Chapter 18

Other Heat Exchange Equipment

INTRODUCTION

The purpose of this chapter is to extend the material presented in the three previous chapters to the design, operation, and predicative calculations of several other types of heat exchangers. The presentation will focus on: evaporators, waste heat boilers, condensers, and quenchers.

As described earlier, heat exchangers are devices used to transfer heat from a hot fluid to a cold fluid. They can be classified by their functions. Faraq and Reynolds[1] provide an abbreviated summary of these units and their functions (see Table 14.1). To this list can be added a host of other exchangers, including parallel corrugated plates, plate units, etc.[2] Also to be included in the list are direct-contact coolers and speciality condensers.[3]

The four general classifications of direct-contact gas–liquid heat transfer operations are:

1. simple gas cooling,
2. gas cooling with vaporization of coolant,
3. gas cooling with partial condensation, and
4. gas cooling with total condensation.

Most of the direct-exchange applications listed above are accomplished with the following devices:

1. Baffle-tray columns.
2. Spray chambers.
3. Packed columns.
4. Crossflow-tray columns.
5. Pipeline contactors.

ILLUSTRATIVE EXAMPLE 18.1

Which of the following is NOT a heat exchanger?

   (a) Reboiler

   (b) Condenser

   (c) Absorber

   (d) Superheater

SOLUTION: A reboiler is connected to the bottom of a fractionating tower and provides the reboil heat necessary for distillation. The heating medium, which may be steam or a hot process fluid, transfers heat to the bottoms product of the column. Therefore, a reboiler is classified as a heat exchanger.

    A condenser (see later section) condenses a vapor or mixture of vapors either alone or in the presence of a non-condensable gas. The cooling medium, which may be cooling water or air, absorbs heat from the hot process vapor. Therefore, a condenser is classified as a heat exchanger.

    An absorber[4] is a mass transfer device, used for separating components from either a liquid or gaseous stream. Therefore, an absorber is NOT a heat exchanger.

    A superheater heats a vapor above its saturation temperature. Therefore a superheater is classified as a heat exchanger.

    Therefore, (c) is the correct answer. ■

    The following topics are treated in this chapter:

    Evaporators

    Waste heat boilers

    Condensers

    Quenchers

Cooling towers do not receive treatment; however, material is available in the literature.[4]

EVAPORATORS

The vaporization of a liquid for the purpose of concentrating a solution is a common operation in the chemical process industry. The simplest device is an open pan or kettle that receives heat from a coil or jacket or by direct firing underneath the pan. Perhaps the traditional unit is the horizontal-tube evaporator in which a liquid (to be concentrated) in the shell side of a closed, vertical cylindrical vessel is evaporated by passing steam or another hot gas through a bundle of horizontal tubes contained in the lower part of the vessel. The liquid level in the evaporator is usually less than half the height of the vessel; the empty space permits disengagement of entrained liquid from the vapor passing overhead.[5]

The describing equation of an evaporator, like that of any heat exchanger, is given by

$$\dot{Q} = UA\Delta T_{\text{lm}} \tag{18.1}$$

The term $(UA)^{-1}$ is equal to the sum of the individual resistances of the steam, the walls of the tubes, the boiling liquid, and any fouling that may be present.

Consider Figure 18.1.[5] Assume F lb of feed to the evaporator per hour, whose solid content is x_F. (The symbol x is employed for weight fraction.) Also, assume the enthalpy of the feed per lb to be h_F. L lb of thick liquor, whose composition in weight fraction of solute is x_L and whose enthalpy is h_L leaves from the bottom of the evaporator. V lb of vapor, having a solute concentration of y_V and an enthalpy of h_V Btu/lb, leaves the unit. In most evaporators, the vapor is pure water, and therefore y_V is zero.

The material balance equations for this evaporator are relatively simple. A total material balance gives

$$F = L + V \tag{18.2}$$

A componential balance leads to
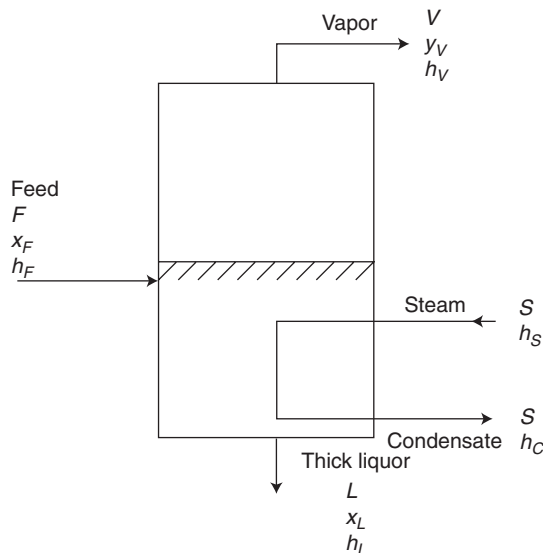
$$Fx_F = Lx_L + Vy_V \tag{18.3}$$

Figure 18.1   Material and enthalpy balance for a single-effect evaporator.

In order to furnish the heat necessary for evaporation, S lb of stream is supplied to the heating surface with an enthalpy of h_S Btu/lb; S lb of condensate with an enthalpy of h_C Btu/lb is leaving as condensate. One simplifying assumption usually made is that, in an evaporator, there is very little cooling of the condensate. This is usually less than a few degrees in practice; the sensible heat recovered from cooling the condensate is so small compared to the latent heat of the steam supplied to the heating surface that the condensate will leave at the condensing temperature of the steam. The enthalpy balance equation is therefore,

$$Fh_F + Sh_S = Vh_V + Lh_L + Sh_C \qquad (18.4)$$

Both Equations (18.1) and (18.4) are applied in tandem when designing and/or predicting the performance of an evaporator.

Mixing Effects

The calculations for an evaporator can be complicated because of an adiabatic solution temperature change. When two or more pure substances are mixed to form a solution, a heat effect usually results. Many have experienced this effect on mixing concentrated sulfuric acid with water. This heat of mixing is defined as the enthalpy change that occurs when two or more pure substances are mixed at constant temperature and pressure to form a solution.

Enthalpy concentration diagrams offer a convenient way to calculate enthalpy of mixing effects and temperature changes associated with this type of process. These diagrams, for a two-component mixture, are graphs of the enthalpy of a binary solution plotted as a function of composition (mole fraction or weight fraction of one component), with the temperature as a parameter. For an ideal solution, isotherms or an enthalpy-concentration diagram would be straight lines. For a real solution, the actual isotherm is displaced vertically from the ideal solution isotherm at a given point by the value of Δh at that point, where Δh is the enthalpy of mixing. With reference to the enthalpy concentration diagram in Figure 18.2, Δh is negative over the entire composition range. This means that heat must be evolved whenever the pure components at the given temperature are mixed to form a solution at the same temperature. Such a system is said to be exothermic. An endothermic system is one for which the heats of solution are positive, i.e., solution at constant temperature is accompanied by the absorption of heat. Organic mixtures often fit this description.[5]

ILLUSTRATIVE EXAMPLE 18.2

Calculate the temperature when 50 lb of pure H_2SO_4 at 25°C is mixed adiabatically with 200 lb of a 50% by weight aqueous H_2SO_4 solution at 25°C. Also calculate the heat effect if the final mixture is at 25°C.

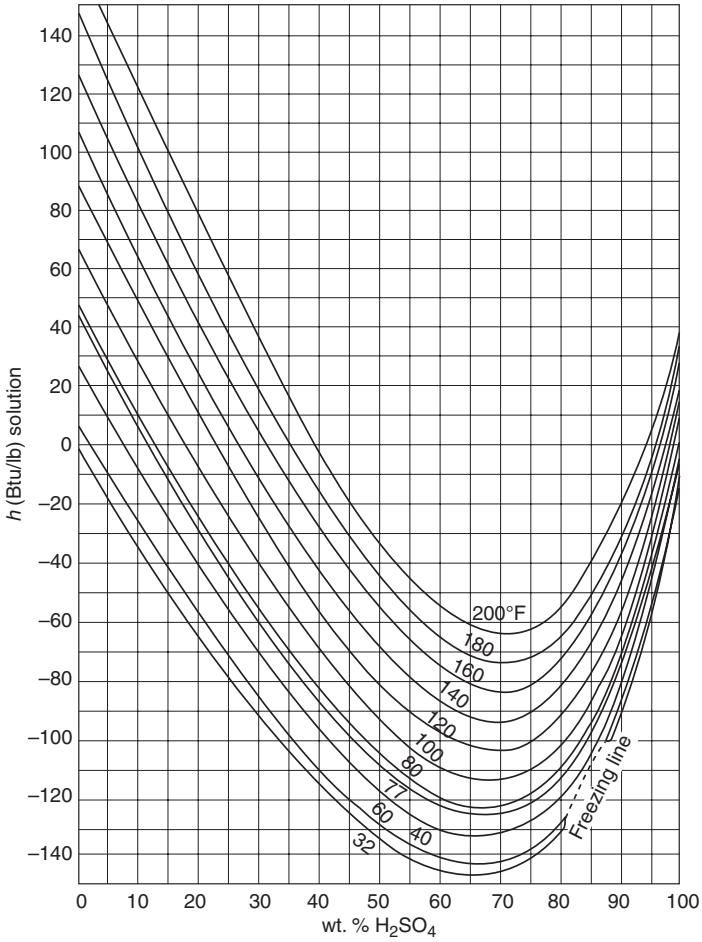**Figure 18.2**   Enthalpy-concentration diagram for H_2SO_4–H_2O. (Adapted from the data of W. D. Ross, *Chem. Eng. Progr.*, 43, 314, 1952.)

SOLUTION: Calculate the weight (or mass) percent H_2SO_4 in the final H_2SO_4–H_2O mixture:

$$H_2SO_4: 50 + 100 = 150 \, lb$$
$$H_2O: 100 \, lb$$
$$\%H_2SO_4 = (150/250)100 = 60\%$$

Construct a straight line between the 50% solution and pure H_2SO_4 at 25°C (77°F) on Figure 18.2 in order to estimate the final temperature in °F at 60% H_2O:

$$T = 140°F$$

Calculate the heat, Q, that needs to be transferred to return the mixture to 25°C. From Figure 18.2,

$$h \text{ at } 140°F = -86 \text{ Btu/lb}$$
$$h \text{ at } 77°F = -121.5 \text{ Btu/lb}$$

so that

$$Q = m\Delta h = m(h_2 - h_1)(250)(-121.5 + 86) = -8875 \text{ Btu}$$

This represents the amount of heat that must be removed since Q is negative.

There is also a heat effect when gases or solids are dissolved in liquids. This effect is normally small but can be significant in some applications, particularly acidic gases in water. This type of calculation should be included in not only evaporator calculations but also for some absorber systems when temperature changes through the unit can be significant. ■

ILLUSTRATIVE EXAMPLE 18.3

A single-effect evaporator is to concentrate 10,000 lb/h of a 10% NaOH solution to 75%. The feed enters at 120°F and the evaporator is to operate at an absolute pressure of 14.7 psi. The 75% NaOH solution leaves at the evaporator equilibrium temperature. For what heat transfer rate (Btu/h) should the evaporator be designed? Also calculate the area requirement in the evaporator if the overall heat transfer coefficient is 500 Btu/h · ft² · °F and 103 psig (340°F) saturated steam is employed in the steam chest. The NaOH–H$_2$O enthalpy-concentration diagram is provided in Figure 18.3. The enthalpy of saturated steam at 14.7 psia is approximately 1150 Btu/lb.

SOLUTION: Assume a basis of one hour of operation. Calculate the flow rate of steam, V, and the 75% NaOH–H$_2$O solution, L, leaving the evaporator. From a NaOH balance,

$$(10,000)(0.1) = (0.75)(L)$$
$$L = 1333.3 \text{ lb/h}$$

From an overall material balance,

$$F = L + V$$
$$V = 10,000 - 1333 = 8667 \text{ lb/h}$$

Estimate the enthalpy of solution entering the unit, h_F, in Btu/lb, see Figure 18.3,

$$h_F = 81 \text{ Btu/lb solution}$$

Estimate the enthalpy of the 75% NaOH solution, h_L, leaving the unit:

$$h_L = 395 \text{ Btu/lb solution}$$

The evaporator heat required, $\dot{Q}$, in Btu/h is then

$$\dot{Q} = \Sigma \dot{m}_i h_i = (8667)(1150) + (1333)(395) - (10,000)(81) = 9,683,600 \text{ Btu/h}$$
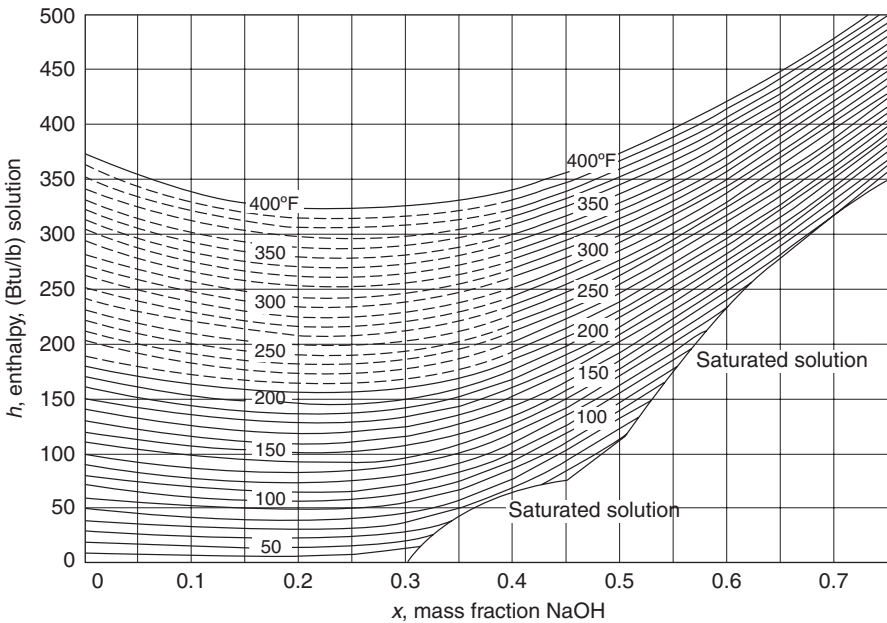
Figure 18.3   Enthalpy-concentration diagram for NaOH–H₂O. [Adapted from W. L. MᴄCᴀʙᴇ, *Trans. AIChE*, 31, 129, 1935; R. H. Wɪʟsᴏɴ and W. L. MᴄCᴀʙᴇ, *Ind. Eng. Chem.*, 34, 558, 1942.]

Finally, the area of the evaporator in ft² may be calculated:

$$\dot{Q} = UA\Delta T \tag{18.1}$$

Rearranging and substituting,

$$A = 9{,}683{,}600/(500)(340 - 212) = 151.3 \text{ ft}^2 \qquad \blacksquare$$

One of the principal operating expenses of evaporators is the cost of steam for heating. A considerable reduction in those costs can be achieved by operating a battery of evaporators in which the overhead vapor from one evaporator (or "effect") becomes the heating medium in the steam chest of the next evaporator, thus saving both the cost on condensing the vapor from the first unit and supplying heat for the second. Several evaporators may operate in a battery in this fashion.[2,5,6]

Basically, a multiple-effect evaporator may be thought of as a number of resistances, in series, to the flow of heat. The main resistances are those to heat transfer across the heating surface of each effect in the evaporator and across the final condenser if a surface condenser is used. The resistances of the heating surfaces are equivalent to the reciprocal of the product of area and overall heat transfer coefficient $(1/UA)$ for each effect.

Neglecting all resistances but those noted above and assuming they are equal, it can be seen that if the number of resistances (effects) is doubled, the flow of heat (steam consumption) will be cut in half. With half as much heat, each effect will evaporate about half as much water. But since there are twice as many effects, the total

evaporation will be the same. Thus, under these simplifying assumptions, the same output would be obtained regardless of the number of effects (each of equal resistances), and the steam consumption would be inversely proportional to the number of effects or the total heating surface installed.

   This resistance concept is also useful in understanding the design and operation of such units. The designer places as many resistances (effects) in series as can be afforded in order to reduce the steam consumption. One can also show that the lowest total area required arises when the ratio of temperature drop to area is the same for each effect.

ILLUSTRATIVE EXAMPLE 18.4

Given the triple-effect evaporator information below, calculate the temperature drop across the heating surface in the first effect:

Effect	U, Btu/h $\cdot$ ft^2 $\cdot$ °F	A, ft^3
1	240	125
2	200	150
3	125	160

The condensing stream temperature in the first effect and the vapor leaving the third effect are 400°F and 120°F, respectively.

SOLUTION: Assuming the resistances across each effect are the same,

$$R_1 = \frac{1}{U_1 A_1} = \frac{1}{(240)(125)} = 3.333 \times 10^{-5}$$

$$R_2 = \frac{1}{U_2 A_2} = \frac{1}{(200)(150)} = 3.333 \times 10^{-5}$$

$$R_3 = \frac{1}{U_3 A_3} = \frac{1}{(125)(160)} = 5.0 \times 10^{-5}$$

$$\sum R = 11.666 \times 10^{-5}$$

One may therefore write,

$$\frac{\Delta T_1}{\Delta T_{\text{TOTAL}}} = \frac{R_1}{\sum R}$$

Substituting

$$\Delta T_1 = \frac{(400 - 120)(3.333 \times 10^{-5})}{(11.666 \times 10^{-5})}$$

$$= 80°F \qquad \blacksquare$$

ILLUSTRATIVE EXAMPLE 18.5

Qualitatively describe what an evaporator does.

SOLUTION: This is an open-ended question that could just have easily been placed in Chapter 23. Standiford[7] provides an answer to this example.

The requirements for the correct functioning of any evaporator are:

1. It must transfer a great deal of heat—on the order of 1000 Btu/lb of water evaporated. This, more than anything else, determines the type, size, and cost of each effect of the evaporator.

2. It must efficiently separate the vapor from the residual liquid. What is efficient in one evaporator may be many orders of magnitude different from what is efficient in another (e.g., from loss of salt value at only a few dollars a ton, to lithium chemicals valued at a dollar a pound, to radioactive waste). Separation may be important solely because of the value of product lost, pollution problems, or because of fouling or corrosion in the equipment in which the vapor is condensed or in which the condensate is subsequently used.

3. It must make as efficient use of the available heat or mechanical energy as is economically feasible. This means using the vapor evaporated in one part (effect) of the evaporator as the heating steam in another effect that is operating at a lower temperature (as in a multiple-effect evaporator), or compressing the vapor evolved so that it can be used as the heating medium in the same evaporator, or by employing a combination of these. As in most other cases, efficiency is usually gained only as capital cost is increased and, for evaporators, the designer has a wide range to choose from.

4. It must meet conditions dictated by the characteristics of either the liquid being evaporated or the product. In a crystallizing evaporator,[4] crystal size, shape, and purity may be of the utmost importance. If a salting or scaling liquid is to be handled, the evaporator type selected must be capable of dealing with it. Product quality characteristics that may also be important are corrosiveness and degradation at high temperature, long holdup time, or contact with certain metals.

Other considerations are the size of the operation, the foaming characteristics of the liquor, the need for special types of materials of construction such as polished stainless steel (required for many food products), and easy access for cleaning.   ■

ILLUSTRATIVE EXAMPLE 18.6 (adapted from Badger and Banchero[8])

An evaporator is to be fed with 5000 lb of solution containing 2% solids by weight. The feed, F, is at a temperature of 100°F. It is to be concentrated to a solution of 5% solute by weight in an evaporator operating at a pressure of 1 atm in the vapor space. In order to carry out the evaporation, the heating surface is supplied with steam at 5 psig (227°F) and the overall heat transfer coefficient of the evaporator, U, is 280 Btu/h · ft^2 · °F. What is the mass of vapor produced, the total mass of steam required, and the surface area required? *Neglect* enthalpy of solution effects.

SOLUTION: In order to simplify the solution, it will be assumed that the solution is so dilute that its boiling point is the same as the boiling point of water and that its heat capacity and latent enthalpy are the same as that of water. Under these circumstances, the thermal properties of the solution (both feed and product) and of the steam can be taken from the steam tables. This results in the following values for pertinent quantities on a per hour

basis (see Figure 18.1):

$$F = 5000\,\text{lb}$$
$$x_F = 0.02$$

Total solids in feed $= (5000)(0.02) = 100\,\text{lb}$ (componential balance)

Total water in feed $= 5000 - 100 = 4900\,\text{lb}$ (componential balance)

$$T_F = 100°\text{F}$$
$$h_F = 68\,\text{Btu/lb (estimated from steam tables at }100°\text{F)}$$
$$x_L = 0.05$$

Total solids in liquor $= (5000)(0.02) = 100\,\text{lb}$

$$L = 100/(0.05) = 2000\,\text{lb}$$
$$T_L = 212°\text{F (at 1 atm)}$$
$$h_L = 180\,\text{Btu/lb (estimated from steam tables)}$$
$$V = 5000 - 2000 = 3000\text{ (overall balance)}$$
$$T_V = 212°\text{F}$$
$$h_V = 1156\,\text{Btu/lb (estimated from steam tables at 1.0 atm)}$$
$$T_S = 227°\text{F}$$
$$h_S = 1156\,\text{Btu/lb (estimated from steam tables at }227°\text{F}$$
$$\text{and 5 psig)}$$
$$T_C = 227°\text{F (condensate)}$$
$$h_C = 195\,\text{Btu/lb (estimated from steam tables)}$$

The following enthalpy (energy) balance results (S representing steam):

$$F(h_F) + S(h_S) = V(h_V) + L(h_C) + S(h_C)$$

Substituting and solving yields

$$(5000)(68) + 1156(S) = (3000)(1150) + (2000)(180) + 195(S)$$
$$S = 3611\,\text{lb}$$

The total heat requirement is

$$Q = (3611)(1156 - 195) = 3,470,000\,\text{Btu}$$

The required area is (assuming all of the above is based on one hour)

$$A = Q/U\Delta T = 3,470,000/(280)(227 - 212) = 826\,\text{ft}^2 \qquad \blacksquare$$

ILLUSTRATIVE EXAMPLE 18.7 (adapted from Badger and Banchero[8])

An evaporator is fed with 5000 lb/h of a 20% solution of sodium hydroxide at 100°F. This is to be concentrated to a 40% solution. The evaporator is supplied with saturated steam at 5 psig. Although the unit operates with the vapor space at a pressure of 4 in Hg absolute, the boiling

temperature of the solution in the evaporator is 198°F (due to the superheat created by the exposed heating element). The overall heat transfer coefficient is $400\,\text{Btu/h} \cdot \text{ft}^2 \cdot °\text{F}$. Calculate the steam rate and the required heat transfer area.

SOLUTION: Assume a basis of one hour. The following data is known:

$$F = 5000\,\text{lb}; \quad x_F = 0.20$$
$$T_F = 100°\text{F}; \quad x_L = 0.40$$
$$x_V = y_V = 0.00$$

From the steam tables (see Appendix) at 228°F and 5 psig,

$$h_S = 1156\,\text{Btu/lb}$$
$$h_C = 196\,\text{Btu/lb}$$
$$h_{\text{vap}} = 1156 - 196 = 960\,\text{Btu/lb}$$

In addition,

$$\text{Boiling point of water at 4 in Hg absolute} = 125.4°\text{F}$$
Enthalpy of saturated steam at $125°\text{F} = 1116\,\text{Btu/lb}$ (estimated from steam tables)

$$h_F(100°\text{F}) \simeq 55\,\text{Btu/lb (from Figure 18.3)}$$
$$h_L(198°\text{F}) \simeq 177\,\text{Btu/lb (from Figure 18.3)}$$

A componential material balance yields

$$Fx_F = Lx_L + Vy_V; \; y_V = 0.0$$
$$(5000)(0.20) = (0.40)(L)$$
$$L = 2500\,\text{lb}$$
$$V = 2500\,\text{lb}$$

An enthalpy balance yields

$$Fh_F + S(h_S - h_C) = Vh_V + Lh_L$$

In calculating h_V, the enthalpy of vapor leaving the solution, it should be remembered that this vapor is in equilibrium with the boiling solution at a pressure of 4 in Hg absolute and therefore, is superheated in comparison with vapor in equilibrium with water at the same pressure. Since the heat capacity of superheated steam in this range may be assumed to be $0.46\,\text{Btu/lb} \cdot °\text{F}$, then

$$h_V = 1116 + (0.46)(198.0 - 125) = 1150\,\text{Btu/lb}$$

Substituting into the enthalpy balance gives

$$5000(55.0) + S(1156 - 196) = (2500)(1150) + (2500)(177)$$

Solving for S, the steam rate, yields

$$S = 3170\,\text{lb/h}$$

The area may now be calculated using the standard heat exchanger equation.

$$\dot{Q} = UA\Delta T$$
$$(3170)(1156 - 196) = 400(A)(228 - 198)$$
$$A = 254\,\text{ft}^2 \qquad\qquad\qquad ∎$$

WASTE HEAT BOILERS

Energy has become too valuable to discard. As a result, waste heat and/or heat-recovery boilers are now common in many process plants. As the chemical processing industries become more competitive, no company can afford to waste or dump thermal energy. This increased awareness of economic considerations has made waste-heat boilers one of the more important products of the boiler industry. The term "waste heat boiler" includes units in which steam is generated primarily from the sensible heat of an available hot flue or hot gas stream rather than by solely firing fuel.

An obvious by-product of incineration processes is thermal energy—in many cases, a large amount of thermal energy. The total heat load generated by a typical hazardous waste incinerator, for example, is in the range of 10 to 150 million Btu/h. While waste heat boilers are capable of recovering 60–70% of this energy, the effort may or may not be justified economically. In assessing the feasibility of recovery, a number of factors must be taken into account; among these are the amount of heat wasted, the fraction of that heat that is realistically recoverable, the irregularity in availability of heat, the cost of equipment to recover the heat, and the cost of energy. The last factor is particularly critical and may be the most important consideration in a decision involving whether or not to harness the energy generated by a particular incinerator. Other important considerations are the incinerator capacity and nature of the waste/fuel being handled. Generally, heat recovery on incinerators of less than 5 million Btu/h may not be economical because of capital cost considerations. Larger capacity incinerators may also be poor candidates for heat recovery if steam is not needed at the plant site or if the combustion gases are highly corrosive; in the latter case, the maintenance cost of the heat recovery equipment may be prohibitive.

The main purpose of a boiler is to convert a liquid, usually water, into a vapor. In most industrial boilers, the energy required to vaporize the liquid is provided by the direct firing of a fuel in the combustion chamber. The energy is transferred from the burning fuel in the combustion chamber by convection and radiation to the metal wall separating the liquid from the combustion chamber. Conduction then takes place through the metal wall and conduction/convection into the body of the vaporizing liquid. In a waste heat boiler, no combustion occurs in the boiler itself; the energy for vaporizing the liquids is provided by the sensible heat of hot gases which are usually product (flue) gases generated by a combustion process occurring elsewhere in the system. The waste heat boilers found at many facilities make use of the flue gases for this purpose.

In a typical waste heat boiler installation, the water enters the unit after it has passed through a water treatment plant or the equivalent. This boiler feed water is sent to heaters/economizers and then into a steam drum. Steam is generated in the boiler by indirectly contacting the water with hot combustion (flue) gases. These hot gases may be around 2000°F. The steam, which is separated from the water in the steam drum, may pass through a superheater, and is then available for internal use or export. The required steam rate for the process or facility plus the steam temperature and pressure are the key design and operating variables on the water side. The inlet and outlet flue gas temperatures also play a role, but it is the chemical properties of the flue gas that can significantly impact boiler performance. For example, acid gases can arise due to the presence of any chlorine or sulfur in the fuel or waste. The principal combustion product of chlorine is hydrogen chloride, which is extremely corrosive to most metal heat transfer surfaces. This problem is particularly aggravated if the temperature of the flue gas is below the dewpoint temperature of HCl (i.e., the temperature at which the HCl condenses). This usually occurs at temperatures of about 300°F. In addition to acid gases, problems may also arise from the ash of incineration processes; some can contain a fairly high concentration of alkali metal salts that have melting points below 1500°F. The lower melting point salts can slag and ultimately foul (and, in some cases, corrode) boiler tubes and/or heat transfer surfaces.

Boilers may be either *fire tube* or *water tube* (water-wall). Both are commonly used in practice; the fire-tube variety is generally employed for smaller applications ($<15 \times 10^6$ Btu/h). In the fire-tube waste heat boiler, the hot gases from the process are passed through the boiler tubes. The bundle of tubes is immersed in the water to be vaporized; the vaporizing water and tube bundles are encased in a large insulated container called a shell. The steam generated is stored in a surge drum, usually located above the shell and connected to the shell through vertical tubes called risers. Because of construction constraints, steam pressure in fire-tube boilers is usually limited to around 1000 psia.

Fire-tube boilers are compact, low in initial cost, and easy to modularize based on plant requirements. However, they are also slow to respond to changes in demand for steam (load) compared to water-tube boilers, and the circulation is slower. Also, stresses are greater in fire-tube boilers because of their rigid design and subsequent inability to expand and contract easily. Fire-tube boilers usually range in size from less than 2 to 50 million Btu/h. Most industrial fire-tube boilers are either horizontal return tube (HRT), scotch marine, or firebox. Fire-tube boilers are usually directly fired with fuels, either liquid or gaseous. They also serve to recover heat from incinerators fired on waste fuels.

In the water-tube waste heat boiler, the water is contained inside the tubes and the hot flue gases flow through the tube bundle, usually in a direction perpendicular to the tubes (cross-flow). Because of the increased turbulence that accompanies cross-flow, the overall heat transfer coefficient for water-tube boilers is higher than that for fire-tube boilers. This advantage is somewhat offset, however, because it is more difficult to clean the outside surfaces of the tubes than the inside surfaces. As a result, heat transfer losses and maintenance problems due to flue gas fouling tend to be greater in water-tube boilers.

Water-tube boilers can be physically divided into two sections, the furnace and the convection pass. Furnaces (fireboxes, combustion chambers, etc.) will vary in configuration and size, but their function is to contain the flaming combustion gases and transfer the heat energy to the water-cooled walls. The convection pass contains the superheaters, reheater, economizer, and air preheater heat exchangers, where the heat of the combustion flue gases is used to increase the temperature of the steam, water, and combustion air. The *superheaters* and *reheaters* are designed to increase the temperature of the steam generated within the tubes of the furnace walls. Steam flows inside the tubes and flue gas passes along the outside surface of the tubes. The *economizer* is normally a counterflow heat exchanger designed to recover energy from the flue gas after the superheater and the reheater. The boiler economizer is a tube bank type, hot-gas-to-water heat exchanger. It increases the temperature of the water entering the steam drum. The *air heater* is not a portion of the steam–water circuit, but serves a key role in the steam generator system to provide additional heat transfer and efficiency. In many cases, especially in a high pressure boiler, the temperature of the flue gas leaving the economizer is still quite high. The air heater recovers much of this energy and adds it to the combustion air. Heating the combustion air prior to its entrance to the furnace reduces fuel usage.

Water-tube boilers respond quickly to changes in demand for steam due to improved water circulation. They can withstand much higher operating pressures and temperatures than fire-tube boilers. In addition, the water-tube boiler design is safer. They can also burn a wide variety of fuels as well as wastes and have the ability to expand and contract more easily than fire-tube boilers. The major drawback is that water-tube boilers are more expensive to install. They also require more complicated furnaces and repair techniques.

Describing Equations

The design of waste heat boilers involve calculations that are based on energy balances and estimates of the rates of heat transfer. Although some units operate in an unsteady-state or cyclical mode, the calculation procedures are invariably based on steady-state conditions.

In heat transfer equipment, there is no shaft work, and potential and kinetic energy effects are small in comparison with the other terms in an energy balance equation. Heat flow to or from the surroundings is not usually desired in practice and is usually reduced to a small magnitude by suitable insulation. It is therefore customary to consider this heat loss or gain negligible in comparison to the heat transfer through the walls of the tubes from the hot combustion gases to the water in the boilers. Thus, all the sensible heat lost by the hot gases may be assumed transferred to the steam.

The heart of the heat transfer calculation is the heat flux, which is based on the area of the heating surface and is a function of the temperature difference driving force and the overall heat transfer coefficient, as discussed earlier. This relatively simple equation can be used to estimate heat transfer rates, area requirements, and temperature changes in a number of heat transfer devices. However, problems develop if more exact

calculations are required. The properties of the fluid (viscosity, thermal conductivity, heat capacity, and density) are important parameters in these calculations. Each of these properties, especially viscosity, is temperature dependent. Since a temperature profile, in which the temperature varies from point to point, exists in a flowing stream undergoing heat transfer, a problem arises in the choice of temperature at which the properties should be evaluated. When temperature changes within the stream become large, the difficulty of calculating heat transfer quantities is increased. Because of these effects, the entire subject of heat transfer to fluids with phase change (as in a waste heat boiler) is complex and, in practice, is treated empirically rather than theoretically.[3]

The rigorous design and/or performance evaluation of a waste heat boiler is an involved procedure. Fortunately, several less rigorous methods are available in the literature. One approach that is fairly simple and yet reasonably accurate has been devised by Ganapathy.[9] This method provides a technique for sizing waste heat boilers of the fire-tube type and involves the use of a performance evaluation chart (see Figure 18.4) that is based on fundamental heat transfer equations plus some simplifying assumptions.

Details regarding Ganapathy's methods are now discussed, employing this author's notation. Earlier equations may be combined and applied to the boiler where the temperature outside the tubes (T_c) can be assumed constant due to the
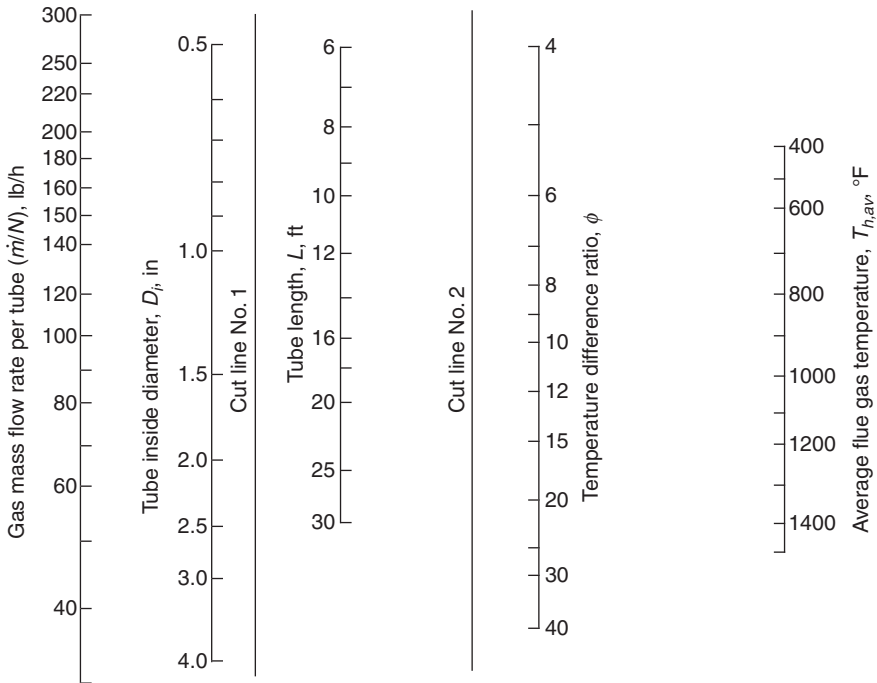

Figure 18.4   Waste heat boiler performance evaluation chart.[9]

water-to-steam phase change. The result is once again,

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h1} - T_{h2}) = U_i A_i \frac{(T_{h1} - T_c) - (T_{h2} - T_c)}{\ln[(T_{h1} - T_c)/(T_{h2} - T_c)]} \tag{18.5}$$

where (employing Ganapathy's notation) T_c is the boiling temperature of the water. Note that $U_i A_i$ was used in Equation (18.5) instead of $U_o A_o$. The two expressions are equal. In this procedure, it is slightly more convenient to use the inside heat transfer area rather than the outside as the basis for U. Since the outside film coefficient, h_o, associated with a boiling liquid is much greater than the inside coefficient for the flue gas, the inside resistance predominates and is responsible for about 95% of the total resistance. The inside overall heat transfer coefficient may therefore be simplified by the approximate relationship

$$U_i = 0.95 h_i \tag{18.6}$$

The total inside heat transfer area for the boiler is given by

$$A_i = \pi D_i N L \tag{18.7}$$

where N is the number of tubes. The Dittus–Boelter equation may be used to solve explicitly for h_i (the inside film coefficient):

$$h_i = 0.023 \frac{k^{0.6} G^{0.8} c_p^{0.4}}{D_i^{0.2} \mu^{0.4}} \tag{18.8}$$

and the gas mass flux, G, may be represented in terms of $\dot{m}_h$

$$G = \frac{4\dot{m}_h}{D_i^2 N} \tag{18.9}$$

Equations (18.5) through (18.9) may be combined to give

$$\ln\left(\frac{T_{h1} - T_c}{T_{h2} - T_c}\right) = \frac{U_i A_i}{\dot{m}_h c_{ph}} = \frac{(C)(L)[F(T)]}{D_i^{0.8}(\dot{m}_h/N)^{0.2}} \tag{18.10}$$

where $F(T) = k^{0.6}/c_{ph}^{0.6} \mu^{0.4}$ and C is a constant equal to 0.0833 when D_i is in feet, or 0.608 when D_i is in inches.

Keeping in mind that this procedure is used for engineering design purposes and is not intended for rigorous analytical calculations, the value of $F(T)$ does not, in practice, vary over a very large range for most hot gas streams. The performance evaluation chart shown in Figure 18.4 is based on Equation (18.10) for typical values of $F(T)$.

The following describes a procedure for the use of the chart as a design tool. It is assumed in this procedure that the inside tube diameter and the number of tubes have

been chosen and the tube length is to be determined. Two later Illustrative Examples further demonstrate the use of Figure 18.4.

1. From the inlet and desired outlet gas temperature, and the water saturation (boiling) temperature, calculate the arithmetic average gas temperature, $T_{h,av}$, and the value of the temperature difference ratio, ϕ, where

$$\phi = \frac{T_{h1} - T_c}{T_{h2} - T_c} \qquad (18.11)$$

Mark those points on the appropriate axes.

2. Draw a straight line connecting these points and extend the line to the left to cut line No. 2. Mark this point B.

3. Mark the value of the flue gas mass flow rate per tube and the value of the inside diameter on the appropriate axes. (For design purposes, a good starting value of $\dot{m}_h/N$ is 80–150 lb/h.)

4. Draw a straight line connecting these points and extend the line to the right to cut line No. 1. Mark this point A.

5. Connect points A and B by a straight line. The intersection of this line with the L axis yields the appropriate tube length.

The chart could be used in similar fashion as a performance evaluation tool. In this case, the outlet gas temperature would be unknown and hence would have to be estimated in order to determine the average gas temperature. In order to avoid a time consuming trial-and-error procedure involving the gas outlet temperature and the average gas temperature, Equation (18.12) may be used to estimate $T_{h,av}$ without too much loss of accuracy:

$$T_{h,av} = 0.5(T_{h1} + T_c) \qquad (18.12)$$

ILLUSTRATIVE EXAMPLE 18.8

What factors should be considered when assessing whether it is feasible to implement energy recovery using a waste heat boiler?

SOLUTION: As discussed above, important factors include:

1. the amount of heat wasted,
2. the fraction of heat that could realistically be recovered,
3. irregularity in availability of heat (scheduling),
4. cost of equipment,
5. cost of energy,
6. incinerator capacity, and
7. nature of waste handled. ∎

ILLUSTRATIVE EXAMPLE 18.9

Discuss the following with respect to waste heat boilers:

1. The effect of fouling on the water/steam side.
2. The effect of fouling on the gas side.
3. Monitoring procedures to follow to account for fouling and/or scaling.

SOLUTION:

1. The exit gas temperature will increase, thus resulting in loss of energy recovery and reduced steam production. Tube wall temperatures can increase significantly, leading to tube failures due to scale formation.
2. This leads to loss of steam output and can increase the gas side pressure drop. However, it does not significantly increase the tube wall temperatures.
3. In monitoring the boiler performance, one should be aware of increases in exit gas temperatures, tube wall temperatures or loss of steam production for the same gas inlet conditions. ∎

ILLUSTRATIVE EXAMPLE 18.10

Using Ganapathy's method, determine the required "length" of a waste heat boiler to be used to cool hot gases (average heat capacity $= 0.279\,\text{Btu/lb} \cdot °\text{F}$) from 2000 to 550°F and generate 30,000 lb/h of steam at 330°F from water at 140°F. The boiler contains 800 1.5-inch ID tubes.

SOLUTION: Calculate the temperature difference ratio, ϕ:

$$\phi = \frac{2000 - 330}{550 - 330} = 7.59$$

The average gas temperature is

$$T_{h,\text{av}} = 0.5(T_{h1} + T_{h2}) = (0.5)(2000 + 550) = 1275°\text{F}$$

From the steam tables, see Appendix:

$$h_{\text{steam}} = 1187.7\,\text{Btu/lb}$$
$$h_{\text{water}} = 107.89\,\text{Btu/lb}$$

Therefore,

$$\dot{Q} = \dot{m}(h_{\text{steam}} - h_{\text{water}}) = 30{,}000(1187.7 - 107.89) = 32.39 \times 10^6\,\text{Btu/h}$$

$$\dot{m}_h = \dot{Q}/c_p(T_{h1} - T_{h2}) = 32.39 \times 10^6/(0.279)(2000 - 550) = 80{,}070\,\text{lb/h}$$

$$\dot{m}_h/N = 80{,}700/800 = 100\,\text{lb/h} \text{ per tube}$$

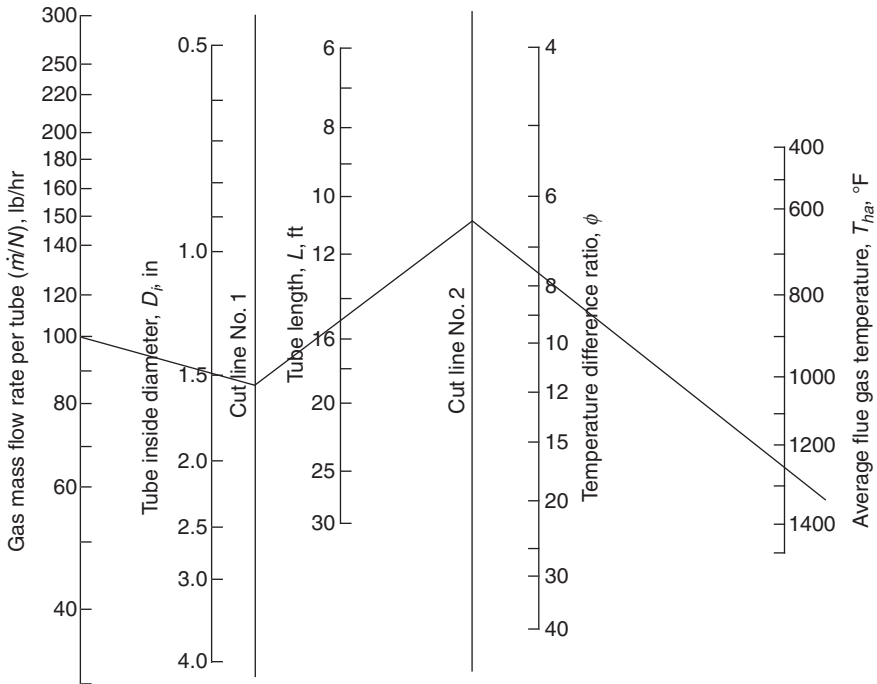From Figure 18.5, the tube length is $L = 15$ ft. ∎

Figure 18.5   Chart for Illustrative Example 18.10.

ILLUSTRATIVE EXAMPLE 18.11

Refer to Illustrative Example 18.10. Provide the details of the calculations and results on Ganapathy's performance chart (Figure 18.4).

SOLUTION: Refer to Figure 18.5 for details.   ■

ILLUSTRATIVE EXAMPLE 18.12

120,000 lb/h of flue gas from an incinerator is to be cooled from 1800 to 500°F in a waste heat boiler. If 2-inch ID tubes and a flow rate of 150 lb/h through each tube is to be used, estimate the required heat transfer area, tube length, heat duty, and water mass flow rate. Water at 200°F is available for the steam generator; saturated steam at 80 psia is needed. The average heat capacity of the flue gas is 0.26 Btu/lb · °F. Use Ganapathy's method to solve the problem.

SOLUTION: From the steam tables (see Appendix), the boiling point of water at 80 psia is 312°F. The temperature difference ratio, ϕ, is

$$\phi = \frac{T_{h1} - T_c}{T_{h2} - T_c} = \frac{1800 - 312}{500 - 312} = 7.9$$

The average gas temperature is

$$T_{h,\text{av}} = (T_{h1} + T_{h2})/2 = (1800 + 500)/2 = 1150°F$$

On the nomograph (see Figure 18.4), connect $(\dot{m}/N) = 150$ with $D_i = 2$ and extend the line to intersect cut line No. 1 at point A. Connect $\phi = 7.9$ with $T_{h,\text{av}} = 1150$ and extend the line to intersect cut line No. 2 at point B. The line connecting points A and B intersects the L line close to 21.5 ft. The number of tubes, N, is

$$N = \frac{\dot{m}}{\dot{m}/N} = 120,000/150 = 800$$

The total heat transfer area (inside) is given by

$$A_i = NLD_i = (800)(21.5)(2/12) = 2866 \, \text{ft}^2$$

Note: This is only one design; several alternatives are possible by changing $\dot{m}/N$ or L. If L has to be limited because of space constraints, the nomograph may be used in reverse to calculate $\dot{m}/N$.

The heat duty is

$$\dot{Q} = \dot{m}c_p(T_{h1} - T_{h2}) = (120,000)(0.26)(1800 - 500) = 4.06 \times 10^7 \, \text{Btu/h}$$

The required water flow rate is obtained from an enthalpy balance. The enthalpy of water at 200°F is 168.1 Btu/lb. The value for the water's enthalpy was obtained from the saturated steam tables. Note that, since pressure has very little effect on the enthalpy of liquids, the enthalpy may be assumed the same as the enthalpy of liquid water at 200°F and its saturation pressure. The enthalpy of steam at 80 psia is 1183.1 Btu/lb. The required water flow is, therefore,

$$\dot{m} = \frac{\dot{Q}}{h_2 - h_1} = \frac{4.06 \times 10^7}{1183.1 - 168.1} = 40,000 \, \text{lb/h} \qquad \blacksquare$$

ILLUSTRATIVE EXAMPLE 18.13

The sulfuric acid dew point for a flue gas discharge from a boiler is 285°F. Is the downstream economizer safer from corrosion if the feed water temperature is 230°F and flue gas is above 600°F or when the feed water is at 275°F and flue gas temperature is 350°F? Provide a qualitative answer.

SOLUTION: In a typical economizer, the water-side film heat transfer coefficient is 50–150 times higher than the gas side film coefficient. Therefore, the tube wall temperature will be close to the water temperature. Therefore, the gas temperature (600°F or 350°F) will not be a factor. Condensation is more likely to occur when the feed water temperature is at 275°F; a slight temperature excursion can lead to problems, e.g., tube failure due to corrosion. $\blacksquare$

CONDENSERS[10,11]

Condensation can be accomplished by increasing pressure or decreasing temperature (removing heat), or both. In practice, condensers operate through extraction of heat. Condensers differ in the means of removing heat and the type of device used. The two different means of condensing are direct contact (or contact), where the cooling medium with vapors and condensate are intimately mixed and combined, and indirect (or surface), where the cooling medium and vapors/condensate are separated by a surface area of some type. The reader is referred to Chapter 12 for additional details.

Contact condensers are simpler, less expensive to install, and require less auxiliary equipment and maintenance. The condensate/coolant from a contact condenser has a volume flow 10 to 20 times that of a surface condenser. This condensate cannot be reused and may pose a waste disposal problem unless the dilution of any pollutant is sufficient to meet regulatory requirements. Some typical contact condensers are shown in Figure 18.6.

Surface condensers form the bulk of the condensers used in industry. Some of the applicable types of surface condensers are: shell and tube, double pipe, spiral plate, flat plate, air-cooled, and various extended surface tubular units. This section focuses on shell and tube condensers (see also Chapter 16) because they are so widely used in industry and have been standardized by the Tubular Exchanger Manufacturers Association (TEMA).

Condensing can be accomplished in either the shell or tubes. The economics, maintenance, and operational ramifications of the allocation of fluids are extremely important, especially if extended surface tubing is being considered. The designer should be given as much latitude as possible in specifying the condenser.
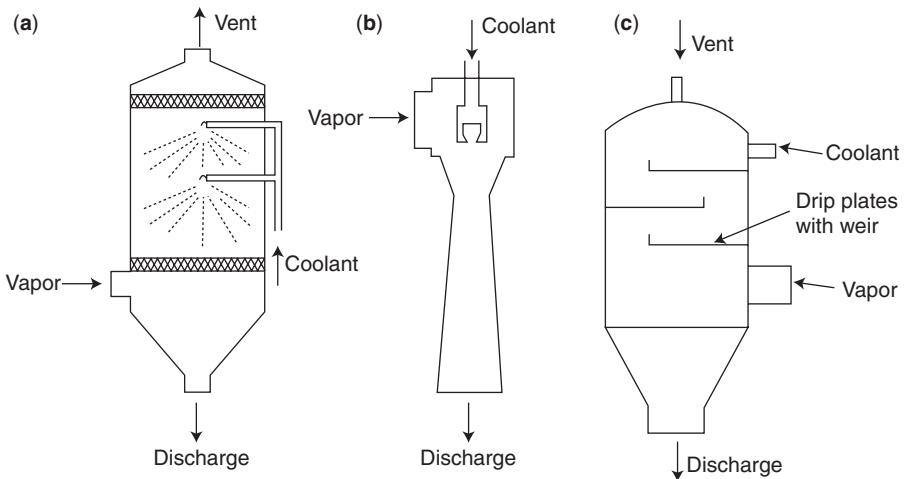

Figure 18.6   Contact condensers: (a) spray; (b) jet; (c) barometric.

Air-cooled condensers consist of a rectangular bank of high finned tubes, a fan, a plenum for even distribution of air to the rectangular face of the tube bank, a header for vapor inlet and condensate outlet, and a steel supporting structure. Fins are usually aluminum, 0.5 to 0.625 inch high, applied to a bare tube by tension winding, or soldering, or cold-extruded from the tube itself. Bimetallic tubes can be used to provide process corrosion resistance on the inside and aluminum extruded fins on the outside. Headers for the tube side frequently contain removable gasketed plugs corresponding to each tube end and are used for access to the individual tube-to-tube sheet joint for maintenance.

Nearly all tubular exchangers employ roller expanding of the tube ends into tube holds drilled into tube sheets as a means of providing a leak-proof seal. TEMA requires that tube holes be of close tolerance and contain two concentric grooves. These grooves give the tube joint greater strength but do not increase sealing ability and may actually decrease it. Research is currently being done to clarify what is the best procedure to follow for a given set of materials, pressures, temperatures, and loadings. Welding or soldering of tubes to tube sheets may be performed for additional leak-tightness or in lieu of roller expanding.

The design of contact condensers involves calculating the quantity of coolant required to condense and subcool the vapor, and proper sizing of the discharge piping and hotwell. The calculation of coolant flow rate is available in the literature.[11] Use of a contact condenser requires consideration of coolant availability, any liquid waste disposal, or treatment facilities. Contact condensers are relatively efficient scrubbers as well as condensers and have the lowest equipment cost. Reputable manufacturers of the various types of contact condensers should be consulted for sizing and layout recommendations.

The heat transfer for a surface condenser is governed by the standard relationship:

$$\dot{Q} = U_o A \Delta T_{lm} \tag{18.13}$$

where $\dot{Q}$ = total heat load, Btu/h

U_o = overall heat-transfer coefficient, Btu/h · ft^2 · °F

ΔT_{lm} = log mean temperature difference, °F

A = surface area, ft^2

The first determination is the heat load. In the simplest case, it is the latent heat to condense plus the subcooling load. If a condensing range exceeding 10–20°F exists, gas cooling or desuperheating and intermediate liquid subcooling must be accounted for. The next step is specifying the coolant flow rate and temperatures to balance the vapor heat load. Temperature crosses where the outlet coolant temperature is higher than the vent and condensate outlet temperature should be avoided. Fluids are usually in counterflow, but in some cases, as with a fluid near its freezing point, parallel flow is used to afford a greater degree of safety from freezing the condenser. Frequently, there is a combination of parallel and counterflow as in a two-pass tube side, single-pass shell tubular condenser. The procedure is to apply a correction (described in Chapter 16)

to the LMTD. Strictly, the correction factor applies only to the subcooling or desuper-heating zones, but it is sometimes applied to condensers.

ILLUSTRATIVE EXAMPLE 18.14

Discuss the effect of a non-condensable gas in a vapor mixture.

SOLUTION: If there is non-condensable gas present or a vapor mixture of miscible and immiscible components with a large condensing range, diffusion of the condensing vapor through another vapor or a gas must be considered. The presence of a noncondensable gas in a vapor stream reduces the film coefficient; the amount of reduction is related to the size of the gas cooling load relative to the total load. The greater the ratio of sensible heat to the total heat, the closer the heat-transfer coefficient approaches that of cooling the gas only. The calculation procedure is essentially that established by Colburn and Hougen.[12] It is a trial-and-error method performed for at least five points along the condensing curve.   ■

ILLUSTRATIVE EXAMPLE 18.15

Provide some typical condensation heat transfer coefficients.

SOLUTION: Some typical condensing coefficients are presented in Table 18.1.[10]   ■

ILLUSTRATIVE EXAMPLE 18.16

Discuss coolant selection.

Table 18.1   Condensation Heat Transfer Coefficient

Fluid	h, Btu/ft$^2 \cdot$ h $\cdot$ °F
Steam	1500
Steam/10% gas	600
Steam/20% gas	400
Steam/40% gas	220
Pure light hydrocarbons	250
Mixed light hydrocarbons	175
Medium hydrocarbons	100
Medium hydrocarbons with steam	125
Pure organic solvents	250

SOLUTION: The choice of a coolant will depend upon the particular plant and the efficiency required of the condenser. The most common coolant is the primary plant coolant, usually cooling tower or river water. It has been shown[13] that the vapor outlet temperature is critical to the efficiency of a condenser. In some instances, use of chilled brine or a boiling refrigerant can achieve a collection efficiency that will be sufficient without additional devices. It is not unusual to specify multiple-stage condensing, where condensers, usually two, connected in series use cooling mediums with successively lower temperatures. For example, a condenser using cooling-tower water can be used prior to a unit using chilled water or brine—thereby achieving maximum efficiency while minimizing use of chilled water.   ∎

QUENCHERS

Hot gases must often be cooled before being discharged to the atmosphere or entering another device(s) which normally is not designed for very high temperature operation ($>500°F$). These gases are usually cooled either by recovering the energy in a waste heat boiler, as discussed in an earlier section of this chapter, or by quenching. Both methods may be used in tandem. For example, a waste heat boiler can reduce an exit gas temperature down to about $500°F$; a water quench can then be used to further reduce the gas temperature to around $200°F$, as well as saturate the gas with water. This secondary cooling and saturation can later eliminate the problem of water evaporation and can also alleviate other potential problems.[3]

Although quenching and the use of a waste heat boiler are the most commonly used methods for gas cooling applications, there are several other techniques for cooling hot gases. All methods may be divided into two categories: *direct-contact* and *indirect-contact* cooling. The direct-contact cooling methods include (a) dilution with ambient air, (b) quenching with water, and (c) contact with high heat capacity solids. Among the indirect contact methods are (d) natural convection and radiation from ductwork, (e) forced-draft heat exchangers, and (f) the aforementioned waste heat boilers.

With the *dilution* method, the hot gaseous effluent is cooled by adding sufficient ambient air that results in a mixture of gases at the desired temperature. The *water quench* method uses the heat of vaporization of water to cool the gases. When water is sprayed into the hot gases under conditions conducive to evaporation, the energy contained in the gases evaporates the water, and this results in a cooling of the gases. The hot exhaust gases may also be quenched using submerged exhaust quenching. This is another technique employed in some applications. In the *solids contact* method, the hot gases are cooled by giving up heat to a bed of ceramic elements. The bed in turn is cooled by incoming air to be used elsewhere in the process. *Natural convection and radiation* occur whenever there is a temperature difference between the gases inside a duct and the atmosphere surrounding it. Cooling hot gases by this method requires only the provision of enough heat transfer area to obtain the desired amount of cooling. In *forced-draft heat exchangers*, the hot gases are cooled by forcing cooling fluid past the barrier separating the fluid from the hot gases. These methods are further discussed below.

Dilution with Ambient Air

The cooling of gases by dilution with ambient air is the simplest method that can be employed. Essentially, it involves the mixing of ambient air with a gas of known volume and temperature to produce a low-temperature mixture that can be admitted to another device. In designing such a system, the amount of ambient air required to provide a gaseous mixture of the desired temperature is first determined.

The quantity of air required for cooling may be calculated directly from an enthalpy balance, i.e., the heat "lost" by the gas is equal to that "gained" by the dilution air. The design of the vessel or duct to accomplish this mixing process is based on a residence time of approximately 0.8–1.5 s (based on the combined flow rate) at the average temperature.

Quenching with Liquids

When a large volume of hot gas is to be cooled, a method other than dilution with ambient air should be used. This is usually the case and the cooling method most often used is liquid quenching.

Cooling by liquid quenching is essentially accomplished by introducing the hot gases into a liquid (usually water) contacting device. When the water evaporates, the energy necessary to vaporize the water is obtained at the expense of the hot gas, resulting in a reduction in the gas temperature. The temperature of the gases discharged from the quencher is at the adiabatic saturation temperature of the gas if the operation is adiabatic and the gas leaves the quencher saturated with water vapor.[3] (A saturated gas contains the maximum water vapor possible at that temperature; any increase in water content will result in condensation.) Simple calculational and graphical procedures are available for estimating the adiabatic saturation temperature of a gas.[3]

There are four types of liquid quenchers that may be employed: spray towers, venturi scrubbers, submerged exhaust tanks, and packed towers (generally a poor choice). The venturi scrubber is actually an air pollution control device but can also be used simultaneously for both particulate removal from, and quenching of, the hot gases.[11]

Contact with High Heat Capacity Solids

Inert ceramics with high heat capacities have, in a few cases, been used to cool hot effluent gases from one unit to temperatures suitable for another process unit. Besides cooling the gases by absorbing heat, this method also allows a certain amount of energy recovery by using the heat absorbed by the solids to subsequently preheat an airstream being introduced to another process unit. In one such system, the solid elements are packed into beds called *regenerative thermal oxidizers* (*RTOs*).[13] These chambers operate on a *regenerative* principle—alternately absorbing, storing, and recycling heat energy. The chambers are thus used in a cyclic fashion. In the first half-cycle, the process air is passed through the previously heated ceramic

elements in a particular chamber, simultaneously cooling the bed and transferring heat into the gas that is then moved on to another unit. At any point during the cycle, the temperature at the bed varies over 1000°F from the cold side (i.e., the inlet side for the ambient air) to the hot side. In the second half-cycle, hot gases are introduced to the hot side of the bed, flow through the bed in the opposite direction of the cold flow in the first half-cycle, the heat lost by the gases is absorbed by the bed that is brought to its maximum average temperature for the start of the next cycle. Typical cycle times range from 2–10 min. Manufacturers claim over 90% thermal energy recovery with this device.

Natural Convection and Radiation

When a hot gas flows through a duct, the duct becomes hot and heats the surrounding air. As the air becomes heated, natural drafts are formed, carrying the heat away from the duct. This phenomenon is *natural convection*. Heat is also discharged from the hot duct to its surroundings by *radiant energy*. Both of these modes of heat transfer have already been discussed in Chapters 12 and 13, respectively.

The rate of heat transfer, $\dot{Q}$, is determined by the amount of heat to be removed from the hot gaseous effluent entering the system. For any particular basic process, the mass flow rate of gaseous effluent and its maximum temperature are fixed. The cooling system must therefore be designed to recover sufficient heat to lower the effluent temperature to the operating temperature of the device to be used. The rate of heat transfer can be calculated by the enthalpy difference of the gas at the inlet and outlet of the cooling system.

Forced-Draft Cooling

As discussed in Chapters 9 and 10, heat transfer by convection is due to fluid motion. Cold fluid adjacent to a hot surface receives heat, which is imparted to the bulk of the fluid by mixing. With *natural convection*, the heated fluid adjacent to the hot surface rises and is replaced by colder fluid. By agitating the fluid, mixing occurs at a much higher rate than with natural currents, and heat is taken away from the hot surface at a much higher rate. In most process applications, this agitation is induced by circulating the fluid at a rapid rate past the hot surface. This method of heat transfer is called *forced convection*. Since forced convection transfers heat much faster than natural convection, most process applications use forced-convection heat exchangers. Whenever possible, heat is exchanged between the hot and cold streams to reduce the heat input to the process. There are, however, many industrial applications where it is not feasible to exchange heat in this fashion, and a cooling fluid such as water or air is used, and the heat removed from the stream by the coolant is transferred to the environment. When water is used, the heat is taken from the process stream in a shell-and-tube cooler, and the heat picked up by the water is transferred to the atmosphere in a cooling tower. When air is used as the cooling medium in either shell-and-tube or fin-tube coolers, the heated air is discharged to the atmosphere and is not recirculated through the

cooler. Because there are so many different types of forced-convection heat exchangers, it is impossible to present a specific design procedure unless a specific type is chosen for discussion.

ILLUSTRATIVE EXAMPLE 18.17

Discuss the equipment employed for most convection–radiation cooling systems.

SOLUTION: For most convection–radiation systems, the only equipment used is sufficient ductwork to provide the required heat transfer area. Unless the temperature of the gases discharged is exceptionally high, or if there are corrosive gases or fumes present, black iron ductwork is generally satisfactory. The temperature of the duct wall, T_w, can be determined for any portion of the ductwork. If T_w proves to be greater than black iron can withstand, either a more heat-resistant material should be used for that portion of the system or a portion of the cooled gas should be recirculated to lower the gas temperature at the cooling system inlet.   ■

ILLUSTRATIVE EXAMPLE 18.18

It is proposed to cool 144,206 lb/h of a hot flue combustion gas by using ambient air at 70°F. Calculate the quantity (mass, mole, and volume basis) of air required to cool the gases from 2050°F to an acceptable temperature of 560°F. Assume an average flue gas heat capacity of 0.3 Btu/lb·°F.

SOLUTION: Under adiabatic conditions,

$$\dot{Q}_{flue} = \dot{Q}_{air}$$

$$\dot{Q}_{flue} = \dot{m}_{flue}\bar{c}_p\,\Delta T = (144{,}206)(0.3)(2050 - 560)$$

$$= 64.5 \times 10^6\,\text{Btu/h}$$

Since $\dot{Q}_{flue} = \dot{Q}_{air}$,

$$64.5 \times 10^6 = \dot{m}_{air}(0.243)(560 - 70)$$

or

$$\dot{m}_{air} = 541{,}700\,\text{lb/h; MW} = 29$$

$$= 18{,}680\,\text{lb mol/h}$$

$$= 7{,}215{,}000\,\text{ft}^3/\text{h}\,(70°\text{F})$$

Note: The combined mass (and volume) flow of the gas is now significantly higher. This can adversely impact on the economics for the downstream equipment.   ■

ILLUSTRATIVE EXAMPLE 18.19

Design the air quench tank in the previous example if a 1.5-s residence time is required.

SOLUTION: The total mass flow rate of the gas is now

$$\dot{m} = 144{,}200 + 541{,}700$$
$$= 685{,}900 \, \text{lb/h}$$

The volumetric flow at 560°F (assuming air) is

$$q = \dot{m}_t RT/P(\text{MW}) = (685{,}900)(0.73)(1020)/(1.0)(29)$$
$$= 1.76 \times 10^7 \, \text{ft}^3/\text{h}$$
$$= 4890 \, \text{f}^3/\text{s}$$

The volume of the tank is, therefore,

$$V_t = (4890)(1.5)$$
$$= 7335 \, \text{ft}^3 \qquad \blacksquare$$

ILLUSTRATIVE EXAMPLE 18.20

Calculate the physical dimensions of the tank in the previous example.

SOLUTION: The physical dimensions of the tank are usually set by minimizing surface (materials) cost. The total surface is given by

$$S = 2(\pi D^2/4) + \pi DH; \quad H = \text{tank height}$$

To minimize the total surface, the derivative of S with respect to D is set equal to zero:

$$\frac{dS}{dD} = \pi D + \pi H = 0$$

or

$$D = H$$

The dimensions of the tank may now be calculated:

$$V_t = (\pi D^2/4)(H) = (\pi D^3/4)$$
$$D = 21.06 \, \text{ft}$$
$$H = 21.06 \, \text{ft}$$

Note: The size of the tank is excessive. This is another reason why air quenching is rarely used. $\qquad \blacksquare$

ILLUSTRATIVE EXAMPLE 18.21

It is proposed to reduce the temperature of 10% of the 144,206 lb/h of flue gas described in Illustrative Example 18.18 to 550°F using a *solids contact* method that operates on a 1-h

cooling cycle. Assume the average heat capacities of the flue gas and solid to be 0.3 and 0.88 Btu/lb · °F, respectively, over the temperature range in question. The initial temperature of the solids is 70°F. Assuming an approach temperature of 40°F, what mass of solid must be provided in order to cool the flue gas to the required temperature during each hour of operation.

SOLUTION: Set up an enthalpy balance for both the flue gas and solid:

$$\Delta h_{\text{flue}} = \Delta h_{\text{solid}}$$

For the flue gas, the enthalpy change for one hour of operation is

$$\Delta h_f = mc_p(T_2 - T_1) = (0.1)(144{,}206)(0.3)(2050 - 550)$$
$$= 6{,}489{,}000 \text{ Btu}$$

For the solids,

$$\Delta h_s = m_s(0.88)(510 - 70)$$

Since $\Delta h_s = \Delta h_f$,

$$m_s = 6{,}489{,}000/(0.88)(440) = 16{,}759 \text{ lb solids} \qquad \blacksquare$$

ILLUSTRATIVE EXAMPLE 18.22

With reference to the previous example, design a radiative heat exchanger to cool the entire combustion gases (MW = 28.27) from 2050 to 180°F. The ambient air temperature is 60°F and an overall heat transfer coefficient for the cooler of 1.5 Btu/h · ft^2 · °F may be assumed to apply.

SOLUTION: Based on the data provided in the previous examples,

$$\dot{Q} = (144{,}206)(0.3)(2050 - 180)$$
$$= 80.9 \times 10^6 \text{ Btu/h}$$

The long-mean temperature difference is given by

$$\Delta T_{\text{lm}} = [(2050 - 60) - (180 - 60)]/\ln[(2050 - 60)/(180 - 60)]$$
$$= 666°F$$

The radiative surface area required is

$$A = \dot{Q}/U\Delta T_{\text{lm}} = 80.9 \times 10^6/(1.5)(666)$$
$$= 80{,}980 \text{ ft}^2$$

The volumetric flow, q_a, at inlet conditions is given by

$$q_a = (144{,}206/28.27)(379)(2050 + 460)/(60 + 460)$$

$$= 9{,}332{,}000 \, \text{ft}^3/\text{h}$$

Duct or pipe velocities of 60 ft/s for this type of application are typical. Assume this value to apply at inlet (2050°F) conditions. The duct area and diameter may now be calculated:

$$A_{\text{duct}} = (9{,}332{,}000)/(3600)(60)$$

$$= 43.2 \, \text{ft}^2$$

$$D = 7.42 \, \text{ft}$$

The length of required heat exchange ducting is then

$$L = A/\pi D = 80{,}980/(\pi)(7.42)$$

$$= 3476 \, \text{ft}$$

The practicality of this method of cooling is questionable. ∎

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