
Natural Gas Gathering

Course No: R04-002

Credit: 4 PDH

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NATURAL GAS GATHERING

1. INTRODUCTION

This course provides guidance on the process of natural gas gathering. Basic physic laws, constants and formulas are provided at the beginning of this course to get a better understanding of their applications in the example problems illustrated later in the course. Several discussion points are introduced and illustrated further through the example calculations provided. While some of the values and formulas used may be familiar to those who more advanced in the topic being presented; they may serve as a refresher or a reference to others. Many of these values and formulas depend on using charted curves which in most cases were developed from equations (provided below) intended to be adapted into software programs.

2. PHYSIC LAWS

Avogadro Law: The number of moles is the same for all gas existing at the same pressure, volume and temperature. This shows that ideal gas represents a state of matter. When an actual gas is lowered towards zero pressure, the relationship $PV = \text{moles} \times RT$ holds true. If the pressure of an actual gas is lowered towards zero pressure, the temperature will converge to the same fixed value.

From this a universal gas constant (\overline{R}) was developed.

Boyles Law: If the temperature of a fixed mass of gas is held constant and the volume of the gas is varied, the pressure exerted by the gas varies in a relation such that the pressure x volume remains essentially constant (absolute pressure x volume = constant). Therefore, two different states of gas at the same temperature can be expressed in the following equation: $\text{pressure}_1 \times \text{volume}_1 = \text{pressure}_2 \times \text{volume}_2$; or $(P_1)(V_1) = (P_2)(V_2)$

Charles Law: The pressure of a gas varies directly with the temperature when the volume is held constant. Similarly, the volume varies directly with the temperature when the pressure is held constant.

$$\frac{V_1}{V_2} = \frac{T_1}{T_2} \quad \text{or} \quad \frac{P_1}{P_2} = \frac{T_1}{T_2}$$

Boyles & Charles Law: When there is a mixed gas, the total pressure of the mixed gas at a certain temperature is determined by the sum of the volumes for each separate gas in the mixture. The equation can be written as shown below:

$$\frac{\text{Total Pressure of mixed gas}}{\text{Total moles of mixed gas}} = \frac{\text{Partial pressure of separate gas}}{\text{Moles of separate gas}}$$

One mole has a mass numerically equal to the molecular weight of the substance.

Guy Lussac Law: Universal gas constant (\bar{R}) = (Pressure x volume) / (number of moles x °Rankine)

R is the amount of work which accompanies a one-lb gas system when it expands, or is compressed, “reversibly” (without friction), with a possible change of pressure and possible change in temperature of one °Rankine.

$$\text{Universal gas constant (R)} = \frac{PV}{T}$$

$$\text{Ideal universal gas constant } (\bar{R}) = 1545 \text{ ft-lb}_{(\text{force})} / \text{lb}_{(\text{mass})} \times \text{°Rankine}$$

$$\text{Individual gas constant } R_g = 1545 / \text{molecular weight}$$

Individual gas constant R_g is expressed in external work when a pound of gas is heated at a constant pressure ($c_p - c_v$) = $R / 778$

$$\text{Molar gas constant for water vapor } (R_{(\text{stm})}) = 1545/18.02 = 85.74 \frac{\text{ft-lb}}{\text{lb}_{(\text{mass})} \times \text{°Rankine}}$$

$$\text{Molar gas constant for air } (R_{(\text{air})}) = 1545/28.95 = 53.37 \frac{\text{ft-lb}}{\text{lb}_{(\text{mass})} \times \text{°Rankine}}$$

Gibbs-Dalton Law: The pressure, internal energy and entropy of a mixed gas are equal to the sums of the separate gases when each gas has the same volume and temperature of the mixed gas.

3. PHYSIC CONSTANTS AND FORMULAS

In this course, we will assume physic ideal gas formulas to be the same as actual gas.

- A. Absolute humidity is the mass (lb) of dry bulb air divided by the volume (ft³) dry bulb dry air in a given quantity of a mixture of air and water vapor; also referred to as density.
- B. Absolute temperature is the state where there is no molecular motion of individual atoms. In the real world it is necessary to have an absolute temperature with a positive value.
- C. Brake horse power (HP) = 2,543 Btu/hr

- D. British thermal unit (BTU) = In the form of energy, it is 778 ft-lb of work. Work is defined as the amount of one pound of force applied on a body that moves one foot in the same direction as the force is applied.
- E. British thermal unit (BTU) = In the form of heat, it is the quantity of heat needed to raise one lb. of water one °F.
- F. Bone dry air is air without water vapor.
- G. Cubic feet per pound (ft³/lb) of ideal gas at 60° and 14.7 psia = 379.49 ft³ / lb-molecular weight.
- H. Degree Rankine (°R) is the absolute temperature that relates to the Fahrenheit scale. The formula is degree Fahrenheit (°F) + 460°.
- I. Degree Kelvin (K) is the absolute temperature that relates to the Celsius scale. The formula is degree Celsius (°C) + 273°.
- J. Dew point temperature is the temperature where the partial pressure of the water vapor is at the saturation pressure of the mixture of water vapor and air.
- K. Dry bulb temperature is the temperature where the mixture of water vapor and air is in equilibrium. Therefore the water vapor is not at its saturation temperature.
- L. Enthalpy (Btu/lb) = The amount of heat released when a substance changes phase with no change in temperature.
- M. Entropy (Btu/lb) = Energy of a substance that is no longer available to perform work. Specific entropy reference of steam is zero at 32 °F.
- N. Liquid flow (lb/hr) = Liquid gallons per minute x 8.337 lb./gallon x 60 minutes/hour x specific gravity of the liquid at the flowing temperature.
- O. One Btu = In the form of mechanical work is 778 ft-lb force.
- P. One horsepower (HP) = 42.4 Btu/min = 500 ft-lb/second = 0.7457 kw
- Q. One horsepower-hour (heat power) = 2,545 Btu
- R. One horsepower-hour = 1.980 x 10⁶ ft-lb
- S. One horsepower = 33,000 ft-lb of work / minute.
- T. One joule (J) for energy = 0.00094845 British thermal unit (Btu)
- U. One joule (J) for work = 0.7376 ft-lb force = one Newton-meter

- V. One Kilogram-Calorie = The quantity of heat need to raise one kilogram of water one °C.
- W. One Kilogram-Calorie = 3.968 Btu
- X. One Kilowatt Hour = 3412.75 Btu
- Y. One ton of heat = The amount of heat needed to melt one ton of ice in one day (12,000,000 Btu/hr).
- Z. Pound per square inch absolute (psia) = Pounds per square inch gage (psig) plus atmospheric pressure (usually accepted to be 14.7 psia). In compressor design calculations, atmospheric pressure is generally accepted to be 14.4 psia.
- AA. Relative humidity (ϕ) is the mass (lb) of water vapor in dry bulb air divided by the maximum amount (lb) of water vapor that the dry bulb air is capable of holding at a specific temperature and pressure. With water vapor and dry bulb air at the same temperature and pressure the relative humidity of 50% is one-half the amount of water vapor the dry bulb air is capable of holding. At 100% relative humidity the water vapor condenses (dew point) from the dry bulb air:
- BB. Specific Gravity of gas (dimensionless) at 60 °F and 14.7 psia = Gas molecular weight / air molecular weight (28.95). For most engineering calculations 60 °F is used as the standard temperature, although some use 68 °F. In this course we will use 60 °F.
- CC. Specific Heat of gas (Btu / lb °F) (c_p) is the amount of heat in Btu needed to raise the temperature of one pound of gas one °F.
- DD. Specific Humidity (SH) (humidity ratio) is the ratio of the mass of water vapor divided by the mass of dry bulb air plus mass of water vapor
- EE. Thermal Conductivity (Btu/hr-ft-°F) K is is the ability of a substance to transfer heat, between two same size areas having a heat differential, per unit length.
- FF. Viscosity (absolute viscosity) centipoises (cP) = Viscosity centistokes (kinematic viscosity) x specific gravity (SG).
- GG. Wet bulb temperature is the temperature where the partial pressure of water vapor is greater than the pressure of the surrounding dry bulb temperature. Therefore, water evaporates.
- HH. Wet Standard atmospheric coefficient = 6.11. This dimensionless coefficient is used in formulas to determine the vapor pressure and saturation pressure at elevations above sea level.

FIG. 16 - 1

Physical Constants of Hydrocarbons¹

No.	Compound	Formula	Molecular weight	Boiling point °F., 14.696 psia	Vapor pressure, 100°F., psia	Freezing point, °F., 14.696 psia	Critical constants			
							Pressure, psia	Temperature, °F.	Volume, cu ft./lb	
1	Methane	CH ₄	16.043	-258.69	(5000)	-296.46 ^d	667.8	-116.63	0.0991	
2	Ethane	C ₂ H ₆	30.070	-127.48	(800)	-297.89 ^d	707.8	90.09	0.0788	
3	Propane	C ₃ H ₈	44.097	-43.67	190.	-305.84 ^d	616.3	206.01	0.0737	
4	n-Butane	C ₄ H ₁₀	58.124	31.10	51.6	-217.05	550.7	305.65	0.0702	
5	Isobutane	C ₄ H ₁₀	58.124	10.90	72.2	-255.29	529.1	274.98	0.0724	
6	n-Pentane	C ₅ H ₁₂	72.151	96.92	15.570	-201.51	488.6	385.7	0.0675	
7	Isopentane	C ₅ H ₁₂	72.151	82.12	20.44	-255.83	490.4	369.10	0.0679	
8	Neopentane	C ₅ H ₁₂	72.151	49.10	35.9	2.17	464.0	321.13	0.0674	
9	n-Hexane	C ₆ H ₁₄	86.178	155.72	4.956	-139.58	436.9	453.7	0.0688	
10	2-Methylpentane	C ₆ H ₁₄	86.178	140.47	6.767	-244.63	436.6	435.83	0.0681	
11	3-Methylpentane	C ₆ H ₁₄	86.178	145.89	6.098	—	453.1	448.3	0.0681	
12	Neohexane	C ₆ H ₁₄	86.178	121.52	9.856	-147.72	446.8	420.13	0.0667	
13	2,3-Dimethylbutane	C ₆ H ₁₄	86.178	136.36	7.404	-199.38	453.5	440.29	0.0665	
14	n-Heptane	C ₇ H ₁₆	100.205	209.17	1.620	-131.05	396.8	512.8	0.0691	
15	2-Methylhexane	C ₇ H ₁₆	100.205	194.09	2.271	-180.89	396.5	495.00	0.0673	
16	3-Methylhexane	C ₇ H ₁₆	100.205	197.32	2.130	—	408.1	503.78	0.0646	
17	3-Ethylpentane	C ₇ H ₁₆	100.205	200.25	2.012	-181.48	419.3	513.48	0.0665	
18	2,2-Dimethylpentane	C ₇ H ₁₆	100.205	174.54	3.492	-190.86	402.2	477.23	0.0665	
19	2,4-Dimethylpentane	C ₇ H ₁₆	100.205	176.89	3.292	-182.63	396.9	475.95	0.0668	
20	3,3-Dimethylpentane	C ₇ H ₁₆	100.205	186.91	2.773	-210.01	427.2	505.85	0.0662	
21	Triptane	C ₇ H ₁₆	100.205	177.58	3.374	-12.82	428.4	496.44	0.0636	
22	n-Octane	C ₈ H ₁₈	114.232	258.22	0.537	-70.18	360.6	564.22	0.0690	
23	Diisobutyl	C ₈ H ₁₈	114.232	228.39	1.101	-132.07	360.6	530.44	0.0676	
24	Isooctane	C ₈ H ₁₈	114.232	210.63	1.708	-161.27	372.4	519.46	0.0656	
25	n-Nonane	C ₉ H ₂₀	128.259	303.47	0.179	-64.28	332.	610.68	0.0684	
26	n-Decane	C ₁₀ H ₂₂	142.286	345.48	0.0597	-21.36	304.	652.1	0.0679	
27	Cyclopentane	C ₅ H ₁₀	70.135	120.65	9.914	-136.91	653.8	461.5	0.059	
28	Methylcyclopentane	C ₆ H ₁₂	84.162	161.25	4.503	-224.44	548.9	499.35	0.0607	
29	Cyclohexane	C ₆ H ₁₂	84.162	177.29	3.264	43.77	591.	536.7	0.0586	
30	Methylcyclohexane	C ₇ H ₁₄	98.189	213.68	1.609	-195.87	503.5	570.27	0.0600	
31	Ethylene	C ₂ H ₄	28.054	-154.62	—	-272.45 ^d	729.8	48.58	0.0737	
32	Propene	C ₃ H ₆	42.081	-53.90	226.4	-301.45 ^d	669.	196.9	0.0689	
33	1-Butene	C ₄ H ₈	56.108	20.75	63.05	-301.63 ^d	583.	295.6	0.0685	
34	Cis-2-Butene	C ₄ H ₈	56.108	38.69	45.54	-218.06	610.	324.37	0.0668	
35	Trans-2-Butene	C ₄ H ₈	56.108	33.58	49.80	-157.96	595.	311.86	0.0680	
36	Isobutene	C ₄ H ₈	56.108	19.59	63.40	-220.61	580.	292.55	0.0682	
37	1-Pentene	C ₅ H ₁₀	70.135	85.93	19.115	-265.39	590.	376.93	0.0697	
38	1,2-Butadiene	C ₄ H ₆	54.092	51.53	(20.)	-213.16	(653.)	(339.)	(0.0649)	
39	1,3-Butadiene	C ₄ H ₆	54.092	24.06	(60.)	-164.02	628.	306.	0.0654	
40	Isoprene	C ₅ H ₈	68.119	93.30	16.672	-230.74	(558.4)	(412.)	(0.0650)	
41	Acetylene	C ₂ H ₂	26.038	-119 ^e	—	-114. ^d	890.4	95.31	0.0695	
42	Benzene	C ₆ H ₆	78.114	176.17	3.224	41.96	710.4	552.22	0.0531	
43	Toluene	C ₇ H ₈	92.141	231.13	1.032	-138.94	595.9	605.55	0.0549	
44	Ethylbenzene	C ₈ H ₁₀	106.168	277.16	0.371	-138.91	523.5	651.24	0.0564	
45	o-Xylene	C ₈ H ₁₀	106.168	291.97	0.264	-13.30	541.4	675.0	0.0557	
46	m-Xylene	C ₈ H ₁₀	106.168	282.41	0.326	-54.12	513.6	651.02	0.0567	
47	p-Xylene	C ₈ H ₁₀	106.168	281.05	0.342	55.86	509.2	649.6	0.0572	
48	Styrene	C ₈ H ₈	104.152	293.29	(0.24)	-23.10	580.	706.0	0.0541	
49	Isopropylbenzene	C ₉ H ₁₂	120.195	306.34	0.188	-140.82	465.4	676.4	0.0570	
50	Methyl Alcohol	CH ₃ O	32.042	148.1(2)	4.63(22)	-143.82(22)	1174.2(21)	462.97(21)	0.0589(21)	
51	Ethyl Alcohol	C ₂ H ₅ O	46.069	172.92(22)	2.3(7)	-173.4(22)	925.3(21)	469.58(21)	0.0580(21)	
52	Carbon Monoxide	CO	28.010	-313.6(2)	—	-340.6(2)	507.(17)	-220.(17)	0.0532(17)	
53	Carbon Dioxide	CO ₂	44.010	-109.3(2)	—	—	1071.(17)	87.9(23)	0.0342(23)	
54	Hydrogen Sulfide	H ₂ S	34.076	-76.6(24)	394.0(6)	-117.2(7)	1306.(17)	212.7(17)	0.0459(24)	
55	Sulfur Dioxide	SO ₂	64.059	14.0(7)	88.(7)	-103.9(7)	1145.(24)	315.5(17)	0.0306(24)	
56	Ammonia	NH ₃	17.031	-28.2(24)	212.(7)	-107.9(2)	1636.(17)	270.3(24)	0.0681(17)	
57	Air	N ₂ O ₂	28.964	-317.6(2)	—	—	547.(2)	-221.3(2)	0.0517(3)	
58	Hydrogen	H ₂	2.016	-423.0(24)	—	—	434.8(24)	-399.8(17)	0.5167(24)	
59	Oxygen	O ₂	31.999	-297.4(2)	—	—	361.8(24)	736.9(24)	-181.1(17)	0.0382(24)
60	Nitrogen	N ₂	28.013	-320.4(2)	—	—	346.0(24)	493.0(24)	-232.4(24)	0.0514(17)
61	Chlorine	Cl ₂	70.906	-29.3(24)	158.(7)	-149.8(24)	1118.4(24)	291.(17)	0.0281(17)	
62	Water	H ₂ O	18.015	212.0	0.9492(12)	32.0	3208.(17)	705.6(17)	0.0500(17)	
63	Helium	He	4.003	—	—	—	—	—	—	
64	Hydrogen Chloride	HCl	36.461	-121(16)	925.(7)	-173.6(16)	1198.(17)	124.5(17)	0.0208(17)	

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4. NATURAL GAS FLOW CALCULATIONS

Most natural gas flows are composed of more than one common gas. Usually, what is given by the user is the flow in standard cubic foot per minute (SCFM) and the composition in molecular % (volume %). Standard temperature and pressure for gas are 60 °F and 14.7 psia, respectively. In most calculations the flow needs to be in pounds in hour (lb/hr).

Example 1

Given the natural gas flow of 1555 ACFM, operating pressure of 100 psia and temperature of 90 °F, find the SCFM.

In this case, the value is given in actual cubic feet per minute (ACFM) which needs to be converted to SCFM.

$$\text{SCFM} = \text{ACFM} \times \frac{\text{Oper. Press. (psia)}}{14.7 \text{ psia}} \times \frac{60^\circ + 460^\circ}{\text{Oper. Temp.} + 460^\circ}$$

$$\text{SCFM} = 1,555 \text{ ACFM} \times \frac{100 \text{ psia}}{14.7 \text{ psia}} \times \frac{60^\circ \text{F} + 460^\circ \text{F}}{90^\circ \text{F} + 460^\circ \text{F}} = 10,001 \text{ SCFM}$$

Example 2

Given a natural gas flow of 10,000 SCFM and the molecular weight of gas as 18.84, find the flow in lb/hr.

$$\text{Flow (lb/hr)} = \text{SCFM} \times 60 \text{ minutes} \times 24 \text{ hrs} \times \text{molecular weight} / 9100.$$

$$\text{Flow (lb/hr)} = 10,000 \times 60 \times 24 \times 18.84 / 9100 = 29,813 \text{ lb/hr}$$

Example 3

Given a mixed natural gas composition of Methane 80 mole % and Ethane 20 mole %, find the specific gravity (SG) of the mixed gas.

SG for individual gases = Gas molecular weight / air molecular weight. (Using Boyles & Charles Law, one mole has a mass numerically equal to the molecular weight of the substance. Therefore, one lb of air is 28.95.)

$$\text{Specific gravity Methane} = 16.04 / 28.95 = 0.554$$

$$\text{Specific gravity Ethane} = 30.07 / 28.95 = 1.039$$

$$\text{Mixed gas specific gravity (SG)} = (\text{mol}\%_1 \times \text{SG}_1) + (\text{mol}\%_2 \times \text{SG}_2)$$

$$\text{Mixed gas specific gravity (SG)} = (0.80 \times 0.554) + (0.20 \times 1.039) = 0.651$$

Example 4

Given mixed natural gas flow of 10,000 SCFM, a composition of Methane mole % of 80 and an Ethane mole % of 20, find the total mixed gas molecular weight, weight % of each gas, and flow (lb/hr) of each gas.

Composition	<u>Mole% (Volume %)</u>	<u>Molecular Weight</u>	<u>Pseudo Weight (Mole% x Mol Wt)</u>
Methane:	0.80	16.04	12.83
Ethane:	0.20	30.07	6.01

The mixed natural gas composition molecular weight is $12.83 + 6.01 = 18.84$.

Weight % of each separate gas = Pseudo Weight / Total Molecular Weight

Weight % of Methane = $12.83 / 18.84 = 0.681$

Weight % of Ethane = $6.01 / 18.84 = 0.319$

Flow of mixed gas (lb/hr) = (SCFM x 60 minutes x 24 hrs x molecular weight) / 9100.

Flow of mixed gas (lb/hr) = $(10,000 \times 60 \times 24 \times 18.84) / 9100 = 29,813$ lb/hr

Flow of each separate gas = Flow of mixed gas / Weight % of each gas

Flow of Methane = $29,813$ lb/hr x $0.681 = 20,303$ lb/hr

Flow of Ethane = $29,813$ lb/hr x $0.319 = 9,510$ lb/hr

Example 5

Given a natural gas with a total flow of 29,813 lb/hr, molecular weight of 18.84 and the composition is methane and ethane, find the flow of each component.

Let lb/hr of methane = X

Let lb/hr of ethane = Y

$X + Y = 29,813$ lb/hr

$Y = 29,813$ lb/hr - X

Total moles = $29,813$ lb/hr / $18.84 = 1582$ moles

$X / 16.04 + Y / 30.07 = 1,582$ moles

$1.875 X$ lb/hr + Y lb/hr = $47,571$ lb/hr

$1.875 X$ lb/hr + $(29,813$ lb/hr - X) = $47,571$ lb/hr

$0.875 X$ lb/hr = $17,758$ lb/hr

Methane lb/hr = $20,295$ lb/hr

Ethane lb/hr = $29,813$ lb/hr - $20,295$ lb/hr = $9,518$ lb/hr

Example 6

Note: If the quantity of flow (lb/hr or mole %), temperature and pressure of a mixed natural gas stream are known, the mixed density (ρ), specific heat (C_p), viscosity (Z) and thermal conductivity (K) can be determined. This is illustrated below.

Given a total mixed natural gas flow consisting of 20,251 lb/hr methane, 9,562 lb/hr ethane, an operating pressure of 900 psig and a temperature of 140 °F, find the mixed natural gas density, specific heat, viscosity and thermal conductivity.

$$\text{The density of gas lb/ft}^3 (\rho) = \frac{\text{Mol Wt} \times \text{Oper. Press. (psia)}}{1545/144 \times (\text{Oper. Temp.} + 460^\circ) \times \text{Compressibility factor}}$$

When the operating pressure of a gas is below 250 psig, the compressibility does not play a major role. Therefore, for relatively low pressures, it can be omitted for approximate density calculations. For the purpose of this example, compressibility will be included in the following calculation:

To obtain critical pressures and temperatures, refer to Fig. 16-1 included at the beginning of this course.

Methane Critical Pressure = 667.8 psia
Methane Critical Temperature = -116.63 °F
Ethane Critical Pressure = 707.8 psia
Ethane Critical Temperature = 90.09 °F

Reduced pressure methane:

$$P_r = P / P_c = (900 \text{ psig} + 14.7 \text{ psia}) / (667.8 \text{ psia}) = 1.370 \text{ psia}$$

Reduced temperature methane:

$$T_r = T / T_c = (140 \text{ °F} + 460 \text{ °F}) / (-116.63 \text{ °F} + 460 \text{ °F}) = 1.747 \text{ °F}$$

Reduced pressure ethane:

$$P_r = P / P_c = (900 \text{ psig} + 14.7 \text{ psia}) / (707.8 \text{ psia}) = 1.292 \text{ psia}$$

Reduced temperature ethane:

$$T_r = T / T_c = (140 \text{ °F} + 460 \text{ °F}) / (90.09 \text{ °F} + 460 \text{ °F}) = 1.091 \text{ °F}$$

Using Fig. 16-4 shown on the following page the compressibility factors can be determined as follows:

Compressibility factor (Z) for methane = 0.94

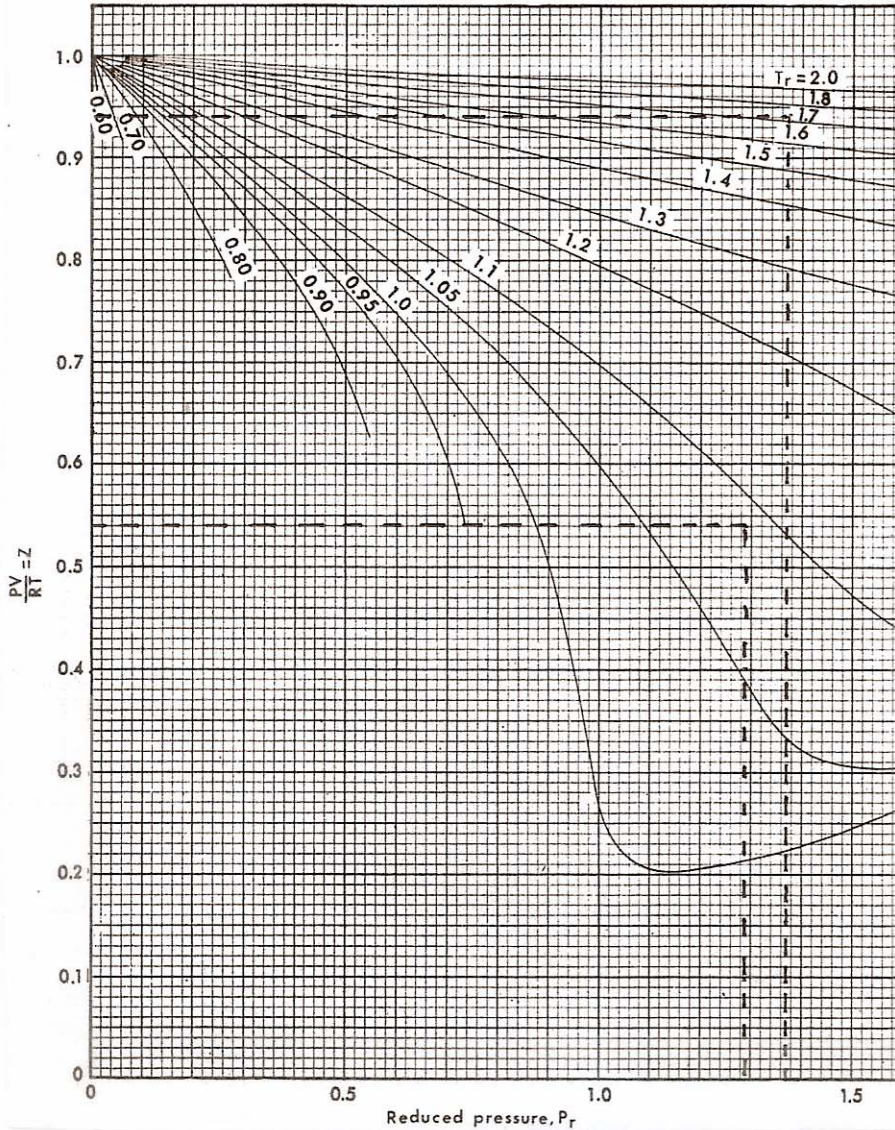
Compressibility factor (Z) for ethane = 0.54

$$\text{Density of methane lb/ft}^3 (\rho) = \frac{16.04 \times (900 \text{ psig} + 14.7 \text{ psia})}{10.73 \times (140 \text{ °F} + 460 \text{ °F}) \times 0.94} = 2.42 \text{ lb/ft}^3$$

$$\text{Density of ethane lb/ft}^3 (\rho) = \frac{30.07 \times (900 \text{ psig} + 14.7 \text{ psia})}{10.73 \times (140 \text{ °F} + 460 \text{ °F}) \times 0.54} = 7.91 \text{ lb/ft}^3$$

FIG. 16-4

Compressibility factors at low reduced pressures



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$$\text{Mixed gas density lb/ft}^3 (\rho) = \frac{\text{Total lb/hr of mixed gas}}{\frac{\text{lb/hr gas}_1}{\rho_1} + \frac{\text{lb/hr gas}_2}{\rho_2}}$$

$$\text{Mixed gas density lb/ft}^3 (\rho) = \frac{29,813}{\frac{20,251}{2.42} + \frac{9,562}{7.91}} = 3.11 \text{ lb/ft}^3$$

For the following calculations use the values shown below, which were taken from reference materials.

	<u>Specific heat (c) Btu / lb/ °F</u>	<u>Viscosity (cP) centipoise</u>	<u>Thermal conductivity (K) Btu/hr ft °F</u>
Methane	0.5611	0.0133	0.0257
Ethane	0.4867	0.0143	0.0209

Mixed gas specific heat (Btu/lb °F), $c_p = c_1 \times \text{wt}\%_1 + c_2 \times \text{wt}\%_2$

Mixed gas specific heat (Btu/lb °F), $c_p = (0.5611 \times 0.681) + (0.4867 \times 0.319) = 0.5374 \text{ Btu/lb } ^\circ\text{F}$

Mixed gas viscosity (centipoise), $cP = \frac{cP_1 \times \text{mol}\%_1 \times \text{mol wt}_1^{1/2} + cP_2 \times \text{mol}\%_2 \times \text{mol wt}_2^{1/2}}{\text{mol}\%_1 \times \text{mol wt}_1^{1/2} + \text{mol}\%_2 \times \text{mol wt}_2^{1/2}}$

Mixed gas viscosity (centipoise), $cP = \frac{(0.0133 \times 0.80 \times 16.04^{1/2}) + (0.0143 \times 0.20 \times 30.07^{1/2})}{(0.80 \times 16.04^{1/2}) + (0.20 \times 30.07^{1/2})}$

Mixed gas viscosity (centipoise), $cP = 0.0136 \text{ centipoise}$

Mixed gas thermal conductivity (Btu/hr-ft-°F), $K = K_1 \times \text{wt}\%_1 + K_2 \times \text{wt}\%_2$

Mixed gas thermal conductivity (Btu/hr-ft-°F), $K = 0.0257_1 \times 0.681_1 + 0.0209_2 \times 0.319_2$

Mixed gas thermal conductivity (Btu/hr-ft-°F), $K = 0.0242 \text{ Btu/hr-ft-}^\circ\text{F}$

Example 7

Note: Most mixed natural gas arrives at the gas gathering plant containing water vapor. This is illustrated below.

Given a mixed natural gas flow of 10,000 lb/hr, an operating pressure of 300 psia, a total molecular weight of 20 and mole % of the water vapor (steam) to be 10%, determine the condensing temperature of water vapor arriving at the gathering plant.

Moles of total mixed gas = $10,000 / 20 = 500 \text{ moles}$

Moles of water vapor = $500 \times 0.10 = 50 \text{ moles}$

$\frac{\text{Operating pressure total gas}}{\text{Moles of total gas}} = \frac{\text{Partial pressure of separate gas}}{\text{Moles of separate gas}}$

$\frac{300 \text{ psia}}{500 \text{ moles}} = \frac{\text{Partial pressure of water vapor}}{50 \text{ moles of water vapor}}$

Partial pressure of water vapor = $\frac{300 \text{ psia} \times 50 \text{ moles}}{500 \text{ moles}} = 30 \text{ psia}$

Condensing temperature of water vapor (steam) at a partial pressure of 30 psia is found in Keenan and Keyes steam tables to be 250 °F.

Example 8

Given a mixture of water vapor and natural gas with a total flow of 10,000 lb/hr, a total molecular weight of 20, an operating pressure of 300 psia, a temperature of 140 °F entering the gathering plant and a temperature of 120 °F at a water separator, find the amount of water vapor (steam) entering the gathering plant and at the separator. The pressure drop between the mixture entering the gathering plant and the separator is 2.0 psi.

1) Find the amount of water vapor entering the gathering plant:

Total moles of water vapor and natural gas = 10,000 lb/hr / 20 = 500 moles.

$$\frac{\text{Operating pressure of water vapor and natural gas}}{\text{Moles of water vapor and natural gas}} = \frac{\text{Partial pressure of water vapor}}{\text{Moles of water vapor}}$$

Partial pressure of water at 140 °F found in Keenan and Keyes steam tables is equal to 2.89 psia.

$$\frac{300 \text{ psia}}{500 \text{ moles}} = \frac{2.89 \text{ psia}}{\text{Moles of water vapor}}$$

$$\text{Moles of water vapor} = \frac{500 \text{ moles} \times 2.89 \text{ psia}}{300 \text{ psia}} = 4.82 \text{ moles}$$

$$\text{Flow (lb/hr) of water vapor} = 4.82 \text{ moles/hr} \times 18.02 \text{ lb/mole} = 87 \text{ lb/hr}$$

Total moles of water vapor and natural gas	=	500.00 moles
Moles of water vapor	=	<u>4.82 moles</u>
Moles of natural gas	=	495.18 moles

Check with the following formula:

Flow (lb/hr) of water vapor:

$$= \text{moles of non-condensable gases} \times \text{mol wt of steam} \times \frac{\text{vapor pressure of steam}}{\text{oper. pres.} - \text{vapor pres. of steam}}$$

$$= 495.18 \text{ moles} \times 18.02 \times \frac{2.89}{300 \text{ psia} - 2.89 \text{ psia}} = 87 \text{ lb/hr}$$

2) Find the amount of water vapor entering the separator:

Partial pressure of water vapor at 120 °F found in Keenan and Keyes steam tables is equal to 1.69 psia.

$$\frac{298 \text{ psia}}{500 \text{ moles}} = \frac{1.69 \text{ psia}}{\text{Moles of steam}}$$

$$\text{Moles of water vapor} = \frac{500 \text{ moles} \times 1.69 \text{ psia}}{298 \text{ psia}} = 2.84 \text{ moles}$$

$$\text{Flow (lb/hr) water vapor} = 2.84 \text{ moles/hr} \times 18.02 \text{ lb/mole} = 51 \text{ lb/hr}$$

Total moles of water vapor and natural gas	= 500.00 moles
Moles of water vapor	= <u>2.84 moles</u>
Moles natural gas	= 497.16 moles

Check with the following formula:

Flow (lb/hr) of water vapor:

$$= \text{moles of non-condensable gases} \times \text{mol wt of steam} \times \frac{\text{vapor pressure of steam}}{\text{oper. pres.} - \text{vapor pres. of steam}}$$

$$= 497.16 \text{ moles} \times 18.02 \times \frac{1.69}{298 \text{ psia} - 1.69 \text{ psia}} = 51 \text{ lb/hr}$$

Example 9

Discussion Point:

To determine the dew point temperature (evaporation temperature) in unsaturated air at dry bulb temperature, air is circulated over a wick dipped in water. Because the wick contains water and the partial pressure of the water vapor in the wick is greater than that of the water vapor in the circulating dry bulb air, the water evaporates from the wick. The evaporation of water vapor requires energy (Btu of latent heat).

When the air is circulated over a wick at the same dry bulb temperature as the air, the wick absorbs some of the water by lowering its initial temperature. The lowering of the wick temperature causes energy to flow from the air to the wick; thereby, reducing the air temperature. The greater the amount of air circulated over the wick, the greater will be the weight of water evaporated. This temperature is known as the wet bulb temperature.

The formulas shown below are based on sea level pressure. They can also be used for elevations above sea level. The atmospheric pressure decreases in pressure 4% for every 1,000-ft increase in elevation above sea level. For example the atmospheric pressure at an elevation of 3,000 ft is 13.01 psia. The pressure coefficient at 3,000 ft is: $(13.01 \text{ psia} / 14.7 \text{ psia} \times 6.11) = 5.4$. The less the pressure coefficient is from that at sea level (which is 6.11), the less the actual vapor pressure and saturation pressure will be from sea level. However, the relative humidity will not change.

Given an ambient dry bulb temperature of 85 °F at 14.7 psia atmospheric pressure and a dew point dry bulb temperature of 65 °F (determined with a calibration device), find the relative humidity, absolute humidity (dry bulb air density) and specific humidity.

1) Calculate the Relative Humidity:

T_f = Ambient dry bulb temperature in °F = 85 °F

T_{df} = Dew point dry bulb temperature °F = 65 °F

Standard atmospheric coefficient = 6.11

T_c = Ambient dry bulb temperature °C

$T_c = 5/9 (T_f - 32)$

$T_c = 5/9 (85 - 32) = 29.44 \text{ °C}$

T_{dc} = Dew point dry bulb temperature °C

$T_{dc} = 5/9 (T_{df} - 32)$

$T_{dc} = 5/9 (65 - 32) = 18.33 \text{ °C}$

E_s (saturation pressure in millibars) = $6.11 \times 10^{(7.5 \times T_c / 237.7 + T_c)}$

E_s (saturation pressure in millibars) = $6.11 \times 10^{(7.5 \times 29.44 / 237.7 + 29.44)} = 40.98 \text{ millibars}$

E (actual pressure in millibars) = $6.11 \times 10^{(7.5 \times T_{dc} / 237.7 + T_{dc})}$

E (actual pressure in millibars) = $6.11 \times 10^{(7.5 \times 18.33 / 237.7 + 18.33)} = 21.04 \text{ millibars}$

Relative humidity (RH) in percent = $(E/E_s) \times 100 = 21.04/40.98 \times 100 = 52\%$

2) Calculate the Absolute Humidity:

Gas Law: $D = P/T R_w$

P = Partial pressure of the water vapor in the mixture of the water vapor and air in Pascals (P_a)

R_w = Molar gas constant for water vapor = 461.5 (J/kg x °Kelvin)

R_a = Molar gas constant for air = 287.0 (J/kg x °Kelvin)

P_{ss} = Saturation pressure of the water vapor at the temperature of the mixture of water vapor and air.

$P_{ss} = 0.5959$ psia from Keenan and Keys steam tables.

T_w = Temperature in °Kelvin

$T_w = T^{\circ}\text{C} + 273^{\circ} = T^{\circ}\text{K}$

P_w = Water vapor pressure in Pascals (P_a)

P_w = Relative humidity (ϕ) x saturation pressure (P_{ss})

$P_w = 0.52 \times 0.5959$ psia x 6,895 $P_a/\text{psia} = 2,137 P_a$

D_w = Density of water vapor in dry bulb air in kg/m^3

$D_w = P_w / (T_w \times R_w)$

$D_w = 2,137 P_a / ((29.44^{\circ}\text{C} + 273^{\circ}) \times 461.5) \text{ J}/\text{kg} \times ^{\circ}\text{Kelvin} = 0.0153 \text{ kg}/\text{m}^3$

Absolute humidity (ρ) = $0.0153 \text{ kg}/\text{m}^3 \times 0.062428 (\text{lb}/\text{kg})/(\text{ft}^3/\text{m}^3) = 0.000955 \text{ lb}/\text{ft}^3$

Check with the formula: $\frac{\text{Mol Wt} \times P_a}{10.73 \times ^{\circ}\text{R}} = \frac{18.02 \times (0.52 \times 0.5959)}{10.73 \times (85 + 460)} = 0.000955 \text{ lb}/\text{ft}^3$

3) Calculate the Specific Humidity:

Partial pressure of dry bulb air (P_a) = $14.7 \text{ psia} - 0.5959 \text{ psia} = 14.1041 \text{ psia}$

Partial pressure of dry bulb air (P_a) = $14.1041 \text{ psia} \times 6,895 P_a / \text{psia} = 97,248 P_a$

Density (D_a) (density of dry bulb air) = $97,248 P_a / ((29.44^{\circ}\text{C} + 273^{\circ}) \times 287.0) \text{ J}/\text{kg} \times ^{\circ}\text{Kelvin}$

Density (D_a) (density of dry bulb air) = $1.1204 \text{ kg}/\text{m}^3$

Density (D_a) (density of dry bulb air) = $1.1204 \text{ kg}/\text{m}^3 \times 0.062428 (\text{lb}/\text{kg})/(\text{ft}^3/\text{m}^3) = 0.0700 \text{ lb}/\text{ft}^3$

Check with the formula: $\frac{\text{Mol Wt} \times P_a}{10.73 \times ^{\circ}\text{R}} = \frac{28.95 \times 14.1041}{10.73 \times (85 + 460)} = 0.0698 \text{ lb}/\text{ft}^3$

Specific humidity (SH) = $\frac{0.000955 \text{ lb}/\text{ft}^3}{0.07000 \text{ lb}/\text{ft}^3 + 0.000955 \text{ lb}/\text{ft}^3} = 0.01346 \text{ lb water vapor} / \text{lb dry bulb air}$

Example 10

Discussion Point:

To gather natural gas from different locations, it must be increased in pressure to move toward the gathering plant in a pipeline. In most cases the natural gas is compressed in a reciprocating compressor (non-centrifugal) which increases the temperature. This temperature must be lowered to meet pipeline requirements. The reciprocating compressor is discussed later in this course.

Usually the method of cooling is by forcing ambient air across finned tubes in an air-cooler bundle. The size of the air-cooler relates to how much the capital cost is to the user. To a large degree, the size of the air-cooler depends on the temperature of the ambient air which varies by local weather conditions.

The amount of heat to be dissipated by the air-cooler is set by the user by adjusting the natural gas flow, composition, temperature in and temperature out.

The temperature of the air inlet and air outlet of the air cooler is an important economic factor in the thermal design of the air-cooler. The closer the inlet and outlet air temperatures are to the inlet and outlet natural gas temperatures, the larger the air-cooler will be.

By forcing a greater quantity of air through the cooling section, the outlet air rise temperature will be less than with a smaller airflow. With the larger airflow the outlet air temperature approach to the natural gas inlet will be less than with a smaller airflow. In the thermal design of the air cooler, there is an economic consideration. The capital cost of the air-cooler should be weighed against the electric cost (horsepower) it takes to force the ambient air through the tube bundle by the fan. Also, the pressure drop of the natural gas through the air-cooler tubes is another consideration as the compressor cost may be more to move the natural gas down the pipeline. Some users set the electric (horsepower) evaluation high as this is a continuing cost.

Hydraulic fan drives are advantageous where low fan speed can achieve the required cooling. This low fan speed will result in using less horsepower than at a higher speed.

Given the heat to be exchanged, the temperature of natural gas inlet and outlet, the temperature ambient air temperature in and out and an estimated overall heat transfer coefficient, find the area of required heat transfer surface in an air-cooler.

Heat to be exchanged (Q) = 12,000,000 Btu/hr
 Temperature of natural gas in = 170 °F
 Temperature of natural gas out = 120 °F
 Temperature of ambient air in = 90 °F
 Temperature of ambient air out = 110 °F
 Estimated overall heat transfer coefficient (U) (Btu/hr-ft²-°F) = 2.6

$$\text{Area} = \frac{\text{Heat to be exchanged}}{\text{Estimated overall heat transfer coefficient} \times \text{Log mean temperature difference}}$$

$$\text{Log mean temperature difference (LMTD)} = \frac{\text{GTDD} - \text{LTTD}}{\text{Log}_{10} \frac{\text{GTDD}}{\text{LTTD}}}$$

Greater thermal temperature difference (GTDD) (Hot to Hot) = 170 °F - 110 °F = 60 °F
 Lesser thermal temperature difference (LTTD) (Cold to Cold) = 120 °F - 90 °F = 30 °F

$$\text{LMTD} = \frac{60\text{ }^{\circ}\text{F} - 30\text{ }^{\circ}\text{F}}{\text{Log}_{10} \frac{60\text{ }^{\circ}\text{F}}{30\text{ }^{\circ}\text{F}}} = 43.28\text{ }^{\circ}\text{F}$$

$$\text{Area} = \frac{12,000,000}{2.6 \times 43.28} = 106,640\text{ ft}^2$$

Example 11

Discussion Point:

The use of humidifiers, in conjunction with air-coolers is rare, although it can be a valuable tool. This application seems to be a lost art because the thermal designer may not be familiar with bone dry air, dry bulb temperature, wet bulb temperature, specific humidity, absolute humidity and relative humidity. Using humidifiers in ambient dry bulb air of 100 °F to 110°F can be lowered to 80 °F into the air-cooler fin tube section. In this course humidifiers with mist pads will be addressed.

Humidifiers are usually installed below the fan deck between the air-cooler support legs. Louvers are installed between the supports where there are no humidifiers and closed, when needed, so the ambient air is drawn by the fan through the humidifiers.

The humidifiers have spray nozzles that distribute water into mist pads. When the ambient mixture of water vapor and air at the dry bulb temperature is mixed in the mist pads with water vapor from spray nozzles, some of the water will evaporate and increase the relative humidity of the air. The energy (latent heat) of the evaporating water lowers the dry bulb temperature of the mixture of the water vapor and air temperature.

There are water collection trays below the humidifiers to catch the water that is not evaporated. Make-up water is needed to replenish the water evaporated. If there is well water available near the air cooler, it would be an advantage.

Another advantage of humidifiers is the air-cooler can be controlled over a wide range of ambient temperatures where the air-cooler may be undersized for operation during hot weather and oversized during cold weather.

Given a natural gas flow of 200,000 lb/hr, an inlet temperature of 170 °F, an outlet temperature of 120 °F and an ambient air temperature of 100 °F; if the thermal design of the air-cooler indicates that a required dry bulb air temperature of 84 °F is needed into the air-cooler fin tube section with an outlet dry bulb temperature of 94 °F, find the values needed to size the humidifier and the amount of water vapor the humidifier adds to the ambient air.

1) Find the humidifier sizing criteria:

The mixture of water vapor and ambient air are in equilibrium at the inlet temperature of 100 °F into the humidifier.

Heat to be exchanged = Flow of natural gas x specific heat of gas at average temperature x (temperature in – temperature out).

Heat to be exchanged = 200,000 lb/hr x 0.54 Btu/lb °F x (170 °F – 120 °F) = 5,400,000 Btu/hr.

The amount of air needed through the humidifier = $\frac{\text{Heat to be exchanged}}{\text{Specific heat of air x temp. difference}}$.

The amount of air needed = $\frac{5,400,000 \text{ Btu/hr}}{0.24 \text{ Btu/lb-}^\circ\text{F} \times (100 \text{ }^\circ\text{F} - 84 \text{ }^\circ\text{F})}$ = 1,406,250 lb/hr

Based on an efficient factor of 0.75 for the mist pads in the humidifier, the humidifier has to cool the dry bulb temperature of water vapor air mixture from 100 °F to 78.7 °F to achieve a dry bulb temperature of 84 °F.

$(100 \text{ }^\circ\text{F} - 84 \text{ }^\circ\text{F}) / 0.75 = 21.3 \text{ }^\circ\text{F}$. Hence, $(100 \text{ }^\circ\text{F} - 21.3 \text{ }^\circ\text{F}) = 78.7 \text{ }^\circ\text{F}$.

The size of the humidifier is determined from published literature and is based on the above air flow, an ambient temperature of 100 °F and a temperature through the humidifier of 78.7 °F.

2) Find the humidifier make-up water:

Heat exchanged by the dry bulb air through the humidifier = 5,400 000 Btu/hr.

The amount of water evaporated is equal to the heat exchanged / latent Btu/lb at the average dry bulb air temperature (from Keenan and Keys steam tables).

Amount of water evaporated = $\frac{5,400,000 \text{ Btu/hr}}{1,042 \text{ Btu/lb}}$ = 5,182 Lb/hr

Make-up water (GPM) = $\frac{5,182 \text{ lb/hr}}{8.337 \text{ lb /gallon} \times 60 \text{ min. / hr} \times \text{S.G.}} = \frac{5182 \text{ lb/hr}}{500 \times 0.955} = 10.85 \text{ GPM}$

5. COMPRESSOR CALCULATIONS

The other kind of energy that crosses a boundary besides heat is work. The definition of work is the result of a force acting on a moving body. The force must be combined with a continuous motion to be work (horsepower). The net flow of work per lb of gas =

$P_2V_2 - P_1V_1$. Multiplying the work done on one pound of natural gas or head by the number of pounds of natural compressed gas and dividing by 33,000, results in horsepower. One horsepower = 33,000 ft-lb of work/minute.

In most instances natural gas must be compressed at compressor stations to send the natural gas from the field in pipelines to the processing plant or from the processing plant to the customer.

In this course we will discuss the reciprocating compressor, which is the piston type with a suction stroke and discharge stroke. In the suction stroke (expansion cycle) the pressure decreases, the volume increases and density decreases. In the discharge stroke (compression cycle) the pressure increases, the volume decreases and the density increases.

The other type of natural gas compression is the turbine (centrifugal fan), which will not be discussed in this course.

The calculations for compressors in this course will be based on the standard base gas condition of 60 °F and 14.7 psia. It should be noted that the values used in most published curves for compressor horsepower calculations are 60 °F and 14.4 psia,

In compressor calculations, it is important to know the mode of operation of the compressor.

If the compressor does not have interior cooling, the process is adiabatic. This means, that heat is not added or removed from the system. If a cooling system is installed, where the exterior walls of the compressor has sufficient cooling to theoretically hold the gas at the same temperature of inlet and outlet, the process is polytropic (isothermal). This means that the specific heat of the gas in and out is constant.

In either the adiabatic process or polytropic process, if there is friction in the system, they are irreversible.

Although in some practices the values of “k” and “n” are interchangeable, this is not actually correct. The value of “n” can be larger or smaller than “k”.

In the polytropic process the coefficient “n” is used. The value “n” is determined by the compressor manufacturer, which depends on the volumetric efficiency. The volumetric efficiency is defined as the ratio of volume of gas drawn into the cylinder to the volume of the piston displacement, with both volumes taken at the same pressure and temperature.

In the adiabatic process a “k” value is used. It is important for two concepts to be understood. The first concept is that the specific heat (c_p) of gas is the amount of heat in Btu that it takes to raise one pound of gas one °F. In this case the pressure is held constant and the volume is allowed to vary while heat is added. While the heat is added,

in addition to raising the temperature, the gas expands and external work is done. This specific heat at constant pressure is denoted c_p .

The external work done = (Pressure x (volume₂ – volume₁)) / 778.

The second concept is if the volume of gas is held constant while heat is added, all the heat is used in increasing the internal energy by raising the temperature. This specific heat is denoted as c_v .

The external work done when a pound of gas is heated at constant pressure, is equal to the gas constant R / 778, or Mol wt x c_p - Mol wt x c_v = 1545 / 778

Rearranging: Mol wt x c_v = Mol wt x c_p – 1.986

k = specific heat at constant pressure (c_p) / specific heat at constant volume (c_v). = c_p / c_v .

There has been a lot of literature published based on the compression of air which will be briefly discussed but should not be confused with the main topic of this course, which is about natural gas.

For air at 60 °F the specific heat is 0.244 Btu / lb-°F

$$K \text{ for air} = \frac{\text{Mol wt} \times c_p}{\text{Mol wt} \times c_v} = \frac{28.95 \times 0.244}{28.95 \times 0.244 - 1.986} = 1.391$$

This value for k may be acceptable to use for air and ideal gases. However, most natural gas contains more than one gas. Therefore, it is necessary to find the c_p and c_v of the gas mixture.

Example 12

Find k if the natural gas has a composition of 80 mole% methane, 20 mole% of ethane and if the suction temperature is 100 °F and suction pressure is 50 psia.

<u>Component</u>	<u>Mole Fraction</u>	<u>Molecular Weight</u>	<u>Specific Heat at 100 °F</u>	<u>Fraction Specific Heat</u>
Methane	0.80	16.04	0.5393 Btu/lb-°F	0.4314 Btu/lb-°F
Ethane	0.20	30.07	0.3562 Btu/lb-°F	0.0712 Btu/lb-°F

$$k = \frac{\text{mol wt Methane} \times \text{fraction specific heat} + \text{mol wt Ethane} \times \text{fraction specific heat}}{\text{mol wt Methane} \times \text{fraction specific heat} + \text{mol wt Ethane} \times \text{fraction specific heat} - 1.986}$$

$$k = \frac{(16.04 \times 0.4314) + (30.07 \times 0.0712)}{(16.04 \times 0.4314) + (30.07 \times 0.0712) - 1.986} = 1.28$$

Example 13

Using the polytropic process (isothermal) for a compressor, find the discharge temperature (P_d) (without friction), given a suction temperature (T_s) of 100 °F, suction pressure (P_s) of 14.7 psia, discharge pressure (P_d) of 50 psia, and a gas composition of 80 mole% methane and 20 mole% ethane. Assume the volumetric efficiency (η) is 1.20.

Temperature out (°Rankine): $T_d = T_s (P_d / P_s)^{(n-1 / n)}$

$$\begin{aligned} T_d &= (100 \text{ °F} + 460^\circ) (50.0 \text{ psia} / 14.7 \text{ psia})^{(1.20 - 1 / 1.20)} \\ T_d &= 560 \text{ °Rankine} \times 3.40^{(1.20 - 1 / 1.20)} = 560 \text{ °Rankine} \times 3.40^{(0.167)} \\ T_d &= 560 \text{ °Rankine} \times 1.23 = 687 \text{ °Rankine} \\ \text{Temperature out } (T_d) \text{ in } ^\circ\text{F} &= 687 \text{ °Rankine} - 460^\circ = 227 \text{ °F} \end{aligned}$$

Example 14

Using the adiabatic process, find the discharge temperature (without friction) of natural gas which has a composition of 80 mole% methane and 20 mole% ethane. The intake suction temperature (T_s) is 100 °F, the suction pressure (P_s) is 14.7 psia and discharge pressure (P_d) is 50 psia. Use the k value of 1.28 calculated in Example 12.

Temperature out (°Rankine): $T_d = T_s (P_d / P_s)^{(k-1 / k)}$

$$\begin{aligned} T_d &= (100 \text{ °F} + 460^\circ) (50.0 \text{ psia} / 14.7 \text{ psia})^{(1.28 - 1 / 1.28)} \\ T_d &= 560 \text{ °Rankine} \times 3.40^{(1.28 - 1 / 1.28)} = 560 \text{ °Rankine} \times 3.40^{(0.219)} \\ T_d &= 560 \text{ °Rankine} \times 1.307 = 732 \text{ °Rankine} \end{aligned}$$

Temperature out (T_d) in °F = 732 °Rankine - 460 ° = 272 °F

It is apparent that the temperature of the natural gas out is lower using the polytropic (isothermal) process (as indicated in Example 13) than the adiabatic system. This results in reduced compressor horsepower because the gas is expanded less.

Example 15

Note: In some literature the polytropic (isothermal) process is shown to have a reversible compression cycle although the value of n is typically equal to 1 and PV^n is equal to a constant. In this case, the calculations shown in Example 15 below would apply and are only presented for clarification purposes only. However, in the actual operation of a compressor the system cannot be reversible.

Given a flow of 1,000 actual ft³/min of methane at 14.7 psia and a temperature of 85 °F; if the flow is compressed to 100 psia, find the compressor brake horsepower (bhp) using the reversible polytropic (isothermal) process.

$$P_s V_s = RT_s = 14.7 \text{ psi} \times 144 \text{ in}^2 / \text{ft}^2 \times (V_s) = 1545 / 16.04 \times (85 \text{ °F} + 460^\circ)$$

$$V_s = 24.80 \text{ ft}^3 / \text{lb}$$

$$\text{Work (Wk)} = P_s V_s (\ln_e) \frac{P_s}{P_d} = 14.7 \text{ psia} \times 144 \times 24.80 \text{ ft}^3 / \text{lb} (\ln_e) \frac{14.7 \text{ psia}}{100 \text{ psia}} = 52,497 \times (-1.92)$$

$$\text{Work (Wk)} = -100,794 \text{ ft-lb} / \text{lb}$$

$$\text{Amount of methane compressed (m)} = \frac{1,000 \text{ ft}^3 / \text{min}}{24.80 \text{ ft}^3 / \text{lb}} = 40.32 \text{ lb} / \text{min}$$

Multiplying the work done on one pound of gas or head by the number of pounds compressed per minute and dividing by 33,000 ft-lb of work / min gives horsepower.

$$\text{Brake horsepower} = \text{Wk} \times \text{m} / 33,000 \text{ ft-lb of work} / \text{min}$$

$$\text{Brake Horsepower} = (100,794 \text{ ft-lb} / \text{lb} \times 40.32 \text{ lb} / \text{min}) / 33,000 \text{ ft-lb of work} / \text{min}$$

$$\text{Brake Horsepower} = 123.2$$

The indicated gage horsepower = brake horsepower x mechanical efficiency.

Example 16

Using the nonreversible polytropic (isothermal) process, if an actual flow of 1,000 ft³/min of methane at 14.7 psia and a temperature of 85 °F is compressed per minute to 100 psia, find the compressor brake horsepower (bhp). Assume the volumetric efficiency (n) is 1.20.

$$P_s V_s = RT_s = 14.7 \text{ psi} \times 144 \text{ in}^2 / \text{ft}^2 \times (V_s) = 1545/16.04 \times (85 \text{ °F} + 460^\circ)$$

$$V_s = 24.80 \text{ ft}^3 / \text{lb}$$

$$\text{Work (WK)} = (n / n-1)(P_s V_s)(1 - (P_d / P_s)^{n-1/n})$$

$$\text{Work (WK)} = (1.20/0.20) \times (14.7 \text{ psi} \times 144 \text{ in}^2/\text{ft}^2 \times 24.80 \text{ ft}^3/\text{lb}) \times (1 - (100 \text{ psia}/14.7 \text{ psia})^{1.20-1/1.20})$$

$$\text{Work (WK)} = 6.00 \times 52,497 (1-1.377)$$

$$\text{Work (WK)} = -118,748 \text{ ft-lb} / \text{lb}$$

$$\text{Amount of methane compressed (m)} = \frac{1,000 \text{ ft}^3 / \text{min}}{24.80 \text{ ft}^3 / \text{lb}} = 40.32 \text{ lb} / \text{min}$$

Multiplying the work done on one pound of gas or head, by the number of pounds compressed per minute and dividing by 33,000 ft-lb of work / min, gives horsepower.

$$\text{Brake horsepower} = \text{Wk} \times \text{m} / 33,000 \text{ ft-lb of work} / \text{min}$$

$$\text{Brake Horsepower} = (118,748 \text{ ft-lb} / \text{lb} \times 40.32 \text{ lb} / \text{min}) / 33,000 \text{ ft-lb of work} / \text{min}$$

$$\text{Brake Horsepower} = 145.1$$

The indicated gage horsepower = brake horsepower x mechanical efficiency.

Example 17

Using the adiabatic process, if an actual flow of 1,000 ft³/min of methane at 14.7 psia and a temperature of 85 °F is compressed per minute to 100 psia, and given a calculated k value of 1.28, find the compressor brake horsepower (bhp).

$$P_s V_s = RT_s = 14.7 \text{ psi} \times 144 \text{ in}^2 / \text{ft}^2 \times (V_s) = 1545/16.04 \times (85 \text{ °F} + 460^\circ)$$

$$V_s = 24.80 \text{ ft}^3 / \text{lb}$$

$$\text{Work (Wk)} = (k / k-1)(P_s V_s)(1 - (P_d / P_s)^{k-1/k})$$

$$\text{Work (WK)} = (1.28/0.28) \times (14.7 \text{ psi} \times 144 \text{ in}^2/\text{ft}^2 \times 24.80 \text{ ft}^3/\text{lb}) \times (1 - (100 \text{ psia}/14.7 \text{ psia})^{1.28-1/1.28})$$

$$\text{Work (WK)} = 4.57 \times 52,497 (1-1.522)$$

$$\text{Work (WK)} = -125,234 \text{ ft-lb} / \text{lb}$$

$$\text{Amount of methane compressed (m)} = \frac{1,000 \text{ ft}^3 / \text{min}}{24.80 \text{ ft}^3 / \text{lb}} = 40.32 \text{ lb} / \text{min}$$

Multiplying the work done on one lb of gas or head, by the number of pounds compressed per minute and dividing by 33,000 ft-lb of work / min gives horsepower.

$$\text{Brake horsepower} = \text{Wk} \times \text{m} / 33,000 \text{ ft-lb of work} / \text{min}$$

$$\text{Brake Horsepower} = (125,234 \text{ ft-lb} / \text{lb} \times 40.32 \text{ lb} / \text{min}) / 33,000 \text{ ft-lb of work} / \text{min}$$

$$\text{Brake Horsepower} = 153.0$$

The indicated gage horsepower = brake horsepower x mechanical efficiency.

In evaluating whether a cooling system should be installed to cool a compressor and the cost associated with the system installation, it is useful to determine the ratio between the two efficiencies or amount of work to be done by the different systems.

The ratio of adiabatic process efficiency and polytropic (isothermal) process efficiency may be shown as:

$$\frac{\dot{\eta}_{(ad)}}{\dot{\eta}_{(iso)}} = \frac{W_{k(ad)}}{W_{k(iso)}}$$

6. CONCLUSION

This course provides guidance on how to provide for the existing and future electrical generation.

Natural gas is abundant in the United States. Therefore, the United States needs to remain competitive in the world in providing clean energy. Also, companies that have fleets of cars could cross over from gasoline to cleaning burning natural gas. This would result in long run profitability.

Waiting on nuclear power for energy seems to be at a standstill in getting government and environmental approval. Also, bio-fuels are not yet profitable to the producer without getting a government subsidy. Therefore, it is logical to use our most abundant and cost effective clean energy fuel which is natural gas.

7. REFERENCES

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