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Sections 1-16

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SECTION 7

Separators and Filters

PRINCIPLES OF SEPARATION

Three principles used to achieve physical separation of gas and liquids or solids are momentum, gravity settling, and coa-

lescing. Any separator may employ one or more of these principles, but the fluid phases must be "immiscible" and have different densities for separation to occur.

FIG. 7-1

Nomenclature

| | |
|---|---|
| A = area, ft^2 | M = mass flow, lb/sec |
| A_p = particle or droplet cross sectional area, ft^2 | M_p = mass of droplet or particle, lb |
| C = empirical constant for separator sizing, ft/hr | MW = molecular weight, lb/lb mole |
| C^* = empirical constant for liquid-liquid separators, $(\text{bbl} \cdot \text{cp})/(\text{ft}^2 \cdot \text{day})$ | P = system pressure, psia |
| C' = drag coefficient of particle, dimensionless (Fig. 7-3) | Q = estimated gas flow capacity, MMscfd per ft^2 of filter area |
| ρ_g = gas phase density, lb/ft^3 | Q_A = actual gas flow rate, ft^3/sec |
| ρ_l = liquid phase density, droplet or particle, lb/ft^3 | R = gas constant, $10.73 (\text{psia} \cdot \text{ft}^3)/(\text{°R} \cdot \text{lb mole})$ |
| D_i = separator inlet nozzle diameter, in. | Re = Reynolds number, dimensionless |
| D_p = droplet diameter, ft | S_h = specific gravity of heavy liquid, water = 1.0 |
| D_v = inside diameter of vessel, ft | S_L = specific gravity of light liquid, water = 1.0 |
| G_m = maximum allowable gas mass-velocity necessary for particles of size D_p to drop or settle out of gas, $\text{lb}/(\text{hr} \cdot \text{ft}^2)$ | T = system temperature, °R |
| g = acceleration due to gravity, 32.2 ft/sec^2 | t = retention time, minutes |
| H_L = width of liquid interface area, ft | U = volume of settling section, bbl |
| J = gas momentum, $\text{lb}/(\text{ft} \cdot \text{sec}^2)$ | μ = viscosity of continuous phase, cp |
| K = empirical constant for separator sizing, ft/sec | V_t = critical or terminal gas velocity necessary for particles of size D_p to drop or settle out of gas, ft/sec |
| K_{CR} = proportionality constant from Fig. 7-4 for use in Eq 7-5, dimensionless | W = total liquid flow rate, bbl/day |
| L = seam to seam length of vessel, ft | W_c = flow rate of light condensate liquid, bbl/day |
| L_L = length of liquid interface area, ft | Z = compressibility factor, dimensionless |

Filter Separators: A filter separator usually has two compartments. The first compartment contains filter-coalescing elements. As the gas flows through the elements, the liquid particles coalesce into larger droplets and when the droplets reach sufficient size, the gas flow causes them to flow out of the filter elements into the center core. The particles are then carried into the second compartment of the vessel (containing a vane-type or knitted wire mesh mist extractor) where the larger droplets are removed. A lower barrel or boot may be used for surge or storage of the removed liquid.

Flash Tank: A vessel used to separate the gas evolved from liquid flashed from a higher pressure to a lower pressure.

Line Drip: Typically used in pipelines with very high gas-to-liquid ratios to remove only free liquid from a gas stream, and not necessarily all the liquid. Line drips provide a place for free liquids to separate and accumulate.

Liquid-Liquid Separators: Two immiscible liquid phases can be separated using the same principles as for gas and liquid separators. Liquid-liquid separators are fundamentally the same as gas-liquid separators except that they

must be designed for much lower velocities. Because the difference in density between two liquids is less than between gas and liquid, separation is more difficult.

Scrubber or Knockout: A vessel designed to handle streams with high gas-to-liquid ratios. The liquid is generally entrained as mist in the gas or is free-flowing along the pipe wall. These vessels usually have a small liquid collection section. The terms are often used interchangeably.

Separator: A vessel used to separate a mixed-phase stream into gas and liquid phases that are "relatively" free of each other. Other terms used are scrubbers, knockouts, line-drips, and decanters.

Slug Catcher: A particular separator design able to absorb sustained in-flow of large liquid volumes at irregular intervals. Usually found on gas gathering systems or other two-phase pipeline systems. A slug catcher may be a single large vessel or a manifolded system of pipes.

Three Phase Separator: A vessel used to separate gas and two immiscible liquids of different densities (e.g. gas, water, and oil).

Momentum

Fluid phases with different densities will have different momentum. If a two phase stream changes direction sharply, greater momentum will not allow the particles of the heavier phase to turn as rapidly as the lighter fluid, so separation occurs. Momentum is usually employed for bulk separation of the two phases in a stream.

Gravity Settling

Liquid droplets will settle out of a gas phase if the gravitational force acting on the droplet is greater than the drag force of the gas flowing around the droplet (see Fig. 7-2). These forces can be described mathematically using the terminal or free settling velocity.

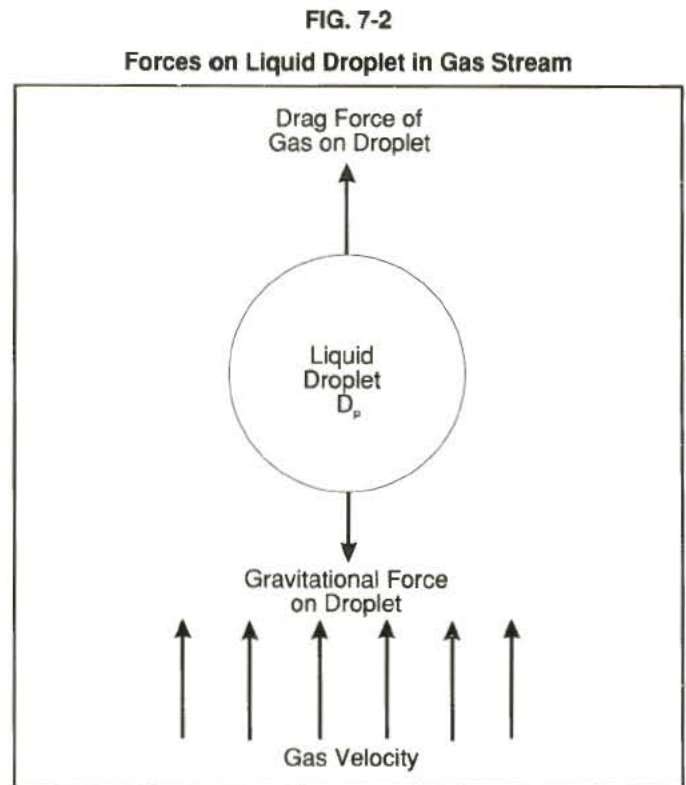
$$V_t = \sqrt{\frac{2 g M_p (\rho_l - \rho_g)}{\rho_l \rho_g A_p C'}} = \sqrt{\frac{4 g D_p (\rho_l - \rho_g)}{3 \rho_g C'}} \quad \text{Eq 7-1}$$

The drag coefficient has been found to be a function of the shape of the particle and the Reynolds number of the flowing gas. For the purpose of this equation particle shape is considered to be a solid, rigid sphere.

Reynolds number is defined as:

$$Re = \frac{1,488 D_p V_t \rho_g}{\mu} \quad \text{Eq 7-2}$$

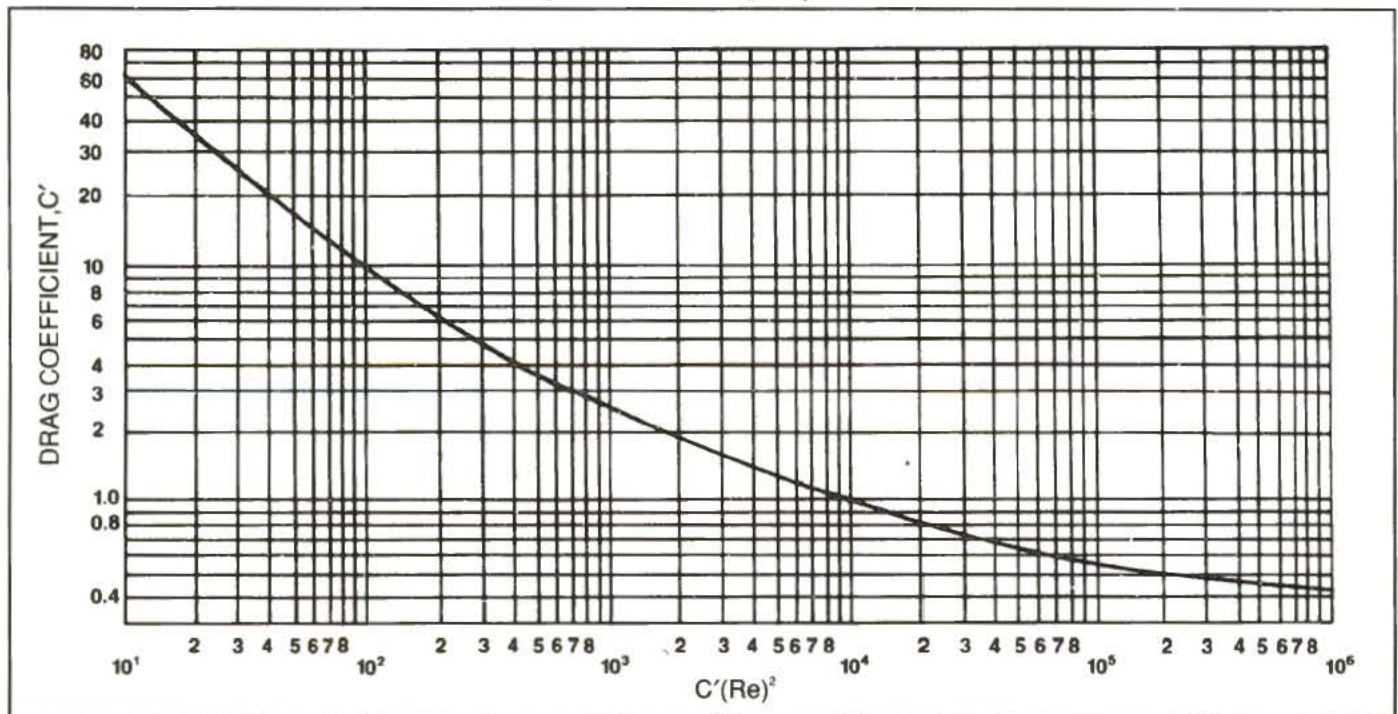
In this form, a trial and error solution is required since both particle size, D_p , and terminal velocity, V_t , are involved. To avoid trial and error, values of the drag coefficient are presented in Fig. 7-3 as a function of the product of drag coefficient, C' , times the Reynolds number squared; this technique



eliminates velocity from the expression¹. The abscissa of Fig. 7-3 is given by:

$$C' (Re)^2 = \frac{(0.95) (10^8) \rho_g D_p^3 (\rho_l - \rho_g)}{\mu^2} \quad \text{Eq 7-3}$$

FIG. 7-3
Drag Coefficient of Rigid Spheres¹³



Gravity Settling - Limiting Conditions

As with other fluid flow phenomena, the drag coefficient reaches a limiting value at high Reynolds numbers.

Newton's Law—For relatively larger particles (approximately 1000 microns and larger) the gravity settling is described by Newton's law (Fig. 7-4). The limiting drag coefficient is 0.44 at Reynolds numbers above about 500. Substituting $C' = 0.44$ in Eq 7-1 produces the Newton's law equation expressed as:

$$V_t = 1.74 \sqrt{\frac{g D_p (\rho_f - \rho_g)}{\rho_g}} \quad \text{Eq 7-4}$$

An upper limit to Newton's law is where the droplet size is so large that it requires a terminal velocity of such magnitude that excessive turbulence is created. The maximum droplet which can settle out can be determined by:

$$D_p = K_{CR} \left[\frac{\mu^2}{g \rho_g (\rho_f - \rho_g)} \right]^{0.33} \quad \text{Eq 7-5}$$

For the Newton's law region, the upper limit to Reynolds number is 200,000 and $K_{CR} = 18.13$.

Stokes' Law—At low Reynolds numbers (less than 2), a linear relationship exists between drag coefficient and the Reynolds number (corresponding to laminar flow). Stokes' law applies in this case and Eq 7-1 can be expressed as:

$$V_t = \frac{1,488 g D_p^2 (\rho_f - \rho_g)}{18 \mu} \quad \text{Eq 7-6}$$

The droplet diameter corresponding to a Reynolds number of 2 can be found using a value of 0.0080 for K_{CR} in Eq 7-5.

The lower limit for Stokes' law applicability is a droplet diameter of approximately 3 microns. The upper limit is about 100 microns.

A summary of these equations is presented in Fig. 7-4.

Coalescing

Very small droplets such as fog or mist cannot be separated practically by gravity. These droplets can be coalesced to form larger droplets that will settle by gravity. Coalescing devices in separators force gas to follow a tortuous path. The momentum of the droplets causes them to collide with other droplets or the coalescing device, forming larger droplets. These larger droplets can then settle out of the gas phase by gravity. Wire mesh screens, vane elements, and filter cartridges are typical examples of coalescing devices.

SEPARATOR DESIGN AND CONSTRUCTION

Separators are usually characterized as vertical, horizontal, or spherical. Horizontal separators can be single or double barrel and can be equipped with sumps or boots.

Parts of a Separator

Regardless of shape, separation vessels usually contain four major sections, plus the necessary controls. These sections are shown for horizontal and vertical vessels in Fig. 7-5. The primary separation section, A, is used to separate the main por-

tion of free liquid in the inlet stream. It contains the inlet nozzle which may be tangential, or a diverter baffle to take advantage of the inertial effects of centrifugal force or an abrupt change of direction to separate the major portion of the liquid from the gas stream.

The secondary or gravity section, B, is designed to utilize the force of gravity to enhance separation of entrained droplets. It consists of a portion of the vessel through which the gas moves at a relatively low velocity with little turbulence. In some designs, straightening vanes are used to reduce turbulence. The vanes also act as droplet collectors, and reduce the distance a droplet must fall to be removed from the gas stream.

The coalescing section, C, utilizes a coalescer or mist extractor which can consist of a series of vanes, a knitted wire mesh pad, or cyclonic passages. This section removes the very small droplets of liquid from the gas by impingement on a surface where they coalesce. A typical liquid carryover from the mist extractor is less than 0.1 gallon per MMscf.

The sump or liquid collection section, D, acts as receiver for all liquid removed from the gas in the primary, secondary, and coalescing sections. Depending on requirements, the liquid section should have a certain amount of surge volume, for degassing or slug catching, over a minimum liquid level necessary for controls to function properly. Degassing may require a horizontal separator with a shallow liquid level while emulsion separation may also require higher temperature, higher liquid level, and/or the addition of a surfactant.

Separator Configurations

Factors to be considered for separator configuration selection include:

- How well will extraneous material (e.g. sand, mud, corrosion products) be handled?
- How much plot space will be required?
- Will the separator be too tall for transport if skidded?
- Is there enough interface surface for three-phase separation (e.g. gas/hydrocarbon/glycol liquid)?
- Can heating coils or sand jets be incorporated if required?
- How much surface area is available for degassing of separated liquid?
- Must surges in liquid flow be handled without large changes in level?
- Is large liquid retention volume necessary?

Vertical Separators

Vertical separators, Fig. 7-6, are usually selected when the gas-liquid ratio is high or total gas volumes are low. In the vertical separator, the fluids enter the vessel striking a diverting baffle which initiates primary separation. Liquid removed by the inlet baffle falls to the bottom of the vessel. The gas moves upward, usually passing through a mist extractor to remove suspended mist, and then the "dry" gas flows out. Liquid removed by the mist extractor is coalesced into larger droplets which then fall through the gas to the liquid reservoir in the bottom. The ability to handle liquid slugs is typically obtained by increasing height. Level control is not critical and liquid level can fluctuate several inches without affecting operating efficiency. Mist extractors can significantly reduce the required diameter of vertical separators.

As an example of a vertical separator, consider a compressor suction scrubber. In this service the vertical separator:

FIG. 7-4

Gravity Settling Laws and Particle Characteristics

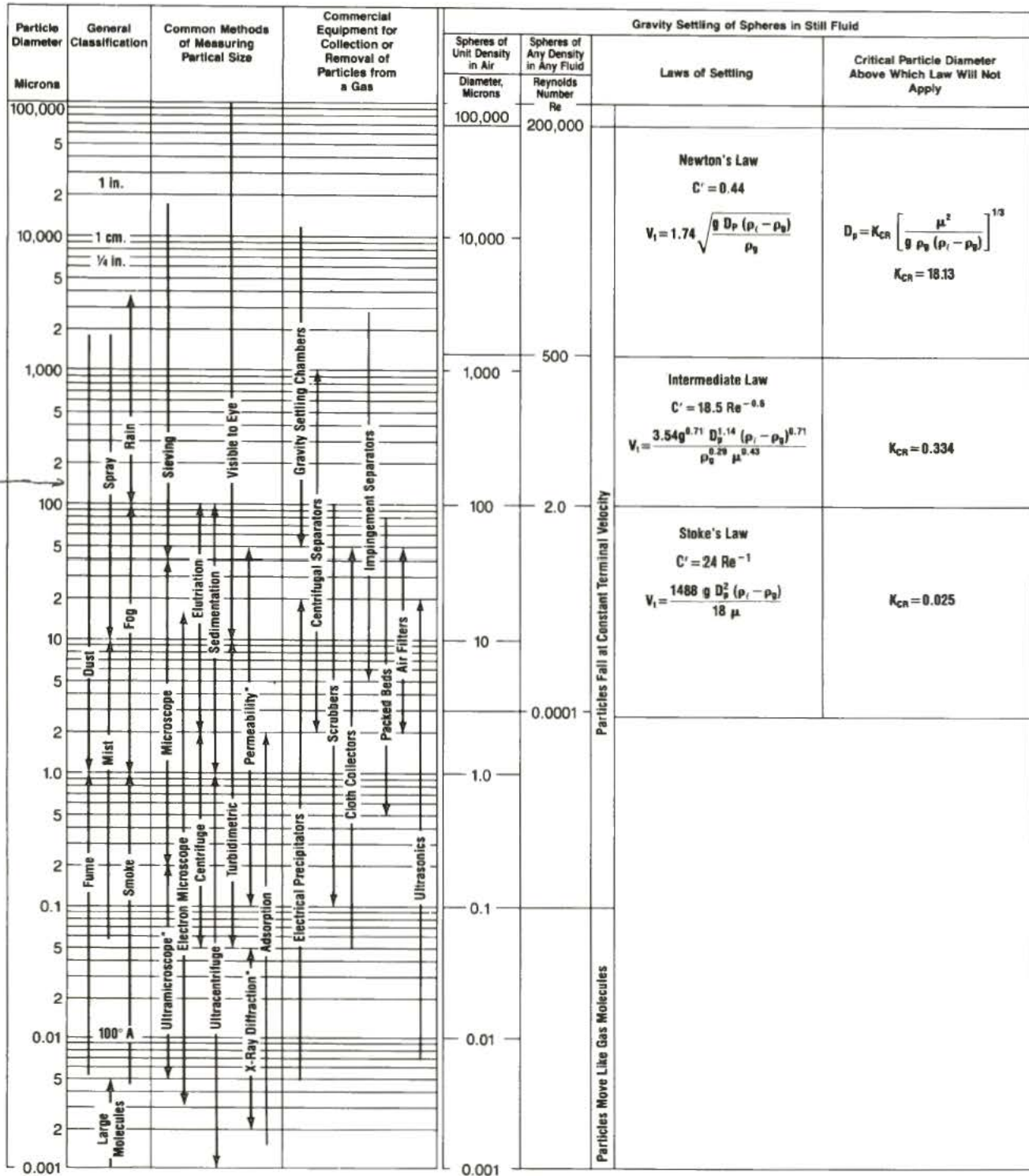
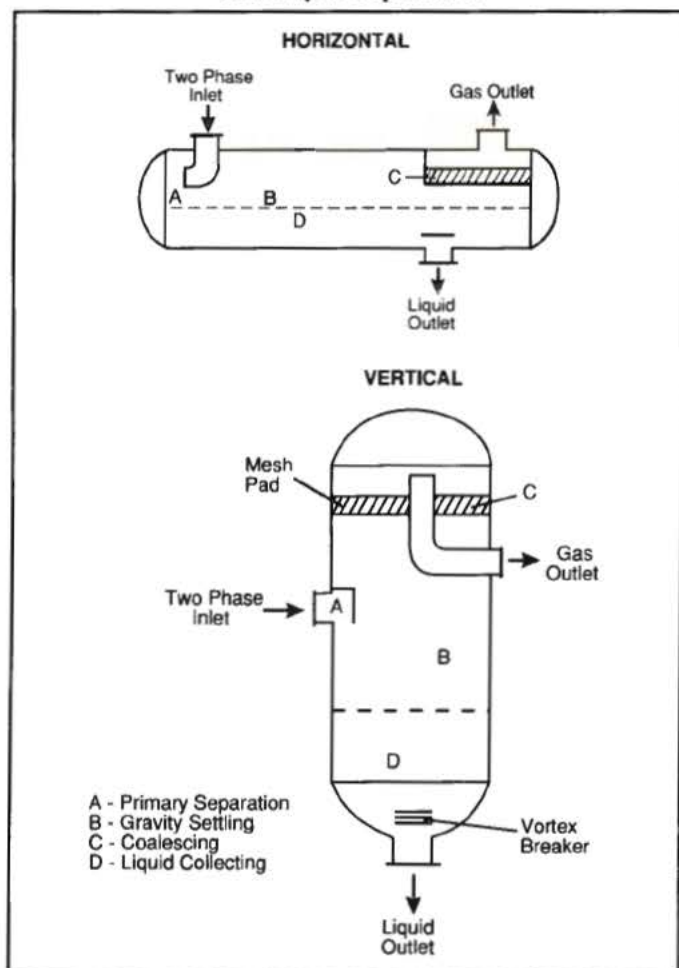


FIG. 7-5
Gas-Liquid Separators



- Does not need significant liquid retention volume.
- The liquid level responds quickly to any liquid that enters—thus tripping an alarm or shutdown.
- The separator occupies a small amount of plot space.

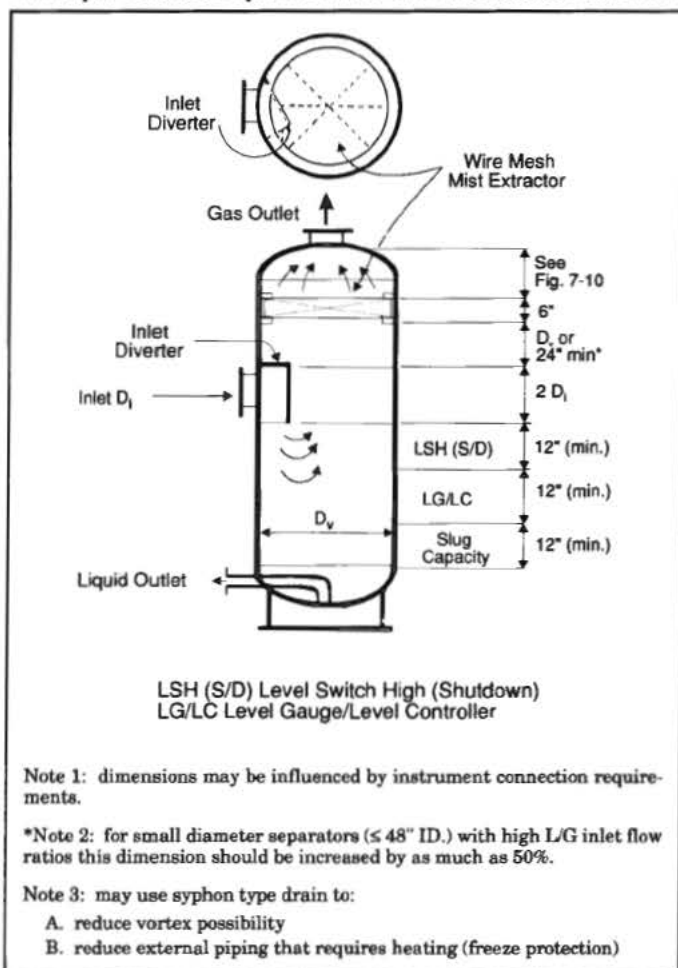
Horizontal Separators

Horizontal separators are most efficient where large volumes of total fluids and large amounts of dissolved gas are present with the liquid. The greater liquid surface area in this configuration provides optimum conditions for releasing entrapped gas. In the horizontal separator, Fig. 7-7, the liquid which has been separated from the gas moves along the bottom of the vessel to the liquid outlet. The gas and liquid occupy their proportionate shares of shell cross-section. Increased slug capacity is obtained through shortened retention time and increased liquid level. Fig. 7-7 also illustrates the separation of two liquid phases (glycol and hydrocarbon). The denser glycol settles to the bottom and is withdrawn through the "boot." The glycol level is controlled by a conventional level control instrument.

In a double barrel separator, the liquids fall through connecting flow pipes into the external liquid reservoir below. Slightly smaller vessels may be possible with the double barrel horizontal separator where surge capacity establishes the size of the lower liquid collection chamber.

FIG. 7-6

Example Vertical Separator with Wire Mesh Mist Extractor



As an example of a horizontal separator consider a rich amine flash tank. In this service:

- There is relatively large liquid surge volume leading to longer retention time (this allows more complete release of the dissolved gas and, if necessary, surge volume for the circulating system).
- There is more surface area per liquid volume to aid in more complete degassing.
- The horizontal configuration would handle a foaming liquid better than a vertical.
- The liquid level responds slowly to changes in liquid inventory.

Spherical Separators

These separators are occasionally used for high pressure service where compact size is desired and liquid volumes are small. Fig. 7-8 is a schematic for an example spherical separator. Factors considered for a spherical separator are:

- compactness;
- limited liquid surge capacity;
- minimum steel for a given pressure.

FIG. 7-7

Example Horizontal Three-Phase Separator with Wire Mesh Mist Extractor

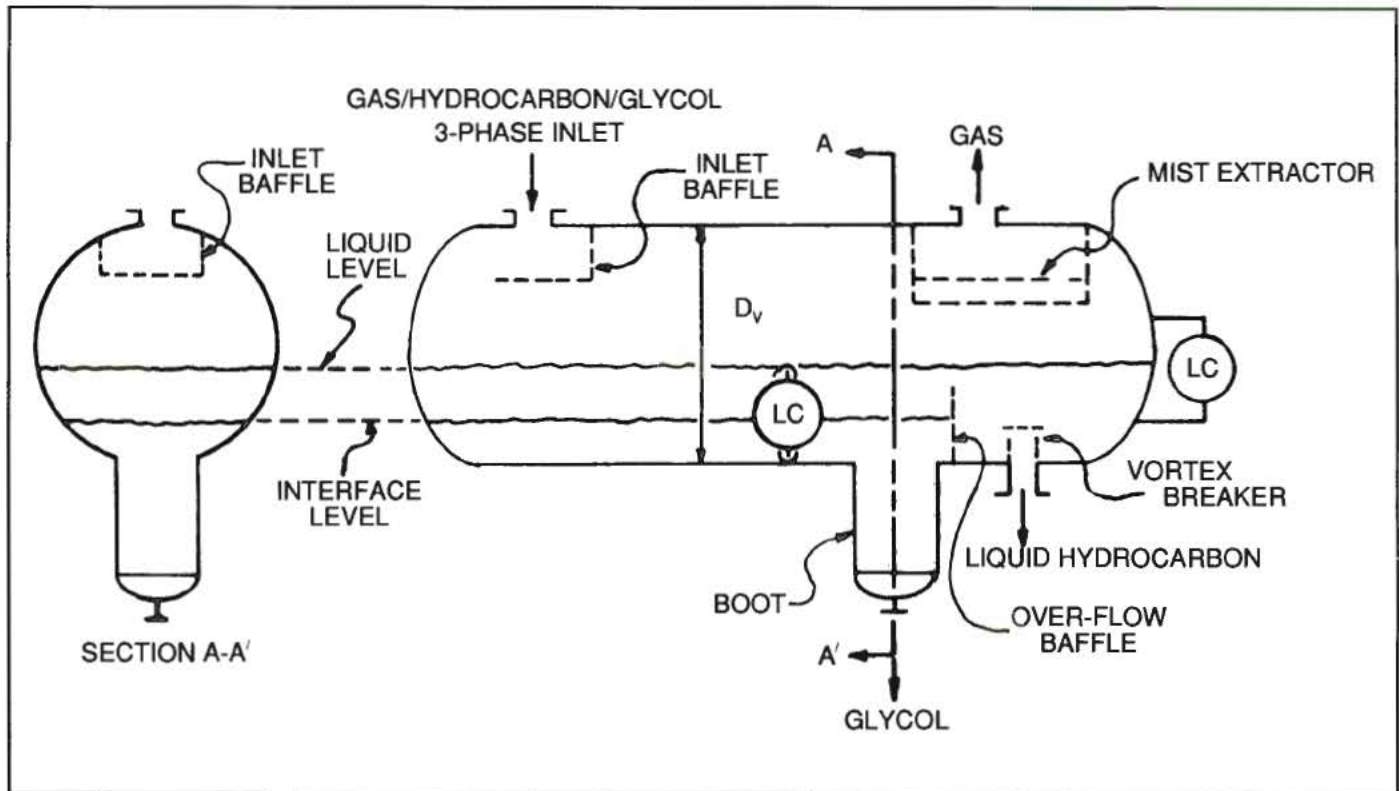
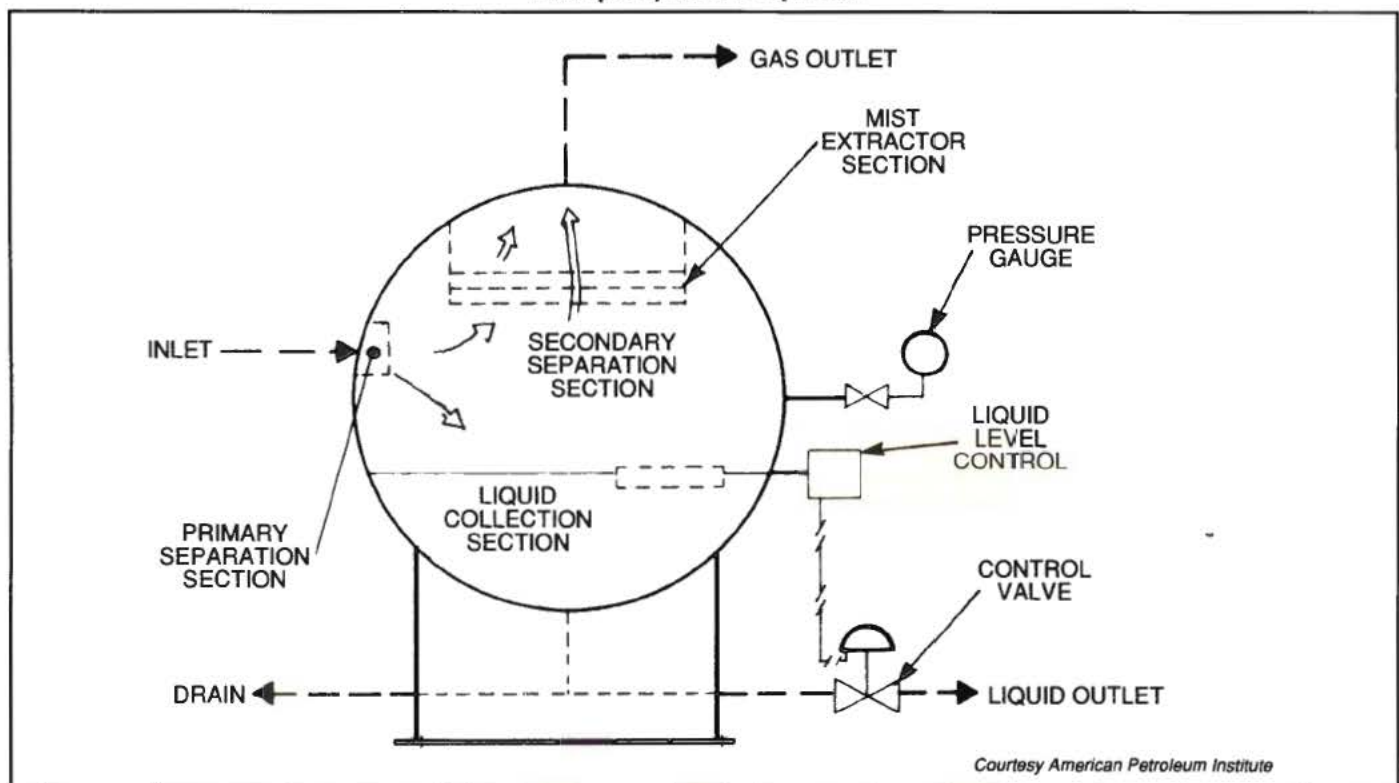


FIG. 7-8

Example Spherical Separator³



Courtesy American Petroleum Institute

GAS-LIQUID SEPARATOR DESIGN

FIG. 7-9

Specifying Separators

Separator designers need to know pressure, temperature, flow rates, and physical properties of the streams as well as the degree of separation required. It is also prudent to define if these conditions all occur at the same time or if there are only certain combinations that can exist at any time. If known, the type and amount of liquid should also be given, and whether it is mist, free liquid, or slugs.

For example, a compressor suction scrubber designed for 70-150 MMscfd gas at 400-600 psig and 65-105°F would require the separator manufacturer to offer a unit sized for the worst conditions, i.e., 150 MMscfd at 400 psig and 105°F. But the real throughput of the compressor varies from 150 MMscfd at 600 psig, 105°F to 70 MMscfd at 400 psig, 65°F. Because the high volume only occurs at the high pressure, a smaller separator is acceptable. Conversely, a pipeline separator could be just the opposite because of winter to summer flow changes.

Basic Design Equations

Separators without mist extractors are designed for gravity settling using Eq 7-1. Values for the drag coefficient are given in Fig. 7-3 for spherical droplet particles. Typically the sizing is based upon removal of 150 micron diameter droplets.

Most vertical separators that employ mist extractors are sized using equations that are derived from Eq 7-1. The two most common are the critical velocity equation:

$$V_t = K \sqrt{\frac{\rho_L - \rho_g}{\rho_g}} \quad \text{Eq 7-7}$$

and the correlation developed by Souders and Brown² to relate vessel diameter to the velocity of rising vapors which will not entrain sufficient liquid to cause excessive carryover:

$$G_m = C \sqrt{\rho_g (\rho_L - \rho_g)} \quad \text{Eq 7-8}$$

Note that if both sides of Eq 7-7 are multiplied by gas density, it is identical to Eq 7-8 when:

$$C = 3600 K \quad \text{Eq 7-9}$$

Some typical values of the separator sizing factors, K and C, are given in Fig. 7-9. Separators are sized using these equations to calculate vessel cross-sectional areas that allow gas velocities at or below the gas velocities calculated by Eq 7-7 or 7-8.

Horizontal separators with mist extractors are sized using Eqs 7-7 and 7-8 except that a factor is added for the length, L, of the gas flow path³. In horizontal separators, the gas drag force does not directly oppose the gravitational settling force. The true droplet velocity is assumed to be the vector sum of the vertical terminal velocity and the horizontal gas velocity. Hence, the minimum length of the vessel is calculated by assuming the time for the gas to flow from the inlet to the outlet is the same as the time for the droplet to fall from the top of the vessel to the surface of the liquid. In calculating the gas capacity of horizontal separators, the cross-sectional area of that portion of the vessel occupied by liquid (at maximum level) is subtracted from the total vessel cross-sectional area. Separators can be any length, but the ratio of seam-to-seam length to the diameter of the vessel, L/D_v, is usually in the range of 2:1 to 4:1.

Typical K & C Factors for Sizing Woven Wire Demisters

| Separator Type | K Factor (ft/sec) | C Factor (ft/hr) |
|-------------------------------|----------------------|---------------------|
| Horizontal (w/vert. pad) | 0.40 to 0.50 | 1440 to 1800 |
| Spherical | 0.20 to 0.35 | 720 to 1260 |
| Vrt. or Horiz. (w/horiz. pad) | 0.18 to 0.36 | 648 to 1260 |
| @ Atm. Pressure | 0.36 | 1260 |
| @ 300 psig | 0.33 | 1188 |
| @ 600 psig | 0.30 | 1080 |
| @ 900 psig | 0.27 | 972 |
| @ 1500 psig | 0.21 | 756 |
| Wet Steam | 0.25 | 900 |
| Most vapors under vacuum | 0.20 | 720 |
| Salt & Caustic Evaporators | 0.15 | 540 |

Note:

- (1) K = 0.35 @ 100 psig – Subtract 0.01 for every 100 psi above 100 psig.
- (2) For glycol and amine solutions, multiply K by 0.6 - 0.8.
- (3) Typically use one-half of the above K or C values for approximate sizing of vertical separators without woven wire demisters.
- (4) For compressor suction scrubbers and expander inlet separators multiply K by 0.7 - 0.8.

$$V_t = K \sqrt{\frac{\rho_L - \rho_g}{\rho_g}} \left(\frac{L}{10} \right)^{0.56} \quad \text{Eq 7-10}$$

$$G_m = C \sqrt{\rho_g (\rho_L - \rho_g)} \left(\frac{L}{10} \right)^{0.56} \quad \text{Eq 7-11}$$

Frequently separators without mist extractors are sized using Eq 7-7 and 7-8 with a constant (K or C) of typically one-half of that used for vessels with mist extractors. Although combining the drag coefficient and other physical properties into an empirical constant is unsound, it can be justified since:

- Selection of the droplet diameter (separation efficiency) is arbitrary. Even if the diameter can be selected on a rational basis, little information is available on the mass distribution above and below the selected size.
- Liquid droplets are not rigid spherical particles in dilute concentration (unhindered settling).

Note: A number of the "separator" sizing equations given only size the separation *element* (mist extractor, Eqs 7-7, 7-8, 7-10, 7-11, and vane separator, Eq 7-13): these equations *do not* directly size the actual separator containment *vessel*.

Thus, for example, a 24 in. diameter wire mesh mist extractor might be installed in a 36 in. diameter vessel because the liquid surge requirements dictated a larger vessel.

Separators without Mist Extractors

This is typically a horizontal vessel which utilizes gravity as the sole mechanism for separating the liquid and gas phases. Gas and liquid enter through the inlet nozzle and are slowed to a velocity such that the liquid droplets can fall out of the gas phase. The dry gas passes into the outlet nozzle and the liquid is drained from the lower section of the vessel.

To design a separator without a mist extractor, the minimum size diameter droplet to be removed must be set. Typically this diameter is in the range of 150 to 2,000 microns (one micron is 10^{-4} cm or 0.00003937 inch).

The length of vessel required can then be calculated by assuming that the time for the gas to flow from inlet to outlet is the same as the time for the liquid droplet of size D_p to fall from the top of the vessel to the liquid surface. Eq 7-12 then relates the length of the separator to its diameter as a function of this settling velocity (assuming no liquid retention):

$$L = \frac{4 Q_A}{\pi V_t D_v} \quad \text{Eq 7-12}$$

If the separator is to be additionally used for liquid storage, this must also be considered in sizing the vessel.

Example 7-1—A horizontal gravity separator (without mist extractor) is required to handle 60 MMscfd of 0.75 specific gravity gas (MW = 21.72) at a pressure of 500 psig and a temperature of 100°F. Compressibility is 0.9, viscosity is 0.012 cp, and liquid specific gravity is 0.50. It is desired to remove all entrainment greater than 150 microns in diameter. No liquid surge is required.

$$\begin{aligned} \text{Gas density, } \rho_g &= \frac{P(\text{MW})}{RTZ} = \frac{(514.7)(21.72)}{(10.73)(560)(0.90)} \\ &= 2.07 \text{ lb/ft}^3 \end{aligned}$$

$$\text{Liquid density, } \rho_l = 0.5(62.4) = 31.2 \text{ lb/ft}^3$$

$$\text{Mass flow, } M = \frac{(60)(10^6)(21.72)}{(379)(24)(3600)} = 39.8 \text{ lb/sec}$$

$$\text{Particle diameter, } D_p = \frac{(150)(0.00003937)}{12} = 0.000492 \text{ ft}$$

From Eq 7-3,

$$\begin{aligned} C'(\text{Re})^2 &= \frac{(0.95)(10^6)\rho_g D_p^3(\rho_l - \rho_g)}{\mu^2} \\ &= \frac{(0.95)(10^6)(2.07)(0.000492)^3(31.2 - 2.07)}{(0.012)^2} \\ &= 4738 \end{aligned}$$

From Fig. 7-3, Drag coefficient, $C' = 1.40$

$$\begin{aligned} \text{Terminal velocity, } V_t &= \sqrt{\frac{4gD_p(\rho_l - \rho_g)}{3\rho_g C'}} \\ &= \sqrt{\frac{4(32.2)(0.000492)(29.13)}{3(2.07)(1.40)}} \\ &= \sqrt{0.212} = 0.46 \text{ ft/sec} \end{aligned}$$

$$\text{Gas flow, } Q_A = \frac{M}{\rho_g} = \frac{39.80}{2.07} = 19.2 \text{ ft}^3/\text{sec}$$

Assume a diameter, $D_v = 3.5$ ft

$$\begin{aligned} \text{Vessel length, } L &= \frac{4Q_A}{\pi V_t D_v} = \frac{4(19.2)}{\pi(0.46)(3.5)} \\ &= 15.2 \text{ ft} \end{aligned}$$

Other reasonable solutions are as follows:

| Diameter, ft. | Length, ft. |
|---------------|-------------|
| 3.5 | 15.2 |
| 4.0 | 13.3 |
| 4.5 | 11.8 |
| 5.0 | 10.6 |

Usually vessels up through 24 in. diameter have nominal pipe dimensions while larger vessels are rolled from plate with 6 in. internal increments in diameter.

Example 7-2—What size vertical separator without mist extractor is required to meet the conditions used in Example 7-1?

$$A = \frac{Q_A}{V_t} = \frac{19.2}{0.46} = 41.7 \text{ ft}^2$$

$D_v = 7.29$ ft minimum; use 90 inch ID

Separators With Wire Mesh Mist Extractors

Wire mesh pads are frequently used as entrainment separators for the removal of very small liquid droplets and, therefore, a higher overall percentage removal of liquid. Removal of droplets down to 10 microns or smaller may be possible with these pads. The pad is generally horizontal with the gas and entrained liquid passing vertically upward. Performance is adversely affected if the pad is tilted more than 30 degrees from the horizontal⁴. Liquid droplets impinge on the mesh pad, coalesce, and fall downward through the rising gas stream. Wire mesh pads are efficient only when the gas stream velocity is low enough that re-entrainment of the coalesced droplets does not occur. Figs. 7-10 and 7-11 illustrate a typical wire mesh installation in vertical and horizontal vessels.

Eqs 7-7 and 7-10 define the maximum gas velocity as a function of the gas density and the liquid density. A value for K can be found from Fig. 7-9. Firmly secure the top and bottom of the pad so that it is not dislodged by high gas flows, such as when a pressure relief valve lifts.

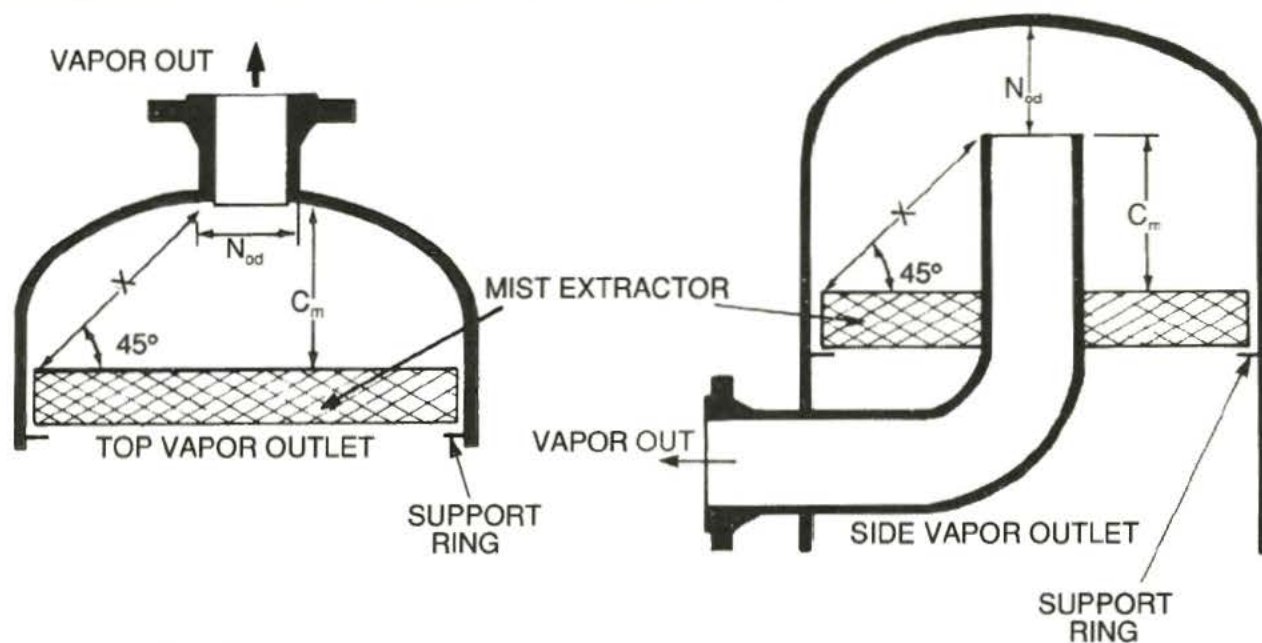
In plants where fouling or hydrate formation is possible or expected, mesh pads are typically not used. In these services vane or centrifugal type separators are generally more appropriate. Most installations will use a six inch thick pad with 9 to 12 lb/ft³ bulk density. Minimum recommended pad thickness is four inches⁴. Manufacturers should be contacted for specific designs.

Wire mesh pads can be used in horizontal vessels. A typical installation is shown in Fig. 7-11. The preferred orientation of the mesh pad is in the horizontal plane. When installed in a vertical orientation, the pad is reported to be less efficient. Problems have been encountered where liquid flow through the pad to the sump is impaired due to dirt or sludge accumulation causing a higher liquid level on one side, providing the serious potential of the pad being dislodged from its mounting brackets making it useless, or forcing parts of it into the outlet pipe. The retaining frame must be designed to hold the mist pad in place during emergency blowdown or other periods of anticipated high vapor velocity.

The pressure drop across a wire mesh pad is sufficiently low (usually less than an inch of water) to be considered negligible for most applications. The effect of the pressure drop becomes significant only in the design of vacuum services and for equipment where the prime mover is a blower or a fan. Manufacturers should be contacted for specific information.

FIG. 7-10

Example Minimum Clearance — Mesh Type Mist Eliminators



MINIMUM EXTRACTOR CLEARANCE, C_m :

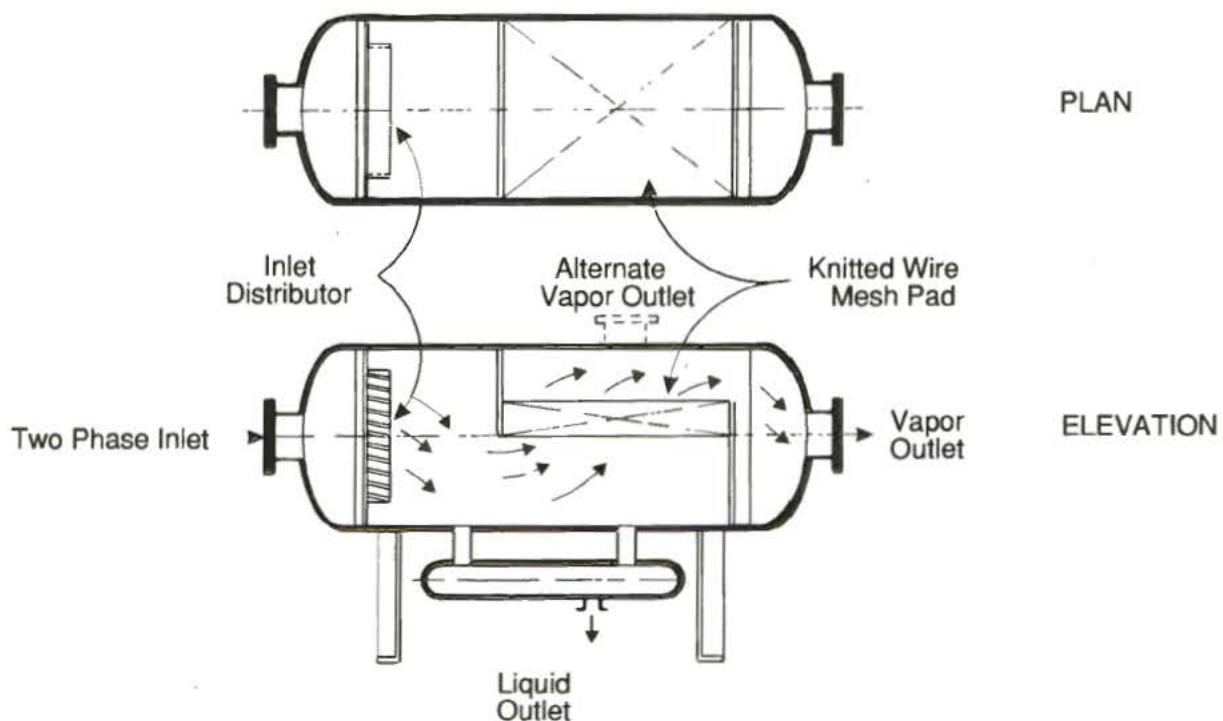
$$C_m = 0.707 X \text{ or } \frac{M_{od} - N_{od}}{2}$$

WHERE:

M_{od} = MIST EXTRACTOR OUTSIDE DIAMETER
 N_{od} = NOZZLE OUTSIDE DIAMETER

FIG. 7-11

Horizontal Separator with Knitted Wire Mesh Pad Mist Extractor and Lower Liquid Barrel



Example 7-3 — What size vertical separator equipped with a wire mesh mist extractor is required for the conditions used in Examples 7-1 and 7-2?

$$K = 0.31 \text{ ft/sec (from Fig. 7-9 for 500 psig)}$$

$$V_t = 0.31 \sqrt{\frac{31.2 - 2.07}{2.07}} = 1.16 \text{ ft/sec}$$

$$A = \frac{Q_A}{V_t} = \frac{19.2}{1.16} = 16.5 \text{ ft}^2$$

$$D_v = 4.59 \text{ ft minimum; use 60 in. ID vessel}$$

Separators with Vane Type Mist Extractors

Vanes differ from wire mesh in that they do not drain the separated liquid back through the rising gas stream. Rather, the liquid can be routed into a downcomer, which carries the fluid directly to the liquid reservoir. A vertical separator with a typical vane mist extractor is shown in Fig. 7-12.

The vanes remove fluid from the gas stream by directing the flow through a tortuous path. A cross-section of a typical vane unit is shown in Fig. 7-13. The liquid droplets, being heavier than the gas, are subjected to inertial forces which throw them against the walls of the vane. This fluid is then drained by gravity from the vane elements into a downcomer.

Vane type separators generally are considered to achieve the same separation performance as wire mesh, with the added advantage that they do not readily plug and can often be housed in smaller vessels. As vane type separators depend upon inertial forces for performance, turndown can sometimes be a problem.

Vane type separator designs are proprietary and are not easily designed with standard equations. Manufacturers of vane type separators should be consulted for detailed designs of their specific equipment. However, a gas momentum equation⁵ can be used to estimate the approximate face area of a vane type mist extractor similar to that illustrated in Fig. 7-13.

$$J = \rho_g V_t^2 = 20 \text{ lb/(ft} \cdot \text{sec}^2) \quad \text{Eq 7-13}$$

where gas velocity, V_t , is the velocity through the extractor cross-section.

Separators with Centrifugal Elements

There are several types of centrifugal separators which serve to separate solids as well as liquids from a gas stream. These devices are proprietary and cannot be readily sized without detailed knowledge of the characteristics of the specific internals. The manufacturer of such devices should be consulted for assistance in sizing these types of separators. Use care in selecting a unit as some styles are not suitable in some applications. A typical centrifugal separator is shown in Fig. 7-14. The main advantage of a centrifugal separator over a filter (or filter separator) is that much less maintenance is involved. Disadvantages of centrifugal separators are:

- some designs do not handle slugs well,
- efficiency is not as good as other types of separators,
- pressure drop tends to be higher than vane or clean knitted mesh mist extractors, and
- they have a narrow operating flow range for highest efficiency.

FIG. 7-12

Example Vertical Separator with Vane Type Mist Extractor

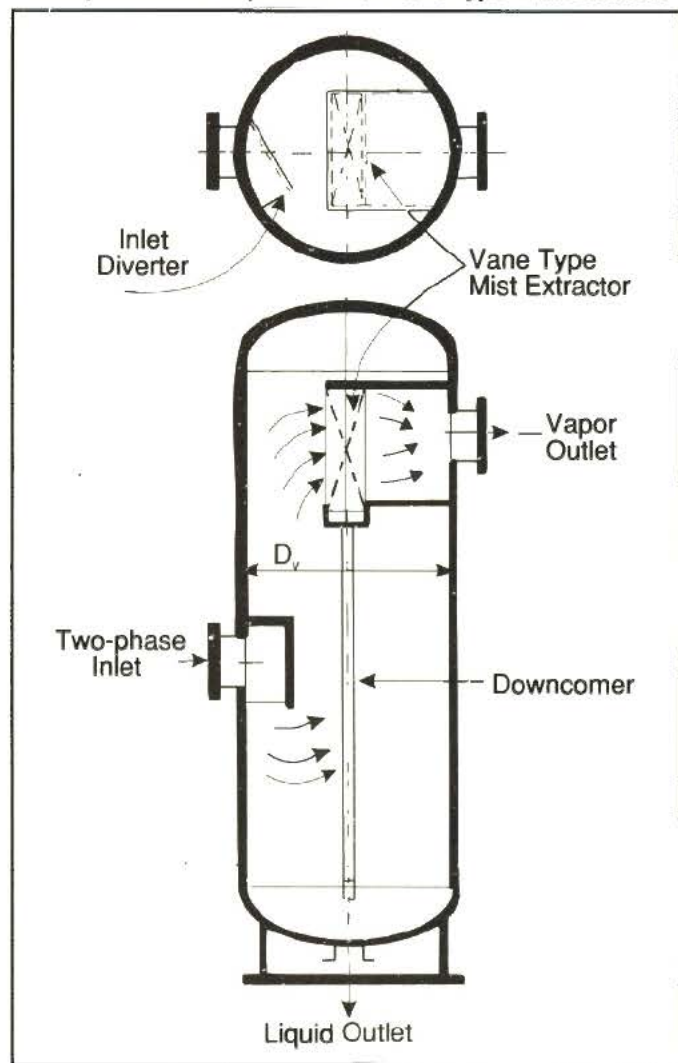


FIG. 7-13

Cross Section of Example Vane Element Mist Extractor Showing Corrugated Plates with Liquid Drainage Traps

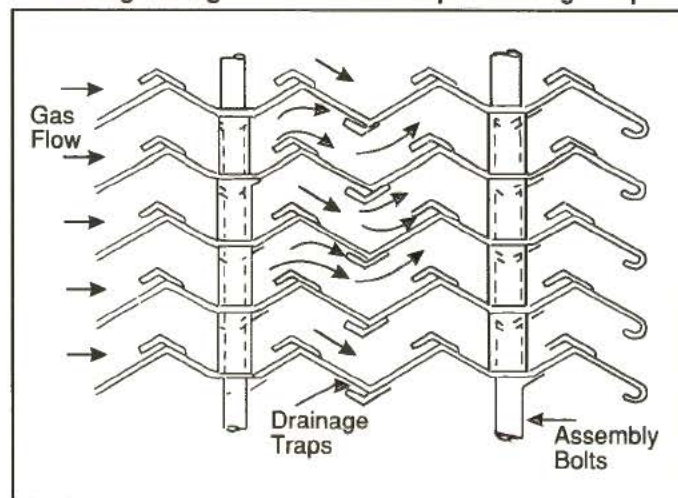
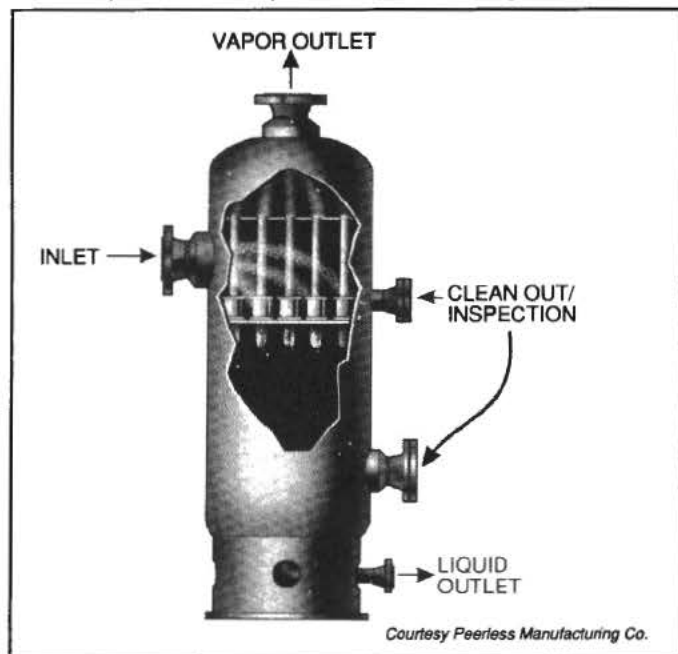


FIG. 7-14

Example Vertical Separator with Centrifugal Elements



Filter Separators

General — This type of separator has a higher separation efficiency than the centrifugal separator, but it uses filter elements, which must periodically be replaced. An example filter separator is shown in Fig. 7-15. Gas enters the inlet nozzle and passes through the filter section where solid particles are filtered from the gas stream and liquid particles are coalesced into larger droplets. These droplets pass through the tube and are entrained into the second section of the separator, where a final mist extraction element removes these coalesced droplets from the gas stream.

The design of filter separators is proprietary and a manufacturer should be consulted for specific size and recommendations. The body size of a horizontal filter separator for a typical application can be estimated by using 1.3 for the value of K in Eq 7-7. This provides an approximate body diameter for a unit designed to remove water (other variables such as viscosity and surface tension enter into the actual size determination). Units designed for water will be smaller than units sized to remove light hydrocarbons.

Example 7-4 — A filter separator is required to handle a flow of 60 MMscfd at conditions presented in Example 7-1. Estimate the diameter of a filter separator.

$$V_t = 1.3 \sqrt{\frac{31.2 - 2.07}{2.07}} = 4.88 \text{ ft/sec}$$

$$A = \frac{Q_A}{V_t} = \frac{19.2}{4.88} = 3.93 \text{ ft}^2$$

$$D_v = 2.2 \text{ ft} = 26.9 \text{ in. minimum}$$

Use 30 inch ID separator.

In many cases the vessel size will be determined by the filtration section rather than the mist extraction section. The

filter cartridges coalesce the liquid mist into droplets which can be easily removed by the mist extractor section. A design consideration commonly overlooked is the velocity out of these filter tubes into the mist extraction section. If the velocity is too high, the droplets will be sheared back into a fine mist that will pass through the extractor element. A maximum allowable velocity for gas exiting the filter tube attachment pipe can be estimated using the momentum Eq 7-13 with a value of 1250 for J . Light hydrocarbon liquids or low pressure gas should be limited to even less than this value. No published data can be cited since this information is proprietary with each filter separator manufacturer.

Design — The most common and efficient agglomerator is composed of a tubular fiber glass filter pack which is capable of holding the liquid particles through submicron sizes. Gas flows into the top of the filter pack, passes through the elements and then travels out through the tubes. Small, dry solid particles are retained in the filter elements and the liquid coalesces to form larger particles. Liquid agglomerated in the filter pack is then removed by a mist extractor located near the gas outlet.

The approximate filter surface area for gas filters can be estimated from Fig. 7-16. The figure is based on applications such as molecular sieve dehydrator outlet gas filters. For dirty gas service the estimated area should be increased by a factor of two or three.

The efficiency of a filter separator largely depends on the proper design of the filter pack, i.e., a minimum pressure drop while retaining an acceptable extraction efficiency. A pressure drop of approximately 1-2 psi is normal in a clean filter separator. If excessive solid particles are present, it may be necessary to clean or replace the filters at regular intervals when a pressure drop in excess of 10 psi is observed. However, as a rule, 25 psi is recommended as a maximum as the cartridge units might otherwise collapse. Removal of the filter pack is easily achieved by using a quick-opening closure.

Various guarantees are available from filter separator manufacturers such as one for 100 percent removal of liquid droplets 8 microns and larger and 99.5 percent removal of particles in the 0.5-8 micron range. However, guarantees for the performance of separators and filters are very difficult to verify in the field.

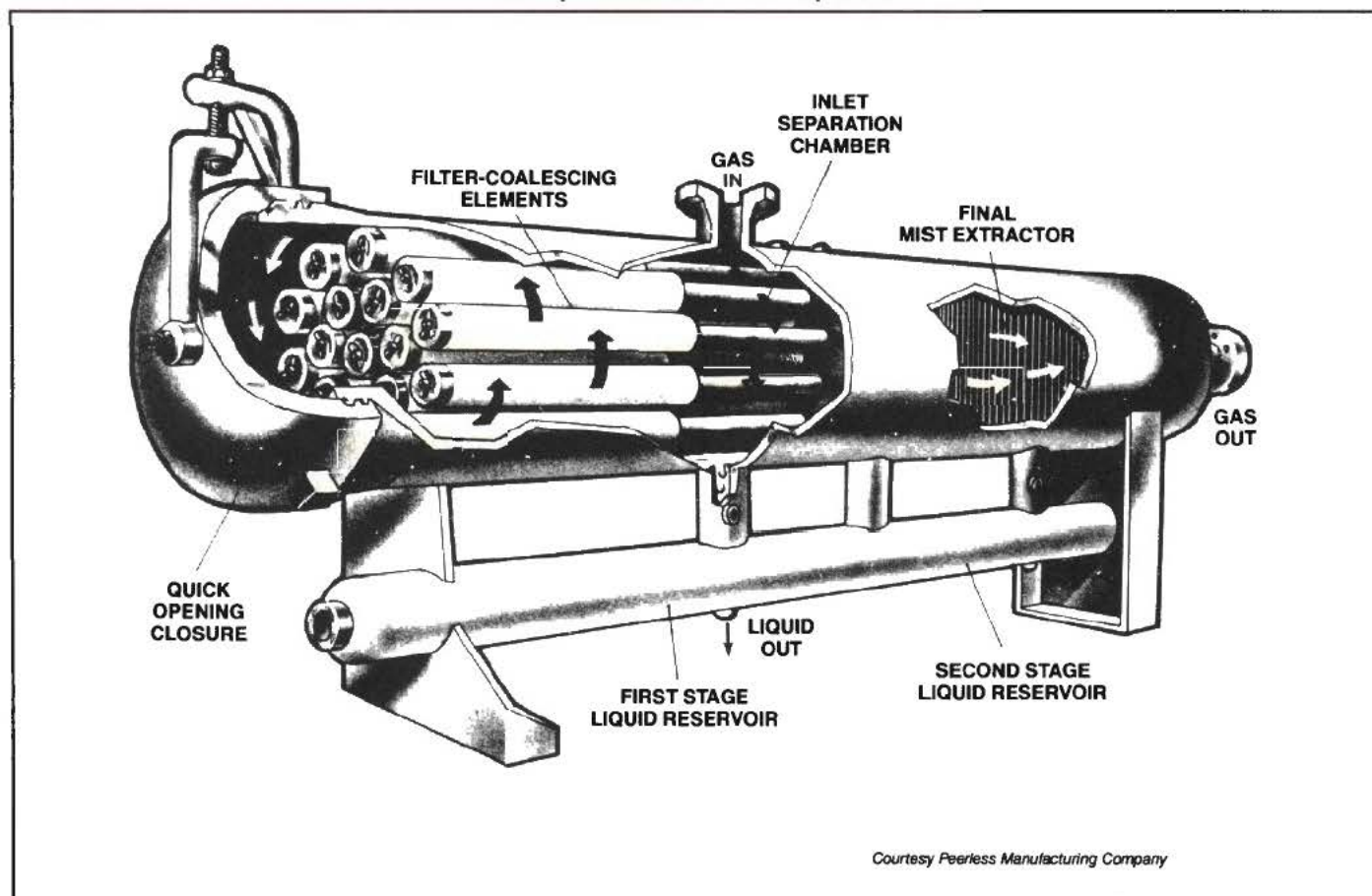
While most dry solid particles about ten microns and larger are removable, the removal efficiency is about 99 percent for particles below approximately ten microns.

For heavy liquid loads, or where free liquids are contained in the inlet stream, a horizontal filter separator with a liquid sump, which collects and dumps the inlet free-liquids separately from coalesced liquids, is often preferred.

LIQUID-LIQUID SEPARATOR DESIGN

Liquid-liquid separation may be divided into two broad categories of operation. The first is defined as "gravity separation" where the two immiscible liquid phases separate within the vessel by the differences in density of the liquids. Sufficient retention time must be provided in the separator to allow for the gravity separation to take place. The second category is defined as "coalescing separation." This is where small particles of one liquid phase must be separated or removed from a large quantity of another liquid phase. Different types of internal construction of separators must be provided for each

FIG. 7-15
Example Horizontal Filter-Separator



Courtesy Peerless Manufacturing Company

type of liquid-liquid separation. The following principles of design for liquid-liquid separation apply equally for horizontal or vertical type separators. Horizontal vessels have some advantage over vertical ones for liquid-liquid separation, due to the larger interface area available in the horizontal style, and the shorter distance particles must travel to coalesce.

There are two factors which may prevent two liquid phases from separating due to differences in specific gravity:

- If droplet particles are so small they may be suspended by Brownian movement. This is defined as a random motion which is greater than directed movement due to gravity for particles less than 0.1 micron in diameter.
- The droplets may carry electric charges due to dissolved ions, and these charges can cause the droplets to repel each other rather than coalesce into larger particles and settle by gravity.

Effects due to Brownian movement are usually small and proper chemical treatment will usually neutralize any electric charges. Then settling becomes a function of gravity and viscosity in accordance with Stokes' law. The settling velocity of spheres through a fluid is directly proportional to the difference in densities of the sphere and the fluid, and inversely proportional to the viscosity of the fluid and the square of the diameter of the sphere (droplet), Eq 7-6. The liquid-liquid separation capacity of separators may be determined⁸ from Eqs 7-14 and 7-15 which were derived from Eq 7-6. Values of C^* are found in Fig. 7-17.

Vertical Vessels:

$$W_{cl} = C^* \left(\frac{S_{hl} - S_{ll}}{\mu} \right) (0.785) D_v^2 \quad \text{Eq 7-14}$$

Horizontal Vessels:

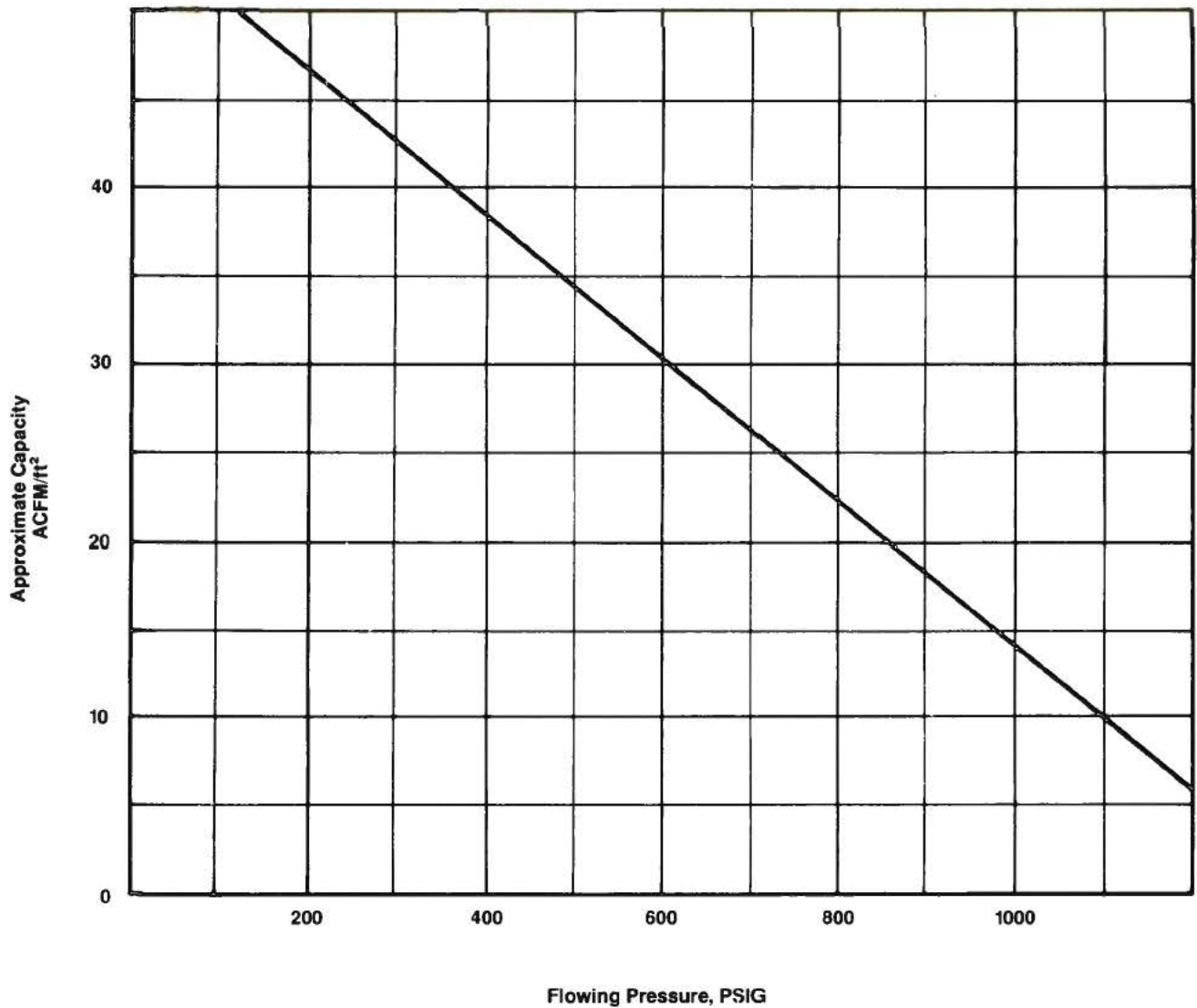
$$W_{cl} = C^* \left(\frac{S_{hl} - S_{ll}}{\mu} \right) L_v H_v \quad \text{Eq 7-15}$$

Since the droplet size of one liquid phase dispersed in another is usually unknown, it is simpler to size liquid-liquid separation based on retention time of the liquid within the separator vessel. For gravity separation of two liquid phases, a large retention or quiet settling section is required in the vessel. Good separation requires sufficient time to obtain an equilibrium condition between the two liquid phases at the temperature and pressure of separation. The liquid capacity of a separator or the settling volume required can be determined¹⁰ from Eq 7-16 using the retention time given in Fig. 7-18.

$$U = \frac{W(t)}{1440} \quad \text{Eq 7-16}$$

The following example shows how to size a liquid-liquid separator.

FIG. 7-16
Approximate Gas Filter Capacity



NOTES FOR FIG. 7-16:

Area of a Filter Element is: $\pi d_e L_e$

Filter Surface Area is:
(No. of Elements) $\pi d_e L_e$

where d_e, L_e are the Filter Element outside diameter and length respectively.

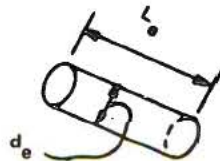


FIG. 7-17

Values of C^* Used in Eq 7-14, 7-15

| Emulsion Characteristic | Droplet Diameter, Microns | Constant ⁹ C^* |
|-------------------------|---------------------------|-----------------------------|
| Free liquids | 200 | 1100 |
| Loose emulsion | 150 | 619 |
| Moderate emulsion | 100 | 275 |
| Tight emulsion | 60 | 99 |

FIG. 7-18

Typical Retention Times for Liquid/Liquid Separation

| Type of Separation | Retention Time |
|--|----------------|
| Hydrocarbon/Water Separators ³ | |
| Above 35° API Hydrocarbon | 3 to 5 min. |
| Below 35° API Hydrocarbon | |
| 100°F and above | 5 to 10 min. |
| 80°F | 10 to 20 min. |
| 60°F | 20 to 30 min. |
| Ethylene Glycol/Hydrocarbon Separators (Cold Separators) ^{11, 14} | 20 to 60 min. |
| Amine/Hydrocarbon Separators ¹¹ | 20 to 30 min. |
| Coalescers, Hydrocarbon/Water Separators ¹¹ | |
| 100°F and above | 5 to 10 min. |
| 80°F | 10 to 20 min. |
| 60°F | 20 to 30 min. |
| Caustic/Propane | 30 to 45 min. |
| Caustic/Heavy Gasoline | 30 to 90 min. |

Example 7-5 — Determine the size of a vertical separator to handle 600 bpd of 55° API condensate and 50 bpd of produced water. Assume the water particle size is 200 microns. Other operating conditions are as follows:

Operating temperature = 80°F
 Operating pressure = 1000 psig
 Water specific gravity = 1.01
 Condensate viscosity = 0.55 cp @ 80°F
 Condensate specific gravity for 55° API = 0.76

From Eq 7-14

$$W_d = C^* \left(\frac{S_{hw} - S_{ll}}{\mu} \right) (0.785) (D_v)^2$$

From Fig. 7-17 for free liquids with water particle diameter = 200 microns, $C^* = 1100$.

$$600 \text{ bbl/day} = 1100 \frac{1.01 - 0.76}{(0.55)} (0.785) (D_v)^2$$

$$(D_v)^2 = \frac{600}{392.5} = 1.53 \text{ ft}^2$$

$$D_v = 1.24 \text{ ft}$$

Using a manufacturer's standard size vessel might result in specifying a 20" OD separator.

Using the alternate method of design based on retention time as shown in Eq 7-16 would give:

$$U = \frac{W(t)}{1440}$$

From Fig. 7-18, use 3 minutes retention time.

$$U = \frac{(650)(3)}{1440} = 1.35 \text{ bbl}$$

$$U = 1.35 (42) = 56.7 \text{ gal}$$

Assuming a 20" OD, 1480 psig working pressure, a vessel would be made from 1.031" wall seamless pipe which holds 13.1 gal/ft. The small volume held in the bottom head can be discounted in this size vessel. The shell height required for the retention volume required would be:

$$\text{Shell height} = \frac{U}{\text{Vol/ft}} = \frac{56.7}{13.1} = 4.3 \text{ ft}$$

This would require a 20" OD x 10' separator to give sufficient surge room above the liquid settling section for any vapor-liquid separation.

Another parameter that should be checked when separating amine or glycol from liquid hydrocarbons is the interface area between the two liquid layers. This area should be sized so the glycol or amine flow across the interface does not exceed approximately 2000 gallons per day per square foot.

The above example indicates that a relatively small separator would be required for liquid-liquid separation. It should be remembered that the separator must also be designed for the vapor capacity to be handled. In most cases of high vapor-liquid loadings that are encountered in gas processing equipment design, the vapor capacity required will dictate a much larger vessel than would be required for the liquid load only. The properly designed vessel has to be able to handle both the vapor and liquid loads. Therefore, one or the other will control the size of the vessel used.

PARTICULATE REMOVAL-FILTRATION

Filtration, in the strictest sense, applies only to the separation of solid particles from a fluid by passage through a porous medium. However, in the gas processing industry, filtration commonly refers to the removal of solids and liquids from a gas stream.

The most commonly used pressure filter in the gas processing industry is the cartridge filter. Cartridge filters are constructed of either a self-supporting filter medium or a filter medium attached to a support core. Depending on the application, a number of filter elements is fitted into a filter vessel. Flow is normally from the outside, through the filter element, and out through a common discharge. When pores in the filter medium become blocked, or as the filter cake is developed, the higher differential pressure across the elements indicates that the filter elements must be cleaned or replaced.

Cartridge filters are commonly used to remove solid contaminants from amines, glycols, and lube oils. Other uses include the filtration of solids and liquids from hydrocarbon vapors and the filtration of solids from air intakes of engines and turbine combustion chambers.

Two other types of pressure filters which also have applications in the gas processing industry include the edge and precoat filter. Edge filters consist of nested metallic discs, enclosed in a pressure cylinder, which are exposed to liquid flow. The spacing between the metal discs determines the solids retention. Some edge filters feature a self-cleaning design in which the discs rotate against stationary cleaning blades. Applications for edge filters include lube oil and diesel fuel filtration as well as treating solvent.

Precoat filters find use in the gas processing industry; however, they are complicated and require considerable attention. Most frequent use is in larger amine plants where frequent replacement of cartridge elements is considerably more expensive than the additional attention required by precoat filters.

The precoat filter consists of a coarse filter medium over which a coating has been deposited. In many applications, the coating is one of the various grades of diatomaceous earth which is mixed in a slurry and deposited on the filter medium. During operation additional coating material is often added continuously to the liquid feed. When the pressure drop across the filter reaches a specified maximum, the filter is taken off-line and back-washed to remove the spent coating and accumulated solids. Applications for precoat filters include water treatment for waterflood facilities as well as amine filtration to reduce foaming. Typical designs for amine plants use 1-2 gpm flow per square foot of filter surface area. Sizes range upward from 10-20 percent of full stream rates⁷.

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NOTES:



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SECTION 17

Fluid Flow and Piping

Few flow problems can be solved with an acceptable degree of accuracy when using equations designed to fit idealized applications. Flow regimes and associated pressure drops are complex phenomena and require complex equations to predict

their relationships. For engineering design purposes, several empirical formulas have been developed to fit particular circumstances in predicting flow capacity and pressure drop.

FIG. 17-1

Nomenclature

| | |
|---|---|
| <p>A = pipe cross sectional area, ft^2 ($A = \pi D^2/4$)</p> <p>c = sum of allowances for corrosion, erosion, etc., in., Fig. 17-23</p> <p>C = design parameter used in Hazen and Williams formula, Eq. 17-33</p> <p>C_1 = discharge factor from Fig. 17-8</p> <p>C_2 = size factor from Fig. 17-9</p> <p>d = internal diameter of pipe, in.</p> <p>D = internal diameter of pipe, feet</p> <p>d_o = outside pipe diameter, in.</p> <p>E = pipeline efficiency factor (fraction)</p> <p>E' = longitudinal weld joint factor from ANSI B31.3, Fig. 17-23</p> <p>E'' = longitudinal joint factor from ANSI B31.8, Fig. 17-24</p> <p>F'' = construction type design factor used in ANSI B31.8, Fig. 17-24</p> <p>F_{pv} = volume correction for a non-ideal fluid due to compressibility from Eq 17-13</p> <p>$\sqrt{1/f_f}$ = transmission factor</p> <p>f_f = Fanning friction factor</p> <p>f_m = Moody friction factor ($f_m = 4.0 f_f$)</p> <p>f_n = single phase friction factor for Dukler calculation, from Eq 17-44</p> <p>f_{tpr} = friction factor ratio for Dukler calculation, Fig. 17-17</p> <p>g = acceleration due to gravity, 32.2 ft/sec^2</p> <p>g_c = gravitational constant, $32.2 (\text{ft} \cdot \text{lbm})/(\text{lbf} \cdot \text{sec}^2)$</p> <p>$H$ = total energy of a fluid at a point above a datum, from Eq 17-1</p> <p>H_{Ld} = liquid holdup fraction (Dukler), Fig. 17-18</p> <p>H_{Le} = liquid holdup fraction (Eaton), Fig. 17-20</p> <p>H_{Lf} = liquid holdup fraction (Flanigan), Fig. 17-19</p> <p>h_L = loss of static pressure head due to fluid flow, feet of fluid</p> <p>I_L = liquid inventory in pipe, ft^3, from Eq 17-57</p> <p>L = length of line, feet</p> <p>L_m = length of line, miles</p> <p>MW = molecular weight</p> <p>N_x = Fig. 17-16 horizontal coordinate, ft/sec</p> <p>N_y = Fig. 17-16 vertical coordinate, ft/sec</p> <p>N_E = abscissa of Eaton correlation, Fig. 17-20</p> | <p>N_{Lv} = liquid velocity number, from Eq 17-53</p> <p>N_{gv} = gas velocity number, from Eq 17-54</p> <p>N_d = pipe diameter number, from Eq 17-55</p> <p>N_L = liquid viscosity number, from Eq 17-56</p> <p>P = pressure, psia</p> <p>P_1 = inlet pressure, psia</p> <p>P_2 = outlet pressure, psia</p> <p>P_{avg} = average pressure, psia, from Eq 17-16</p> <p>P_b = base absolute pressure, psia (ANSI 2530 specification: $P_b = 14.73 \text{ psia}$)</p> <p>P_i = internal design pressure, psig</p> <p>ΔP_{100} = pressure drop, psi/100 ft equivalent pipe length</p> <p>ΔP_e = elevation component of pressure drop, psi</p> <p>ΔP_f = frictional component of pressure drop, psi</p> <p>ΔP_t = total pressure drop, psi</p> <p>q = flow rate, gal./min</p> <p>Q = flow rate of gas, cubic feet per day at base conditions</p> <p>Q_L = liquid volumetric flow rate at flowing conditions, ft^3/sec</p> <p>Q_g = gas volumetric flow rate at flowing conditions, ft^3/sec</p> <p>Re = Reynolds number</p> <p>Re_y = mixture Reynolds number for Dukler calculation, from Eq 17-45</p> <p>S = specific gravity of flowing gas (air = 1.0)</p> <p>S' = allowable stress, psi, Fig. 17-23</p> <p>S'' = specified minimum yield strength, psi, Fig. 17-24</p> <p>T = absolute temperature of flowing gas, $^{\circ}\text{R}$</p> <p>T'' = temperature derating factor used in ANSI B31.8, Fig. 17-24</p> <p>T_{avg} = average temperature, $^{\circ}\text{R}$, [$T_{avg} = 1/2 (T_{in} + T_{out})$]</p> <p>$T_b$ = base absolute temperature, $^{\circ}\text{R}$ (ANSI 2530 specification: $T_b = 520^{\circ}\text{R}$)</p> <p>$t$ = thickness, in., Figs. 17-23, 17-24</p> <p>t_m = minimum required wall thickness, in., Fig. 17-23</p> <p>V = single phase fluid velocity, ft/sec</p> <p>V_{sg} = superficial gas velocity, ft/sec, from Eq 17-36</p> <p>V_{sL} = superficial liquid velocity, ft/sec, from Eq 17-35</p> <p>V_m = mixture velocity, ft/sec, from Eq 17-46</p> <p>W = mass flow, lb/hr</p> |
|---|---|

FIG. 17-1 (Cont'd)

X_A = Aziz fluid property correction factor (horizontal axis, Fig. 17-16)
 Y_A = Aziz fluid property correction factor (vertical axis, Fig. 17-16)
 Y' = coefficient found in Table 304.1.1, ANSI B31.3, Fig. 17-23
 Z_{avg} = average compressibility factor
 Z_e = pipeline vertical elevation rise, ft
 ϵ = absolute roughness, ft
 λ = flowing liquid volume fraction
 μ_e = single phase fluid viscosity, lb_m/(ft • sec)
 μ = single phase fluid viscosity, cp
 μ_g = gas viscosity, cp

μ_L = liquid viscosity, cp
 μ_n = mixture viscosity for Dukler calculation, cp
 ρ = single phase fluid density, lb/ft³
 ρ_{avg} = average density, lb/ft³ [$\rho_{avg} = 1/2 (\rho_{in} + \rho_{out})$]
 ρ_a = air density at 60°F and 14.7 psia, 0.0764 lb/ft³
 ρ_w = water density at 60°F and 14.7 psia, 62.4 lb/ft³
 ρ_g = gas density, lb/ft³
 ρ_L = liquid density, lb/ft³
 ρ_k = two phase mixture density for Dukler calculation, lb/ft³
 σ = interfacial tension at flowing conditions, dyne/cm
 σ_{wa} = interfacial tension of air and water at 60°F and 14.7 psia, 72.4 dyne/cm

Bernoulli's Theorem

The Bernoulli Theorem¹ is a mathematical derivation based on the law of conservation of energy. This theorem states that the total energy of a fluid at any particular point above a datum plane is the sum of the elevation head, the pressure head, and the velocity head. Stated mathematically:

$$H = Z_e + \frac{144 P}{\rho} + \frac{V^2}{2g} \quad \text{Eq 17-1}$$

If there are no friction losses and no energy is added to or taken from the system, H is constant for any point in the fluid. In reality, whenever fluid is moving there is friction loss (h_L). This loss describes the difference in total energy at two points in the system. Expressing the energy levels at Point 1 versus Point 2 then becomes:

$$Z_{e1} + \frac{144 P_1}{\rho_1} + \frac{V_1^2}{2g} = Z_{e2} + \frac{144 P_2}{\rho_2} + \frac{V_2^2}{2g} + h_L \quad \text{Eq 17-2}$$

All practical formulas for fluid flow are derived from the above. Modifications to Eq 17-2 have been proposed by many investigators to account for the friction losses.

Fluid Physical Properties

The physical properties of a flowing fluid must be known to predict pressure drop in piping. The two properties entering into the solution of most fluid flow problems are viscosity and density.

Viscosity expresses the readiness with which a fluid flows when it is acted upon by an external force. Two types of viscosity measurements are used, absolute and kinematic. Absolute viscosity is a measure of a fluid's internal resistance to deformation or shear. Kinematic viscosity is the ratio of absolute viscosity to mass density. The absolute viscosity will be used for all calculations in this section.

Viscosity is temperature dependent. The viscosity of most liquids decreases with an increase in temperature, whereas that of gases increases. Pressure has almost no effect on the viscosity of liquids or near perfect gases. On the other hand, the viscosity of saturated or slightly superheated vapors is changed appreciably by pressure changes. The viscosity of steam is readily available, but the viscosity of other vapors may not be known.

Specific volume is the inverse of density. Specific gravity of a liquid is the ratio of the density of the liquid at a specified

temperature to the density of water at 60°F. The specific gravity of gas is defined as the ratio of the molecular weight of the gas to the molecular weight of air.

$$S = \frac{MW(\text{gas})}{MW(\text{air})} \quad \text{Eq 17-3}$$

Flow in Pipes and Reynolds Number

At low velocities, fluid molecules or particles carried by the fluid move in a reasonably straight line. Velocity of the fluid is maximum at the center of the pipe and zero at the pipe wall. This flow pattern is referred to as laminar. If the velocity is increased it will reach a critical point where fluid particles begin to show a random motion transverse to the direction of flow. This is the critical velocity. This random motion is typical of what is referred to as turbulent flow. Above the critical velocity the flow is considered to be completely turbulent even though there is always a boundary layer at the pipe wall where flow is laminar. In the turbulent zone the velocity profile is more nearly straight across the face of the pipe.

Reynolds developed a dimensionless number that may be considered as the ratio of the dynamic forces of mass flow to the shear stress due to viscosity. The Reynolds number is:

$$Re = \frac{DV\rho}{\mu_e} \quad \text{Eq 17-4}$$

If the Reynolds number is less than 2000, flow may be considered laminar. If it is above 4000, the flow is turbulent. In the zone between 2000 and 4000 the flow could be either turbulent or laminar, but cannot be predicted by the Reynolds number.

If a non-circular conduit is encountered, the Reynolds number can be approximated by using an equivalent diameter for D. The equivalent diameter would equal four (4) times the hydraulic radius. The hydraulic radius is defined as:

$$\text{Hydraulic Radius} = \frac{\text{Area of Flowing Fluid}}{\text{Wetted Perimeter}} \quad \text{Eq 17-5}$$

This conversion would not apply to extremely narrow shapes where the width is small relative to the length. In such cases an approximation may be used wherein one-half the width of the passage equal to the hydraulic radius.

Pressure Loss Due to Friction

Flow is always accompanied by friction. This friction results in a loss of energy available for work. A general equation for

pressure drop due to friction is the Darcy-Weisbach² (often referred to as simply the Darcy) equation. This equation can be rationally derived by dimensional analysis, with the exception of the friction factor, f_m , which must be determined experimentally. Expressed in feet of fluid this equation is:

$$h_L = \frac{f_m L V^2}{2 g D} \quad \text{Eq 17-6}$$

Converting to pounds per square inch, the equation becomes:

$$\Delta P_f = \frac{\rho f_m L V^2}{(144) D (2g_c)} \quad \text{Eq 17-7}$$

It should be noted that the Moody friction factor³, f_m , is used in the equations above. Some equations are shown in terms of the Fanning friction factor, f_f , which is one fourth of f_m ($f_m = 4.0 f_f$). A graph of both Fanning and Moody friction factors as a function of Reynolds number appears in Fig. 17-2.

The Darcy-Weisbach equation is valid for both laminar and turbulent flow of any liquid, and may also be used for gases with certain restrictions. When using this equation, changes in elevation, velocity, or density must be accounted for by applying Bernoulli's theorem. The Darcy-Weisbach equation must be applied to line segments sufficiently short such that fluid density is essentially constant over that segment. The overall pressure drop is the sum of the ΔP_f values calculated

for the individual segments. For gas applications the segmental length may be relatively short, as compared to liquid applications, since many gas applications involve compressible gases where gas densities vary with pressure.

Friction Factor and Effect of Pipe Roughness

When the fluid flow is laminar ($Re < 2000$), the friction factor has a direct relationship to the Reynolds number, such that:

$$f_m = 64/Re \quad \text{or} \quad f_f = 16/Re \quad \text{Eq 17-8}$$

Pipe roughness has no effect on the friction factor in laminar flow.

Substitution of the formula for Reynolds number, Eq 17-4, into Eq 17-8, yields the following:

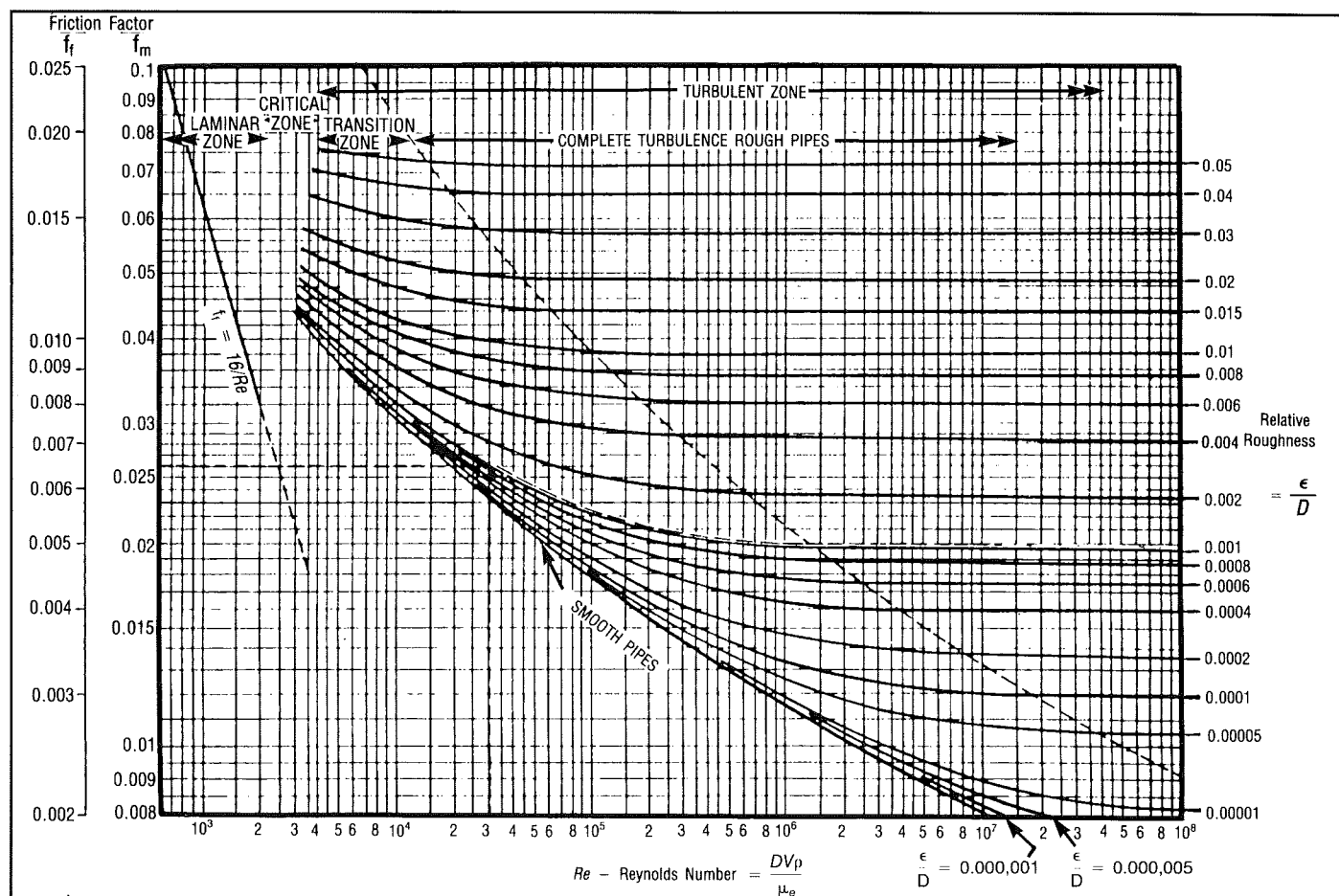
$$f_m = \frac{64 \mu_e}{DV\rho} = \left(\frac{64}{V\rho} \right) \left(\frac{\mu}{1488} \right) \left(\frac{12}{d} \right) \quad \text{Eq 17-9}$$

This expression can then be substituted for the friction factor in Eq 17-7, resulting in the following formula for pressure loss in pounds per square inch:

$$\Delta P_f = 0.000668 \frac{\mu LV}{d^2} \quad \text{Eq 17-10}$$

Eq 17-10 is commonly known as Poiseuille's law for laminar flow.

FIG. 17-2
Friction Factors⁵



When the flow is turbulent, the friction factor depends on the Reynolds number and the relative roughness of the pipe, ϵ/D , which is the roughness of the pipe, ϵ , over the pipe diameter, D . Fig. 17-2 incorporates the relative roughness of the pipe into the determination of the friction factor. Fig. 17-3 indicates relative roughness and friction factors for various piping materials. These figures are based on the iterative solution of the following equation developed by Colebrook⁴.

$$\frac{1}{\sqrt{f_m}} = -2 \log_{10} \left(\frac{\epsilon}{3.7 D} + \frac{2.51}{\text{Re} \sqrt{f_m}} \right) \quad \text{Eq 17-11}$$

Equivalent Length of Valves and Fittings

The pressure drop effects of valves and fittings can be accounted for by addition of the "equivalent lengths" of the fittings to the actual piping lengths. This augmented pipe length is then used in any of the following pressure drop calculation techniques. A table of equivalent lengths for a number of representative valves and fittings appears in Fig. 17-4.

Compressibility of Gases

For more accurate values of Z , refer to Section 23. For more approximate calculations, the value of the average compressibility factor, Z_{avg} , may be calculated from the following relations:

$$Z_{\text{avg}} = \frac{1}{(F_{\text{pv}})^2} \quad \text{Eq 17-12}$$

and

$$F_{\text{pv}} = 1 + \left[\frac{(P_{\text{avg}}) (3.444) (10^5) (10^{(1.785)(S)})}{T_{\text{avg}}^{3.825}} \right] \quad \text{Eq 17-13}$$

Fig. 17-5 contains a plot of the deviation factor, F_{pv} , virtually identical to those calculated by this equation.

An estimate for Z_{avg} at pressures below 100 psi is:

$$Z_{\text{avg}} = \frac{1}{1 + 0.0002 P_{\text{avg}}} \quad \text{Eq 17-14}$$

SINGLE PHASE FLOW

Transmission Line Gas Flow

Isothermal Flow — The steady-state, isothermal flow behavior of gas in pipelines is defined by a general energy equation of the form:

$$Q = 38.77 \left(\frac{T_b}{P_b} \right) E \sqrt{\frac{1}{f_f}} \left[\frac{P_1^2 - P_2^2}{S L_m T_{\text{avg}} Z_{\text{avg}}} \right]^{0.5} d^{2.5} \quad \text{Eq 17-15}$$

This equation is completely general for steady-state flow, and adequately accounts for variations in compressibility factor, kinetic energy, pressure, and temperature for any typical line section. However, the equation as derived involves an unspecified value of the transmission factor, $\sqrt{1/f_f}$. The correct representation of this friction factor is necessary to the validity of the equation.

The friction factor is fundamentally related to the energy lost due to friction. In the derivation of the general energy equation, all irreversibilities and non-idealities, except for those covered by the real gas law, have been collected into the friction loss term.

Empirical methods historically and currently used to calculate or predict the flow of gas in a pipeline are the result of various correlations of the transmission factor substituted into the general energy equation.

Examination of the relationships presented by various authors shows that their forms differ primarily in the inherent or specified representation of the transmission factor which defines the energy lost in resistance to flow for various pipe sizes, roughnesses, flow conditions, and gases.

To obtain Eq 17-15, which is convenient for general calculations, a number of simplifying assumptions have been made. For other than pipeline sections with a very high pressure gradient, the change in the kinetic energy of the gas is not significant, and is assumed equal to zero. It is also assumed that the gas temperature is constant at an average value for the section considered; the compressibility factor is constant at the value characterized by the average gas temperature and pressure; and in the term giving the effect of elevation change, the pressure is constant at the average value. In the range of conditions to which pipeline flow equations are ordinarily applied, averages are usually sufficiently accurate. Average temperatures are calculated as indicated in Fig. 17-1.

The average pressure in the line can be computed by:

$$P_{\text{avg}} = \frac{2}{3} \left(P_1 + P_2 - \frac{P_1 P_2}{P_1 + P_2} \right) \quad \text{Eq 17-16}$$

In the absence of field data indicating otherwise, an efficiency factor, E , of 1.0 is usually assumed.

The AGA Equations — The AGA Equations were developed to approximate partially and fully turbulent flow using two different transmission factors. The fully turbulent flow equation accounts for the relative pipe roughness, ϵ/D , based on the rough-pipe law.⁴ This equation uses the following transmission factor:

$$\sqrt{1/f_f} = 4 \log_{10} \left(\frac{3.7 D}{\epsilon} \right) \quad \text{Eq 17-17}$$

When the transmission factor for fully turbulent flow is substituted in the general energy equation (Eq 17-15), the AGA Equation for fully turbulent flow becomes:

$$Q = 38.77 \left(\frac{T_b}{P_b} \right) E \left[4 \log_{10} \left(\frac{3.7 D}{\epsilon} \right) \right] \left[\frac{P_1^2 - P_2^2}{S L_m T_{\text{avg}} Z_{\text{avg}}} \right]^{0.5} d^{2.5} \quad \text{Eq 17-18}$$

The partially turbulent flow equation is based on the smooth-pipe law⁴ and is modified to account for drag-inducing elements. The transmission factor for this equation is:

$$\sqrt{1/f_f} = 4 \log_{10} \frac{R_e}{\sqrt{1/f_f}} - 0.6 \quad \text{Eq 17-19}$$

FIG. 17-3

Relative Roughness of Pipe Materials and Friction Factors for Complete Turbulence⁵

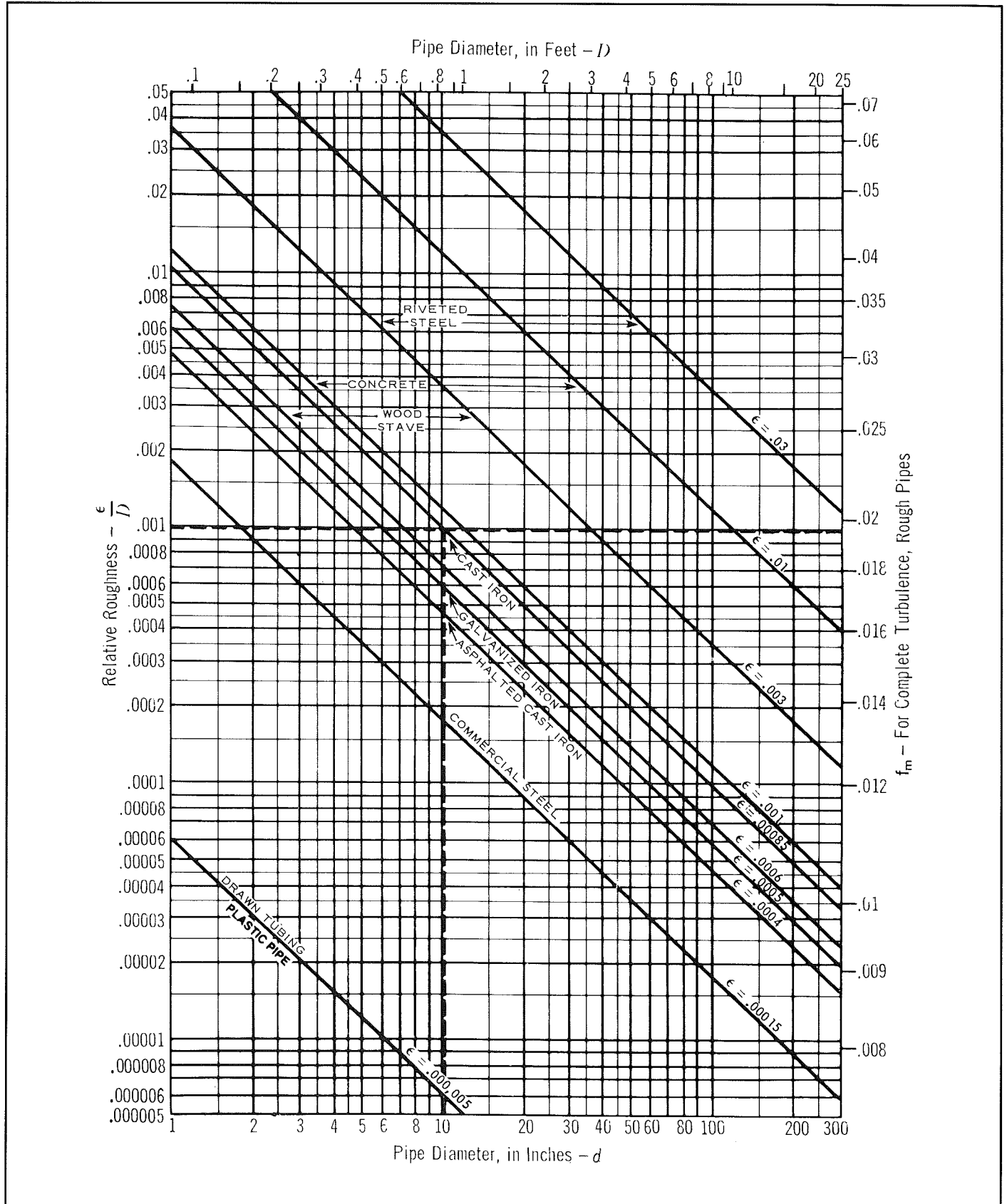


FIG. 17-4

Equivalent Length of Valves and Fittings in Feet

| Nominal Pipe size in. | | Globe valve or ball check valve | Angle valve | Swing check valve | Plug cock | Gate or ball valve | 45° ell | | Short rad. ell | | Long rad. ell | | Hard T | | Soft T | | 90° miter bends | | | Enlargement | | | | | Contraction | | | | | | | | | | | |
|-----------------------|-----|---------------------------------|-------------|-------------------|-----------|--------------------|---------|----------|----------------|----------|---------------|----------|--------|----------|--------|----------|-----------------|---------|---------|------------------------------|-----------|-----------|-----------|-----------|-------------|-----------|-----------|-----------|-----------|--|--|--|--|--|--|--|
| | | | | | | | Welded | Threaded | Welded | Threaded | Welded | Threaded | Welded | Threaded | Welded | Threaded | 2 miter | 3 miter | 4 miter | Sudden | | Std. red. | | | Sudden | | | Std. red. | | | | | | | | |
| | | | | | | | | | | | | | | | | | | | | Equiv. L in terms of small d | | | | | | | | | | | | | | | | |
| | | | | | | | | | | | | | | | | | | | | d/D = 1/4 | d/D = 1/2 | d/D = 3/4 | d/D = 1/2 | d/D = 3/4 | d/D = 1/4 | d/D = 1/2 | d/D = 3/4 | d/D = 1/2 | d/D = 3/4 | | | | | | | |
| 1½ | 55 | 26 | 13 | 7 | 1 | 1 | 2 | 3 | 5 | 2 | 3 | 8 | 9 | 2 | 3 | | | 5 | 3 | 1 | 4 | 1 | 3 | 2 | 1 | 1 | - | | | | | | | | | |
| 2 | 70 | 33 | 17 | 14 | 2 | 2 | 3 | 4 | 5 | 3 | 4 | 10 | 11 | 3 | 4 | | | 7 | 4 | 1 | 5 | 1 | 3 | 3 | 1 | 1 | - | | | | | | | | | |
| 2½ | 80 | 40 | 20 | 11 | 2 | 2 | - | 5 | - | 3 | - | 12 | - | 3 | - | | | 8 | 5 | 2 | 6 | 2 | 4 | 3 | 2 | 2 | - | | | | | | | | | |
| 3 | 100 | 50 | 25 | 17 | 2 | 2 | | 6 | | 4 | 14 | 4 | 4 | | | | | 10 | 6 | 2 | 8 | 2 | 5 | 4 | 2 | 2 | - | | | | | | | | | |
| 4 | 130 | 65 | 32 | 30 | 3 | 3 | | 7 | | 5 | 19 | 5 | | | | | | 12 | 8 | 3 | 10 | 3 | 6 | 5 | 3 | 3 | - | | | | | | | | | |
| 6 | 200 | 100 | 48 | 70 | 4 | 4 | | 11 | | 8 | 28 | 8 | | | | | | 18 | 12 | 4 | 14 | 4 | 9 | 7 | 4 | 4 | 1 | | | | | | | | | |
| 8 | 260 | 125 | 64 | 120 | 6 | 6 | | 15 | | 9 | 37 | 9 | | | | | | 25 | 16 | 5 | 19 | 5 | 12 | 9 | 5 | 5 | 2 | | | | | | | | | |
| 10 | 330 | 160 | 80 | 170 | 7 | 7 | | 18 | | 12 | 47 | 12 | | | | | | 31 | 20 | 7 | 24 | 7 | 15 | 12 | 6 | 6 | 2 | | | | | | | | | |
| 12 | 400 | 190 | 95 | 170 | 9 | 9 | | 22 | | 14 | 55 | 14 | 28 | 21 | 20 | | | 37 | 24 | 8 | 28 | 8 | 18 | 14 | 7 | 7 | 2 | | | | | | | | | |
| 14 | 450 | 210 | 105 | 80 | 10 | 10 | | 26 | | 16 | 62 | 16 | 32 | 24 | 22 | | | 42 | 26 | 9 | - | - | 20 | 16 | 8 | - | - | | | | | | | | | |
| 16 | 500 | 240 | 120 | 145 | 11 | 11 | | 29 | | 18 | 72 | 18 | 38 | 27 | 24 | | | 47 | 30 | 10 | - | - | 24 | 18 | 9 | - | - | | | | | | | | | |
| 18 | 550 | 280 | 140 | 160 | 12 | 12 | | 33 | | 20 | 82 | 20 | 42 | 30 | 28 | | | 53 | 35 | 11 | - | - | 26 | 20 | 10 | - | - | | | | | | | | | |
| 20 | 650 | 300 | 155 | 210 | 14 | 14 | | 36 | | 23 | 90 | 23 | 46 | 33 | 32 | | | 60 | 38 | 13 | - | - | 30 | 23 | 11 | - | - | | | | | | | | | |
| 22 | 688 | 335 | 170 | 225 | 15 | 15 | | 40 | | 25 | 100 | 25 | 52 | 36 | 34 | | | 65 | 42 | 14 | - | - | 32 | 25 | 12 | - | - | | | | | | | | | |
| 24 | 750 | 370 | 185 | 254 | 16 | 16 | | 44 | | 27 | 110 | 27 | 56 | 39 | 36 | | | 70 | 46 | 15 | - | - | 35 | 27 | 13 | - | - | | | | | | | | | |
| 30 | - | - | - | 312 | 21 | 21 | | 55 | | 40 | 140 | 40 | 70 | 51 | 44 | | | | | | | | | | | | | | | | | | | | | |
| 36 | - | - | - | - | 25 | 25 | | 66 | | 47 | 170 | 47 | 84 | 60 | 52 | | | | | | | | | | | | | | | | | | | | | |
| 42 | - | - | - | - | 30 | 30 | | 77 | | 55 | 200 | 55 | 98 | 69 | 64 | | | | | | | | | | | | | | | | | | | | | |
| 48 | - | - | - | - | 35 | 35 | | 88 | | 65 | 220 | 65 | 112 | 81 | 72 | | | | | | | | | | | | | | | | | | | | | |
| 54 | - | - | - | - | 40 | 40 | | 99 | | 70 | 250 | 70 | 126 | 90 | 80 | | | | | | | | | | | | | | | | | | | | | |
| 60 | - | - | - | - | 45 | 45 | | 110 | | 80 | 260 | 80 | 190 | 99 | 92 | | | | | | | | | | | | | | | | | | | | | |

Substituting $\sqrt{1/f_f}$ from Eq 17-19 into Eq 17-15 does not provide an equation which can be solved directly. For partially turbulent flow a frictional drag factor must also be applied to account for the effects of pipe bends and irregularities. These calculations are beyond the scope of this book and the AGA "Steady Flow in Gas Pipelines"⁶ should be consulted for a detailed treatment of partially turbulent flow.

The Weymouth Equation — The Weymouth Equation, published in 1912⁷, evaluated the coefficient of friction as a function of the diameter.

$$f_f = \frac{0.008}{d^{1/3}} \quad \text{Eq 17-20}$$

$$\sqrt{1/f_f} = 11.18 d^{1/6} \quad \text{Eq 17-21}$$

When the friction factor, f_f , is substituted in the general energy equation, Weymouth's Equation becomes:

$$Q = (433.5) \left(\frac{T_b}{P_b} \right) E \left[\frac{P_1^2 - P_2^2}{S L_m T_{avg} Z_{avg}} \right]^{0.5} d^{2.667} \quad \text{Eq 17-22}$$

The Weymouth formula for short pipelines and gathering systems agrees more closely with metered rates than those calculated by most other formulae. However, the degree of error increases with pressure. If the Q calculated from the Weymouth formula is multiplied by $\sqrt{1/Z}$, where Z is the compressibility factor of the gas, the corrected Q will closely approximate the metered flow. Fig. 17-5 shows a plot of the deviation factor, $\sqrt{1/Z}$, of a common gas and can be used safely if exact data is not available.

The equation cannot be generally applied to any variety of diameters and roughness, and in the flow region of partially

developed turbulence, it is not valid. The Weymouth Equation may be used to approximate fully turbulent flow by applying correction factors determined from the system to which it is to be applied. Graphs showing gas flow calculations based on the Weymouth equation are shown in Fig. 17-6a and 17-6b.

Panhandle A Equation — In the early 1940s Panhandle Eastern Pipe Line Company developed a formula for calculation of gas flow in transmission lines which has become known as the Panhandle A Equation. This equation uses the following expressions of Reynolds number and transmission factor.

$$Re = 1.934 \frac{QS}{d} \quad \text{Eq 17-23}$$

$$\sqrt{1/f_f} = 7.211 \left(\frac{QS}{d} \right)^{0.07305} = 6.872 (Re)^{0.07305} \quad \text{Eq 17-24}$$

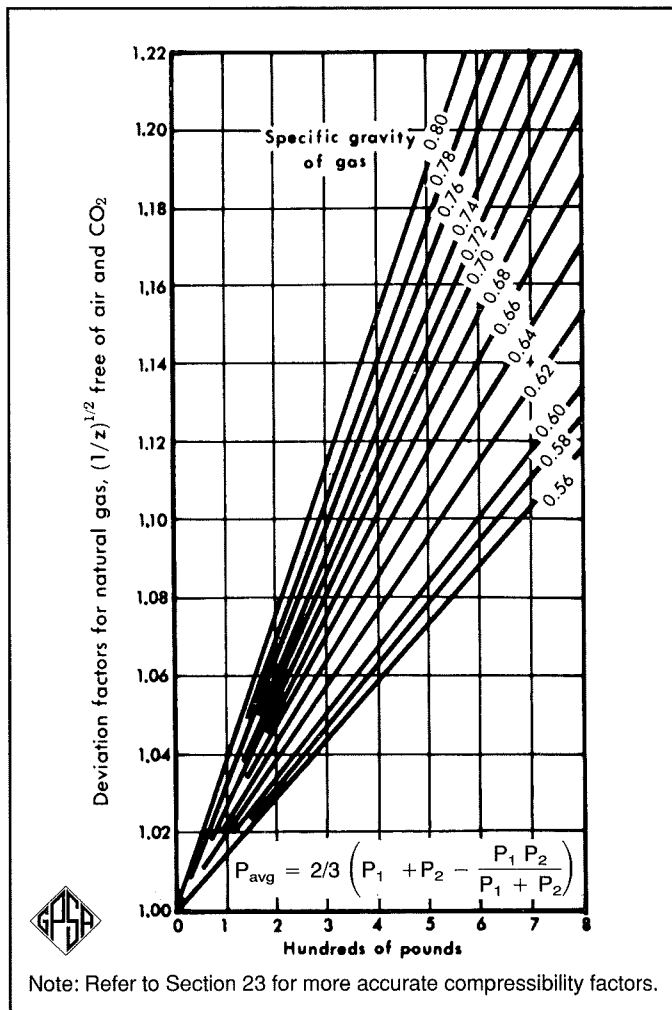
The transmission factor assumes a Reynolds number value from 5 to 11 million based on actual metered experience.

Substituting Eq 17-24 for $\sqrt{1/f_f}$ in the general energy equation (Eq 17-15), the Panhandle A Equation becomes:

$$Q = 435.87 \left(\frac{T_b}{P_b} \right)^{1.0788} E \left[\frac{P_1^2 - P_2^2}{S^{0.853} L_m T_{avg} Z_{avg}} \right]^{0.5392} d^{2.6182} \quad \text{Eq 17-25}$$

This equation was intended to reflect the flow of gas through smooth pipes. When "adjusted" with an efficiency factor, E , of about 0.90, the equation is a reasonable approximation of the partially turbulent flow equation. The equation becomes less accurate as flow rate increases. Many users of the Panhandle A Equation assume an efficiency factor of 0.92.

FIG. 17-5
Deviation Factors⁸



Panhandle B Equation — A new or revised Panhandle Equation was published in 1956. This revised equation is known as the Panhandle B Equation and is only slightly Reynolds number dependent. Therefore, it more nearly approximates fully turbulent flow behavior. The transmission factor used here is:

$$\sqrt{1/f_f} = 16.70 \left(\frac{QS}{d} \right)^{0.01961} = 16.49 (Re)^{0.01961} \quad \text{Eq 17-26}$$

Substituting Eq 17-23 for $\sqrt{1/f_f}$ in the general energy equation (Eq 17-15), the Panhandle B Equation becomes:

$$Q = 737 \left(\frac{T_b}{P_b} \right)^{1.02} E \left[\frac{P_1^2 - P_2^2}{S^{0.961} L_m T_{avg} Z_{avg}} \right]^{0.51} d^{2.53} \quad \text{Eq 17-27}$$

The equation can be adjusted through the use of an efficiency term that makes it applicable across a relatively limited range of Reynolds numbers. Other than this, however, there are no means for adjustment of the equation to correct it for variations in pipe surface. Adjusted to an average flowing Reynolds number, the equation will predict low flow rates at low Reynolds numbers, and high flow rates at high Reynolds

numbers, as compared to a fully turbulent flow equation. Efficiencies based on the Panhandle B equation decrease with increasing flow rate for fully turbulent flow. The efficiency factor, E , used in the Panhandle B equation generally varies between about 0.88 and 0.94.

Conclusions — The successful application of these transmission line flow equations in the past has largely involved compensation for discrepancies through the use of adjustment factors, usually termed "efficiencies." These efficiencies are frequently found in practice by determining the constant required to cause predicted gas equation behavior to agree with flow data. As a result, the values of these factors are specific to particular gas flow equations and field conditions, and, under many circumstances, vary with flow rate in a fashion that obscures the real nature of flow behavior in the pipe.

The Reynolds number dependent equations, such as the Panhandle equations, utilize a friction factor expression which yields an approximation to partially turbulent flow behavior in the case of the Panhandle A equation, and an approximation to fully turbulent behavior in the case of the Panhandle B.

These equations suffer from the substitution of a fixed gas viscosity value into the Reynolds number expression, which, in turn, substituted into the flow equation, results in an expression with a preconditioned bias.

Regardless of the merits of various gas flow equations, past practices may dictate the use of a particular equation to maintain continuity of comparative capacities through application of consistent operating policy. A summary of comparisons between transmission factors used in the above gas equations are shown in Fig. 17-7.

Reference should be made to "Steady Flow in Gas Pipelines"⁶, published by American Gas Association, for a complete analysis of steady flow in gas pipelines.

Low Pressure Gas Flow

Gas gathering often involves operating pressures below 100 psi. Some systems flow under vacuum conditions. For these low pressure conditions, equations have been developed that give a better fit than the Weymouth or Panhandle Equations. Two such formulae are:

The Oliphant Formula⁹ for gas flow between vacuum and 100 psi:

$$Q = 42 (24) \left(d^{2.5} + \frac{d^3}{30} \right) \left(\frac{14.4}{P_b} \right) \left(\frac{T_b}{520} \right) \left[\left(\frac{0.6}{S} \right) \left(\frac{520}{T} \right) \left(\frac{P_1^2 - P_2^2}{L_m} \right) \right]^{1/2} \quad \text{Eq 17-28}$$

The Spitzglass Formula for gas flow below 1 psig at 60°F.

$$Q = (24) (3550) \left[\frac{27.69 (P_1 - P_2) d^5}{SL \left(1 + \frac{3.6}{d} + 0.03 d \right)} \right]^{1/2} \quad \text{Eq 17-29}$$

Plant Piping Gas Flow

For estimating pressure drop in short runs of gas piping, such as within plant or battery limits, a simplified formula for compressible fluids is accurate for fully turbulent flow, assuming the pressure drop through the line is not a significant fraction of the total pressure (i.e. no more than 10%).

FIG. 17-6a
Gas Flow Based On Weymouth Formula

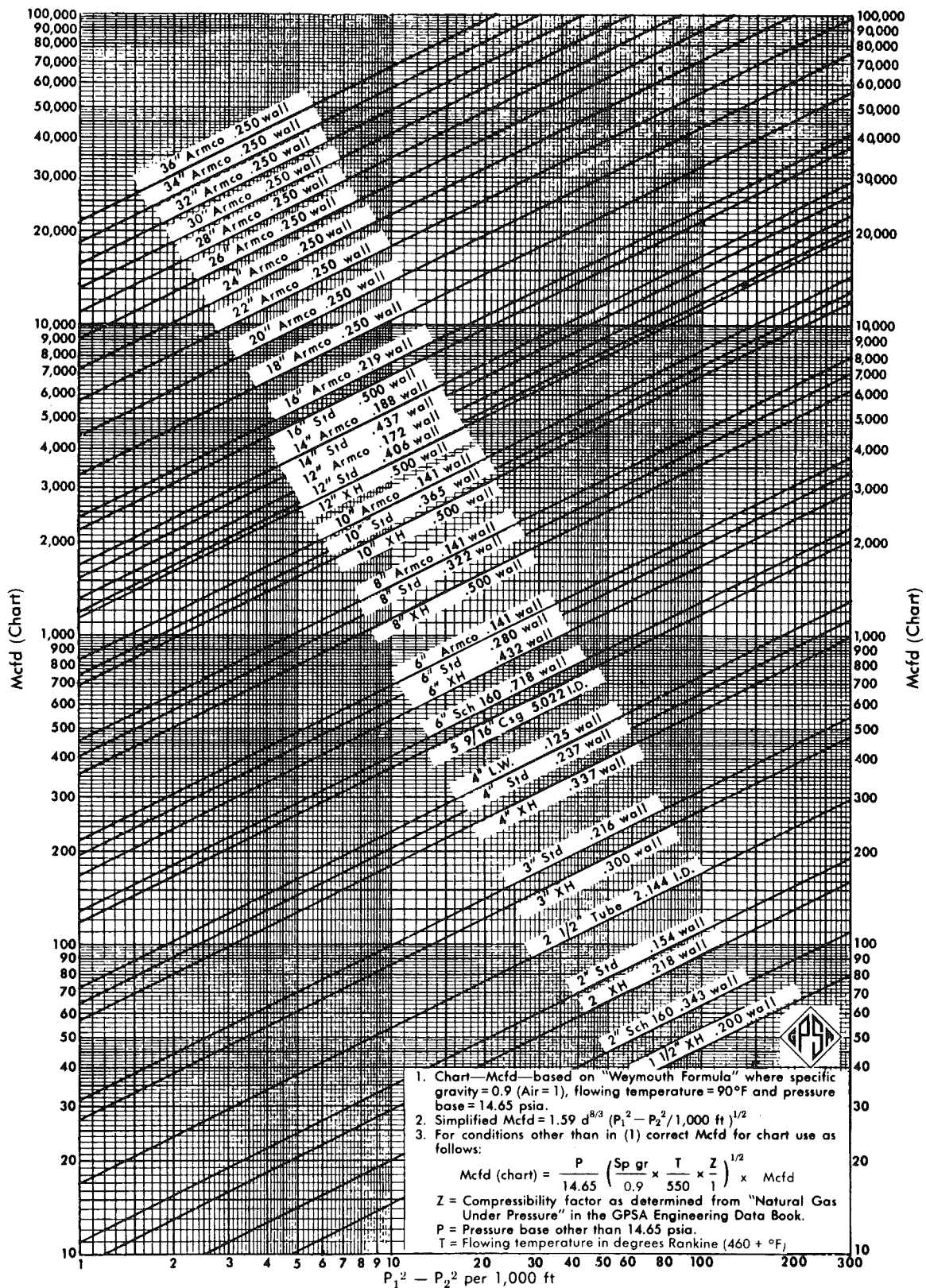


FIG. 17-6b

Gas Flow Based On Weymouth Formula

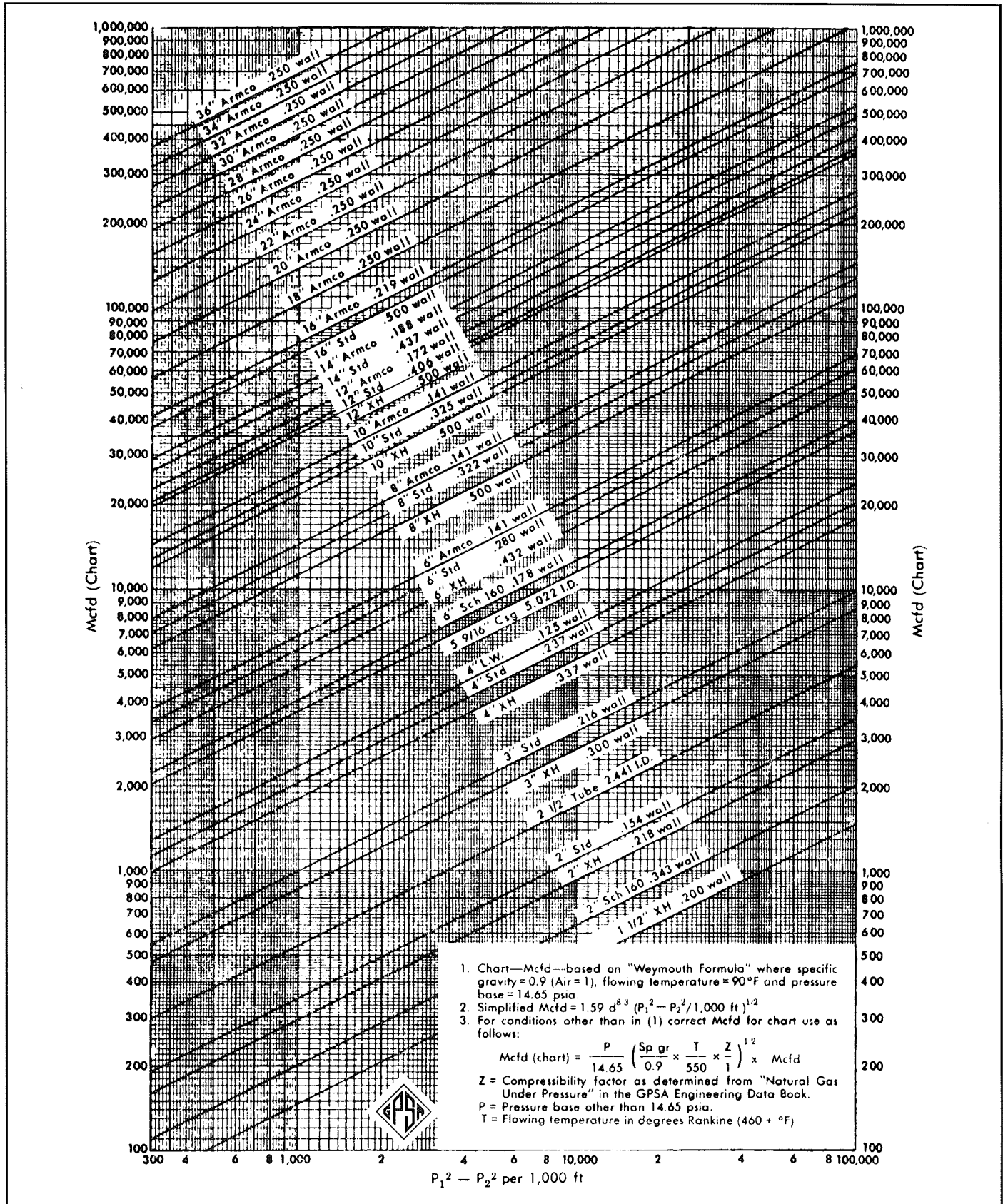
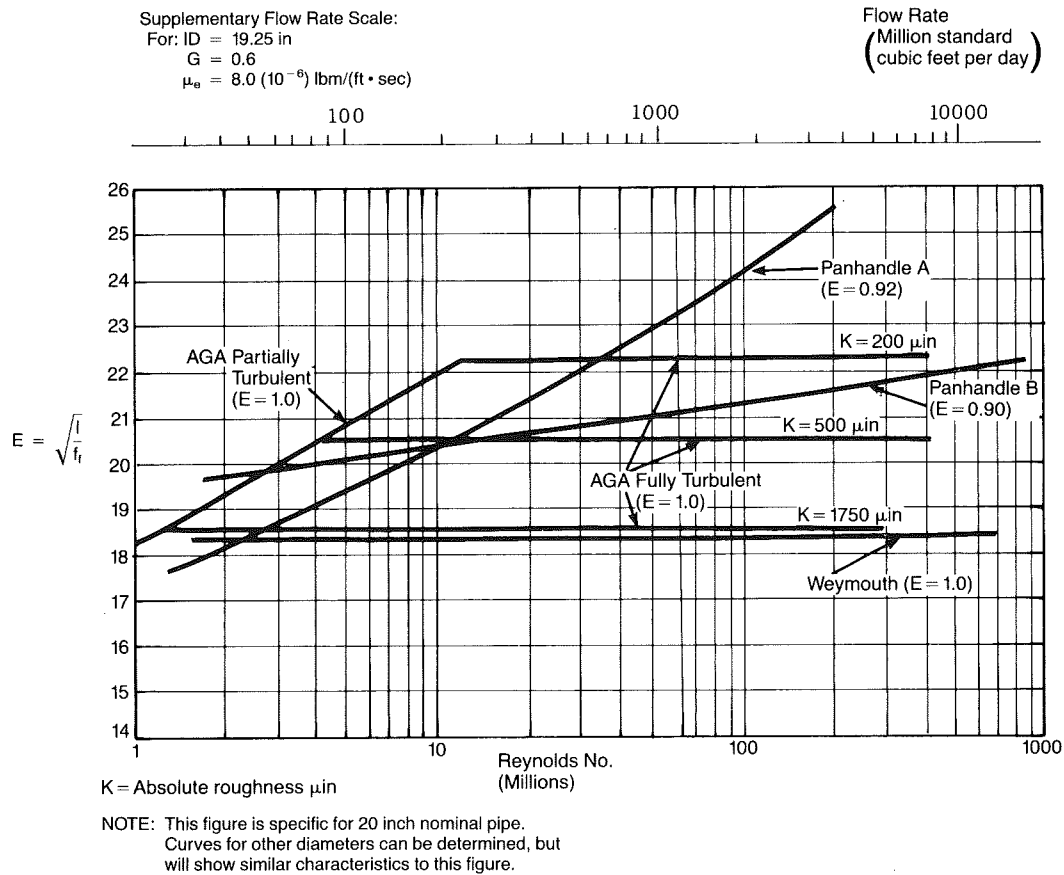


FIG. 17-7

Comparison of Gas Equation Transmission Factors for Nominal 20 Inch Pipe



The following method is a simplification of the Darcy formula, which eliminates calculation of f_m , the Moody friction factor. This simplification was checked over a wide range of flows and densities for pressure drops of 0.25 to 1.5 psi/100 ft. Density was varied over a range of 100 to 1; flows varied over a range of 75 to 1. Pressure variation was from atmospheric to 1000 psia. The error from using the simplified approach as compared to the actual friction factor calculated in the Darcy formula was from zero to 5%, with the simplified approach giving consistently lower calculated pressure drop for a given flow.

The Darcy formula can be written in the simple form:

$$\Delta P_{100} = \frac{W^2}{\rho} \left(\frac{0.000336 f_m}{d^5} \right) \quad \text{Eq 17-30}$$

Simplifying, $C_1 = W^2 (10^{-9})$, and $C_2 = \frac{336,000 f_m}{d^5}$, then

$$\Delta P_{100} = \frac{C_1 C_2}{\rho} \quad \text{Eq 17-31}$$

$$C_1 = \frac{(\Delta P_{100}) \rho}{C_2} = \text{discharge factor from chart, Fig. 17-8}$$

$$C_2 = \frac{(\Delta P_{100}) \rho}{C_1} = \text{size factor Fig. 17-9}$$

C_2 incorporates the friction factor, assuming clean steel. Using this simplified approach, new lines can be sized by setting the desired ΔP_{100} and solving for C_2 with a given flow. For a given flow and pipe size, ΔP_{100} can be solved directly.

Example 17-1 — Calculate the pressure drop in a 10-in., Schedule 40 pipe for a flow of 150,000 lb/hr of methane. Temperature is 60°F and pressure is 750 psia. The compressibility factor is 0.905 (from Fig. 23-3).

Solution Steps

$$\rho = \frac{16.042 (750)}{10.73 (460 + 60) (0.905)} = 2.38 \text{ lb/ft}^3$$

C_1 from Fig. 17-8 is 22.5

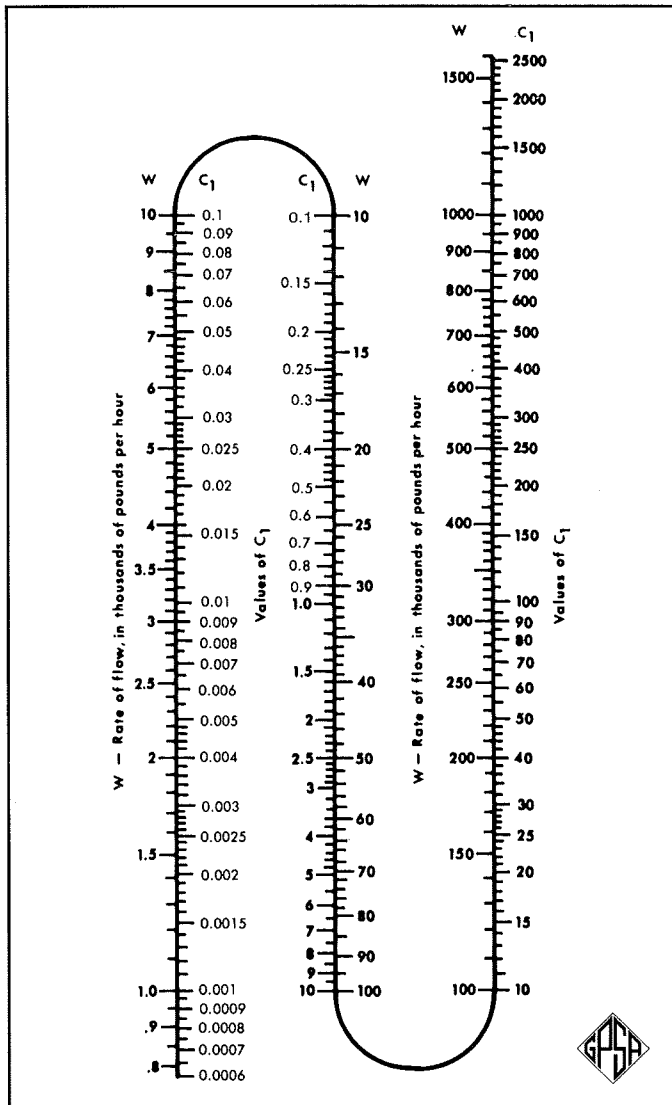
C_2 from Fig. 17-9 is 0.0447

$$\begin{aligned} \Delta P_{100} &= \frac{C_1 C_2}{\rho} = \frac{22.5 (0.0447)}{2.38} \\ &= 0.423 \text{ psi/100 ft using Eq 17-31} \end{aligned}$$

Example 17-2 — Calculate the required line size (of Schedule 40 pipe) to give $\Delta P_{100} = 1$ psi or less when flowing 75,000 lb/hr of methane at 400 psia and 100°F. The compressibility factor is 0.96 (from Fig. 23-8).

FIG. 17-8

Simplified Flow Formula for Compressible Fluids⁵ Values of C_1



Solution Steps

$$\rho = \frac{16.062 (400)}{10.73 (460 + 100) (0.96)} = 1.11 \text{ lb/ft}^3$$

C_1 from Fig. 17-8 is 5.6

$$C_2 = \frac{(\Delta P_{100}) \rho}{C_1} = \frac{1 (1.11)}{5.6} = 0.20$$

From Fig. 17-9 the smallest size of Schedule 40 pipe with C_2 less than 0.20 is 8-in. pipe. For 8 in. Sch 40 pipe, C_2 is 0.146. The actual pressure drop can then be calculated as:

$$\Delta P_{100} = \frac{5.6 (0.146)}{1.11} = 0.74 \text{ psi/100 ft}$$

using Eq 17-31 for the above flow conditions.

Liquid Flow

For the calculation of pressure drop in liquid lines, the Darcy-Weisbach method, Eq 17-6, can be used. The calculation is simplified for liquid flows since the density can reasonably be assumed to be a constant. As a result, the Darcy-Weisbach calculation can be applied to a long run of pipe, rather than segmentally as dictated by the variable density in gas flow. In addition, several graphical aids are available for pressure drop calculation. Elevation pressure drops must be calculated separately using Eq 17-32. These elevation pressure gains or losses are added algebraically to the frictional pressure drops.

$$\Delta P_e = (0.00694) \rho_L Z_e \quad \text{Eq 17-32}$$

Water — A graph showing pressure drop for water per 100 feet as a function of flow rate in gallons per minute and pipe size is shown in Fig. 17-10. These data are based on the Hazen and Williams empirical formula¹⁰ using a "C" constant of 100 which is commonly used for design purposes in welded and seamless steel pipe.

Hazen and Williams formula for flow of water:

$$q = 0.442 d^{2.63} C \left(\frac{P_1 - P_2}{L} \right)^{0.54} \quad \text{Eq 17-33}$$

Where:

$C = 140$ for new steel pipe

$C = 130$ for new cast iron pipe

$C = 100$ is often used for design purposes to account for pipe fouling, etc.

Hydrocarbon — A graph showing pressure drop for hydrocarbons per 100 feet as a function of flow rate in gallons per minute and pipe size is shown in Fig. 17-11. This graph assumes a specific gravity of 1.0 (water). To correct for different liquid densities, the value read from Fig. 17-11 must be multiplied by the actual specific gravity to obtain the correct pressure loss.

Steam Flow

Fig. 17-12 contains a graphical representation of Fritzsche's formula¹¹ for calculating pressure drop in steam lines. Fritzsche's formula and instructions for the chart usage are given in Fig. 17-12.

The Babcock formula⁵ for steam flow is:

$$\Delta P_f = 3.63 (10^{-8}) \left(\frac{d + 3.6}{d^6} \right) \frac{W^2 L}{\rho} \quad \text{Eq 17-34}$$

Fire Stream Flow

Fig. 17-13 is a table permitting rapid computation of the behavior of various sized fire nozzles. The table also includes the estimated pressure drop in 100 feet of 2-1/2 inch diameter fire hose.

TWO PHASE FLOW

Two-phase flow presents several design and operational difficulties not present in single phase liquid or vapor flow. Frictional pressure drops are much harder to calculate. For cross-country pipelines, a terrain profile is needed to calculate elevation pressure drops. At the downstream end of a pipeline, it is frequently necessary to separate the liquid and vapor phases in a separator. The presence of liquid slugs complicates this process, and a slug catcher may be required.

FIG. 17-9
Simplified Flow Formula for Compressible Fluids⁵
Values of C₂

| Nominal pipe size in. | Schedule number | Value of C ₂ | Nominal pipe size in. | Schedule number | Value of C ₂ | Nominal pipe size in. | Schedule number | Value of C ₂ |
|-----------------------|-----------------|-------------------------|-----------------------|-----------------|-------------------------|---|-----------------|-------------------------|
| 1/8 | 40 s | 7 920 000. | 6 | 40 s | 0.610 | 16 | 10 | 0.004 63 |
| | 80 x | 26 200 000. | | 80 x | 0.798 | | 20 | 0.004 83 |
| 1/4 | 40 s | 1 590 000. | | 120 | 1.015 | | 30 s | 0.005 04 |
| | 80 x | 4 290 000. | | 160 | 1.376 | | 40 x | 0.005 49 |
| 3/8 | 40 s | 319 000. | | ... xx | 1.861 | | 60 | 0.006 12 |
| | 80 x | 718 000. | 8 | 20 | 0.133 | | 80 | 0.007 00 |
| 1/2 | 40 s | 93 500. | | 30 | 0.135 | | 100 | 0.008 04 |
| | 80 x | 186 100. | | 40 s | 0.146 | | 120 | 0.009 26 |
| | 160 | 430 000. | | 60 | 0.163 | | 140 | 0.010 99 |
| | ... xx | 11 180 000. | | 80 x | 0.185 | | 160 | 0.012 44 |
| 3/4 | 40 s | 21 200. | | 100 | 0.211 | 18 | 10 | 0.002 47 |
| | 80 x | 36 900. | | 120 | 0.252 | | 20 | 0.002 56 |
| | 160 | 100 100. | | 140 | 0.289 | | ... s | 0.002 66 |
| | ... xx | 627 000. | | ... xx | 0.317 | | 30 | 0.002 76 |
| 1 | 40 s | 5 950. | | 160 | 0.333 | | ... x | 0.002 87 |
| | 80 x | 9 640. | 10 | 20 | 0.039 7 | | 40 | 0.002 98 |
| | 160 | 22 500. | | 30 | 0.042 1 | | 60 | 0.003 35 |
| | ... xx | 114 100. | | 40 s | 0.044 7 | | 80 | 0.003 76 |
| 1 1/4 | 40 s | 1 408. | | 60 x | 0.051 4 | | 100 | 0.004 35 |
| | 80 x | 2 110. | | 80 | 0.056 9 | | 120 | 0.005 04 |
| | 160 | 3 490 | | 100 | 0.066 1 | | 140 | 0.005 73 |
| | ... xx | 13 640. | | 120 | 0.075 3 | | 160 | 0.006 69 |
| 1 1/2 | 40 s | 627. | | 140 | 0.090 5 | 20 | 10 | 0.001 41 |
| | 80 x | 904. | | 160 | 0.105 2 | | 20 s | 0.001 50 |
| | 160 | 1 656. | 12 | 20 | 0.015 7 | | 30 x | 0.001 61 |
| | ... xx | 4 630. | | 30 | 0.016 8 | | 40 | 0.001 69 |
| 2 | 40 s | 169. | | ... s | 0.017 5 | | 60 | 0.001 91 |
| | 80 x | 236. | | 40 | 0.018 0 | | 80 | 0.002 17 |
| | 160 | 488. | | ... x | 0.019 5 | | 100 | 0.002 51 |
| | ... xx | 899. | | 60 | 0.020 6 | | 120 | 0.002 87 |
| 2 1/2 | 40 s | 66.7 | | 80 | 0.023 1 | | 140 | 0.003 35 |
| | 80 x | 91.8 | | 100 | 0.026 7 | | 160 | 0.003 85 |
| | 160 | 146.3 | 14 | 120 | 0.031 0 | 24 | 10 | 0.000 534 |
| | ... xx | 380.0 | | 140 | 0.035 0 | | 20 s | 0.000 565 |
| 3 | 40 s | 21.4 | | 160 | 0.042 3 | | ... x | 0.000 597 |
| | 80 x | 28.7 | | | | | 30 | 0.000 614 |
| | 160 | 48.3 | | 10 | 0.009 49 | | 40 | 0.000 651 |
| | ... xx | 96.6 | | 20 | 0.009 96 | | 60 | 0.000 741 |
| 3 1/2 | 40 s | 10.0 | | 30 s | 0.010 46 | | 80 | 0.000 835 |
| | 80 x | 13.2 | | 40 | 0.010 99 | | 100 | 0.000 972 |
| | | | | ... x | 0.011 55 | | 120 | 0.001 119 |
| 4 | 40 s | 5.17 | | 60 | 0.012 44 | | 140 | 0.001 274 |
| | 80 x | 6.75 | | 80 | 0.014 16 | | 160 | 0.001 478 |
| | 120 | 8.94 | | 100 | 0.016 57 | Note: The letters s, x, and xx in the columns of Schedule Numbers indicate Standard, Extra Strong, and Double Extra Strong pipe respectively. | | |
| | 160 | 11.80 | | 120 | 0.018 98 | | | |
| | ... xx | 18.59 | | 140 | 0.021 8 | | | |
| 5 | 40 s | 1.59 | | 160 | 0.025 2 | | | |
| | 80 x | 2.04 | | | | | | |
| | 120 | 2.69 | | | | | | |
| | 160 | 3.59 | | | | | | |
| | ... xx | 4.93 | | | | | | |

FIG. 17-10
Pressure Drop for Flowing Water

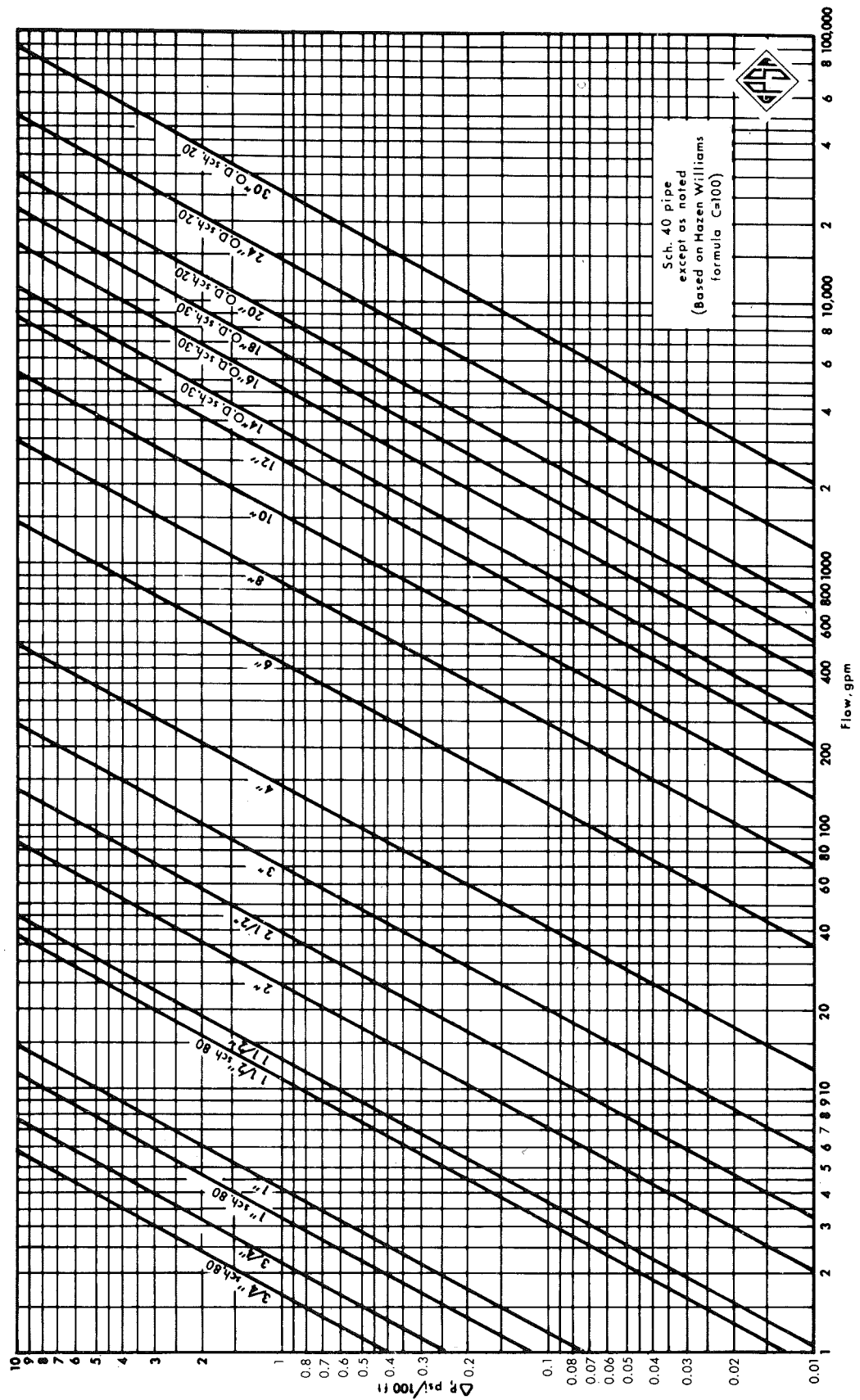
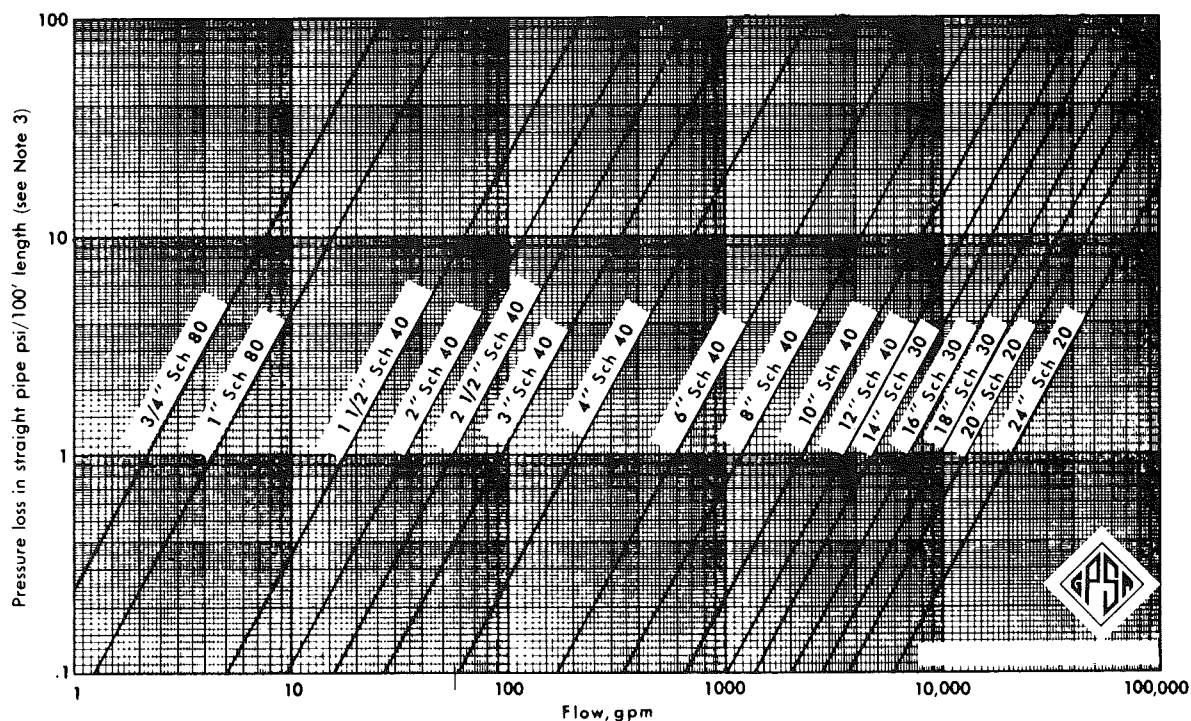


FIG. 17-11

Pressure Drop for Hydrocarbon Liquids in Smooth Pipe

**Notes:**

1. Curves calculated using the Fanning equation:

$$\Delta P_{100} = \frac{5.35 f_f q^2 \rho_L}{d^5 \rho_w}$$

2. For this chart $\epsilon = 0.00015$ ft

3. This chart assumes specific gravity = 1.0. For other values multiply pressure loss from chart by actual specific gravity (ρ_L/ρ_w).

4. For viscous fluids, use this chart only if the flow is fully turbulent (see Fig. 17-2). For viscous fluids in the partially turbulent range, use Fig. 17-2 to obtain f_m and use Eq. 17-7 to calculate ΔP_f .

Flow Regime Determination

Several empirical flow regime maps have been presented that determine vapor-liquid flow patterns as a function of fluid properties and flow rates. Diagrams of these flow patterns are shown in Fig. 17-14. One map commonly used was developed by Gregory, Aziz, and Mandhane¹² for horizontal flow. This map appears as Fig. 17-15. The coordinates of the map are:

$$\begin{aligned} V_{sL} &= \text{superficial liquid velocity} \\ &= Q_L/A \end{aligned} \quad \text{Eq 17-35}$$

$$\begin{aligned} V_{sg} &= \text{superficial gas velocity} \\ &= Q_g/A \end{aligned} \quad \text{Eq 17-36}$$

Mandhane proposed a fluid property correction to the superficial velocities, but concluded that the fluid property effects are insignificant compared to the errors in the empirical map. The map reports the flow regimes: stratified, wavy, annular mist, bubble, slug, and dispersed. Care should be taken in the interpretation of these flow maps as the regime boundaries are strongly affected by pipe inclination. In particular, horizontal flow regime maps must not be used for vertical flow,

and vertical flow regime maps must not be used for horizontal flow. The Mandhane map given in Fig. 17-15 was developed for horizontal lines flowing air and water at near atmospheric pressure. Inclinations in the range of 0.1-1.0 degrees can cause substantial regime boundary movement. In addition, flow regime boundary adjustment has been observed due to fluid pressure, pipe diameter, and surface tension.^{13,22} The gas density increase caused by high pressure acts to move the slug-mist boundary to lower superficial gas velocities, while increased pipe diameter acts to increase the stratified wavy flow regime at the expense of the slug flow regime. In addition, foamy fluids having a high surface tension have been observed to flow in the dispersed flow regime even though Mandhane would have predicted superficial liquid velocities too low to cause dispersed flow. A flow regime map generated by Taitel and Dukler contains explicit inclination effects and should be used for inclined pipes. This latter method also contains explicit pipe diameter effects but lacks any way of accounting for surface tension.

For vertical flow, the stratified flow regime cannot exist as there is no preferred direction for the liquid to settle. An empirical flow regime map developed by Aziz¹⁴ for vertical upward flow is shown in Fig. 17-16. The coordinates for this flow map are the same as for the Mandhane map in Fig. 17-15

FIG. 17-12

Pressure Drop in Steam Pipes by Fritzsche's Formula¹¹

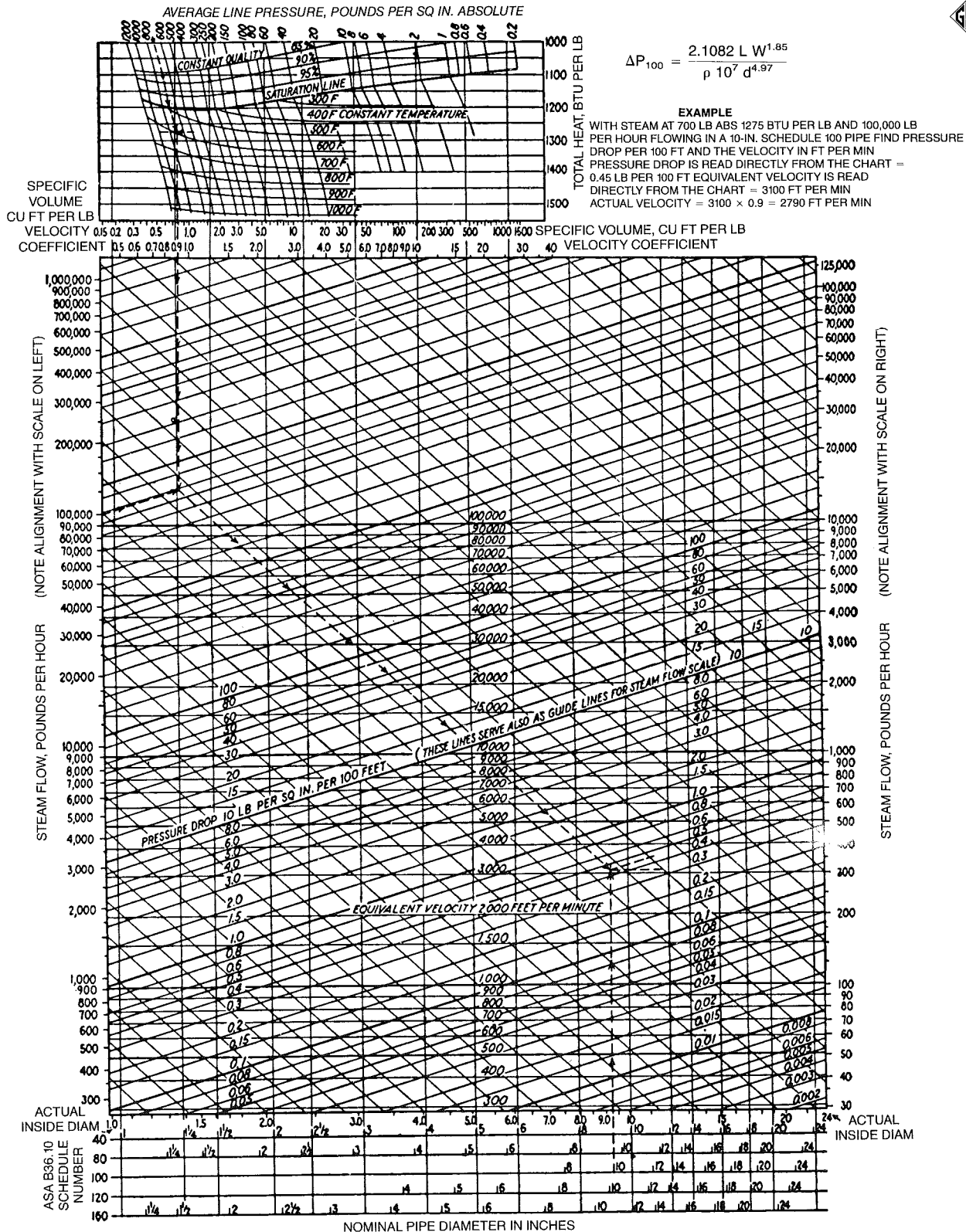


FIG. 17-13

Table of Effective Fire Streams

| Smooth nozzles, size | 3/4" | | | | | | 7/8" | | | | | | 1" | | | | | |
|--|------|-----|-----|-----|-----|-----|------|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|
| Pressure at hydrants, psi. | 32 | 43 | 54 | 65 | 75 | 85 | 34 | 46 | 57 | 69 | 80 | 91 | 37 | 50 | 62 | 75 | 87 | 100 |
| Pressure at nozzle, psi | 30 | 40 | 50 | 60 | 70 | 80 | 30 | 40 | 50 | 60 | 70 | 80 | 30 | 40 | 50 | 60 | 70 | 80 |
| Pres. lost in 100 ft. 2 1/2 in. hose psi | 2 | 3 | 4 | 5 | 5 | 6 | 4 | 6 | 7 | 9 | 10 | 11 | 7 | 10 | 12 | 15 | 17 | 20 |
| Vertical height, ft. | 48 | 60 | 67 | 72 | 76 | 79 | 49 | 62 | 71 | 77 | 81 | 85 | 51 | 64 | 73 | 79 | 85 | 89 |
| Horizontal distance, ft | 37 | 44 | 50 | 54 | 58 | 60 | 42 | 49 | 55 | 61 | 66 | 70 | 47 | 55 | 61 | 67 | 72 | 76 |
| Gal discharge/ min. | 90 | 104 | 116 | 127 | 137 | 147 | 123 | 142 | 159 | 174 | 188 | 201 | 161 | 186 | 208 | 228 | 246 | 263 |

| Smooth nozzles, size | 1 1/8" | | | | | | 1 1/4" | | | | | | 1 3/8" | | | | | |
|--|--------|-----|-----|-----|-----|-----|--------|-----|-----|-----|-----|-----|--------|-----|-----|-----|-----|-----|
| Pressure at hydrants, psi | 42 | 56 | 70 | 84 | 98 | 112 | 49 | 65 | 81 | 97 | 113 | 129 | 58 | 77 | 95 | 116 | 135 | 154 |
| Pressure at nozzle, psi | 30 | 40 | 50 | 60 | 70 | 80 | 30 | 40 | 50 | 60 | 70 | 80 | 30 | 40 | 50 | 60 | 70 | 80 |
| Pres. lost in 100 ft 2 1/2 in hose psi | 12 | 16 | 20 | 24 | 28 | 30 | 19 | 25 | 31 | 37 | 43 | 49 | 28 | 37 | 36 | 56 | 65 | 74 |
| Vertical height, ft | 52 | 65 | 75 | 83 | 88 | 92 | 53 | 67 | 77 | 85 | 91 | 95 | 55 | 69 | 79 | 87 | 92 | 97 |
| Horizontal distance, ft | 50 | 59 | 66 | 72 | 77 | 81 | 54 | 63 | 70 | 76 | 81 | 85 | 56 | 66 | 73 | 79 | 84 | 88 |
| Gal discharge/min | 206 | 238 | 266 | 291 | 314 | 336 | 256 | 296 | 331 | 363 | 392 | 419 | 315 | 363 | 406 | 445 | 480 | 514 |

Using 100 ft of 2 1/2 in. ordinary best quality rubber lined hose between nozzle and hydrant or pump.

The vertical and horizontal distance in above table are "effective" streams. The "effective stream," is one that has not broken up into a spray and which will project three fourths of the water through a circle 10 in. in diameter, at the distance indicated.

The bulk of the stream and extreme drops will carry a greater distance, but the stream is spread out too much to penetrate a hot fire and reach the burning materials before evaporation. The vertical height "h" in feet, which the bulk of the stream will carry, may be computed

by formula $h = 0.91 \frac{V^2}{2g}$

The gallons/minute discharge in the above table check approximately with the formula: $V = C\sqrt{2gH} = \frac{0.321 \times \text{gpm}}{A}$

then $\text{gpm} = \frac{VA}{0.321}$ $V = \text{Velocity in feet per second}$ $g = \text{Acceleration due to gravity,} = 32.16 \text{ ft/sec}^2$
 $A = \text{Area of nozzle, in square inches}$ $C = \text{Coefficient for smooth nozzles} = 0.98$ $H = \text{Head in feet} = \text{pressure} \times 2.31$

Reference: Fire Protection Handbook, 14th Edition, 1976, National Fire Protection Association

FIG. 17-14

Two Phase Flow Regimes

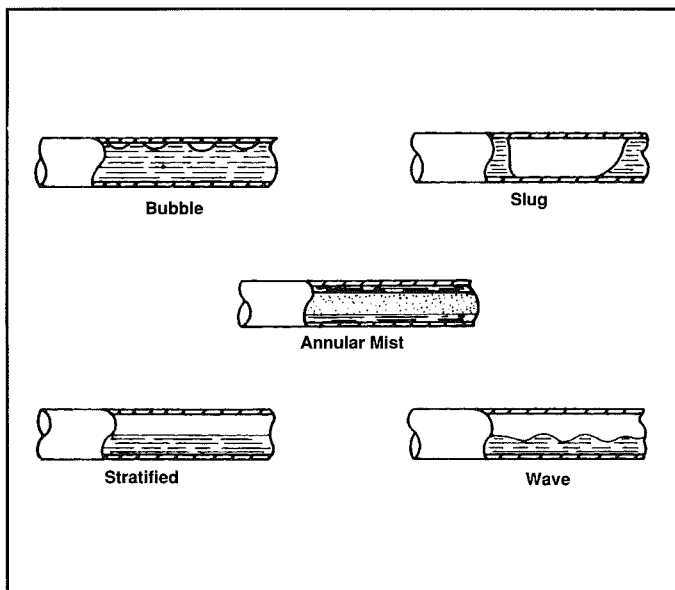


FIG. 17-15

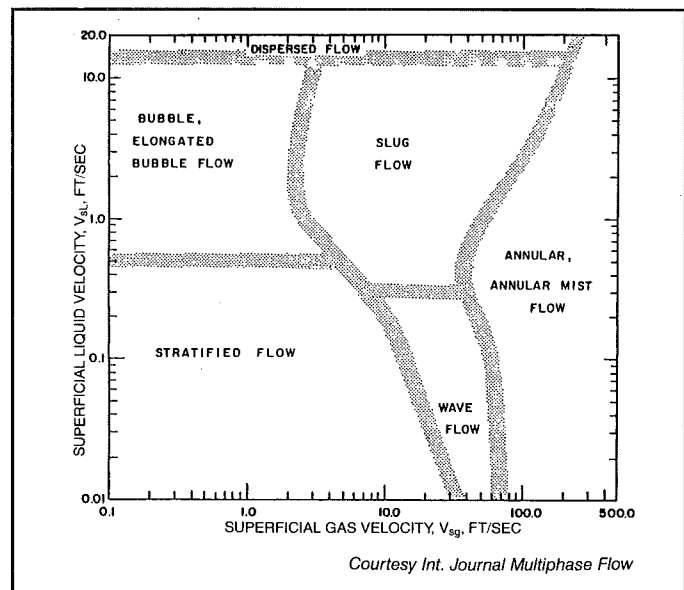
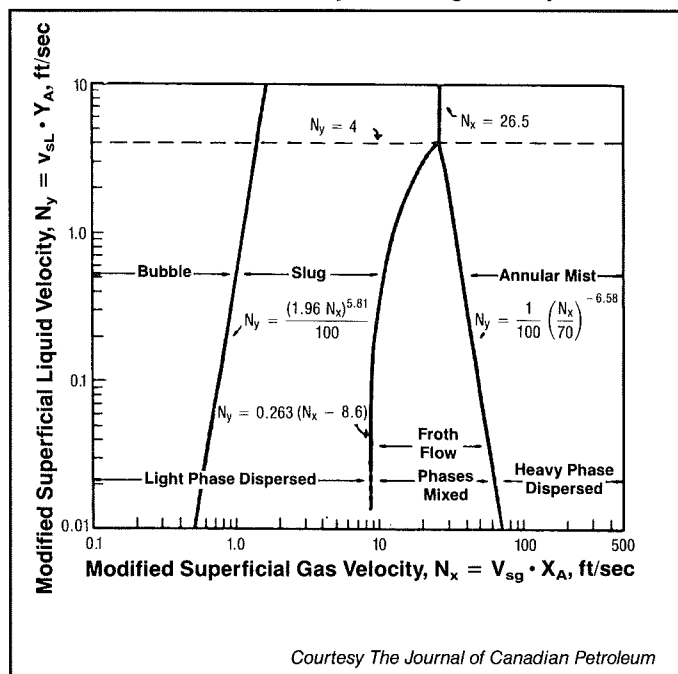
Mandhane et al. Horizontal Flow Regime Map¹²

FIG. 17-16

Aziz et al. Vertical Up-Flow Regime Map¹⁴

except that fluid property corrections are used. The coordinates used in the Aziz vertical map are:

$$N_x = V_{sg} X_A \quad \text{Eq 17-37}$$

$$N_y = V_{sl} Y_A \quad \text{Eq 17-38}$$

$$X_A = \left(\frac{\rho_g}{\rho_a} \right)^{0.333} Y_A \quad \text{Eq 17-39}$$

$$Y_A = \left(\frac{\rho_L \sigma_{wa}}{\rho_w \sigma} \right)^{0.25} \quad \text{Eq 17-40}$$

For pipe inclinations greater than 10-20 degrees, flow regime patterns resemble those of vertical flow more than those of horizontal flow, and the Aziz vertical map should be used. For vertical downward flow, the flow patterns can be more complicated. A generally accepted vertical down-flow map is not available.

Although the designer frequently does not have the choice, avoidance of the slug flow regime in horizontal flow and the slug and froth flow regimes in vertical flow is desirable. Slug flow introduces a flow rate and pressure intermittency that may be troublesome to process control. In some cases, slug flow may be avoided by the choice of smaller pipe sizes. Of course, frictional pressure drop may be increased by use of this smaller pipe. In vertical upflow, oversizing the pipe may result in a bubble flow regime, and a large liquid inventory. This liquid inventory may cause excessive hydrostatic pressure drops.

Example 17-3 — A vapor-liquid mixture is flowing vertically upward in a pipe having an inside diameter of 8.0 inches. The fluid is a hydrocarbon liquid-hydrocarbon vapor mixture. The liquid density is 52 lb/ft³ and the vapor density is 2.0 lb/ft³. The interfacial surface tension is 20 dynes/cm. The liquid volumetric flow rate is 0.17 ft³/sec and the vapor flow rate is

0.5 ft³/sec measured at actual conditions. What flow regime is to be expected?

Solution Steps

$$Y_A = \left[\frac{(52.0)(72.4)}{(62.4)(20)} \right]^{0.25} = 1.32 \quad \text{using Eq 17-40}$$

$$X_A = \left[\frac{(2.0)}{0.0764} \right]^{0.333} (1.32) = 3.91 \quad \text{using Eq 17-39}$$

$$V_{sg} = \frac{0.5}{(\pi/4)(8/12)^2} = 1.43 \text{ ft/sec} \quad \text{using Eq 17-36}$$

$$V_{sl} = \frac{0.17}{(\pi/4)(8/12)^2} = 0.49 \text{ ft/sec} \quad \text{using Eq 17-35}$$

$$N_x = (1.43)(3.91) = 5.60 \text{ ft/sec} \quad \text{using Eq 17-37}$$

$$N_y = (0.49)(1.32) = 0.64 \text{ ft/sec} \quad \text{using Eq 17-38}$$

Fig. 17-16 shows that this flow is in the slug flow regime.

Pressure Drop Calculation

Calculation of pressure drop in two-phase flow lends itself better to computer calculation than to hand calculation. Several two-phase pressure drop correlations are available for both horizontal and vertical flows.^{15, 16, 17} Due to the complexity of two-phase flow, uncertainties associated with pressure drop calculations are much greater than uncertainties in single-phase pressure drop calculations. As a result, errors in calculated two-phase pressure drops in the order of plus or minus twenty percent may normally be anticipated, especially in circumstances where fluid velocities are unusually high or low, where terrain is rugged, or where fluid properties are inadequately known. In addition, different two-phase flow correlations may give significantly different pressure drops. In order to evaluate these differences, several correlations should be used. A method suggested by the American Gas Association¹⁸ can serve as a basis for hand calculation. This method uses a frictional pressure drop calculation generated by Dukler¹⁹ and an elevation pressure drop correlation by Flanigan²⁰.

Frictional Component¹⁸ — Using the Dukler frictional pressure drop calculation method, the frictional pressure drop is given by the equation:

$$\Delta P_f = \frac{f_n f_{tpr} \rho_k V_m^2 L_m}{(0.14623)d} \quad \text{Eq 17-41}$$

where

$$\rho_k = \frac{\rho_L \lambda^2}{H_{Ld}} + \frac{\rho_g (1 - \lambda)^2}{(1 - H_{Ld})} \quad \text{Eq 17-42}$$

and

$$\lambda = \frac{Q_L}{Q_L + Q_g} \quad \text{Eq 17-43}$$

The single phase friction factor, f_n , can be obtained from the correlation¹⁹:

$$f_n = 0.0056 + 0.5 (Re_y)^{-0.32} \quad \text{Eq 17-44}$$

The mixture Reynolds number, Re_y , is calculated according to the equation:

$$Re_y = \frac{(124.0) \rho_k V_m d}{\mu_n} \quad \text{Eq 17-45}$$

Calculation of this Reynolds number requires determination of mixture velocity, V_m , and mixture viscosity, μ_n . These quantities can be determined according to:

$$V_m = V_{sL} + V_{sg} \quad \text{Eq 17-46}$$

$$\mu_n = \mu_L \lambda + \mu_g (1 - \lambda) \quad \text{Eq 17-47}$$

The two-phase friction factor ratio, f_{tpr} , representing a two-phase frictional "efficiency" can be determined by reference to Fig. 17-17 or by the equation:

$$f_{tpr} = 1 + \left[\frac{y}{1.281 - 0.478y + 0.444y^2 - 0.094y^3 + 0.00843y^4} \right] \quad \text{Eq 17-48}$$

where $y = -\ln(\lambda)$.

The remaining quantity to be calculated in the Dukler scheme is an estimate of the liquid holdup, H_{Ld} . This holdup can be estimated using Fig. 17-18. This figure gives liquid holdup as a function of λ and Re_y . Since Re_y is itself a function of liquid holdup, the calculation is, in general, iterative. For most calculations, however, the Re_y line can be used for a first estimate.

Elevation Component — The elevation component of pressure drop can be found using the Flanigan method. In this method, the elevation component is calculated using the equation:

$$\Delta P_e = \frac{\rho_L H_{Lf}}{144} \sum Z_e \quad \text{Eq 17-49}$$

where H_{Lf} is determined from Fig. 17-19 or calculated according to the formula:

$$H_{Lf} = \frac{1}{1 + 0.3264 (V_{sg})^{1.006}} \quad \text{Eq 17-50}$$

The term Z_e is the vertical elevation rise of a hill. The rises are summed. No elevation drops are considered. This is tantamount to ignoring any possible hydrostatic pressure recoveries in downhill sections of pipeline and may lead to a considerable error in the pressure drop analysis.

Once the frictional component or pressure drop is found using the Dukler method, and the elevation component is found using the Flanigan method, the overall two-phase pressure drop is found by summing the friction and elevation components.

$$\Delta P_t = \Delta P_e + \Delta P_f \quad \text{Eq 17-51}$$

Since fluid properties and liquid holdups can change rapidly in a two-phase line, accuracy is improved if this AGA calculation procedure is performed segmentally. The need for segmental calculations is one of the reasons why two-phase calculations are best suited for computer calculations.

Liquid Holdup — The liquid holdup correlation given in Fig. 17-18 is intended only for use in the Dukler friction pressure drop calculation. A correlation by Eaton et al.²¹ is better suited for liquid holdup determination in liquid inventory calculations.

The Eaton²¹ holdup correlation is shown in Fig. 17-20. In this figure, the holdup fraction, H_{Le} , is plotted directly as a function of the dimensionless group, N_E . This dimensionless group is of the form:

$$N_E = \frac{1.84 (N_{Lv})^{0.575} \left(\frac{P_{avg}}{P_b} \right)^{0.05} (N_L)^{0.1}}{N_{gv} (N_d)^{0.0277}} \quad \text{Eq 17-52}$$

where:

$$N_{Lv} = 1.938 V_{sL} \left(\frac{\rho_L}{\sigma} \right)^{0.25} \quad \text{Eq 17-53}$$

$$N_{gv} = 1.938 V_{sg} \left(\frac{\rho_L}{\sigma} \right)^{0.25} \quad \text{Eq 17-54}$$

$$N_d = 10.073 d \left(\frac{\rho_L}{\sigma} \right)^{0.50} \quad \text{Eq 17-55}$$

$$N_L = 0.15726 \mu_L \left(\frac{1}{\rho_L \sigma^3} \right)^{0.25} \quad \text{Eq 17-56}$$

The Eaton correlation has been found reasonably accurate by several investigators, particularly for low holdup flows.

The liquid holdup fraction, H_{Le} , is the fraction of the flow area of the pipe occupied by liquid. To calculate the liquid inventory in the pipe, I_L , the pipe internal volume is multiplied by this holdup fraction.

$$I_L = (28.80) H_{Le} d^2 L_m \quad \text{Eq 17-57}$$

Since holdup fractions may change along the length of the pipe, a segmental calculation is more accurate.

Example 17-4 — A pipeline segment with a 6-inch inside diameter, 0.75 miles long, transports a mixture of gas and oil. The pipeline has a gradual upward slope and rises 100 feet over the 0.75 mile length. The inlet pressure of the pipeline is 400 psia, liquid viscosity is 20 cp, the vapor viscosity is 0.015 cp, and the interfacial surface tension is 15 dynes/cm. The liquid flow rate is 10 ft³/min and the vapor flow rate is 250 actual ft³/min. The density of the liquid phase is 55 lb/ft³, and the density of the gas phase is 1.3 lb/ft³ at operating conditions. What is the pressure at the downstream end of the line segment, and what is the liquid inventory of the line?

Solution Steps

Calculate the flowing liquid volume fraction using Eq 17-43.

$$\lambda = \frac{10}{250 + 10} = 0.038$$

Calculate the mixture viscosity, μ_n , using Eq 17-47.

$$\mu_n = (20) (0.038) + (0.015) (1.0 - 0.038) = 0.774 \text{ cp}$$

For a first guess, assume $H_{Ld} = \lambda$ and estimate ρ_k using Eq 17-42

$$\rho_k = \frac{(55) (0.038)^2}{0.038} + \frac{(1.3) (1.0 - 0.038)^2}{(1.0 - 0.038)} = 3.341 \text{ lb/ft}^3$$

Calculate the superficial velocities and the mixture velocity.

$$V_{sL} = \frac{10.0}{(\pi/4) (6/12)^2 (60)} = 0.849 \text{ ft/sec using Eq 17-35}$$

$$V_{sg} = \frac{250}{(\pi/4) (6/12)^2 (60)} = 21.22 \text{ ft/sec using Eq 17-36}$$

$$V_m = 0.849 + 21.22 = 22.07 \text{ ft/sec using Eq 17-46}$$

FIG. 17-17
Dukler Two-Phase Friction Factor Ratio¹⁹

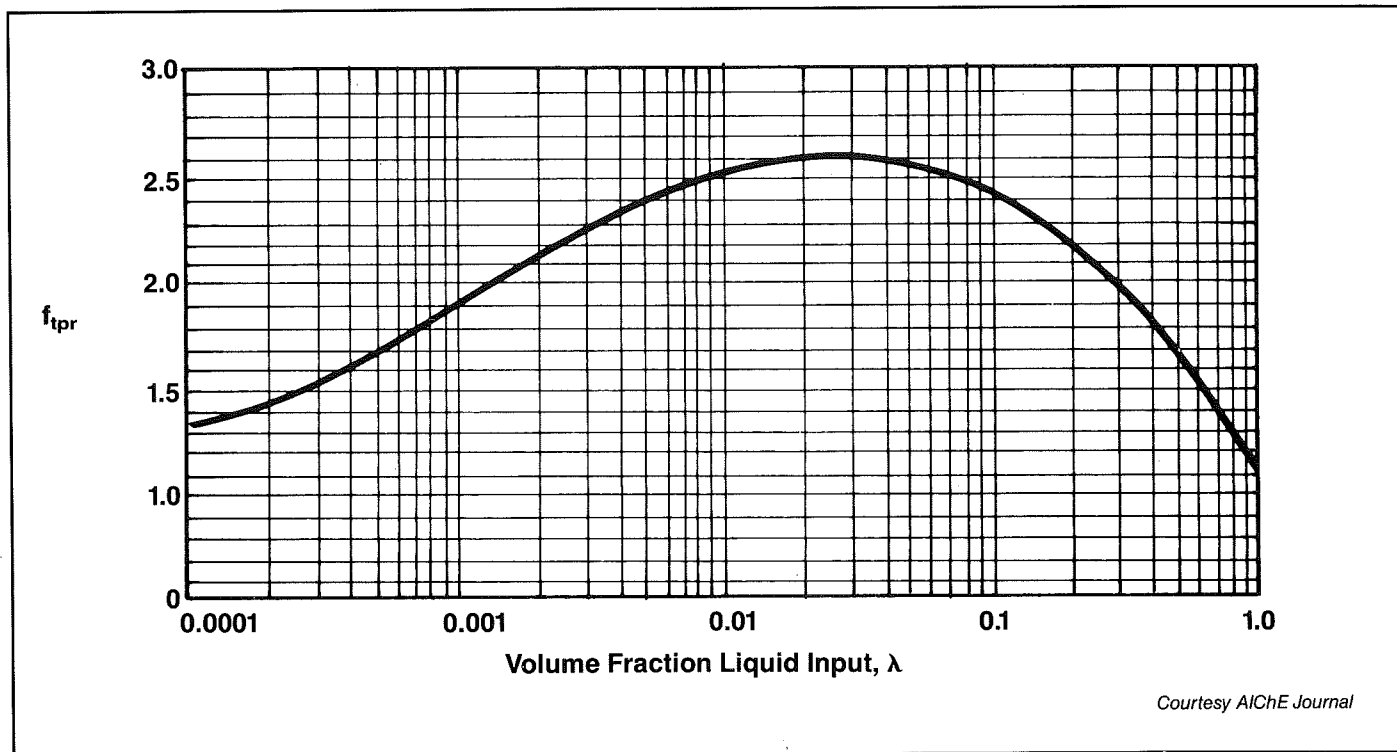


FIG. 17-18
Dukler Liquid Holdup Correlation¹⁹

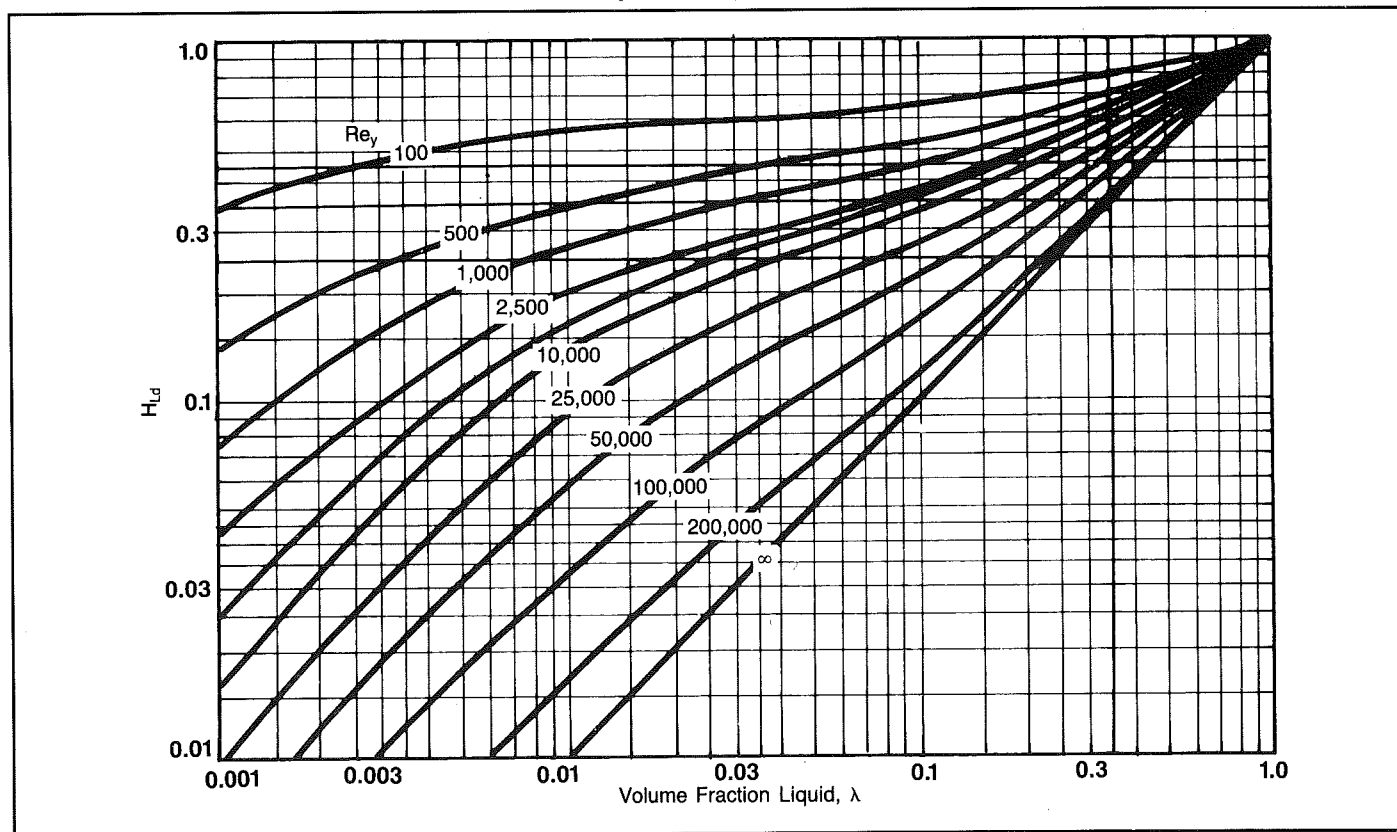
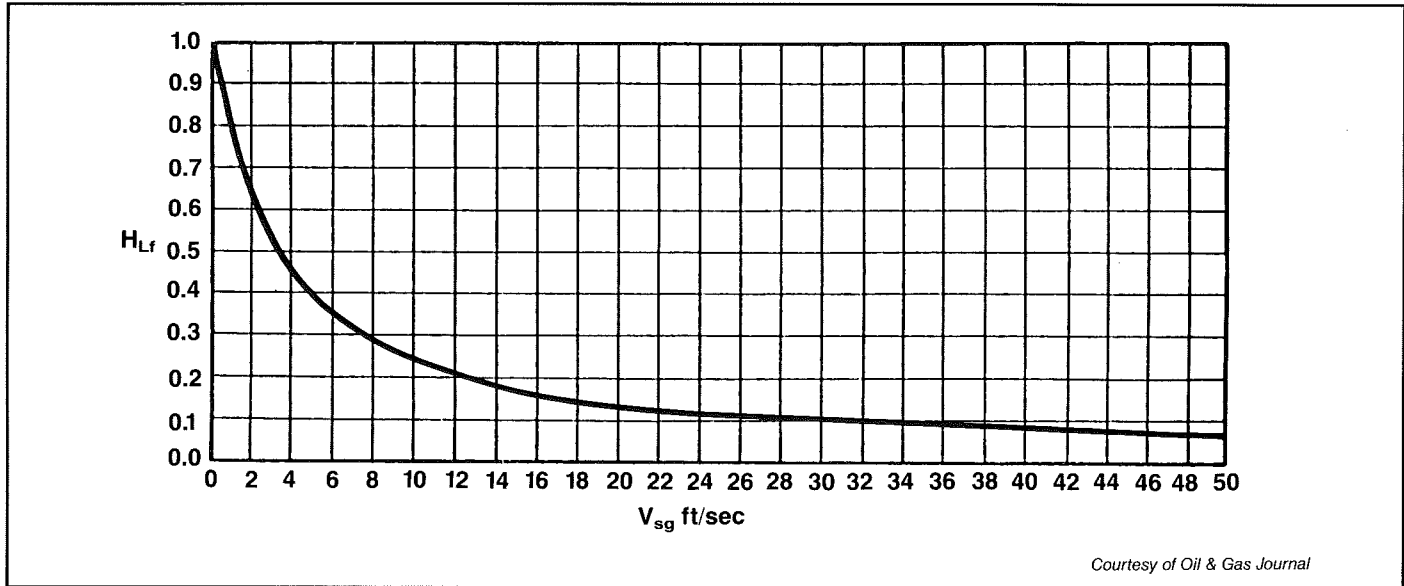


FIG. 17-19
Flanigan Liquid Holdup Correlation²⁰



Calculate an estimate of the mixture Reynolds number, Re_y , using Eq 17-45

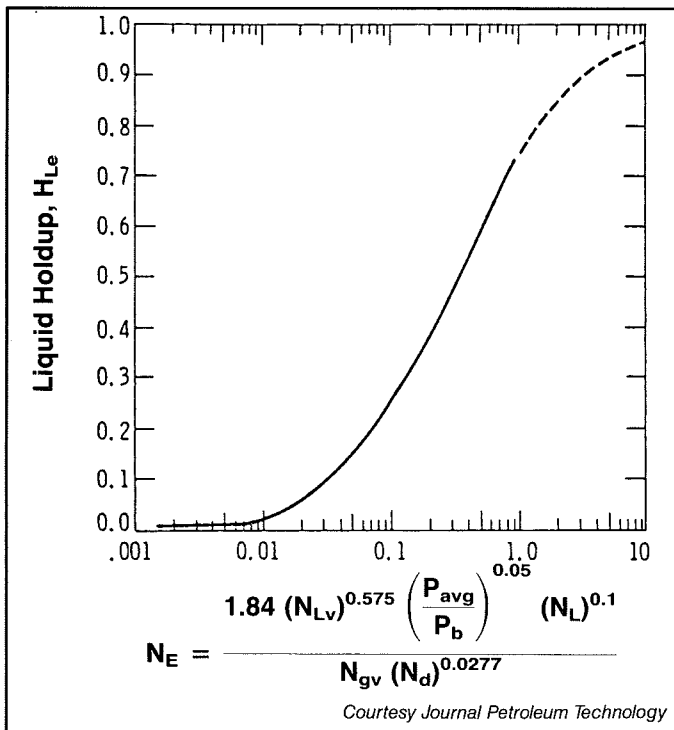
$$Re_y = \frac{(124.0)(3.341)(22.07)(6.0)}{(0.774)} = 70,878$$

From Fig. 17-18, determine a better estimate for the holdup fraction H_{Ld} using $\lambda = 0.038$, $Re_y = 70,878$

$$H_{Ld} = 0.12$$

FIG. 17-20

Eaton Liquid Holdup Correlation²¹



Using this improved H_{Ld} , recalculate ρ_k using Eq 17-42

$$\rho_k = \frac{(55)(0.038)^2}{(0.12)} + \frac{(1.3)(1.0 - 0.038)^2}{(1.0 - 0.12)} = 2.029 \text{ lb/ft}^3$$

Using this improved ρ_k , recalculate Re_y using Eq 17-45

$$Re_y = \frac{(124.0)(2.029)(22.07)(6.0)}{(0.774)} = 43,044$$

From Fig. 17-18 with $\lambda = 0.038$ and $Re_y = 43,044$, $H_{Ld} = 0.16$. Another iteration using $H_{Ld} = 0.16$ indicates $Re_y = 40,923$ and $H_{Ld} = 0.16$.

Calculate the single phase friction factor with $Re_y = 40,923$, using Eq 17-44

$$f_n = 0.0056 + 0.5 (40,923)^{-0.32} = 0.0223$$

Determine the two-phase friction factor, f_{tpr} , from Fig. 17-17 using $\lambda = 0.038$.

$$f_{tpr} = 2.59$$

Now for $\lambda = 0.038$, $H_{Ld} = 0.16$ then $\rho_k = 1.929$ using Eq 17-42.

Calculate the frictional component of pressure drop ΔP_f using Eq 17-41

$$\Delta P_f = \frac{(0.0223)(2.59)(1.929)(22.07)^2(0.75)}{(0.14623)(6.0)} = 46.39 \text{ psi}$$

Find H_{Lf} from Fig. 17-19 using $V_{sg} = 21.22 \text{ ft/sec}$

$$H_{Lf} = 0.13$$

Determine the elevation component of pressure drop, ΔP_e , using Eq 17-49

$$\Delta P_e = \frac{(55)(0.13)(100)}{(144)} = 4.97 \text{ psi}$$

Find the total pressure drop, ΔP_t , using Eq 17-51

$$\Delta P_t = 46.39 + 4.97 = 51.36 \text{ psi}$$

Find the segment discharge pressure

$$P_2 = 400 - 51.36 = 348.64 \text{ psia}$$

The pipeline segment has a discharge pressure of 348.64 psia.

To calculate the liquid inventory, the liquid holdup fraction from Eaton's correlation must be found. First determine the nondimensional parameters:

$$N_{Lv} = (1.938) (0.849) (55/15)^{0.25} = 2.277 \text{ using Eq 17-53}$$

$$N_{gv} = (1.938) (21.22) (55/15)^{0.25} = 56.91 \text{ using Eq 17-54}$$

$$N_d = (10.073) (6.0) (55/15)^{0.5} = 115.73 \text{ using Eq 17-55}$$

$$N_L = (0.15726) (20.0) \left[\frac{1}{(55) (15)^3} \right]^{0.25} = 0.152$$

using Eq 17-56

Determine Eaton's nondimensional abscissa, N_E , using Eq 17-52

$$N_E = \frac{(1.84) (2.277)^{0.575} (400/14.73)^{0.05} (0.152)^{0.1}}{(56.91) (115.73)^{0.0277}} = 0.0444$$

From Fig. 17-20, read the holdup fraction, H_{Le}

$$H_{Le} = 0.14$$

Note that this estimate is close to the H_{Lf} predicted in Fig. 17-19 for elevation pressure drop determination. It also coincides closely with the value of 0.16 from Fig. 17-18.

Calculate the pipeline segment liquid inventory from Eq 17-57

$$I_L = (28.80) (0.14) (6.0)^2 (0.75) = 108.86 \text{ ft}^3$$

The pipeline segment contains 108.86 cubic feet of liquid at any instant.

Liquid Slugging

Purpose of Separators — The slug flow regime is frequently encountered for pipe sizes and flow rates used in process and transmission piping. Liquid slugging introduces an additional design and operational difficulty as liquid and vapor must generally be separated at the downstream end of the two-phase flowline. The downstream separator serves both as a liquid-vapor disengaging device and as a surge vessel to absorb the fluctuating liquid flow rates caused by slugging. In order to size the separator or slug catcher, the length of the incoming slugs must be determined. Slug length calculation methods are not well developed, and there is large uncertainty in slug length determination.

Mechanisms of Slug Generation — Liquid slug lengths are difficult to determine in part because there are at least four identifiable mechanisms for liquid slug generation. Slugs can form as the result of wave formation at the liquid-gas interface in a stratified flow. When the liquid waves grow large enough to bridge the entire pipe diameter, the stratified flow pattern breaks down and a slug flow is formed.

Slugs can also form due to terrain effects. Liquid collects at a sag in the pipeline and blocks the gas flow. The pressure in this blocked gas rises until it blows the accumulated liquid in the sag out as a slug. Changes in pipeline inlet flow rate can also cause slugs. When the inlet flow rate increases, the liquid

inventory in the pipeline decreases, and the excess liquid forms a slug or series of slugs. Finally, pigging can cause very large liquid slugs as the entire liquid inventory of the line is swept ahead of the pig. Of the four mechanisms described, wave growth normally produces the shortest slugs, followed in length by terrain generated slugs. Methods for calculating wave induced slugs were described by Greskovich and Shrier²², and by Brill et al.²³ A preliminary scheme for calculating terrain generated slugs was reported by Schmidt.²⁴ Analytical methods for determining inlet flow rate generated slugs were given by Cunliffe²⁵, and a method of analyzing pigging dynamics was given by McDonald and Baker.²⁶

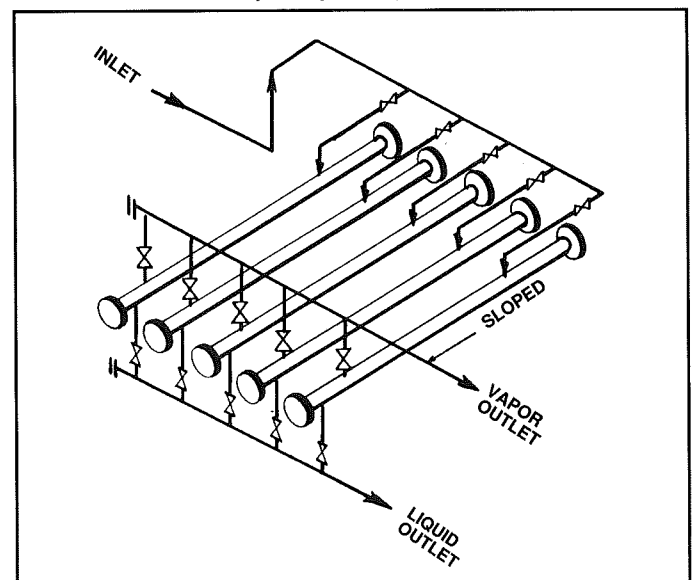
Slug Catchers — Slug catchers are devices at the downstream end or other intermediate points of a pipeline to absorb the fluctuating liquid inlet flow rates through liquid level fluctuation. Slug catchers may be either a vessel or constructed of pipe. All size specifications discussed in Section 7 to provide residence time for vapor-liquid disengagement also apply to vessels used as slug catchers. In addition, sufficient volume must be provided for liquid level fluctuation. Particularly for high pressure service, vessel separators may require very thick walls.

In order to avoid thick wall vessels, slug catchers are frequently made of pipe. Lengths of line pipe tens or hundreds of feet long are used as long, slender horizontal separators. The pipe is generally inclined from one to ten degrees and banks of these slightly inclined pipes are frequently manifolded together. Pipe type slug catchers are frequently less expensive than vessel type slug catchers of the same capacity due to thinner wall requirements of smaller diameter pipe. The manifold nature of multiple pipe slug catchers also makes possible the later addition of additional capacity by laying more parallel pipes. A schematic of a multiple pipe (harp) slug catcher appears in Fig. 17-21. Different pipe inclinations and different manifolding arrangements are favored by different designers.

An example of a line drip catcher is shown in Fig. 17-22. A drip vessel is connected to the incoming pipeline and often laid beneath it. A blow line from the drip vessel is used to blow the liquids out to a storage or surge vessel as they accumulate.

FIG. 17-21

Multiple Pipe Slug Catcher



Pigging — Pipelines are pigged for several reasons. If water is present in the line, it must be removed periodically in order to minimize corrosion. This water accumulates in sags in the pipeline, and these low spots are particularly susceptible to corrosion. Pipelines are also pigged to improve pressure drop-flow rate performance. Water or hydrocarbon liquids that settle in sags in the pipeline constitute partial blockages that

FIG. 17-22

Example Line Drip

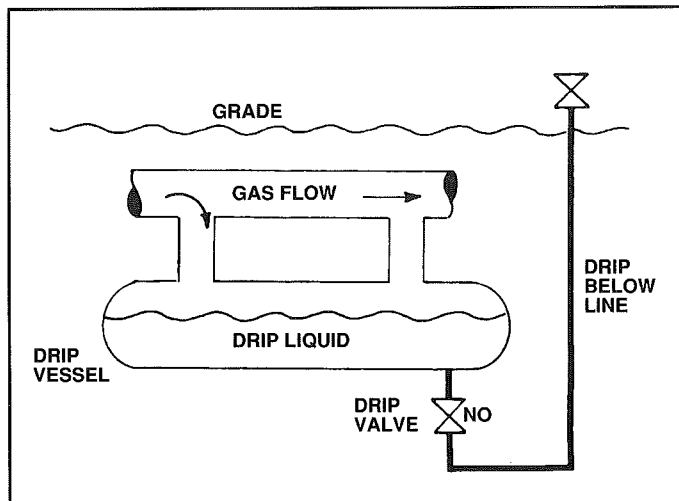


FIG. 17-23

Working Pressures Refinery Piping

To determine allowable internal working pressure for straight sections of pipe in accordance with ANSI B31.3 - 1984, "Code for Pressure Piping, Petroleum Refinery Piping", use the following:

$$t_m = t + c$$

$$t = \frac{P_i d_o}{2(S'E' + P_i Y)} \quad \text{or} \quad P_i = \frac{2 t S'E'}{d_o - 2 t Y}$$

where:

t_m = minimum required thickness, satisfying requirements for pressure, and mechanical, corrosion, and erosion allowances, inches. (The minimum thickness for the pipe selected, considering manufacturer's minus tolerance, shall not be less than t_m).

t = pressure design thickness, in.

c = the sum of the mechanical allowances (thread depth and groove depth), corrosion, and erosion allowances, in.

P_i = internal design pressure, psig.

d_o = outside diameter of pipe, in.

S' = allowable stresses, psi.

E' = longitudinal weld joint factor:

Seamless = 1.000, ERW = 0.85

Y = coefficient having values for ferritic steels as follows:

| |
|-------------------------------|
| 0.4 up to and including 900°F |
| 0.5 for 950°F |
| 0.7 for 1,000°F and above |

increase pressure drop. Pigging can remove these liquids and improve pipeline efficiency. Pigging can also be used as a means of limiting the required slug catcher size. By pigging at frequent intervals, liquid inventory buildup in a pipeline can be reduced, and the maximum slug size can be limited. The required downstream slug catcher size must take into account pigging frequency.

Operational hazards are associated with pigging. The very large slugs swept ahead of the pig may overwhelm inadequately sized downstream facilities. Pigs may also occasionally be destroyed in the pipeline and the resulting debris may damage downstream fittings or equipment. Even worse, the pig may become stuck in the line and require an expensive shutdown for location and removal.

FIG. 17-24

Working Pressures Transmission Lines

To determine allowable internal working pressures for piping outside of refineries and other processing facilities in accordance with ANSI B31.8-1982, "Code of Pressure Piping, Gas Transmission and Distribution Piping," use the following:

$$P_i = \frac{2 S'' t}{d_o} (F'') (E'') (T'')$$

Where

P_i = Design pressure, psig

S'' = Specified minimum yield strength, psi

d_o = Nominal outside diameter, in.

t = Nominal wall thickness, in.

F'' = Construction type design factor, Table 841.1A (see note)

| Type | F'' |
|------|-------|
| A | .72 |
| B | .60 |
| C | .50 |
| D | .40 |

Complete details are covered in Par. 841.15

E'' = Longitudinal joint factor, Table 841.1B

Normally a factor of 1.0 is used for seamless and welded pipe except for the following:

| | |
|--------------------------------------|------|
| Fusion Welded A 134 and A 139 | 0.80 |
| Spiral Welded A 211 | 0.80 |
| Furnace Butt Welded ASTM-A53, API-5L | 0.60 |

T'' = Temperature derating factor, Table 841.1C

| Temp, °F | Factor T'' |
|-------------|--------------|
| 250 or less | 1.000 |
| 300 | 0.967 |
| 350 | 0.933 |
| 400 | 0.900 |
| 450 | 0.867 |

For intermediate temperatures, interpolate for derating factor.

Note: Factor reflecting location of line, proximity to roads, public or private land, etc.

PIPE AND FLANGE DATA

The Petroleum Refinery Piping Code (ANSI B31.3) is used to determine the allowable pressure limits for piping inside refineries and other processing facilities. Refer to Fig. 17-23 for the calculation method.

The table of allowable stresses, Fig. 17-25, is extracted from ANSI B31.3. The designer is strongly urged to consult the latest ANSI B31.3 publication for full description of the code. A tabular compilation of maximum allowable working pressures calculated according to ANSI B31.3 appears in Fig. 17-26.

For piping outside of refineries and other processing facilities, a separate code applies for determining allowable pressure limits, ANSI B31.8. Refer to Fig. 17-24 for the calculation method.

The designer is encouraged to refer to the latest ANSI B31.8 standard for comprehensive code description. A tabular compilation of maximum allowable working pressures computed according to ANSI B31.8 appears in Fig. 17-27.

Fig. 17-28 provides pressure ratings for steel flanges and flanged fittings.

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FIG. 17-25

Allowable Stresses in Tension for Materials (1)

(Excerpted from ANSI B31.3a-1985,
Appendix A, Tables A-1, A-1B)

| Material | Specification | Grade | Class | Factor (E) | Tensile Strength min. psi | Yield Strength min. psi | Notes | BASIC ALLOWABLE STRESSES IN TENSION, psi (1) | | | | | | | | | | | | | | | | | |
|--|---------------|--------|--------|---------------|---------------------------------|-------------------------------|----------------|--|-------------------------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|-------|------|--|
| | | | | | | | | Metal Temperature, °F (7) | | | | | | | | | | | | | | | | | |
| | | | | | | | | (6) Min. Temp. | Min. Temp. to 100 | 200 | 300 | 400 | 500 | 600 | 650 | 700 | 750 | 800 | 850 | 900 | 950 | 1000 | 1050 | 1100 | |
| CARBON STEEL | | | | | | | | | | | | | | | | | | | | | | | | | |
| Seamless Pipe | | | | | | | | | | | | | | | | | | | | | | | | | |
| | ASTM A53 | A | Type S | | 48000 | 30000 | 57, 59 | -20 | 16000 | 16000 | 16000 | 16000 | 16000 | 14800 | 14500 | 14400 | 10700 | 9300 | 7900 | 6500 | 4500 | 2500 | 1600 | 1000 | |
| | ASTM A53 | B | Type S | | 60000 | 35000 | 57, 59 | -20 | 20000 | 20000 | 20000 | 20000 | 18900 | 17300 | 17000 | 16500 | 13000 | 10800 | 8700 | 6500 | 4500 | 2500 | 1600 | 1000 | |
| | ASTM A106 | A | | | 48000 | 30000 | 57 | -20 | 16000 | 16000 | 16000 | 16000 | 16000 | 14800 | 14500 | 14400 | 10700 | 9300 | 7900 | 6500 | 4500 | 2500 | 1600 | 1000 | |
| | ASTM A106 | B | | | 60000 | 35000 | 57 | -20 | 20000 | 20000 | 20000 | 20000 | 18900 | 17300 | 17000 | 16500 | 13000 | 10800 | 8700 | 6500 | 4500 | 2500 | 1600 | 1000 | |
| | ASTM A106 | C | | | 70000 | 40000 | 57 | -20 | 23300 | 23300 | 23300 | 22900 | 21600 | 19700 | 19400 | 19200 | 14800 | 12000 | | | | | | | |
| | ASTM A120 | | | | | | 8 | -20 | 112000 | 11400 | | | | | | | | | | | | | | | |
| | ASTM A333 | 1 | | | 55000 | 30000 | 57, 59 | -50 | 18300 | 18300 | 17700 | 17200 | 16200 | 14800 | 14500 | 14400 | 12000 | 10200 | 8300 | 6500 | 4500 | 2500 | 1600 | 1000 | |
| | ASTM A334 | 1 | | | 55000 | 30000 | 57, 59 | -50 | 18300 | 18300 | 17700 | 17200 | 16200 | 14800 | 14500 | 14400 | 12000 | 10200 | 8300 | 6500 | 4500 | 2500 | 1600 | 1000 | |
| | ASTM A333 | 6 | | | 60000 | 35000 | 57 | -50 | 20000 | 20000 | 20000 | 20000 | 18900 | 17300 | 17000 | 16500 | 13000 | 10800 | 8700 | 6500 | 4500 | 2500 | 1600 | 1000 | |
| | ASTM A334 | 6 | | | 60000 | 35000 | 57 | -50 | 20000 | 20000 | 20000 | 20000 | 18900 | 17300 | 17000 | 16500 | 13000 | 10800 | 8700 | 6500 | 4500 | 2500 | 1600 | 1000 | |
| | API 5L | A | | | 48000 | 30000 | 57, 59 | -20 | 16000 | 16000 | 16000 | 16000 | 16000 | 14800 | 14500 | 14400 | 10700 | 9300 | 7900 | 6500 | 4500 | 2500 | 1600 | 1000 | |
| | API 5L | B | | | 60000 | 35000 | 57, 59 | -20 | 20000 | 20000 | 20000 | 20000 | 18900 | 17300 | 17000 | 16500 | 13000 | 10800 | 8700 | 6500 | 4500 | 2500 | 1600 | 1000 | |
| | API 5LX | X42 | | | 60000 | 42000 | 51, 55 | -20 | 20000 | 20000 | 20000 | 20000 | 18900 | 17300 | 17000 | 16500 | 13000 | 10800 | 8700 | 6500 | 4500 | 2500 | 1600 | 1000 | |
| | API 5LX | X46 | | | 63000 | 46000 | 51, 55 | -20 | 21000 | 21000 | 21000 | 21000 | 18900 | 17300 | 17000 | 16500 | 13000 | 10800 | 8700 | 6500 | 4500 | 2500 | 1600 | 1000 | |
| | API 5LX | X52 | | | 66000 | 52000 | 51, 55 | -20 | 22000 | 22000 | 22000 | 22000 | 18900 | 17300 | 17000 | 16500 | 13000 | 10800 | 8700 | 6500 | 4500 | 2500 | 1600 | 1000 | |
| | API 5LX | X52 | | | 72000 | 52000 | 51, 55 | -20 | 24000 | 24000 | 24000 | 24000 | 18900 | 17300 | 17000 | 16500 | 13000 | 10800 | 8700 | 6500 | 4500 | 2500 | 1600 | 1000 | |
| Electric Resistance Welded Pipe | | | | | | | | | | | | | | | | | | | | | | | | | |
| | ASTM A53 | A | Type E | 0.85 | 48000 | 30000 | 57, 59 | -20 | 13600 | 13600 | 13600 | 13600 | 13600 | 12600 | 12300 | 12250 | 9100 | 7900 | 6700 | 5500 | 3800 | 2150 | 1350 | 850 | |
| | ASTM A53 | B | Type E | 0.85 | 60000 | 35000 | 57, 59 | -20 | 17000 | 17000 | 17000 | 17000 | 16100 | 14700 | 14500 | 14000 | 11000 | 9200 | 7350 | 5500 | 3800 | 2150 | 1350 | 850 | |
| | ASTM A120 | | | 0.85 | | | 8 | -20 | 10200 | 9800 | | | | | | | | | | | | | | | |
| LOW AND INTERMEDIATE ALLOY STEEL & STAINLESS STEEL (4.40) - Seamless Pipe | | | | | | | | | | | | | | | | | | | | | | | | | |
| 3 1/2 Ni | ASTM A333 | 3 | | | 65000 | 35000 | | -150 | 21700 | 19600 | 19600 | 18700 | 17800 | 16800 | 16300 | 15500 | 13900 | 11400 | 9000 | 6500 | 4500 | 2500 | 1600 | 1000 | |
| 3 1/2 Ni | ASTM A334 | 3 | | | 65000 | 35000 | | -150 | 21700 | 19600 | 19600 | 18700 | 17800 | 16800 | 16300 | 15500 | 13900 | 11400 | 9000 | 6500 | 4500 | 2500 | 1600 | 1000 | |
| Ni-Cr-Cu-Al | ASTM A333 | 4 | | | 60000 | 35000 | | -150 | 20000 | 19100 | 18200 | 17300 | 16400 | 15500 | 15000 | | | | | | | | | | |
| 2 1/4 Ni | ASTM A333 | 7 | | | 65000 | 35000 | | -100 | 21700 | 19600 | 19600 | 18700 | 17800 | 16800 | 16300 | 15500 | 13900 | 11400 | 9000 | 6500 | 4500 | 2500 | 1600 | 1000 | |
| 2 1/4 Ni | ASTM A334 | 7 | | | 65000 | 35000 | | -100 | 21700 | 19600 | 19600 | 18700 | 17800 | 16800 | 16300 | 15500 | 13900 | 11400 | 9000 | 6500 | 4500 | 2500 | 1600 | 1000 | |
| 9 Ni | ASTM A333 | 8 | | | 100000 | 75000 | 47 | -320 | 31700 | 31700 | | | | | | | | | | | | | | | |
| 9 Ni | ASTM A334 | 8 | | | 100000 | 75000 | | -320 | 31700 | 31700 | | | | | | | | | | | | | | | |
| 18Cr-8Ni Pipe | ASTM A376 | TP304 | | | 75000 | 30000 | 20,26,28,31,36 | -425 | 20000 | 20000 | 20000 | 18700 | 17500 | 16400 | 16200 | 16000 | 15600 | 15200 | 14900 | 14600 | 14400 | 13800 | 12200 | 9700 | |
| 18Cr-8Ni Pipe | ASTM A376 | TP304H | | | 75000 | 30000 | 26, 31, 36 | -325 | 20000 | 20000 | 20000 | 18700 | 17500 | 16400 | 16200 | 16000 | 15600 | 15200 | 14900 | 14600 | 14400 | 13800 | 12200 | 9700 | |
| NOTE: These notes are requirements of the Code. Those marked with an asterisk (*) restate requirements found in the text of the Code. The other Notes are limitations or special requirements applicable to particular materials. Full interpretation of these notes will require reference to ANSI/ASME B31.3a — 1984 Edition. | | | | | | | | | | | | | | | | | | | | | | | | | |

NOTE: These notes are requirements of the Code. Those marked with an asterisk (*) restate requirements found in the text of the Code. The other Notes are limitations or special requirements applicable to particular materials. Full interpretation of these notes will require reference to ANSI/ASME B31.3a — 1984 Edition.

- (1)* The stress values in Table A-1 and the design stress values in Table A-2 are basic allowable stresses in tension in accordance with 302.3.1(a). For pressure design, the stress values from Table A-1 are multiplied by the appropriate quality factor E (E_c from Table A-1A, or E_j from Table A-1B). Stress values in shear and bearing are stated in 302.3.1(b); those in compression in 302.3.1(c).
- (3)* This casting quality factor can be enhanced by supplementary examination in accordance with 302.3.3(c) and Table 302.3.3C. The higher factor from Table 302.3.3C may be substituted for this factor in pressure design equations.
- (4)* In shaded areas, stress values printed in italics exceed two-thirds of the expected yield strength at temperature. All other stress values in shaded areas are equal to 90% of expected yield strength at temperature. See 302.3.2(d)(4) and 302.3.2(d) [Note (3)].
- (6)* The minimum temperature shown is that design minimum temperature for which the material is normally suitable without impact testing other than that required by the material specification. However, the use of a material at a design minimum temperature below -20°F [-29°C] is established by rules elsewhere in this Code, including any necessary impact test requirements.
- (7)* A single bar (|) in these Stress Tables indicates there are conditions other than stress which affect usage above or below the temperature, as described in other referenced notes. A double bar (||) after a tabled

- stress indicates that use of the material is prohibited above that temperature. A double bar (||) before the stress value for "Min. Temp. to 100°F" indicates that the use of the material is prohibited below the listed minimum temperature. At temperatures where there are no stress values, the material may be used in accordance with 323.2 unless prohibited by a double bar (||).
- (8)* There are restrictions on the use of this material in the text of the Code.
- (20) For pipe sizes NPS 8 and larger and for wall thicknesses of Schedule 140 or heavier, the minimum specified tensile strength is 70.0 ksi [483 MPa].
- (26) These unstabilized grades of stainless steel have increasing tendency to intergranular carbide precipitation as the carbon content increases above 0.03%.
- (28) For temperatures above 1000°F [538°C], these stress values apply only when the carbon content is 0.04% or higher.
- (30) For temperatures above 1000°F [538°C], these stress values may be used only if the material has been heat treated at a temperature of 2000°F [1090°C] minimum.
- (31) For temperatures above 1000°F [538°C], these stress values may be used only if the material has been heat treated by heating to a minimum temperature of 1900°F [1040°C] and quenching in water or rapidly cooling by other means.

- (36) The specification permits this material to be furnished without solution heat treatment or with other than a solution heat treatment. When the material has not been solution heat treated the minimum temperature shall be -20°F [-29°C] unless the material is impact tested per 323.3.
- (40) The stress values for austenitic stainless steels in these Tables may not be applicable if the material has been given a final heat treatment other than that required by the material specification and any overriding requirements of this Code called for by Notes (30) or (31).
- (47) If no welding is employed in fabrication of piping from these materials, the stress values may be increased to 33.3 ksi [230 MPa].
- (51) Special P-Numbers SP-1, SP-2, and SP-3 of carbon steels are not included in P-No. 1 because of possible high carbon, high manganese combination which would require special consideration in qualification. Qualification of any high carbon, high manganese grade may be extended to other grades in its group.
- (55) Pipe produced to this specification is not intended for high temperature service. The stress values apply to either non-expanded or cold expanded material in the as-rolled, normalized, or normalized and tempered condition.
- (57)* Conversion of carbides to graphite may occur after prolonged exposure to temperatures over 800°F [425°C] (see Appendix F).
- (59)* For temperatures above 900°F [480°C], consider the advantages of killed steel [see Appendix F].

FIG. 17-26

Design Properties and Allowable Working Pressures for Piping
ASTM A106, grade B seamless pipe—Petroleum Refinery Piping Code
for Pressure Piping ANSI B31.3-1984—Corrosion allowance = 0.05

| Nom pipe size in. | Sch. No. | Weight of pipe lb./ft. | O.D. in. | Wall thk. in. | I D (d) in. | d ⁵ | Flow area sq ft | Allowable working pressures for temperatures (in °F) not to exceed. | | | | | | |
|----------------------------|-------------|------------------------------|-------------|---------------------|-------------------|----------------|-----------------------|---|------|------|------|------|------|------|
| | | | | | | | | -20 to 100 | 200 | 300 | 400 | 500 | 600 | 700 |
| 1/2 | S40 | .851 | .840 | .109 | .622 | .0931 | .00211 | 2258 | 2258 | 2258 | 2258 | 2134 | 1953 | 1863 |
| 3/4 | S40 | 1.131 | 1.050 | .113 | .824 | .3799 | .00371 | 1933 | 1933 | 1933 | 1933 | 1827 | 1672 | 1595 |
| | X80 | 1.474 | | .154 | .742 | .2249 | .00300 | 3451 | 3451 | 3451 | 3451 | 3261 | 2985 | 2847 |
| 1 | S40 | 1.679 | 1.315 | .133 | 1.049 | 1.2700 | .00600 | 2103 | 2103 | 2103 | 2103 | 1988 | 1819 | 1735 |
| | X80 | 2.172 | | .179 | .957 | .8027 | .00499 | 3468 | 3468 | 3468 | 3468 | 3277 | 3000 | 2861 |
| | 160 | 2.844 | | .250 | .815 | .3596 | .00362 | 5720 | 5720 | 5720 | 5720 | 5405 | 4948 | 4719 |
| | XX | 3.659 | | .358 | .599 | .0771 | .00196 | 9534 | 9534 | 9534 | 9534 | 9010 | 8247 | 7866 |
| 1-1/2 | S40 | 2.718 | 1.900 | .145 | 1.610 | 10.820 | .01414 | 1672 | 1672 | 1672 | 1672 | 1580 | 1446 | 1379 |
| | X80 | 3.632 | | .200 | 1.500 | 7.594 | .01225 | 2777 | 2777 | 2777 | 2777 | 2625 | 2402 | 2291 |
| | 160 | 4.866 | | .281 | 1.338 | 4.288 | .00976 | 4494 | 4494 | 4494 | 4494 | 4247 | 3887 | 3707 |
| | XX | 6.409 | | .400 | 1.100 | 1.611 | .00660 | 7228 | 7228 | 7228 | 7228 | 6831 | 6253 | 5963 |
| 2 | S40 | 3.653 | 2.375 | .154 | 2.067 | 37.72 | .02330 | 1469 | 1469 | 1469 | 1469 | 1388 | 1270 | 1212 |
| | X80 | 5.022 | | .218 | 1.939 | 27.41 | .02050 | 2488 | 2488 | 2488 | 2488 | 2351 | 2152 | 2053 |
| | 160 | 7.445 | | .343 | 1.687 | 13.74 | .01556 | 4600 | 4600 | 4600 | 4600 | 4347 | 3979 | 3795 |
| | XX | 9.030 | | .436 | 1.503 | 7.67 | .01232 | 6284 | 6284 | 6284 | 6284 | 5939 | 5436 | 5185 |
| 3 | S40 | 7.58 | 3.500 | .216 | 3.068 | 271.80 | .05130 | 1640 | 1640 | 1640 | 1640 | 1550 | 1419 | 1353 |
| | X80 | 10.25 | | .300 | 2.900 | 205.10 | .04587 | 2552 | 2552 | 2552 | 2552 | 2412 | 2207 | 2105 |
| | 160 | 14.33 | | .438 | 2.624 | 124.40 | .03755 | 4122 | 4122 | 4122 | 4122 | 3895 | 3566 | 3401 |
| | XX | 18.58 | | .600 | 2.300 | 64.36 | .02885 | 6089 | 6089 | 6089 | 6089 | 5754 | 5267 | 5024 |
| 4 | S40 | 10.79 | 4.500 | .237 | 4.026 | 1058.0 | .08840 | 1439 | 1439 | 1439 | 1439 | 1360 | 1244 | 1187 |
| | X80 | 14.99 | | .337 | 3.826 | 819.8 | .07986 | 2275 | 2275 | 2275 | 2275 | 2150 | 1968 | 1877 |
| | 160 | 22.51 | | .531 | 3.438 | 480.3 | .06447 | 3978 | 3978 | 3978 | 3978 | 3760 | 3441 | 3282 |
| | XX | 27.54 | | .674 | 3.152 | 311.1 | .05419 | 5307 | 5307 | 5307 | 5307 | 5015 | 4590 | 4378 |
| 6 | S40 | 18.98 | 6.625 | .280 | 6.065 | 8206. | .2006 | 1205 | 1205 | 1205 | 1205 | 1139 | 1042 | 994 |
| | X80 | 28.58 | | .432 | 5.761 | 6346. | .1810 | 2062 | 2062 | 2062 | 2062 | 1948 | 1783 | 1701 |
| | 160 | 45.30 | | .718 | 5.187 | 3762. | .1469 | 3753 | 3753 | 3753 | 3753 | 3546 | 3246 | 3097 |
| | XX | 53.17 | | .864 | 4.897 | 2816. | .1308 | 4659 | 4659 | 4659 | 4659 | 4403 | 4030 | 3844 |
| 8 | S40 | 28.56 | 8.625 | .322 | 7.981 | 32380. | .3474 | 1098 | 1098 | 1098 | 1098 | 1037 | 950 | 906 |
| | X80 | 43.4 | | .500 | 7.625 | 25775. | .3171 | 1864 | 1864 | 1864 | 1864 | 1761 | 1612 | 1537 |
| | XX | 72.4 | | .875 | 6.875 | 15360. | .2578 | 3554 | 3554 | 3554 | 3554 | 3359 | 3074 | 2932 |
| | 160 | 74.7 | | .906 | 6.813 | 14679. | .2532 | 3699 | 3699 | 3699 | 3699 | 3496 | 3200 | 3052 |
| 10 | S40 | 40.5 | 10.750 | .365 | 10.020 | 101000. | .5475 | 1022 | 1022 | 1022 | 1022 | 966 | 884 | 843 |
| | X60 | 54.7 | | .500 | 9.750 | 88110. | .5185 | 1484 | 1484 | 1484 | 1484 | 1403 | 1284 | 1224 |
| | 160 | 115.7 | | 1.125 | 8.500 | 44371. | .3941 | 3736 | 3736 | 3736 | 3736 | 3531 | 3232 | 3082 |
| 12 | S | 49.6 | 12.750 | .375 | 12.000 | 248800 | .7854 | 888 | 888 | 888 | 888 | 839 | 768 | 732 |
| | X | 65.4 | | .500 | 11.750 | 223970 | .7528 | 1245 | 1245 | 1245 | 1245 | 1177 | 1077 | 1027 |
| | 160 | 160.3 | | 1.312 | 10.126 | 106461 | .5592 | 3699 | 3699 | 3699 | 3699 | 3496 | 3200 | 3052 |
| 14 | 10 | 36.7 | 14.000 | .250 | 13.500 | 448400 | .9940 | 486 | 486 | 486 | 486 | 460 | 421 | 401 |
| | S30 | 54.6 | | .375 | 13.250 | 408394 | .9575 | 807 | 807 | 807 | 807 | 763 | 698 | 666 |
| | X | 72.1 | | .500 | 13.000 | 371290 | .9211 | 1132 | 1132 | 1132 | 1132 | 1069 | 979 | 934 |
| 16 | 10 | 42.1 | 16.000 | .250 | 15.500 | 894660 | 1.310 | 425 | 425 | 425 | 425 | 402 | 368 | 351 |
| | S30 | 62.6 | | .375 | 15.250 | 824801 | 1.268 | 705 | 705 | 705 | 705 | 666 | 609 | 581 |
| | S40 | 82.8 | | .500 | 15.000 | 759375 | 1.227 | 987 | 987 | 987 | 987 | 933 | 854 | 815 |
| 18 | 10 | 47.4 | 18.000 | .250 | 17.500 | 1641309 | 1.670 | 377 | 377 | 377 | 377 | 357 | 326 | 311 |
| | S | 70.6 | | .375 | 17.250 | 1527400 | 1.622 | 625 | 625 | 625 | 625 | 591 | 541 | 516 |
| | X | 93.5 | | .500 | 17.000 | 1419900 | 1.575 | 876 | 876 | 876 | 876 | 828 | 757 | 722 |
| 20 | 10 | 52.7 | 20.000 | .250 | 19.500 | 2819500 | 2.074 | 339 | 339 | 339 | 339 | 321 | 293 | 280 |
| | S20 | 78.6 | | .375 | 19.250 | 2643352 | 2.021 | 562 | 562 | 562 | 562 | 531 | 486 | 464 |
| | X30 | 104.1 | | .500 | 19.000 | 2476099 | 1.969 | 787 | 787 | 787 | 787 | 743 | 680 | 649 |
| 24 | 10 | 63.4 | 24.000 | .250 | 23.500 | 7167030 | 3.012 | 282 | 282 | 282 | 282 | 267 | 244 | 233 |
| | S20 | 94.6 | | .375 | 23.250 | 6793832 | 2.948 | 468 | 467 | 467 | 467 | 442 | 404 | 386 |
| | X | 125.5 | | .500 | 23.000 | 6436300 | 2.883 | 660 | 654 | 654 | 654 | 618 | 565 | 539 |

Note: The above allowable working pressures are calculated from Fig. 17-23 using a reduction in t_m to 87.5% of the wall thickness shown above to recognize mill wall tolerance of 12.5%.

FIG. 17-27

Gas Transmission and Distribution Piping
Code for Pressure Piping ANSI B31.8-1982
Carbon Steel and High Yield Strength Pipe

(Values apply to A106, API 5L and API 5LX pipe having the same specified minimum yield strength as shown)

| Allowable Working Pressures up to 250°F, in psig | | | | | | | | | | | | | | | | | | | | | | | | |
|---|-----------------|--------------|----------------------------------|--------|--------|--------|--------|------------------|--------|--------|--------|--------|------------------|--------|--------|--------|--------|------------------|--------|--------|--------|--------|-----|--|
| Nom Pipe Size | O.D. | Wall Thk. | Construction Type Design Factors | | | | | | | | | | | | | | | | | | | | | |
| | | | Type A, F = 0.72* | | | | | Type B, F = 0.60 | | | | | Type C, F = 0.50 | | | | | Type D, F = 0.40 | | | | | | |
| | | | GR.B | | | | | GR.B | | | | | GR.B | | | | | GR.B | | | | | | |
| | | | 35,000 | 42,000 | 46,000 | 52,000 | 60,000 | 35,000 | 42,000 | 46,000 | 52,000 | 60,000 | 35,000 | 42,000 | 46,000 | 52,000 | 60,000 | 35,000 | 42,000 | 46,000 | 52,000 | 60,000 | | |
| 2 | (STD) 2.375 | .154 | 3268 | | | | | 2723 | | | | | 2270 | | | | | 1816 | | | | | | |
| | | .218 | 4626 | | | | | 3855 | | | | | 3213 | | | | | 2570 | | | | | | |
| 3 | 3.500 (STD) | .125 | 1800 | | | | | 1500 | | | | | 1250 | | | | | 1000 | | | | | | |
| | | .156 | 2246 | | | | | 1872 | | | | | 1560 | | | | | 1248 | | | | | | |
| | | .188 | 2707 | | | | | 2256 | | | | | 1880 | | | | | 1504 | | | | | | |
| | | .216 | 3110 | | | | | 2592 | | | | | 2160 | | | | | 1728 | | | | | | |
| | | .250 | 3600 | | | | | 3000 | | | | | 2500 | | | | | 2000 | | | | | | |
| | | .281 | 4046 | | | | | 3372 | | | | | 2810 | | | | | 2248 | | | | | | |
| | | .300 | 4320 | | | | | 3600 | | | | | 3000 | | | | | 2400 | | | | | | |
| 4 | 4.500 (STD) | .125 | 1400 | 1680 | 1840 | | | 1167 | 1400 | 1533 | | | 973 | 1167 | 1278 | | | 778 | 933 | 1022 | | | | |
| | | .156 | 1747 | 2097 | 2296 | | | 1456 | 1747 | 1913 | | | 1214 | 1456 | 1595 | | | 971 | 1165 | 1276 | | | | |
| | | .188 | 2105 | 2526 | 2767 | | | 1754 | 2105 | 2306 | | | 1462 | 1755 | 1922 | | | 1170 | 1404 | 1537 | | | | |
| | | .219 | 2453 | 2943 | 3223 | | | 2044 | 2453 | 2686 | | | 1704 | 2044 | 2239 | | | 1363 | 1635 | 1791 | | | | |
| | | .237 | 2654 | 3185 | 3488 | | | 2212 | 2654 | 2907 | | | 1844 | 2212 | 2423 | | | 1475 | 1770 | 1938 | | | | |
| | | .250 | 2800 | 3360 | 3680 | | | 2333 | 2800 | 3067 | | | 1945 | 2333 | 2556 | | | 1556 | 1869 | 2044 | | | | |
| | | .281 | 3147 | 3776 | 4136 | | | 2623 | 3147 | 3447 | | | 2186 | 2622 | 2873 | | | 1748 | 2098 | 2298 | | | | |
| | | .312 | 3494 | 4193 | 4593 | | | 2912 | 3494 | 3827 | | | 2427 | 2912 | 3190 | | | 1941 | 2330 | 2552 | | | | |
| | | .337 | 3774 | 4530 | 4961 | | | 3145 | 3775 | 4134 | | | 2621 | 3146 | 3445 | | | 2097 | 2516 | 2756 | | | | |
| | | 6 | 6.625 (STD) | .156 | 1187 | 1424 | 1560 | 1763 | | 989 | 1187 | 1300 | 1469 | | 824 | 989 | 1083 | 1224 | | 659 | 791 | 866 | 980 | |
| .188 | 1429 | | | 1716 | 1880 | 2124 | | 1192 | 1430 | 1567 | 1770 | | 993 | 1192 | 1306 | 1475 | | 794 | 954 | 1044 | 1180 | | | |
| .219 | 1666 | | | 2000 | 2190 | 2475 | | 1388 | 1666 | 1825 | 2063 | | 1157 | 1389 | 1521 | 1719 | | 926 | 1111 | 1216 | 1375 | | | |
| .250 | 1902 | | | 2282 | 2500 | 2826 | | 1585 | 1902 | 2083 | 2355 | | 1321 | 1585 | 1736 | 1963 | | 1057 | 1268 | 1389 | 1570 | | | |
| .280 | 2130 | | | 2556 | 2799 | 3164 | | 1775 | 2130 | 2333 | 2637 | | 1479 | 1775 | 1944 | 2198 | | 1183 | 1420 | 1555 | 1758 | | | |
| .312 | 2373 | | | 2848 | 3120 | 3527 | | 1978 | 2374 | 2600 | 2933 | | 1649 | 1978 | 2167 | 2449 | | 1319 | 1582 | 1733 | 1959 | | | |
| .375 | 2853 | | | 3424 | 3750 | 4237 | | 2377 | 2853 | 3125 | 3531 | | 1981 | 2378 | 2604 | 2943 | | 1585 | 1902 | 2083 | 2354 | | | |
| .432 | 3287 | | | 3943 | 4319 | 4883 | | 2739 | 3286 | 3599 | 4069 | | 2283 | 2738 | 3000 | 3391 | | 1826 | 2191 | 2400 | 2713 | | | |
| 8 | 8.625 (STD) | | | .156 | 912 | 1094 | 1198 | 1354 | | 760 | 912 | 998 | 1128 | | 633 | 760 | 832 | 940 | | 506 | 608 | 666 | 752 | |
| | | | | .188 | 1098 | 1318 | 1444 | 1632 | | 915 | 1098 | 1203 | 1360 | | 763 | 915 | 1003 | 1133 | | 610 | 732 | 802 | 907 | |
| | | .203 | 1186 | 1424 | 1559 | 1762 | | 989 | 1186 | 1299 | 1469 | | 824 | 989 | 1083 | 1224 | | 659 | 791 | 866 | 979 | | | |
| | | .219 | 1280 | 1535 | 1681 | 1901 | | 1067 | 1280 | 1401 | 1584 | | 889 | 1067 | 1168 | 1320 | | 711 | 853 | 934 | 1056 | | | |
| | | .250 | 1461 | 1753 | 1920 | 2170 | | 1217 | 1461 | 1600 | 1809 | | 1014 | 1217 | 1333 | 1507 | | 812 | 974 | 1067 | 1206 | | | |
| | | .277 | 1618 | 1942 | 2128 | 2405 | | 1349 | 1618 | 1773 | 2004 | | 1124 | 1349 | 1478 | 1670 | | 899 | 1079 | 1182 | 1336 | | | |
| | | .312 | 1823 | 2189 | 2396 | 2709 | | 1520 | 1823 | 1997 | 2258 | | 1266 | 1520 | 1664 | 1881 | | 1013 | 1216 | 1331 | 1505 | | | |
| | | .322 | 1882 | 2258 | 2473 | 2796 | | 1568 | 1882 | 2061 | 2329 | | 1307 | 1568 | 1717 | 1941 | | 1045 | 1254 | 1374 | 1553 | | | |
| | | .344 | 2011 | 2412 | 2642 | 2988 | | 1676 | 2011 | 2202 | 2490 | | 1396 | 1676 | 1835 | 2075 | | 1117 | 1340 | 1468 | 1660 | | | |
| | | .375 | 2191 | 2628 | 2880 | 3256 | | 1826 | 2191 | 2399 | 2713 | | 1521 | 1826 | 1999 | 2261 | | 1217 | 1460 | 1599 | 1808 | | | |
| | | .438 | 2560 | 3071 | 3364 | 3803 | | 2133 | 2560 | 2804 | 3170 | | 1778 | 2133 | 2336 | 2641 | | 1422 | 1706 | 1869 | 2113 | | | |
| | | .500 | 2922 | 3506 | 3840 | 4341 | | 2435 | 2922 | 3200 | 3617 | | 2029 | 2435 | 2667 | 3014 | | 1623 | 1948 | 2133 | 2412 | | | |
| 10 | 10.750 (STD) | .188 | 881 | 1058 | 1158 | 1310 | | 733 | 881 | 965 | 1091 | | 612 | 735 | 804 | 909 | | 490 | 588 | 644 | 728 | | | |
| | | .203 | 959 | 1143 | 1251 | 1415 | | 794 | 952 | 1043 | 1179 | | 661 | 794 | 869 | 983 | | 529 | 635 | 695 | 786 | | | |
| | | .219 | 1026 | 1231 | 1348 | 1525 | | 855 | 1026 | 1124 | 1271 | | 713 | 855 | 936 | 1059 | | 570 | 684 | 749 | 847 | | | |
| | | .250 | 1172 | 1407 | 1540 | 1741 | | 977 | 1172 | 1284 | 1451 | | 814 | 977 | 1070 | 1209 | | 651 | 781 | 856 | 967 | | | |
| | | .279 | 1309 | 1570 | 1719 | 1944 | | 1091 | 1309 | 1433 | 1620 | | 909 | 1091 | 1194 | 1350 | | 727 | 872 | 955 | 1080 | | | |
| | | .307 | 1440 | 1728 | 1892 | 2138 | | 1200 | 1440 | 1577 | 1782 | | 1000 | 1200 | 1314 | 1486 | | 800 | 960 | 1051 | 1189 | | | |
| | | .344 | 1613 | 1935 | 2120 | 2396 | | 1344 | 1613 | 1767 | 1997 | | 1120 | 1344 | 1473 | 1664 | | 896 | 1075 | 1178 | 1331 | | | |
| | | .365 | 1711 | 2054 | 2249 | 2542 | | 1426 | 1711 | 1874 | 2119 | | 1188 | 1426 | 1562 | 1766 | | 951 | 1141 | 1249 | 1412 | | | |
| | | .438 | 2054 | 2464 | 2700 | 3051 | | 1712 | 2054 | 2250 | 2543 | | 1426 | 1712 | 1875 | 2119 | | 1141 | 1369 | 1500 | 1695 | | | |
| | | .500 | 2344 | 2813 | 3081 | 3483 | | 1953 | 2344 | 2567 | 2902 | | 1628 | 1953 | 2140 | 2419 | | 1302 | 1563 | 1712 | 1935 | | | |
| 12 | 12.750 (STD) | .188 | 743 | 892 | 977 | 1104 | | 619 | 743 | 814 | 920 | | 516 | 619 | 678 | 767 | | 413 | 495 | 543 | 613 | | | |
| | | .203 | 803 | 963 | 1055 | 1193 | | 669 | 803 | 879 | 995 | | 558 | 669 | 733 | 829 | | 446 | 535 | 586 | 663 | | | |
| | | .219 | 866 | 1039 | 1138 | 1287 | | 722 | 866 | 948 | 1073 | | 601 | 722 | 790 | 894 | | 481 | 577 | 632 | 715 | | | |
| | | .250 | 988 | 1186 | 1299 | 1468 | | 824 | 988 | 1082 | 1224 | | 686 | 824 | 902 | 1020 | | 549 | 659 | 722 | 816 | | | |
| | | .281 | 1111 | 1332 | 1460 | 1651 | | 926 | 1111 | 1217 | 1376 | | 771 | 926 | 1014 | 1146 | | 617 | 740 | 811 | 917 | | | |
| | | .312 | 1233 | 1480 | 1620 | 1832 | | 1028 | 1233 | 1350 | 1527 | | 856 | 1028 | 1125 | 1273 | | 685 | 822 | 900 | 1018 | | | |
| | | .330 | 1305 | 1566 | 1715 | 1939 | | 1088 | 1305 | 1430 | 1616 | | 906 | 1088 | 1191 | 1346 | | 725 | 870 | 953 | 1077 | | | |
| | | .344 | 1359 | 1631 | 1786 | 2020 | | 1133 | 1359 | 1488 | 1683 | | 944 | 1133 | 1240 | 1403 | | 755 | 906 | 992 | 1122 | | | |
| | | .375 | 1482 | 1779 | 1948 | 2202 | | 1235 | 1482 | 1624 | 1835 | | 1029 | 1235 | 1353 | 1529 | | 824 | 988 | 1082 | 1224 | | | |
| | | .406 | 1606 | 1926 | 2110 | 2385 | | 1338 | 1606 | 1758 | 1988 | | 1115 | 1338 | 1465 | 1656 | | 892 | 1070 | 1172 | 1325 | | | |
| | | .438 | 1732 | 2077 | 2275 | 2572 | | 1443 | 1732 | 1896 | 2144 | | 1203 | 1443 | 1580 | 1786 | | 962 | 1154 | 1264 | 1429 | | | |
| | | .500 | 1976 | 2372 | 2598 | 2936 | | 1647 | 1976 | 2165 | 2447 | | 1373 | 1647 | 1804 | 2039 | | 1098 | 1318 | 1443 | 1631 | | | |
| * Type A construction also applicable to "Liquid Petroleum Transportation Piping Code," ANSI B31.4-1979 | | | | | | | | | | | | | | | | | | | | | | | | |

* Type A construction also applicable to "Liquid Petroleum Transportation Piping Code," ANSI B31.4-1979

FIG. 17-27 (Cont'd.)

Gas Transmission and Distribution Piping

| Nom Pipe Size | O.D. | Wall Thk. | Allowable Working Pressures up to 250°F, in psig | | | | | | | | | | | | | | | | | | | |
|---|-----------------|---|--|-------------------|--------|--------|--------|------------------|------------------|--------|--------|--------|------------------|------------------|--------|--------|--------|------------------|------------------|------|------|------|
| | | | Construction Type Design Factors | | | | | | | | | | | | | | | | | | | |
| | | | Type A, F = 0.72* | | | | | Type B, F = 0.60 | | | | | Type C, F = 0.50 | | | | | Type D, F = 0.40 | | | | |
| | | | GR.B | Type A, F = 0.72* | | | | GR.B | Type B, F = 0.60 | | | | GR.B | Type C, F = 0.50 | | | | GR.B | Type D, F = 0.40 | | | |
| 35,000 | 42,000 | 46,000 | 52,000 | 60,000 | 35,000 | 42,000 | 46,000 | 52,000 | 60,000 | 35,000 | 42,000 | 46,000 | 52,000 | 60,000 | 35,000 | 42,000 | 46,000 | 52,000 | 60,000 | | | |
| 16 | 16.000 (STD) | .219 | 689 | 828 | 905 | 1024 | 1183 | 575 | 689 | 755 | 854 | 986 | 479 | 575 | 629 | 711 | 822 | 383 | 460 | 503 | 569 | 657 |
| | | .250 | 786 | 945 | 1035 | 1170 | 1350 | 656 | 788 | 863 | 975 | 1125 | 547 | 656 | 719 | 813 | 938 | 438 | 525 | 575 | 650 | 750 |
| | | .281 | 886 | 1063 | 1164 | 1316 | 1518 | 738 | 886 | 970 | 1097 | 1265 | 615 | 738 | 809 | 914 | 1054 | 492 | 591 | 647 | 731 | 843 |
| | | .312 | 982 | 1179 | 1291 | 1460 | 1685 | 818 | 982 | 1076 | 1217 | 1404 | 682 | 819 | 896 | 1014 | 1170 | 546 | 655 | 717 | 811 | 936 |
| | | .344 | 1084 | 1300 | 1424 | 1609 | 1858 | 903 | 1084 | 1187 | 1341 | 1548 | 753 | 903 | 989 | 1118 | 1290 | 602 | 722 | 791 | 894 | 1032 |
| | | .375 | 1181 | 1418 | 1553 | 1755 | 2025 | 984 | 1181 | 1294 | 1463 | 1688 | 820 | 984 | 1078 | 1219 | 1407 | 656 | 788 | 863 | 975 | 1125 |
| | | .438 | 1379 | 1655 | 1813 | 2049 | 2365 | 1149 | 1379 | 1511 | 1707 | 1971 | 958 | 1149 | 1259 | 1423 | 1643 | 766 | 919 | 1007 | 1138 | 1314 |
| | | .500 | 1575 | 1890 | 2070 | 2340 | 2700 | 1313 | 1575 | 1725 | 1950 | 2250 | 1094 | 1313 | 1438 | 1625 | 1875 | 875 | 1050 | 1150 | 1300 | 1500 |
| 20 | (STD) 20.000 | .625 | 1969 | 2363 | 2588 | 2925 | 3375 | 1641 | 1969 | 2157 | 2438 | 2812 | 1368 | 1641 | 1796 | 2031 | 2344 | 1094 | 1313 | 1438 | 1625 | 1875 |
| | | .250 | 630 | 756 | 828 | 936 | 1080 | 525 | 630 | 690 | 780 | 900 | 438 | 525 | 575 | 650 | 750 | 350 | 420 | 460 | 520 | 600 |
| | | .281 | 709 | 851 | 931 | 1053 | 1214 | 591 | 709 | 776 | 878 | 1012 | 492 | 591 | 647 | 731 | 843 | 394 | 473 | 518 | 585 | 674 |
| | | .312 | 786 | 943 | 1033 | 1168 | 1348 | 656 | 786 | 861 | 973 | 1123 | 547 | 656 | 719 | 813 | 936 | 438 | 525 | 575 | 650 | 749 |
| | | .344 | 867 | 1040 | 1139 | 1289 | 1486 | 723 | 868 | 950 | 1074 | 1238 | 603 | 723 | 791 | 895 | 1032 | 482 | 578 | 633 | 716 | 826 |
| | | .375 | 945 | 1134 | 1242 | 1404 | 1620 | 788 | 945 | 1035 | 1170 | 1350 | 656 | 788 | 863 | 975 | 1125 | 525 | 630 | 690 | 780 | 900 |
| | | .406 | 1023 | 1228 | 1345 | 1520 | 1754 | 853 | 1023 | 1121 | 1267 | 1462 | 710 | 853 | 934 | 1056 | 1218 | 568 | 682 | 747 | 844 | 974 |
| | | .438 | 1103 | 1324 | 1450 | 1639 | 1892 | 919 | 1103 | 1208 | 1366 | 1577 | 766 | 919 | 1007 | 1138 | 1314 | 613 | 735 | 805 | 910 | 1051 |
| 24 | (STD) 24.000 | .500 | 1260 | 1512 | 1656 | 1872 | 2160 | 1050 | 1260 | 1380 | 1560 | 1800 | 875 | 1050 | 1150 | 1300 | 1500 | 700 | 840 | 920 | 1040 | 1200 |
| | | .625 | 1575 | 1890 | 2070 | 2340 | 2700 | 1313 | 1575 | 1725 | 1950 | 2250 | 1094 | 1313 | 1438 | 1625 | 1875 | 875 | 1050 | 1150 | 1300 | 1500 |
| | | .750 | 1890 | 2268 | 2484 | 2808 | 3240 | 1575 | 1890 | 2070 | 2340 | 2700 | 1313 | 1575 | 1725 | 1950 | 2250 | 1050 | 1260 | 1380 | 1560 | 1800 |
| | | .281 | 590 | 709 | 776 | 877 | 1012 | 492 | 591 | 647 | 731 | 843 | 410 | 492 | 539 | 609 | 703 | 328 | 394 | 431 | 487 | 562 |
| | | .312 | 655 | 786 | 861 | 973 | 1123 | 547 | 656 | 719 | 812 | 936 | 456 | 547 | 599 | 677 | 780 | 365 | 438 | 479 | 542 | 624 |
| | | .344 | 722 | 866 | 949 | 1073 | 1238 | 602 | 722 | 791 | 894 | 1032 | 502 | 602 | 659 | 745 | 860 | 401 | 481 | 527 | 596 | 688 |
| | | .375 | 788 | 945 | 1035 | 1170 | 1350 | 656 | 788 | 862 | 975 | 1125 | 547 | 656 | 719 | 813 | 938 | 438 | 525 | 575 | 650 | 750 |
| | | .406 | 853 | 1024 | 1121 | 1267 | 1462 | 711 | 853 | 935 | 1056 | 1218 | 592 | 711 | 779 | 880 | 1015 | 474 | 569 | 623 | 704 | 812 |
| 26 | (STD) 26.000 | .438 | 919 | 1103 | 1208 | 1366 | 1577 | 766 | 919 | 1006 | 1138 | 1314 | 638 | 766 | 839 | 948 | 1095 | 510 | 613 | 671 | 758 | 876 |
| | | .500 | 1050 | 1260 | 1380 | 1560 | 1800 | 875 | 1050 | 1150 | 1300 | 1500 | 729 | 875 | 958 | 1083 | 1250 | 583 | 700 | 767 | 867 | 1000 |
| | | .625 | 1312 | 1575 | 1725 | 1950 | 2250 | 1094 | 1313 | 1437 | 1625 | 1875 | 911 | 1094 | 1198 | 1354 | 1563 | 729 | 875 | 958 | 1083 | 1250 |
| | | .750 | 1575 | 1890 | 2070 | 2340 | 2700 | 1313 | 1575 | 1725 | 1950 | 2250 | 1094 | 1313 | 1438 | 1625 | 1875 | 875 | 1050 | 1150 | 1300 | 1500 |
| | | .281 | 545 | 654 | 717 | 810 | 934 | 454 | 545 | 597 | 675 | 778 | 379 | 454 | 498 | 563 | 649 | 303 | 363 | 398 | 450 | 519 |
| | | .312 | 604 | 727 | 796 | 900 | 1037 | 503 | 604 | 663 | 750 | 864 | 421 | 505 | 553 | 625 | 720 | 337 | 404 | 442 | 500 | 576 |
| | | .344 | 667 | 800 | 876 | 990 | 1143 | 556 | 666 | 730 | 825 | 953 | 463 | 555 | 606 | 688 | 794 | 370 | 444 | 487 | 550 | 635 |
| | | .375 | 726 | 872 | 955 | 1080 | 1246 | 606 | 727 | 796 | 900 | 1039 | 505 | 606 | 663 | 750 | 866 | 404 | 485 | 531 | 600 | 692 |
| 30 | (STD) 30.000 | .406 | 787 | 945 | 1035 | 1170 | 1349 | 656 | 787 | 863 | 975 | 1124 | 547 | 656 | 719 | 813 | 937 | 437 | 525 | 575 | 650 | 750 |
| | | .438 | 848 | 1018 | 1115 | 1260 | 1456 | 707 | 848 | 929 | 1050 | 1213 | 589 | 707 | 774 | 875 | 1011 | 471 | 565 | 619 | 700 | 809 |
| | | .500 | 969 | 1163 | 1274 | 1440 | 1662 | 808 | 969 | 1062 | 1200 | 1385 | 673 | 808 | 885 | 1000 | 1154 | 538 | 646 | 708 | 800 | 923 |
| | | .625 | 1211 | 1454 | 1592 | 1800 | 2077 | 1010 | 1212 | 1327 | 1500 | 1731 | 841 | 1010 | 1106 | 1250 | 1442 | 673 | 808 | 885 | 1000 | 1154 |
| | | .750 | 1454 | 1745 | 1911 | 2160 | 2492 | 1212 | 1454 | 1592 | 1800 | 2077 | 1010 | 1212 | 1327 | 1500 | 1731 | 808 | 969 | 1062 | 1200 | 1385 |
| | | .312 | 525 | 630 | 690 | 780 | 899 | 438 | 525 | 575 | 650 | 749 | 365 | 438 | 479 | 542 | 624 | 292 | 350 | 383 | 433 | 499 |
| | | .344 | 578 | 693 | 759 | 858 | 991 | 482 | 577 | 632 | 715 | 826 | 401 | 481 | 527 | 596 | 688 | 321 | 385 | 422 | 477 | 550 |
| | | .375 | 630 | 756 | 828 | 936 | 1080 | 525 | 630 | 690 | 780 | 900 | 437 | 525 | 575 | 650 | 750 | 350 | 420 | 460 | 520 | 600 |
| 30 | (STD) 30.000 | .406 | 682 | 819 | 897 | 1014 | 1169 | 568 | 682 | 747 | 845 | 974 | 474 | 569 | 623 | 704 | 812 | 379 | 455 | 498 | 563 | 650 |
| | | .438 | 735 | 882 | 966 | 1092 | 1261 | 612 | 735 | 805 | 910 | 1051 | 510 | 612 | 671 | 758 | 876 | 408 | 490 | 537 | 607 | 701 |
| | | .500 | 840 | 1008 | 1104 | 1248 | 1440 | 700 | 840 | 920 | 1040 | 1200 | 583 | 700 | 767 | 867 | 1000 | 467 | 560 | 613 | 693 | 800 |
| | | .625 | 1050 | 1260 | 1380 | 1560 | 1800 | 875 | 1050 | 1150 | 1300 | 1500 | 729 | 875 | 958 | 1083 | 1250 | 583 | 700 | 767 | 867 | 1000 |
| | | .750 | 1260 | 1512 | 1656 | 1872 | 2160 | 1050 | 1260 | 1380 | 1560 | 1800 | 875 | 1050 | 1150 | 1300 | 1500 | 700 | 840 | 920 | 1040 | 1200 |
| * Type A construction also applicable to "Liquid Petroleum Transportation Piping Code," ANSI B31.4-1979 | | | | | | | | | | | | | | | | | | | | | | |
| Notes: | | 1. All dimensions are in inches 2. See Fig 17-24 | | | | | | | | | | | | | | | | | | | | |

* Type A construction also applicable to "Liquid Petroleum Transportation Piping Code," ANSI B31.4-1979

Notes: 1. All dimensions are in inches
2. See Fig 17-24

FIG. 17-28

Pressure-Temperature Ratings for Pipe Flanges and Flanged Fittings from ANSI B16.5-1981

| CLASS | 150 | 300 | 400 | 600 | 900 | 1500 | 2500 | See Notes | |
|--------------------|---|-----|-----|------|----------------|------|--|-----------|---------------------------|
| Material Group 1.1 | | | | | (Carbon Steel) | | A105 (1)(3), A216-WCB (1), A515-70 (1) A516-70 (1) A350-LF2, A537-C1.1 | | (a) (h) (a) (g) (d) |
| °F | Pressures are in pounds per square inch, gauge (psig) | | | | | | | | |
| -20 to 100 | 285 | 740 | 990 | 1480 | 2220 | 3705 | 6170 | | |
| 200 | 260 | 675 | 900 | 1350 | 2025 | 3375 | 5625 | | |
| 300 | 230 | 655 | 875 | 1315 | 1970 | 3280 | 5470 | | |
| 400 | 200 | 635 | 845 | 1270 | 1900 | 3170 | 5280 | | |
| 500 | 170 | 600 | 800 | 1200 | 1795 | 2995 | 4990 | | |
| 600 | 140 | 550 | 730 | 1095 | 1640 | 2735 | 4560 | | |
| 650 | 125 | 535 | 715 | 1075 | 1610 | 2685 | 4475 | | |
| 700 | 110 | 535 | 710 | 1065 | 1600 | 2665 | 4440 | | |
| 750 | 95 | 505 | 670 | 1010 | 1510 | 2520 | 4200 | | |
| 800 | 80 | 410 | 550 | 825 | 1235 | 2060 | 3430 | | |
| 850 | 65 | 270 | 355 | 535 | 805 | 1340 | 2230 | | |
| 900 | 50 | 170 | 230 | 345 | 515 | 860 | 1430 | | |
| 950 | 35 | 105 | 140 | 205 | 310 | 515 | 860 | | |
| 1000 | 20 | 50 | 70 | 105 | 155 | 260 | 430 | | |
| Material Group 2.1 | | | | | (Type 304) | | A182-F304 (5), A182-F304H A240-304 (5)(6), A351-CF8 (5) A351-CF3 | | — — (f) |
| -20 to 100 | 275 | 720 | 960 | 1440 | 2160 | 3600 | 6000 | | |
| 200 | 235 | 600 | 800 | 1200 | 1800 | 3000 | 5000 | | |
| 300 | 205 | 530 | 705 | 1055 | 1585 | 2640 | 4400 | | |
| 400 | 180 | 470 | 630 | 940 | 1410 | 2350 | 3920 | | |
| 500 | 170 | 435 | 585 | 875 | 1310 | 2185 | 3640 | | |
| 600 | 140 | 415 | 555 | 830 | 1245 | 2075 | 3460 | | |
| 700 | 110 | 405 | 540 | 805 | 1210 | 2015 | 3360 | | |
| 800 | 80 | 395 | 525 | 790 | 1180 | 1970 | 3280 | | |
| 850 | 65 | 390 | 520 | 780 | 1165 | 1945 | 3240 | | |
| 900 | 50 | 385 | 510 | 770 | 1150 | 1920 | 3200 | | |
| 950 | 35 | 375 | 500 | 750 | 1125 | 1870 | 3120 | | |
| 1000 | 20 | 325 | 430 | 645 | 965 | 1610 | 2685 | | |
| 1050 | | 310 | 410 | 620 | 925 | 1545 | 2570 | | |
| 1100 | | 260 | 345 | 515 | 770 | 1285 | 2145 | | |
| 1150 | | 195 | 260 | 390 | 585 | 980 | 1630 | | |
| 1200 | | 155 | 205 | 310 | 465 | 770 | 1285 | | |
| 1250 | | 110 | 145 | 220 | 330 | 550 | 915 | | |
| 1300 | | 85 | 110 | 165 | 245 | 410 | 685 | | |
| 1350 | | 60 | 85 | 125 | 185 | 310 | 515 | | |
| 1400 | | 50 | 65 | 90 | 145 | 240 | 400 | | |
| 1450 | | 35 | 45 | 70 | 105 | 170 | 285 | | |
| 1500 | | 25 | 30 | 50 | 70 | 120 | 200 | | |

Notes: (a) permissible but not recommended for prolonged use above 800°F
 (d) not to be used over 650°F
 (f) not to be used over 800°F
 (g) not to be used over 850°F
 (h) not to be used over 1000°F

Additional Notes: (1) Upon prolonged exposure to temperatures above about 800°F (425°C), the carbide phase of carbon steel may be converted to graphite
 (3) Only killed steel shall be used above 850°F (455°C)
 (5) At temperatures over 1000°F (540°C), use only when the carbon content is 0.04 percent or higher
 (6) For temperatures above 1000°F (540°C), use only if the material is heat treated by heating it to a temperature of at least 1900°F (1040°C) and quenching in water or rapidly cooling by other means