Guide for Pressure-Relieving and Depressuring Systems

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ERRATA

This errata corrects an editorial error in the Fourth Edition of RP 521.

Page 64, Equation 30, shown below is incorrect:

$$U_c = 1.15\frac{\sqrt{gD(\rho_1 - \rho_v)}}{\rho_v(C)}$$

The correct version of Equation 30 is as follows:

$$U_c = 1.15\sqrt{\frac{gD(\rho_1 - \rho_v)}{\rho_v(C)}}$$

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Manufacturing, Distribution and Marketing Department

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FOREWORD

This recommended practice has been developed as a guide for plant engineers in the design, installation, and operation of pressure-relieving and depressuring systems. The text, based on the accumulated knowledge and experience of qualified engineers in petroleum-processing and related industries, recommends economically sound and safe practices for pressure relief.

Before this recommended practice was published, no source of collected information of this type was available for reference. The development of API Recommended Practice 520, *Sizing, Selection, and Installation of Pressure-Relieving Devices in Refineries*, disclosed the existence of detailed information in the files of participating individuals; Recommended Practice 521 is a compilation of these pertinent data and is published as an adjunct to API Recommended Practice 520.

As modern processing units become more complex in design and operation, the levels of energy stored in these units point to the importance of reliable, carefully designed pressure-relieving systems. Suggested solutions to the immediate design, economic, and safety problems involved in pressure-relieving discharge systems are presented herein. Users of this recommended practice are, however, reminded that no publication of this type can be complete, nor can any written document be substituted for qualified engineering analysis.

This edition incorporates both editorial changes and major changes based on experience gained since the third edition was published. In line with the general practice for API publications, metric numbers, unit designations, and formulas have been included in the text.

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.

IMPORTANT INFORMATION CONCERNING USE OF ASBESTOS OR ALTERNATIVE MATERIALS

Asbestos is specified or referenced for certain components of the equipment described in some API standards. It has been of extreme usefulness in minimizing fire hazards associated with petroleum processing. It has also been a universal sealing material, compatible with most refining fluid services.

Certain serious adverse health effects are associated with asbestos, among them the serious and often fatal diseases of lung cancer, asbestosis, and mesothelioma (a cancer of the chest and abdominal linings). The degree of exposure to asbestos varies with the product and the work practices involved.

Consult the most recent edition of the Occupational Safety and Health Administration (OSHA), U.S. Department of Labor, Occupational Safety and Health Standard for Asbestos, Tremolite, Anthophyllite, and Actinolite, 29 *Code of Federal Regulations,* Section 1910.1001; the U.S. Environmental Protection Agency, *National Emission Standard for Asbestos,* 40 *Code of Federal Regulations,* Sections 61.140 through 61.156; and the proposed rule by the U.S. Environmental Protection Agency (EPA) proposing labeling requirements and phased banning of asbestos products, published at 51 *Federal Register* 3738–3759 (January 29, 1986; the most recent edition should be consulted).

There are currently in use and under development a number of substitute materials to replace asbestos in certain applications. Manufacturers and users are encouraged to develop and use effective substitute materials that can meet the specifications for, and operating requirements of, the equipment to which they would apply.

SAFETY AND HEALTH INFORMATION WITH RESPECT TO PARTICULAR PRODUCTS OR MATERIALS CAN BE OBTAINED FROM THE EMPLOYER, THE MANUFACTURER OR SUPPLIER OF THAT PRODUCT OR MATERIAL, OR THE MATERIAL SAFETY DATA SHEET.

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SECTION 1—GENERAL

1.1 Scope

This recommended practice is applicable to pressure-relieving and vapor depressuring systems. The information provided is designed to aid in the selection of the system that is most appropriate for the risks and circumstances involved in various installations. This recommended practice is intended to supplement the practices set forth in API Recommended Practice 520, Part 1, for establishing a basis of design.

This recommended practice provides guidelines for examining the principal causes of overpressure; determining individual relieving rates; and selecting and designing disposal systems, including such component parts as vessels, flares, and vent stacks.

Piping information pertinent to pressure-relieving systems is presented in 5.4.1, but the actual piping should be designed in accordance with ASME B31.3 or other applicable codes.

Health risks may be associated with the operation of pressure-relieving equipment. The discussion of specific risks is outside the scope of this document.

1.2 Referenced Publications

The most recent editions of the following standards, codes, and specifications are cited in this recommended practice. Additional references are listed at the end of Sections 3, 4, and 5 and in the Bibliography, Section 6.

API

RP 520	*Sizing, Selection, and Installation of Pressure-Relieving Devices in Refineries*
Std 526	*Flanged Steel Safety-Relief Valves*
Std 527	*Seat Tightness of Pressure Relief Valves*
Std 2000	*Venting Atmospheric and Low-Pressure Storage Tanks: Nonrefrigerated and Refrigerated*
RP 2003	*Protection Against Ignitions Arising Out of Static, Lightning, and Stray Currents*
Publ 2216	*Ignition Risk of Hydrocarbon Vapors by Hot Surfaces in Open Air*
Publ 2218	*Fireproofing Practices in Petroleum and Petrochemical Processing Plants (out of print)*
Std 2510	*Design and Construction of LP-Gas Installations at Marine and Pipeline Terminals, Natural Gas Processing Plants, Petrochemical Plants, and Tank Farms.*

AGA[1]

Purging Principles and Practice (Catalog Number XK0775)

ASME[2]

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

B31.3	*Chemical Plant and Petroleum Refinery Piping*
PTC 25	*Pressure Relief Devices*

NFPA[3]

30	*Flammable and Combustible Liquid Code*
68	*Guide for Venting Deflagrations*
69	*Explosion Protection Systems*
78	*Lightning Protection Code*
325M	*Fire-Hazard Properties of Flammable Liquids, Gases, and Volatile Solids,* Volume I

1.3 Definition of Terms

Terms used in this recommended practice, as they relate to pressure-relieving systems, are defined in 1.3.1 through 1.3.37. Many of the terms and definitions are taken from API Recommended Practice 520, Part I, and ASME PTC 25.

1.3.1 accumulation: The pressure increase over the maximum allowable working pressure of a 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.

1.3.2 atmospheric discharge: The release of vapors and gases from pressure-relieving and depressuring devices to the atmosphere.

1.3.3 back pressure: The pressure that exists at the outlet of a pressure relief device as a result of the pressure in the discharge system. Back pressure can be either constant or variable. Back pressure is the sum of the superimposed and built-up back pressures.

1.3.4 balanced pressure relief valve: A spring-loaded pressure relief valve that incorporates a means for minimizing the effect of back pressure on the performance characteristics

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[1]American Gas Association, 1515 Wilson Boulevard, Arlington, Virginia 22209.
[2]American Society of Mechanical Engineers, 345 East 47th Street, New York, New York 10017.
[3]National Fire Protection Association, 1 Batterymarch Park, Quincy, Massachusetts 02269.

of the pressure relief valve (see Recommended Practice 520, Part I).

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

1.3.6 built-up back pressure: The increase in pressure in the discharge header that develops as a result of flow after the pressure relief device or devices open.

1.3.7 burst pressure: The inlet static pressure at which a rupture disk device functions.

1.3.8 closed-bonnet pressure relief valve: A pressure relief valve whose spring is totally encased in a metal housing. This housing protects the spring from corrosive agents in the environment and is a means of collecting leakage around the stem or disk guide. The bonnet may or may not be sealed against pressure leakage from the bonnet to the surrounding atmosphere, depending on the type of cap or lifting-lever assembly employed or the specific handling of bonnet venting.

1.3.9 closed disposal system: A disposal system capable of containing pressures that are different from atmospheric pressure.

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

1.3.11 conventional pressure relief valve: A spring-loaded pressure relief valve whose performance characteristics are directly affected by changes in the back pressure on the valve (see Recommended Practice 520, Part I).

1.3.12 design pressure of a vessel: At least the most severe condition of coincident temperature and gauge pressure expected during operation. It may be used in place of the maximum allowable working pressure in all cases where the maximum allowable working pressure has not been established. The design pressure is the pressure used in the design of a vessel to determine the minimum permissible thickness or other physical characteristics of the different parts of the vessel (see also maximum allowable working pressure).

1.3.13 flare: A means of safely disposing of waste gases through the use of combustion. With an elevated flare, the combustion is carried out at the top of a pipe or stack where the burner and igniter are located. A ground flare is similarly equipped except that combustion is carried out at or near ground level. A burn pit differs from a flare in that it is primarily designed to handle liquids.

1.3.14 huddling chamber: An annular pressure chamber in a pressure relief valve located beyond the seat for the purpose of generating a rapid opening.

1.3.15 lift: The actual travel of the disk away from the closed position when a valve is relieving.

1.3.16 maximum allowable accumulated pressure: The sum of the maximum allowable working pressure and the maximum allowable accumulation.

1.3.17 maximum allowable working pressure: 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.

1.3.18 open-bonnet pressure relief valve: A pressure relief valve whose spring is directly exposed to the atmosphere through the bonnet or yoke. Depending on the design, the spring may be protected from contact with vapors or gases discharged by the valve and will be cooled by the free passage of ambient air through and around the spring.

1.3.19 open disposal system: A disposal system that discharges directly from the relieving device to the atmosphere with no containment other than a short tail pipe.

1.3.20 operating pressure: The pressure to which the vessel is usually subjected in service. A pressure vessel is normally designed for a maximum allowable working pressure that will provide a suitable margin above the operating pressure in order to prevent any undesirable operation of the relieving device.

1.3.21 overpressure: 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, assuming no inlet pipe loss to the relieving device.

Note: When the set pressure of the first, or primary, pressure relief valve to open is less than the vessel's maximum allowable working pressure, the overpressure may be greater than 10 percent of the valve's set pressure.

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

1.3.23 pressure relief valve: A generic term applied to relief valves, safety valves, and safety relief valves. A pressure relief valve is designed to automatically reclose and prevent the flow of fluid.

1.3.24 pressure-relieving system: An arrangement of a pressure-relieving device, piping, and a means of disposal intended for the safe relief, conveyance, and disposal of fluids in a vapor, liquid, or gaseous phase. A relieving system may consist of only one pressure relief valve or rupture disk, either with or without discharge pipe, on a single vessel or line. A more complex system may involve many pressure-relieving devices manifolded into common headers to terminal disposal equipment.

1.3.25 quenching: The cooling of a fluid by mixing it with another fluid of a lower temperature.

1.3.26 rated relieving capacity: That portion of the measured relieving capacity permitted by the applicable code or regulation to be used as a basis for the application of a pressure relief device.

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

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

1.3.29 rupture disk device: 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.

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

1.3.31 sealed block valve: A valve that may be sealed fully open or fully closed during normal operation. Sealing may be achieved by applying car seals or positive locks (see the ASME *Boiler and Pressure Vessel Code*, Section VIII, Paragraphs UG-135, M-5, and M-6).

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

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

1.3.34 stamped burst pressure: The value of the pressure differential across the rupture disk at a coincident temperature at which the rupture disk is designed to burst. It is derived from destructive tests performed on 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.

1.3.35 superimposed back pressure: The static pressure that exists at the outlet of a pressure relief device at the time the device is required to operate. It is the result of pressure in the discharge system coming from other sources, and it may be either constant or variable.

1.3.36 vapor depressuring system: A protective arrangement of valves and piping intended to provide for rapid reduction of pressure in equipment by releasing vapors. The actuation of the system may be automatic or manual.

1.3.37 vent stack: The elevated vertical termination of a disposal system that discharges vapors into the atmosphere without combustion or conversion of the relieved fluid.

SECTION 2—CAUSES OF OVERPRESSURE

2.1 General

This section discusses the principal causes of overpressure in refinery equipment and offers guidance in plant design to minimize the effects of these causes. Overpressure is the result of an unbalance or disruption of the normal flows of material and energy that causes the material or energy, or both, to build up in some part of the system. Analysis of the causes and magnitudes of overpressure is, therefore, a special and complex study of material and energy balances in a process system.

The application of the principles outlined in this section will be unique for each processing system. Although efforts have been made to cover all major circumstances, the user is cautioned not to consider the conditions described as the only causes of overpressure. The treatment of overpressure in this recommended practice can only be suggestive. Any circumstance that reasonably constitutes a hazard under the prevailing conditions for a system should be considered in the design. Pressure-relieving devices are installed to ensure that a process system or any of its components are not subjected to pressures that exceed the maximum allowable accumulated pressure.

2.2 Overpressure Criteria

The causes of overpressure, including external fire, are considered to be unrelated if no process or mechanical or electrical linkages exist among them, or if the length of time that elapses between possible successive occurrences of these causes is sufficient to make their classification unrelated. The simultaneous occurrence of two or more conditions that could result in overpressure will not be postulated if the causes are unrelated.

Operator error is considered a potential source of overpressure.

The practices evaluated in this section should be used in conjunction with sound engineering judgment and with full consideration of federal, state, and local rules and regulations.

In addition, some relieving scenarios require the installation of high-integrity protective instrument systems to prevent overpressure and/or over-temperature. If this approach is used, the protective instrument system shall be at least as reliable as a pressure-relief device system, and shall be used only when the use of pressure relief devices is impractical.

Fail-safe devices, automatic start-up equipment, and other conventional control instrumentation should not replace pressure-relieving devices as protection for individual process equipment. However, in the design of some components of a relieving system, such as the blowdown header, flare, and flare tip, favorable instrument response of some percentage of instrument systems can be assumed. The percentage of favorable instrument responses is generally calculated based on the amount of redundancy, maintenance schedules, and other factors that affect instrument reliability.

2.3 Potentials for Overpressure

2.3.1 GENERAL

Pressure vessels, heat exchangers, operating equipment, and piping are designed to contain the system pressure. The design is based on (a) the normal operating pressure at operating temperatures, (b) the effect of any combination of mechanical loadings that is likely to occur, and the differential between the operating, and (c) set pressures of the pressure-relieving device. The process systems designer must define the minimum relief required to prevent the pressure in any piece of equipment from exceeding the maximum allowable accumulated pressure. The principal causes of overpressure listed in 2.3.2 through 2.3.16 will serve as guides to generally accepted safe practices.

2.3.2 CLOSED OUTLETS ON VESSELS

The inadvertent closure of a block valve on the outlet of a pressure vessel while the plant is on stream may expose the vessel to a pressure that exceeds the maximum allowable working pressure. If closure of an outlet block valve can result in overpressure, a pressure relief device is required unless administrative procedures to control valve closure, such as car seals or locks, are in place. Every control valve should be considered as being subject to inadvertent operation. In general, the omission of block valves interposed in vessels in a series can simplify pressure-relieving requirements.

For system capacity design, it may be assumed that control valves which are normally open and functioning at the time of failure and that are not affected by the primary cause of failure will remain in operation at their normal operating positions. See 3.10.4 for additional information.

2.3.3 INADVERTENT VALVE OPENING

The inadvertent opening of any valve from a source of higher pressure, such as high-pressure steam or process fluids, should be considered. This action may require pressure-relieving capacity unless provisions are made for locking or sealing the valve closed.

2.3.4 CHECK-VALVE MALFUNCTION

The failure of a check valve to close must also be considered. For example, where a fluid is pumped into a process system that contains gas or vapor at significantly higher pressures than the design rating of equipment upstream of the pump, cessation of flow accompanied by failure of a check valve in the discharge line will result in a reversal of the liquid's flow. When the liquid has been displaced into a suction system and a high-pressure fluid enters, serious overpressure may result. A single check valve is usually considered acceptable unless a potential exists for backflow of high-pressure fluid to create pressures that exceed the test pressure of the equipment. In these cases, one should consider providing a secondary device to minimize the potential for a reversal of flow. The device may be a nonreturn valve, a power-assisted check valve, a second conventional check valve, or similar equipment. The sizing of pressure-relieving facilities on the suction side to accommodate the peak flow following the failure of a check valve is normally not recommended, since reverse flow through rotating machinery may result in centrifugal forces that are sufficient to destroy mechanical equipment.

2.3.5 UTILITY FAILURE

The consequences that may develop from the loss of any utility service, whether plantwide or local, must be carefully evaluated. The normal utility services that could fail and a partial listing of affected equipment that could cause overpressure are given in Table 1.

Table 1—Possible Utility Failures and Equipment Affected

Utility Failure	Equipment Affected
Electric	Pumps for circulating cooling water, boiler feed, quench, or reflux
	Fans for air-cooled exchangers, cooling towers, or combustion air
	Compressors for process vapor, instrument air, vacuum, or refrigeration
	Instrumentation
	Motor-operated valves
Cooling Water	Condensers for process or utility service
	Coolers for process fluids, lubricating oil, or seal oil
	Jackets on rotating or reciprocating equipment
Instrument air	Transmitters and controllers
	Process-regulating valves
	Alarm and shutdown systems
Steam	Turbine drivers for pumps, compressors, blowers, combustion air fans, or electric generators
	Reboilers
	Reciprocating pumps
	Equipment that uses direct steam injection
	Eductors
Fuel (oil, gas, etc.)	Boilers
	Reheaters (reboilers)
	Engine drivers for pumps or electric generators
	Compressors
	Gas turbines
Inert gas	Seals
	Catalytic reactors
	Purge for instruments and equipment

2.3.6 PARTIAL FAILURE

An evaluation of the effect of overpressure that is attributable to the loss of a particular utility service should include the chain of developments that could occur and the reaction time involved. In situations in which the equipment fails but operates in parallel with equipment that has a different energy source, operating credit may be taken for the unaffected and functioning equipment to the extent that service is maintained. An example would be a cooling-water circulating system that consists of two pumps in parallel service and continuous operation whose drivers have unrelated energy sources. If one of the two energy sources fails, partial credit may be taken for the other power source that continues to function (see 2.2). The quantity of excess vapor generated because of the energy failure then depends solely on the quantity of cooling water lost. Another example would be two cooling-water pumps in parallel service, with one pump providing the full flow of cooling water and the second being in standby service. The second pump has a separate energy source and is equipped with controls for automatic start-up if the first pump fails. No protective credit is taken for the standby pump because the standby device is not considered totally reliable.

After detailed study, full or partial protective credit may be taken for normally operating parallel instrument air compressors and electric generators that have two unrelated sources of energy to the drivers. Manual cut-in of auxiliaries is operator and time dependent and must be carefully analyzed before it is used as insurance against overpressure.

2.3.7 ELECTRICAL OR MECHANICAL FAILURE

The failure of electrical or mechanical equipment that provides cooling or condensation in process streams can cause overpressure in process vessels.

2.3.8 LOSS OF FANS

Fans on air-cooled heat exchangers or cooling towers occasionally become inoperative because of a loss of power or a mechanical breakdown. On cooling towers and air-cooled exchangers where independent operation of the louvers can be maintained, credit for the cooling effect may be obtained by convection and radiation in still air at ambient conditions.

2.3.9 LOSS OF HEAT IN SERIES FRACTIONATION SYSTEMS

In series fractionation (that is, where the bottoms from the first column feed into the second column, and the bottoms from the second feed into the third), the loss of heat input to a column can overpressure the following column. Loss of heat results in some of the light ends mixing with the bottoms and being transferred to the next column as feed. Under this circumstance, the overhead load of the second column may consist of its normal vapor load plus the light ends from the first column. If the second column does not have the condensing capacity for the additional vapor load, excessive pressure could occur.

2.3.10 LOSS OF INSTRUMENT AIR OR ELECTRIC POWER

The complexity of instrument automation on process units requires the provision of reliable and continuous sources of air or electric power, or both, for dependable operation. Where a single instrument air compressor is installed, an air receiver of generous size may suffice if it is supplemented by an emergency pressure-reducing station from the plant air system.

Key electronic or electrical instruments should be interconnected with an emergency electric source of the proper AC or DC voltage. The fail-safe condition of each control valve should be evaluated. Fail-safe refers to control valve action (spring open, spring closed, or fixed position) on the loss of operating air or electric power. To minimize the likelihood of overpressure, each control valve should have its fail-safe characteristics properly established as an integral part of the plant design. The failure position of a control valve in itself is not considered to be adequate relief protection as other failures in an instrument system can cause a control valve to move in a direction opposite its design failure position.

2.3.11 REFLUX FAILURE

The loss of reflux as a result of pump or instrument failure can cause overpressure in a column because of condenser flooding or loss of coolant in the fractionating process.

2.3.12 ABNORMAL HEAT INPUT FROM REBOILERS

Reboilers are designed with a specified heat input. When they are new or recently cleaned, additional heat input above the normal design can occur. In the event of a failure of temperature control, vapor generation can exceed the process system's ability to condense or otherwise absorb the buildup of pressure, which may include noncondensables caused by overheating.

2.3.13 HEAT-EXCHANGER TUBE FAILURE

In shell-and-tube heat exchangers, the tubes are subject to failure from a number of causes, including thermal shock, vibration, and corrosion. Whatever the cause, the result is the possibility that the high-pressure stream will overpressure equipment on the low-pressure side of the exchanger. The ability of the low-pressure system to absorb this release should be determined. The possible pressure rise must be ascertained to determine whether additional pressure relief would be required if flow from the tube rupture were to discharge into the lower-pressure stream.

2.3.14 TRANSIENT PRESSURE SURGES

2.3.14.1 Water Hammer

The probability of hydraulic shock waves, known as *water hammer*, occurring in any liquid-filled system should be carefully evaluated. Water hammer is a type of overpressure that cannot be reasonably controlled by pressure relief valves, since the response time of the valves is normally too slow. The oscillating peak pressures, measured in milliseconds, can rise to many times the normal operating pressure. These pressure waves damage the pressure vessels and piping where proper safeguards have not been incorporated. Water hammer is frequently caused by the action of quick-closing valves. Where water hammer can occur, the use of pulsation dampeners should be considered.

2.3.14.2 Steam Hammer

An oscillating peak pressure surge, called *steam hammer*, can occur in piping that contains compressible fluids. The most common occurrence is generally initiated by rapid valve closure. This oscillating pressure surge occurs in milliseconds, with a possible pressure rise to many times the normal operating pressure, resulting in vibration and violent movement of piping and possible rupture of equipment. Pressure relief valves cannot effectively be used as a protective device because of their slow response time. Avoiding the use of quick-closing valves may prevent steam hammer.

2.3.15 PLANT FIRES

Fire as a cause of overpressure in plant equipment is discussed in 3.15. A provision for initiating a controlled shutdown or installation of a depressuring system for the units can minimize overpressure that results from exposure to external fire.

To limit vapor generation and the possible spread of fire, facilities should also allow for the removal of liquids from the systems. Normally operating product-withdrawal systems are considered superior and more effective for removing liquids from a unit, compared with separate liquid-pulldown systems. Liquid holdup required for normal plant operations, including refrigerants or solvents, can be effective in keeping the vessel wall cool and does not necessarily require systems for its removal. Provisions may be made either to insulate the vessel's vapor space and apply external water for cooling or to depressure the vessel using a vapor depressuring system.

Area design should include adequate surface drainage facilities and a means for preventing the spread of flammable liquids from one operating area to another. Easy access to each area and to the process equipment must be provided for firefighting personnel and their equipment. Fire hydrants, firefighting equipment, and fire monitors should be placed in readily accessible locations.

Credit for insulation can be taken provided that the requirements of 3.15.5 are met.

2.3.16 PROCESS CHANGES/CHEMICAL REACTIONS

In some reactions and processes, loss of process control may result in a significant change in temperature and/or pressure. The result could exceed the intended limits of the materials selected. Thus, where cryogenic fluids are being processed, a reduction in pressure could lower the temperature of the fluids to a level below the minimum allowable design temperature of the equipment, with the attendant risk of a low-temperature brittle failure. For exothermic reactions (for example, decompositions, acid dilutions, polymerizations, for example), excessive temperatures and/or pressures associated with runaway reactions may reduce the allowable stress levels below the design point, or increase the pressure above the maximum allowable working pressure (MAWP). Where normal pressure relieving-devices cannot protect against these situations, controls are needed to warn of changes outside the intended temperature/pressure limits to provide corrective action (see 3.9, 3.10, and 3.13).

SECTION 3—DETERMINATION OF INDIVIDUAL RELIEVING RATES

3.1 Principal Sources of Overpressure

The basis for determining individual relieving rates that result from various causes of overpressure is presented in this section in the form of general considerations and specific proposals. Good engineering judgment, rather than blind adherence to these proposals, should be followed in each case. The results achieved should be economically, operationally, and mechanically feasible, but in no instance should the safety of a plant or its personnel be compromised.

Table 2 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. A more descriptive analysis is provided in the remainder of this section.

Table 2—Bases for Relief Capacities Under Selected Conditions

Item No.	Condition	Pressure Relief Device (Liquid Relief)[a]	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	—	
	Water into hot oil	—	For towers, usually not predictable
	Light hydrocarbons 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
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 non-condensables 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 3.14.3	—
	Lines outside process area shut in	See 3.14.3	—
15	Exterior fire	—	Estimated by the method given in 3.15
16	Power failure (steam, electric, or other)	—	Study the installation to determine the effect of power failure; size the 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 the 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	—

[a]Consideration may be given to the reduction of the relief rate as the result of the relieving pressure being above operating pressure.

3.2 Sources of Overpressure

The liquid or vapor rates used to establish relief requirements are developed by the net energy input. The two most common forms of energy are (a) heat input, which is indirect pressure input through vaporization or thermal expansion, and (b) direct pressure input from higher pressure sources. Overpressure may result from one or both of these sources.

The peak individual relieving rate is the maximum rate that must be relieved to protect equipment against overpressure due to any single cause. The probability of two unrelated failures occurring simultaneously is remote and normally does not need to be considered.

3.3 Effects of Pressure, Temperature, and Composition

Pressure and temperature should be considered to determine individual relieving rates, since they affect the volumetric and compositional behavior of liquids and vapors. Vapor is generated when heat is added to a liquid. The rate at which vapor is generated changes with equilibrium conditions because of the increased pressure in a confined space and the heat content of streams that continue to flow into and out of equipment. In many instances, a volume of liquid may be a mixture of components with different boiling points. Heat introduced into fluids that do not reach their critical temperature under pressure-relieving conditions produces a vapor that is rich in low-boiling components. As heat input is continued, successively heavier components are generated in the vapor. Finally, if the heat input is sufficient, the heaviest components are vaporized.

During pressure relieving, the changes in vapor rates and molecular weights at various time intervals should be investigated to determine the peak relieving rate and the composition of the vapor (see API Recommended Practice 520, Part I). The composition of inflowing streams may also be affected by variations in time intervals and, therefore, requires study.

Relieving pressure may sometimes exceed the critical pressure (or pseudocritical pressure) of the components in the system. In such cases, reference must be made to compressibility correlations to compute the density-temperature-enthalpy relationships for the system fluid. If the overpressure is the result of an inflow of excess material, then the excess mass quantity must be relieved at a temperature determined by equating the incoming enthalpy with the outgoing enthalpy.

In a system that has no other inflow or outflow, if the overpressure is the result of an extraneous excess heat input, the quantity to be relieved is the difference between the initial contents and the calculated remaining contents at any later time. The cumulative extraneous enthalpy input is equal to the total gain in enthalpy by the original contents, whether they remain in the container or are vented. By calculating or plotting the cumulative vent quantity versus time, the maximum instantaneous relieving rate can be determined. This maximum will usually occur near the critical temperature.

In such cases, the assumption of an ideal gas may be too conservative, and Equation 5 (see 3.15.2.1.2) will oversize the safety relief valve. This formula should be used only when physical properties for the fluid are not available.

3.4 Effect of Operator Response

The decision to take credit for operator response in determining maximum relieving conditions requires consideration of those who are responsible for operation and an understanding of the consequences of an incorrect action. A commonly accepted time range for the response is between 10 minutes and 30 minutes, depending on the complexity of the plant. The effectiveness of this response depends on the process dynamics.

3.5 Closed Outlets

To protect a vessel or system from overpressure when all outlets on the vessel or system are blocked, the capacity of the relief device must be at least as great as the capacity of the sources of pressure. If all outlets are not blocked, the capacity of the unblocked outlets may properly be considered. The sources of overpressure include pumps, compressors, high-pressure supply headers, stripped gases from rich absorbent, and process heat. In the case of heat exchangers, a closed outlet can cause thermal expansion (see 3.14) or possibly vapor generation.

The quantity of material to be relieved should be determined at conditions that correspond to the set pressure plus overpressure instead of at normal operating conditions. The required valve capacity is often reduced appreciably when this difference in conditions is considered. The effect of friction drop in the connecting line between the source of overpressure and the system being protected should also be considered in determining the capacity requirement.

3.6 Cooling or Reflux Failure

3.6.1 GENERAL

The required relieving rate is determined by a heat and material balance on the system at the relieving pressure. In a distillation system, the rate may require calculation with or without reflux. Credit is normally not taken for the effect of residual coolant after the cooling stream fails because this effect is time limited and depends on the physical configuration of the piping. However, if the process piping system is unusually large and bare, the effect of heat loss to the surroundings may be considered.

Because of the difficulty in calculating detailed heat and material balances, the simplified bases described in 3.6.2 through 3.6.9 have generally been accepted for determining relieving rates.

3.6.2 TOTAL CONDENSING

The relief requirement is the total incoming vapor rate to the condenser, recalculated at a temperature that corresponds to the new vapor composition at set pressure plus overpressure, and the heat input prevailing at the time of relief. The surge capacity of the overhead accumulator at the normal liquid level is generally limited to less than 10 minutes. If cooling failure exceeds this time, reflux is lost, and the overhead composition, temperature, and vapor rate may change significantly.

3.6.3 PARTIAL CONDENSING

The relief requirement is the difference between the incoming and outgoing vapor rate at relieving conditions. The incoming vapor rate should be calculated on the same basis used in 3.6.2. If the composition or rate of the reflux is changed, the incoming vapor rate to the condenser should be determined for the new conditions.

3.6.4 FAN FAILURE

Because of natural convection effects, credit for a partial condensing capacity of 20 percent to30 percent of normal duty is often used unless the effects at relieving conditions are determined to be significantly different. The capacity of the relief valve is then based on the remaining 70 percent to 80 percent, depending on the service (see 3.6.2 and 3.6.3). However, actual duty available by natural convection is usually a function of air-cooled heat exchanger design. Some designs may allow significantly more credits if a supporting engineering analysis is performed. In addition, reduction in cooling capabilities may also occur if variable pitch fans are used and failure of the pitch mechanism occurs.

3.6.5 LOUVER CLOSURE

Louver closure on air-cooled condensers is considered to be total failure of the coolant with the resultant capacity established as in 3.6.2 and 3.6.3. Louver closure may result from automatic control failure, mechanical linkage failure, or destructive vibration on a manually positioned louver.

3.6.6 OVERHEAD CIRCUIT

In many cases, failure of the reflux that results, for example, from pump shutdown or valve closure will cause flooding of the condenser, which is equivalent to total loss of coolant with the capacity established as in 3.6.2 and 3.6.3. Compositional changes caused by loss of reflux may produce different vapor properties that affect the capacity. A valve sized for total failure of the coolant will usually be adequate for this condition, but each case must be examined in relation to the particular components and system involved.

3.6.7 PUMP-AROUND CIRCUIT

The relief requirement is the vaporization rate caused by an amount of heat equal to that removed in the pump-around circuit. The latent heat of vaporization would correspond to the latent heat under the relieving conditions of temperature and pressure at the point of relief.

3.6.8 OVERHEAD CIRCUIT PLUS PUMP-AROUND

An overhead circuit plus pump-around is usually arranged so that simultaneous failure of the pump-around and the overhead condenser will not occur; however, partial failure of one with complete failure of the other is quite possible. The required relieving capacity is discussed in 3.6.6 and 3.6.7.

3.6.9 SIDESTREAM REFLUX FAILURE

Principles similar to those described in 3.6.6 and 3.6.7 apply in condenser flooding (if a condenser is in the system) or changes in vapor properties resulting from changes in composition. The relieving capacity should be large enough to relieve the vaporization rate caused by the amount of heat normally removed from the system.

3.7 Absorbent Flow Failure

For lean-oil absorption of hydrocarbons, generally no relief requirement results from lean-oil failure. However, in an acid-gas removal unit in which large quantities (25 percent or more) of the inlet vapor may be removed in the absorber, loss of absorbent could cause a pressure rise to relief pressure, since the downstream system may not be adequate to handle the increased flow. The case of a synthesis-gas carbon dioxide removal unit in which the downstream gas goes to a methanator is more complicated to analyze. Any quantity of carbon dioxide above design capability that enters the methanator, as occurs on even partial absorbent failure, produces a rapid temperature rise that usually closes a methanator feed shutoff valve and opens a vent to the atmosphere. If the vent to the atmosphere fails to open, a possibility of overpressure arises.

Each individual case must be studied for its process and instrumentation characteristics. The study should include the effect on downstream process units in addition to the reaction in piping and instrumentation immediately downstream of the absorber.

3.8 Accumulation of Noncondensables

Noncondensables do not accumulate under normal conditions, because they are released with the process streams. However, with certain piping configurations, noncondensables can accumulate to the point that the condenser is blocked. This effect is equal to a total loss of coolant.

3.9 Entrance of Volatile Material Into the System

3.9.1 WATER INTO HOT OIL

Although the entrance of water into hot oil remains a source of potential overpressure, no generally recognized methods for calculating the relieving requirements are available. In a limited sense, if the quantity of water present and the heat available in the process stream are known, the size of the relief valve can be calculated like that of a steam valve. Unfortunately, the quantity of water is almost never known, even within broad limits. Also, since the expansion in volume from liquid to vapor is so great (approximately 1:1,400 at atmospheric pressure) and the speed of vapor generation is essentially instantaneous, it is questionable whether the valve could open fast enough to be of value. Normally, no pressure-relieving device is provided for this contingency. Proper design and operation of the process system are essential in attempts to eliminate this possibility. Some precautions that can be taken are avoiding water-collecting pockets and installing proper steam condensate traps and double blocks and bleeds on water connections to hot process lines.

3.9.2 LIGHT HYDROCARBONS INTO HOT OIL

The information in 3.9.1 applies to the entrance of light hydrocarbons into hot oil even though the ratio of liquid volume to vapor volume may be considerably less than 1:1,400.

3.10 Failure of Process Stream Automatic Controls

3.10.1 GENERAL

Automatic control devices, directly actuated from the process or indirectly actuated from a process variable (for example, pressure, flow, liquid level, or temperature) are used at inlets or outlets of vessels or systems. When the transmission signal or operating medium to a final control element (such as a valve operator) fails, the control devices should assume either a fully open or fully closed position according to their basic design. Final control elements that fail in a stationary position should be assumed to fail fully open or fully closed (See 3.10.5). The failure of a process-measuring element in a transmitter or controller without coincidental failure of the operating power to the final controlled element should be reviewed to determine the effect on the final controlled element. Operation of the manual bypass valve is discussed in 3.16. Possible failure of the control device while the manual bypass valve is fully or partially open deserves to be considered; however, this factor is not intended to cover the condition of an undersized control valve.

In evaluating relief considerations, the designer should assume proper sizing of the control valve and unit operation at or near design unless he knows a specific condition that exists to the contrary. The designer should be alert to temporary start-up or upset conditions when unit operators are using the bypass valve. Since these are upset and off-control conditions, the probability that relieving requirements will arise is usually greater than when the unit is running normally under control with all bypasses closed.

3.10.2 CAPACITY CREDIT

In evaluating relieving requirements due to any cause, any automatic control valves that are not under consideration as causing a relieving requirement and which would tend to relieve the system should be assumed to remain in the position required for normal processing flow. In other words, no credit should be taken for any favorable instrument response. Normal valve position is the expected position of the valve prior to the upset incident, taking into consideration the design capacity and turn down of the system. Therefore, unless the condition of flow through the control valves changes (see 3.10.6), credit may be taken for the normal flow of these valves, corrected to relieving conditions, provided that the downstream system is capable of handling any increased flow. Although controllers actuated by variables other than the system pressure may try to open their valves fully, credit may be taken for such control valves only to the extent permitted by their operating position at normal flow regardless of the valve's initial condition.

3.10.3 INLET CONTROL DEVICES

There may be single or multiple inlet lines fitted with control devices. The scenario to consider is that one inlet valve will be in a fully opened position regardless of the control valve failure position. Opening of this control valve may be caused by instrument failure or misoperation. If the system has multiple inlets, the position of any control device in those remaining lines shall be assumed to remain in its normal operating position. Therefore, the required relief capacity is the difference between the maximum expected inlet flow and the normal outlet flow, adjusted for relieving conditions and considering unit turndown, assuming that the other valves in the system are still in operating position at normal flow (that is, normally open, normally closed, or throttling). If one or more of the outlet valves are closed, or more inlet valves are opened by the same failure that caused the first inlet valve to open, the required relief capacity is the difference between the maximum expected inlet flow and the normal flow from the outlet valves that remain open. All flows should be calculated at relieving conditions. An important consideration is the effect of having a manual bypass on the inlet control valve(s) at least partially open.

Other situations may arise where problems involved in evaluating relief requirements after the failure of an inlet control device are more complex and of special concern (for example, a pressure vessel operating at a high pressure where liquid bottoms are on level control and discharge into a lower pressure system). Usually, when the liquid is let down from the high-pressure vessel into the low-pressure system, only the flashing effect is of concern in the event that the low-pressure system has a closed outlet. However, the designer should also consider that vapors will flow into the low-pressure system if loss of liquid level occurs in the vessel at higher pressure. In this case, if the volume of the source of incoming vapors is large compared with the volume of the low-pressure system or if the source of vapor is unlimited, serious overpressure can rapidly develop. When this occurs, relief devices on the low-pressure system may need to be sized to handle the full vapor flow through the liquid control valve.

In circumstances where process systems involve significant differences in pressure level and the volume of vapor contained by the high-pressure equipment is less than the volume of the low-pressure system, the additional pressure may in some cases be absorbed without overpressure.

In the event of loss of liquid level, the vapor flow into the low-pressure system depends on what the interconnecting system, which usually consists of wide-open valves and piping, passes with a differential pressure based on the normal operating pressure upstream and the relieving pressure on equipment downstream. This pressure drop at initial conditions frequently results in critical flow and may cause the rate to be several times higher than the normal rate of vapor inflow to the high-pressure system. Unless makeup equals outflow, this condition will be of short duration as the upstream reservoir is depleted. Nonetheless, the relief facilities that protect the low-pressure system must be sized to handle peak flow. If the low-pressure side has a large vapor volume, it may prove worthwhile to take credit for the following: The transfer of vapors from the high-pressure system needed to raise the pressure on the downstream side from operating pressure to relieving pressure (normally 110 percent of design pressure) will lower the upstream pressure. This decrease produces a corresponding reduction in the flow that establishes the relieving requirement. Where such credit is taken, an allowance must be made for the normal makeup of vapor to the high-pressure system, which tends to maintain upstream pressure.

3.10.4 OUTLET CONTROL DEVICES

Each outlet control valve should be considered both in the fully opened and fully closed positions for the purposes of relief load determination. This is regardless of the control valve failure position and may be caused by instrument system failure or misoperation. If one or more of the inlet valves are opened by the same failure that caused the outlet valve to close, pressure-relieving devices may be required to prevent overpressure. The required relief capacity is the difference between the maximum inlet and maximum outlet flows. All flows should be calculated at relieving conditions. Also, one should consider the effects of inadvertent closure of control devices by operator action.

For applications involving single outlets with control devices that fail in the closed position, pressure-relieving devices on these outlets may be required to prevent overpressure. The required relief capacity is equal to the maximum expected inlet flow at relieving conditions and should be determined as outlined in 3.5.

For applications involving more than one outlet and a control device that fails in the closed position on an individual outlet, the required relief capacity is the difference between the maximum expected inlet flow and the design flow (adjusted for relieving conditions and considering unit turndown) through the remaining outlets, assuming that the other valves in the system remain in their normal operating position.

For applications involving more than one outlet, each with control devices that fail in the closed position because of the same failure, the required relief capacity is equal to the maximum expected inlet flow at relieving conditions.

3.10.5 FAIL-STATIONARY VALVES

Even though some control devices are designed to remain stationary in the last controlled position, one cannot predict the position of the valve at time of failure.Therefore, the designer should always consider that such devices could be either open or closed: no reduction in relief capacity should be considered when such devices are used.

3.10.6 SPECIAL CAPACITY CONSIDERATIONS

Although control devices, such as diaphragm-operated control valves, are specified and sized for normal operating conditions, they are also expected to operate during upset conditions, including periods when pressure-relieving devices are relieving. Valve design and valve operator capability should be selected to position the valve plug properly in accordance with control signals during abnormal conditions. Since the capacities at pressure-relieving conditions are not the same as those at normal conditions, the capacities of control valves should be calculated for the relieving conditions of temperature and pressure in determining the required relief capacities. In extreme cases, the state of the fluid controlled may change (for example, from liquid to gas or from gas to liquid). The wide-open capacity of a control valve selected to handle a liquid may, for example, differ greatly when it handles a gas. This becomes a matter of particular concern where loss of liquid level can occur, causing the valve to pass high-pressure gas to a system sized to handle only the vapor flashed from the normal liquid entry.

3.11 Abnormal Process Heat Input

The required capacity is the maximum rate of vapor generation at relieving conditions (including any noncondensables produced from overheating) less the rate of normal condensation or vapor outflow. In every case, the designer should consider the potential behavior of the system and each of its components. For example, the fuel or heat-medium control valve or the tube heat flux may be the limiting consideration. To be consistent with the practice used for other causes of overpressure, design values should be used for an item such as valve size. However, built-in overcapacity, which is applicable to the common practice of specifying burners capable of 125 percent of heater design heat input, must be considered.

Where limit stops are installed on valves, the wide-open capacity, rather than the capacity at the stop setting, should normally be used. However, if a mechanical stop is installed and is adequately documented, use of the limited capacity may be appropriate. In shell-and-tube heat-exchange equipment, heat input should be calculated on the basis of clean, rather than fouled, conditions.

3.12 Internal Explosion (Excluding Detonation)

Where overpressure protection against internal explosions caused by ignition of vapor-air mixtures is to be provided, rupture discs or explosion vent panels, not relief valves, should be used. Relief valves react too slowly to protect the vessel against the extremely rapid pressure buildup caused by internal flame propagation. The vent area required is a function of a number of factors including the following:

a. Initial conditions (pressure, temperature, composition).

b. Flame propagation properties of the specific vapors or gases.

c. Volume of the vessel.

d. Pressure at which the vent device activates.

e. Maximum pressure that can be tolerated during a vented explosion incident.

It should also be noted that the peak pressure reached during a vented explosion is usually higher, sometimes much higher, than the pressure at which the vent device activates.

Design of explosion relief systems should follow recognized guidelines such as those contained in NFPA 68 [1]. Simplified rules-of-thumb should not be used as these can lead to inadequate designs. Where the operating conditions of the vessel to be protected are outside the range over which the design procedure applies, explosion vent designs should be based on specific test data, or an alternate means of explosion protection should be used.

Note: Numbers in brackets correspond to references listed in 3.21.

Some alternate means of explosion protection are described in NFPA 69 [2], including explosion containment, explosion suppression, oxidant concentration reduction, and so forth.

Explosion relief systems, explosion containment, and explosion suppression should not be used for cases where detonation is considered a credible risk. In such cases, the explosion hazard should be mitigated by preventing the formation of mixtures that could detonate.

Explosion prevention measures, such as inert gas purging, in conjunction with suitable administrative controls can be considered in lieu of explosion relief systems for equipment in which internal explosions are possible only as a result of air contamination during start-up or shutdown activities.

3.13 Chemical Reaction

The methodology for determining the appropriate size of an emergency vent system for chemical reactions was established by DIERS (Design Institute for Emergency Relief Systems) [3], [4], [5], [6], [7].

The DIERS methodology is based on the following:

a. Defining the design basis upset conditions for the reaction system.

b. Characterizing the systems through bench scale tests simulating the design basis upset conditions.

c. Using vent sizing formula which account for two phase gas/liquid vent flow.

The design basis upset conditions are process specific, but generally include one or more of the following:

a. External fire.

b. Loss of mixing.

c. Loss of cooling.

d. Mischarge of reagents.

Reaction rates are rarely known; therefore, bench scale tests simulating the design basis upset condition are usually required. There are a number of test apparati available for this purpose. With the information obtained from the bench scale tests, the system can be characterized by one of the following terms:

a. Tempered: Tempered systems are those in which the unwanted reaction produces condensable products and whose rate of temperature rise is tempered by liquid boiling at system pressure. Typically, tempered systems are liquid phase reactions in which a reactant (or solvent) is a major portion of the reactor contents.

b. Gassy: Gassy systems are those in which the unwanted reaction produces noncondensable products and whose rate of temperature rise is not tempered by boiling liquid. Gassy systems can be either liquid-phase decompositions or vapor-phase reactions.

c. Hybrid: Hybrid systems are those whose rate of temperature rise due to an unwanted reaction may be tempered by liquid boiling at system pressure, but can also give rise to the generation of noncondensable gas.

Following characterization of the system, the appropriate vent sizing formula can be selected. An excellent discussion of these procedures is contained in Grolmes et al [3]. However, the reader should be cautioned that this is an area with rapidly changing technology, and the most current technology should be used, if available.

If the bench scale simulations indicate the potential for an explosion, the considerations in 3.12 should be applied. It may also be prudent to consider housing the reactor in a specially constructed bay to handle potentially explosive reactions or to increase the equipment design conditions to contain maximum expected temperature and pressure.

Where feasible, a pressure relief device should be used to control overpressure. Where this is infeasible, other design strategies may be employed to control equipment over-stressing. These strategies may include using safety systems such as automatic shutdown systems, inhibitor injection, quench, de-inventorying, alternative power supplies, and depressuring. When this approach is taken, the reliability of the protective system(s) should be addressed in a formal risk analysis. This analysis is outside the scope of this document.

Other forms of reactions that generate heat (dilution of strong acids) should also be evaluated.

3.14 Hydraulic Expansion

3.14.1 CAUSES

Hydraulic expansion is the increase in liquid volume caused by an increase in temperature (see Table 3). It can result from several causes, the most common of which are the following:

a. Piping or vessels are blocked-in while they are filled with cold liquid and are subsequently heated by heat tracing, coils, ambient heat gain, or fire.

b. An exchanger is blocked-in on the cold side with flow in the hot side.

c. Piping or vessels are blocked in while they are filled with liquid at near-ambient temperatures and are heated by direct solar radiation.

In certain installations, such as cooling circuits, the processing scheme, equipment arrangements and methods, and operation procedures make feasible the elimination of the hydraulic expansion-relieving device, which might normally be required on the cooler fluid side of a shell-and-tube exchanger. Typical of such conditions would be multiple-shell units with at least one cold-fluid block valve of the locked-open design on each shell, and a single-shell unit in a given service where the shell can reasonably be expected to remain in service, except on shutdown. In this instance, closing the cold-fluid block valves on the exchanger unit should be controlled by administrative procedures and possibly the addition of signs stipulating the proper venting and draining procedures when shutting down and blocking in. Such cases are acceptable and do not compromise the safety of personnel or equipment, but the designer is cautioned to review each case carefully before deciding that a relieving device based on hydraulic expansion is not warranted.

Table 3—Typical Values of Cubical Expansion Coefficient for Hydrocarbon Liquids and Water at 60°F

Gravity of Liquid (°API)	Value (per °F)
3-34.9	0.0004
35-50.9	0.0005
51-63.9	0.0006
64-78.9	0.0007
79-88.9	0.0008
89-93.9	0.00085
94-100 and lighter	0.0009
Water	0.0001

3.14.2 SIZING AND SET PRESSURE

The capacity requirement is not easy to determine. Since every application will be relieving liquid, the required capacity of the relieving device will be small; specifying an oversized device is, therefore, reasonable. A ¾-inch × 1-inch nominal pipe size (NPS ¾ × NPS 1) relief valve is commonly used. If there is reason to believe that this size is not adequate, the procedure in 3.14.3 can be applied. If the liquid being relieved is expected to flash or form solids while it passes through the relieving device, the procedure in 3.20.1 is recommended.

Proper selection of the set pressure for these relieving devices should include a study of the design rating of all items included in the blocked-in system. The thermal relief pressure setting should never be above the maximum pressure permitted by the weakest component in the system being protected. However, the pressure-relieving device should be set high enough to open only under hydraulic-expansion conditions. When thermal relief valves discharge into a closed system, the effects of back pressure should be considered.

3.14.3 SPECIAL CASES

Two general applications for which thermal relieving devices larger than a ¾-inch × 1-inch nominal pipe size (NPS ¾ × NPS 1) valve might be required are long pipelines of large diameter in uninsulated aboveground installations and large vessels or exchangers operating liquid-full. Long pipelines may be blocked in at or below ambient temperature, and the effect of solar radiation will raise the temperature at a calculable rate. If the total heat transfer rate and thermal-expansion coefficient for the fluid are known, a relieving capacity requirement can be calculated. See Parry [8] for additional information on thermal relief.

For liquid-full systems, expansion rates for the sizing of relief devices that protect against thermal expansion of the trapped liquids can be approximated using the following formula (Equation 1):

$$gpm = \frac{BH}{500GC} \tag{1}$$

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 design data; however, Table 3 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.

This calculation method 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 should be capable of handling the vapor generation rate. If discovery and correction before liquid boiling is expected, then vaporization need not be accounted for in sizing the pressure-relief device.

3.15 External Fire

3.15.1 GENERAL

3.15.1.1 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 4 gives recommended portions of liquid inventory for use in calculations. Portions higher than 25 feet are normally excluded. Also, vessel heads protected by support skirts with limited ventilation are normally not included when determining wetted area.

Table 4—Effects of Fire on the Wetted Surfaces of a Vessel

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 dumped to the normal level in the column bottom; 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	Maximum inventory level up to the height of 25 feet (portions of the wetted area in contact with foundations or the ground are normally excluded)	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	—

Relieving temperatures are often above the design temperature of the equipment being protected. If, however, the elevated temperature is likely to cause vessel rupture, additional protective measures should be considered (see 3.15.4). Also, where exposure to fire results in vapor generation from thermal cracking, alternate sizing methods may be appropriate.

The wetted area for spheres normally includes all area up to the maximum diameter. It may be appropriate to add a percentage of the vessel area to account for vapor generation in piping associated with the vessel under consideration.

3.15.1.2 Effect of Fire on the Unwetted Surface of a Vessel

3.15.1.2.1 Unwetted wall vessels are those in which the internal walls are exposed to a gas, vapor, or super-critical fluid, 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.

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.

As in the case with wetted surfaces, relieving temperatures are often above the design temperature of the equipment being protected. If, however, the elevated temperature is likely to cause vessel rupture, additional protective measures should be considered (See 3.15.4 and 3.15.5).

3.15.1.2.2 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 1 and 2 indicate how quickly an unwetted bare vessel wall might be heated to rupture conditions. Figure 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 1,100°F and about 17 minutes to reach 1,300°F when the plate is exposed to an open fire.

Figure 2 shows the effect of overheating ASTM A 515, Grade 70 steel [9]. 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 1,100°F and about 2 ½ minutes at 1,300°F.

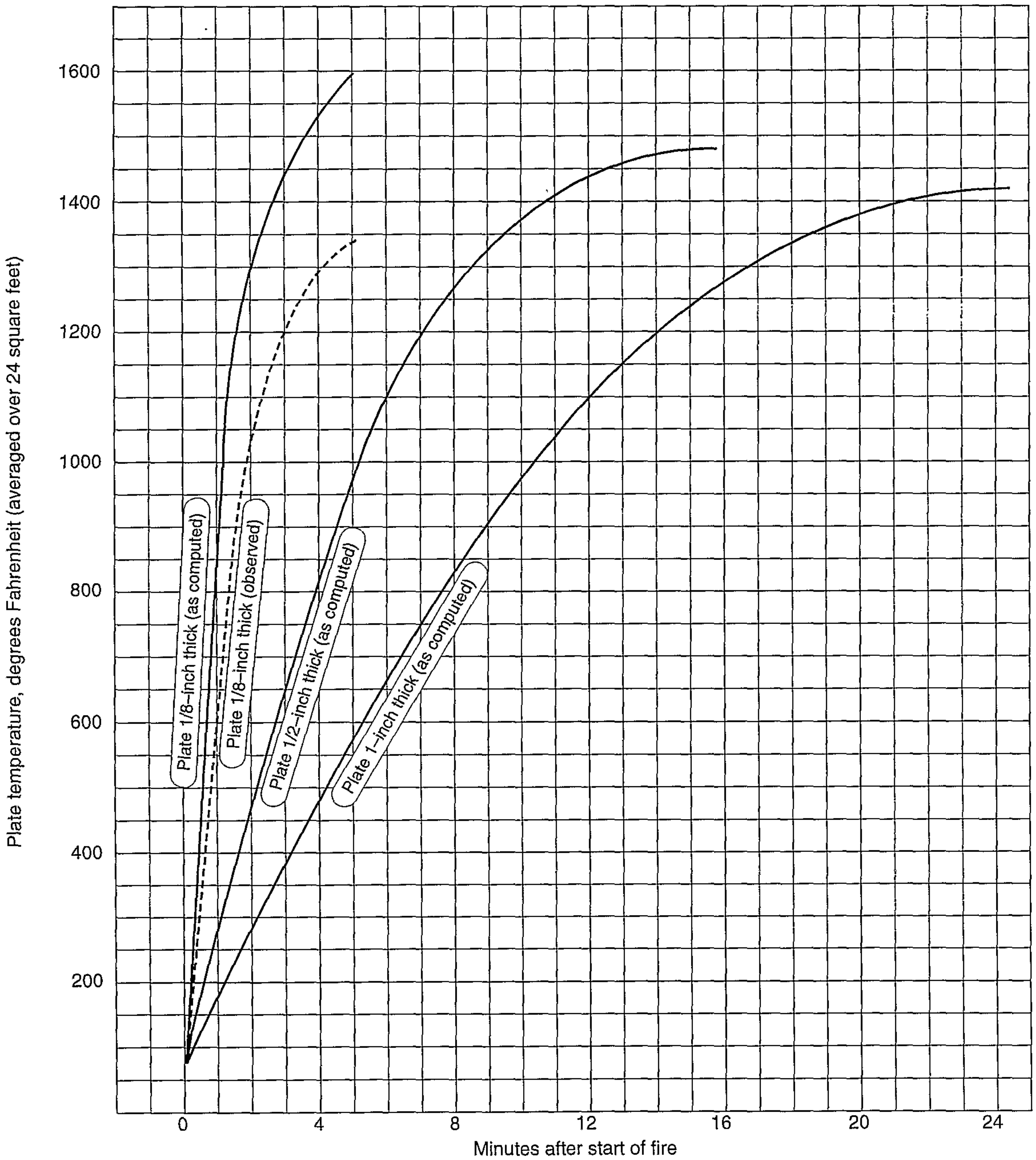


Figure 1—Average Rate of Heating Steel Plates Exposed to Open Gasoline Fire on One Side

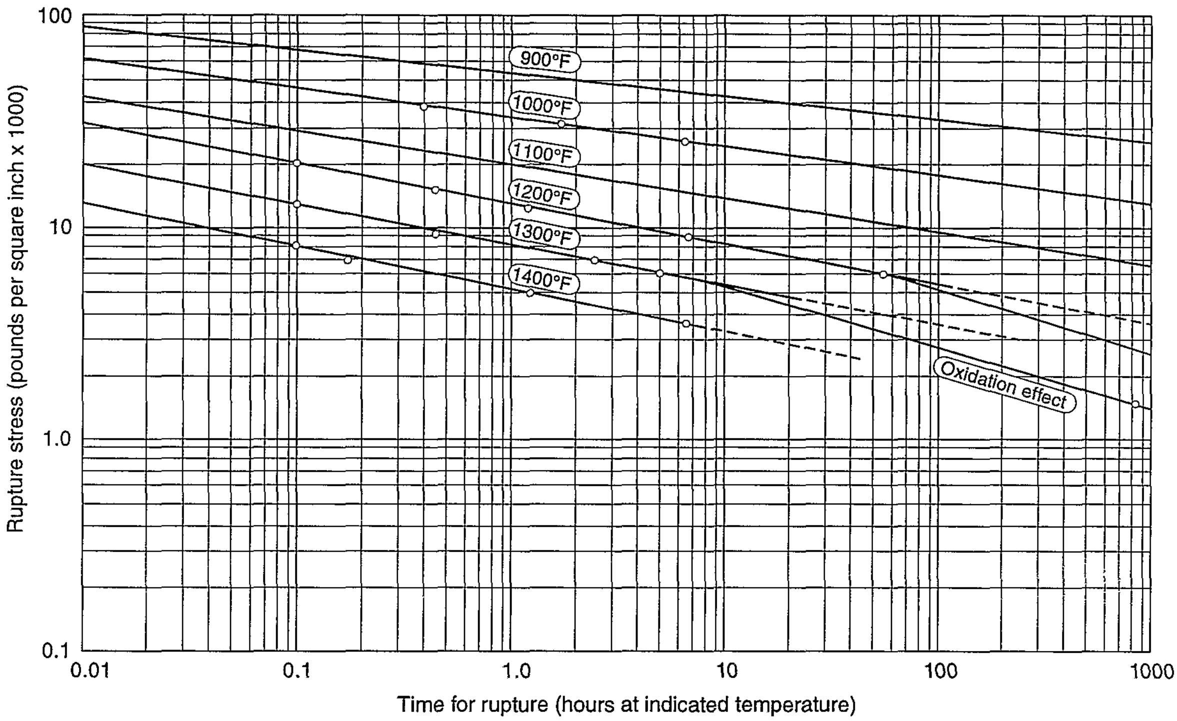


Figure 2—Effect of Overheating Steel (ASTM A 515, Grade 79)

3.15.2 SIZING

3.15.2.1 Heat Absorption Equations

3.15.2.1.1 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 (Equations 2 and 3) are used to evaluate these conditions where there are prompt firefighting efforts and drainage of flammable materials away from the vessels.

Adequate drainage is necessary to control the spread of major spills from one area to another and to control surface drainage and refinery waste water. This can be accomplished by the strategic use of sewers and trenches with adequate capacity and/or by using the natural slope of the land.

$$q = 21{,}000FA^{-0.18} \tag{2}$$

$$Q = 21{,}000FA^{0.82} \tag{3}$$

Where adequate drainage and firefighting equipment do not exist, Equation 4 should be used [10].

$$Q = 34{,}500FA^{0.82} \tag{4}$$

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 5.)

A = total wetted surface, in square feet (see 3.15.1.1 and Table 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.)

The heat absorption equations listed in the following are for process vessels and pressurized storage of liquefied gases. For other storage, whether in pressure vessels or vessels and tanks with a design pressure of 15 pounds per square inch gauge or lower. See API Standard 2000 for recommended heat absorption rates due to external fire exposure.

3.15.2.1.2 See 3.15.1.2 for a discussion of the effect of fire on the unwetted surface of a vessel.

The discharge areas for pressure relief devices on vessels containing super-critical fluids, gases, or vapors exposed to open fires can be estimated using Equation 5. In the use of this equation, no credit has been taken for insulation. Credit for insulation may be taken per Table 5.

$$A = \frac{F'A'}{\sqrt{P_1}} \tag{5}$$

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.

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 (See Equation 6).

$$F' = \frac{0.1406}{CK_d}\left(\frac{(Tw - T_1)^{1.25}}{T_1^{0.6506}}\right) \quad (6)$$

$$C = 520\sqrt{k\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}} \quad (7a)$$

Where:

k = Cp/Cv, the specific heat ratio of gas or vapor (See API Recommended Practice 520, Part I)

K_d = coefficient of discharge (obtainable from the valve manufacturer). K_d is equal to 0.975 for sizing relief valves (See API Recommended Practice 520, Part 1).

T_w = vessel wall temperature, in degrees Rankine.

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

$$T_1 = \left(\frac{P_1}{P_n}\right)T_n \quad (7b)$$

Where:

P_n = normal operating gas pressure, in pounds per square inch absolute.

T_n = normal operating gas temperature, in degrees Rankine.

The recommended maximum vessel wall temperature for the usual carbon steel plate materials is 1,100°F. Where vessels are fabricated from alloy materials, the value for T_w should be changed to a more appropriate recommended maximum.

The relief load can be calculated directly, in pounds per hour, by rearranging Equation 2 from API Recommended Practice 520, Part I, and substituting Equations 5 and 6, which results in Equation as follows:

$$W = 0.1406\sqrt{MP_1}\left(\frac{A'(T_w - T_1)^{1.25}}{T_1^{1.1506}}\right) \quad (8)$$

Where:

M = molecular weight of the gas.

Z and K_b in Equation 2 of API Recommended Practice 520, Part I are assumed equal to 1.

The derivations of Equations 5, 6, and 8 [11] are based on the physical properties of air and the perfect gas laws. The derivations assume 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.

Table 5—Environment Factor

Type of Equipment	Factor F[a]
Bare vessel	1.0[e]
Insulated vessel[b] (With insulation conductance values for fire exposure conditions as follows 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[e]
Depressurizing and emptying facilities[d]	1.0[e]
Earth-covered storage	0.03
Below-grade storage	0.00

Note:
[a]These are suggested values for the conditions assumed in 3.15.2. 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 3.15.4 and 3.15.5.
[b]Insulation should resist dislodgment by firehose streams (3.15.5.2). For the examples, a temperature difference of 1,600°F was used. These conductance values are computed from Equation 9 and are based upon insulation having thermal conductivity of 4 BTU-in/hr-ft^2-°F at 1,000°F and correspond to various thicknesses of insulation between 1 inch and 12 inches. See Equation 9 to determine *F*.
[c]See 3.15.4.2.
[d]See 3.15.4.3.
[e]The environment factor, *F*, in Equations 3 and 4 does not apply to noninsulated vessels. The environment factor should be replaced by 1.0 when calculating heat input to noninsulated vessels.

3.15.2.2 More Rigorous Calculations

Where the preceding assumptions in 3.15.2.1 are not appropriate, more rigorous methods of calculations 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 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. 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 [12, 13]. These procedures require successive iteration.

3.15.3 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 3.15.2 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, providing 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 should be made to obtain a realistic relief flow rate for the relief device. These assumptions may include the following:

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

3.15.3.1 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 pressure relief device 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, 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 13 gives an example of a time-dependent model used to calculate relief requirements for a vessel that is exposed to fire and that contains fluids near or above the critical range.

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.

3.15.3.2 Liquid

The hydraulic expansion formula given in 3.14.3 may be used to complete 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 during sizing and selecting of the pressure relief device. However, the impact of two-phase flow should be considered for the design of the piping system, liquid knockout facilities, and metallurgy (for example, potential for brittle fracture). For some vessels, particularly overfilled steam drums or polymerization reactors, the interim relieving contingency may be the factor that would determine the size of the relieving device. (See 3.15.3.3 for information about mixed-phase flow.)

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

3.15.3.3 Mixed Phase

As stated in 3.15.3.2, 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 concluded an intensive research program to develop methods for the design of emergency relief systems to handle runaway reactions. The interested reader can obtain more information on this subject from Reference [4] and [28].

3.15.4 PROTECTIVE MEASURES EXCLUDING INSULATION

3.15.4.1 General

The determination may be made that a pressure relief valve will not provide sufficient protection for an unwetted wall vessel or a vessel containing high boiling-point liquid. Where a pressure relief valve alone is not adequate, additional protective measures should be considered, such as water sprays (see 3.15.4.2), depressuring (see 3.15.4.3), fireproofing, earth-covered storage, and diversion walls.

Where local jurisdiction permits, it may be appropriate to utilize these protective measures as an alternative to relief devices sized for the fire case under the following circumstances:

a. Vessel contains vapor only or a high boiling-point liquid.
b. An engineering analysis indicates that additional protection provided by the relief device serves little value in reducing the likelihood of vessel rupture (in other words, a thin wall carbon steel vessel).

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 reaction and initiation of firefighting procedures before possible vessel rupture. Operator action may include depressuring, using water sprays, employing firewater monitors, and isolating the source of fuel.

3.15.4.2 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 5) is recommended; however, as stated previously, properly applied water can be very effective.

3.15.4.3 Depressuring Systems

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

a. Manual controls near the vessel may be inaccessible during a fire.
b. Automatic controls could fail in a direction that would prevent depressuring (for example, valves that fail closed).
c. Early initiation of depressuring is desirable to limit vessel stress to acceptable levels commensurate with the vessel wall temperature that may result from a fire.
d. Safe disposal of vented streams should be provided.
e. No credit is recommended when safety valves are being sized for fire exposure.

Further information on depressuring is provided in 3.15.6, 3.19, and 5.2.3.

3.15.4.4 Earth-Covered Storage

Covering a pressure vessel with earth is another effective method of limiting heat input.

3.15.4.5 Limiting Fire Areas With Diversion Walls

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

3.15.5 EXTERNAL INSULATION

3.15.5.1 General

To calculate the fire relief requirement, the advantage of reduced capacity to be gained by using the F factor (Table 5) permitted by the vessel's fire protection insulation system should be used in the absence of other special instructions. To take credit for insulation in reducing heat input from fire exposure, the precautions given in 3.15.5.2 through 3.15.5.4 should be observed.

3.15.5.2 Installation Considerations for External Insulation Systems

The designer should be certain that any system of insulating materials will permit the basic insulating material to function effectively at temperatures up to 1,660°F (904°C) during a fire. This period of exposure may be for up to 2 hours, depending on the adequacy of firefighting provisions, the accessibility of equipment, and the degree of skill and training of the firefighting group. This consideration is especially pertinent to newer installations using foamed or cellular plastic materials that have excellent properties at operating conditions but that (unless they were specially treated and pretested) have melted, vaporized, or otherwise been destroyed at temperatures as low as 500°F (260°C). Although jacketing and coatings may burn off or disintegrate, the insulation system should retain its shape, most of its integrity in covering the vessel, and its insulating value. (See API Publication 2218 for further guidance.)

The finished installation should ensure that fire-protection insulation will not be dislodged when it is subjected to the high-pressure water streams used for fire fighting, such as streams from hand lines or monitor nozzles, if installed. Some criteria that should be considered include the ability of the protected system to withstand direct flame impingement. Fire-insulation, or insulation which is part of a composite system, should be capable of withstanding an exposure temperature of 1,660°F (904°C) for up to two hours. Insulation system materials selection should consider equipment metallurgy while providing required jacket integrity at fire water pressures and fire temperatures. Stainless steel jacketing and banding have demonstrated satisfactory performance in fire situations. On the other hand, jacketing systems that use aluminum exclusively have not demonstrated satisfactory performance. Insulation materials which may decompose during fires should be avoided or suitably protected with layered composite systems.

3.15.5.3 Physical Properties of Insulation Systems

The value of thermal conductivity used in calculating the environmental-factor credit for insulation should be the thermal conductivity of the insulation at the mean temperature between 1,660°F (904°C) and the process temperature expected at relieving conditions (see 3.15.5.4). If reasonably possible, the variation in conductivity due to service and maintenance practices from known laboratory values should be taken into account. Where multiple-layer insulating systems consist of different materials, the physical characteristics of each material under the expected temperature conditions should be examined. Typical values of thermal conductivity for various insulating materials appear in Table 6.

3.15.5.4 Calculation of Environmental Factor for 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.

When an external insulation system is designed to limit fire heat input, it should conform to the insulation considerations of 3.15.5.2.

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 1,660°F (904°C). 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 should 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 (see Table 5) for insulation becomes the following (Equation 9):

$$F = \frac{k(1660 - T_f)}{21,000t} \tag{9}$$

Where:

k = thermal conductivity of insulation (Btu-in/hr-ft^2–°F) at mean temperature.

t = thickness of insulation, in inches.

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

Note that if the pressure relief facilities are based on an environment factor with insulation, these facilities should be rechecked for an environment factor of 1.0 if the insulation is removed.

3.15.6 VAPOR DEPRESSURING

Before the relief requirement is calculated for conditions caused by fire, 2.3.15 and 5.2.3 should be reviewed. In connection with fire protection, particularly in higher-pressure services, the designer should consider vapor depressuring facilities (see 3.19). Unless special provisions are made, a pressure relief valve cannot provide depressuring; it will merely limit the pressure rise to a given value under emergency conditions. In evaluating a vapor depressuring system for fire load, it is particularly worthwhile to consider the possibility of a fire occurring around a vessel that contains both liquid and vapor. The unwetted portion of the vessel will probably reach a temperature at which the strength of the material will be reduced. In this instance, the pressure relief valve will not protect against rupture; whereas, a vapor depressuring system could reduce the pressure to a safe level.

3.15.7 AIR-COOLED EXCHANGERS

One must recognize the problem of heat input to air-cooled coolers and condensers on fire exposure. Although the material in 3.15.7.1 through 3.15.7.4 is offered as a guide, the individual circumstances involved in each situation should be considered.

3.15.7.1 General

Air-cooled exchangers are considered separately, since, unlike shell-and-tube units, their heat-transfer surface is exposed directly to the fire. They are designed for ambient inlet air conditions, and they rapidly lose all cooling and condensing ability when they are exposed to fire-heated air. This

Table 6—Thermal Conductivity Values for Typical Thermal Insulations

Item		Data							
Material/Description		Calcium-Silicate Type I	Calcium-Silicate Type II	Mineral Fiber Mesh Covered (Class II)	Cellular Glass	Perlite, Expanded with Binders and Fiber Roof Board	Glass Fiber Felt	Lightweight Cementitious (see note)	Dense Cementitious (see note)
References (see 3.21)		[14]	[14]	[15] [16]	[17]	[18]	[19]	[20]	[20]
		Thermoconductivity (BTU, in./ft., ft, hr, °F)							
	0°F	-	-	-	0.31	-	-	3.6	12.2
	100°F	-	-	0.30	0.36	-	0.29 (75°F)	3.6	12.0
	200°F	0.45	-	0.36	0.44	0.55	-	3.6	11.8
	300°F	0.50	-	0.42	0.54	0.60	0.40	3.6	11.6
	400°F	0.55	0.66	-	0.65	0.66	-	3.6	11.5
Average	500°F	0.60	0.70	0.63	-	0.74	0.50	3.6	11.3
Temperature	600°F	0.66	0.76	-	-	0.80	-	3.6	11.2
of Insulation	700°F	0.71	0.82	-	-	0.88	0.65	3.6	11.0
	800°F	-	0.94	-	-	-	-	3.6	10.0
	900°F	-	-	-	-	-	-	3.6	10.7
	1,000°F	-	-	-	-	-	-	3.6	10.5
	1,100°F	-	-	-	-	-	-	3.6	10.3
	1,200°F	-	-	-	-	-	-	3.6	10.2
Maximum temperature for use as insulation		1,600°F	1,600°F	1,200°F	800°F	1,200°F	1,200°F	Approx. 1,600°F	Approx. 2,000°F

Note: Thermal conductivities for lightweight and dense cementitious materials are approximate.

happens without power loss so that two simultaneous occurrences need not be postulated. Assuming that the exchangers are treated as vessels (see 3.15.2), the relieving load could be calculated using the bare-tube area installed in the bundle as a basis for establishing the area term. The bare-tube area is used instead of the finned-tube area because most types of fins are destroyed within the first few minutes of exposure to fire. The bare tube area is calculated as follows:

a. Condensing without subcooling. The wetted-surface area is equal to 0.3 times the bare-tube area (based on the bottom 30 percent of the circumference being wetted by the condensate layer).

b. Condensing with subcooling. The condensing section should be treated as in Item a; for the subcooling section, the wetted-surface area is equal to the bare tube area.

c. Gas cooling. The surface area is equal to the bare tube area.

d. Liquid cooling. The wetted surface area is equal to the bare tube area.

In the wetted-surface cases, the area term, A, would be taken to an exponent of 1.0 instead of 0.82 as is done for vessels; the environmental factor for insulation, F, would be 1.0.

Air-cooled exchangers are subject to the same height limitations as other equipment. The so-called chimney effect, where hot combustion products are pulled into air-cooled exchanger plenums, will not occur when the surrounding air is heated above temperatures in the exchanger.

3.15.7.2 Condensing Service

If a fire occurs when the valve is relieving for a related cause, the load on the air cooler due to fire is additive because the system is already at its accumulated relieving pressure. Under these conditions, the following considerations of vapor inventory do not apply. The calculations in 3.15.7.1 for typical hydrocarbon processing units indicate the following: In condensing service the inventory of liquid in the tubes as a result of the bottom 30 percent of the circumference being wetted is less than 1 percent of the total hourly condensing rate. Thus, the maximum total vapor that can be generated over and above normal overhead is less than 1 percent of the normal hourly overhead.

Similarly, calculations on typical systems indicate that this additional volume of total vapor can represent from approximately 10 percent to more than 500 percent of the vapor volume present in the system at the time the fire starts. Assuming that there is a relief capability, based on loss of fans, of 70 percent of the design condensing load (see 3.6.4 and 3.6.5), the maximum pressure to which the system will be exposed is a function of the rate of vaporization of the condensed material as a result of fire. If it takes 1 hour to vaporize the condensed liquid, then the additional vapor load is a maximum of

1.5 percent of the installed capacity. However, assuming the fire heat flux recommended in 3.15.2, vaporization occurs in less than 1 minute, and the additional vapor load can approach 300 percent of the installed capacity, though only for approximately 1 minute to 2 minutes.

The assumption of some intermediate value for flux, such as one which might be valid for boiling hydrocarbons, can reduce the additional load to about 50 percent of the installed capacity, although the time of demand would be lengthened to 5 minutes or 10 minutes. A flux value of 12,000 British thermal units per hour per square foot (37.9 kilowatts per square meter) has been suggested [21] as being more reasonable than the value of 21,000 British thermal units per hour per square foot (66.2 kilowatts per square meter) determined from 3.15.2.

Sloping sections will help reduce inventory, and inventory liquid will continue to drain to the receiver as temperatures rise and the cooler stops condensing. Since air coolers totally lose their cooling ability when they are exposed to fire, the total normal process condensing load that cannot be handled by normal outlets should be the minimum relief requirement for fire exposure on air coolers. In addition, condensate inventory vaporization could increase this load between 50 percent and 300 percent.

Because of wide variations in system characteristics, the lack of any data on fire exposure heat fluxes for this specific type of equipment and the extremely large loadings of a very short duration that can be calculated on some assumed bases, no firm recommendation on handling these relief requirements is made in this publication. The total vapor input under relieving conditions seems a reasonable minimum for condensing services. The designer is cautioned to ascertain the probable behavior of the system resulting from liquid inventory vaporization and to use his engineering judgment to justify additional capacity.

Steady-state conditions are not attained in the case of fire under an air-cooled condenser. The sudden evaporation of the limited quantity of liquid remaining in the tubes causes a brief increase in pressure in the system and may result in safety relief valve discharge, but it has generally been concluded, depending somewhat on assumptions, that the quantity of vapor generated in most cases is insufficient to require increased safety relief valve capacity.

3.15.7.3 Liquid Cooling

The fire loading can become extremely large in the case of liquid coolers, becoming in some cases the dominant loading in sizing major portions of the relief header system (see 3.6.4).

3.15.7.4 Water Spray

A number of installations have had a water-spray installation designed to prevent impingement of the heat from a fire. However, because the reliability of effective water application depends on so many variable factors, adequate or uniform water coverage is not always feasible (see 3.15.4.2).

3.16 Opening Manual Valves

The following applies when a manual valve is inadvertently opened, causing pressure buildup in a vessel: the vessel should have a pressure relief valve large enough to pass a rate equal to the flow through the open valve, less credit for alternative vessel outlets that can reasonably be expected to be operational. The manual valve should be considered as passing its capacity at a full-open position with the pressure in the vessel at relieving conditions. Volumetric or heat-content equivalents may be used if the manual valve admits a liquid that flashes or a fluid that will cause vaporizing of the vessel contents. Only one inadvertently-opened manual valve need be considered at a time.

3.17 Electric Power Failure

3.17.1 GENERAL

Determination of relieving requirements resulting from power failures requires a careful plant or system analysis to evaluate what equipment is affected by the power failure and how failure of the equipment affects plant operation. Careful study and consideration should be given to the material presented in 2.3.5 and 2.3.6. Automatic standby is an excellent device for maximizing the unit's on-stream time, minimizing unit upsets, and ensuring unit production rates but the circuitry, sequences, and components involved are not yet considered sufficiently reliable to permit credit for them in establishing individual relieving requirements.

3.17.2 ANALYSIS

Electric power failure should be analyzed in the following three ways:

a. As a local power failure in which one piece of equipment is affected.
b. As an intermediate power failure in which one distribution center, one motor control center, or one bus is affected.
c. As a total power failure in which all electrically operated equipment is simultaneously affected.

The effects of a local power failure are easily evaluated when individual pieces of equipment, such as pumps, fans, and solenoid valves, are affected. Most of these effects are covered in other sections of this recommended practice. Once the upsetting cause is resolved, the relieving requirements can be determined from these sections. For example, a pump failure can cause a loss of cooling water or a loss of reflux. For the effects of loss of reflux and/or cooling water and relieving requirements, see 3.6. Loss of absorbent is covered in 3.7.

Intermediate power failure may cause more serious effects than either of the other two types of failure. Depending on the method of dividing various pumps and drivers among the electrical feeders, it is possible to lose all the fans at an air cooler at the same time that the reflux pumps are lost. This could then flood the condenser and may void any credit normally taken for the effect of natural convection of the air condenser.

Total power failure requires additional study to analyze and evaluate the combined effects of multiple equipment failures. Special consideration should be given to the effect of the simultaneous opening of relief valves in several services, particularly if the relief valves discharge into a closed header system.

3.18 Heat-Transfer Equipment Failure

3.18.1 REQUIREMENTS

The ASME Code, Section VIII, Division 1, Paragraph UG-133(d), states that "heat exchangers and similar vessels shall be protected with a relieving device of sufficient capacity to avoid overpressure in case of an internal failure." This statement defines a broad problem but also presents the following specific problems:

a. The type and extent of internal failure that can be anticipated.
b. The determination of the amount of relieving capacity required.
c. The selection of a relieving device that will react fast enough to prevent the overpressure.
d. The selection of the proper location for the device so that it senses the overpressure in time to react to it.

3.18.2 PRESSURE CONSIDERATIONS

Complete tube rupture, in which a large quantity of high pressure fluid will flow to the lower pressure exchanger side, is a remote but possible contingency. Minor leakage can seldom overpressure an exchanger during operation. Since standard hydrostatic test pressure is 150 percent of the equipment design pressure, equipment failure, in other words, loss of containment of the low-pressure side to atmosphere, is unlikely to result from a tube rupture where the low pressure side (including upstream and downstream systems) is designed for at least two-thirds of the design pressure of the high pressure side. The use of maximum possible system pressure instead of design pressure may be considered as the design pressure of the high-pressure side on a case-by-case basis where there is a substantial difference in the design and operating pressures for the high-pressure side of the exchanger.

Where the actual test pressure of the low-pressure side is less than 150 percent of the design pressure, this lower pressure should be used to determine whether overpressure protection is needed. Pressure relief for tube rupture is not required where the low-pressure exchanger side (including upstream and downstream systems) is designed at or above this two-thirds criteria.

For new installations, increasing the design pressure of the low-pressure side may reduce risk. Upstream and downstream piping and equipment systems must be thoroughly evaluated when this containment approach is taken.

3.18.3 DETERMINING THE REQUIRED RELIEF FLOW RATE

In practice, an internal failure can vary from a pinhole leak to a complete tube rupture. For the purpose of determining the required relieving flow rate, the following basis should be used:

a. The tube failure is a sharp break in one tube.
b. The tube failure is assumed to occur at the back side of the tubesheet.
c. The high-pressure fluid is assumed to flow both through the tube stub remaining in the tube sheet and through the other longer section of tube.

A simplifying assumption of two orifices may also be used in lieu of the above method since this will produce a larger relief flow rate than the above approach of a long open tube and tube stub.

In determining the relief rate, allowance should be made for any liquid that will flash to vapor either as a result of the pressure reduction or, in the case of volatile fluids being heated, because of the combined effects of pressure reduction and vaporization as the fluid is intimately contacted by the hotter material on the low-pressure side.

For liquids that do not flash when they pass through the opening, the discharge rate through the failure should be computed using incompressible flow formulas. For vapor passing through the ruptured tube opening, compressible flow theories apply. Typical steady-state equations for evaluating the flow rate through an orifice or an open tube end, for gas or non-flashing liquid service, are presented in Crane Technical Paper No. 410 [22], or other fluid flow references.

A two-phase flow method should be used in determining the flow rate through the failure for flashing liquids or two-phase fluids. The flow models developed by DIERS and others can be adapted for this purpose. Additional information concerning these models is available in references [23] and [24].

Two approaches are available for determining the required size of the relief device: (a) steady-state and (b) dynamic analysis. If a steady-state method is used, the relief device size should be based on the gas and/or liquid flow rate passing through the rupture. Capacity credit can be taken for the low-pressure side piping per the guidelines of 3.18.5. A dynamic approach simulates the pressure profile and pressure transients developed in the exchanger from the time of the rupture, and generally will include the response time of the relief device.

This type of analysis is recommended where there is a wide difference in design pressure between the two exchanger sides, especially where the low-pressure side is liquid-full and the high-pressure side contains a gas or fluid which will flash across the rupture. Modeling has shown that under these circumstances transient conditions may produce significant overpressure, even when protected by a pressure relief device [25, 26].

3.18.4 RELIEF DEVICES AND LOCATIONS

The design of piping around the exchanger and the location of the relieving device are both critical factors in protecting the exchanger. Both rupture disks and pressure relief valves should be considered.

The relieving device may need to be located either directly on the exchanger or immediately adjacent on the connected piping. This is especially important if the low-pressure side of the exchanger is liquid-full. In that case the time interval in which the shock wave is transmitted to the relieving device from the point of the tube failure will increase if the device is located remotely. This may result in higher transient overpressure on the exchangers before operation of the rupture disk or relief valve.

An analysis should be made of the time interval needed for the relieving device to open. The opening time for the device used should be verified by the manufacturer and should also be compatible with the requirements of the system.

3.18.5 INFLUENCE OF PIPING AND PROCESS CONDITIONS

To determine the influence of piping either in eliminating the need for a relieving device or in reducing relieving requirements, the configuration of the discharge piping and the contents (liquid or vapor) of the low-pressure side should be considered. Where the low-pressure side is in the vapor phase, full credit can be taken for the vapor-handling capacity of the outlet and inlet lines, provided that the inlet lines do not contain check valves or other equipment that could prevent backflow. Where the low-pressure side is liquid-full, the effective relieving capacity for which the piping system may be credited shall be based on the volumetric flow rate of the low-pressure side liquid that existed prior to the tube rupture. However, if a detailed analysis is performed, a capacity credit may be taken for low-pressure side liquid acceleration.

Where the piping system to the low-pressure side of heat-transfer equipment contains valves, their effect on the capacity of the system when overpressure occurs should be taken into account. Valves provided only for isolation may be assumed to be fully opened. In calculating relieving capacity credit for the piping system, one should consider the valves used for control purposes to be in a position equivalent to the minimum normal flow requirements of the specific process. However, this assumption cannot be made if the valve could automatically close because of the emergency situation.

3.18.6 DOUBLE-PIPE EXCHANGERS

The two types of double-pipe exchangers are those that actually use schedule pipe as the inner tube and those that use gauge tubes, usually in the heavier gauges. Units that use schedule pipe for the inner conduit or tube are no more likely to rupture the inner pipe than any other pipe in the system. Therefore, failure need not be considered a source of pressure relief requirement. Although complete tube rupture may be unlikely, weld failures may occur, especially if the two pipes are made from dissimilar metals. However, the designer is cautioned to evaluate each case carefully and to use sound engineering judgment to decide whether the particular case under study represents an exception. For example, where gauge tubes are used, the designer should determine whether or not they are equivalent to schedule pipe. Other special designs should receive similar consideration.

3.19 Vapor Depressuring

3.19.1 GENERAL

When metal temperature is increased above the specified design temperature due to fire or exothermic or runaway process reactions, the metal temperature may reach a level at which stress rupture could occur. This may be possible even though the system pressure does not exceed the maximum allowable accumulation. The use of vapor depressuring is one method of avoiding such an occurrence. If vapor depressuring is required for both fire and process reasons, the larger requirement must govern the size of the depressuring facilities.

A vapor depressuring system should have adequate capacity to permit reduction of the vessel stress to a level at which stress rupture is not of immediate concern. For sizing, this generally involves reducing the equipment pressure from initial conditions to a level equivalent to 50 percent of the vessels design pressure within approximately 15 minutes. This criteria is based on the vessel wall temperature versus stress to rupture and applies generally to vessels with wall thicknesses of approximately 1 inch (25 millimeters) or more. Vessels with thinner walls generally require a somewhat greater depressuring rate. The required depressuring rate depends on the metallurgy of the vessel, the thickness and initial temperature of the vessel wall, and the rate of heat input.

Many light hydrocarbons will chill to low temperatures as pressure is reduced. Design and depressuring conditions should consider this possibility.

Depressuring is assumed to continue for the duration of the emergency. The valves should remain operable for the duration of the emergency or should fail in a full open position.

Fireproofing of the power supply and valve actuator may be required in a fire zone.

Where fire is controlling, it may be appropriate to limit the application of vapor depressuring to facilities that operate at 250 pounds per square inch gauge (1724 kilopascals gauge) and above, where the size of the equipment and volume of the contents are significant. An alternative is to provide depressuring on all equipment that processes light hydrocarbons and set the depressured rate to achieve 100 pounds per square inch gauge (690 kilopascals) or 50 percent of the vessel design pressure, whichever is lower, in 15 minutes. The reduced operating pressure is intended to permit somewhat more rapid control in situations in which the source of fire is the leakage of flammable materials from the equipment to be depressured. The effect of heat input to process vessels is discussed in 3.15.2 and 3.19.2.

If the temperature rise is from a chemical reaction, refer to 3.13 for guidance on how to estimate the vent size and temperature rise in a reactive system.

3.19.2 VAPOR FLOWS

3.19.2.1 General

To reduce the internal pressure in equipment involved in a fire, vapor should be removed at a rate that will compensate for the following occurrences:

a. Vapor generated from liquid by heat input from the fire.
b. A change in density of the internal vapor during pressure reduction.
c. Liquid flash due to pressure reduction. (This factor applies only when a system contains liquid at or near its saturation temperature.)

The total vapor load for a system to be depressured may be expressed as the sum of the individual occurrences for all equipment involved. Thus, in terms of the loads in Items a–c, Wt = Item a + Item b + Item c and is expressed by Equation 10:

$$Wt = \sum_{i=1}^{m}(W_f t)_i + \sum_{i=1}^{m}(W_d t)_i + \sum_{i=1}^{m}(W_v t)_i \qquad (10)$$

Note: The variables for all equations in this section are defined in 3.19.3.

The combined expression Wt is used because W represents a flow rate per unit of time, and some of the noted vapor quantities are mass quantities that are not influenced by time, namely, $W_d t$ and $W_v t$ (the vapor loads from density change and liquid flash). If the system to be depressured includes more than one vessel, the vapor quantities for each vessel under all three occurrences should be calculated, especially if different molecular weights, latent heats, insulation thicknesses, and vaporization temperatures are involved. The average molecular weight and temperature for Wt—the total vapor relieved from the whole system—should be calculated from the total individual vapor molecular weights and vapor temperatures involved. The vapor loading on the depressuring system for each of the terms in Equation 10 may be estimated in 3.19.2.2 through 3.19.2.4.

3.19.2.2 Vapor From Fire Heat Input

The heat input to equipment during a fire is generally calculated in accordance with 3.15.2; however, the following modifications and limitations can be used to compute loads for a vapor depressuring and pressure-relieving system under fire conditions:

a. The extent of an assumed fire zone will be a function of the design and installation features that permit confining a fire within a given area (see 2.3.15). Although the size of the assumed fire zone may vary, experience generally indicates that a fire which can be confined to approximately 2,500 square feet (232 square meters) of plot area will not affect the design of the main relief headers in processing areas where a depressuring flow discharges into the same relief header.
b. Additional insulation or an increase in the thickness of insulation on individual vessels may also be considered as a means of reducing vapor generation resulting from exposure to fire.
c. During a fire, all feed and output streams to and from the system to be depressured and all internal heat sources within the process are assumed to have ceased. Thus, the vapor generation is only a function of the heat absorbed from the fire and the latent heat of the liquid.

To calculate the vapor load generated by fire, the fire should be assumed to be in progress throughout the depressuring period. The weight of vapor generated by the fire during the depressuring interval in a vessel, i, of the system may be determined by Equation 11:

$$(W_f t)_i = t(Q/\lambda)_i \qquad (11)$$

This calculation should be repeated for all vessels in the system if significant differences in vapor and liquid properties are involved.

3.19.2.3 Vapor From Density Change and Liquid Flash

The calculations of vapor loads caused by vapor density change and those that result from liquid flash cannot be completely separated. To determine the vapor quantities contributed by these causes, the liquid inventory and vapor volume of the system must be known. This would include all liquid and vapor in any directly connected facilities outside the fire area that cannot be isolated under fire conditions as well as all liquid and vapor contained in equipment located in the assumed fire area. Although liquid inventory and vapor volume depend on plant design, the following assumptions may be made to estimate these values:

a. The liquid inventory of fractionating columns can be estimated as the normal column bottom and draw-off tray capacity, plus a holdup per tray, equal to the weir height plus 2 inches (50 millimeters), or its design quantity, if known.
b. Normal operating levels may be used as the basis for computing the inventory of accumulators.
c. To obtain an initial, rapid approximation for standard shell-and-tube heat exchangers, one-third of the total shell volume should be assumed to be occupied by the tube bundle. For condensers and heat exchangers in vaporizing service, 80 percent of the volume involved should be assumed to be vapor. The remainder should be assumed to be liquid.
d. All liquid in heaters should be included in the estimate, regardless of temperature. If the heater is in vaporizing service, one should assume 80 percent of the tube volume past the normal point of vaporization to be vapor.

Only after the vapor and liquid volumes in the system have been determined can one estimate the respective loadings they contribute to depressuring.

One can determine the weight of vapor to be removed from a given vapor space in a vessel, i, to compensate for the reduced vapor density at the lower pressure by using Equation 12 or 13:

$$(W_d t)_i = 0.0932 V_i \left[\left(\frac{PM}{zT}\right)_a - \left(\frac{PM}{zT}\right)_b\right]_i \tag{12}$$

In metric units:

$$(W_d t)_i = 0.1205 V_i \left[\left(\frac{PM}{zT}\right)_a - \left(\frac{PM}{zT}\right)_b\right]_i \tag{13}$$

Note: V_i is assumed not to increase significantly as a result of liquid flash. This calculation should be repeated for each vessel in the system if different vapor properties are involved.

Since the calculation of the vapor load caused by liquid flash depends on liquid quantity and liquid properties in the system, the preceding data are also valid for this calculation. In systems that contain liquid at saturation conditions, the temperature of the liquid should be reduced to obtain the required reduction in pressure. To reduce pressure, one can remove vapor at a rate the vapor generation rate by heat input from the fire to compensate for the flash vaporization of some liquid. Without this allowance for flash vaporization, the required reduction in pressure is not possible. Only the liquid inventory that is at or near its saturation temperature need be considered for liquid flash. Two methods are shown for calculating the rate at which vapor must be withdrawn in order to reduce the temperature within a time interval, t, to a point at which the corresponding liquid vapor pressure will equal the desired final pressure. The first method applies only to relatively pure chemicals and to narrow-boiling-range hydrocarbons; the second covers liquids that consist of mixtures of hydrocarbons with a wider boiling range. For pure chemicals or hydrocarbons with narrow boiling ranges, the amount of liquid flash in a vessel, i, of the system may be conservatively approximated by equating the heat of the flashed vapor with the heat loss of the average liquid quantity as shown in Equation 14:

$$(W_v t)_i \lambda_i \approx \left[(W_a t) - \frac{Q_i t}{2\lambda_i} - \frac{(W_v t)_i}{2}\right] C_{pi}(T_a - T_b)_i \tag{14}$$

Rearranging Equation 14 as follows in Equation 15 yields the amount of liquid flash:

$$(W_v t)_i \approx \left[(W_a t)_i - \frac{Q_i t}{2\lambda_i}\right]\left[\frac{2c_{pi}(T_a - T_b)_i}{2\lambda_i + c_{pi}(T_a - T_b)_i}\right] \tag{15}$$

Note: $W_a t$ is used only for consistency, and W_a has no physical significance. If a more rigorous calculation is desired, the same approach may be applied in stepwise form.

Equation 15 cannot be used for liquids consisting of a mixture of hydrocarbons that have a wide boiling range because the liquid properties and composition change as the liquid is vaporized. If more accurate fluid data are not available, a series of simplified adiabatic flash calculations should be made between the initial pressure and the final pressure while neglecting the simultaneous fire effect. The simplified adiabatic flash calculation is a stepwise procedure that, by repeatedly applying Equation 16, yields a weight fraction flashed from the liquid quantity that was originally in the system during the required pressure decrease. This process assumes that the vapors flashed in each step are totally removed from the system to be depressured before the next step occurs. The correction for the fire is made in Equation 17 in which the average of the remaining liquid quantity is used (that is, the original liquid quantity in the system minus half of the quantity vaporized by fire during the total depressuring period) instead of the total liquid quantity that was originally in the system. This will compensate to some extent for neglecting the fire vaporization effects on the composition for each flash step.

To determine the approximate amount of liquid vaporized from a mixture, an equilibrium phase diagram is required and a graphical solution is employed. The procedure uses the following equation:

$$(\Delta T_n)_i = \left[\frac{\lambda_n(\Delta W_v t)_n}{(W_L t)_{n-l} - (\Delta W_v t)_n (C_p)_n}\right]_i \tag{16}$$

3.19.2.4 Example for Vapor From Density Change and Liquid Flash

Figure 3 is an example of an equilibrium phase diagram for a given liquid. This example starts at initial conditions when 0 percent of the liquid is vaporized; $T_a = 492°F$ (256°C); and

P_a = 415 pounds per square inch absolute (2861 kilopascals absolute).

The conditions after Step $n = 1$ are as follows:

Five percent of the liquid is vaporized, and 95 percent of the liquid remains; T_1 = 486°F (252°C); and P_1 = 362 pounds per square inch absolute (2496 kilopascals absolute).

The development of the diagram continues stepwise until depressuring is completed at the following conditions after Step $n = 5$ at b: Thirty percent of the liquid is vaporized, and 70 percent of the liquid remains; T_b = 461°F (238°C); and P_b = 215 pounds per square inch absolute (1,482 kilopascals absolute).

For convenience, the weight percent vaporized was assumed to be equal to the volume percent vaporized. By assuming that an incremental part of the liquid (for example, 5 percent) was vaporized during each step, the change in the liquid temperature can be computed using Equation 16. Since the remaining liquid will have a saturation temperature and pressure along the 5-percent-vaporized line of the phase diagram, and the temperature change has been determined using Equation 16, the pressure change is also known. The process is repeated in incremental steps until the pressure, P_b, at the end of the depressuring period is obtained. In Figure 3, the desired end pressure was reached when the weight fraction vaporized of the initial liquid in the vessel, i, was $X_i \approx 0.30$. Substituting this value, X_i, in Equation 17 for the last term in Equation 15 gives the estimated weight of liquid flashed as a result of the depressuring from the vessel, i, of the system during a simultaneous fire.

$$(W_v t)_i \approx \left[(W_a t)_i - \frac{Q_i t}{2\lambda_i}\right](X_i) \tag{17}$$

3.19.3 NOMENCLATURE

The variables used in the equations throughout this section are defined as follows:

a = original condition at the start of the depressuring time interval, assumed to be the saturated vapor-liquid equilibrium condition with respect to temperature and pressure.
b = depressured condition at the end of the depressuring time interval.
c_p = average specific heat of the liquid, in British thermal units per pound per degree Fahrenheit (kilojoules per kilogram per degree Celsius).
d = relating to the density change of the vapor due to pressure reduction.
f = relating to vaporization from the fire.
i = individual vessel of the system if more than one vessel is involved and requires separate consideration because of differing fluid properties, insulation for fire effect, or related factors.
L = liquid.
M = molecular weight of the vapor.
m = total number of vessels in the depressuring system.
n = depressuring step of many steps between the original condition and the depressured condition.
n–1 = depressuring step preceding Step n.
P = absolute pressure, in pounds per square inch absolute (kilopascals absolute).
Q = total heat absorption (input) to the wetted surface, in British thermal units per hour (kilojoules per hour).
T = absolute temperature of the liquid or vapor, in degrees Rankine (Kelvin).
t = depressuring time interval, in hours (usually assumed to be 0.25 hour).
V = volume available for the vapor, in cubic feet (cubic meters).
v = relating to liquid flash or vapor generated from pressure reduction.
W = vapor flow rate per unit of time, in pounds per hour (kilograms per hour).
Wt = weight of liquid or vapor, in pounds (kilograms).
X = weight fraction of the initial liquid in the system vaporized as a result of depressuring (dimensionless).
z = compressibility factor (dimensionless).
Δ = difference, for example, $\Delta T_n = T_{n-1} - T_n$.
λ = average latent heat of the liquid, in British thermal units per pound (kilojoule per kilogram).

3.20 Special Considerations for Individual Valves

Sizing procedures for pressure safety valves are covered in API Recommended Practice 520, Part I, with the exception of the circumstances covered in 3.20.1 through 3.20.3.

3.20.1 LIQUID-VAPOR MIXTURE AND SOLIDS FORMATION

A pressure relief valve handling a liquid at vapor-liquid equilibrium or a mixed-phase fluid will produce vapor due to flashing as the fluid moves through the valve. The vapor generation may reduce the effective mass flow capacity of the valve and should be taken into account. Liquid carryover may result from foaming or inadequate vapor-liquid disengaging. The designer is cautioned to investigate the effects of flow reduction or choking. Choking occurs at a point in any flowing compressible or flashing fluid where the available pressure-drop increment is totally used up by accelerating the flashing fluid. Therefore, no additional pressure difference is available to overcome the friction in the incremental line length. See API Recommended Practice 520, Part I and references 23, 24, and, 27 for further discussion on this subject.

Some fluids (for example, carbon dioxide and wet propane) may form solids when they are discharged through the relieving device. No uniformly accepted method has been established for reducing the possibility of plugging.

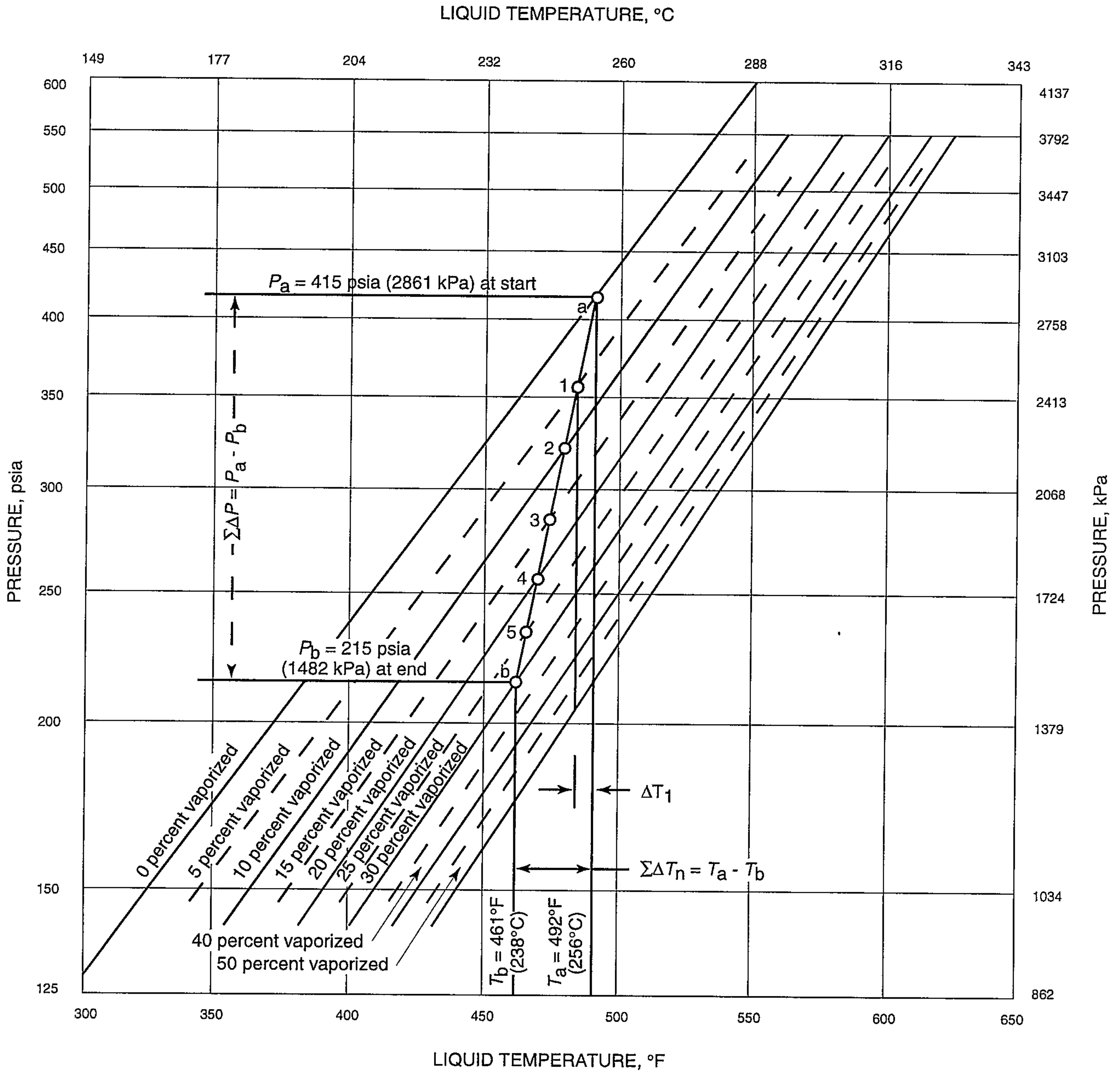


Figure 3—Equilibrium Phase Diagram for a Given Liquid

3.20.2 LOCATION OF A PRESSURE-RELIEVING DEVICE IN A NORMALLY LIQUID SYSTEM

Where valves or other devices are sized to relieve vapors caused by vapor entry or generation of vapor in a normally all-liquid system (see 3.10, 3.12, 3.13, and 3.19), care should be taken to locate the device so that it actually relieves vapor and is not required to relieve the volumetric equivalent of the vapor as liquid.

3.20.3 MULTIPLE VALVES

3.20.3.1 Basis

By virtue of its broad use as a reference or base code, the ASME Code may be considered the authority for minimum safety and relief valve requirements and performance. Other regulations may be controlling in certain areas or installations. These codes or regulations do not always concur with the ASME Code with regard to the sizing, application, setting, and use of relief devices. However, since the ASME Code is the most widely used, its stipulations are germane to any discussion of installations of multiple-pressure relief valves with or without staggered set pressures. For more detailed information, see the ASME Code, Section VIII, Division 1, Paragraph UG-125 through Paragraph UG-136, Appendix 11, and Appendix M.

Steam-generating equipment, when required to be built in accordance with Section I of the ASME Code, should be protected in accordance with that code and is not treated in this recommended practice.

3.20.3.2 Justification

The considerations that make a multiple-valve installation with staggered settings desirable include the sizing factor and valve leakage, the pressure vessel requirements, the inlet pressure characteristics of the pressure relief valve, the reactive thrust at relief, and the range of relieving rates for various contingencies.

In sizing pressure relief valves, the designer should explore possible sources of overpressure, establish the governing flow rate, and select the required orifice area. Although the governing flow may result from a single factor or a combination of circumstances, the difficulty of anticipating simultaneous occurrences tends to encourage conservative sizing (oversizing). As the size of process units increases, the calculated area required, often cannot be obtained in a single-valve body of rated, commercial design. Hence, multiple valves are needed simply to handle the capacity. Minor fluctuations in the controlled vessel pressure can approach or enter the operating range of a single safety relief valve. This creates continuous leakage that sustains itself until the pressure in the system drops low enough to enable the spring to force the valve closed. The larger the valve, the lower its lift will be to handle this small flow rate, and the greater the leakage at any given lift. Chatter and seat damage often accompany this circumstance. This problem is compounded with multiple valves uniformly set; however, multiple valves with staggered settings may provide a solution. If feasible, the lowest set valve should be the smallest one that can be selected on the basis of a reasonable relieving requirement or a reasonable portion of the total requirement. The higher set valves open only under conditions that require the combined orifice areas to handle the generated flow.

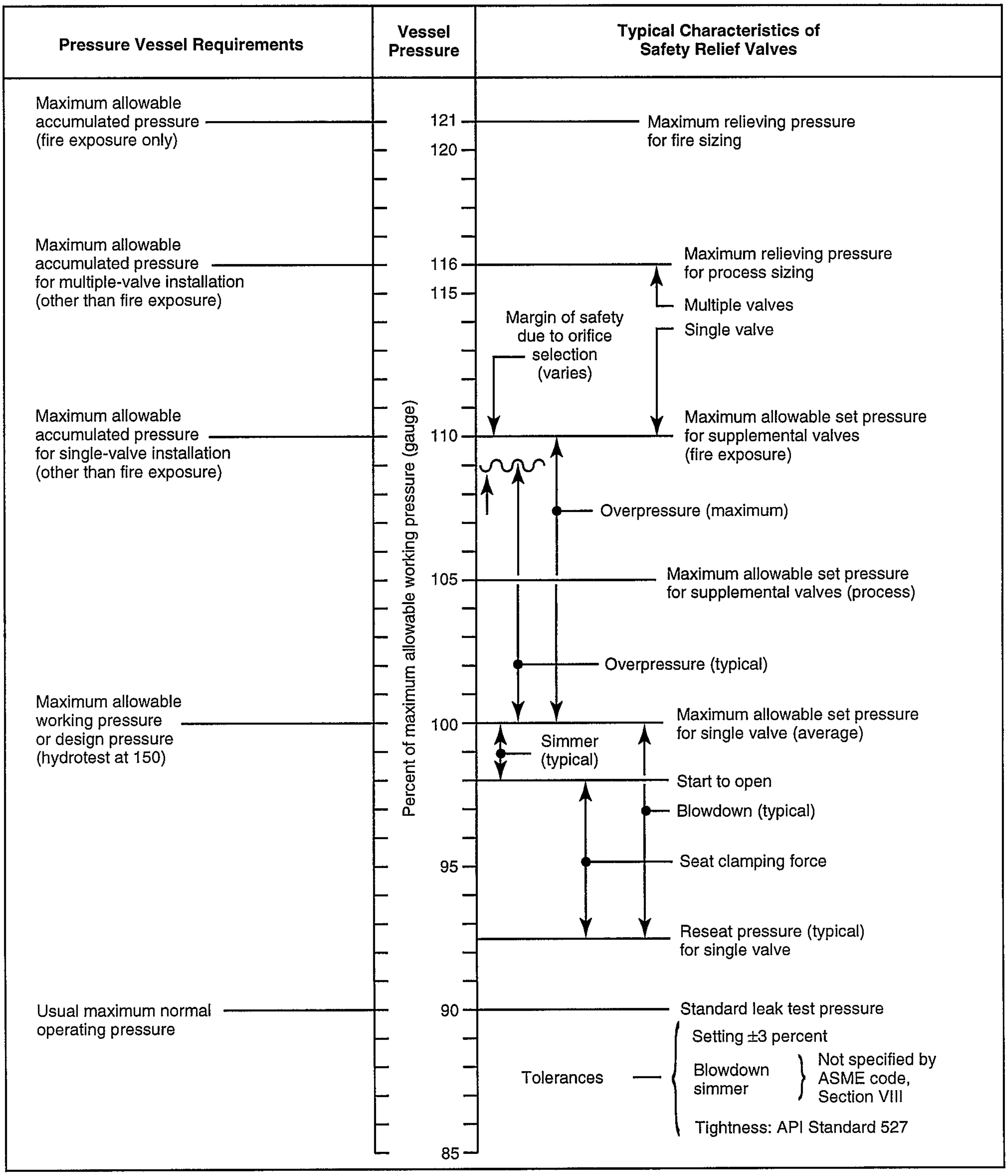
3.20.3.3 Application and Practice

The pressure vessel design requirements are dictated by the pressure conditions of the process, the operational pressure relief requirements, and the specifications of Section VIII of the ASME Code, which are illustrated in Figure 4. The most pertinent requirement, the maximum allowable working pressure, is the baseline of the chart and is the highest pressure at which the primary (first-to-open) pressure relief valve may be set to open. There is some latitude in establishing set pressures within the specified limits.

The inlet pressure characteristics of the pressure relief valve or valves are illustrated in the right-hand columns of Figure 4. The usual installation of a safety relief valve or valves on a processing pressure vessel and the special case of a supplemental valve for unexpected external heat input, such as fire conditions, are shown. In all cases, starving of the inlet to the pressure relief valve is assumed to have been avoided by limiting flow losses (caused by friction between the system being protected and the valve inlet) as recommended in API Recommended Practice 520, Part II.

The maximum probable mechanical stresses and forces resulting from the discharge flow through a pressure relief valve are given in API Recommended Practice 520, Part II. These discussions consider steady-state conditions and not the momentary, instantaneous forces that result when the valve first opens. Both impact forces and adjustment forces result as the inlet and outlet piping attempts to reach a force equilibrium for the relieving conditions of pressure and temperature. The problem is compounded if multiple valves are operating simultaneously. Since the orifices installed should be sized for the largest quantity to be relieved, the probability of on-off action (chatter) is increased for uniformly set multiple valves or a single large valve.

The use of multiple valves can frequently be accomplished easily and economically within the limitations of the ASME Code. See API Recommended Practice 520, Part I, for the allowable value of P_1, the pressure used in relief valve sizing. This specifies the set pressure staggering allowed by the ASME Code for both operating and fire contingencies. In considering multiple safety relief valve releases, the effects of back pressure should be evaluated with all valves releasing concurrently under that single contingency. The normal design approach is to consider all safety relief valves flowing



Notes:
1. The operating pressure may be any lower pressure required.
2. The set pressure and all other values related to it may be moved downward if the operating pressure permits.
3. This figure conforms with the requirements of Section VIII, Division 1, of the ASME Code.
4. The pressure conditions shown are for safety relief valves installed on a pressure vessel (vapor phase).

Figure 4—Pressure Levels

simultaneously, whether they be staggered set valves on one vessel or several safety relief valves on various vessels that should also release under this same contingency. The aggregate rate of flow determines the back pressure in the system. Any increase in back pressure in the system that results from the contingency would be considered to be built-up back pressure. Where conventional valves are employed, these pressure increases on the discharge side of any valve would need to be limited to the restriction on built-up back pressure discussed in API Recommended Practice 520, Part I. Within these limitations, the changes in back pressure caused by flow that results from one valve opening before another need not be considered as superimposed back pressure on other valves.

3.21 References

1. NFPA 68, *Guide for Venting Deflagrations*, National Fire Protection Association, Quincy, Massachusetts, 1988.

2. NFPA 69, *Explosion Prevention Systems*, National Fire Protection Association, Quincy, Massachusetts, 1986.

3. M. A. Grolmes, J. C. Leung, and H. K. Fauske, "Reactive Systems Vent Sizing Evaluations" International Symposium on Runaway Reactions, March 7–9, 1989, Cambridge, Massachusetts, pp. 451–476.

4. H. G. Fisher, et al, "Emergency Relief System Design Using DIERS Technology," American Institute of Chemical Engineers, New York, 1992.

5. H. G. Fisher, "An Overview of Emergency Relief System Design Practice," *Plant Operations Progress,* January 1991.

6. J. C. Leung, "Overpressure During Emergency Relief Venting in Bubbly and Churn Turbulent Flow," *AIChE Journal,* Volume 33, Number 6, June 1987.

7. J. C. Leung, "Simplified Vent Sizing Equations for Emergency Relief Requirements in Reactors and Storage Vessels," *AIChE Journal,* Volume 32, Number 10, October 1986.

8. Cyril F. Parry, *Relief Systems Handbook,* Institution of Chemical Engineers, Rugby, Warwickshire, U.K., 1992.

9. H. R. Wharton, *"Digest of Steels for High-Temperature Service,"* Timken Steel, 1946.

10. F. J. Heller, "Safety Relief Valve Sizing: API Versus CGA Requirements Plus a New Concept for Tank Cars," 1983 Proceedings—Refining Department, Volume 62, American Petroleum Institute, Washington, D.C., pp. 123–135.

11. J. J. Duggan, C. H. Gilmour, and P. F. Fisher, "Requirements for Relief of Overpressure in Vessels Exposed to Fire," *Transactions of the ASME,* 1944, Volume 66, pp. 1–53.

12. I. Heitner, T. Trautmauis, and M. Morrissey, "Relieving Requirements for Gas Filled Vessels Exposed to Fire," 1983 *Proceedings—Refining Department,* Volume 62, American Petroleum Institute, Washington, D.C., pp. 112–122.

13. J. O. Francis and W. E. Shackelton, "A Calculation of Relieving Requirements in the Critical Region," *1985 Proceedings—Refining Department,* Volume 64, American Petroleum Institute, pp. 179–182.

14. ASTM C-533-85, *Standard Specification for Calcium Silicate Block and Pipe Thermal Insulation,* Reapproved 1990.

15. ASTM C-592-80, *Mineral Fiber Blanket Insulation and Blanket-Type Pipe Insulation (Metal Mesh Covered) (Industrial Type),* American Society for Testing Materials, Dallas, Texas.

16. ASTM C-612-93, *Standard Specifications for Mineral Fiber Block and Board Thermal Insulation, (Type III, Type IV, and Type V),* American Society for Testing Materials, Dallas, Texas.

17. ASTM C-552-91, *Standard Specification for Cellular Glass Thermal Insulation,* American Society for Testing Materials, Dallas, Texas.

18. ASTM C-610-95, *Molded Expanded Perlite Block and Pipe Thermal Insulation,* American Society for Testing Materials, Dallas, Texas.

19. ASTM C-1086-90a, *Glass Fiber Felt Thermal Insulation,* American Society for Testing Materials, Dallas, Texas.

20. T. Z. Harmathy and L. W. Allen, "Thermal Properties of Selected Masonry Unit Concretes, Title Number 70-15," *ACI Journal,* February 1973.

21. P. P. M. Brown and D. W. France, "How to Protect Air Cooled Exchangers from Overpressure," *Hydrocarbon Processing,* August 1975.

22. Crane Engineering Division Technical Paper 410, "Flow of Fluids Through Valves, Fittings, and Pipe."

23. J. C. Leung, "Size Safety Relief Valves for Flashing Liquids," *Chemical Engineering Progress,* February, 1992.

24. J. C. Leung, and F. N. Nazario, "Two-Phase Flashing Flow Methods and Comparison," *Journal of Loss Prevention Process Industries,* Volume 3, Number 2, 1990.

25. D. W. Fowler, T. R. Herndon, and R. C. Wahrmund, "An Analysis of Potential Overpressure of Heat Exchanger Shell Due to a Ruptured Tube," Paper presented at the American Society of Mechanical Engineers Petroleum Division Conference, September 22–25, 1968.

26. V. H. Sumaria, J. A. Rovnak, I. Heitner, and R. J. Herbert, "Model to Predict Transient Consequences of a Heat Exchanger Tube Rupture," *Proceedings—Refining Department,* 1976, Volume 55, American Petroleum Institute, Washington, D.C., pp. 631–654.

27. L. L. Simpson, "Estimate Two-Phase Flow in Safety Devices," *Chemical Engineering,* August 1991.

28. H. K. Fauske, et al, "Emergency Relief Vent Sizing for Fire Emergencies Involving Liquid Filled Atmospheric Storage Vessels," *Plant Operations Progress,* Volume 5, Number 4, October 1986.

SECTION 4—SELECTION OF DISPOSAL SYSTEMS

4.1 General

The selection of a disposal method is subject to many factors that may be specific to a particular location or an individual unit. The purpose of a disposal system is to conduct the relieved fluid to a location where it may be safely discharged. Disposal systems generally consist of piping and vessels. All components should be suitable in size, pressure rating, and material for the service conditions intended. This section outlines the general principles and design approach for determining the most suitable type of disposal system.

4.2 Fluid Properties That Influence Design

4.2.1 PHYSICAL AND CHEMICAL PROPERTIES

The flash point, flammable limits, and ignition temperatures of certain flammable liquids, gases, and solids are listed in NFPA 325M [1] (see also 6.1). Additional data on the flammability characteristics of pure compounds and mixtures, in both air and atmospheres that contain varying amounts of inert gases and water vapor, are found in the U.S. Bureau of Mines Bulletin 627 [2]. This reference also provides information on explosive limits and presents a method for calculating the flammability characteristics of mixtures, based on the properties of pure compounds.

Note: Numbers in brackets correspond to references in 4.7.

Consideration should be given to any phase change—either vaporization of liquid or condensation of vapor—that occurs in the fluid when the pressure is reduced or as a result of cooling. With autorefrigeration, vaporization of volatile liquids may be incomplete unless facilities are provided to add the necessary heat for vaporization.

Caution should be exercised to avoid mixing chemicals that may react in flare headers. Routing reactive materials to a flare header has caused high flare pressures that have resulted in flare header ruptures. Materials that react violently when mixed with water (such as, alkyls, sodium, potassium, and silanes) should be routed to a segregated header which does not contain water.

4.2.2 PHYSIOLOGICAL AND NUISANCE PROPERTIES

The physiological and nuisance properties of material released from pressure-relieving and depressuring systems should be studied to establish the proper type of disposal system.

4.2.3 RECOVERY VALUE

The monetary value of refinery wastes may warrant special means of collection for return to the process, as is the case with costly solvents, for example. An economic engineering evaluation can determine whether the recovery value of the material justifies the installation of a recovery system. If a recovery system is justified, refer to 5.5 for guidance. To avoid loss of valuable process material, the pressure-relieving device should be set sufficiently above the normal operating pressure to give a reliable margin of differential pressure (see Figure 4).

4.3 Atmospheric Discharge

4.3.1 GENERAL

In many situations, pressure-relief vapor streams may be safely discharged directly to the atmosphere if environmental regulations permit such discharges. This has been demonstrated by many years of safe operation with atmospheric releases from properly installed vapor pressure relief valves. Technical work sponsored by API [3] has also shown that within the normal operational range of conventional safety relief devices, well-defined flammable zones can be predicted for most such vapor releases. With proper recognition of the appropriate design parameters, vapor releases to the atmosphere can provide for the highest degree of safety. Atmospheric discharge eliminates the significant problems associated with analysis of system loads, proper sizing of piping, mechanical design criteria, and considerations of the back pressure on safety relief valves where closed release systems are used. Where feasible, this arrangement offers significant advantages over alternative methods of disposal because of its inherent simplicity, dependability, and economy. The decision to discharge hydrocarbons or other flammable or hazardous vapors to the atmosphere requires careful attention to ensure that disposal can be accomplished without creating a potential hazard or causing other problems, such as the formation of flammable mixtures at grade level or on elevated structures, exposure of personnel to toxic vapors or corrosive chemicals, ignition of relief streams at the point of emission, excessive noise levels, and air pollution.

4.3.2 FORMATION OF FLAMMABLE MIXTURES

4.3.2.1 General

To evaluate the potential hazards of flammable mixtures that result from atmospheric discharge of hydrocarbons, the physical state of the released material is of primary importance; for example, the behavior of a vapor emission is entirely different from that of a liquid release. Between these two extremes are situations involving liquid-vapor mixtures in which mists or sprays are formed. Vapors, mists, and liquids each introduce special considerations in analyzing the risk associated with atmospheric relief.

4.3.2.2 Vapor Emission

When hydrocarbon relief streams comprised entirely of vapors are discharged to the atmosphere, mixtures in the flammable range will unavoidably occur downstream of the outlet as the vapor mixes with air. Under most circumstances in which individual safety relief valves discharge vertically upward through their own stacks, this flammable zone will be confined to a rather limited definable pattern at elevations above the level of release. At exit velocities from the safety relief valve stack, the jet momentum forces of release will usually be dominant [3]. Under these conditions, the air entrainment rate will be very high, and the released gases will then be diluted to below the lower flammable limit before the release passes out of the jet-dominated portion if the Reynolds number meets the criterion of Equation 18:

$$Re > (1.54 \times 10^4)\left(\frac{\rho_j}{\rho_\infty}\right) \tag{18}$$

Where:

Re = Reynolds number, calculated at the vent outlet.
ρ_j = density of the gas at the vent outlet.
ρ_∞ = density of the air.

Note that Equation 18 may not be valid where jet velocity is less than 40 feet per second or when the jet-to-wind velocity ratio is below 10.

On the other hand, if the release is at too low a velocity and has too low a Reynolds number, jet entrainment of air will be limited, and the released material will be wind dominated. Principles of atmospheric dispersion will then determine the dilution rate and the distance flammable conditions can occur. Under these conditions, flammable mixtures can possibly occur at grade or at distant ignition sources. A complete evaluation requires consideration of the following:

a. The velocity and temperature of the exit gas.
b. The molecular weight and quantity of the exit gas.
c. The prevailing meteorological conditions, especially any adverse conditions peculiar to the site.
d. The local topography and the presence of nearby structures.
e. The elevation at which the emission enters the atmosphere.

Previous technical investigations [4] have demonstrated the rapid dispersion caused by the turbulent mixing that results from dissipation of energy in a high-velocity gas jet. For a situation in which a safety relief valve is flowing at or close to full capacity, discharge velocities through independent atmospheric stacks usually exceed 500 feet per second (150 meters per second). The studies on the discharge of jets into still air indicate that gases with velocities of 500 feet per second (150 meters per second) or more have sufficient energy in the jet to cause turbulent mixing with air and effect dilution in accordance with Equation 19:

$$W/W_o = 0.264(y/D) \tag{19}$$

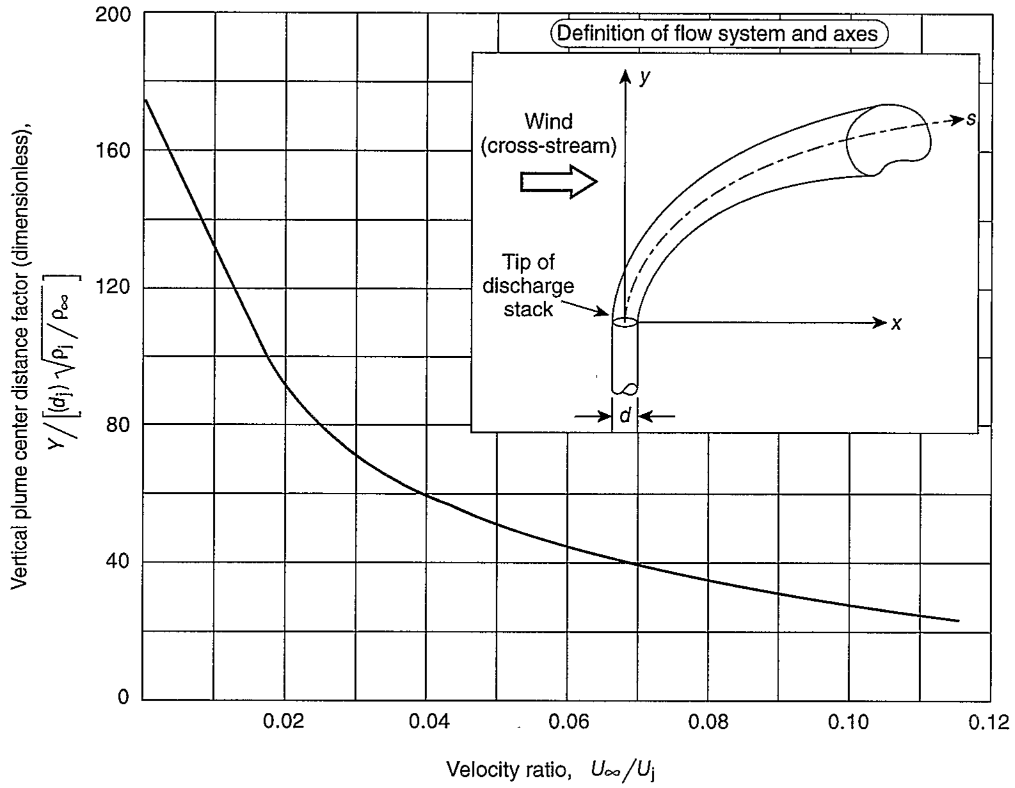
Where:

W = weight flow rate of the vapor-air mixture at distance y from the end of the tail pipe.
W_o = weight flow rate of the relief device discharge, in the same units as W.
y = distance along the tail pipe axis at which W is calculated.
D = tail pipe diameter, in the same units as y.

Equation 19 indicates that the distance, y, from the exit point at which typical hydrocarbon relief streams are diluted to their lower flammable limit (approximately 3 weight percent) occurs approximately 120 diameters from the end of the discharge pipe, measured along the axis. In essence, when hydrocarbon vapors are diluted with air to approximately 3 weight percent, the concentration of the resultant mixture will be at or below the lower flammable limit. This value actually varies from 3.0 percent for methane to 3.6 percent for hexane. When figured on a volumetric basis, which is more commonly used than weight percent to express limits of flammability, these percentages are equivalent to 5.3 and 1.2, respectively. For materials that do not have combustion characteristics similar to light hydrocarbons, the extent of a flammable mixture may differ considerably from 120 diameters. Based on these data, it can be concluded that where discharge velocities are achieved, the hazard of flammable concentrations below the level of the discharge point is negligible. This confirms the many years of experience with vapor releases from safety relief valves discharging directly to the atmosphere without accumulating flammable concentrations.

Through the years, skepticism arose about the validity of this past investigation, even though experience had indicated that large flammable volumes were not created by safety valves releasing vapor directly to the air. There was concern because any system designed for a discharge velocity of 500 feet per second (150 meters per second) at maximum conditions would have some lower discharge velocity under other conditions.

Although a high discharge velocity is characteristic of a relief valve when it is flowing at design capacity, one cannot assume that a relief valve is flowing at full capacity. For example, even though the initial release may be at a high velocity, once a spring-loaded safety relief valve has opened, kinetic forces will be sufficient to offset the spring-closing force until the flow has been reduced to approximately 25



U_∞ = wind speed, in feet per second (meters per second).
U_j = jet exit velocity, in feet per second (meters per second).
y = vertical distance, in feet (meters).
ρ_j = fluid density inside the tip exit, in pounds per cubic foot (kilograms per cubic meter).
ρ_∞ = density of the ambient air, in pounds per cubic foot (kilograms per cubic meter).
d_j = inside diameter of the tip (jet exit diameter), in feet (meters).
$^a y$ = distance factor multiplied by $d_j(\rho_j/\rho_\infty)^{1/2}$.

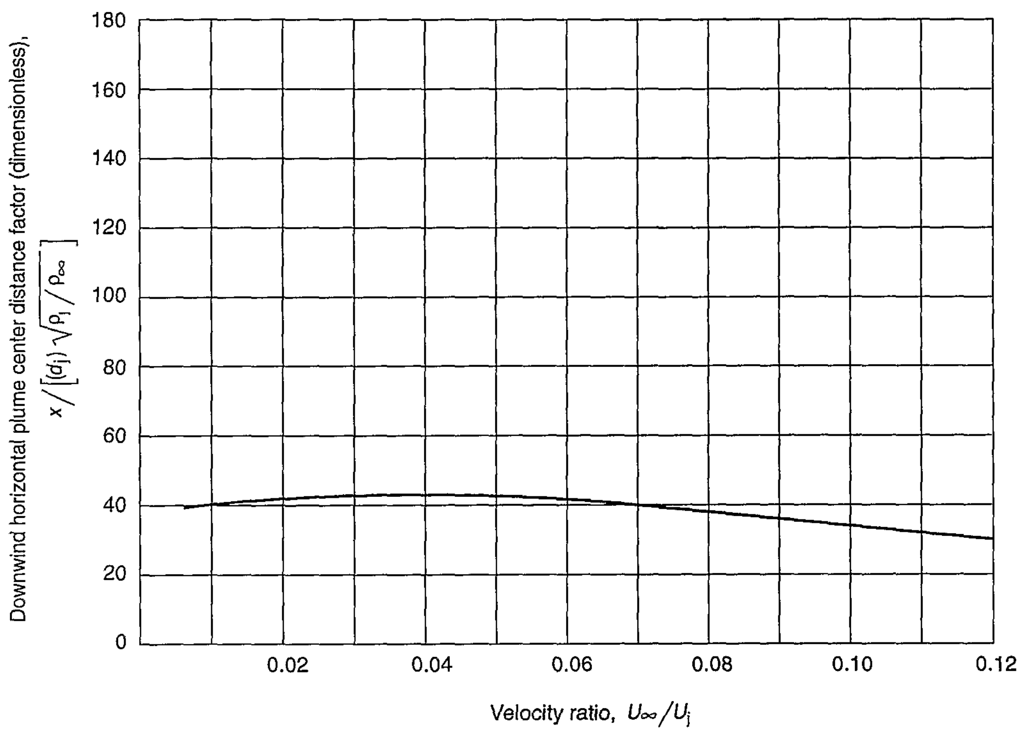
Figure 5—Maximum Downwind Vertical Distance From Jet Exit to Lean-Flammability Concentration Limit (Petroleum Gases)

percent of the valve's rated capacity. Reduced flow rates may occur as the conditions affecting relief are corrected. In many cases, overpressure may result from a minor operating upset, causing the flow rate to be appreciably less than the design capacity. The probability of these situations occurring can often be minimized by using two or more relief valves and staggering the set pressure to provide for sequential operation. Using a common vent stack for several relief valves can also result in a discharge at a relatively low velocity if only one valve should be operating.

Because of these concerns, studies were undertaken at the Battelle Institute [3] to evaluate the effects of reduced velocities of discharge at the point where the safety relief valve was about to reseat at approximately one-quarter on the valve's rated capacity. Also covered in these studies were the effects of the temperature and molecular weight of the hydrocarbon gases as they affected the zone of flammability under various ratios of exit velocity to wind velocity. These studies verified that vapors released from safety relief valves through their individual stacks were safely dispersed even when the valves were operating at only 25 percent of their full capacity, which would correspond to the reseat level of the valves. As long as the minimal value given by Equation 18 was exceeded, the release would be jet dominated and diluted outside the flammable range, within the jet pattern. For the most part, vent velocities would be greater than 100 feet per second (30 meters per second), even at the 25 percent release rate.

Other studies of the safety of tanker venting [5] had shown the same jet momentum dilution effects where release velocities exceeded 100 feet per second (30 meters per second). One would expect that only for low-set safety-relief valves, or for multiple valves manifolded into a common vent stack, would Reynolds number of the released gases be below the minimum necessary for jet momentum effects.

Figures 5, 6, and 7 demonstrate the limits of flammability vertically, horizontally, and along the main axis of the jet. The axial and vertical distance in still air is indicated to be somewhat greater than the 120 diameters indicated by previous

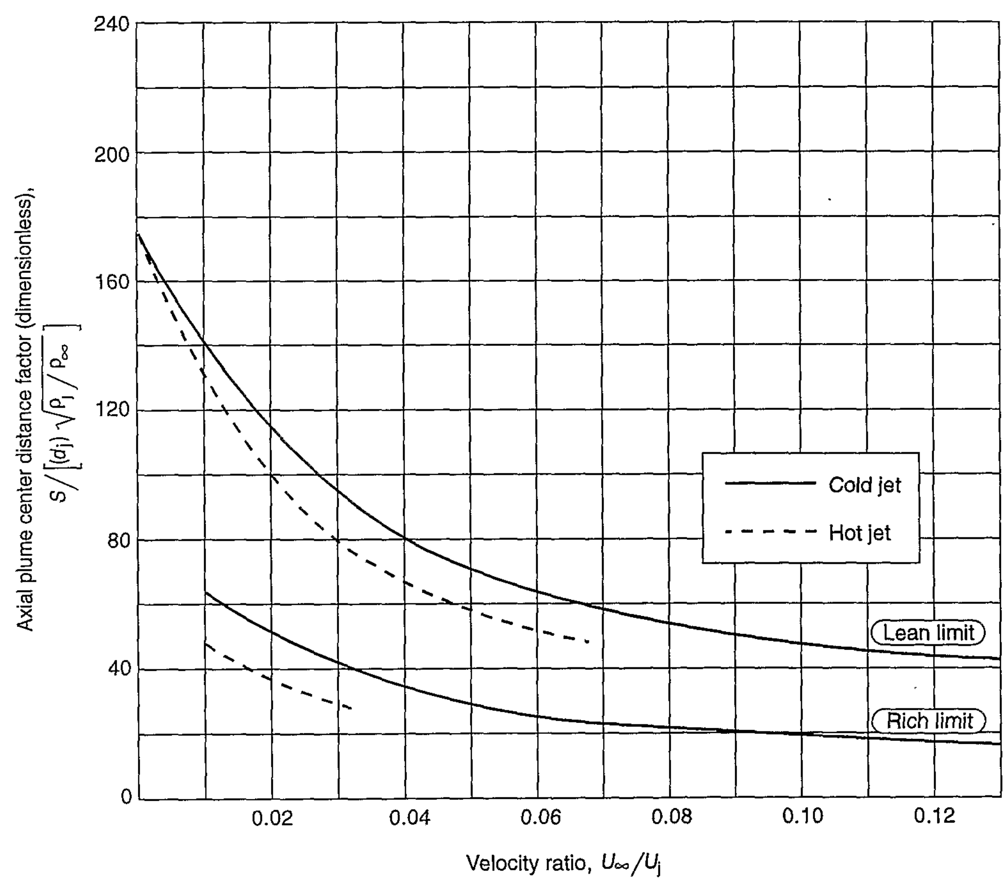


x = horizontal distance, in feet (meters).

[a]x = distance factor multiplied by $d_j(\rho_j/\rho_\infty)^{1/2}$.

Note: See Figure 5 for the nomenclature of the variables used above and for a definition of flow system and axes.

Figure 6—Maximum Downwind Horizontal Distance[a] From Jet Exit to Lean-Flammability Concentration Limit (Petroleum Gases)



S = distance along jet center from tip apex. in feet (meters).

[a]S = distance factor multiplied by $d_j(\rho_j/\rho_\infty)^{1/2}$.

Note: See Figure 5 for the nomenclature of the variables used above and for a definition of flow system and axes.

Figure 7—Axial Distance[a] to Lean- and Rich-Flammability Concentration Limits (Petroleum Gases)

still-air studies. However, the horizontal limit of the flammable envelope is shown to be essentially independent of the wind velocity and would be significantly lower than the axial distances indicated by the previous study.

The studies demonstrated the adequacy of the general industry practice of locating safety relief valve stacks that discharge to the atmosphere at least 50 feet (15 meters) horizontally from any structures or equipment running to a higher elevation than the discharge point. In most cases, this would be adequate to prevent flammable vapors from reaching the higher structures. With these jet momentum releases, there should also be no concern about large clouds of flammable vapors or flammable conditions existing at levels below the release level of the stack. These recent studies have generally verified long-standing experience relative to the safety of vapor releases vertically to the atmosphere from atmospheric safety relief valve discharge stacks.

4.3.2.3 Mist Emission

Mists, as referred to in this recommended practice, result from condensation following vapor relief. Fine sprays associated with relief streams that contain liquids are considered in 4.3.2.4. Condensed mists are finely divided; the diameter of most drops is less than 10 micrometers, with few larger than 20 micrometers. Mechanical sprays do not usually contain many drops less than 100 micrometers in diameter.

Whether vapors will condense in appreciable quantities when they are released to the atmosphere will depend on the stream composition, atmospheric temperature, and exit velocity. The assumption is frequently made that if the lowest anticipated atmospheric temperature is below the dew point of a released hydrocarbon, significant condensation will occur. This approach ignores two important effects associated with the release of vapors. As vapors are depressured across the safety relief valve, they are superheated, and the tendency for immediate condensation to occur is minimized in the highly rich zone at the point of emission. More important is the combined dilution effect of air and light components normally present in discharges from safety relief valves. Rapid dilution tends to lower the dew point of individual components to a point below the ambient temperature.

Loudon [6] gives a method of calculating whether condensation of a discharge from a safety relief valve has occurred. These calculations indicate that most emissions do not condense, regardless of molecular weight, although heavy-molecular-weight hydrocarbons may condense in the range previously noted (10 micrometers to 20 micrometers). This approach is supported by experience with refinery relief installations involving discharge to the atmosphere of vapor streams covering a wide range of conditions.

In cases in which the vapor discharges from safety relief valves condense, consideration should be given to how this condensation influences the formation of a flammable atmosphere. Combustible-liquid mists in air are capable of propagating flame when they are ignited, even though the liquid is so involatile that no appreciable amount of vapor is formed at the ambient temperature. Mists of flammable liquids can thus present a hazard even at temperatures well below the flash point. Burgoyne [7] has shown that for flammable condensed mists, the weight percent lower flammability limit and the burning velocity are the same as for the corresponding vapor. According to Saletan [8], the ignition energy required to ignite a mist in air at ambient temperatures and pressures is approximately 10 times that needed to ignite a vapor.

In cases in which calculations indicate that vapor discharges from safety relief valves may condense, coalescence may possibly produce droplets that rapidly settle to grade rather than disperse as a mist similar to vapors. The hydrocarbon partial pressure at which the calculated adiabatic air-mixing curve intersects the dew-point curve should be considered indicative of bounding the region in which coalescence seems unlikely. Although no conclusive data are now available, condensation at hydrocarbon partial pressure of 5 pounds per square inch absolute (34 kilopascals absolute) or less should be treated as finely divided mists without coalescence. In the absence of coalescence, the effect of gravity should be negligible, since the free-fall velocity of 10-micrometer hydrocarbon particles in air is approximately 0.01 foot per second (3 millimeters per second). Therefore, even with very light wind, the discharge from an elevated location will travel a considerable distance before it reaches grade.

Based on the foregoing factors pertaining to the dispersion and combustion characteristics of a mist, it can be concluded that as long as the condensate remains in finely divided form and is airborne, the mixture can be treated for flammability and dispersion characteristics as though it were completely vaporized. Because of the extremely small size of the droplets, use of the methods described in 4.3.2.2 can give an order of magnitude of the concentration at various distances from the point of emission. As noted above, the same weight percentages of hydrocarbons are necessary to make a mist flammable as would be necessary to make a vapor flammable. It can, therefore, also be concluded that as long as the minimum Reynolds number of the release is in excess of that required by Equation 18, the envelope of flammability for the mist will be within the same confined predictable limits as those for a vapor. Therefore, although condensation may create problems relative to air pollution, if only hydrocarbon-flammability considerations are involved, the risk is no higher from an area explosion potential than if the vapors had not condensed.

4.3.2.4 Liquid Emission

Unlike discharges composed of vapor or mist, which rapidly disperse when they are vented to the atmosphere at high velocity, liquid discharges settle to grade. If volatile components are present, a flammable atmosphere may result. The risk of fire or explosion may be high if appreciable quantities of liquid hydrocarbon are released to the atmosphere when the ambient temperature is at or above the flash point of the liquid. Theoretically, liquids that have a flash point above the maximum anticipated ambient temperature would not vaporize enough to create a flammable atmosphere. However, widespread spraying of oil droplets could create concern in an emergency and constitute a serious nuisance. Also, minor fires might occur if liquid came in contact with very hot lines or equipment. Therefore, all liquid relief streams should generally be disposed of by one of the methods described in 4.6.

To minimize the possibility of a release of flammable liquid, all safety relief valves that vent vapor to the atmosphere should be located so that the valve inlet connects to the vapor space of vessels or lines. In some instances, additional safeguards are warranted. For example, during unit upsets, liquid levels may increase and flood the vessels that are partially or totally filled with vapor during normal operations. The potential for such a situation can be greatly minimized by locating safety relief valves at a point in the process system where the probability of liquid occurring at the safety relief valve inlet is considered negligible because of factors related to time and the system's liquid capacity. For example, a safety relief valve located on top of a large fractionating column will present far less risk of liquid release than a valve positioned on the overhead receiver, which might flood in a matter of minutes. In other situations, high-level alarms or other instrumentation may provide a valuable safeguard against high-liquid levels reaching the safety relief valve inlet.

In summary, a rigorous analysis should be made of the various causes of overpressure on any system containing flammable liquid in which pressure relief valves that vent to the atmosphere are included in the design. All possibilities that might allow liquid to gain entrance to the pressure relief valve should be determined, and appropriate safeguards should be taken to prevent this occurrence.

4.3.3 EXPOSURE TO TOXIC VAPORS OR CORROSIVE CHEMICALS

4.3.3.1 Toxic Vapors

Although most vapor streams would be harmful to breathe at high concentrations, the majority present little or no risk to personnel when they are discharged from safety relief valves at a remote location. The average person can tolerate short-term exposure to most hydrocarbon vapors at concentration levels equivalent to or above the lower flammable limit. Facilities should be designed to avoid a flammable atmosphere.

Certain refinery streams may contain vapors that are dangerous at extremely low concentrations; for example, hydrogen sulfide vapors can cause unconsciousness within seconds following exposure to a concentration above 1000 parts per million. This is approximately one-tenth the concentration representing the lowest flammable limits of any hydrocarbon. Where toxic materials are present in a relief stream, an investigation should be made to predict the maximum downwind concentration at any location where personnel may be exposed. Special attention should be given to adjacent elevated structures that may lie within the path of the plume and will thus be subject to relatively high concentrations.

Each situation in which toxic vapors may be released to the atmosphere warrants careful analysis. Since toxicity varies greatly for different materials, the maximum concentration that can be tolerated should first be determined. Based on the length of exposure, the maximum tolerated level may vary for different locations. A higher concentration might pose less risk at locations that could be quickly and safely evacuated as opposed to those locations where personnel must remain on duty or cannot readily leave. Of further importance is the probable duration of a release. Most emergencies that cause overpressure on equipment can be controlled within 5 to 10 minutes. The duration of an emergency will vary, depending on the process and equipment involved. For example, when the source of overpressure can be eliminated by shutting down a pump or compressor, the duration of relief should be shorter than if a fractionator column were to overpressure. A period of 10 minutes to 30 minutes should be sufficient to control any emergency situation short of a catastrophe.

The actual exposure of an individual to a release is difficult to determine accurately but should be estimated. Where toxic safety releases would have a sufficiently high Reynolds number, they would meet the dispersion/dilution criteria for jet momentum releases described in 4.3.2.2. The materials in the release would be expected to be diluted 30 to 50:1 before the jet momentum effects were lost. In considering further dispersion from the end of the jet, one should take into account that the released materials have already been diluted to at least this level. From this level of mixing, calculations of ground concentrations could be evaluated based on the techniques given in 4.3.6.

4.3.3.2 Corrosive Chemicals

Certain chemicals, such as phenols, that are liquid at ambient conditions may create a serious hazard to personnel if they are discharged from safety relief valves to the atmosphere. When process systems contain such chemicals, atmospheric relief will not be safe unless valves can be installed at locations where freedom from release of such materials can be assured. Many of the same considerations discussed in 4.3.2.4 concerning avoidance of liquid releases apply to corrosive chemicals.

4.3.4 IGNITION OF A RELIEF STREAM AT THE POINT OF EMISSION

4.3.4.1 Sources of Ignition

The possibility of accidental ignition of the outflow of hydrocarbon vapors from a safety relief valve can best be analyzed in terms of the possible causes of ignition covered in 4.3.4.1.1 through 4.3.4.1.4.

4.3.4.1.1 The possible existence of outside ignition sources such as open flames, hot surfaces, and nonclassified electrical equipment installed in surrounding areas and on structures—will be known or can be anticipated. With jet momentum releases from safety relief valves, emission points can be located so that the flammable pattern evolved will not reach such sources. This becomes more difficult where wind-dominated low-velocity releases are involved, since flammable patterns could extend considerable distances from the release point. Also in these instances, the ignition potential from temporary sources, such as automotive equipment or hot-work activities, should be recognized. With normal atmospheric releases, outside ignition sources can be readily avoided by the proper location of vents. On the other hand, with low-velocity, low-momentum releases, a careful design check should be made of conditions at various emissions rates and atmospheric conditions to avoid the potential of ignition by outside sources.

4.3.4.1.2 Discharges from open atmospheric vents have been known to be ignited by lightning. Except for emergency discharges associated with power outages that may occur during thunderstorms, the probability of lightning occurring simultaneously with the opening of a relief valve is negligible. Intermittent discharges over long periods and continuous discharges (for example, from leaking relief valves) increase the probability of lightning ignition. For additional information about lightning, see the recommendations in NFPA 78.

4.3.4.1.3 For general information on electrostatics, see Eichel's article [9] and API Recommended Practice 2003. During high-velocity discharges from gas wells to the atmosphere, static discharges are developed that are sufficient to cause sparks and ignition [10]. The condensate zone in the jet of well-head gas apparently tends to produce a high level of charge, although ignition does not actually occur. Another theory relating to static ignition proposes that gas flow through a piping system during venting induces a static charge on any solid or liquid particles in the pipe stream that contact the pipe wall. As the gas reaches the sharp edges of the vent outlet, static discharges may occur either by complete electrical breakdown (spark discharge) or by partial electrical breakdown (corona discharge). There is a lack of documented information on the ignition of relief valve vapor discharges attributed to the development of electric potential at the discharge point. The experience of pipeline companies (who customarily discharge natural gas to the atmosphere at low elevations) includes gas pressures as high as 900 pounds per square inch gauge (6200 kilopascals gauge) and discharge rates as high as 650,000 pounds per hour (82 kilograms per second) from a single vent stack. The probability of ignition by static electricity is, therefore, very low because of a relatively weak charge buildup in the jet and reasonable isolation from the well-grounded vent stack.

This conclusion pertains to hydrocarbon vapor releases. Experience indicates that streams with a high hydrogen content are susceptible to ignition by static electricity as a result of the described mechanism because of electrostatic discharges at the sharp edge of the vent outlet. The National Aeronautics and Space Administration investigated this phenomenon [11] and found that such electrostatic discharges can be prevented by installing a toroidal ring on the vent stack outlet. The ring inhibits the flow of current at the vent stack lip by removing the cause of turbulence characteristics of a sharply defined vent exit.

Ignition of hydrogen from atmospheric vents may also result from the chemical reaction between hydrogen and iron oxides frequently found in vessels and piping. When a stream containing extremely small particles of ferrous oxide (FeO) or iron (Fe) is brought into close contact with oxygen present in the atmosphere, an exothermic reaction occurs that under ideal conditions may provide sufficient energy to ignite a hydrogen-air mixture. The energy requirement has been experimentally determined at 0.017 millijoule (approximately 5 percent of that necessary to ignite a methane-air mixture). This quantity of energy could conceivably be imparted to a tiny particle as a result of the heat released in the reaction of either FeO or Fe with oxygen (O_2). Furthermore, if the ratio of surface area to mass were high enough, a temperature sufficient for ignition might be reached. Also, because of the wide explosive range of hydrogen—4 to 75 volume percent—flammable atmospheres are formed very close to the point of release. This along with hydrogen's very low ignition energy increases the probability of ignition.

4.3.4.1.4 Relief streams, which are above the autoignition temperature on the upstream side of the valve, may ignite spontaneously on contact with air unless sufficient cooling occurs before a flammable vapor-air mixture is formed. For this reason, these hot streams should usually be routed to a closed system, cooler, or quench tower. Under some circumstances, with proper location of the discharge stack, ignition can be tolerated. Under these conditions, the radiation effects discussed in 4.3.4.3 should be evaluated. (See API Publication 2216 for more information.)

4.3.4.2 Explosive Release of Energy

Should a quantity of gas accumulate and then ignite, the possible explosive release of energy in the atmosphere can cause concern about using atmospheric relief. Where unconfined jet momentum releases are involved, as with a normal safety relief valve, there is little potential for the accumulation of large vapor clouds. The total potential hazard would be related to the total quantity of hydrocarbon-air mixture that would accumulate within the flammable envelope downstream of the point of emission. With jet momentum releases, the total volume can be calculated. In a typical case, the flammable zone would be in the range of 40 to 120 diameters downstream but could vary depending on densities and ratios of jet-to-wind velocity. The mixture in this zone might contain an average of about 6 percent hydrocarbon, which would represent 3 seconds of the emitted outflow. The volume within the flammable range at any time would be relatively small compared with the total gas volume emitted and would considerably limit the problem even if an ignition did occur.

If the release rate does not achieve jet momentum and dilution is not achieved, significant vapor clouds can occur. In these cases, care should be taken to avoid any confinement of the released gases since the degree of confinement determines the pressure rise if accidental ignition should occur. Evaluating such confinement should include the proximity of buildings or high concentrations of equipment that produce confinement. The total potential hazard from such sources would then be related to the total quantity of gas released [12].

4.3.4.3 Radiation Effects

Wherever large quantities of flammables are vented, the potential heat release is sufficient to warrant considering its effects on personnel and equipment even though ignition of the discharge from safety relief valves is highly improbable. Once allowable thermal radiation levels have been established, the required distance from various exposure locations to the point of emission may be calculated (see 4.4.2.3 for information on evaluating thermal radiation effects).

4.3.5 EXCESSIVE NOISE LEVELS

The noise generated by a pressure relief valve discharging to the atmosphere can be loud. The noise levels produced by gases at the point of atmospheric discharge can be approximated by reference to 5.4.4.3. Since emergency relief is typically infrequent and of short duration, the noise may not be subject to regulation. In many areas, regulatory authorities define allowable levels of noise exposure for personnel or at property limits. Where no regulatory limits are prescribed, the proposed standards of the American Conference of Governmental Industrial Hygienists [13] may be applied.

The allowable noise intensity and duration should be evaluated at areas where operating personnel would normally work or at property limits. Where two or more pressure relief valves can discharge to the atmosphere simultaneously, the combined effects will need to be evaluated. For design information on noise levels associated with atmospheric discharge, see 5.4.4.3.

4.3.6 AIR POLLUTION

The continuing problem of air pollution has become a factor that warrants serious consideration. Regulations pertaining to air pollution usually provide exemption for discharges that occur only under emergency conditions; however, effluent concentrations at grade level or other locations obviously should be controlled even though the acceptance level for limited and occasional emergency discharge can be much higher than that for prolonged or continuous emissions. Methods for calculating the grade-level concentration to determine whether air pollution could exist are discussed in Gifford's article [14].

4.4 Disposal by Flaring

4.4.1 GENERAL

The primary function of a flare is to use combustion to convert flammable, toxic, or corrosive vapors to less objectionable compounds. Selection of the type of flare and the special design features required will be influenced by several factors, including the availability of space; the characteristics of the flare gas, namely, composition, quantity, and pressure level; economics, including both the initial investment and operating costs; and public relations. Public relations may be a factor if the flare can be seen or heard from residential areas or navigable waterways.

4.4.2 COMBUSTION PROPERTIES

4.4.2.1 Flame Properties

4.4.2.1.1 A flame is a rapid self-sustaining chemical reaction that occurs in a distinct reaction zone. The two basic types of flames are (a) the diffusion flame, which is found in conventional flares and occurs on ignition of a fuel jet issuing into air, and (b) the aerated flame, which occurs when fuel and air are premixed before ignition. The burning velocity, or flame velocity, is the speed at which a flame front travels through the unburned combustible mixture.

4.4.2.1.2 In the case of a flare, the flame front is normally at the top of the stack; however, at low gas velocities, back mixing of air occurs in the top of the stack. Experiments [15] have shown that if a sufficient flow of combustible gas is maintained to produce a flame visible from ground level, there will usually not be significant back mixing of air into the stack. At lower gas flows, there is the possibility of combustion at a flame front located part of the way down the stack with a resultant high stack temperature. Or there can be flame extinguishment with subsequent formation of an explosive mixture in the stack and ignition from the pilot light.

In an aerated flame from a premixing device, a phenomenon known as flashback may occur. This results from the linear velocity of the combustible mixture becoming less than the flame velocity, causing the flame to travel back to the point of mixture.

In the case of either aerated or diffusion flames, if the fuel flow rate is increased until it exceeds the flame velocity at every point, the flame will be lifted above the burner until a new stable position in the gas stream above the port is reached as a result of turbulent mixing and dilution with air. This phenomenon is called blowoff. (Extinguishment of the flame is referred to as blowout.) With a proper tip design, the flame of the main stream can be anchored in the boundary regions where velocity gradient would otherwise far exceed the critical value for blowoff. There is evidence [16, 17, 18] that flame stability can be maintained at relatively high velocities depending on the discharge properties and the type of tip used. Both blowoff and flashback velocities are greater for fuels that have high burning velocities. Small amounts of hydrogen in a hydrocarbon fuel widen the stability range because blowoff velocity increases much faster than flashback velocity. Designs may be based on velocities of 0.5 Mach or higher, if pressure drop, noise, and other factors permit.

4.4.2.2 Smoke

Many hydrocarbon flames are luminous because of incandescent carbon particles formed in the flames. Under certain conditions, these particles are released from luminous flames as smoke. The exact reasons and mechanisms by which smoke is formed are still not fully understood. Many different processes have been suggested, but a discussion of them is beyond the scope of this recommended practice. However, it is safe to say that smoke is formed during the combustion of hydrocarbons only when the system is fuel rich, either overall or locally. Observation has revealed that suppression of the hydrogen atom concentration in the flames accompanies the suppression of smoke formation [19]. Smoke formation could possibly be reduced by reactions that will consume hydrogen atoms or render them ineffective.

The ways in which water vapor reduces smoke from flares have been discussed by Smith [20]. Briefly, one theory suggests that steam separates the hydrocarbon molecules, thereby minimizing polymerization, and forms oxygen compounds that burn at a reduced rate and temperature that are not conducive to cracking and polymerization. Another theory claims that water vapor reacts with the carbon particles to form carbon monoxide, carbon dioxide, and hydrogen, thereby removing the carbon before it cools and forms smoke.

4.4.2.3 Radiation

The information in 4.4.2.3.1 through 4.4.2.3.3 pertains to investigations regarding the effects of thermal radiation on humans and equipment.

4.4.2.3.1 Many investigations have been undertaken to determine the effect of thermal radiation on human skin. Using human subjects, Stoll and Greene [21] found that with an intensity of 2000 British thermal units per hour per square foot (6.3 kilowatts per square meter), the pain threshold was reached in 8 seconds and blistering occurred in 20 seconds. On the bare skin of white rats, an intensity of 2000 British thermal units per hour per square foot (6.3 kilowatts per square meter) produced burns in less than 20 seconds. The same report indicated that an intensity of 7500 British thermal units per hour per square foot (23.7 kilowatts per square meter) caused burns on the bare skin of white rats in approximately 6 seconds. Table 7 gives Buettner's [22] exposure times necessary to reach the pain threshold as a function of radiation intensity. These experimental data were derived from tests given to people who were radiated on the forearm at room temperature. The data state that burns follow the pain threshold fairly quickly. Buettner's data agree well with those of Stoll and Greene.

Table 7—Exposure Times Necessary to Reach the Pain Threshold

Radiation Intensity		Time to Pain Threshold
British Thermal Units per Hour per Square Foot	Kilowatts per Square Meter	(seconds)
550	1.74	60
740	2.33	40
920	2.90	30
1500	4.73	16
2200	6.94	9
3000	9.46	6
700	11.67	4
6300	19.87	2

Since the allowable radiation level is a function of the length of exposure, factors involving reaction time and human mobility should be considered. In emergency releases, a reaction time of 3–5 seconds may be assumed. Perhaps 5 seconds more would elapse before the average individual could seek cover or depart from the area, which would result in a total exposure period ranging from 8 to 10 seconds.

As a basis of comparison, the intensity of solar radiation is in the range of 250–330 British thermal units per hour per square foot (0.79–1.04 kilowatts per square meter). Solar radiation may be a factor for some locations, but its effect added to flare radiation will have only a minor impact on the acceptable exposure time. Regardless of its impact, solar radiation should be considered in the total radiation computation.

Table 8—Recommended Design Total Radiation

Permissible Design Level (K) British Thermal Units per Hour per Square Foot	Kilowatts per Square Meter	Conditions
5000	15.77	Heat intensity on structures and in areas where operators are not likely to be performing duties and where shelter from radiant heat is available (for example, behind equipment)
3000	9.46	Value of K at design flare release at any location to which people have access (for example, at grade below the flare or a service platform of a nearby tower); exposure should be limited to a few seconds, sufficient for escape only
2000	6.31	Heat intensity in areas where emergency actions lasting up to 1 minute may be required by personnel without shielding but with appropriate clothing
1500	4.73	Heat intensity in areas where emergency actions lasting several minutes may be required by personnel without shielding but with appropriate clothing
500	1.58	Value of K at any location where personnel with appropriate clothing may be continuously exposed

Notes:
1. On towers or other elevated structures where rapid escape is not possible, ladders must be provided on the side away from the flare, so the structure can provide some shielding when K is greater than 2000 British thermal units per hour per square foot (6.31 kilowatts per square meter).
2. Solar radiation contribution varies by geographical location and is generally in the range of 250 to 330 BTU/hr/ft² (0.79 to 1.04 kW/m²).

Another factor to be considered regarding thermal radiation levels is that clothing provides shielding, allowing only a small part of the body to be exposed to full intensity. In the case of radiation emanating from an elevated point, standard personnel protective measures, such as wearing of a hard hat, may reduce thermal exposure.

There are practical differences between laboratory tests and full-scale field exposure [3, 23]. Heat radiation is frequently the controlling factor in the spacing of equipment such as elevated and ground flares. Table 8 presents recommended design total radiation levels for personnel at grade or on adjacent platforms. The extent and use of personal protective equipment may be considered as a practical way of extending the times of exposure beyond those listed.

The effects of thermal radiation on the general public, who might be exposed at or beyond the plant boundaries, should be considered.

4.4.2.3.2 In most cases, equipment can safely tolerate higher degrees of heat density than those defined for personnel. However, if any items vulnerable to overheating problems are involved, such as construction materials that have low melting points (for example, aluminum or plastic), heat-sensitive streams, flammable vapor spaces, or electrical equipment, then the effect of radiant heat on them may need to be evaluated. When this evaluation is required, the necessary heat balance is performed to determine the resulting surface temperature for comparison with acceptable temperatures for the equipment [23].

4.4.2.3.3 The following equation by Hajek and Ludwig [18] may be used to determine the minimum distance from a flare to an object whose exposure to thermal radiation must be limited.

$$D = \sqrt{\frac{\tau F Q}{4\pi K}} \tag{20}$$

Where:

D = minimum distance from the midpoint of the flame to the object being considered, in feet (meters).
τ = fraction of heat intensity transmitted.
F = fraction of heat radiated.
Q = heat release (lower heating value), in British thermal units per hour (kilowatts).
K = allowable radiation, in British thermal units per hour per square foot (kilowatts per square meter).

Refer to C.3.6, Note 3, for further information on the use of τ.

The F factor allows for the fact that not all the heat released in a flame can be transferred by radiation. Measurements of radiation from flames indicate that the fraction of heat radiated (radiant energy per total heat of combustion) increases toward a limit, similar to the increase in the burning rate with increasing flame diameter.

Data from the U.S. Bureau of Mines [24] for radiation from gaseous-supported diffusion flames are given in Table 9. These data apply only to the radiation from a gas. If liquid droplets of hydrocarbon larger than 150 micrometers in size are present in the flame, the values in Table 9 should be somewhat increased.

Two methods are present in Appendix C for considering radiation levels. The example in C.2 is the simple approach that has been used for many years. It uses Figures 8 and 9 to determine an estimated flame length with the flame radiation center being at the flame midpoint. A flame under the influence of wind will tilt in the direction the wind is blowing. The lateral wind effect is obtained from Figure 10, which relates horizontal and vertical displacement of the flame center to the ratio of lateral wind velocity to stack velocity.

The correction for the location of the flame center will be quite significant when radiation levels are examined. Flame length varies with emission velocity and heat release. Information on this subject is limited and is usually based on

Table 9—Radiation From Gaseous Diffusion Flames

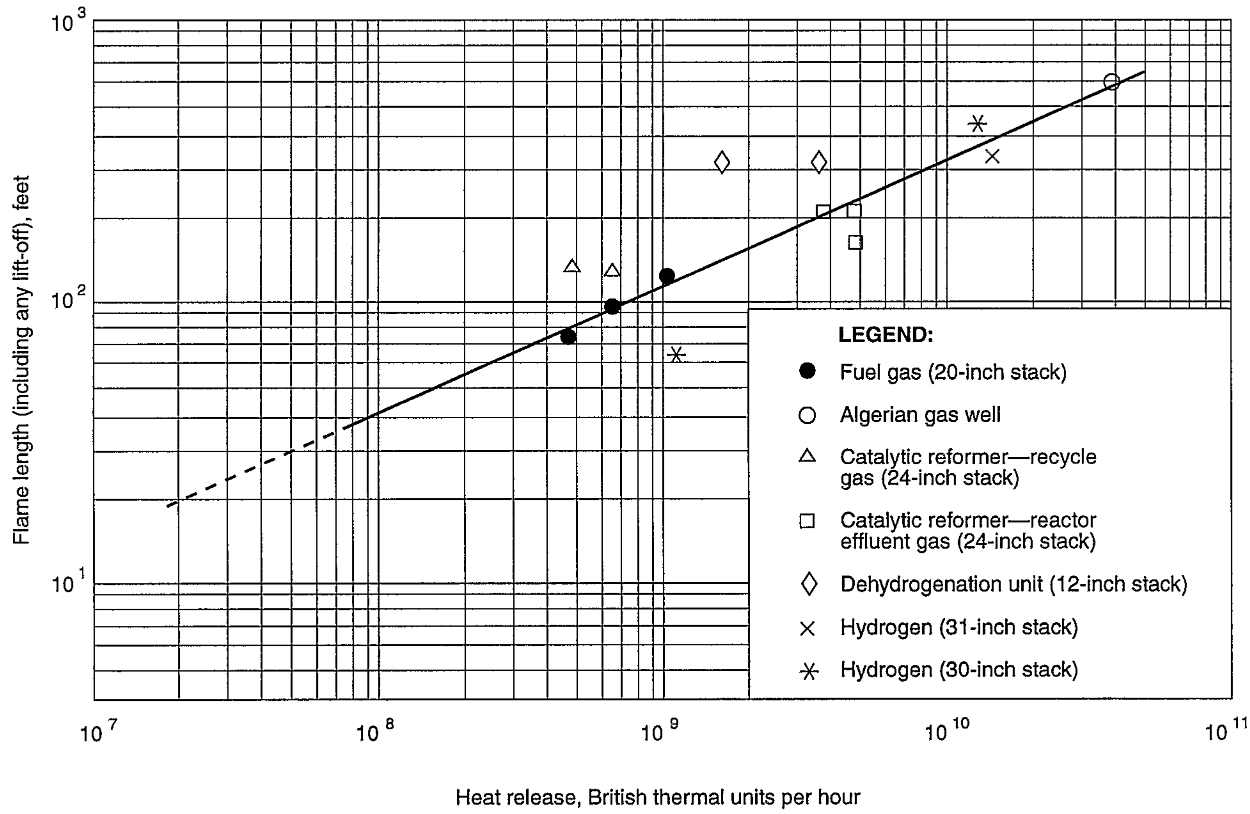
Gas	Burner Diameter (centimeters)	Radiative Output x 100 Thermal Output
Hydrogen	0.51	9.5
	0.91	9.1
	1.90	9.7
	4.10	11.1
	8.40	15.6
	20.30	15.4
	40.60	16.9
Butane	0.51	21.5
	0.91	25.3
	1.90	28.6
	4.10	28.5
	8.40	29.1
	20.30	28.0
	40.60	29.9
Methane	0.51	10.3
	0.91	11.6
	1.90	16.0
	4.10	16.1
	8.40	14.7
Natural Gas	20.30	19.2
(95 percent CH_4)	40.60	23.2

visual observations in connection with emergency discharges to flares. Figures 8 and 9 were developed from some plant-scale experimental work on flame lengths covering relatively high release rates of various mixtures of hydrogen and hydrocarbons.

Several formulas for calculating flame length are presented in the literature [3, 23, 25, 26, 27]. Several formulas for approximating flame tilt are also presented. Each formula has its own special range of applicability and should be used with caution, particularly since the combined impact of several factors (radiation, radiant heat fraction, flame length and center, and flame tilt) must be considered.

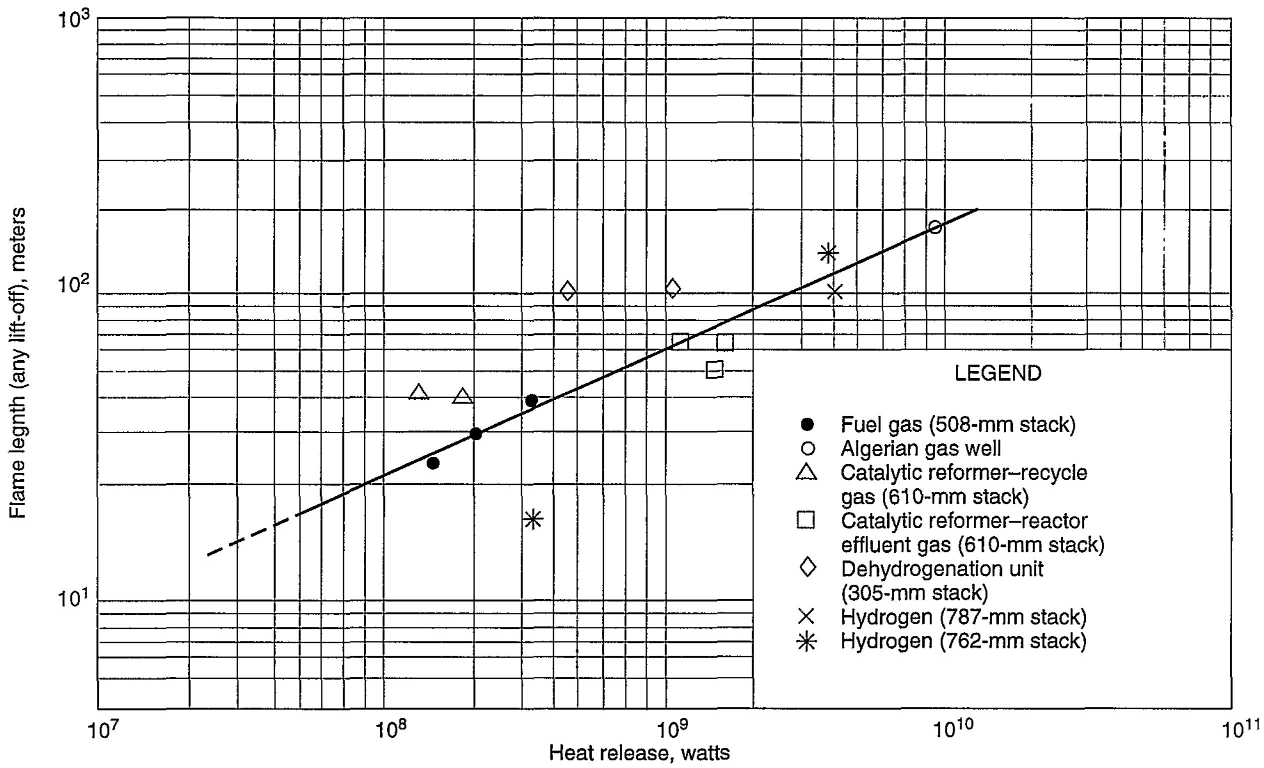
The example in C.3 is another approach to calculating the probable radiation effects, using the more recent method of Brzustowski and Sommer [23]. The principal difference between these methods is the location of the flame center. The curves and graphs necessary to simplify the calculations are included in Appendix C.

There are other methods that can be utilized to calculate radiation from flares. More sophisticated models that consider wind velocity, exit flare gas velocity, flame shape, and



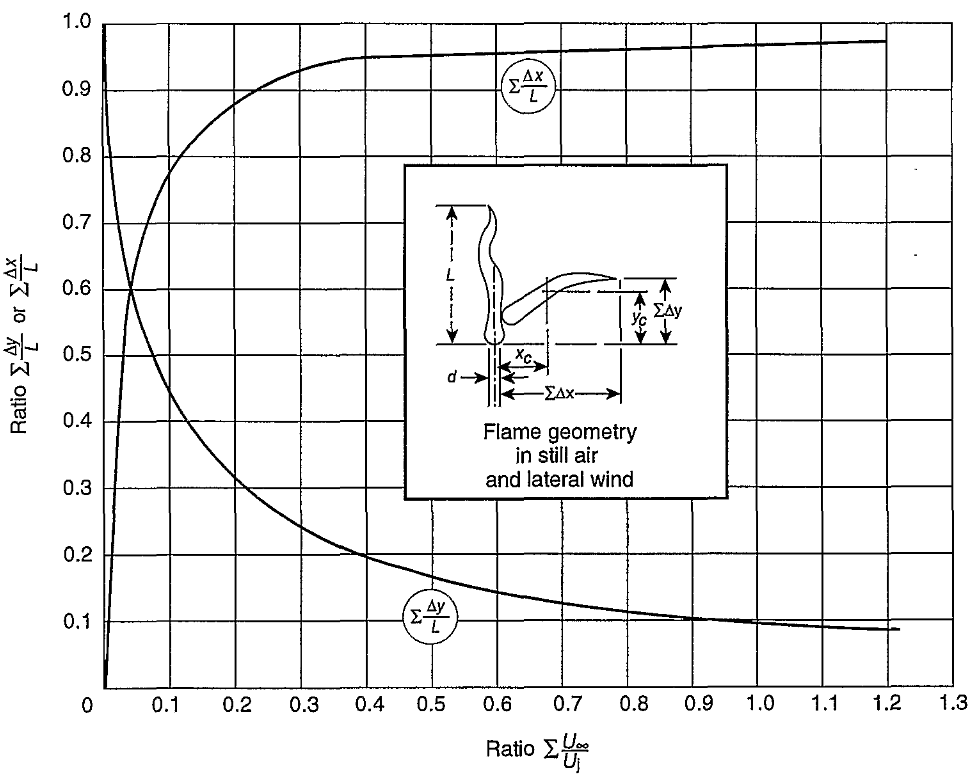
Note: Multiple points indicate separate observations or different assumptions of heat content.

Figure 8—Flame Length Versus Heat Release: Industrial Sizes and Releases (Customary Units)



Note: This figure was developed from Figure 8. Multiple points indicate separate observations or different assumptions of heat content.

Figure 9—Flame Length Versus Heat Release: Industrial Sizes and Releases (SI Units)



U_x = Lateral wind velocity.
U_j = exit gas velocity from stack.

Figure 10—Approximate Flame Distortion Due to Lateral Wind on Jet Velocity From Flare Stack

flame segmental analysis may be appropriate for special cases, especially with large release systems.

Most flare manufacturers have developed proprietary radiation programs based on empirical values. The emissivity (fraction of heat radiated) values used in these programs are specific to the equations used, and may not be interchangeable with the emissivity values used in the API calculation procedure. These programs have not been subject to review and verification in the open literature. The user is cautioned to assess the applicability of these methods to his or her particular situation.

4.4.2.4 Atmospheric Dispersion

Atmospheric dispersion is discussed in 4.3.2 and 4.3.6.

4.4.3 COMBUSTION METHODS

Disposal of combustible gases, vapors, and liquids by burning is generally accomplished in flares. Flares are used for environmental control of continuous flows of excess gases and for large surges of gases in an emergency. The flare is usually required to be smokeless for the gas flows that are expected to occur from normal day-to-day operations. This is usually 15 to 20 percent of the maximum gas flow, but some environmentally sensitive areas require 100 percent smokeless or even a fully enclosed flare. Various techniques are available for producing smokeless operation, most of which are based on the premise that smoke is the result of a fuel-rich condition and is eliminated by promoting uniform air distribution throughout the flames (see 4.4.2.2). The next section provides a description of the most common techniques employed for providing smokeless operation. In addition to smokeless operating requirements, stricter flaring regulations (federal, state and local) are constantly evolving and in most areas typically include low noise levels, limits on smoking reliefs, continuous pilot monitoring, and limits on tip exit velocities/minimum heat content of the flare gas. Current regulations should always be consulted for detailed flaring requirements.

4.4.3.1 Flare Systems Designs

Smokeless operation is normally the overriding requirement when designing the burner for a flare system. Almost every flare design is aimed at inducing smokeless operation under a certain set of flare gas or utility availability conditions. To promote even air distribution throughout the flames (and thus prevent smoke formation), energy is required to create turbulence and mixing of the combustion air within the flare gas as it is being ignited. This energy may be present in the gases, in the form of pressure, or it may be exerted on the system through another medium such as injecting high-pressure steam, compressed air, or low-pressure blower air into the gases as they exit the flare tip. To create conditions favorable for smokeless combustion, flare designs range in complexity from a simple open pipe with an ignition source to integrated staged flare systems with complex control systems. Following is a short summary of the most common types of flaring systems.

4.4.3.1.1 The simplest flare tip design is commonly referred to as a utility or pipe flare tip and may consist of little more than a piece of pipe fitted with a flame retention device for flame stability at higher exit velocities (the upper portion is typically stainless steel to endure the high flame temperatures) and a pilot for gas ignition. This plain design has no special features to prevent smoke formation, and consequently should not be used in applications where smokeless operation is required unless the gases being flared are not prone to smoking, such as methane or hydrogen. Flare tips of this style, as a minimum, should include a flame retention device (to increase flame stability at high flow rates) and one or more pilots (depending upon the diameter of the tip). Windshields/heatshields are usually added on flare tips to reduce flame lick on the outside of the tip. An inner refractory lining is also common with larger diameter tips to minimize thermal degradation caused by internal burning at low rates (known as burnback).

4.4.3.1.2 Flare tips which use steam to control smoking are the most common form of smokeless flare tip. The steam can be injected through a single pipe nozzle located in the center of the flare, through a series of steam/air injectors in the flare, through a manifold located around the periphery of the flare tip, or a combination of all three, as appropriate for a particular application (see Figure 11). The steam is injected into the flame zone to create turbulence and/or aspirate air into the flame zone via the steam jets. This improved air distribution combined with the steam water-gas shift interaction reacts more readily with the flare gases to eliminate fuel rich conditions which result in smoke formation. Proprietary tip designs are available from various manufacturers which offer unique steam injection methods and varying resultant steam efficiencies. The amount of steam required is primarily a function of the gas composition, flow rate, and steam pressure and flare tip design and is normally in the range of 0.25 to 1.0 pound of steam per pound of gas flared. See Table 10. Although steam is normally provided from a 100 to 150 pounds per square inch supply header, special designs are available for utilizing steam pressure in the range of 30 pounds per square inch. The major impact of lower steam pressure is a reduction in steam efficiency during smokeless turndown conditions.

In cold climates, an internal steam nozzle may cause condensate to enter the flare header, collect, and freeze. In some instances, this has resulted in complete blockage of the flare or flare header.

4.4.3.1.3 High-pressure air can also be used to prevent smoke formation. This approach is less common because

Table 10—Suggested Injection Steam Rates

Gases Being Flared	Steam Required (pound of steam per pound of gas)
Paraffins	
Ethane	0.10–0.15
Propane	0.25–0.30
Butane	0.30–0.35
Pentane plus	0.40–0.45
Olefins	
Ethylene	0.40–0.50
Propylene	0.50–0.60
Butane	0.60–0.70
Diolefins	
Propadiene	0.70–0.80
Butadiene	0.90–1.00
Pentadiene	1.10–1.20
Acetylenes	
Acetylene	0.50–0.60
Aromatics	
Benzene	0.80–0.90
Toluene	0.85–0.95
Xylene	0.90–1.00

Note: The suggested amount of steam that should be injected into the gases being flared in order to promote smokeless burning can be determined from this table: The given values provide a general guideline for the quantity of steam required.

compressed air is usually more expensive than steam. However, in some situations with low smokeless capacities, it may seem preferable, for example, in arctic or low-temperature applications where steam could freeze and plug the flare tip/stack. Also, other applications include desert or island installations where there is a shortage of water for steam, or where the waste flare gas stream would react with water. The same injection methods described for steam (4.4.3.1.2) are used with compressed air. The air is usually provided at 100 pounds per square inch and the mass quantity required is approximately 20 percent greater than required by steam since the compressed air does not produce the water-gas shift reaction that occurs with steam.

4.4.3.1.4 High-pressure water, while quite uncommon, is also used to control smoking, especially for horizontal flare applications and when large quantities of waste water or brine are to be eliminated. One pound of water (at 50 to 100 pounds per square inch) is usually required for each pound of gas flared. Freeze protection is required in cold climates and, because of the difficulty in controlling the water flow at low flaring rates, usually requires a staged water spray injection system.

4.4.3.1.5 A low-pressure forced air system is usually the first alternative evaluated if insufficient on-site utilities are

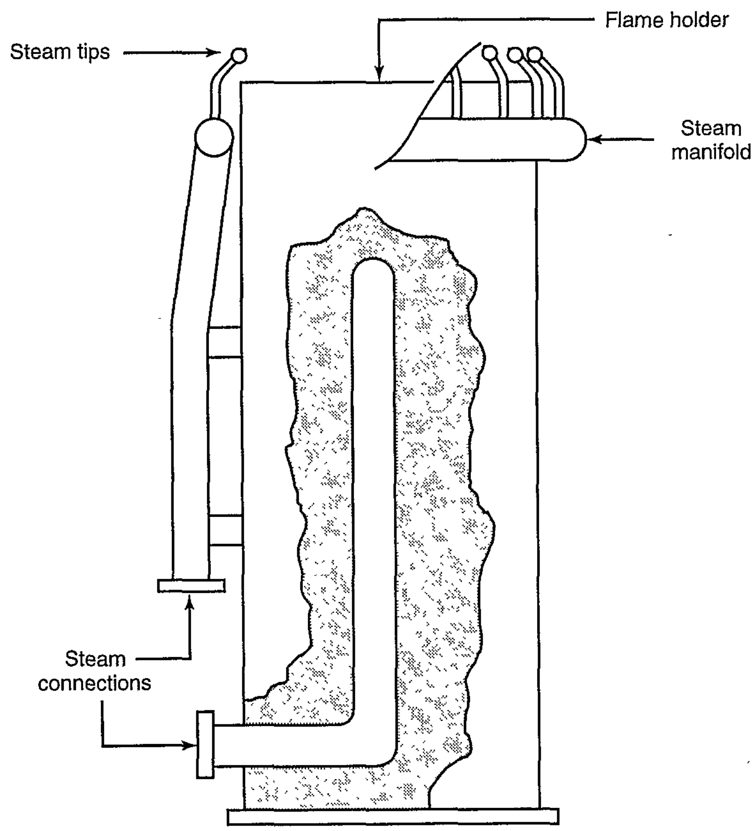


Figure 11—Steam Injected Smokeless Flare Tip

available to aid in producing smokeless operation. The system creates turbulence in the flame zone by injecting low-pressure air supplied from a blower across the flare tip as the gases are being ignited, thus promoting even air distribution throughout the flames. Usually air at 2 inches to 6 inches WC pressure flows coaxially with the flare gas to the flare tip where the two are mixed. This system has a higher initial cost due to the requirement for a dual stack and an air blower. See Figure 12. However, this system has much lower operating costs than a steam-assisted design (only needs power for a blower). The additional quantity of air supplied by the blower for smokeless operation is normally 10 to 30 percent of the stoichiometric air required for saturated hydrocarbons and 30 to 40 percent of the stoichiometric air required for unsaturated hydrocarbons.

4.4.3.1.6 A high-pressure system does not require any utilities such as steam or air to promote smokeless flaring. Instead, these systems utilize pressure energy available within the flare gas itself (typically 5 pounds per square inch to 20 pounds per square inch minimum at the flare tip) to eliminate fuel rich conditions and resulting smoke within the flames. High pressure system limitations are also present but vary by manufacturer and nature of design. By injecting the flare gas into the atmosphere at a high pressure, turbulence is created in the flame zone, which promotes even air distribution throughout the flames. Since no external utilities are required, these systems are normally advantageous for disposing of very large gas releases, both from the economics of smokeless operation and the control of flame shape. The individual tips used have relatively small capacities, and larger system designs may require that many tips be manifolded together. Maintaining sufficient tip pressure during turndown conditions is critical and often requires that a staging system be employed to proportionately control the number of flare tips in service with relationship to the gas flowing. Staged-flare systems can be mounted either at grade or elevated; however, the larger systems may require ground level designs since numerous tips are required (it is not uncommon to have more than 300 tips in a large staged-flare system) and the tips must be evenly spaced to allow air entry into the system.

Staged flares should provide back-up for system failures by inclusion of by-passes or emergency vents. By-passes around control valves are a common safety measure. This is typically done with a rupture disk or like device.

4.4.3.1.7 All of the preceding descriptions have been for flare equipment to dispose of exothermic flare gases; that is, gases that have a high enough heating value (usually greater than 200 Btu/Scf for unassisted flares and 300 Btu/Scf for assisted flares) to sustain combustion on their own without any auxiliary fuel additions. Endothermic gases may be disposed of in thermal incineration systems; however, there are situations where the preferred approach is to use a special flare design. These flares utilize auxiliary fuel gas to burn the flare gases. With small gas flow rates, simple enrichment of the flare gases by adding fuel gas in the flare header to raise the net heating value of the mixture may be sufficient. In other situations, it may be necessary to add a fuel gas injection manifold around the flare tip (similar to a steam manifold) and build a fire around the exit end of the flare tip that the gases must flow through. Dilute ammonia and high CO_2 composition gases with small amounts of H_2S are common applications.

4.4.3.1.8 High-pressure fuel gas can also be used to prevent smoke formation by entraining outside air into the flare flame and generating turbulence to assist overall combustion. Usually, the injection methods are similar to steam tips, but special high performance tips are used to reduce the amount of assist gas. Typical assist gas is 1 volume of assist per 5 volumes of flare gas, based on normal paraffinic gas, for example, propane, butane. The gas pressure for fuel gas assist is typically 75 pounds per square inch (minimum) with 150 pounds per square inch preferred.

4.4.3.2 Enclosed Ground Flares

Ground flares encompass a broad range of vastly different types of flare systems. In general, any of the flare tips or systems discussed in 4.4.3.1 may be mounted atop an elevated stack or mounted at grade. With increasingly strict requirements regarding flame visibility, emissions, and noise, enclosed ground flares can offer the advantages of hiding flames, monitoring emissions, and lowering noise. However, the initial cost often makes them undesirable for large releases when compared to elevated systems. With an enclosed ground flare system, a variety of tips or burners may be utilized and are enclosed or hidden behind a refractory lined carbon steel shell or a radiation fence, depending upon the size of the system. A significant disadvantage with a ground flare is the potential accumulation of a vapor cloud in the event of a flare malfunction; special safety dispersion systems are usually included in the ground flare system. For this reason, instrumentation for monitoring and controlling ground flares is typically more stringent than with an elevated system. These flares are typically the most expensive because of the size of the shell or fence and the additional instrumentation which may be required to monitor these key parameters. Another significant limitation is that enclosed ground flares have significantly less capacity than elevated flares.

4.4.3.3 Elevated Flares

The most common type of flare system currently in use is an elevated flare. In these systems the flare tip is mounted atop the stack, which reduces ground level radiation and the

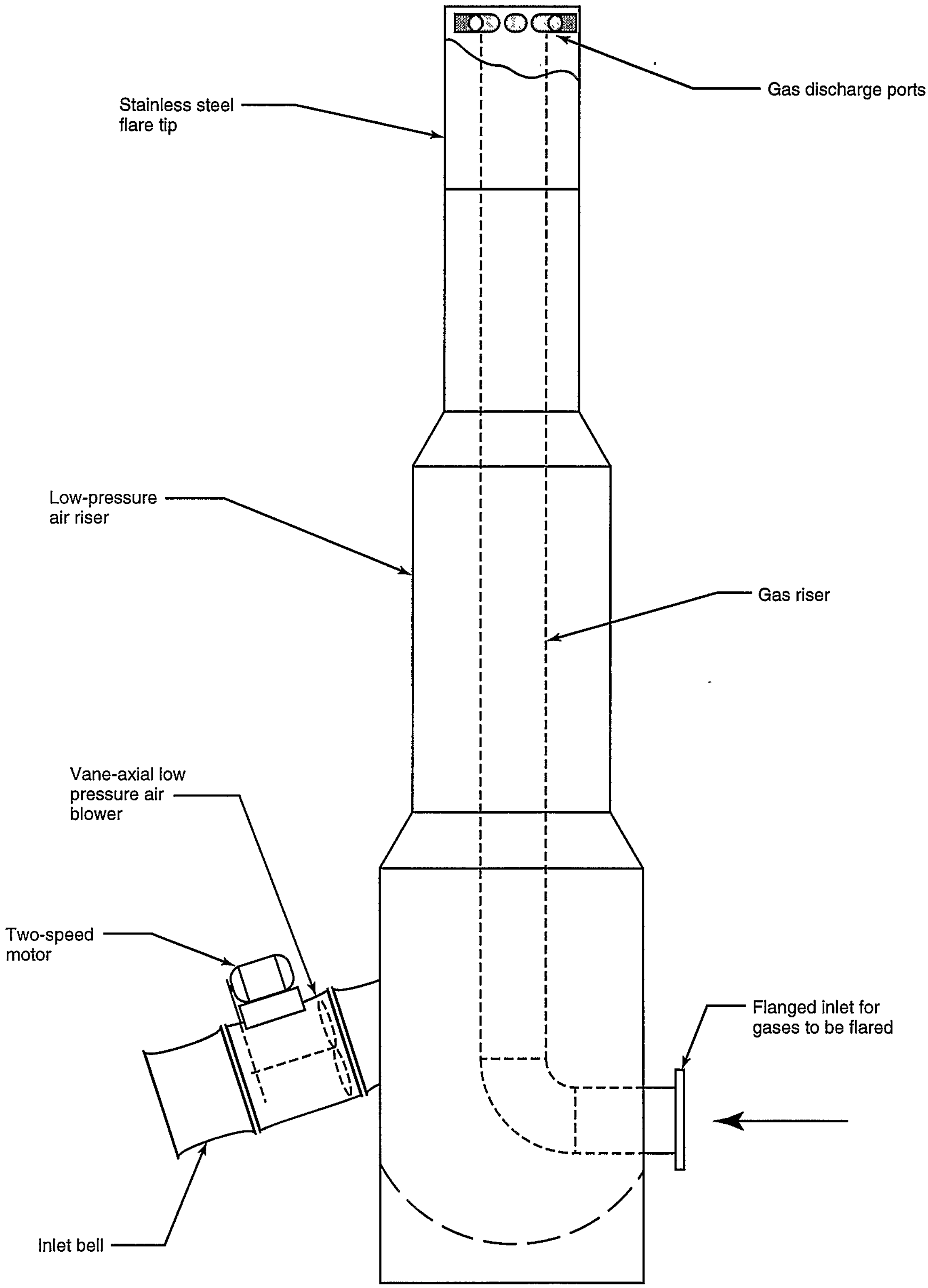


Figure 12—Typical Air-Assisted Flare System

toxicity dispersion profile. There are three common stack support methods as listed below:

a. Self-supported. Self-supported stacks are normally the most desirable. However, they are also the most expensive because of greater material requirements needed to ensure structural integrity over the anticipated conditions (wind, seismic, and the like). They require only enough land area for the foundation and the ability to meet safe ground level thermal radiation levels, but are normally limited (economically versus alternatives) to a stack height of 200 to 300 feet. See Figure 13.

b. Guy-wire supported. These are the least expensive but require the largest land area due to the guy-wire radius requirements. Typical guy-wire radius is equal to one-half the overall stack height. Guyed stacks of heights of 600 feet to 800 feet have been used. See Figure 14.

c. Derrick supported. Used only on larger stacks where self-supported is not practical, or available land area excludes a guy-wire design. Some derrick designs allow the flare stack and tip to be lowered to grade on movable trolleys for inspection and maintenance. This self-lowering design is especially useful when multiple stacks are installed on the same derrick. In locations where land is not available, the multi-flare derrick can be used. See Figure 15.

4.4.3.4 Auxiliary Flaring Equipment

4.4.3.4.1 Liquid seals are very common components in flare systems. Often they are included as integral portions of the base of an elevated stack. Some of the many reasons for using a liquid seal are as follows:

a. Flashback protection. Liquid seals are used to provide positive protection against flame propagation into plant piping. A properly designed and maintained liquid seal offers excellent flashback protection for very flammable gases, such as acetylene, ethylene oxide, and hydrogen. If the flare gas has oxygen inherently contained within it, the addition of a liquid seal as close to the flame source as possible is recommended. Flashback protection must be specifically designed into the seal. The presence of liquid-leg seal is not assurance of this protection.

b. Maintain positive header pressure. Liquid seals are used to provide a means of ensuring that a positive pressure is always maintained on the flare header. This ensures that any leaks in the flare header will result in gas leakage into the atmosphere and not air leakage into the header. Air leakage into a flare header can result in a potentially hazardous gas/air mixture with the possibility of an explosion in the flare stack or header.

c. Direct flare gas flow. Liquid seals are used to direct gases between two or more flare systems. For example, the normal flow of gas may be to an enclosed flare; however, when the capacity of the enclosed flare is reached, the seal would be designed such that the back pressure of the flare gas would exceed the depth of the liquid seal. Thus, the gas would begin to flow to another flare system sized to handle the additional flow. The same logic is also used between a flare gas recovery system and a flare system.

4.4.3.4.2 For safety purposes, a precommissioning and subsequent continuous purge with an oxygen free gas is desirable through the flare stack. The prepurge displaces any existing air from the stack, and the continuous purge ensures that atmospheric air does not enter the stack through the flare tip during low-flow conditions. The requirements for a continuous purge may be eliminated if a liquid seal is located near the base of the stack (refer to 4.4.3.4.1). This requires special precautions in the design of the stack to assure viability in the event of an internal space explosion. It also may allow air to infiltrate to the liquid seal, which for some compounds carries other requirements.

Air present in the stack can create a potentially explosive mixture with incoming flare gas during low-flare gas loads. There are two common types of mechanical seals usually located at/or below the flare tip that are used to reduce the amount of continuous purge gas required to prevent air infiltration into the flare stack:

a. Diffusion-type seal. This type of seal uses the difference in the molecular weights of the purge gas and infiltrating air to form a gravity seal which prevents the air from entering into the stack. A baffled cylinder arrangement forces the incoming air through two 180-degree bends (one bend up and one bend down) before it can enter into the flare stack. If the purge gas is lighter than air, the purge gas will accumulate in the top of the seal and prevent the air from infiltrating the system. If the purge gas is heavier than air, the purge gas will accumulate in the bottom of the seal and prevent air from infiltrating. This seal normally reduces the purge gas velocity required through the tip to 0.01 feet per second. Also, with some purge gas compositions, this rate will limit oxygen levels below the device to less than 0.1 percent. However, these low purge rates do not prevent burnback inside the flare tip, which results in short tip life. This effect is called cancer because the deterioration of the metal wall of the flare tip of the molecular seal is hidden until the flame burns through the tip or seal requiring shutdown for immediate maintenance. Most molecular seals are purged at rates of 0.5 feet per second to keep the flame out of the flare tip and insure proper flare life. Flare tips with velocity seals built into their top section can eliminate this burnback at much lower purge rates. See Figure 16.

b. Velocity seal. This seal works under the premise that infiltrating air enters through the flare tip and hugs the inner wall of the flare tip. The velocity seal is a cone shaped obstruction with single or multiple baffles, which forces the air away from the wall where it encounters the focused purge gas flow and is swept out of the tip. This seal normally reduces the purge gas velocity through the tip to between 0.02 feet per

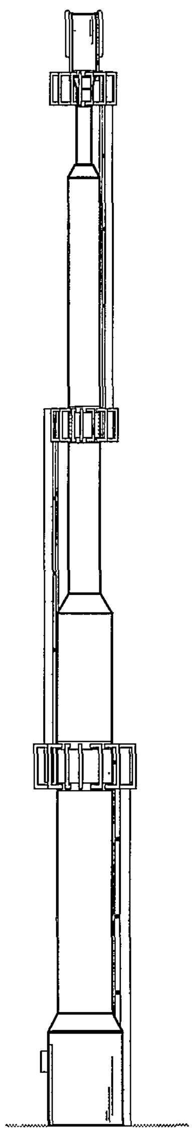


Figure 13—Self-Supported Structure

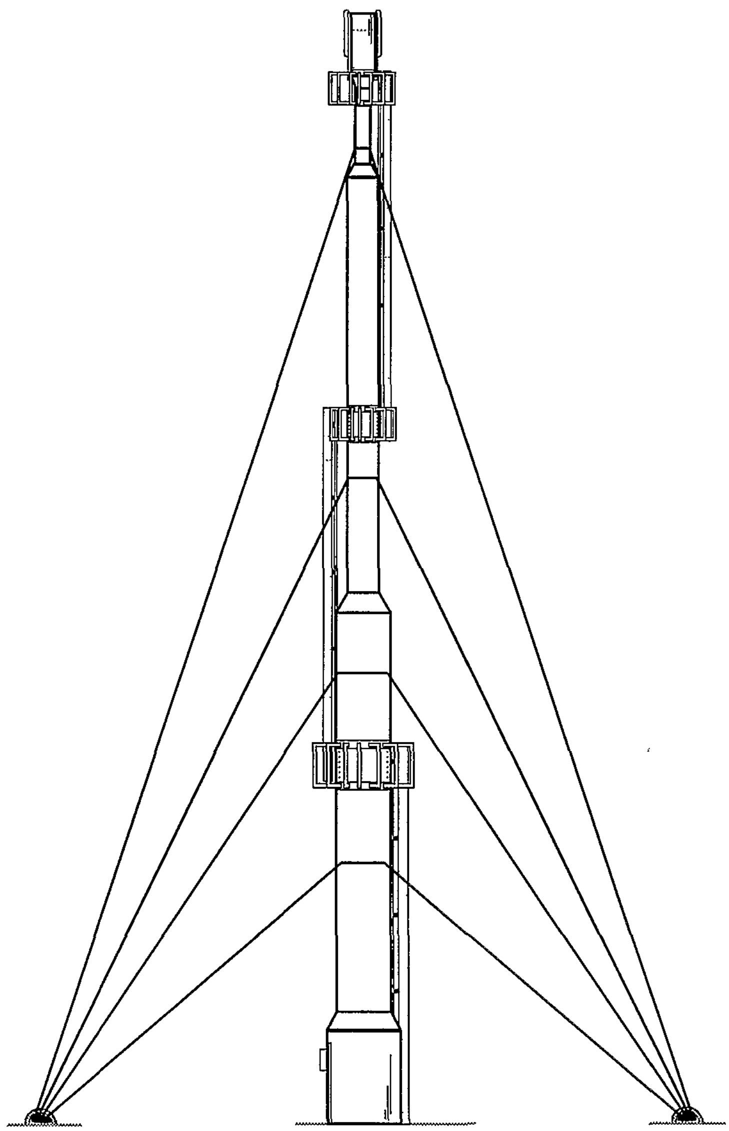


Figure 14—Guyed-Supported Structure

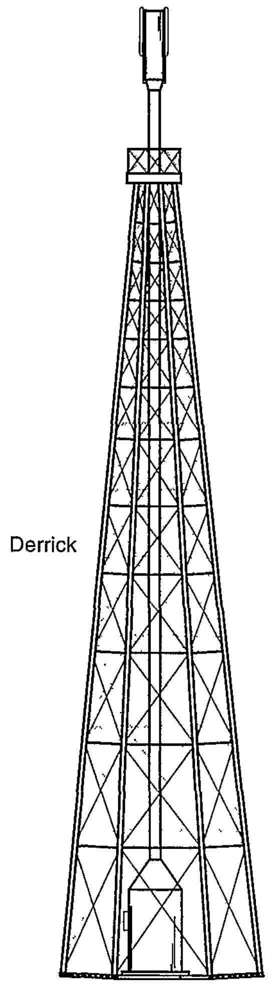


Figure 15—Derrick-Supported Structure

second and 0.04 feet per second, which keeps oxygen concentrations below the seal to 4 percent to 8 percent (approximately 50 percent of the limiting oxygen concentration required to create a flammable mixture). See Figure 17.

Without either one of these seals, the purge gas velocity in the tip required to prevent air infiltration into the stack should be determined using the procedure described 5.4.3.2.2.

CAUTION: Purge reduction seals are not flame arrestors; that is, they will not stop a flash back. They are designed as energy conservation devices to reduce purge gas flows required to prevent flashbacks by reducing air infiltration into the stack.

4.4.3.4.3 The following methods of controlling steam (or compressed air, water, and so on) for smokeless flare control are common (many other strategies are possible).

a. Manual operation. Manual control usually involves remote operation of a steam valve by operating personnel assigned to a unit from which the flare is readily visible. This method is satisfactory if short-term smoking can be tolerated when a sudden increase in flaring occurs. With a manual arrangement, close supervision is required to ensure that the steam flow is reduced following the correction of an upset. Operating costs can be excessive if monitoring is not timely.
b. Television monitoring with manual control. The philosophy is the same as with manual operation except a television-monitoring system is added so that control room operators can monitor and control the steam flow.
c. Feed forward control system pressure, mass flow or velocity. By measuring the amount of flare gas flowing to the flare, the steam rate can be automatically adjusted to compensate for rate changes. This system may not be desirable if the type of hydrocarbon flared frequently changes (in other words, paraffin to olefin or aromatic, and so forth).
d. Feedback system using on infrared sensor. Infrared sensors can be used to detect smoke formation in the flames and automatically adjust the steam control valve to compensate. A disadvantage of this system is that infrared waves are absorbed by moisture, and the resultant feedback signal is reduced in rainy or foggy conditions.

At low flaring rates, fluctuations in either pressure or flow are so minute that very sensitive instruments are required to provide sufficient steam for smokeless combustion while at the same time avoiding waste. Therefore, controls must be carefully sized, precisely adjusted, and properly installed to obtain satisfactory operation.

4.4.3.4.4 A pilot ignition panel is an integral part of any flare system utilizing flame-front generation for ignition. A flame-front generator will fill an ignition line leading from the panel to the flare pilot with a flammable mixture of air and fuel gas. A spark is then introduced at the panel to ignite the mixture and send a flame-front through the piping to ignite the pilot. These panels can be operated manually, or can be automated for pilot reignition when pilot flame-out is detected. Electronic ignition systems, which do not require flame-front propagation, are also used. These systems typically ignite the pilot gas at the pilot itself.

4.4.3.4.5 Several methods of pilot monitoring are available, including thermocouples installed within the pilot head, ionization monitoring within the pilot head, and remote infrared or optical monitors. Further details concerning their design may be found in 5.4.

4.4.3.4.6 Aircraft warning lights are usually required only when flare heights are greater than 200 feet or when the stack is close to an airport. The type of lights and number required are regulated by governmental agencies (such as the Federal Aviation Administration). Consideration should be given to maintenance accessibility in determining the types of lights to use.

4.4.3.5 Flaring Toxic Gases

The flaring of toxic gases requires special considerations. Some information can be obtained from a test program sponsored by the Environmental Protection Agency (EPA) through the Chemical Manufacturers Association (CMA). The destruction efficiency for certain combustible toxic material in a properly operated flare may be in the range of 98 percent [28].

Depending upon the gases being flared and the flare style being used, the minimum allowable net lower heating value should be in the range of 200 to 300 Btu/Scf. If the Btu/Scf value can drop below this range, a special flare design may be required. (See 4.4.3.1.7.)

To ensure safe operation during periods when the flare may not have a flame present, ground level concentration calculations for hazardous components should be performed assuming the flare as a vent only. Other safeguards may be necessary to mitigate groundlevel exposure hazards. Reliable continuous pilot monitoring is considered critical when flaring toxic gases.

4.4.3.6 Burn Pits

Burn pits normally require excavation or bermed areas to contain liquid hydrocarbons or other objectionable materials produced by incomplete combustion. Seepage from a poorly designed or maintained burn pit can pose a threat to groundwater supplies.

4.5 Disposal to a Lower-Pressure System

4.5.1 GENERAL

Discharge of the relieved material to the same or another system at lower pressure may be a safe and economical

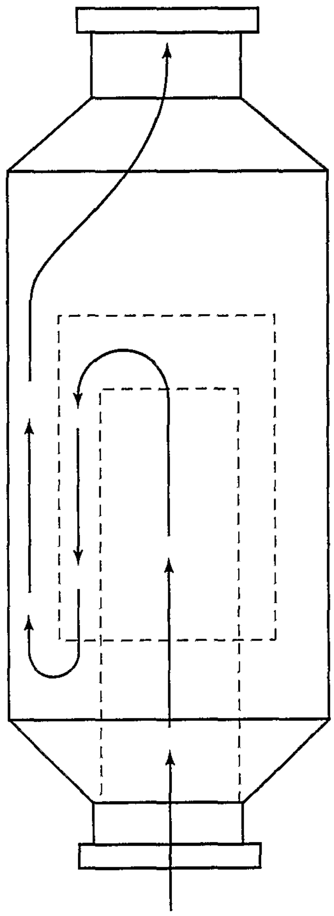


Figure 16—Purge Reduction Seal—Molecular Type

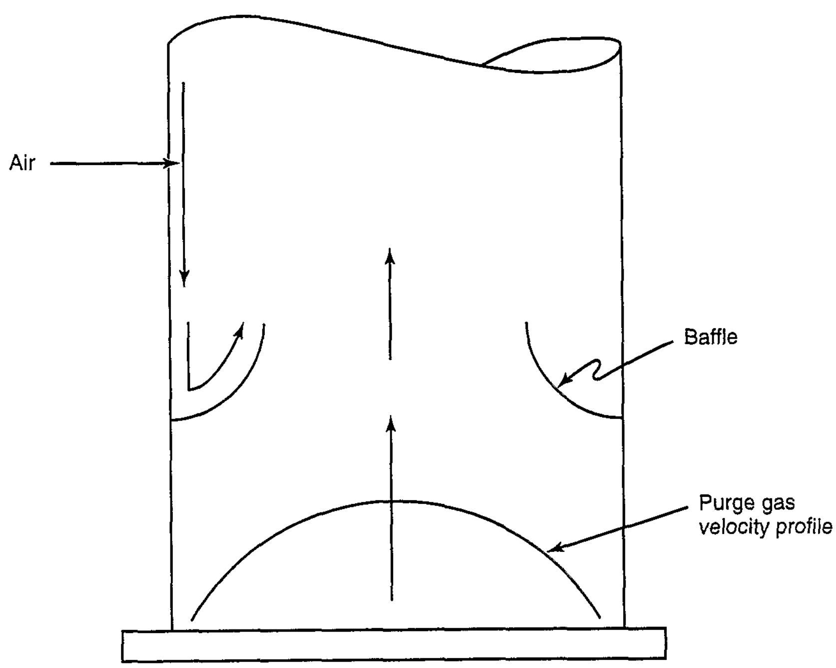


Figure 17—Purge Reduction Seal—Velocity or Venturi

method provided that the receiving system is designed for the additional load.

4.5.2 SEWER

Nonvolatile liquid discharges from relief valves may be piped to sewer drains provided that the sewer system has adequate capacity and is properly sealed and vented. Caution should be exercised to avoid discharging volatile, toxic, or hot fluids into a sewer.

4.5.3 PROCESS

The particular type of process unit selected will determine whether a lower-pressure process system exists that can safely receive material relieved from a higher pressure system. This will usually be true with liquid reliefs (for example, liquid relieved from the discharge side of a pump being disposed to the suction side). Selection of the type of valve to be used (that is, a balanced or a conventional valve) will depend on the back pressure—constant, variable, or built-up—of the lower pressure system.

4.6 Disposal of Liquids and Condensable Vapors

The selection of a disposal system for liquids and condensable vapors, not covered in 4.3 through 4.5, is determined on the basis of the items covered in 4.6.1 through 4.6.4.

4.6.1 TEMPERATURE

4.6.1.1 General

Only a comprehensive study of the plot plan and individual safety relief valve data can determine the most desirable system for a particular plant. The methods of coping with temperature problems given in 4.6.1.2 through 4.6.1.4 are not meant to be limiting. They are included merely to illustrate a principle of separation of discharges.

4.6.1.2 Ambient

Nonvolatile liquids at ambient temperatures may be discharged into a separate closed relief header that discharges into a sump from which the liquids are recovered. Volatile or nonvolatile liquids may be alternately discharged into the regular closed disposal system. The liquid will be disengaged at the knockout drum before the flare (see 5.4.2 for additional details).

4.6.1.3 Hot

Hot liquids and vapor may be cooled and condensed by one of the methods covered in 4.6.1.3.1 through 4.6.1.3.3.

4.6.1.3.1 Pressure relief valves that discharge hot condensable hydrocarbon vapors or liquids may be piped into a separate header that terminates in a quench drum. In this service, quench can reduce the temperature of the relief stream and may permit the use of less-expensive materials in downstream equipment. Cooling also condenses some of the less-volatile components and can reduce or prevent the release of hot condensable vapors to the atmosphere. A quench drum is a vessel equipped to spray a quenching liquid down through the hot discharged gases as they pass at reduced velocity through the drum. The quenching fluid may be water, gas oil, or another suitable liquid. The used liquid collects in the bottom of the drum for subsequent removal.

One type of quench drum is a vertical vessel containing baffles that is connected by a means of a conical transition to a vent stack or flare. The condensable hydrocarbon material is fed into the drum below the baffles. Water is introduced into the drum above the baffles at a rate that depends on the temperature and the amount of hydrocarbon material being fed to the quench drum. The water, spilling over the baffles, desuperheats and condenses the hydrocarbon vapor, knocks out entrained hydrocarbon liquid, and cools down the hydrocarbon liquid collected in the bottom of the drum. The uncondensed vapor and any steam formed pass up the vent stack or enter a flare system (see Figure D.2).

4.6.1.3.2 The submerged discharge system is not extensively used in present-day design. Care should be taken in its use and location when noncondensable gases that could escape to the atmosphere are present. Cooling of hot liquid and condensation of vapor by submerged discharge in a large body of cold liquid may have limited utility when considered as a disposal method to a lower-pressure system in the same process unit. Occasionally, steam is mixed with the effluent in sufficiently large quantities to make the discharge noncombustible. In this type of design, the pressure-relieving system on a unit that handles heavy hydrocarbons generally serves a dual purpose—as a disposal system for the pressure-relieving devices and as a dropout or blowdown system for furnaces and vessels.

The submerged discharge system is a relieving system that terminates in parallel laterals submerged in a water-filled sump. Holes are cut in the bottom of the laterals throughout their length, imparting downward flow to the discharged effluent to obtain maximum agitation, cooling, and condensing. Provisions must be made to maintain a liquid level in the sump while the blowdown system is being used. The discharge is drained from this sump into a separator, where the oil and condensed vapors are removed from the water.

4.6.1.3.3 The use of shell-and-tube heat exchangers or coil-in-box coolers has the merit of separating cooled or condensed material immediately. In addition, the coil-in-box cooler can remove some heat, which is desirable in emergencies when no cooling water is flowing.

4.6.1.4 Cold

Low-temperature fluids require considerations similar to those outlined for hot streams, particularly if there is a possibility of low-boiling liquids entering the disposal system. Autorefrigeration will occur as liquid boils at the reduced pressure. If the equilibrium temperature is sufficiently low, piping and drums fabricated of materials designed for low temperature may be required to eliminate the risk of brittle fracture [29]. In such circumstances, consideration should be given either to a completely separate low-temperature system or isolation of the stream until it reaches a knockout drum where the liquid can disengage. Vapors vented off the drum can often be safely combined with other disposal systems if, in the absence of liquid, the heat pickup (of the piping system) from the surrounding atmosphere will prevent temperatures from dropping to a dangerously low level.

4.6.2 HAZARDOUS PROPERTIES

The safe disposal of material that has toxic, acidic, alkaline, or corrosive properties may require chemical neutralization, absorption, or reaction in a special disposal system. Dilution with air or water to a safe level may be satisfactory in some cases.

4.6.3 VISCOSITY AND SOLIDIFICATION

In the selection of a disposal system for liquids and condensable vapors, the production of highly viscous or solid materials warrants consideration. The design of a disposal system for such materials may require steam tracing of valves and discharge lines. The formation of gums, polymers, coke, or ice that might prevent safe operation of the discharge system should also be considered in the design.

4.6.4 MISCIBILITY

Solubility or miscibility of the material with water and avoidance of the formation of emulsions should be considered in the selection of a disposal system.

4.7 References

1. NFPA 325 M, *Fire Hazard Properties of Flammable Liquids, Gases, and Volatile Solids*, National Fire Protection Association, Quincy, Massachusetts, 1991.

2. M. G. Zabetakis, *Flammability Characteristics of Combustible Gases and Vapors,* U.S. Bureau of Mines Bulletin 627, U.S. Department of the Interior, Washington, D.C., 1965.

3. V. O. Hoehne, R. G. Luce, and L. W. Miga, *The Effect of Velocity, Temperature, and Gas Molecular Weight on Flammability Limits in Wind-Blown Jets of Hydrocarbon Gases,* Report to the American Petroleum Institute, Battelle Memorial Institute, Columbus Ohio, 1970.

4. J. F. Taylor, H. L. Grimmett, and E. C. Comings, "Isothermal Free Jets of Air Mixing with Air," *Chemical Engineering Progress, 1951,* Volume 47, pp. 175–180.

5. *International Safety Guide for Oil Tankers and Terminals,* International Chamber of Shipping/Oil Companies International Marine Forum, Witherby and Co., London, 4th edition, 1996.

6. D. E. Loudon, "Requirements for Safe Discharge of Hydrocarbons to Atmosphere," *API Proceedings,* 1963, Volume 43, American Petroleum Institute, Washington, D.C., pp. 418–433.

7. J. H. Burgoyne, "Mist and Spray Explosions," *Chemical Engineering Progress,* 1957, Volume 53, Number 3, pp. 121–124.

8. D. I. Saletan, "The Theory Behind Static Electricity Hazards in Process Plants," *Chemical Engineering,* 1959, Volume 66, Number 1, pp. 99–102.

9. F. G. Eichel, "Electrostatics," *Chemical Engineering,* March 13, 1967, pp. 153–167.

10. J. W. Young, et al., "Electrostatic and Gas Blowout," *Canadian Mining Meeting Bulletin Number 63,* 1960, p. 456.

11. *Toroidal Ring Prevents Gas Ignition at Vent Stack Outlet,* National Aeronautics and Space Administration Technical Brief 67-10098, National Technical Information Service, Springfield, Virginia, 1967.

12. *Guidelines for Evaluating the Characteristics of Vapor Cloud Explosions, Flash Fires, and BLEVES,* Center for Chemical Process Safety, American Institute of Chemical Engineers, New York, 1994

13. "American Conference of Governmental Industrial Hygienists' Proposed Threshold Limit Value for Noise," *American Industrial Hygiene Association Journal,* November-December 1968, Volume 26, Number 26, pp. 537–540.

14. F. A. Gifford, Jr., "Atmospheric Dispersion Calculations Using the Generalized Garrison Plume Model," *Nuclear Safety,* December 1960, pp. 56–59.

15. H. W. Husa, "How to Compute Safe Purge Rates," *Hydrocarbon Processing and Petroleum Refiner,* 1964, Volume 43, Number 5, pp. 179–182.

16. S. H. Tan, "Flare System Design Simplified," *Hydrocarbon Processing and Petroleum Refiner,* 1967, Volume 46, Number 1, pp. 172–176.

17. G. R. Kent, "Practical Design of Flare Stacks," *Hydrocarbon Processing and Petroleum Refiner,* 1964, Volume 43, Number 8, pp. 121–125.

18. J. D. Hajek and E. E. Ludwig, "How to Design Safe Flare Stacks," Part 1, *Petro/Chem Engineer,* 1960, Volume 32, Number 6, pp. C31–C38; Part 2, *Petro/Chem Engineer,* 1960, Volume 32, Number 7, pp. C44–C51.

19. J. R. Arthur, "Some Reactions of Atomic Hydrogen in Flames," *Nature,* 1950, Volume 165, pp. 557–558.

20. W. R. Smith, "Eliminate Air-Polluting Smoke," *Petroleum Processing,* June 1954, pp. 887–881.

21. A. M. Stoll and L. C. Green, *The Production of Burns by Thermal Radiation of Medium Intensity,* Paper Number 58-A-219, American Society of Mechanical Engineers, New York, 1958.

22. J. K. J. Buettner, *Heat Transfer and Safe Exposure Time for Man in Extreme Thermal Environment,* Paper Number 57-SA-20, American Society of Mechanical Engineers, New York, 1957.

23. T. A. Brzustowski and E. C. Sommer, Jr., "Predicting Radiant Heating from Flares," *Proceedings–Division of Refining,* 1973, Volume 53, American Petroleum Institute, Washington, D.C., pp. 865–893.

24. M. G. Zabetakis and D. S. Burgess, *Research on the Hazards Associated with the Production and Handling of Liquid Hydrogen,* U.S. Bureau of Mines Rept. Invest. 5707, U.S. Department of the Interior, Washington, D.C., 1961.

25. R. Schwartz and M. Keller, "Environmental Factors Versus Flare Application," Paper Number 13d, presented at the Symposium on Loss Prevention in the Chemical Industry, 83rd National Meeting of the American Institute of Chemical Engineers, Houston, Texas, March 22, 1977.

26. L. D. Cleveland, *Oil and Gas Journal,* August 4, 1952.

27. R. D. Reed, "Design and Operation of Flare Systems," Chemical Engineering Progress, June 1968, Volume 64, Number 6.

28. EPA-600/2-86-080, "Evaluation of The Efficiency of Industrial Flares: H_2S Gas Mixtures and Pilot Assisted Flares," Environmental Protection Agency, Washington, D.C., September 1986.

29. *Prevention of Brittle Fracture of Pressure Vessels,* 1st edition, ANSI/API Publication 920-1992, American Petroleum Institute, Washington, D.C., 1990.

4.8 Bibliography

4.8.1 G. A. Chamberlain, "Developments in Design Methods for Predicting Thermal Radiation From Flares," *Chem Eng Res Des,* Volume 65, July 1987, pp. 299–309.

4.8.2 *Gas Processors Suppliers Association Data Book,* Section 5, 10th Edition, Tulsa, Oklahoma, 1987.

SECTION 5—DISPOSAL SYSTEMS

5.1 General

The preceding sections have outlined contingencies to be considered in determining individual relieving requirements, together with methods of calculating depressuring or relieving rates. A discussion of alternative arrangements for disposal of valve discharges has also been presented. This section provides general principals and guidelines for the design of disposal systems.

5.2 Definition of System Load

5.2.1 GENERAL

The first requirement in designing a disposal system is to define the loadings to be handled. Where the disposal system serves only a single-pressure relief valve or depressuring valve, the design basis for the disposal system would normally correspond to the design loading of the pressure relief valve or to the maximum flow rate of the depressuring valve.

Where the disposal system serves more than one pressure relief or depressuring valve, loadings should be calculated for all contingencies that may affect any of the pressure relief or depressuring valves. This means that each pressure relief valve tied into the system should have a flow rate, a temperature, and fluid properties calculated for each pertinent contingency.

For disposal systems that serve more than one pressure relief or depressuring valve, the geographic location of each source should be defined. Combinations of sources into one or more headers for disposal should be studied or defined. Based on the specified or assumed combination arrangement, the loadings for each section of header can then be defined. Maximum load is not necessarily the largest number of pounds per hour (kilograms per second): It is the flow that imposes the greatest head loss in flowing through the system. Thus, a flow of 100,000 pounds per hour (12.6 kilograms per second) of a vapor with a molecular weight of 19 at a temperature of 300°F (149°C) develops a greater head loss and is a greater load than a flow at 150,000 pounds per hour (18.9 kilograms per second) of a vapor with a molecular weight of 44 at a temperature of 100°F (38°C).

After defining the loadings to be handled in the disposal system, the maximum allowable back pressure under each contingency should be defined for each pressure relief valve. The allowable back pressure will be a function of (a) the type and make of pressure relief valve, (b) the set pressure of the valve, and (c) the relieving rate required for each contingency.

When a pressure relief valve has a set pressure below the maximum allowable working pressure of the protected equipment, an increase in back pressure over normal limits may be permissible (see API Recommended Practice 520, Part I). However, maximum allowable back pressure may be limited by API Standard 526. Where a single contingency affects a number of pressure relief valves, the timing and duration of relieving become significant factors in the combined load. If large relieving rates are involved, it may be worthwhile to estimate the time scale for the system load. This may require calculating the time lag between the initiation of the contingency and the opening of each pressure relief valve, the effects of depletion of inventory and change in composition on relieving rates, and the effects of increased equipment pressure on the supply of material and energy to the process system. Consideration may also be given to the capability for and timing of operator intervention as a means of reducing system loads.

5.2.2 LOADS FROM PRESSURE RELIEF VALVES

The contingencies to be considered in defining relieving requirements are discussed in Section 2. To define the system load, the simultaneous occurrence of two or more unrelated contingencies need not be assumed. For example, it is generally not necessary to assume blocked outlets on process systems under fire conditions. However, under some arrangements of process equipment, a fire could possibly result in a failure of local wiring or instrument air piping, leading to the closure of valves that block off the process system. Each individual contingency should therefore be reviewed for possible resultant effects.

Particular study is required for cases of failure of major utilities, such as power or cooling water. Complete failure of electrical power, cooling water, or steam to an entire plant should be considered. Where utility sources are believed to be unreliable or are not backed up by a spare unit, the effect of complete failure should be studied. This type of study, with reference to electrical power failure, commonly results in a design based on the failure of one bus, although loss of an entire distribution center or of the incoming line is occasionally used as a basis for design.

The capability of steam systems to pick up standby turbine loads should be reviewed in conjunction with the overall installed boiler capacity and the normal standby capacity immediately available.

The most common basis for analyzing water or steam failure is the failure of one lateral rather than the entire supply. Instrument air failure is commonly considered to be a plantwide failure unless automatic makeup from an uninterrupted source is provided. Failure of the power supply to electronic or electrical instruments may also be considered plantwide unless proper standby power supplies are provided.

To define the combined relieving loads under fire exposure, the probable maximum extent of a fire should be estimated. This may be done on the basis of the actual layout of facilities, considering the location of sources of combustibles, the provision of drainage, and the effects of natural barriers. Facilities that handle only gaseous fluids may be assumed to generate more localized fires than those that handle liquid combustibles. In the absence of any other governing factors, consideration of a fire incident is frequently limited to a ground area of 2,500 square feet to 5,000 square feet (230 square meters to 460 square meters).

5.2.3 LOADS FROM DEPRESSURING SYSTEMS

Depressuring valves are normally assumed to be either 100 percent open or 100 percent closed. The maximum load from an open depressuring valve normally corresponds to the flow capacity of the valve at the maximum pressure of the protected equipment; this may be the maximum accumulated pressure of the equipment.

Where the same equipment is provided with both pressure relief valves and depressuring valves, only the larger of the pressure relief valve load or the depressuring valve capacity needs to be considered as the disposal system load.

If the capacity of a vapor-depressuring valve should exceed the normal vapor flow rate within the protected equipment or if the depressuring rate is additive to normal flows within the equipment, considerable liquid entrainment may occur. Therefore, disposal systems for depressuring valves should generally provide for liquid carryovers.

5.3 System Arrangement

5.3.1 GENERAL

Once the various combinations of loads have been defined for all pertinent contingencies and the corresponding allowable back pressures have been determined for all relief valves, selection of the disposal system can proceed. The factors influencing the choice of the disposal system are discussed in Section 4.

In selecting the arrangement of the disposal system or systems, special attention should be given to situations where relief valves can discharge flashing liquids or where a combination of cold liquid and hot vapor discharge may result in vaporization of the liquid. Such situations may generate additional vapor loads beyond those that correspond to the relieving loads (see also 4.6.1.4 for special considerations in handling liquids that are capable of autorefrigeration).

5.3.2 SINGLE-VALVE DISPOSAL SYSTEMS

Where only a single pressure-relief valve or a single depressuring valve is connected to the disposal system, the outlet may also be to the atmosphere, to another system operating at lower pressure, or to a local flare.

If the outlet is connected to a lower-pressure system, the allowable pressure drop in the disposal system should generally be based on the maximum allowable working pressure of the lower-pressure equipment. However, a reduced back pressure (for example, normal operating pressure in the lower-pressure equipment) may be used if it can be shown that (a) none of the contingencies causing a relieving load would also overpressure the lower-pressure equipment, and (b) the load (the required capacity of the valve) imposed by the higher-pressure relief valve would not result in overpressuring the lower-pressure equipment.

Each pressure relief valve that vents directly to the atmosphere should normally have an individual vent pipe sized for a relatively high exit velocity (at the design relieving rate); however, the outlet piping should not be smaller than the pressure-relief valve outlet flange. The developed back pressure of this system should include all pressure losses, such as exit losses, friction losses, and kinetic energy loss. On-off-type pressure relief valves will generate instantaneous loads that are equivalent to the full valve capacity. The design of the disposal system should be checked for adequacy under such conditions.

Where the atmospheric vent handles combustible vapors, the outlet from the vent should be elevated approximately 10 feet (3 meters) above any adjacent equipment, building, chimney, or other structure. Provisions should be made for drainage of each vent pipe so that liquid cannot accumulate in the vent.

5.3.3 MULTIPLE-VALVE DISPOSAL SYSTEM

For disposal to a flare or to a remote atmospheric vent stack, combining the discharges from a number of pressure relief or depressuring valves is usually economical. The specific arrangement of the headers and the routing of piping for the multiple-valve system is normally a question of minimizing investment. This requires taking into consideration the system loads, the back-pressure limitations, the requirement for special materials, and other design parameters discussed in preceding sections.

In a multiple-valve system that services a single unit—from the standpoint of economics, safety, or other pertinent factors—it is frequently desirable to isolate certain pressure relief or depressuring streams. This involves one or more of the following situations:

a. The occurrence of corrosive materials.
b. Significant differences in the pressure levels of the equipment connected to the system.
c. Pressure relief or depressuring streams that may subject piping to abnormally high or low temperatures.
d. Reactive materials (See 4.2.1).

In defining the header arrangements, consideration should be given to any requirements for separate shutdowns or separate maintenance on the protected equipment. It is usually not advisable to route pressure relief valve headers from one operating area through another area where major maintenance shutdowns are performed separately. Furthermore, it is usually advisable to be able to isolate headers that serve separate process areas from the disposal system, rather than to be required to isolate individual pressure relief valves within a common process area.

Multiple pressure-relief valve disposal systems that handle combustible vapors should not be used for venting air or steam during the start-up of process equipment. Any tie-ins of process vents to the multiple-valve system should be accompanied by strict instructions against using such tie-ins for venting air to avoid flammable mixtures in a system.

Most multiple-valve systems involve collecting pressure-relief valve discharges from various elevations. In general, laterals and headers should be arranged so that the outlet from each relief valve is not a liquid trap. All collecting piping should be considered subject to the inflow of liquid and should avoid liquid traps. If it is not practical to arrange the piping so that laterals and headers drain to a remote knockout drum, a local knockout drum is usually required. This local knockout drum normally need not be sized for efficient vapor-liquid separation at the maximum flow rate, but only for collecting the probable maximum liquid carryover from any valves that could discharge liquids. The use of traps or other devices with operating mechanisms should be avoided.

If the liquids to be handled include water, or oil with a relatively high pour point, a provision should be made to avoid solidification in the system. Likewise, the introduction of high-viscosity oils may require protection against low ambient temperature, particularly on instrument leads.

5.4 Design of Disposal System Components

5.4.1 PIPING

5.4.1.1 General

In general, the design of disposal piping should conform to the requirements of ASME B31.3. Installation details and criteria pertinent to pressure relief devices should conform to those specified in API Recommended Practice 520, Part II, and in the ASME Code.

5.4.1.2 Design of Relief Device Inlet Piping

The inlet piping includes all components and fittings that comprise the flow passage between the entrance to the vessel nozzle and the face of the inlet flange of the pressure relief valve or device. The first consideration is to comply with the mandatory requirements of the ASME Code by providing full inlet area and evaluating the reduction in relief valve capacity caused by any rupture disk devices, block valves, or other components.

The inlet system should be self-draining and designed to prevent excessive pressure loss, which causes chattering with consequent reduction of flow and damage to pipe joints and seating surfaces. Pilot-operated valves may require consideration of the location of the pressure pickup to ensure proper sensing for stable operation of the main valve where inlet pressure loss is excessive. API Recommended Practice 520, Part II, and nonmandatory requirements of the ASME Code limit the total pressure drop due to nonrecoverable losses to 3 percent of the set pressure. Pressure drop through the burst rupture disk should be taken into account when calculating the inlet losses to the pressure relief valve. The pressure relief valve should also be located as close to the source of pressure as is practicable, and oversizing should be avoided.

Section 2 of API Recommended Practice 520, Part II, provides information on the design of inlet piping for pressure relief devices. Several sizing methods have been developed that minimize inlet piping calculations in most cases and allow the designer to quickly identify marginal situations [1,2].

Note: Numbers in brackets correspond to references in 5.6.

In addition to flow considerations, the vessel nozzle and other inlet piping should be designed to withstand thermal loadings, reaction forces resulting from valve operation, vibration, dead weight, and externally applied loadings.

The strength of the inlet piping is less than that of the valve because the inlet piping has a smaller section modulus. Any moments created by loads applied to the outlet flange and by the reactive force of the discharging fluid will transmit bending stresses and rotational forces to the inlet piping. Design for reactive force is discussed in API Recommended Practice 520, Part II.

5.4.1.3 Design of Relief Device Discharging Piping

5.4.1.3.1 The basic criterion for sizing the discharge piping and the relief manifold is that the back pressure (which may exist or be developed at any point in the system) not reduce the relieving capacity of any of the pressure-relieving devices below the amount required to protect the corresponding vessels from overpressure. Thus, the effect of superimposed or built-up back pressure on the operating characteristics of the valves should be carefully examined. The discharge piping system should be designed so that the built-up back pressure caused by the flow through the valve under consideration does not reduce the capacity below that required of any pressure relief valve that may be relieving simultaneously. Where conventional safety relief valves are used, the relief manifold system should be sized to limit the built-up back pressure to approximately 10 percent of the set pressure of each pressure relief valve that may be receiving concurrently. Additionally, the effect of superimposed back-pressure from other valves upon the set pressure should be considered.

With balanced pressure relief valves, higher manifold pressures may be used. These balanced valves—bellows, piston, or pilot operated with the pilot vented independently—function independently of back pressure; however, their capacity begins to decrease when the back pressure exceeds 30 percent to 50 percent of the set pressure, in pounds per square inch absolute (kilopascals absolute). Refer to API Recommended Practice 520, Part I, and to manufacturer's curves for the effects of this back pressure. Additionally, the back pressure should not exceed the rating tabulated in API Standard 526, which may be lower than the outlet flange rating.

When discharge manifolds and relief headers are sized, the relief contingency that produces the greatest back pressure should 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 instrument air failure.

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.

In addition to the back-pressure criterion, the determination of the flow rate to be considered forms the basis for discharge-line sizing. In general, laterals and tailpipes from individual devices are sized based on the rated capacity of the device, which is consistent with the value used to size the inlet piping (see API Recommended Practice 520, Part II). Common header systems and manifolds in multiple-device installations are generally sized based on the worst-case cumulative required capacities of all devices that may reasonably be expected to discharge simultaneously in a single overpressure event (in other words, for certain scenarios, it may be appropriate to assume some level of favorable instrument and/or operation response). See API Recommended Practice 520, Part I, for discussion and determination of rated capacities. Causes of overpressure events are discussed in Section 2 of this recommended practice; required relieving capacities for various events are discussed in Section 3.

Simple rules cannot be expected to cover all installations. Good engineering judgment should be applied to select the flow basis most appropriate to each case. (See the ASME Code, Section VIII, Division I, Appendix M-8.)

In designing vapor depressuring systems, precise pressure drop calculations are usually not necessary. The only limits on built-up back pressure, in addition to those mentioned above, are as follows:

a. The ratings of fittings exposed to back pressure should not be exceeded.
b. The source that might reasonably be depressured concurrently should be capable of entering the header when its depressuring valve is opened.
c. Back-flow from the header into any connected process should be avoided.

When the maximum vapor-relieving requirement has been established and the maximum allowable header back pressure has been defined (as determined by the type of valves in the system and the applicable code requirements), the selection of line size is then reduced to fluid flow calculations. Several methods can be used to calculate the size of discharge piping when the flow conditions are known. These range from treating the flow as isothermal, with appropriate allowances for kinetic energy effects, to the more rigorous solutions afforded by the adiabatic approach. A number of methods listed in 6.6 should permit the user to select the method best suited to his needs. In the absence of any preference, the following methods are recommended.

5.4.1.3.2 Vapor flow in relief discharge piping is characterized by rapid changes in density and velocity; consequently, the flow should be rated 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 stepwise manner for each pipe segment of constant diameter. The isothermal flow equation based on inlet pressure [3] is as follows in Equation 21:

$$\frac{fL}{D} = \frac{1}{M_1^2}\left[1-\left(\frac{P_2}{P_1}\right)^2\right] - \ln\left(\frac{P_1}{P_2}\right)^2 \qquad (21)$$

The equation can be transposed to the following (Equation 22) for outlet pressure:

$$\frac{fL}{D} = \frac{1}{M_2^2}\left[\left(\frac{P_1}{P_2}\right)^2\right]\left[1-\left(\frac{P_2}{P_1}\right)^2\right] - \ln\left(\frac{P_1}{P_2}\right)^2 \qquad (22)$$

Where:

f = Moody friction factor.
L = equivalent length of pipe, in feet (meters).
D = pipe inside diameter, in feet (meters).
M_1 = Mach number at pipe inlet.
M_2 = Mach number at pipe outlet.
P_1 = pipe inlet pressure, in pounds per square inch absolute (kilopascals absolute).
P_2 = pipe outlet pressure, in pounds per square inch absolute (kilopascals absolute).

The outlet Mach number is given by Equation 23 and Equation 24:

$$M_2 = 1.702 \times 10^{-5}\left(\frac{W}{P_2 D^2}\right)\left(\frac{ZT}{kM_w}\right)^{0.5} \qquad (23)$$

Where:

W = gas flow rate, in pounds per hour (kilograms per hour).
Z = gas compressibility factor.
T = absolute temperature, in degrees Rankine (Kelvin).
M_w = gas molecular weight.
k = ratio of specific heats (Cp/Cv)

In metric units:

$$M_2 = 3.23 \times 10^{-5}\left(\frac{W}{P_2 D^2}\right)\left(\frac{ZT}{kM_w}\right)^{0.5} \qquad (24)$$

Both graphical and computerized methods have been developed for solving Equations 21 and 22 and calculating pipe inlet pressure [3, 4]. Figure 18 is a typical graphical representation of Equation 21. 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 those cases, a check should be made to see if the flow is critical. The critical pressure at the pipe outlet can be determined by setting $M_2 = 1.0$ (sonic flow) in Equation (23) as follows in Equation 25:

$$P_{Critical} = 1.702 \times 10^{-5}\left(\frac{W}{D^2}\right)\left(\frac{ZT}{kM_w}\right)^{0.5} \qquad (25)$$

In metric units Equation 26:

$$P_{Critical} = 3.23 \times 10^{-5}\left(\frac{W}{D^2}\right)\left(\frac{ZT}{kM_w}\right)^{0.5} \qquad (26)$$

Where:

$P_{Critical}$ = critical pressure, in pounds per square inch absolute (kilopascals absolute).

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 21 with P_2 equal to the critical pressure.

5.4.1.3.3 A rapid solution for sizing depressuring lines is offered below, using the method developed by Lapple [5]. This method employs a theoretical critical mass flow based on an ideal nozzle and adiabatic flow conditions and assumes a known upstream low velocity source pressure. The mass flux, where vapor $k = C_p/C_v = 1.00$, can be determined using Equation 27.

$$G_{Ci} = 12.6\ P_1\left(\frac{M_w}{ZT_1}\right)^{0.5} \qquad (27)$$

In metric units (Equation 28):

$$G_{Ci} = 6.7P_1\left(\frac{M_w}{ZT_1}\right)^{0.5} \tag{28}$$

Where:

G_{ci} = critical mass flux, in pounds per second per square foot (kilograms per second per square meter).
P_1 = pressure at the upstream low velocity source, (see Figure 19) in pounds per square inch absolute (kilopascals absolute).
M_w = molecular weight of the vapor.
T_1 = upstream temperature, in degrees Rankine (Kelvin).
Z = compressibility factor.

The compressibility factor should be taken at flow conditions and, thus, will change as the fluid moves down the line with resulting pressure drop. A stepwise calculation may be employed to allow for this variation. An accurate solution using this method would be tedious, but sufficiently accurate results can usually be obtained by performing the calculation over relatively large increments of pipe lengths, using an average compressibility factor over those lengths. Regardless of which equation is used, actual mass flux (G) is a function of critical mass flux (G_{ci}), frictional resistance (N), and the ratio of downstream to upstream pressure. These relationships are plotted in Figure 19. (Similar charts for adiabatic cases with ratios of specific heats of 1.4 and 1.8 have been developed by Lapple [5].) In the area below the diagonal line in Figure 19, the ratio G:G_{ci} remains constant, which indicates that sonic flow has been established. The total frictional resistance for use with the chart is expressed by Equation 29:

$$N = fL/D + \Sigma K \tag{29}$$

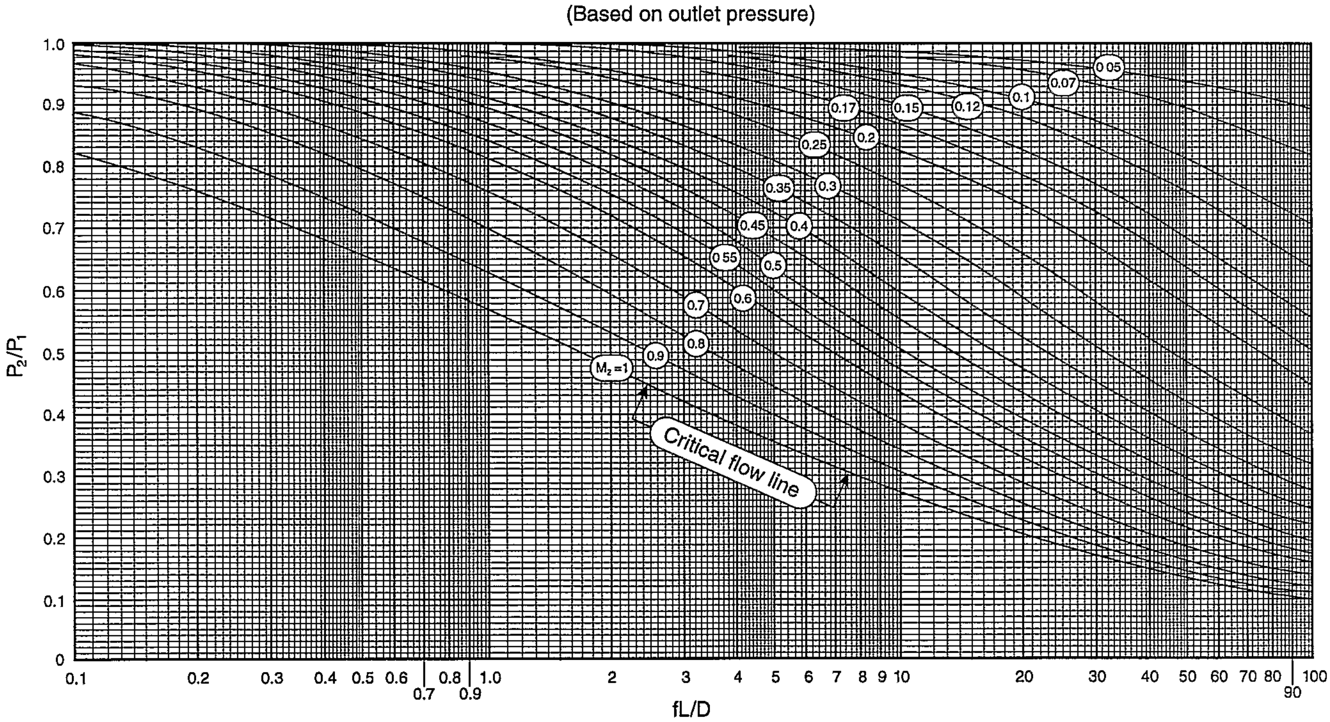
Where:

N = line resistance factor (dimensionless).
f = Moody friction factor.
L = actual length of the line, in feet (meters).
D = diameter of the line, in feet (meters).
K = resistance coefficients of fittings.

If a Fanning friction factor is used, $N = 4fL/D$.

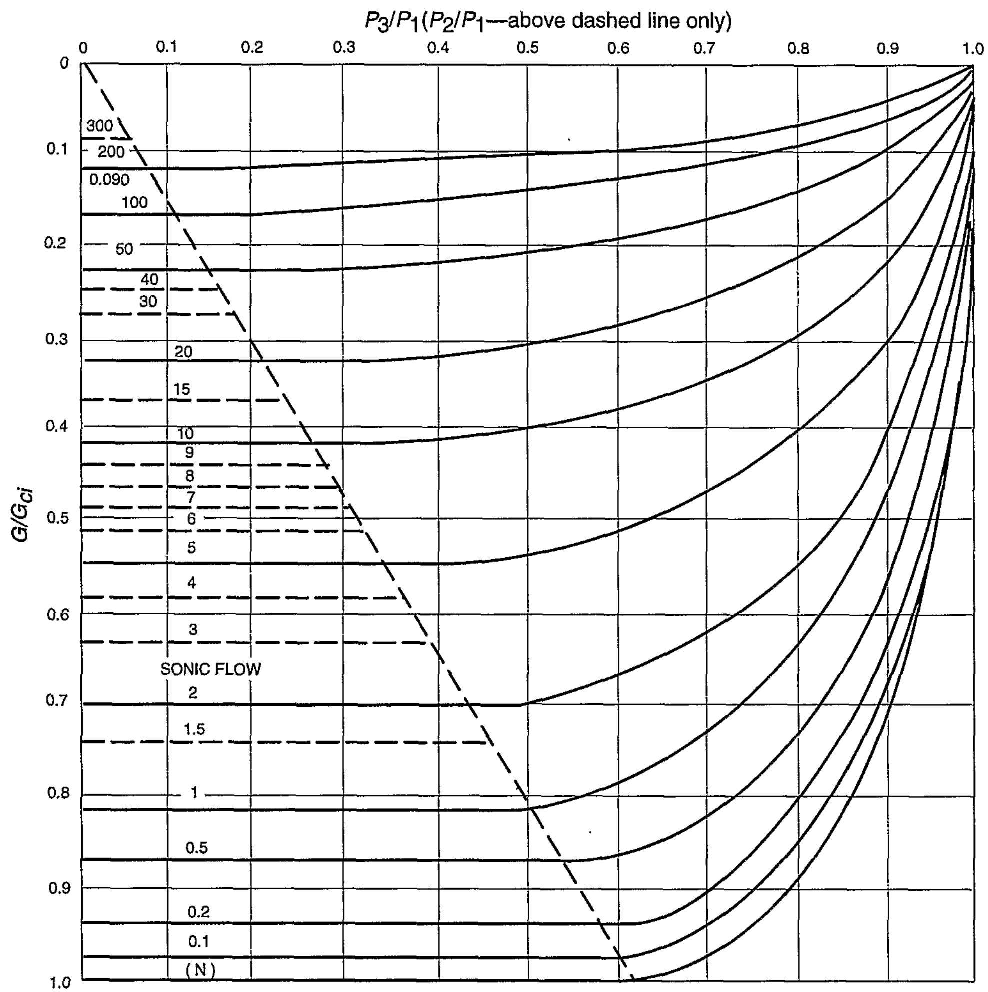
These methods assume that there are no enlargements or contractions in the piping, and no variation in the Mach number that results from a change in area. Coulter [6] provides a more comprehensive treatment of ideal gas flow through sudden enlargements and contractions.

Another method of calculating pressure drops for ideal gases at high velocities is the use of Fanno lines. Fanno lines are the loci of enthalpy/entropy conditions that result from adiabatic flow with friction in a pipe of constant cross-section. Fanno lines extend into both supersonic and subsonic flow zones. For relief disposal systems, only the subsonic



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

Figure 18—Isothermal Flow Chart



Note:
Equations 30 and 31 are based on adiabatic flow and a vapor $k=C_p/C_v$ approaching 1.00. For adiabatic flow with a k=1.40 vapor, the critical mass flux is 12.9% higher than calculated by Equations 30 and 31.

Note: Figure 19 is used as follows:

Calculate N (number of velocity heads)

$$N = (fL/D) + \Sigma K$$

Where:

f = Moody friction factor (dimensionless).
L = length of equivalent pipe, in feet (meters).
D = internal diameter of pipe, in feet (meters).
K = resistance coefficient of fittings from Table 11.

Calculate P_3 / P_1 or P_2 / P_1

Where:

P_3 = pressure in reservoir into which pipe discharges 14.7 pounds per square inch absolute (101 kilopascals absolute) with atmosphere discharge.
P_1 = pressure at upstream low velocity source, in pounds per square inch absolute (kilopascals absolute).
P_2 = pressure in the pipe at the exit or any point distance L downstream from the source, in pounds per square inch absolute (kilopascals absolute).

Calculate G_{ci}

$$G_{ci} = 12.6\ P_1 (M_w/zT_1)^{0.5} \quad (30)$$

In metric units,

$$G_{ci} = 6.7\ P_1 (M_w/zT_1)^{0.5} \quad (31)$$

Where:

G_{ci} = critical mass flux, in pounds per second per square foot (kilograms per second per square square meter).
M_w = molecular weight of the vapor.
T_1 = temperature at upstream low velocity source, in degrees Rankine (Kelvin).
z = compressibility factor.

From P_3 / P_1 or P_2 / P_1 and N, read G/G_{ci}

Calculate G in pounds per second per square foot (kilograms per square meter).

Calculate W [actual flow in pounds per second (kilograms per second)].

Where:

$W = G$ x cross-sectional area of pipe [in square feet (square meters)]

Figure 19—Adiabatic Flow of k=1.00 Compressible Fluids Through Pipes at High Pressure Drops

Table 11—Typical Values for Pipe Fittings

Fitting	K	Fitting	K
Globe valve, open	9.7	90-degree double-miter elbow	0.59
Typical depressuring valve, open	8.5	Threaded tee through run	0.50
Angle valve, open	4.6	Fabricated tee through run	0.50
Swing check valve, open	2.3	Lateral through run	0.50
180-degree close-threaded return	1.95	90-degree triple-miter elbow	0.46
Threaded or fabricated tee through branch	1.72	45-degree single-miter elbow	0.46
90-degree single-miter elbow	1.72	180-degree welded return	0.43
Welded tee through branch	1.37	45-degree threaded elbow	0.43
90-degree standard-threaded elbow	0.93	Welded tee through run	0.38
60-degree single-miter elbow	0.93	90-degree welded elbow	0.32
45-degree lateral through branch	0.76	45-degree welded elbow	0.21
90-degree long-sweep elbow	0.59	Gate valve, open	0.21
Rupture disk, subcritical flow	1.5[3]		

			d/d′		
Contraction or Enlargement	0	0.2	0.4	0.6	0.8
Contraction (ANSI)	-	-	0.21	0.135	0.039
Contraction (sudden)	0.5	0.46	0.38	0.29	0.12
Enlargement (ANSI)	-	-	0.90	0.50	0.11
Enlargement (sudden)	1.0	0.95	0.74	0.41	0.11

Notes:
1. Except for rupture disk data, this table is taken from the *Tube-Turn Catalogue and Engineering Data Book No. 211*, Chemetron Corp., Louisville, Kentucky.
2. *K* may vary with nominal pipe diameter. The values above are typical only.
3. $K = 1.5$ has been used successfully for design purposes. Other *K* values have also been reported [15]. A rupture disk manufacturer should be consulted for specific data, if required.

flow is of interest. The use of Fanno lines permits the calculation of pressure drops for ideal gases under adiabatic or isothermal flow conditions, with the total piping resistance as a parameter [7]. In general, the velocity in gas discharge piping cannot exceed the sonic or critical velocity limit. (This limit is shown on Lapple's charts [5] or on Fanno lines.)

In most disposal systems, the gases being handled are not ideal. For gases, deviations from the ideal are expressed as compressibility factors, which in turn are normally correlated with reduced pressure and reduced temperature. For hydrocarbon gases, the compressibility factor is less than 1.0 if the reduced temperature does not exceed 2.0 and the reduced pressure does not exceed about 6.0. Since most refinery pressure relief valve disposal systems fall within these limits, the compressibility of the gases will usually be less than 1.0. As long as compressibility is less than 1.0, the pressure drop calculated for an ideal gas will be larger than that calculated for the same gas incorporating the compressibility factor.

If one desires to make a rigorous calculation of pressure drop, including the effect of compressibility, an incremental or stepwise approach is usually required. Most nonideal gases also exhibit some deviations in specific heat ratio (k) or in the polytropic exponent (n) from the ideally constant values. Rigorous pressure drop calculations should also take into account these deviations.

For most applications, the pressure drops that are calculated assuming ideal gases under isothermal flow conditions will exceed those calculated by more rigorous procedures. In any design of a disposal system, the sizing of piping based on ideal gas flows under isothermal conditions will normally be adequate. However, for very high pressure or high- or low-temperature situations, the possible effects of deviations from ideality should be checked.

In any calculation method, the total frictional resistance should include the length of piping and the equivalent length of all fittings, valves, expansion or contraction losses, and any other flow resistances. The frictional resistance of fittings and some other items in the piping system can also be expressed in terms of K factors. Table 11 shows typical K factors for pipe fittings and for enlargements and contractions.

The friction factor, f, enters into all calculations of pressure drop. At high gas flow velocities, which usually prevail in the design of disposal systems, the friction factor approaches a constant number that depends only on pipe size and internal roughness. Table 12 shows the limiting friction factors for common sizes of steel pipe. The friction factors in Table 12 apply only at high gas velocities (that is, high Reynolds numbers).

For preliminary studies, it is often necessary to assume K's or an equivalent length of fittings, expansion loops, and the like. Based on actual layouts, these elements can add equivalent length equal to 100 percent or more of the geographical length of the pipe.

Where gas flow velocity in long runs of piping approaches the critical flow limit, it is often economical to increase the pipe size in steps or progressively along the run. In general, a

Table 12—Typical Factors for Clean Steel Pipe (Based on Equivalent Roughness of 0.00015 Feet)

Nominal Pipe Size (inches)	Moody Friction Factor (f)
NPS 2, Schedule 40	0.0195
NPS 3, Schedule 40	0.0178
NPS 4, Schedule 40	0.0165
NPS 6, Schedule 40	0.0150
NPS 8 × 1/4-inch wall	0.0140
NPS 10 × 1/4-inch wall	0.0135
NPS 12 × 1/4-inch wall	0.0129
NPS 14 × 1/4-inch wall	0.0126
NPS 16 × 1/4-inch wall	0.0123
NPS 20 × 1/4-inch wall	0.0119
NPS 24 × 1/4-inch wall	0.0115
NPS 30 × 1/4-inch wall	0.0110
NPS 36 × 1/4-inch wall	0.0107

Note: NPS = nominal pipe size. The above friction factors apply at high Reynolds numbers, namely, above 1×10^6 for NPS 24 and larger, scaling down to 2×10^5 for NPS 2.

calculation of pressure drop is required for each section of uniform size. The piping directly connected to a relief valve should not be smaller than the size of the outlet flange.

If the system includes mixed-phase fluids, the line sizing is more complex. Mixed-phase flow situations may have either a constant liquid/vapor ratio, as in discharges that carry entrained liquids, or a variable liquid/vapor ratio, as in flashing liquid discharges. In either case the pressure drop calculation usually involves an incremental or stepwise approach. The usual calculation procedure involves the following:

a. Evaluating the liquid/vapor ratio and the physical properties of each phase at given or assumed conditions of pressure, enthalpy, and flow.
b. Defining the flow pattern.
c. Calculating the vapor-phase flow pressure drop.
d. Calculating the two-phase flow pressure drop, which is commonly expressed as a function of the vapor-phase flow pressure drop.

Methods are available for flashing flow calculations (see 6.4 and 6.5).

From very limited information, the foregoing calculation method appears to give calculated pressure drops that are higher than those which would be observed under the specified conditions. The method may, therefore, be considered conservative with respect to the adequacy of sizing of piping.

5.4.1.3.4 The mechanical design of the disposal system warrants the same attention as that given to the design of piping systems that handle process fluids. The problems encountered in the design of discharge piping from pressure relief or depressuring valves are frequently more complex than those encountered in the design of a process system, since discharge piping may be subject to a greater range of temperature, pressure, and shock caused by the wide range of operating conditions. In addition, the disposal system may at one time or another contain any material handled in the process system.

The major stresses to which the discharge piping of a relieving system is subject are results of thermal expansion or contraction from the entry of cold or hot materials and thrust developed by the discharge fluid. In relieving systems that serve typical refinery process units, temperatures may range from well below zero to several hundred degrees. Designing for flexibility is more complicated than it is for process piping systems because thrust as well as thermal expansion must be controlled.

Most situations make it possible to maintain stress levels in relieving systems within allowable limits over the full temperature range by providing guides, anchors, and appropriate piping configurations.

Special attention to stresses is recommended where piping constructed of carbon steel may be cooled below its transition temperature. Cooling may be caused by the entry of cold materials or by autorefrigeration, which occurs when the pressure is reduced on low-boiling liquids. Reference should be made to ASME B31.3 for material specifications, allowable stresses, and impact test requirements for carbon steel piping materials that may be used for temperatures as low as -50°F (-46°C). Stress relieving of welded piping systems has proven beneficial as a supplementary precaution in reducing the risk of brittle fracture of carbon steel piping that may operate below its transition temperature. Where temperatures below -50°F (-46°C) are possible, the usual practice is to construct relief lines of materials that will exhibit ductile behavior at the minimum anticipated operating temperature.

The design of discharge piping requires careful analysis of the possible imposition of both thermal and mechanical stresses on the associated pressure relief valves. The stresses set up in the pressure relief valves may cause malfunction or leakage of the valves (see API Recommended Practice 520, Part II). Forces on the valve may be controlled by proper anchors, supports, and provisions for flexibility of discharge piping.

Discharge piping, which is supported by the outlet of the pressure relief valve instead of being supported separately, will induce stresses in associated pressure relief valves and inlet piping. Forced alignment of the discharge piping imposes similar stresses. Discharge piping, including short tail pipes, should be examined, supported, and carefully aligned as requirements dictate. Strains sufficient to cause mechanical failure usually occur first at the inlet piping; however, moments at much lower levels can cause serious malfunction and leakage of the pressure relief valve. Stresses may also be imposed on the disposal piping as a result of reaction forces created when the pressure relief valves are discharging. Provisions should be made for anchoring or restraining disposal lines related to these valves where analysis indicates that this is necessary. A formula for computing reactive loads due to the operation of pressure relief valves is given in API Recommended Practice 520, Part II.

Shock loading should also be considered in relief lines. Shock loading may result either from the sudden release of a compressible fluid into a multidirectional piping system or from the impact action of liquid slugs at points of change in direction. Reaction forces can occur at each change of direction in the piping.

5.4.1.3.5 The design of appropriate and adequate anchors, guides, and supports for a pressure-relieving discharge piping system is complex. There are several methods of calculating piping flexibility; reference should be made to ASME B31.3 for a background discussion. Once the range of relieving conditions to be handled is established, the problems are no different from those for most other piping systems, other than also having to consider thrust forces.

Experience has shown that carefully considered answers to the following questions are needed to permit the design of a satisfactory system of anchors, guides, and supports:

a. What are the probable combinations of relieving conditions that the manifold will need to handle? What sort of temperature ranges do these conditions impose, considering changes in the ambient temperature? What are the probable inlet conditions, in terms of thermal movement, when these reliefs occur?
b. What are the probable magnitude and sources of any liquid slugs?
c. Are there any valves that could release large volumes of high-pressure gas and produce shock loads? If so, where are they located?

In general, it is preferable to select anchor points so that header movements and the resultant forces and moments are not imposed on the bodies or the discharge piping of safety relief valves. Where valves discharge to the atmosphere, the tailpipe configuration should be checked for discharge reaction forces to ensure that it will not be overstressed.

5.4.1.3.6 Disposal-system piping should be self-draining toward the discharge end. Pocketing of discharge lines should be avoided. Where pressure relief valves handle viscous materials or materials that can solidify as they cool to ambient temperature, the discharge line should be steam-traced. A small drain pot or drip leg may be necessary at low points in lines that cannot be sloped continuously to the knockout or blowdown drum. The use of traps or other devices with operating mechanisms should be avoided.

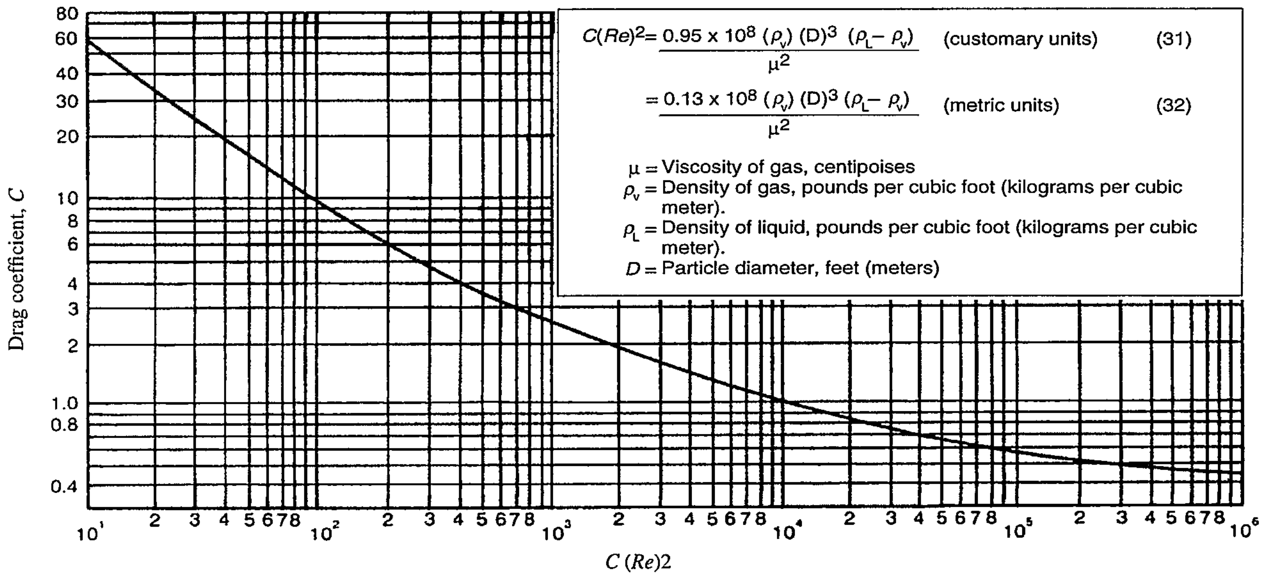
5.4.1.3.7 Many design details and features merit particular emphasis with respect to relieving systems. The following points are not to be taken as definitive or restrictive:

a. The laterals from individual relieving devices should normally enter a header from above. This tends to keep any liquids that may flow or develop in the header out of the laterals to each valve.
b. Laterals that lead from individual valves located at an elevation above the header should drain to the header. Locating a safety valve below the header elevation in closed systems should be avoided wherever possible. Laterals from individual valves that must be located below the header should be arranged to rise continuously to the top of the header entry point; however, means should be provided to prevent liquid accumulation on the discharge side of these valves.
c. A slope of ¼ inch in 10 feet (21 millimeters in 10 meters) is suggested for all laterals and headers, taking into account piping deflections between supports.
d. Where individual valves are vented to the atmosphere, an adequate drain hole [a nominal pipe size of ½ inch (NPS ¼) is usually considered suitable] should be provided at the low point to ensure that no liquid collects downstream of the valve. The vapor flow that occurs through this hole during venting is generally not considered significant, but each case should be checked to see if the drain connection should be piped to a safe location. Vapors escaping from the drain hole should not be allowed to impinge against the vessel shell, since accidental ignition of such vent streams can seriously weaken the shell.
e. The use of angle entry—an entry at 45 degrees (0.79 radian) or even 30 degrees (0.52 radian) to the header axis—for laterals is much more common in relieving systems than in most process piping systems. The two main reasons for this approach are (1) lower pressure drop (including velocity head losses), and (2) reduced reaction forces. Since laterals in relieving systems can often be sized at velocities approaching sonic, pressure losses or recoveries caused by velocity change can become a significant factor in system analysis. These densities can produce large reaction forces.
f. The use of valves to section the header system for maintenance or safety should be considered. Such valves should be provided with locking or sealing devices. Where valves cannot be justified, the provision for blinding should be studied. In locating sectioning valves or blinds, extreme caution should be exercised in their use to ensure that equipment which is operating is not isolated from its relieving system. If valves are used in the header system, they should be mounted so that they cannot fail in the closed position (an example would be a gate falling into its closed position).

5.4.2 DRUMS AND SEALS

5.4.2.1 Sizing a Knockout Drum

Sizing a knockout drum is generally a trial-and-error process. The first step is to determine the drum size required for liquid entrainment separation. Liquid particles will separate (a) when the residence time of the vapor or gas is equal to or greater then the time required to travel the available vertical height at the dropout velocity of the liquid particles, and (b)



Note: Refer to the section on particle dynamics in the *Chemical Engineers' Handbook* [9].

Figure 20—Determination of Drag Coefficient

when the vertical gas velocity is sufficiently low to permit the liquid dropout to fall. This vertical height is usually taken as the distance from the liquid surface. The vertical velocity of the vapor and gas should be low enough to prevent large slugs of liquid from entering the flare. Since the flare can handle small liquid droplets, the allowable vertical velocity in the drum may be based on that necessary to separate droplets from 300 micrometers to 600 micrometers in diameter. The dropout velocity [9] of a particle in a stream is calculated using Equation 30 as follows:

$$U_c = 1.15 \frac{\sqrt{gD(\rho_l - \rho_v)}}{\rho_v(C)} \qquad (30)$$

Where:

U_c = dropout velocity, in feet per second (meters per second).

g = acceleration due to gravity, at 32 feet per second per second (9.8 meters per second per second).

D = particle diameter, in feet (meters).

ρ_l = density of the liquid at operating conditions, in pounds per cubic foot (kilograms per cubic meter).

ρ_v = density of the vapor at operating conditions, in pounds per cubic foot (kilograms per cubic meter).

C = drag coefficient (see Figure 20).

This basic equation is widely accepted for all forms of entrainment separation.

The second step in sizing a knockout drum is to consider the effect any liquid contained in the drum may have on reducing the volume available for vapor/liquid disengagement. This liquid may result from (a) condensate that separates during a vapor release, or (b) liquid streams that accompany a vapor release. The volume occupied by the liquid should be based on a release that lasts 20 to 30 minutes. Any accumulation of liquid retained from a prior release (from pressure relief valves or other sources) should be added to the liquid indicated in Items a and b to determine the available vapor-disengaging space. However, it would usually not be necessary to consider the following volumes relative to vapor disengaging in the following situation: that in which the knockout drum is used to contain large liquid dumps from pressure relief valves from other sources where there is no significant flashing, and the liquid can be removed promptly.

Economics of vessel design in selecting a drum size, which may influence the choice between a horizontal and a vertical drum. When large liquid storage is desired and the vapor flow is high, a horizontal drum is often more economical.

Although horizontal and vertical knockout drums are available in many designs, the differences are mainly in how the path of the vapor is directed. The various designs include the following:

a. A horizontal drum with the vapor entering one end of the vessel and exiting at the top of the opposite end (no internal baffling).

b. A vertical drum with the vapor inlet nozzle on a diameter of the vessel and the outlet nozzle at the top of the vessel's vertical axis. The inlet stream should be baffled to direct the flow downward.

c. A vertical vessel with a tangential nozzle.

d. A horizontal drum with the vapor entering at each end on the horizontal axis and a center outlet.

e. A horizontal drum with the vapor entering in the center and exiting at each end on the horizontal axis.

f. A combination of a vertical drum in the base of the flare stack and a horizontal drum upstream to remove the bulk of the liquid entrained in the vapor. This combination permits the use of larger values for the numerical constant in the velocity equation.

The following sample calculations have been limited to the simplest of the designs, Items a and b. The calculations for Items d and e would be similar, with one-half the flow rate determining one-half the vessel length. The normal calculations would be used for Item c and will not be duplicated here.

The following conditions should be assumed:

a. A single contingency results in the flow of 200,000 pounds per hour (25.2 kilograms per second) of a fluid with a liquid density of 31 pounds per cubic foot (496.6 kilograms per cubic meter) and a vapor density of 0.18 pound per cubic foot (2.9 kilograms per cubic meter), both at flowing conditions.

b. The pressure is 2 pounds per square inch gauge (13.8 kilopascals gauge), and the temperature is 300°F (149°C).

c. The viscosity of the vapor is 0.01 centipoise.

d. The fluid equilibrium results in 31,000 pounds per hour (3.9 kilograms per second) of liquid and 169,000 pounds per hour (21.3 kilograms per second) of vapor.

In addition, 500 gallons (1.89 cubic meters) of storage for miscellaneous drainings from the units is desired. The schematic in Figure 21 applies. The droplet size selected as allowable is 0.000984 foot (300 micrometers) in diameter.

The vapor rate, R_v, in actual cubic feet per second (cubic meters per second), is determined as follows:

$$R_v = \frac{169,000 \text{ pounds per hour}}{(3600 \text{ seconds per hour})(0.18 \text{ pounds per cubic feet})} = 261$$

In metric units,

$$R_v = \frac{21.3 \text{ kilograms per second}}{2.9 \text{ kilograms per cubic meter}} = 7.34$$

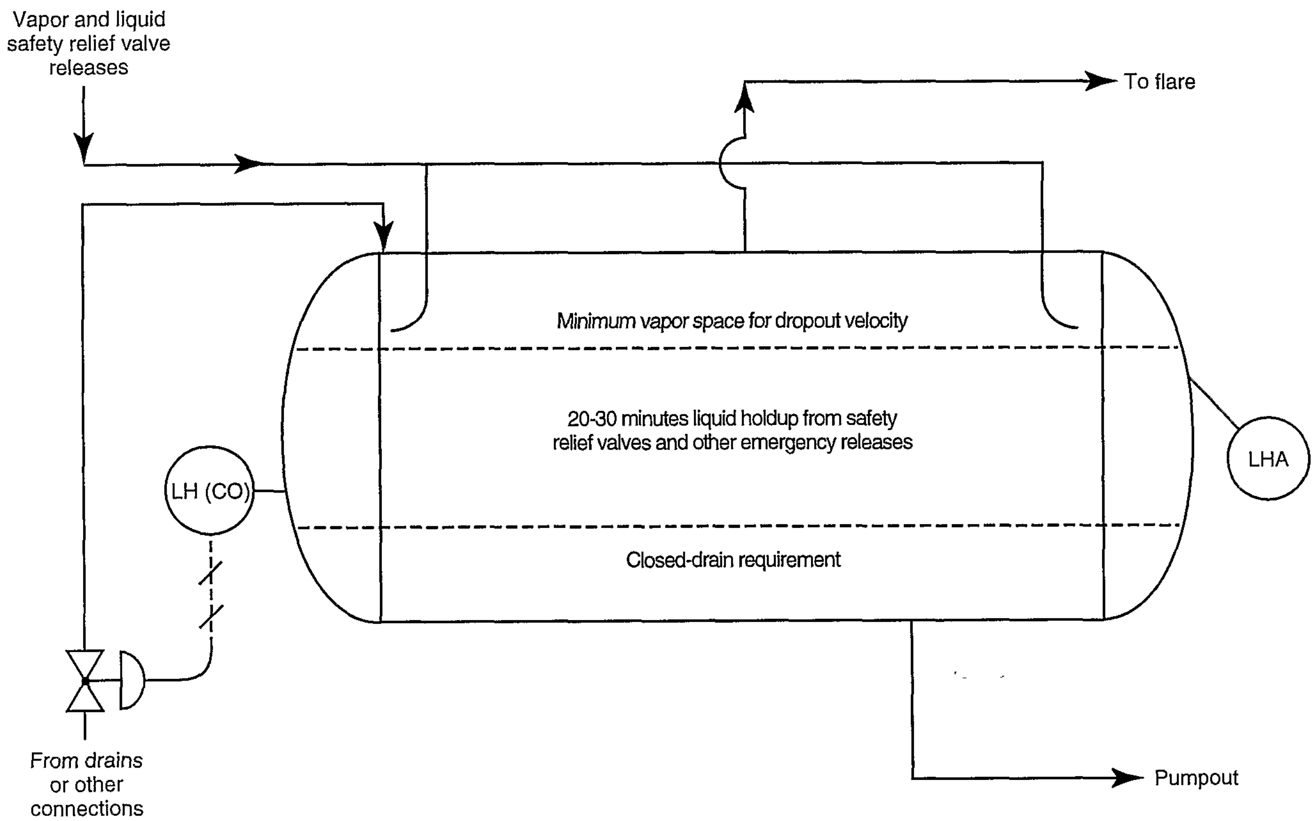


Figure 21—Flare Knockout Drum

The drag coefficient, C, is determined from Figure 20 using the following equation:

$$C(Re)^2 = \frac{0.95 \times 10^8 (0.18)(0.000984)^3 (31 - 0.18)}{(0.01)^2} = 5,021$$

In metric units:

$$C(Re)^2 = \frac{0.13 \times 10^8 (2.9)(300 \times 10^{-6})^3 (496.6 - 2.9)}{(0.01)^2} = 5,025$$

From Figure 20, $C = 1.3$.

The dropout velocity, U_c, in feet per second (meters per second), is calculated as follows:

$$U_c = 1.15\left[\frac{(32.2)(0.000984)(31 - 0.18)}{(0.18)(1.3)}\right]^{0.5} = 2.35$$

In metric units:

$$U_c = 1.15\left[\frac{(9.8)(300 \times 10^6)(496.6 - 2.9)}{(2.9)(1.3)}\right]^{0.5} = 0.71$$

A horizontal vessel with an inside diameter, D_i, and a cylindrical length, L, should be assumed. This gives the following total cross-sectional area, A_t:

$$A_t = \frac{\pi}{4}(D_i)^2 \qquad (33)$$

Liquid holdup for a 30-minute release from the single contingency, in addition to the slop and drain volume, is desired. The volume in the heads is neglected for simplicity. The liquid holdup required, A_{LI} in square feet (square meters), is calculated as follows:

a. The slop and drain volume of 500 gallons (1.89 cubic meters) will occupy a bottom segment as follows:

$$A_{L1} = \left(\frac{500}{7.48 \text{ (gallons per cubic foot)}}\right)\left(\frac{1}{L}\right) \qquad (34)$$

In metric units:

$$A_{L1} = (1.89)\left(\frac{1}{L}\right) \qquad (35)$$

b. A total of 31,000 pounds per hour (3.9 kilograms per second) of condensed liquids with a density of 31 pounds per cubic foot (496.6 kilograms per cubic meter) accumulated for 30 minutes will occupy a cross-sectional segment (see Item a) as follows:

$$A_{L2} = \left(\frac{31,000}{31}\right)\left(\frac{30}{60 \text{ minutes per hour}}\right)\left(\frac{1}{L}\right) \qquad (36)$$

In metric units:

$$A_{L2} = \left(\frac{3.9}{496.6}\right)(60 \text{ seconds per minute})(30)\left(\frac{1}{L}\right) \qquad (37)$$

The cross-sectional area remaining for the vapor flow is expressed as follows:

$$A_v = A_t - (A_{LI} + A_{L2}) \qquad (38)$$

The vertical depths of the liquid and vapor spaces are determined using standard geometry, where h_{LI} = depth of slops and drains; $h_{LI} + h_{L2}$ = depth of all liquid accumulation; and h_v = remaining vertical space for the vapor flow.

The total drum diameter is calculated using the following equation:

$$h_t = h_{L1} + h_{L2} + h_v \qquad (39)$$

The adequacy of the vapor space is verified as follows: The vertical drop available for liquid dropout is equal to h_v, in inches (centimeters). The liquid dropout time, in seconds, is determined as follows:

$$\theta = \left(\frac{h_v}{12 \text{ inches per foot}}\right)\left(\frac{I}{U_c \text{ feet per second}}\right) \qquad (40)$$

In metric units:

$$\theta = \left(\frac{h_v}{100 \text{ centimeters per meter}}\right)\left(\frac{I}{U_c \text{meters per second}}\right) \qquad (41)$$

The velocity of N vapor passes, in feet per second (meters per second), based on one vapor pass, is determined as follows:

$$U_v = \left(\frac{260 \text{ cubic feet per second}}{N \text{ vapor passes}}\right) \times \left(\frac{1}{A_v \text{ square feet}}\right) \qquad (42)$$

In metric units:

$$U_v = \left(\frac{7.34 \text{ cubic meters per second}}{N \text{ vapor passes}}\right) \times \left(\frac{1}{A_v \text{ square meters}}\right) \qquad (43)$$

The drum length required, in feet (meters), is determined as follows:

$$L_{min} = (U_v \text{ feet per second}) \times (\theta \text{ seconds})(N \text{ vapor passes}) \qquad (44)$$

In metric units:

$$L_{min} = (U_v \text{ meters per sec.}) \times (\theta \text{ seconds})(N \text{ vapor passes}) \qquad (45)$$

Table 13—Optimizing the Size of a Horizontal Knockout Drum (Customary Units)

Trial No.	Assumed Drum Inside Diameter, D_i (feet)	Assumed Drum Cylindrical Length, L (feet)	Cross-Sectional Area (square feet) A_t	A_{L1}	A_{L2}	A_v	Vertical Depth of Liquid and Vapor Spaces (inches) h_{L1}	$h_{L1}+h_{L2}$	h_v	h_t	Liquid Dropout Time, 0 (seconds)	Vapor Velocity, u_v (ft. per sec.)	Required Drum Length, L_{min} (feet)
1	8.0	19.0	50.26	3.25	26.32	20.69	11.75	55.0	41.0	96	1.45	12.73	18.5
2	7.5	20.5	44.17	3.26	24.39	16.52	11.50	54.0	36.0	90	1.28	15.74	20.2
3	7.0	22.5	38.48	2.97	22.23	13.28	11.00	52.3	31.7	84	1.13	19.58	22.1
4	6.5	25.0	33.18	2.67	20.00	10.51	10.5	50.4	27.6	78	0.98	24.74	24.3

Note: The data in this table are in accordance with the example given in text for one pass vapor flow. It can be concluded from this table that (a) all of the drum sizes above would fulfill the design requirements, (b) the most suitable drum size should be selected according to the design pressure, material requirements, and corrosion allowance as well as layout, transportation, and other considerations, and (c) the choice of two-pass flow, as shown in Figure 21, is optional.

Table 14—Optimizing the Size of a Horizontal Knockout Drum (SI Units)

Trial No.	Assumed Drum Inside Diameter, D_i (m)	Assumed Drum Cylindrical Length, L (m)	Cross-Sectional Area (m^2) A_t	A_{L1}	A_{L2}	A_v	Vertical Depth of Liquid and Vapor Spaces (cm) h_{L1}	$h_{L1}+h_{L2}$	h_v	h_t	Liquid Dropout Time, 0 (seconds)	Vapor Velocity, u_v (m per sec.)	Required Drum Length, L_{min} (m)
1	2.44	5.79	4.67	0.33	2.45	1.89	30	140	104	244	1.45	3.9	5.6
2	2.29	6.25	4.10	0.30	2.27	1.53	29	137	91	229	1.28	4.8	6 2
3	2.13	6.86	3.57	0.28	2.07	1.22	28	133	81	213	1.13	6.0	6.7
4	1.98	7.62	3.08	0.25	1.86	.98	27	128	70	198	0.98	7.5	7.4

Note: The data in this table are in accordance with the example given in text for one pass vapor flow. The values in this table are rounded-off conversions of the values in Table 13. It can be concluded from this table that (a) all of the drum sizes above would fulfill the design requirements, (b) the most suitable drum size should be selected according to the design pressure, material requirements, and corrosion allowance as well as layout, transportation, and other considerations, and (c) the choice of two-pass flow, as shown in Figure 21, is optional.

L_{min} must be less than or equal to the above assumed cylindrical drum length, L; otherwise, the calculation must be repeated with a newly assumed cylindrical drum length.

Tables 13 and 14 summarize the preceding calculations for 1 pass for horizontal drums with various inside diameters to determine the most economical drum size. Drum diameters in 6-inch (15-centimeter) increments are assumed, in accordance with standard head sizes.

If a vertical vessel is considered, the vapor velocity is equal to the dropout velocity, which is 2.35 feet per second (0.71 meter per second). The required cross-sectional area, A_{cs}, of the drum, in square feet (square meters), is determined as follows:

$$A_{cs} = \frac{260 \text{ cubic feet per second}}{2.35} = 110.6 \text{ square feet} \quad (46)$$

In metric units:

$$A_{cs} \doteq \frac{7.34 \text{ cubic meters per second}}{0.71} = 10.3 \text{ square meters} \quad (47)$$

The drum diameter, D, in feet (meters), is determined as follows:

$$D = \sqrt{(110.6 \text{ square feet})(4/\pi)} = (11.9 \text{ feet}) \quad (48)$$

In metric units,

$$D = \sqrt{(10.3 \text{ square meters})(4/\pi)} = 3.6 \text{ meters} \quad (49)$$

5.4.2.2 Sizing a Seal Drum

In sizing a seal drum, the maximum exit back pressure allowable in the vent header should first be determined (see 5.4.1.3). This back pressure will set the maximum distance, h (see Equations 50 and 51), that the inlet pipe is submerged. The vessel-free area for gas flow above the liquid level should be at least three times the inlet pipe cross-sectional area to prevent surges of gas flow to the flare.

$$h = \frac{144P}{\rho} \quad (50)$$

In metric units:

$$h = \frac{102P}{\rho} \quad (51)$$

Where:

h= distance, in feet (meters).

P= maximum header exit pressure, in pounds per square inch gauge (kilopascals gauge).

ρ= sealing liquid density, in pounds per cubic foot (kilograms per cubic meter).

The area for the gas above the liquid surface should equal at least that of a circle with a diameter, D, that is equal to $2d$, where d equals the diameter of the inlet gas pipe. This can be derived assuming a vertical vessel that has an internal area equal to $(\pi D^2)/4$ and an inlet pipe with an area equal to $(\pi d^2)/4$. The annular area is $(\pi/4)(D_2 - d_2)$. Since the suggested ratio is 1:3:

$$(D^2 - d^2) = 3d^2 \text{ or } D^2 = 4d^2 \text{ and } D = 2d \quad (52)$$

The height, H, of the vapor space in a vertical seal drum should be approximately 0.5 to 1.0 times the diameter, D, to provide disengaging space for entrained seal liquid. A minimum dimension of 3 feet (1 meter) is suggested.

In some situations, special considerations may affect the size of a seal drum. One such occurrence would be a large flow of hot vapor into the vent header. The vacuum created when this vapor cools may pull sufficient liquid into the header to break the seal, thus allowing air to be drawn into the flare system. To prevent this occurrence, the inlet line should be constructed to form a vacuum leg. The total vertical height of the inlet leg at the seal drum will be determined by the maximum vacuum expected. The volume of liquid in the inlet line at the maximum vacuum should be obtained from the seal drum. This requirement may necessitate an increase in the size of the drum.

5.4.2.3 Sizing a Quench Drum

The sizing criteria for quench drums depend so closely on the design of the drum internals, the liquid loading, the amount of condensation, and other features specific to the particular installation that no generally meaningful sizing rules can be established. A common criterion is to reduce the temperature of the stream so that the exit liquid and vapor will not exceed the range of 150°F to 200°F (66°C to 93°C) and, typically, to assume that no more than 40 percent to 50 percent of the liquid fed will be vaporized. Scheiman's articles [10, 11] cover the sizing criteria for one type of internals frequently used in this service.

5.4.2.4 Design Details

Design details that may be applicable to seal drums include the following:

a. Antiswirl or antivortex baffles should be used on the liquid outlet lines.
b. Internally extended liquid outlet nozzles should be used so that sediment will settle out in the drums, not in low spots in the lines.
c. Antifreeze, siphon-type drains should be used for normal manual drains where a freezing problem exists.
d. Provisions should be made for water leg or boot and water removal if three-phase separation is expected.
e. Handholes (4-inch to 8-inch nozzles) should be present on the bottom of the drum to permit thorough cleaning. These nozzles should have 1½-inch or 2-inch nominal pipe size (NPS 1½ or NPS 2) valves in the blind flange to permit complete draining of the vessels before opening.
f. Allowance should be made for blinding, venting, purging (steaming), and preparing the vessel for entry where manways are provided.
g. Provisions should be made for heating the contents of the vessel where cold weather, autorefrigeration, viscous, or congealing liquids may introduce problems. If internal coils are needed, consideration should be given to coil drainage. The coils should have a generous corrosion allowance and adequate support to prevent mechanical failure. Because one side of the vessel shell cannot be inspected, heating jackets that use the vessel shell as one wall should be avoided.
h. Most knockout drums and seal drums will be operating at relatively low pressures. To ensure sound construction, a minimum design pressure of 50 pounds per square inch gauge (345 kilopascals gauge) is suggested. The vessel should be designed according to the specifications of the ASME Code, although a code stamp may not be justified if the vessel pressure can never exceed 15 pounds per square inch gauge (103 kilopascals gauge). A vessel with a design pressure of 50 pounds per square inch gauge (345 kilopascals gauge) should not rupture if an explosion occurs. Stoichiometric hydrocarbon-air mixtures can produce peak explosion pressures on the order of seven to eight times the absolute operating pressure. Most flare seal drums operate in the range of 0 pounds per square inch gauge to 5 pounds per square inch gauge (0 kilopascals gauge to 34 kilopascals gauge).
i. In designing vessel nozzles, attachments, supports, and internals, one should consider shock loadings that result from thermal effects, slugs of liquid, or gas expansion.
j. Try cocks for liquid-level detection may be desirable in addition to or instead of level gauges.
k. Facilities to provide for continuous removal or intermittent manual skimming of hydrocarbons that may accumulate should be considered. Constant skimming by means of continuous addition of seal liquid and overflow to drain can be used. Provisions for periodically raising the level of the seal

liquid to force lighter fluid out through a skimmer connection are permissible. The designer is cautioned to review the proposed system to ensure that lighter material cannot build up to the point at which a false (nondesign) sealing effect is established.

l. Instrumentation components should be the simplest and most rugged available and should be easily maintained (externally mounted and valved). The use of seals instead of valves and of valves instead of traps is preferred, primarily because of the nature of the materials handled and the conditions under which these components must operate. On-off valves with large flow areas are frequently preferred to small-passage throttling valves.

m. Provisions for establishing and maintaining an adequate seal level are recommended.

n. Where corrosion exists at the seal fluid/vapor interface, an adequate corrosion allowance should be used. Such corrosion can occur even in hydrocarbon systems that use water as the seal fluid or in areas where water can collect at low points in the system.

In addition to these common details, some details are specific to the various types of equipment. Knockout drums may be of the horizontal or vertical type; and they should be provided with a pump or draining facilities and instrumentation to remove the accumulated liquids to a tank, sewer, or other location. The actual type of disposal used will depend on the characteristics and hazards associated with the liquids removed. The design of liquid-removal facilities for a knockout drum depends on the size of the vessel and the extent or probability of liquid occurring in the system.

In the simplest system, the vessel may have only a manually operated drain valve and a liquid-level sight glass for reference. A liquid-removal pump is frequently used. A high-level alarm, a manual starting switch, and an automatic shutoff switch to the pump motor are generally provided. More elaborate arrangements may also have high- and low-level alarms and level controls that operate a motorized drain valve or a liquid-removal pump. Where a drain valve is used, the on-off type is more common; however, a throttling type may be employed. The high liquid level in the drum is limited so that the cross-sectional area of vapor passage is not reduced. The low liquid limit is established to prevent vapor from entering the liquid-removal system.

The seal drum should be located between the stack and the other header drums and as close to the flare stack as is practical. A variation of a seal drum is often incorporated into the base of the flare stack where the flare line enters the stack. The configuration of a seal drum may consist of a vessel partially filled with a sealing liquid (for example, water).

The problem of surging in seal drums can be minimized by the use of slots or V notches on the end of the dip pipe so that increasing flow area is provided as the gas flow increases, utilizing a principle similar to that involved in the design of a bubble cap. Occasionally, one might want to increase the inlet line inside the drum to reduce gas velocity and allow enough circumference for the slots. The desired sealing level may also be maintained by means of an automatic controller operating on the liquid supply line. Low- and high-level alarms are sometimes used for warning in case the liquid is not maintained within the desired levels. An adequately-sized drain line with shutoff valves should be provided for removing the liquid.

The quenching fluid in a quench drum or tower may be water, gas oil, or another suitable liquid. The used liquid collects in the bottom of the drum for subsequent removal. The requirement for quenching may be monitored by a temperature or flow switch in the relief discharge header. The liquid in the bottom of the drum or tower may be automatically controlled. Removed liquid may be cooled and recycled, dispatched to sewers, or sent to equipment for recovery of condensed-vapor components. Alarms may be provided to signal operators in the event that design liquid levels are exceeded. Heating equipment should be provided in the liquid collection zone to keep the system operative in situations where low ambient temperatures may be a factor.

5.4.3 FLARES

5.4.3.1 Sizing

Factors governing the sizing of flares are covered in 5.4.3.1.1 through 5.4.3.1.5. General considerations involved in the calculation of these requirements are discussed in Section 4. Examples covering the full design of a flare stack are given in Appendix C. Note that flare diameter calculations are based on a basic flare. Most commercial flares have flame retainers that restrict flow area by 2 percent to 10 percent.

5.4.3.1.1 Flare stack diameter is generally sized on a velocity basis, although pressure drop should be checked. One may want to permit a velocity of up to 0.5 Mach for a peak, short-term, infrequent flow, with 0.2 Mach maintained for the more normal and possibly more frequent conditions for low-pressure flares, depending on the following: (a) volume ratio of maximum conceivable flare flow to anticipated average flare flow; (b) the probable timing, frequency, and duration of those flows; and (c) the design criteria established for the project to stabilize flare burning. However, sonic velocity operation may be appropriate for high-pressure flares. Smokeless flares should be sized for the conditions under which they are to operate smokelessly. Equation 23 or 24 can be used to calculate the Mach number (see 5.4.1.3.2). Velocity limitations imposed by reference [12] do not apply to flares in emergency relief service.

Pressure drops as large as 2 pounds per square inch (14 kilopascals) have been satisfactorily used at the flare tip. Too low a tip velocity can cause heat and corrosion damage. The burning of the gases becomes quite slow, and the flame is greatly influenced by the wind. The low-pressure area on the downwind side of the stack may cause the burning gases to be drawn down along the stack for 10 feet (3 meters) or more. Under these conditions, corrosive materials in the stack gases may attack the stack metal at an accelerated rate, even though the top 8 feet–10 feet (2.4 meters–3 meters) of the flare is usually made of corrosion-resistant material.

5.4.3.1.2 The flare stack height is generally based on the radiant heat intensity generated by the flame. Equation 20 in Section 4 applies. The recommended levels of radiation intensity, *K*, are given in Table 7.

The quality of combustion affects the radiation characteristics. Use of the fraction of heat radiated, *F*, based on the U.S. Bureau of Mines data given in Table 9, is considered to result in a reasonable but conservative stack height.

5.4.3.1.3 Another factor to be considered is the effect of wind in tilting the flame, thus varying the distance from the center of the flame, which is considered to be the origin of the total radiant heat release, with respect to the plant location under consideration. A generalized curve for approximating the effect of wind is given in Figure 10.

5.4.3.1.4 Where there is concern about the resulting atmospheric dispersion if the flare were to be extinguished, the information referred to in 4.3.1 and in Gifford's article [7] may be used to calculate the probable concentration at the point in question.

5.4.3.1.5 Ground Flare

A complete description of a type of ground flare, including design procedures, is provided by Miller et al. [15]. (See 4.4.3.2).

5.4.3.2 Design Details

An extensive bibliography on flare systems is included in this recommended practice. Specific design details are covered in 5.4.3.2.1 through 5.4.3.2.4.

5.4.3.2.1 Smoke-free operation of flares can be achieved by various methods, including steam injection, injection of high-pressure waste gas, forced draft air, operation of flares as a premix burner, or distribution of the flow through many small burners. The most common type of smokeless flare involves steam injection. For more detailed information on the theory of smokeless burning with steam injection, refer to 4.4.3.

The amount of steam required for smokeless burning will depend on the maximum vapor flow at which smokeless burning is to be achieved and the composition of the mixture. The composition involves both the percentage of unsaturates and the molecular weight. Figure 22, which is based on an open steam pipe that terminates inside a flare stack, may be used to determine the steam requirements as a function of the unsaturate content. In some cases, certain proprietary systems that use other methods of steam injection are more efficient. As indicated, the flare may be designed for various degrees of smokelessness.

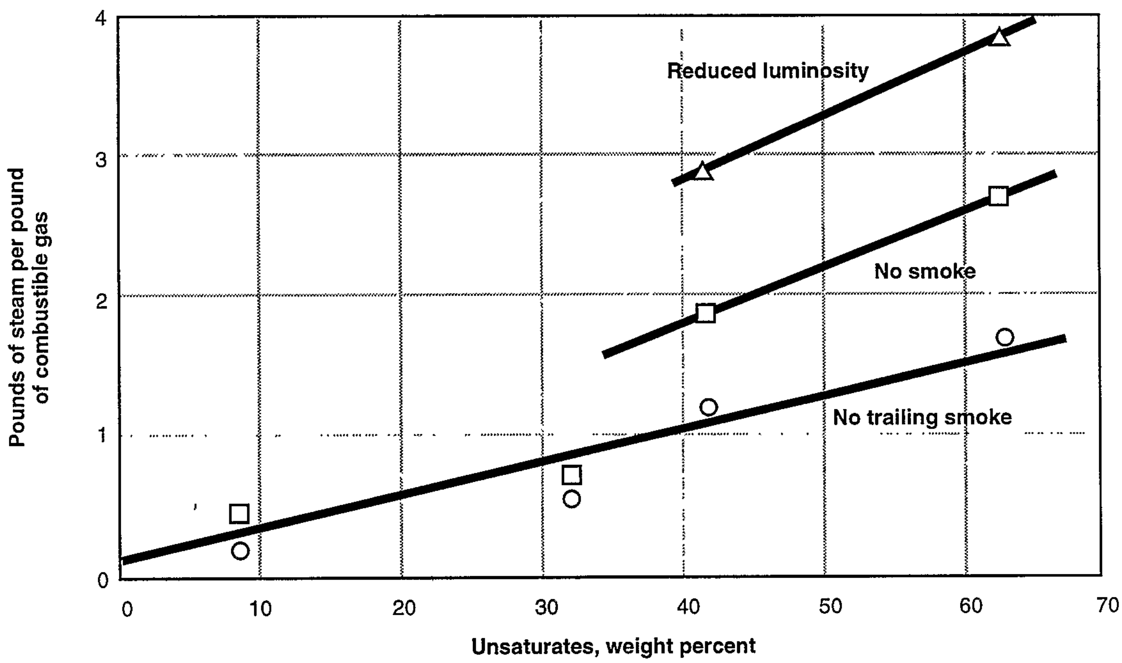


Figure 22—Required Steam Rates for Elevated Flares

Insofar as the effect of molecular weight is concerned, the higher the molecular weight of a hydrocarbon, the lower the ratio of steam to carbon dioxide (CO_2) and the greater the tendency to smoke. The amount of steam [16] that must be injected to maintain a constant steam-CO_2 ratio as molecular weight increases can be calculated from Equation 53, which is based on a steam-CO_2 weight ratio of approximately 0.7. See Table 10 for typical steam rates.

$$W_{steam} = W_{HC}\left(0.68 - \frac{10.8}{M}\right) \quad (53)$$

Where:

W_{steam} = steam, in pounds per hour (kilograms per second).
W_{HC} = hydrocarbon, in pounds per hour (kilograms per second).
M = molecular weight.

In any event, if a proprietary smokeless flare is purchased, the manufacturer should be consulted about the minimum necessary steam rate.

5.4.3.2.2 The most common method of preventing propagation of flame into the flare system as a result of the entry of air is to install a seal drum as described in 5.4.2.2. Flame arresters are occasionally used for flashback protection; however, they are subject to plugging, and their application is limited. The hazards associated with an obstruction in a flare system are of such a serious nature that flame arresters could be recommended only when vapors are noncorrosive and are dry and free from any liquid that might congeal. Although these situations are rarely encountered with flare systems, an additional disadvantage to a flame arrester that is pertinent to flare systems is that during the cooling which follows a warm discharge, air may be drawn back through the flame arrester into the system.

Proprietary systems that utilize differences in vapor density have been successfully installed and should be more reliable than a flame arrester. Such systems, however, may require continuous purging and may be subject to other limitations that should be established with the supplier.

Alternatively, the continuous introduction of purge gas can be used to prevent flashback. Studies [13] have shown that a safe condition exists in situations that involve hydrocarbon-air mixtures if a positive flow of oxygen-free gas is maintained, allowing the oxygen concentration to be no greater than 6 percent at a point 25 feet (7.6 meters) from the flare tip. Once the required quantity of purge gas has been established, the injection rate should be controlled by a fixed orifice to ensure that the supply remains constant and is not subject to instrument malfunction or maladjustment.

If the gas in the stack (for example, hydrogen) is lighter than air, the pressure in the bottom of the stack can be lower than atmospheric, even with some outflow from the top of the stack. The purge gas flow should be high enough to counteract this situation. Gases or vapors with unusually high-burning velocities, such as hydrogen and acetylene, should be considered for the possibility of flashback (see 4.4.3.4.1 and 4.4.3.4.2).

5.4.3.2.3 To ensure ignition of flare gases, continuous pilots with a means of remote ignition are recommended for all flares. The most commonly used type of igniter is the flame-front propagation type which uses a spark from a remote location to ignite a flammable mixture. Pilot igniter controls are located near the base of elevated flares and at least 100 feet (30 meters) from ground flares (see 4.4.3.4.4).

5.4.3.2.4 The fuel gas supply to the pilots and igniters should be highly reliable. Since normal plant fuel sources may be upset or lost, it is desirable to provide a backup system connected to the most reliable alternative fuel source, with a provision for automatic cut-in on low pressure. The use of a waste gas with low-energy content or with unusual burning characteristics should be avoided. Parallel instrumentation for pressure reduction is frequently justifiable. The flare fuel system should be carefully checked to ensure that hydrates cannot present a problem. Because of small lines, long exposed runs, large vertical rises up the stack, and pressure reductions, use of a liquid knockout pot or scrubber after the last pressure reduction is frequently warranted. If at all feasible in terms of distance, relative location, and cost, a low-pressure alarm should be installed on the fuel supply after the last regulator or control valve so that operators will be warned of any loss of fuel to the pilots.

5.4.4 VENT STACKS

5.4.4.1 Sizing

The size of a vent stack is determined by the available pressure drop and by any minimum velocity required to prevent hazardous conditions due to combustible or toxic material at grade or working levels. Calculation methods applicable to a vent stack that discharges hazardous materials are given in Section 4. Normally, a size is selected that will result in a high discharge velocity; for example, a velocity of 500 feet per second (152 meters per second) will provide excellent dispersion. The size should be checked to ensure that sonic flow is not established or, if it is, that allowance has been made for the pressure discontinuity at the discharge end in calculating pressure drop.

Vent stacks or the tips of vent stacks are generally sized for an exit velocity of at least 500 feet per second (152 meters per second) at the maximum relief rate. A sample calculation is presented in this section.

For this calculation, the following conditions should be assumed: The maximum relief rate, W, is 250,000 pounds per hour (31.5 kilograms per second). The molecular weight of the vapor, M, is 44. The temperature of the vapor just inside the vent tip, T, is 650°R (361°K). The exit velocity, V, is 500 feet per second (152 meters per second). The pressure of the vapor just inside the vent tip, P, is 14.7 pounds per square inch absolute (101 kilopascals absolute). The gas constant, R, is 10.7 (8.3 for metric units). The density, ρ, is then calculated as follows in Equations 54 and 55:

$$\rho = \frac{MP}{RT} = \frac{(44)(14.7)}{(10.7)(650)} = 0.1 \text{ pounds per cubic foot} \quad (54)$$

In metric units:

$$\rho = \frac{MP}{RT} = \frac{(44)(101)}{(8.3)(361)} = 1.48 \text{ kilograms per cubic meter} \quad (55)$$

The tip area, A_T, is determined as follows (Equations 56 and 57):

$$A_T = \frac{W}{3600\rho V} = \frac{250,000}{(3600)(0.1)(500)} = 1.39 \text{ square feet} \quad (56)$$

In metric units:

$$A_T = \frac{W}{\rho V} = \frac{31.5}{(1.48)(152)} = 0.14 \text{ square meter} \quad (57)$$

Thus, the pipe diameter should be about 16 inches nominal pipe size (NPS 16).

5.4.4.2 Design Details

Once the vent stack has been sized in accordance with the recommendations in 5.4.4.1 and the height has been established in accordance with the principles in 4.3, design is primarily a structural problem. If the vent stack is in a location remote from other facilities, the use of a guyed stack will usually be as satisfactory as, and more economical than, providing a structure to support the stack. Vent stacks are frequently located in a process area that contains equipment connected to the stack. The stack can often be supported from a fractionating tower, chimney, or other tall structure in the unit. Such an arrangement provides for economical discharge at a safe elevation.

The height of the vent stack is selected so that the concentration of vapor at a point of interest is well below the lower flammable limit of the vapor. Flammability consideration can be satisfied with 0.1 times to 0.5 times the lower flammable limit. Toxicity consideration may require much lower concentrations on certain applications and is, therefore, the controlling factor. The radiant heat intensity for vent stacks should also be checked in the event that a relieving vapor should ignite. This is done by the same means used for flare stacks, and the same limits apply for radiant heat intensity. Radiant heat levels sometimes take precedence over dispersion in determining stack height.

In every vent stack installation, careful consideration should be given to two potential problems: (a) accumulation of liquid in lines that terminate at the vent stack, and (b) accidental ignition by lightning. Accumulation of liquid in lines to the vent stack may result from leakage into the system of high-molecular-weight vapors that condense at ambient temperature. If appreciable quantities of liquid collect, they will subsequently be discharged to the atmosphere when vapors are released into the system.

To avoid liquid accumulation, pockets should be prevented from occurring in the lines, and the system should be sloped to a low-point drain. These drains can be installed to function automatically by using a properly designed seal. The height of the seal should provide a head equivalent to at least 1¾ times the back pressure under the maximum relief load to avoid release of vapor through the seal. As an alternative to a sealed drain, a small disengaging drum may be installed at the base of the vent stack. This type of installation is recommended where significant quantities of liquid may occur.

The possibility that vapors from the vent stack may be accidentally ignited by lightning or other sources usually makes a remote-controlled snuffing steam connection desirable on the vent stack. This is especially true in locations where the incidence of lightning is high or where access to the point of discharge would be difficult with conventional fire-extinguishing equipment. It is frequently impractical to size the steam supply line for a rate that is sufficient to extinguish a fire under maximum venting conditions. However, steam would still be essential, since, in most cases, vent fires occur when the only flow to the system consists of leakage or minor venting. Furthermore, unless steam is supplied, if ignition occurs when venting at or near the maximum design load, the fire will likely continue to burn when the cause of overpressure is corrected, with an accompanying reduction in venting.

5.4.4.3 Noise

The noise level at 100 feet (30 meters) from the point of discharge to the atmosphere can be approximated by Equation 58:

$$L_{100\,(30)} = L \text{ (from Figure 23)} + 10 \log_{10} (1/2\, MC^2) \quad (58)$$

Figure 23 illustrates the noise intensity measured as the sound pressure level at 100 feet (30 meters) from the stack tip versus the pressure ratio across the safety valve.

The following symbols are used in the procedure for calculating the noise level:

M = mass flow through the valve, in slugs per second (kilograms per second).

C = speed of sound in the gas at the valve, in feet per second (meters per second).

Equation 59 is computed in feet per second:

$$C = 223\left(\frac{kT}{\text{molecular weight}}\right)^{0.5} \tag{59}$$

Equation 60 is computed in meters per second:

$$C = 91.2\left(\frac{kT}{\text{molecular weight}}\right)^{0.5} \tag{60}$$

Where:

k = ratio of the specific heats in the gas.

T = gas temperature in degrees Rankine (Kelvin).

An example of calculating, in customary units, the noise level at 100 feet from the point of discharge to the atmosphere is presented in Items a–d:

a. Calculate $\frac{1}{2}MC^2$ in watts. Divide the weight flow (pounds per second) by 32 to obtain M. Multiply $\frac{1}{2}MC^2$ (foot-pounds per second) by 1.36 to obtain $\frac{1}{2}MC^2$ in watts.

b. Calculate $10 \log_{10}(\frac{1}{2}MC^2)$.

c. In Figure 23, enter PR as the abscissa and read the ordinate.

d. Add Items b and c to obtain the average sound pressure level at 100 feet, L_{100}, in decibels. Assume the following:

M = 1 slug per second

= 32 pounds per second

k = 1.4

Molecular weight = 29

T = 560 degrees Rankine

PR = 48/16 = 3

$$C = 223\left[\frac{(1.4)(560)}{29}\right]^{0.5} = 1{,}159 \text{ ft/sec}$$

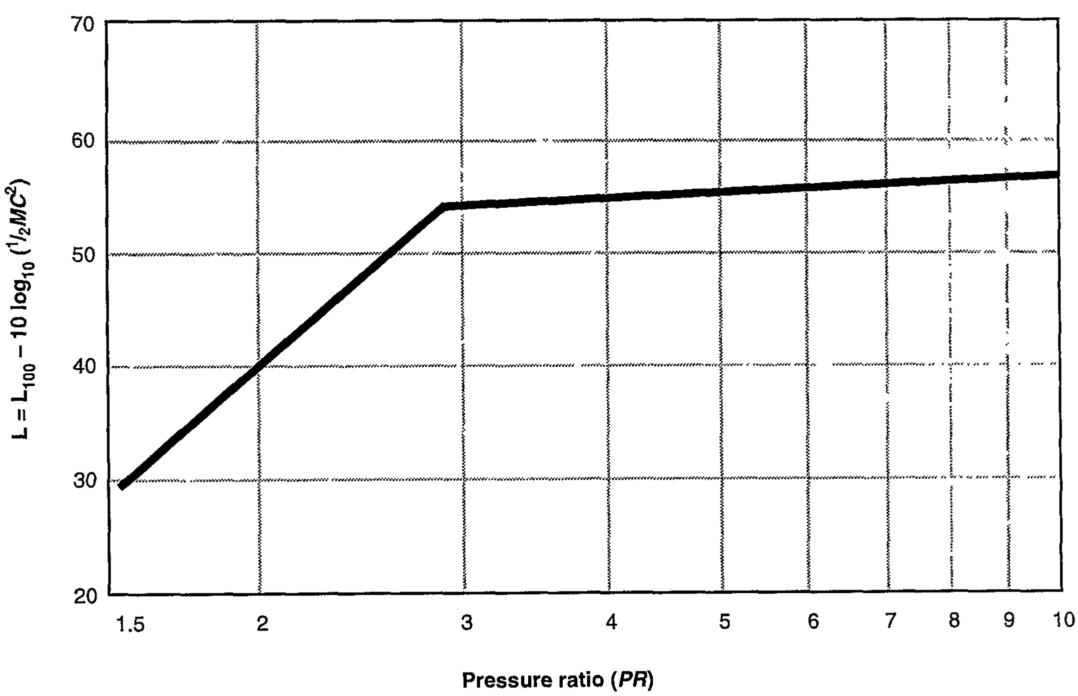


Figure 23—Noise Intensity at 100 Feet (30 Meters) From the Stack Tip

The results are as follows:

a. $\frac{1}{2}MC^2 = (\frac{1}{2})(1)(1159)^2(1.36) = (9.1)(10^5)$.
b. $10 \log_{10}(\frac{1}{2} MC^2) \approx 60$.
c. From Figure 23, at $PR = 3$, the ordinate = 54.
d. L_{100} at 100 feet = 54 + 60 = 114 decibels.

An example of calculating, in metric units, the noise level at 30 meters from the point of discharge to the atmosphere is presented in Items a–d as follows:

a. Calculate $\frac{1}{2} MC^2$ in watts.
b. Calculate $10 \log_{10}(\frac{1}{2} MC^2)$.
c. In Figure 23, enter PR as the abscissa and read the ordinate.
d. Add Items b and c to obtain the average sound pressure level at 30 meters, L_{30}, in decibels. Assume the following:

$$M = 14.6 \text{ kilograms per second}$$

$$k = 1.4$$

$$\text{Molecular weight} = 29$$

$$T = 311° \text{ Kelvin}$$

$$PR = 48/16 = 3$$

$$C = 91.2\left[\frac{(1.4)(311)}{29}\right]^{0.5}$$

$$= 353 \text{ meters per second}$$

The results are as follows:

a. $\frac{1}{2}MC^2 = (\frac{1}{2})(14.6)(353)^2 = (9.1)(10^5)$.
b. $10 \log_{10}(\frac{1}{2} MC^2) \approx 60$.
c. From Figure 23, at $PR = 3$, the ordinate = 54.
d. L_{30} at 30 meters = 54 + 60 = 114 decibels.

Note: These calculations are based on spherical spreading of the sound. If distances much larger than the height of the vent aboveground are of concern, add 3 decibels to the calculated result to correct for hemispherical diffusion.

By applying the following equations (61 and 62), the noise level can be adjusted for distances that differ from the 100-foot (30-meter) reference boundary:

$$Lp = L_{100} - 20 \log_{10}(r/100) \quad (61)$$

In metric units:

$$Lp = L_{30} - 20 \log_{10}(r/30) \quad (62)$$

Where:

Lp = sound pressure level at distance r, in decibels.
$L_{100(30)}$ = sound pressure level at 100 feet (30 meters), in decibels.
r = distance from the sound source (stack tip), in feet (meters).

For distances greater than 1,000 feet (305 meters), some credit may be taken for molecular noise absorption. When safety valves prove to be excessively noisy during operation, the sound can be deadened by the application of insulation around the valve body and the downstream pipe up to approximately five pipe diameters from the valve.

5.5 Flare Gas Recovery Systems

5.5.1 GENERAL

Environmental and economic considerations have resulted in the use of flare gas recovery systems to capture and compress flare gases for other uses. Many times the recovered flare gas is treated and routed to the refinery fuel gas system. Depending upon flare gas composition, recovered gas may have other uses.

5.5.2 SAFETY CONSIDERATIONS

5.5.2.1 Path to Flare

Flare systems are used for both normal process releases and emergency releases. Emergency streams, such as those from pressure relief valves, depressuring systems, and so on, must always have flow paths to the flare available at all times. The design of flare gas recovery systems shall not compromise this path. Several methods of accomplishing this are described in 5.5.3.3.

5.5.2.2 Back Flow

Because flare gas recovery systems usually involve compressors which take their suction directly from the flare header, the potential for back flow of air from the flare into the compressors at low flare-gas loads must be considered. Typically, oxygen content of the flare gas stream should be measured and provisions must be made to shut down the flare gas compressors if potentially dangerous conditions exist.

5.5.2.3 Flare Gas Characteristics

Flare gases can have widely varying compositions which must be evaluated during specification of recovery systems. The potential for materials which are not compatible with the flare gas-treating systems or ultimate destinations must be determined. For example, streams containing acid gases typically are routed directly to the flare, thereby bypassing the recovery system.

5.5.3 DESIGN CONSIDERATIONS

Figure 24 shows a conceptual design for a flare gas recovery system. Typically, the system consists of one or more reciprocating compressors whose suction is directly connected to the flare header. The compressed gas is usually routed to some type of treating system appropriate for the gas composition, then to fuel gas or processing systems.

5.5.3.1 Sizing

Flare-gas recovery systems are seldom sized for emergency flare loads. Usually, economics dictate that capacity be

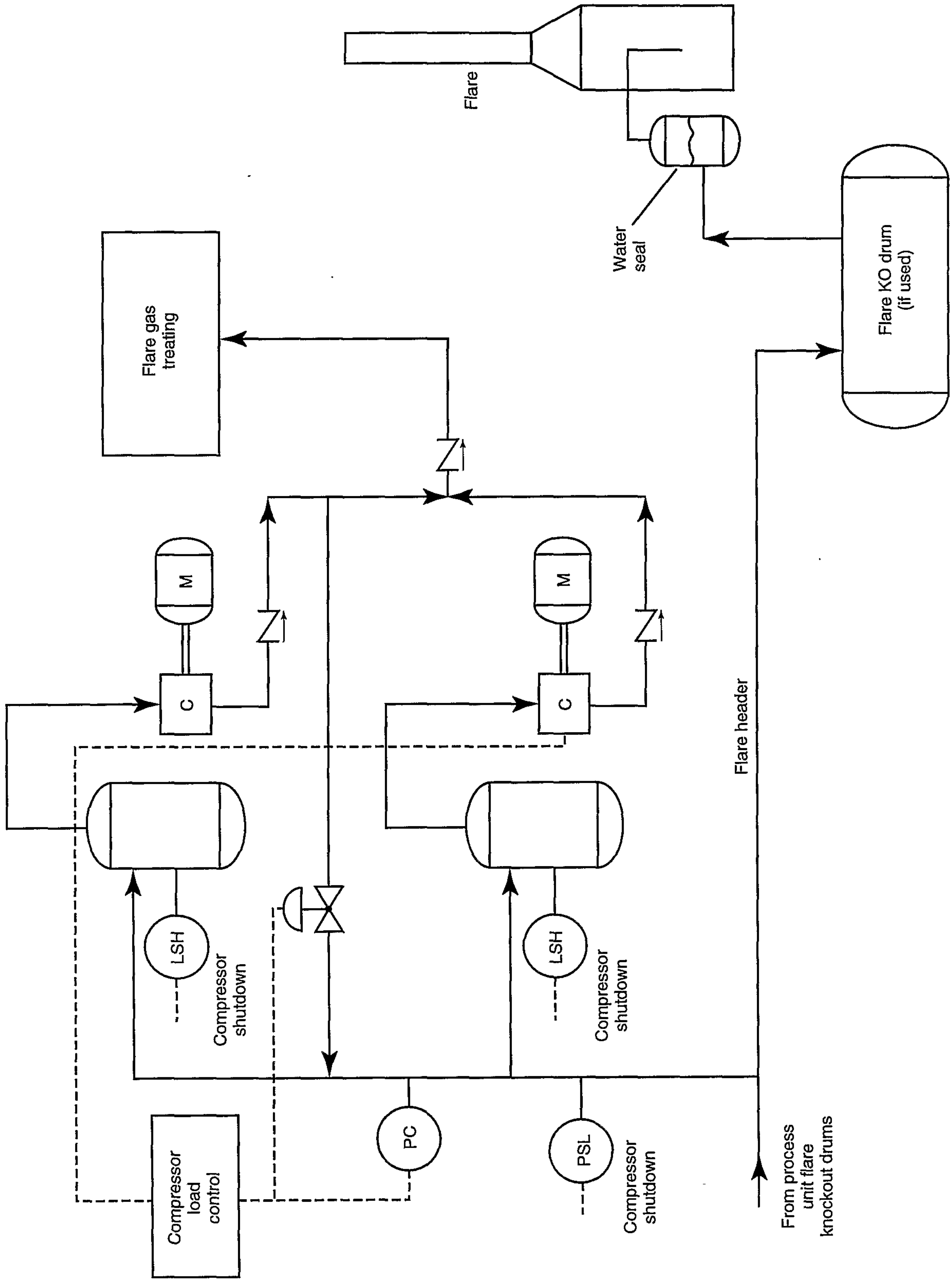


Figure 24—Typical Flare Gas Recovery System

provided for some normal flare rate, above which gas is flared. Flare loads vary widely over time, and the normal rate may represent some average flare load, or a frequently encountered maximum load. Actual loads on these systems will vary widely, and they must be designed to operate over a wide range of dynamically changing loads. Flare gas recovery systems often are installed to comply with local regulatory limits on flare operation and, therefore, must be sized to conform to any such limits.

5.5.3.2 Location

Typically, flare gas recovery systems are located on the main-flare header downstream of all unit header tie-ins and at a point where header pressure does not vary substantially with load. Locations upstream of process unit tie-ins should be carefully considered because of the potential for back-flow and high-oxygen concentrations. Limited downstream tie-ins for material not suitable for recovery may be required.

5.5.3.3 Flare Tie-In

As discussed in 5.5.2, a major consideration in flare recovery system design is preservation of a path to the flare for emergency releases. The flare gas recovery system must be designed as a side stream from the flare header. Main flare flow should not be through any compressor knock out or suction piping. The tie-in line to the flare gas recovery system should come off the top of the flare line to minimize the possibility of liquid entrance.

Some method of ensuring a positive pressure on the flare gas recovery system must also be provided. Figure 25 shows some methods of doing this while preserving a reliable open path to the flare.

5.5.3.3.1 The most positive and preferred method for preventing air ingress is the installation of a water seal vessel between the flare knockout drum and the flare itself. The seal provides a relatively constant low backpressure on the flare header and provides a narrow, but usually adequate control range for the flare-gas recovery control system. The water seal should be designed to function over the pressure for which the flare gas recovery system is designed to operate. At higher release rates, flare gas flows through the seal and out the flare. Design provisions must be made to maintain the seal level, prevent high flare rates from carrying the seal water up the flare, and prevent seal freeze-up. See Figure D.1 for typical seal drum design.

5.5.3.3.2 If process requirements are such that the narrow operating ranges afforded by water seals cannot be accepted, an alternate method is to use a fail-open control valve to regulate the suction pressure of the flare-gas recovery system. A positive path to the flare is provided by installing a low-pressure, high-capacity pilot-operated pressure relief valve around the control valve. The sensing line for the pressure relief valve pilot shall be provided with a clean gas purge and a backflow preventer.

The sizes of the control and pressure relief valves can become quite large. The flare header system must also be studied to verify that the backpressure imposed by the pressure relief device (assuming the control valve is closed) at full header load will not induce unacceptable back pressures on devices releasing into the headers at the processing units.

Alternatives to use of a pressure relief valve are installation of nonreclosing devices such as rupture discs or breaking-pin devices. These installations must also be carefully reviewed to ensure that the devices operate at as low a pressure as possible and that they do not cause unacceptable back pressure.

5.5.3.3.3 If a control valve must be used in the flare line to regulate flare-gas recovery system suction pressure, the control valve should be of a fail-open design and be interlocked to go fully open upon a higher-than-normal header pressure, high-oxygen content, or when the compressors are unloaded or shutdown. These interlocks are not a substitute for a positive path around the control valve described in 5.5.3.3.2.

5.5.3.4 Back Flow Protection

Provisions must be made to prevent back flow of air from the flare into the flare-gas recovery system. All compressors should be equipped with highly reliable low-suction-pressure shutdown controls. Consideration should also be given to installation of additional instrumentation in the section of header between the flare and the compressor suction take-off to detect reverse flow and automatically shut down the flare gas recovery system.

5.5.3.5 Flare-Gas Recovery Controls

5.5.3.5.1 Flare-gas recovery systems must operate over wide ranges, usually within very narrow suction pressure bands. A typical system might operate over a suction pressure range of 2 to 5 inches of water to 10 to 12 inches of water. The flare-gas recovery compressors should be equipped with several stages of unloaders and a compressor recycle valve. Suction pressure is maintained by pressure control of a recycle valve, with additional loading and unloading of the compressors when limits of valve opening or closing or suction pressure are reached. Usually, the controls are set up to sequentially load and unload the compressors.

5.5.3.5.2 The possibility of significant liquid in flare systems is usually quite high. Liquid knock-out vessels should be provided for the compressors with automatic shutdown of the compressors on high suction drum levels. Other mechanical protection systems may also be required for the compressors. These systems may either shutdown or just unload the compressors. Refer to API Standard 618 for guidance on compressor protection.

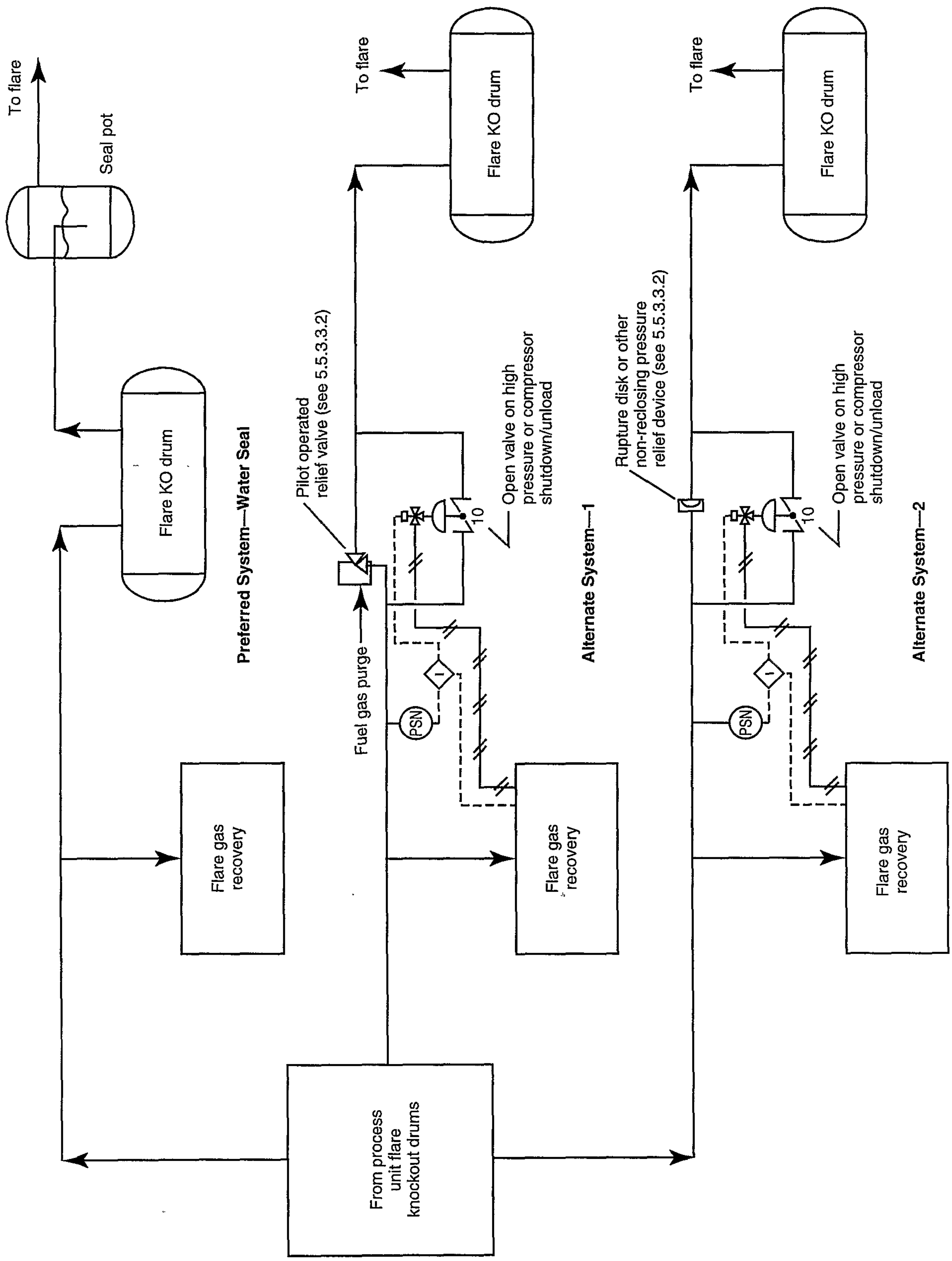


Figure 25—Flare Gas Recovery Inlet Pressure

5.6 Referenc s

1. O. Cox, Jr., and M. L. Weirick, "Sizing Safety Valve Inlet Lines," *Chemical Engineering Progress*, November, 1980.

2. B. A. Van Boskirk, "Sensitivity of Relief Valves to Inlet and Outlet Line Lengths," *Chemical Engineering*, August, 1982.

3. H. Y. Mak, "New Method Speeds Pressure-Relief Manifold Design," *Oil and Gas Journal*, November 20, 1978.

4. P. Kandell, "Program Sizes Pipe and Flare Manifolds for Compressible Flow," *Chemical Engineering*, June 29, 1981.

5. C. E. Lapple, "Isothermal and Adiabatic Flow of Compressible Fluids," *Transactions of the American Institute of Chemical Engineers*, 1943, Volume 39, pp. 385–432.

6. B. M. Coulter, *Compressible Flow Manual*, Fluid Research Publishing Company, Melbourne, Florida.

7. D. M. Kirkpatrick, "Simpler Sizing of Gas Piping," *Hydrocarbon Processing*, December, 1969.

8. J. E. Huff and K. R. Shaw, "Measurement of Flow Resistance of Rupture Disc Devices," *Plant/Operations Progress*, July, 1992.

9. R. H. Perry, C. H. Chilton, and S. D. Kirkpatrick, editors, *Chemical Engineers Handbook*, 5th edition, McGraw-Hill, New York, 1972, pp. 5–61.

10. A. D. Scheiman, "How to Size Shower Deck Baffled Towers Quicker: Part I—Tower Diameter," *Petro-Chem Engineer*, 1965, Volume 37, Number 3, pp. 28–33.

11. A. D. Scheiman, "How to Size Shower Deck Baffled Towers Quicker: Part 2—Tower Tangent Length," *Petro/Chem Engineer*, 1965, Volume 37, Number 4, pp. 75, 78–79.

12. CFR 60.18, "General Control Device Requirements," *Code of Federal Regulations*, Government Printing Office, Washington, D.C.

13. H. W. Husa, "How to Compute Safe Purge Rates," *Hydrocarbon Processing and Petroleum Refiner*, 1964, Volume 43, Number 5, pp. 179–182.

14. F. A. Gifford, Jr., "Atmospheric Dispersion Calculations Using the Generalized Garrison Plume Model," *Nuclear Safety*, December, 1960, pp. 56–59.

15. P.D. Miller, Jr., H. J. Hibshman, and J. R. Connel, "The Design of Smokeless, Nonluminous Flares," *API Proceedings*, 1958, Volume 38, American Petroleum Institute, Washington, D.C., pp. 276–281.

16. S.H. Tan, "Flare Design Simplified," *Hydrocarbon Processing and Petroleum Refiner*, 1967, Volume 46, Number 1, pp. 172–176.

SECTION 6—BIBLIOGRAPHY

6.1 Disposal Systems

Armistead, G., Jr., *Safety in Petroleum Refining and Related Industries*, 2nd edition, John G. Simmonds, New York, 1959.

Bosanquet, C. H., W. F. Carey, and F. M. Halton, *Journal of the Institution of Mechanical Engineers*, Volume 162, Number 3, (1950), London, p. 355.

Fristrom, R. M., *Chemical and Engineering News*, Volume 41, Number 41 (October 14, 1963), pp. 150–160.

Gordier, R. L., *Studies on Fluid Jets Discharging Normally Into Moving Liquid*, Technical Paper Number 28, Series B. St. Anthony Falls Hyd. Lab., University of Minnesota, 1959.

Karlovitz, B., *Chemical Engineering Progress*, Volume 61, Number 8, (1965), pp. 56–62.

Keagy, W. R., and A. E. Weller, *A Study of Freely Expanding Inhomogeneous Jets*, Paper Number 89, Heat Transfer and Fluid Mechanics Institute, University of California, Berkeley, 1949.

Keffer, J. F. and W. D. Baines, *Journal of Fluid Mechanics*, Volume 15, (1963), p. 487.

Loudon, D. E. *API Proceedings*, 1963, Volume 43, American Petroleum Institute, Washington, D.C., pp. 418–433.

Manual on Disposal of Refinery Wastes—Volume on Atmospheric Emissions, American Petroleum Institute, Washington, D.C., 1977.

Miller, P. D., Jr., H. J. Hibshman, and J. R. Connell, *API Proceedings*, 1958, Volume 38, American Petroleum Institute, Washington, D.C., pp. 276–281.

Pipkin, O. A., and C. M. Sliepcevich, *Industrial and Engineering Chemistry Fundamentals*, Volume 3, Number 2, (1964), pp. 147–154.

Wohl, K., N. M, Kapp, and C. Gazley, *Proceedings: 3rd Symposium (International) on Combustion and Flame Explosion Phenomena, Madison, Wisconsin, 1948*, Williams and Wilkins, Baltimore, 1949, pp. 3–21.

6.2 Drums and Separators

Kerns, G. D., *Petroleum Refiner*, Volume 39, Number 7 (1960), pp. 168–170.

Lapadula, E. J., *Chemical Engineering,* Volume 70, Number 16, (August 5, 1963), pp. 128–129.

Montross, C. F., *Chemical Engineering,* Volume 60, Number 10, (1953), pp. 213–236.

Niemoyer, F. R., *Hydrocarbon Processing and Petroleum Refiner,* Volume 40, Number 6, (1961), pp. 155–156.

Scheiman, A. D., *Hydrocarbon Processing and Petroleum Refiner,* Volume 42, Number 10, (1963), pp. 165–168.

Scheiman, A. D., *Hydrocarbon Processing and Petroleum Refiner,* Volume 43, Number 5, (1964), pp. 155–160.

Scheiman, A. D., *Petro/Chem Engineer,* Volume 37, Number 3, (1965), pp. 28–33.

Scheiman, A. D., *Petro/Chem Engineer,* Volume 37, Number 4, (1965), pp. 75, 78–79.

Smith, V. *Oil and Gas Journal,* Volume 57, Number 1, (1959), pp. 97–101.

Watkins, R. N., *Hydrocarbon Processing,* Volume 46, Number 11, (1967), pp. 253–256.

6.3 Flares and Stacks

Boudurtha, F. T., *Chemical Engineering,* Volume 65, Number 25, (December 15, 1958), pp. 177–180.

Boudurtha, F. T., "The Behavior of Dense Stack Gases," Paper presented at the meeting of the Air Pollution Control Association, New York, 1955.

Brzustowski, T. A., and E. C. Sommer, Jr., *API Proceedings—Division of Refining,* Volume 53, 1973, American Petroleum Institute, Washington, D.C., pp. 865–893.

"The Calculation of Atmospheric Dispersion from a Stack," CONCAWE, The Hague, The Netherlands, 1966.

Cleveland, L. D., *Oil and Gas Journal,* August 4, 1952.

Daniels, E. K. J. R Lutz, and L. A Castler, *API Proceedings,* Volume 38, 1958, American Petroleum Institute, Washington, D.C., pp. 291–292.

Grumer, J., A. Strasser, J. M.Singer, and P. M. Gussey, "Hydrogen Flare Stack Diffusion Flames, Low Flow Instability, Burning Rates, and Dilution Limits," U.S. Bureau of Mines Preprint, ERC No. 4030, U.S. Department of the Interior, Washington, D.C.,1968.

Hajek, J. D. and E. E. Ludwig, *Petro/Chem Engineer,* Volume 32, Number 6, (1960), pp. C31–C38.

Hajek, J. D., and Ludwig, E. E., *Petro/Chem Engineer,* Volume 32, Number 7, 1960, pp. C44–C51.

Hannaman, J. R. and A. J. Etingen, *Petroleum Processing* March 1956, pp. 66–69.

Hewson, E. W., *Transactions of the American Society of Mechanical Engineers,* Volume 77, Number 10, 1955, pp. 1163–1172.

Hoehne, V. O., R. G. Luce, and L. W. Miga, "The Effect of Velocity, Temperature, and Gas Molecular Weight on Flammability Limits in Wind-Blown Jets of Hydrocarbon Gases," Report to the American Petroleum Institute. Battelle Memorial Institute, Columbus, Ohio, 1970.

Husa, H. W., *Hydrocarbon Processing and Petroleum Refiner,* Volume 43, Number 5, 1964, pp. 179–182.

International Safety Guide for Oil Tankers and Terminals, International Chamber of Shipping/Oil Companies International Marine Forum, Witherby and Co., London, 1978.

Johnson, J. H., *Petroleum Refiner,* Volume 26, Number 6, 1947, p. 91.

Kent, G. R., *Hydrocarbon Processing and Petroleum Refiner,* Volume 43, Number 8, 1964, pp. 121–125.

Kent, G. R., *Hydrocarbon Processing,* Volume 47, Number 6, 1968, pp. 119–130.

Miller, P. D., Jr., H. J. Hibschman, and J. R. Connell, *API Proceedings,* Volume 38, American Petroleum Institute, 1958, Washington, D.C., pp. 276–281.

Reed, R. D., *Chemical Engineering Progress,* Volume 64, Number 6, June 1968, pp. 53–57.

Schwartz, R. and M. Keller, "Environmental Factors Versus Flare Application," Paper Number 13d, presented at the Symposium on Loss Prevention in the Chemical Industry, 83rd National Meeting of the American Institute of Chemical Engineers, Houston, Texas, March 22, 1977.

"Session on Refinery Flare Design," *API Proceedings,* 1952, Volume 32M, American Petroleum Institute, Washington, D.C., pp. 143–164.

Sherlock, R. H. and E. J. Lesher, *Transactions of the American Society of Mechanical Engineers,* Volume 77, Number 1, 1955, pp. 1–9.

Smith, W. R., *Petroleum Processing,* June 1954, pp. 867–880.

Smolen, W. H., *Petroleum Processing,* Volume 6, Number 9 1951, pp. 978–982.

Stankiewicz, E. J., *Combustion,* Volume 26, Number 8, February 1955, pp. 51–55.

Steinbock, R. S., *Chemical Engineering,* Volume 59, Number 2, 1952, pp. 202–203.

Steinbock, R. S., *Chemical Engineering,* Volume 59, Number 3, 1952, pp. 144–147.

Steinbock, R. S., *Chemical Engineering,* Volume 59, Number 4, 1952, pp. 154–155.

Tan, S. H., *Hydrocarbon Processing and Petroleum Refiner,* 1967, Volume 46, Number 1, pp. 172–176.

6.4 Flashing Flow in Pipes

Anderson, R. J. and T. W. F. Russell, *Chemical Engineering,* Volume 72, Number 25, December 6, 1965, pp. 139–144.

Anderson, R. J. and T. W. F. Russell, *Chemical Engineering,* Volume 72, Number 26, December 20, 1965, pp. 99–104.

Anderson, R. J. and T. W. F. Russell, *Chemical Engineering,* Volume 73, Number 1, January 3, 1966, pp. 87–90.

Baker, O., *Oil and Gas Journal,* Volume 53, Number 12, July 26, 1954, pp. 185–190, 192, 195.

Baker, O., *Oil and Gas Journal,* Volume 56, Number 45, November 10, 1958, pp. 156-157, 159–161, 163, 165, 167.

Baker, O., et al. AGA/API Report NX-28, October 1970.

Chavez, J. A., *Oil and Gas Journal,* Volume 57, Number 35, August 24, 1959, pp. 100–102.

Chenoweth, J. M. and M. W. Martin, *Petroleum Refiner,* Volume 34, Number 10, 1955, pp. 151–155.

Coates, J. and B. S. Pressburg, *Chemical Engineering,* Volume 66, Number 18, September 7, 1959, pp. 153–154.

Dukler, A. E., M. Wicks, III, and R. G. Cleveland, *American Institute of Chemical Engineers Journal,* Volume 10, Number 1, 1964, pp. 38–51.

First, K. E. and J. E. Huff, "Design Charts for Two-Phase Flashing Flow in Emergency Pressure Relief Systems," Paper presented at AIChE International Symposium on Runaway Reactions, March 7–9, 1989.

Flanigan, O. *Oil and Gas Journal,* Volume 56, Number 10, (March 10, 1958): pp. 132–133, 136, 140–141.

Flanigan, O., *Oil and Gas Journal,* Volume 57, Number 49, November 30, 1959, pp. 60–66.

Leung, J. C., "Size Safety Relief Valves for Flashing Liquids," *Chemical Engineering Progress,* February, 1992.

Leung, J. C. and M. A Grolmes, "The Discharge of Two-Phase Flashing Flow in a Horizontal Duct." *AIChE Journal,* Volume 33, Number 3, March, 1987.

Leung, J. C., "A Generalized Correlation for One-Component Homogeneous Equilibrium Flashing Choked Flow," *AIChE Journal,* Volume 32, Number 10, October, 1986.

Lockhart, R. W. and R. C. Martinelli, *Chemical Engineering Progress,* Volume 45, Number 1, 1949, pp. 39–48.

Paige, P. M., *Chemical Engineering,* Volume 74, Number 17, August 14, 1967, pp. 159–164.

Richter, S. H., *Proceedings—Refining Department,* 1978, Volume 57, American Petroleum Institute, Washington, D.C., pp. 23–35.

Ros, N. C. J., *Journal of Petroleum Technology,* Volume 13, Number 10, 1961, pp. 1037–1049.

6.5 Flashing Flow in Valves

Baumann, H. D., "The Introduction of a Critical Flow Factor for Valve Sizing," Preprint No. T 34.3.62-1, Paper presented at the 18th Annual Conference of the Instrument Society of America, Chicago, October 1962.

Driskell, L. R., *Instrumentation Technology,* Volume 14, Number 6, 1967), pp. CVS1–CVS12.

Hanssen, A. J., "Accurate Valve Sizing for Flashing Fluids," Technical Bulletin Number 108, Conoflow Corp., Blackwood, N.J., 1964.

Kent, G. R., *Control Engineering,* Volume 13, Number 5, 1966, pp. 87–92.

Kent, G. R., *Control Engineering,* Volume 13, Number 6, 1966, pp. 69–73.

Sheldon, C. W. and C. B. Schuder, *Instruments and Control Systems,* Volume 38, Number 1, 1965, pp. 134–137.

6.6 Piping

Altorfer, H. A., *Oil and Gas Journal,* Volume 53, Number 32, December 13, 1954, pp. 126–131.

Conison, J. E., *Oil and Gas Journal,* Volume 52, Number 44, March 8, 1954, pp. 119–120.

Conison, J. E., *Oil and Gas Journal,* Volume 52, Number 45, March 15, 1954, p. 115.

Conison, J. E., *Oil and Gas Journal,* Volume 52, Number 47, March 29, 1954, p. 123.

Conison, J. E., *Oil and Gas Journal,* Volume 52, Number 48, April 5, 1954, p. 123.

Driskell, L. R. *Petroleum Refiner,* Volume 39, Number 7, 1960, pp. 127–132.

"Flow of Fluids Through Valves, Fittings, and Pipe," Technical Paper Number 410, Crane Co., New York, 1957, pp. 2–14, 2–15, A-19–A-21.

Lapple, C. E. *Transactions of the American Institute of Chemical Engineers,* Volume 39, 1943, American Institute of Chemical Engineers, New York, pp. 385–432.

Lipinski, F., *Oil and Gas Journal,* Volume 56, Number 14, April 14, 1958, p. 128.

Lipinski, F., *Oil and Gas Journal,* Volume 56, Number 18, May 5, 1958, p. 129.

Lipinski, F., *Oil and Gas Journal,* Volume 56, Number 20, May 19, 1958, p. 187.

Missen, R. W., *Chemical Engineering,* Volume 69, Number 22, October 29, 1962, pp. 101–102.

Parish, C. E. and D. Cornell, *Petroleum Refiner,* Volume 34, Number 2, 1955, pp. 121–123.

Roberts, R. W. and D. Cornell, *Petroleum Refiner,* Volume 34, Number 7, 1955, pp. 141–143.

Steshko, N., *Oil and Gas Journal,* Volume 54, Number 41, February 13, 1956, p. 141.

Steshko, N., *Oil and Gas Journal,* Volume 54, Number 46, March 19, 1956, pp. 265–266.

Steshko, N., *Oil and Gas Journal,* Volume 54, Number 47, March 26, 1956, p. 153.

Tan, S. H., *Hydrocarbon Processing and Petroleum Refiner,* Volume 46, Number 10, 1967, pp. 149–154.

6.7 Piping Guides and Anchors

Brock, J. E., *Heating, Piping and Air Conditioning,* Volume 36, Number 1, 1964, pp. 152–155.

Brock, J. E., *Heating, Piping and Air Conditioning,* Volume 39, Number 1, 1967, pp. 144–147.

Parker, C. S., *Heating, Piping and Air Conditioning,* Volume 39, Number 11, 1967, pp. 93–100.

6.8 Pressure Relief Valves

Beard, C. S. and F. D., Marton. *Regulators and Relief Valves,* Instrument Publishing Company, Pittsburgh, 1959, pp. 103–130.

Bigham, J. E., *Chemical Engineering,* Volume 65, Number 3, February 10, 1958, pp. 133–136.

Conison, J. E., *Petroleum Refiner,* Volume 34, Number 7, 1955, pp. 137–140.

Heller, F. J., *Oil and Gas Journal,* Volume 52, Number 49, April 12, 1954, p. 159.

Heller, F. J., *Oil and Gas Journal,* Volume 52, Number 52, May 3, 1954, p. 111.

Heller, F. J., *Oil and Gas Journal,* Volume 53, Number 3, May 24, 1954, pp. 263–264.

Leung, J. C., "Size Safety Relief Valves for Flashing Liquids," *Chemical Engineering Progress,* February, 1992.

Leung, J. C. and F. N. Nazario, "Two-Phase Flashing Flow Evaluations Based on DIERS, API, and ASME Methodologies," Paper presented at AIChE Loss Prevention Symposium, April 2–6, 1989.

Puleo, P. A., *Petroleum Refiner,* Volume 39, Number 10, 1960, pp. 157–162.

Weber, C. G., Chemical Engineering, Volume 62, Number 10, 1955, pp. 170–174.

Weber, C. G., "Relief Valve Selecting and Sizing," Paper presented at the 7th Southeast Regional Instrument Society of America Conference, Charlotte, North Carolina, April 20, 1961.

Weber, C. G., *Oil and Gas Equipment,* Volume 7, Number 11, September 1961, pp. 12–13.

6.9 Rupture Disks

Bigham, J. E., *Chemical Engineering,* Volume 65, Number 7, April 7, 1958, pp. 143–145.

Boyle, W. J., Jr., *Chemical Engineering Progress,* Volume 63, Number 8, 1967, pp. 61–66.

Brown, J. F. W., *Transactions of the Institution of Chemical Engineers,* Volume 36, April 1958, London, pp. 81–86.

Diss, E., H. Karma., and C. Jones, *Chemical Engineering,* Volume 68, Number 19, September 18, 1961, pp. 187–188.

Lowenstein, J. G., *Chemical Engineering,* Volume 65, Number 1, January 13, 1958, pp. 157–158.

Luker, J. A. and M. J Leibson, *Journal of Chemical and Engineering Data,* Volume 4, Number 2, April 1959, pp. 133–136.

Monday, G., *Chemical Process Hazards with Special Reference to Plant Design,* Symp. Soc. Number 15, Volume 2, 1963, Institution of Chemical Engineers, London, p. 46.

Solter, R. L., L. L. Fike, and F. A Hansen, *API Proceedings,* Volume 43, 1963, American Petroleum Institute, Washington, D.C., pp. 448–456.

Tarver, M. S., *Chemical and Metallurgical Engineering,* Volume 51, November 1944, pp. 108–112.

Tarver, M. S., *Chemical and Metallurgical Engineering,* Volume 51, December 1944, pp. 99–104.

Weil, J. A., *Transactions of the American Society of Mechanical Engineers, Series E: Journal of Applied Mechanics,* Volume 81, Number 4, December 1959, pp. 621–624.

Wright, H. R., *Transactions of the Institution of Chemical Engineers,* Volume 36, April 1958, London, pp. 69–78.

6.10 Surge and Pressure Transients

Bergeron, L., *Water Hammer in Hydraulics and Wave Surges in Electricity,* John Wiley & Sons, New York, 1961.

Burnett, R. R., *Oil and Gas Journal,* Volume 58, Number 18, May 2, 1960, pp. 153–160.

Coccio, C. L., *Combustion,* Volume 38, Number 8, February 1967, pp. 18–22.

Lindberg, G. A., *Pipeline Engineer,* Volume 37, Number 3, 1965, pp. 132–135.

Rouleau, W. T., *Transactions of the American Society of Mechanical Engineers, Series D: Journal of Basic Engineering,* Volume 82, Number 4, December 1960, pp. 912–920.

Sarlat, I. M. and T. L. Wilson, *Transactions of the American Society of Mechanical Engineers, Series D: Journal of Basic Engineering,* Volume 84, Number 3, September 1962, pp. 363–368.

Shapiro, A. H., *Dynamics and Thermodynamics of Compressible Fluid Flow,* Volume 2, Chapter 24, Ronald Press, New York, 1953.

Signor, C. W., *American Society of Heating, Refrigeration and Air Conditioning Engineers Journal,* Volume 2, Number 4, 1960, p. 69.

Wilkinson, J. F., D. V. Holliday, and E. F. Batey, *Oil and Gas Journal,* Volume 63, Number 2, January 11, 1965, pp. 94–95.

6.11 Systems

Block, B., *Chemical Engineering,* Volume 69, Number 2, January 22, 1962, pp. 111–118.

Bluhm, W. C., *API Proceedings,* Volume 41, 1961, American Petroleum Institute, Washington, D.C., pp. 169–179.

Case, R. C., *Proceedings—Division of Refining,* Volume 50, 1970, American Petroleum Institute, Washington, D.C., pp. 1082–1092.

Conison, J. E., *API Proceedings,* Volume 43, 1963, American Petroleum Institute, Washington, D.C., pp. 434–447.

Doyle, W. H., *National Safety Congress Transactions,* Volume 5, pp. 30–34; Volume 19: pp. 25–29, 1964.

Fowler, D. W., T. R. Herndon, and R. C Wahrmund, "An Analysis of Potential Overpressure of a Heat Exchanger Shell Due to a Ruptured Tube," Paper presented at the American Society of Mechanical Engineers Joint Annual Petroleum Mechanical Engineering Conference, Dallas, Texas, September 22–26, 1968.

General American Tank Car Manual, GATX, Chicago,1961.

Katz, D. L. and N. E Sylvander, *Engineering Research Bulletin Number 31,* University of Michigan Press, Ann Arbor, 1948.

Loudon, D. E., *API Proceedings,* Volume 43, 1963, American Petroleum Institute, Washington, D.C., pp. 418–433.

"Plant and Design Safety," CEP Reprint, American Institute of Chemical Engineers, New York, 1965.

Rase, H. F. and M. H. Barrow, *Project Engineering of Process Plants,* Chapter 24, Wiley & Sons, New York, 1957.

Sumaria, V. H., J. A. Rovnak, I. Heitner, and R.J. Herbert, *Proceedings—Refining Department,* Volume 55, 1976, American Petroleum Institute, Washington, D.C., pp. 631–654.

APPENDIX A—DETERMINATION OF FIRE-RELIEF REQUIREMENTS

A.1 Background

The problem of estimating fire-relief requirements for storage tanks was first recognized in 1928 when the 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: The numbers in brackets refer to references listed in A.4.

Table A-1 contains data from ten fire tests and one actual fire. 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 A-1 are average rates on the wetted surface.

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

Test	Source	Type of Exposure	Fuel	Vessel Capacity (BBL)	Total Area (ft^2)	Wetted Area (ft^2)	Total Heat Input (BTU/Hr)	Temperature of Surface (°F)	BTU/Hr/ Ft^2 Wetted	Ref.
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

Note: BBL = barrels; ft^2 = square feet; BTU/hr = British Thermal units per hour.

[a]This represents an actual fire.

[b]References are as follows

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., Northbrook, Illinois, 1938.
5. *Safety Memorandum 89*, Rubber Reserve Corporation, Washington, DC, 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.

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 to 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 into the development of Equations 1 and 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 that in some cases 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 by these tests are extremely unlikely to occur in an actual refinery fire.

A.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 heater, 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 much 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.

A.3 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 A-1 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, one can obtain the latent heat from Figure A-1. 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 3.15.3.2.

Other recognized sources [4] of latent-heat data or methods of calculating latent heat of vaporization should be used where Figure A-1 does not apply.

A.4 References

1. N. E. Sylvander and D. L. Katz, "Investigation of Pressure Relieving Systems," *Engineering Research Bulletin Number 31,* University of Michigan, Ann Arbor, April 1948.

2. *Recommended Practice for the Design and Construction of Pressure Relieving Systems for Process Equipment and Pressure Storage in Refineries* (tentative), American Petroleum Institute, August, 1954.

3. F. J. Heller, "Safety Relief Valve Sizing: API Versus CGA Requirements Plus a New Concept for Tank Cars," *1983 Proceedings—Refining Department,* Volume 62, American Petroleum Institute, Washington, D.C., pp. 123–135.

4. Publication 999 (English Edition), *Technical Data Book—Petroleum Refining,* American Petroleum Institute, Washington, D.C.

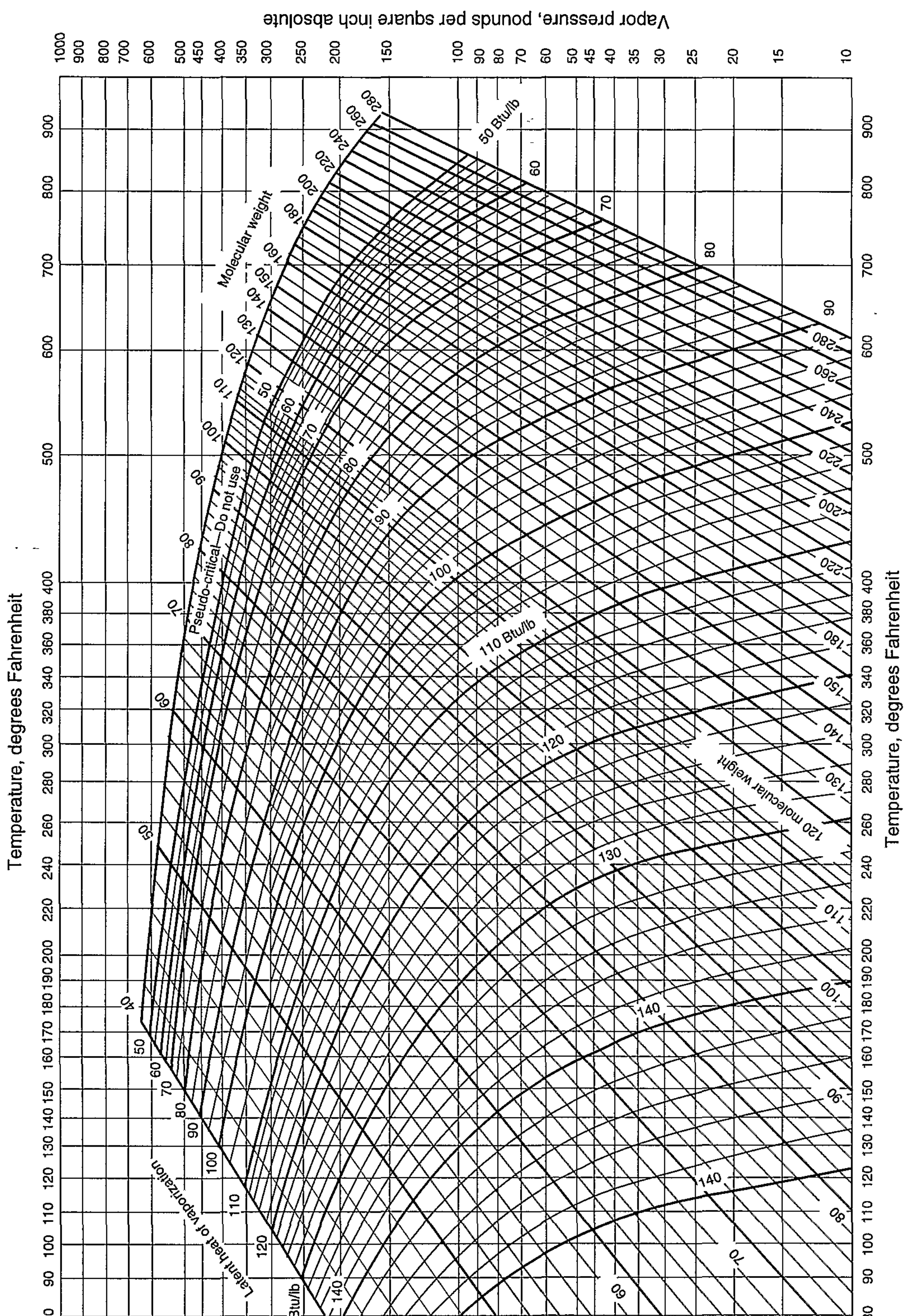


Figure A.1—Vapor Pressure and Heat of Vaporization of Pure Single-Component Paraffin Hydrocarbon Liquids

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 should 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 unless closure of these valves is positively controlled as described in Recommended Practice 520, Part II, Section 2.3 and Section 4.
b. The set pressure of the first pressure relief device to be actuated should 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 should 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 hydroelectric-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 in the preceding.

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

If the procedure outlined in the following list 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. These steps should be taken:

a. The pressure profile should be developed for the processing conditions that will result in the maximum pressure drop (normal 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.

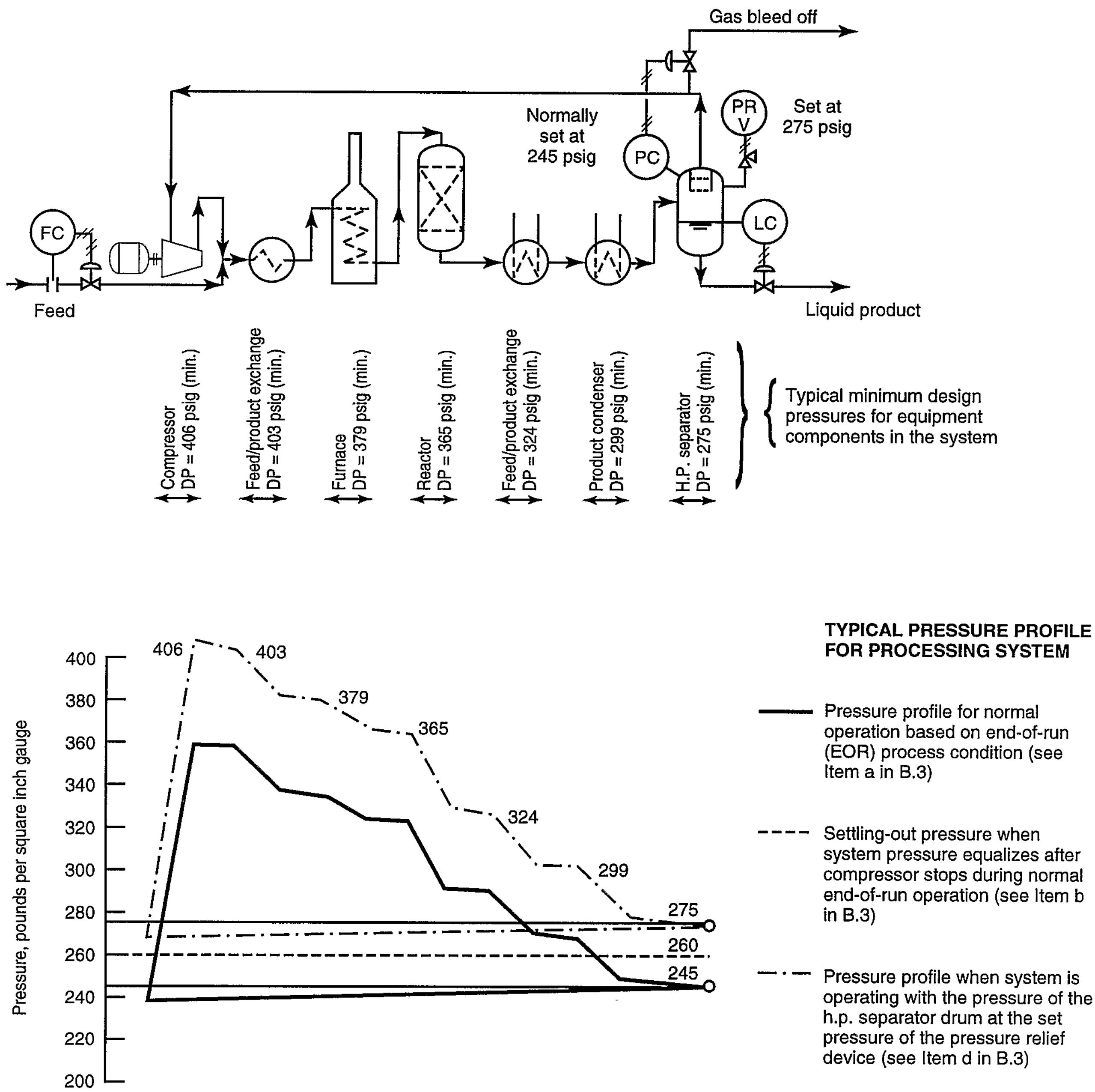


Figure B.1—Typical Flow Scheme of a System Involving a Single Pressure Relief Device Serving Components in a Process System With Typical Pressure Profiles

APPENDIX C—SAMPLE CALCULATIONS FOR SIZING A FLARE STACK

C.1 General

This appendix presents examples of the two methods used to size flare stacks based on the effects of radiation. The first method covered is the simple approach presented in Section 4; the second is the more specific approach using Brzustowski's and Sommer's method. The height and location of the flare stack should be considered, based on gas dispersion if the flame is extinguished (see 4.4.2.4).

C.2 Example 1: Sizing a Flare Stack Using the Simple Approach

C.2.1 BASIC DATA

In this example, the material flowing is hydrocarbon vapors. The flow rate, W, is 100,000 pounds per hour (45,455 kilograms per hour). The average molecular weight of the vapors, M_w is 46.1. The flowing temperature, T, is 760°R (300°F) [422K (149°C)]. The compressibility factor, z, is 1.0. The heat of combustion is 21,500 British thermal units per pound (5×10^3 kilojoules per kilogram). The ratio of the specific heats in the gas, k, is 1.1. The following pressure at the flare tip is 14.7 pounds per square inch absolute (101.3 kilopascals absolute). The design wind velocity is 20 miles per hour (29.3 feet per second) [32.2 kilometers per hour (approximately 8.9 meters per second)].

C.2.2 CALCULATION OF FLARE DIAMETER

The Mach number is determined as follows (see 5.4.3.1.2):

$$\text{Mach} = (1.702)(10^{-5})\left(\frac{W}{P_2 D^2}\right)\left(\frac{zT}{kM_w}\right)^{0.5}$$

In metric units:

$$\text{Mach} = (3.23)(10^{-5})\left(\frac{W}{P_2 D^2}\right)\left(\frac{zT}{kM_w}\right)^{0.5}$$

For Mach = 0.2, the flare diameter is calculated as follows:

$$0.2 = (1.702)(10^{-5})\left(\frac{100,000}{14.7D^2}\right)\sqrt{\frac{760}{(1.1)(46.1)}}$$

$D^2 = 2.24$
$D = 1.5$ feet (inside diameter)

In metric units:

$$0.2 = (3.23)(10^{-5})\left(\frac{45,445}{101.3D^2}\right)\sqrt{\frac{422}{(1.1)(46.1)}}$$

$D^2 = 0.209$
$D = 0.46$ meter (inside diameter)

For Mach = 0.5, the flare diameter is calculated as follows:

$D^2 = 0.897$
$D = 0.95$ foot (inside diameter)

In metric units:

$D^2 = 0.0833$
$D = 0.29$ meter (inside diameter)

C.2.3 CALCULATION OF FLAME LENGTH

The heat liberated, Q, in British thermal units per hour (kilowatts), is calculated as follows (see Figures 8 and 9):

$Q = (100,000)(21,500)$
$= 2.15 \times 10^9$ British thermal units per hour

In metric units:

$Q = (12.6)(50 \times 10^3)$
$= 6.3 \times 10^5$ kilowatts

From Figures 8 and 9, flame length, L, is 170 feet (52 meters). See Figure C.1.

C.2.4 SIMPLE CALCULATION OF FLAME DISTORTION CAUSED BY WIND VELOCITY

The vapor flow rate is determined as follows:

$$\text{Flow} = \left(\frac{100,000}{3,600}\right)\left(\frac{379.1}{46.1}\right)\left(\frac{760}{520}\right)$$

= 333.9 actual cubic feet per second

In metric units:

$$\text{Flow} = \left(\frac{45,455}{3,600}\right)\left(\frac{22.4}{46.1}\right)\left(\frac{422}{273}\right)$$

= 9.46 actual cubic meters per second

The flame distortion caused by wind velocity is calculated as follows (see Figure 10):

$$\frac{U_\infty}{U_j} = \frac{\text{Wind velocity}}{\text{Flare tip velocity}}$$

The flare tip exit velocity, U_j, may be determined as follows (see C.3.3 for another method of calculating U_j):

$$U_j = \frac{\text{Flow}}{(\pi d^2)/4}$$

For Mach = 0.2:

$$U_j = \frac{333.9}{[\pi(1.5)^2]/4}$$
$$= 189 \text{ feet per second}$$

In metric units:

$$U_j = \frac{9.46}{[\pi(0.46)^2]/4}$$
$$= 56.9 \text{ meters per second}$$

For Mach = 0.5:

$$U_j = \frac{333.9}{[\pi(0.95)^2]/4}$$
$$= 471 \text{ feet per second}$$

In metric units:

$$U_j = \frac{9.46}{[\pi(0.29)^2]/4}$$
$$= 143.2 \text{ meters per second}$$

At Mach = 0.2:

$$\frac{U_\infty}{U_j} = \frac{29.3}{189} = 0.155$$
$$\Sigma\frac{\Delta y}{L} = 0.35$$
$$\Sigma\frac{\Delta x}{L} = 0.85$$
$$\Sigma\Delta y = (0.35)(170) = 59.5 \text{ feet}$$
$$\Sigma\Delta x = (0.85)(170) = 144.5 \text{ feet}$$

In metric units:

$$\frac{U_\infty}{U_j} = \frac{8.9}{56.9} = 0.156$$
$$\Sigma\frac{\Delta y}{L} = 0.35$$
$$\Sigma\frac{\Delta y}{L} = 0.85$$
$$\Sigma\Delta y = (0.35)(52) = 18.2 \text{ meters}$$
$$\Sigma\Delta x = (0.85)(52) = 44.2 \text{ meters}$$

At Mach = 0.5:

$$\frac{U_\infty}{U_j} = \frac{29.3}{471} = 0.062$$
$$\Sigma\frac{\Delta y}{L} = 0.53$$
$$\Sigma\frac{\Delta x}{L} = 0.72$$
$$\Sigma\Delta y = (0.53)(170) = 90.1 \text{ feet}$$
$$\Sigma\Delta x = (0.72)(170) = 122.4 \text{ feet}$$

In metric units:

$$\frac{U_\infty}{U_j} = \frac{8.9}{143.2} = 0.062$$
$$\Sigma\frac{\Delta y}{L} = 0.53$$
$$\Sigma\frac{\Delta x}{L} = 0.72$$
$$\Sigma\Delta y = (0.53)(52) = 27.6 \text{ meters}$$
$$\Sigma\Delta x = (0.72)(52) = 37.4 \text{ meters}$$

C.2.5 CALCULATION OF REQUIRED FLARE STACK HEIGHT

For the basis of the calculations used in this section, see 4.4.2.3. See Figure C.1 for dimensional references.

The design basis for these calculations is as follows:

The fraction of heat radiated, F, is 0.3. The heat liberated (see C.2.3), Q, is 2.15×10^9 British thermal units per hour (6.3×10^5 kilowatts). The maximum allowable radiation, K, at 150 feet (45.7 meters) from the flare stack is 2,000 British thermal units per hour per square foot (6.3 kilowatts per square meter).

In Equation 20 in 4.4.2.3.3, the value of τ should be assumed to be 1.0. The distance from the flame center to the grade-level boundary (that is, the object being considered), D, is then calculated as follows:

$$D = \sqrt{\frac{\tau F Q}{4\pi K}}$$
$$= \sqrt{\frac{(0.3)\ (2.15)\ (10^9)}{(4)\ (3.14)\ (2,000)}} = 160.2\text{feet}$$

In metric units:

$$D = \sqrt{\frac{\tau F Q}{4\pi K}}$$
$$= \sqrt{\frac{(0.3)\ (6.3)\ (10^5)}{(4)\ (3.14)\ (6.3)}} = 48.9\text{meters}$$

The physical arrangement shown in Figure C.1 is the basis of the remaining calculations in this section.

At Mach = 0.2, the flare stack height, H, is calculated as follows:

$$H' = H + \tfrac{1}{2}\Sigma\Delta y$$

$$R' = R - \tfrac{1}{2}\Sigma\Delta x$$

$$\Sigma\Delta y = 59.5 \text{ feet}$$

$$\Sigma\Delta x = 144.5 \text{ feet}$$

(See C.2.4.)

$$R' = 150 - \tfrac{1}{2}(144.5)$$

$$= 78 \text{ feet}$$

$$D^2 = R'^2 + H'^2$$

$$(160)^2 = 78^2 + H'^2$$

$$H'^2 = 25{,}600 - 6{,}084$$

$$= 19{,}516$$

$$H' = 140 \text{ feet}$$

$$H = 140 - \tfrac{1}{2}(60)$$

$$= 110 \text{ feet}$$

In metric units:

$$H' = H + \tfrac{1}{2}\Sigma\Delta y$$

$$R' = R - \tfrac{1}{2}\Sigma\Delta x$$

$$\Sigma\Delta y = 18.2 \text{ meters}$$

$$\Sigma\Delta x = 44.2 \text{ meters}$$

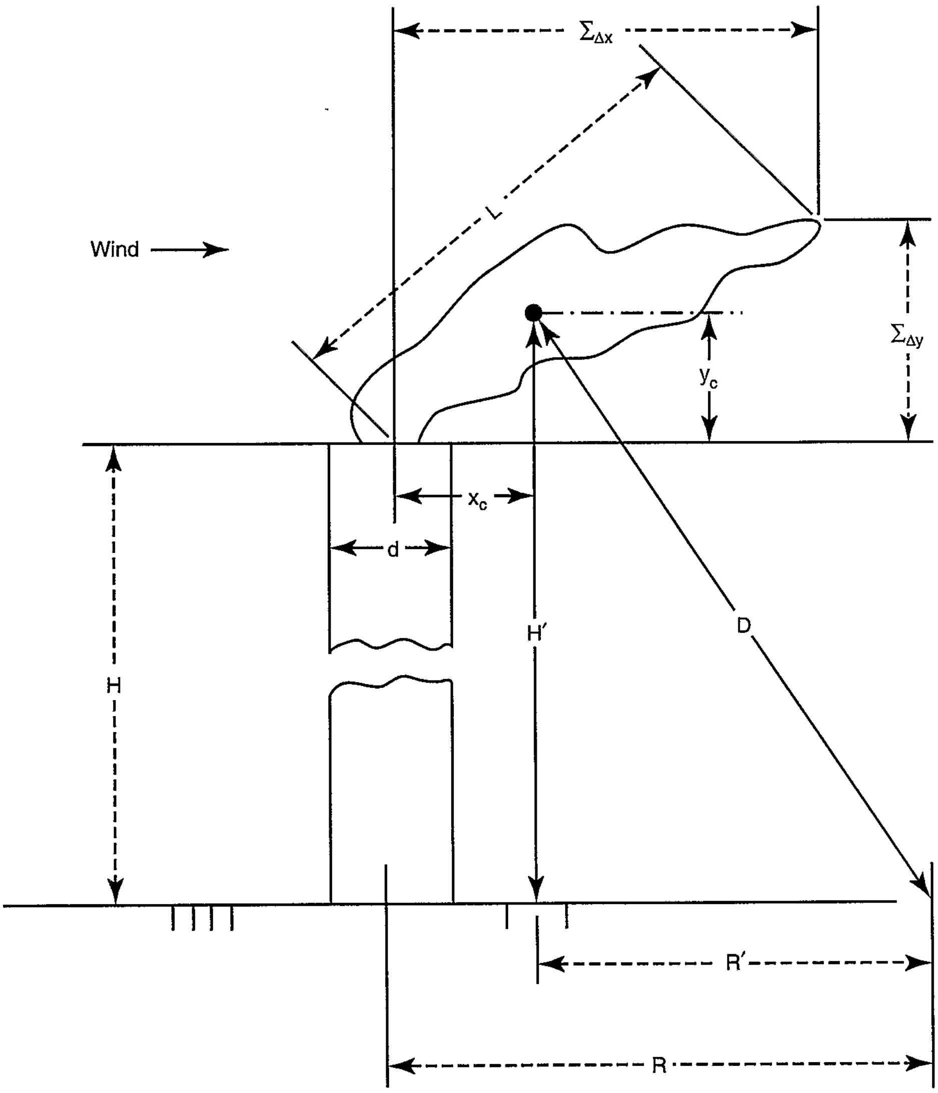


Figure C.1—Dimensional References for Sizing a Flare Stack

(See C.2.4.)

$$R' = 45.7 - \tfrac{1}{2}(44.0)$$
$$= 23.7 \text{ meters}$$
$$D^2 = R'^2 + H'^2$$
$$(48.9)^2 = (23.7)^2 + H'^2$$
$$H'^2 = 2{,}391.2 - 561.7$$
$$= 1829.5$$
$$H' = 42.8 \text{ meters}$$
$$H = 42.8 - \tfrac{1}{2}(18.2)$$
$$= 33.7 \text{ meters}$$

At Mach = 0.5, H is calculated as follows:

$$H' = H + \tfrac{1}{2}\Sigma\Delta y$$
$$R' = R - \tfrac{1}{2}\Sigma\Delta x$$
$$\Sigma\Delta y = 90.1 \text{ feet}$$
$$\Sigma\Delta x = 122.4 \text{ feet}$$

(See C.2.4.)

$$R' = 150 - \tfrac{1}{2}(122)$$
$$= 89 \text{ feet}$$
$$D^2 = R'^2 + H'^2$$
$$(160)^2 = (89)^2 + H'$$
$$H'^2 = 25{,}600 - 7{,}921$$
$$= 17{,}679$$
$$H' = 133 \text{ feet}$$
$$H = 133 - \tfrac{1}{2}(90)$$
$$= 88 \text{ feet}$$

In metric units:

$$H' = H + \tfrac{1}{2}\Sigma\Delta y$$
$$R' = R - \tfrac{1}{2}\Sigma\Delta x$$
$$\Sigma\Delta y = 27.6 \text{ meters}$$
$$\Sigma\Delta x = 37.4 \text{ meters}$$

(See C.2.4.)

$$R' = 45.7 - \tfrac{1}{2}(37.4)$$
$$= 27.0 \text{ meters}$$
$$D^2 = R'^2 + H'^2$$
$$(48.9)^2 = (27.0)^2 + H'^2$$
$$H'^2 = 2391.2 - 729$$
$$= 1662.2$$
$$H' = 40.8 \text{ meters}$$
$$H = 40.8 - \tfrac{1}{2}(27.6)$$
$$= 27 \text{ meters}$$

C.3 Example 2: Sizing a Flare Using Brzustowski's and Sommer's Approach [see 4.7, Reference 23].

C.3.1 BASIC DATA

In this example, the material flowing is hydrocarbon vapors. The flow rate, W, is 1,000,000 pounds per hour (126 kilograms per second). The molecular weight of the flare gas, M_j, is 46.1, and the molecular weight of air, M_∞, is 29. The normal average wind speed, U_∞, is 20 miles per hour (29.3 feet per second) [32.2 kilometers per hour (8.9 meters per second)]. The velocity of the flare gas at the flare tip, U_j, is measured in feet per second (meters per second). The inside diameter of the flare tip, d_j, is measured in feet (meters). The pressure of the flare gas at the flare tip, P_j, is 15.7 pounds per square inch absolute (108 kilopascals absolute). The average relative humidity, γ, is 50 percent. The heat of combustion is 21,500 British thermal units per pound (50×10^3 kilojoules per kilogram). The ratio of the specific heats in the gas, k, is 1.1. The compressibility factor, z, is 1.0. The lower-explosive-limit concentration of the flare gas in air, C_L, measured as a volume fraction, is 0.021 (see C.3.6, Note 1). The temperature of the flare gas, T_j, is 760°R (422°K), and temperature of the air, T_∞, is 520°R (289°K).[1]

The fraction by which the flame radiation is reduced when transmitted through the atmosphere is indicated by τ. The fraction of heat radiated is indicated by F. The heat release, Q, is measured in British thermal units per hour (kilowatts), and the allowable radiation intensity, K, is measured in British thermal units per hour per square foot (kilowatts per square meter).

C.3.2 CALCULATION OF FLARE DIAMETER

The Mach number is determined as follows (see 5.4.1.3.2):

$$\text{Mach} = (1.702)(10^{-5})\frac{W}{P_j d_j^2}\sqrt{\frac{zT_j}{kM_j}} \qquad (33)$$

In metric units:

$$\text{Mach} = 3.23 \times 10^{-5}\frac{W}{P_j d_j^2}\sqrt{\frac{zT_j}{kM_j}} \qquad (34)$$

For Mach = 0.5, the flare diameter is calculated as follows:

$$0.5 = (1.702)(10^{-5})\frac{1{,}000{,}000}{15.7 d_j^2}\sqrt{\frac{760}{(1.1)(46.1)}}$$

$$d_j^2 = 8.39$$

$$d_j = 2.90 \text{ feet}$$

[1]For more information about the method used in this example, see T.A. Brzustowski and E.C. Sommer, Jr., (see 4.7 reference 23).

In metric units:

$$0.5 = 3.23 \times 10^{-5} \frac{454,545}{108 d_j^2} \sqrt{\frac{422}{(1.1)(46.1)}}$$

$$d_j^2 = 0.78$$

$$d_j = 0.88 \text{ meter}$$

C.3.3 LOCATION OF FLAME CENTER

The tip exit velocity, U_j, is calculated as follows:

$$\text{sonic velocity} = 223\left(\frac{kT_j}{M_j}\right)^{0.5}$$

$$= 223\left(\frac{(1.1)\ (760)}{46.1}\right)^{0.5}$$

$$= 949.64 \text{ feet per second}$$

$$U_j = \text{jet Mach number} \times \text{sonic velocity}$$

$$= (0.5)(949.64)$$

$$= 475 \text{ feet per second}$$

In metric units:

$$\text{sonic velocity} = 91.2\left(\frac{kT_j}{M_j}\right)^{0.5}$$

$$= 91.2\left(\frac{(1.1)\ (422)}{46.1}\right)^{0.5}$$

$$= 289.4 \text{ meters per second}$$

$$U_j = \text{jet Mach number} \times \text{sonic velocity}$$

$$=(0.5)(289.4)$$

$$=144.7 \text{ meters per second}$$

The lower-explosive-limit concentration parameter for the flare gas, $\overline{C}_L$, is calculated as follows:

$$\overline{C}_L = C_L\left(\frac{U_j}{U_\infty}\right)\left(\frac{M_j}{M_\infty}\right)$$

$$= (0.021)\left(\frac{475}{29.3}\right)\left(\frac{46.1}{29}\right)$$

$$= 0.542$$

In metric units:

$$\overline{C}_L = C_L\left(\frac{U_j}{U_\infty}\right)\left(\frac{M_j}{M_\infty}\right)$$

$$= (0.021)\left(\frac{144.7}{8.9}\right)\left(\frac{46.1}{29}\right)$$

$$= 0.542$$

The parameter for jet thrust and wind thrust, d_jR, is calculated as follows (see C.3.6, Note 2):

$$d_jR = d_j\left(\frac{U_j}{U_\infty}\right)\left(\frac{T_\infty M_j}{T_j}\right)^{0.5}$$

$$d_jR = (2.9)\left(\frac{475}{29.3}\right)\left[\frac{(520)(46.1)}{760}\right]^{0.5} = 264$$

In metric units:

$$d_jR = d_j\left(\frac{U_j}{U_\infty}\right)\left(\frac{T_\infty M_j}{T_j}\right)^{0.5}$$

$$= (0.88)\left(144.7\frac{475}{8.9}\right)\left[\frac{(289)(46.1)}{422}\right]^{0.5} = 80.4$$

The horizontal and vertical distances from the flare tip to the flame center, x_c and y_c, respectively, are determined as follows:

From Figure C.2.A,

$$x_c = 58 \text{ feet}$$

From Figure C.2.B,

$$x_c = 17.7 \text{ meters}$$

From Figure C.3.A,

$$y_c = 100 \text{ feet}$$

From Figure C.3.B,

$$y_c = 30 \text{ meters}$$

C.3.4 CALCULATION OF THE DISTANCE FROM THE FLAME CENTER TO THE OBJECT OR POINT BEING CONSIDERED

The design basis for this calculation is as follows: The fraction of heat radiated, F, is 0.3. The heat liberated (see C.2.3), Q, is 2.15×10^{10} British thermal units per hour (6.3×10^6 kilowatts). The assumed maximum allowable radiation (see 4.4.1.3), K, is 3000 British thermal units per hour per square foot (9.5 kilowatts per square meter).

In Equation 29 in Section 4, the value of τ should be assumed to be 1.0 (see C.3.6, Notes 3 and 4). The distance from the flame center to the object or point being considered (that is, the distance to the limit of the radiant heat intensity, such as grade level, an equipment platform, or a plant boundary), D, is then calculated as follows:

$$D = \sqrt{\frac{\tau FQ}{4\pi K}}$$

$$= \sqrt{\frac{(1.0)(0.3)(2.15)(10^{10})}{(4)(3.14)(3000)}}$$

$$= 414 \text{ feet}$$

In metric units:

$$D = \sqrt{\frac{\tau FQ}{4\pi K}}$$

$$= \sqrt{\frac{(1.0)(0.3)(6.3)(10^{6})}{(4)(3.14)(9.5)}} = 126 \text{ meters}$$

Thus, at a maximum allowable K of 3,000 British thermal units per hour per square foot (9.5 kilowatts per square meter), $D = 414$ feet (126 meters) from the flame center.

Similarly, at a maximum allowable K of 2,000 British thermal units per hour per square foot (6.3 kilowatts per square meter), $D = 507$ feet (154.5 meters) from the flame center.

C.3.5 DETERMINATION OF FLARE STACK HEIGHT

The limiting height of the flare stack depends on the design criteria selected and the facilities near the flare. At grade level, directly under the flame center, with K up to 3,000 British thermal units per hour per square foot (9.5 kilowatts per square meter), the minimum flare stack height, H, is determined as follows:

$$H = D - y_c$$

$$= 414 - 100$$

$$= 314 \text{ feet}$$

In metric units:

$$H = D - y_c$$

$$= 126 - 30$$

$$= 96 \text{ meters}$$

At grade level, at a radius of 150 feet (45.7 meters) from the base of the flare stack, with K up to 2,000 British thermal units per hour per square foot (6.3 kilowatts per square meter) and following the general arrangement shown in Figure C.1, H is determined as follows:

$$H' = H + y_c$$

$$R' = R - x_c$$

$$D^2 = R'^2 + H'^2$$

$$= (507)^2$$

$$D^2 = (R - x_c)^2 + (H + y_c)^2$$

$$(H + 100)^2 = (507)^2 - (150 - 58)^2$$

$$= 257{,}049 - 8{,}464$$

$$= 248{,}585$$

$$H = 499 - 100$$

$$= 399 \text{ feet}$$

In metric units:

$$H' = H + y_c$$

$$R' = R - x_c$$

$$D^2 = R'^2 + H'^2$$

$$= (154.5)^2$$

$$D^2 = (R - x_c)^2 + (H + y_c)^2$$

$$(H + 30)^2 = (154.5)^2 - (45.7 - 17.7)^2$$

$$= 23{,}870.3 - 784$$

$$= 23{,}086.3$$

$$H = 152 - 30 = 122 \text{ meters}$$

$$= 122 \text{ meters}$$

Similarly, other alternatives should be reviewed. For a new flare installation within a maze of existing facilities, the heat intensity can be calculated directly as follows:

$$K = \frac{\tau FQ}{4\pi D^2}$$

C.3.6 EXPLANATORY NOTES

1. Lower explosive limits for pure components may be obtained from AGA Catalog No. XK0775 or from NFPA 325M. The lower explosive limits of mixtures may be calculated as follows:

$$C_L = \Sigma\left(\frac{y_1}{C_{L_1}}\right) + \left(\frac{y_2}{C_{L_2}}\right) + \dots + \left(\frac{y_n}{C_{L_n}}\right)^{-1}$$

Where:

C_{L_1} = lower explosive concentration of the component in air.

y_1 = mole fraction (or volume fraction) of the component in the mixture.

2. The graphs in Figures C.2.A through C.3.B are based on two independent variables, C_L and a modified form of d_jR. The variable d_jR was modified (from that proposed in the Brzustowski and Sommer article) to include gas and air temperatures and molecular weights instead of densities. The ideal-gas law was assumed. Some adjustments were

made in the graph curves over the C_L range from 0.5 to 1.5 to smooth out discontinuities. No significant difference, compared with hand-calculated results, is introduced with the data smoothing. See the original article (4.7, Reference 23) for the details of the hand-calculation procedure.

3. Brzustowski and Sommer recommend the use of the fraction of heat intensity transmitted, τ, to correct the radiation impact. The following is quoted from the original article (see 4.7, Reference 23):

Note: In the case of flares, atmospheric absorption attenuates K by about 10–20 percent over distances of 500 feet (150 meters). The empirical equation presented here as Equation C-1 was obtained by cross-plotting absorptivities calculated from the Hottel charts. It is strictly applicable only when a luminous hydrocarbon flame is radiating at 2,240°F (1227°C), the dry bulb ambient temperature is 80°F (27°C), the relative humidity is more than 10 percent, and the distance from the flame is between 100 feet and 500 feet (30 meters and 50 meters); however, the equation can be used to estimate the order of magnitude of τ under a wider range of conditions.

$$\tau = 0.79\left(\frac{100}{r}\right)^{1/16}\left(\frac{100}{D}\right)^{1/16} \qquad \text{(C-1)}$$

In metric units:

$$\tau = 0.79\left(\frac{100}{r}\right)^{1/16}\left(\frac{30.5}{D}\right)^{1/16} \qquad \text{(C-2)}$$

Where:

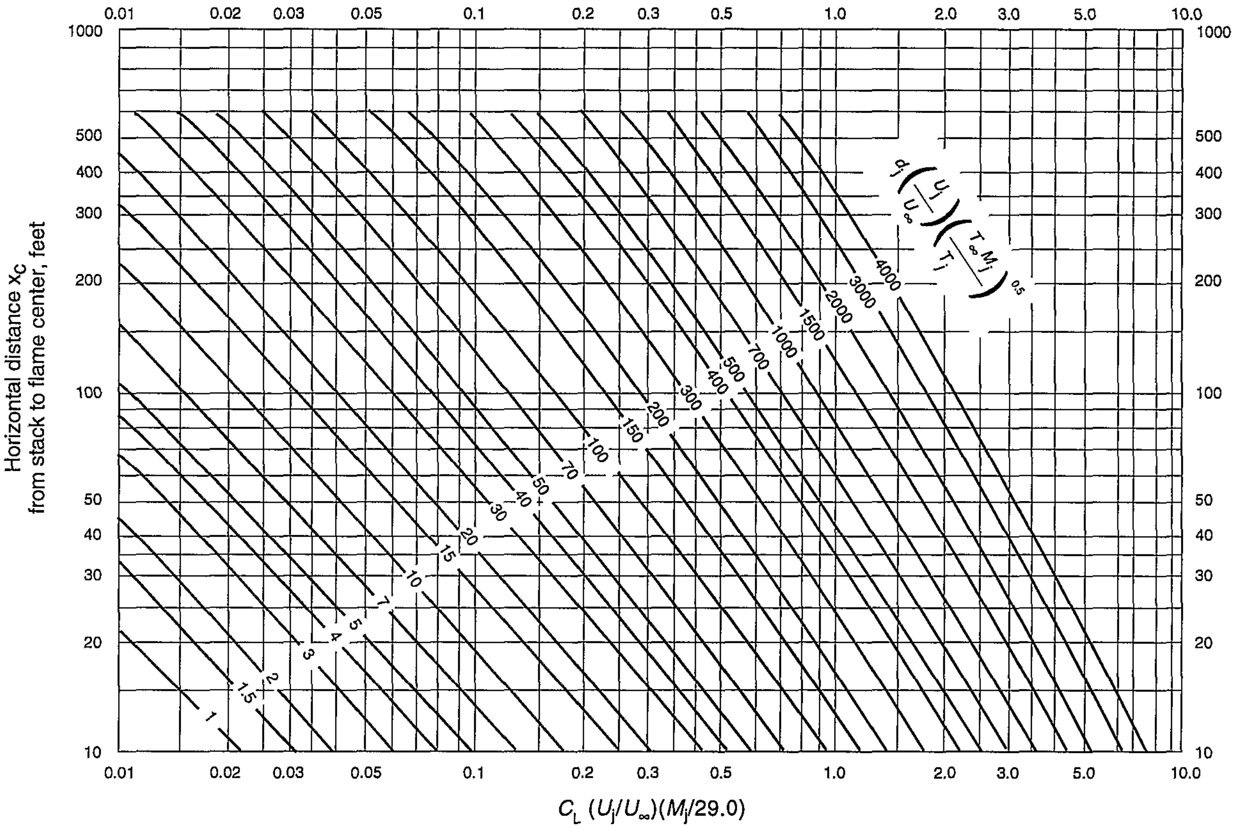
τ = fraction of K transmitted through the atmosphere.

r = relative humidity, percent.

D = distance from flame to illuminated area, in feet (meters).

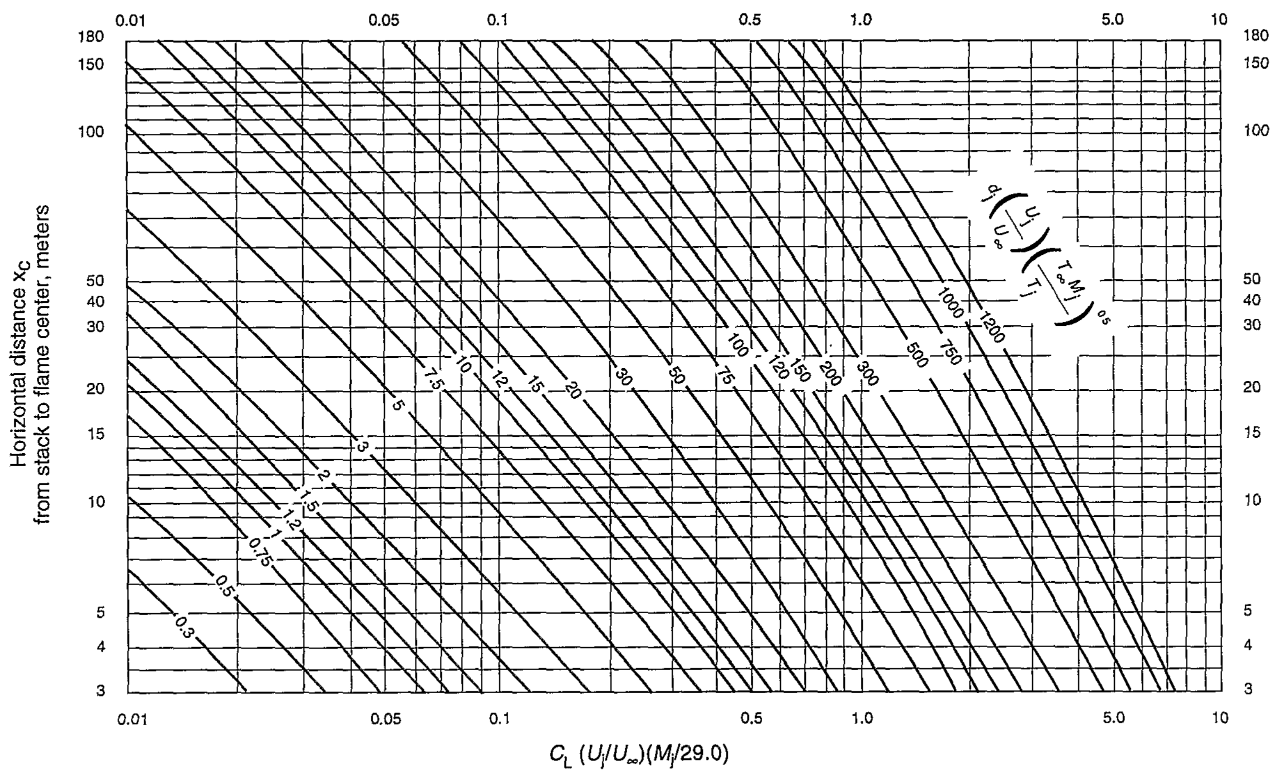
Equations C-1 and C-2 should prove adequate for most flare gases, except H_2 and H_2S which burn with little or no luminous radiation. If the anticipated design conditions are very different from those under which Equations C-1 and C-2 were derived, the designer should go back to the Hottel charts.

4. Where steam injection is used at a rate of about 0.3 pound (kilogram) of steam per pound (kilogram) of flare gas, then the fraction of heat radiated, F, is decreased by 20 percent.



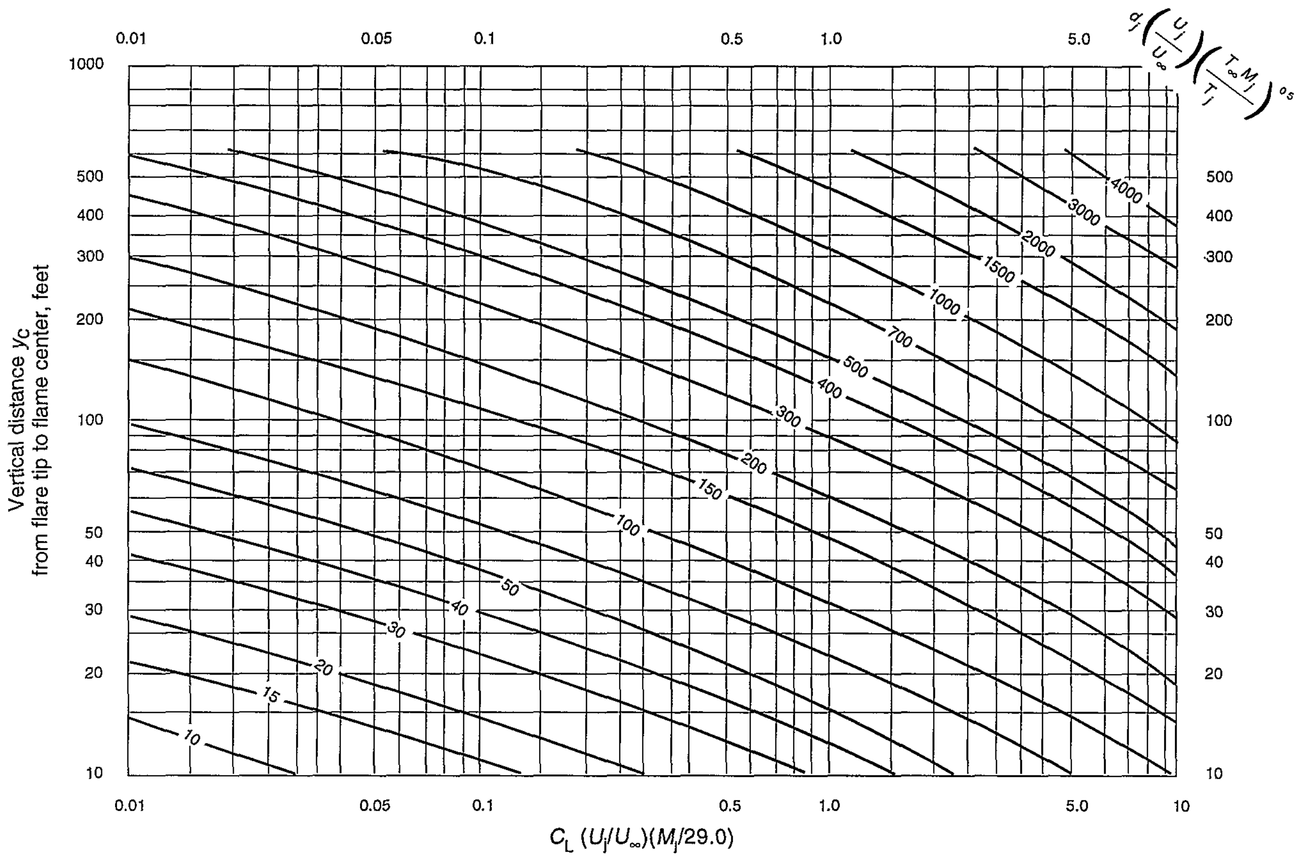
C_L = lower explosive limit, volume fraction.
d_j = inside diameter of flare tip, in feet.
M_j = molecular weight of the gas.
T_j = temperature of the gas, in degrees Rankine.
T_∞ = temperature of the air, in degrees Rankine.
U_j = gas exit velocity, in feet per second.
U_∞ = average wind velocity, in feet per second.

Figure C.2.A—Flame Center for Flares and Ignited Vents: Horizontal Distance x_c (Customary Units)



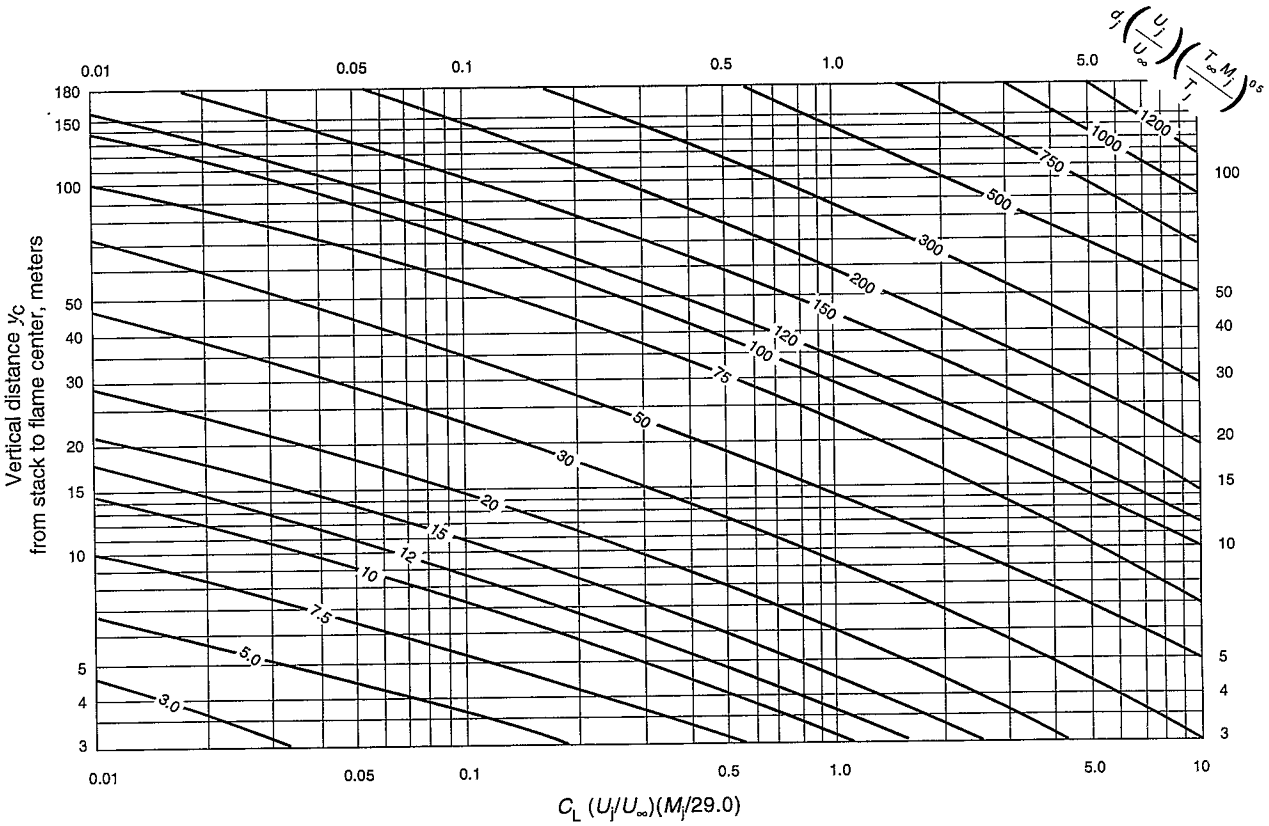
C_L = lower explosive limit, volume fraction.
d_j = inside diameter of flare tip, in meters.
M_j = molecular weight of the gas.
T_j = temperature of the gas, in degrees Kelvin.
T_∞ = temperature of the air, in degrees Kelvin.
U_j = gas exit velocity, in meters per second.
U_∞ = average wind velocity, in meters per second.

Figure C.2.B—Flame Center for Flares and Ignited Vents: Horizontal Distance x_c (SI Units)



C_L = lower explosive limit, volume fraction.
d_j = inside diameter of flare tip, in feet.
M_j = molecular weight of the gas.
T_j = temperature of the gas, in degrees Rankine.
T_x = temperature of the air, in degrees Rankine.
U_j = gas exit velocity, in feet per second.
U_x = average wind velocity, in feet per second.

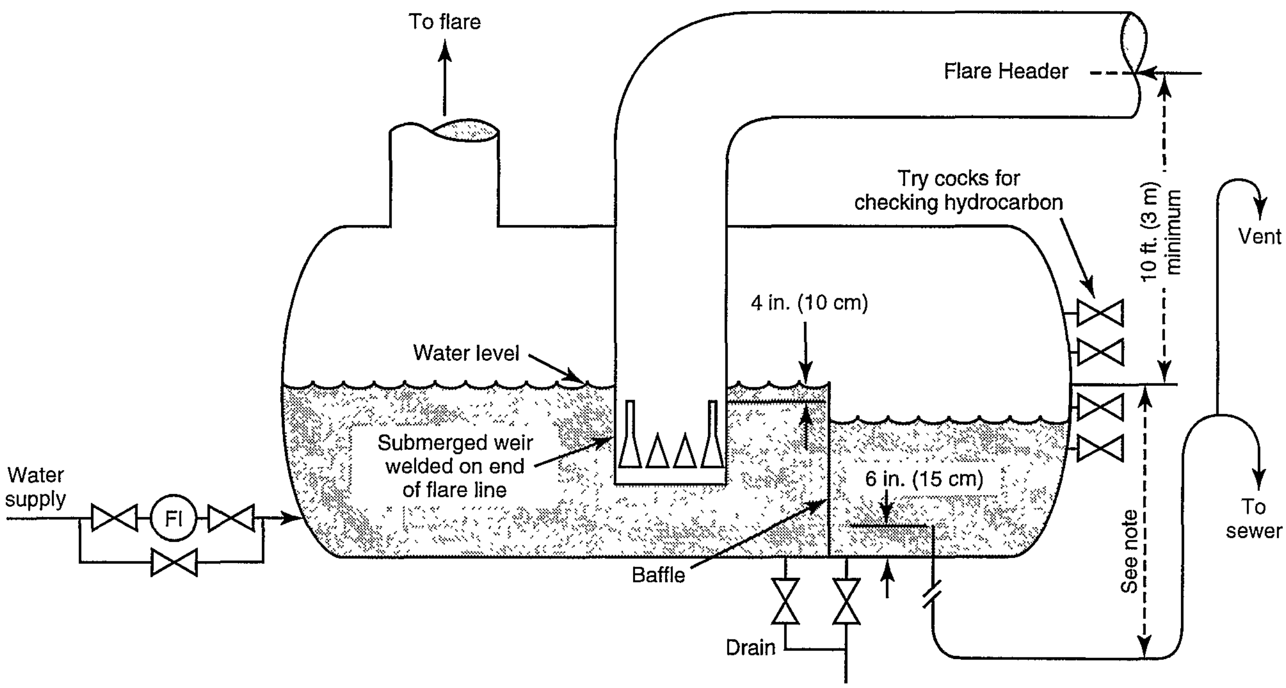
Figure C.3.A—Flame Center for Flares and Ignited Vents: Vertical Distance y_c (Customary Units)



C_L = lower explosive limit, volume fraction.
d_j = inside diameter of flare tip, in meters.
M_j = molecular weight of the gas.
T_j = temperature of the gas, in degrees Kelvin.
T_x = temperature of the air, in degrees Kelvin.
U_j = gas exit velocity, in meters per second.
U_x = average wind velocity, in meters per second.

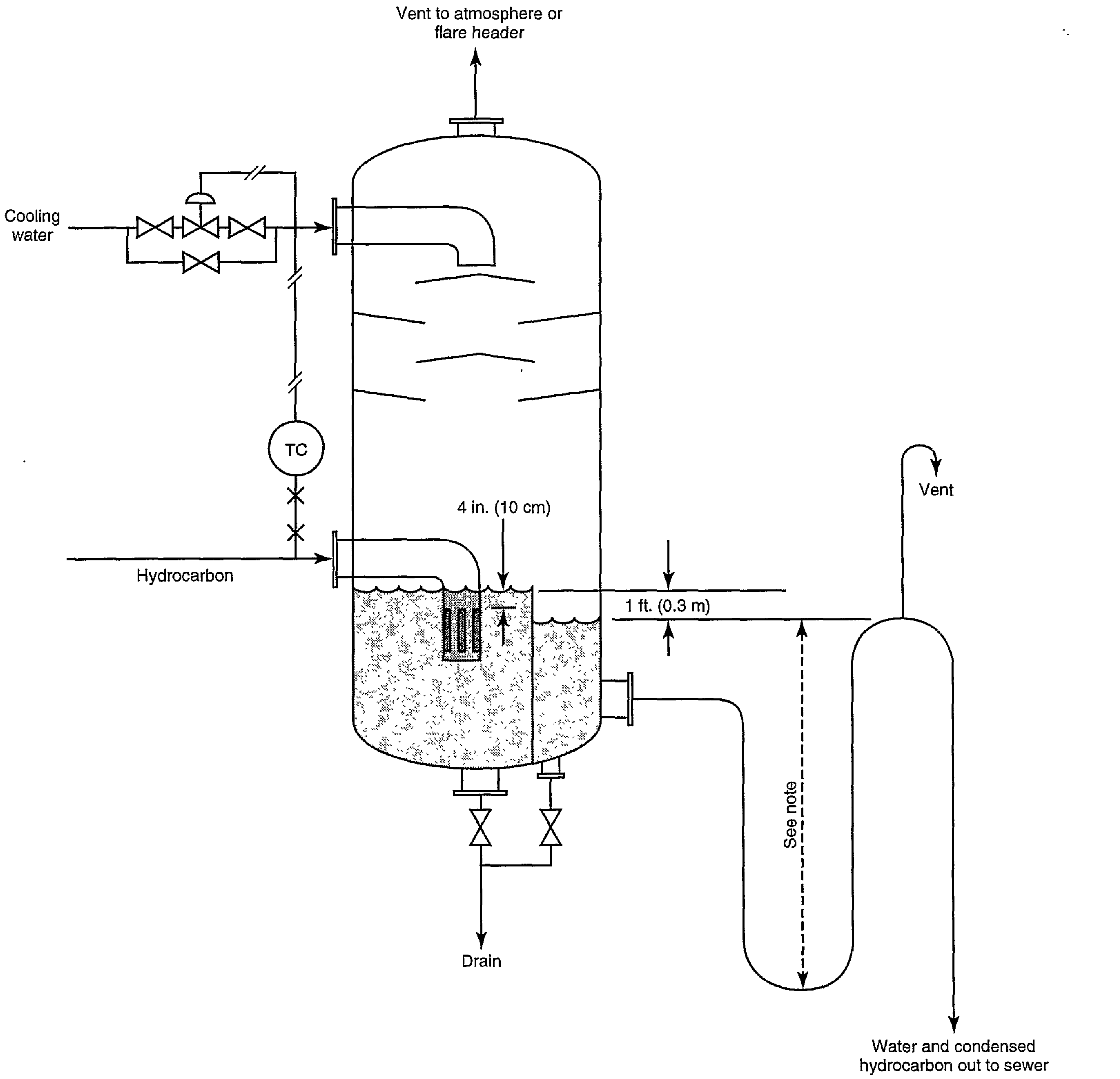
Figure C.3.B—Flame Center for Flares and Ignited Vents: Vertical Distance y_c (SI Units)

APPENDIX D—TYPICAL DETAILS AND SKETCHES



Note: The sewer seal should be designed for a minimum of 175 percent of the drum's maximum operating pressure.

Figure D.1—Flare Stack Seal Drum



Note: This figure represents an operable system arrangement and its components. The arrangement of the system will vary with the performance required. Correspondingly, the selection of types and quantities of components, as well as their applications, should match the needs of the particular plant and its specifications.

Figure D.2—Quench Drum

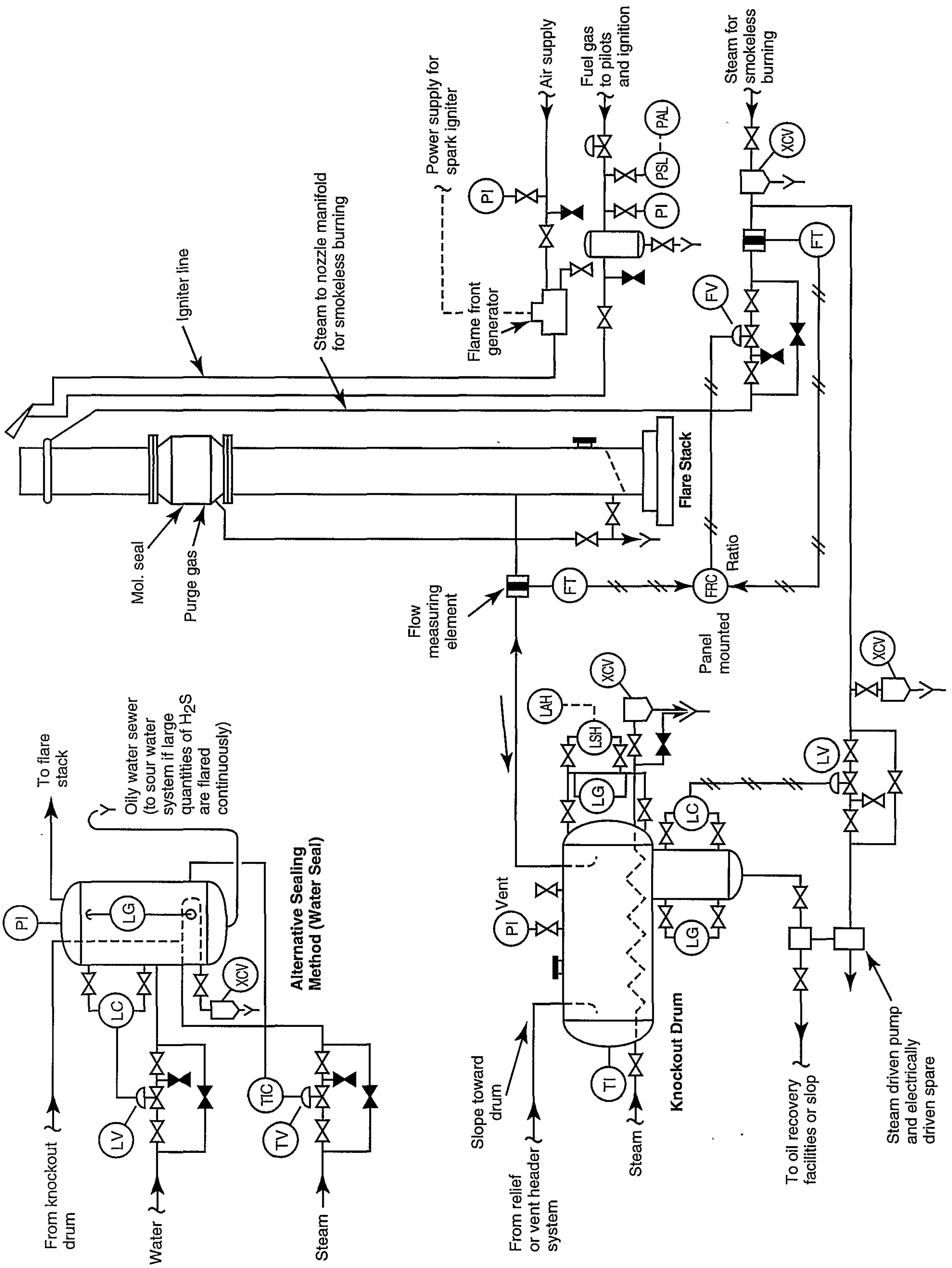


Figure D.3—Typical Flare Installation