SECTION 5

CHEMICAL AND PROCESS PLANT ENGINEERING

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For additional calculation procedures in chemical engineering, please refer to the following sections in this handbook: Sec. 3, Mechanical Engineering; Sec. 4, Electrical Engineering; Sec. 6, Water and Waste-Water Engineering. Each section contains a number of calculation procedures pertinent to the content of Sec. 5, Chemical Engineering, but size limitations prevent their repetition in Sec. 5.

REFERENCES

Allen and Shonnard—*Green Engineering: Environmentally Conscious Design of Chemical Processes*, Prentice Hall; Biegler, Grossmann and Westerberg—*Systematic Methods of Chemical Process Design*, Prentice Hall; Branan—*Pocket Guide to Chemical Engineering*, Gulf Professional Publishing; Branan—*Rules of Thumb for Chemical Engineers*, Butterworth-Heinemann; Chopey and Hicks—*Handbook of Chemical Engineering Calculations*, McGraw-Hill; Cutlip and Shacham—*Problem Solving in Chemical Engineering with Numerical Methods*, Prentice Hall; Darby—*Chemical Engineering Fluid Mechanics*, Marcel Dekker; Deen—*Analysis of Transport Phenomena*, Oxford University Press; Felder and Rousseau—*Elementary Principles of Chemical Processes*, Wiley; Fogler—*Elements of Chemical Reaction Engineering*, Prentice Hall; Griskey—*Chemical Engineers' Portable Handbook*, McGraw-Hill; Hessel, Hardt and Löwe—*Chemical Micro Process Engineering: Fundamentals, Modelling and Reactions*, Wiley; Hill—*An Introduction to Chemical Engineering Kinetics and Reactor Design*, Wiley; Himmelblau—*Basic Principles and Calculations in Chemical Engineering*, Prentice Hall; Levenspiel—*Chemical Reaction Engineering*, Wiley; Ludwig—*Applied Process Design for Chemical and Petrochemical Plants*, Gulf Professional Publishing; McCabe, Smith and Harriott—*Unit Operations of Chemical Engineering*, McGraw-Hill; Missen, Mims and Saville—*Introduction to Chemical Reaction Engineering and Kinetics*, Wiley; Ogden—*Handbook of Chemical Engineering*, Research and Education Association; Perry and Green—*Perry's Chemical Engineers' Handbook*, McGraw-Hill; Peters and Timmerhaus—*Plant Design and Economics for Chemical Engineers*, McGraw-Hill; Reynolds, Theodorc and Jeris—*Handbook of Chemical and Environmental Engineering Calculations*, Wiley-Interscience; Smith, Van Ness and Abbott—*Introduction to Chemical Engineering Thermodynamics*, McGraw-Hill; Smith—*Chemical Engineering Kinetics*, McGraw-Hill; Swindin—*Pumps in Chemical Engineering*, Wexford College Press; Thomas—*Simulation of Industrial Processes for Control Engineers*, Butterworth-Heinemann; Turton, Bailie, Whiting et al.—*Analysis, Synthesis, and Design of Chemical Processes*, Prentice Hall; Varma and Morbidelli—*Mathematical Methods in Chemical Engineering*, Oxford University Press; Veasey, Wilson, Squires et al.—*The Physical Separation and Recovery of Metals from Wastes*, T & F STM; Walas—*Chemical Process Equipment, Selection and Design*, Butterworth-Heinemann; Wilkes—*Fluid Mechanics for Chemical Engineers*, Prentice Hall.

Chemical Engineering

ANALYSIS OF A SATURATED SOLUTION

If 1000 gal (3785.4 L) of water is saturated with potassium chlorate ($KClO_3$) at 80°C (176°F), determine (*a*) the weight, lb, of $KClO_3$ that will precipitate if the solution is cooled to 30°C (86°F) and (*b*) the weight of $KClO_3$ that will precipitate if one-half the 1000 gal (3785.4 L) of water is evaporated at 100°C (212°F).

Calculation Procedure

1. Compute the precipitate when the solution is cooled. When a solid is dissolved in water (or any other solvent liquid), the resulting solution is termed *saturated* when at a given temperature the solvent cannot dissolve any more of the solid. Most solvents dissolve (hold) more solids at higher temperatures than at lower temperatures. Thus, when the solution temperature is lowered or a portion of the solvent is evaporated, the solution becomes *supersaturated* and solid material may precipitate. This is the basis of *crystallization*, a chemical engineering operation frequently used to produce a purer or more crystalline product.

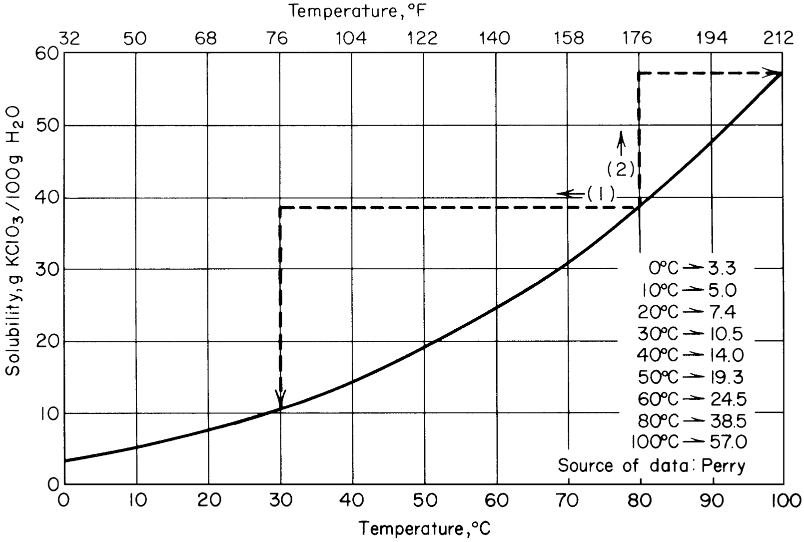


FIGURE 1 Solubility of $KClO_3$.

Referring to Fig. 1, obtain these solubilities: at 80°C (176°F), $KClO_3$ solubility = 38.5 g per 100 g H_2O; at 30°C (86°F), $KClO_3$ solubility = 10.5 g per 100 g of H_2O.

The weight of the water at 80°C (176°F) = (1000 gal H_2O)(0.97183 g H_2O per cm^3 H_2O) × (1 lb/454 g) = 8103 lb (3683.2 kg). Now, the weight of $KClO_3$ that any solvent can dissolve at a given temperature = weight of solvent at the given temperature, lb (solubility of $KClO_3$ at the given temperature, g per 100 g of the solvent). Or, at 80°C (176°F), weight of $KClO_3$ dissolved by the water = (8103 lb of water)(38.5 g $KClO_3$ per 100 g of H_2O) = 3119 lb (1417.7 kg) of $KClO_3$. And at 30°C (86°F) with the same quantity of water but the reduced solubility, the weight of $KClO_3$ that can be dissolved = (8103)(10.5 g per 100 g) = 851 lb (386.8 kg) of $KClO_3$.

When the temperature of the water (solvent) is reduced from 80 to 30°C (176 to 86°F), the weight of $KClO_3$ precipitated = weight of $KClO_3$ dissolved at 80°C (176°F) – weight of $KClO_3$ dissolved at 30°C (86°F), or 3119 – 851 = 2271 lb (1032.3 kg) of $KClO_3$ precipitated.

Note that the same procedure can be followed for any similar solution, i.e., any similar solvent and solid. Neither the solvent nor the solid need be the ones considered here.

2. Compute the precipitate when a portion of the solvent is evaporated. Since half the solvent (water in this case) is evaporated, the weight of water remaining = 8103/2 = 4051.5 lb (1841.6 kg). Using the solubility of $KClO_3$ as before, except that the solvent temperature is 100°C (212°F), we see the weight of $KClO_3$ dissolved = 4051.5(57 g $KClO_3$ per 100 g H_2O) = 2309 lb (1047.3 kg) of $KClO_3$. Then the weight of $KClO_3$ precipitated by the evaporation = weight of $KClO_3$ dissolved in 1000 gal (3785.0 L) of water at 80°C (176°F) – weight of $KClO_3$ dissolved in 500 gal (1892.5 L) of water at 100°C (212°F) = 3119 – 2309 = 810 lb (367.4 kg) of $KClO_3$ precipitated.

TERNARY LIQUID SYSTEM ANALYSIS

For a liquid mixture of 20 weight percent water, 30 weight percent acetic acid, and 50 weight percent isopropyl ether, determine the composition of the two phases (e.g., the ether layer and the water layer) and the amount of acetic acid that must be added to the system to form a one-phase (single-layer) solution.

Calculation Procedure

1. Compute the composition of the two layers. When two pure liquids are mixed, they will dissolve in each other to some degree. If they are completely soluble in each other, such as water and acetic acid, they are *miscible.*

If their mutual solubilities are zero, they are called immiscible. Between these extremes, liquids are partially miscible.

Addition of a third liquid component often affects the mutual solubilities of the two original liquids. The third liquid may be more soluble in one liquid than in another. This difference in solubilities is the basis of the chemical engineering operation termed *liquid-liquid extraction.*

The third liquid may cause immiscible liquids to become completely miscible, or the third liquid may produce miscibility only in certain concentration ranges. Such interrelationships can be shown graphically, as with the two parts in Fig. 2.

The *phase envelope*, Fig. 2, separates the two-phase region from the one-phase region. Note, Fig. 2, that the acetic acid and water are completely miscible, as indicated by the phase envelope not

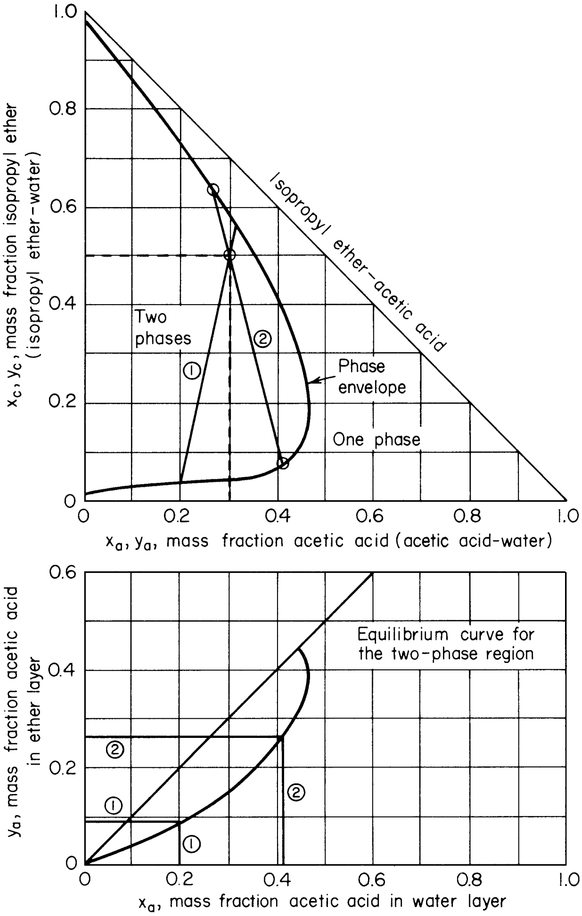


FIGURE 2 Liquid-system phase-envelope plot. (*After Anderson and Wenzel*—Introduction to Chemical Engineering, *McGraw-Hill.*)

touching the horizontal axis at any point. Likewise, the isopropyl ether and acetic acid are completely miscible. But water and isopropyl ether are virtually immiscible, as indicated by little of the vertical axis being free of the two-phase region of the phase envelope, Fig. 2.

The composition of the two phases, for the mixture in the two-phase region, is found on the phase envelope line itself, Fig. 2. Toward the lower part of the phase envelope, Fig. 2, is the water-rich layer, and toward the top of the phase envelope line is the ether-rich layer.

Plot on the upper portion of Fig. 2 the given values of acetic acid and isopropyl ether, that is, 30 weight percent, 50 weight percent. Through this point, draw a tie line to intersect the phase envelope at two points, line 1, Fig. 2. Read the values: *lower point*—acetic acid in water layer = 20 weight percent; *upper point*—acetic acid in isopropyl ether layer = 31.5 weight percent.

Transferring the lower intersection point to the bottom diagram for tie line 1 shows that equilibrium exists between a layer that is 20 weight percent acetic acid in water and a layer that is 9 weight percent (not 31.5 weight percent) acetic acid in isopropyl ether.

Draw a second tie line, 2, Fig. 2, as shown. Line 2 gives a check between the upper and lower diagrams, giving *water layer* – $x_a = 0.415$; $x_c = 0.065$; $x_w = 0.520$; *ether layer* – $y_a = 0.270$; $y_c = 0.650$; $x_w = 0.080$.

2. Compute the amount of acetic acid that must be added to form a one-phase system. The water/ether ratio remains unchanged at water/ether = 0.20/0.50 = 0.40. Then the total system is: water + ether + acid = 1.000; ∴ ether (weight percent) = [1.000 – acid (weight percent)]/1.40.

Assume that the acid = 0.350. Then ether = (1.000 – 0.350)/1.40 = 0.464. Checking against the upper diagram in Fig. 2, this point ($x_a = 0.350$, $x_c = 0.464$) falls inside the two-phase region. Hence, the assumption was incorrect.

As a second trial, assume acid = 0.380. Then ether = (1.000 – 0.380)/1.40 = 0.443. Checking $x_a = 0.380$, $x_c = 0.443$ in the upper diagram of Fig. 2 shows that the point falls exactly on the phase envelope line. Hence, it is at the minimum one-phase region.

DETERMINING THE HEAT OF MIXING OF CHEMICALS

How many Btu's of heat are released (generated) when 1000 lb (453.6 kg) of water at 80°C (176°F) is mixed with (1) 500 lb (226.8 kg) of aluminum bromide, $AlBr_3$; (2) 750 lb (340.3 kg) of barium nitrate, $Ba(NO_3)_2$; and (3) 1000 lb (453.6 kg) of dextrin, $C_{12}H_{20}O_{10}$?

Calculation Procedure

1. Compute the heat released when $AlBr_3$ is dissolved in water. When two or more substances are mixed, heat is usually generated or absorbed. The heat released or absorbed may be small when two similar organic liquids are mixed or very large when strong acids are mixed in water. The heat evolved (or absorbed) during the mixing of liquids is often called the *heat of dilution*, whereas the heat from the mixing of solids is often termed the *heat of solution*. Data for heats of solution for both organic and inorganic liquids and solids are given in Perry, Lange, and similar reference works.

Thus, at 80°C (176°F) the solubility of $AlBr_3$ in water is 126 g per 100 g of water. The weight of $AlBr_3$ that will dissolve in 1000 lb (454.5 kg) of water = (126/100)1000 = 1260 lb (572.7 kg). Standard references show that the heat of solution for $AlBr_3$ is 85.3 kg·cal per g·mol $AlBr_3$. Since the total Btu/(lb·mol) = 1.8 g·cal/g, the $AlBr_3$ in this solution can release [85.3 kg·cal/(g·mol)][1000 cal/(kg·cal)]1.8 = 153,540 Btu/(lb·mol) (357.1 kJ/mol).

The weight of 1 lb·mol of $AlBr_3$ = (27 + 79.9 × 3) = 266.7 lb (121.2 kg). Hence, the heat evolved when 500 lb (227.3 kg) of $AlBr_3$ is dissolved in water is {500 lb $AlBr_3$/[266.7 lb/(lb·mol)]} [153,540 Btu/(lb·mol)] = 287,800 Btu (303.6 kJ).

2. Compute the heat released in dissolving $Ba(NO_3)_2$ in water. At 80°C (176°F), the solubility of $Ba(NO_3)_2$ in water is 27.0 lb/lb (12.3 kg/kg) of water. The weight of $Ba(NO_3)_2$ that will dissolve in 1000 lb (454.5 kg) of water = (27/100)1000 = 270 lb (122.7 kg). Since 750 lb (340.9 kg) of $Ba(NO_3)_2$ is available for dissolving, the weight that will not dissolve is 750 – 270 = 480 lb (218.2 kg).

The heat of solution of $Ba(NO_3)_2$ is –10.2 kg·cal/(g·mol) of $Ba(NO_3)_2$. As in step 1, the weight of 1 lb·mol of $Ba(NO_3)_2$ = 137.34 + 2(14.0 + 3 × 16) = 261.4. Then, as in step 1, the heat released = [480 lb $Ba(NO_3)_2$/261.4 lb/(lb·mol)][–10.2 kg·cal/(g·mol) × 1000 g·cal/(kg·cal) × 1.8 [Btu/(lb·mol)/[g·cal/(g·mol)] = –33,300 Btu (–35,131.5 J).

The negative heat release means that 33,300 Btu (35,131.5 J) of heat must be added to the system to maintain the solution temperature at 80°C (176°F) because a fall in temperature would reduce the solubility of the $Ba(NO_3)_2$ in water and thus change the resulting solution.

3. Compute the heat released in dissolving $C_{12}H_{20}O_{10}$ in water. Perry indicates that there is no solubility limit for $C_{12}H_{20}O_{10}$ in water. By following the same procedure as in step 1, the heat released = [1000 lb $C_{12}H_{20}O_{10}$/324.2 lb/(lb·mol)] × {268 g·cal/(g·mol) × 1.8[Btu/(lb·mol)]/[g·cal/(g·mol)]} = 1488 Btu (1569.8 J) released.

Related Calculations Use the general procedure to determine the heat of mixing of any material dissolved in another.

CHEMICAL EQUATION MATERIAL BALANCE

Ethylene oxide is produced by the catalytic reaction of ethylene and oxygen: $C_2H_4 + {}^1/_2O_2 \rightarrow (CH_2)_2O$. For each 100 lb (45.5 kg) of ethylene, (1) how much ethylene oxide is produced, (2) how much oxygen is required, and (3) what are the quantities of ethylene oxide and ethylene in the product if there is a 20 percent deficiency of oxygen?

Calculation Procedure

1. Compute the quantity of ethylene oxide produced. The two most frequently met calculations in day-to-day chemical engineering are the *material balance* (discussed here) and the *energy balance*, discussed later. In a chemical process, a balance is the same as any other type of balance, i.e., an equating of input, output, and accumulation or loss: Input – output = ± accumulation.

Such a balance may be written around a single item of chemical process equipment, a portion of a process, or an entire chemical plant. A balance may be used to check experimental data or to determine an unknown quantity of some process stream.

For purposes of balance calculations, chemical processes are classified as *steady-state*, i.e., input = output, no accumulation; *unsteady-state*, i.e., input ≠ output, a ± accumulation; *batch process*, i.e., system is loaded, no further ± accumulation; continuous process, i.e., continuous input and output. Chemical processes may be further classed as physical, in which there is no chemical reaction, or chemical, in which a chemical change occurs. To analyze chemical reactions, the principles of chemical equation balances must be understood.

A *stoichiometrically balanced reaction* is one in which the reactants are exactly proportioned to give a product free of excess reactants, as in $C_2H_4 + {}^1/_2O_2 \rightarrow (CH_2)_2O$. An *excess reactant* is one present in excess of the stoichiometric quantity, such as if there were more than 0.5 mol of oxygen in the above equation.

The *degree of completion* is the percentage of the limiting reactant that reacts. The *limiting reactant* is the one present in less than stoichiometric proportion, so that the other reactant is in excess.

To determine how much ethylene oxide is produced, find the molecular weight of C_2H_4 = 2(12) + 4(1) = 28. The moles of C_2H_4 = 100/28 = 3.571.

Referring to the reaction equation shows that for each mole of C_2H_4, 1 mol of $(CH_2)_2O$ is produced, having a molecular weight of 2(12) + 4(1) + 16 = 44. Then the weight of $(CH_2)_2O$ = 44(3.571) = 157.14 lb (71.4 kg).

2. ***Compute the amount of oxygen required.*** The molecular weight of O_2 = 16(2) = 32. Referring to the reaction equation shows that ½ mol of oxygen is needed for each mole of ethylene, C_2H_4. Hence, the weight of oxygen needed = ½(32)(3.571) = 57.14 lb (25.9 kg).

3. ***Compute the product mix for a reactant deficiency.*** Referring to step 2, we see that a 20 percent oxygen deficiency means that there was 0.80(½) = 0.40 mol of oxygen available. Rewriting the equation gives $0.2C_2H_4 + 0.8C_2H_4 + 0.40_2 \rightarrow 0.8(CH_2)O + 0.2C_2H_4$. Hence, the ethylene oxide $(CH_2)_2O$ in the product = 0.8(157.14) = 125.71 lb (57.14 kg). And the ethylene, C_2H_4, in the product = 0.2(100) = 20 lb (9.1 kg).

Related Calculations Use this general procedure for any chemical equation balance similar to that analyzed here.

BATCH PHYSICAL PROCESS BALANCE

A load of clay containing 35 percent moisture on a wet basis weighs 2000 lb (909.1 kg). If the clay is dried to a 15 percent moisture content (on a wet basis), how much water is evaporated in the drying process?

Calculation Procedure

1. ***Compute the initial moisture content.*** The 2000 lb (909.1 kg) of wet clay contains 35 percent moisture, or 2000 (0.35) = 700 lb (318.2 kg) of water. Thus, the dry clay weighs 2000 – 700 = 1300 lb (590.9 kg).

2. ***Compute the weight after drying.*** Set up the relation y lb of wet clay + x lb of water = 1300 lb (590.9 kg) of dry clay. But the final batch contains 15 percent moisture. Hence, the water = $0.15y$. Therefore, the dry clay = $(1.00 - 0.15)y = 0.85y = 1300$. Solving, we find y = 1529 lb (694.9 kg) of wet (15 percent moisture) clay. And since $y + x$ = 2000 lb (909.1 kg), $x = 2000 - y = 2000 - 1529$ = 471 lb (214.1 kg) of water evaporated.

Related Calculations Use this general procedure for any batch physical process balance involving evaporation or drying of a solid.

Where the rate of feed is given, a steady-state physical process balance can be analyzed. Thus, if the 2000 lb (909.1 kg) of clay in the above process were fed to the dryer in 1 h, the rate of evaporation would be 471 lb/h (214.1 kg/h) of water.

STEADY-STATE CONTINUOUS PHYSICAL BALANCE WITH RECYCLE AND BYPASS

Feed to a distillation tower is 1000 lb·mol/h (0.126 kg·mol/s) of a solution of 35 mole percent ethylene dichloride (EDC) in xylene. There is not any accumulation in the tower. The overhead distillate stream contains 90 mole percent ethylene dichloride, and the bottoms stream contains 15 mole percent ethylene dichloride. Cooling water to the overhead condenser is adjusted to give a reflux ratio of 10:1 (10 mol reenters the column for each mole of overhead product). Heat to the reboiler,

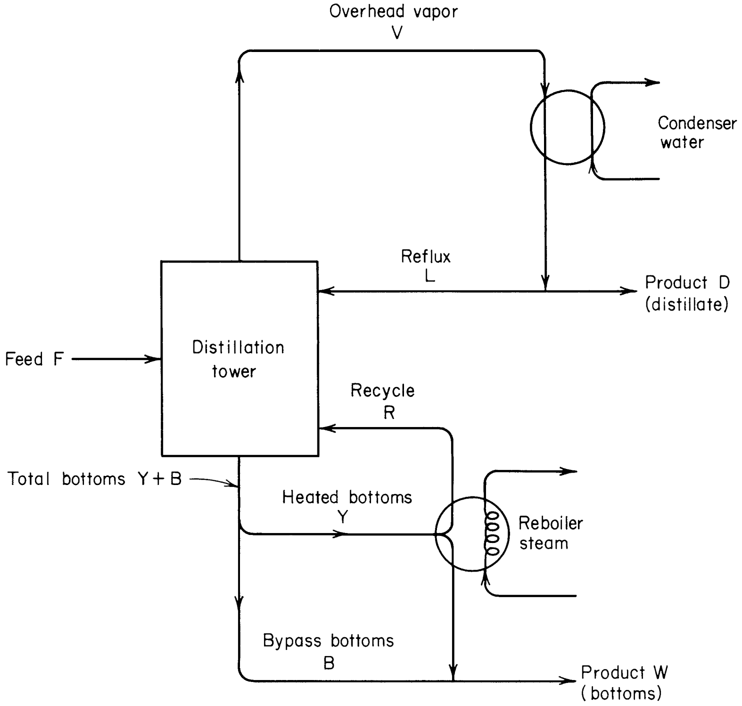


FIGURE 3 Distillation tower flow.

Fig. 3, is adjusted so that the recycle ratio is 5:1 (5 mol reenters the column for each mole of bottom product), with a 2:15 bypass (2 mol bypasses the reboiler for each 15 mol that passes through the reboiler). Determine the flow rate of the overhead product, bottoms product, overhead reflux reentering the column, bottoms recycle reentering the column, bottoms bypassing the reboiler, and the total bottoms.

Calculation Procedure

1. Compute the bottoms product. Since this is a physical system with no change in chemical composition and no accumulation, any component may be followed through the system. Having been given the important values of the ethylene dichloride (X_F, X_D, X_W), use them as the basis of the calculation, with X representing the moles of ethylene dichloride in each stream.

Set up a total material balance thus: $F = D + W = 1000$ lb·mol/h (0.126 kg·mol/s), Eq. 1, where F = feed; D = product (i.e., distillate); W = product (i.e., bottoms), all expressed in lb·mol/h as shown in Fig. 3.

An ethylene dichloride balance is $FX_F = DX_D + WX_W$. Substituting given values, we have $1000(0.35) = 0.90D + 0.15W = 350$, Eq. 2. Solving Eqs. 1 and 2 simultaneously gives $D = 1000 - W$; $W = (350 - 0.90D)/0.15$; $D = 1000 - (350 - 0.90D)/0.15$; $D = 266.67$ lb·mol/h (0.034 kg·mol/s) distillate product; $W = 733.33$ lb·mol/h (0.092 kg·mol/s) bottoms product.

2. Compute the reflux flow rate. Taking the tower overhead as a separate system, Fig. 3, we find $X_L/X_D = 10 = L/D$. Hence, $L = 10D = 10(266.67) = 2666.7$ lb·mol/h (0.34 kg·mol/s) reflux.

3. Analyze the condenser. A total material balance around the condenser is input = output; or $V = D + L$, Fig. 3. Hence, $V = 266.67 + 2666.7 = 2933.37$ lb·mol/h (0.37 kg·mol/s) overhead vapor.

4. Analyze the tower reboiler. Taking the tower reboiler as a separate system. Fig. 3, gives $X_R/X_W = 5 = R/W$; hence, $R = 5W = 5(733.33) = 3666.7$ lb·mol/h (0.46 kg·mol/s) bottoms recycle.

Also, $X_B/X_Y = 2/15 = B/Y$; $2Y = 15B$, Eq. 3. And a total material balance around the reboiler is input = output, or $Y = R + (W - B)$, Eq. 4. Solving Eqs. 3 and 4 simultaneously gives $B = 2/15(Y)$; $Y = R + [W - 2/15(Y)]$; $Y = [15(3666.7) + 15(733.3)]/17 = 3882.35$ lb·mol/h (0.49 kg·mol/s) reboiled bottoms. Then $Y + B = 4399.97$ lb·mol/h (0.55 kg·mol/s) total bottoms.

Related Calculations Use this general procedure to analyze distillation towers handling liquids similar to those considered here.

STEADY-STATE CONTINUOUS PHYSICAL PROCESS BALANCE

The distillation tower of the previous calculation procedure has the temperature and thermal conditions shown in Fig. 4. The reboiler is heated by steam that condenses at 280°F (137.8°C). Cooling water enters the overhead condenser at 70°F (21.1°C) and leaves at 120°F (48.9°C). In the condenser, the

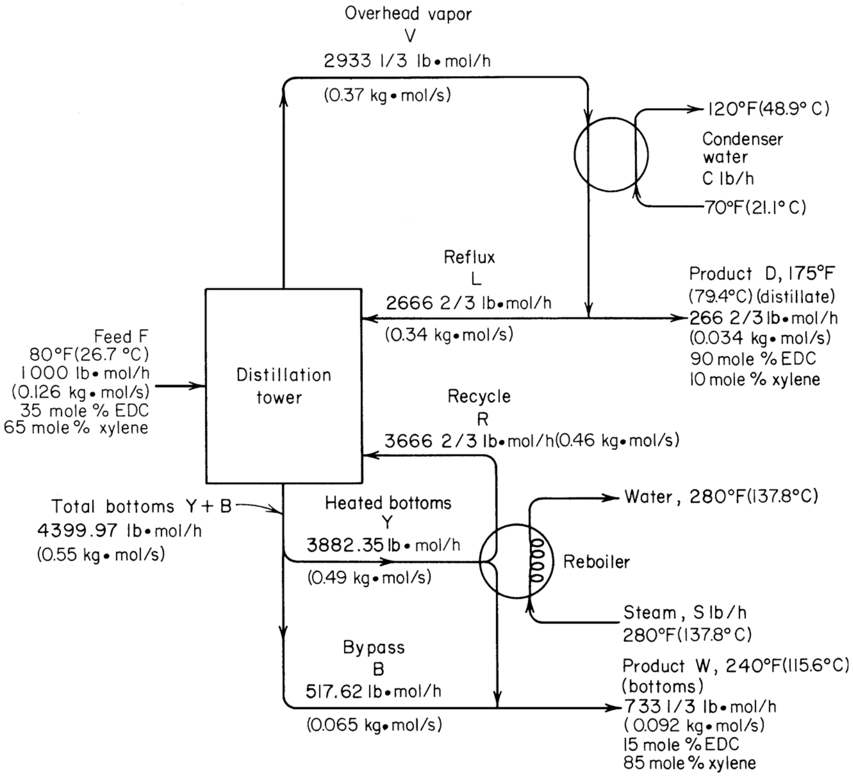


FIGURE 4 Distillation-tower flow quantities and flow rates.

overhead vapor condenses at 184°F (84.4°C) before being cooled at 175°F (79.4°C), the temperature of the liquid reflux and distillate product. The heat of condensation ΔH of the overhead vapor is 14,210 Btu/(lb·mol) [33.1 kJ/(kg·mol)], as given in a standard reference work, and 14,820 Btu/(lb·mol) [34.5 kJ/(kg·mol)] for the tower bottoms. The heat capacity of all liquid streams in this installation is 40 Btu/(lb·mol·°F) [167.4 kJ/(kg·mol·K)]. Determine the steam and cooling-water flow rates required.

Calculation Procedure

1. Set up a heat balance for the column. Thus, heat in = heat in feed + heat in steam. Let the temperature basis for the calculation = 80°F (26.7°C) = t_b. Then, the heat in the feed = ΔH_F = (feed rate, lb/h) [heat capacity of the feed, Btu/(lb·mol·°F)](feed temperature, °F – temperature basis for the calculation, °F) = 1000(40)(80 – 80) = 0.

The enthalpy of vaporization of the steam is, from the steam tables, 924.74 Btu/lb (2151 kJ/kg) when there is complete condensation of the steam and the condensate leaves the reboiler at 280°F (137.8°C). Then the heat given up by the condensation of S lb (kg) of steam is $\Delta H_S = 924.74S$ Btu/h (271.0 W).

The heat out = heat in distillate + heat in bottoms + heat in water, all expressed in Btu/h (W). Using the same procedure as for the heat in the feed, we see the heat in the distillate = $\Delta H_D/DC\Delta t_d$, where C = distillate heat capacity, Btu/(lb·mol·°F) [kJ/(kg·°C)]; Δt_d = temperature change of the distillate, °F (°C). Or, ΔH_D = 266.7(40)(175 – 80) = 1,010,000 Btu/h (295.9 kW). Likewise, for the bottoms, ΔH_W = 733.3(40)(240 – 80) = 4,690,000 Btu/h (1374.2 kW).

For the water, ΔH_w = heat to condense overhead vapor + heat absorbed when cooling the condensed vapor from 184 to 175°F (84.4 to 79.4°C), all expressed in Btu/h (W). With a flow of 2933.3 lb/h (0.37 kg/s) of vapor, ΔH_w = 2933.3 [14,210 + 40(184 – 175)] = 42,700,000 Btu/h (12.5 MW).

2. Analyze the heat balance. The heat balance is heat in = heat out, or $\Delta H_S = \Delta H_D + \Delta H_W + \Delta H_w$. Thus, 924.74$S$ = 1,010,000 + 4,690,000 + 42,700,000; S = 52,300 lb/h (6.59 kg/s) of steam.

For the water, $\Delta H_W = Cc\Delta t_c$, where C = water flow rate, lb/h; c = specific heat of water = 1.0 Btu/(lb·°F). Substituting gives $\Delta H_W = C(1)(120 - 70)$ = 42,700,000; C = 854,000 lb/h (107.6 kg/s) of water.

DETERMINING THE CHARACTERISTICS OF AN IMMISCIBLE SOLUTION

For steam distillation of 2-bromoethylbenzene, the vapor temperature is 222.4°F (105.8°C). Analysis shows 0.16 lb (0.073 kg) of 2-bromoethylbenzene (BB) per lb (kg) of vapor. Saturated steam is used in the distillation process. Determine the pressure in the still and how far from ideal the actual conditions are.

Calculation Procedure

1. Compute the pressure in the still. Each component of an immiscible mixture of liquids exerts a vapor pressure that is independent of its concentration and equal to the vapor pressure of the pure substance—but only if stratification is avoided by vigorous mixing or boiling. The major industrial application of immiscible systems is in steam distillation of high-molecular-weight heat-sensitive organic materials. The mixture of water (steam) and an organic substance will boil when the total solution pressure equals atmospheric pressure. Since the organic material must exert some vapor pressure, it vaporizes with the steam, at a greatly reduced temperature.

The relationship for immiscible components A and B is $w_A/w_B = y_AM_A/y_BM_B = P_{VA}M_A/P_{VB}M_B$, where $w_{A,B}$ = weight of component A, B in vapor; $M_{A,B}$ = molecular weight of component A, B; $y_{A,B}$ = vapor-phase mole fraction of component A, B; $P_{VA,VB}$ = vapor pressure of component A, B.

The vapor pressure of BB at 222.4°F (105.8°C) is, from Perry's—*Chemical Engineers' Handbook*, 20 mmHg, and the vapor pressure of water at 222.4°F (105.8°C) is 938 mmHg. Hence, the total pressure (ideal) in the still is 938 + 20 = 958 mmHg.

2. Compare the ideal to the actual conditions. If conditions in the still were ideal (i.e., exactly according to theory), the weight of the BB in the vapor would be, according to step 1, $w_{BB} = (P_{V,BB}/P_{V,H_2O})(M_{BB}M_{H_2O}) = (20/938)(185/18) = 0.219$ lb (0.0995 kg), versus 0.16 lb (0.073 kg) actual, as given.

Or, by computing the ideal BB vapor pressure for 0.16 lb of BB per lb (0.07 kg/kg) of vapor, from the relation in step 1, $(P_{V,BB}/P_{H_2O})(185/18) = (P_{V,BB}/938)(185/18) = 0.16$ lb/lb (0.07 kg/kg). Solving, we find $P_{V,BB} = (0.16)(938)(18/185) = 14.6$ mmHg versus 20 mmHg actual.

The divergence between the actual and ideal most likely means that the time of contact between the steam and the BB is insufficient to reach equilibrium. Also, the total pressure should be 938 + 14.6 = 952.6 mmHg, not the 958 mmHg of the ideal case.

Related Calculations This procedure is valid for immiscible solutions of all types resembling the one considered here.

Plant engineers and designers in the chemical processing industry must be extremely careful about making changes in chemical processes or waste disposal. Seemingly routine decisions changing a process or disposal method can run into trouble under the Toxic Substance Control Act (TSCA). This act gives the U.S. Environmental Protection Agency (EPA) information and control over commercial chemicals.

Fines as high as $23,000 a day can be levied when TSCA rules are not obeyed. EPA applies rigid formulas when enforcing the act. Violation can result in million-dollar assessments.

It is important that engineers submit a premanufacture notification (PMN) in accordance with Section 5 of TSCA *before* manufacturing or importing a new chemical substance. A new chemical substance, as defined by Matthew Kuryla in *Chemical Engineering* magazine, is one that does not appear on an EPA list known as the TSCA Inventory. This list is constantly changing. Further, a portion of the list is confidential. To make a comprehensive search of the list requires a written request to EPA certifying a bona fide intent to manufacture or import a chemical.

Unless an exemption applies, a manufacturer must file a PMN with EPA *before* commencing production or importation of a new chemical. (In certain instances, a PMN must also be filed for existing chemical production that falls under a regulation known as the "significant new use rule.") The PMN must include the identity of the chemical, information about its proposed use and quantity, its by-products, and all available data concerning potential worker exposure and environmental or public-health effects.

Ninety days after filing a PMN, a company may commence manufacture or import of the chemical if Notice of Commencement (NOC) is filed. When EPA receives an NOC, it places the chemical on the TSCA Inventory. The chemical is then no longer considered new, but an "existing" chemical subject to other TSCA rules.

A number of chemicals and processes are exempt from the PMN requirements of TSCA, including: (1) foods, drugs, and cosmetics (including their intermediates); (2) pesticides (but not their intermediates); (3) chemicals used solely for research and development purposes, in small quantities; (4) chemicals manufactured solely for export; (5) impurities unintentionally present in another chemical; (6) by-products whose only commercial purpose is for burning as fuel, disposal as waste, or reclamation; (7) nonisolated intermediates (i.e., those mixed with other products and reactants) or incidental reaction by-products.

Besides these exemptions, TSCA provides a specialized PMN process for certain limited uses of a new chemical. These rules are known as the "test market," "low volume," and "polymer" exemptions. Such exemptions are subject to detailed rules of their own. They do not apply across the board, and frequently do not have specific, quantitative limits, notes Matthew Kuryla in *Chemical Engineering*.

PUMP SELECTION FOR CHEMICAL PLANTS

Choose a pump to handle 26,000 gal/min (1640 L/s) of water at 60°F (15.6°C) in a chemical plant when the total dynamic head is 37 ft (11.3 m) of water. What is the required hp input to the pump if the pump efficiency is 85 percent? What type of pump should be used if the rotational speed is limited to 880 r/min?

Calculation Procedure

1. Determine the required power input to the pump. A quick way to determine the power input to a pump handling water at normal atmospheric temperatures is to use Fig. 5. Enter on the left at the total dynamic head, 37 ft (11.3 m), and project to the right to the required pump capacity, 26,000 gal/min (1640 L/s). At the intersection with the hp stem, read the required power input as 285 hp (212.6 kW).

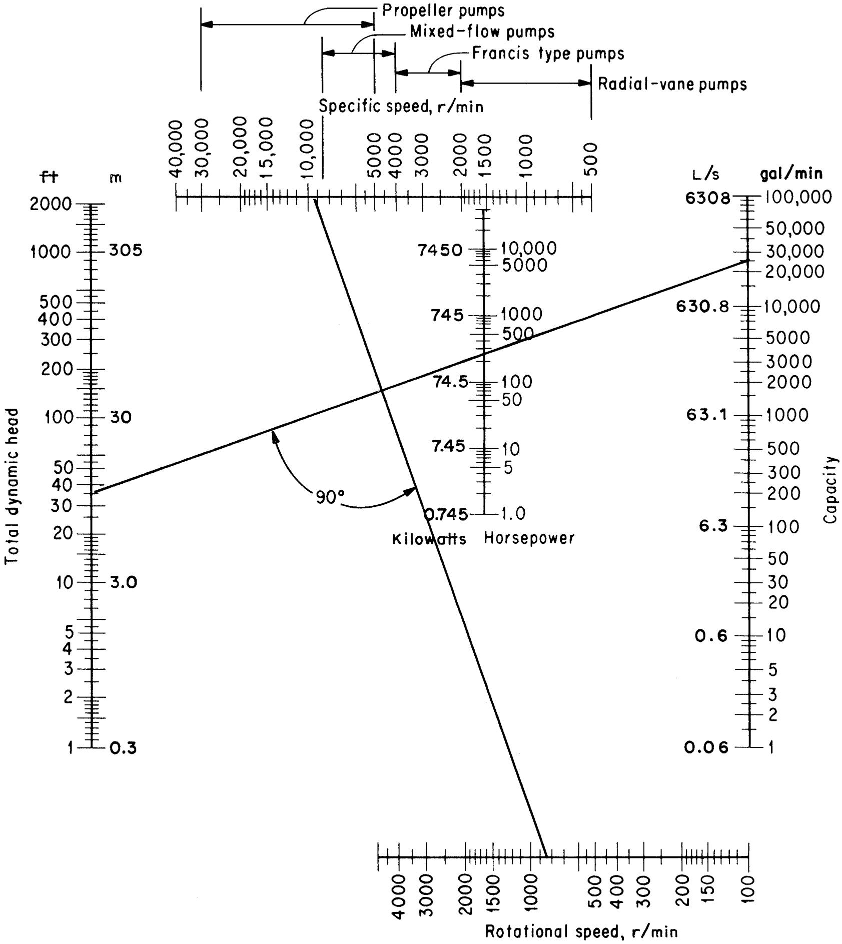


FIGURE 5 Pump hp and type selection chart. (*Power.*)

2. Select the type of pump to use. From the rotational speed, 880 r/min on the bottom stem, draw a straight line at right angles to the first construction line, as shown. At the intersection with the top stem, read the type of pump as a propeller pump having a specific speed of 9500 r/min.

Related Calculations Note that this pump application chart applies to rotating-type centrifugal pumps. Where a reciprocating pump is desired, use the methods given in Sec. 3 of this handbook. The chart in Fig. 5 was developed by H. W. Hamm and was first presented in *Power* magazine.

Pending environmental regulations will strictly limit pump leakage in chemical and process plants of all kinds. Today's laws require plant operators to report leakage of toxic substances of 0.0001% of the pump's capacity.

There are both national (EPA) and state laws controlling pump leakage. For example, the state of New Jersey has a Toxic Catastrophe Prevention Act (TCPA) which strictly controls pump leakage. This and similar state environmental laws controlling pump seal leakage of toxic materials will probably become stricter in the future. For this reason, careful selection of pump shaft seals is important to every engineer working with toxic materials.

Typical toxic materials whose leakage must be prevented from pumps are sulfuric and nitric acid. Where water-flushed seals are used to contain leakage of such materials, the acidic flush water must be treated before disposal. Leakage of toxic materials must be prevented both while a pump is operating and while it is idle. Often, a seal that prevents leakage while the pump is operating will allow leakage when the pump is shut down. Such leakage is just as unlawful as leakage while the pump is operating.

For these reasons, engineers must carefully specify leak-free seals when choosing pumps handling toxic materials. The best seals can only be chosen after thorough study and consultation with both the pump and seal manufacturers.

CRUSHER POWER INPUT DETERMINATION

A chemical process requires the crushing of 240 tons/h (217.7 t/h) of quartz. The quartz feed used is such that 80 percent passes a 3-in (7.6-cm) screen and 80 percent of the product must pass a $^1/_4$-in (0.64-cm) screen. Determine the power input to the crusher.

Calculation Procedure

1. Compute the crusher capacity in tons/min. Use the relation $t_m = t_h/60$, where t_m = crusher capacity, tons/min; t_h = crusher capacity, tons/h. Substituting yields $t_m = 240/60 = 4$ tons/min (3.63 t/min).

2. Determine the material work index. The work index for any material that will be crushed is the total energy, kWh/ton, needed to reduce the feed to a size so that 80 percent of the product will pass through a 100-μm screen. Standard references such as Perry's—*Chemical Engineers' Handbook* list work indexes for various materials. For quartz having a specific gravity of 2.65, Perry gives the work index $W_i = 13.57$ kWh/ton (14.96 kWh/t).

3. Compute the raw-material and product mesh sizes. Use the relation $d_r = s/12$, where d_r = mesh size, ft, for feed; s = mesh opening measure used, in. For the product, $d_p = s/12$, where the symbols are the same as before except that the mesh opening is that used for the product. Substituting gives $d_r = 3/12 = 0.25$; $d_p = 0.25/12 = 0.0208$ (0.0064 m).

4. Compute the required power input to the crusher. Use the relation $hp = 1.46 t_m W_i(1/d_p^{0.5} - 1/d_r^{0.5})$, where the symbols are as given earlier. Substituting gives $hp = 1.46(4)(13.57)(1/0.208^{0.5} - 1/0.25^{0.5}) = 391$ hp (291.6 kW). A 400-hp (298.3 kW) motor would be used to drive this crusher.

Related Calculations Use this general procedure, known as the bond crushing law and work index, to determine the power input required for commercially available grinders and crushers of all types. The result obtained is valid for all usual preliminary calculations.

COOLING-WATER FLOW RATE FOR CHEMICAL-PLANT MIXERS

A kneader used in a chemical plant requires 300-hp (223.7-kW) input per 1000 gal (3785.0 L) of material kneaded. If this kneader handles 3000 lb (1360.8 kg) of a chemical having a density of 65 lb/ft^3 (1041.2 kg/m^3), determine the quantity of cooling water required in gal/min and gal/h if the maximum allowable temperature rise of the water during passage through the kneader is 25°F (13.9°C).

Calculation Procedure

1. Convert the kneader load to gallons. Since the power-input requirements of chemical mixers are normally stated in hp/gal, the kneader load must be converted to gal. Use the relation, load, gal = load weight, lb (7.48 gal/ft^3 water)/load density, lb/ft^3. For this kneader, load, gal = 3000(7.48)/65 = 345 gal (1306.0 L).

2. Compute the required power input. Use this relation: power input *hp* = hp input per 1000 gal (load, gal)/1000. For this kneader, power input *hp* = 300(345)/1000 = 103.5 hp (77.2 kW).

3. Compute the heat that must be removed. Since 1 hp = 2545 Btu/h (745.9 W), the heat that must be removed = (103.5 hp)(2545) = 263,407.5 Btu/h (77,145.5 W).

4. Compute the cooling-water flow rate. With an allowable temperature rise of 25°F (–3.9°C), and a specific heat of 1 Btu/(h·°F) (0.293 W), the cooling-water flow rate required = (263,407.5 Btu/h)/[(25°F)(1.0)(8.33 lb/gal of water)(60 min/h)] = 21.1 gal/min, or 21.2(60 min/h) = 1265 gal/h (4788 L/h).

Related Calculations Use the general procedure given here for any of the usual chemical mixers, such as paddles, turbines, propellers, disks, cones, change cans, dispersers, tumbling mixers, mixing rolls, masticators, pug mills, and mixer-extruders. Consult Perry's—*Chemical Engineers' Handbook* for suitable power-input data for mixers of various types.

LIQUID-LIQUID SEPARATION ANALYSIS

Size a liquid-liquid separator or decanter by using gravitational force for continuous separation of two liquids, the first of which has a density of 47 lb/ft^3 (752.5 kg/m^3) and the second liquid a density of 81 lb/ft^3 (1296.8 kg/m^3). Both liquids flow into the separator at a rate of 50 gal/min (189.3 L/min). The time required for settling is 35 min. What size separator is required to handle this flow? How far above the separator bottom should overflow of the heavier liquid be located?

Calculation Procedure

1. Compute the liquid holdup volume. Since there are two liquids, a light one and a heavy one, entering the separator, the holdup volume = (number of liquids entering)(liquid flow rate into the separator, gal/min)(holdup time, min). Or for this separator, total holdup volume = 2(50)(35) = 3500 gal (13,247.5 L).

2. Determine the separator tank volume. Usual design practice is to make the separator tank volume 10 to 25 percent greater than the required holdup volume. Using a volume 20 percent greater than the required holdup volume gives a required tank volume of 1.20(3500 gal) = 4200 gal (15,897.0 L).

3. Size the separator tank. Most decanter-type separator tanks are sized so that the tank diameter and height are approximately equal. Selecting a 10-ft (3.05-m) diameter and 10-ft (3.05-m) high tank gives a total tank volume of (head area, ft^2)(height, ft) = $(d^2\pi/4)h$, where d and h are the diameter and height of the tank, ft, respectively. Or, volume = $(10^2\pi/4)(10) = 785.4\ \text{ft}^3$ ($2.22\ \text{m}^3$). Since 1 gal (3.8 L) of liquid occupies $0.13\ \text{ft}^3$, the capacity of this tank = 785.4/0.13 = 5850 gal (22,142.2 L). This is sufficient to store the holdup liquid but somewhat oversize.

Try a 9-ft (2.74-m) diameter and high tank. By the same method, the tank capacity is 4250 gal (16,086.3 L). This is closer to the required holdup capacity. Hence, a 9-ft (2.74-m) tank will be used.

4. Compute the liquid depth in the tank. Use the relation D_1 ft = 4(holdup volume, gal)/$7.48\pi d^2$, where D_1 = liquid depth, ft. So $D_1 = 4(3500)/7.48\pi 9^2 = 7.34$ ft (2.24 m).

5. Determine the height of the heavy-liquid overflow. Assume that the two liquids interface midway between the vessel bottom and the liquid surface. Then the height of the heavy liquid = 7.34/2 = 3.67 ft (1.12 m).

To find the height of the heavy-liquid overflow, solve $H_h = H_1 + (D_1 - H_1)$(density of lighter liquid, lb/ft^3)/(density of heavier liquid, lb/ft^3), where H_h = height of heavy-liquid overflow above tank bottom, ft; H_1 = height of heavy liquid in tank, ft; other symbols as before. Solving gives $H_h = 3.67 + (7.34 - 3.67)(47/81) = 5.80$ ft (1.77 m). This is the distance measured to the inside lower surface of the overflow pipe from the tank bottom.

The *continuous decanter* is a popular type of static separator for immiscible liquids of many types. This type of separator is fed from the top and vented to the open air through both the light- and heavy-liquid overflow lines.

Process Plant Engineering

DESIGNING STEAM TRACING FOR PIPING

A 10-in (25-cm) stainless-steel pipe is conveying phthalic anhydride at 300°F (148.9°C). The line is to be steam-traced by using heat-transfer cement to attach the tracing line to the main pipe which is insulated with 1.5 in (3.81 cm) of calcium silicate insulation to maintain the bulk temperature at 300°F (148.9°C). The process fluid is stagnant in the pipe with an average, inside, natural convection coefficient of 20 Btu/(h·ft²·°F) [113.6 W/(m²·K)]. Determine the number of parallel tracers required, the heat transferred to the process fluid, and the steam consumption, using 150 lb/in² (gage) (1034 kPa) saturated steam. Other key data are as follows: ambient temperature $T_{\text{amb}} = -10$°F (–23.3°C); supply-steam temperature $T_s = 366$°F (185.6°C); process fluid temperature $t_p = 300$°F (148.9°C); thermal conductivity of stainless-steel pipe wall $K = 9.8$ Btu/(h·ft²·°F·in) [1.4 W/(m·K)]; pipe-wall thickness = 0.165 in (0.413 cm); ID = 0.43 in (1.08 cm); heat-transfer coefficient between the insulation and ambient atmosphere $h_{\text{air}} = 2$ Btu/(h·ft²·°F) [11.4 W/(m²·K)].

Calculation Procedure

1. Compute the overall heat-transfer coefficient between the wall and the ambient atmosphere. Use the relation

$$\frac{1}{h_o} = \frac{1}{h_{\text{air}}} + \frac{x_{\text{ins}}}{k_{\text{ins}}}$$

where the symbols are h_o = overall heat-transfer coefficient between the pipe wall and the air, Btu/(h·ft^2·°F) [W/(m^2·K)]; h_{air} = heat-transfer coefficient between the ambient air and insulation, Btu/(h·ft^2·°F) [W/(m^2·K)]; x_{ins} = insulation thickness, in (cm); k_{ins} = thermal conductivity of insulation, Btu/(h·ft^2·°F·in) [W/(m^2·K·cm)]. Entering the given values, we find $1/h_o = 1/2.0 + 1.5/0.3$; h_o = 0.182 Btu/(h·ft^2·°F) [1.03 W/(m^2·K)].

2. Determine the constants A and B and the ratio B/A. The dimensionless constant $A = (h_o + h_i)/Kt$, where h_i = process-fluid heat-transfer coefficient, Btu/(h·ft^2·°F) [W/(m·K)]; K = thermal conductivity of pipe or vessel wall, Btu/(h·ft^2·°F·ft) [W/(m^2·K)], other symbols as before. Substituting gives $A = (0.182 + 20)/(9.8)(0.165/12) = 150$; $\sqrt{A} = (150)^{1/2} = 12.25$. And $B = (h_iT_p + h_oT_{\text{amb}})/Kt$, where T_p = pipe or vessel-wall temperature, °F (°C); other symbols as before. Substituting, we find $B = [(20)(300) + (0.182)(-10)]/9.8(0.165/12) = 44{,}510$. Then $B/A = 44{,}510/150 = 296.7$. This corresponds to the equilibrium temperature, 296.7°F (147.1°C), of the pipe wall in the absence of steam tracing.

3. Determine the temperature of the pipe wall at the tracer. The temperature of the pipe wall at the tracer T_o is usually set equal to the lowest temperature in the tracer, namely the saturated steam temperature at the tracer outlet. Normally, trapping distances are based on a 10 percent or 10-lb/in^2 (gage) (68.9-kPa) pressure drop, whichever is greater.

Assume a 10 percent pressure drop for this tracer circuit. The outlet pressure = 135 lb/in^2 (gage) (930.8 kPa), corresponding to a saturated steam temperature of T_o = 358°F (181.1°C) from the steam tables.

4. Calculate the steam tracing half-pitch. Use a value of heat transferred to the process fluid, Btu/h (W)/length of the tracer, equivalent ft (m) of 100 Btu/(h·ft) (8.92 W/m) of tracer to compute L = one-half the steam tracing pitch, ft (m). Then, $L = [Q/z - 2(T_o - B/A)h_i/\sqrt{A}\,]/2h_i(B/A - T_p)$, where the symbols are as given earlier. Substituting, we find $L = [100 - 2(358 - 296.7)(20)/12.25]/2(20)(2.967 - 300) = 0.76$ ft (0.23 m).

Checking the assumption of the 10 percent pressure drop, we see $\tanh(\sqrt{A}L) = \tanh(12.25 \times 0.76) = \tanh 9.3 \approx 1.0$. Therefore, the assumption was valid and $L = 0.76$ is correct. If it were not correct, another pressure drop would have to be assumed and L computed again until a suitable value were obtained.

For certain applications, another value for Q/z different from 100 may be preferable, depending on the reheat time required if the tracing steam supply were lost. Hence, the designer must verify that the heatup time possibly meets this requirement while not producing an uneconomical tracing design.

5. Compute the minimum required pitch and distance between tracers. The minimum required pitch $= 2L = 2(0.76) = 1.52$ ft (0.46 m). Determine the number of parallel tracers from $\pi D/2L$, where the symbols are as defined earlier. For this installation, $N = \pi(10.75/2)/1.52 = 1.85$. With two parallel tracers the resultant distance between tracers is $\pi D/2 = \pi(10.75/12)/2 = 1.4$ ft (0.42 m).

6. Determine the total heat transferred by the tracer. The total heat transferred by the tracer $H_T = Q_T/2 = [2(20.182)/12.25](358 - 296.7) \tanh (12.25 \times 0.7) = 202$ Btu/(h·ft) (18 W/m).

7. Determine the tracer steam consumption. The steam consumption is $w' = (Q_T/2)\Delta H$, where ΔH = change in enthalpy of the steam from the tracer inlet to the tracer outlet. Or, $w' = 202/865 = 0.234$ lb/(h·ft) [0.0000967 kg/(s·m)].

8. Find the maximum equivalent feet of tracing run per steam trap. Using the tracer circuit pressure-drop assumed in step 3, compute the equivalent length, ft, from $z = [1.48(10^{11})(D^5)(\Delta P)/fw'^2\,(V_v - V_L)]^{13} = (1.48)(10^{11})(0.43/12)^5(15)/(0.012)(0.234)^2(3.02 - 0.018) = 405$ maximum equivalent ft (123.4 m).

The equation used in this step is derived by integrating the Darcy equation for a fluid with a changing specific volume. No account is taken of the fact that that fluid has two phases, since the largest portion of the total pressure drop is taken where the fluid is nearly all vapor. Further, the steam pressure is taken to be that at the outlet of the tracer circuit, thereby a somewhat conservative steam specific volume for the circuit.

Related Calculations This procedure can be used for piping in a variety of applications, including chemical, petroleum, food, textile, marine, steel, etc. The ultimate use of the medium in the pipe has little or no effect on the calculation. This procedure is the work of Carl G. Bertram, Vikram J. Desai, and Edward Interess, the Badger Co., as reported in *Chemical Engineering* magazine.

STEAM TRACING A VESSEL BOTTOM TO KEEP THE CONTENTS FLUID

The bottom of a 4-ft (1.22-m) diameter, stainless-steel, solvent-recovery column holds a liquid that freezes at 320°F (160°C) and polymerizes at 400°F (204.4°C). The bottom head must be traced to keep the material fluid after a shutdown. Determine the required pitch of the tracing, using 150 lb/in^2 (gage) (1034 kPa) saturated steam. The ambient temperature is –20°F (–28.9°C), the supply steam temperature T_s = 366°F (185.6°C), thermal conductivity of stainless steel = 9.8 Btu/(h·ft^2·°F·ft) [16.95 W/(m·K)], insulation thickness = 2 in (5.1 cm), thermal conductivity of insulation = 0.3 Btu/(h·ft^2·°F·ft) [0.52 W/(m·K)], and wall thickness = 0.375 in (0.95 cm). The heat-transfer coefficient between the insulation and the air is 2.0 Btu/(h·ft^2·°F) [11.4 W/(m^2·K)], and the inside convection coefficient is 20.

Calculation Procedure

1. Compute the process-fluid heat-transfer coefficient and the overall heat-transfer coefficient Use the relation $1/h_o = 1/h_{\text{air}} + x_{\text{ins}}/k_{\text{ins}}$, where the symbols are as defined in the previous calculation procedure. Substituting, we get $1/h_o = {}^1\!/_2 + 2/0.3$; h_o = 0.14 Btu/(h·ft^2·°F) [0.79 W/(m^2·K)].

The process-fluid heat-transfer coefficient h_i = 20 Btu/(h·ft^2·°F) [113.6 W/(m^2·K)], assumed.

2. Determine the constants A and B and the ratio B/A. To determine the value of A, solve $A = (h_o + h_t)/Kt = (0.14 + 20)/(9.8)(0.375/12) = 65.6$, dimensionless. Also, $B = (h_iT_p + h_oT_{\text{amb}})/Kt = [(20)(320) + (0.14)(–20)]/(9.8)(0.375/12) = 20{,}900$. Then $B/A = 20{,}900/65.6 = 318$.

3. Compute the tracer-steam outlet temperature. The tracer steam is supplied at 50 lb/in^2 (gage) (344.7 kPa). Assuming a 15-lb/in^2 (gage) (103.4-kPa) pressure drop in the tracer system, we see the outlet pressure = 150 – 15 = 135 lb/in^2 (gage) (930.8 kPa). The corresponding saturated-steam temperature is, from the steam tables, T_o = 358°F (181°C). This is the tracer outlet steam temperature.

4. Calculate the adjusted temperature ratio. Use the relation $(T_{\text{mid}} - B/A)/(T_o - B/A) = (320 - 318)/(358 - 318) = 0.05$.

5. Determine the required tracing pitch. From Fig. 6, with the adjusted temperature ratio of 0.05, $\sqrt{A}(L) = 3.7$ when $\alpha = 1$. (Here α = a parameter = x/L, where x = distance along the pipe or vessel wall, ft.) Then, by solving for $L = 3.7/65.6^{1/2}$, L = 0.46 ft (0.14 m) = 5.5 in (13.97 cm).

The maximum allowable pitch for tracing the bottom of the column, Fig. 7, is 21, or 2(5.5) = 11 in (27.9 cm). A typical tracing layout is shown in Fig. 7.

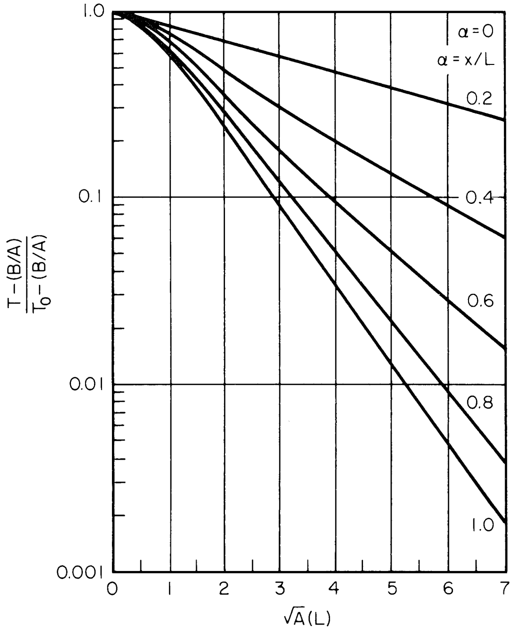


FIGURE 6 Graphical solution for steam-tracing design. (*Chemical Engineering.*)

Related Calculations This procedure can be used to design steam tracing for a variety of tanks and vessels used in chemical, petroleum, food, textile, utility, and similar industries. The medium heated can be liquid, solid, vapor, etc. As with the previous calculation procedure, this procedure is the work of Carl G. Bertram, Vikram J. Desai, and Edward Interess, the Badger Company, as reported in *Chemical Engineering* magazine.

DESIGNING STEAM-TRANSMISSION LINES WITHOUT STEAM TRAPS

Design a steam line for transporting a minimum of 6.0×10^5 lb/h (2.7×10^5 kg/h) and a maximum of 8.0×10^5 lb/h (3.6×10^5 kg/h) of saturated steam at 205 lb/in^2 (gage) and 390°F (1413 kPa and 198.9°C). The line is 3000 ft (914.4 m) long, with eight 90° elbows and one gate valve. Ambient temperatures range from –40 to 90°F (–40 to 32.2°C). The line is to be designed to operate without steam traps. Insulation 3-in (7.6-cm) thick with a thermal conductivity of 0.48 Btu·in/(h·ft^2·°F) [0.069 W/(m·K)] will be used on the exterior of the line.

Calculation Procedure

1. Size the pipe by using a suitable steam velocity for the maximum flow rate. The minimum acceptable steam velocity in a transmission line which is not fitted with steam traps is 110 ft/s (33.5 m/s). Assuming, for safety purposes, a steam velocity of 160 ft/s (48.8 m/s) to use in sizing this transmission line, compute the pipe diameter in inches from $d = 0.001295 f\rho LV^2/\Delta P$, where f = friction factor for the

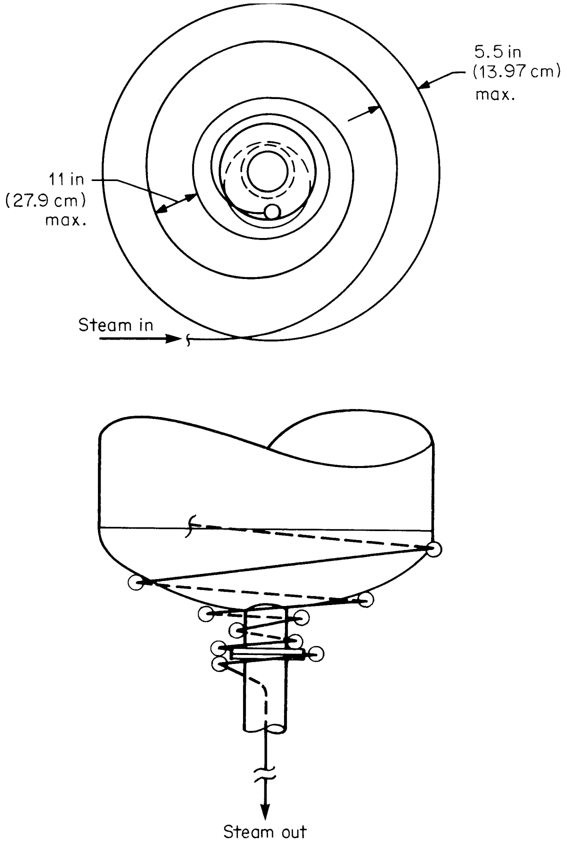


FIGURE 7 Steam-traced vessel bottom. (*Chemical Engineering.*)

pipe (=0.0105, assumed); ρ = density of the steam, lb/ft^3 (kg/m^3) [=0.48 (7.7) for this line]; L = length of pipe, ft, including the equivalent length of fittings [=3500 ft (1067 m) for this pipe]; V = steam velocity, ft/s [=160 ft/s (48.8 m/s) for this line]; ΔP = pressure drop in the line between inlet and outlet, lb/in^2 [=25 lb/in^2 (172.4 kPa) assumed for this line]. Substituting yields d = 0.001295(0.0105)(0.48)(3500)(160)^2/25 = 22.94 in (58.3 cm); use 24-in (61-cm) schedule 40 pipe, the nearest standard size.

2. Check the actual steam velocity in the pipe chosen. The actual velocity of the steam in the pipe can be found from $V = Q/A$, where V = steam velocity, ft/s (m/s); Q = flow rate for steam, lb/s (kg/s); A = cross-sectional area of pipe, ft^2 (m^2). Substituting gives V = (800,000 lb/h ÷ 3600 s/h)(2.08 ft^3/lb for steam at the entering pressure)/2.94 ft^2 = 157.6 ft/s (48.0 m/s) for maximum-flow conditions; V = (600,000/3600)(2.08)/2.94 = 117.9 ft/s (35.9 m/s) for minimum-flow conditions.

3. Compute the pressure drop in the pipe for each flow condition. Use the relation $\Delta P = 0.001295 f\rho LV^2/D$, where the symbols are the same as in step 1. Substituting, we find ΔP = 0.001295(0.0105)(0.48)(3500)(157.6)^2/24 = 23.2 lb/in^2 (159.9 kPa) for maximum-flow conditions. For minimum-flow conditions by the same relation, ΔP = 0.001295 (0.0105)(0.48)(3500)(117.9)^2/ 24 = 13.23 lb/in^2 (91.2 kpa). The pressure at the line outlet will be 220.0 – 23.2 = 196.5 lb/in^2 (1356.9 kPa) for the maximum-flow condition and 220.0 – 13.2 = 206.8 lb/in^2 (1425.9 kPa) for minimum-flow conditions.

4. Compute the steam velocity at the pipe outlet. Use the velocity relation in step 1. Hence, for maximum-flow conditions, V = (800,000/3600)(2.30)/2.94 = 173.8 ft/s (52.9 m/s). Likewise, for minimum-flow conditions, V = (600,000/3600)(2.19)/2.94 = 124.1 ft/s (37.8 m/s).

5. Determine the enthalpy change in the steam at maximum temperature-difference conditions. First, the heat loss from the insulated pipe must be determined for the maximum temperature-difference condition from $Q_m = h\Delta tA$, where Q_m = heat loss at maximum flow rate, Btu/h (W); h = overall coefficient of heat transfer for the insulated pipe, Btu·in/(h·ft^2·°F) [W·cm/(m^2·°C)]; Δt = temperature difference when the minimum ambient temperature prevails, °F (°C); A = insulated area of pipe exposed to the outdoor air, ft^2 (m^2). Substituting yields Q_m = 0.16 (430)(3362)(6.28) = 1,452,599 Btu/lb (3378.7 MJ/kg). In this relation, 430°F = 390°F steam temperature ± (–40°F) ambient temperature; 3362 = pipe length including elbows and valves, ft; 6.28 = area of pipe per ft of pipe length, ft^2.

The enthalpy change for the maximum temperature difference will be the largest with the minimum steam flow. This change, in Btu/lb (J/kg) of steam, is $\Delta h_{\max} = Q_m/F$, where F = flow rate in the line, lb/h, or $\Delta h_{\max}$ = 1,452,599/600,000 = 2.42 Btu/lb (5631 J/kg).

The minimum enthalpy at the pipe line outlet = inlet enthalpy – enthalpy change. For this pipe line, $h_{0\min}$ = 1199.60 – 2.42 = 1197.18 Btu/lb (278.6 kJ/kg).

6. Determine the enthalpy change in the steam at the minimum temperature-difference conditions. As in step 5, $Q_{\min} = h\Delta tA$, or $Q_{\min}$ = 0.16(300)(3362)(6.28) = 1,013,441 Btu/lb (2357.3 mJ/kg). Then $\Delta h_{\min}$ = 1,013,441/800,000 = 1.26 Btu/lb (2946.6 J/kg). Also, $h_{2\max}$ = 1199.60 – 1.26 = 1198.34 Btu/lb (2787.3 kJ/k).

7. Determine the steam conditions at the pipe outlet. From step 3, the pressure at the transmission line outlet at minimum flow and lowest ambient temperature is 206.8 lb/in^2 (1425.9 kPa), and the enthalpy is 1197.18 Btu/lb (2784.6 kJ/kg). Checking this condition on a Mollier chart for steam, we find that the steam is wet because the condition point is below the saturated-vapor line.

From steam tables, the specific volume of the steam is 2.22 ft^3/lb of total mass, while the specific volume of the condensate is 0.0000342 ft^3/lb of total mass. Thus, the percentage of condensate per volume = 100(0.0000342)/2.22 = 0.00154 percent condensate per volume. The percentage volume of dry steam therefore = 100(1.00000 – 0.00154) = 99.99846 percent dry steam per volume.

Since the velocity under these steam conditions is 124.1 ft/s (37.8 m/s), the steam will exist as a fine mist because such a status prevails when the steam velocity exceeds 110 ft/s (33.5 m/s). In the fine-mist condition, the condensate cannot be collected by a steam trap. Hence, no steam traps are required for this transmission line as long as the pressure and velocity conditions mentioned above prevail.

Related Calculations Some energy is lost whenever a steam trap is used to drain condensate from a steam transmission line. This energy loss continues for as long as the steam trap is draining the line. Further, a steam-trap system requires an initial investment and an ongoing cost for routine maintenance. If the energy loss and trap-system costs can be reduced or eliminated, many designers will take the opportunity to do so.

A steam transmission line carries energy from point 1 to point 2. This energy is a function of temperature, pressure, and flow rate. Along the line, energy is lost through the pipe insulation and through steam traps. A design that would reduce the energy loss and the amount of required equipment would be highly desirable.

The designer's primary concern is to ensure that steam conditions stay as close to the saturated line as possible. The steam state in the line changes according to the change in pressure due to a pressure drop and the change in enthalpy due to a heat loss through insulation. These changes of condition are plotted in Fig. 8, a simplified Mollier chart for steam. Point 1 is defined by P_1 and T_1 steam conditions. Because of the variability of such parameters as flow rate and ambient

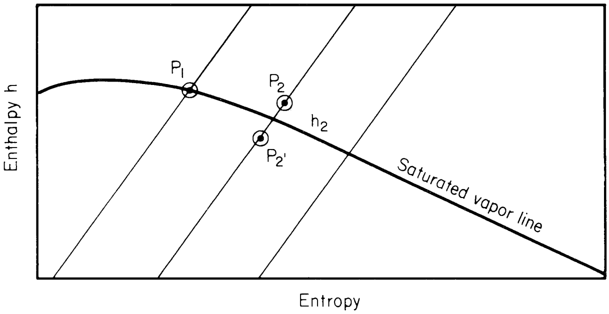


FIGURE 8 Simplified Mollier chart showing changes in steam state in a steam-transmission line.

temperature, the designer should consider extreme conditions. Thus, P_2 would be defined by the minimum pressure drop produced by the minimum flow rate. Similarly, h_2 would be defined by the maximum heat loss produced by the lowest ambient temperature. Point 2 on the *h-s* diagram is defined by the above P_2 and h_2. If point 2 is above or on the saturated-steam line, no condensate is generated and steam traps are not required.

In some cases (small pressure drop, large heat loss), point 2′ is below the saturated-steam line, and some condensate is generated. The usual practice has been to provide trap stations to collect this condensate and steam traps to remove it. However, current research in two-phase flow demonstrates that the turbulent flow, produced by normal steam velocities and reasonable steam qualities, disperses any condensate into a fine mist equally distributed along the flow profile. The trap stations do not collect the condensate, and once again, the steam traps are not required. For velocities greater than 110 ft/s (33.5 m/s) and a steam fraction more than 98 percent by volume, the condensate normally generated in a transmission line exists as a fine mist that cannot be collected by steam-trap stations.

A few basic points should be followed when a steam line is operated without steam traps. All lines must be sloped. If a line is long, several low points may be required. Globe valves are used on drains for each low point and for a drain at the end of the line. Since trap stations are not required, drain valves should be located as close to the line as possible to avoid freezing. Vents are placed at all high points. All vents and drains are opened prior to warming the line. Once steam is flowing from all vent valves, they are closed. As each drain valve begins to drain steam only, it is partially closed so that it may still bleed condensate if necessary. When full flow is established, all drain valves are shut. If the flow is shut down, all valves are opened until the pipe cools and are then closed to isolate the line from the environment. The above procedure would be the same if the steam traps were on the line.

In summary, it can be demonstrated that steam traps are not required for steam transmission lines, provided that one of the following parameters is met:

1. Steam is saturated or superheated.
2. Steam velocity is greater than 110 ft/s (33.5 m/s), and the steam fraction more than 98 percent by volume.

By using the above design, the steam energy normally lost through traps is saved, along with the construction, maintenance, and equipment costs for the traps, drip leg, strainers, etc., associated with each trap station.

This calculated procedure can be used for steam transmission lines in chemical plants, petroleum refineries, power plants, marine installations, factories, etc. The procedure is the work of Mileta Mikasinovic and David R. Dautovich, Ontario Hydro, and reported in *Chemical Engineering* magazine.

Leaks of hazardous materials from underground piping and tanks can endanger lives and facilities. To reduce leakage dangers, the EPA now requires all underground piping through which hazardous chemicals or petrochemicals flow to be designed for double containment. This means that the inner pipe conveying the hazardous material is contained within an outer pipe, giving the "double-containment" protection.

Likewise, underground tanks are governed by the new Underground Storage Tank (UST) laws. The UST laws also cover underground piping. By December 1998, all existing underground piping conveying hazardous materials will have to be retrofitted to double-containment systems to comply with EPA requirements.

Double containment of piping brings a host of new problems for the engineering designer. Expansion of the inner and outer pipes must be accommodated so that there is no interference between the two. While prefabricated double-containment piping can solve some of these problems, engineers are still faced with considerations of soil loading, pipe expansion and contraction, and fluid flow. Careful study of the EPA requirements is needed before any double-containment design is finalized. Likewise, local codes and laws must be reviewed prior to starting and before finalizing any design.

LINE SIZING FOR FLASHING STEAM CONDENSATE

REFERENCES

[1] O. Baker, *Oil & Gas J.*, July 26, 1954; [2] S. G. Bankoff, *Trans. ASME*, vol. C82, 265 (1960); [3] M. W. Benjamin and J. G. Miller, *Trans. ASME*, vol. 64, 657 (1942); [4] J. M. Chenoweth and M. W. Martin, *Pet. Ref.*, vol. 34, 151 (1955); [5] A. E. Dukler, M. Wickes, and R. G. Cleveland, *AIChE J.*, Vol. 10, 44(1964); [6] E. C. Kordyban, *Trans. ASME*, Vol. D83, 613 (1961); [7] R. W. Lockhart and R. C. Martinelli, *Chem. Eng. Prog.*, Vol. 45, 39 (1949); [8] P. M. Paige, *Chem. Eng.*, p. 159, Aug. 14, 1967.

A reboiler in an industrial plant is condensing 1000 lb/h (0.13 kg/s) of steam of 600 lb/in^2 (gage)(4137 kPa) and returning the condensate to a nearby condensate return header nominally at 200 lb/in^2 (gage) (1379 kPa). What size condensate line will give a pressure drop of (1 lb/in^2)/100 ft (6.9 kPa/30.5 m) or less?

Calculation Procedure

1. Use a graphical method to determine a suitable pipe size. Flow in condensate-return lines is usually two-phase, i.e., comprised of liquid and vapors. As such, the calculation of line size and pressure drop can be done by using a variety of methods, a number of which are listed below. Most of these methods, however, are rather difficult to apply because they require extensive physical data and lengthy computations. For these reasons, most design engineers prefer a quick graphical solution to two-phase flow computations. Figure 9 provides a rapid estimate of the pressure drop of flashing condensate, along with a determination of fluid velocity. To use Fig. 9, take these steps.

Enter Fig. 9 near the right-hand edge at the steam pressure of 600 lb/in^2 (gage) (4137 kPa) and project downward to the 200-lb/in^2 (gage) (1379-kPa) end-pressure curve.

From the intersection with the end-pressure curve, project horizontally to the left to intersect the 1000-lb/h (0.13-kg/s) curve. Project vertically from this intersection to one or more trial pipe sizes to find the pressure loss for each size.

Trying the 1-in (2.5-cm) pipe diameter first shows that the pressure loss—[3.0 lb/in^2 (gage)]/100 ft(20.7 kPa/ 30.5 m) exceeds the desired [1 lb/in^2 (gage)]/100 ft (6.9 kPa/30.5 m). Projecting to the next larger standard pipe size, 1.5 in (3.8 cm), gives a pressure drop of [2 lb/in^2 (gage)]/100 ft (1.9 kPa/30.5 m). This is within the desired range. The velocity in this size pipe will be 16.5 ft/s (5.0 m/s).

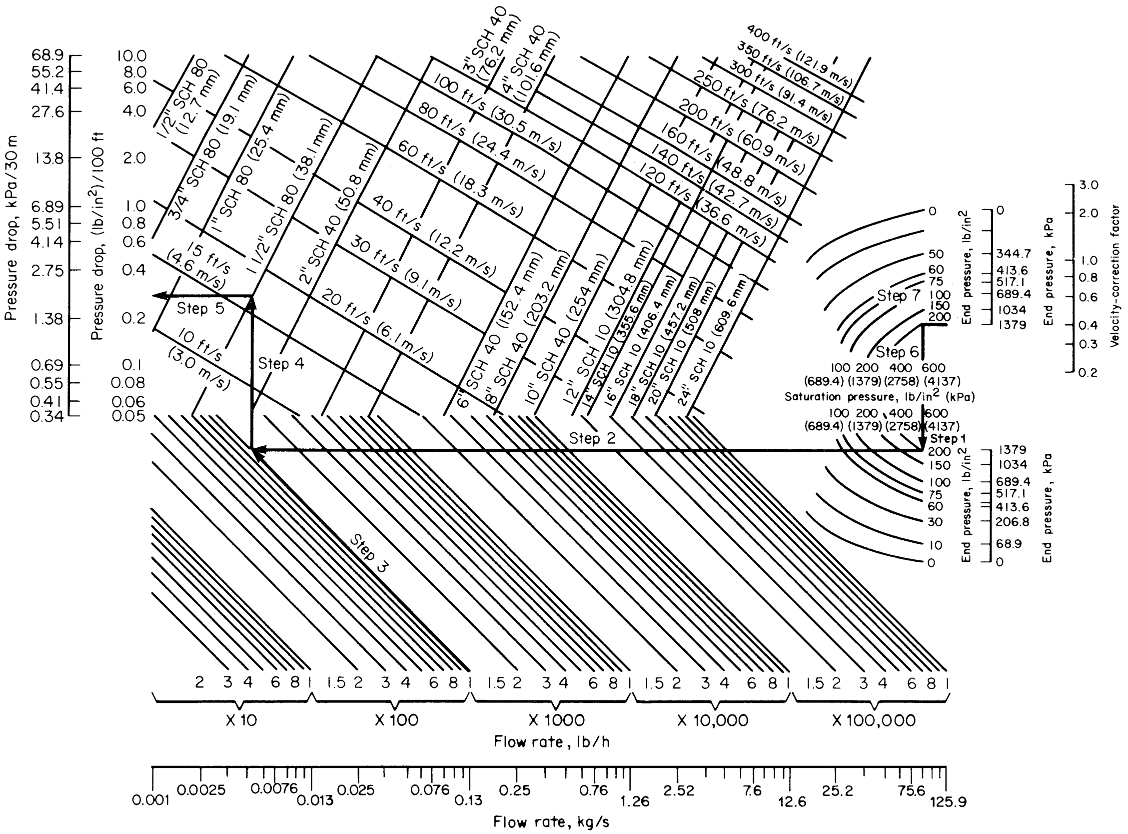


FIGURE 9 Flashing steam condensate line-sizing chart. Divide by 10^4 to obtain numerical values for flow rate measured in kg/s. (*Chemical Engineering.*)

2. ***Determine the corrected velocity in the pipe.*** At the right-hand edge of Fig. 9, project upward from 600-lb/in^2 (gage) (4137-kPa) to 200-lb/in^2 (gage) (1379-kPa) end pressure to read the velocity correction factor as 0.41. Thus, the actual velocity of the flashing mixture in the pipe = 0.41(16.5) = 6.8 ft/s (2.1 m/s).

Related Calculations This rapid graphical method provides pressure-drop values comparable to those computed by more sophisticated techniques for two-phase flow [1–8]. Thus, for the above conditions, the Dukler [5] no-slip method gives [0.22 lb/in2 (gage)]/100 ft (1.52 kPa/30.5 m), and the Dukler constant-slip method gives [0.25 lb/in2 (gage)]/100 ft (1.72 kPa/30.5 m).

The chart in Fig. 9 is based on the simplifying assumption of a single homogeneous phase of fine liquid droplets dispersed in the flashed vapor. Pressure drop is computed by Darcy's equation for single-phase flow. Steam-table data were used to calculate the isenthalpic flash of liquid condensate from a saturation pressure to a lower end pressure; the average density of the resulting liquid-vapor mixture is used as the assumed homogeneous fluid density. Flows within the regime of Fig. 9 are characterized as either in complete turbulence or in the transition zone near complete turbulence.

Pressure drops for steam-condensate lines can be determined by assuming that the vapor-liquid mix throughout the lines is represented by the mix for conditions at the end pressure. This assumption conforms to conditions typical of most actual condensate systems, since condensate lines are sized for low-pressure drop, with most flashing occurring across the steam trap or control valve at the entrance.

If the condensate line is to be sized for a considerable pressure drop, so that continuous flashing occurs throughout its length, end conditions will be quite different from those immediately downstream of the trap. In such cases, an iterative calculation should be performed, involving a series of pressure-drop determinations across given incremental lengths.

This iteration is begun at the downstream end pressure and worked back to the trap, taking into account the slightly higher pressure, and thus the changing liquid-vapor mix, in each successive upstream incremental pipe length. The calculation is complete when the total equivalent length for the incremental lengths equals the equivalent length between the trap and the end-pressure point. This operation can be performed by using Fig. 9.

Results from Fig. 9 have also been compared to those calculated by a method suggested by a Paige [8] and based on the work of Benjamin and Miller [3]. Paige's method assumed a homogeneous liquid-vapor mixture with no liquid holdup, and thus it is similar in approach to the present method. However, Paige suggests calculation of the liquid-vapor mix based on an isentropic flash, whereas Fig. 9 is based on an isenthalpic flash; and this is believed to be more representative of steam-condensate collecting systems.

For the example, the Paige method gives (0.26 lb/in^2)/100 ft (1.79 kPa/30.5 m) at the terminal pressure and (0.25 lb/in^2)/100 ft (1.72 kPa/30.5 m) at a point 1000 ft (305 m) upstream of the terminal pressure, owing to the slightly higher pressure, which suppresses flashing.

The method given here is valid for sizing lines conveying flashing steam used in power plants, factories, air-conditioning systems, petroleum refineries, ships, heating systems, etc. Further, Fig. 9 is designed so that it covers the majority of steam-condensate conditions met in these applications. This calculation procedure is the work of Richard P. Ruskin, Process Engineer, Arthur G. McKee & Co., as reported in *Chemical Engineering* magazine.

SAVING ENERGY LOSS FROM STORAGE TANKS AND VESSELS

Fuel oil at 12° API with a viscosity of 50 SSF (0.01068 mm^2/s) at 122°F (50°C) is stored at 300°F (148.9°C) in a 20-ft (6.1-m) diameter by 30-ft (9.1-m) high carbon-steel tank at atmospheric pressure. The oil level in the tank is 18 ft (5.4 m); the air temperature is 70°F (21.1°C). Determine the heat loss to the environment for two situations: (*a*) Total surface of the tank is uninsulated and black in color. Wind velocity is 0 mi/h (0 km/h). Surface emissivity of the tank is 0.9. Thermal conductivity of the ground under the tank is 0.8 Btu/(h·ft^2·°F·ft) [1.38 W/(m·K)]. (*b*) Roof of tank is uninsulated and is coated with aluminum paint. Sidewall is insulated with calcium silicate, or equivalent, and has a surface emissivity of 0.8. Wind velocity is 30 mi/h (48.0 km/h). The tank contents are not agitated. Thermal conductivity of the ground is 0.8 Btu/(h·ft^2·°F·ft) [1.38 W/(m·K)].

Calculation Procedure

1. Determine the heat loss from the wetted surface inside the tank. For situation (*a*), the wetted area inside the tank is $A_L = \pi DH_L$, where A_L = wetted area, ft^2 (m^2); D = tank diameter, ft (m); H_L = liquid height, ft (m). For this tank, $A_L = \pi 20(18) = 1130.9$ ft^2 (105.1 m^2).

2. Find the temperature difference between the stored liquid and the atmospheric air. The oil temperature T_i is 300°F (148.9°C), and the air temperature T_A is 70°F (21.1°C). Hence, $\Delta T_W = \Delta(T_i - T_A) = 300 - 70 = 230$°F (127.8°C).

3. Determine the heat loss from the tank. Enter Fig. 10 at the bottom with the temperature difference of 230°F (127.8°C), project vertically upward to the unit heat loss curve q_T, and read

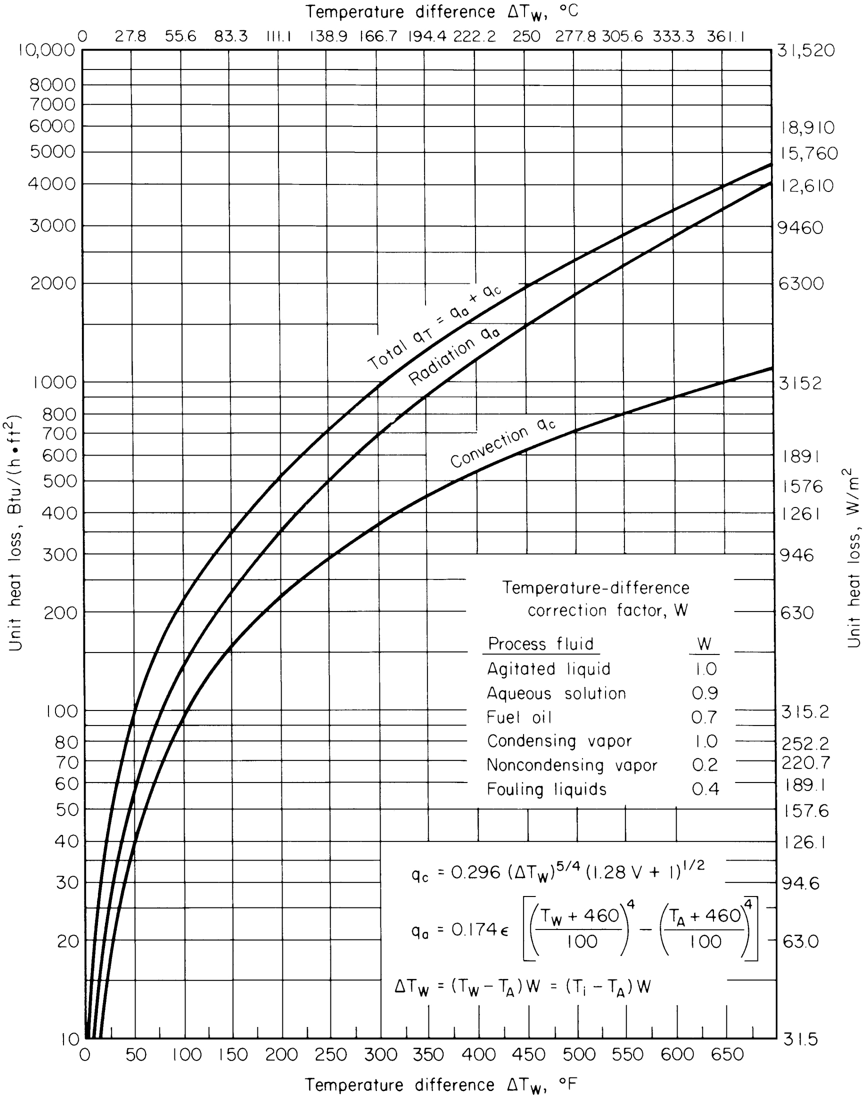


FIGURE 10 Heat losses from uninsulated tanks depend on nature of tank contents. Values in chart are for wind velocity of zero, surface emissivity of 0.9, and ambient air temperature of 70°F (21.1°C). (*Chemical Engineering.*)

q_T = 664 Btu/(h·ft^2) (2086.6 W/m^2). Then the total heat loss from the tank $Q_L = A_L q_T$ = 1130(664) = 750,320 Btu/h (219,896.3 W).

4. Compute the heat loss from the dry inside surface and tank roof surface. The area of the vessel $A_V = \pi D H_V + \pi D^2/4$, where the symbols are as above except that they apply to the dry surfaces of the tank. Then $A_V = \pi(20)(30 - 18) + \pi(20)^2/4$ = 1068 ft^2 (99.2 m^2), where A_V = area.

The temperature difference, by using the correction factor W from Fig. 10 for a noncondensing vapor and the temperature differences in step 2, is $\Delta T_W = (T_i - T_A)\,W$, or ΔT_W = (300 – 70) (0.2) = 46°F(7.7°C).

Next, the unit heat loss is found from Fig. 10 to be 84.3 Btu/(h·ft^2) (265.9 W/m^2) for a temperature difference of 46°F(7.7°C). Then the total heat loss from the dry inside surface in contact with the vapor is $Q_V = 1068(84.3) = 90{,}032.4$ Btu/h (26,385.8 W).

5. Compute the heat loss through the tank bottom to the ground. The heat loss to the ground through the tank bottom, in Btu/h is $O_G = 2dk_G(T_L - T_G)$, where k_G = thermal conductivity of the ground, Btu/(h·ft^2·°F·ft); T_L = liquid temperature, °F; T_G = ground temperature, °F; other symbols as before. Assuming that the ground temperature equals the air temperature, and with $k_G = 0.8$, we see $O_G = 2(20)(0.8)(300 - 70) = 7360$ Btu/h (2156.4 W).

6. Compute the total heat loss from the tank. The total heat loss from the tank will be the sum of the losses from the liquid, vapor, and ground areas of the tank, or $Q_T = Q_L + Q_V + Q_G = 750{,}320 + 90{,}032 + 7360 = 847{,}712$ Btu/h (248,438.9 W).

7. Compare the results by using exact equations. Figure 10 gives the exact algebraic equations for the total heat loss from uninsulated tanks. Substituting in these equations gives an exact total heat loss of 894,122 Btu/h (261,977 W). This is a difference of 4.7 percent from the approximate solution obtained by using Fig. 10. Most working engineers would be willing to accept such a difference in view of the savings in time and labor obtained by using the graphic solution.

8. Determine the insulation thickness needed for the tanks. For situation (*b*), the wetted surface of the tank $A_L = 1130$ ft^2 (105.1 m^2). This represents the interior circumferential area of the tank wetted by the fuel oil to a height of 18 ft (5.4 m).

Heat loss from the tank to the ambient air is a function of the temperature difference between the tank wall and the air, or $\Delta T_I = T_W - T_A = 300 - 70 = 230$°F (110°C).

From Fig. 11, find the recommended insulation thickness as 1.5 in (3.8 cm) and the wind correction factor for a 30-mi/h (48-km/h) wind and 1.5-in (3.8-cm) insulation as 1.10.

9. Correct the unit heat loss for wind velocity. From step 4, $\Delta T_W = 46$; then $q_L = 46(1.10) = 50.6$ Btu/(h·ft^2) (159.6 W/m^2).

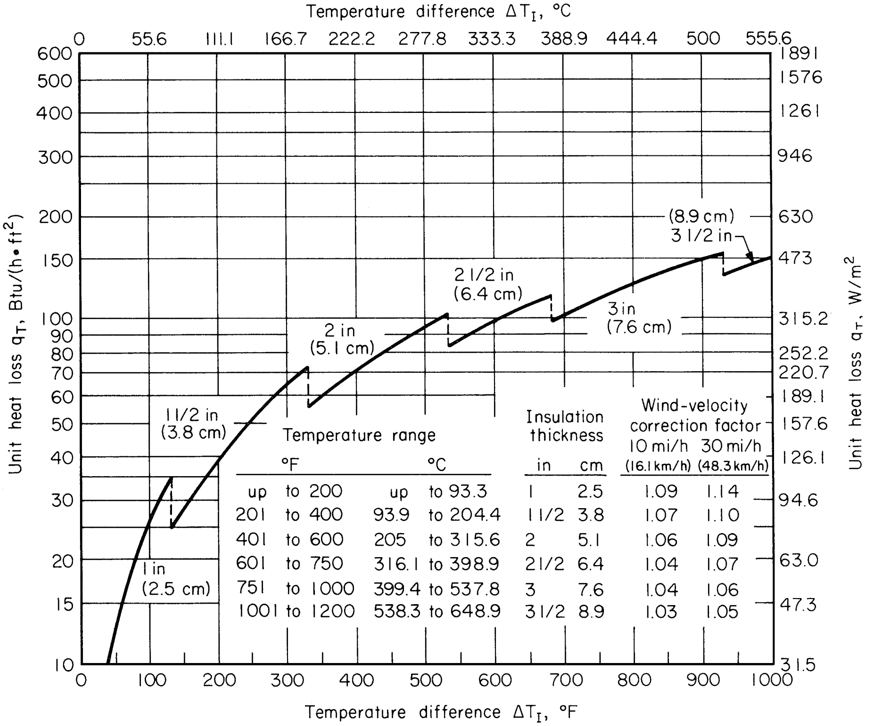
10. Compute the heat loss from liquid and vapor in the tank. Heat loss from the liquid is $Q_L = A_L q_L = 1130(50.6) = 57{,}178$ Btu/h (16,757.2 W).

Now the heat loss from the insulated dry-side surface which contacts the vapor is computed from $A_V = \pi(20)(30 - 18) = 754$ ft^2 (70.0 m^2). Then $Q_V = A_V q_V = 754(50.6) = 38{,}152$ Btu/h (11,181.2 W).

TABLE 1 Air-Pollution Control Criteria

Substance	Maximum ground concentration, ppm*	Lower explosive limit, ppm	Odor threshold, ppm
Acetylene	—	2.5	—
Ammonia	100	15.5	53
Amylene	—	1.7	2.3
Benzene	50	1.4	1.5
Butane	—	1.9	5,000
Carbon monoxide	100	12.5	Odorless
Ethylene	—	2.8	—
Hydrogen sulfide	30	4.3	0.1
Methanol	200	6.7	410
Propane	—	2.1	20,000
Sulfur dioxide	10	—	3.0

*8-h exposure.



Temperature range				Insulation thickness		Wind-velocity correction factor	
°F		°C		in	cm	10 mi/h (16.1 km/h)	30 mi/h (48.3 km/h)
up	to 200	up	to 93.3	1	2.5	1.09	1.14
201	to 400	93.9	to 204.4	1 1/2	3.8	1.07	1.10
401	to 600	205	to 315.6	2	5.1	1.06	1.09
601	to 750	316.1	to 398.9	2 1/2	6.4	1.04	1.07
751	to 1000	399.4	to 537.8	3	7.6	1.04	1.06
1001	to 1200	538.3	to 648.9	3 1/2	8.9	1.03	1.05

FIGURE 11 Insulated tanks, covered with calcium silicate, have heat losses based on negligible resistance to heat flow on process side. Values in chart are for wind velocity of zero, emissivity of 0.8, ambient air temperature of 70°F (21.1°C). (*Chemical Engineering.*)

11. Determine the heat loss from the uninsulated roof. The area of the roof is $A_R = \pi D^2/4 = \pi(20)^2/4 = 314$ ft^2 (28.3 m^2). Correcting the temperature difference for a noncondensing vapor by using Fig. 10, we find $\Delta T_W = (T_i - T_A)W = 230(0.20) = 46$°F (7.77°C), where T_i = tank contents temperature, °F.

Using this corrected temperature difference and Fig. 10, we see the unit heat loss for radiation and convection can be found after the emissivity is determined. From the ASHRAE *Guide,* the surface emissivity is 0.40 for bright aluminum-painted surfaces at temperatures in the 50 to 100°F (10 to 37.7°C) range.

The unit heat loss from the roof for radiation and convection is found by using the corrected temperature difference and Fig. 10 and correcting the radiation loss for an emissivity of 0.4 and the convection loss for the wind velocity: $q_a = 54.2(0.4) = 21.7$ Btu/(h·ft^2) (68.4 W/m^2); $q_c = 35.5[(1.28)(30) + 1]^{0.5} = 222.8$ Btu/(h·ft^2) (702.3 W/m^2). Then $q_R = 21.7 + 222.8 = 244.5$ Btu/(h·ft^2) (770.7 W/m^2). Hence, for the roof, $Q_R = A_R q_R$, where A_R = roof area, ft^2. Or, $Q_R = 314(244.5) = 76{,}773$ Btu/h (22,499.9 W).

12. Determine the total heat loss from the tank. The total heat loss from the tank is the sum of the component heat losses. Or, since the heat loss to the ground is the same as in situation (*a*), $Q_T = 57{,}543 + 38{,}529 + 76{,}773 + 7360 = 180{,}405$ Btu/h (52,871.3 W).

Using the algebraic equations gives $Q_T = 147{,}945$ Btu/h (43,358.2 W). This is a difference greater than in the first situation, but the time savings accrued from using the approximations are significant.

Related Calculations This procedure can be used for a variety of insulated and uninsulated tanks and vessels used to store oil, chemicals, food, water, and similar liquids in almost any industry. Table 1 lists the typical conditions encountered with such tanks and vessels, the factors which can be neglected in the insulation calculations, and the exact procedure to follow for both graphical and algebraic methods. The methods given here are the work of Richard Hughes and Victor Deumango of the Badger Company and reported in *Chemical Engineering* magazine.

SAVING ENERGY COSTS BY RELOCATING HEAT-GENERATING UNITS

A vacuum pump is driven by a 10-hp (7.5-kW) electric motor and is located in a refrigerated packing room. Determine the energy saving if the vacuum pump and motor are moved out of the room into a noncooled area. Find the energy cost if only the motor is removed from the room. The refrigeration unit has a coefficient of performance of 2.5, the cost of electricity is \$0.05 per kilowatthour, the vacuum pump operates 75 percent of the time, and the packing room is cooled 2000 h/year.

Calculation Procedure

1. Determine the annual cooling load for the motor and pump. Use Fig. 12 to find the heat gain for continuously operated electric motors and equipment. Entering Fig. 12 at 10 hp (7.5 kW) at the bottom and projecting up to the top curve, we see that the amount of heat generated is 30,000 Btu/h (8790 W).

2. Adjust the cooling load for the actual operating time. The pump operates 75 percent of the packing-room annual schedule of 2000 h. Hence, annual cooling load = 0.75(30,000 Btu/h)(2000 h/year) = 45×10^6 Btu/year (47.5 kJ/year).

3. Compute the energy saved. With a coefficient of performance (COP) of 2.5, the energy saved = (annual cooling load, Btu/year) (2.93×10^{-4} kWh/Btu)/COP, or (45×10^6 Btu/year) (2.93×10^{-4} kWh/Btu)/2.5 = 5274 kWh/year. At \$0.05 per kilowatthour for electric power, the annual saving = (\$0.05) (5274) = \$263.70.

4. Determine the saving if only the motor is removed. With only the motor removed from the refrigerated room, the saving is smaller. Figure 12 shows a 4500-Btu/h (1318.5-W) heat gain for a 10-hp (7.5-kW) motor alone. From the assumptions from steps 3 and 4, annual cooling load = 0.75 (4500 Btu/h)(2000 h/year) = 6.75×10^6 Btu/year (7.12 kJ/year). Then, energy saved = (6.75×10^6 Btu/year)(7.12 kJ/year) (2.93×10^{-4} kWh/Btu)/2.5 = 786 kWh/year. The monetary saving is (\$0.05)(786) = \$39.30 per year.

Related Calculations This method of calculation can be used for any type of motor-driven equipment located in air-conditioned or refrigerated areas. The total heat gain depends on the nameplate (brake) horsepower of the motor and on the motor efficiency. Or, total heat gain, Btu/h (W) = [2545 Btu/(hp·h)] (P/E), where P = brake horsepower; E = motor efficiency.

Heat given off by a motor or powered equipment can also be expressed as motor heat gain, Btu/h = [2545 Btu/(hp·h)]P)(1 − E)/E. Also, equipment heat gain, Btu/h = [2545 Btu/(hp·h)]P. Because smaller motors are less efficient, they contribute more of the total heat gain than large motors. For example, 36 percent of the heat gain is contributed by a 0.25-hp (0.19-kW) motor, but only 9 percent by a 250-hp (186.5-kW) motor.

Use this procedure for motor-driven equipment in commercial buildings, factories, ships, aircraft, cold-storage warehouses, and other installations where the heat given off by the

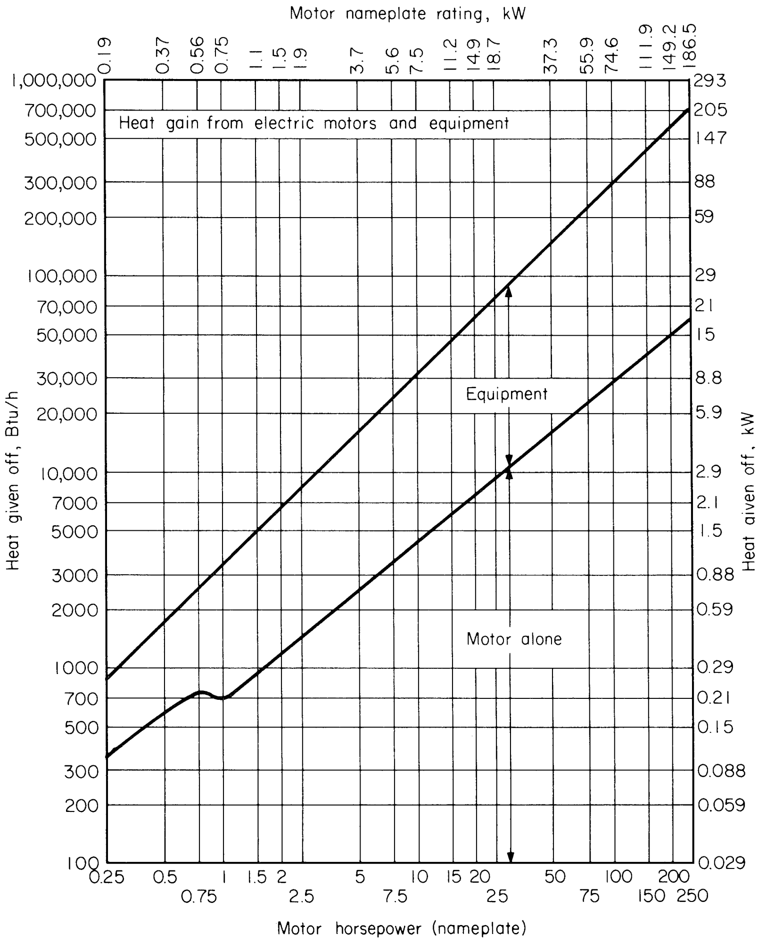


FIGURE 12 Heat gain from electric motors and equipment. (*Chemical Engineering.*)

motor or equipment will place an extra or unwanted load on the air-conditioning system. The procedure given here is the work of Walter A. Hendrix and William G. Moran, Engineering Experiment Station, Georgia Institute of Technology, as reported in *Chemical Engineering* magazine.

ENERGY SAVINGS FROM VAPOR RECOMPRESSION

Determine the energy savings possible in a plant where 15-lb/in^2 (gage) (103.4-kPa) steam is vented to the atmosphere while 5000 lb/h (2250 kg/h) of 40-lb/in^2 (gage) (275.8-kPa) steam is used from the boiler in another process, if the vented steam is recompressed in an electrically driven compressor to the 40-lb/in^2 (gage) (275.8-kPa) level. The boiler feedwater temperature is 80°F (26.7°C), the boiler efficiency is 80 percent, the boiler operates 8000 h/year, the cost of 150,000-Btu/gal no. 6 fuel oil is \$1.00 per gallon, and electricity costs \$0.03 per kilowatthour.

Calculation Procedure

1. Compute the annual heat input to the boiler. The annual heat input to the boiler $H = W(\Delta h)T/e$, where H = heat input, Btu/year (W); W = weight of steam used, lb/h (kg/h); Δh = enthalpy change in the boiler = enthalpy of steam – enthalpy of the feedwater, both expressed in Btu/lb (J/kg); T = annual operating time of the boiler, h; e = boiler efficiency, expressed as a percentage. Substituting, we find $H = 5000(1176 - 48)(8000)/0.80 = 5.6 \times 10^{10}$ Btu/year (5.91×10^{10} J/year).

2. Find the annual fuel cost for generating the steam in the boiler. The annual fuel cost $C = HP/h_v$, where C = annual fuel cost; P = price per gallon of fuel oil; h_v = fuel heating value, Btu/gal; other symbols as before. Substituting gives $C = (5.6 \times 10^{10})(\$1.00)/150{,}000 = \$373.333$ per year.

3. Determine the recompression energy input. When 15-lb/in^2 (gage) (103.4-kPa) waste steam is compressed to 40 lb/in^2 (gage) (275.8 kPa) by an electrically driven compressor, the compressor ratio is $c_r = P_d$ = discharge pressure, lb/in^2 (abs) (kPa)/p_i = inlet pressure, lb/in^2 (abs) (kPa). Substituting gives $c_r = (40 + 14.7)/(15 + 14.7) = 1.84$. To find the recompression energy input, enter Fig. 13 at the computed compression ratio and project vertically to the inlet pressure curve. At the left read the energy input as 66 Btu/lb (153.5 kJ/kg) of steam recompressed.

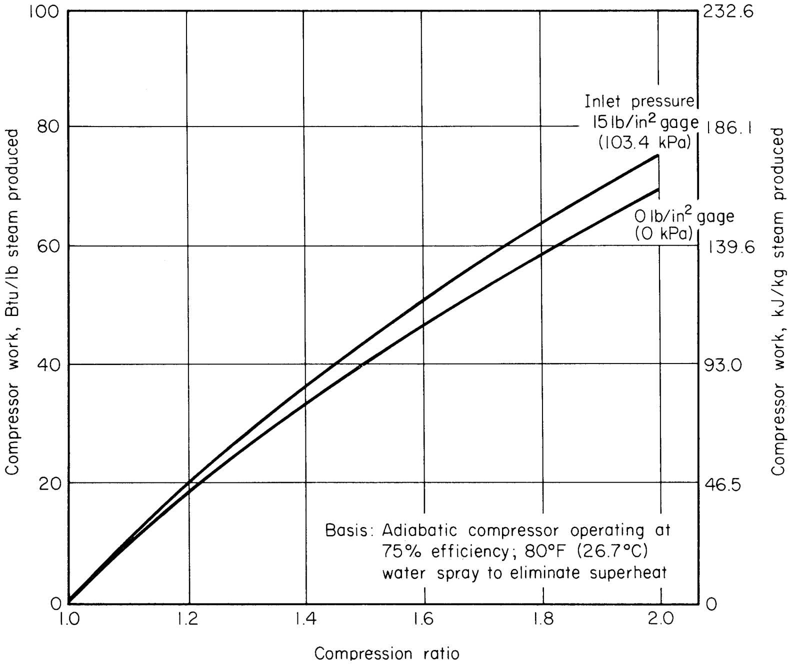


FIGURE 13 Recompression work input. (*Chemical Engineering*.)

4. Compute the energy cost of recompression. The energy cost of recompression = (Btu/lb for recompression) (WT) (0.000293 kWh/Btu) ($0.03 per kWh) = $23,000 per year.

5. Find the annual energy cost saving from recompression. The annual energy cost saving for this installation would be $373,333 – $23,200 = $350,133.

Related Calculations It is common practice in many industrial plants to vent any steam at pressures below 20 lb/in^2 (gage) (137.9 kPa) to the atmosphere. At the same time, there may be several users that require somewhat higher-pressure steam, 30 to 50 lb/in^2 (gage) (206.9 to 344.6 kPa). Rather than reducing high-pressure boiler steam to supply these needs, it is possible to compress the waster low-pressure vapor to a higher pressure so that it can be reused. Although energy must be supplied to the compressor to raise the steam pressure, this operation typically requires only 5 to 10 percent of the energy necessary to generate the same steam in a boiler. In practice, this principle is limited to situations where the compressor inlet pressure is above 14.7 lb/in^2 (abs) (101.4 kPa) and the compression ratio is less than about 2.0, owing to physical limitations of the compressors used.

Vapor recompression has been used for years as a means of lowering the steam requirements of evaporators. In this application, the overhead vapors are compressed and recycled to an evaporator steam chest where they evaporate more liquid. In this way, a single-effect evaporator can achieve a steam economy equivalent to an evaporator with up to 15 effects.

Figure 13 indicates the energy required to compress steam as a function of the compression ratio and inlet pressure. It is based on adiabatic compression with a compressor efficiency of 75 percent. Since the steam leaving the compressor is superheated, it is also assumed that water at 80°F (26.7°C) is sprayed into the steam to eliminate the superhead. The graph can be used in conjunction with standard steam tables to estimate the energy saving possible from employing vapor recompression.

The procedure given here can be used for any application—industrial, commercial, residential, marine, etc.—in which recompression of steam might prove economical. This procedure is the work of George Whittlesey and John D. Muzzy, School of Chemical Engineering, Georgia Institute of Technology, as reported in *Chemical Engineering* magazine.

EFFECTIVE STACK HEIGHT FOR DISPOSING PLANT GASES AND VAPORS

Acetylene (molecular weight 26) is being emitted from a process at 100 lb/min (0.76 kg/s). What stack height is needed to achieve an allowable downwind concentration of 40 percent of the lower explosive limit of 2.5?

Calculation Procedure

1. Convert the flow rate to moles. To convert from lb/min to mol/h, use the expression $M = 60L/m$, where M = flow rate, mol/h; L = flow rate, lb/min; m = molecular weight of the flowing gas or vapor. Substituting gives $M = 60\ (100)/26 = 230$ mol/h.

2. Determine the allowable concentration of the vapor. Use the relation $A = E_L L_e$, where A = allowable concentration downwind of the stack, in percent. Or, $A = 2.5(0.40) = 1.00$ volume percent.

3. Find the required stack height. Use Fig. 14 to determine the stack height, entering at the top of the chart at the lower explosive limit of 1 volume percent and projecting vertically downward to

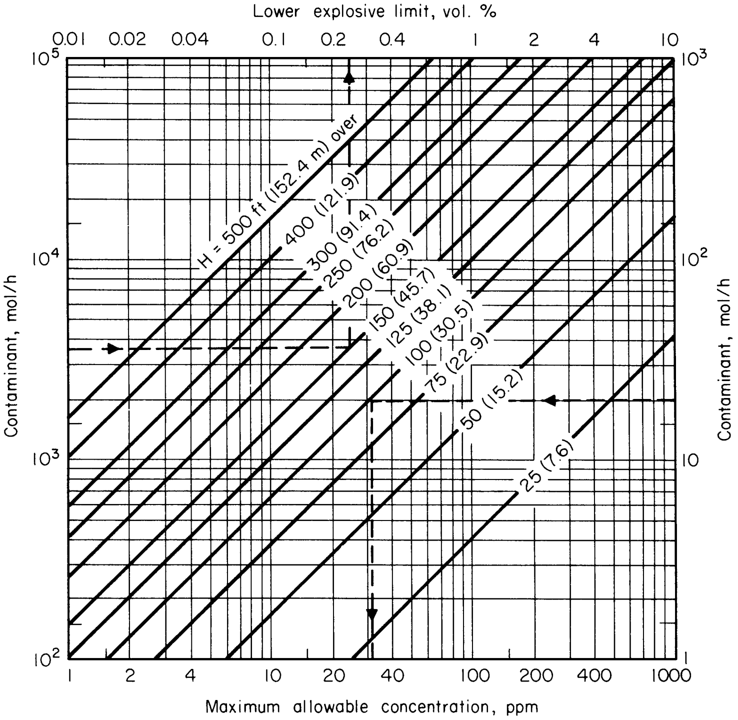


FIGURE 14 Stack height needed to dispose of plant gases and vapors. (*Chemical Engineering.*)

the contaminant flow rate of 230 mol/h. The intersection is just below the stack height of 25 ft (7.6 m). Use a height of 25 ft (7.6 m) because this height is accepted in industry as the minimum allowable. This is the effective height for a 1-mi/h (1.6-km/h) wind.

4. Find the distance downwind from the stack. The distance downwind from the stack where the maximum concentration will be found varies with the turbulence conditions in the area. Turbulence parameters as given by Bosanquet-Pearson are as follows:

	p	q	p/q
Low turbulence	0.02	0.04	0.50
Average turbulence	0.50	0.08	0.63
Moderate turbulence	0.10	0.16	0.63

Note: p and q are the vertical and horizontal dimensionless diffusion coefficients, respectively.

The distance downwind from the stack for maximum concentration of the effluent is given by $d = H/2p$, where d = distance, ft (m); H = stack height, ft (m); p = vertical diffusion coefficient. For low turbulence, $d = 25/2(0.02) = 625$ ft (190.6 m). For moderate turbulence, $d = 25/2(0.1) = 125$ ft (38.1 m).

Related Calculations When designing emission control systems, be sure to consult the local air-pollution control ordinance (if any) for the criteria which must be met. If an odorous pollutant is being emitted, the design basis will be the concentration below the odor threshold (Table 1) at ground level outside the plant.

Do not design for emission directly to the atmosphere in areas where atmospheric temperature inversions are known to occur. Use, instead, a closed system to rid the plant of the gas or vapor. Extreme care in design is required to avoid the possibility of a legal nuisance suit. Be sure that all applicable ordinances are reviewed before any final design work is begun.

Figure 14 is based on the well-known Bosanquet-Pearson formula. Two solutions plotted on Fig. 14 are for a contaminant flow rate of 3800 mol/h and a lower explosive limit of 0.25 volume percent, requiring a stack height of 150 ft (45.7 m); and a contaminant flow rate of 20 mol/h and a maximum allowable concentration of 31 ppm, requiring a stack height of 75 ft (22.9 m).

As a general guide for stack design, the following relations are given by the above formula:

1. Concentration of an effluent downwind from a source is directly proportional to the discharge quantity.
2. It is impossible to alter materially the downwind ground-level concentration of a contaminant by diluting the effluent.
3. Concentrations downwind of a stack are inversely proportional to wind speed; doubling the wind speed cuts pollutant concentration by half.
4. Pollutant concentration is inversely proportional to the square of the stack height. Doubling the stack height reduces the maximum ground-level concentration to one-fourth the previous level.
5. Location of the maximum ground-level concentration depends on atmospheric stability. When atmospheric conditions are unstable—i.e., wind speeds are low or there is an inversion—the maximum concentration occurs close to the stack. As the wind speed increases and the inversion disappears, the maximum ground-level concentrations move farther away from the stack.
6. Figure 14 is based on a wind speed of 1 mi/h (1.6 km/h) and $p/q = 1$. Further, a smooth, level terrain was ensured for the equation from which the chart is plotted.
7. The effective height to which the plume from a stack rises before it begins to turn downward is the actual height of the stack plus the plume rise created by the sum of the exit velocity and the difference in density above the plume. Use the relation $H_v = 4.77(Q_1 V/1.5)^{0.5}$, where H_V = plume rise due to exit velocity, ft; V = stack exit velocity, ft/s; $Q_1 = Q_V T_1/530$, in which Q_V = stack exit volume, ft^3/s; T_1 = 18.3 (molecular weight of contaminant).

This procedure can be used for a variety of gases and vapors, including acetylene, ammonia, amylene, benzene, butane, carbon monoxide, ethylene, hydrogen sulfide, methanol, propane, and sulfur dioxide. Pertinent data for these effluents are given in Table 1.

This calculation procedure is the work of John D. Constance, Consultant, as reported in *Chemical Engineering* magazine.

Where plant gases or vapors pollute the local environment, expensive pollution-abatement equipment may be required. On the west coast of the United States certain chemical and refining plants are gaining emissions credits for their stacks by eliminating pollution elsewhere.

Thus, at the time of writing, one chemical company is paying $700 each for pre-1972 cars. Each car and light truck purchased is cut up for scrap, thereby eliminating the smoke and pollution such vehicles emit. Emissions credits are issued to the chemical plant for each vehicle scrapped. This approach to emissions control is in accord with federal regulations requiring companies to either reduce their own pollution or obtain emissions credits by reducing other pollutants generated in the area.

SAVINGS POSSIBLE FROM USING LOW-GRADE WASTE HEAT FOR REFRIGERATION

An industrial plant presently exhausts low-pressure steam to the atmosphere. What would the annual savings be if a mechanical chiller having a coefficient of performance (COP) of 4.0 producing an average of 150 tons/year (527.4 kW) of refrigeration for 4000 h were replaced by an absorption refrigeration unit using the exhaust steam? The cost of electricity is 3.0 cents per kilowatthour.

Calculation Procedure

1. Sketch the refrigeration system being considered. Figure 15 shows the absorption refrigeration unit being considered. The heat input from the exhaust steam is indicated as Q_C.

In an absorption refrigeration unit, the mechanical vapor compressor is replaced by the generator which uses steam or hot water to revaporize the refrigerant. But absorption refrigeration units are characterized by a low COP, compared to the mechanical type. Hence, to be competitive, an absorption unit must use low-grade waste heat to power the generator. The low-pressure steam available in this plant would be ideal for this purpose.

2. Determine the hourly savings possible. Use Fig. 16 to find the hourly savings. Enter at the refrigeration load of 150 tons (527.4 kW) on the left, and project vertically to the COP value of 4.

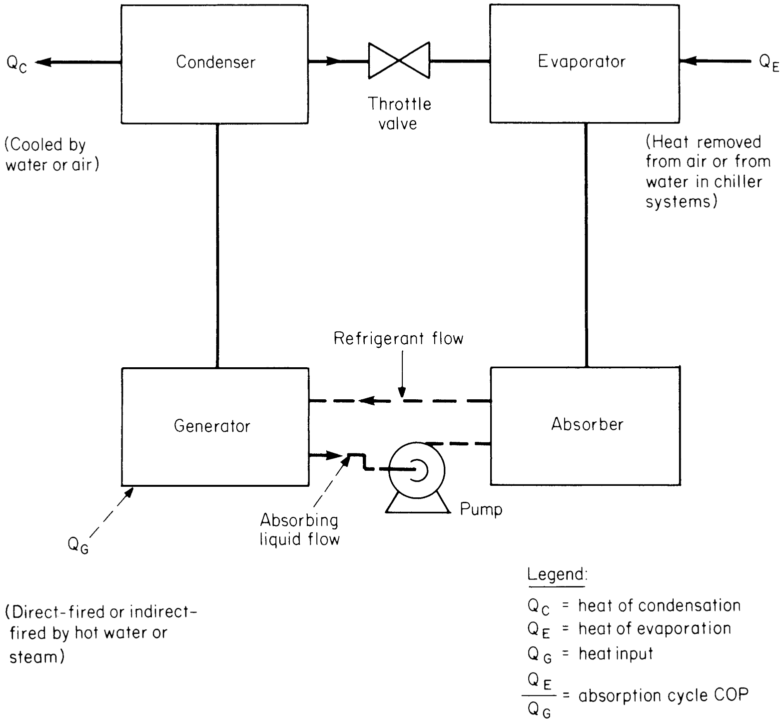


FIGURE 15 Absorption refrigeration system. (*Chemical Engineering.*)

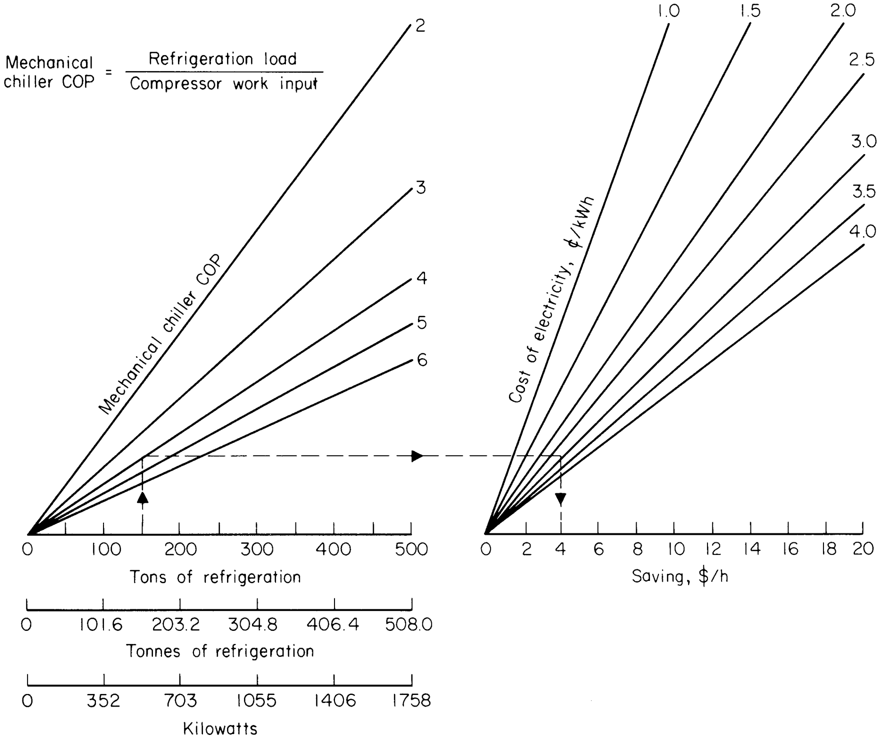


FIGURE 16 Savings using low-grade heat. (*Chemical Engineering.*)

From the intersection with this curve, project horizontally to the right to intersect the electricity cost curve of 3 cents per kilowatthour. At the bottom read the saving as $4 per hour.

3. Compute the annual savings. Use this relation: annual savings, $ = (hourly savings)(annual number of operating hours) = ($4)(4000) = $16,000 per year.

Related Calculations This procedure can be used for an absorption refrigeration system by using waste heat in the generator. The heat can be in the form of exhaust steam, hot waste liquids, warm air, etc. The source of the heat is not important provided (1) the temperature of the heating medium is high enough for use in the generator, (2) the supply of heat is steady, and (3) the heat is not chargeable to the refrigeration process.

Given the above criteria as guidelines, the procedure given here can be used for absorption refrigeration machines in industrial plants, commercial building, ships, and domestic applications. Where the COP of an equivalent mechanical refrigeration system is not known, it can be approximated by applying data from known similar installations. Where the electricity cost exceeds the plotted values, use half the actual cost and multiply the result by 2.

This procedure is the work of Guillermo H. Hoyos, Universidad de los Andes, and John D. Muzzy, School of Chemical Engineering, Georgia Institute of Technology, as reported in *Chemical Engineering* magazine.

EXCESS-AIR ANALYSIS TO REDUCE WASTE-HEAT LOSSES

The fuel input to the primary reformer of an ammonia plant is a mixture of natural gas, tail gas, and naphtha with these flow rates and compositions:

	Natural gas	Tail gas	Naphtha
Fuel no.	1	2	3
Flow rate, kg/h	6800	3420	7800
C:H weight ratio	3.20	2.40	5.35
Weight % carbon	72.3	17.4	84.2
Volume % incombustibles	2.8	53.0	nil

How much excess air is being fed to the reformer if the flue gas analysis shows 3.5 percent O_2 and 11 percent CO_2 on a dry basis?

Calculation Procedure

1. Calculate the hydrogen:carbon weight ratio. The given ratios are for carbon:hydrogen, or the inverse of the required ratios. Hence, the carbon:hydrogen ratio r for each fuel can be found by inverting the given values. Or, for fuel no.1, natural gas, $r_1 = 1/3.20 = 0.3125$. For fuel no. 2, tail gas, $r_2 = 1/2.40 = 0.4167$; for fuel no. 3, naphtha, $r_3 = 1/5.35 = 0.1869$.

2. Determine the weight of carbon in the fuel. The weight of carbon in the fuel c = fraction of carbon × mass flow rate. Or, for natural gas, fuel no. 1, $c_1 = 0.723 \times 6800 = 4915$ kg/h (10,837.7 lb/h), where 0.723 is the weight percent of carbon in the natural gas, expressed as a decimal and given in the table above. Then, for fuel no. 2, tail gas, $c_2 = 0.174 \times 3450 = 600.3$ kg/h (1323.4 lb/h). For fuel no. 3, naphtha, $c_3 = 0.842 \times 7800 = 6568$ kg/h (14,478.8 lb/h).

3. Determine the mole fraction (volume fraction for gases) of incombustibles in the fuel. Divide the given percentage of incombustibles by 100 to get the mole fraction for each fuel. Or, for natural gas, $a_i = 2.8/100 = 0.028$; for tail gas, $a_i = 53.0/100 = 0.53$; for naphtha, a_i = nil/100 = nil.

4. Compute the value R for the fuel. Here R is defined as $R = \Sigma(1 - a_i)r_i c_i / \Sigma(1 - a_i)c_i$, where the summations are over all fuels in the mixture. Substituting yields $R = [(1 - 0.028)(0.3125)(4916) + (1 - 0.53)(0.4167)(600) + (1 - 0)(0.1869)(6568)]/[(1 - 0.028)(4916) + (1 - 0.53)(600) + (1 - 0)(6568)] = 0.2441$.

5. Calculate the percentage of excess air. Use the relation, percentage of excess air = [100(%O_2 in dry flue gas)]/(%CO_2 in dry flue gas)(1 + 3R). Substituting the previously calculated values gives percentage of excess air = 100(3.5)/11[1 + 3(0.2441)] = 18.36, say 18.4 percent.

Related Calculations Combustion processes require at least stoichiometric air to get complete fuel utilization, but excess air should be limited to keep the waste heat carried away by flue gases to a minimum. Because it is linked to energy usage, excess air is a key factor in evaluating combustion processes.

The method given here is useful for determining the percentage of excess air being fed to a combustion process. It can be used for both liquid and gaseous fuels, singly or in mixtures, as well as for fuels that contain incombustibles. The method only requires readily available information such as fuel characteristics, flow rates, and flue-gas analysis and avoids the laborious solution of simultaneous material-balance equations that is usually needed.

This calculation procedure is the work of S. Michael Antony, IFFCO, Ltd., as reported in *Chemical Engineering* magazine.

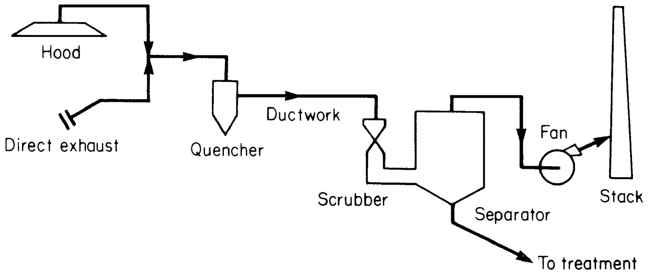


FIGURE 17 Scrubber-separator in an air-pollution control system. (*Chemical Engineering.*)

ESTIMATING SIZE AND COST OF VENTURI SCRUBBERS

Determine the size and cost of a venturi scrubber to handle 100,000 actual ft^3/min (47.2 m^3/s) of gas entering the venturi in an air-pollution control system. The scrubber, Fig. 17, is to remove all particles larger than 0.6 μm.

Calculation Procedure

1. Determine the scrubber base cost. The cost of a venturi scrubber depends on the volumetric flow, operating pressure, and materials of construction. Figure 18 gives flange-to-flange costs—covering the venturi, elbow, separator, pumps, and controls—for base systems [i.e., constructed of

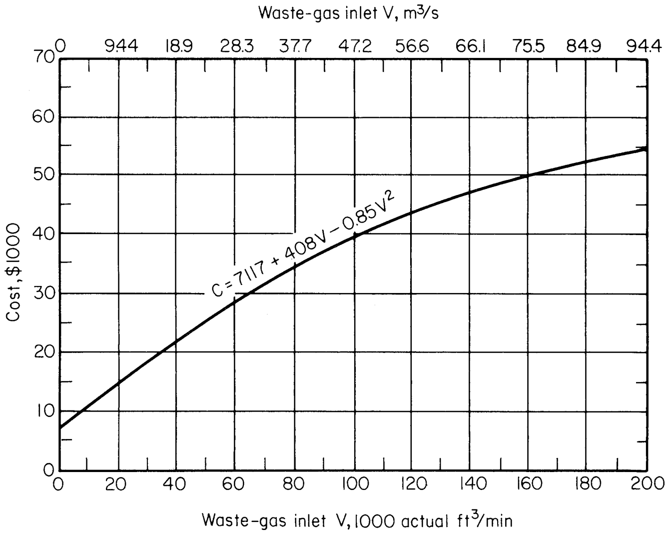


FIGURE 18 Base scrubber cost, flange-to-flange construction of 0.125-in (0.32-cm) carbon steel. (*Chemical Engineering magazine and Fuller Company.*)

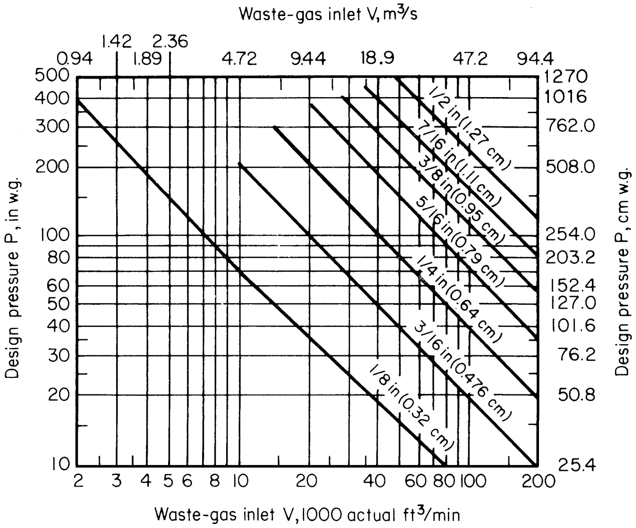


FIGURE 19 Flow rate and design pressure dictate scrubber metal thickness. (*Chemical Engineering magazine and Fuller Company.*)

0.125-in (0.32-cm) carbon steel] for different volumetric flow rates. Do not extend the curve or equation beyond 200,000 actual ft^3/min (94.4 m^3/s).

Whether a thickness other than 0.125 in (0.32 cm) is required can be determined from Fig. 19, the thickness being a function of design operating pressure and shell diameter. The curves in Fig. 19 include a safety factor of 2, but no allowance is made for corrosion and erosion. Figure 19 is used after the base cost is determined from Fig. 18.

Entering Fig. 18 at 100,000 ft^3/min (47.2 m^3/s) at the bottom and projecting vertically upward to the curve, we find a base cost of $39,400. From the equation, the exact cost would be $39,417.

2. Find the pressure drop in the scrubber. For an efficiency that removes all particles larger than 0.6 μm, Fig. 20 shows that a 35-in water gage (w.g.) (88.9-cm w.g.) pressure drop is needed. Note that as the particle size removed decreases, the pressure drop required for particle removal increases.

3. Compute the metal thickness and scrubber cost. At the 35-in w.g. (88.9-cm w.g.) pressure drop, Fig. 19 shows that a scrubber metal thickness of 0.25 in (0.64 cm) is required. (Always round *up* to the next standard metal thickness when you use this procedure.) For 0.25-in (0.64-cm) carbon steel, Fig. 21 gives a cost adjustment factor of 1.6. Hence, the scrubber cost will be 1.6 × $39,400 = $63,040.

If the scrubber metal is to be 304 or 316 stainless steel, multiply the above cost estimate, $63,040, by 2.3 or 3.2, respectively. And if the scrubber is to be equipped with a fiberglass lining to reduce wear of the metal, multiply the Fig. 18 estimate of $39,400 by 0.15 to obtain $5910, and add this to $63,040 to arrive at an estimate of $68,950.

4. Determine the cost of a rubber-lined scrubber. If the scrubber is to be lined with 0.1857-in (0.476-cm) rubber, or any other thickness rubber, determine the internal surface area of the scrubber from Fig. 22. For 100,000 actual ft^3/min (47.2 m^3/s), the scrubber internal area is 1500 ft^2 (139.4 m^2). The unit cost for lining a scrubber with this thickness rubber is $4.69 per ft^2 ($50.48 per m^2). Hence, the total cost = (1500 ft^2)($4.69/ft^2) = $7035. Adding this to the $63,040 gives an estimate of $70,075.

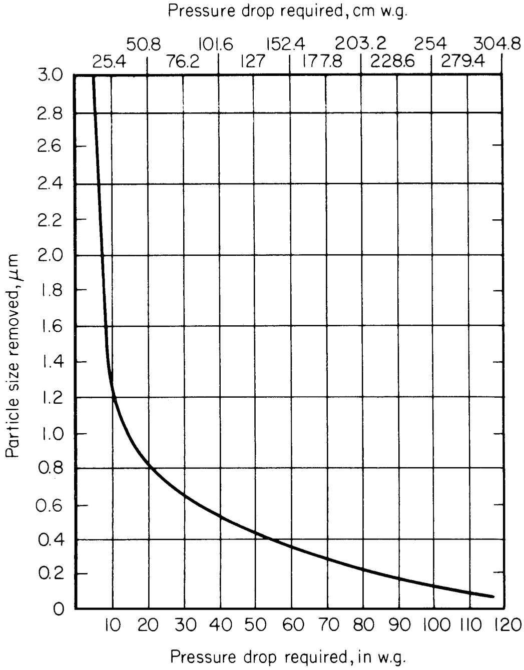


FIGURE 20 Correlation gives efficiency performance of venturi scrubbers. (*Chemical Engineering magazine and AIChE.*)

5. Determine the number and diameter of the scrubber trays. If the separator is to be equipped for gas cooling, the number of trays needed must be determined, based on an average of 5 lb (2.3 kg) of water removed per square foot of tray area, with an outlet gas temperature about 40°F (22.2°C) higher than the inlet water temperature. [This is valid for typical scrubber outlet-gas temperatures of 200°F (93.3°C) or less, cooling-water temperature of about 70°F (21.1°C), and superficial gas velocities of 600 ft/min (3.05 m/s).] The total water to be removed is determined from the difference between the absolute humidities of the inlet and outlet gas streams.

Find the diameter of each tray from Fig. 22 for a flow of 100,000 ft^3/min as 13.5 ft (4.1 m). Figure 23 gives a cost of $14,000 per tray. This includes the cost of the tray plus the cost of additional separator height to contain the tray.

If the separator requires six trays to achieve the dehumidification required, the $70,075 estimate will be increased by $84,000 (= 6 × $14,000) to $154,075, say $154,000. (If the chart in Fig. 18 were used instead of the equation, the total estimate would be $153,000 after rounding.) To update the estimate to current costs, use a suitable cost index, as detailed below.

Related Calculations Venturi scrubbers are highly efficient in removing submicron dust particles from gas streams.

Basically, the gas stream accelerates in the converging section of the venturi to maximum velocity in the throat, where it is sprayed by a scrubbing liquor. The faster velocity of the gas stream atomizes the liquor and promotes collisions between the particles and the droplets. Agglomeration in the diverging section produces droplets, with entrapped particles, of a size easily removed by mechanical means.

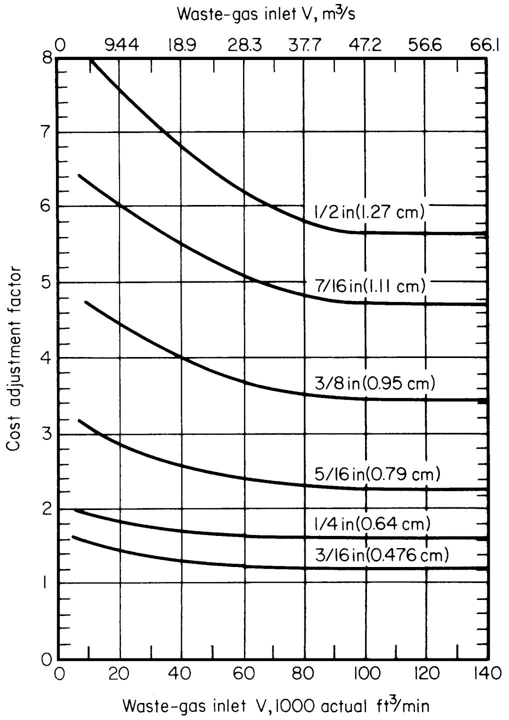


FIGURE 21 Factors adjust scrubber cost for flow rate and metal thickness. (*Chemical Engineering magazine and Fuller Company.*)

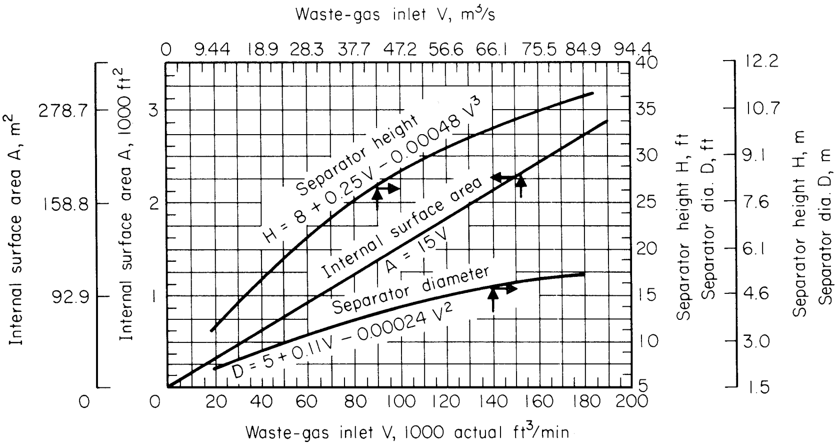


FIGURE 22 Scrubber internal surface area and separator dimensions. (*Chemical Engineering magazine and Fuller Company.*)

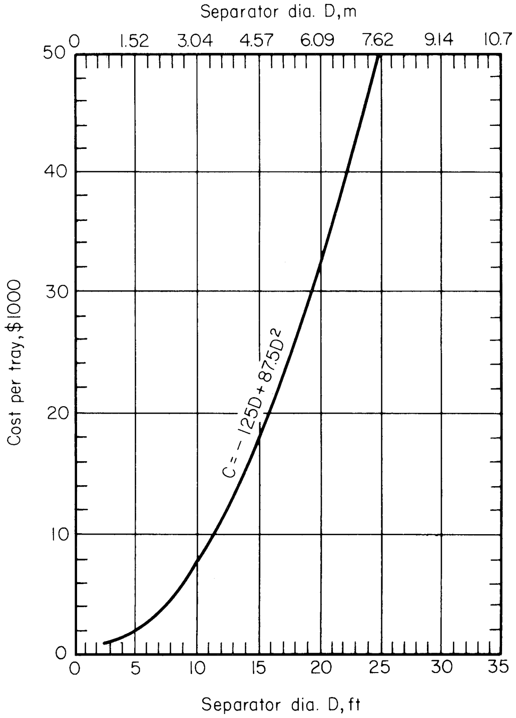


FIGURE 23 Cost per tray of separator internal cooler. (*Chemical Engineering magazine and Fuller Company.*)

Collection efficiency depends on the venturi pressure drop, which is a function of gas-stream throat velocity and scrubbing-liquor flow rate. The smaller the particles, the higher the pressure drop required. Venturis are normally operated at pressure drops between 6 in (15.2 cm) and 80 in (203.2 cm) water gage, depending on the characteristics of the dust, and at liquor flow rates of 3 to 20 gal/min (0.19 to 1.26 L/s) per 1000 actual ft^3/min (0.47 m^3/s). Collection efficiencies range from 99+ percent for 1-μm and larger particles to between 90 and 99 percent for those less than 1 μm.

Precise pressure drops can be obtained from Fig. 20, which represents 100 percent removal of particle sizes indicated for a particular pressure drop. For instance, to remove all particles 0.4 μm (10^{-6} m) and larger requires a pressure drop of 55 in (139.7 cm) w.g. (Also $P_d = 15.4d^{-139}$, with d the diameter in micrometers, also gives the pressure drop.)

A separator—normally a cylindrical tank having a low tangential inlet and a centered top gas outlet—located immediately downstream of the scrubber removes the agglomerated liquor drops from the gas stream by a cyclonic motion that forces them to impinge on the tank wall. Slurry settles into a bottom cone, from which most of it is sent to the water treatment facility, with the cleaner liquid above the sediment being recycled to the venturi (Fig. 17).

In hot processes, a considerable amount of water is vaporized in the scrubber and upstream equipment (particularly the quencher). Unless this vapor is removed, it must be handled by the fan (commonly a radial-tip fan), which therefore must be of higher horsepower and so is more costly to operate.

A gas cooler can be incorporated into the separator to cool and dehumidify the gas stream. Such a cooler can be one of several types, including one in which the gas stream passes through spray banks of cooling water and then impinges on baffles, and another in which the stream rises through perforated holes or bubble caps in trays flooded with cooling water.

The cost data in this procedure are based on 1977 values. They can be easily updated to the current year by using a suitable cost index for the year in which the estimate is being made. Thus, if the cost index (*Chemical Engineering, EN-R, Marshall & Stevens*, etc.) is 229 for the year 1977 and 245 for the design year, the cost of the scrubber in the design year will be \$154,000(245/229) = \$164,759, say \$165,000.

This procedure is the work of William M. Vatavuk of the U.S. Environmental Protection Agency and Robert B. Neveril of Gard, Inc., as reported in *Chemical Engineering* magazine, using data from an AIChE paper and Fuller Company.

Toxic air pollutants are now strictly controlled by the eleven titles of the U.S. Clean Air Act Amendments (CAAA) passed in 1990. Title III deals with 189 chemicals designated as hazardous air pollutants (HAP), which may be emitted by chemical process plants. EPA is required to issue control standards based on maximum achievable control technologies (MACT) for sources designated to be "major" or "area" generators of hazardous pollutants.

Any facility that emits 10 or more tons per year of any single HAP, or 25 or more tons per year of any combination of HAPs is considered a major source. Alternatively, an area source is any facility that routinely emits HAPs but is not classified a major source. EPA is required to ensure that 90 percent of the emissions from the 30 most serious area-source pollutants are regulated by the year 2000. For these reasons, scrubbers like those discussed above are becoming more important to chemical engineers worldwide.

Title IV of the act will impose strict SO_2 and NO_x control requirements on electric utilities. Title V requires operators of plants to apply for, and obtain, permits. These operating permits, issued by the states, will establish emission limits, permit fees, and monitoring and reporting requirements.

Volatile organic compounds (VOC) comprise half of all regulated toxic air pollutants. VOC are regulated by Title I, which imposes strict compliance requirements on particulate matter and ozone precursors, including some VOC. These regulations may exceed MACT requirements under Title III. Data on toxic air pollutants as given here are from K. Sampeth Kumar, Rodney L. Pennington, and Jan T. Zmuda, Research-Cottrell Co., a subsidiary of Air & Water Technologies Corp., as reported in *Chemical Engineering.*

SIZING DESUPERHEATER-CONDENSERS ECONOMICALLY

A reactor exhausts 27,958 lb/h (3.52 kg/s) of isobutane with a small amount of *n*-butane at 200°F (93.3°C) and 85 lb/in^2 (gage) (586.0 kPa). This gas becomes saturated at 130°F (54.4°C) and condenses completely at 125°F (51.7°C). The gas is to be cooled and condensed by a horizontal counterflow heat exchanger like that in Fig. 24 using well water at 65°F (18.3°C) inlet temperature and an outlet temperature of 100°F (37.8°C). How much heat-transfer area is required in this exchanger?

Calculation Procedure

1. Check for condensation at the hot end. Desuperheater-condensers are widely used in the process, petrochemical, chemical, and power industries. Figure 24 shows a horizontal in-shell design which might be used as a high-pressure feed heater, an inter- or aftercondenser in a steam-jet ejector system, or a gas cooler in a compressor system.

Conventional design practice splits the heat load and sizes the desuperheating and condensing zones separately. This assumes that the superheated vapor cools as if it were a dry gas, which is true only when the tube-wall temperature T_w in the desuperheating zone is greater than the vapor's saturation temperature T_{sat}.

When T_w is less than T_{sat}, the superheated vapor condenses directly, in the same way as saturated vapor. Because the heat flux in condensation is much greater than in desuperheating, this situation requires less heat-transfer area than the conventional design would prescribe.

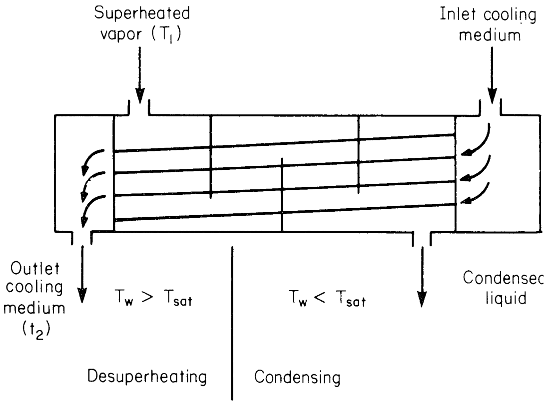


FIGURE 24 Horizontal in-shell desuperheater-condenser. (*Chemical Engineering.*)

For a counterflow desuperheater-condenser with vapor on the shell side like that in Fig. 24, the energy balance for the desuperheating zone is $h_d(T_1 - T_w) = U_d(T_1 - t_2)$, where h_d is the desuperheating heat-transfer coefficient; U_d is the overall heat-transfer coefficient; T_1 is the vapor inlet temperature; t_2 is the cooling-medium outlet temperature. Using this energy-balance equation and the data from Fig. 25, we find the tube-wall temperature at the hot end of the unit is $45.2(200 - T_w) = 42.3(200 - 100)$; $T_w = 106.4$°F (41.3°C). Since T_w is less than T_{sat}, condensation does take place as soon as the superheated vapor enters the shell.

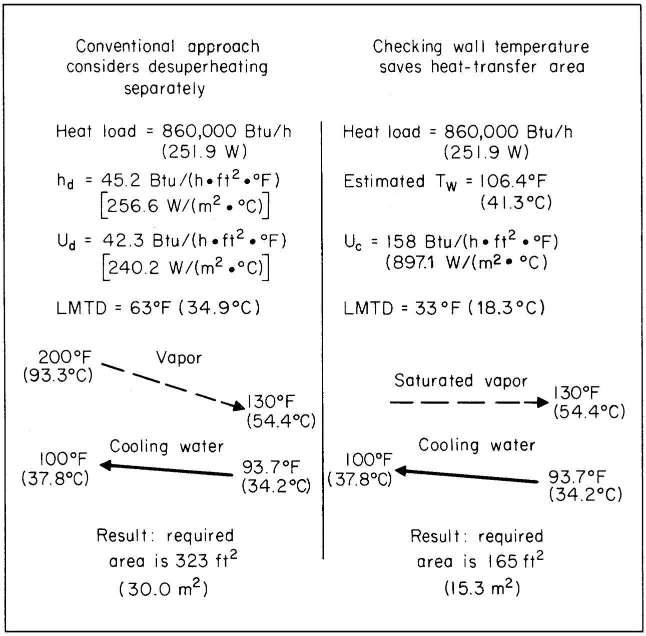


FIGURE 25 Example shows importance of checking wall temperature. (*Chemical Engineering.*)

2. Determine the area required for the condensing load. Since condensation does take place, the entire desuperheating load, 860,000 Btu/h (251.9 W), should be treated as a condensing load, by using the condensing heat-transfer coefficient shown in Fig. 25. The heat-transfer area required for this load is (Btu/h)/U_c(LMTD); or $A = 860{,}000/158(33) = 165$ ft^2 (15.3 m^2).

3. Compare the conventional approach to this approach. Using the conventional approach, such as that in Kern—*Process Heat Transfer*, McGraw-Hill, and in the heat-exchanger calculation procedures given elsewhere in this handbook (see index), shows that the heat duty for the exchanger is split into two zones—condensing and desuperheating. The area required for the condensing zone is 523 ft^2 (48.6 m^2); for the desuperheating zone it is 323 ft^2 (30 m^2). Important data for the desuperheating zone are shown in Fig. 25.

The total heat-transfer area, when the approach given here is used, will be the condensing-zone area from the conventional approach + desuperheating area computed in step 2. Or, $A_{\text{total}} = 523 + 165 = 688$ ft^2 (63.9 m^2). This compares with 523 + 323 = 846 ft^2 (78.6 m^2) for the conventional approach, or 23 percent greater area.

Note that the desuperheat was a sizable fraction (over 20 percent) of the total heat load in this exchanger, which is cooling an organic vapor. With steam, the importance of desuperheating is generally less.

Related Calculations Energy conservation studies often show a longer breakeven period and a smaller payout because equipment costs are excessive. Higher equipment costs lead to greater cost of money for the heat-recovery unit or units. Hence, it is important that any equipment chosen to conserve heat be sized properly. The procedure given here shows how a saving of nearly 25 percent can be made in the area of certain types of heat exchangers. Such savings can significantly reduce the required investment, leading to an earlier breakeven and higher payout. Thus, energy conservation will be easier to justify when this procedure is used.

Since this procedure is relatively simple, it should be applied in selecting a heat exchanger which involves desuperheating. The method is applicable in land, marine, chemical, petrochemical, and process heat-exchanger selection. This procedure is the work of P. S. V. Kurmarao, Ph.D., EDC (Heat Exchangers) Bharat Heavy Electricals Ltd., as reported in *Chemical Engineering* magazine.

SIZING VERTICAL LIQUID-VAPOR SEPARATORS

Find the diameter needed for a vertical vessel to separate a liquid having a density $d = 58.0$ lb/ft^3 (928.6 kg/m^3) from 2000 mol/h of vapor having a molecular weight of 25.0 at an operating temperature of 300°F (148.7°C) and 250 lb/in^2 (gage) (1723.5 kPa). The compressibility factor $Z = 1.0$ for the vapor.

Calculation Procedure

1. Find the vapor volumetric flow rate V. Using the gas law $PV = nRTZ$, let $V =$ ft^3/s and $n = 2000/3600$ mol/s. Solving yields $V = nRTZ/P = (2000/3600)(10.73)(760)(1.0)/264.7 = 17.1$ ft^3/s (48,393 m^3/s). In this equation, $R =$ the gas constant; $T =$ absolute temperature, °R = 460 + operating temperature, °F; $P =$ absolute pressure of the vapor, lb/in^2 (abs).

2. Determine the density of the vapor d_v. Use the relation $d_v =$ (mol/h) (molecular weight)/volumetric flow rate, lb/h. Or, $d_v = (2000)(25)/(17.1)(3600 \text{ s/h}) = 0.812$ lb/ft^3 (13.0 kg/m^3).

3. Compute the terminal vapor velocity v_t. Use the relation $v_t = K'[(d - d_v)/d_v]^{0.5}$, where K' is a constant which ranges between 0.1 and 0.35, with 0.227 being the value for many satisfactory designs and recommended except when special considerations are warranted. Substituting, we find $v_t = 0.227[(58.0 - 0.812)/0.812]^{0.5} = 1.91$ ft/s (0.58 m/s).

4. Find the allowable vapor velocity v_a. Use the relation $v_a = 0.15v_t$, where the constant 0.15 is based on an allowable vapor velocity of 15 percent of v_t to ensure good liquid disentrainment during the normal flow surges. For usual designs, researchers have determined that v_a should be 15 percent of v_t. By substituting, $v_a = 0.15(1.91) = 0.286$ ft/s (0.087 m/s).

5. Determine the separator cross-sectional area and diameter. The separator cross-sectional area $A = V/v_a = 17.1/0.286 = 59.8$ ft^2 (5.6 m^2). Then the separator diameter $D = [(4)(59.8)/\pi]^{0.5} = 8.7$ ft (2.65 m). A diameter of 9 ft (2.74 m) would be chosen.

Related Calculations Vertical liquid-vapor separators are used primarily to disengage a liquid from a vapor when the volume of the first is small compared with that of the second. The separation is accomplished by providing an environment (i.e., a vessel) in which the liquid particles are directed by the force of gravity rather than the force of the flowing vapor.

Devices have been developed to agglomerate liquid particles in a vapor stream and enhance disentrainment. Some act as baffles, causing multiple changes in the direction of vapor flow. Inertia keeps the liquid particles from changing direction, and they impinge on the baffles. As the particles coalesce on the baffles, they agglomerate into droplets, which fall because of gravity.

Other devices, such as packing and grids, provide a large surface area for liquid coalescence and agglomeration. One such device that has gained wide acceptance—because it is highly efficient and relatively inexpensive and causes negligible pressure drop—is the mist elimination pad. Usually a mesh formed by knitting metal wire, it comes in a variety of standard thicknesses and densities. For general process-separator and compressor-suction knockout-pot services, a stainless-steel pad of 4-in (10.2-cm) thickness and nominal 9-lb/ft^3 (144.1-kg/m^3) density is the most economical.

A separator equipped with a mist eliminator can be considerably smaller in diameter than one not having it. Indeed, design practice permits ignoring the 15 percent safety factor and letting the allowable vapor velocity be equal to the terminal vapor velocity (that is, $v_a = v_t$).

In the separator above, therefore, the required cross-sectional area of the vertical separator A now becomes $A = 18.1/1.91 = 9.0$ ft^2 (2.74 m^2). And the separator diameter D becomes $D = [(4 \times 9.0)/\pi]^{0.5} = 3.4$ ft (1.04 m). A diameter of 3 ft 6 in (1.07 m) would now be chosen.

The height of the liquid level in a vertical separator, Fig. 26, depends primarily on the residence time dictated by process considerations. Suppose that the residence time for the above separator is chosen as 5 min. For the 3.5-ft (1.07-m) diameter separator, the cross-sectional area is $A = (\pi/4)(3.5)^2 = 9.62$ ft^2 (0.89 m^2).

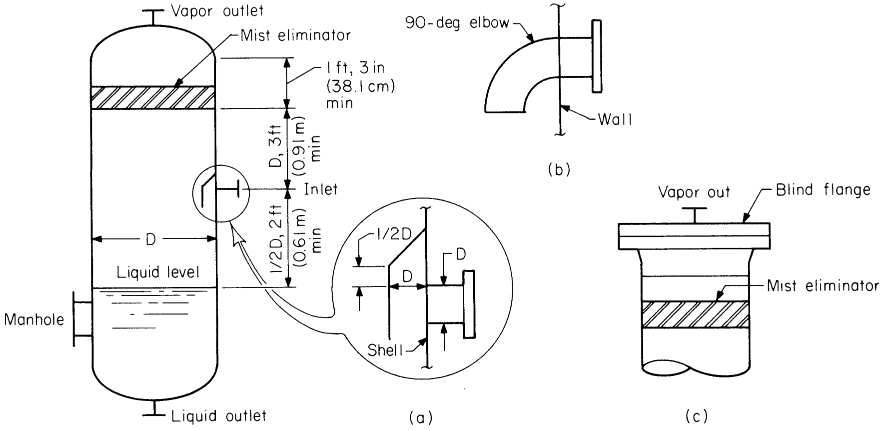


FIGURE 26 (*a*) Inlet configuration, diameter ≧ 30 in (9.1 cm). (*b*) Inlet configuration, diameter < 30 in (9.1 cm). (*c*) Top head configuration of pipe separators, diameter < 30 in (9.1 cm). (*Chemical Engineering.*)

Assuming that liquid is entering the separator at a rate of 2000 gal/h (2.1 L/s), or 4.64 ft^3/ min (0.00219 m^3/s), we find the liquid level for a 5-min residence time is $L = (ft^3/min)$ (residence time, min)/A = 4.46(5)/9.62 = 2.32 ft (0.71 m). Choose a sump height of 2.5 ft (0.76 m). A vertical separator is usually specified when a short liquid holdup time is permitted.

The following procedure is standard in the process design of vertical liquid-vapor separators. A standardized design procedure and vessel configuration saves much engineering time. A separator is usually relatively inexpensive, and the application of a rigorous, sophisticated procedure to achieve an optimum design is seldom warranted. Only in special cases, such as when a separator is built of extra thick laminated shells, does it become economical to attain an optimum design, because the saving in fabrication cost can be significant.

The standard procedure stipulates the following:

1. The allowable vapor velocity v_a in a separator shall be equal to the terminal velocity v_t, calculated by rounding up the vessel diameter to the nearest 6 in (15.2 cm), when a mist eliminator is used. However, v_a shall be no greater than 15 percent of v_t when the separator is not equipped with a mist eliminator.
2. The disengaging space, the distance between any inlet and the bottom of the mist elimination pad (see Fig. 26), shall be equal to the diameter of the separator. However, when the diameter of the separator is less than 3 ft 0 in (0.91 m), the height of the disengaging space shall be a minimum of 3 ft (0.91 m).
3. The distance between the inlet nozzle and the maximum liquid level shall be equal to one-half the vessel diameter, or a minimum of 2 ft (0.61 m).
4. The dimension between the top tangent line of the separator and the bottom of the mist elimination pad shall be a minimum of 1 ft 3 in (38.1 cm) (Fig. 26).
5. Vessel diameters 3 ft 0 in (0.91 m) and larger shall be specified in increments of 6 in (15.2 cm). Diameters of shell plate vessels shall be specified as inside diameters. Vessel lengths shall be specified in 3-in (7.6-cm) increments.
6. Separators of 30-in (76.2-cm) diameter and smaller shall be specified as fabricated from pipe. Diameter dimensions shall represent pipe outside diameters. Top heads shall be specified as full-diameter flanges, with blind flange covers (Fig. 26). Bottom heads shall be standard heads or pipe caps.
7. Inlets shall have an internal arrangement to divert flow downward. Vessels 3 ft 0 in (0.91 m) and larger shall have a hood, attached to the shell, covering the inlet nozzle (Fig. 26).
8. Outlets shall have antivortex baffles.
9. Mist elimination pads shall be specified as 4-in (10.2-cm) thick, nominal 9-lb/ft^3 (144.1-kg/m^3) density and stainless steel. Spiral-wound pads are not acceptable.

The method given here is valid for vertical separators used in process, chemical, petrochemical, power, marine, and a variety of other plants. This procedure is the work of Arthur Gerunda, Vice President of Commercial Development, The Heyward-Robinson Co., as reported in *Chemical Engineering* magazine.

SIZING A HORIZONTAL LIQUID-VAPOR SEPARATOR

Design a horizontal vessel to separate 7000 gal/h (7.36 L/s) of liquid having a density of 60 lb/ft^3 (960.6 kg/m^3) from 1000 mol/h of vapor having a molecular weight of 28 if the holding time for the liquid is to be 8 min when the operating temperature is 100°F (37.8°C), the operating pressure is 300 lb/in^2 (gage) (2068.2 kPa), and $Z = 1.0$.

Calculation Procedure

1. Find the volumetric flow rate V. Using the gas law as in the previous calculation procedure with $n = 1000/3600$ mol/s and $A = 1.0$, we get $V = (1000/3600)(10.73)(560)(1.0) = 5.3$ ft^3/s (15,004 m^3/s).

2. Compute the density of the vapor d_v. As in the previous calculation procedure, d_v = (mol/h) (molecular weight)/(V) (3600) = 1.47 lb/ft^3 (23.5 kg/m^3).

3. Determine the terminal vapor velocity v_t. Use the relation $v_t = 0.227[(d_l - d_v)/d_v]^{0.5}$, where d_l = density of liquid. Or $v_t = 0.227[(60 - 1.47)/1.47]^{0.5} = 1.43$ ft/s (0.44 m/s). The constant 0.227 is obtained in the same way as described in the previous calculation procedure.

4. Decide what sets the size of the separator. Either the rate of liquid separation from the vapor or the liquid holding time will set the size of the separator. *By liquid separation,* $D = [V/3(\pi/4)(0.15)(V_t)]^{0.5}$ where D = separator diameter, ft. Or, $D = [5.3/(4\pi/4)(0.15 \times 1.43)]^{0.5} = 2.8$ ft (0.85 m).

By holding time, $D = [t_h V_i/3(\pi/4) f]^{0.333}$, where t_h = holding time, min; V_i = volumetric flow rate of the liquid, ft^3/min; f = fraction of the separator area occupied by the liquid. Assuming an L/D ratio, i.e., separator length/diameter, of 4, and with 7000 gal/h = 15.6 ft^3/min (0.01 m^3/s), make a first approximation with an assumed liquid-space area f of 0.70: $D = [8(15.6)/4(\pi/4)(0.70)]^{0.333} = 3.84$ ft (1.17 m).

Since the larger diameter is set by the holding time, this is the determining factor in the sizing of the separator.

Next, examine a 4-ft (1.22-m) diameter vessel with $f = 0.70$. With $L/D = 4$, $L = 4D = 4(4) = 16$ ft (4.9 m). With the area of the vapor space = 30 percent (that is, 1.00 – 0.70), the fractional height of the vapor space $f_{hv} = 0.342$, from the geometry of the tank. The height of the vapor space then is 0.342 (46 in) = 15.7 in (39.9 cm), where the 46 in (116.8 cm) is the approximate actual internal diameter of the vessel. Make the vapor-space height 18 in (45.7 cm). This gives a vapor-space area fraction f_{av} of 0.36 and a liquid-space area fraction f_{al} of 0.64.

The holding time t_h now is $t_h = [(\pi/4)(4^2)(0.64)(16)]/15.6 = 8.25$ min. Hence, the final separator size is as follows: diameter = 4 ft (1.22 m), length = 16 ft (4.88 m), and liquid height = 2.5 ft (0.76 m). As with a vertical separator, when a horizontal separator is equipped with a mist elimination pad, the allowable vapor velocity v_a can be taken to be the same as the terminal velocity v_t in the vessel diameter calculations. Figure 27 shows some typical arrangements of mist eliminators in horizontal liquid-vapor separators.

Related Calculations The chief concern in designing a horizontal liquid separator is to have the vapor velocity sufficiently low to give the liquid particles just enough time to settle out before the vapor leaves the vessel. Figure 28 shows the approximate traverse of a liquid particle for which

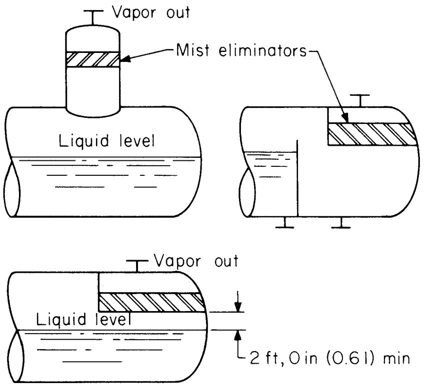


FIGURE 27 Mist eliminators in horizontal separators. (*Chemical Engineering.*)

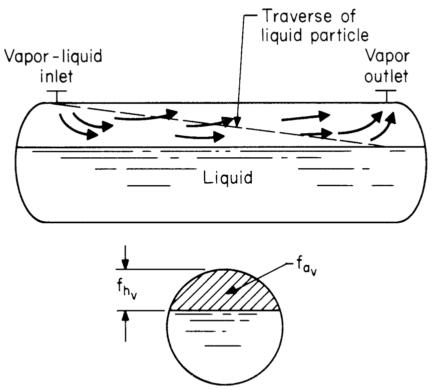


FIGURE 28 Traverse of liquid particle. (*Chemical Engineering.*)

the minimum time has been allowed for its disentrainment from the vapor. Indicated in the cross section are the fraction of area f_{av} and height f_{hv} taken up by the vapor space.

As with a vertical separator, empirical findings have shown that for safe design the allowable vapor velocity v_a in a horizontal separator should be no greater than 15 percent of the calculated terminal velocity v_t. Another restriction found necessary is that f_{av} be no less than 15 percent of the cross-sectional area.

For horizontal separators *L*/*D* ratios are dictated by economics and plot restrictions. As a general guide, the following provides economic designs:

Operating pressure		
lb/in^2 (gage)	kPa	*L*/*D* ratio
0–250	0–1723.5	3.0
251–500	1730.4–3447.0	4.0
501 and higher	3423.9 and higher	5.0

For a first trial size, set the liquid level at the centerline of the separator, so that $f_h = f_a = 0.5$. Now, $D = [V/(3)(\pi/4)(0.15 \times v_t)]^{0.5}$, or $D = [V/(0.35 \times v_t)]^{0.5}$.

These equations provide a good starting point for trial-and-error calculations to determine the size of a liquid-vapor horizontal separator operating at less than 251 lb/in^2 (gage) (1730.4 kPa). Note that the terminal vapor velocity v_t has, in effect, been replaced by the allowable vapor velocity v_a, because v_t is multiplied by the safety design factor, 0.15.

The following specifications are generally standard in the design of horizontal separators: (1) The maximum liquid level shall provide a minimum vapor space height of 15 in (38.1 cm) but not be below the centerline of the separator. (2) The volume of dished heads is not taken into account in vessel sizing calculations. (3) Inlet and outlet nozzles shall be located as closely as practical to the vessel tangent lines. (4) Liquid outlets shall have antivortex baffles.

When the size of a horizontal separator is set by the holdup time for the liquid, the diameter of the vessel must be determined by trial-and-error calculations. If f_{al} = the fraction of area occupied by the liquid, the holdup time t_h is given by $t_h = [(\pi/4)D^2 f_{al} L]V_t$. Here, L = vessel length and V_t = volumetric flow rate of the liquid. When the operating pressure is below 251 lb/in^2 (gage) (1730.4 kPa), $L/D = 3.0$. Solving for D yields $D = [t_h V_t / 3(\pi/4) f_{al}]^{0.333}$.

This procedure is the work of Arthur Gerunda, Vice-President, Commercial Development, Heyward-Robinson Co., as reported in *Chemical Engineering* magazine.

SIZING RUPTURE DISKS FOR GASES AND LIQUIDS

What diameter rupture disk is required to relieve 50,000 lb/h (6.3 kg/s) of hydrogen to the atmosphere from a pressure of 80 lb/in^2 (gage) (551.5 kPa)? Determine the diameter of a rupture disk required to relieve 100 gal/min (6.3 L/s) of a liquid having a specific gravity of 0.9 from 200 lb/in^2 (gage) (1378.8 kPa) to atmosphere.

Calculation Procedure

1. Determine the rupture disk diameter for the gas. For a gas, use the relation $d = (W/146P)^{0.5}(1/Mw)^{0.25}$, where d = minimum rupture-disk diameter, in; W = relieving capacity, lb/h; P = relieving pressure, lb/in^2 (abs); Mw = molecular weight of gas being relieved. By substituting, $d = [50{,}000/146(94.7)]^{0.5}(1/2)^{0.25} = 1.60$ in (4.1 cm).

2. Find the rupture-disk diameter for the liquid. Use the relation $d = 0.236(Q)^{0.5}(Sp)^{0.25}/P^{0.25}$, where the symbols are the same as in step 1 except that Q = relieving capacity, gal/min; Sp = liquid specific gravity. So $d = 0.236(100)^{0.5}(0.0)^{0.25}/(214.7)^{0.25} = 0.60$ in (1.52 cm).

Related Calculations Rupture disks are used in a variety of applications—process, chemical, power, petrochemical, and marine plants. These disks protect pressure vessels from pressure surges and are used to separate safety and relief valves from process fluids of various types.

Pressure-vessel codes give precise rules for installing rupture disks. Most manufacturers will guarantee rupture disks they size according to the capacities and operating conditions set forth in a purchase requisition or specification.

Designers, however, often must know the needed size of a rupture disk long before bids are received from a manufacturer so the designer can specify vessel nozzles, plan piping, etc.

The equations given in this procedure are based on standard disk sizing computations. They provide a quick way of making a preliminary estimate of rupture-disk diameter for any gas or liquid whose properties are known. The procedure is the work of V. Ganapathy, Bharat Heavy Electricals Ltd., as reported in *Chemical Engineering* magazine.

TIME NEEDED TO EMPTY A STORAGE VESSEL WITHOUT DISHED ENDS

How long will it take to empty a 10-ft (3-m) diameter spherical tank filled to a height of 8 ft (2.4 m) with ethanol, a newtonian fluid, if the drain is a short 2-in (5.1-cm) diameter tube of double extra-strong pipe?

Calculation Procedure

1. Determine the discharge coefficient for the drain. Figure 29 shows that the discharge coefficient is $C_d = 0.80$ for a short, flush-mounted tube.

2. Compute the discharge time. Substitute the appropriate values in the equation in Fig. 29 for spherical storage tanks. Or, $t = (2)^{0.5}(\pi)(8)^{1.5}[10 - (0.6 \times 8)]/[3(0.8)(1.774/144)(32.2)^{0.5}] = 3116$ s, or 51.9 min.

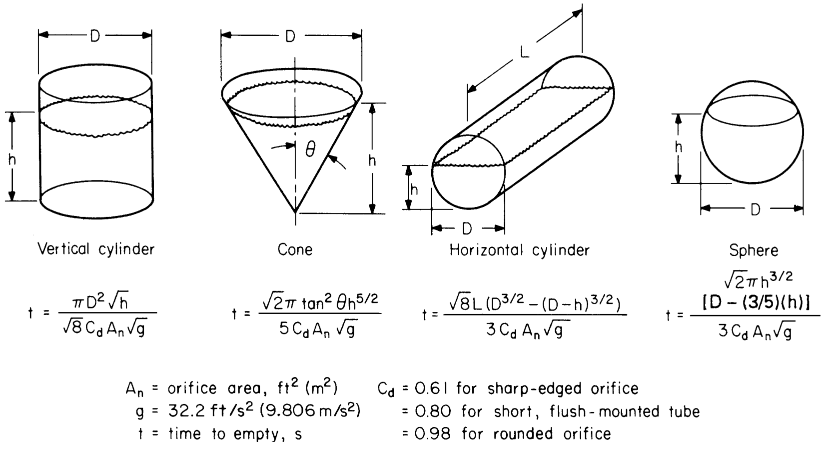


FIGURE 29 Time to empty tanks. (*Chemical Engineering.*)

Related Calculations Figure 29 gives the equations for computing the emptying time for four common tank geometrics. The discharge coefficient C_d is constant for newtonian fluids in turbulent flow, but the coefficient depends on the shape of the orifice. Water flowing through sharp-edged orifices of 0.25-in (0.64-cm) diameter, or larger, is always turbulent. Thus, the assumption of a constant C_d is valid for most practical applications. Figure 29 lists accepted C_d values.

The relations given here are valid for storage tanks used in a variety of applications—chemical and petrochemical plants, power plants, waterworks, ships and boats, aircraft, etc. This procedure is the work of Thomas C. Foster, as reported in *Chemical Engineering* magazine.

COST ESTIMATION OF HEAT EXCHANGERS AND STORAGE TANKS VIA CORRELATIONS

Using correlations, estimate the cost of a fixed-head, carbon-steel heat exchanger rated for 150 lb/in^2 (gage) (1034 kPa) having a total heat-transfer area of 1500 ft^2 (139.4 m^2). Using the same approach, estimate the cost of a cone-roof storage tank made of carbon steel having a total capacity of 677,000 gal (2,562,445 L). Show how to update the costs from the base year (cost index = 200.8) to a year in which the cost index is 265.

Calculation Procedure

1. Compute the base cost of the head exchanger. Using Table 2, substitute the area A in the relation $C_B - \exp\,[8.551 - 0.30863 \ln A + 0.06811\,(\ln A)^2]$. Or, $C_B = \exp\,[8.551 - 0.30863 \ln 1500 + 0.06811\,(\ln 1500)^2] = \$20{,}670$.

2. Determine the exchanger-type cost factor for the heat exchanger. Again, by using Table 2 for a fixed-head exchanger, $F_D = \exp\,(-1.1156 + 0.090606 \ln A)$, where F_D = exchanger-type cost factor. Substituting yields $F_D = \exp\,(-1.1156 + 0.090606 \ln 1500) = 0.6357$.

3. Find the design-pressure cost factor for the exchanger. From Table 2, the design-pressure cost factor for a pressure in the 100 to 300-lb/in^2 (gage) range (700 to 2100-kPa range) is $F_P = 0.7771 + 0.04981 \ln A$. Substituting, we find $F_P = 0.7771 + 0.04981 \ln 1500 = 1.1414$.

TABLE 2 Correlations for Costs of Heat Exchangers*

USCS units	SI units
Base cost for carbon-steel, floating-head, 100 lb/in^2 (gage) exchanger:	Base cost for carbon-steel, floating-head, 700-kN/m^2 exchanger:
$C_B = \exp\,[8.551 - 0.30863 \ln A + 0.06811\,(\ln A)^2]$	$C_B = \exp\,[8.202 + 0.01506 \ln A + 0.06811\,(\ln A)^2]$
Exchanger-type cost factor:	Exchanger-type cost factor:
Fixed-head: $F_D = \exp\,(-1.1156 + 0.0906 \ln A)$	Fixed-head: $F_D = \exp\,(-0.9003 + 0.0906 \ln A)$
Kettle reboiler: $F_D = 1.35$	Kettle reboiler: $F_D = 1.35$
U-tube: $F_D = \exp\,(-0.9816 + 0.0830 \ln A)$	U-tube: $F_D = \exp\,(-0.7844 + 0.0830 \ln A)$
Design-pressure cost factor:	Design-pressure cost factor:
100 to 300 lb/in^2 (gage): $F_P = 0.7771 + 0.04981 \ln A$	700–2100 kN/m^2: $F_P = 0.8955 + 0.04981 \ln A$
300 to 600 lb/in^2 (gage): $F_P = 1.0305 + 0.07140 \ln A$	2100–4200 kN/m^2: $F_P = 1.2002 + 0.07140 \ln A$
600 to 900 lb/in^2 (gage): $F_P = 1.1400 + 0.12088 \ln A$	4200–6200 kN/m^2: $F_P = 1.4272 + 0.12088 \ln A$
A in ft^2; lower limit: 150 ft^2, upper limit: 12,000 ft^2	A in m^2; lower limit: 14 m^2, upper limit: 1100 m^2.

**Chemical Engineering.*

TABLE 3 Material-of-Construction Cost Factors for Heat Exchangers*

Material	USCS units, A in ft^2 $F_M = g_1 + g_2 \ln A$		SI units, A in m^2 $F_M = g_1 + g_2 \ln A$	
	g_1	g_2	g_1	g_2
Stainless steel 316	0.8608	0.23296	1.4144	0.23296
Stainless steel 304	0.8193	0.15984	1.1991	0.15984
Stainless steel 347	0.6116	0.22186	1.1388	0.22186
Nickel 200	1.5092	0.60859	2.9553	0.60859
Monel 400	1.2989	0.43377	2.3296	0.43377
Inconel 600	1.2040	0.50764	2.4103	0.50764
Incoloy 825	1.1854	0.49706	2.3665	0.49706
Titanium	1.5420	0.42913	2.5617	0.42913
Hastelloy	0.1549	1.51774	3.7614	1.51774

**Chemical Engineering.*

4. Find the materials-of-construction cost factor. The materials-of-construction cost factor, F_M, for carbon steel is unity, or 1.0. Factors for other materials of construction are shown in Table 3.

5. Compute the heat-exchanger cost. Use the relation $C_E = C_B F_D F_P F_M$, where C_E = exchanger cost. Or, C_E = (\$20,670)(0.6357)(1.1414)(1.0) = \$15,000.

6. Update the heat-exchanger cost. The base-year cost index—for 1976, the year on which the above costs are based—is 200.8. For the year in which the cost estimate is being made, the cost index is 265 (obtained from any of the standard, widely accepted cost indices). Updating the heat-exchanger cost reveals C_{EU} = \$15,000(265/200.8) = \$19,796. In this relation, the updated cost is $C_{EU} = C_E$ (current-year equipment cost index/base-year cost index).

7. Compute the storage-tank cost. Using Table 4, apply the relation $C_B = \exp\,[11.362 - 0.6104 \ln V + 0.045355 (\ln V)^2]$, where C_B = base cost of field-erected tank in carbon steel; V = tank volume, gal. Substituting gives $C_B = \exp\,[11.362 - 0.6104 \ln 677.000 + 0.045355 (\ln 677{,}000)^2]$ = \$84,300. Updating the cost, as before, we find C_{BU} = \$84,300(265/200.8) = \$111,252. Table 5 shows materials of construction cost factors for storage tanks.

Related Calculations The approach given here correlates the cost of shell-and-tube heat exchangers and heat-transfer area. This contrasts with cost estimation procedures that take into

TABLE 4 Correlations for Costs of Storage Tanks*

USCS units	SI units
Base cost for carbon-steel, shop-fabricated tanks:	Base cost for carbon-steel, shop-fabricated tanks:
$C_B = \exp\,[2.331 + 1.3673 \ln V - 0.063088 (\ln V)^2]$	$C_B = \exp\,[7.994 + 0.6637 \ln V - 0.063088 (\ln V)^2]$
V in gallons; lower limit: 1300 gal, upper limit: 21,000 gal	V in m^3; lower limit: 5 m^3, upper limit: 80 m^3
Base cost for carbon-steel, field-erected tanks:	Base cost for carbon-steel, field-erected tanks:
$C_B = \exp\,[11.362 - 0.6104 \ln V + 0.045355 (\ln V)^2]$	$C_B = \exp\,[9.369 - 0.1045 \ln V + 0.045355 (\ln V)^2]$
V in gallons; lower limit: 21,000 gal, upper limit: 11,000,000 gal	V in m^3; lower limit: 80 m^3, upper limit: 45,000 m^3

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TABLE 5 Material-of-Construction Cost Factors for Storage Tanks*

Material of construction	Cost factor F_M
Stainless steel 316	2.7
Stainless steel 304	2.4
Stainless steel 347	3.0
Nickel	3.5
Monel	3.3
Inconel	3.8
Zirconium	11.0
Titanium	11.0
Brick-and-rubber- or brick-and-polyester-lined steel	2.75
Rubber- or lead-lined steel	1.9
Polyester, fiberglass-reinforced	0.32
Aluminum	2.7
Copper	2.3
Concrete	0.55

**Chemical Engineering*.

account shell diameter, number and length of the tubes, types of heads, and other construction details. The accuracy of the simple correlation of cost versus area is sufficient for preliminary cost estimates.

Correlations for base cost are given in both USCS and SI units in the accompanying tables. The base-cost basis for the equipment is given in each table. While heat-exchanger costs are based on area, storage-tank costs are based on the total tank volume. The tank volume is calculated (for the base cost) from residence time, a fixed overcapacity factor of 20 percent, and volumetric flow rate.

Omitted from the cost estimation procedure given here are the number and sizes of nozzles and manholes and other design details. These details cause variations in cost that are usually within the accuracy of preliminary estimates.

Data on the cost of shell-and-tube heat exchangers in a wide range of heat-transfer areas and design pressures were used in developing the correlations for 10 different materials of construction and three design types. PDQ$, Inc., supplied these and the cost data for cylindrical carbon-steel tanks having cone roofs and flat bottoms in a wide range of volumes. The cost of field-erected tanks includes the cost of platforms and ladders, but not of foundations and other installation materials (piping, electric instrumentation, etc.). The cost of the shop-fabricated tanks does not include any of the installation materials.

This procedure is the work of Armando B. Corripio, Louisiana State University, and Katherine S. Chrien and Lawrence B. Evans, both of the Massachusetts Institute of Technology, as reported in *Chemical Engineering* magazine.

ESTIMATING CENTRIFUGAL-PUMP AND ELECTRIC-MOTOR COST BY USING CORRELATIONS

Determine the cost of a ductile-steel pump to deliver 1430 gal/min (90.2 L/s) at a differential head of 77 ft·lbf/lb (230.2 J/kg). A horizontally split case one-stage pump running at 3550 r/min is specified. The specific gravity of the fluid being pumped is 0.952

TABLE 6 Correlations for Costs of Centrifugal Pumps*

USCS units

Base cost for one-stage, 3550 r/min, VSC cast-iron pump:

$$C_B = \exp\,[8.3949 - 0.6019 \ln S + 0.0519(\ln S)^2]$$

Here, $S = Q\sqrt{H}$, with Q in gal/min and H in ft·lbf/lb (ft of head).

Cost factor for pump type:

$$F_T = \exp\,[b_1 + b_2 \ln S + b_3 (\ln S)^2]$$

Type	b_1	b_2	b_3
One-stage, 1750-r/min, VSC	5.1029	–1.2217	0.0771
One-stage, 3550-r/min, HSC	0.0632	0.2744	–0.0253
One-stage, 1750-r/min, HSC	2.0290	–0.2371	0.0102
Two-stage, 3550-r/min, HSC	13.7321	–2.8304	0.1542
Multistage, 3550-r/min, HSC	9.8849	–1.6164	0.0834

SI units

Base cost for one-stage, 3550-r/min, VSC cast-iron pump:

$$C_B = \exp\,[7.2234 + 0.3451 \ln S + 0.0519(\ln S)^2]$$

Here, $S = Q\sqrt{H}$, with Q in m^3/s, and H in J/kg or m^2/s^2.

Cost factor for pump type:

$$F_T = \exp\,[b_1 + b_2 \ln S + b_3 (\ln S)^2]$$

Type	b_1	b_2	b_3
One-stage, 1750-r/min, VSC	0.3740	0.1851	0.0771
One-stage, 3550-r/min, HSC	0.4612	–0.1872	–0.0253
One-stage, 1750-r/min, HSC	0.7147	–0.0510	0.0102
Two-stage, 3550-r/min, HSC	0.7445	–0.0167	0.1542
Multistage, 3550-r/min, HSC	2.0798	–0.0946	0.0834

**Chemical Engineering* and Richardson Engineering Services, Inc.

Calculation Procedure

1. Determine the size parameter S. The size parameter is defined as $S = QH^{0.5}$, where Q = design capacity of the pump, gal/min (m^3/s), and H is the required head for the pump, ft·lb/lb or J/kg. Substituting, we get $S = 1430(77)^{0.5} = 12{,}550$, closely.

2. Find the pump base cost C_B. Use the base cost relation from Table 6, or $C_B = \exp\,[8.3949 - 0.6019 \ln S + 0.0519 (\ln S)^2]$. So $C_B = \exp\,[8.3949 - 0.6019 \ln 12{,}550 + 0.0519 (\ln 12{,}550)^2] =$ \$1536.

3. Compute the pump design-type factor F_T. Table 6 shows that the design-type factor for a one-stage, 3550-r/min HSC pump is found from $F_T = \exp\,[b_1 + b_2 \ln S + f_3 (\ln S)^2]$. Substituting the values given in the table, we see $F_T = \exp\,[0.0632 + 0.2744 \ln 12{,}550 - 0.0253 (\ln 12{,}550)^2] = 1.491$.

4. Find the materials-of-construction factor F_M. From Table 7 for ductile iron, $F_M = 1.15$.

5. Compute the pump cost C_P with base plate and coupling. Use the relation $C_P = C_B F_T F_M$, where the symbols are as given above. Thus, $C_P =$ (\$1536)(1.491)(1.15) = \$2630.

6. Determine the required horsepower for the motor. Use the relation $P_B = pQH/33{,}000N_P$, where P_B = bhp input to pump; p = fluid density, lb/gal; N_P = pump efficiency, percent; other symbols as given earlier. In this method of cost estimating, $N_Q = -0.316 + 0.24015 (\ln Q) - 0.01199 (\ln Q)^2$.

Find the fluid density from p = specific gravity (8.33 lb/gal) = 0.952(8.33) = 7.93 lb/gal (0.94 kg/L).

TABLE 7 Cost Factors for Material of Construction*

(Source: Monsanto Co.'s FLOWTRAN pump-costing subprogram.)

Material	Cost factor F_M
Cast steel	1.35
304 or 316 fittings	1.15
Stainless steel, 304 or 316	2.00
Cast Gould's alloy no.20	2.00
Nickel	3.50
Monel	3.30
ISO B	4.95
ISO C	4.60
Titanium	9.70
Hastelloy C	2.95
Ductile iron	1.15
Bronze	1.90

**Chemical Engineering.*

TABLE 8 Correlation for Cost of Electric Motors*

Cost of 60-Hz standard-voltage motor and insulation, discounted:
$C_M = \exp [a_1 + a_2 \ln P + a_3(\ln P)^2]$
P is the nominal size in horsepower

	Coefficients				
	a_1	a_2	a_3	hp limits	kW limits
Open, drip-proof:					
3600 r/min	4.8314	0.09666	0.10960	1–7.5	0.75–5.6
	4.1514	0.53470	0.05252	7.5–250	5.6–186.5
	4.2432	1.03251	–0.03595	250–700	186.5–522.2
1800 r/min	4.7075	–0.01511	0.22888	1–7.5	0.75–5.6
	4.5212	0.47242	0.04820	7.5–250	5.6–186.5
	7.4044	–0.06464	0.05448	250–600	186.5–447.6
1200 r/min	4.9298	0.30118	0.12630	1–7.5	0.75–5.6
	5.0999	0.35861	0.06052	7.5–250	5.6–186.5
	4.6163	0.88531	–0.02188	250–500	186.5–373.0
Totally enclosed, fan-cooled:					
3600 r/min	5.1058	0.03316	0.15374	1–7.5	0.75–5.6
	3.8544	0.83311	0.02399	7.5–250	5.6–186.5
	5.3182	1.08470	–0.05695	250–400	186.5–298.4
1800 r/min	4.9687	–0.00930	0.22616	7.5–250	5.6–186.5
	4.5347	0.57065	0.04609		
1200 r/min	5.1532	0.28931	0.14357	1–7.5	0.75–5.6
	5.3858	0.31004	0.07406	7.5–350	5.6–261.1
Explosion-proof:					
3600 r/min	5.3934	–0.00333	0.15475	1–7.5	0.75–5.6
	4.4442	0.60820	0.05202	7.5–200	5.6–149.2
1800 r/min	5.2851	0.00048	0.19949	1–7.5	0.75–5.6
	4.8178	0.51086	0.05293	7.5–250	5.6–186.5
1200 r/min	5.4166	0.31216	0.10573	1–7.5	0.75–5.6
	5.5655	0.31284	0.07212	7.5–200	5.6–149.2

**Chemical Engineering.*

TABLE 9 Flow, Head, and Power Limits for Centrifugal Pumps*

	Flow, gal/min (m^3/s)		Head, ft·lbf/lb (J/kg)		Motor hp, upper limit	Motor kW
	Lower limit	Upper limit	Lower limit	Upper limit		
One-stage, 3550 r/min, VSC	50 (0.00315)	900 (0.568)	50 (150)	400 (1200)	75	55.95
One-stage, 1750 r/min, VSC	50 (0.00315)	3500 (0.2208)	50 (150)	200 (600)	200	149.2
One-stage, 3550 r/min, HSC	100 (0.00631)	1500 (0.0946)	100 (300)	450 (1350)	150	111.9
One-stage, 1750 r/min, HSC	250 (0.01577)	5000 (0.3155)	50 (150)	500 (1500)	250	186.5
Two-stage, 3550 r/min, HSC	50 (0.00315)	1100 (0.0694)	300 (900)	1100 (3300)	250	186.5
Multistage, 3550 r/min, HSC	100 (0.00631)	1500 (0.0946)	650 (2000)	3200 (9600)	1450	1081.7

**Chemical Engineering.*

The pump efficiency, from the previous relation, is $N_P = -0.316 + 0.24015 (\ln 1430) - 0.01199 (\ln 1430)^2 = 0.796$.

Substituting in the power relation yields $P_B = 7.93 (1430)(77)/33{,}000(0.796) = 33.2$ hp (24.8 kW). A 40-hp (29.8·kW) motor is required for this pump.

TABLE 10 Correlations for Cost of Drive Coupling*

Cost of belt-drive coupling:

$$C_C = \exp (3.689 + 0.8917 \ln P)$$

Cost of chain-drive coupling:

$$C_C = \exp (5.329 + 0.5048 \ln P)$$

Cost of variable-speed-drive coupling:

$$C_C = 1/[1.562 \times 10^{-4} + (7.877 \times 10^{-4}/P)]$$

Upper limit = 75 hp; S = nominal motor size in hp

**Chemical Engineering* and U.S. Bureau of Mines.

7. Compute the cost of the electric motor. Use the appropriate correlation from Table 8. Assume a 3600-r/min totally enclosed fan-cooled motor is needed. Then the motor cost $C_M = \exp [3.8544 + 0.8331 (\ln P_B) + 0.02399 (\ln P_B)^2] = \exp [3.8544 + 0.8331 (\ln 40) + 0.02399 (\ln 40)^2] = \$1410$.

8. Determine the total cost of the pump and motor. Find the sum of $C_P + C_M$. Or, $C_P + C_M = \$2630 + \$1410 = \$4040$.

9. Compute the pump power consumption. Use the relation $U_M + 0.80 = 0.0319 (\ln P_B) - 0.00182 (\ln P_B)^2$ to find the efficiency of the motor. By substituting, $N_M = 0.80 + 0.0319 (\ln 33.2) - 0.00182 \ln (33.2)^2 = 0.889$. Then the pump power consumption $P_C = P_B/N_M = 33.2/0.889 = 37.3$ hp (27.8 kW).

Related Calculations This procedure can be used for centrifugal pumps and electric motors in a variety of industries and applications provided the pump and motor are of the type listed in the tables. Typical industries and applications include chemical, petroleum, petrochemical, power, marine, air-conditioning, heating, and food processing.

Data on the cost of centrifugal pumps and electric motors were taken from Vol. 4 of the data book by Richardson Engineering Services (Solana Beach, CA), *Process Plant Construction Estimating Standards*. The material-of-construction cost factors for pumps were taken from Monsanto Co.'s FLOWTRAN pump-costing subprograms.

Although the cost of a pump includes the cost of the driver coupling, cost correlations for belt-, chain-, and variable-speed drive couplings were obtained from the U.S. Bureau of Mines equipment-costing program. These correlations were escalated from their original data of 1967 to the first quarter of 1979 by using the chemical engineering pumps and compressors index ratio of 270/11.2 = 2.43. All other cost data were for the first quarter of 1979, when the pumps and compressors index was 270 and the electrical equipment index was 175.5. To update the costs to the year in which an estimate is being made, simply apply the current index, as detailed in the preceding procedure.

Table 9 gives the flow, head, and power limits for the centrifugal pumps considered in this procedure. Table 10 shows the correlations for the cost of drive couplings for the pumps.

This procedure is the work of Armando B. Corripio, Louisiana State University; Katherine S. Chrien of J.S. Dweck, Consultant, Inc.; and Lawrence B. Evans, Massachusetts Institute of Technology, as reported in *Chemical Engineering* magazine.

DETERMINING THE FRICTION FACTOR FOR FLOW OF BINGHAM PLASTICS

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A coal slurry is being pumped through a 0.4413-m (18-in) diameter schedule 20 pipeline at a flow rate of 400 m^3/h. The slurry behaves as a Bingham plastic, with the following properties (at the relevant temperature): $\tau_0 = 2$ N/m^2 (0.0418 lbf/ft^2); $\mu_\infty = 0.03$ Pa·s (30 cP); $\rho = 1500$ kg/m^3 (93.6 lbm/ft^3). What is the Fanning friction factor for this system?

Calculation Procedure

1. Determine the Bingham Reynolds number and the Hedstrom number. Engineers today often must size pipe or estimate pressure drops for fluids that are nonnewtonian in nature—coal suspensions, latex paint, or printer's ink, for example. This procedure shows how to find the friction factors needed in such calculations for the many fluids that can be described by the Bingham-plastic flow mode. The method is convenient to use and applies to all regimes of pipe flow.

A Bingham plastic is a fluid that exhibits a yield stress; that is, the fluid at rest will not flow unless some minimum stress τ_0 is applied. Newtonian fluids, in contrast, exhibit no yield stress, as Fig. 30 shows.

The Bingham-plastic flow model can be expressed in terms of either shear stress τ versus shear rate $\dot{\gamma}$, as in Fig. 30, or apparent viscosity η versus shear rate:

$$\tau = \tau_0 + \mu_\infty \dot{\gamma} \tag{1}$$

$$\eta = \frac{\tau}{\dot{\gamma}} = \frac{\tau_0}{\dot{\gamma}} + \mu_\infty \tag{2}$$

Equation 2 means that the apparent viscosity of a Bingham plastic depends on the shear rate. The parameter μ_∞ is sometimes called the coefficient of rigidity, but it is really a limiting viscosity. As Eq. 2 shows, apparent viscosity approaches μ_∞ as shear rate increases indefinitely. Thus, the Bingham plastic behaves almost like a newtonian at sufficiently high shear rates, exhibiting a viscosity of μ_∞ at such conditions. Table 11 shows values of τ_0 and μ_∞ for several actual fluids.

For any incompressible fluid flowing through a pipe, the friction loss per unit mass F can be expressed in terms of a Fanning friction factor f:

$$F = \frac{2fLv^2}{D} \tag{3}$$

where L is the lengths of the pipe section, D is its diameter, and v is the fluid velocity.

An exact description of friction loss for Bingham plastics in fully developed laminar pipe flow was first published by Buckingham [1]. His expression can be rewritten in dimensionless form

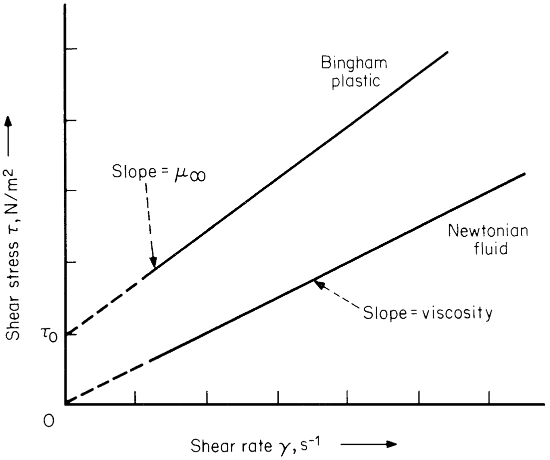


FIGURE 30 Bingham plastics exhibit a yield stress. (*Chemical Engineering.*)

as follows:

$$f_L = \frac{16}{N_{\rm Re}}\left(1 + \frac{N_{\rm He}}{6N_{\rm Re}} - \frac{N_{\rm He}^4}{3f_L^3 N_{\rm Re}^7}\right) \tag{4}$$

where $N_{\rm Re}$ is the Bingham Reynolds number (Dvp/μ_∞) and $N_{\rm He}$ is the Hedstrom number $(D^2 p\tau_0/\mu^2)$. Equation 4 is implicit in f_L, the laminar friction factor, but can be readily solved either by Newton's method or by iteration. Since the last term in Eq. 4 is normally small, the value of f obtained by omitting this term is usually a good starting point for iterative solution.

For this pipeline

$$N_{\rm Re} = \frac{4Q\rho}{\pi D\mu_\infty} = \frac{4(400)(1/3)(600)(1500)}{\pi(0.4413)(0.03)} = 16{,}030$$

$$N_{\rm He} = \frac{D^2\rho\tau_0}{\mu_\infty^2} = \frac{(0.4413)^2(1500)(2)}{(0.03)^2} = 649{,}200$$

2. Find the friction factor f_L for the laminar-flow regime. Substituting the values for $N_{\rm Re}$ and $N_{\rm He}$ into Eq. 4, we find $f_L = 0.007138$.

TABLE 11 Values of τ_0 and μ_∞

Fluid	τ_0, N/m^2	μ_∞, Pa·s	Ref.
Blood (45% hematocrit)	0.005	0.0028	[6]
Printing-ink pigment in varnish (10% by wt.)	0.4	0.25	[7]
Coal suspension in methanol (35% by vol.)	1.6	0.04	[8]
Finely divided galena in water (37% by vol.)	4.0	0.057	[9]
Molten chocolate (100°F)	20	2.0	[10]
Thorium oxide in water (50% by vol.)	300	0.403	[11]

3. Determine the friction factor f_T for the turbulent-flow regime. Equation 4 describes the laminar-flow sections. An empirical expression that fits the turbulent-flow regime is

$$f_T = 10^a N_{\mathrm{Re}}^{-0.193} \tag{5}$$

where

$$a = -1.378\,[1 + 0.146\ \exp(-2.9 \times 10^{-5} N_{\mathrm{He}})] \tag{6}$$

We now have friction-factor expressions for both laminar and turbulent flow. Equation 6 does not apply when N_{He} is less than 1000, but this is not a practical constraint for most Bingham plastics with a measurable yield stress.

When N_{He} is above 300,000, the exponential term in Eq. 6 is essentially zero. Thus, $a = -1.378$ here, and Eq. 5 becomes

$$\begin{aligned} f_T &= 10^{-1.378}(16{,}030)^{-0.193} \\ &= 0.006463 \end{aligned}$$

4. Find the friction factor f. Combine the f_L and f_T expressions to get a single friction factor valid for all flow regimes:

$$f = (f_L^m + f_T^m)\frac{1}{m} \tag{7}$$

where f_L and f_T are obtained from Eqs. 4 and 5, and the power m depends on the Bingham Reynolds number:

$$m = 1.7 + \frac{40{,}000}{N_{\mathrm{Re}}} \tag{8}$$

The values of f predicted by Eq. 7 coincide with Hanks's values in most places, and the general agreement is excellent. Relative roughness is not a parameter in any of the equations because the friction factor for nonnewtonian fluids, and particularly plastics, is not sensitive to pipe roughness.

Substituting yields $m = 1.7 + 40{,}000/16{,}030 = 4.20$, and $f = [(0.007138)^{4.20} + (0.006463)^{4.20}]^{1/4.20} = 0.00805$.

If m had been very large, the bracketed term above would have approached zero. Generally, when N_{Re} is below 4000, Eq. 8 should be solved by taking f equal to the greater of f_L and f_T.

Related Calculations This procedure is valid for a variety of fluids met in many different industrial and commercial applications. The procedure is the work of Ron Darby, Professor of Chemical Engineering, Texas A & M University, College of Engineering, and Jeff Melson, Undergraduate Fellow, Texas A & M, as reported in Chemical Engineering magazine. In their report they cite works by Hanks and Pratt [2], Hanks and Dadia [3], Churchill [4], and Churchill and Usagi [5] as important in the procedure described and presented here.

TIME NEEDED TO EMPTY A STORAGE VESSEL WITH DISHED ENDS

A tank with a 6-ft (1.8-m) diameter cylindrical section that is 16 ft (4.9 m) long has elliptical ends, each with a depth of 2 ft (0.7 m), and is half-full with ethanol, a newtonian fluid. How long will it take to empty the tank if it is set horizontally and fitted at the bottom with a drain consisting of a short tube of 2-in (5.1-cm) double extrastrong pipe? How long will it take to empty the tank if it is set vertically and fitted at the bottom with a drainpipe of 2-in (5.1-cm) double extrastrong pipe? The drain system extends 4 ft (1.2 m) below the dished bottom and has an equivalent length of 250 ft (76.2 m).

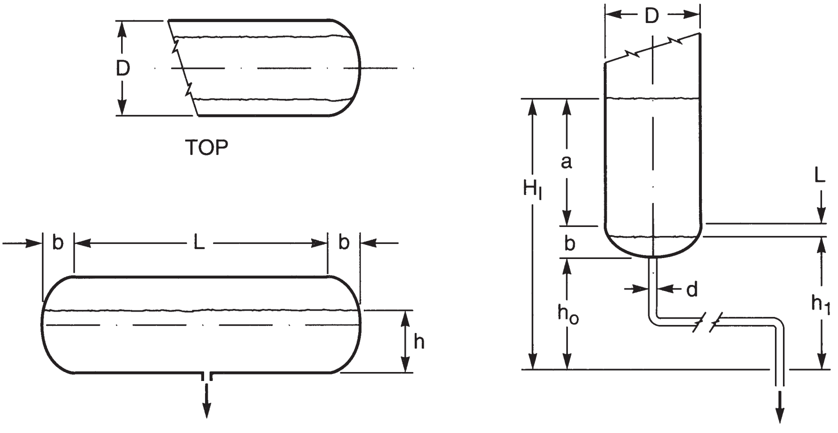
Calculation Procedure

1. Determine the discharge coefficient for, and orifice area of, the drain tube. Figure 29 shows the discharge coefficient is $C_d = 0.80$ for a short, flush-mounted tube. Baumeister, in *Mark's Standard Handbook for Mechanical Engineers,* indicates the internal section area of the tube is $A_n = 1.774$ in^2 (11.4 cm^2).

2. Compute the discharge time for the tank in a horizontal position. Substitute the appropriate values in the equation for t_p shown under the storage tanks in Fig. 31. Thus, $t_p = [(8)^{0.5}/[3(0.80)(1.774/144)(32.2)^{0.5}]\{16[(6)^{1.5} - (6 - 3)^{1.5}] + [2\pi(3)^{1.5}/6][6 - (3/5)(3)]\} = 2948$ s, or 49.1 min.

3. Determine the internal diameter and friction factor for the drainpipe. From Baumeister, *Mark's Standard Handbook for Mechanical Engineers,* the internal diameter of the pipe is $d = 1.503$ in (3.8 cm), or 0.125 ft (0.038 m) and the Moody friction factor is $f = 0.020$ for the equivalent length, $l = 250$ ft (76.2 m), of pipe.

4. Compute the initial and final height above the drainpipe outlet for the cylindrical section. Initial height of the liquid is $H_1 = a + b + h_o = [(16/2) + 2 + 4] = 14.0$ ft (4.3 m). Final height is $H_F = b + h_o = 2 + 4 = 6$ ft (1.8 m).



TOP

FRONT
Horizontal cylinder with dished ends

Vertical cylinder with dished end
and drainpipe system

$$t_c = \frac{D^2}{d^2}\{[(2/g)\,(1 + [fl/d])]^{1/2}\,(H_I{}^{1/2} - H_F{}^{1/2})\}$$

$$t_e = C\{[(2 \text{ X } H_2{}^2/5) - (4 \text{ X } B \text{ X } H_2/3) + 2E^2\}(H_2{}^{1/2}) - [(2 \text{ X } H_1{}^2/5) - (4 \text{ X } B \text{ X } H_1/3) + 2E^2](H_1{}^{1/2})\}$$

$$t_p = \frac{\sqrt{8}}{3C_dA_n\sqrt{g}}\,\{L[D^{3/2} - (D - h)^{3/2}] + \frac{bph^{3/2}}{D}[D - (3h/5)]\}$$

FIGURE 31 Time to drain tanks. (*a*) Top and (*b*) front view of horizontal cylinder with dished ends. (*c*) Vertical cylinder with dished-end and drainpipe system. (*Chemical Engineering*).

5. Compute the time required to drain the cylindrical section of the tank. Substitute the appropriate values in the equation for t_c shown under the storage tanks in Fig. 31. Hence, $t_c = [(6)^2/(0.125)^2]\{(2/32.2)[1 + (0.020 \times 250/0.125)]\}^{0.5}[(14)^{0.5} - (6)^{0.5}] = 4751$ s, or 79.2 min.

6. Compute the initial and final liquid height above the drainpipe outlet for the elliptically dished head. Initial height of the liquid is $H_1 = b + h_o = 2 + 4 = 6$ ft (1.8 m). Final height is $H_2 = h_o = 4$ ft (1.2 m).

7. Compute how long it will take to empty the dished bottom of the tank. In order to solve for t_e it is necessary to determine the following values: $B = h_o + b = 4 + 2 = 6$ ft (1.8 m); $E^2 = h_o^2 + 2bh_o = (4)^2 + 2(2)(4) = 32$ ft^2 (3.0 m^2); $C = [D/(db)]^2\{[1/(2g)][1+ (f_1/d)]\}^{0.5} = [6/(0.125 \times 2)]^2\{[1/(2 \times 32.2)][1 + ([0.02 \times 250\ ]/0.125)]\}^{0.5} = 459.6$, s/ft$^{5/2}$ (s/m$^{5/2}$).

Then, use the values for B, E^2, C, and other relevant dimensions to find t_e from the equation shown under the storage tanks in Fig. 31. Thus, $t_e = 459.6\ [(2 \times 4^2/5) - (4 \times 6 \times 4/3) + 2(32)](4)^{0.5} - [(2 \times 6^2/5) - (4 \times 6 \times 6/3) + 2(32)](6^{0.5}) = 1073$ s, or 17.9 min.

8. Compute the time it will take to drain the half-full vertical tank. Total time is $t_v + t_c + t_e = 4751 + 1073 = 5824$ s, or 96.1 min.

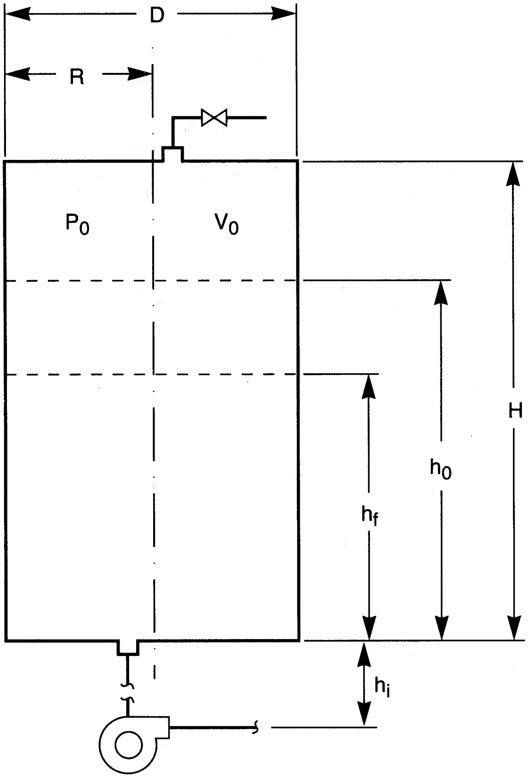
> **Related Calculations** Figure 31 shows the equation for computing the emptying time for a horizontal cylindrical tank with elliptically dished ends and equations for calculating the emptying time for a vertical cylindrical tank with an elliptically dished bottom end fitted with a drain system. The symbols A_n, g, t, and C_d are defined as in the previous problem for a storage vessel without dished ends, except that A_n is now the drainpipe internal area.
>
> The term associated with the second pair of brackets in the equation for t_p accounts for the dished ends of the horizontal tank. For hemispherical ends $b = D/2$ and for flat ends, $b = 0$.
>
> When seeking the time required to drain a portion of the cylindrical part of the vertical tank use the formula for t_c with the appropriate values for H_1 and H_F and other pertinent variables. To find the time it takes to drain a portion of the dished bottom of the vertical tank use the formula for t_e with given values of H_1 and H_2 and other applicable variables.
>
> The relations given here are valid for storage tanks used in a variety of applications—chemical and petrochemical plants, power plants, waterworks, ships and boats, aircraft, etc. The procedure for a horizontal cylindrical tank with dished ends is the work of Jude T. Sommerfeld, and the procedure for a vertical cylindrical tank with a dished bottom end is the work of Mahnoosh Shoael and Jude T. Sommerfeld, as reported in *Chemical Engineering* magazine.

CHECKING THE VACUUM RATING OF A STORAGE VESSEL

Check the vacuum rating of a cylindrical flat-ended process tank which is 12.75 ft (3.9 m) tall and 4 ft (1.22 m) in diameter. It contains fresh water at 190°F (87.8°C) and is located where $g = 32.0$ ft/s^2 (9.8 m/s^2) and the atmospheric pressure is 14.7 lb/in^2 (101.3 kPa). What is its maximum vacuum when the tank is gravity-drained? Find both the final tank vacuum and final height of liquid above the tank bottom when the tank is initially 75 percent full, first with gravity drain and then using a pumped drain, each discharging to the atmosphere. The pumped-drain piping system consists of double extra-strong 2-in (5.1-cm) pipe with an equivalent length of 75 ft (22.9 m) and a pump which can discharge 40 gal/min (151 L/min) with its suction centerline 2 ft (0.61 m) below the tank bottom and has a net positive suction head of 4.5 ft (1.4 m).

Calculation Procedure

1. Find the density of the fresh water. From a suitable source such as Baumeister, *Mark's Standard Handbook for Mechanical Engineers*, freshwater density $\rho = 60.33$ lb/ft^3 (966.7 kg/m^3) at the given conditions.



# rho	$P_s = 2.036Hd(g/[144g_c)$, in Hg|(kPa)
# rho	$h_{fg} = 144P_o(1 - \{V_o/[V_o + pR^2(h_o - h_{fg})]\}n)/(eg/g_c)$, ft|(m)
# pi gamma	$P_g = 29.92h_{fg}eg/(144P_og_c)$, in Hg|(kPa)
#	$h_{fg} = NPSH + F_1 + V_p - h_i - 144P_o\,\{V_o/[V_o + pR^2(h_o - h_{fp})]\}n/(eg/g_c)$, ft|(m)
#	$P_p = 2.036P_o\{1 - [V_o/(V_o + pR^2h_o)]n\}$, in Hg|(kPa)

FIGURE 32 Vacuum rating of tanks.

2. Compute the maximum vacuum rating with gravity drain. A shortcut method gives the maximum vacuum rating by use of the equation for P_s in Fig. 32 where H = overall vertical dimension of the tank; ρ = density of the fresh water; g = acceleration due to gravity at the tank's location; and g_c = 32.174 lb·ft/lb·s^2, a conversion factor. Substituting appropriate values gives P_s = 2.036(12.75)(60.33)(32.0/[144 × 32.174]) = 10.82 in Hg (36.64 kPa).

3. Compute the head space volume when the tank is 75 percent full. Airspace volume V_o = (1.00 – 0.75)(H × 0.7854 D^2) = 0.25(12.75 × 0.7854 × 4^2) = 40.1 ft^3 (1.14 m^3).

4. Compute the final height of liquid above the tank bottom created by gravity drain. By trial and error, the final height of liquid can be computed by solving the equation for h_{fg} shown in Fig. 32 where the ambient atmospheric pressure P_o = 14.7 lb/in^2 (101.3 kPa); tank radius R = 2 ft (0.61 m); initial fluid height above tank bottom $h_o = 0.75H$ = 0.75(12.75) = 9.56 ft (2.91 m); ratio of molar specific heats, C_p/C_v, is $\gamma = 1.4$ for diatomic gases. Assuming a reasonable initial value of h_{fg} on the right-hand side of the equation and substituting appropriate other values, too, gives $h_{fg} = 144(14.7)(1 - \{40.1/[40.1 + \pi(2^2)(9.56 - h_{fg})]\}1.4)/(60.33 \times 32.0/32.174) = 7.18$ ft (2.19 m), where the left-hand side value is used to repeat the interation process until the h_{fg} values on either side of the equal sign are in close agreement.

5. Compute the final tank vacuum under gravity drain. Use the equation for P_g shown in Fig. 32 to find the final tank vacuum. Thus, $P_g = 29.92 \times 7.18 \times 60.33 \times 32.0/(144 \times 14.7 \times 32.174) = 6.09$ in Hg (20.62 kPa).

6. Compute the water velocity in the pumped-drain system piping. Pump's flow rate $q = (40$ gal/min)/([7.48 gal/ft^3][60 s/min]) = 0.0891 ft^3/s (0.0025 m^3/s). The sectional area of the 2 in (5.1 cm) double extra-strong pipe is $a = 0.7854d^2 = 0.7854(1.503/12)^2 = 0.0123$ ft^2 (0.00114 m^2). Thus, water mean velocity $v = q/a = 0.0891/0.0123 = 7.24$ ft/s (2.2 m/s).

7. Determine the fluid's viscosity. From Baumeister, *Mark's Standard Handbook for Mechanical Engineers*, fresh water at 190°F (87.8°C) has a dynamic viscosity of $\mu = 6.75 \times 10^{-6}$ lb·s/ft^2 (323.2×10^{-6} Pa·s).

8. Compute the Reynold's number for the drain system piping. Using pertinent values found previously, Reynold's number = Re = $\rho vd/\mu = (60.33 \times 7.24)(1.1503/12)/(6.75 \times 10^{-6}) = 8{,}088{,}690$.

9. Find the friction factor for the drain system piping. Baumeister, *Mark's Standard Handbook for Mechanical Engineers*, indicates that the relative roughness factor for the drain piping is $\epsilon/d = 150 \times 10^{-6}/0.12 = 0.0012$, hence the Moody friction factor is $f = 0.02$ for the above value of N_{R}.

10. Compute the friction loss of the pumped-drain piping system. Friction loss $F_1 = (fL/d)(v^2/2g)$ where F_1 is in feet (m) of fresh water and L is the equivalent length of the piping system in ft (m). Substituting, $F_1 = (0.02 \times 75/0.125)(7.24^2/[2 \times 32.174]) = 9.77$ ft (1.98 m).

11. Compute the final height of liquid above the tank bottom created by the pumped drain. In the equation for h_{fp} shown in Fig. 32, the net positive suction head (NPSH) = 4.5 ft (1.37 m); freshwater vapor pressure $V_p = 22.29$ ft (6.79 m) of water (Baumeister, *Mark's Standard Handbook for Mechanical Engineers*); height between tank bottom and centerline of pump suction $h_i = 2$ ft (0.61 m). Substituting appropriate values gives $h_{fp} = 4 + 9.77 + 22.29 - 2 - 144(14.7)\{40.1/[40.1 + \pi(2^2)(9.56 - h_{fp})]\}1.4/(60.33 \times 32.0/32.174) = 6.94$ ft (2.12 m), by trial and error, as was done for the gravity drain. From 4 to 10 trials should do it.

12. Compute the final tank vacuum for the pumped drain. Final tank vacuum is found by solving the equation for P_p shown in Fig. 32, thus, $P_p = 2.036(14.7)\{1 - (40.1/[40.1 + (\pi \times 2^2)(9.56)])1.4\} = 19.44$ in Hg (65.83 kPa).

Related Calculations Specifying an appropriate vacuum rating could prevent the collapse of a storage vessel as the contents are being drained while the vent is inadvertently blocked. Vacuum ratings range from full vacuum to no vacuum. A full vacuum rating is advisable for tanks such as those for steam-sterilized sanitary service and those with pumped discharge. Tanks with vents that cannot be blocked require no vacuum rating.

That the maximum gravity-drain vacuum rating occurs at 100 percent full capacity was borne out by the above calculations for P_s and P_g. However, the pumped-drain vacuum rating P_p, under more favorable conditions, still turned out to be 3.19 times greater than P_s, the maximum for gravity drain. This varies with pump capacity and the drain system piping size. These calculations assume ideal gas behavior in the head space above the fluid surface and the process is considered isothermal for drain times longer than 5 min. If the initial fluid height is set too low, it is possible for the tank to be emptied by pump drain before maximum vacuum occurs. The calculations presume a centrifugal pump will not deliver if the NPSH requirements are not met and then backflow into the tank starts. Use the equation for P_p to find the final tank vacuum if it is expected that the tank will be emptied before backflow occurs.

The procedure presented here allows the designer to choose a vacuum rating appropriate to the tank. However, it is suggested that the designer perform applicable code calculations before making a final decision on the vacuum rating for a tank. This presentation is based upon an article by Barry Wintner of Life Sciences International, and which appeared in *Chemical Engineering* magazine.

DESIGNING PRISMATIC PRESSURE VESSELS

A closed-top tank filled with fresh water, Fig. 33*a*, is constructed of 0.3125 in (0.79 cm) medium steel-plate sides. Plate's allowable bending stress = 19,330 lb/in^2 (133.3 MPa) and modulus of elasticity = 27.9 × 10^6 lb/in^2 (192.4 Gpa). Where should horizontal stiffeners be located and what size should they be if their allowable bending stress is 16,000 lb/in^2 (110.32 Gpa)?

Calculation Procedure

1. Check out an initial tentative height for the lowest plate, panel No. 1. Assume the vertical dimension of panel No. 1 is one-third the tank height, or $b_1 = H/3 = 135/3 = 45$ in (1.14 m). Variation of the liquid's horizontal pressure is symbolized by the inclined line shown in Fig. 33*b*. Average pressure distribution over the entire panel is considered to be represented by the pressure head at a level two-thirds down the panel, as indicated by h_1 in Fig. 33*c*. Thus, the applied uniform pressure is $p_1 = h_1\gamma$, where p_1 is in lb/in^2 (Pa); h_1 is in in (m); specific weight γ is in lb/in^3 (N/m^3). Hence, $p_1 = [H - (b_1/3)][\gamma] = [135 - (45/3)][62.4/1728] = 120(0.0361) = 4.33$ lb/in^2 (29.85 kPa).

Then, since the panel dimension ratio $a/b = 60/45 = 1.33$ select, by interpolation from Table 12, deflection parameter $\alpha = 0.0213$ and bending parameter $\beta = 0.4173$. The equation for calculating the panel height is $b = (s_b t^2/\beta p)^{1/2}$, where b is in in (m); allowable bending stress $s_b = 19{,}330$ lb/in^2 (133.3 MPa); plate thickness $t = 0.3125$ in (0.79×10^{-2} m); other symbols are as determined previously. Substituting, $b_1 = [(19{,}330 \times 0.3125^2)/(0.4173 \times 4.33)]^{1/2} = 32.32$ in (0.82 m). Since the result is much smaller than the assumed value, it is necessary to iterate again. Solutions emerge quickly for low-pressure gradients such as those for water or oil.

TABLE 12 Deflection and Bending Parameters (Machine Design).

Dimension ratio a/b	Deflection parameter α	Bending parameter β
1.0	0.0138	0.3078
1.2	0.0188	0.3834
1.4	0.0226	0.4356
1.6	0.0251	0.4680
1.8	0.0267	0.4872
2.0	0.0277	0.4974
2.2	0.0282	0.4999
∞	0.0284	0.5000

2. Review a revised tentative height for panel No. 1. Repeating the previous procedure, but with $b_1 = 29$ in (0.74 m), $p_1 = [H - (b_1/3)](0.0361) = [135 - (29/3)](0.0361) = 4.48$ lb/in^2 (30.89 kPa). Also, $a/b_1 = 60/29 = 2.07$, $\alpha = 0.0279$, and $\beta = 0.4979$. Then $b_1 = [(19{,}330 \times 0.3125^2)/(0.4979 \times 4.48)]^{1/2} = 29.1$ in (0.74 m), and is tentatively acceptable as 29 in (0.74 m).

3. Check panel No. 1 for bending stress. Although the calculated value is very close to the assumed value of b_1, it is best to check the actual plate stress by $s_b = \beta p b^2/t^2$, or $s_{b1} = 0.4979 \times 4.48 \times$

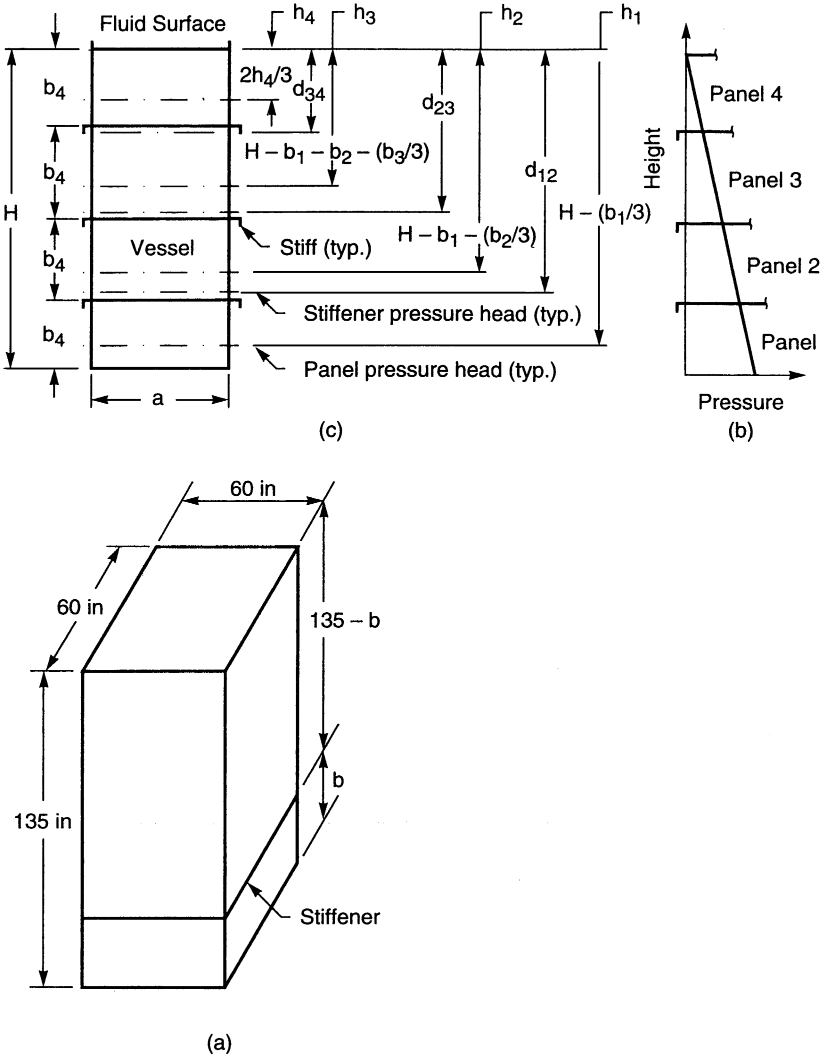


FIGURE 33 (*a*) Tank dimensions. (*b*) Tank pressure distribution. (*c*) Tank elevation.

$29^2/0.3125^2 = 19{,}210$ lb/in^2 (132.4 MPa). This is less than the allowable bending stress; hence, the panel height is still tentatively acceptable as 29 in (0.74 m).

4. Check panel No. 1 for deflection. Although the bending stress is acceptable, the panel should now be checked for deflection, which may not exceed one-half the plate thickness $t/2 = 0.3125/2 = 0.156$ in (0.40 cm). The formula for deflection is $\delta = \alpha p b^4/Et^3$, where δ = plate deflection, in (m); modulus of elasticity $E = 27.9 \times 10^6$ lb/in^2 (192.4 Gpa); other variables are as previously determined. Thus, $\delta_1 = 0.0279 \times 4.48 \times 29^4/(27.9 \times 10^6 \times 0.3125^3) = 0.104$ in (0.26 cm). Since the deflection is within the allowable limit, the value of $b_1 = 29$ in (0.74 m) is now acceptable.

5. Size the stiffener for panel No. 1. The stiffener is continuous around the tank; therefore, the ends are practically fixed. Uniform loading along the length a of the stiffener is considered to be the sum of the pressure loading acting on half of the panels on either side of the stiffener. Average pressure exists at the horizontal centerline of the loaded area. This centerline is below the fluid surface at a distance $d_{12} = b_4 + b_3 + (b_2/2) + [(b_2 + b_1)/2] = 33 + 39 + (34/2) + [(34 + 29)/4] = 104.75$ in (2.66 m). Average pressure $p_{a1} = 0.0361d_{12} = 0.0361 \times 104.75 = 3.78$ lb/in^2 (26.06 kPa). Thus, the stiffener is subjected to a uniform load $W_1 = p_{a1}(b_1 + b_2)/2 = 3.78(29 + 34)/2 = 119.07$ lb/in (20.85 kN/m). The required composite section modulus of the stiffener and plate can be calculated from $S = WL^2/(12s_a)$, where S = section modulus, in^3 (m^3); $L = a$ = length of the stiffener, in (m); s_a = stiffener allowable bending stress, lb/in^2 (Pa). Thus, $S_1 = 119.07 \times 60^2/(12 \times 16{,}000) = 2.24$ in^3 (3.79×10^{-5} m^3).

6. Check out an initial tentative height for panel No. 2. Assume $b_2 = (H - b_1)/3 = (135 - 29)/3 = 35.3$ in (0.90 m). Applied pressure $p_2 = h_2(0.0361) = [H - b_1 - (b_2/3)](0.0361) = [135 - 29 - (35.3/3)](0.0361) = 3.40$ lb/in^2 (23.44 kPa). And, $a/b = 60/35.3 = 1.70$, where $\alpha = 0.0259$ and $\beta = 0.4776$. Substituting in the equation for panel height $b_2 = [(19{,}330 \times 0.3125^2)/(0.4776 \times 3.40)]^{1/2} = 34.1$ in (0.87 m). This is so close to the assumed value that a tentative value of $b_2 = 34$ in (0.86 m) can be set.

7. Check panel No. 2 for bending stress. Substituting in the equation for bending stress $s_{b2} = 0.4776 \times 3.40 \times 34^2/0.3125^2 = 19{,}220$ lb/in^2 (132.5 MPa). This is within the allowable limit of the bending stress.

8. Check panel No. 2 for deflection. Substituting in the equation for deflection $\delta_2 = 0.0259 \times 3.40 \times 34^4/(27.9 \times 10^6 \times 0.3125^3) = 0.138$ in (0.35 cm). Since this value is less than the allowable deflection of 0.156 in (0.40 cm), the height of panel 2 is now acceptable as 34 in (0.86 m).

9. Size the stiffener for panel No. 2. Pressure head $d_{23} = b_4 + (b_3/2) + [(b_3 + b_2)/4] = 33 + (39/2) + [(39 + 34)/4] = 70.75$ in (1.80 m). Average pressure $p_{a2} = 0.0361d_2 = 0.0361(70.75) = 2.55$ lb/in^2 (17.56 kPa). Uniform load $W_2 = p_{a2}(b_2 + b_3)/2 = 2.55(34 + 39)/2 = 93.08$ lb/in (16.30 kN/m). Composite section modulus $S_2 = W_2a^2/(12s_a) = 93.08 \times 60^2/(12 \times 16{,}000) = 1.75$ in^3 (2.94×10^{-5} m^3).

10. Check out an initial tentative height for panel No. 3. Following the previous procedures, assume panel height $b_3 = (135 - 29 - 34)/3 = 24$ in (0.61 m). Applied pressure $p_3 = [135 - 29 - 34 - (24/3)](0.0361) = 2.31$ lb/in^2 (15.93 kPa). Then, since $a/b = 60/24 = 2.50$, $\alpha = 0.0284$ and $\beta = 0.5000$ for infinity because corresponding values for $a/b = 2.50$ are not much less. Substituting in the equation for panel height $b_3 = [19{,}330 \times 0.3125^2/(0.5000 \times 2.31)]^{1/2} = 40.4$ in (1.23 m). Because the result is much greater than the assumed value, it is necessary to iterate again.

11. Check out a revised tentative height for panel No. 3. Repeating the previous procedure, but with $b_3 = 48$ in (1.22 m), $p_3 = h_3(0.0361) = [135 - 29 - 34 - (48/3)](0.0361) = 2.02$ lb/in^2 (13.94 kPa). Also, $a/b = 60/48 = 1.25$ and, by interpolation, $\alpha = 0.0201$ and $\beta = 0.4023$. Again, solving for $b_3 = [19{,}330 \times 0.3125^2/(0.4023 \times 2.02)]^{1/2} = 48.2$ in (1.22 m). This result is very close to the assumed value; hence $b_3 = 48$ in (1.22 m) is tentatively acceptable.

12. Check panel No. 3 for bending stress. The actual plate stress $s_{b3} = 0.4023 \times 2.02 \times 48^2/(0.3125)^2 = 19{,}170$ lb/in^2 (132.17 MPa). This stress is less than the allowable 19,330 lb/in^2 (133.3 MPa), hence the height of panel No. 3 may be less, but not more, than 48 in (1.22 m).

13. Check panel No. 3 for deflection. Substituting in the equation for deflection $\delta_3 = 0.0201 \times 2.02 \times 48^4/(27.9 \times 10^6 \times 0.3125^3) = 0.253$ in (0.64 cm). Since this deflection exceeds 0.156 in (0.40 cm), it is necessary to select a smaller value for b_3.

14. Check panel No. 3 for deflection with its height reduced. With $b_3 = 39$ in (0.99 m), $h_3 = 135 - 29 - 34 - (39/3) = 59$ in (1.5 m) and $p_3 = 59 \times 0.0361 = 2.13$ lb/in^2 (14.69 kPa). Also, for $a/b = 60/39 = 1.54$ the value of $\alpha = 0.0243$. Then, $\delta_3 = 0.0243 \times 2.13 \times 39^4/(27.9 \times 10^6 \times 0.3125^3) = 0.141$ in

(0.36 cm). This deflection is close enough to 0.156 in (0.40 cm) to stop iterating for a value of b_3 greater than 39 in (0.99 m) and less than 48 in (1.22 m) wherein the bending stress does not exceed the allowable value.

15. Size the stiffener for panel No. 3. Pressure head $d_{34} = (b_4/2) + [(b_4 + b_3/4] = (33/2) + [(33 + 39)/4] = 34.5$ in (0.88 m). Average pressure $p_{a3} = 0.0361d_3 = 0.0361 \times 34.5 = 1.25$ lb/in^2 (8.62 kPa). Uniform load $W_3 = p_{a3}(b_3 + b_4)/2 = 1.25(39 + 33)/2 = 45$ lb/in (7.88 kN/m). Composite section modulus $S_3 = W_3a^2/(12s_a) = 45.0 \times 60^2/(12 \times 16{,}000) = 0.84$ in^3 (1.42×10^{-5} m^3).

16. Determine if there is a need for reinforcing panel No. 4. The height of the uppermost panel is $b_4 = 135 - 29 - 34 - 39 = 33$ in (0.84 m). Applied water pressure $p_4 = h_4(0.0361) = (2 \times 33/3)(0.0361) = 0.79$ lb/in^2 (5.45 kPa). Panel dimension ratio $a/b = 60/33 = 1.82$, hence $\alpha = 0.0268$ and $\beta = 0.4882$. Then, $b_4 = [19{,}330 \times 0.3125^2/(0.4882 \times 0.79)]^{1/2} = 70.0$ in (1.78 m). Since this calculated height is much greater than the actual height of the panel, it shows that panel No. 4 is adequate without an additional stiffener.

Related Calculations Stiffeners divide the vessel faces into rectangular plates whose edges are considered to be fixed. Plate thickness can be minimized to an optimum size which provides panel heights within preferred magnitudes while maintaining maximum stress or deflection just within allowable limits. For more accuracy in obtaining values for α and β and to avoid interpolation, values may be obtained from graphs plotted from the listed values.

The solution to this example is based upon the method shown in an article written by R. Jay Smith and G.L.B. Knight in the November 26, 1981, issue of *Machine Design* magazine. Both authors were associated with the Davy McKee Corp., Lakeland, Florida.

MINIMUM-COST PRESSURE VESSELS

Find the diameter that minimizes the cost to construct a pressure tank with the following design characteristics: volume $V = 1400$ ft^3 (40 m^3); allowable stress, $s = 16{,}000$ lb/in^2 (110 MPa); welding efficiency $e = 85$ percent; internal pressure $p = 400$ lb/in^2 (2760 kPa); corrosion allowance $t_c = 0.05$ in (1.27×10^{-3} m). Also, find the length-to-diameter ratio of the tank.

Calculation Procedure

1. Compute the approximate diameter. Use the equation for the approximate diameter, $D_a = [V(0.2898V^{1/3} + Z_1)/(1.426V^{1/3} + Z_2)]^{1/3}$, where $Z_1 = 0.2829t_c[(se/p) - 0.6]$ and $Z_2 = 0.2175t_c[(se/p) - 0.6]$. Substituting $Z_1 = 0.2829 \times 0.05[(16{,}000 \times 0.85/400) - 0.6] = 0.4724$ and $Z_2 = 0.2175 \times 0.05[(16{,}000 \times 0.85/400) - 0.6] = 0.3632$. Then $D_a = \{1400[0.2898(1400)^{1/3} + 0.4724]/[1.426(1400)^{1/3} + 0.3632]\}^{1/3} = 6.83$ ft (2.08 m).

2. Compute the exact diameter. The equation for the exact diameter is $D = [V(0.4244D + Z_1)/(2.088D + Z_2)]^{1/3}$. For a first trial use the approximate diameter; hence, $D = \{1400[(0.4244 \times 6.83) + 0.4724]/[(2.088 \times 6.83) + 0.3632)\}^{1/3} = 6.87$ ft (2.09 m). Next, try $D = 6.87$ ft (2.09 m). Thus, $D = \{1400[(0.4244 \times 6.87) + 0.4724]/[(2.088 \times 6.87) + 0.3632]\}^{1/3} = 6.86$ ft (2.09 m). Iterating once more, $D = \{1400[(0.4244 \times 6.86) + 0.4724]/[(2.088 \times 6.86) + 0.3632]\}^{1/3} = 6.86$ ft (2.09 m); the exact diameter.

3. Determine the length-to-diameter ratio of the tank. The length-to-diameter ratio $L/D = 4V/\pi D^3 = (4 \times 1400)/(\pi \times 6.86^3) = 5.52$.

Related Calculation The approximate equation directly provides the size of a diameter within 3 percent, or less, of the exact size. Greater accuracy can be obtained by iterating the exact equation which converges rapidly. The length-to-diameter ratio varies accordingly. For metric calculations, the coefficient for Z_1 should be 3.420 instead of 0.2829 and for Z_2 it should be 2.629 instead of 0.2175.

This example is based on an article written by J. Zigrang and N.D. Sylvester, professor and dean, respectively, at the University of Tulsa, Tulsa, Oklahoma. The piece appeared in the December 12, 1985, issue of *Machine Design* magazine.