

GPSA ENGINEERING DATABOOK ERRATA (2004 FPS Edition)

PAGE	DESCRIPTION
3-10	Correct typo
7-15	Change V_1 to U and gl to W
13-9	Change Eq. 13-16a
14-13	Change reference to Fig. 14-26
20-21	Change wording
20-46	Change Fig. 20-87 reference
23-11	Change Fig. 23-3 reference
23-31	Change Eq. 23-22

FIG. 3-10

Orifice Meter Installation Requirements without a Flow Conditioner

Minimum straight unobstructed meter tube length from the upstream and downstream side of the orifice plate (in multiples of published internal pipe diameter, D_1)

Diameter ratio β	a. Single 90° elbow. b. Two 90° elbows in the same plane with SEP > 30D ₁ . c. Two 90° elbows in perpendicular planes with SEP > 15D ₁ .	Two 90° elbows in the same plane "SEP" configuration SEP ≤ 10D ₁ .	Two 90° elbows in perpendicular planes, 5D ₁ ≤ SEP ≤ 15D ₁ .	Two 90° elbows in perpendicular planes, 5D ₁ ≤ SEP ≤ 15D ₁ .	Two 90° elbows in perpendicular planes, 5D ₁ ≤ SEP ≤ 15D ₁ .	Single 90° Tee used as an elbow but not as a header element	a. Single 45° elbow. b. Two 45° elbows in the same "SEP" configuration SEP ≥ 22D ₁ .	Gate valve at least 50% open	Concentric reducer	Any other configuration (catch all category)*	Downstream meter tube length
	UL	UL	UL	UL	UL	UL	UL	UL	UL	UL	DL
	6	10	10	10	50	19	30	17	6	70	2.8
0.30	11	10	12	32	50	32	30	19	6	108	3.0
0.40	16	10	13	44	50	44	30	21	6	145	3.2
0.50	30	30	18	44	95	44	30	25	7	145	3.5
0.60	44	44	30	44	95	44	30	30	9	145	3.9
0.67	44	44	44	44	95	44	44	35	11	145	4.2
0.75	44	44	44	44	95	44	44	44	13	145	4.5
Recommended length for maximum range $\beta \leq 0.75$	44	44	44	44	95	44	44	44	13	145	4.5

UL – Minimum meter tube length upstream of the orifice plate in internal pipe diameter, D_1 (See Figure 3-9). Straight length shall be measured from the downstream end of the curved portion of the nearest (or only) elbow or of the tee or the downstream end of the conical portion of reducer or expander.

DL – Minimum downstream meter tube length in internal pipe diameters, D_1 (See Figure 3-9).

SEP – Separation distance between piping elements in internal pipe diameter, D_1 , measured from the downstream end of the curved portion of the upstream elbow to the upstream end of the curved portion of the downstream elbow.

Note : The tolerance on specified lengths for UL and DL is $\pm 0.25D_1$.

* – This installation exhibits strong effect of Reynolds number and pipe roughness on the recommended length due to rate of decay of swirl. The present recommendations have been developed for high Reynolds numbers and smooth pipes to capture the worst case.

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Removal of very small droplets may require the use of specialized internals or the application of electrostatic fields to promote coalescence.

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 type of liquid-liquid separators. The following principles of design for liquid-liquid separation apply equally for horizontal or vertical separators. Horizontal vessels have some advantage over verticals 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 that may prevent two liquid phases from separating due to differences in specific gravity:

- If droplet particles are so small that they may be suspended by Brownian movement. This is defined as a random motion that 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. 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 Stoke's 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), as noted in Eq 7-3. The liquid-liquid separation capacity of separators may be determined from Equations 7-13 and 7-14, which were derived from Equation 7-3.⁹ Values of C^* are found in Fig. 7-23.

Vertical vessels:

$$W_{cl} = C^* \left(\frac{S_{hl} - S_{ll}}{\mu} \right) (0.785) D_v^2 \quad \text{Eq 7-13}$$

Horizontal vessel:

$$W_{cl} = C^* \left(\frac{S_{hl} - S_{ll}}{\mu} \right) L_1 H_1 \quad \text{Eq 7-14}$$

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 determined from Eq 7-12 using the retention time given in Fig. 7-22.

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

FIG. 7-23
Values of C^* Used in Equations 7-13 and 7-14

Emulsion Characteristic	Droplet Diameter, microns	Constant, ¹⁰ C^*
Free Liquids	200	1,100
Loose Emulsion	150	619
Moderate Emulsion	100	275
Tight Emulsion	60	99

Example 7-3—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 =	1,000 psig
Water specific gravity =	1.01
Condensate viscosity =	0.55 cp @ 80°F
Condensate specific gravity for 55° API =	0.76

From Eq 7-13, $W_{cl} = C^*[(S_{hl} - S_{ll}) / \mu] \times 0.785 \times D_v^2$

From Fig. 7-23 for free liquids with water particle diameter = 200 microns, $C^* = 1,100$

600 bbl/day = $1,100 \times [1.01 - 0.76] / 0.55 \times 0.785 \times D_v^2$

$D_v^2 = 660 / 392.5 = 1.53 \text{ ft}^2$

$D_v = 1.24 \text{ feet}$

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

Using the alternate method of design based on retention time as shown in Equation 7-12 should give:

$U = W(t) / 1440$

From Fig. 7-22, use 3 minutes retention time

$U = 650 \times 3 / 1440 = 1.35 \text{ bbl}$

$U = 1.35 \times 42 = 56.7 \text{ gallons}$

Assuming a 20-inch diameter and 1,480 psig working pressure, a vessel would be made from a 1.031-inch 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:

Shell height = $U / \text{vol/ft} = 56.7 / 13.1 = 4.3 \text{ feet}$

This would require a 20 inch OD x 10 foot 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 2,000 gal/day/ft².

lation. The closest k value column may be safely used without a second interpolation.

Volumetric efficiencies for "high speed" separable compressors in the past have tended to be slightly lower than estimated from Eq 13-14. Recent information suggests that this modification is not necessary for all models of high speed compressors.

In evaluating efficiency, horsepower, volumetric efficiency, etc., the user should consider past experience with different speeds and models. Larger valve area for a given swept volume will generally lead to higher compression efficiencies.

Equivalent Capacity

The net capacity for a compressor, in cubic feet per day @ 14.4 psia and suction temperature, may be calculated by Eq. 13-16a which is shown in dimensioned form:

$$\text{MMcfd} = \frac{\text{PD} \frac{\text{ft}^3}{\text{min}} \cdot 1440 \frac{\text{min}}{\text{d}} \cdot \frac{\text{VE}\%}{100} \cdot P_s \frac{\text{lb}}{\text{in}^2} \cdot 10^{-6} \frac{\text{MMft}^3}{\text{ft}^3} \cdot Z_{14.4}}{14.4 \frac{\text{lb}}{\text{in}^2} \cdot Z_s} \quad \text{Eq 13-16a}$$

which can be simplified to Eq. 13-16b when $Z_{14.4}$ is assumed to equal 1.0.

$$\text{MMcfd} = \frac{\text{PD} \cdot \text{VE} \cdot P_s \cdot 10^{-6}}{Z_s} \quad \text{Eq 13-16b}$$

For example, a compressor with 200 cu ft/min piston displacement, a volumetric efficiency of 80%, a suction pressure of 75 psia, and suction compressibility of 0.9 would have a capacity of 1.33 MMcfd at 14.4 psia and suction temperature. If compressibility is not used as a divisor in calculating cu ft/min, then the statement "not corrected for compressibility" should be added.

In many instances the gas sales contract or regulation will specify some other measurement standard for gas volume. To convert volumes calculated using Equation 13-16 (i.e. at 14.4 psia and suction temperature) to a P_L and T_L basis, Eq 13-17 would be used:

$$\text{MMscfd at } P_L, T_L = (\text{MMcfd from Eq 13-16}) \left(\frac{14.4}{P_L} \right) \left(\frac{T_L}{T_s} \right) \left(\frac{Z_r}{Z_s} \right) \quad \text{Eq 13-17}$$

Discharge Temperature

The temperature of the gas discharged from the cylinder can be estimated from Eq 13-18, which is commonly used but not recommended. (Note: the temperatures are in absolute units, °R or K.) Eqs 13-31 and 13-32 give better results.

$$T_d = T_s (r^{(k-1)/k}) \quad \text{Eq 13-18}$$

The discharge temperature determined from Eq 13-18 is the theoretical value. While it neglects heat from friction, irreversibility effects, etc., and may be somewhat low, the values obtained from this equation will be reasonable field estimates.

Rod Loading

Each compressor frame has definite limitations as to maximum speed and load-carrying capacity. The load-carrying capacity of a compressor frame involves two primary considerations: horsepower and rod loading.

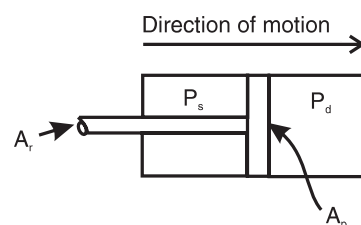
The horsepower rating of a compressor frame is the measure of the ability of the supporting structure and crankshaft to withstand torque (turning force) and the ability of the bearings to dissipate frictional heat. Rod loads are established to limit the static and inertial loads on the crankshaft, connecting rod, frame, piston rod, bolting, and projected bearing surfaces.

Good design dictates a reversal of rod loading during each stroke. Non-reversal of the loading results in failure to allow bearing surfaces to part and permit entrance of sufficient lubricant. The result will be premature bearing wear or failure.

Rod loadings may be calculated by the use of Eqs 13-19 and 13-20.

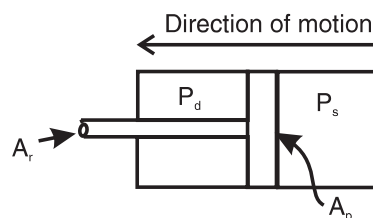
$$\begin{aligned} \text{Load in compression} &= P_d A_p - P_s (A_p - A_r) \\ &= (P_d - P_s) A_p + P_s A_r \end{aligned} \quad \text{Eq 13-19}$$

Rod in Compression



$$\begin{aligned} \text{Load in tension} &= P_d (A_p - A_r) - P_s A_p \\ &= (P_d - P_s) A_p - P_d A_r \end{aligned} \quad \text{Eq 13-20}$$

Rod in Tension



Using Eqs. 13-19 and 13-20, a plus value for the load in both compression and tension indicates a reversal of loads based on gas pressure only. Inertial effects will tend to increase the degree of reversal.

The true rod loads would be those calculated using internal cylinder pressures after allowance for valve losses. Normally, the operator will know only line pressures, and because of this, manufacturers generally rate their compressors based on line-pressure calculations.

A further refinement in the rod-loading calculation would be to include inertial forces. While the manufacturer will consider inertial forces when rating compressors, useful data on this point is seldom available in the field. Except in special cases, inertial forces are ignored.

A tail-rod cylinder would require consideration of rod cross-section area on both sides of the piston instead of on only one side of the piston, as in Eqs 13-19 and 13-20.

changer to accomplish an economizing step. Fig. 14-18 shows an example economizer using a heat exchanger. The heat exchanger is a chiller which uses some of the condensed refrigerant to subcool the balance of the condensed refrigerant stream. The refrigerant used for the chilling is then fed to the interstage (or second stage) of the refrigeration compressor. The subcooled refrigerant is then used for process chillers. The subcooled refrigerant produces less unusable vapor when flashed to suction drum conditions than a refrigerant stream that is not subcooled. Thus the use of the heat exchanger effectively shifts vapor from the low stage of compression to the high stage, thus saving power. The resultant process impact is very similar to the flash economization previously discussed.

Design and Operating Considerations

The following are some of the important parameters that should be considered while designing any refrigeration system to provide a safe, reliable, and economical operation.

Oil Removal — Oil removal requirements from evaporators are related to the type of the refrigerant, lubricant, evaporator, and compressor used in the refrigeration cycle. Fig. 14-27 illustrates the application of an oil reclaimer in a propane refrigerant cycle. In order to remove oil from the refrigerant, a slip stream of refrigerant from the bottom of the chiller is drained into the reclaimer where hot propane refrigerant from the compressor discharge is used to evaporate the refrigerant into the compressor suction. The oil is removed from the bottom of the reclaimer. Similar arrangements can be utilized for other hydrocarbon and ammonia refrigerants. Operation may be designed for either manual or automatic.

Where halocarbon refrigerants and/or synthetic lubricants are employed, it is imperative that the oil reclaimer system be approved by the compressor manufacturer.

Liquid Surge and Storage — All refrigeration systems should have a liquid surge and storage vessel, commonly called a receiver. A surge vessel is required on all systems where the operating charge in the evaporator(s) and the condenser(s) varies due to variable load conditions. In addition to accommodating a varying refrigerant charge, the receiver drains the condenser(s) of liquid so that the effective condensing surface is not reduced by liquid backing up. The refrigerant charge in a receiver may vary over a wide range, from a minimum at full load to a maximum at no load.

Systems with inadequate surge vessels often cause problems as they lose the liquid seal due to load variations that always occur. Surge vessels or receivers are relatively inexpensive and when sizing them, consideration should be given to: (1) a volume equal to 100% of the system inventory at 80% full level, and (2) the availability and quantity of refrigerant makeup.

Vacuum Systems — Refrigeration systems can operate with a suction pressure below atmospheric pressure. These vacuum systems require special considerations:

- Where hydrocarbons are used with reciprocating compressors (which employ rod “packing”), air can enter the compressor and possibly form a hazardous mixture. Extreme care should be taken where such systems are used. These systems must have a manual or automatic purge system. Double acting packing should be employed.
- Where halocarbons such as R-11, R-114, R-113, and other low pressure, high volume refrigerants are employed with centrifugal compressors, the deep vacuums may “draw” air and moisture through flanges, seals, etc. This

water-oxygen combination in the presence of halocarbons forms acid and causes “crevice corrosion” of the tubes along with some other problems. A positive purge system must be employed and frequent monitoring of the moisture content in the refrigerant is suggested.

- High pressure halocarbons R-12, R-22, and others are employed in vacuum systems with reciprocating, centrifugal, and screw compressors. These systems will suffer from the same corrosion problems as defined above, but to a lesser extent. However, since they are generally

FIG. 14-18
Heat Exchanger Economizing

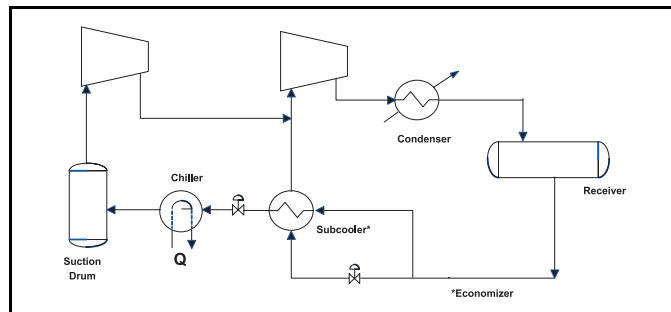
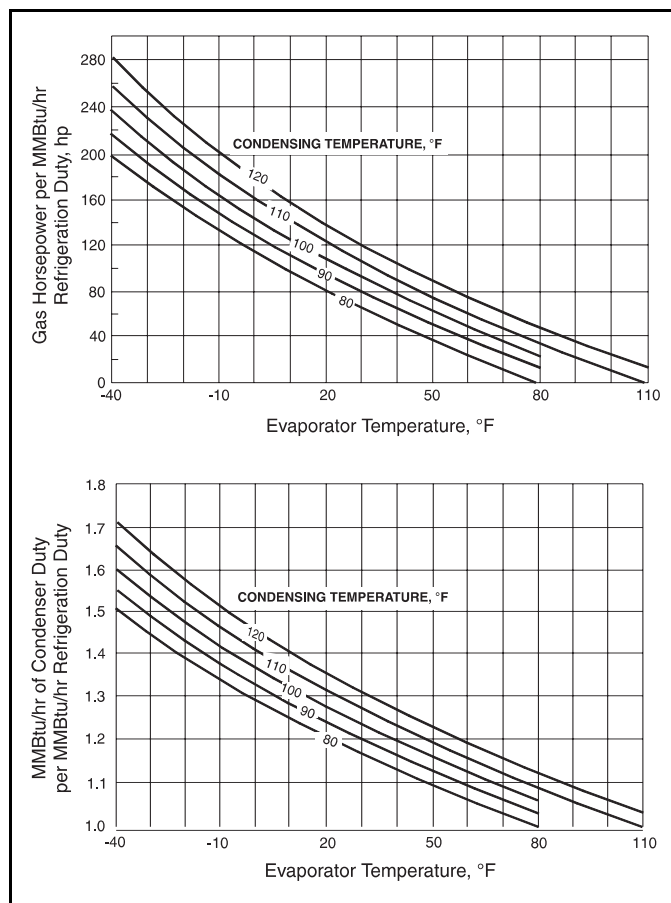


FIG. 14-19
Gas Horsepower and Condenser Duty for Two Stage R-22 Refrigeration



point depression is plotted against the weight percent methanol in Fig. 20-59.

Maddox *et al.*²⁷ presents a method of estimating the required inhibitor concentration for both methanol and EG. The method is iterative but converges easily after a few iterations.

Figs. 20-60 thru 20-64 provide a comparison of various inhibitor correlations with experimental data.^{28,29,30} Experimental data at very high inhibitor concentrations is limited.

Once the required inhibitor concentration has been calculated, the mass of inhibitor solution required in the water phase may be calculated from Eq 20-8

$$m_I = \frac{X_R \cdot m_{H_2O}}{X_L - X_R} \quad \text{Eq 20-8}$$

The amount of inhibitor to be injected not only must be sufficient to prevent freezing of the inhibitor water phase, but also must be sufficient to provide for the equilibrium vapor phase content of the inhibitor and the solubility of the inhibitor in any liquid hydrocarbon. The vapor pressure of methanol is high enough that significant quantities will vaporize. Methanol vaporization losses may be estimated from Fig. 20-65.³¹ Fig. 20-65 is extrapolated above 700 psia. Recent studies indicate Fig. 20-65 may underestimate vapor phase methanol losses at higher pressures. Glycol vaporization losses are generally very small and are typically ignored in calculations.

Inhibitor losses to the hydrocarbon liquid phase are more difficult to predict. Solubility is a strong function of both the water phase and hydrocarbon phase compositions. Fig. 20-66 presents experimental data^{32,33,34} showing solubility of methanol in a paraffinic hydrocarbon liquid as a function of temperature and methanol concentration. Methanol solubility in naphthenic hydrocarbons is slightly less than paraffinic, but solubility in aromatic hydrocarbons may be four to six times higher than in paraffins.

Solubility of EG in the liquid hydrocarbon phase is extremely small.²⁹ A solubility of 0.3 lb per 1000 gal. (U.S.) of NGL is often used for design purposes. However, entrainment and other physical losses may result in total losses significantly higher than this.

Example 20-10—100 MMscf/d of natural gas leaves an offshore platform at 100°F and 1200 psia. The gas comes onshore at 40°F and 900 psia. The hydrate temperature of the gas is 65°F. Associated condensate production is 10 Bbl/MMscf. The condensate has an API gravity of 50 and a MW of 140. Calculate the amount of methanol and 80 wt% EG inhibitor required to prevent hydrate formation in the pipeline.

Solution Steps:

Methanol

- Calculate the amount of water condensed per day
from Fig. 20-4, $W_{in} = 53.0 \text{ lb / MMscf}$
 $W_{out} = 9.5 \text{ lb / MMscf}$
 $\Delta W = 43.5 \text{ lb / MMscf}$
Water condensed = $(100)(53 - 9.5) = 4350 \text{ lb/day}$
- Calculate required methanol inhibitor concentration from Eq 20-5 and 20-7 (with Fig. 20-59).
 $d = 25^\circ\text{F}$ $MW = 32$
Solving for X_I ,
 $X_I = 0.255$, Eq 20-5
 $X_I = 0.275$, Eq. 20-7 (use this value in subsequent calculations)

- Calculate mass rate of inhibitor solution in water phase from Eq. 20-8 (assume 100% methanol is injected)

$$m_I = \frac{X_R \cdot m_{H_2O}}{X_L - X_R} = \frac{(0.275)(4350)}{(1 - 0.275)} = 1650 \text{ lb/day}$$

- Estimate vaporization losses from Fig. 20-65.

$$@ 40^\circ\text{F and 900 psia, losses} = 1.05 \frac{\text{lb/MMscf}}{\text{wt\% MeOH}}$$

$$\text{daily losses} = (1.05)(100)(27.5) = 2890 \text{ lb/day}$$

- Estimate losses to hydrocarbon liquid phase from Fig. 20-66.

$$@ 40^\circ\text{F and 27.5 wt\% MeOH, } x_{\text{MeOH}} \approx 0.2 \text{ mol\%}$$

lb • mols of condensate per day—

$$\left(\frac{100 \text{ MMscf}}{\text{day}} \right) \left(\frac{10 \text{ bbl}}{\text{MMscf}} \right) \left(\frac{(350)(0.78) \text{ lb}}{\text{bbl}} \right) \left(\frac{1 \text{ lb} \cdot \text{mol}}{140 \text{ lb}} \right)$$

$$= 1950 \text{ lb-mols/day}$$

$$\text{lb} \cdot \text{mols methanol} = (1950)(0.002)$$

$$= 3.9 \text{ lb} \cdot \text{mols/day}$$

$$\text{lb methanol} = (3.9)(32) = 125 \text{ lb/day}$$

$$\text{Total methanol injection rate} = 1650 + 2890 + 125$$

$$= 4665 \text{ lb/day}$$

Methanol left in the gas phase can be recovered by condensation with the remaining water in downstream chilling processes. Likewise, the methanol in the condensate phase can be recovered by water by downstream water washing.

80 wt% EG

- Calculate required inhibitor concentration from Eq 20-6.

$$d = 25^\circ\text{F} \quad MW = 62 \quad K_H = 2335$$

$$\text{Solving for } X_I, \quad X_I = 0.40$$

- Calculate mass rate of inhibitor in water phase from Eq. 20-8.

$$\dot{m}_I = \frac{(0.40)(4350)}{(0.8 - 0.40)} = 4350 \text{ lb / day}$$

Vaporization and liquid hydrocarbon losses are negligible.

Inhibitor losses represent a significant operating cost and can cause problems in downstream process units. Efficient inhibitor separation should be provided.

Low Dosage Hydrate Inhibitors (LDHIs)—LDHIs can provide significant benefits compared to thermodynamic inhibitors including:

- Significantly lower inhibitor concentrations and therefore dosage rates. Concentrations range from 0.1 to 1.0 weight percent polymer in the free water phase, whereas alcohols can be as high as 50%
- Lower inhibitor loss caused by evaporation, particularly compared to methanol
- Reduced capital expenses through decreased chemical storage and injection rate requirements; and no need for regeneration because the chemicals are not currently recovered. These are especially appropriate for offshore where weight and space are critical to costs
- Reduced operating expenses in many cases through decreased chemical consumption and delivery frequency
- Increased production rates, where inhibitor injection capacity or flowline capacity is limited

Gas Stripping

One simple method of dehydrating liquid hydrocarbons is counter current stripping with a dry gas. This method is currently used to dry condensate produced offshore prior to export from the production platform. The contactor is usually trayed. Stripping gas rates depend on the condensate rate, the amount of entrained water in the condensate, stripper temperature and pressure.

Advantages of this process are simplicity and low capital cost. Disadvantages include the requirement for a dry natural gas stream, and the coincidental stripping of some of the volatile hydrocarbons from the condensate. The stripping gas may be recycled to the gas dehydration unit or it can be used as fuel gas.

SOLID DESICCANT DEHYDRATION

Several solid desiccant processes are available to dry liquid hydrocarbons.

Liquid velocity is usually 3-5 ft/minute through solid desiccant beds with a minimum travel of at least 5 feet to ensure good distribution. Direction of flow can be upflow or downflow in the adsorption cycle.

Special care must be taken in designing the bed supports in the liquid dehydrator vessels to prevent desiccant loss, desiccant damage, and to ensure proper distribution. Layers of ceramic balls are installed in decreasing size from the support screen. The support ball sizes may vary with the type and size of solid desiccant used but the layers of support balls should never be graduated in size more than twice the diameter of the balls being supported.

The regeneration of solid desiccant beds is very similar to gas dehydrators with the following exceptions:

- Liquid draining and filling time must be allowed.
- Pressuring and depressuring must be done carefully to avoid bed movement.
- Adequate bed cooling is required before liquid re-entry to minimize flashing.

It is important to prevent movement of the bed particles to prevent attrition that would require premature replacement. Also, desiccant dust particles can cause downstream plugging, equipment damage, and excessive filter maintenance. Liquid and vapor velocities must be controlled carefully and flashing of liquids or accelerated blow-down rates that would "lift" or "float" all or portions of the bed should be avoided.

Desiccant bed life can be extended by doing several or all of the following activities:

- Prevent the desiccant particles from moving.
- Keep contaminants out of the dehydrating portion of the bed by upstream conditioning or by providing a sacrificial layer of less expensive desiccant to act as a catcher of any compounds such as amine, glycol or oil.
- Prevent overheating the bed to reduce the formation of carbon during the regeneration cycle.
- Analyze the heating/cooling regeneration temperature cycles to minimize the time the bed is at elevated temperatures. This will also minimize energy requirements.

A typical heating/cooling regeneration temperature cycle plot is shown in Fig. 20-86, with a description of the stage activities. There are typically four (4) distinct stages in a normal cycle:

Stage 1—First bed-heating stage

Stage 2—Desorption stage

Stage 3—Second bed-heating stage

Stage 4—Bed-cooling stage

For a period of time after the heat source is introduced into a desiccant bed being dehydrated, the bed must be heated to a temperature where the water will start to be desorbed (Stage 1). As the water is desorbed (Stage 2), the bed temperature will usually rise only a few degrees because the regeneration gas heat is utilized to provide the heat of vaporization of the water being removed. The completion of the water desorption stage is characterized by a rapid increase in bed temperature measured as the outlet temperature. At this point the heating may be discontinued while bed heating will continue from residual heat in the heating cycle (Stage 3). As the unheated regeneration gas stream continues to pass through the bed, the bed will be cooled (Stage 4).

At near ambient pressures, regeneration of silica gel and alumina can be accomplished at 300°F. Molecular sieve requires 500-550°F to maintain the low dewpoint potential, and the higher temperatures may increase desiccant life by providing more complete removal of adsorbed hydrocarbons.

Capacity and performance data for new solid desiccants are usually presented based on a static test. Under operating conditions (dynamic) the performance data may be significantly different. Typically the effective capacity at operating conditions is about one-half of the capacity at equilibrium (static) conditions for most solid desiccants. This operating characteristic must be considered when designing a dehydration system and can be influenced by careful design and good control of operating parameters such as temperatures, contaminant levels, regeneration cycles, and desiccant selection. Solid desiccant manufacturers should be consulted for the most current product information and design criteria.

Molecular Sieve

Molecular sieve is not normally used for liquid dehydration because the required level of water removal is usually moderate and the cost of molecular sieve is considerably more than other types of suitable desiccants, such as activated alumina. However, in extreme cases where the moisture content of the liquid must be kept at an unusually low concentration, molecular sieve should be considered.

Molecular sieve may be used for removing other undesirable compounds, such as H₂S, COS, mercaptans, etc., from liquid streams. Dehydration may be a secondary benefit of using this type of treating method.

Refer to the discussion of molecular sieve for gas dehydration elsewhere in this Section for more information.

Activated Alumina

There are several types of alumina available for use as a solid desiccant. Alumina is widely used for drying liquid product streams following gas processing, treating, or fractionation. Most alumina desiccants will produce a dewpoint below -100°F if applied properly. Alumina tends to adsorb heavy hydrocarbons which are difficult to remove during regeneration. Alumina is alkaline and is subject to reaction with mineral acids which are found in some well treating fluids.

Example 23-1 — Pure component properties

Using Fig. 24-27, the P- H diagram for propane, calculate the density of propane vapor at 200°F and 100 psia.

Solution Steps

On the P-H diagram at the intersection of the $T = 200^{\circ}\text{F}$, $P = 100$ psia lines read $v = 1.5 \text{ ft}^3/\text{lbm}$. Then: $\rho = 1/1.5 = 0.667 \text{ lbm ft}^{-3}$

Using the EZ*THERMO⁹⁰ version of the SRK⁹¹ equation of state, ρ is $0.662 \text{ lb}/\text{ft}^3$, from which $v = (1/0.662) = 1.51 \text{ ft}^3/\text{lbm}$.

For propane at 200°F and 100 psia using data from Fig. 24-27.

$$Z = \frac{MW \cdot P}{R T_p} = \frac{(44.10)(100)}{(10.73)(458.67 + 200)(0.667)} = 0.936$$

The SRK calculation gives $\rho = 0.662 \text{ lb}/\text{ft}^3$, and $Z = 0.941$.

Gas Mixtures

GPA Standard 2172, "Calculation of Gross H Value, Relative Density and Compressibility Factor for N Gas Mixtures from Compositional Analysis" contains additional information regarding the calculation of Z-factors for mixtures at pressures below 150 psia.

Minor Amounts of Non-hydrocarbons — Fig 23-4¹ shows Z-factors for typical sweet natural gases. Using Z-factors from Fig. 23-4 should yield mixture volumes (densities) within 2% to 3% of the true values for reduced temperatures from slightly greater than 1.0 to the limits of the chart for both temperature and pressure. The chart has been prepared from data for binary mixtures of methane with ethane, propane and butane and data for natural gas mixtures. All mixtures have average molecular weights less than 40, and all gases contain less than 10% nitrogen and less than 2% combined hydrogen sulfide and carbon dioxide. Fig. 23-4 applies for temperatures 20 °F or more above saturation up to pressures of 10,000 psia.

Appreciable Amount of Non-Hydrocarbons — Figure 23-4 does not apply for gases or vapors with more than 2% H₂S and/or CO₂ or more than 20% nitrogen. Use other methods for

vapors that have compositions atypical of natural gases mixtures or for mixtures containing significant amounts of water and/or acid gases, and for all mixtures as saturated fluids, other methods should be employed.

Figure 23-4 provides reasonably accurate gas Z-factors for natural gases with high nitrogen content, up to 50% (or even higher) when using the molar average pseudo-criticals from Eqs 23-3a and 23-4a. The same approach applies to gas condensate fluids containing appreciable amounts of heptanes and heavier components. Critical temperatures and pressures for heptane and heavier fractions can be estimated from molecular weight and relative density, or average boiling point and relative density, using correlations presented in this section.

Figs. 23-5, 23-6 and 23-7 provide Z-factors for low molecular weight natural gases. These figures cover a wide range of molecular weights (15.95 to 26.10), temperatures (-100 to 1000 °F) and pressures (up to 5,000 psia). For gases with molecular weights between those shown in Figs. 23-5 through 23-7, linear interpolation between adjacent charts is sufficient to compute the Z-factors.

In general, Z-factors for gases with less than 5% noncondensable non-hydrocarbons, such as nitrogen, carbon dioxide, and hydrogen sulfide, result with less than 5% error. When the molecular weight is above 20 and the Z-factor is below 0.6, errors as large as 10% may occur.

Effect of Acid Gas Content — Natural gases containing H₂S and/or CO₂ exhibit different Z-factor behavior than do sweet gases. Wichert and Aziz³ present a calculation procedure to account for these differences. Their method uses the standard gas Z-factor chart (Fig. 23-4) and provides accurate sour gas Z-factors that contain as much as 85% total acid gas. Wichert and Aziz define a "critical temperature adjustment factor," ϵ , that is a function of the concentrations of CO₂ and H₂S in the sour gas. This correction factor adjusts the pseudo-critical temperature and pressure of the sour gas according to the equations:

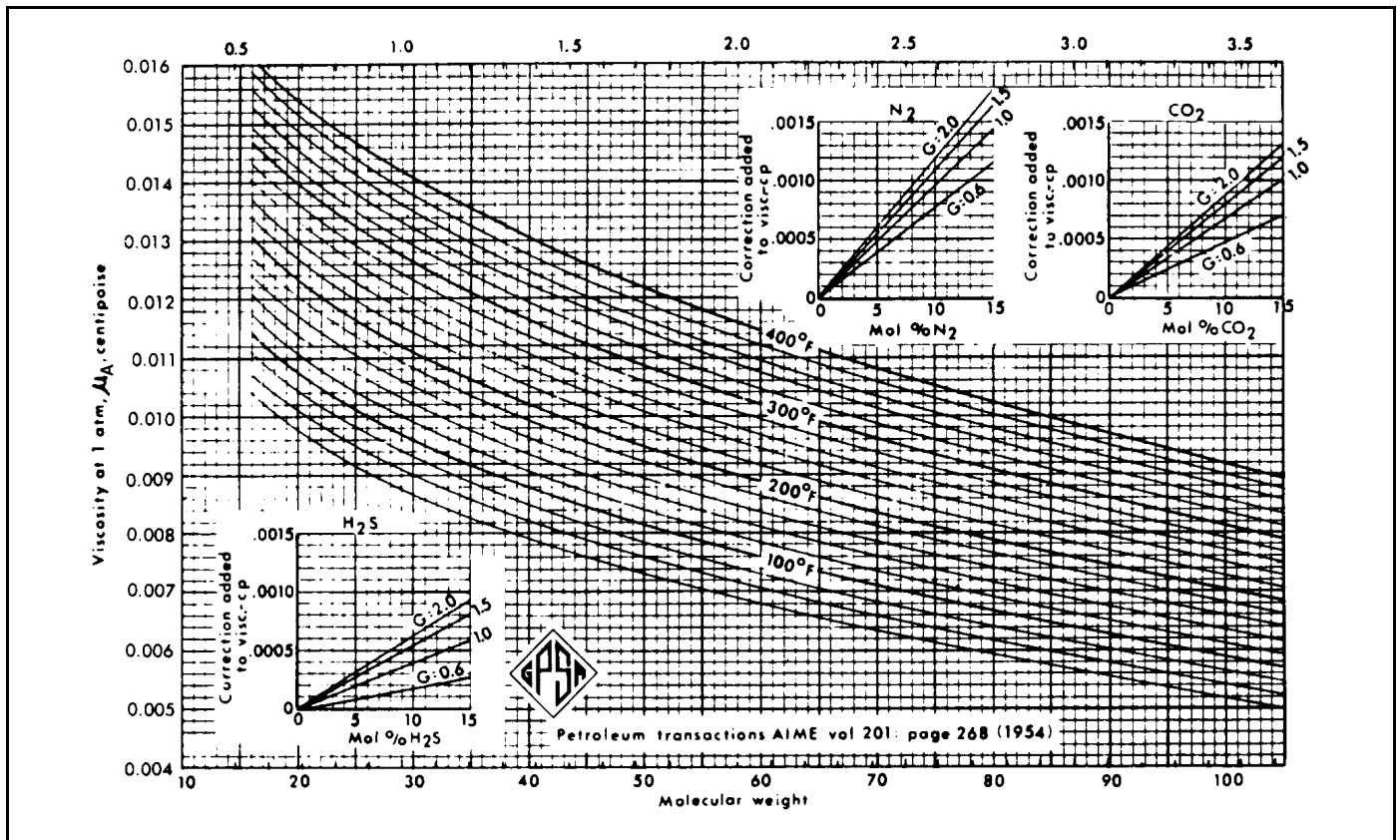
$$T_c' = T_c - \epsilon \quad \text{Eq 23-6}$$

FIG. 23-3**Calculation of Pseudo-critical Temperature and Pressure for a Natural Gas Mixture**

Component	Mole Fraction, y_i	Component Critical Temperature, T_{ci} , °R	Pseudocritical Temperature, T_{pc} , °R	Component Critical Pressure, P_{ci} , psia	Pseudocritical Pressure, P_{pc} , psia	Component Molecular Weight, MW	Mixture Molecular Weight, $y_i \cdot MW$
CH ₄	0.8319	343.0	285.3	666.4	554.4	16.043	13.346
C ₂ H ₆	0.0848	549.6	46.6	706.5	59.9	30.070	2.550
C ₃ H ₈	0.0437	665.7	29.1	616.0	26.9	44.097	1.927
iC ₄ H ₁₀	0.0076	734.1	5.58	527.9	4.01	58.123	0.442
nC ₄ H ₁₀	0.0168	765.3	12.86	550.6	9.25	58.123	0.976
iC ₅ H ₁₂	0.0057	828.8	4.72	490.4	2.80	72.150	0.411
nC ₅ H ₁₂	0.0032	845.5	2.71	488.6	1.56	72.150	0.231
nC ₆ H ₁₄	0.0063	913.3	5.75	436.9	2.75	86.177	0.543
			$T_{pc} = 392.62$		$P_{pc} = 661.57$		$MW_m = 20.426$
$G = 20.426/28.9625 = 0.705$							

FIG. 23-22

Viscosity of Paraffin Hydrocarbon Gases at One Atmosphere



from Fig. 23-23. Figure 23-24 is better when the reduced temperature is greater than 1.0. Figure 23-28 gives the viscosity of hydrocarbon liquids containing dissolved gases. Note that Fig. 1-7 gives conversion factors for viscosity.

Calculation of Gas Mixture Viscosity

Example 23-8 — Determine the viscosity of a gas at 1,000 psia and 100 °F with the properties: MW = 22, $T_c = 409^\circ\text{R}$, $P_c = 665$ psia.

Solution Steps

$$G^{id} = 22/28.9625 = 0.760$$

From Fig. 23-22 at 100 °F:

$$\mu_A = 0.0105 \text{ centipoise}$$

Then:

$$T_r = \frac{100 + 459.67}{409} = 1.37 \quad P_r = \frac{1000}{665} = 1.50$$

Note: If the composition of the gas is available, calculate the pseudo-critical temperatures and pressures as outlined in this section.

Because $T_r > 1.0$, Fig. 23-24 is better than Fig. 23-23 to obtain the correction for elevated pressure to the viscosity at one atmosphere.

From Fig. 23-24:

$$\mu/\mu_A = 1.21$$

Hence, the viscosity at 1000 psia and 100°F is:

$$\mu = (1.21)(0.0105) = 0.0127 \text{ centipoise}$$

The method of Dean and Stiel⁴¹ is best for calculating the viscosity of a gaseous mixture with large amounts of non-hydrocarbons. This method is particularly useful for handling natural gas with high CO₂ content. Compared to 30 CO₂-N₂ mixtures, this method has an average deviation of 1.21% at pressures up to 3525 psia. The method uses a factor, ξ , defined as:

$$\xi = 5.4402 \left[\frac{T_{cm}^{1/6}}{P_{cm}^{2/3} \left(\sum_i y_i MW_i \right)^{1/2}} \right] \quad \text{Eq 23-20}$$

If $T_r > 1.5$, then

$$\xi\mu_A = 166.8(10^{-5})(0.1338 T_r - 0.0932)^{5/9} \quad \text{Eq 23-21}$$

If $T_r \leq 1.5$,

$$\xi\mu_A = 34.0(10^{-5})(T_r^{8/9}) \quad \text{Eq 23-22}$$

In either case, μ_A is $\xi\mu_A$ divided by ξ .

Equations 23-20 through 23-22 predict the viscosity of pure gases as well as mixtures. To apply the Dean and Stiel⁴¹ method to mixtures, calculate the pseudo-critical volumes, Z -