GPSA ENGINEERING DATABOOK ERRATA
(2004 FPS Edition)

PAGE	DESCRIPTION
4-18	Figure 4-24, Missing text
4-22	Figure 4-32, Change 0.082 to 0.82
4-24	Figure 4-34, Missing text
5-18	Equations 5-24, 5-28, Corrected
8-7	Example 8-6, Correct text
8-22	Example 8-9, Correct text
12-1	Figure 12-1, Change figure reference
12-6	Figure 12-4, Missing text
12-16	Figure 12-17, Missing text
13-8	Figure 13-10, Missing text
13-15	Example 13-3, Change figure reference
18-16	Figure 18-14, Missing text
20-29	Figure 20-56, Missing text
21-7	Figure 21-5, Correct viscosity for MDEA
21-26	Desorex text, Correct units
23-24	Figure 23-16, Missing text
25-10	Methane-Ethane Binary, Change x-axis scale

FIG. 4-22

Typical Responses Obtained When Determining Ultimate Gain and Ultimate Period

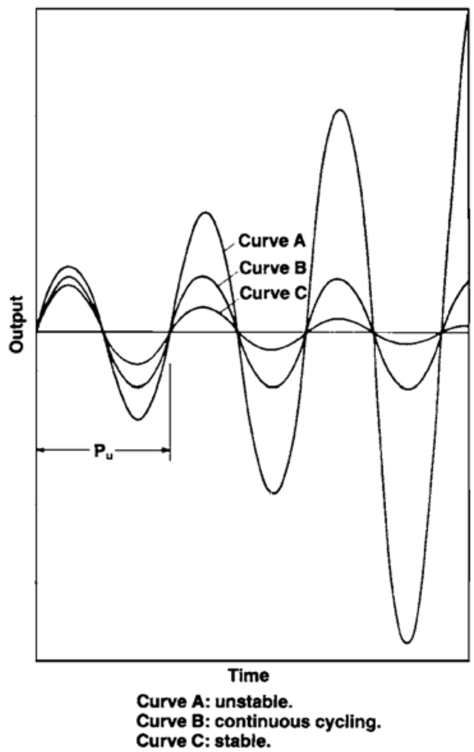


FIG. 4-23

Ziegler-Nichols Settings for 1/4 Decay Response[1]

Mode	K_p	or	PB(%)	T_i	T_d
(P)	0.5 K_u		2(PB_u)	max.	zero
(PI)	0.45 K_u		2.2(PB_u)	P_u/1.2	zero
(PD)	0.6 K_u		1.65(PB_u)	max.	P_u/8.0
(PID)	0.6 K_u		1.65(PB_u)	P_u/2.0	P_u/8.0

FIG. 4-24

Typical Controller Settings

Process	Gain	PB(%)	Integral		Derivative
			T_i (sec)	min/repeat	T_d (sec)
Flow	0.6-0.8	167-125	3.0-1.8	0.05-0.03	0.0
Pressure	5.0	20.0	120-60	2.0-1.0	0.0
Temp.	1.0-2.0	100-50	120-30	2.0-0.5	6.0-12
Level	0.8-1.2	125-83	600-300	10.0-5.0	0.6-1.2

Fig. 4-23 gives relative controller gain, integral time, and derivative time for the various control mode combinations for quarter-decay response as related to ultimate controller gain setting, K_u, and ultimate period P_u. Gain settings are also shown in units of proportional band, PB.

Fig. 4-24 shows some typical settings for various types of process controllers.

Example 4-2 — An example using the Ziegler-Nichols method is given below:

For a certain temperature control system, the ultimate sensitivity K_u was found to be 0.4 psi per deg F, and the ultimate period P_u was found to be two minutes. A three mode PID controller is required.

Using Fig. 4-24:

Proportional gain K_p:

$$K_p = 0.6\,K_u = 0.6\,(0.4\text{ psi/°F}) = 0.24\text{ psi/°F}$$

Integral time constant T_i:

$$T_i = P_u/2,\ T_i = 2/2 = 1.0\text{ minute}$$

Derivative time constant T_d:

$$T_d = P_u/8,\ T_d = 2/8 = 0.25\text{ minutes}$$

Control Mode Considerations

The process control engineer has the responsibility for matching the many and variable characteristics of the process to be controlled with the most effective control hardware available.[6] Fig. 4-25 provides guidelines for choosing the mode of control for various types of applications based upon the process reaction rate and size and speed of load changes.

Special considerations should be made in applying a “split-range” controller. A common example is a column temperature controller on a cryogenic demethanizer. In this system the first half (0-50%) of the controller output actuates the “free” heat exchange with the incoming feed, and the second half (50-100%) of the controller output actuates the supplemental heat from the hot oil system. Adaptive gain control may be required since the heating value of the hot oil is much greater than that of the gas used in the heat exchange.

EMBEDDED ADVANCED CONTROL

Embedded advanced control will usually give an improved plant performance over that achievable with traditional techniques. By introducing Embedded Advanced Control, a high level of reliability and security is provided to maximize control system uptime. Since embedded advanced control tools have direct access to controller I/O, they may access process measurements and actuators with no communication jitter or delay. This allows use of these tools on the fastest processes.

CONTROL VALVES

Selecting the proper control valve for each application involves many factors. The valve body design, actuator style, and plug characteristic are critical items for selection. Proper valve sizing is necessary for accurate, efficient, economical process control. In areas where personnel will be affected, noise prediction and control becomes a significant factor.

Engineering application guidelines, nomographs, and equations presented in the following pages may be used to determine the correct control valve configuration, size and flow characteristics, and to predict noise levels for most applications. The material presented here may also be used to evaluate the performance of valves installed in existing plants.

The equations given in this section are used to calculate the flow coefficient (C_v or C_g) required for a valve to pass the rewith the listed Cv should then be used in the chosen sizing equation to calculate a revised, required C_v. This iteration process continues until the calculated C_v and equals the manufactuer's listed C_v.

4. For a new valve selection a valve size is typically chosen such that the maximum, calculated C_v is close to 75% to 85% of valve travel. This allows for process variability while maintaining flow capability. The minimum, calculated C_v should typically occur at or about 10% of valve travel.
5. F_p is the Piping Geometry Factor. It corrects the sizing equations for the effects of fittings such as reducers and expanders that are attached to the valve body ends. F_p values can be determined via test or calculated per the ANSI/ISA S75.01 standard. If the valve has no such fittings attached, e.g., the nominal value size and nominal pipe size are the same, then F_p = 1.0. Refer to the full standard for the F_p calculations in cases where fittings do exist.

Other valve configurations, such as ball and butterfly valves, can be sized in a similar manner using the unique X_c and C_v values derived by the manufacturers.

FIG. 4-32

Typical C_V, X_C and F_L Values for Valves*

Valve Style	Body Size, Inches	Flow Characteristic					
		Equal Percentage			Linear		
		C_v	X_c	F_L	C_v	X_c	F_L
Globe	1	8	0.74	0.88	17	0.61	0.84
	1-1/2	17	0.69	0.84	30	0.70	0.82
	2	25	0.70	0.85	62	0.68	0.77
	2-1/2	49	0.66	0.84	84	0.71	0.81
	3	66	0.66	0.82	118	0.70	0.82
	4	125	0.67	0.82	181	0.74	0.82
	6	239	0.74	0.85	367	0.78	0.84
	8	268	0.60	0.85	526	0.74	0.87
Ball	1	16	0.53	0.86	–	–	–
	2	59	0.53	0.81	–	–	–
	3	120	0.50	0.80	–	–	–
	4	195	0.52	0.80	–	–	–
	6	340	0.52	0.80	–	–	–
	8	518	0.54	0.82	–	–	–
	10	1000	0.47	0.80	–	–	–
	12	1530	0.49	0.78	–	–	–
Butterfly	2	60	0.37	0.69	–	–	–
	3	111	0.40	0.69	–	–	–
	4	238	0.40	0.69	–	–	–
	6	635	.040	0.69	–	–	–
	8	1020	0.40	0.69	–	–	–
	10	1430	0.40	0.69	–	–	–
	12	2220	0.40	0.69	–	–	–
	14	2840	0.40	0.69	–	–	–
	16	3870	0.40	0.69	–	–	–

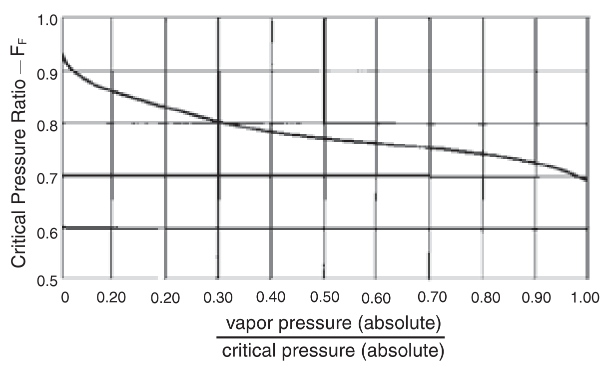
*At approximately 70% of valve travel. Maximum valve capacity may be estimated using the values given in this figure in conjunction with Fig. 4-29. For a more detailed analysis of capacity capabilities of a given valve at other percentages of travel, consult the valve manufacturer's data.

through this calculation accounting for the variation in F_1 and valve-rated C_v due to valve style, size, trim, flow direction, etc.

4. Select the higher of the two calculated C_v's as the required C_v.
5. From the valve manufacturer's sizing data, select a specific valve type and size such that the listed C_v is equal to or greater than the calculated C_v.
6. See the previous section on Cavitation and consult the manufacturer's data for appropriate valve cavitation operating limits.

FIG. 4-33

Critical Pressure Ratios for All Liquids, F_F



Use this curve for all liquids. Determine the ratio by dividing the pressure which may be obtained from Figure 4-34. From the ratio thus calculated the critical pressure ratio, F_F, may be read from this curve.

FIG. 4-34

Critical Pressure of Various Liquids

psia			
Ammonia	1636	Isobutane	529
Argon	706	Isobutylene	580
n-Butane	551	Methane	668
Carbon Dioxide	1071	Nitrogen	493
Carbon Monoxide	508	Nitrous Oxide	1048
Chlorine	1118	Oxygen	737
Dowtherm A	465	Phosgene	823
Ethane	708	Propane	616
Ethylene	731	Propylene	667
Fluorine	809	Refrigerant 11	635
Helium	33	Refrigerant 12	597
Hydrogen	188	Refrigerant 22	716
Hydrogen Chloride	1205	Water	3208

FIG. 4-35

Liquid Valve Sizing Equations
Use Fig. 4-36 for value of Numerical Constants, N

Flow Basis and Units	Equation
Nonvaporizing Mass Flow with Specific Weight, γ_1	$w = N_6 F_p C_v \sqrt{(P_1 - P_2)\gamma_1}$
Nonvaporizing Volumetric Flow with Specific Gravity, G_f	$q = N_1 F_p C_v \sqrt{\frac{P_1 - P_2}{G_f}}$
Vaporizing Mass Flow with Specific Weight, γ_1	$w = N_6 F_L C_v \sqrt{(P_1 - F_F P_v)\gamma_1}$
Vaporizing Volumetric Flow with Specific Gravity, G_f	$q = N_1 F_L C_v \sqrt{\frac{P_1 - F_F P_v}{G_f}}$

FIG. 4-36

Numerical Constants for Liquid Flow Equations

Constant		Units Used in Equations					
N		w	q	p, Δp	d, D	γ_1	v
N1	0.0865 0.865 1.00	– – –	m^3/h m^3/h gpm	kPa bar psia	– – –	– – –	– – –
N_6	2.73 27.3 63.3	kg/h kg/h lb/h	– – –	kPa bar psia	– – –	kg/m^3 kg/m^3 lb/ft^3	– – –

INSTALLATION, TROUBLESHOOTING, AND CALIBRATION

Installation and Troubleshooting

Control system troubleshooting logically falls into two categories: (1) the repair of control systems that previously functioned well, and (2) the successful modification of poorly commissioned systems that have never worked properly due to improper application, poor design, faulty hardware, or improper operating procedures (Fig. 4-37). Different techniques are employed for each category.

Failed Systems

- Control system malfunctions normally are reported by the process operator. A discussion with the operator should yield some clues as to the source of the problem, since he has probably been observing it for several hours, or days.
- The next step is to use the "process of elimination" to localize the problem. If replacement of an element with a known good one causes the problem to disappear, this is usually conclusive! Often this simple approach of parts changing will save time by avoiding a detailed system analysis. However, if the situation permits, the "bad" part should be temporarily re-installed to verify a "hard" failure rather than a "hung-up" condition which is often reset by the procedure of substitution.

FIG. 5-20

Fraction of Heat Radiated Values for Flared Gases

Carbon Monoxide	0.075
Hydrogen	0.075
Hydrogen Sulfide	0.070
Ammonia	0.070
Methane	0.10
Propane	0.11
Butane	0.12
Ethylene	0.12
Propylene	0.13

The maximum value of ε for any gas is 0.13.

fraction of heat radiated values for the most frequently flared gases is shown in Fig. 5-20.

To calculate the intensity of radiation at different locations, it is necessary to determine the length of the flame and its angle in relation to the stack (see Fig. 5-21). A convenient expression to estimate length of flame, L_f, is shown below, based on information from equipment suppliers.

$$L_f = (10)\,(d)\sqrt{\frac{\Delta P_w}{55}} \qquad \textbf{Eq 5-22}$$

or from API RP 521,

$$L_f = 3.94\,[(Q_r)\,(10^{-6})]^{0.474} \qquad \textbf{Eq 5-23}$$

For conventional (open pipe) flares, an estimate of total flare pressure drop is 1.5 velocity heads based on nominal flare tip diameter. The pressure drop equivalent to 1 velocity head is given by:

$$\Delta P_w = \frac{(27.7)\,\rho\,V^2}{(2\,g_c)\,(144)} = \frac{\rho\,V^2}{334.8} \qquad \textbf{Eq 5-24}$$

ΔP_w is the pressure drop at the tip in inches of water. After determining tip diameter, d, using Eq 5-25, and the maximum required relieving capacity, flame length for conditions other than maximum flow can be calculated using Eq 5-22 and Eq 5-24.

Common practice is to use tip velocities of up to Mach 0.5 for short term emergency flows and Mach 0.2 for maximum continuous flowing.

$$d = \left(\sqrt{\frac{1.702 \cdot 10^{-5} \cdot W}{P_2 \cdot M} \cdot \left(\frac{Z \cdot T}{k \cdot MW}\right)^{0.5}}\right) \cdot 12 \qquad \textbf{Eq 5-25}$$

Sonic velocity of a gas is given by:

$$a = 223\sqrt{k\,\frac{T}{MW}} \qquad \textbf{Eq 5-26}$$

The center of the flame is assumed to be located at a distance equal to 1/3 the length of the flame from the tip.

The angle of the flame results from the vectorial addition of the velocity of the wind and the gas exit velocity.

$$\theta = \tan^{-1}\left(\frac{V_w}{V_{ex}}\right) \qquad \textbf{Eq 5-27}$$

$$V_{ex} = 550\sqrt{\frac{\Delta P_W}{55}} \qquad \textbf{Eq 5-28}$$

Note: API gives a greater lean angle

The coordinates of the flame center with respect to the tip are:

$$X_c = (L_f / 3)\,(\sin\theta) \qquad \textbf{Eq 5-29}$$

$$Y_c = (L_f / 3)\,(\cos\theta) \qquad \textbf{Eq 5-30}$$

The distance from any point on the ground level to the center of the flame is:

$$R = \sqrt{(X - X_c)^2 + (H_s + Y_c)^2} \qquad \textbf{Eq 5-31}$$

Equations 5-21 and 5-31 allow radiation to be calculated at any location.

The stack height results from considering the worst position vertically below the center of the flame for a given condition of gas flow and wind velocities (see Fig. 5-21).

$$R^2 = (H_s + Y_c)^2 \qquad \textbf{Eq 5-32}$$

$$R = (H_s + Y_c) \qquad \textbf{Eq 5-33}$$

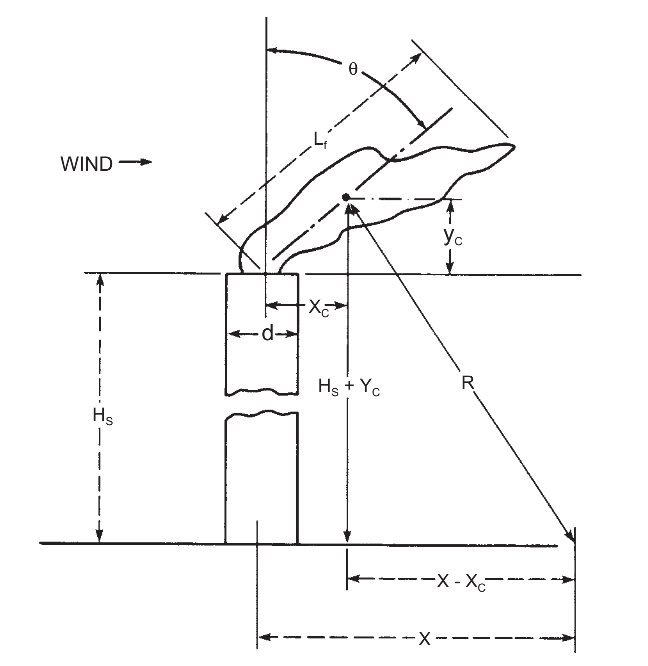
$$H_s = (R - Y_c) \qquad \textbf{Eq 5-34}$$

$$H_s = R - [(L_f / 3)\,(\cos\theta)] \qquad \textbf{Eq 5-35}$$

This method assumes that for different wind velocities the length of the flame remains constant. In reality this is not true.

FIG. 5-21

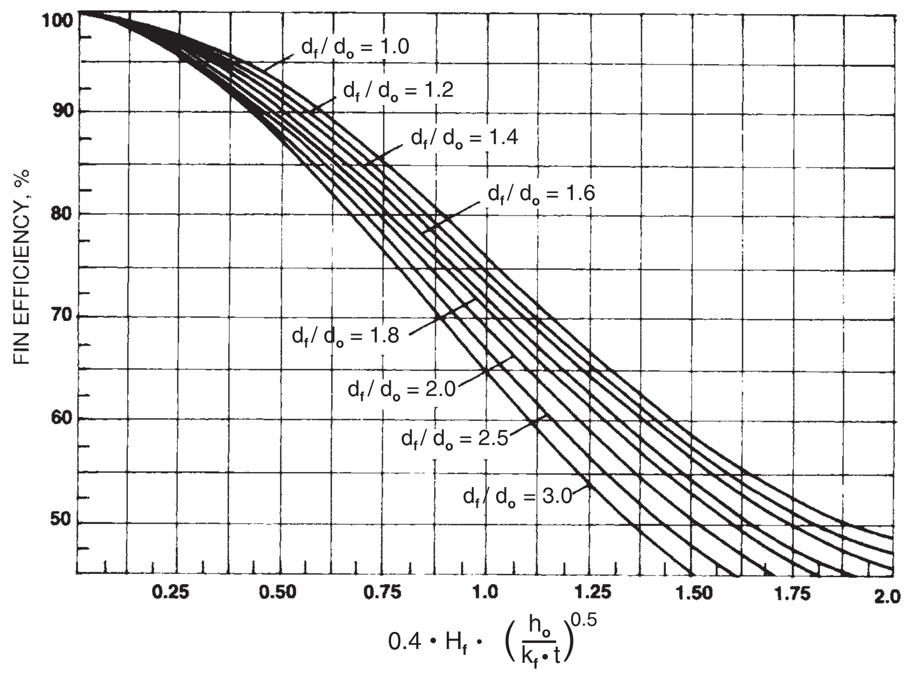
Dimensional References for Sizing a Flare Stack



Courtesy American Petroleum Institute

FIG. 8-6

Fin Efficiency Chart[4]



Example 8-6 — What is the radiant heat flux to a 3 ft length of a 2 ft ID firetube when the combustion gases inside the tube are at 2800°F and the firetube wall is at 300°F? Assume 20% excess air is used.

Solution Steps

$$F = \frac{\text{curved surface area}}{\text{total surface area}} = \frac{\pi \bullet D \bullet L}{\pi \bullet D \bullet L + 2\,(\pi \bullet D^2/4)}$$

$$= \frac{2\,(3)}{2\,(3) + 2\,(4/4)} = 0.75$$

From Fig. 8-10, $P_{CO_2} + P_{H_2O} = 0.24$

From Fig. 8-11, $L = 2.0$ ft, so $P \bullet L = 0.48$

From Fig. 8-12, $\varepsilon_1 = 0.12$

From Fig. 8-9, $\varepsilon_2 = 0.79$ (steel, oxidized at 1100°F)

Equation 8-14, $$\frac{Q}{A} = \frac{0.173\,(10^{-8})\,0.75\,(3260^4 - 760^4)}{(1/0.12 + 1/0.79 - 1)}$$

$$= 16{,}990 \text{ Btu/(hr} \bullet \text{sq ft)}$$

Note that T is in °R.

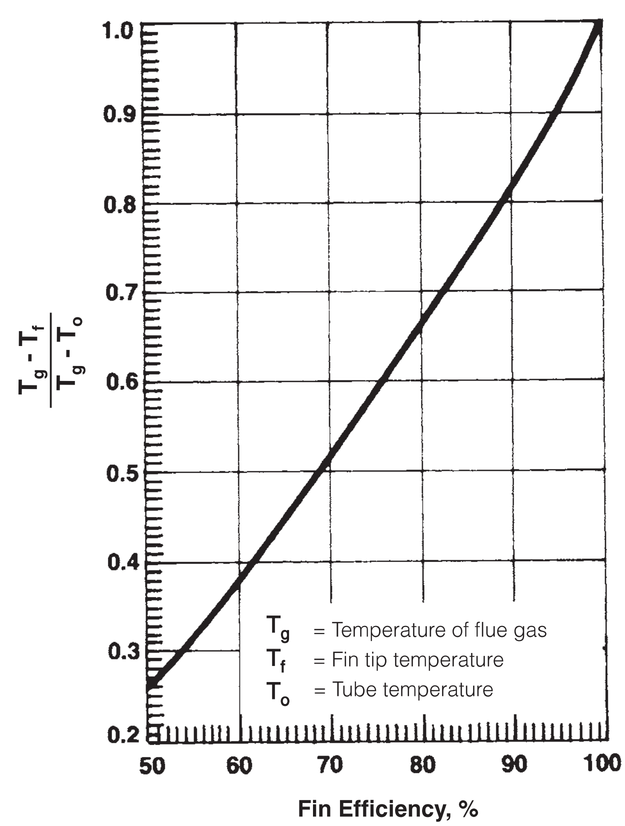
Heat Losses

Heat losses from equipment surfaces occur primarily by radiation and convection. Fig. 8-13 gives the combined heat transfer coefficient, h_c+h_r, in terms of the wind velocity and the temperature difference between the surface and the surrounding air.

Example 8-7 — How much heat can be saved per linear foot by covering an 8 in. Sch 40 steam header, carrying 15 psig steam at 250°F, with a 1 in. thick layer of block insulation? Assume ambient conditions are 30°F with a 15 mph wind.

FIG. 8-7

Fin Tip Temperature[5]



cally when natural gas is the fuel, the fins are 1 in. high, 0.06 in. thick and up to 72 fins per linear ft. For oil fired heaters where soot deposition is possible, the fins are 1 in. high, 0.105 in. thick and not more than 36 fins per ft. Often the first finned row has fewer, shorter, and thicker fins to reduce the fin tip temperature. Where ash and soot fouling are expected, a lane is left every four or five rows for soot blowers. These are tubes equipped with nozzles that direct steam against the tubes. Soot blowing is intermittent and is seldom used more than once every shift.

The fins compensate for the low flue gas heat transfer coefficient. Typically, the heat flux in the convection section is 2000–4000 Btu/(hr • sq ft) of finned surface or 12,000–24,000 Btu/ (hr • sq ft) on a bare tube basis.

Cast iron tube supports can be used below 800°F and 25% chrome – 12% nickel is good up to 2000°F. With high vanadium or sodium levels in the fuel oil, 50% chrome – 50% nickel must be used.

The distance between supports for horizontal tubes should be the lesser of 35 outside tube diameters or 20 ft. The distance between supports on vertical tubes should not exceed either 70 tube diameters or 40 ft. Usually the return bends are external to the tube sheets. This prevents flue gases from bypassing the tube fins.

Fig. 8-25 shows approximate external heat transfer coefficients for 3, 4, and 6 in. steel pipe arranged in staggered rows and surrounded by combustion gases.

Example 8-9 — Design the convection section for the 10 MMBtu/hr regeneration gas heater of Example 8-8. The heat loss is assumed to be 2% of the heat release. Use six 4 in. Sch 80 tubes on 8 in. center-to-center spacing with 8 ft effective length in each row. After two rows of bare shock tubes use finned pipe, 36 fins/ft, 1.25 in. high, 0.105 in. thick. Assume pipe wall temperatures of 200 to 470°F across the finned part of the convection section and average values of 480° and 500°F for the two shock rows.

Solution Steps

Fig. 8-26 summarizes the design of both the radiant and convection sections. A trial and error solution for assumed temperatures is required. Details follow for the converged solution.

$$Q_{total} = \text{duty/NTE} = 10\,(10^6)/0.80 = 12.5 \text{ MMBtu/hr}$$

$$r = 970 \text{ lb flue gas/MMBtu (Fig. 8-24)}$$

$$\text{Flue gases flow rate} = 12.5(970) = 12{,}125 \text{ lb/hr}$$

Assume that the setting loss of 2% or 0.25 MMBtu/hr occurs in the radiant section.

The heat content (LHV) rate of the combustion gases leaving radiant section:

$$Q_{radiant\ exit} = 12.5 - 6.525 - 0.25 = 5.725 \text{ MMBtu/hr}$$

The enthalpy (without latent heat) of the exit gas from radiant section:

$$H = 5.725\,(10^6)/12{,}125 = 472.1 \text{ Btu/lb}$$

$$T_g = 1730°F \text{ (Fig. 8-20, Flue Gas – LHV)}$$

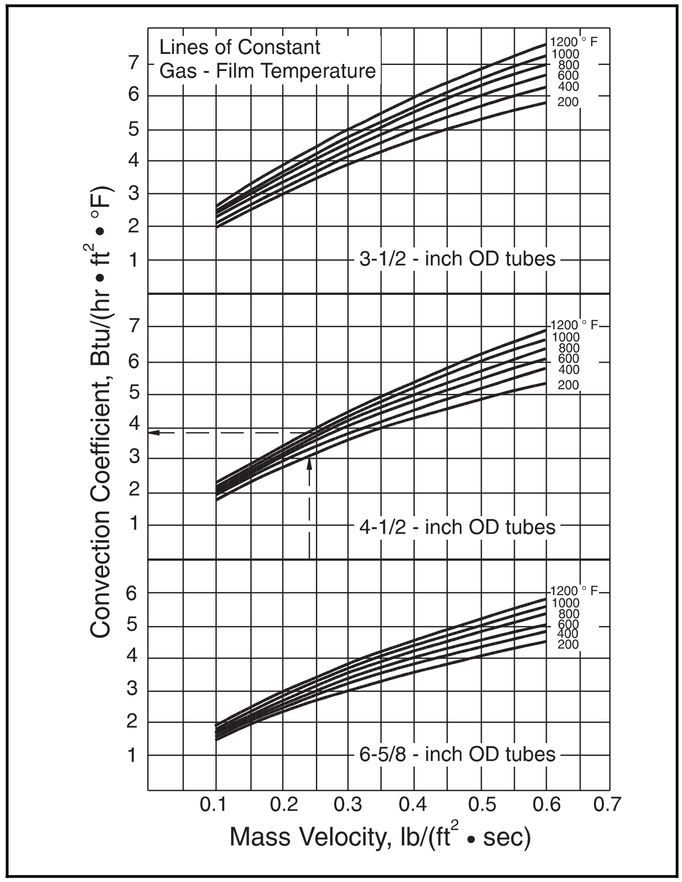
Convection Section:

$$\text{Area for gas flow} = \text{(no. of tubes) (L) (spacing – D)}$$

$$= (6)\,(8)\left[\frac{8}{12} - \frac{4.5}{12}\right] = 14.0 \text{ sq ft}$$

FIG. 8-25

Flue Gas Convection-Coefficients for Flow Across Staggered Banks of Bare Tubes[9]



$$G_g = 12{,}125/(14.0)\,(3600) = 0.241 \text{ lb/(sec • sq ft)}$$

First shock row. Assume the average gas temperature is 1625°F and tube wall temperature is 500°F.

$$T_g \text{ mean} = \frac{500 + 1625}{2} = 1062°F$$

$$h_o = 3.8 \text{ Btu/(hr • sq ft • °F) (Fig. 8-25)}$$

$$A = 1.178 \text{ sq ft per linear ft} \quad \text{(Example 8-8)}$$

$$A_{tubes} = 48\,(1.178) = 56.54 \text{ sq ft}$$

$$Q_c = h_o A \Delta T = (3.8)\,(56.54)\,(1625 - 500)$$

$$= 0.242 \text{ MMBtu/hr}$$

$$I = Q/A = 10{,}000 \text{ Btu/(hr • sq ft) (Example 8-8)}$$

$$Q_r = (Q/A)\,(A) = 10{,}000\,(56.54) = 0.565 \text{ MMBtu/hr}$$

$$Q_c + Q_r = (0.242 + 0.565)\,10^6 = 0.807 \text{ MMBtu/hr}$$

$$Q_{exit\ gases} = (5.725 - 0.807)\,10^6 = 4.918 \text{ MMBtu/hr}$$

$$H_{exit\ gases} = 4.918\,(10^6)/12{,}125 = 405.6 \text{ Btu/lb}$$

$$T_g\text{(exit)} = 1520°F \text{ (Fig. 8-20, Flue Gas – LHV)}$$

SECTION 12

Pumps & Hydraulic Turbines

Pumps

The most common types of pumps used in gas processing plants are centrifugal and positive displacement. Occasionally regenerative turbine pumps, axial-flow pumps, and ejectors are used.

Modern practice is to use centrifugal rather than positive displacement pumps where possible because they are usually less costly, require less maintenance, and less space. Conventional centrifugal pumps operate at speeds between 1200 and 8000 rpm. Very high speed centrifugal pumps, which can operate

FIG. 12-1

Nomenclature

A = cross-sectional area of plunger, piston, or pipe, sq in.
a = cross-sectional area of piston rod, sq in.
AC = alternating current
bbl = barrel (42 U.S. gallons)
bhp = brake horsepower
C = constant (Fig. 12-19)
C_p = specific heat at average temperature, Btu/(lb • °F)
cfs = cu ft/sec
D = displacement of reciprocating pump, gpm
DC = direct current
d = impeller diameter, in.
e = pump efficiency, fraction
g = 32.2 ft/sec^2 (acceleration of gravity)
gpm = U.S. gallons/minute
H = total equipment head, ft of fluid
h = head, ft of fluid pumped
hyd hp = hydraulic horsepower
k = factor related to fluid compressibility (Fig. 12-19)
K = type of pump factor (Eq. 12-18)
L = length of suction pipe, ft
L_s = stroke length, in.
m = number of plungers or pistons
NPSH = net positive suction head of fluid pumped, ft
NPSHA = NPSH available, ft
NPSHR = NPSH required, ft
n = speed of rotation, revolutions/minute (rpm)
n_s = specific speed (See Fig. 12-2 for units)
N = Polytropic exponent of charge gas. (For nitrogen, N = 1.4)
ΔP = differential pressure, psi
P = pressure, psia or psig
P_{vp} = liquid vapor pressure at pumping temperature, psia
psi = lb/sq in.
psia = lb/sq in. absolute
psig = lb/sq in. gauge
Q = rate of liquid flow, gpm
r = ratio of internal volume of fluid between valves, when the piston or plunger is at the end of the suction stroke, to the piston or plunger displacement.
RD = relative density to water at standard temperature
s = slip or leakage factor for reciprocating and rotary pumps
S = suction specific speed (units per Eq 12-7)
sp gr = specific gravity at average flowing conditions. Equal to RD
T = torque, ft lb
t_r = temperature rise, °F
u = impeller peripheral velocity, ft/sec
VE = volumetric efficiency, fraction
VE_o = overall volumetric efficiency
VE_ρ = volumetric efficiency due to density change
VE_l = volumetric efficiency due to leakage
V_{pd} = pulsation dampener volume, in^3
v = liquid mean velocity at a system point, ft/sec
z = elevation of a point of the system above (+) or below (–) datum of the pump. For piping, the elevation is from the datum to the piping centerline; for vessels and tanks, the elevation is from the datum to the liquid level.

Greek:

ρ = density at average flowing conditions, lb/ft^3
ρ_i = inlet density, lb/ft^3
ρ_o = outlet density, lb/ft^3
Δ = allowable pressure fluctuations as a percentage of mean pressure

Subscripts:

a = acceleration
ave = with P, average pressure in pulsating flow
bep = best efficiency point, for maximum impeller diameter
c = compression
d = discharge of pump
dv = discharge vessel
D = displacement
f = friction
i = inlet of equipment
l = leakage
max = with P, maximum acceptable peak pressure in pulsating flow
min = with P, minimum acceptable valley pressure in pulsating flow
o = outlet of equipment
ov = overall
p = pressure
r = rise

packed fractionating towers, consider only the piping and exclude such vessels from the system.

5. Add the static head to the suction vessel pressure, then subtract the frictional head losses in the suction piping. This gives the total pressure (or head) of liquid at the pump suction flange.
6. Add the discharge vessel pressure, the frictional head losses in the discharge piping system, and the discharge static head. This gives the total pressure (or head) of liquid at the pump discharge. According to the type of capacity and head controls, pump type and energy conservation, required for the particular situation, provide a head and/or a flow additional margin to provide a good control. A control valve to throttle the discharge or to recirculate the flow, or a variable speed motor, etc. may be the options to provide good control.
7. Calculate the required pump total head by subtracting the calculated pump suction total pressure from the calculated pump discharge total pressure and converting to head.
8. It is prudent to add a safety factor to the calculated pump head to allow for inaccuracies in the estimates of heads and pressure losses, and pump design. Frequently a safety factor of 10% is used, but the size of the factor used for each pump should be chosen with consideration of:
   - The accuracy of the data used to calculate the required head
   - The cost of the safety factor
   - The problems which might be caused by installing a pump with inadequate head.

Example 12-1 — Liquid propane, at its bubble point, is to be pumped from a reflux drum to a depropanizer. The maximum flow rate is expected to be 360 gpm. The pressures in the vessels are 200 and 220 psia respectively. The specific gravity of propane at the pumping temperature (100°F) is 0.485. The elevations and estimated frictional pressure losses are shown on Fig. 12-6. The pump curves are shown in Fig. 12-7. The pump nozzles elevations are zero and the velocity head at nozzles is negligible.

Required differential head is determined as follows:

Absolute Total Pressure at Pump Suction

Reflux drum		200.0 psia
Elevation	20 ft. • 0.485/2.31 =	+4.2 psi
Friction	piping	–0.5 psi
	valves	–0.2 psi
		203.5 psia
		= 188.8 psig

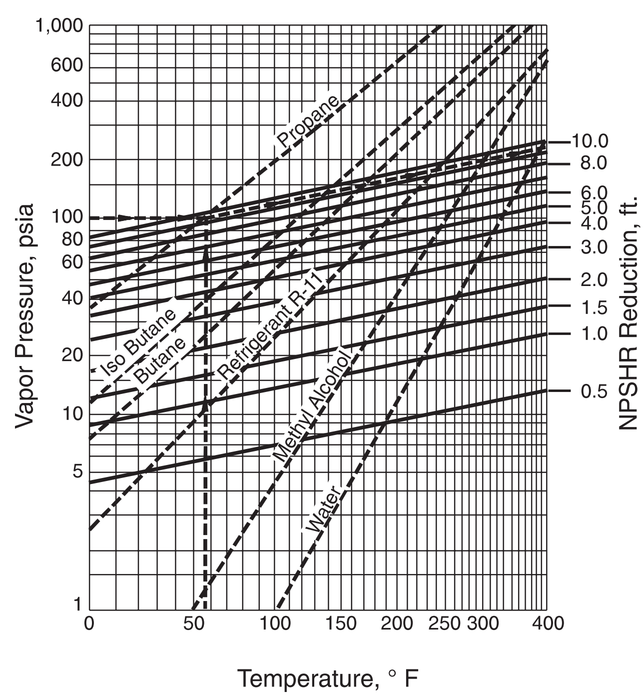
Absolute Total Pressure at Pump Discharge

Tower		220.0 psia
Elevation	74 ft • 0.485/2.31 =	+15.5 psi
Friction	piping	+3.0 psi
	valves	+2.0 psi
	orifice	+1.2 psi
	filter	+13.0 psi
	check valve	+1.0 psi
	control valve	+9.0 psi
		264.7 psia
		= 250.0 psig

FIG. 12-4
Datum elevation

Pump type	Standard	Datum elevation
Centrifugal, horizontal	API 610[1] Hydraulic Institute[5]	Shaft centerline
Centrifugal, vertical in-line	API 610[1]	Suction nozzle centerline
Centrifugal, other vertical	API 610[1]	Top of the foundation
Centrifugal, vertical single suction, volute and diffused vane type	Hydraulic Institute[5]	Entrance eye to the first stage impeller
Centrifugal, vertical double suction	Hydraulic Institute[5]	Impeller discharge horizontal centerline
Vertical turbine. Line shaft and submersible types	AWWA E101[18]	Underside of the discharge head or head baseplate
Reciprocating	Hydraulic Institute[5]	Suction nozzle centerline
Rotary	Hydraulic Institute[5]	Reference line or suction nozzle centerline

FIG. 12-5
NPSHR Reduction for Centrifugal Pumps Handling Hydrocarbon Liquids and High Temperature Water



This chart has been constructed from test data obtained using the liquids shown. For applicability to other liquids refer to the Hydraulic Institute Standards.[5]

FIG. 12-17

Check List for Centrifugal Pump Troubles and Causes

Trouble:	Possible Causes:
1. Failure to deliver liquid	a. Wrong direction of rotation b. Pump not primed c. Suction line not filled with liquid d. Air or vapor pocket in suction line e. Inlet to suction pipe not sufficiently submerged f. Available NPSH not sufficient g. Pump not up to rated speed h. Total head required greater than head which pump is capable of delivering
2. Pump does not deliver rated capacity	a. Wrong direction of rotation b. Suction line not filled with liquid c. Air or vapor pocket in suction line d. Air leaks in suction line or stuffing boxes e. Inlet to suction pipe not sufficiently submerged. f. Available NPSH not sufficient g. Pump not up to rated speed h. Total head greater than head for which pump designed j. Foot valve too small k. Foot valve clogged with trash m. Viscosity of liquid greater than that for which pump designed n. Mechanical defects: (1) Wearing rings worn (2) Impeller damaged (3) Internal leakage resulting from defective gaskets o. Discharge valve not fully opened
3. Pump does not develop rated discharge pressure	a. Gas or vapor in liquid b. Pump not up to rated speed c. Discharge pressure greater than pressure for which pump designed d. Viscosity of liquid greater than that for which pump designed e. Wrong rotation f. Mechanical defects: (1) Wearing rings worn (2) Impeller damaged (3) Internal leakage resulting from defective gaskets
4. Pump loses liquid after starting	a. Suction line not filled with liquid b. Air leaks in suction line or stuffing boxes c. Gas or vapor in liquid d. Air or vapor pockets in suction line e. Inlet to suction pipe not sufficiently submerged f. Available NPSH not sufficient g. Liquid seal piping to lantern ring plugged h. Lantern ring not properly located in stuffing box
5. Pump overloads driver	a. Speed too high b. Total head lower than rated head c. Excessive recirculation d. Either or both the specific gravity and viscosity of liquid different from that for which pump is rated e. Mechanical defects: (1) Misalignment (2) Shaft bent (3) Rotating element dragging (4) Packing too tight
6. Vibration	a. Starved suction (1) Gas or vapor in liquid (2) Available NPSH not sufficient (3) Inlet to suction line not sufficiently submerged (4) Gas or vapor pockets in suction line b. Misalignment c. Worn or loose bearings d. Rotor out of balance (1) Impeller plugged (2) Impeller damaged e. Shaft bent f. Improper location of control valve in discharge line g. Foundation not rigid
7. Stuffing boxes overheat	a. Packing too tight b. Packing not lubricated c. Wrong grade of packing d. Insufficient cooling water to jackets e. Box improperly packed.
8. Bearings overheat	a. Oil level too low b. Improper or poor grade of oil c. Dirt in bearings d. Dirt in oil e. Moisture in oil f. Oil cooler clogged or scaled g. Failure of oiling system h. Insufficient cooling water circulation i. Insufficient cooling air k. Bearings too tight m. Oil seals too close fit on shaft n. Misalignment
9. Bearings wear rapidly	a. Misalignment b. Shaft bent c. Vibration d. Excessive thrust resulting from mechanical failure inside the pump e. Lack of lubrication f. Bearings improperly installed g. Dirt in bearings h. Moisture in oil j. Excessive or insufficient cooling of bearings

For double acting cylinders, the percent clearance is based on the total clearance volume for both the head end and the crank end of a cylinder. These two clearance volumes are not the same due to the presence of the piston rod in the crank end of the cylinder. Sometimes additional clearance volume (external) is intentionally added to reduce cylinder capacity.

The term "volumetric efficiency" refers to the actual pumping capacity of a cylinder compared to the piston displacement. Without a clearance volume for the gas to expand and delay the opening of the suction valve(s), the cylinder could deliver its entire piston displacement as gas capacity. The effect of the gas contained in the clearance volume on the pumping capacity of a cylinder can be represented by:

$$VE = 100 - r - C\left[\frac{Z_s}{Z_d}(r^{1/k}) - 1\right] \qquad \textbf{Eq 13-14}$$

Volumetric efficiencies as determined by Eq. 13-14 are theoretical in that they do not account for suction and discharge valve losses. The suction and discharge valves are actually spring-loaded check valves that permit flow in one direction only. The springs require a small differential pressure to open. For this reason, the pressure within the cylinder at the end of the suction stroke is lower than the line suction pressure and, likewise, the pressure at the end of the discharge stroke is higher than line discharge pressure.

One method for accounting for suction and discharge valve losses is to reduce the volumetric efficiency by an arbitrary amount, typically 4%, thus modifying Eq. 13-14 as follows:

$$VE = 96 - r - C\left[\frac{Z_s}{Z_d}(r^{1/k}) - 1\right] \qquad \textbf{Eq 13-15}$$

When a non-lubricated compressor is used, the volumetric efficiency should be corrected by subtracting an additional 5% for slippage of gas. This is a capacity correction only and, as a first approximation, would not be considered when calculating compressor horsepower. The energy of compression is used by the gas even though the gas slips by the rings and is not discharged from the cylinder.

If the compressor is in propane, or similar heavy gas service, an additional 4% should be subtracted from the volumetric efficiency. These deductions for non-lubricated and propane performance are both approximate and, if both apply, cumulative.

Fig. 13-10 provides the solution to the function $r^{1/k}$. Values for compression ratios not shown may be obtained by interpo-

FIG. 13-10

Values of $r^{1/k}$

Compression Ratio	k, isentropic exponent C_p/C_v								
	1.10	**1.14**	**1.18**	**1.22**	**1.26**	**1.30**	**1.34**	**1.38**	**1.42**
1.2	1.180	1.173	1.167	1.161	1.156	1.151	1.146	1.141	1.137
1.4	1.358	1.343	1.330	1.318	1.306	1.295	1.285	1.276	1.267
1.6	1.533	1.510	1.489	1.470	1.452	1.436	1.420	1.406	1.392
1.8	1.706	1.675	1.646	1.619	1.594	1.572	1.551	1.531	1.513
2.0	1.878	1.837	1.799	1.765	1.733	1.704	1.677	1.652	1.629
2.2	2.048	1.997	1.951	1.908	1.870	1.834	1.801	1.771	1.742
2.4	2.216	2.155	2.100	2.050	2.003	1.961	1.922	1.886	1.852
2.6	2.384	2.312	2.247	2.188	2.135	2.086	2.040	1.999	1.960
2.8	2.550	2.467	2.393	2.326	2.264	2.208	2.156	2.109	2.065
3.0	2.715	2.621	2.537	2.461	2.391	2.328	2.270	2.217	2.168
3.2	2.879	2.774	2.680	2.595	2.517	2.447	2.382	2.323	2.269
3.4	3.042	2.926	2.821	2.727	2.641	2.563	2.492	2.427	2.367
3.6	3.204	3.076	2.961	2.857	2.764	2.679	2.601	2.530	2.465
3.8	3.366	3.225	3.100	2.987	2.885	2.792	2.708	2.631	2.560
4.0	3.526	3.374	3.238	3.115	3.005	2.905	2.814	2.731	2.655
4.2	3.686	3.521	3.374	3.242	3.124	3.016	2.918	2.829	2.747
4.4	3.846	3.668	3.510	3.368	3.241	3.126	3.021	2.926	2.839
4.6	4.004	3.814	3.645	3.493	3.357	3.235	3.123	3.022	2.929
4.8	4.162	3.959	3.779	3.617	3.473	3.342	3.224	3.116	3.018
5.0	4.319	4.103	3.912	3.740	3.587	3.449	3.324	3.210	3.106
5.2	4.476	4.247	4.044	3.863	3.700	3.554	3.422	3.303	3.193
5.4	4.632	4.390	4.175	3.984	3.813	3.659	3.520	3.394	3.279
5.6	4.788	4.532	4.306	4.105	3.925	3.763	3.617	3.485	3.364
5.8	4.943	4.674	4.436	4.224	4.035	3.866	3.713	3.574	3.448
6.0	5.098	4.815	4.565	4.343	4.146	3.968	3.808	3.663	3.532
6.2	5.252	4.955	4.694	4.462	4.255	4.069	3.902	3.751	3.614
6.4	5.406	5.095	4.822	4.579	4.363	4.170	3.996	3.839	3.696
6.6	5.560	5.235	4.949	4.696	4.471	4.270	4.089	3.925	3.777
6.8	5.713	5.374	5.076	4.813	4.578	4.369	4.181	4.011	3.857
7.0	5.865	5.512	5.202	4.928	4.685	4.468	4.272	4.096	3.937

Gas Pulsation Control

Pulsation is inherent in reciprocating compressors because suction and discharge valves are open during only part of the stroke.

Pulsation must be damped (controlled) in order to:

a. provide smooth flow of gas to and from the compressor,
b. prevent overloading or underloading of the compressors, and
c. reduce overall vibration.

There are several types of pulsation chambers. The simplest one is a volume bottle, or a surge drum, which is a pressure vessel, unbaffled internally and mounted on or very near a cylinder inlet or outlet.

A manifold joining the inlet and discharge connections of cylinders operating in parallel can also serve as a volume bottle.

Performance of volume bottles is not normally guaranteed without an analysis of the piping system from the compressor to the first process vessel.

Volume bottles are sized empirically to provide an adequate volume to absorb most of the pulsation. Several industry methods were tried in an effort to produce a reasonable rule-of-thumb for their sizing. Fig. 13-20 may be used for approximate bottle sizing.

Example 13-3

Indicated suction pressure = 600 psia

Indicated discharge pressure = 1400 psia

Cylinder bore = 6 in

Cylinder stroke = 15 in

Swept volume = $\pi\,(6^2/4)\,(15) = 424$ cu in

From Fig. 13-20:

At 600 psi inlet pressure, the suction bottle multiplier is approximately 7.5. Suction-bottle volume = (7.5) (424) = 3,180 cu in.

NOTE: When more than one cylinder is connected to a bottle, the sum of the individual swept volumes is the size required for the common bottle.

FIG. 13-19

Sectional View of a Cylinder Equipped with a Hand-Operated Valve Lifter and Variable-Volume Clearance

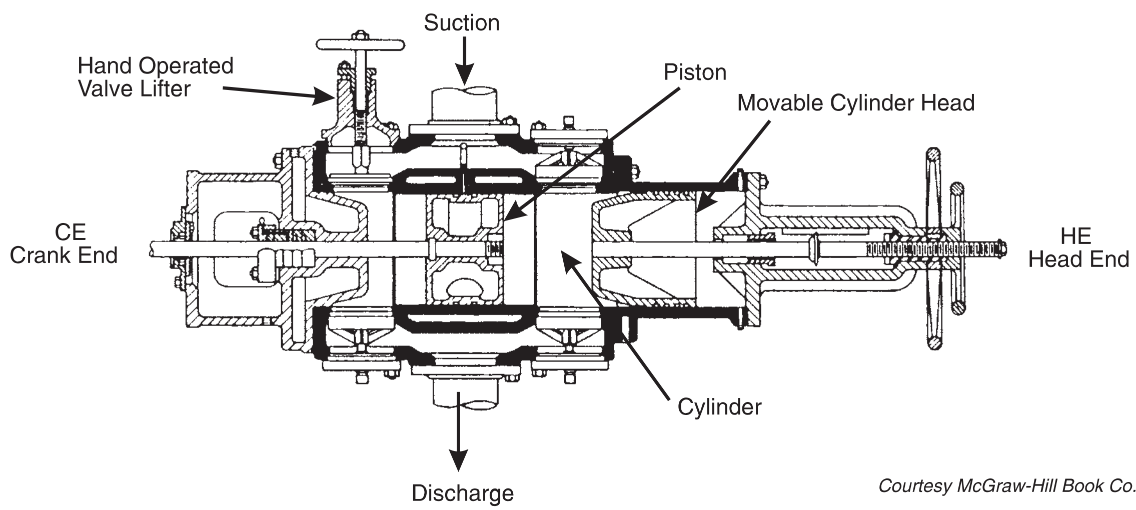


FIG. 13-20

Approximate Bottle Sizing Chart

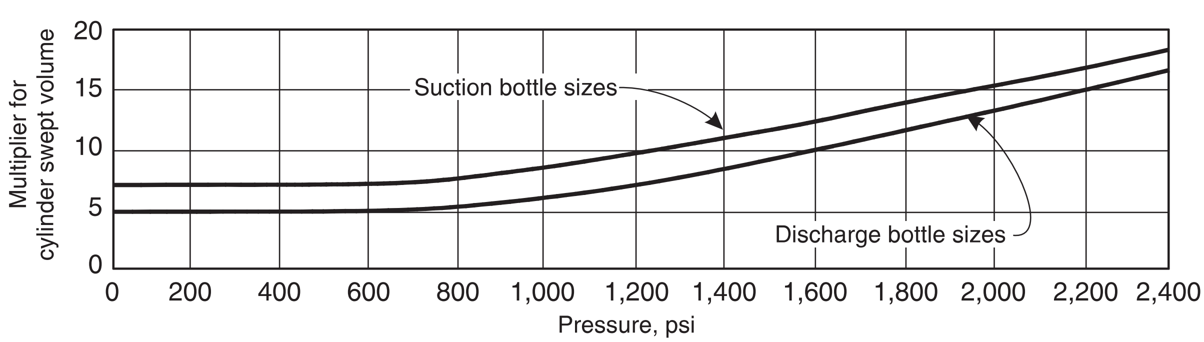


FIG. 18-14

Types of Ion-Exchange Processes

(A)	$Ca(HCO_3)_2$	→	Cation	→	$NaHCO_3$
			Na^{+1}		
	$CaSO_4$	→	Exchanger	→	Na_2SO_4
(B)	$Ca(HCO_3)_2$	→	Cation	→	H_2CO_3
			H^{+1}		
	$CaSO_4$	→	Exchanger	→	H_2SO_4
(C)	Na_2SO_4	→	Anion	→	NaCl
			Cl^{-1}		
	$NaHCO_3$	→	Exchanger	→	NaCl
(D)	H_2CO_3	→	Anion	→	H_2O
			OH^{-1}		
	H_2SO_4	→	Exchanger	→	H_2O

Conventional Softening — Process (A)

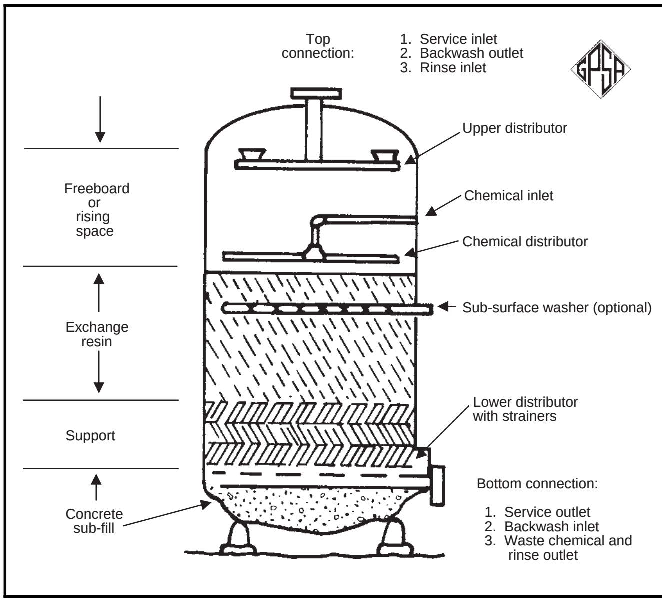
Dealkalization by Split Stream Softening — Blending Effluents from (A) and (B)

Dealkalization by Anion Exchange — Process (C) preceded by (A)

Demineralizing — Combination of (B) and (D)

FIG. 18-15

Typical Ion-Exchange Bed



tricity, etc.) and water treating chemicals. Heat (energy) costs are relatively independent of the feed water composition.

Reverse Osmosis — involves separating water from dissolved solids by forcing the water to pass through a semi-permeable membrane which retains most of the dissolved solids. As illustrated in Fig. 18-16, this is accomplished by providing sufficient pressure on the system feedwater to overcome the normal osmotic pressure and produce a reasonable flow rate through the membrane.

A typical brackish water with a dissolved solids content of 1500 ppmw will have an osmotic pressure of about 15 psi; seawater with a dissolved solids content of about 35000 ppmw has an osmotic pressure of about 350 psi. The applied pressure for brackish water purification is typically in the range of 400-600 psig and for seawater purification, in the range of 800-1000 psig. Recovery of product (desalted) water with reverse osmosis units ranges from 50 to 90% of the feedwater depending upon the feedwater composition, the product water quality requirement, and the number of stages utilized.

Operating costs consist mainly of pumping costs (the pressure drop across the membrane may be from 250 psi to 1000 psi, depending upon dissolved solids content and membrane selection) and membrane cleaning and replacement costs. For water containing from about 250 to 1500 ppmw dissolved solids, an economic comparison of ion exchange and reverse osmosis is frequently necessary to select the more cost effective process. Reverse osmosis has been successfully employed for desalination of seawater. In many cases, the reverse osmosis product water must be treated by one of the ion exchange processes if high quality boiler feedwater is required.

A pretreatment system is needed to avoid fouling or excessive degradation of the membrane. Typically, pretreatment will include filtration to remove suspended particles and addition of chemicals to prevent scaling and biological growth. Because the optimum operating temperature for reverse osmosis systems is about 75-80°F, it is frequently desirable to heat the feedwater. This represents an additional operating cost; however, because reverse osmosis is a continuous process which does not require regenerant chemicals, the cost of disposing of the waste water from the reverse osmosis system may be less than that of waste water from an ion exchange unit.

Electrodialysis — involves separating water from dissolved solids by passing the dissolved solids (ions) through a semi-permeable membrane which is relatively impervious to water. This is accomplished by means of a direct current electrical field which transports the ions through the membranes. Fig. 18-17 shows a basic electrodialysis system with alternating cation-selective and anion-selective membranes.

Recovery of product (deionized) water with electrodialysis units ranges from 50 to 90% of the feedwater depending upon the number of stages and degree of recirculation utilized. Operating costs consist mainly of power costs (typically 6-10 kwh/1000 gallons of product water) and membrane cleaning and replacement costs. Based upon combined capital and operating costs, the electrodialysis process is most economical when used to desalt brackish water (1000 to 5000 ppmw dissolved solids) to a product water concentration of about 500 ppmw dissolved solids.

A pretreatment system is usually needed to prevent fouling or degradation of the membranes. Electrodialysis units can operate over a pH of 1 to 13 and at temperatures up to about 110°F.

Deaeration (Degasifying) — Although other gases (e.g. H_2S, ammonia, methane) can be present in source or makeup water, the dissolved gases of primary concern in boiler feedwater and steam condensate are oxygen and carbon dioxide. Both are highly corrosive and should be removed to the greatest extent possible because the presence of these gases can result in significant damage to piping and equipment and the resulting corrosion products can foul boiler heat transfer surfaces. If a steam condensate treater (polisher) is utilized, high concentrations of corrosion products increase its load and oxygen can attack the ion exchange resin of the treater, espe-

FIG. 20-56

Physical Properties of Selected Glycols and Methanol

	Ethylene Glycol	Diethylene Glycol	Triethylene Glycol	Tetraethylene Glycol	Methanol
Formula	$C_2H_6O_2$	$C_4H_{10}O_3$	$C_6H_{14}O_4$	$C_6H_{18}O_5$	CH_3OH
Molecular Weight	62.1	106.1	150.2	194.2	32.04
Boiling Point* at 760 mm Hg, °F	387.1	472.6	545.9	597.2	148.1
Boiling Point* at 760 mm Hg, °C	197.3	244.8	285.5	314	64.5
Vapor Pressure at 77°F (25°C) mm Hg	0.12	<0.01	<0.01	<0.01	120
Density (g/cc) at 77°F (25°C)	1.110	1.113	1.119	1.120	0.790
(g/cc) at 140°F (60°C)	1.085	1.088	1.092	1.092	
Pounds Per Gallon at 77°F (25°C)	9.26	9.29	9.34	9.34	6.59
Freezing Point, °F	8	17	19	22	–144.0
Pour Point, °F	—	–65	–73	–42	
Viscosity in centipoise at 77°F (25°C)	16.5	28.2	37.3	44.6	0.52
at 140°F (60°C)	4.68	6.99	8.77	10.2	
Surface Tension at 77°F (25°C), dynes/cm	47	44	45	45	22.5
Refractive Index at 77°F (25°C)	1.430	1.446	1.454	1.457	0.328
Specific Heat at 77 °F (25°C) Btu/(lb•°F)	0.58	0.55	0.53	0.52	0.60
Flash Point, °F (PMCC)	240	255	350	400	53.6
Fire Point, °F (C.O.C.)	245	290	330	375	

* Glycols decompose at temperatures below their atmospheric boiling point. Approximate decomposition temperatures are:

Ethylene Glycol 329°F   Triethylene Glycol 404°F   Diethylene Glycol 328°F   Tetraethylene Glycol 460°F

dilute KHI solution, or by changing the KHI carrier fluid to ethylene glycol.

- The KHI and water from the KHI solution will form separate phases if the inhibited fluid is above the lower critical solution temperature (LCST) of the KHI solution.
- The KHI polymer suffers degradation effects at temperatures above 480°F.

Antiagglomerant Inhibitors — Antiagglomerants were developed out of the necessity to extend the range of subcooling for LDHIs beyond that of KHIs, and AAs can achieve subcooling of greater than 40°F. Unlike KHIs, which delay the formation of hydrates, AAs allow their formation at normal rates, but as small nonagglomerating hydrate crystals that are dispersed into an oil or condensate preventing the formation and accumulation of large hydrate crystals. Thus, AAs are suitable only in the presence of liquid hydrocarbon. The mechanism of dispersion is emulsification with the AAs acting as emulsification agents.

AAs Compared to Methanol or Glycols — The comparisons of AAs are similar for KHIs except AAs achieve greater subcooling.

AA Screening Considerations — Although AAs are applicable under most producing conditions, certain conditions must be considered when evaluating a potential application. These conditions include water salinity, emulsification and de-mulsification (i.e., separation), pipeline hydraulics, water cuts, material compatibility, water treating, and downstream impacts.

- Some AAs have a maximum salinity criterion that is normally not exceeded with produced water.
- Since AAs are based on dispersing (i.d., emulsifying) polar hydrate crystals in a nonpolar oil or condensate phase (i.e., continuous phase), they may sometimes require a de-emulsifier for oil and water separation. Further, the addition of a heater upstream or heat coil inside a separator may be required to melt the hydrate crystals.
- Since AAs form crystals that are then dispersed in the liquid hydrocarbon phase, careful consideration of the potential impact on viscosity should be considered including steady state flow, shut-in flow and restart conditions.
- An additional consideration for AAs is that the water cuts (i.e., percent water in the liquids) should be less than 50%. Higher water cuts can invert the emulsion (i.e., change the continuous liquid phase from liquid hydrocarbon to water) and make the AA ineffective.
- AAs can impact the performance of some metallurgy and elastomers, so impacts on existing hardware should be reviewed.

FIG. 21-5

Physical Properties of Gas Treating Chemicals

	Monoethanol-amine	**Diethanol-amine**	**Triethanol-amine**	**Diglycol®-amine**	**Diisopropanol-amine**	**Selexol®**
Formula	$HOC_2H_4NH_2$	$(HOC_2H_4)_2NH$	$(HOC_2H_4)_3N$	$H(OC_2H_4)_2NH_2$	$(HOC_3H_6)_2NH$	Polyethylene glycol derivative
Molecular Wt	61.08	105.14	148.19	105.14	133.19	280
Boiling point @ 760 mm Hg, °F	338.9	516.2 (decomposes)	680 (decomposes)	430	479.7	518
Freezing point, °F	50.9	82.4	72.3	9.5	107.6	–20
Critical constants						
Pressure, psia	868	474.7	355	547.11	546.8	–
Temperature, °F	662	827.8	957.7	756.6	750.6	–
Density @ 20°C, gm/cc.	1.018	1.095	1.124	1.058 @ 60°F	0.999 @ 30°C	1.031 @ 77°F
Weight, lb/gal	8.48 @ 60°F	9.09 @ 60°F	9.37 @ 68°F	8.82 @ 60°F		8.60 @ 77°F
Specific gravity 20°C/20°C	1.0179	1.0919 (30/20°C)	1.1258	1.0572	0.989 @ 45°C/20°C	–
Specific heat @ 60°F, Btu/lb/°F	0.608 @ 68°F	0.600	0.70	0.571	0.69 @ 30°C	0.49 @ 41°F
Thermal conductivity						
Btu/[(hr • sq ft • °F)/ft] @ 68°F	0.148	0.127	–	0.121	–	0.11 @ 77°F
Latent heat of vaporization, Btu/lb	180 @ 760 mmHg	288 @ 73 mmHg	230 @ 73 mmHg	219 @ 760 mmHg	185 @ 760 mmHg	–
Heat of reaction, Btu/lb of Acid Gas						
H_2S			–400	–674	–	–190 @ 77°F
CO_2			–630	–850	–	–160 @ 77°F
Viscosity, cp	24.1 @ 68°F	350 @ 68°F (at 90% wt. solution)	1013 @ 68°F (at 95% wt. solution)	40 @ 60°F	870 @ 86°F 198 @ 113°F 86 @ 129°F	5.8 @ 77°F
Refractive index, N_d 68°F	1.4539	1.4776	1.4852	1.4598	1.4542 @ 113°F	–
Flash point, COC, °F	200	298	365	260	255	304

	Propylene Carbonate	**Methyldiethanol-amine**	**Sulfolane®**	**Methanol**	**10% Sodium Hydroxide**
Formula	$C_3H_6CO_3$	$(HOC_2H_4)_2NCH_3$	$C_4H_8SO_2$	CH_3OH	
Molecular Wt	102.09	119.16	120.17	32.04	19.05
Boiling point @ 760 mm Hg, °F	467	477	545	148.1	217
Freezing point, °F	–56.6	–9.3	81.7	–143.8	14
Critical constants					
Pressure, psia	–		767.3	1153.9	
Temperature, °F	–		1013.8	464	
Density @ 20°C, gm/cc.	1.2057				
Weight, lb/gal		8.68	10.623 @ 30°C/30°C		9.254
Specific gravity 20°C/20°C	1.203	1.0418	1.268	0.7917	1.110
Specific heat @ 60°F, Btu/lb/°F	0.335	0.535	0.35 @ 30°C	0.59 @ 5°-10°C	0.897
Thermal conductivity					
Btu/[(hr • sq ft • °F)/ft] @ 68°F	0.12 @ 50°F	0.159	0.114 @ 100°F	0.124	
Latent heat of vaporization, Btu/lb	208 @ 760 mmHg	204	225.7 @ 212°F	474 @ 760 mmHg	
Heat of reaction, Btu/lb of Acid Gas					
H_2S	–				
CO_2					
Viscosity, cp	1.67 @ 100°F	1.01 @ 68°F 33.8 @ 104°F	10.3 @ 86°F 6.1 @ 122°F 2.5 @ 212°F 1.4 @ 302°F 0.97 @ 392°F	0.6 @ 68°F	1.83 @ 68°F 0.97 @ 122°F 0.40 @ 212°F
Refractive index, N_d 68°F	1.4209	1.469	1.481 @ 86°F	1.3286	
Flash point, COC, °F	270	265	350	58	

pregnate reacts with the mercury to produce a mercury sulfide that is fixed in the carbon microstructure.

Merespec™

Johnson Matthey Catalysts supplies MereSpec™ fixed bed absorbents for removal of traces, elemental and organic, of mercury from hydrocarbon liquids and gases. The absorbents have been shown to be capable of providing the outlet mercury concentration normally specified for LNG production and are in service in several European locations including an offshore oil/gas production platform. Merespec™ is a trademark of Synetix.

Desorex

Activated carbon provides only a limited storage capacity for the strictly physical adsorption of mercury. Desorex HGD2S and HGD4S from Donau Carbon can be employed to bind mercury through the process of chemical adsorption involving oxidation and adsorption in the form of stable compound or fixation in metallic form as an amalgam. Many reference installations of these Desorex products for the purification of natural gases to levels as low as 10 $\mu g/m^3$ of mercury have been realized over a long period of time.

HgSIV

UOP supplies HgSIV adsorbents which are molecular sieves coated with elemental silver. Mercury in the gas is trapped by amalgamation with the silver. The adsorbent also serves the dual function of dehydrating the gas. HgSIV is regenerated thermally, just like molecular sieves for dehydration. This material can be added as a layer to existing molecular sieve dryers[58]. However, one must take care to appropriately handle the regeneration gas in this case, as it will contain mercury.

CMG 275 and 273

Another mercury trapping material labeled CMG 275 was developed by IFP and Procatalyse (today Axens) which is sulfur supported onto mesoporous alumina. The advantage of mesoporous alumina based product is its resistance to capillary condensation[59]. The larger pore size of this material, compared to carbon based trapping materials permits utilization under near dew point conditions. In another material from Axens labeled CMG 273 the trapping component is anchored on the alumina carrier making it completely insoluble in liquid hydrocarbons and water. The material has been subjected at gas plant sites to both DEA and liquid hydrocarbon carry-over with no active phase leaching. This same material has been used to eliminate elemental mercury from LPG and full range condensates.

Organic Mercury Removal

Removal of all forms of organic mercury compounds from natural gases and liquids requires firstly the conversion of the compounds to elemental metallic mercury followed by trapping materials to remove the metallic mercury formed. This requires in the first stage some hydrogen for the organo-mercury hydrogenolysis with a suitable catalyst. The first stage catalyst such as MEP 841 also traps arsenic and lead impurities in the feed. The two stage impurities removal process is called RAM and is available from Axens.

ACID GAS INJECTION

In some cases, it is possible to compress the acid gas as generated from a chemical or a physical solvent process and inject it into a depleted, non-producing, or even a producing reservoir[60]. A key consideration is the phase behavior of the acid gas mixture. Depending on temperature, pressure, and composition, the acid gas may be injected as a liquid or gas; or as a dense phase. Dehydration is usually necessary to avoid corrosion or hydrate formation. However, in many cases, a minimum in water-holding capacity occurs with respect to pressure. Thus, it may be possible to compress the gas to a given level, cool it, and drop out the liquid water. Further compression increases the capacity of acid gas to hold water, so that water drop out should not occur in the downstream piping or well. The phase behavior of acid gas mixtures is complex, and careful consideration must be given to the design of the project.[61,62]

There have been a number of successful acid gas injection projects, with sulfur contents of 1 to more than 70 tons per day[63].

LIQUID HYDROCARBON TREATING

As a guide in the selection of the method of treating to be used, the following characteristics of each are given:

Regenerated Caustic

1. Can handle large volumes of hydrocarbon.
2. Primarily for removing methyl and ethyl mercaptans.
3. Capable of producing a doctor-sweet product.
4. Reduces the total sulfur content of treated product.

Perco Solid Copper Chloride

1. Can treat gasoline streams with relatively high mercaptan content.
2. Suited for small flow rate.
3. Sulfur content not reduced.
4. Water content must be only that of saturation.
5. Hydrogen sulfide must be removed ahead of contact with bed.
6. Excess regeneration oxygen may cause corrosion downstream of bed. Gasoline with components that may be affected by oxygen, such as olefins, should not be treated with this process.
7. Capable of producing doctor-sweet product.

Batch Caustic Wash

1. Can use a single wash.
2. Best suited for streams with low mercaptan content (if mercaptan removal is important).
3. Primarily for removing trace amounts of hydrogen sulfide and methyl and ethyl mercaptans.
4. Disposing of spent caustic can be a major consideration.
5. Relatively high caustic consumption per gallon of product.

Solid Potassium Hydroxide

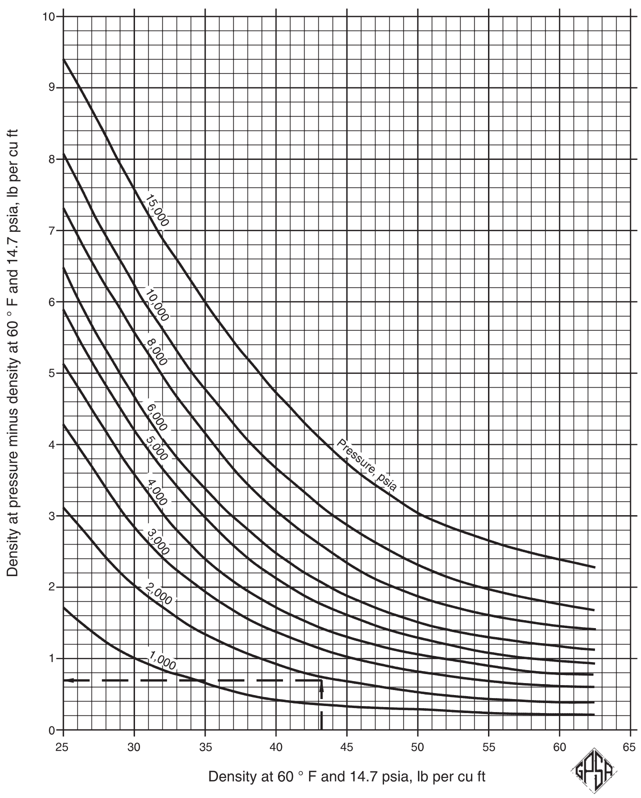
1. Low installation and operation costs.
2. Acts as a desiccant as well as removing the sulfur compounds.
3. Suitable for removing trace amounts of H_2S.

Molecular Sieve

1. Can handle large or small streams.
2. Reduces total sulfur content by removing hydrogen sulfide, mercaptans, and partially removing organic sulfur in the same adsorber vessel.

FIG. 23-15

Density Correction for Compressibility of Hydrocarbon Liquids



9. Correct the density at 60°F and pressure to the actual temperature using Fig. 23-17. Add the correction to the density from Step 8.

This procedure is not valid in the critical region. Mixtures at temperatures greater than 150 °F that contain more than 60 mol% methane or more than 80 mol% CO_2 are problem areas. Outside the near-critical region, calculated densities usually are within 5% of experimental data[35] and errors are rarely greater than 8%. The best accuracy occurs for mixtures containing mostly C_5+ with relatively small amounts of dissolved gaseous components (errors are usually less than 3%). Note that densities of C_2+, C_3+, CO_2+, or C_4+ mixtures can be calculated by this procedure at various temperatures and pressures, and that the gaseous components need not be present.

Example 23-3 — Fig. 23-16 illustrates the procedure outlined above.

$$\text{Density of } C_3+ = \frac{\text{Wt of } C_3+}{\text{Vol of } C_3+} = \frac{44.836\ \text{lbm}}{1.0128\ \text{ft}^3}$$

$$= 44.275\ \text{lbm/ft}^3$$

$$\text{Wt \% } C_2 \text{ in } C_2+ = \left(\frac{100\ (0.567)}{0.567 + 44.836}\right) = 1.25\%$$

$$\text{Density of } CO_2+ = \frac{45.403 + 17.485}{(45.403/44.0) + 0.3427} = 45.75\ \text{lbm/ft}^3$$

Wt% of CH_4 in Total = 100(3.352)/66.241 = 5.1%

Pseudo-density of mixture at 60°F and 14.7 psia from Fig. 23-14 = 42.9 1bm/ft^3

Pressure correction to 1760 psia from Fig. 23-15 = +0.7

Density at 60°F and 1760 psia = 42.9 + 0.7 = 43.6 lb/ft^3

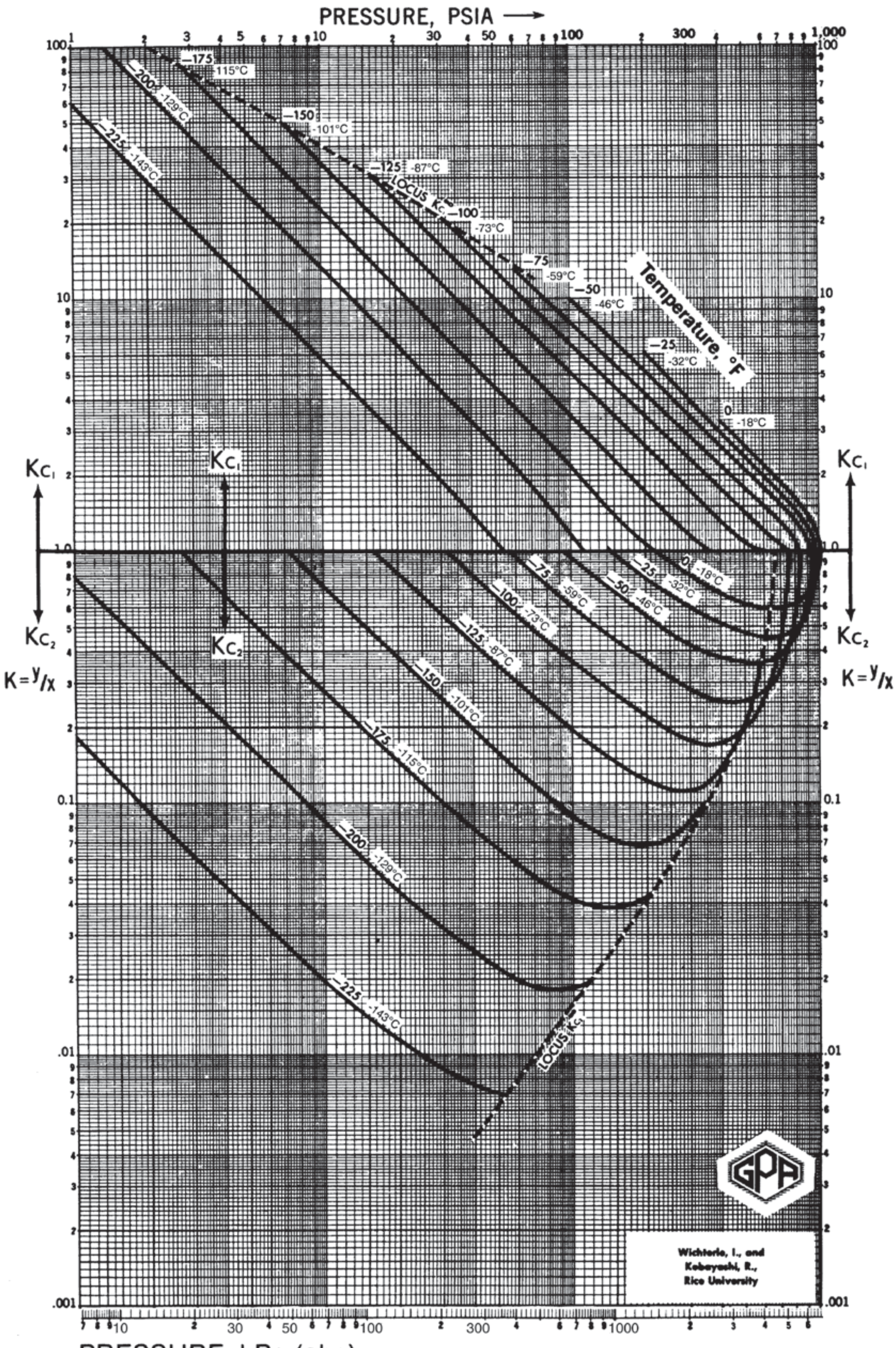
Temperature correction to 120 °F from Fig. 23-17 = -1.8

Density at 120 °F and 1760 psia = 43.6- 1.8 = 41.8 lb/ft^3

FIG. 23-16

Calculation of Liquid Density of a Mixture at 120°F and 1760 psia

(1)	(2)	(3)	(4)=(2)•(3)	(5)	(6)=(4)/(5)
Component	**Mole Fraction**	**Molecular Weight**	**Weight, lb**	**Density (60°F), lb/cu ft**	**Volume, cu ft**
Methane	0.20896	16.043	3.352	–	–
Carbon Dioxide	0.39730	44.010	17.485	51.016	0.3427
Ethane	0.01886	30.070	0.567	–	–
Propane	0.02387	44.097	1.053	31.619	0.0333
n-Butane	0.03586	58.123	2.084	36.423	0.0572
n-Pentane	0.02447	72.150	1.766	39.360	0.0449
n-Hexane	0.01844	86.177	1.589	41.400	0.0384
n-Heptane	0.02983	100.204	2.989	42.920	0.0696
n-Octane	0.02995	114.231	3.421	44.090	0.0776
n-Decane	0.18208	142.285	25.907	45.790	0.5658
n-Tetradecane	0.03038	198.394	6.027	47.815	0.1260
Total	1.00000		66.240		



METHANE-ETHANE BINARY