GPSA ENGINEERING DATABOOK ERRATA
(2004 SI Edition)

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wellhead
The assembly of fittings, valves, and controls located at the surface and connected to the flow lines, tubing, and casing of the well so as to control the flow from the reservoir.

wet gas
(1) A gas containing water, or a gas which has not been dehydrated. (2) A term synonymous with rich gas. Refer to definition of "rich gas".

Wobbe number
A number proportional to the heat input to a burner at constant pressure. In British practice, it is the gross heating value of a gas divided by the square root of its gravity. Widely used in Europe, together with a measured or calculated flame speed, to determine interchangeability of fuel gases.

Conversion Factors

In these tables, factors for conversion, including conversions to the International System of Units (SI), are based on ASTM Standard for Metric Practice, E380-91. The latest edition of this publication should be studied for more detail on the SI system, including definitions and symbols.

In calculating derived factors in the tables that follow, exact conversions were used, when available, rather than the 7-digit round-offs listed in ASTM E380 conversion tables. Derived factors given below are rounded to the same number of significant digits as the source factors.

In any conversion of fundamental measurement units, some confusion may result due to redefinition of units used in earlier tables. For example, in 1959 a small refinement was made in the definition of the yard, which changed its length from 3600/3937 meter (or 1 inch = 25.4000508 mm) to 0.9144 m exactly (or 1 inch = 25.4 mm exactly). The tables below are based on the new definition, but one should be aware that where U.S. land measurements are concerned, the old relationship applies. Refer to ASTM E380-91, note 13, for more detail.

Energy Units Conversion

Confusion may arise in the definition of units for heat or energy. In the tables below, the Btu (IT) and calorie (IT) are used. These are the heat units recommended by the International Conference on the Properties of Steam, as defined:

1 Btu (IT) = 1055.055 852 62 joule (exactly)
1 Calorie (IT) = 4.186 800 joule (exactly)

For information only, other definitions that may be used elsewhere:

1 Btu (Mean) = 1055.87 joule
1 Btu (39°F) = 1059.67 joule
1 Btu (60°F) = 1054.68 joule
1 Btu (Thermochemical) = 1054.350 joule
1 calorie (Mean) = 4.190 02 joule
1 calorie (15°C) = 4.185 80 joule
1 calorie (20°C) = 4.181 90 joule
1 calorie (Thermochemical) = 4.184 000 joule

The fundamental relationship between the Btu and the calorie:

$$\frac{\text{gram–pound relationship}}{\text{Fahrenheit–Celsius scale relationship}}$$

or: $$\text{Btu} \times \frac{453.592\ 29}{1.8} = \text{calorie (IT, mean, or other)}$$

1 therm = 100,000 Btu = 105.5056 x 10^6 J
= 105,505.6 kJ = 1.055 056 x 10^5 kJ.
(Btu denotes British Thermal Units)
(ref: Physical Properties of Natural Gases, Gas Unie, 1988 p. 23)

FIG. 1-2

Conversion Factor Tables

Velocity
(Length/unit of time)

ft/sec	ft/min	Miles/hr (U.S. Statute)	m/sec	m/min	km/hr
1	60	0.6818182	0.3048	18.288	1.09728
0.01666667	1	0.01136364	5.08 x 10^{-3}	0.3048	0.018288
1.466667	88	1	0.44704	26.8224	1.609344
3.280840	196.8504	2.236936	1	60	3.6
0.05468066	3.280840	0.03728227	0.016667	1	0.06
0.9113444	54.68066	0.6213712	0.2777778	16.66667	1

Energy

Ft-lbf	Kgf-meter	Btu (IT)	Kilo-calorie (IT)	Hp-hr	Kilowatt-hr	joule (J)
1	0.1382550	1.285068 x 10^{-3}	3.238316 x 10^{-4}	5.050505 x 10^{-7}	3.766161 x 10^{-7}	1.355818
7.233014	1	9.294911 x 10^{-3}	2.342278 x 10^{-3}	3.653037 x 10^{-6}	2.724070 x 10^{-6}	9.806650
778.1692	107.5858	1	0.2519958	3.930148 x 10^{-4}	2.930711 x 10^{-4}	1055.056
3088.025	426.9348	3.968321	1	1.559609 x 10^{-3}	1.163 x 10^{-3}	4186.8
1980000	273744.8	2544.434	641.1865	1	0.7456999	2684520.
2655224	367097.8	3412.142	859.8452	1.341022	1	3600000.
0.7375621	0.1019716	9.478171 x 10^{-4}	2.388459 x 10^{-4}	3.725061 x 10^{-7}	2.777778 x 10^{-7}	1

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 much 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 μm 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:

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

Horizontal vessel:

$$W_{cl} = C^* \left(\frac{S_{hl} - S_{ll}}{\mu} \right) L_l\, H_l \qquad \textbf{Eq 7-14}$$

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.

Example 7-3 — Determine the size of a vertical separator to handle 100 m^3/day of 0.76 relative density condensate and 10 m^3/day of produced water. Assume the water particle size is 200 μm. Other operating conditions are as follows:

Operating temperature = 25°C
Operating pressure = 6900 kPa(ga)
Water relative density = 1.01
Condensate viscosity = 0.55 mPa • s @ 25°C
Condensate relative density = 0.76

From Eq 7-14

$$W_{cl} = C^* \left(\frac{S_{hl} - S_{ll}}{\mu} \right) (0.785)\, (D_v)^2$$

From Fig. 7-17 for free liquids with water particle diameter = 200 microns, C* = 1880.

FIG. 7-23

Values of C* Used in Equations 7-13 and 7-14

Emulsion Characteristic	Droplet Diameter, μm	Constant,[10] C*
Free Liquids	200	1880
Loose Emulsion	150	1060
Moderate Emulsion	100	470
Tight Emulsion	60	170

$$100 m^3/day = 1880 \frac{1.01 - 0.76}{(0.55)} (0.785)\, (D_v)^2 (10^{-6})$$

$$(D_v)^2 = (1.43)\, (10^5)$$

$$D_v = 390 \text{ mm}$$

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

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

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

$$U = \frac{(110)\,(3)}{1440} = 0.23\ m^3$$

A 390 mm diameter vessel will hold about 0.12 m^3 per 1000 mm of height. The small volume held in the bottom head can be discounted in this size vessel. The shell height required for the retention volume required would be:

$$\text{Shell height} = \frac{0.23}{0.12} = 1.9 \text{ m} = 1900 \text{ mm}$$

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 100 m^3 per day per m^2.

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

Use base values from Fig. 9-11 for $(r_i)_1$ conditions.

$$f = \left(\frac{\mu_2}{\mu_1}\right)^{0.47}\left(\frac{k_1}{k_2}\right)^{0.67}\left(\frac{Cp_1}{Cp_2}\right)^{0.33}\left(\frac{G_1}{G_2}\right)^{0.8}\left(\frac{D_{i2}}{D_{i1}}\right)^{0.2}$$

$$= \left(\frac{0.21}{0.726}\right)^{0.47}\left(\frac{0.136}{0.135}\right)^{0.67}\left(\frac{2.09}{2.51}\right)^{0.33}\left(\frac{903}{537}\right)^{0.8}\left(\frac{15.7}{12.6}\right)^{0.2}$$

$$= 0.840$$

Basis: (Inside Area)

$(r_i)_2 = f(r_i)_1$ and $(r_i)_1 = 0.000\ 67$ from Fig. 9-11

$= (0.840)\ (0.000\ 67)$

$= 0.000\ 563\ (m^2 \bullet °C)/W$

From Fig. 9-10 (see Note †), the ratio of the second to the first pressure drop is:

$(\Delta P_i)_2 = (f)\ (\Delta P_i)_1$

Use base values from Fig. 9-11 for $(\Delta P_i)_1$ conditions.

$$f = \left(\frac{\mu_2}{\mu_1}\right)^{0.2}\left(\frac{G_2}{G_1}\right)^{1.8}\left(\frac{\rho_1}{\rho_2}\right)\left(\frac{D_{i1}}{D_{i2}}\right)^{1.2}\left(\frac{Np_2}{Np_1}\right)$$

$$= \left(\frac{0.21}{0.726}\right)^{0.2}\left(\frac{537}{903}\right)^{1.8}\left(\frac{751}{614}\right)\left(\frac{12.6}{15.7}\right)^{1.2}\left(\frac{1}{1}\right)$$

$$= 0.285$$

$(\Delta P_i)_2 = (f)\ (\Delta P_i)_1 = (0.285)\ (1.37) = 0.390$ kPa/m

For a 9.15 m tube length the total 9.15(0.390) = 3.569 kPa

5. Calculate the shell side pressure drop and resistance to heat transfer with the relationships shown in Fig. 9-10, the values shown in Fig. 9-11, and the data shown in Fig. 9-13.

$$G = \frac{(215\ 784)}{(3600)(101\ 283/1\ 000\ 000)} = 591.8\ kg/(m^2 \bullet s)$$

From Fig. 9-10 (see Note †), the ratio of the new to the old resistance is:

$(r_o)_2 = (f)\ (r_o)_1$

Use base values from Fig. 9-11 for $(r_o)_1$ conditions.

$$f = \left(\frac{\mu_2}{\mu_1}\right)^{0.27}\left(\frac{k_1}{k_2}\right)^{0.67}\left(\frac{Cp_1}{Cp_2}\right)^{0.33}\left(\frac{G_1}{G_2}\right)^{0.6}\left(\frac{D_{o2}}{D_{o1}}\right)^{0.4}$$

$$= \left(\frac{0.34}{0.549}\right)^{0.27}\left(\frac{0.132}{0.133}\right)^{0.67}\left(\frac{2.33}{2.27}\right)^{0.33}\left(\frac{646.4}{591.8}\right)^{0.6}\left(\frac{19.05}{15.7}\right)^{0.4}$$

$$= 0.998$$

$(r_o)_2 = (f)(r_o)_1 = (0.998)(0.000\ 49) = 0.000\ 49\ (m^2 \bullet °C)/W$

From Fig. 9-10 (see Note †), the ratio of the new to the old pressure drop is:

$(\Delta P_o)_2 = (f)\ (\Delta P_o)_1$

Use base values from Fig. 9-11 for $(\Delta P_o)_1$ conditions. Obtain tube rows crossed between baffle window centroids from Fig. 9-13.

$RC_2 = 23$ ($RC_1 = 10$ per note on Fig. 9-11)

Obtain the number of crossflow spaces, which is one more than the number of baffles, from Fig. 9-12.

$SP_2 = 19$ [$SP_1 = 1$ per note on Fig. 9-11 since $(\Delta P_o)_1$ is for one baffle space.]

$$f = \left(\frac{\mu_2}{\mu_1}\right)^{0.15}\left(\frac{G_2}{G_1}\right)^{1.85}\left(\frac{\rho_1}{\rho_2}\right)\left(\frac{D_{o1}}{D_{o2}}\right)^{0.15}\left(\frac{SP_2}{SP_1}\right)\left(\frac{RC_2}{RC_1}\right)$$

$$= \left(\frac{0.34}{0.549}\right)^{0.15}\left(\frac{591.8}{646.4}\right)^{1.85}\left(\frac{750}{660}\right)\left(\frac{15.7}{19.05}\right)^{0.15}\left(\frac{19}{1}\right)\left(\frac{23}{10}\right)$$

$$= 38.3$$

$(\Delta P_o)_2 = (f)\ (\Delta P_o)_1 = (38.3)\ (1.7) = 65.1$ kPa

6. Calculate the tube metal resistance.

$D_o = 19.05$ mm

$D_i = 15.75$ mm

$k_w = 50\ W/(m \bullet °C)$ from Fig. 9-8.

$$r_w = \frac{D_o}{2 \bullet 1000\ k_w} ln\left[\frac{D_o}{D_i}\right]$$

$$= 0.000\ 036(m^2 \bullet °C)/W$$

7. Calculate the overall heat transfer coefficient.

$$\Sigma r = r_i\left(\frac{A_o}{A_i}\right) + r_o + r_w + r_{fo} + r_{fi}\left(\frac{A_o}{A_i}\right)$$

$$= 0.000\ 563\left(\frac{0.0182}{0.0151}\right) + 0.000\ 49 + 0.000\ 036 + 0.000\ 35$$

$$+ 0.000\ 2\left(\frac{0.0182}{0.0151}\right) = 0.0018$$

$$U = \frac{1}{\Sigma r} = \frac{1}{0.001\ 78} = 555.6\ W/(m^2 \bullet °C)$$

8. Compare the required heat transfer coefficient calculated in step 3 to the value calculated in step 7. (Available U = 561.8; required U = 570.4) The available value is 1.6% less than the required value, and the calculated pressure drops are less than the pressure drops allowed in Fig. 9-12. Therefore, by these calculations, the unit will perform adequately.

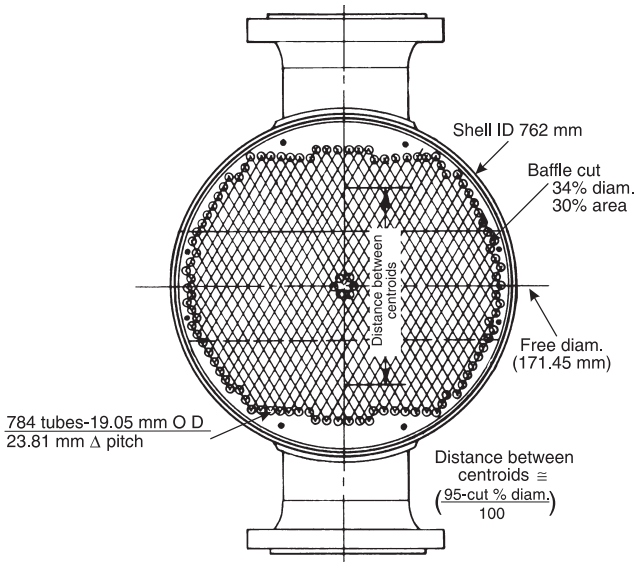
CONDENSERS

The purpose of a condenser is to change a fluid stream from the vapor state to the liquid state by removing the heat of vaporization. The fluid stream may be a pure component or a mixture of components. Condensation may occur on the shell side or the tube side of an exchanger oriented vertically or horizontally.

Condensing the overhead vapors of a distillation column is an example of condensing a mixed vapor stream. A vertical exchanger flanged directly to the top of the column might be used. The condensed liquid drains back into the column countercurrent to the vapor entering the condenser. The major concerns in designing this type exchanger are keeping the vapor velocity sufficiently low to prevent flooding the exchanger and evaluating an appropriate temperature profile at the condensing surface to determine an effective temperature difference. The technical literature addresses criteria for flooding determination[1] and special flow characteristics of falling liquid films. A useful estimate for determining an effective temperature difference can be made by assuming an isothermal condensate film at the saturation temperature of the last condensate formed. If the condensing temperature range ex-

FIG. 9-13

Heat Exchanger Detail Design Results



Leakage – use TEMA tolerances		
tube hole = 576 • 19.05 • π • 0.4	=	13 789 mm^2
shell crack = 762π • 4.445 • 0.60	=	6 385 mm^2
		20 174 mm^2
Window flow area:		
window area = 0.237 • $(762)^2$	=	137 613 mm^2
tubes in window = 208 • 0.785 • $(19.05)^2$	=	59 255 mm^2
		78 358 mm^2
	leak	20 174 mm^2
		98 532 mm^2
Cross flow area:		
Free area = 171.45 mm • 473.08	=	81 109 mm^2
Leakage	=	20 174 mm^2
Net cross flow area	=	101 283 mm^2
BP = baffle pitch = 473.08 mm		
FD = free diameter = 171.45 mm		
Note: window area ≅ cross flow area		

ceeds 5°C, consulting a specialist is recommended for a more rigorous calculation procedure.

The condensing of a pure component occurs at a constant temperature equal to the saturation temperature of the incoming vapor stream. Frequently a vapor enters a condenser superheated and must have the sensible heat removed from the vapor before condensation can occur. If the condensing surface temperature is greater than the incoming vapor saturation temperature, the superheat in the vapor is transferred to the cold surface by a sensible heat transfer mechanism ("drywall" condition). If the condensing surface temperature is less than the saturation temperature of the incoming vapor, a condensate film will be formed on the cold surface. The sensible heat is removed from the vapor at the condensate-vapor interface by vaporizing (flashing) condensate so that the heat of vaporization is equal to the sensible heat removed from the vapor. Under this "wet wall" condition, the effective temperature of the vapor is the saturation temperature, and the effective heat transfer mechanism is condensation. The determination of the point in the desuperheating zone of a condenser where "drywall" conditions cease and "wet wall" conditions begin is a trial and error procedure. A method frequently employed to give a safe approximation of the required surface is to use the condensing coefficient and the CMTD based on the vapor saturation temperature to calculate the surface required for both the desuperheating zone and the condensing zone.

The following Example 9-2 will illustrate the use of the heat release curve to calculate the surface required and the LMTD for each zone in a condenser for a pure component application.

Example 9-2 — A propane refrigerant condenser is required to condense the vapor stream using the heat release curve as shown in Fig. 9-14. This stream enters the condenser superheated and leaves the condenser as a subcooled liquid. Assume that a single-tube pass, single-shell pass, counterflow exchanger is used so that LMTD correction factors do not apply. Note that the propane is on the shell side. The overall heat transfer coefficients for each zone are as follows:

Desuperheating: (82°C to 42°C)

$$U_v = 396.6\ W/(m^2 \cdot °C)$$

$$h_v = 630.4\ W/(m^2 \cdot °C)$$

Condensing: (42°C to 42°C)

$$U_c = 794.4\ W/(m^2 \cdot °C)$$

Subcooling: (42°C to 35°C)

$$U_L = 649.7\ W/(m^2 \cdot °C)$$

Solution Steps

1. Calculate the surface temperature (outside wall) on the vapor side at the refrigerant stream inlet using the following equation:[2]

FIG. 9-14

Propane Condensing Curve

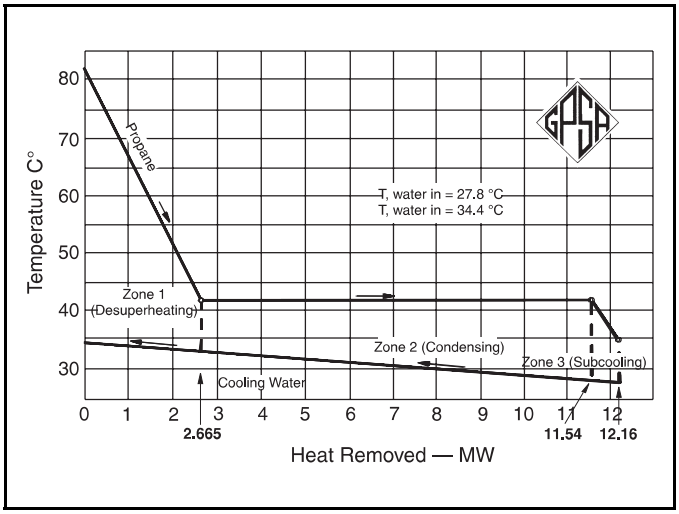


FIG. 10-20

Altitude and Atmospheric Pressures[8]

Altitude above Sea Level		Barometer		Atmospheric Pressure	
meters	feet	mm Hg abs.	in Hg abs.	kPa (abs)	psia
0	0	760.0	29.92	101.325	14.696
153	500	746.3	29.38	99.49	14.43
305	1000	733.0	28.86	97.63	14.16
458	1500	719.6	28.33	95.91	13.91
610	2000	706.6	27.82	94.18	13.66
763	2500	693.9	27.32	92.46	13.41
915	3000	681.2	26.82	90.80	13.17
1068	3500	668.8	26.33	89.15	12.93
1220	4000	656.3	25.84	87.49	12.69
1373	4500	644.4	25.37	85.91	12.46
1526	5000	632.5	24.90	84.32	12.23
1831	6000	609.3	23.99	81.22	11.78
2136	7000	586.7	23.10	78.19	11.34
2441	8000	564.6	22.23	75.22	10.91
2746	9000	543.3	21.39	72.39	10.50
3050	10 000	522.7	20.58	69.64	10.10

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Sound power level cannot be measured directly but must be calculated from sound pressure levels (SPL) dB. In metric terms, this is known as Lw.

Sound pressure level, known as SPL, or Lp in metric terminology, is the audible noise given in decibels that relates to intensity at a point some distance from the noise source. It can be related to octave bands or as an overall weighted level dB(A).

Weighted sound levels relate the decibel (loudness) to a frequency. Ears can easily pick up high-frequency noises (both intensity and direction) but are relatively insensitive to low-frequency noise. For a stereo system high-frequency speakers must be very carefully located in a room for best results, but low-frequency bass speakers can be placed anywhere, even out of sight.

There are three basic weighting systems: A, B and C. The "A" system, dB(A), most closely relates to our ear, the "B" system, dB(B), has some specific uses and the "C" system, dB(C), is considered unweighted.

The dB(A) is the most common weighting system. It expresses sound levels with a single number instead of having to specify the noise of each octave band.

Note that the sound range of ACHEs (at close range) is typically between 80 and 105 dB(A).

ACHE Noise

Whether the concern is for overall plant noise or the noise exposure of plant workers in the vicinity of the fans, a different type of noise specification must be used.

Overall, noise limitations from an ACHE are typically a sound power level (PWL) specification for each unit. This limits the contribution of each unit (typically two fans) to the plant noise as a whole. This is usually needed where noise at the plant boundary is considered. Contributions of each part of the plant must be carefully controlled if overall plant noise is limited. PWLs can be expressed as weighted level dB(A) or sometimes even by limitations on each octave band.

If worker protection is the main concern, a limitation of sound pressure level at 1 m below the bundle will probably be imposed as "SPL dB(A) at 1 m". The OSHA limitation is 90 dB(A) for eight-hour exposure, but 85 dB(A) down to 80 dB(A) is not uncommon.

Predicting Fan Noise

Each fan manufacturer has proprietary equations for predicting fan noise. API Guidelines use the general formula:

$$PWL = 56 + 30 \log \left(\frac{MPM}{304.8} \right) + 10 \log HP \qquad \textbf{Eq 10-4}$$

This calculates PWL as dB(A)

Proprietary noise equations are based on actual tests at various speeds and operating conditions considering the following effects:

- Fan diameter
- Fan tip speed
- Blade Type
- Blade pitch angle
- Inlet conditions
- Horsepower

Note: Logs are common logs (base 10).

For example:

4.267 meter fan

237 RPM (3177 m/min tip speed)

25.1 HP

Find sound power level

PWL= 56 + 30 log 10.4 + 10 log 25.1

= 100.5 dB(A)

- When considering multiple noise sources (fans) use the relation:

$PWL_N = PWL + 10 \log N$ **Eq 10-5**

The sound power level for 2 adjacent fans is the PWL of one fan plus 10 log 2 or $PWL_2 = PWL + 3$.

A doubling of the noise source adds 3 dB.

- Noise attenuates with distance by the equation:

$SPL \text{ (at distance R)} = PWL - 20 \log (3.28R)$ **Eq 10-6**

Where R is in meters from the center of the source. Measure R as a "line of sight" distance.

Consider the noise an observer at grade hears at 15.24 m from an operating ACHE with the fan on and the line-of-sight distance from grade to the center of the fan is actually 18.9 m. Remember what the ear hears is SPL, the noise energy is PWL.

Assume PWL = 100.5 dB(A).

SPL = 100.5 - 20 log 62 = 64.7 dB(A)

This also assumes background noise is at least 10 dB quieter.

Note: If both fans were running, the SPL would have been 67.7 dB(A).

When considering the noise at 1 m beneath the unit, the drive system and motor noise become dominant at lower tip speeds.

Factors that influence this noise are:

- Motor noise
- Belt or gear noise
- Bearing noise
- Reflected noise from supports
- Background noise

Gear noise is especially significant in a forced draft unit.

Noise Testing

Frequently, the ACHE must be tested for confirmation that its noise does not exceed specifications imposed by the purchaser. There are two basic types of tests normally performed before shipment:

a) Measure SPL dB(A) (Sound Pressure Level) at "1 m below the bundle or fan guard" — depending on whether the unit is induced or forced draft.

b) Measure PWL (Sound Power Level) using "hemispherical power level test." PWL is specified as either a dB(A) weighted value or by octave bands.

The "SPL at 1 m" test is by far the most common and least expensive. Usually, only one or two measurements are required. The answer is immediate and read directly from the noise meter.

The hemispherical test is far more complicated and expensive. Several hours, many technicians and a large crane are required to perform this test. Full details of the test are given in API Recommended Practice 631 M, June, 1981.[7]

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6. Rubin, Frank L., “Winterizing Air Cooled Heat Exchangers,” Hydrocarbon Processing, October 1980, pp. 147-149.
7. API Standard 631 M, First Edition, "Measurement of Noise from Air-Cooled Heat Exchangers," June, 1981.
8. John M. Campbell Co. "Gas Conditioning and Processing, Vol. 2," Eighth Edition.

Answer: t_2 = 130°C (approximately) from Fig. 13-32.

Fig. 13-34 gives the approximate brake power required for the compression. It includes compressor efficiencies in the range of 60 to 70%.

Example 13-5 — Given: Weight flow, w, = 30 000 kg/h

head = 150 kN•m/kg

Find: Brake Power

Answer: Gp = 1700 kW from Fig. 13-34.

Fig. 13-37 predicts the approximate number of compressor wheels required to produce the head. If the number of wheels is not a whole number, use the next highest number.

Calculating Performance

When more accurate information is required for compressor head, gas horsepower, and discharge temperature, the equations in this section should be used. This method applies to a gas mixture for which a P-H diagram chart is not available. To calculate the properties of the gas, see Figs. 13-6 and 13-7. All values for pressure and temperature in these calculation procedures are the absolute values. Unless otherwise specified, volumes of flow in this section are actual volumes.

To calculate the inlet volume:

$$Q = \frac{(w)\,(8.314)\,(T_1)\,(Z_1)}{(M)\,(P_1)} \qquad \textbf{Eq 13-25}$$

If we assume the compression to be isentropic (reversible adiabatic, constant entropy), then:

$$H_{is} = \frac{ZRT}{M\,(k-1)/k}\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \qquad \textbf{Eq 13-26}$$

Since these calculations will not be wheel-by-wheel, the head will be calculated across the entire machine. For this, use the average compressibility factor:

FIG. 13-25

Typical Centrifugal Compressor Cutaway

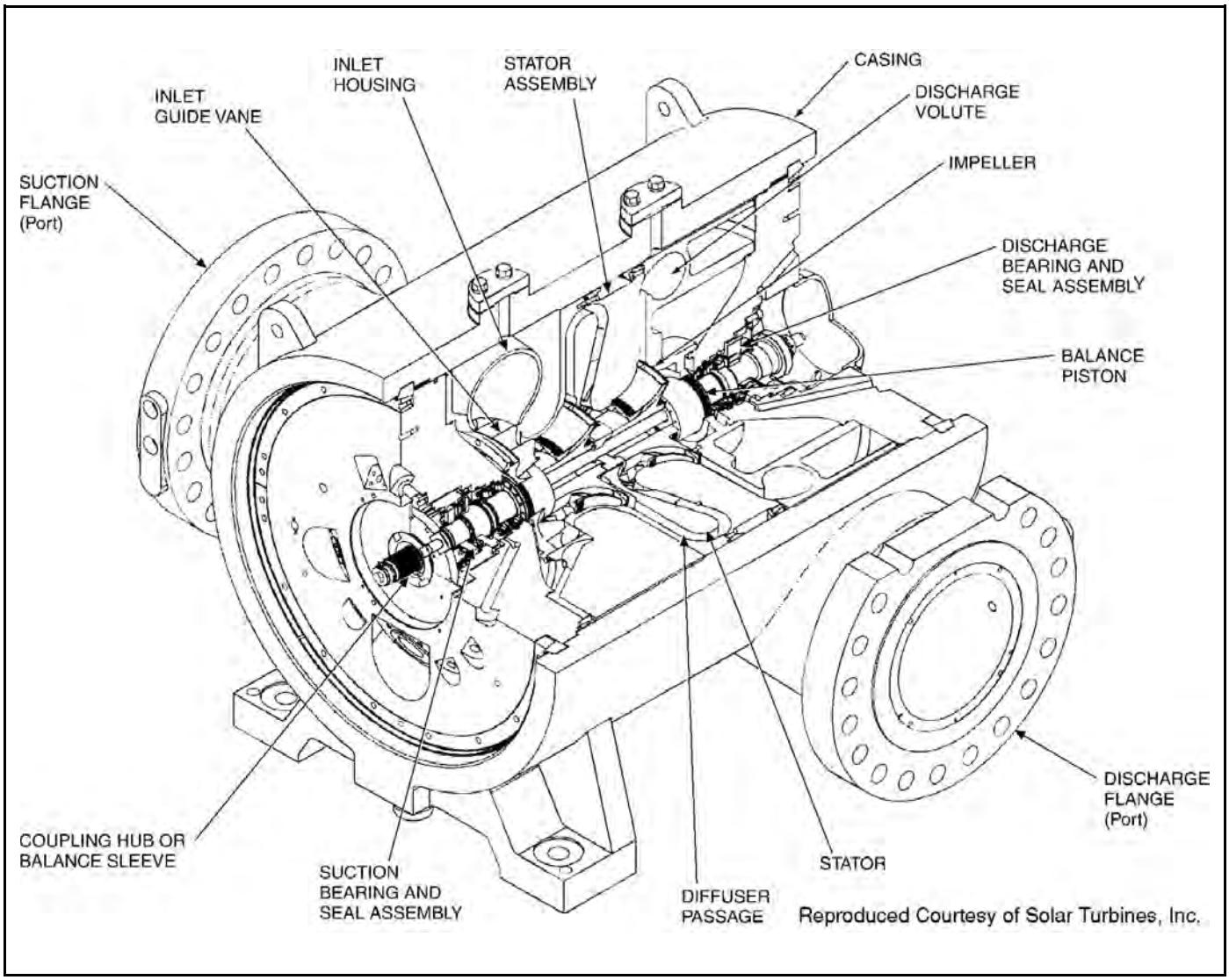
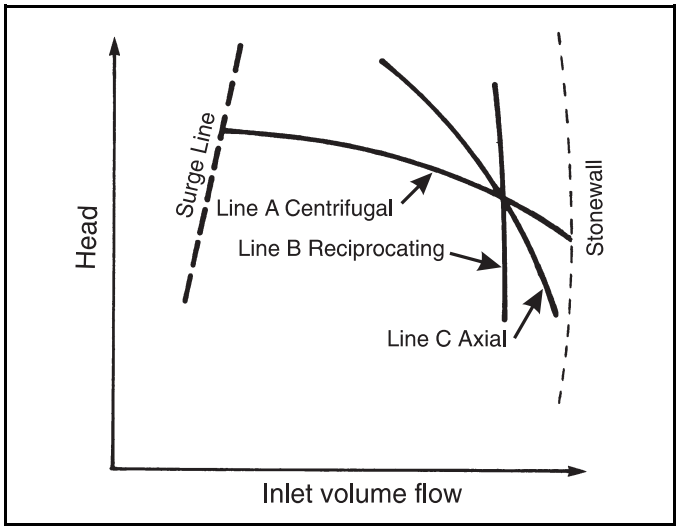


FIG. 13-27

Compressor Head



$$Z_{avg} = \frac{Z_1 + Z_2}{2}$$

The heat capacity ratio, k, is normally determined at the average suction and discharge temperature (see Figs. 13-7 and 13-8).

Isentropic Calculation

To calculate the head:

$$H_{is} = \frac{Z_{avg} R T_1}{M\,(k-1)/k}\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \qquad \textbf{Eq 13-27a}$$

which can also be written in the form:

$$H_{is} = \frac{8.314}{M}\,\frac{Z_{avg} T_1}{(k-1)/k}\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \qquad \textbf{Eq 13-27b}$$

The gas horsepower can now be calculated from:

$$Ghp = \frac{(w)\,(H_{is})}{(\eta_{is})\,(3\,600)} \qquad \textbf{Eq 13-28}$$

FIG. 13-28

Compressor Performance, Low Compression Ratio

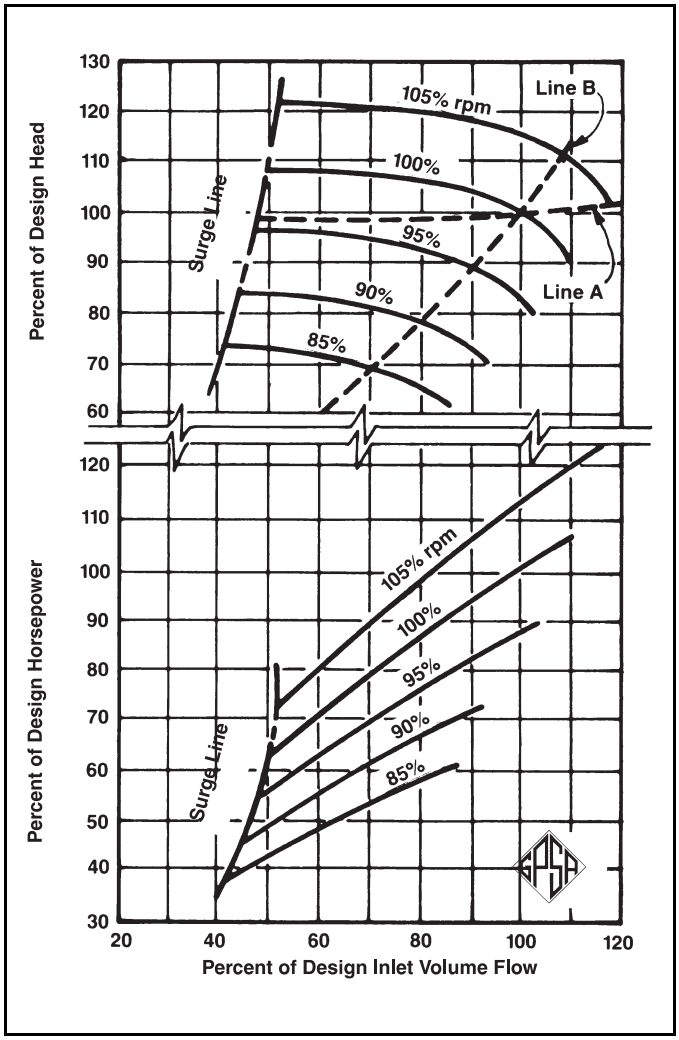


FIG. 13-29

Compressor Performance, Higher Compression Ratio

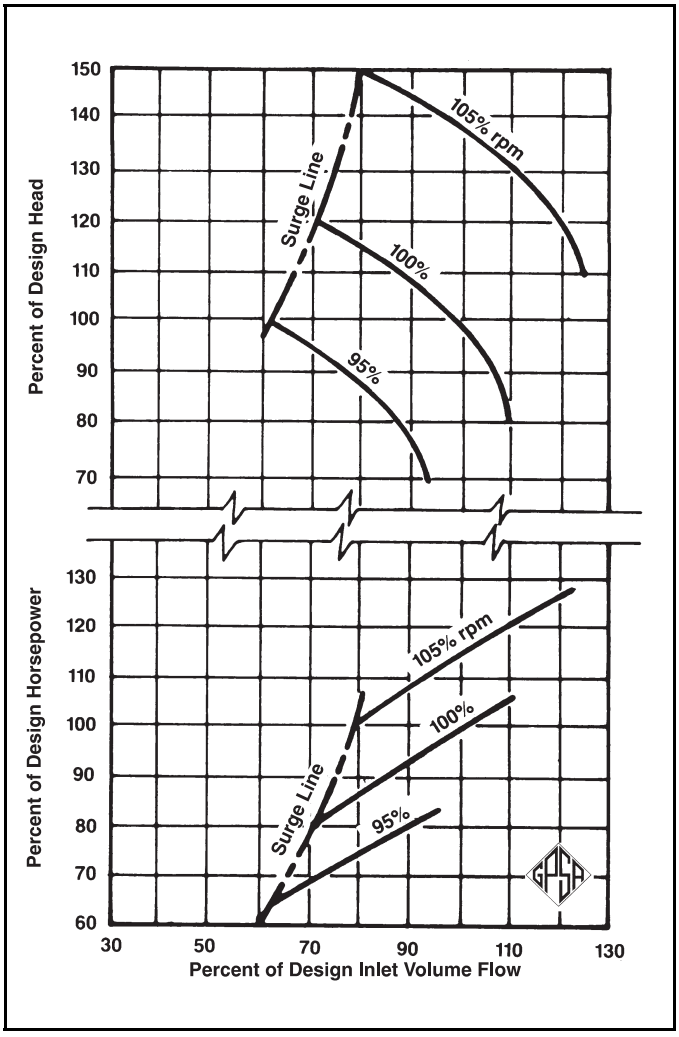
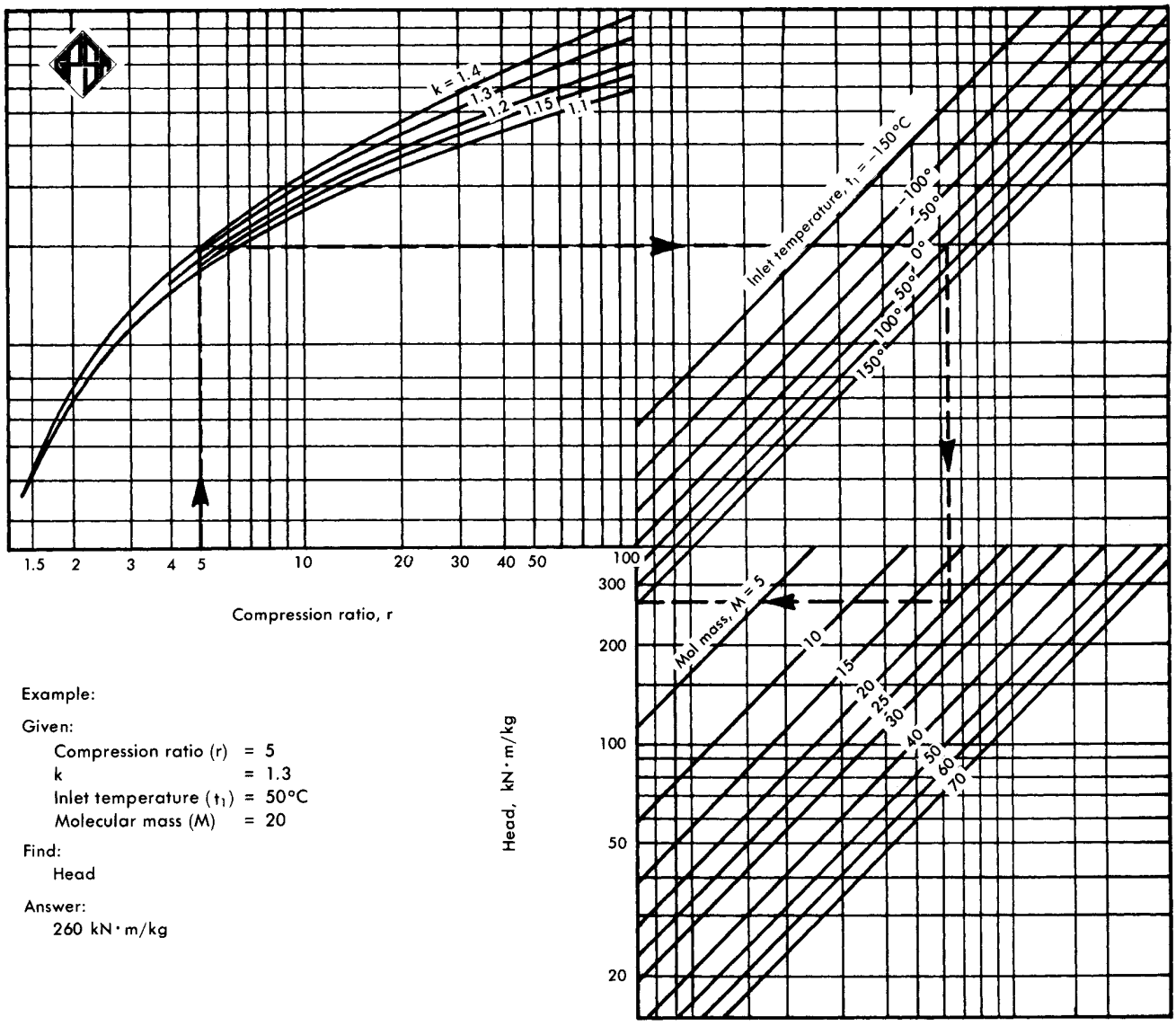


FIG. 13-33

Head

Z = 1



Example:

Given:

Compression ratio (r)	=	5
k	=	1.3
Inlet temperature (t_1)	=	50°C
Molecular mass (M)	=	20

Find:

Head

Answer:

260 kN · m/kg

FIG. 13-34

Power Determination

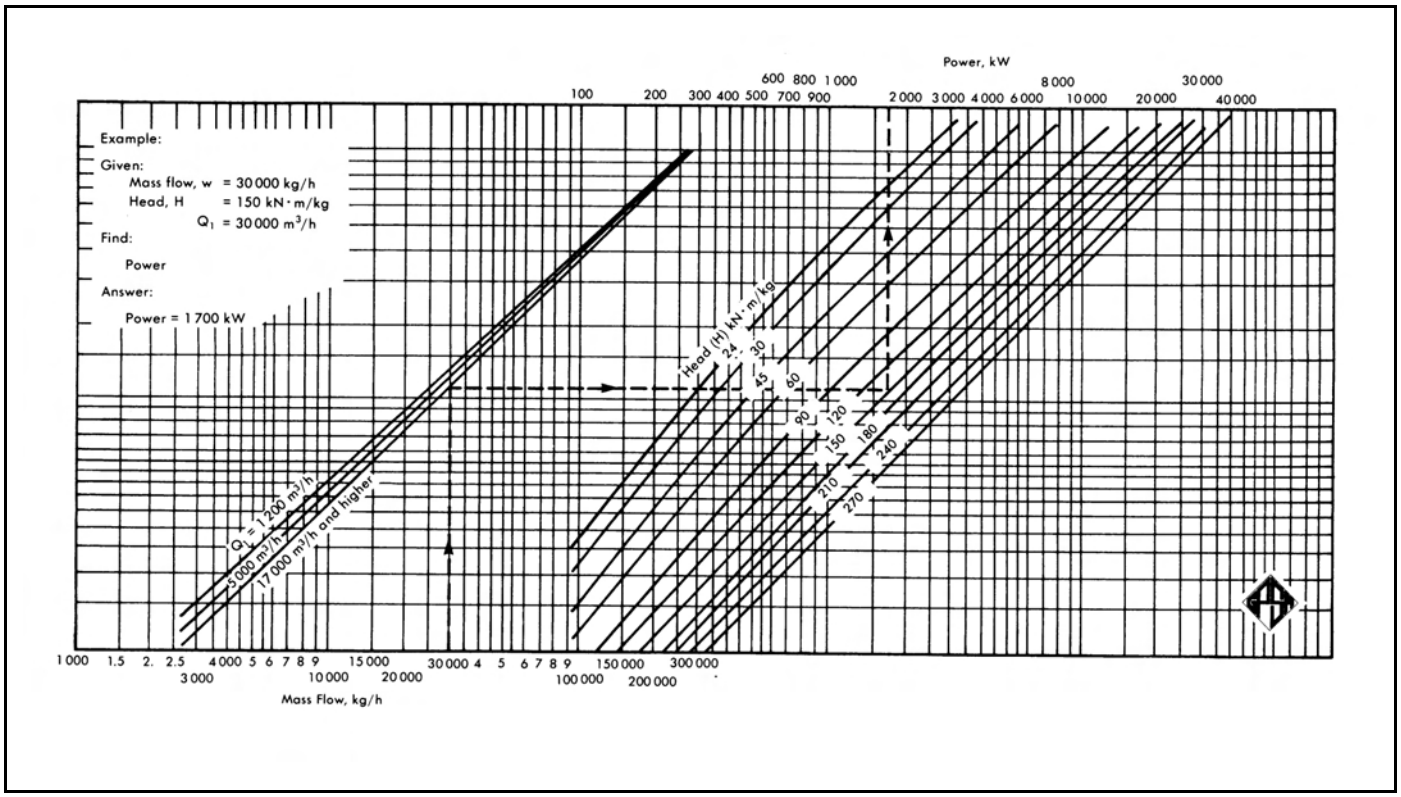
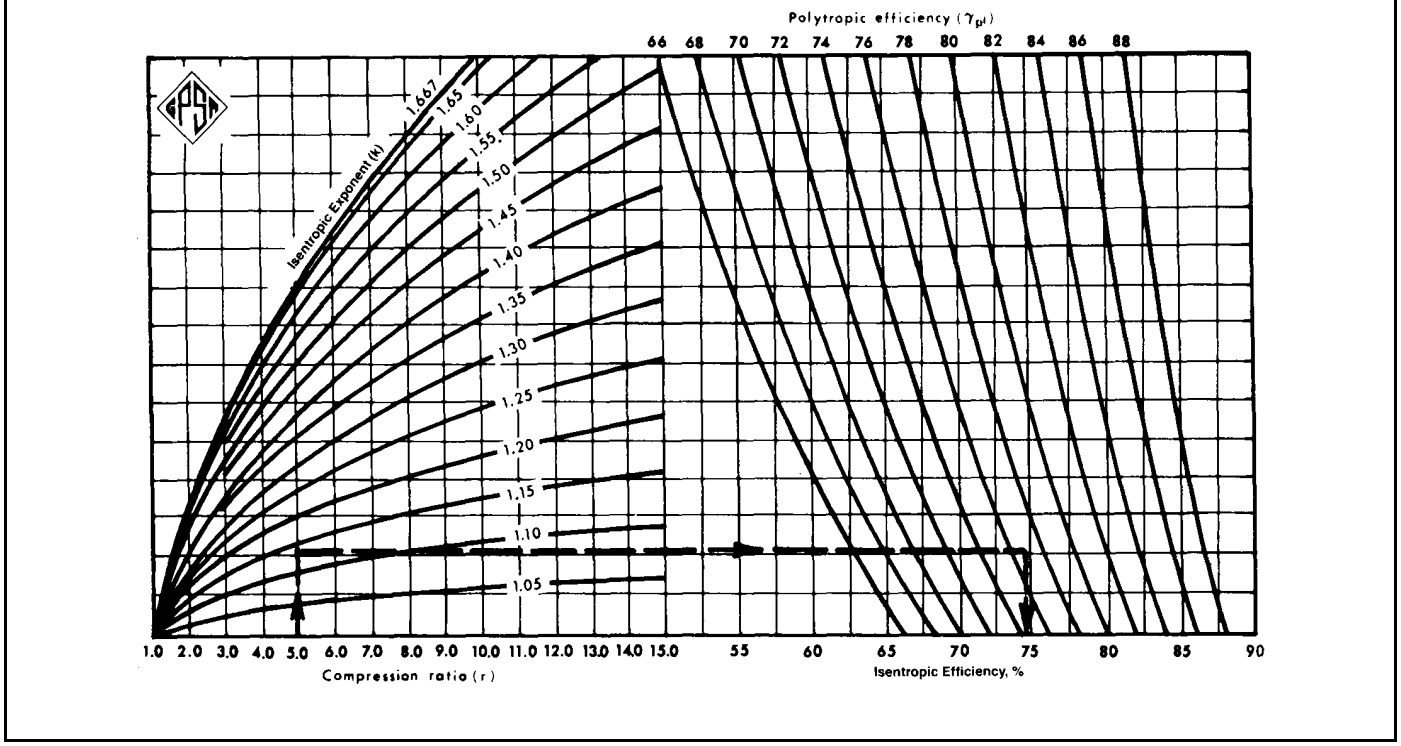


FIG. 13-35

Efficiency Conversion



The approximate theoretical discharge temperature can be calculated from:

$$\Delta T_{ideal} = T_1 \left[\left(\frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right] \qquad \textbf{Eq 13-29}$$

$$T_2 = T_1 + \Delta T_{ideal} \qquad \textbf{Eq 13-30}$$

The actual discharge temperature can be approximated:

$$\Delta T_{actual} = T_1 \frac{\left[\left(\frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right]}{\eta_{is}} \qquad \textbf{Eq 13-31}$$

$$T_2 = T_1 + \Delta T_{actual} \qquad \textbf{Eq 13-32}$$

Polytropic Calculation

Sometimes compressor manufacturers use a polytropic path instead of isentropic. Polytropic efficiency is defined by:

$$\frac{n}{(n-1)} = \left[\frac{k}{(k-1)} \right] \eta_p \qquad \textbf{Eq 13-33}$$

(See Fig. 13-35 for conversion of isentropic efficiency to polytropic efficiency.)

The equations for head and gas horsepower based upon polytropic compression are:

$$H_p = \frac{Z_{avg} R T_1}{M\,(n-1)/n} \left[\left(\frac{P_2}{P_1} \right)^{(n-1)/n} - 1 \right] \qquad \textbf{Eq 13-34a}$$

which also can be written in the form:

$$H_p = \frac{8.314}{M} \frac{Z_{avg} T_1}{(n-1)/n} \left[\left(\frac{P_2}{P_1} \right)^{(n-1)/n} - 1 \right] \qquad \textbf{Eq 13-34b}$$

$$Ghp = \frac{(w)\,(H_p)}{(\eta_p)\,(3\,600)} \qquad \textbf{Eq 13-35}$$

Polytropic and isentropic head are related by

$$H_p = \frac{H_{is}\,\eta_p}{\eta_{is}} \qquad \textbf{Eq 13-36}$$

The approximate actual discharge temperature can be calculated from:

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\left(\frac{n-1}{n} \right)} \qquad \textbf{Eq 13-37}$$

Mechanical Losses

After the gas horsepower has been determined by either method, horsepower losses due to friction in bearings, seals, and speed increasing gears must be added.

Fig. 13-36 shows losses related to the shaft speed and casing size for conventional multistage units.

Bearings and seal losses can also be roughly computed from Scheel's equation:

$$\text{Mechanical losses} = (Gp)^{0.4} \qquad \textbf{Eq 13-38}$$

To calculate the brake horsepower:

$$Bhp = Gp + \text{mechanical losses} \qquad \textbf{Eq 13-39}$$

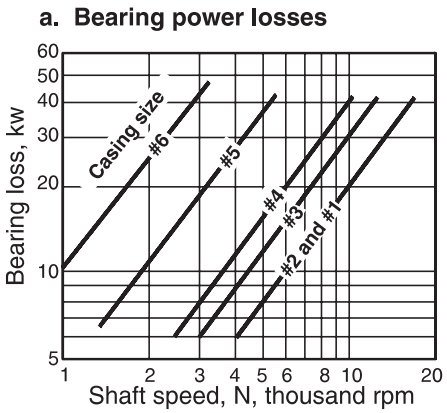
Compressor Speed

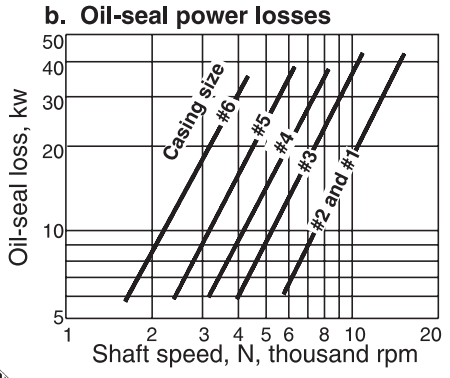
The basic equation for estimating the speed of a centrifugal compressor is:

FIG. 13-36

Mechanical Losses

Casing Size	Max Flow (inlet m^3/h)	Nominal Speed (rpm)
1	12 700	10 500
2	33 900	8 200
3	56 000	6 400
4	93 400	4 900
5	195 000	3 600
6	255 000	2 800





Courtesy Chemical Engineering Magazine

$$N = (N_{nominal}) \sqrt{\frac{H\,total}{(No.\,of\,wheels)\,(H\,max/\,wheel)}} \qquad \textbf{Eq 13-40}$$

where the number of wheels is determined from Fig. 13-37.

Nominal speeds to develop 30 000 N • m/kg of head/wheel can be determined from Fig. 13-23. However, to calculate the maximum head per wheel, the following equation based on molecular weight (or more accurately, density) can be used.

$$H\,max/stage = 15{,}000 - 1{,}500\,(M)^{0.35} \qquad \textbf{Eq 13-41}$$

This equation will give a head of 30 000 N • m/kg for a gas when M = 30 and 33 000 N • m/kg when M = 16.

economic advantage versus methanol recovered by distillation. At cryogenic conditions (below –40°C) methanol usually is preferred because glycol's viscosity makes effective separation difficult.

Ethylene glycol (EG), diethylene glycol (DEG), and triethylene glycol (TEG) glycols have been used for hydrate inhibition. The most popular has been ethylene glycol because of its lower cost, lower viscosity, and lower solubility in liquid hydrocarbons.

Physical properties of methanol and methanol-water mixtures are given in Fig. 20-34 through Fig. 20-37. Physical properties of the most common glycols and glycol-water mixtures are given in Fig. 20-38 through Fig. 20-49. Tabular information for the pure glycols and methanol is provided in Fig. 20-56.

To be effective, the inhibitor must be present at the very point where the wet gas is cooled to its hydrate temperature. For example, in refrigeration plants glycol inhibitors are typically sprayed on the tube-sheet faces of the gas exchangers so that it can flow with the gas through the tubes. As water condenses, the inhibitor is present to mix with the water and prevent hydrates. Injection must be in a manner to allow good distribution to every tube or plate pass in chillers and heat exchangers operating below the gas hydrate temperature.

The viscosities of ethylene glycol and its aqueous solutions increase significantly as temperature decreases, and this must be allowed for in the rating of refrigeration-plant exchangers and chillers.

The inhibitor and condensed water mixture is separated from the gas stream along with a separate liquid hydrocarbon stream. At this point, the water dewpoint of the gas stream is essentially equal to the separation temperature. Glycol-water solutions and liquid hydrocarbons can emulsify when agitated or when expanded from a high pressure to a lower pressure, e.g., JT expansion valve. Careful separator design will allow nearly complete recovery of the diluted glycol for regeneration and reinjection. Fig. 20-57 shows a flow diagram for a typical EG injection system in a refrigeration plant.

The regenerator in a glycol injection system should be operated to produce a regenerated glycol solution that will have a freezing point below the minimum temperature encountered in the system. This is typically 75-80 wt%. Fig. 20-58 shows the freezing point of various concentrations of glycol water solutions.

The minimum inhibitor concentration in the free water phase may be approximated by Hammerschmidt's equation.[25]

$$d = \frac{K_H X_I}{MW_I (1 - X_I)} \qquad \textbf{Eq 20-5}$$

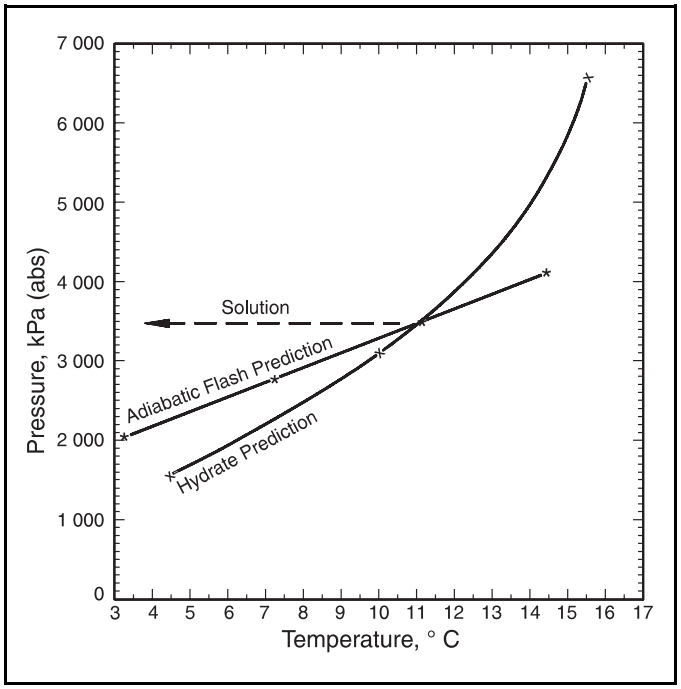
$$X_I = \frac{d MW_I}{K_H + d MW_I} \qquad \textbf{Eq 20-6}$$

Where K_H (glycols) = 1297 to 2222 and
K_H (methanol) = 1297

The K_H range of 1297 to 2222 for glycols reflects the uncertainty in the value of this parameter. At equilibrium, such as for a laboratory test, 1297 is applicable as illustrated on Fig. 20-60. In some field operations, however, hydrate formation has been prevented with glycol concentrations corresponding with K_H values as high as 2222. This is because hydrate suppression with glycols depends on the system's physical and flow characteristics (i.e., the system dynamics, configuration, location and method of glycol injection, amount of free water, etc.) as well as the properties of the gas and the glycol. Therefore, in the absence of reliable field-test data, a system should be designed for a K_H of 1297. Once the system is operating, the glycol concentration can be reduced to tolerable levels.

FIG. 20-30

Solution Sketch for Example 20-8



Eq 20-5 and Eq 20-6 should not be used beyond 20-25 wt% for methanol and 60-70 wt% for the glycols. For methanol concentrations up to about 50%, the Nielsen-Bucklin equation[26] provides better accuracy:

$$d = -72.0 \ln(x_{H_2O}) \qquad \textbf{Eq 20-7}$$

Note that "x_{H_2O}" in Eq 20-7 is a mole fraction, not a mass fraction. Expressing mole fraction in terms of mass fraction, dewpoint 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 required in the water phase may be calculated from Eq 20-8

$$m_I = \frac{X_R \bullet m_{H_2O}}{X_L - X_R} \qquad \textbf{Eq 20-8}$$

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 4800 kPa (abs). 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.

tor countercurrent to the gas flow. Water-rich glycol is removed from the bottom of the contactor, passes through the reflux condenser coil, flashes off most of the soluble gas in the flash tank, and flows through the rich-lean heat exchanger to the regenerator. In the regenerator, absorbed water is distilled from the glycol at near atmospheric pressure by application of heat. The regenerated lean glycol exits the surge drum, is partly cooled in the lean-rich exchanger and is pumped through the glycol cooler before being recirculated to the contactor.

Evaluation of a TEG system involves first establishing the minimum TEG concentration required to meet the outlet gas water dewpoint specification. Fig. 20-68 shows the water dewpoint of a natural gas stream in equilibrium with a TEG solution at various temperatures and TEG concentrations. Fig. - 20-68 can be used to estimate the required TEG concentration for a particular application or the theoretical dewpoint depression for a given TEG concentration and contactor temperature. Actual outlet dewpoints depend on the TEG circulation rate and number of equilibrium stages, but typical approaches to equilibrium are 6–11°C. Equilibrium dewpoints are relatively insensitive to pressure and Fig. 20-68 may be used up to 10 300 kPa (abs) with little error.

Fig. 20-68 combines the results published by Parrish, et al.[35] covering TEG purity up to 99.99 mass percent and those reported by Bucklin and Won in 1987[52] covering TEG purity up

FIG. 20-61

Hydrate Inhibition with Methanol: Hammerschmidt vs. Experimental Data[28]

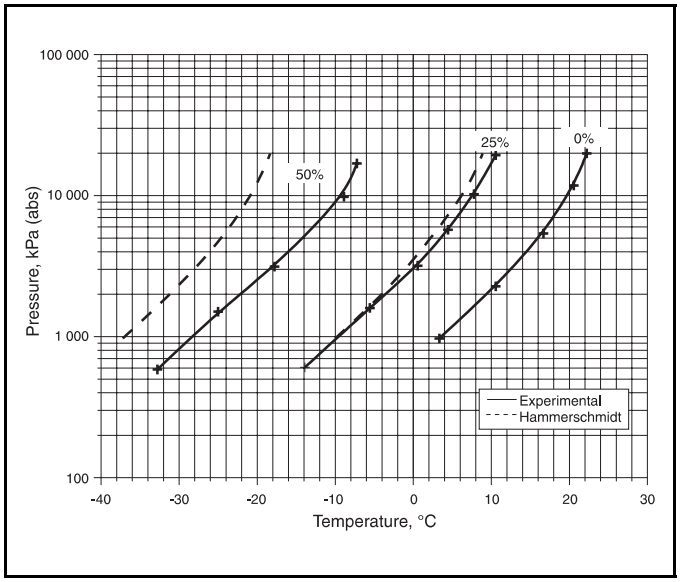


FIG. 20-62

Hydrate Inhibition with Methanol: Nielsen & Bucklin vs. Experimental Data[28]

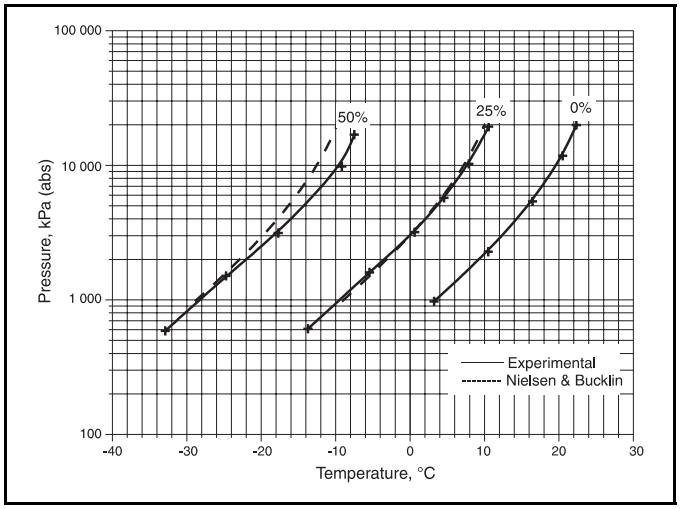


FIG. 20-63

Hydrate Inhibition with Methanol: Nielsen & Bucklin[26] vs. Experimental Data[29]

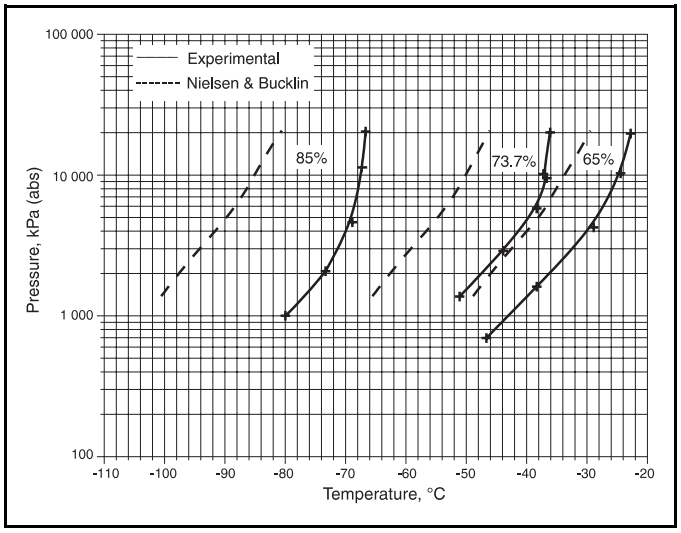
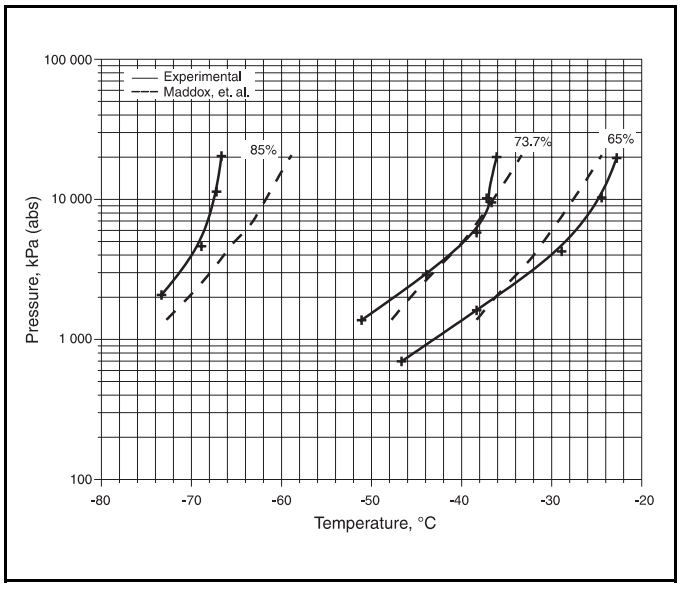


FIG. 20-64

Hydrate Inhibition with Methanol: Maddox et al.[27] vs. Experimental Data[29]



The solubilities of pure H_2S and CO_2 in pure TEG are given in Fig. 20-76. These charts apply to pure TEG or to the lean TEG, which is essentially pure. For the rich-TEG solution leaving the bottom of a contactor, it is approximately correct to apply them to the TEG portion.

Heavier paraffin hydrocarbons are essentially insoluble in TEG. Aromatic hydrocarbons, however, are very soluble in TEG, and significant amounts of aromatic hydrocarbons may be absorbed in the TEG at contactor conditions. This may present an environmental or safety hazard when they are discharged from the top of the regenerator.

Vapor-liquid equilibrium constants (K-values) for benzene, toluene, ethylbenzene, and o-xylene in TEG solutions are presented in RR-131.[43] This data indicates that at typical contactor conditions approximately 10-30% of the aromatics in the gas stream may be absorbed in the TEG solution. Based on the data in RR-131 at 6900 kPa (abs), 38°C and a TEG circulation rate of 25 liters TEG/kg H_2O absorbed, the approximate percentage absorption of aromatics from the gas into the rich TEG are calculated as:

Benzene	10%
Toluene	14%
Ethylbenzene	19%
O-xylene	28%

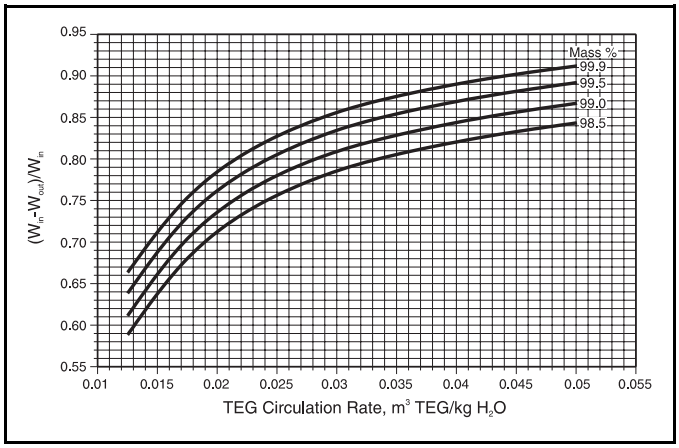
Aromatic absorption increases with increasing pressure and decreasing temperature. Aromatic absorption is directly related to TEG circulation rate. Higher circulation rates result in increased absorption. Aromatic absorption is essentially independent of the number of contacts in the absorber so one method of minimizing aromatic absorption is to use taller contactors and minimize TEG circulation rates.

Most of the aromatic components will be stripped from the TEG solution in the regenerator.

Flash tank sizing should be sufficient to degas the glycol solution and skim entrained liquid hydrocarbons, if necessary. A minimum retention time of 3-5 minutes is required for degassing. If liquid hydrocarbons are to be removed as well, retention times of 20-30 minutes may be required for adequate separation. Flash tank pressures are typically less than 520 kPa (abs).

FIG. 20-69

Water Removal vs. TEG Circulation Rate at Various TEG Concentrations (N = 1.0)



Regenerator sizing requires establishing the reboiler duty and, when high TEG concentrations are required, providing sufficient stripping gas.

A quick estimate of reboiler duty can be made using Eq 20-10.

$$Q = (0.116)(L_g) \qquad \textbf{Eq 20-10}$$

Eq 20-10 is approximate and usually gives values which are higher than the actual duty. A more rigorous determination of reboiler duty is shown in Example 20-12.

Example 20-12 — Determine reboiler duty for conditions in the previous example. Assume the rich TEG temperature entering the regenerator is 150°C and the reboiler temperature is 200°C.

Glycol Reboiler Duty: Basis 1 m^3 TEG

Sensible Heat:

$$Q_s = m\, C_p\, \Delta T$$

FIG. 20-70

Water Removal vs. TEG Circulation Rate at Various TEG Concentrations (N = 1.5)

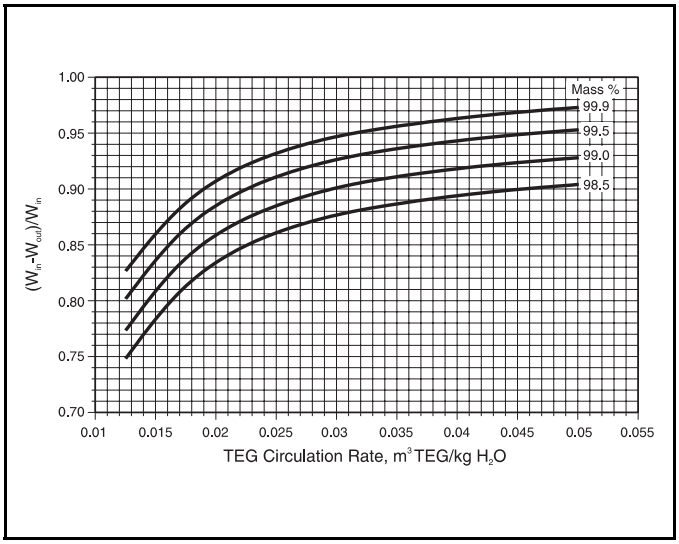
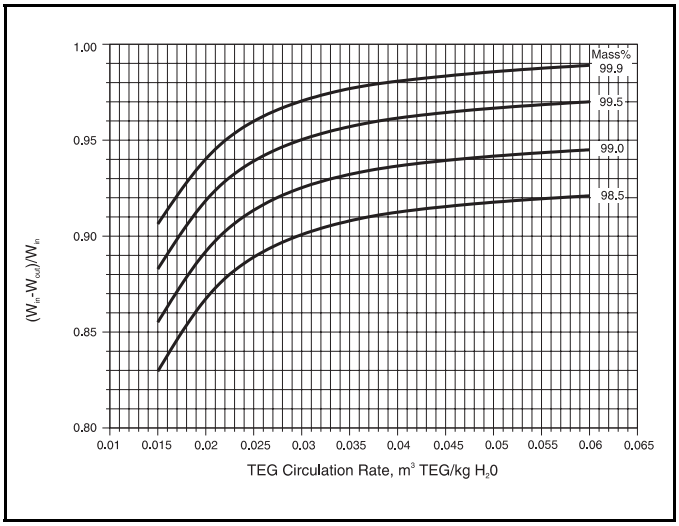


FIG. 20-71

Water Removal vs. TEG Circulation Rate at Various TEG Concentrations (N = 2.0)



and the period of regeneration should be decreased by multiplying it times the ratio V / V_{min}. A method more exact than Fig. 20-88 for calculating the minimum superficial velocity is to use Eq 20-16, but to consider it in terms of $(\Delta P/L)_{min}$ and V_{min} instead of $(\Delta P/L)_{max}$ and V_{max}.

FIG. 20-84

Mole Sieve Capacity Correction for Unsaturated Inlet Gas

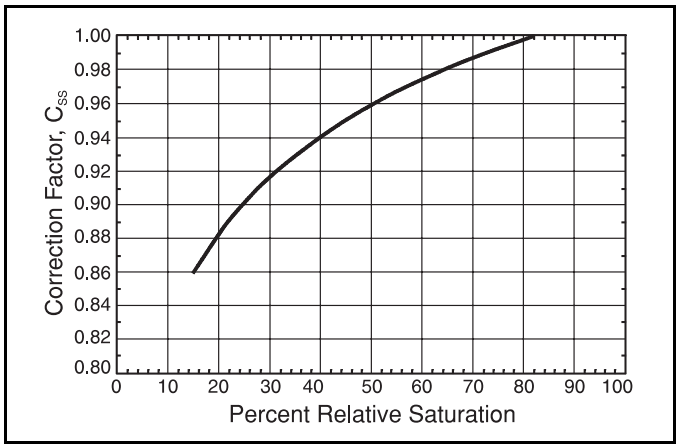


FIG. 20-85

Mole Sieve Capacity Correction for Temperature

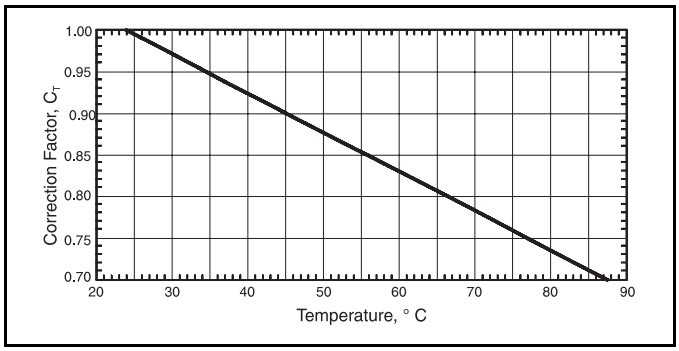
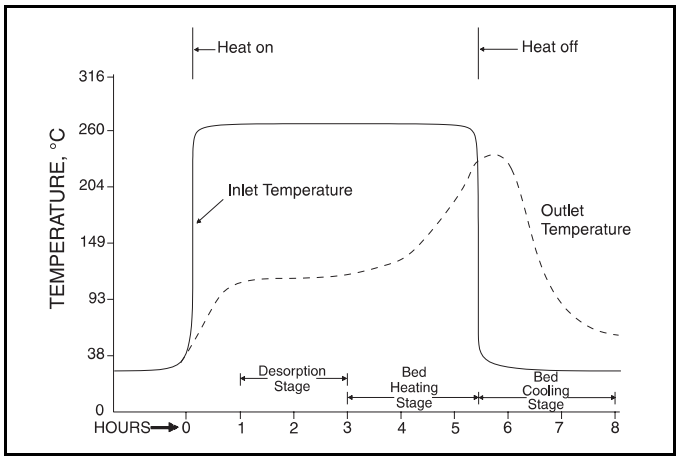


FIG. 20-86

Inlet and Outlet Temperatures During Typical Solid Desiccant Bed Regeneration Period



General Comments

The regeneration cycle frequently includes depressuring/repressuring to match the regeneration gas pressure and/or to maximize the regeneration gas volume to meet the velocity criterion. In these applications, the rate of depressuring or repressuring should not exceed 350 kPa per/minute. Some applications, termed pressure swing adsorption, regenerate the bed only with depressurization and sweeping the bed with gas just above atmospheric pressure.

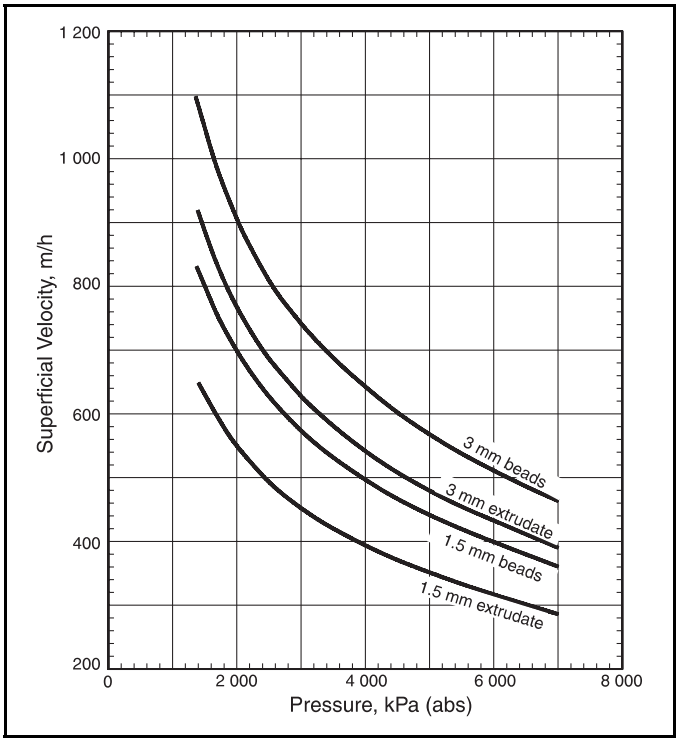
Moisture analyzers for very low water contents require care to prevent damage to the probes. When inserted into the beds, sample probes and temperature probes must be installed to reach the center of the gas phase.

Solid desiccant towers are insulated externally or possibly internally. Internal refractory requires careful installation and curing, usually before the desiccant is installed. It saves energy but the greatest benefit is it can dramatically reduce the required heating and cooling times. This is often an important benefit for systems where regeneration times are limited. The primary disadvantage is the potential for wet gas bypassing the desiccant through cracks and defects in the insulation during the adsorption cycle.

Example 20-13: 2.85×10^6 Sm^3/day of natural gas with a molecular weight of 18 is to be processed for ethane recovery in a turbo-expander plant. It is water saturated at 4140 kPa (abs) and 38°C and must be dried to –101°C dew point. Determine the water content of the gas, and the amount of water that must be removed; and do a preliminary design of a molecular-sieve dehydration system consisting of two towers with down-flow dehydration in one tower and up-flow regeneration in the other. Use 4A molecular sieve of 3.2 mm beads (i.e., 4x8 mesh). The regeneration gas is part of the plant's residue gas, which

FIG. 20-87

Allowable Velocity for Mole Sieve Dehydrator



Chemistry — The overall equilibrium reactions applicable for H_2S and CO_2 and primary and secondary amines are shown below with a primary amine[9]. A qualitative estimation of the velocity of the reaction is given.

For hydrogen sulfide removal

$$RNH_2 + H_2S \leftrightarrow RNH_3^+ + HS^- \text{ Fast} \qquad \textbf{Eq. 21-1}$$

$$RNH_2 + HS^- \leftrightarrow RNH_3^+ + S^{--} \text{ Fast} \qquad \textbf{Eq. 21-2}$$

The overall reactions between H_2S and amines are simple since H_2S reacts directly and rapidly with all amines to form the bisulfide by Eq. 21-1 and the sulfide by Eq. 21-2.

For carbon dioxide removal

$$2\,RNH_2 + CO_2 \leftrightarrow RNH_3^+ + RNHCOO^- \text{ Fast} \qquad \textbf{Eq. 21-3}$$

$$RNH_2 + CO_2 + H_2O \leftrightarrow RNH_3^+ + HCO_3^- \text{ Slow} \qquad \textbf{Eq. 21-4}$$

$$RNH_2 + HCO_3^- \leftrightarrow RNH_3^+ + CO_3^{--} \text{ Slow} \qquad \textbf{Eq. 21-5}$$

FIG. 21-3

Approximate Guidelines for Amine Processes[1]

	MEA	DEA[9]	DGA®	Sulfinol	MDEA[9]
Acid gas pickup, m^3/100L @ 38°C, normal range[2]	2.3–3.2	2.85–6.4	3.5–5.40	3.0–12.75	2.2–6.4
Acid gas pickup, mol/mol amine, normal range[3]	0.33–0.40	0.20–0.80	0.25–0.38	NA	0.20–0.80
Lean solution residual acid gas, mol/mol amine, normal range[4]	0.12 ±	0.01 ±	0.06 ±	NA	0.005–0.01
Rich solution acid gas loading, mol/mol amine, normal range[3]	0.45–0.52	0.21–0.81	0.35–0.44	NA	0.20–0.81
Solution concentration, wt%, normal range	15–25	30–40	50–60	3 components. Varies.	40–50
Approximate reboiler heat duty, kJ/L lean solution[5]	280–335	235–280	300–360	100–210	220–250
Steam heated reboiler tube bundle, approx. average heat flux, Q/A = MJ/(h • m^2)[6]	100–115	75–85	100–115	100–115	75–85
Direct fired reboiler fire tube, average heat flux, Q/A = MJ/(h • m^2)[6]	90–115	75–85	90–115	90–115	75–85
Reclaimer, steam bundle or fire tube, average heat flux, Q/A = MJ/(h • m^2)[6]	70–90	NA[7]	70–90	NA	NA[7]
Reboiler temperature, normal operating range, °C[8]	107–127	110–127	121–132	110–138	110–132
Heats of reaction;[10] approximate: kJ/kg H_2S	1420	1290	1570	N/A	1230
kJ/kg CO_2	1920	1700	2000	N/A	1425
NA — not applicable or not available					

NOTES:

1. These data alone should not be used for specific design purposes. Many design factors must be considered for actual plant design.
2. Dependent upon acid gas partial pressures and solution concentrations.
3. Dependent upon acid gas partial pressures and corrosiveness of solution. Might be only 60% or less of value shown for corrosive systems.
4. Varies with stripper overhead reflux ratio. Low residual acid gas contents require more stripper trays and/or higher reflux ratios yielding larger reboiler duties.
5. Varies with stripper overhead reflux ratios, rich solution feed temperature to stripper and reboiler temperature.
6. Maximum point heat flux can reach 230 to 285 MJ/(h • m^2) at highest flame temperature at the inlet of a direct fired fire tube. The most satisfactory design of firetube heating elements employs a zone by zone calculation based on thermal efficiency desired and limiting the maximum tube wall temperature as required by the solution to prevent thermal degradation. The average heat flux, Q/A, is a result of these calculations.
7. Reclaimers are not used in DEA and MDEA systems.
8. Reboiler temperatures are dependent on solution conc. flare/vent line back pressure and/or residual CO_2 content required. It is good practice to operate the reboiler at as low a temperature as possible.
9. According to Total.
10. The heats of reaction vary with acid gas loading and solution concentration. The values shown are average.[10.]