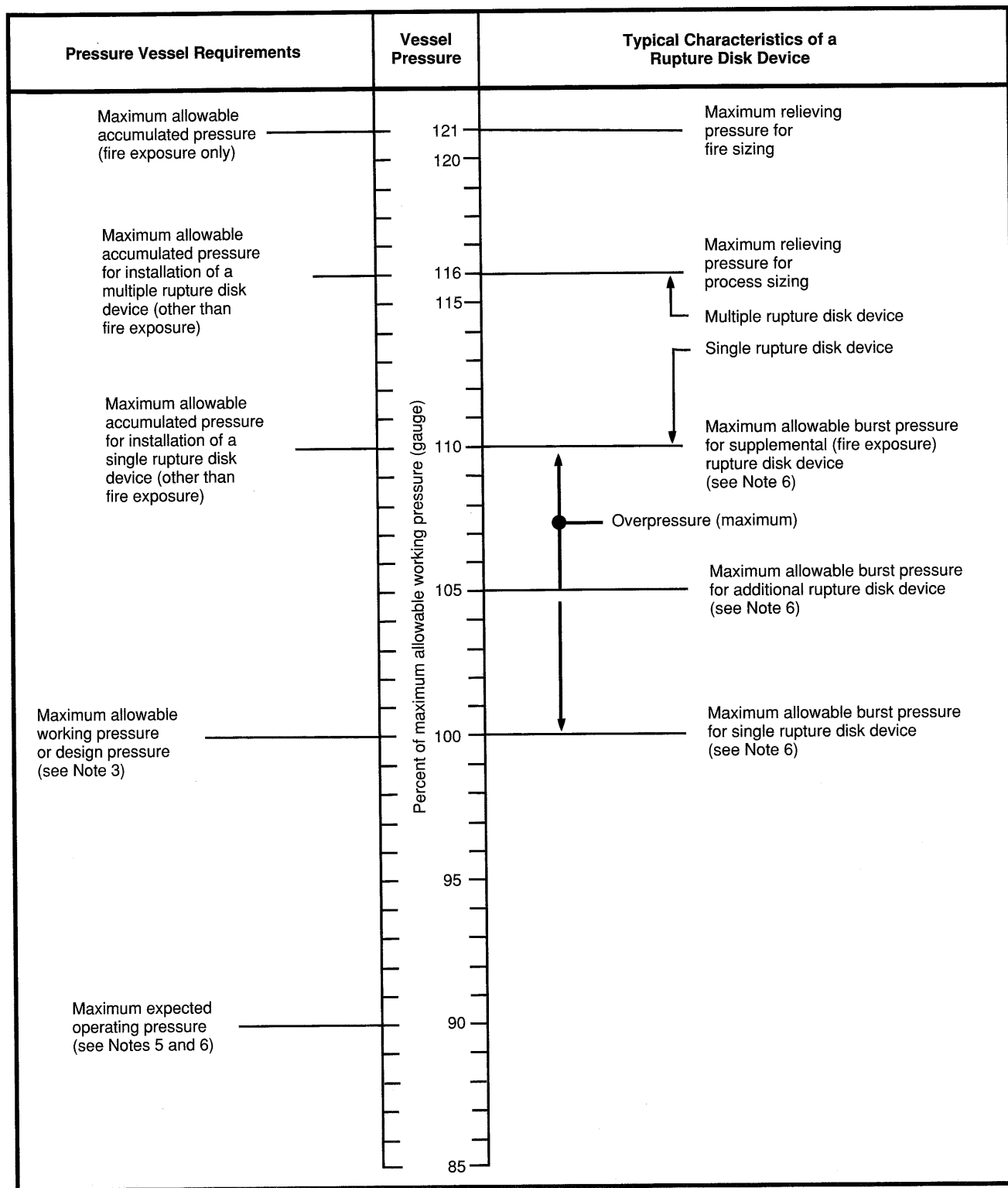


Figure 24—Typical Relationship Between Lift of Disk in a Modulating-Action Pilot-Operated Relief Valve and Vessel Pressure



Notes:

1. This figure conforms with the requirements of Section VIII of the ASME Boiler and Pressure Vessel Code.
2. The pressure conditions shown are for rupture disk devices installed on a pressure vessel.
3. The margin between the maximum allowable working pressure and the operating pressure must be considered in the selection of a rupture disk.

4. The allowable burst-pressure tolerance will be in accordance with the applicable code.

5. The operating pressure may be higher or lower than 90 depending on the rupture disk design.

6. The stamped burst pressure of the rupture disk may be any pressure at or below the maximum allowable burst pressure.

Figure 25—Pressure-Level Relationships for Rupture Disk Devices

A detailed description of the various causes of overpressure is given in API Recommended Practice 521.

3.2 Process Causes (Excluding Fire)

An imbalance of fluid flow rates in and out of process equipment and vessels during operation results in an accumulation of liquids or vapors and may produce pressure that exceeds the operating pressure. If the pressure is not controlled, it must be relieved through a relieving device at a pressure that does not exceed a certain percentage of the allowable accumulation specified in the ASME Code.

The simplest case is that of a pressure vessel into which liquid is pumped. The discharge pressure developed by the

pump may be greater than the maximum allowable working pressure of the vessel. In this case, a pressure relief device must be provided to protect the vessel against excessive pressure when the vessel becomes full. A means of warning the operator against overfilling the vessel may be provided by the designer. A means of automatically shutting down the pump may also be provided; nevertheless, a pressure relief device must be installed.

A more involved relief problem is a fractionating tower in which vapor may be generated and accumulated as a result of maloperation of the condensing and reflux system or the reboiler and vapor-generating system. The fractionating tower will experience rising pressure that the pressure relief device will limit.

Table 1—Bases for Relief Capacities Under Selected Conditions

Item No.	Condition	Pressure Relief Device (Liquid Relief)	Pressure Relief Device (Vapor Relief) ^a
1	Closed outlets on vessels	Maximum liquid pump-in rate	Total incoming steam and vapor plus that generated therein at relieving conditions
2	Cooling water failure to condenser	—	Total vapor to condenser at relieving conditions
3	Top-tower reflux failure	—	Total incoming steam and vapor plus that generated therein at relieving conditions less vapor condensed by sidestream reflux
4	Sidestream reflux failure	—	Difference between vapor entering and leaving section at relieving conditions
5	Lean oil failure to absorber	—	None, normally
6	Accumulation of noncondensables	—	Same effect in towers as found for Item 2; in other vessels, same effect as found for Item 1
7	Entrance of highly volatile material	—	For towers, usually not predictable
	Water into hot oil	—	For heat exchangers, assume an area twice the internal cross-sectional area of one tube to provide for the vapor generated by the entrance of the volatile fluid due to tube rupture
	Light hydrocarbons into hot oil	—	—
8	Overfilling storage or surge vessel	Maximum liquid pump-in rate	—
9	Failure of automatic controls	—	Must be analyzed on a case-by-case basis
10	Abnormal heat or vapor input	—	Estimated maximum vapor generation including noncondensables from overheating
11	Split exchanger tube	—	Steam or vapor entering from twice the cross-sectional area of one tube; also same effects found in Item 7 for exchangers
12	Internal explosions	—	Not controlled by conventional relief devices but by avoidance of circumstances
13	Chemical reaction	—	Estimated vapor generation from both normal and uncontrolled conditions
14	Hydraulic expansion	—	—
	Cold fluid shut in	See C.2	—
	Lines outside process area shut in	See C.2	—
15	Exterior fire	—	Estimate by the method given in D.5
16	Power failure (steam, electric, or other)	—	Study the installation to determine the effect of power failure; size relief valve for the worst condition that can occur
	Fractionators	—	All pumps could be down, with the result that reflux and cooling water would fail
	Reactors	—	Consider failure of agitation or stirring, quench or retarding stream; size valves for vapor generation from a runaway reaction
	Air-cooled exchangers	—	Fans would fail; size valves for the difference between normal and emergency duty
	Surge vessels	Maximum liquid inlet rate	—

^aConsideration may be given to the suppression of vapor production as the result of the device's relieving pressure being above operating pressure, assuming constant heat input. (Procedures for sizing pressure relief devices are presented in Section 4.)

Equipment that is completely full of liquid (that is, hydrostatic) presents a problem of liquid hydraulic expansion if it is blocked in while heat is added. In many cases, this potential risk can be controlled by design or procedures (see API Recommended Practice 521). To protect piping or equipment against overpressure resulting from ambient temperature changes, a nominal size relief device (for example, a 3/4-inch x 1-inch relief valve) will normally suffice. A method of calculating the required expansion rate is given in Appendix C.

Equipment that is not capable of withstanding external atmospheric pressure should be provided with a vacuum relief device if it is likely to be subjected to vacuum.

3.3 Fire

3.3.1 GENERAL

It is assumed that the user will comply with all safety and fire protection measures, as recommended in API Recommended Practice 2001. This assumption is important in estimating the heat input rate of a fire, since conditions such as surface drainage and firewater application play a vital role in limiting heat input.

Any pressure vessel in an operating plant that handles or processes flammable liquids or gases may be exposed to fire at some time in its life. This hazard may exist even though the contents of the vessel itself are not flammable; therefore, for pressure vessels that contain liquids, a pressure relief device should be provided to relieve overpressure generated by fire heat input.

For pressure vessels that contain only vapor, consideration should be given to providing a means for reducing pressure to near atmospheric and reliably restricting the fire heat input, since a pressure-relieving device may not protect the vessel from bursting when it is exposed to fire.

A fire may occur if any flammable material from refinery equipment is accidentally ignited. The source of the liquid or gas release that feeds the fire may be either the vessel exposed to the flame or an adjacent operating or storage vessel. The release may result from leaking joints in a pipeline or other equipment or from operational mishaps. The escaping flammable fluids may be carried some distance from the source of the leak. Liquids may be carried by the natural slope of the ground surface; gases or vapors may be carried by air currents; fluids escaping a pressure source may be carried by a jetting stream.

With the sudden release of a considerable quantity of flammable material that was mostly contained as a liquid at or above its vapor pressure corresponding to the surrounding atmospheric (ambient) temperature, a portion of the outflow discharged as a liquid will flash. Unless drained away, the re-

maining liquid will form a pool and provide a source of fuel if it is ignited (see Appendix D).

If an open, free-burning fire occurs, heat will be absorbed by the vessel and other equipment exposed to the flame either by radiation or by direct contact from hot gases or flames or both. If the heat absorption continues long enough, the contents of the vessel will be heated, and the pressure will rise until the pressure relief device opens. The release of vapors through the pressure relief device will serve to limit the maximum pressure in the vessel. If the rate of vapor generation is greater than the rated capacity of the valve, the pressure will build up beyond the permissible accumulation and may reach a pressure that is unsafe for the vessel. Considering the possibilities of fire exposure is therefore a necessity in determining the size of a pressure relief device. The heat input to a vessel under these conditions may be estimated by the method given in D.5.1.

If a flammable material leak (caused by the failure of a pipeline or nipple or by a gasket blowing out between flanges) is ignited, a jet fire may result. No attempt has been made to evaluate this type of fire.

3.3.2 EFFECT OF FIRE ON THE WETTED SURFACE OF A VESSEL

If the critical pressure of the fluid exceeds the relieving pressure, radiation to the wetted surface of a partially filled vessel (assuming no internal insulation) will be absorbed as sensible heat; the vessel and liquid temperatures will be increased and will essentially be equal. At the boiling point, the radiation will be absorbed by the liquid almost totally as latent heat. The vessel temperature may rise slightly higher than the boiling temperature, depending on the boiling mechanism, but for practical purposes, the temperatures are assumed to be equal. The heat absorption and consequent vapor generation will cause the pressure in the vessel to rise until it reaches the relieving pressure of the pressure relief device provided.

As long as the vapor quantity generated under these conditions is below the flow capacity of the pressure relief valve, the valve when opened will discharge until the pressure in the vessel drops to the valve reseating pressure and the valve closes. Intermittent valve operations will continue in this range of heat input.

Internal insulation is likely to seriously affect the temperature of the vessel wall by reducing the amount of heat absorbed by the liquid. In this case, the vessel wall may quickly reach temperatures at which the stress level from pressure is adequate to cause short-term creep rupture (see D.3).

The user should consider other methods of protecting vessels against fire exposure in addition to the use of pressure relief devices (see D.8).

3.3.3 EFFECT OF FIRE ON THE UNWETTED SURFACE OF A VESSEL

Unwetted vessel surfaces exposed to fire do not have the benefit of the latent heat absorption of liquid to mitigate a rise in metal temperature. Where only vapor is present to

absorb heat internally, the wall temperature can rise quite rapidly; this may soon lead to vessel rupture. Pressure relief valves may provide protection for only a short period; other devices would be required to prevent vessel rupture (see D.3, D.5.2, and D.8).

SECTION 4—PROCEDURES FOR SIZING

4.1 Determination of Relief Requirements

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

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

Appendix D provides relieving flow rates for fire conditions. Table 1 lists a number of common operational conditions for which overpressure protection may be required. This list is by no means complete; each plant may have unique features that must be considered in addition to those listed in Table 1. (See API Recommended Practice 521 for a detailed discussion of relief requirements.)

Pressure relief valves may be sized using the equations presented in 4.3 through 4.5 as appropriate for vapors, gases, or liquids. These equations are used to calculate the effective nozzle area necessary to achieve a required flow rate through the valve. A valve is then chosen for the application that has an effective area equal to or greater than the calculated required effective area.

The effective areas and assumed discharge coefficient, $K_d = 0.975$, are generally different from actual orifice areas and discharge coefficients that are used to determine certified valve capacities. However, effective areas calculated using the equations in 4.3 through 4.5 will result in the selection of valves with certified capacities that equal or exceed the required capacities.

The effective-area concept allows for the selection of valve size independent of the manufacturer. Standard effective orifice areas and corresponding letter designations may be found in API Standard 526.

4.2 Relieving Pressure

4.2.1 GENERAL

Relieving pressure, shown as P_1 in the various sizing equations, is the inlet pressure of the pressure relief device at relieving conditions. The relieving pressure is the total of set pressure plus overpressure plus atmospheric pressure. The examples cited in this section for the determination of relieving pressure refer to pressure relief valves; however, they are also applicable to rupture disk devices. (See Figures 1 and 25 for pressure-level relationships for these types of devices.)

The allowable overpressure is established from the accumulation permitted by the applicable code. The allowable overpressure may vary for different applications depending on the relationship of the set pressure to the maximum allowable working pressure of the vessel or system that is protected. Allowable overpressure is the same as allowable accumulation only when the set pressure is equal to the maximum allowable working pressure.

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

Sections 4.2.2 through 4.2.4 discuss methods of determining the relieving pressure for pressure relief valves in gas and vapor service. Standard atmospheric pressure (14.7 pounds per square inch absolute) is used for gauge/absolute pressure conversion in these sections. For design, barometric pressure corresponding to site elevation should be used.

Relieving pressure for pressure relief valves in liquid service is determined in a manner similar to that used for vapor service except that the relieving pressure is expressed in gauge rather than absolute units. In the case of ASME-application liquid service valves (that is, for protection of a liquid-full vessel), maximum accumulated pressure is limited to 110 percent of the maximum allowable working pressure for operating contingencies—the same constraint as for vapor service. In the case of non-ASME-application liquid service valves (that is, for protection of piping without vessels in-

cluded), 25-percent overpressure is generally specified.

Table 2 summarizes the maximum accumulation and set pressure for pressure relief valves specified in accordance with the ASME Code.

4.2.2 OPERATING CONTINGENCIES

4.2.2.1 Single-Valve Installation

In accordance with the requirements of the ASME Code, accumulated pressure shall be limited to 110 percent of the maximum allowable working pressure in vessels that are protected by a single pressure relief valve sized for operating (nonfire) contingencies. The set pressure of the valve shall not exceed the maximum allowable working pressure.

Note: Allowable accumulation is 3 pounds per square inch when the maximum allowable working pressure is between 15 and 30 pounds per square inch gauge.

Table 3 shows an example determination of relieving pressure for a single valve whose set pressure is less than or equal to the vessel's maximum allowable working pressure.

4.2.2.2 Multiple-Valve Installation

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

In accordance with the requirements of the ASME Code, accumulated pressure shall be limited to 116 percent of the maximum allowable working pressure in vessels that are protected by multiple valves sized for operating (nonfire) contingencies. The set pressure of the first valve shall not exceed the maximum allowable working pressure. The set pressure of the additional valve or valves shall not exceed 105 percent of the maximum allowable working pressure.

Note: Allowable accumulation is 4 pounds per square inch when the maximum allowable working pressure is between 15 and 25 pounds per square inch gauge.

Table 4 shows an example determination of relieving pressure for a multiple-valve installation in which the set pressure of the first valve is equal to the maximum allowable

working pressure of the vessel, and the set pressure of the additional valve is 105 percent of the vessel's maximum allowable working pressure.

4.2.3 FIRE CONTINGENCIES

4.2.3.1 General

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

Single or multiple valves sized for fire may also be utilized for relieving requirements attributed to secondary operating (nonfire) contingencies, if applicable, provided that the constraint of 110-percent (of the maximum allowable working pressure) accumulated pressure for the nonfire contingencies is observed.

4.2.3.2 Single-Valve Installation

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

Table 5 shows an example determination of relieving pressure for a single valve whose set pressure is less than or equal to the vessel's maximum allowable working pressure.

4.2.3.3 Multiple-Valve Installation

A multiple-valve installation requires the combined capacity of two or more valves to alleviate overpressure from a fire. The set pressure of the first valve to open shall not exceed the maximum allowable working pressure. The set pressure of the last valve to open shall not exceed 105 percent of the maximum allowable working pressure.

Table 6 shows an example determination of relieving pressure for a multiple-valve installation in which the set pressure of the first valve is equal to the vessel's maximum allowable working pressure, and the set pressure of the additional valve is 105 percent of the vessel's maximum allowable working pressure.

Table 2—Set Pressure and Accumulation Limits for Pressure Relief Valves

Contingency	Single-Valve Installations		Multiple-Valve Installations	
	Set Pressure (percent)	Maximum Accumulated Pressure (percent)	Set Pressure (percent)	Maximum Accumulated Pressure (percent)
Nonfire only				
First valve	100	110	100	116
Additional valve(s)	—	—	105	116
Fire only				
First valve	100	121	100	121
Additional valve(s)	—	—	105	121
Supplemental valve	—	—	110	121

Note: All values are percentages of the maximum allowable working pressure.

Table 3—Example Determination of Relieving Pressure for a Single Valve (Operating Contingencies)

Characteristic	Value
Valve Set Pressure Less Than MAWP	
Protected vessel MAWP, psig	100
Maximum accumulated pressure, psig	110
Valve set pressure, psig	90
Allowable overpressure, psi	20
Relieving pressure, P_1 , psia	124.7
Valve Set Equal to MAWP	
Protected vessel MAWP, psig	100
Maximum accumulated pressure, psig	110
Valve set pressure, psig	100
Allowable overpressure, psi	10
Relieving pressure, P_1 , psia	124.7

4.2.3.4 Supplemental-Valve Installation

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

Supplemental valves are used only in addition to valves sized for operating (nonfire) contingencies.

Table 7 shows an example determination of relieving pressure for a supplemental-valve installation in which the set pressure of the first (nonfire) valve does not exceed the vessel's maximum allowable working pressure (see 4.2.1 for determination of relieving pressure), and the set pressure of the supplemental valve is 110 percent of the vessel's maximum allowable working pressure.

4.2.4 STEAM SERVICE

Valves for pressure vessels in steam service, excluding boilers designed in accordance with Section I of the ASME

Table 4—Example Determination of Relieving Pressure for a Multiple-Valve Installation (Operating Contingencies)

Characteristic	Value
First Valve (Set Pressure Equal to MAWP)	
Protected vessel MAWP, psig	100
Maximum accumulated pressure, psig	116
Valve set pressure, psig	100
Allowable overpressure, psi	16
Relieving pressure, P_1 , psia	130.7
Additional Valve (Set Pressure Equal to 105 Percent of MAWP)	
Protected vessel MAWP, psig	100
Maximum accumulated pressure, psig	116
Valve set pressure, psig	105
Allowable overpressure, psi	11
Relieving pressure, P_1 , psia	130.7

Table 5—Example Determination of Relieving Pressure for a Single Valve (Fire Contingencies)

Characteristic	Value
Valve Set Pressure Less Than MAWP	
Protected vessel MAWP, psig	100
Maximum accumulated pressure, psig	121
Valve set pressure, psig	90
Allowable overpressure, psi	31
Relieving pressure, P_1 , psia	135.7
Valve Set Pressure Equal to MAWP	
Protected vessel MAWP, psig	100
Maximum accumulated pressure, psig	121
Valve set pressure, psig	100
Allowable overpressure, psi	21
Relieving pressure, P_1 , psia	135.7

Code, should be specified as discussed above, depending on the contingency.

4.3 Sizing for Gas or Vapor Relief

4.3.1 CRITICAL FLOW BEHAVIOR

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

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

Table 6—Example Determination of Relieving Pressure for a Multiple-Valve Installation (Fire Contingencies)

Characteristic	Value
First Valve (Set Pressure Equal to MAWP)	
Protected vessel MAWP, psig	100
Maximum accumulated pressure, psig	121
Valve set pressure, psig	100
Allowable overpressure, psi	21
Relieving pressure, P_1 , psia	135.7
Additional Valve (Set Pressure Equal to 105 Percent of MAWP)	
Protected vessel MAWP, psig	100
Maximum accumulated pressure, psig	121
Valve set pressure, psig	105
Allowable overpressure, psi	16
Relieving pressure, P_1 , psia	135.7

Table 7—Example Determination of Relieving Pressure for a Supplemental-Valve Installation (Fire Contingencies)

Characteristic	Value
Supplemental Valve (Set Pressure Equal to 110 Percent of MAWP)	
Protected vessel MAWP, psig	100
Maximum accumulated pressure, psig	121
Valve set pressure, psig	110
Allowable overpressure, psi	11
Relieving pressure, P_1 , psia	135.7

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

The critical flow pressure ratio in absolute units may be estimated using the ideal gas relationship in Equation 1:

$$\frac{P_{cf}}{P_1} = \left[\frac{2}{k+1} \right]^{k/(k-1)} \quad (1)$$

Where:

P_{cf} = critical flow throat pressure, in pounds per square inch absolute.

P_1 = upstream relieving pressure, in pounds per square inch absolute.

k = ratio of specific heats for any ideal gas.

The sizing equations for pressure relief valves in vapor or gas service fall into two general categories depending on whether the flow is critical or subcritical. If the pressure downstream of the throat is less than or equal to the critical flow pressure, P_{cf} , then critical flow will occur, and the procedures in 4.3.2 should be applied. If the downstream pressure exceeds the critical flow pressure, P_{cf} , then subcritical flow will occur, and the procedures in 4.3.3 should be applied. (See Table 8 for typical critical flow pressure ratio values.)

4.3.2 SIZING FOR CRITICAL FLOW

4.3.2.1 General

Pressure relief valves in gas or vapor service that operate under critical flow conditions (see 4.3.1) may be sized using Equations 2–4. Each of the equations may be used to calculate the effective discharge area, A , required to achieve a required flow rate through a pressure relief valve. A valve that has an effective discharge area equal to or greater than the calculated value of A is then chosen for the application.

$$A = \frac{W}{CK_d P_1 K_b} \sqrt{\frac{TZ}{M}} \quad (2)$$

$$A = \frac{V \sqrt{TZM}}{6.32CK_d P_1 K_b} \quad (3)$$

$$A = \frac{V \sqrt{TZG}}{1.175CK_d P_1 K_b} \quad (4)$$

Where:

A = required effective discharge area of the valve, in square inches (see 1.2.2).

W = required flow through the valve, in pounds per hour.

C = coefficient determined from an expression of the ratio of the specific heats of the gas or vapor at standard conditions. This can be obtained from Figure 26 or Table 9.

K_d = effective coefficient of discharge = 0.975 for use in Equations 2–4.

P_1 = upstream relieving pressure, in pounds per square inch absolute. This is the set pressure plus the allowable overpressure (see 4.2) plus atmospheric pressure.

K_b = capacity correction factor due to back pressure. This can be obtained from the manufacturer's literature or estimated from Figure 27. The back-pressure correction factor applies to balanced-bellows valves only.

Note: See 4.3.3 for applications that involve superimposed back pressure of a magnitude that will cause subcritical flow.

T = relieving temperature of the inlet gas or vapor, in degrees Rankine (degrees Fahrenheit + 460).

Z = compressibility factor for the deviation of the actual gas from a perfect gas, a ratio evaluated at inlet conditions.

M = molecular weight of the gas or vapor. Various handbooks carry tables of molecular weights of materials, but the composition of the flowing gas or vapor is seldom the same as that listed in tables. This value should be obtained from the process data. Table 8 lists values for some common fluids.

V = required flow through the valve, in standard cubic feet per minute at 14.7 pounds per square inch absolute and 60°F.

G = specific gravity of gas referred to air
= 1.00 for air at 14.7 pounds per square inch absolute and 60°F.

The value of the coefficient C can be evaluated from the expression of the ratio of the specific heats of the gas or vapor as shown in Figure 26.

The ratio of specific heats of any ideal gas and possibly the ratio of specific heats of a diatomic actual gas can be found in any acceptable reference work.

Table 9 complements Figure 26 where $k = C_p/C_v$. When k cannot be determined, it is suggested that $C = 315$.

While ideal gas law behavior is generally acceptable for the majority of refinery applications, Appendix E should be referred to for unusual situations in which deviation from ideal behavior is significant.

4.3.2.2 Example

In this example, the following relief requirements are given:

- Required hydrocarbon vapor flow, W , caused by an operational upset, of 53,500 pounds per hour.
- Molecular weight of hydrocarbon vapor [a mixture of butane (C_4) and pentane (C_5)], M , of 65.
- Relief temperature, T , of 627°R (167°F).
- Relief valve set at 75 pounds per square inch gauge, the design pressure of the equipment.
- Back pressure of 0 pounds per square inch gauge.

In this example, the following data are derived:

- Permitted accumulation of 10 percent.
 - Relieving pressure, P_1 , of $75 \times 1.1 + 14.7 = 97.2$ pounds per square inch absolute.
 - Calculated compressibility, Z , of 0.84. (If a calculated compressibility is not available, $Z = 1.0$ should be used.)
 - Critical back pressure (from Table 8) of $97.2 \times 0.59 = 57.3$ pounds per square inch absolute (42.6 pounds per square inch gauge).
- Note: Since the back pressure (0 pounds per square inch gauge) is less than the critical back pressure (42.6 pounds per square inch gauge), the relief valve setting is based on the critical flow equation (see Equation 2 and 4.3.1 and 4.3.2).
- $C_p/C_v = k$ (from Table 8) of 1.09. From Table 9, $C = 326$.
 - Capacity correction due to back pressure, K_b , of 1.0.

The size of a single pressure relief valve is derived from Equation 2 as follows:

$$A = \frac{53,500}{326 \times 0.975 \times 97.2} \sqrt{\frac{627 \times 0.84}{65}} = 4.93 \text{ square inches}$$

Table 8—Properties of Gases

Gas	Molecular Weight	Specific Heat Ratio ($k = C_p/C_v$) at 60°F and 1 Atmosphere	Critical Flow Pressure Ratio at 60°F and 1 Atmosphere	Specific Gravity at 60°F and 1 Atmosphere	Critical Constants		Condensation Temperature 1 Atmosphere (°F)	Flammability Limits (volume percent in air mixture)	References
					Pressure (psia)	Temperature (°F)			
Methane	16.04	1.31	0.54	0.300	667	-117	-259	5.0-15.0	1
Ethane	30.07	1.19	0.57	0.356	708	90	-127	2.9-13.0	1
Ethylene	28.05	1.25	0.56	0.139	731	49	-155	2.7-36.0	1
Propane	44.10	1.13	0.58	0.507	615	206	-44	2.0-9.5	1
Propylene	42.08	1.15	0.57	0.518	672	198	-54	2.0-10.6	2, 3
Isobutane	58.12	1.10	0.59	0.563	528	274	11	1.8-8.5	1
<i>n</i> -Butane	58.12	1.09	0.59	0.584	549	306	31	1.5-9.0	1
1-Butane	56.11	1.11	0.58	0.600	586	296	21	1.6-9.3	2, 3
Isopentane	72.15	—	—	0.625	490	369	82	1.3-8.0	1
<i>n</i> -Pentane	72.15	—	—	0.631	488	386	97	1.4-8.3	1
1-Pentane	70.13	—	—	0.646	510	377	86	1.5-8.7	1
<i>n</i> -Hexane	86.18	—	—	0.664	437	454	156	1.1-7.7	1
Benzene	78.11	—	—	0.882	710	552	176	1.4-7.1	2, 3
<i>n</i> -Heptane	100.20	—	—	0.688	397	513	209	1.0-7.0	1
Toluene	92.14	—	—	0.874	596	606	231	1.2-7.1	2, 3
<i>n</i> -Octane	114.23	—	—	0.707	361	564	258	0.8-6.5	1
<i>n</i> -Nonane	128.26	—	—	0.722	332	611	303	0.7-5.6	1
<i>n</i> -Decane	142.28	—	—	0.734	305	653	345	0.7-5.4	1
Air	28.96	1.40	0.53	0.875	547	-221	-318	—	2, 3
Ammonia	17.03	1.31	0.54	0.616	1636	271	-28	16.0-25.0	2, 3
Carbon dioxide	44.01	1.29	0.55	0.818	1071	88	-109	—	2, 3
Hydrogen	2.02	1.41	0.53	—	190	-400	-423	4.0-75.0	2, 3
Hydrogen sulfide	34.08	1.32	0.54	0.801	1300	213	-77	4.3-45.5	2, 3
Sulfur dioxide	64.06	1.27	0.55	1.394	1143	316	14	—	2, 3
Steam	18.02	—	—	1.000	3199	705	212	—	2, 3

^aEstimated.

References:

- "Physical Constants of Hydrocarbons C_1 to C_{10} ," ASTM Special Technical Publication No. 109A, Philadelphia, Pa., 1963.
- "International—Critical Tables," McGraw-Hill Book Co., Inc., New York.
- "Engineering Data Book," Gas Processors Suppliers Association, 1977.
- API Technical Data Book—Petroleum Refining, Fifth edition.

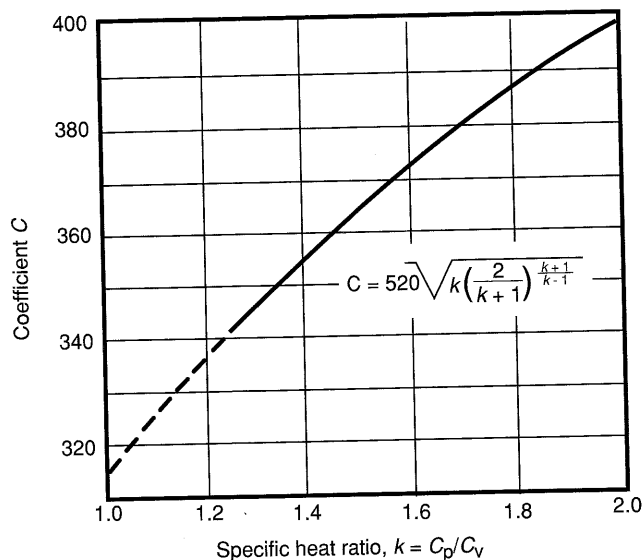


Figure 26—Curve for Evaluating Coefficient C in the Flow Equation From the Specific Heat Ratio Assuming Ideal Gas Behavior

See API Standard 526, which also provides a purchase specification sheet for flanged steel safety relief valves (see Figure 28).

Select a "P" letter orifice size (6.38 square inches).

4.3.3 SIZING FOR SUBCRITICAL FLOW: GAS OR VAPOR OTHER THAN STEAM

4.3.3.1 General

When the ratio of back pressure to inlet pressure exceeds the critical pressure ratio P_{cf}/P_1 , the flow through the pressure relief valve is subcritical (see 4.3.1). Equations 5-7 may be used to calculate the required effective discharge area for a conventional relief valve that has its spring setting adjusted to compensate for superimposed back pressure and for sizing a pilot-operated relief valve.

Note: Balanced-bellows relief valves that operate in the subcritical region should be sized using Equations 2-4. The back pressure correction factor for this application should be obtained from the valve manufacturer.

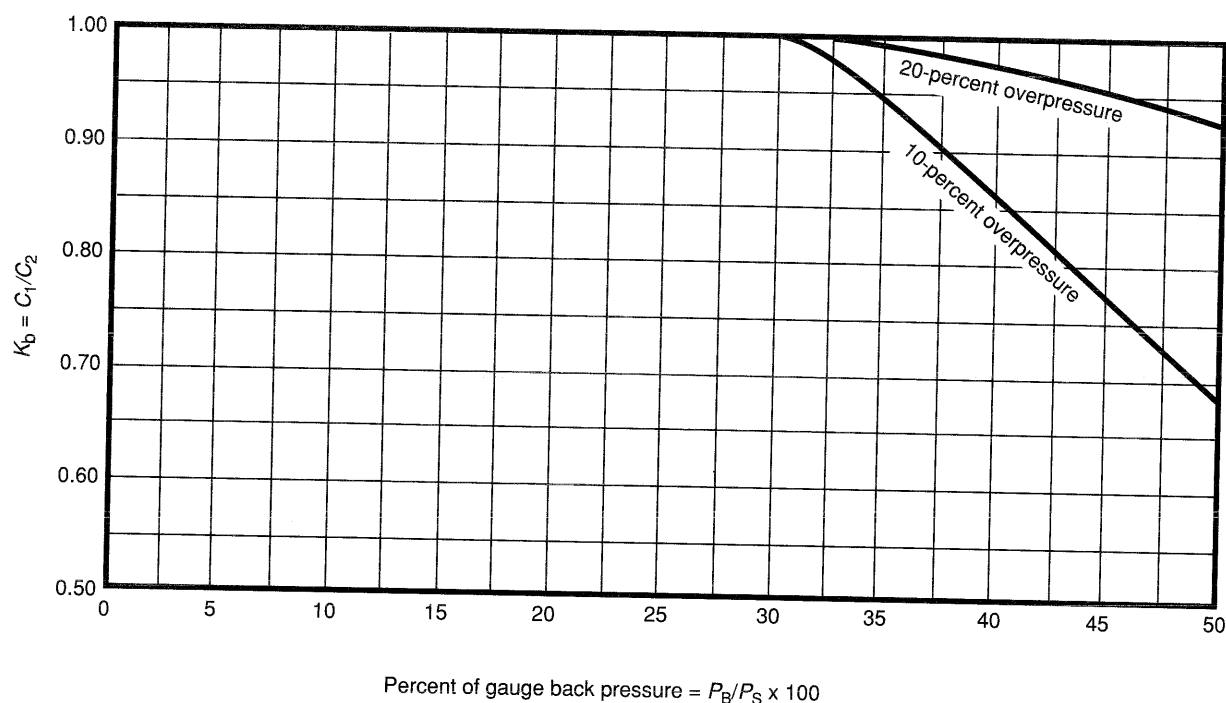
$$A = \frac{W}{735 F_2 K_d} \sqrt{\frac{ZT}{MP_1(P_1 - P_2)}} \quad (5)$$

$$A = \frac{V}{4645.2 F_2 K_d} \sqrt{\frac{ZTM}{P_1(P_1 - P_2)}} \quad (6)$$

Table 9—Values of Coefficient C

k	C	k	C	k	C	k	C
1.01	317 ^a	1.31	348	1.61	373	1.91	395
1.02	318	1.32	349	1.62	374	1.92	395
1.03	319	1.33	350	1.63	375	1.93	396
1.04	320	1.34	351	1.64	376	1.94	397
1.05	321	1.35	352	1.65	376	1.95	397
1.06	322	1.36	353	1.66	377	1.96	398
1.07	323	1.37	353	1.67	378	1.97	398
1.08	325	1.38	354	1.68	379	1.98	399
1.09	326	1.39	355	1.69	379	1.99	400
1.10	327	1.40	356	1.70	380	2.00	400
1.11	328	1.41	357	1.71	381	-	-
1.12	329	1.42	358	1.72	382	-	-
1.13	330	1.43	359	1.73	382	-	-
1.14	331	1.44	360	1.74	383	-	-
1.15	332	1.45	360	1.75	384	-	-
1.16	333	1.46	361	1.76	384	-	-
1.17	334	1.47	362	1.77	385	-	-
1.18	335	1.48	363	1.78	386	-	-
1.19	336	1.49	364	1.79	386	-	-
1.20	337	1.50	365	1.80	387	-	-
1.21	338	1.51	365	1.81	388	-	-
1.22	339	1.52	366	1.82	389	-	-
1.23	340	1.53	367	1.83	389	-	-
1.24	341	1.54	368	1.84	390	-	-
1.25	342	1.55	369	1.85	391	-	-
1.26	343	1.56	369	1.86	391	-	-
1.27	344	1.57	370	1.87	392	-	-
1.28	345	1.58	371	1.88	393	-	-
1.29	346	1.59	372	1.89	393	-	-
1.30	347	1.60	373	1.90	394	-	-

^aInterpolated value, since C becomes indeterminate as k approaches 1.00.



C_1 = capacity with back pressure.
 C_2 = rated capacity with zero back pressure.
 P_B = back pressure, in pounds per square inch gauge.
 P_S = set pressure, in pounds per square inch gauge.

Note: The curves above represent a compromise of the values recommended by a number of relief valve manufacturers and may be used when the make of the valve or the actual critical flow pressure point for the vapor or gas is unknown. When the make is known, the manufacturer should be consulted for the correction factor. These curves are for set pressures of 50 pounds per square inch gauge and above. They are limited to back pressure below critical flow pressure for a given set pressure. For subcritical flow back pressures below 50 pounds per square inch gauge, the manufacturer must be consulted for values of K_b .

Figure 27—Back Pressure Sizing Factor, K_b , for Balanced-Bellows Pressure Relief Valves (Vapors and Gases)

$$A = \frac{V}{863.63 F_2 K_d} \sqrt{\frac{ZTG}{P_1(P_1 - P_2)}} \quad (7)$$

Where:

A = required effective discharge area of the valve, in square inches (see 1.2.2).

W = required flow through the valve, in pounds per hour.

F_2 = coefficient of subcritical flow (see Figure 29 for values)

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

k = ratio of the specific heats.

r = ratio of back pressure to upstream relieving pressure, P_2/P_1 .

K_d = effective coefficient of discharge
= 0.975 for use in Equations 5-7.

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

ing inlet conditions.

T = relieving temperature of the inlet gas or vapor, in degrees Rankine (degrees Fahrenheit + 460).

M = molecular weight of the gas or vapor. Various handbooks carry tables of molecular weights of materials, but the composition of the flowing gas or vapor is seldom the same as that listed in the tables. This value should be obtained from the process data. Table 8 lists values for some common fluids.

P_1 = upstream relieving pressure, in pounds per square inch absolute. This is the set pressure plus the allowable overpressure (see 4.2) plus atmospheric pressure, in pounds per square inch absolute.

P_2 = back pressure, in pounds per square inch absolute.

V = required flow through the valve, in standard cubic feet per minute at 14.7 pounds per square inch absolute and 60°F.

G = specific gravity of gas referred to air
= 1.00 for air at 14.7 pounds per square inch absolute and 60°F.

PRESSURE RELIEF VALVE SPECIFICATION SHEET		PAGE _____ OF _____				
		SHEET NO. _____ REQUISITION NO. _____ JOB NO. _____ DATE _____ REVISED _____ BY _____				
General	(See 4.3.2)	(See 4.3.3)	(See 4.3.4)	(See 4.4)	(See 4.5.1)	
1. Item number.	<u>Example 1</u>	<u>Example 2</u>	<u>Example 3</u>	<u>Example 4</u>	<u>Example 5</u>	
2. Tag number.	_____	_____	_____	_____	_____	
3. Service, line, or equipment number.	_____	_____	_____	_____	_____	
4. Number required.	_____	_____	_____	_____	_____	
5. Full nozzle, semi-nozzle, or other.	_____	_____	_____	_____	_____	
6. Design type.						
a. Safety, relief, or safety relief.	<u>SRV</u>	<u>SRV</u>	<u>SRV</u>	<u>SV</u>	<u>RV</u>	
b. Conventional, bellows, or pilot operated.	<u>Conv.</u>	<u>Conv.</u>	<u>Conv.</u>	<u>Conv.</u>	<u>Bellows</u>	
7. Bonnet type.	_____	_____	_____	_____	_____	
Connections						
8. Size (inlet/outlet).	<u>4/6</u>	<u>4/6</u>	<u>4/6</u>	<u>3/6</u>	<u>4/6</u>	
9. Flange class, ANSI or threaded.	<u>150/150</u>	<u>150/150</u>	<u>150/150</u>	<u>900/150</u>	<u>150/150</u>	
10. Type facing.	<u>/</u>	<u>/</u>	<u>/</u>	<u>/</u>	<u>/</u>	
Material						
11. Body/bonnet.	<u>/</u>	<u>/</u>	<u>/</u>	<u>/</u>	<u>/</u>	
12a. Seal/disk.	<u>/</u>	<u>/</u>	<u>/</u>	<u>/</u>	<u>/</u>	
12b. Resilient rubber seal.	<u>/</u>	<u>/</u>	<u>/</u>	<u>/</u>	<u>/</u>	
13. Guide/rings.	<u>/</u>	<u>/</u>	<u>/</u>	<u>/</u>	<u>/</u>	
14. Spring.	<u>/</u>	<u>/</u>	<u>/</u>	<u>/</u>	<u>/</u>	
15. Bellows.	<u>/</u>	<u>/</u>	<u>/</u>	<u>/</u>	<u>/</u>	
Accessories						
16. Cap, screwed or bolted.	_____	_____	_____	_____	_____	
17. Lever, plain or packed.	_____	_____	_____	_____	_____	
18. Gag.	_____	_____	_____	_____	_____	
19. _____	_____	_____	_____	_____	_____	
20. _____	_____	_____	_____	_____	_____	
Basis of Selections						
21. Code.	<u>ASME VIII</u>	<u>ASME VIII</u>	<u>ASME VIII</u>	<u>ASME VIII</u>	<u>ASME VIII</u>	
22. Fire.	<u>Oper. fail</u>	<u>Oper. fail</u>	<u>Oper. fail</u>	<u>Oper. fail</u>	<u>Oper. fail</u>	
23. Other.	_____	_____	_____	_____	_____	
Service Conditions						
24. Fluid and state.	<u>HC vapor</u>	<u>HC vapor</u>	<u>HC vapor</u>	<u>Steam</u>	<u>Crude oil</u>	
25. Required capacity per valve and units.	<u>53,500 lb/hr</u>	<u>53,500 lb/hr</u>	<u>53,500 lb/hr</u>	<u>153,500 lb/hr</u>	<u>1800 gpm</u>	
26. Molecular weight or specific gravity at flowing temperature.	<u>65</u>	<u>65</u>	<u>65</u>	_____	<u>0.90 SG</u>	
27. Viscosity at flowing temperature and units.	<u>—</u>	<u>—</u>	<u>—</u>	_____	<u>2000 SSU</u>	
28. Operating pressure, psig, and set pressure, psig.	<u>50/75</u>	<u>50/75</u>	<u>50/75</u>	<u>1460/1600</u>	<u>225/250</u>	
29. Operating temperature, °F, and flowing temperature, °F.	<u>100/167</u>	<u>100/167</u>	<u>100/167</u>	<u>593/606</u>	<u>100/100</u>	

Figure 28—Sample of Completed Pressure Relief Valve Specification Sheet

PRESSURE RELIEF VALVE SPECIFICATION SHEET		PAGE ____ OF ____ SHEET NO. _____ REQUISITION NO. _____ JOB NO. _____ DATE _____ REVISED _____ BY _____				
Service Conditions - Continued						
30. Constant back pressure, psig.	0	55	55	0	0	0
31. Variable back pressure, psig.	0	0	0	0	0	0-50
32. Differential set pressure.	75	20	20	1600	250	250
33. Allowable overpressure, as a percent.	10	10	10	10	10	10
34. Compressibility factor.	0.84	0.84	0.84	0.84	0.84	0.84
35. Ratio of specific heats.	1.09	1.09	1.09	1.09	1.09	1.09
Orifice Area						
36. Calculated, in square inches.	4.93	5.60	5.60	1.705	4.267	4.267
37. Selected, in square inches.	6.38	6.38	6.38	1.838	4.43	4.43
38. Orifice designation.	P	P	P	K	N	N
39. Manufacturer's model number.						
40. Manufacturer.						
Note: Purchaser should specify requirements for items left blank.						

Note: See Appendix G for a blank sheet.

Figure 28—Sample of Completed Pressure Relief Valve Specification Sheet—Continued

4.3.3.2 Example

In this example, the following relief requirements are given:

- Required hydrocarbon vapor flow, W , caused by an operational upset, of 53,500 pounds per hour.
- Molecular weight of hydrocarbon vapor [a mixture of butane (C_4) and pentane (C_5)], M , of 65.
- Relief temperature, T , of 627°R (167°F).
- Relief valve set at 75 pounds per square inch gauge, the design pressure of the equipment.
- Constant back pressure of 55 pounds per square inch gauge. The spring setting of the valve should be adjusted according to the amount of constant back pressure obtained.

In this example, the following data are derived:

- Permitted accumulation of 10 percent.
- Relieving pressure, P_1 , of $75 \times 1.1 + 14.7 = 97.2$ pounds per square inch absolute.
- Calculated compressibility, Z , of 0.84. (If a calculated compressibility is not available, $Z = 1.0$ should be used.)
- Critical back pressure (from Table 8) of $97.2 \times 0.59 =$

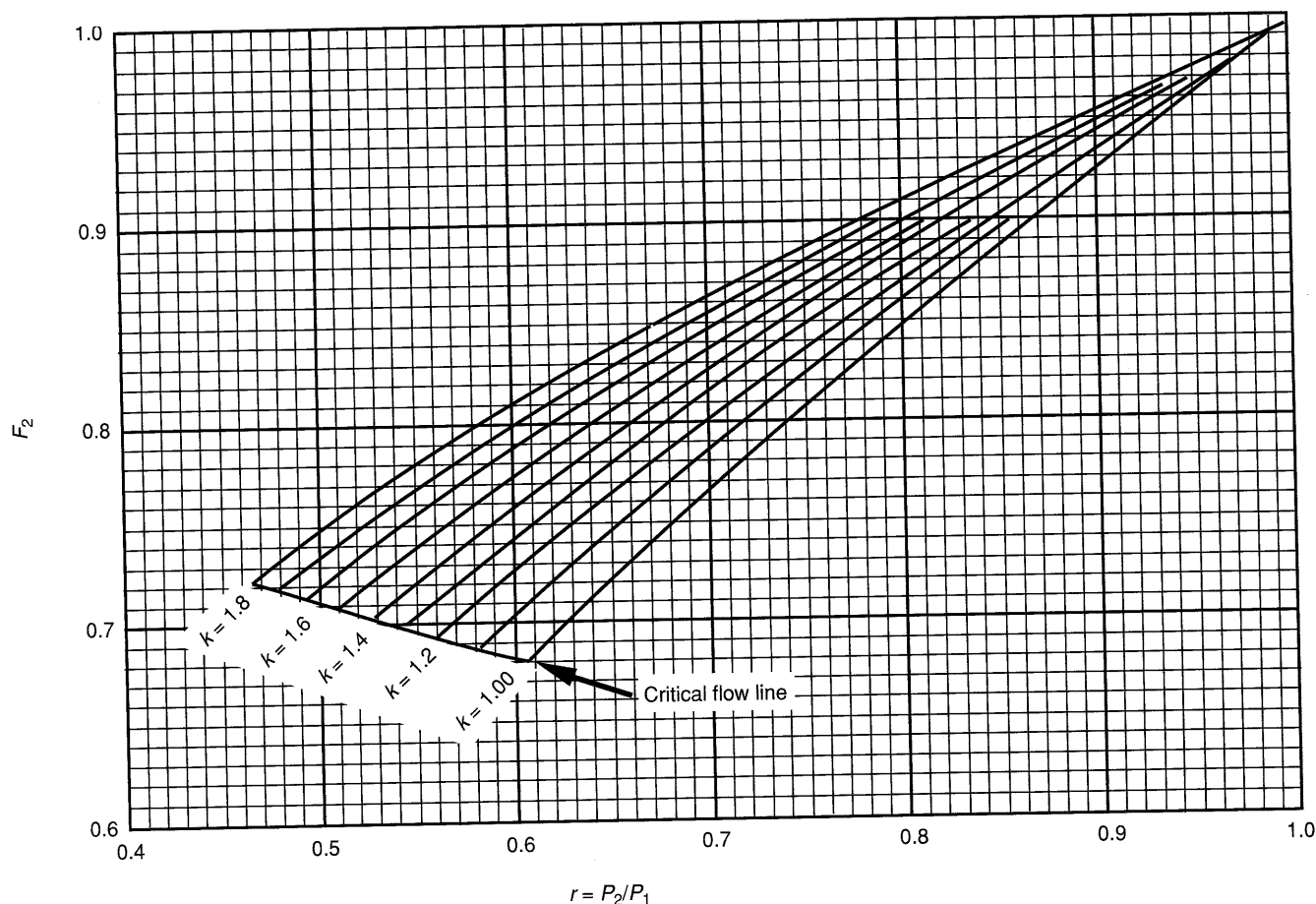
57.3 pounds per square inch absolute (42.6 pounds per square inch gauge).

Note: Since the back pressure (55 pounds per square inch gauge) is greater than the critical back pressure (42.6 pounds per square inch gauge), the relief valve setting is based on the subcritical flow equation (see Equation 5 and 4.3.1 and 4.3.3).

- Permitted built-up back pressure of $0.10 \times 75 = 7.5$ pounds per square inch.
- Total back pressure of $55 + 7.5 = 62.5$ pounds per square inch gauge.
- $C_p/C_v = k$ (from Table 8) of 1.09.
- $P_2/P_1 = (62.5 + 14.7)/97.2 = 0.794$.
- Coefficient of subcritical flow, F_2 , of 0.86 (from Figure 29).

The size of a single pressure relief valve is derived from Equation 5 as follows:

$$A = \frac{53,500}{735 \times 0.86 \times 0.975} \sqrt{\frac{0.84 \times 627}{65 \times 97.2(97.2 - 77.2)}} = 5.60 \text{ square inches}$$

Figure 29—Values of F_2 for Subcritical Flow

See API Standard 526, which also provides a purchase specification sheet for flanged steel safety relief valves (see Figure 28).

Select a "P" letter orifice size (6.38 square inches).

4.3.4 ALTERNATE SIZING PROCEDURE FOR SUBCRITICAL FLOW

4.3.4.1 General

Critical flow Equations 2-4 may be used to calculate the required discharge area of a pressure relief valve used in subcritical service. The area obtained using this sizing procedure is identical to the area obtained using the subcritical flow equations. (The capacity correction factor due to back pressure is derived by setting the subcritical flow equation equal to the critical flow equation and algebraically solving for K_b .) This alternate sizing procedure allows the designer to use the familiar critical flow equation to calculate the same area obtained with the subcritical flow equation. A graphical presentation of the capacity correction factor, K_b , is given in Figure 30. It should be noted that this correction factor is used only for the sizing of conventional (nonbalanced) relief

valves that have their spring settings adjusted to compensate for the superimposed back pressure. The correction factor should not be used to size balanced-type valves.

4.3.4.2 Example

In this example, the following relief requirements are given:

- Required hydrocarbon vapor flow, W , caused by an operational upset, of 53,500 pounds per hour.
- Molecular weight of hydrocarbon vapor [a mixture of butane (C_4) and pentane (C_5)], M , of 65.
- Relief temperature, T , of 627°R (167°F).
- Relief valve set at 75 pounds per square inch gauge, the design pressure of the equipment.
- Constant back pressure of 55 pounds per square inch gauge. The spring setting of the valve should be adjusted according to the amount of constant back pressure obtained.

In this example, the following data are derived:

- Permitted accumulation of 10 percent.
- Relieving pressure, P_1 , of $75 \times 1.1 + 14.7 = 97.2$ pounds

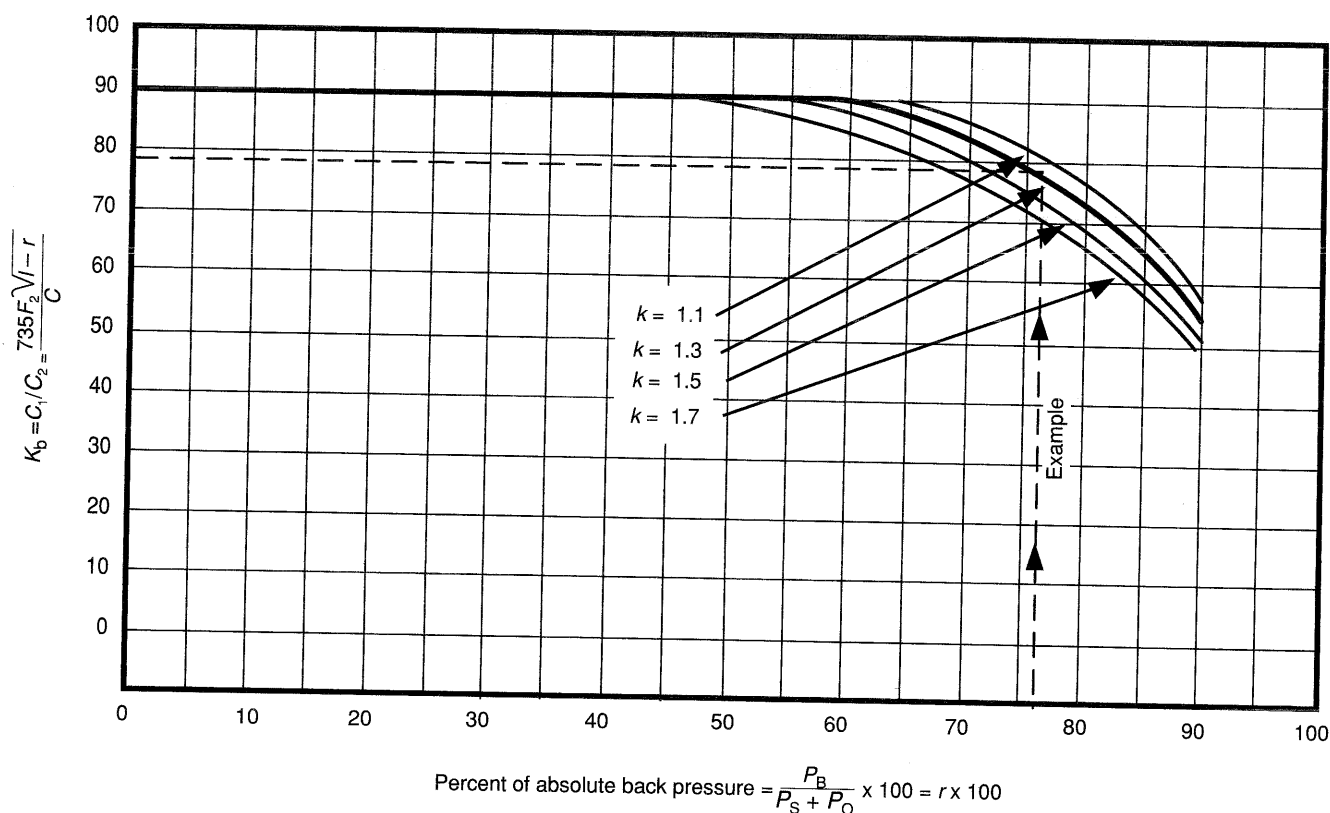
per square inch absolute.

c. Calculated compressibility, Z , of 0.84. (If a calculated compressibility is not available, $Z = 1.0$ should be used.)

d. Critical back pressure (from Table 8) of $97.2 \times 0.59 = 57.3$ pounds per square inch absolute (42.6 pounds per square inch gauge).

Note: Since the back pressure (55 pounds per square inch gauge) is greater than the critical back pressure (42.6 pounds per square inch gauge), the sizing of the relief valve is based on subcritical flow. The back pressure correction factor, K_b , should be determined using the critical flow formulas (see Equations 2-4).

e. Built-up back pressure of $0.10 \times 75 = 7.5$ pounds per square inch.



C_1 = capacity with back pressure.

C_2 = rated capacity with zero back pressure.

P_B = back pressure, in pounds per square inch absolute.

P_S = set pressure, in pounds per square inch absolute.

P_O = overpressure, in pounds per square inch.

Set pressure (MAWP) = 100 pounds per square inch gauge.

Overpressure = 10 pounds per square inch.

Superimposed back pressure (constant) = 70 pounds per square inch gauge.

Spring set = 30 pounds per square inch.

Built-up back pressure = 10 pounds per square inch.

Percent absolute back pressure = $\frac{(70 + 10 + 14.7)}{(100 + 10 + 14.7)} \times 100 = 76$

K_b (follow dotted line) = 0.89 (from the curve).

Capacity with back pressure = 0.89 (rated capacity without back pressure).

Note: This chart is typical and suitable for use only when the make of the valve or the actual critical flow pressure point for the vapor or gas is unknown; otherwise, the valve manufacturer should be consulted for specific data. This correction factor should be used only in the sizing of conventional (non-balanced) pressure relief valves that have their spring setting adjusted to compensate for the superimposed back pressure. It should not be used to size balanced-type valves.

Figure 30—Constant Back Pressure Sizing Factor, K_b , for Conventional Safety Relief Valves (Vapors and Gases Only)

- f. Total back pressure of $55 + 7.5 + 14.7 = 77.2$ pounds per square inch absolute.
- g. $C_p/C_v = k$ of 1.09.
- h. $P_2/P_1 = 77.2/97.2 = 0.794$.
- i. Back pressure correction factor, K_b , of 0.88 (from Figure 30).
- j. Coefficient determined from an expression of the ratio of the specific heats of the gas or vapor at standard conditions, C , of 326 (from Table 9).

The size of the relief valve is derived from Equation 2 as follows:

$$A = \frac{53,500}{326(0.975)(97.2)(0.88)} \sqrt{\frac{627(0.84)}{65}}$$

$$= 5.60 \text{ square inches}$$

Note: This area requirement is the same as that obtained using the subcritical flow equation (see Equation 5).

4.4 Sizing for Steam Relief

4.4.1 GENERAL

Pressure relief valves in steam service may be sized using Equation 8.

$$A = \frac{W}{51.5P_1K_dK_NK_{SH}} \quad (8)$$

Where:

A = required effective discharge area, in square inches (see 1.2.2).

W = required flow rate, in pounds per hour.

P_1 = upstream relieving pressure, in pounds per square inch absolute. This is the set pressure plus the allowable overpressure (see 4.2) plus the atmospheric pressure, in pounds per square inch absolute.

K_d = effective coefficient of discharge
= 0.975 for use in Equation 8.

K_N = correction factor for Napier equation (see Reference 1)

= 1 where $P_1 \leq 1515$ pounds per square inch absolute

= $(0.1906P_1 - 1000)/(0.2292P_1 - 1061)$ where $P_1 > 1515$ pounds per square inch absolute and ≤ 3215 pounds per square inch absolute.

K_{SH} = superheat steam correction factor. This can be obtained from Table 10. For saturated steam at any pressure, $K_{SH} = 1.0$.

Table 10—Superheat Correction Factors, K_{SH}

Set Pressure (pounds per square inch gauge)	Temperature (degrees Fahrenheit)									
	300	400	500	600	700	800	900	1000	1100	1200
15	1.00	0.98	0.93	0.88	0.84	0.80	0.77	0.74	0.72	0.70
20	1.00	0.98	0.93	0.88	0.84	0.80	0.77	0.74	0.72	0.70
40	1.00	0.99	0.93	0.88	0.84	0.81	0.77	0.74	0.72	0.70
60	1.00	0.99	0.93	0.88	0.84	0.81	0.77	0.75	0.72	0.70
80	1.00	0.99	0.93	0.88	0.84	0.81	0.77	0.75	0.72	0.70
100	1.00	0.99	0.94	0.89	0.84	0.81	0.77	0.75	0.72	0.70
120	1.00	0.99	0.94	0.89	0.84	0.81	0.78	0.75	0.72	0.70
140	1.00	0.99	0.94	0.89	0.85	0.81	0.78	0.75	0.72	0.70
160	1.00	0.99	0.94	0.89	0.85	0.81	0.78	0.75	0.72	0.70
180	1.00	0.99	0.94	0.89	0.85	0.81	0.78	0.75	0.72	0.70
200	1.00	0.99	0.95	0.89	0.85	0.81	0.78	0.75	0.72	0.70
220	1.00	0.99	0.95	0.89	0.85	0.81	0.78	0.75	0.72	0.70
240	—	1.00	0.95	0.90	0.85	0.81	0.78	0.75	0.72	0.70
260	—	1.00	0.95	0.90	0.85	0.81	0.78	0.75	0.72	0.70
280	—	1.00	0.96	0.90	0.85	0.81	0.78	0.75	0.72	0.70
300	—	1.00	0.96	0.90	0.85	0.81	0.78	0.75	0.72	0.70
350	—	1.00	0.96	0.90	0.86	0.82	0.78	0.75	0.72	0.70
400	—	1.00	0.96	0.91	0.86	0.82	0.78	0.75	0.73	0.70
500	—	1.00	0.96	0.92	0.86	0.82	0.78	0.75	0.73	0.70
600	—	1.00	0.97	0.92	0.87	0.82	0.79	0.75	0.73	0.70
800	—	—	1.00	0.95	0.88	0.83	0.79	0.76	0.73	0.71
1000	—	—	1.00	0.96	0.89	0.84	0.78	0.76	0.73	0.71
1250	—	—	1.00	0.97	0.91	0.85	0.80	0.77	0.74	0.71
1500	—	—	—	1.00	0.93	0.86	0.81	0.77	0.74	0.71
1750	—	—	—	1.00	0.94	0.86	0.81	0.77	0.73	0.70
2000	—	—	—	1.00	0.95	0.86	0.80	0.76	0.72	0.69
2500	—	—	—	1.00	0.95	0.85	0.78	0.73	0.69	0.66
3000	—	—	—	—	1.00	0.82	0.74	0.69	0.65	0.62

4.4.2 EXAMPLE

In this example, the following relief requirement is given:
 W = saturated steam at 153,500 pounds per square inch gauge set pressure
 1600 pounds per square inch gauge set pressure
 with 10 percent accumulation.

In this example, the following data are derived:

- Relieving pressure, P_1 , of $1600 \times 1.1 + 14.7 = 1774.7$ pounds per square inch absolute.
- Effective coefficient of discharge, K_d , of 0.975.
- Correction factor for Napier equation, K_N , of $[0.1906(1774.7) - 1000]/[0.2293(1774.7) - 1061] = 1.01$.
- Superheat steam correction factor, K_{SH} , of 1.0.

The size of the relief valve is derived from Equation 8 as follows:

$$A = \frac{153,500}{51.5(1774.7)(0.975)(1.01)(1)} = 1.705 \text{ square inches}$$

See API Standard 526, which also provides a purchase specification sheet for flanged steel safety relief valves (see Figure 28).
 Select a "K" orifice valve (1.838 square inches), that is, a 3K6 safety valve.

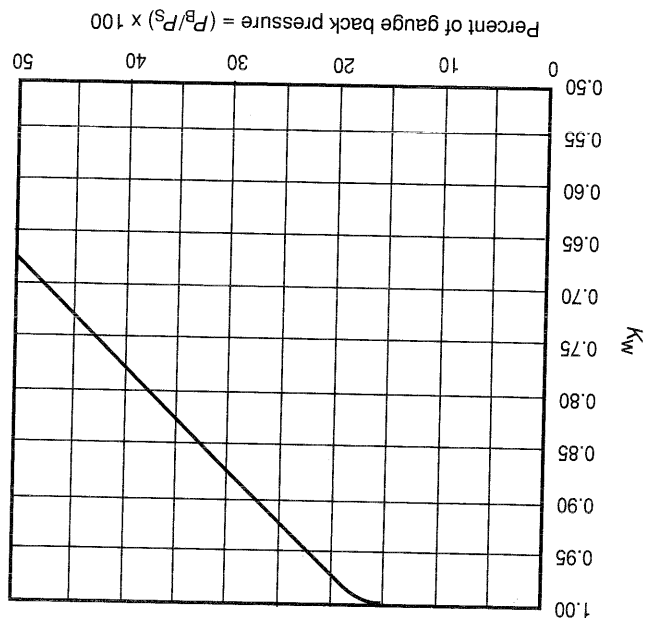


Figure 31—Capacity Correction Factor, K_w , Due to Back Pressure on Balanced-Bellows Pressure Relief Valves in Liquid Service

K_w = correction factor due to back pressure.
 P_b = back pressure, in pounds per square inch gauge.
 P_s = set pressure, in pounds per square inch gauge.

Note: The curve above represents values recommended by various manufacturers. This curve may be used when the manufacturer is not known. Otherwise, the manufacturer should be consulted for the applicable correction factor.

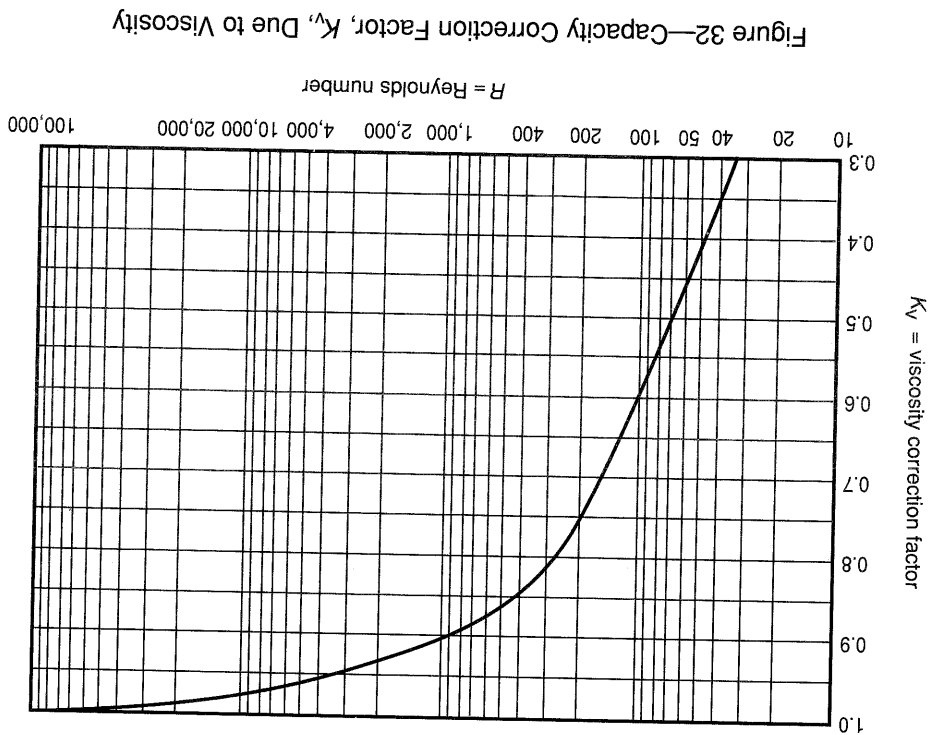


Figure 32—Capacity Correction Factor, K_v , Due to Viscosity

4.5 Sizing for Liquid Relief: Relief Valves Requiring Liquid Capacity Certification

4.5.1 GENERAL

Section VIII, Division I, of the ASME Code requires that capacity certification be obtained for pressure relief valves designed for liquid service. The procedure for obtaining capacity certification includes determining the coefficient of discharge for the design of liquid relief valves at 10 percent overpressure. Valves that require a capacity in accordance with the ASME Code may be sized using Equation 9.

$$A = \frac{Q}{38K_d K_w K_v} \sqrt{\frac{G}{P_1 - P_2}} \quad (9)$$

Where:

A = required effective discharge area, in square inches.

Q = flow rate, in U.S. gallons per minute.

K_d = effective coefficient of discharge that should be obtained from the valve manufacturer. For a preliminary sizing estimation, a discharge coefficient of 0.65 can be used.

K_w = correction factor due to back pressure. If the back pressure is atmospheric, $K_w = 1$. Balanced-bellows valves in back pressure service will require the correction factor determined in Figure 31. Conventional valves require no special correction.

K_v = correction factor due to viscosity as determined from Figure 32.

G = specific gravity of the liquid at the flowing temperature referred to water = 1.0 at 70°F.

P_1 = upstream relieving pressure, in pounds per square inch gauge. This is the set pressure plus allowable overpressure.

P_2 = total back pressure, in pounds per square inch gauge.

When a relief valve is sized for viscous liquid service, it should first be sized as it was for nonviscous-type application so that a preliminary required discharge area, A , can be obtained. From manufacturers' standard orifice sizes, the next larger orifice size should be used in determining the Reynold's number, R , from either of the following relationships:

$$R = \frac{Q(2800G)}{\mu\sqrt{A}} \quad (10)$$

or

$$R = \frac{12,700Q}{U\sqrt{A}} \quad (11)$$

Where:

Q = flow rate at the flowing temperature, in U.S. gallons per minute.

G = specific gravity of the liquid at the flowing temperature referred to water

= 1.00 at 70°F.

μ = absolute viscosity at the flowing temperature, in centipoises.

A = effective discharge area, in square inches (from manufacturers' standard orifice areas).

U = viscosity at the flowing temperature, in Saybolt Universal seconds.

Note: Equation 11 is not recommended for viscosities less than 100 Saybolt Universal seconds.

After the value of R is determined, the factor K_v is obtained from Figure 32. K_v is applied to correct the preliminary required discharge area. If the corrected area exceeds the chosen standard orifice area, the above calculations should be repeated using the next larger standard orifice size.

4.5.2 EXAMPLE

In this example, the following relief requirements are given:

- Required crude-oil flow caused by blocked discharge, Q , of 1800 gallons per minute.
- Specific gravity, G , of 0.90 (viscosity at the flowing temperature is 2000 Saybolt Universal seconds).
- Relief valve set at 250 pounds per square inch gauge, the design pressure of the equipment.
- Back pressure variable from 0 to 50 pounds per square inch gauge.

In this example, the following data are derived:

- Overpressure of 10 percent.
- Relieving pressure, P_1 , of $1.10 \times 250 = 275$ pounds per square inch gauge.
- Back pressure of $(50/250) \times 100 = 20$ percent.

A balanced-bellows valve is indicated, since back pressure is variable. (From Figure 31, $K_w = 0.97$.)

The manufacturer's effective coefficient of discharge $K = 0.75$.

Sizing first for no viscosity correction ($K_v = 1.0$), the size of the relief valve is derived from Equation 9 as follows:

$$A = \frac{1800}{38.0 \times 0.75 \times 0.97 \times 1.0} \sqrt{\frac{0.90}{275 - 50}} = 4.118 \text{ square inches}$$

See Equation 11. An area of 4.34 square inches ("N" orifice) should be used.

4.7 Sizing for Two-Phase Liquid/Vapor Relief

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

Calculations for determining properties and handling liquid/vapor phases are available; see Section 5 for emerging technical literature that can be used to size relieving devices. For information about saturated water, see specifically Section VIII, Appendix 11, of the ASME Code.

A reasonable, conservative method of sizing for two-phase liquid/vapor relief is as follows:

- Determine the amount of liquid that flashes by an isenthalpic (adiabatic) expansion from the relieving condition either to the critical downstream pressure for the flashed vapor or to the back pressure, whichever is greater.

4.6 Sizing for Liquid Relief: Relief Valves Not Requiring Capacity Certification

Before the ASME Code made provisions for capacity certification, valves were generally sized for liquid service using Equation 12. This method assumes a coefficient of discharge, $K_d = 0.62$, and 25 percent overpressure. An additional capacity correction factor, K_p , was obtained from Figure 33 for relieving pressures other than 25 percent overpressure. This sizing method may be used where capacity certification is not required.

Where:

$$A = 4.267 \text{ square inches}$$

$$A = \frac{A_r}{K_v} = \frac{4.118}{0.965}$$

From Figure 32, $K_v = 0.965$.

$$R = \frac{12,700 \times 1800}{2000 \sqrt{4.34}} = 5487$$

See API Standard 526, which also provides a purchase specification sheet for flanged steel safety relief valves (see Figure 28).

Select an "N" orifice pressure relief valve (4.34 square inches), that is, a 4N6 pressure relief valve.

Where:

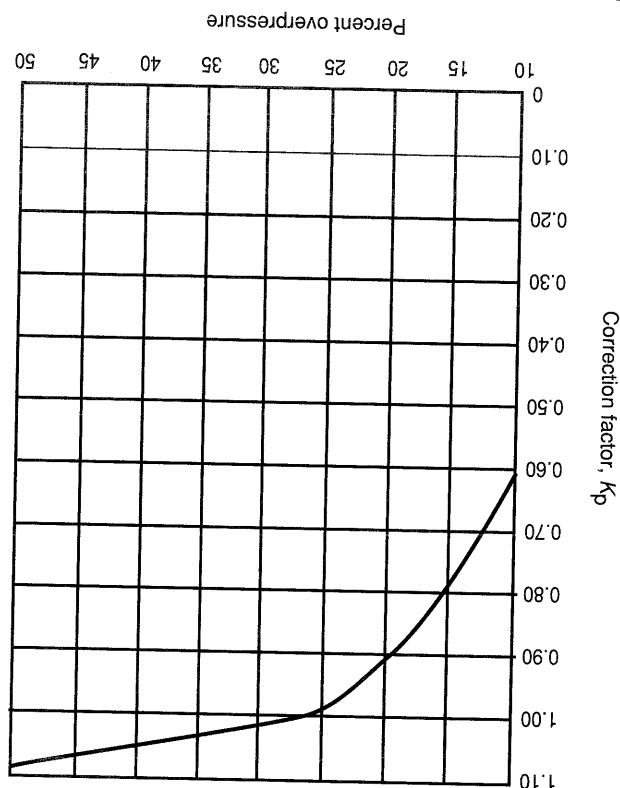
$$A = \frac{38 K_d K_w K_v K_p}{\bar{Q}} \sqrt{1.25 p - p_b} \quad (12)$$

A = required effective discharge area, in square inches.
 $\bar{Q}$ = flow rate, in U.S. gallons per minute.
 K_d = effective coefficient of discharge that should be obtained from the valve manufacturer. For a preliminary sizing estimation, a discharge coefficient of 0.62 can be used.
 K_w = correction factor due to back pressure. If back pressure is atmospheric, $K_w = 1$. Balanced-bellows valves in back-pressure service will require the correction factor determined from Figure 31. Conventional valves require no special correction.
 K_v = correction factor due to viscosity as determined from Figure 32.
 K_p = correction factor due to overpressure. At 25 percent overpressure, $K_p = 1.0$. For overpressures other than 25 percent, K_p is determined from Figure 33.

G = specific gravity of the liquid at the flowing temperature referred to water.
 p = set pressure, in pounds per square inch gauge.
 p_b = total back pressure, in pounds per square inch gauge.

Figure 33—Capacity Correction Factors Due to Overpressure for Relief and Safety-Relief Valves in Liquid Service

Note: The curve above shows that up to and including 25 percent overpressure, capacity is affected by the change in lift, the change in the orifice discharge coefficient, and the change in overpressure. Above 25 percent, capacity is affected only by the change in overpressure. Valves operating at low overpressures tend to chatter; therefore, overpressures of less than 10 percent should be avoided.



- b. Calculate individually the orifice area required to pass the flashed vapor component, using Equations 2-7 as appropriate, according to service, type of valve, and whether the back pressure is greater or less than the critical downstream pressure.
- c. Calculate individually the orifice area required to pass the unflashed liquid component using Equation 9. The pressure drop ($P_1 - P_2$) is the inlet relieving pressure minus the back pressure.
- d. Add the individual areas calculated for the vapor and liquid components to obtain the total orifice area, A , that is required.
- e. Select a pressure relief valve that has an effective discharge area equal to or greater than the total calculated orifice area. The designer should recheck the back pressure that will exist for the specific relief valve selected, with its particular discharge installation, by examining the vapor generation downstream of the pressure relief valve nozzle. Where appropriate, corrections can be applied to the individual orifice areas previously calculated. Furthermore, selecting a balanced pressure relief valve is often desirable to minimize the effect of flashed vapor on the valve capacity.

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

4.8 Sizing for Rupture Disk Devices

4.8.1 RUPTURE DISK DEVICES USED INDEPENDENTLY

Rupture disk devices may be used alone or in combination with a pressure relief valve in gas or vapor service or in liquid service. The sizing of a rupture disk device used alone is based on the equations applicable for pressure relief valves using for all fluids an effective coefficient of discharge $K_d = 0.62$.

The required discharge area, A in square inches, is calculated using the appropriate equation for the flowing medium (see Equations 2-7 for gas or vapor, Equation 8 for steam, and Equation 9 for liquid). The rupture disk device selected should be the nominal pipe size whose area is equal to or greater than the required discharge area calculated by the appropriate equation.

For rupture disk devices that have a structural member (for example, a knife blade or vacuum support) that reduces the effective discharge area after bursting, the projected area of the structural member is deducted from the flow area of the pipe to determine the net discharge area of the burst rupture disk.

Users should be aware of the following limitations in using $K_d = 0.62$ to determine the capacity of a given size of

rupture disk, or conversely, the required area for a given flow quantity:

- a. The rupture disk device is used in a size and pressure range that the manufacturer has determined will give a satisfactory opening for the style of rupture disk in the particular fluid service.
- b. The rupture disk device is installed in a short piping system that does not add significantly to the flow resistance of the burst rupture disk device.

If a rupture disk device discharges into a vent system or a closed relief system, it will usually not contribute significantly to the pressure loss obtained in the discharge piping. The sizing of the inlet and discharge piping becomes a line sizing problem that uses the relieving rate and the maximum allowable inlet pressure defined by the code. In general, a pressure loss through the rupture disk device of approximately 75 pipe diameters may be used. The manufacturer should be consulted if more accurate values are required. This problem is similar to the line sizing of a process line except that the effect of volumetric expansion on the pressure loss must be considered. This will include an acceleration effect (the vapor is exiting the discharge piping at a higher velocity) as well as the effect of changing density. If the vent line subsequently discharges into headers of varying sizes, critical flow restrictions must be considered.

4.8.2 RUPTURE DISK DEVICES USED IN COMBINATION WITH PRESSURE RELIEF VALVES

An important application of a rupture disk device is at the inlet of a pressure relief valve. The sizing of the pressure relief valve/rupture disk device combination requires that the pressure relief valve first be sized to meet the required relieving capacity. The certified and published capacity of the pressure relief valve used alone is then multiplied by the combination capacity factor, K_c , to determine the capacity of that combination. (See 2.6.2 for further information on the combination capacity factor.)

The nominal size of the rupture disk device installed at the inlet of the pressure relief valve must be equal to or greater than the nominal size of the inlet connection of the valve to permit sufficient flow capacity and valve performance.

The design of the piping from the protected vessel to the inlet of the pressure relief valve is crucial to the proper functioning of the valve. Users should consult applicable engineering codes for guidance on inlet piping design. Unless the pressure relief device is installed directly on the vessel, a good practice is to analyze the frictional pressure loss from the vessel to the valve inlet at the rated relieving capacity to comply with recommended limits. An inlet pipe sized larger than necessary for the inlet to the pressure relief valve is often required; this may dictate a rupture disk device to match the pipe size.

SECTION 5—REFERENCES

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APPENDIX A—RUPTURE DISK DEVICE SPECIFICATION SHEET

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

Line No.	Instruction
1.	Fill in item number.
2.	Fill in user's rupture disk device identification number.
3.	Specify service, line, or equipment on which rupture disk device is to be mounted.
4.	Specify the applicable code or codes.
5.	Indicate flowing fluid and state (liquid, gas, or vapor).
6.	Specify quantity of fluid that rupture disk device is required to relieve at relieving conditions and the fluid units (such as pounds per hour, gallons per minute, and cubic feet per minute).
7.	Specify the molecular weight or specific gravity of the fluid at the flowing temperature.
8.	Specify viscosity and units of viscosity at the flowing temperature.
9.	Specify operating pressure, in pounds per square inch gauge.
10.	Specify maximum allowable working pressure or design pressure, in pounds per square inch gauge.
11.	Specify coincident temperature, in degrees Fahrenheit, at burst pressure.
12.	Specify relieving temperature, in degrees Fahrenheit.
13.	Specify superimposed back pressure, in pounds per square inch gauge.
14.	Specify vacuum pressure, in pounds per square inch absolute, present in the inlet or outlet at any one time.
15.	Specify allowable overpressure, as a percent.
16.	Specify compressibility factor.
17.	Give specific heat ratio as $k(C_p/C_v)$.
18.	Specify whether or not fragmentation is allowed.
19.	Specify type of holder (flanged, screwed, or union).
20.	Specify number of holders.
21.	Specify nominal pipe size.
22.	Specify American National Standards Institute (ANSI) class of flanged connections for inlet and outlet flanges. If device has threaded connections, write in "threaded."
23.	Specify flange facing for inlet and outlet.
24.	Specify piping connection, schedule/bore.
25.	Specify inlet holder material (such as 316ss, cs, or Monel).
26.	Specify outlet holder material (such as 316ss, cs, or Monel).
27.	Specify gauge-tap size and location for inlet, outlet, or both (may not be required).
28.	If eyebolts or jackscrews are required, specify which ones and the material of each.
29.	If a telltale assembly is required, specify the material.
30.	Specify quantity (include spares).
31.	Give size expressed as a nominal pipe size.
32.	Specify type, such as conventional domed or reverse buckling.
33.	Specify burst pressure at burst temperature [pounds per square inch gauge at temperature (in degrees Fahrenheit)].
34.	Fill in manufacturing range per manufacturer's standards.
35.	Specify allowable burst tolerance (normally ± 5 percent of the stamped burst pressure).
36.	Fill in manufacturer's name.
37.	Fill in previous lot number. If the rupture disk device has been previously ordered, the manufacturer will have a lot number specifying exactly how the disks were manufactured.

Line No.	Instruction
38.	Fill in model number of rupture disk.
39.	Fill in model number of holder.
40.	Fill in model number of accessories or other items.
41.	Specify disk material; that is, the type of metal or other material.
42.	Specify vacuum support material, if required.
43.	Specify liner material, if required.
44.	Specify coating material, if required.
45.	Specify ring material, if required.
46.	Specify top section material.
47.	Specify seal material.

RUPTURE DISK DEVICE SPECIFICATION SHEET		PAGE <u>1</u> OF <u>2</u>	
		SHEET NO. _____ REQUISITION NO. _____ JOB NO. _____ DATE _____ REVISED _____ BY _____	
General			
1. Item number.	_____	_____	_____
2. Tag number	_____	_____	_____
3. Service, line, or equipment number.	_____	_____	_____
Service Conditions			
4. Design code (ASME, DIN, etc.).	_____	_____	_____
5. Fluid and state.	_____	_____	_____
6. Required capacity and units.	_____	_____	_____
7. Molecular weight or specific gravity at flowing temperature.	_____	_____	_____
8. Viscosity at flowing temperature.	_____	_____	_____
9. Operating pressure, psig.	_____	_____	_____
10. Maximum allowable working pressure or design pressure, psig.	_____	_____	_____
11. Burst temperature, °F.	_____	_____	_____
12. Relieving temperature, °F.	_____	_____	_____
13. Superimposed back pressure, psig.	_____	_____	_____
14. Vacuum pressure present, psia, inlet or outlet.	_____	_____	_____
15. Allowable overpressure, as a percent.	_____	_____	_____
16. Compressibility factor.	_____	_____	_____
17. Ratio of specific heats.	_____	_____	_____
18. Fragmentation allowed (yes or no).	_____	_____	_____
19. Design type.	_____	_____	_____
20. Number required.	_____	_____	_____
Connections			
21. Size.	_____	_____	_____
22. Flange class (ANSI).	_____	_____	_____
23. Facing type (flat, RTJ, etc.)	_____	_____	_____
24. Piping connections, schedule/bore.	_____	_____	_____
Rupture Disk Holders			
25. Holder material, inlet.	_____	_____	_____
26. Holder material, outlet.	_____	_____	_____
27. Gauge tap size, inlet or outlet.	_____	_____	_____
Rupture Disk Holder Accessories			
28. Eyebolts, jackscrews, material.	_____	_____	_____
29. Telltale assembly, material.	_____	_____	_____
Rupture Disks			
30. Quantity (include spares).	_____	_____	_____
31. Size (nominal pipe size).	_____	_____	_____
32. Type.	_____	_____	_____
33. Specified burst condition (psig/temperature, °F).	_____	_____	_____
34. Manufacturing range.	_____	_____	_____
35. Allowable burst tolerance.	_____	_____	_____

Figure A-1—Rupture Disk Device Specification Sheet

RUPTURE DISK DEVICE SPECIFICATION SHEET	PAGE <u> 2 </u> OF <u> 2 </u> SHEET NO. _____ REQUISITION NO. _____ JOB NO. _____ DATE _____ REVISED _____ BY _____
Manufacturer's Data	
36. Manufacturer.	_____
37. Previous lot number.	_____
38. Model number (rupture disk).	_____
39. Model number (holder).	_____
40. Model number (other).	_____
Materials - All Rupture Disks	
41. Disk material.	_____
42. Vacuum support.	_____
43. Liner.	_____
44. Coatings.	_____
45. Rings.	_____
Materials - Composite Rupture Disks.	
46. Top section.	_____
47. Seal.	_____
Notes:	

Figure A-1—Rupture Disk Device Specification Sheet—Continued

APPENDIX B—SPECIAL SYSTEM DESIGN CONSIDERATIONS

B.1 Single Pressure Relief Device Protecting Several Components In a Process System

In some situations, a single pressure relief device may be desirable to protect several equipment components in a process system. For this system to meet the intent of Section VIII of the ASME Code with respect to overpressure protection, the following four criteria must be satisfied:

- a. No means can exist for blocking any of the equipment components being protected from the installation of the single pressure relief device.
- b. The set pressure of the first pressure relief device to be actuated must be at or below the lowest design pressure of any equipment component being protected in the system.
- c. The accumulated pressure when the pressure relief device is discharging may be as high as 10 percent (or 21 percent for fire) above the lowest design pressure for any equipment component in the system. If multiple relief devices are used in a single installation, then accumulated pressure may be as high as 16 percent (or 21 percent for fire) above the lowest design pressure for any equipment component in the system.
- d. The operating pressure in any equipment component must never exceed its design pressure when the pressure relief device protecting the system is not discharging.

B.2 Description of a Typical Process System

A typical process system that may be provided with only one pressure relief device is a hydrotreater reactor recycle-gas loop. Such a system may contain the following main equipment components:

- a. A recycle-gas compressor.
- b. A feed/effluent heat exchanger.
- c. A fired heater.
- d. A reactor.

- e. An effluent condenser.
- f. A separator drum.
- g. Interconnecting piping.
- h. Piping for liquid feed, product, and purge gas.

Figure B-1 is a schematic for the typical process system indicated above.

B.3 Procedure to Calculate the Design Pressure of Equipment Components

If the procedure outlined below is followed, the design pressure of any equipment component in the system will never be exceeded unless the pressure in the system actuates the pressure relief device. The following steps should be taken:

- a. The pressure profile should be developed for the processing conditions that will result in the maximum pressure drop (normally end-of-run conditions with fouled equipment).
- b. The settling-out pressure that develops when the compressor stops during the maximum pressure drop case should be calculated. The separator drum should be assumed to be operating at normal operating pressure before compressor stoppage, and the purge gas line should be assumed to be closed to conserve gas.
- c. The minimum design pressure of the separator drum should be calculated as 1.05 times the settling-out pressure. This will provide an adequate differential between the operating pressure and set pressure of the pressure relief device for a compressor shutdown contingency.
- d. The pressure profile should be developed for the system with the pressure of the separator drum at the set pressure of the pressure relief device. Assuming equal volumetric gas flow, the pressure gradient will be proportional to the change in absolute pressure.

Note: The minimum design pressure of each equipment component is the inlet pressure for each equipment component as determined in Item d.

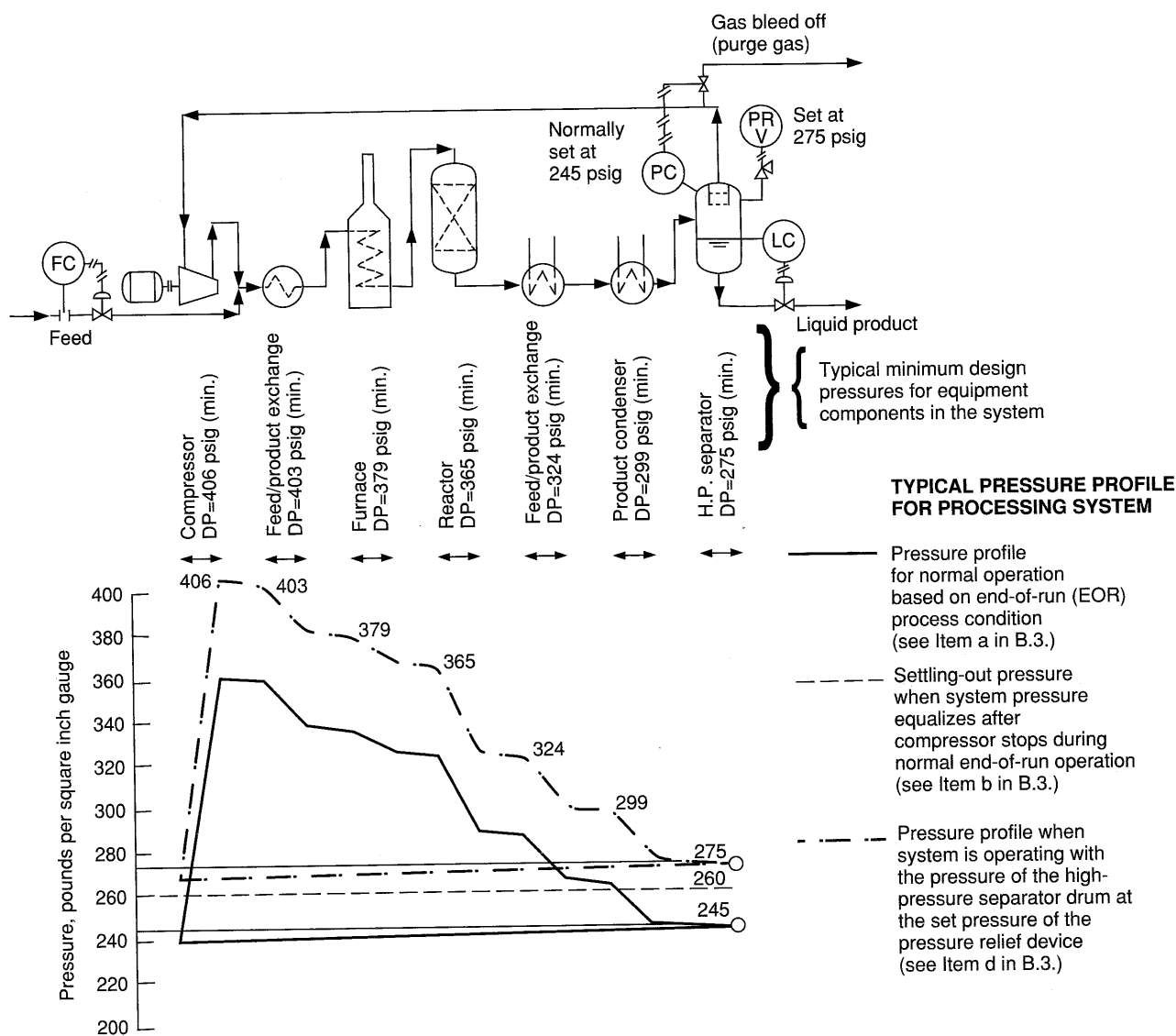


Figure B-1—Typical Flow Scheme of a System Involving a Single Pressure Relief Device Serving Components in a Process System With Typical Pressure Profiles

APPENDIX C—PRINCIPAL CAUSES OF OVERPRESSURE

C.1 Bases for Relief Capacities

The premise of this document is that the designer has performed an analysis of the conditions that can cause overpressure of the equipment protected by the pressure relief device being sized. Table 1 lists some common occurrences that may require overpressure protection. This table is not intended to be all-inclusive or complete in suggesting maximum relief capacities; it is merely recommended as a guide. Should the designer find a need to pursue the flow-rate analysis in more detail, a more descriptive analysis can be found in API Recommended Practice 521.

C.2 Hydraulic Expansion

C.2.1 BACKGROUND

Where liquid-full equipment can be blocked in and continued heat input cannot be avoided, a pressure relief device shall be provided. The rate of expansion depends primarily on the rate of heat input and the liquid properties.

Liquid expansion rates for the sizing of relief devices that protect heat exchangers, condensers, and coolers against thermal expansion of trapped liquids can be approximated using the following formula:

$$gpm = \frac{BH}{500GC}$$

Where:

gpm = flow rate at the flowing temperature, in U.S. gallons per minute.

B = cubical expansion coefficient per degree Fahrenheit for the liquid at the expected temperature. This information is best obtained from the process de-

sign data; however, Table C-1 shows typical values for hydrocarbon liquids and water at 60°F.

H = total heat transfer rate, in British thermal units per hour. For heat exchangers, this can be taken as the maximum exchanger duty during operation.

G = specific gravity referred to water
= 1.00 at 60°F. Compressibility of the liquid is usually ignored.

C = specific heat of the trapped fluid, in British thermal units per pound per degree Fahrenheit.

C.2.2 PRECAUTIONS

The user should realize that the method described in C.2.1 will provide only short-term protection in some cases. If the blocked-in liquid has a vapor pressure higher than the relief design pressure, then the pressure relief device must be capable of handling the vapor generation rate. The circumstances may not warrant concerns such as the time required to reach the boiling point versus the probability of discovery and the time for correction by an operator; however, if timely correction cannot be expected, then vaporization should be accounted for in sizing the pressure relief device.

Table C-1—Typical Values for Hydrocarbon Liquids and Water at 60°F

Liquid	Value
3-34.9 deg API gravity	0.0004
35-50.9 deg API gravity	0.0005
51-63.9 deg API gravity	0.0006
64-78.9 deg API gravity	0.0007
79-88.9 deg API gravity	0.0008
89-93.9 deg API gravity	0.00085
94-100 deg API gravity and lighter	0.0009
Water	0.0001

APPENDIX D—DETERMINATION OF FIRE-RELIEF REQUIREMENTS

D.1 Background

The problem of estimating fire-relief requirements for storage tanks was first recognized in 1928 when the National Fire Protection Association (NFPA) requested API to recommend that a table of minimum emergency relief capacities for a series of tank capacities be included in the *NFPA Suggested Ordinance Regulating the Use, Handling, Storage and Sale of Flammable Liquids and the Products Thereof*.

It was later recognized that tank capacities did not provide the best basis for estimating the amount of vapor to be handled. Since the heat was absorbed almost entirely by radiation, the area exposed—not the volume of the tank contents—seemed to be the important factor. Many of the tanks were large and could never be expected to be entirely surrounded by fire; the assumption was therefore made that the larger the area of the container, the less the likelihood that the tank would be fully exposed to radiation. In other words, the larger the surface area of the tank shells, the lower the average unit heat absorption rate from a fire.

By 1948 several different equations [1] were in general use, prompting the API Subcommittee on Pressure-Relieving Systems to develop an equation for determining the heat absorbed from open fires using the test data available at the time. The resultant equation has remained in general use since its publication in 1954 [2], and its development is documented in a paper presented by F. J. Heller in 1983 [3].

Note: Numbers in brackets correspond to references in Section 5.

Table D-1 contains data from 16 fire tests and one actual fire. Data from these tests were considered in the development of Equations D-1 and D-2.

These data result from tests in which means were provided to measure the total heat absorbed by a vessel by (a) computing the heat required to bring the liquid contents to the boiling range and (b) measuring the amount of liquid contents evaporated in a given time. The unit heat absorption rates in Table D-1 are average rates on the wetted surface.

Examinations of detailed reports on these tests indicate that the setup for Tests 4, 5, and 8 was arranged to provide continuous and complete flame envelopment of the small vessels; under these conditions, maximum average heat input rates of 30,400-32,500 British thermal units per hour per square foot were realized. The environmental conditions set up for Tests 1, 3, 6, 7, 9, and 10 allowed the flame to be subjected to air currents and wind. All other factors were conducive to maintaining maximum heat input, a condition that should not exist in a refinery. Under these conditions, the maximum average heat input rates varied greatly. Test 2 differed from Test 1 in that drainage away from the equipment was provided. The maximum heat input rate is reduced by 60

percent when drainage is provided; this fact was incorporated in the development of Equations D-1 and D-2. Test 11 gives an indication of the effect of a large area on average heat input during an actual fire.

The test reports mentioned in some cases that the tests were delayed until the arrival of a calm day so that the wind would not blow the flames away from the vessel. Copious supplies of fuel were available. In most cases, the fuel was maintained by dikes in a pool beneath the vessel and was not allowed to drain away as it normally would. In the Rubber Reserve Corporation tests, a 2-inch gasoline line, running full, was required to keep the fuel supplied during the test. Without these special adverse conditions, the maximum heat absorption values obtained in these tests are extremely unlikely to occur in an actual refinery fire.

D.2 Nature of an Open Fire

The nature of an open fire of flammable fluid, as related to test data, is important. This kind of fire differs from the fire in the firebox of a boiler or still, where the fuel and air are mixed by means other than the convection currents caused by the heated gases. The flame will accordingly have a core of flammable vapor, either unmixed with air or insufficiently mixed to burn. Combustion occurs on the exterior envelope of this core. Because the actual combustion zone is on the rich side, a considerable amount of black smoke is generated. This envelope of soot may serve to mask a considerable portion of the flame.

Hot gases from the combustion rise, and the air that supports the combustion flows in at the bottom. The flame mass is quite turbulent; as masses of the burning vapor tumble and billow, the smoky mantle is displaced and the bright flame can be seen intermittently. This flame is not a blazing white, as it would be in a furnace; it is red or orange, indicating a lower temperature than that of a furnace flame.

Flames of this type tend to rise because of their temperature; however, they can also be blown aside by the wind and may be blown so far from a vessel that the heating effect on the vessel is small.

D.3 Effect of Fire on the Unwetted Surface of a Vessel

D.3.1 GENERAL

Unwetted wall vessels are those in which the internal walls are exposed to a single fluid, vapor, or gas or are internally insulated regardless of the contained fluids. These include vessels that contain separate liquid and vapor phases under normal conditions but become single phase (above the critical) at relieving conditions.

Table D-1—Comparison of Heat-Absorption Rates in Fire Tests

Test	Source	Type of Exposure	Fuel	Vessel Capacity (barrel)	Total Area (ft ²)	Wetted Area (ft ²)	Total Heat Input (Btu/Hr)	Temperature of Surface (°F)	Btu/Hr/Ft ² Wetted	Reference
1	Hottel, average of 36 tests	6-inch thick metal stack	Gasoline	Conning Tower	296	123	3,760,000		30,500	1
2	Hottel, average of 13 tests	6-inch thick metal stack	Gasoline	Conning Tower	296	123	2,139,000		17,400	2
3	Standard Oil Company of California	Heating water in drum	Naphtha	2.6		26	416,000		16,000	3
4	Standard Oil Company of California	Heating water in tank	Naphtha	33	206	105	3,370,000	70-212	32,000	3
5	Underwriters - Laboratories, Inc.	Water flowing over plate	Gasoline		24	24	780,000	76	32,500	4
6	Rubber Reserve Corporation Test No. 17	Heating water in tank	Gasoline	119	568	400	9,280,000	300	23,200	5
7	Rubber Reserve Corporation Test No. 17	Generating steam in tank	Gasoline	199	568	400	8,400,000		21,000	5
8	Rubber Reserve Corporation Test No. 17	Water flowing in 3/4-inch standard pipe	Gasoline		9.0	9.0	274,000		30,400	5
9	API Project Test No. 1	Heating water in tank	Kerosene	0.88	16.2	6.1	95,800	300	15,700	6
10	API Project Test No. 2	Heating water in tank	Kerosene	0.88	16.2	6.1	102,500	320	16,800	6
11 ^a	Report to API on 38 ft. butane sphere	Plant fire	Butane	5,000	4,363	4,363	23,560,000		5,400	7
12	Lauderback		Chemical waste		100	100	3,210,000		32,100	8
13	NFPA (Tulsa)		Cutback jet fuel	238	773	303	8,736,000		23,000	9
14	Union Carbide (1938)		Propane	71.4	242	132	2,300,000		17,400	10
15	Union Carbide (1938)		Propane	71.4	242	176	4,993,000		28,400	10
16	Fetterly	Liquefied petroleum gas container	Wood saturated with kerosene	7.7	83	57.8	1,350,000		23,300	11

^aThis represents an actual fire.

References:

1. H. C. Hottel, Private communication to API Subcommittee on Pressure-Relieving Systems, January 1948.
2. H. C. Hottel, Private communication to API Subcommittee on Pressure-Relieving Systems, December 1950.
3. F. L. Maker, Private communication to API Subcommittee on Pressure-Relieving Systems regarding 1925 tests, December 22, 1950.
4. "Opacity of Water to Radiant Heat Energy," *Research Bulletin 3*, Underwriters Laboratory, Inc., 1938.
5. Safety Memorandum 89, Rubber Reserve Corporation, Washington, D.C., May 1944.
6. University of Michigan, Unpublished tests made for API Subcommittee

on Pressure-Relieving Systems, June 1947.

7. Anonymous report to API Subcommittee on Pressure-Relieving Systems regarding a fire (not a test), June 1941.
8. J. J. Duggan, C. H. Gilmour, and P. F. Fisher, "Requirements for Relief of Overpressure on Vessels Exposed to Fire," *Transactions of the ASME*, 1944, Volume 66, pp. 1-53.
9. "Large Scale Fire Exposure Tests to Evaluate 'Unox' Foam for Fire Exposure Protection," Fire Research Laboratory Report No. FRL-62, Process Safety Department, Carbide and Carbon Chemicals Co., a Division of Union Carbide and Carbon Corporation, Dec. 8, 1954.
10. Lauderback.
11. National Fire Protection Association.

Vessels may be designed to have internal insulation. A vessel should be considered internally insulated when the internal wall can become insulated by the deposition of coke or other materials as a result of the contained fluids.

D.3.2 CHARACTERISTICS

A characteristic of a vessel with an unwetted internal wall is that heat flow from the wall to the contained fluid is low as a result of the resistance of the contained fluid or any internal insulating material. Heat input from an open fire to the bare outside surface of an unwetted vessel may, in time, be sufficient to heat the vessel wall to a temperature high enough to rupture the vessel. Figures D-1 and D-2 indicate how quickly an unwetted bare vessel wall might be heated to rupture conditions. Figure D-1 illustrates the rise in temperature that occurs with time in the unwetted plates of various thicknesses exposed to open fire; for example, an unwetted steel plate 1 inch thick would take about 12 minutes to reach approximately 1100°F and about 17 minutes to reach 1300°F when the plate is exposed to an open fire.

Figure D-2 shows the effect of overheating ASTM A 515, Grade 70 steel [4]. The figure indicates that at a stress of 15,000 pounds per square inch, an unwetted steel vessel would rupture in about 7 hours at 1100°F and about 2½ minutes at 1300°F.

D.4 Effect of Fire on the Wetted Surface of a Vessel

The surface area wetted by a vessel's internal liquid contents is effective in generating vapor when the area is exposed to fire. To determine vapor generation, only that portion of the vessel that is wetted by its internal liquid and is equal to or less than 25 feet above the source of flame needs to be recognized. The term *source of flame* usually refers to ground grade but could be at any level at which a substantial spill or pool fire could be sustained. Various classes of vessels are operated only partially full. Table D-2 gives recommended portions of liquid inventory for use in calculations. Portions higher than 25 feet are normally excluded.

D.5 Heat Absorption Equations

D.5.1 HEAT ABSORPTION ACROSS THE WETTED SURFACE OF A VESSEL

The amount of heat absorbed by a vessel exposed to an open fire is markedly affected by the type of fuel feeding the fire, the degree to which the vessel is enveloped by the flames (a function of vessel size and shape), and fireproofing measures. The following equivalent formulas are used to evaluate these conditions where there are prompt fire-fighting efforts and drainage of flammable materials away from the vessel:

$$q = 21,000FA^{-0.18} \quad (D-1)$$

$$Q = 21,000FA^{0.82} \quad (D-2)$$

Where adequate drainage and fire-fighting equipment do not exist, Equation D-2 becomes the following [3]:

$$q = 21,000FA^{-0.18}$$

Where:

q = average unit heat absorption, in British thermal units per hour per square foot of wetted surface.

Q = total heat absorption (input) to the wetted surface, in British thermal units per hour.

F = environment factor. (Values for various types of installation are shown in Table D-3.)

A = total wetted surface, in square feet (see D.4). (The expression $A^{-0.18}$, or $1/A^{0.18}$, is the area exposure factor or ratio. This ratio recognizes the fact that large vessels are less likely than small ones to be completely exposed to the flame of an open fire.)

D.5.2 HEAT ABSORPTION ACROSS THE UNWETTED SURFACE OF A VESSEL

D.5.2.1 Simple Equations

See D.3 for a discussion of the effect of fire on the unwetted surface of a vessel.

The discharge areas for pressure relief valves on gas-containing vessels exposed to open fires can be determined using the following formula:

$$A = \frac{F' A'}{\sqrt{P_1}} \quad (D-3)$$

F' can be determined from the following relationship. The recommended minimum value of F' is 0.01; when the minimum value is unknown, $F' = 0.045$ should be used.

$$F' = \frac{0.1406 (T_w - T_1)^{1.25}}{CK_D T_1^{0.6506}} \quad (D-4)$$

Where:

A = effective discharge area of the valve, in square inches.

A' = exposed surface area of the vessel, in square feet.

P_1 = upstream relieving pressure, in pounds per square inch absolute. This is the set pressure plus the allowable overpressure plus the atmospheric pressure.

C = coefficient determined by the ratio of the specific heats of the gas at standard conditions. This can be obtained from Figure 26 or Table 9.

K_D = coefficient of discharge (obtainable from the valve manufacturer). The maximum allowable K_D established by ASME is 0.975.

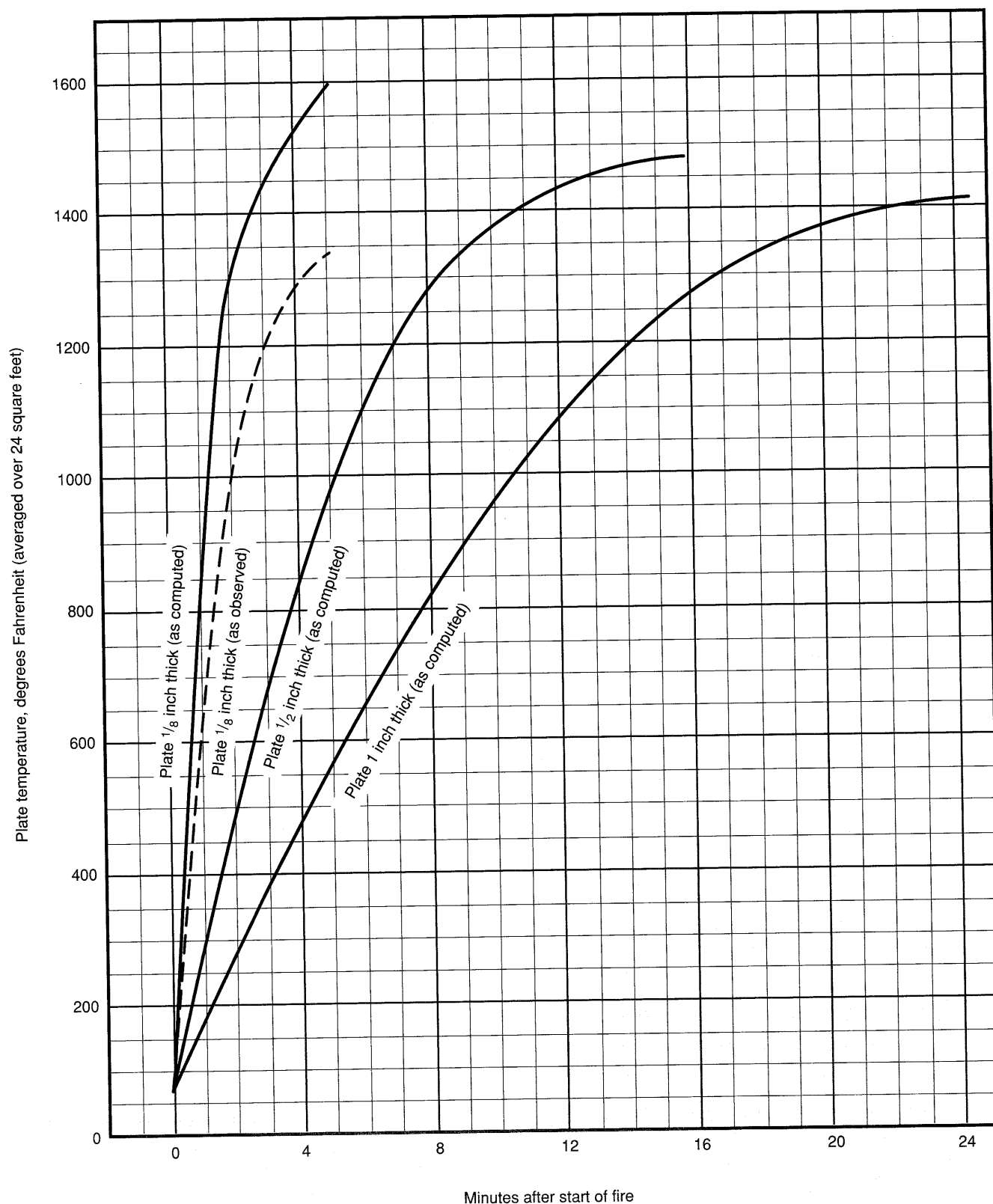


Figure D-1—Average Rate of Heating Steel Plates Exposed to Open Gasoline Fire on One Side

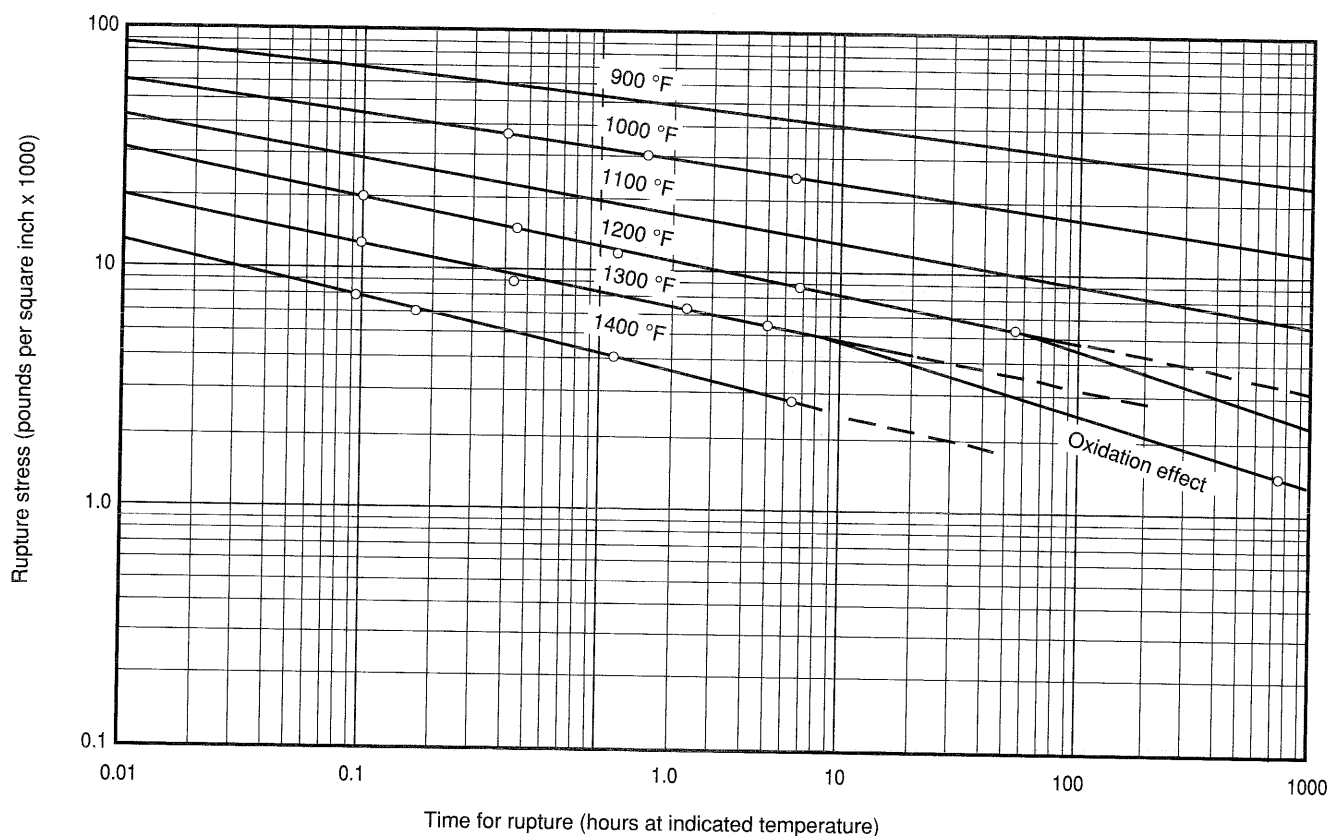


Figure D-2—Effect of Overheating Steel (ASTM A 515, Grade 70)

T_{ω} = vessel wall temperature, in degrees Rankine.

T_1 = gas temperature, absolute, in degrees Rankine, at the upstream pressure, determined from the following relationship:

$$T_1 = \frac{P_1}{P_{\eta}} T_{\eta}$$

Where:

P_{η} = normal operating gas pressure, in pounds per square inch absolute.

T_{η} = normal operating gas temperature, in degrees Rankine.

The recommended maximum vessel wall temperature for the usual carbon steel plate materials is 1100°F. Where vessels are fabricated from alloy materials, the value for T_{ω} should be changed to a more appropriate recommended maximum.

D.5.2.2 Rearrangement of Simple Equations

The relief load can be calculated directly in pounds per hour by rearranging Equation 2 and substituting Equations D-3 and D-4, which results in the following equation:

$$W = 0.1406 \sqrt{MP_1} \left(A' \frac{(T_{\omega} - T_1)^{1.25}}{T_1^{1.1506}} \right) \quad (D-5)$$

Where:

M = molecular weight of the gas.

Z and K_b in Equation 2 are assumed to equal 1.

D.5.2.3 Discussion of Simple Equations

The derivation of Equations D-3 and D-5 [5] is based on the physical properties of air and the perfect gas laws. The derivation assumes that the vessel is uninsulated and has no mass, that the vessel wall temperature will not reach rupture stress, and that there is no change in fluid temperature. These assumptions should be reviewed to ensure that they are appropriate for any particular situation.

D.5.2.4 More Rigorous Calculations

Where the assumptions in D.5.2.3 are not appropriate, more rigorous methods of calculation may be warranted. In such cases, the necessary physical properties of the containing fluid may need to be obtained from actual data or estimated from equations of state. The effects of vessel mass

Table D-2—Wetted Surface Area of a Vessel Based on Fire Heat Absorbed

Class of Vessel	Portion of Liquid Inventory	Remarks
Liquid-full, such as treaters	All up to the height of 25 feet	—
Surge drums, knockout drums, process vessels	Normal operating level up to the height of 25 feet	—
Fractionating columns	Normal level in bottom plus liquid holdup from all trays; total wetted surface up to the height of 25 feet	Level in reboiler is to be included if the reboiler is an integral part of the column
Working storage	Average inventory level up to the height of 25 feet	For tanks of 15 psig operating pressure or less; see API Standard 2000
Spheres and spheroids	Up to the maximum horizontal diameter or up to the height of 25 feet, whichever is greater	—

and insulation may need to be considered. The pressure-relieving rate is based on an unsteady state. As the fire continues, the vessel wall temperature and the contained gas temperature and pressure increase with time. The pressure relief valve will open at the set pressure, or if the pressure is set too high, the vessel will rupture. With the loss of fluid on relief, the temperatures will further increase at the relief pressure. If the fire is of sufficient duration, the temperature will increase until vessel rupture occurs. Procedures are available for estimating the changes in average vessel wall and contained fluid temperatures that occur with time and the maximum relieving rate at the set pressure [6, 7]. The procedures require successive iteration.

D.5.2.5 Additional Protective Measures

The determination may be made that a pressure relief valve will not provide sufficient protection for an unwetted wall vessel, and vessel rupture could occur before or too soon after initial relief. Where a pressure relief valve alone is not adequate, additional protective measures should be considered, including insulation (D.8.3.1), water sprays (D.8.3.3), and depressuring (D.8.2) to avoid vessel rupture.

Where calculations indicate that rupture would not occur prior to relief, a rupture disk device could also be considered.

The design should allow sufficient time for operator action and initiation of fire-fighting procedures before possible vessel rupture. Operator action may include depressuring, using water sprays, and employing firewater monitors.

D.6 Fluids To Be Relieved

A vessel may contain liquids or vapors or fluids of both phases. The liquid phase may be subcritical at operating temperature and pressure and may pass into the critical or supercritical range during the duration of a fire as the temperature and pressure in the vessel increase.

The quantity and composition of the fluid to be relieved during a fire depend on the total heat input rate to the vessel

under this contingency and on the duration of the fire.

The total heat input rate to the vessel may be computed by means of one of the formulas in Section D.5 using the appropriate values for wetted or exposed surfaces and for the environment factor.

Once the total heat input rate to the vessel is known, the quantity and composition of the fluid to be relieved can be calculated, provided that enough information is available on the composition of the fluid contained in the vessel.

If the fluid contained in the vessel is not completely specified, assumptions must be made to obtain a realistic relief flow rate for the relief device. These assumptions may include the following:

- An estimation of the latent heat of boiling liquid and the appropriate molecular weight of the fraction vaporized.
- An estimation of the thermal expansion coefficient if the relieving fluid is a liquid below its boiling temperature, a gas, or a supercritical fluid.

D.6.1 VAPOR

For pressure and temperature conditions below the critical point, the rate of vapor formation—a measure of the rate of vapor relief required—is equal to the total rate of heat absorption divided by the latent heat of vaporization. The vapor to be relieved is the vapor that is in equilibrium with the liquid under conditions that exist when the valve is relieving at its accumulated pressure.

The latent-heat and molecular-weight values used in calculating the rate of vaporization should pertain to the conditions that are capable of generating the maximum vapor rate.

The vapor and liquid composition may change as vapors are released from the system. As a result, temperature and latent-heat values could change, thus affecting the required size of the pressure relief device. On occasion, a multicomponent liquid may be heated at a pressure and temperature that exceed the criticals for one or more of the individual components. For example, vapors that are physically or

chemically bound in solution may be liberated from the liquid upon heating. This is not a standard latent-heating effect but is more properly termed degassing or dissolution. Vapor generation is determined by the rate of change in equilibrium caused by increasing temperature.

For these and other multicomponent mixtures that have a wide boiling range, a time-dependent model may have to be developed where the total heat input to the vessel not only causes vaporization but also raises the temperature of the remaining liquid, keeping it at its boiling point.

Reference 7 gives an example of a time-dependent model used to calculate relief requirements for a vessel exposed to fire that contains fluids near the critical range or above.

The recommended practice of finding a relief vapor flow rate from the heat input to the vessel and from the latent heat of liquid contained in the vessel becomes invalid near the critical point of the fluid, where the latent heat approaches zero and the sensible heat dominates.

When no accurate latent-heat value is available for these hydrocarbons near the critical point, a minimum value of 50 British thermal units per pound is sometimes acceptable as an approximation.

When pressure-relieving conditions are above the critical point, the rate of vapor discharge depends only on the rate at which the fluid will expand as a result of the heat input.

D.6.2 LIQUID

The hydraulic expansion formula given in Appendix C may be used to compute the initial liquid-relieving rate in a liquid-filled system when the liquid is still below its boiling

point. However, this rate is valid for a very limited time, after which vapor generation will become the determining contributor in the sizing of the pressure relief device.

There is an interim time period between the liquid expansion and the boiling vapor relief in which mixtures of both phases need to be relieved simultaneously, either as flashing, bubble, slug, froth, or mist flow until sufficient vapor space is available inside the vessel for phase separation. This mixed-phase condition is usually neglected, but for some vessels, particularly overfilled steam drums or polymerization reactors, the limiting relieving contingency may be the factor that would determine the size of the relieving device. (See D.6.3 for information about mixed-phase flow.)

Should a pressure relief device be located in the liquid zone of a vessel exposed to fire conditions, the pressure relief device must be able to pass a volume of liquid equivalent to the displacement caused by vapor generated by the fire.

D.6.3 MIXED PHASE

As stated in D.6.2, mixed-phase flow may sometimes be the limiting relieving contingency and thus will determine the size of the pressure relief device. This is particularly true for reactors during runaway reactions that may be caused by lack of cooling or excess heat input (for example, under fire).

The Design Institute for Emergency Relief Systems recently concluded an intensive research program to develop methods for the design of emergency relief systems to handle runaway reactions. The interested reader is advised to study some introductory publications [8-12] on this subject in Chemical Engineering Progress, August 1985.

Table D-3—Environment Factor

Type of Equipment	Factor F ^a
Bare vessel	1.0
Insulated vessel ^b (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
Water-application facilities, on bare vessel ^c	1.0
Depressurizing and emptying facilities ^d	1.0

^aThese are suggested values for the conditions assumed in D.5.1. When these conditions do not exist, engineering judgment should be exercised either in selecting a higher factor or in providing means of protecting vessels from fire exposure as suggested in D.8.

^bInsulation shall resist dislodgement by fire-hose streams. For the examples, a temperature difference of 1600°F was used. These conductance values are based on insulation having thermal conductivity of 4 BTU/hr-Ft²-°F per inch at 1600°F and correspond to various thicknesses of insulation between 1 and 12 inches.

^cSee D.8.3.3.

^dSee D.8.2.

D.7 Data on Latent Heat of Vaporization of Hydrocarbons

Different hydrocarbon liquids have different latent heats of vaporization even though hydrocarbons as a group behave similarly to one another. The latent heat of vaporization of a pure single-component liquid decreases as the temperature at vaporization increases and the latent heat becomes zero at the critical temperature and pressure for that liquid.

Figure D-3 shows the vapor pressures and latent heats of the pure single-component paraffin-hydrocarbon liquids. This chart is directly applicable to such liquids and applies as an approximation to paraffin-hydrocarbon mixtures composed of two components whose molecular weights vary no more than propane to butane and butane to pentane.

The chart may also be applicable to isomer hydrocarbons, aromatic or cyclic compounds, or paraffin-hydrocarbon mixtures of components that have slightly divergent molecular weights. The equilibrium temperature should be calculated. Using the relationship for the calculated temperature versus vapor pressure, the latent heat can then be obtained from Figure D-3. The molecular-weight relationship as shown by the chart is not to be used in such cases; the molecular weight of the vapor should be determined from the vapor-liquid equilibrium calculation.

For cases that involve mixtures of components that have a wide boiling range or widely divergent molecular weights, a rigorous series of equilibrium calculations may be required to estimate vapor generation rates, as discussed in D.6.1.

Other recognized sources [13] of latent-heat data or methods of calculating latent heat of vaporization should be used where Figure D-3 does not apply.

D.8 Protecting Vessels Against Fire Exposure

The measures described in D.8.1 through D.8.3 for protecting vessels against fire exposure are contingent on proper drainage away from the vessel so that pools of fuel cannot accumulate beneath them.

Improper drainage under fire conditions will limit the effectiveness of any of these measures in reducing the intensity of heat absorption by the vessels.

D.8.1 EFFECTIVENESS OF PRESSURE RELIEF DEVICES AS RELATED TO FIRE EXPOSURE

The effect of fire exposure on the unwetted surface of vessels is described in D.3. A pressure relief device does not prevent weakening and failure of a vessel that becomes locally overheated and overstressed at an unwetted surface. It will only prevent the internal pressure from rising beyond

the allowable accumulation pressure. A vessel may be protected against such failure by (a) depressurizing the vessel and (b) limiting the heat input.

D.8.2 DEPRESSURING SYSTEMS

Controlled depressuring of the vessel reduces internal pressure and stress in the vessel walls. It also guards against the potential addition of fuel to the fire should the vessel rupture. The design of depressuring systems should recognize the following factors:

- Manual controls near the vessel may be inaccessible during an emergency.
- Unless anticipated, automatic controls could fail in a direction that would prevent depressuring (for example, valves that fail closed).
- Early initiation of depressuring is desirable to limit vessel stress to acceptable levels commensurate with the vessel wall temperature that results from a fire.
- Safe disposal of vented streams must be provided.
- No credit is recommended when safety valves are being sized for fire exposure.

Further information on depressuring is provided in API Recommended Practice 521.

D.8.3 METHODS OF LIMITING HEAT INPUT FROM FIRE

D.8.3.1 External Insulation

Limiting the heat input from fires by external insulation reduces both the rise of the vessel wall temperature and the generation of vapor inside the vessel. Insulation may also reduce the problem of disposing of the vapors and the expense of providing an exceptionally large relieving system to conduct the effluent to a point of disposal.

The insulation must be fire resistant and protected from dislodgement by fire-hose streams (see API Recommended Practice 521).

Where insulation or fireproofing is applied, the heat absorption can be computed by assuming that the outside temperature of the insulation jacket or other outer covering has reached an equilibrium temperature of 1660°F. With this temperature and the operating temperature for the inside of the vessel, together with the thickness and conductivity of the fire-protection coating, the average heat transfer rate to the contents can be computed. It must be kept in mind that the thermal conductivity of the insulation increases with the temperature, and a mean value should be used.

For insulated vessels, the environment factor for insulation becomes the following:

$$F = \frac{k(1660 - T_f)}{21,000t}$$

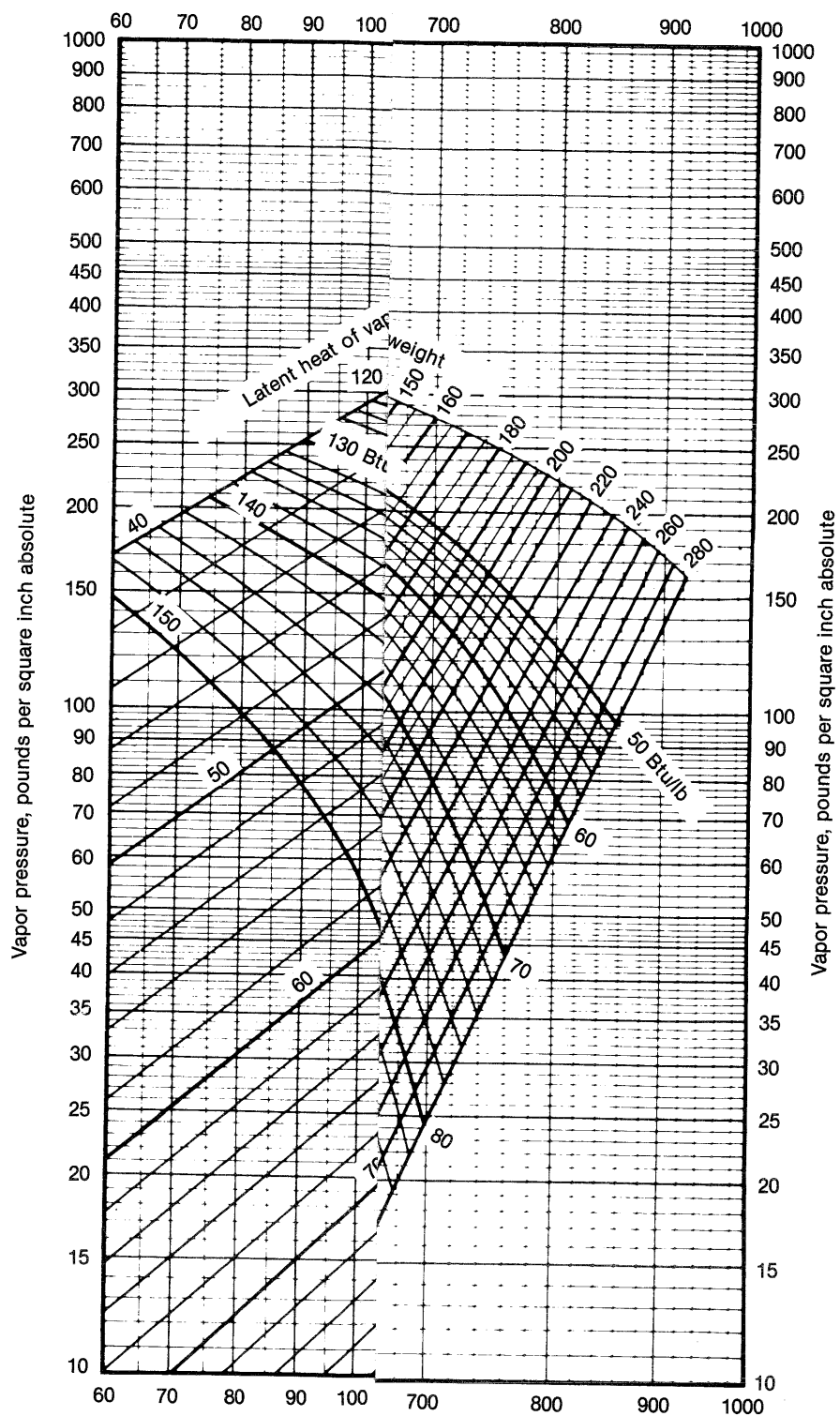


Figure D-3—Vapor Pressure and Heat of Vaporization of Pure Single-Component Paraffin Hydrocarbon Liquids

Where:

k = thermal conductivity of insulation, in British thermal units per hour per square foot per degree Fahrenheit per inch at mean temperature.

T_f = temperature of vessel contents at relieving conditions, in degrees Fahrenheit.

t = thickness of insulation, in inches.

D.8.3.2 Earth-Covered Storage

Covering a pressure vessel with earth is another effective method of limiting heat input. The reduction of heat absorption due to the earth cover can be calculated as suggested in D.8.3.1.

D.8.3.3 Cooling the Surface of a Vessel With Water

Under ideal conditions, water films covering the metal surface can absorb most incident radiation. The reliability of water application depends on many factors. Freezing weather, high winds, clogged systems, undependable water supply, and vessel surface conditions can prevent uniform water coverage. Because of these uncertainties, no reduction in environment factor (see Table D-3) is recommended; however, as stated previously, properly applied water can be very effective.

D.8.3.4 Limiting Fire Areas With Diversion Walls

Diversion walls can be provided to deflect vessel spills from other vessels.

APPENDIX E—REVIEW OF FLOW EQUATIONS USED IN SIZING PRESSURE RELIEF VALVES

E.1 Development of Flow Equations

The development of the vapor flow equation has been outlined in various places including standard thermodynamic texts and papers presented to API. Basically, the equation is determined by a mass and energy balance around the relief valve nozzle. Since the pressure changes as the vapor is accelerated in the nozzle, a relationship between pressure and volume is required to describe the changes in energy that occur. This pressure-volume relationship is described along an isentropic path to permit calculation of the maximum flow that can be obtained in a nozzle and serves as a reference point to determine the efficiency of an actual nozzle. The equation obtained is rearranged algebraically to include only those variables that are readily available at the inlet of the relief valve.

The vapor flow equation used to determine the capacity of a pressure relief valve was developed under the following assumptions:

- That the ideal gas laws adequately described the pressure-volume relationship of the expanding vapor.
- That no heat was transferred to or from the nozzle of the pressure relief valve (that is, adiabatic flow).
- That the vapor expansion followed an isentropic path.

The assumption in Item b is representative of the conditions obtained in the nozzle of a pressure relief valve and does not impact the capacity equation to a significant degree. The assumption that the vapor expands isentropically only provides a convenient means to determine the maximum capacity of a particular nozzle; the actual capacity of the nozzle is determined by a flow test and the ratio of actual flow, and the theoretical flow, K_D , becomes a derating factor applied to the flow equation.

The assumption that the vapor obeys the ideal gas laws refers only to the pressure-volume relationship that is obtained during an isentropic expansion. This relationship can specifically be described by the following equation:

$$PV^k = \text{constant}$$

Where:

P = pressure.

V = volume.

k = specific heat ratio.

This relationship affects the capacity equation through the coefficient C , which is a function of the specific heat ratio. (See Figure 26.)

Even though many vapors encountered in refinery service do not follow the ideal gas laws, in most cases a safety relief valve (SRV) is adequately sized based on this assumption. However, there may be unusual situations where deviations from ideal behavior are significant. In those cases, an isentropic expansion coefficient is used to characterize the actual pressure-temperature relationship that exists in the SRV nozzle. Since this coefficient is used in the same way as the ideal specific heat ratio, the form of the vapor sizing equation is identical. The coefficient C is calculated for a real gas using the isentropic expansion coefficient n instead of the specific heat ratio k .

Determining the isentropic expansion coefficient for a real gas is somewhat complicated because it is a function of both pressure and temperature. The coefficient can generally be obtained from an equation of state that describes the pressure-volume relationship along any thermodynamic path but is restricted to an isentropic expansion path. Since many of the relieving streams encountered are at their dew points (saturated condition), Figure E-1 gives the isentropic expansion coefficient for paraffin-hydrocarbons as a function of pressure. (The temperature dependence of the isentropic expansion coefficient is contained implicitly by specifying the vapor to be at its dew point.)

Figure E-2 gives the coefficient C as a function of the isentropic expansion coefficient. This figure is an extension of Figure 26 and provides for expansion coefficients that are less than 1.0.

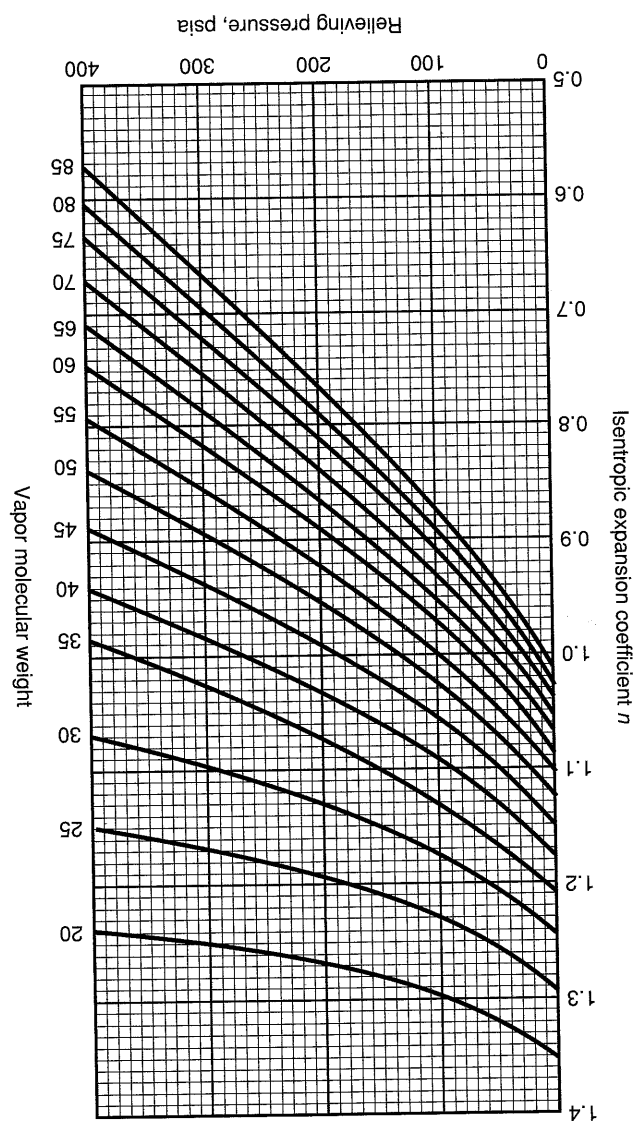


Figure E-1—Curves for Paraffin Hydrocarbons
Expanding From Relieving Pressures at or
Near Saturation to Critical Flow Pressures

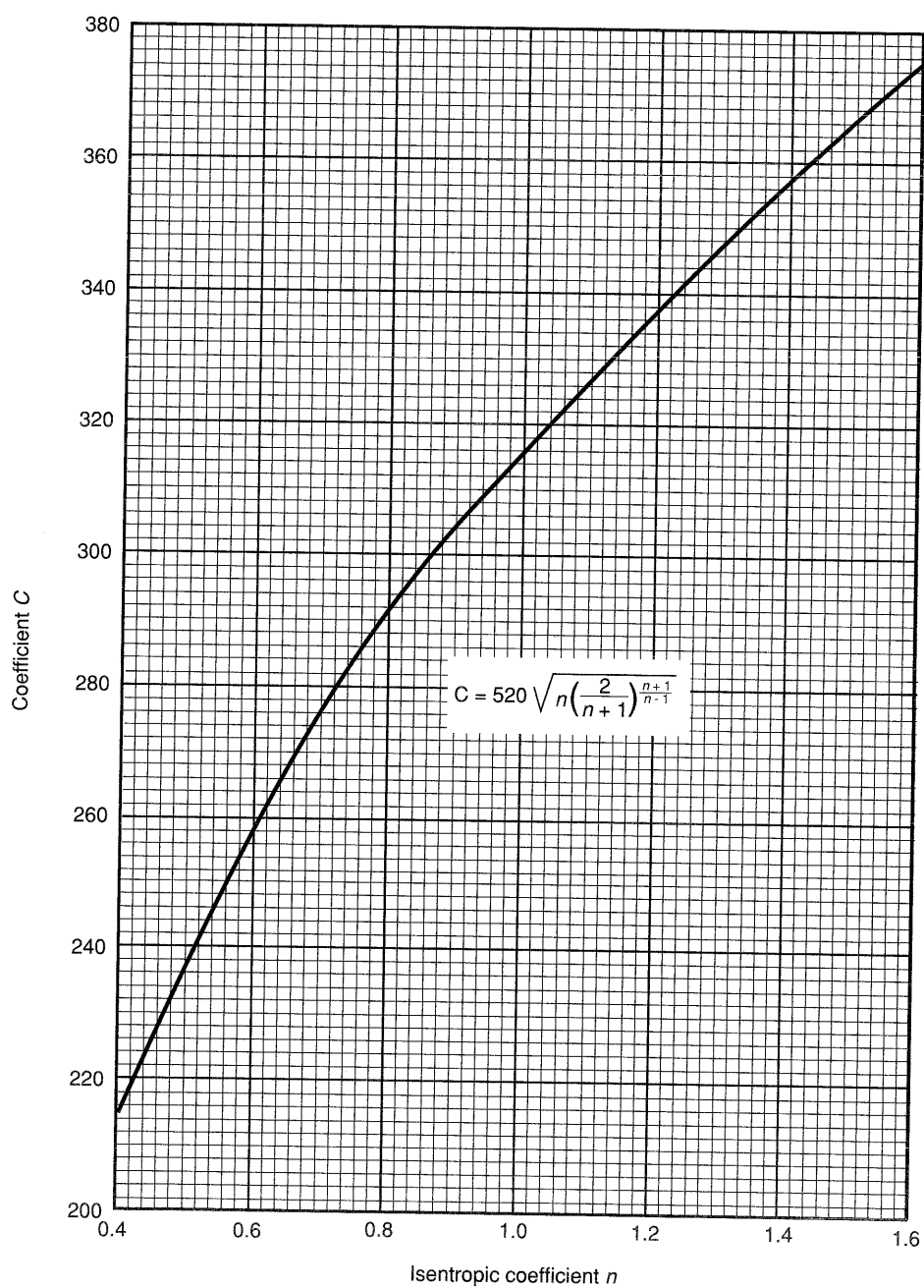


Figure E-2—Curve for Evaluating Coefficient C in Safety Relief Valve Flow Formula From Isentropic Coefficient n

APPENDIX F—DESIGN OF RELIEF MANIFOLDS

F.1 Introduction

The design and selection of relief disposal systems is discussed in API Recommended Practice 521.

This appendix deals specifically with the design of the inlet and outlet manifolds connected to a pressure relief device. Excessive pressure loss in either the inlet or outlet manifold may result in loss of protection. Acceptable pressure losses depend on the type of device used and its set pressure.

F.2 Design of Inlet Manifolds

Section 2 of API Recommended Practice 520, Part II, provides information on the design of inlet piping for pressure relief devices. The designs of some pressure relief devices are more sensitive to the effects of inlet pressure losses than others. Several sizing methods have been developed that minimize inlet piping calculations in most cases and allow the designer to quickly identify marginal situations [14, 15].

F.3 Design of Outlet (Discharge) Manifolds

F.3.1 GENERAL

Where a number of pressure relief devices discharge into the same manifold, caution should be exercised to ensure that there is not excessive developed pressure in the manifold. The allowable back pressure on a pressure relief device varies with design and is discussed in more detail in Section 5 of API Recommended Practice 520, Part II and Section 5 of API Recommended Practice 521.

When discharge manifolds and relief headers are sized, the relief contingency that produces the greatest back pressure must be identified. Any single relief contingency may involve several pressure relief devices. Typical relief contingencies that may be considered include cooling water failure, power failure, and valve malfunction.

The design of pipe anchors and supports on discharge manifolds may require special consideration. Sudden changes in flow rate and temperature can produce large reaction forces; if liquids are present in the relief system, the momentum forces can be significant. API Recommended Practice 520, Part II, discusses this subject in more detail.

F.3.2 VAPOR FLOW

Vapor flow in relief discharge piping is characterized by rapid changes in density and velocity; consequently, the flow should be treated as compressible. Several methods for calculating the size of discharge piping have been developed using isothermal or adiabatic flow equations. Actual flow

conditions in relief systems will normally be somewhere between isothermal and adiabatic conditions. For most cases, the slightly more conservative isothermal equations are recommended; however, the adiabatic flow equations may be preferable for some less common applications (for example, cryogenic conditions).

The sizing of relief discharge piping can usually be simplified by starting at the system outlet, where the pressure is known, and working back through the system to verify acceptable back pressure at each pressure relief device. Calculations are performed in a step-wise manner for each pipe segment of constant diameter. The isothermal flow equation based on inlet pressure [16] is as follows:

$$\frac{fL}{D} = \left(\frac{1}{M_1^2} \right) \left[1 - \left(\frac{P_2}{P_1} \right)^2 \right] - \ln \left(\frac{P_1}{P_2} \right)^2$$

The equation can be transposed to the following for outlet pressure:

$$\frac{fL}{D} = \left(\frac{1}{M_2^2} \right) \left[\left(\frac{P_1}{P_2} \right)^2 \left[1 - \left(\frac{P_2}{P_1} \right)^2 \right] - \ln \left(\frac{P_1}{P_2} \right)^2 \right] \quad (\text{F-1})$$

Where:

f = Moody friction factor.

L = equivalent length of pipe, in feet.

D = pipe inside diameter, in feet.

M_1 = Mach number at pipe inlet.

M_2 = Mach number at pipe outlet.

$\ln$ = log to base e, or log normal.

P_1 = pipe inlet pressure, in pounds per square inch absolute.

P_2 = pipe outlet pressure, in pounds per square inch absolute.

The outlet Mach number is given by the following equation:

$$M_2 = 1.702 \times 10^{-5} \left(\frac{W}{P_2 D^2} \right) \left(\frac{ZT}{Mw} \right)^{(1/2)} \quad (\text{F-2})$$

Where:

W = gas flow rate, in pounds per hour.

Z = gas compressibility factor.

T = absolute temperature, in degrees Rankine.

Mw = gas molecular weight.

Both graphical and computerized methods have been developed for solving Equations F-1 and F-2 and calculating pipe inlet pressure [17, 18]. Figure F-1 is a typical graphical representation of Equation F-1. The figure may be used to

calculate the inlet pressure, P_1 , for a line segment of constant diameter where the outlet pressure is known. If the relief system is to be operated at high pressure, the flow may be sonic in some parts of the system. In these cases, a check should be made to see if the flow is critical. The critical pressure at the pipe outlet is given by the following equation:

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

Where:

P_{Critical} = critical pressure, in pounds per square inch absolute.

d = pipe inside diameter, in inches.

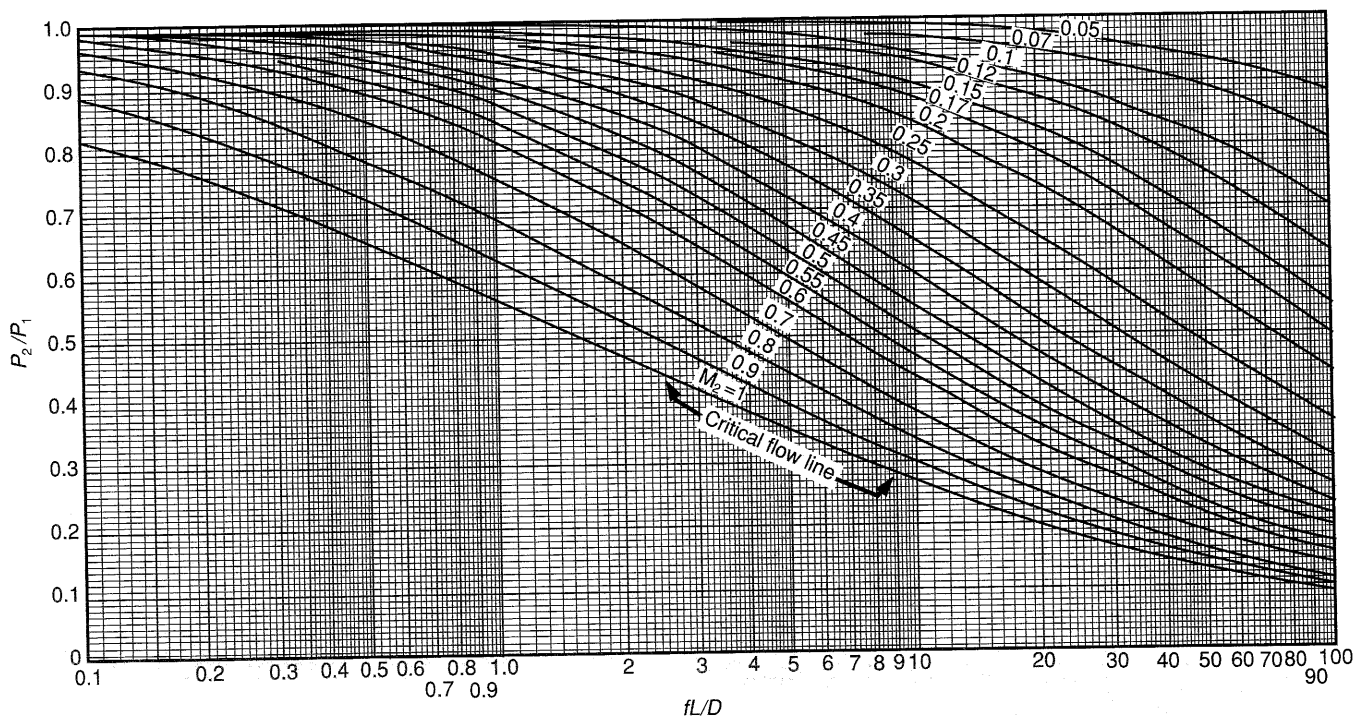
If the critical pressure is less than the pipe outlet pressure, the flow is subsonic. If the critical pressure is greater than the pipe outlet pressure, the flow is sonic and $M_2 = 1$. Therefore, the pipe inlet pressure, P_1 , is calculated from Equation F-1 with P_2 equal to the critical pressure.

API Recommended Practice 521 presents another isothermal calculation method based on the Lapple Chart [16]. This method utilizes upstream pressure.

F.3.3 MIXED-PHASE FLOW

The sizing of relief discharge piping that handles mixed-phase flow is generally complex. API Recommended Practice 521 discusses the major aspects of mixed-phase flow and provides additional references in Sections 6.1, 6.4, and 6.5.

(Based on outlet pressure)



Note: Reprinted with permission from the *Oil and Gas Journal*, Nov. 20, 1978, p. 166.

Figure F-1—Isothermal Flow Chart

APPENDIX G—PRESSURE RELIEF VALVE SPECIFICATION SHEET

A line-by-line description of the information to be provided on the pressure relief valve specification sheet is provided in this appendix, followed by a typical blank specification sheet. (See Figure G-1).

Line No.	Instruction
1.	Fill in item number.
2.	Fill in user's pressure relief valve identification number.
3.	Specify service, line, or equipment on which valve is to be mounted.
4.	Specify number of valves required.
5.	Give description of valve inlet (full nozzle, semi-nozzle, or other type).
6a.	Specify whether valve is classified as safety, relief, or safety relief type.
6b.	Specify whether valve is conventional, balanced bellows, or pilot operated.
7.	Specify open or closed bonnet.
8.	Specify nominal pipe size of inlet and outlet connections.
9.	Specify ANSI class of flanged connections for inlet and outlet flanges. If the valve has threaded connections, write in "threaded."
10.	Specify flange facing for inlet and outlet.
11.	Specify material of body and bonnet.
12a.	Specify material of seat or nozzle and disk.
12b.	If a resilient seat seal is required, specify material of seal, otherwise, "no."
13.	Specify material of guide and adjusting ring or rings.
14.	Specify material of spring.
15.	Specify material of bellows.
16.	If valve is to have a cap and no lever, specify screwed or bolted cap.
17.	If valve is to have a lifting lever, specify whether plain or packed.
18.	If a test gag is required, write in "yes;" otherwise, "no."
19-20.	Specify other accessories that are required (for example, wire seal).
21.	Specify the applicable code.
22.	Write in "yes" if fire governs a selection of the valve.
23.	Specify basis of selection if other than fire, such as cooling water failure, blocked discharge, thermal relief, or other.
24.	Indicate flowing fluid and state (liquid, gas, or vapor).
25.	Specify quantity of fluid that the valve is required to relieve at relieving conditions and fluid units (such as pounds per hour, gallons per minute, or cubic feet per minute).
26.	Specify the molecular weight or specific gravity of the fluid at the flowing temperature.
27.	Specify viscosity and units of viscosity at the flowing temperature.
28.	Specify operating pressure, in pounds per square inch gauge, and set pressure, in pounds per square inch gauge.
29.	Specify operating temperature, in degrees Fahrenheit, and temperature at which valve relieves, in degrees Fahrenheit.
30.	Specify the amount of superimposed back pressure that normally exists on the valve outlet, in pounds per square inch gauge.
31.	If back pressure is variable, specify the range of variation, in pounds per square inch gauge.
32.	Specify the differential set pressure. For conventional valves, the differential set pressure is the set pressure less the constant back pressure. For balanced valves, the differential pressure and the set pressure are the same.

Line No.	Instruction
33.	Specify the overpressure allowed, as a percent.
34.	Specify the compressibility factor, if used.
35.	Give the specific heat ratio as $k_p = C_p/C_v$.
36.	Specify the calculated orifice area, in square inches.
37.	Specify the selected orifice area, in square inches.
38.	Specify the letter designation of the selected orifice.
39.	Fill in the manufacturer's model or type numbers, if desired.
40.	Fill in the name of the manufacturer, if desired.

PRESSURE RELIEF VALVE SPECIFICATION SHEET		PAGE <u>1</u> OF <u>2</u> SHEET NO. _____ REQUISITION NO. _____ JOB NO. _____ DATE _____ REVISED _____ BY _____				
General						
1. Item number.	_____	_____	_____	_____	_____	_____
2. Tag number.	_____	_____	_____	_____	_____	_____
3. Service, line or equipment number.	_____	_____	_____	_____	_____	_____
4. Number required.	_____	_____	_____	_____	_____	_____
5. Full nozzle, semi-nozzle, or other.	_____	_____	_____	_____	_____	_____
6. Design type.	_____	_____	_____	_____	_____	_____
a. Safety, relief, or safety relief.	_____	_____	_____	_____	_____	_____
b. Conventional, bellows, or pilot operated.	_____	_____	_____	_____	_____	_____
7. Bonnet type.	_____	_____	_____	_____	_____	_____
Connections.						
8. Size (inlet/outlet).	____/____	____/____	____/____	____/____	____/____	____/____
9. Flange class, ANSI or threaded.	____/____	____/____	____/____	____/____	____/____	____/____
10. Type facing.	____/____	____/____	____/____	____/____	____/____	____/____
Material						
11. Body/bonnet.	____/____	____/____	____/____	____/____	____/____	____/____
12a. Seal/disk.	____/____	____/____	____/____	____/____	____/____	____/____
12b. Resilient rubber seal.	____/____	____/____	____/____	____/____	____/____	____/____
13. Guide/rings.	____/____	____/____	____/____	____/____	____/____	____/____
14. Spring.	____/____	____/____	____/____	____/____	____/____	____/____
15. Bellows.	____/____	____/____	____/____	____/____	____/____	____/____
Accessories						
16. Cap, screwed or bolted.	_____	_____	_____	_____	_____	_____
17. Lever, plain or packed.	_____	_____	_____	_____	_____	_____
18. Gag.	_____	_____	_____	_____	_____	_____
19. _____	_____	_____	_____	_____	_____	_____
20. _____	_____	_____	_____	_____	_____	_____
Basis of Selections						
21. Code.	_____	_____	_____	_____	_____	_____
22. Fire.	_____	_____	_____	_____	_____	_____
23. Other.	_____	_____	_____	_____	_____	_____
Service Conditions						
24. Fluid and state.	_____	_____	_____	_____	_____	_____
25. Required capacity per valve and units	_____	_____	_____	_____	_____	_____
26. Molecular weight or specific gravity at flowing temperature.	_____	_____	_____	_____	_____	_____
27. Viscosity at flowing temperature and units.	_____	_____	_____	_____	_____	_____
28. Operating pressure, psig, and set pressure, psig.	_____	_____	_____	_____	_____	_____

Figure G-1—Pressure Relief Valve Specification Sheet

PAGE <u> 2 </u> OF <u> 2 </u>	
PRESSURE RELIEF VALVE SPECIFICATION SHEET	SHEET NO. _____ REQUISITION NO. _____ JOB NO. _____ DATE _____ REVISED _____ BY _____
Service Conditions - Continued	
30. Constant back pressure, psig.	_____ _____ _____ _____ _____
31. Variable back pressure, psig.	_____ _____ _____ _____ _____
32. Differential set pressure.	_____ _____ _____ _____ _____
33. Allowable overpressure, as a percent.	_____ _____ _____ _____ _____
34. Compressibility factor.	_____ _____ _____ _____ _____
35. Ratio of specific heats.	_____ _____ _____ _____ _____
Orifice Area	
36. Calculated, in square inches.	_____ _____ _____ _____ _____
37. Selected, in square inches.	_____ _____ _____ _____ _____
38. Orifice designation.	_____ _____ _____ _____ _____
39. Manufacturer's model number.	_____ _____ _____ _____ _____
40. Manufacturer.	_____ _____ _____ _____ _____
Note:	

Figure G-1—Pressure Relief Valve Specification Sheet—Continued

Sizing, Selection, and Installation Of Pressure-Relieving Devices in Refineries

Part II—Installation

API RECOMMENDED PRACTICE 520
THIRD EDITION, NOVEMBER 1988



American National Standard

ANSI/API RP 520, Pt. II-1992

American Petroleum Institute
1220 L Street, Northwest
Washington, D.C. 20005



H. C. Lee
DEC

Sizing, Selection, and Installation Of Pressure-Relieving Devices in Refineries

Part II—Installation

Refining Department

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