

HEAT EXCHANGERS

Heat exchangers are devices that facilitate the *exchange of heat* between two fluids that are at different temperatures while keeping them from mixing with each other. Heat exchangers are commonly used in practice in a wide range of applications, from heating and air-conditioning systems in a household to chemical processing and power production in large plants. Heat exchangers differ from mixing chambers in that they do not allow the two fluids involved to mix.

Heat transfer in a heat exchanger usually involves *convection* in each fluid and *conduction* through the wall separating the two fluids. In the analysis of heat exchangers, it is convenient to work with an *overall heat transfer coefficient* U that accounts for the contribution of all these effects on heat transfer. The rate of heat transfer between the two fluids at a location in a heat exchanger depends on the magnitude of the temperature difference at that location, which varies along the heat exchanger.

Heat exchangers are manufactured in a variety of types, and thus we start this chapter with the *classification* of heat exchangers. We then discuss the determination of the overall heat transfer coefficient in heat exchangers, and the *log mean temperature difference* (LMTD) for some configurations. We then introduce the *correction factor* F to account for the deviation of the mean temperature difference from the LMTD in complex configurations. Next we discuss the effectiveness–NTU method, which enables us to analyze heat exchangers when the outlet temperatures of the fluids are not known. Finally, we discuss the selection of heat exchangers.

OBJECTIVES

The objectives of this chapter are to:

- Recognize numerous types of heat exchangers and classify them.
- Develop an awareness of fouling on surfaces and determine the overall heat transfer coefficient for a heat exchanger.
- Perform a general energy analysis on heat exchangers.
- Obtain a relation for the logarithmic mean temperature difference for use in the LMTD method, and modify it for different types of heat exchangers using the correction factor.
- Develop relations for effectiveness and analyze heat exchangers when outlet temperatures are not known using the effectiveness–NTU method.
- Know the primary considerations in the selection of heat exchangers.

22-1 ■ TYPES OF HEAT EXCHANGERS

Different heat transfer applications require different types of hardware and different configurations of heat transfer equipment. The attempt to match the heat transfer hardware to the heat transfer requirements within the specified constraints has resulted in numerous types of innovative heat exchanger designs.

The simplest type of heat exchanger consists of two concentric pipes of different diameters, as shown in Fig. 22-1, called the **double-pipe** heat exchanger. One fluid in a double-pipe heat exchanger flows through the smaller pipe while the other fluid flows through the annular space between the two pipes. Two types of flow arrangement are possible in a double-pipe heat exchanger: in **parallel flow**, both the hot and cold fluids enter the heat exchanger at the same end and move in the *same* direction. In **counter flow**, on the other hand, the hot and cold fluids enter the heat exchanger at opposite ends and flow in *opposite* directions.

Another type of heat exchanger, which is specifically designed to realize a large heat transfer surface area per unit volume, is the **compact** heat exchanger. The ratio of the heat transfer surface area of a heat exchanger to its volume is called the *area density* β . A heat exchanger with $\beta > 700 \text{ m}^2/\text{m}^3$ (or $200 \text{ ft}^2/\text{ft}^3$) is classified as being compact. Examples of compact heat exchangers are car radiators ($\beta \approx 1000 \text{ m}^2/\text{m}^3$), glass-ceramic gas turbine heat exchangers ($\beta \approx 6000 \text{ m}^2/\text{m}^3$), the regenerator of a Stirling engine ($\beta \approx 15,000 \text{ m}^2/\text{m}^3$), and the human lung ($\beta \approx 20,000 \text{ m}^2/\text{m}^3$). The flow passages in these compact heat exchangers are usually small and the flow can be considered to be laminar. Compact heat exchangers enable us to achieve

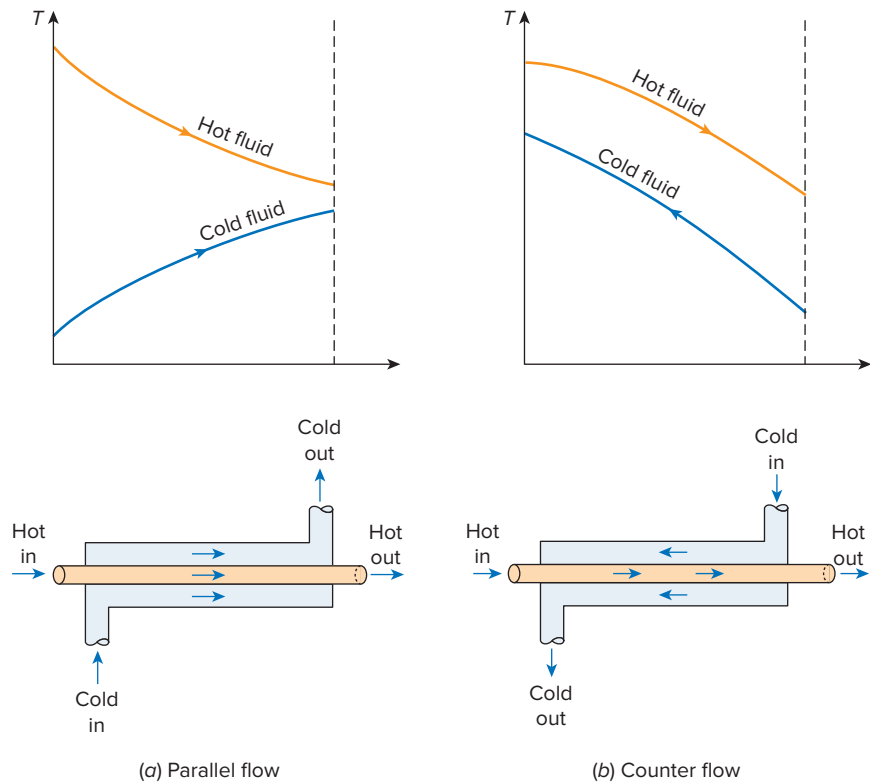


FIGURE 22-1

Different flow regimes and associated temperature profiles in a double-pipe heat exchanger.

high heat transfer rates between two fluids in a small volume, and they are commonly used in applications with strict limitations on the weight and volume of heat exchangers (Fig. 22–2).

The large surface area in compact heat exchangers is obtained by attaching closely spaced *thin plate* or *corrugated fins* to the walls separating the two fluids. Compact heat exchangers are commonly used in gas-to-gas and gas-to-liquid (or liquid-to-gas) heat exchangers to counteract the low heat transfer coefficient associated with gas flow with increased surface area. In a car radiator, which is a water-to-air compact heat exchanger, for example, it is no surprise that fins are attached to the air side of the tube surface.

Another example of compact heat exchangers widely used today in industrial applications such as chemical processing, fuel processing, waste heat recovery, and refrigeration is the **printed circuit heat exchanger** referred to as PCHE (Fig. 22–3). PCHE derives its name from the manufacturing process in which the flat metal plates that form the core of this heat exchanger are chemically etched to a depth of 1 to 3 mm. The chemically etched plates are then stacked together and joined by diffusion bonding. There may be different etching patterns on the plates to make the heat exchanger as a parallel, counter or cross flow heat exchanger. Furthermore, Li et al. (2011) reported that unlike conventional heat exchangers with straight flow channels, the flow channels in PCHE can be made in zigzag, S-shape, or aero foil shape in order to induce flow turbulence and hence enhance the heat transfer coefficient. PCHEs have high surface density typically greater than $2500 \text{ m}^2/\text{m}^3$. The PCHEs are usually made from stainless steel, titanium, copper, nickel, and nickel alloys and can withstand operating pressures up to 500 bar. The major advantages of printed circuit heat exchangers are their wide operating temperature range, which is from -250°C to 900°C , very high heat transfer coefficients, and their size, which is about four to six times smaller and lighter than conventional heat exchangers. One of the major disadvantages of using PCHEs is very high pressure drop and that they require very clean fluid to be passed through them otherwise blockages can occur easily in the fine channel spacing (0.5 to 2 mm).

In compact heat exchangers, the two fluids usually move *perpendicular* to each other, and such flow configuration is called **cross-flow**. The cross-flow is further classified as *unmixed* and *mixed* flow, depending on the flow configuration, as shown in Fig. 22–4. In (a) the cross-flow is said to be *unmixed*

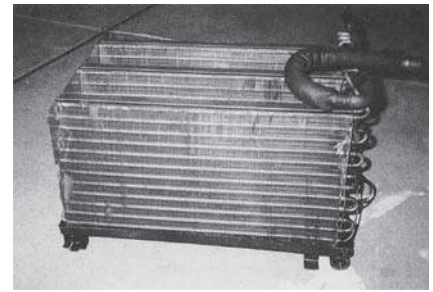


FIGURE 22–2

A gas-to-liquid compact heat exchanger for a residential air-conditioning system.

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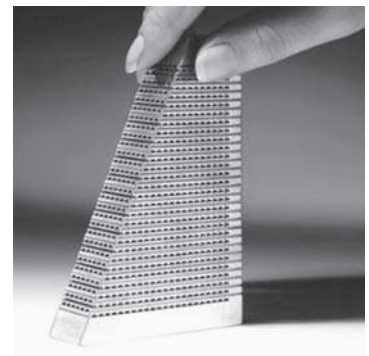


FIGURE 22–3

Photograph of a 'Core' section from a printed circuit heat exchanger (Heatric, a Meggitt Company, Dorset, UK).

Courtesy of Heatric, a Meggitt Company, Dorset UK

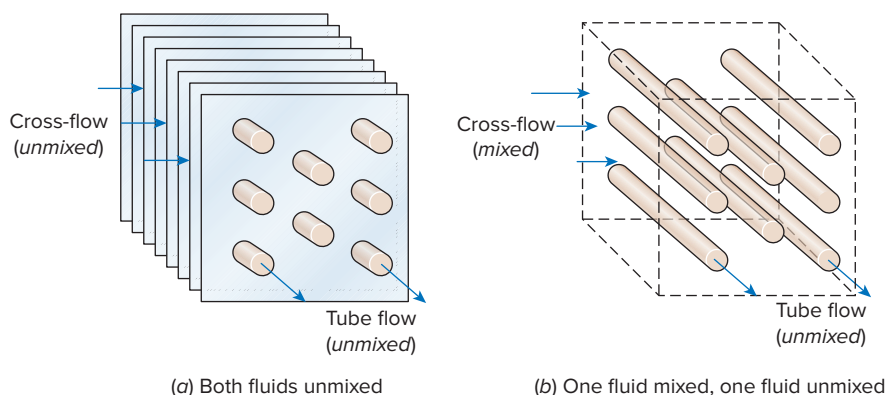
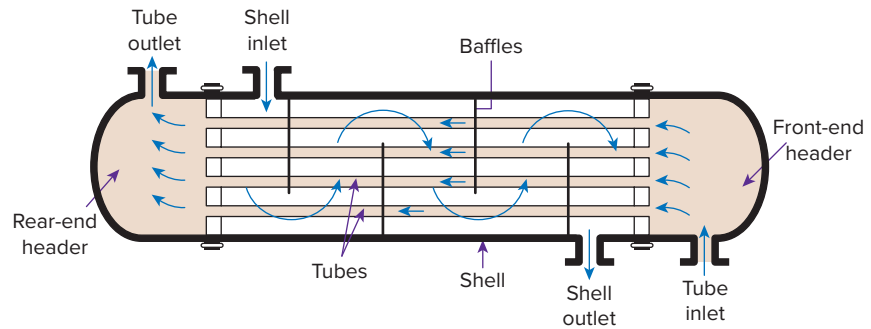


FIGURE 22–4

Different flow configurations in cross-flow heat exchangers.

**FIGURE 22-5**

The schematic of a shell-and-tube heat exchanger (one-shell pass and one-tube pass).

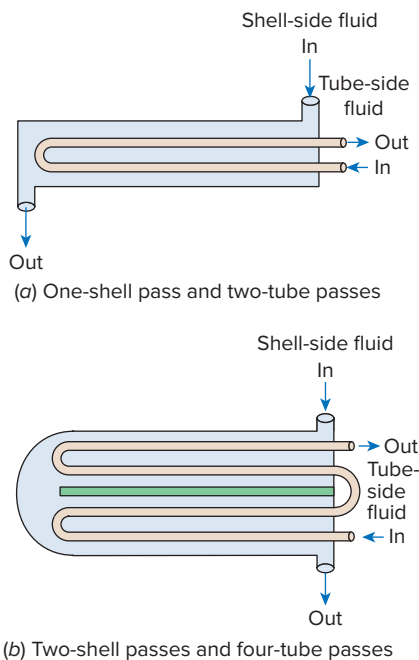
since the plate fins force the fluid to flow through a particular interfin spacing and prevent it from moving in the transverse direction (i.e., parallel to the tubes). The cross-flow in (b) is said to be *mixed* since the fluid now is free to move in the transverse direction. Both fluids are unmixed in a car radiator. The presence of mixing in the fluid can have a significant effect on the heat transfer characteristics of the heat exchanger.

Perhaps the most common type of heat exchanger in industrial applications is the **shell-and-tube** heat exchanger, shown in Fig. 22-5. Shell-and-tube heat exchangers contain a large number of tubes (sometimes several hundred) packed in a shell with their axes parallel to that of the shell. Heat transfer takes place as one fluid flows inside the tubes while the other fluid flows outside the tubes through the shell. *Baffles* are commonly placed in the shell to force the shell-side fluid to flow across the shell to enhance heat transfer and to maintain uniform spacing between the tubes. Despite their widespread use, shell-and-tube heat exchangers are not suitable for use in automotive and aircraft applications because of their relatively large size and weight. Note that the tubes in a shell-and-tube heat exchanger open to some large flow areas called *headers* at both ends of the shell, where the tube-side fluid accumulates before entering the tubes and after leaving them.

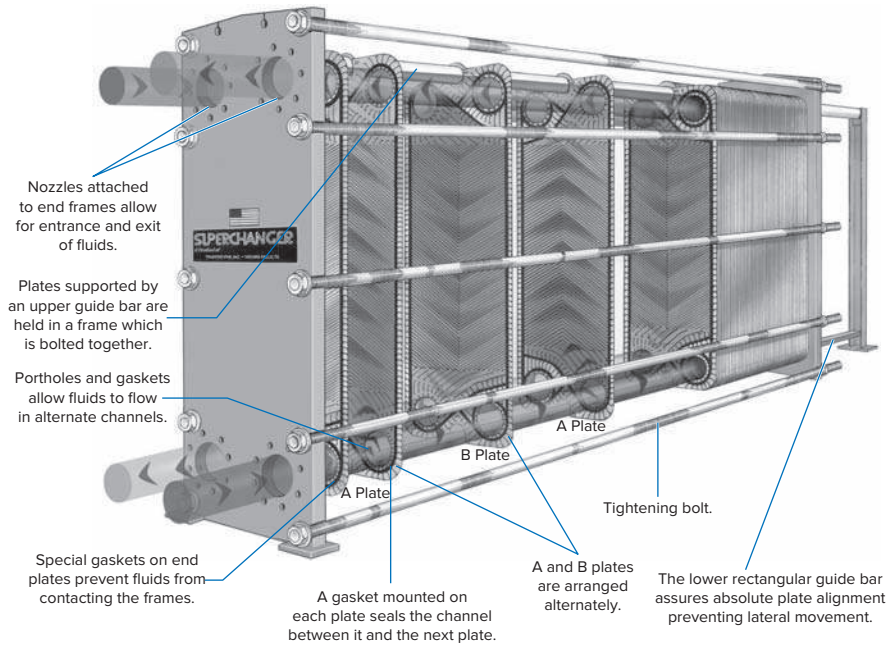
Shell-and-tube heat exchangers are further classified according to the number of shell and tube passes involved. Heat exchangers in which all the tubes make one U-turn in the shell, for example, are called *one-shell-pass and two-tube-passes* heat exchangers. Likewise, a heat exchanger that involves two passes in the shell and four passes in the tubes is called a *two-shell-passes and four-tube-passes* heat exchanger (Fig. 22-6).

An innovative type of heat exchanger that has found widespread use is the **plate and frame** (or just plate) heat exchanger, which consists of a series of plates with corrugated flat flow passages (Fig. 22-7). The hot and cold fluids flow in alternate passages, and thus each cold fluid stream is surrounded by two hot fluid streams, resulting in very effective heat transfer. Also, plate heat exchangers can grow with increasing demand for heat transfer by simply mounting more plates. They are well suited for liquid-to-liquid heat exchange applications, provided that the hot and cold fluid streams are at about the same pressure.

Another type of heat exchanger that involves the alternate passage of the hot and cold fluid streams through the same flow area is the **regenerative**

**FIGURE 22-6**

Multipass flow arrangements in shell-and-tube heat exchangers.

**FIGURE 22-7**

A plate-and-frame liquid-to-liquid heat exchanger.

Courtesy of Tranter PHE, Inc.

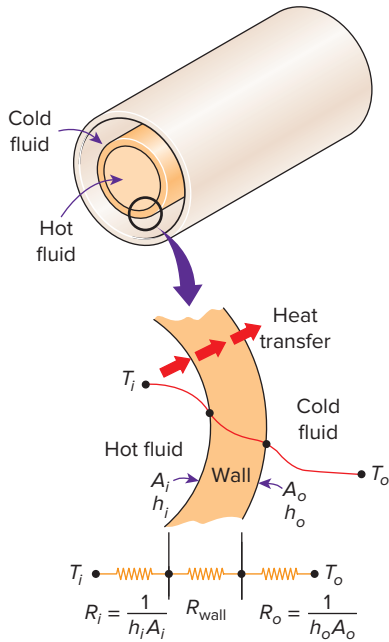
heat exchanger. The *static*-type regenerative heat exchanger is basically a porous mass that has a large heat storage capacity, such as a ceramic wire mesh. Hot and cold fluids flow through this porous mass alternately. Heat is transferred from the hot fluid to the matrix of the regenerator during the flow of the hot fluid, and from the matrix to the cold fluid during the flow of the cold fluid. Thus, the matrix serves as a temporary heat storage medium.

The *dynamic*-type regenerator involves a rotating drum and continuous flow of the hot and cold fluid through different portions of the drum so that any portion of the drum passes periodically through the hot stream, storing heat, and then through the cold stream, rejecting this stored heat. Again the drum serves as the medium to transport the heat from the hot to the cold fluid stream.

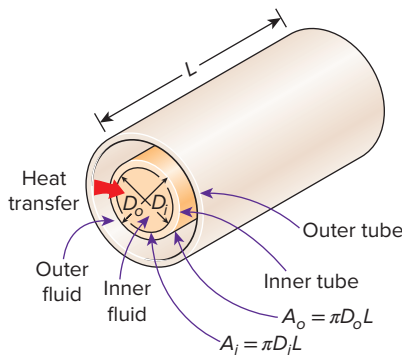
Heat exchangers are often given specific names to reflect the specific application for which they are used. For example, a *condenser* is a heat exchanger in which one of the fluids is cooled and condenses as it flows through the heat exchanger. A *boiler* is another heat exchanger in which one of the fluids absorbs heat and vaporizes. A *space radiator* is a heat exchanger that transfers heat from the hot fluid to the surrounding space by radiation.

22-2 ■ THE OVERALL HEAT TRANSFER COEFFICIENT

A heat exchanger typically involves two flowing fluids separated by a solid wall. Heat is first transferred from the hot fluid to the wall by *convection*, through the wall by *conduction*, and from the wall to the cold fluid again by *convection*. Any radiation effects are usually included in the convection heat transfer coefficients.


FIGURE 22-8

Thermal resistance network associated with heat transfer in a double-pipe heat exchanger.


FIGURE 22-9

The two heat transfer surface areas associated with a double-pipe heat exchanger (for thin tubes, $D_i \approx D_o$ and thus $A_i \approx A_o$).

The thermal resistance network associated with this heat transfer process involves two convection and one conduction resistances, as shown in Fig. 22–8. The analysis is very similar to the material presented in Chap. 17. Here the subscripts i and o represent the inner and outer surfaces of the inner tube. For a double-pipe heat exchanger, the *thermal resistance* of the tube wall is (also see Eq. 17–39)

$$R_{\text{wall}} = \frac{\ln(D_o/D_i)}{2\pi kL} \quad (22-1)$$

where k is the thermal conductivity of the wall material and L is the length of the tube. Then the *total thermal resistance* becomes (also see Eq. 17–43)

$$R = R_{\text{total}} = R_i + R_{\text{wall}} + R_o = \frac{1}{h_i A_i} + \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{1}{h_o A_o} \quad (22-2)$$

The A_i is the area of the inner surface of the wall that separates the two fluids, and A_o is the area of the outer surface of the wall. In other words, A_i and A_o are surface areas of the separating wall wetted by the inner and the outer fluids, respectively. When one fluid flows inside a circular tube and the other outside of it, we have $A_i = \pi D_i L$ and $A_o = \pi D_o L$ (Fig. 22–9).

In the analysis of heat exchangers, it is convenient to combine all the thermal resistances in the path of heat flow from the hot fluid to the cold one into a single resistance R , and to express the rate of heat transfer between the two fluids as

$$\dot{Q} = \frac{\Delta T}{R} = U A_s \Delta T = U_i A_i \Delta T = U_o A_o \Delta T \quad (22-3)$$

where A_s is the surface area and U is the **overall heat transfer coefficient**, whose unit is $\text{W/m}^2\cdot\text{K}$, which is identical to the unit of the ordinary convection coefficient h . Canceling ΔT , Eq. 22–3 reduces to

$$\frac{1}{U A_s} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o} = R = \frac{1}{h_i A_i} + R_{\text{wall}} + \frac{1}{h_o A_o} \quad (22-4)$$

Perhaps you are wondering why we have two overall heat transfer coefficients U_i and U_o for a heat exchanger. The reason is that every heat exchanger has two heat transfer surface areas A_i and A_o , which, in general, are not equal to each other.

Note that $U_i A_i = U_o A_o$, but $U_i \neq U_o$ unless $A_i = A_o$. Therefore, the overall heat transfer coefficient U of a heat exchanger is meaningless unless the area on which it is based is specified. This is especially the case when one side of the tube wall is finned and the other side is not, since the surface area of the finned side is several times that of the unfinned side.

When the wall thickness of the tube is small and the thermal conductivity of the tube material is high, as is usually the case, the thermal resistance of the tube is negligible ($R_{\text{wall}} \approx 0$) and the inner and outer surfaces of the tube are almost identical ($A_i \approx A_o \approx A_s$). Then Eq. 22–4 for the overall heat transfer coefficient simplifies to

$$\frac{1}{U} \approx \frac{1}{h_i} + \frac{1}{h_o} \quad (22-5)$$

where $U \approx U_i \approx U_o$. The individual convection heat transfer coefficients inside and outside the tube, h_i and h_o , are determined using the convection relations discussed in Chap. 19 for internal and external forced convection flows, respectively.

The overall heat transfer coefficient U in Eq. 22-5 is dominated by the *smaller* convection coefficient, since the inverse of a large number is small. When one of the convection coefficients is *much smaller* than the other (say, $h_i \ll h_o$), we have $1/h_i \gg 1/h_o$, and thus $U \approx h_i$. Therefore, the smaller heat transfer coefficient creates a *bottleneck* on the path of heat transfer and seriously impedes heat transfer. This situation arises frequently when one of the fluids is a gas and the other is a liquid. In such cases, fins are commonly used on the gas side to enhance the product UA and thus the heat transfer on that side.

Representative values of the overall heat transfer coefficient U are given in Table 22-1. Note that the overall heat transfer coefficient ranges from about $10 \text{ W/m}^2\cdot\text{K}$ for gas-to-gas heat exchangers to about $10,000 \text{ W/m}^2\cdot\text{K}$ for heat exchangers that involve phase changes. This is not surprising, since gases have very low thermal conductivities, and phase-change processes involve very high heat transfer coefficients.

When the tube is *finned* on one side to enhance heat transfer, the total heat transfer surface area on the finned side becomes

$$A_s = A_{\text{total}} = A_{\text{fin}} + A_{\text{unfinned}} \quad (22-6)$$

TABLE 22-1

Representative values of the overall heat transfer coefficients in heat exchangers

Type of heat exchanger	U , $\text{W/m}^2\cdot\text{K}^*$
Water-to-water	850–1700
Water-to-oil	100–350
Water-to-gasoline or kerosene	300–1000
Water-to-brine	600–1200
Feedwater heaters	1000–8500
Steam-to-light fuel oil	200–400
Steam-to-heavy fuel oil	50–200
Steam condenser	1000–6000
Freon condenser (water cooled)	300–1000
Ammonia condenser (water cooled)	800–1400
Alcohol condensers (water cooled)	250–700
Gas-to-gas	10–40
Gas-to-brine	10–250
Oil-to-oil	50–400
Organic vapors-to-water	700–1000
Organic solvents-to-organic solvents	100–300
Water-to-air in finned tubes (water in tubes)	30–60 [†]
	400–850 [†]
Steam-to-air in finned tubes (steam in tubes)	30–300 [†]
	400–4000 [‡]

*Multiply the listed values by 0.176 to convert them to $\text{Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$.

[†]Based on air-side surface area.

[‡]Based on water- or steam-side surface area.

where A_{fin} is the surface area of the fins and A_{unfinned} is the area of the unfinned portion of the tube surface. For short fins of high thermal conductivity, we can use this total area in the convection resistance relation $R_{\text{conv}} = 1/hA_s$ since the fins in this case will be very nearly isothermal. Otherwise, we should determine the effective surface area from

$$A_s = A_{\text{unfinned}} + \eta_{\text{fin}} A_{\text{fin}} \quad (22-7)$$

where η_{fin} is the fin efficiency which can be obtained from the relationships discussed in Chap. 17. This way, the temperature drop along the fins is accounted for. Note that $\eta_{\text{fin}} = 1$ for isothermal fins, and thus Eq. 22-7 reduces to Eq. 22-6 in that case. Note that for the finned surface, the relevant area (A_i or A_o) in Eq. 22-4 should be calculated from Eq. 22-7.

Fouling Factor

The performance of heat exchangers usually deteriorates with time as a result of accumulation of *deposits* on heat transfer surfaces. The layer of deposits represents *additional resistance* to heat transfer and causes the rate of heat transfer in a heat exchanger to decrease. The net effect of these accumulations on heat transfer is represented by a **fouling factor** R_f , which is a measure of the *thermal resistance* introduced by fouling.

The most common type of fouling is the *precipitation* of solid deposits in a fluid on the heat transfer surfaces. You can observe this type of fouling even in your house. If you check the inner surfaces of your teapot after prolonged use, you will probably notice a layer of calcium-based deposits on the surfaces at which boiling occurs. This is especially the case in areas where the water is hard. The scales of such deposits come off by scratching, and the surfaces can be cleaned of such deposits by chemical treatment. Now imagine those mineral deposits forming on the inner surfaces of fine tubes in a heat exchanger (Fig. 22-10) and the detrimental effect it may have on the flow passage area and the heat transfer. To avoid this potential problem, water in power and process plants is extensively treated and its solid contents are removed before it is allowed to circulate through the system. The solid ash particles in the flue gases accumulating on the surfaces of air preheaters create similar problems.

Another form of fouling, which is common in the chemical process industry, is *corrosion* and other *chemical fouling*. In this case, the surfaces are fouled by the accumulation of the products of chemical reactions on the surfaces. This form of fouling can be avoided by coating metal pipes with glass or using plastic pipes instead of metal ones. Heat exchangers may also be fouled by the growth of algae in warm fluids. This type of fouling is called *biological fouling* and can be prevented by chemical treatment.

In applications where it is likely to occur, fouling should be considered in the design and selection of heat exchangers. In such applications, it may be necessary to select a larger and thus more expensive heat exchanger to ensure that it meets the design heat transfer requirements even after fouling occurs. The periodic cleaning of heat exchangers and the resulting down time are additional penalties associated with fouling.



FIGURE 22-10

Precipitation fouling of ash particles on superheater tubes.

From/it/Steam: its generation and use./xit/Babcock & Wilcox./xit/ Reprinted by permission.

The fouling factor is obviously zero for a new heat exchanger and increases with time as the solid deposits build up on the heat exchanger surface. The fouling factor depends on the *operating temperature* and the *velocity* of the fluids, as well as the length of service. Fouling increases with *increasing temperature* and *decreasing velocity*.

The overall heat transfer coefficient relation Eq. 22–4 or 22–5, for an unfinned double-pipe heat exchanger, is valid for clean surfaces and needs to be modified to account for the effects of fouling on both the inner and the outer surfaces of the tube. For an unfinned double-pipe heat exchanger, it can be expressed as

$$\frac{1}{UA_s} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o} = R = \frac{1}{h_i A_i} + \frac{R_{f,i}}{A_i} + \frac{\ln(D_o/D_i)}{2\pi k L} + \frac{R_{f,o}}{A_o} + \frac{1}{h_o A_o} \quad (22-8)$$

where $R_{f,i}$ and $R_{f,o}$ are the fouling factors at those surfaces.

Representative values of fouling factors are given in Table 22–2. More comprehensive tables of fouling factors are available in handbooks. As you would expect, considerable uncertainty exists in these values, and they should be used as a guide in the selection and evaluation of heat exchangers to account for the effects of anticipated fouling on heat transfer. Note that most fouling factors in the table are of the order of $10^{-4} \text{ m}^2\cdot\text{K}/\text{W}$, which is equivalent to the thermal resistance of a 0.2-mm-thick limestone layer ($k = 2.9 \text{ W}/\text{m}\cdot\text{K}$) per unit surface area. Therefore, in the absence of specific data, we can assume the surfaces to be coated with 0.2 mm of limestone as a starting point to account for the effects of fouling.

TABLE 22–2

Representative fouling factors
(thermal resistance due to fouling
for a unit surface area)

Fluid	$R_f, \text{ m}^2\cdot\text{K}/\text{W}$
Distilled water, sea-water, river water, boiler feedwater:	
Below 50°C	0.0001
Above 50°C	0.0002
Fuel oil	0.0009
Transformer, lubricating or hydraulic oil	0.0002
Quenching oil	0.0007
Vegetable oil	0.0005
Steam (oil-free)	0.0001
Steam (with oil traces)	0.0002
Organic solvent vapors, natural gas	0.0002
Engine exhaust and fuel gases	0.0018
Refrigerants (liquid)	0.0002
Refrigerants (vapor)	0.0004
Ethylene and methylene glycol (antifreeze) and amine solutions	0.00035
Alcohol vapors	0.0001
Air	0.0004

(Source: Tubular Exchange Manufacturers Association.)

EXAMPLE 22–1 Overall Heat Transfer Coefficient of a Heat Exchanger

Hot oil is to be cooled in a double-tube counter-flow heat exchanger. The copper inner tubes have a diameter of 2 cm and negligible thickness. The inner diameter of the outer tube (the shell) is 3 cm. Water flows through the tube at a rate of 0.5 kg/s, and the oil through the shell at a rate of 0.8 kg/s. Taking the average temperatures of the water and the oil to be 45°C and 80°C, respectively, determine the overall heat transfer coefficient of this heat exchanger.

SOLUTION Hot oil is cooled by water in a double-tube counter-flow heat exchanger. The overall heat transfer coefficient is to be determined.

Assumptions 1 The thermal resistance of the inner tube is negligible since the tube material is highly conductive and its thickness is negligible. 2 Both the oil and water flow are fully developed. 3 Properties of the oil and water are constant.

Properties The properties of water at 45°C are (Table A–15)

$$\rho = 990.1 \text{ kg}/\text{m}^3 \quad \text{Pr} = 3.91$$

$$k = 0.637 \text{ W}/\text{m}\cdot\text{K} \quad \nu = \mu/\rho = 0.602 \times 10^{-6} \text{ m}^2/\text{s}$$

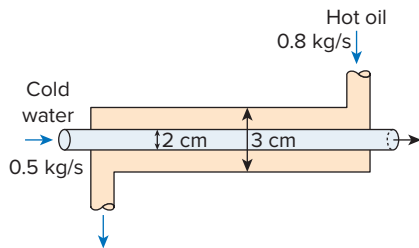


FIGURE 22–11
Schematic for Example 22–1.

The properties of oil at 80°C are (Table A–19)

$$\begin{aligned}\rho &= 852 \text{ kg/m}^3 & \text{Pr} &= 499.3 \\ k &= 0.138 \text{ W/m}\cdot\text{K} & \nu &= 3.794 \times 10^{-5} \text{ m}^2/\text{s}\end{aligned}$$

Analysis The schematic of the heat exchanger is given in Fig. 22–11. The overall heat transfer coefficient U can be determined from Eq. 22–5:

$$\frac{1}{U} \approx \frac{1}{h_i} + \frac{1}{h_o}$$

where h_i and h_o are the convection heat transfer coefficients inside and outside the tube, respectively, which are to be determined using the forced convection relations.

The hydraulic diameter for a circular tube is the diameter of the tube itself, $D_h = D = 0.02 \text{ m}$. The average velocity of water in the tube and the Reynolds number are

$$V = \frac{\dot{m}}{\rho A_c} = \frac{\dot{m}}{\rho (\frac{1}{4} \pi D^2)} = \frac{0.5 \text{ kg/s}}{(990.1 \text{ kg/m}^3) [\frac{1}{4} \pi (0.02 \text{ m})^2]} = 1.61 \text{ m/s}$$

and

$$\text{Re} = \frac{VD}{\nu} = \frac{(1.61 \text{ m/s})(0.02 \text{ m})}{0.602 \times 10^{-6} \text{ m}^2/\text{s}} = 53,490$$

which is greater than 10,000. Therefore, the flow of water is turbulent. Assuming the flow to be fully developed, the Nusselt number can be determined from

$$\text{Nu} = \frac{hD}{k} = 0.023 \text{ Re}^{0.8} \text{Pr}^{0.4} = 0.023(53,490)^{0.8} (3.91)^{0.4} = 240.6$$

Then,

$$h = \frac{k}{D} \text{Nu} = \frac{0.637 \text{ W/m}\cdot\text{K}}{0.02 \text{ m}} (240.6) = 7663 \text{ W/m}^2\cdot\text{K}$$

Now we repeat the analysis above for oil. The properties of oil at 80°C are

$$\begin{aligned}\rho &= 852 \text{ kg/m}^3 & \nu &= 37.5 \times 10^{-6} \text{ m}^2/\text{s} \\ k &= 0.138 \text{ W/m}\cdot\text{K} & \text{Pr} &= 490\end{aligned}$$

The hydraulic diameter for the annular space is

$$D_h = D_o - D_i = 0.03 - 0.02 = 0.01 \text{ m}$$

The average velocity and the Reynolds number in this case are

$$V = \frac{\dot{m}}{\rho A_c} = \frac{\dot{m}}{\rho [\frac{1}{4} \pi (D_o^2 - D_i^2)]} = \frac{0.8 \text{ kg/s}}{(852 \text{ kg/m}^3) [\frac{1}{4} \pi (0.03^2 - 0.02^2)] \text{ m}^2} = 2.39 \text{ m/s}$$

and

$$\text{Re} = \frac{VD}{\nu} = \frac{(2.39 \text{ m/s})(0.01 \text{ m})}{3.794 \times 10^{-5} \text{ m}^2/\text{s}} = 630$$

which is less than 2300. Therefore, the flow of oil is laminar. Assuming fully developed flow, the Nusselt number on the tube side of the annular

space Nu_i corresponding to $D_i/D_o = 0.02/0.03 = 0.667$ can be determined from Table 22–3 by interpolation to be

$$Nu = 5.45$$

and

$$h_o = \frac{k}{D_h} Nu = \frac{0.138 \text{ W/m}\cdot\text{K}}{0.01 \text{ m}} (5.45) = 75.2 \text{ W/m}^2\cdot\text{K}$$

Then the overall heat transfer coefficient for this heat exchanger becomes

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{7663 \text{ W/m}^2\cdot\text{K}} + \frac{1}{75.2 \text{ W/m}^2\cdot\text{K}}} = 74.5 \text{ W/m}^2\cdot\text{K}$$

Discussion Note that $U \approx h_o$ in this case, since $h_i \gg h_o$. This confirms our earlier statement that the overall heat transfer coefficient in a heat exchanger is dominated by the smaller heat transfer coefficient when the difference between the two values is large.

To improve the overall heat transfer coefficient and thus the heat transfer in this heat exchanger, we must use some enhancement techniques on the oil side, such as a finned surface.

TABLE 22–3

Nusselt number for fully developed laminar flow in a circular annulus with one surface insulated and the other isothermal (Kays and Perkins, 1972)

D_i/D_o	Nu_i	Nu_o
0.00	—	3.66
0.05	17.46	4.06
0.10	11.56	4.11
0.25	7.37	4.23
0.50	5.74	4.43
1.00	4.86	4.86

EXAMPLE 22–2 Effect of Fouling on the Overall Heat Transfer Coefficient

A double-pipe (shell-and-tube) heat exchanger is constructed of a stainless steel ($k = 15.1 \text{ W/m}\cdot\text{K}$) inner tube of inner diameter $D_i = 1.5 \text{ cm}$ and outer diameter $D_o = 1.9 \text{ cm}$ and an outer shell of inner diameter 3.2 cm . The convection heat transfer coefficient is given to be $h_i = 800 \text{ W/m}^2\cdot\text{K}$ on the inner surface of the tube and $h_o = 1200 \text{ W/m}^2\cdot\text{K}$ on the outer surface. For a fouling factor of $R_{f,i} = 0.0004 \text{ m}^2\cdot\text{K/W}$ on the tube side and $R_{f,o} = 0.0001 \text{ m}^2\cdot\text{K/W}$ on the shell side, determine (a) the thermal resistance of the heat exchanger per unit length and (b) the overall heat transfer coefficients U_i and U_o based on the inner and outer surface areas of the tube, respectively.

SOLUTION The heat transfer coefficients and the fouling factors on the tube and shell sides of a heat exchanger are given. The thermal resistance and the overall heat transfer coefficients based on the inner and outer areas are to be determined.

Assumptions The heat transfer coefficients and the fouling factors are constant and uniform.

Analysis (a) The schematic of the heat exchanger is given in Fig. 22–12. The thermal resistance for an unfinned shell-and-tube heat exchanger with fouling on both heat transfer surfaces is given by Eq. 22–8 as

$$R = \frac{1}{UA_s} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o} = \frac{1}{h_i A_i} + \frac{R_{f,i}}{A_i} + \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{R_{f,o}}{A_o} + \frac{1}{h_o A_o}$$

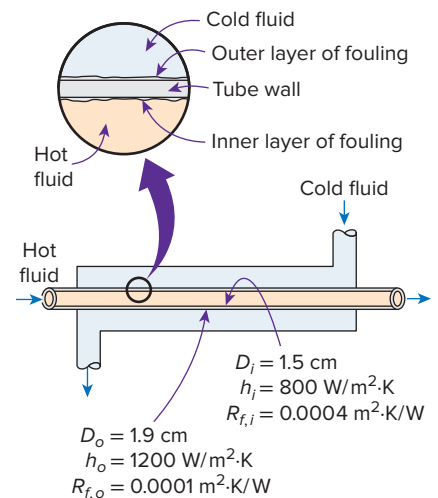


FIGURE 22–12

Schematic for Example 22–2.

where

$$A_i = \pi D_i L = \pi(0.015 \text{ m})(1 \text{ m}) = 0.0471 \text{ m}^2$$

$$A_o = \pi D_o L = \pi(0.019 \text{ m})(1 \text{ m}) = 0.0597 \text{ m}^2$$

Substituting, the total thermal resistance is determined to be

$$\begin{aligned} R &= \frac{1}{(800 \text{ W/m}^2\cdot\text{K})(0.0471 \text{ m}^2)} + \frac{0.0004 \text{ m}^2\cdot\text{K/W}}{0.0471 \text{ m}^2} \\ &\quad + \frac{\ln(0.019/0.015)}{2\pi(15.1 \text{ W/m}\cdot\text{K})(1 \text{ m})} \\ &\quad + \frac{0.0001 \text{ m}^2\cdot\text{K/W}}{0.0597 \text{ m}^2} + \frac{1}{(1200 \text{ W/m}^2\cdot\text{K})(0.0597 \text{ m}^2)} \\ &= (0.02654 + 0.00849 + 0.0025 + 0.00168 + 0.01396) \text{ K/W} \\ &= \mathbf{0.0532^\circ\text{C/W}} \end{aligned}$$

Note that about 19 percent of the total thermal resistance in this case is due to fouling and about 5 percent of it is due to the steel tube separating the two fluids. The rest (76 percent) is due to the convection resistances.

(b) Knowing the total thermal resistance and the heat transfer surface areas, the overall heat transfer coefficients based on the inner and outer surfaces of the tube are

$$U_i = \frac{1}{RA_i} = \frac{1}{(0.0532 \text{ K/W})(0.0471 \text{ m}^2)} = \mathbf{399 \text{ W/m}^2\cdot\text{K}}$$

and

$$U_o = \frac{1}{RA_o} = \frac{1}{(0.0532 \text{ K/W})(0.0597 \text{ m}^2)} = \mathbf{315 \text{ W/m}^2\cdot\text{K}}$$

Discussion Note that the two overall heat transfer coefficients differ significantly (by 27 percent) in this case because of the considerable difference between the heat transfer surface areas on the inner and the outer sides of the tube. For tubes of negligible thickness, the difference between the two overall heat transfer coefficients would be negligible.

22-3 ■ ANALYSIS OF HEAT EXCHANGERS

Heat exchangers are commonly used in practice, and an engineer often finds himself or herself in a position to *select a heat exchanger* that will achieve a *specified temperature change* in a fluid stream of known mass flow rate, or to *predict the outlet temperatures* of the hot and cold fluid streams in a *specified heat exchanger*.

In upcoming sections, we discuss the two methods used in the analysis of heat exchangers. Of these, the *log mean temperature difference* (or LMTD) method is best suited for the first task and the *effectiveness–NTU* (the number of transfer units) method for the second task. But first we present some general considerations.

Heat exchangers usually operate for long periods of time with no change in their operating conditions. Therefore, they can be modeled as *steady-flow* devices. As such, the mass flow rate of each fluid remains constant, and the fluid properties such as temperature and velocity at any inlet or outlet remain the same. Also, the fluid streams experience little or no change in their velocities and elevations, and thus the kinetic and potential energy changes are negligible. The specific heat of a fluid, in general, changes with temperature. But, in a specified temperature range, it can be treated as a constant at some average value with little loss in accuracy. Axial heat conduction along the tube is usually insignificant and can be considered negligible. Finally, the outer surface of the heat exchanger is assumed to be *perfectly insulated*, so that there is no heat loss to the surrounding medium, and any heat transfer occurs between the two fluids only.

The idealizations stated above are closely approximated in practice, and they greatly simplify the analysis of a heat exchanger with little sacrifice from accuracy. Therefore, they are commonly used. Under these assumptions, the *first law of thermodynamics* requires that the rate of heat transfer from the hot fluid be equal to the rate of heat transfer to the cold one. That is

$$\dot{Q} = \dot{m}_c c_{pc} (T_{c, \text{out}} - T_{c, \text{in}}) \quad (22-9)$$

and

$$\dot{Q} = \dot{m}_h c_{ph} (T_{h, \text{in}} - T_{h, \text{out}}) \quad (22-10)$$

where the subscripts *c* and *h* stand for *cold* and *hot* fluids, respectively, and

$\dot{m}_c, \dot{m}_h$ = mass flow rates

c_{pc}, c_{ph} = specific heats

$T_{c, \text{out}}, T_{h, \text{out}}$ = outlet temperatures

$T_{c, \text{in}}, T_{h, \text{in}}$ = inlet temperatures

Note that the heat transfer rate $\dot{Q}$ is taken to be a positive quantity, and its direction is understood to be from the hot fluid to the cold one in accordance with the second law of thermodynamics.

In heat exchanger analysis, it is often convenient to combine the product of the mass flow rate and the specific heat of a fluid into a single quantity. This quantity is called the **heat capacity rate** and is defined for the hot and cold fluid streams as

$$C_h = \dot{m}_h c_{ph} \quad \text{and} \quad C_c = \dot{m}_c c_{pc} \quad (22-11)$$

The heat capacity rate of a fluid stream represents the rate of heat transfer needed to change the temperature of the fluid stream by 1°C as it flows through a heat exchanger. Note that in a heat exchanger, the fluid with a *large* heat capacity rate experiences a *small* temperature change, and the fluid with a *small* heat capacity rate experiences a *large* temperature change. Therefore, *doubling* the mass flow rate of a fluid while leaving everything else unchanged will *halve* the temperature change of that fluid.

With the definition of the heat capacity rate above, Eqs. 22-9 and 22-10 can also be expressed as

$$\dot{Q} = C_c (T_{c, \text{out}} - T_{c, \text{in}}) \quad (22-12)$$

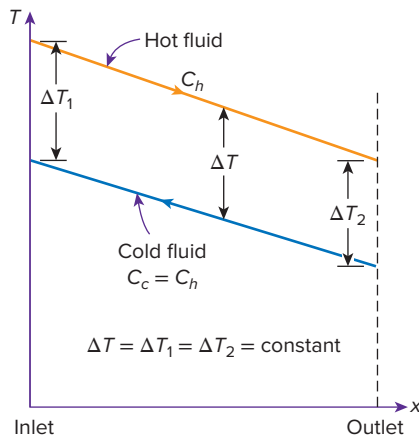
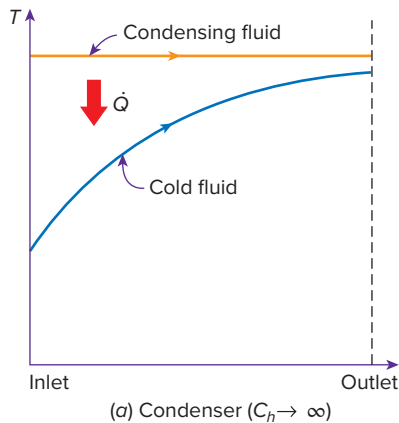
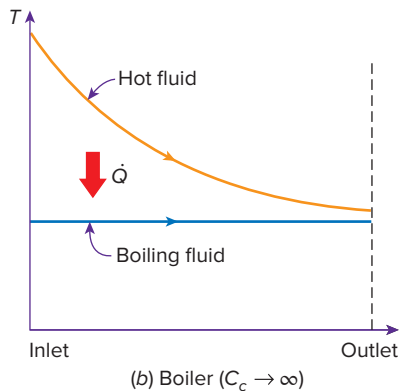


FIGURE 22-13

Two fluid streams that have the same capacity rates experience the same temperature change in a well-insulated heat exchanger.



(a) Condenser ($C_h \rightarrow \infty$)



(b) Boiler ($C_c \rightarrow \infty$)

FIGURE 22-14

Variation of fluid temperatures in a heat exchanger when one of the fluids condenses or boils.

and

$$\dot{Q} = C_h(T_{h,\text{in}} - T_{h,\text{out}}) \quad (22-13)$$

That is, the heat transfer rate in a heat exchanger is equal to the heat capacity rate of either fluid multiplied by the temperature change of that fluid. Note that *the only time the temperature rise of a cold fluid is equal to the temperature drop of the hot fluid is when the heat capacity rates of the two fluids are equal to each other* (Fig. 22-13).

Two special types of heat exchangers commonly used in practice are *condensers* and *boilers*. One of the fluids in a condenser or a boiler undergoes a phase-change process, and the rate of heat transfer is expressed as

$$\dot{Q} = \dot{m}h_{fg} \quad (22-14)$$

where $\dot{m}$ is the rate of evaporation or condensation of the fluid and h_{fg} is the enthalpy of vaporization of the fluid at the specified temperature or pressure.

An ordinary fluid absorbs or releases a large amount of heat essentially at constant temperature during a phase-change process, as shown in Fig. 22-14. The heat capacity rate of a fluid during a phase-change process must approach infinity since the temperature change is practically zero. That is, $C = \dot{m}c_p \rightarrow \infty$ when $\Delta T \rightarrow 0$, so that the heat transfer rate $\dot{Q} = \dot{m}c_p \Delta T$ is a finite quantity. Therefore, in heat exchanger analysis, a condensing or boiling fluid is conveniently modeled as a fluid whose heat capacity rate is *infinity*.

The rate of heat transfer in a heat exchanger can also be expressed in an analogous manner to Newton's law of cooling as

$$\dot{Q} = UA_s \Delta T_m \quad (22-15)$$

where U is the overall heat transfer coefficient, A_s is the heat transfer surface area, and ΔT_m is an appropriate mean temperature difference between the two fluids. Here the surface area A_s can be determined precisely using the dimensions of the heat exchanger. However, the overall heat transfer coefficient U and the temperature difference ΔT between the hot and cold fluids, in general, may vary along the heat exchanger.

The average value of the overall heat transfer coefficient can be determined as described in the preceding section by using the average convection coefficients for each fluid. It turns out that the appropriate form of the average temperature difference between the two fluids is *logarithmic* in nature, and its determination is presented in Sec. 22-4. It should be noted that the average temperature difference ΔT_m is dependent on the heat exchanger flow arrangement and its type of construction.

22-4 ■ THE LOG MEAN TEMPERATURE DIFFERENCE METHOD

Earlier, we mentioned that the temperature difference between the hot and cold fluids varies along the heat exchanger, and it is convenient to have a *mean temperature difference* ΔT_m for use in the relation $\dot{Q} = UA_s \Delta T_m$.

In order to develop a relation for the equivalent average temperature difference between the two fluids, consider the *parallel-flow double-pipe*

heat exchanger shown in Fig. 22–15. Note that the temperature difference ΔT between the hot and cold fluids is large at the inlet of the heat exchanger but decreases exponentially toward the outlet. As you would expect, the temperature of the hot fluid decreases and the temperature of the cold fluid increases along the heat exchanger, but the temperature of the cold fluid can never exceed that of the hot fluid no matter how long the heat exchanger is.

Assuming the outer surface of the heat exchanger to be well insulated so that any heat transfer occurs between the two fluids, and disregarding any changes in kinetic and potential energy, an energy balance on each fluid in a differential section of the heat exchanger can be expressed as

$$\delta\dot{Q} = -\dot{m}_h c_{ph} dT_h \quad (22-16)$$

and

$$\delta\dot{Q} = \dot{m}_c c_{pc} dT_c \quad (22-17)$$

That is, the rate of heat loss from the hot fluid at any section of a heat exchanger is equal to the rate of heat gain by the cold fluid in that section. The temperature change of the hot fluid is a *negative* quantity, and so a *negative sign* is added to Eq. 22–16 to make the heat transfer rate $\dot{Q}$ a positive quantity. Solving the equations above for dT_h and dT_c gives

$$dT_h = -\frac{\delta\dot{Q}}{\dot{m}_h c_{ph}} \quad (22-18)$$

and

$$dT_c = \frac{\delta\dot{Q}}{\dot{m}_c c_{pc}} \quad (22-19)$$

Taking their difference, we get

$$dT_h - dT_c = d(T_h - T_c) = -\delta\dot{Q} \left(\frac{1}{\dot{m}_h c_{ph}} + \frac{1}{\dot{m}_c c_{pc}} \right) \quad (22-20)$$

The rate of heat transfer in the differential section of the heat exchanger can also be expressed as

$$\delta\dot{Q} = U(T_h - T_c) dA_s \quad (22-21)$$

Substituting this equation into Eq. 22–20 and rearranging give

$$\frac{d(T_h - T_c)}{T_h - T_c} = -U dA_s \left(\frac{1}{\dot{m}_h c_{ph}} + \frac{1}{\dot{m}_c c_{pc}} \right) \quad (22-22)$$

Integrating from the inlet of the heat exchanger to its outlet, we obtain

$$\ln \frac{T_{h,out} - T_{c,out}}{T_{h,in} - T_{c,in}} = -UA_s \left(\frac{1}{\dot{m}_h c_{ph}} + \frac{1}{\dot{m}_c c_{pc}} \right) \quad (22-23)$$

Finally, solving Eqs. 22–9 and 22–10 for $\dot{m}_c c_{pc}$ and $\dot{m}_h c_{ph}$ and substituting into Eq. 22–23 give, after some rearrangement,

$$\dot{Q} = UA_s \Delta T_{lm} \quad (22-24)$$

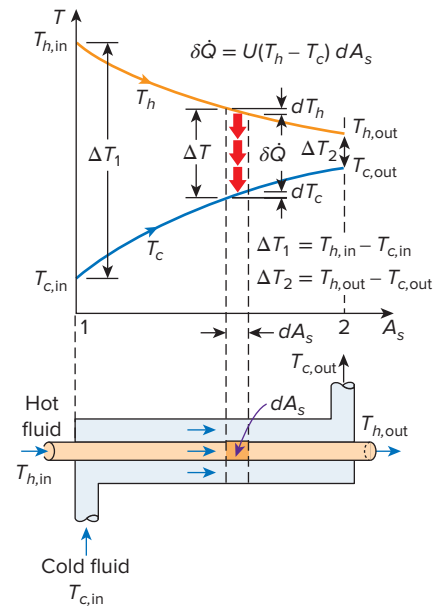


FIGURE 22–15

Variation of the fluid temperatures in a parallel-flow double-pipe heat exchanger.

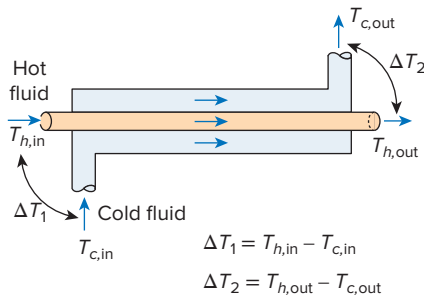
where

$$\Delta T_{\text{lm}} = \frac{\Delta T_1 - \Delta T_2}{\ln (\Delta T_1 / \Delta T_2)} \quad (22-25)$$

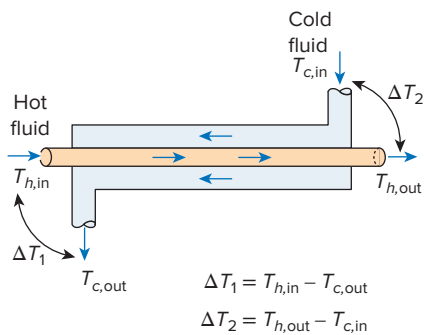
is the **log mean temperature difference**, which is the suitable form of the average temperature difference for use in the analysis of heat exchangers. Here ΔT_1 and ΔT_2 represent the temperature difference between the two fluids at the two ends (inlet and outlet) of the heat exchanger. It makes no difference which end of the heat exchanger is designated as the inlet or the outlet (Fig. 22-16). It should be noted that Eqs. 22-24 and 22-25 are good for any heat exchanger provided that the end point temperature differences are defined properly. For multipass and cross-flow heat exchangers, as will be shown later, the log mean temperature difference should be corrected through a correction factor.

The temperature difference between the two fluids decreases from ΔT_1 at the inlet to ΔT_2 at the outlet. Thus, it is tempting to use the arithmetic mean temperature $\Delta T_{\text{am}} = \frac{1}{2}(\Delta T_1 + \Delta T_2)$ as the average temperature difference. The logarithmic mean temperature difference ΔT_{lm} is obtained by tracing the actual temperature profile of the fluids along the heat exchanger and is an *exact* representation of the *average temperature difference* between the hot and cold fluids. It truly reflects the exponential decay of the local temperature difference.

Note that ΔT_{lm} is always less than ΔT_{am} . Therefore, using ΔT_{am} in calculations instead of ΔT_{lm} will overestimate the rate of heat transfer in a heat exchanger between the two fluids. When ΔT_1 differs from ΔT_2 by no more than 40 percent, the error in using the arithmetic mean temperature difference is less than 1 percent. But the error increases to undesirable levels when ΔT_1 differs from ΔT_2 by greater amounts. Therefore, we should always use the *logarithmic mean temperature difference* when determining the rate of heat transfer in a heat exchanger.



(a) Parallel-flow heat exchangers



(b) Counter-flow heat exchangers

FIGURE 22-16

The ΔT_1 and ΔT_2 expressions in parallel-flow and counter-flow heat exchangers.

Counter-Flow Heat Exchangers

The variation of temperatures of hot and cold fluids in a counter-flow heat exchanger is given in Fig. 22-17. Note that the hot and cold fluids enter the heat exchanger from opposite ends, and the outlet temperature of the *cold fluid* in this case may exceed the outlet temperature of the *hot fluid*. In the limiting case, the cold fluid will be heated to the inlet temperature of the hot fluid. However, the outlet temperature of the cold fluid can *never* exceed the inlet temperature of the hot fluid, since this would be a violation of the second law of thermodynamics.

The relation already given for the log mean temperature difference is developed using a parallel-flow heat exchanger, but we can show by repeating the analysis for a counter-flow heat exchanger that is also applicable to counter-flow heat exchangers. But this time, ΔT_1 and ΔT_2 are expressed as shown in Fig. 22-16.

For specified inlet and outlet temperatures, the log mean temperature difference for a counter-flow heat exchanger is always greater than that for a parallel-flow heat exchanger. That is, $\Delta T_{\text{lm, CF}} > \Delta T_{\text{lm, PF}}$, and thus a smaller surface area (and thus a smaller heat exchanger) is needed to achieve a

specified heat transfer rate in a counter-flow heat exchanger, assuming the same value of the overall heat transfer coefficient. Therefore, it is common practice to use counter-flow arrangements in heat exchangers.

In a counter-flow heat exchanger, the temperature difference between the hot and the cold fluids remains constant along the heat exchanger when the *heat capacity rates* of the two fluids are *equal* (i.e., $\Delta T = \text{constant}$ when $C_h = C_c$ or $\dot{m}_h c_{ph} = \dot{m}_c c_{pc}$). Then we have $\Delta T_1 = \Delta T_2$, and the log mean temperature difference relation gives $\Delta T_{lm} = \frac{0}{0}$, which is indeterminate. It can be shown by the application of l'Hôpital's rule that in this case we have $\Delta T_{lm} = \Delta T_1 = \Delta T_2$, as expected.

A *condenser* or a *boiler* can be considered to be either a parallel- or counter-flow heat exchanger since both approaches give the same result.

Multipass and Cross-Flow Heat Exchangers: Use of a Correction Factor

The log mean temperature difference ΔT_{lm} relation developed earlier is limited to parallel-flow and counter-flow heat exchangers only. Similar relations are also developed for *cross-flow* and *multipass shell-and-tube* heat exchangers, but the resulting expressions are too complicated because of the complex flow conditions.

In such cases, it is convenient to relate the equivalent temperature difference to the log mean temperature difference relation for the counter-flow case as

$$\Delta T_{lm} = F \Delta T_{lm,CF} \quad (22-26)$$

where F is the **correction factor**, which depends on the *geometry* of the heat exchanger and the inlet and outlet temperatures of the hot and cold fluid streams. The $\Delta T_{lm,CF}$ is the log mean temperature difference for the case of a *counter-flow* heat exchanger with the same inlet and outlet temperatures and is determined from Eq. 22-25 by taking $\Delta T_1 = T_{h,in} - T_{c,out}$ and $\Delta T_2 = T_{h,out} - T_{c,in}$ (Fig. 22-18).

The correction factor is less than unity for a cross-flow and multipass shell-and-tube heat exchanger. That is, $F \leq 1$. The limiting value of $F = 1$ corresponds to the counter-flow heat exchanger. Thus, the correction factor F for a heat exchanger is a *measure of deviation of the ΔT_{lm} from the corresponding values for the counter-flow case*.

The correction factor F for common cross-flow and shell-and-tube heat exchanger configurations is given in Fig. 22-19 versus two temperature ratios P and R defined as

$$P = \frac{t_2 - t_1}{T_1 - t_1} \quad (22-27)$$

and

$$R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{(\dot{m}c_p)_{\text{tube side}}}{(\dot{m}c_p)_{\text{shell side}}} \quad (22-28)$$

where the subscripts 1 and 2 represent the *inlet* and *outlet*, respectively. Note that for a shell-and-tube heat exchanger, T and t represent the *shell-* and *tube-side* temperatures, respectively, as shown in the correction factor charts.

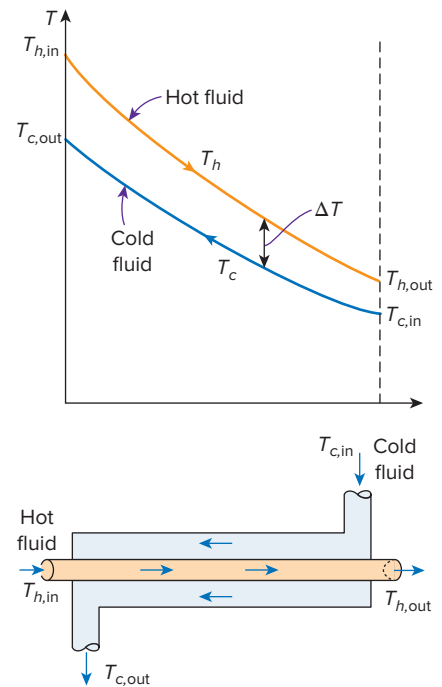
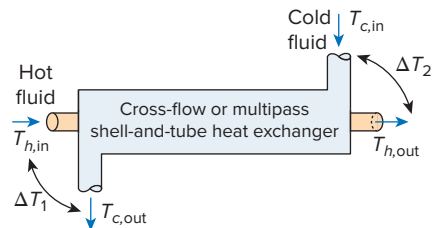


FIGURE 22-17

The variation of the fluid temperatures in a counter-flow double-pipe heat exchanger.



Heat transfer rate:

$$\dot{Q} = UA_s F \Delta T_{lm,CF}$$

where $\Delta T_{lm,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$

$$\Delta T_1 = T_{h,in} - T_{c,out}$$

$$\Delta T_2 = T_{h,out} - T_{c,in}$$

and $F = \dots$ (Fig. 22-19)

FIGURE 22-18

The determination of the heat transfer rate for cross-flow and multipass shell-and-tube heat exchangers using the correction factor.

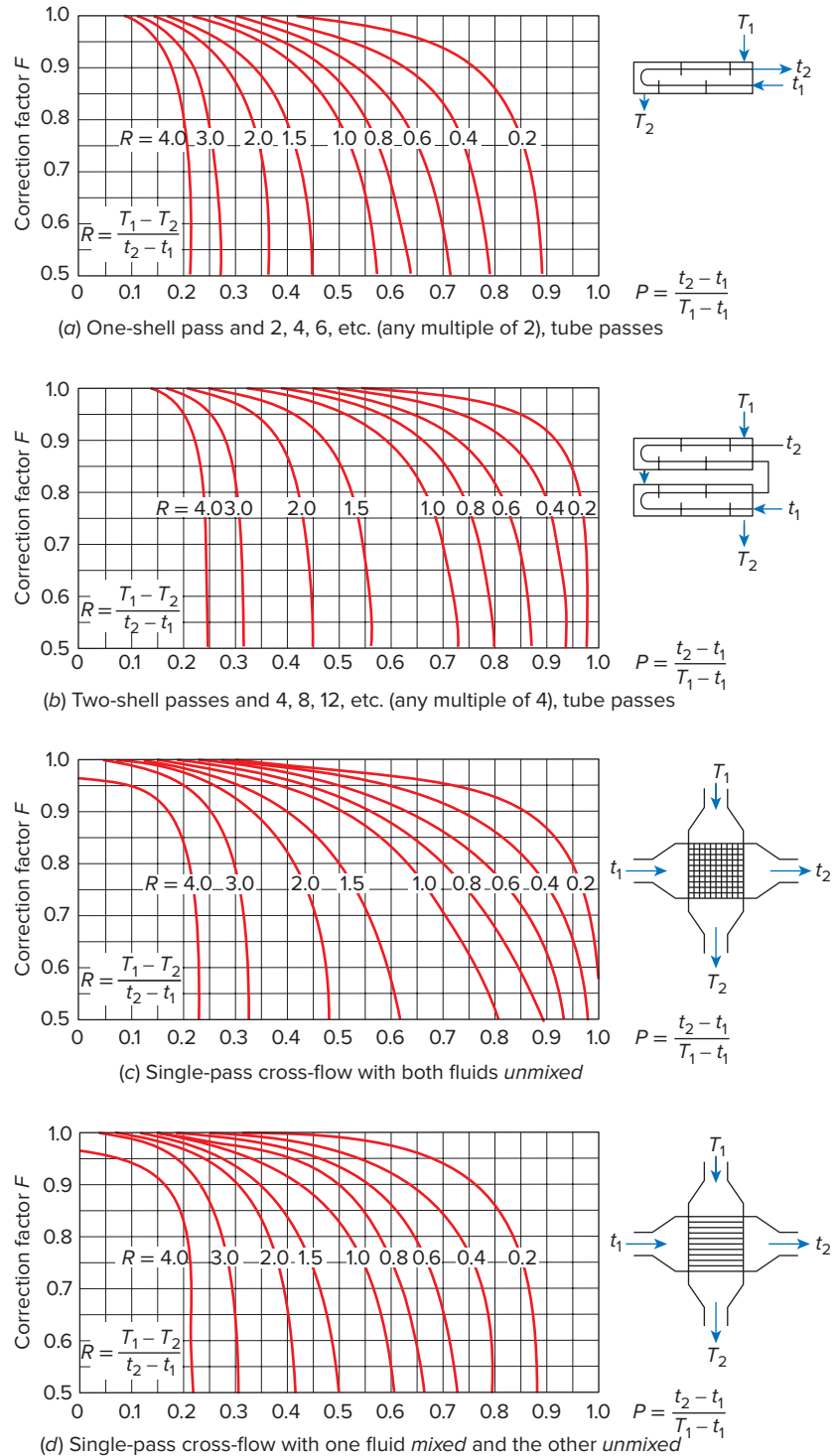


FIGURE 22-19

Correction factor F charts for common shell-and-tube and cross-flow heat exchangers.

From R. A. Bowman, A. C. Mueller, and W. M. Nagle. "Mean Temperature Difference in Design." *Transactions of the ASME* 62, 1940, Figs. 1–2, p. 284–285 and Figs. 7–8, p. 288–289. Reprinted by permission of ASME International.

It makes no difference whether the hot or the cold fluid flows through the shell or the tube. The determination of the correction factor F requires the availability of the inlet and the outlet temperatures for both the cold and hot fluids.

Note that the value of P ranges from 0 to 1. The value of R , on the other hand, ranges from 0 to infinity, with $R = 0$ corresponding to the phase-change (condensation or boiling) on the shell-side and $R \rightarrow \infty$ to phase-change on the tube side. The correction factor is $F = 1$ for both of these limiting cases. Therefore, the correction factor for a condenser or boiler is $F = 1$, regardless of the configuration of the heat exchanger.

EXAMPLE 22–3 The Condensation of Steam in a Condenser

Steam in the condenser of a power plant is to be condensed at a temperature of 30°C with cooling water from a nearby lake, which enters the tubes of the condenser at 14°C and leaves at 22°C . The surface area of the tubes is 45 m^2 , and the overall heat transfer coefficient is $2100\text{ W/m}^2\cdot\text{K}$. Determine the mass flow rate of the cooling water needed and the rate of condensation of the steam in the condenser.

SOLUTION Steam is condensed by cooling water in the condenser of a power plant. The mass flow rate of the cooling water and the rate of condensation are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well insulated so that heat loss to the surroundings is negligible. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties The heat of vaporization of water at 30°C is $h_{fg} = 2431\text{ kJ/kg}$ and the specific heat of cold water at the average temperature of 18°C is $c_p = 4184\text{ J/kg}\cdot\text{K}$ (Table A-15).

Analysis The schematic of the condenser is given in Fig. 22–20. The condenser can be treated as a counter-flow heat exchanger since the temperature of one of the fluids (the steam) remains constant.

The temperature difference between the steam and the cooling water at the two ends of the condenser is

$$\Delta T_1 = T_{h,\text{in}} - T_{c,\text{out}} = (30 - 22)^\circ\text{C} = 8^\circ\text{C}$$

$$\Delta T_2 = T_{h,\text{out}} - T_{c,\text{in}} = (30 - 14)^\circ\text{C} = 16^\circ\text{C}$$

That is, the temperature difference between the two fluids varies from 8°C at one end to 16°C at the other. The proper average temperature difference between the two fluids is the *log mean temperature difference* (not the arithmetic), which is determined from

$$\Delta T_{\text{lm}} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{8 - 16}{\ln(8/16)} = 11.5^\circ\text{C}$$

This is a little less than the arithmetic mean temperature difference of $\frac{1}{2}(8 + 16) = 12^\circ\text{C}$. Then the heat transfer rate in the condenser is determined from

$$\dot{Q} = UA_s \Delta T_{\text{lm}} = (2100\text{ W/m}^2\cdot\text{K})(45\text{ m}^2)(11.5^\circ\text{C}) = 1.087 \times 10^6\text{ W} = 1087\text{ kW}$$

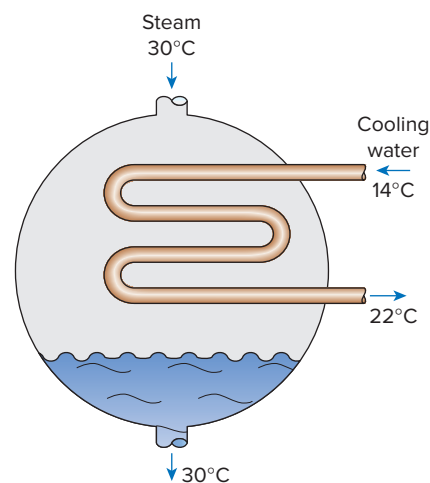


FIGURE 22–20
Schematic for Example 22–3.

Therefore, steam will lose heat at a rate of 1087 kW as it flows through the condenser, and the cooling water will gain practically all of it, since the condenser is well insulated.

The mass flow rate of the cooling water and the rate of the condensation of the steam are determined from $\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{cooling water}} = (\dot{m}h_{fg})_{\text{steam}}$ to be

$$\dot{m}_{\text{cooling water}} = \frac{\dot{Q}}{c_p(T_{\text{out}} - T_{\text{in}})} = \frac{1087 \text{ kJ/s}}{(4.184 \text{ kJ/kg}\cdot\text{K})(22 - 14)^\circ\text{C}} = 32.5 \text{ kg/s}$$

and

$$\dot{m}_{\text{steam}} = \frac{\dot{Q}}{h_{fg}} = \frac{1087 \text{ kJ/s}}{2431 \text{ kJ/kg}} = 0.45 \text{ kg/s}$$

Therefore, we need to circulate about 72 kg of cooling water for each 1 kg of steam condensing to remove the heat released during the condensation process.

EXAMPLE 22-4 Heating Water in a Counter-Flow Heat Exchanger

A counter-flow double-pipe heat exchanger is to heat water from 20°C to 80°C at a rate of 1.2 kg/s. The heating is to be accomplished by geothermal water available at 160°C at a mass flow rate of 2 kg/s. The inner tube is thin-walled and has a diameter of 1.5 cm. If the overall heat transfer coefficient of the heat exchanger is 640 W/m²·K, determine the length of the heat exchanger required to achieve the desired heating.

SOLUTION Water is heated in a counter-flow double-pipe heat exchanger by geothermal water. The required length of the heat exchanger is to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well insulated so that heat loss to the surroundings is negligible. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 There is no fouling. 5 Fluid properties are constant.

Properties We take the specific heats of water and geothermal fluid to be 4.18 and 4.31 kJ/kg·K, respectively.

Analysis The schematic of the heat exchanger is given in Fig. 22-21. The rate of heat transfer in the heat exchanger can be determined from

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} = (1.2 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot\text{K})(80 - 20)^\circ\text{C} = 301 \text{ kW}$$

Noting that all of this heat is supplied by the geothermal water, the outlet temperature of the geothermal water is determined to be

$$\begin{aligned}\dot{Q} &= [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{geothermal}} \longrightarrow T_{\text{out}} = T_{\text{in}} - \frac{\dot{Q}}{\dot{m}c_p} \\ &= 160^\circ\text{C} - \frac{301 \text{ kW}}{(2 \text{ kg/s})(4.31 \text{ kJ/kg}\cdot\text{K})} \\ &= 125^\circ\text{C}\end{aligned}$$

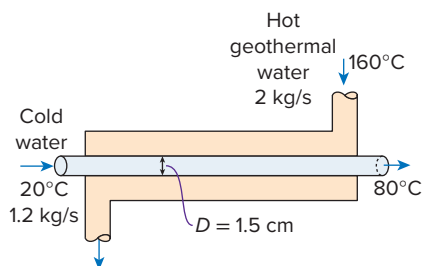


FIGURE 22-21

Schematic for Example 22-4.

Knowing the inlet and outlet temperatures of both fluids, the logarithmic mean temperature difference for this counter-flow heat exchanger becomes

$$\Delta T_1 = T_{h, \text{in}} - T_{c, \text{out}} = (160 - 80)^\circ\text{C} = 80^\circ\text{C}$$

$$\Delta T_2 = T_{h, \text{out}} - T_{c, \text{in}} = (125 - 20)^\circ\text{C} = 105^\circ\text{C}$$

and

$$\Delta T_{\text{lm}} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{80 - 105}{\ln(80/105)} = 91.9^\circ\text{C}$$

Then the surface area of the heat exchanger is determined to be

$$\dot{Q} = UA_s \Delta T_{\text{lm}} \longrightarrow A_s = \frac{\dot{Q}}{U \Delta T_{\text{lm}}} = \frac{301,000 \text{ W}}{(640 \text{ W/m}^2\cdot\text{K})(91.9^\circ\text{C})} = 5.12 \text{ m}^2$$

To provide this much heat transfer surface area, the length of the tube must be

$$A_s = \pi D L \longrightarrow L = \frac{A_s}{\pi D} = \frac{5.12 \text{ m}^2}{\pi(0.015 \text{ m})} = \mathbf{109 \text{ m}}$$

Discussion The inner tube of this counter-flow heat exchanger (and thus the heat exchanger itself) needs to be over 100 m long to achieve the desired heat transfer, which is impractical. In cases like this, we need to use a plate heat exchanger or a multipass shell-and-tube heat exchanger with multiple passes of tube bundles.

EXAMPLE 22–5 Heating of Glycerin in a Multipass Heat Exchanger

A two-shell passes and four-tube passes heat exchanger is used to heat glycerin from 20°C to 50°C by hot water, which enters the thin-walled 2-cm-diameter tubes at 80°C and leaves at 40°C (Fig. 22–22). The total length of the tubes in the heat exchanger is 60 m. The convection heat transfer coefficient is $25 \text{ W/m}^2\cdot\text{K}$ on the glycerin (shell) side and $160 \text{ W/m}^2\cdot\text{K}$ on the water (tube) side. Determine the rate of heat transfer in the heat exchanger (a) before any fouling and (b) after fouling with a fouling factor of $0.0006 \text{ m}^2\cdot\text{K/W}$ occurs on the outer surfaces of the tubes.

SOLUTION Glycerin is heated in a two-shell passes and four-tube passes heat exchanger by hot water. The rate of heat transfer for the cases of fouling and no fouling are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well insulated so that heat loss to the surroundings is negligible. 3 Changes in the kinetic and potential energies of fluid streams are negligible. 4 Heat transfer coefficients and fouling factors are constant and uniform. 5 The thermal resistance of the inner tube is negligible since the tube is thin-walled and highly conductive.

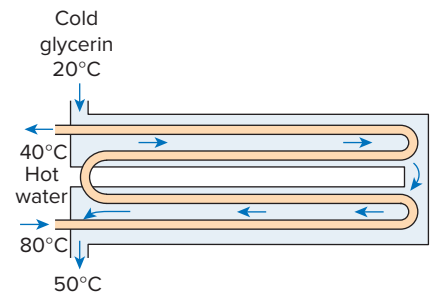


FIGURE 22–22
Schematic for Example 22–5.

Analysis The tubes are said to be thin-walled, and thus it is reasonable to assume the inner and outer surface areas of the tubes to be equal. Then the heat transfer surface area becomes

$$A_s = \pi DL = \pi(0.02 \text{ m})(60 \text{ m}) = 3.77 \text{ m}^2$$

The rate of heat transfer in this heat exchanger can be determined from

$$\dot{Q} = UA_s F \Delta T_{\text{lm, CF}}$$

where F is the correction factor and $\Delta T_{\text{lm, CF}}$ is the log mean temperature difference for the counter-flow arrangement. These two quantities are determined from

$$\Delta T_1 = T_{h, \text{in}} - T_{c, \text{out}} = (80 - 50)^\circ\text{C} = 30^\circ\text{C}$$

$$\Delta T_2 = T_{h, \text{out}} - T_{c, \text{in}} = (40 - 20)^\circ\text{C} = 20^\circ\text{C}$$

$$\Delta T_{\text{lm, CF}} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{30 - 20}{\ln(30/20)} = 24.7^\circ\text{C}$$

and

$$\left. \begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{40 - 80}{20 - 80} = 0.67 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{20 - 50}{40 - 80} = 0.75 \end{aligned} \right\} F = 0.91 \quad (\text{Fig. 22-19b})$$

(a) In the case of no fouling, the overall heat transfer coefficient U is

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{160 \text{ W/m}^2\cdot\text{K}} + \frac{1}{25 \text{ W/m}^2\cdot\text{K}}} = 21.6 \text{ W/m}^2\cdot\text{K}$$

Then the rate of heat transfer becomes

$$\dot{Q} = UA_s F \Delta T_{\text{lm, CF}} = (21.6 \text{ W/m}^2\cdot\text{K})(3.77 \text{ m}^2)(0.91)(24.7^\circ\text{C}) = \mathbf{1830 \text{ W}}$$

(b) When there is fouling on one of the surfaces, we have

$$\begin{aligned} U &= \frac{1}{\frac{1}{h_i} + \frac{1}{h_o} + R_f} = \frac{1}{\frac{1}{160 \text{ W/m}^2\cdot\text{K}} + \frac{1}{25 \text{ W/m}^2\cdot\text{K}} + 0.0006 \text{ m}^2\cdot\text{K/W}} \\ &= 21.3 \text{ W/m}^2\cdot\text{K} \end{aligned}$$

and

$$\dot{Q} = UA_s F \Delta T_{\text{lm, CF}} = (21.3 \text{ W/m}^2\cdot\text{K})(3.77 \text{ m}^2)(0.91)(24.7^\circ\text{C}) = \mathbf{1805 \text{ W}}$$

Discussion Note that the rate of heat transfer decreases as a result of fouling, as expected. The decrease is not dramatic, however, because of the relatively low convection heat transfer coefficients involved.

EXAMPLE 22–6 Cooling of Water in an Automotive Radiator

A test is conducted to determine the overall heat transfer coefficient in an automotive radiator that is a compact cross-flow water-to-air heat exchanger with both fluids (air and water) unmixed (Fig. 22–23). The radiator has 40 tubes of internal diameter 0.5 cm and length 65 cm in a closely spaced plate-finned matrix. Hot water enters the tubes at 90°C at a rate of 0.6 kg/s and leaves at 65°C. Air flows across the radiator through the interfin spaces and is heated from 20°C to 40°C. Determine the overall heat transfer coefficient U_i of this radiator based on the inner surface area of the tubes.

SOLUTION During an experiment involving an automotive radiator, the inlet and exit temperatures of water and air and the mass flow rate of water are measured. The overall heat transfer coefficient based on the inner surface area is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Changes in the kinetic and potential energies of fluid streams are negligible. 3 Fluid properties are constant.

Properties The specific heat of water at the average temperature of $(90 + 65)/2 = 77.5^\circ\text{C}$ is $4.195 \text{ kJ/kg}\cdot\text{K}$ (Table A–15).

Analysis The rate of heat transfer in this radiator from the hot water to the air is determined from an energy balance on water flow:

$$\begin{aligned}\dot{Q} &= [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{water}} = (0.6 \text{ kg/s})(4.195 \text{ kJ/kg}\cdot\text{K})(90 - 65)^\circ\text{C} \\ &= 62.93 \text{ kW}\end{aligned}$$

The tube-side heat transfer area is the total surface area of the tubes, and is determined from

$$A_i = n\pi D_i L = (40)\pi(0.005 \text{ m})(0.65 \text{ m}) = 0.408 \text{ m}^2$$

Knowing the rate of heat transfer and the surface area, the overall heat transfer coefficient can be determined from

$$\dot{Q} = U_i A_i F \Delta T_{\text{lm, CF}} \longrightarrow U_i = \frac{\dot{Q}}{A_i F \Delta T_{\text{lm, CF}}}$$

where F is the correction factor and $\Delta T_{\text{lm, CF}}$ is the log mean temperature difference for the counter-flow arrangement. These two quantities are found to be

$$\Delta T_1 = T_{h, \text{in}} - T_{c, \text{out}} = (90 - 40)^\circ\text{C} = 50^\circ\text{C}$$

$$\Delta T_2 = T_{h, \text{out}} - T_{c, \text{in}} = (65 - 20)^\circ\text{C} = 45^\circ\text{C}$$

$$\Delta T_{\text{lm, CF}} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{50 - 45}{\ln(50/45)} = 47.5^\circ\text{C}$$

and

$$\left. \begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{65 - 90}{20 - 90} = 0.36 \\ R &= \frac{T_2 - T_1}{t_2 - t_1} = \frac{20 - 40}{65 - 90} = 0.80 \end{aligned} \right\} F = 0.97 \quad (\text{Fig. 22–19c})$$

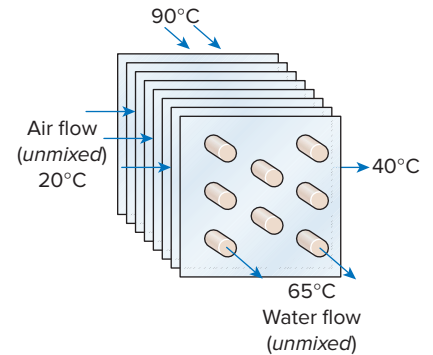


FIGURE 22–23

Schematic for Example 22–6.

Substituting, the overall heat transfer coefficient U_i is determined to be

$$U_i = \frac{\dot{Q}}{A_i F \Delta T_{\text{lm,CF}}} = \frac{62,930 \text{ W}}{(0.408 \text{ m}^2)(0.97)(47.5^\circ\text{C})} = \mathbf{3347 \text{ W/m}^2 \cdot \text{K}}$$

Discussion Note that the overall heat transfer coefficient on the air side will be much lower because of the large surface area involved on that side.

22-5 ■ THE EFFECTIVENESS-NTU METHOD

The log mean temperature difference (LMTD) method discussed in Sec. 22-4 is easy to use in heat exchanger analysis when the inlet and the outlet temperatures of the hot and cold fluids are known or can be determined from an energy balance. Once ΔT_{lm} , the mass flow rates, and the overall heat transfer coefficient are available, the heat transfer surface area of the heat exchanger can be determined from

$$\dot{Q} = UA_s \Delta T_{\text{lm}}$$

Therefore, the LMTD method is very suitable for determining the *size* of a heat exchanger to realize prescribed outlet temperatures when the mass flow rates and the inlet and outlet temperatures of the hot and cold fluids are specified.

With the LMTD method, the task is to *select* a heat exchanger that will meet the prescribed heat transfer requirements. The procedure to be followed by the selection process is

1. Select the type of heat exchanger suitable for the application.
2. Determine any unknown inlet or outlet temperature and the heat transfer rate using an energy balance.
3. Calculate the log mean temperature difference ΔT_{lm} and the correction factor F , if necessary.
4. Obtain (select or calculate) the value of the overall heat transfer coefficient U .
5. Calculate the heat transfer surface area A_s .

The task is completed by selecting a heat exchanger that has a heat transfer surface area equal to or larger than A_s .

A second kind of problem encountered in heat exchanger analysis is the determination of the *heat transfer rate* and the *outlet temperatures* of the hot and cold fluids for prescribed fluid mass flow rates and inlet temperatures when the *type* and *size* of the heat exchanger are specified. The heat transfer surface area of the heat exchanger in this case is known, but the *outlet temperatures* are not. Here the task is to determine the heat transfer performance of a specified heat exchanger or to determine if a heat exchanger available in storage will do the job.

The LMTD method could still be used for this alternative problem, but the procedure would require tedious iterations, and thus it is not practical. In an attempt to eliminate the iterations from the solution of such problems,

Kays and London came up with a method in 1955 called the **effectiveness–NTU method**, which greatly simplified heat exchanger analysis.

This method is based on a dimensionless parameter called the **heat transfer effectiveness** ε , defined as

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{\text{Actual heat transfer rate}}{\text{Maximum possible heat transfer rate}} \quad (22-29)$$

The *actual* heat transfer rate in a heat exchanger can be determined from an energy balance on the hot or cold fluids and can be expressed as

$$\dot{Q} = C_c(T_{c, \text{out}} - T_{c, \text{in}}) = C_h(T_{h, \text{in}} - T_{h, \text{out}}) \quad (22-30)$$

where $C_c = \dot{m}_c c_{pc}$ and $C_h = \dot{m}_h c_{ph}$ are the heat capacity rates of the cold and hot fluids, respectively.

To determine the maximum possible heat transfer rate in a heat exchanger, we first recognize that the *maximum temperature difference* in a heat exchanger is the difference between the *inlet* temperatures of the hot and cold fluids. That is

$$\Delta T_{\max} = T_{h, \text{in}} - T_{c, \text{in}} \quad (22-31)$$

The heat transfer in a heat exchanger will reach its maximum value when (1) the cold fluid is heated to the inlet temperature of the hot fluid or (2) the hot fluid is cooled to the inlet temperature of the cold fluid. These two limiting conditions will not be reached simultaneously unless the heat capacity rates of the hot and cold fluids are identical (i.e., $C_c = C_h$). When $C_c \neq C_h$, which is usually the case, the fluid with the *smaller* heat capacity rate will experience a larger temperature change, and thus it will be the first to experience the maximum temperature, at which point the heat transfer will come to a halt. Therefore, the maximum possible heat transfer rate in a heat exchanger is (Fig. 22–24)

$$\dot{Q}_{\max} = C_{\min}(T_{h, \text{in}} - T_{c, \text{in}}) \quad (22-32)$$

where $C_{\min}$ is the smaller of C_h and C_c . This is further clarified by Example 22–7.

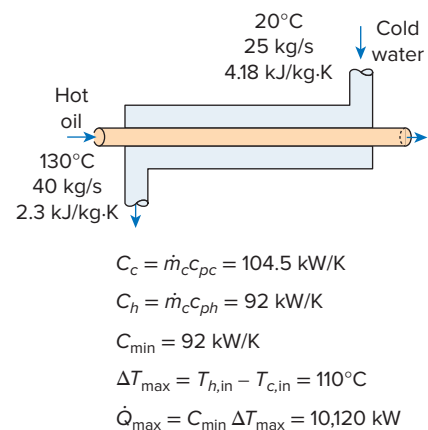


FIGURE 22–24

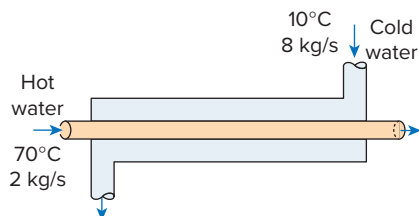
The determination of the maximum rate of heat transfer in a heat exchanger.

EXAMPLE 22–7 Upper Limit for Heat Transfer in a Heat Exchanger

Cold water enters a counter-flow heat exchanger at 10°C at a rate of 8 kg/s , where it is heated by a hot-water stream that enters the heat exchanger at 70°C at a rate of 2 kg/s . Assuming the specific heat of water to remain constant at $c_p = 4.18 \text{ kJ/kg}\cdot\text{K}$, determine the maximum heat transfer rate and the outlet temperatures of the cold- and the hot-water streams for this limiting case.

SOLUTION Cold- and hot-water streams enter a heat exchanger at specified temperatures and flow rates. The maximum rate of heat transfer in the heat exchanger and the outlet temperatures are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well insulated so that heat loss to the surroundings is negligible. 3 Changes

**FIGURE 22–25**

Schematic for Example 22–7.

in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heat of water is given to be $c_p = 4.18 \text{ kJ/kg}\cdot\text{K}$.

Analysis A schematic of the heat exchanger is given in Fig. 22–25. The heat capacity rates of the hot and cold fluids are

$$C_h = \dot{m}_h c_{ph} = (2 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot\text{K}) = 8.36 \text{ kW/K}$$

and

$$C_c = \dot{m}_c c_{pc} = (8 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot\text{K}) = 33.4 \text{ kW/K}$$

Therefore,

$$C_{\min} = C_h = 8.36 \text{ kW/K}$$

which is the smaller of the two heat capacity rates. Then the maximum heat transfer rate is determined from Eq. 22–32 to be

$$\begin{aligned}\dot{Q}_{\max} &= C_{\min}(T_{h,\text{in}} - T_{c,\text{in}}) \\ &= (8.36 \text{ kW/K})(70 - 10)^\circ\text{C} \\ &= \mathbf{502 \text{ kW}}\end{aligned}$$

That is, the maximum possible heat transfer rate in this heat exchanger is 502 kW. This value would be approached in a counter-flow heat exchanger with a *very large* heat transfer surface area.

The maximum temperature difference in this heat exchanger is $\Delta T_{\max} = T_{h,\text{in}} - T_{c,\text{in}} = (70 - 10)^\circ\text{C} = 60^\circ\text{C}$. Therefore, the hot water cannot be cooled by more than 60°C (to 10°C) in this heat exchanger, and the cold water cannot be heated by more than 60°C (to 70°C), no matter what we do. The outlet temperatures of the cold and the hot streams in this limiting case are determined to be

$$\dot{Q} = C_c(T_{c,\text{out}} - T_{c,\text{in}}) \longrightarrow T_{c,\text{out}} = T_{c,\text{in}} + \frac{\dot{Q}}{C_c} = 10^\circ\text{C} + \frac{502 \text{ kW}}{33.4 \text{ kW/K}} = \mathbf{25^\circ\text{C}}$$

$$\dot{Q} = C_h(T_{h,\text{in}} - T_{h,\text{out}}) \longrightarrow T_{h,\text{out}} = T_{h,\text{in}} - \frac{\dot{Q}}{C_h} = 70^\circ\text{C} - \frac{502 \text{ kW}}{8.36 \text{ kW/K}} = \mathbf{10^\circ\text{C}}$$

Discussion Note that the hot water is cooled to the limit of 10°C (the inlet temperature of the cold-water stream), but the cold water is heated to 25°C only when maximum heat transfer occurs in the heat exchanger. This is not surprising, since the mass flow rate of the hot water is only one-fourth that of the cold water, and, as a result, the temperature of the cold water increases by 0.25°C for each 1°C drop in the temperature of the hot water.

You may be tempted to think that the cold water should be heated to 70°C in the limiting case of maximum heat transfer. But this will require the temperature of the hot water to drop to -170°C (below 10°C), which is impossible. Therefore, heat transfer in a heat exchanger reaches its maximum value when the fluid with the smaller heat capacity rate (or the smaller mass flow rate when both fluids have the same specific heat value) experiences the

maximum temperature change. This example explains why we use $C_{\min}$ in the evaluation of $\dot{Q}_{\max}$ instead of $C_{\max}$.

We can show that the hot water will leave at the inlet temperature of the cold water and vice versa in the limiting case of maximum heat transfer when the mass flow rates of the hot- and cold-water streams are identical (Fig. 22–26). We can also show that the outlet temperature of the cold water will reach the 70°C limit when the mass flow rate of the hot water is greater than that of the cold water.

The determination of $\dot{Q}_{\max}$ requires the availability of the *inlet temperature* of the hot and cold fluids and their *mass flow rates*, which are usually specified. Then, once the effectiveness of the heat exchanger is known, the actual heat transfer rate $\dot{Q}$ can be determined from

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = \varepsilon C_{\min} (T_{h, \text{in}} - T_{c, \text{in}}) \quad (22-33)$$

where

$$\text{if } C_c = C_{\min}: \quad \varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_c (T_{c, \text{out}} - T_{c, \text{in}})}{C_c (T_{h, \text{in}} - T_{c, \text{in}})} = \frac{T_{c, \text{out}} - T_{c, \text{in}}}{T_{h, \text{in}} - T_{c, \text{in}}}$$

$$\text{if } C_h = C_{\min}: \quad \varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_h (T_{h, \text{in}} - T_{h, \text{out}})}{C_h (T_{h, \text{in}} - T_{c, \text{in}})} = \frac{T_{h, \text{in}} - T_{h, \text{out}}}{T_{h, \text{in}} - T_{c, \text{in}}}$$

Therefore, the effectiveness of a heat exchanger enables us to determine the heat transfer rate without knowing the *outlet temperatures* of the fluids.

The effectiveness of a heat exchanger depends on the *geometry* of the heat exchanger as well as the *flow arrangement*. Therefore, different types of heat exchangers have different effectiveness relations. Below we illustrate the development of the effectiveness ε relation for the double-pipe *parallel-flow* heat exchanger.

Equation 22–23 developed in Sec. 22–4 for a parallel-flow heat exchanger can be rearranged as

$$\ln \frac{T_{h, \text{out}} - T_{c, \text{out}}}{T_{h, \text{in}} - T_{c, \text{in}}} = -\frac{UA_s}{C_c} \left(1 + \frac{C_c}{C_h} \right) \quad (22-34)$$

Also, solving Eq. 22–30 for $T_{h, \text{out}}$ gives

$$T_{h, \text{out}} = T_{h, \text{in}} - \frac{C_c}{C_h} (T_{c, \text{out}} - T_{c, \text{in}}) \quad (22-35)$$

Substituting this relation into Eq. 22–34 after adding and subtracting $T_{c, \text{in}}$ gives

$$\ln \frac{T_{h, \text{in}} - T_{c, \text{in}} + T_{c, \text{in}} - T_{c, \text{out}} - \frac{C_c}{C_h} (T_{c, \text{out}} - T_{c, \text{in}})}{T_{h, \text{in}} - T_{c, \text{in}}} = \frac{UA_s}{C_c} \left(1 + \frac{C_c}{C_h} \right)$$

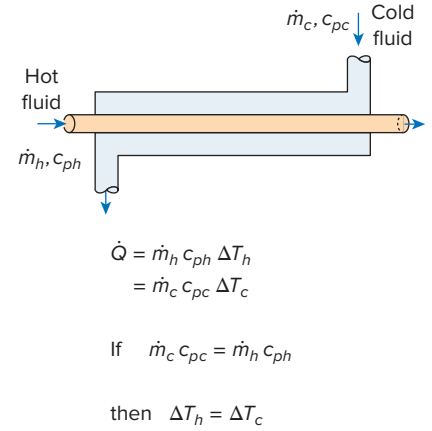


FIGURE 22–26

The temperature rise of the cold fluid in a heat exchanger will be equal to the temperature drop of the hot fluid when the heat capacity rates of the hot and cold fluids are identical.

which simplifies to

$$\ln \left[1 - \left(1 + \frac{C_c}{C_h} \right) \frac{T_{c, \text{out}} - T_{c, \text{in}}}{T_{h, \text{in}} - T_{c, \text{in}}} \right] = -\frac{UA_s}{C_c} \left(1 + \frac{C_c}{C_h} \right) \quad (22-36)$$

We now manipulate the definition of effectiveness to obtain

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{C_c(T_{c, \text{out}} - T_{c, \text{in}})}{C_{\min}(T_{h, \text{in}} - T_{c, \text{in}})} \longrightarrow \frac{T_{c, \text{out}} - T_{c, \text{in}}}{T_{h, \text{in}} - T_{c, \text{in}}} = \varepsilon \frac{C_{\min}}{C_c}$$

Substituting this result into Eq. 22-36 and solving for ε gives the following relation for the effectiveness of a *parallel-flow* heat exchanger:

$$\varepsilon_{\text{parallel flow}} = \frac{1 - \exp \left[-\frac{UA_s}{C_c} \left(1 + \frac{C_c}{C_h} \right) \right]}{\left(1 + \frac{C_c}{C_h} \right) \frac{C_{\min}}{C_c}} \quad (22-37)$$

Taking either C_c or C_h to be $C_{\min}$ (both approaches give the same result), the relation above can be expressed more conveniently as

$$\varepsilon_{\text{parallel flow}} = \frac{1 - \exp \left[-\frac{UA_s}{C_{\min}} \left(1 + \frac{C_{\min}}{C_{\max}} \right) \right]}{1 + \frac{C_{\min}}{C_{\max}}} \quad (22-38)$$

Again $C_{\min}$ is the *smaller* heat capacity rate and $C_{\max}$ is the larger one, and it makes no difference whether $C_{\min}$ belongs to the hot or cold fluid.

Effectiveness relations of the heat exchangers typically involve the *dimensionless* group $UA_s/C_{\min}$. This quantity is called the **number of transfer units NTU** and is expressed as

$$\text{NTU} = \frac{UA_s}{C_{\min}} = \frac{UA_s}{(\dot{m}c_p)_{\min}} \quad (22-39)$$

where U is the overall heat transfer coefficient and A_s is the heat transfer surface area of the heat exchanger. Note that NTU is proportional to A_s . Therefore, for specified values of U and $C_{\min}$, the value of NTU *is a measure of the heat transfer surface area A_s* . Thus, the larger the NTU, the larger the heat exchanger.

In heat exchanger analysis, it is also convenient to define another dimensionless quantity called the **capacity ratio c** as

$$c = \frac{C_{\min}}{C_{\max}} \quad (22-40)$$

It can be shown that the effectiveness of a heat exchanger is a function of the number of transfer units NTU and the capacity ratio c . That is

$$\varepsilon = \text{function} (UA_s/C_{\min}, C_{\min}/C_{\max}) = \text{function} (\text{NTU}, c)$$

TABLE 22–4

Effectiveness relations for heat exchangers: $NTU = UA_s/C_{\min}$ and $c = C_{\min}/C_{\max} = (\dot{m}c_p)_{\min}/(\dot{m}c_p)_{\max}$

Heat exchanger type	Effectiveness relation
1 Double pipe:	
Parallel-flow	$\varepsilon = \frac{1 - \exp[-NTU(1 + c)]}{1 + c}$
Counter-flow	$\varepsilon = \frac{1 - \exp[-NTU(1 - c)]}{1 - c \exp[-NTU(1 - c)]} \quad (\text{for } c < 1)$
	$\varepsilon = \frac{NTU}{1 + NTU} \quad (\text{for } c = 1)$
2 Shell-and-tube:	
One-shell pass 2, 4, . . . tube passes	$\varepsilon_1 = 2 \left\{ 1 + c + \sqrt{1 + c^2} \frac{1 + \exp[-NTU_1 \sqrt{1 + c^2}]}{1 - \exp[-NTU_1 \sqrt{1 + c^2}]} \right\}^{-1}$
n -shell passes $2n, 4n, . . .$ tube passes	$\varepsilon_n = \left[\left(\frac{1 - \varepsilon_1 c}{1 - \varepsilon_1} \right)^n - 1 \right] \left[\left(\frac{1 - \varepsilon_1 c}{1 - \varepsilon_1} \right)^n - c \right]^{-1}$
3 Cross-flow (single-pass)	
Both fluids unmixed	$\varepsilon = 1 - \exp \left\{ \frac{NTU^{0.22}}{c} [\exp(-cNTU^{0.78}) - 1] \right\}$
$C_{\max}$ mixed, $C_{\min}$ unmixed	$\varepsilon = \frac{1}{c} (1 - \exp \{-c[1 - \exp(-NTU)]\})$
$C_{\min}$ mixed, $C_{\max}$ unmixed	$\varepsilon = 1 - \exp \left\{ -\frac{1}{c} [1 - \exp(-cNTU)] \right\}$
4 All heat exchangers with $c = 0$	$\varepsilon = 1 - \exp(-NTU)$

Source: W. M. Kays and A. L. London. *Compact Heat Exchangers*, 3rd ed. McGraw-Hill, 1984.

Effectiveness relations have been developed for a large number of heat exchangers, and the results are given in Table 22–4. The effectivenesses of some common types of heat exchangers are also plotted in Fig. 22–27 on the next page.

More extensive effectiveness charts and relations are available in the literature. The dashed lines in Fig. 22–27*f* are for the case of $C_{\min}$ unmixed and $C_{\max}$ mixed and the solid lines are for the opposite case. The analytic relations for the effectiveness give more accurate results than the charts, since reading errors in charts are unavoidable, and the relations are very suitable for computerized analysis of heat exchangers.

We make these observations from the effectiveness relations and charts already given

1. The value of the effectiveness ranges from 0 to 1. It increases rapidly with NTU for small values (up to about $NTU = 1.5$), but rather slowly

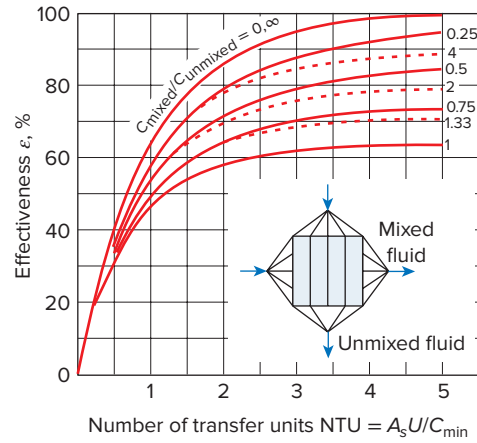
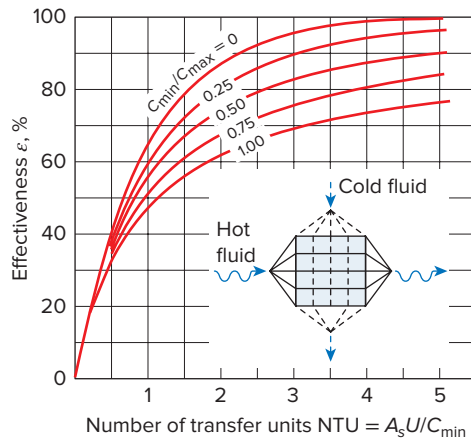
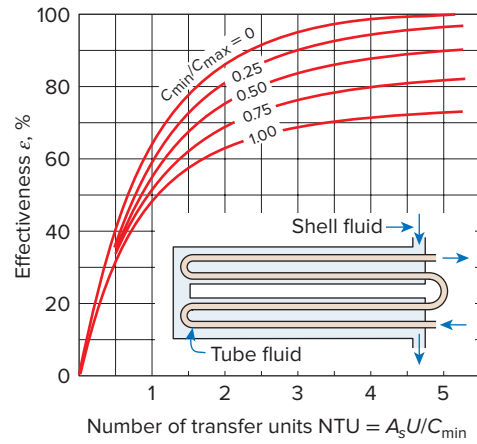
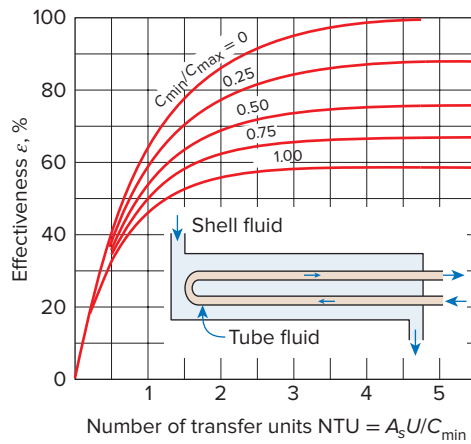
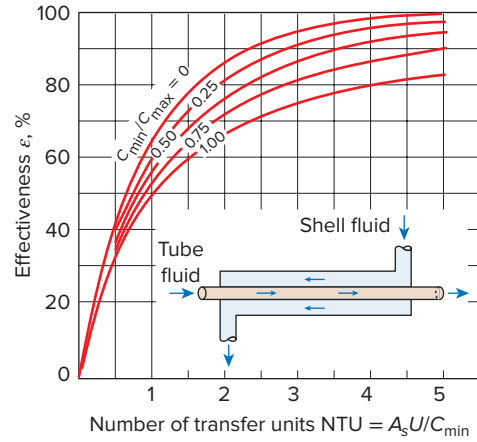
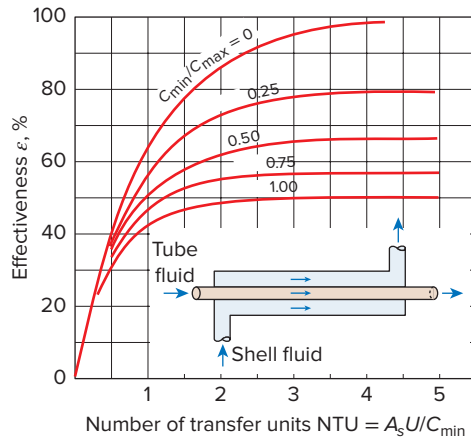


FIGURE 22-27

Effectiveness for heat exchangers.

From Kays and London, 1984.

for larger values. Therefore, the use of a heat exchanger with a large NTU (usually larger than 3) and thus a large size cannot be justified economically, since a large increase in NTU in this case corresponds to a small increase in effectiveness. Thus, a heat exchanger with a very high effectiveness may be desirable from a heat transfer point of view but undesirable from an economical point of view.

2. For a given NTU and capacity ratio $c = C_{\min}/C_{\max}$, the *counter-flow* heat exchanger has the *highest* effectiveness, followed closely by the cross-flow heat exchangers with both fluids unmixed. As you might expect, the lowest effectiveness values are encountered in parallel-flow heat exchangers (Fig. 22–28).
3. The effectiveness of a heat exchanger is independent of the capacity ratio c for NTU values of less than about 0.3.
4. The value of the capacity ratio c ranges between 0 and 1. For a given NTU, the effectiveness becomes a *maximum* for $c = 0$ and a *minimum* for $c = 1$. The case $c = C_{\min}/C_{\max} \rightarrow 0$ corresponds to $C_{\max} \rightarrow \infty$, which is realized during a phase-change process in a *condenser* or *boiler*. All effectiveness relations in this case reduce to

$$\varepsilon = \varepsilon_{\max} = 1 - \exp(-NTU) \quad (22-41)$$

regardless of the type of heat exchanger (Fig. 22–29). Note that the temperature of the condensing or boiling fluid remains constant in this case. The effectiveness is the *lowest* in the other limiting case of $c = C_{\min}/C_{\max} = 1$, which is realized when the heat capacity rates of the two fluids are equal. In industrial applications such as hot water cooling or chilled water heating pertaining to heating, ventilating, and air conditioning (HVAC) industry, the hot or cold fluid is channeled from one location to another with the pipe immersed in ambient environment. Typical examples of this flow situation are the fluid carrying heat exchanger pipe exposed to ambient air or submerged in large liquid mediums such as a lake. In such cases, since the effective change in reference temperature of the ambient environment is virtually zero and the relative mass of the ambient environment is infinitely large, the maximum heat capacity rate $C_{\max}$ is a very large number. Thus the capacity ratio, $c = C_{\min}/C_{\max} \rightarrow 0$ and hence Eq. 22–41 is applicable in cases when the heat exchanger is in contact with the ambient environment.

Once the quantities $c = C_{\min}/C_{\max}$ and $NTU = UA_s/C_{\min}$ have been evaluated, the effectiveness ε can be determined from either the charts or the effectiveness relation for the specified type of heat exchanger. Then the rate of heat transfer $\dot{Q}$ and the outlet temperatures $T_{h, \text{out}}$ and $T_{c, \text{out}}$ can be determined from Eqs. 22–33 and 22–30, respectively. Note that the analysis of heat exchangers with unknown outlet temperatures is a straightforward matter with the effectiveness–NTU method but requires rather tedious iterations with the LMTD method.

We mentioned earlier that when all the inlet and outlet temperatures are specified, the *size* of the heat exchanger can easily be determined using the LMTD method. Alternatively, it can also be determined from the effectiveness–NTU method by first evaluating the effectiveness ε from its definition (Eq. 22–29) and then the NTU from the appropriate NTU relation in Table 22–5.

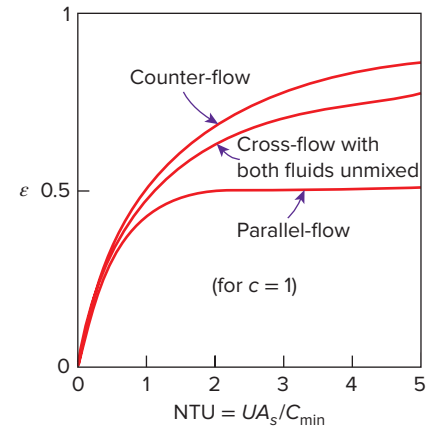


FIGURE 22–28

For a specified NTU and capacity ratio c , the counter-flow heat exchanger has the highest effectiveness and the parallel-flow the lowest.

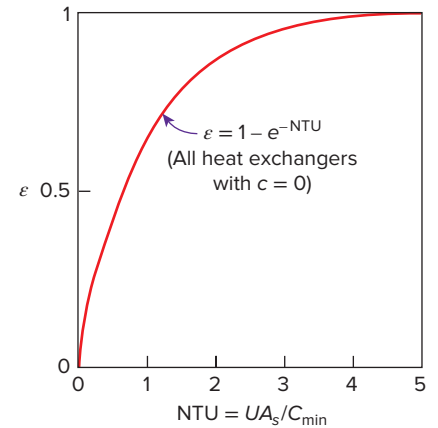


FIGURE 22–29

The effectiveness relation reduces to $\varepsilon = \varepsilon_{\max} = 1 - \exp(-NTU)$ for all heat exchangers when the capacity ratio $c = 0$.

TABLE 22-5

NTU relations for heat exchangers: $NTU = UA_s/C_{\min}$ and $c = C_{\min}/C_{\max} = (\dot{m}c_p)_{\min}/(\dot{m}c_p)_{\max}$

Heat exchanger type	NTU relation
1 Double-pipe:	
Parallel-flow	$NTU = -\frac{\ln[1 - \varepsilon(1 + c)]}{1 + c}$
Counter-flow	$NTU = \frac{1}{c - 1} \ln \left(\frac{\varepsilon - 1}{\varepsilon c - 1} \right) \quad (\text{for } c < 1)$
	$NTU = \frac{\varepsilon}{1 - \varepsilon} \quad (\text{for } c = 1)$
2 Shell and tube:	
One-shell pass 2, 4, . . . tube passes n -shell passes	$NTU_1 = -\frac{1}{\sqrt{1 + c^2}} \ln \left(\frac{2/\varepsilon_1 - 1 - c - \sqrt{1 + c^2}}{2/\varepsilon_1 - 1 - c + \sqrt{1 + c^2}} \right)$
$2n, 4n, . . .$ tube passes	$NTU_n = n(NTU)_1$ To find effectiveness of the heat exchanger with one-shell pass use, $\varepsilon_1 = \frac{F - 1}{F - c}$ where $F = \left(\frac{\varepsilon_n c - 1}{\varepsilon_n - 1} \right)^{1/n}$
3 Cross-flow (single-pass):	
$C_{\max}$ mixed, $C_{\min}$ unmixed	$NTU = -\ln \left[1 + \frac{\ln(1 - \varepsilon c)}{c} \right]$
$C_{\min}$ mixed, $C_{\max}$ unmixed	$NTU = -\frac{\ln [c \ln(1 - \varepsilon) + 1]}{c}$
4 All heat exchangers with $c = 0$	
	$NTU = -\ln(1 - \varepsilon)$

Source: W. M. Kays and A. L. London. *Compact Heat Exchangers*, 3rd ed. McGraw-Hill, 1984.

Note that the relations in Table 22-5 are equivalent to those in Table 22-4. Both sets of relations are given for convenience. The relations in Table 22-4 give the effectiveness directly when NTU is known, and the relations in Table 22-5 give the NTU directly when the effectiveness ε is known.

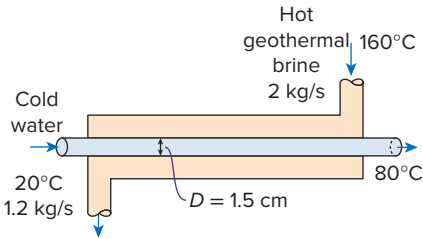


FIGURE 22-30
Schematic for Example 22-8.

EXAMPLE 22-8 The Effectiveness–NTU Method

A counter-flow double-pipe heat exchanger is to heat water from 20°C to 80°C at a rate of 1.2 kg/s (Fig. 22-30). The heating is to be accomplished by geothermal water available at 160°C at a mass flow rate of 2 kg/s. The inner tube is thin-walled and has a diameter of 1.5 cm. The overall heat transfer coefficient of the heat exchanger is 640 W/m²·K. Using the effectiveness–NTU method, determine the length of the heat exchanger required to achieve the desired heating.

SOLUTION Water is heated in a counter-flow double-pipe heat exchanger by geothermal water. Using the ε -NTU method, the required length of the heat exchanger is to be determined.

Analysis In the effectiveness-NTU method, we first determine the heat capacity rates of the hot and cold fluids and identify the smaller one.

$$C_h = \dot{m}_h c_{ph} = (2 \text{ kg/s})(4.31 \text{ kJ/kg}\cdot\text{K}) = 8.62 \text{ kW/K}$$

$$C_c = \dot{m}_c c_{pc} = (1.2 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot\text{K}) = 5.02 \text{ kW/K}$$

Therefore,

$$C_{\min} = C_c = 5.02 \text{ kW/K}$$

and

$$c = C_{\min}/C_{\max} = 5.02/8.62 = 0.582$$

Then the maximum heat transfer rate is determined from Eq. 22-32 to be

$$\begin{aligned}\dot{Q}_{\max} &= C_{\min}(T_{h,\text{in}} - T_{c,\text{in}}) \\ &= (5.02 \text{ kW/K})(160 - 20)^\circ\text{C} \\ &= 702.8 \text{ kW}\end{aligned}$$

That is, the maximum possible heat transfer rate in this heat exchanger is 702.8 kW. The actual rate of heat transfer is

$$\begin{aligned}\dot{Q} &= [\dot{m}_c c_p (T_{\text{out}} - T_{\text{in}})]_{\text{water}} = (1.2 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot\text{K})(80 - 20)^\circ\text{C} \\ &= 301.0 \text{ kW}\end{aligned}$$

Thus, the effectiveness of the heat exchanger is

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{301.0 \text{ kW}}{702.8 \text{ kW}} = 0.428$$

Knowing the effectiveness, the NTU of this counter-flow heat exchanger can be determined from Fig. 22-27b or the appropriate relation from Table 22-5 for $c < 1$. We choose the latter approach for greater accuracy:

$$\text{NTU} = \frac{1}{c-1} \ln \left(\frac{\varepsilon-1}{\varepsilon c-1} \right) = \frac{1}{0.582-1} \ln \left(\frac{0.428-1}{0.428 \times 0.582-1} \right) = 0.651$$

Then the heat transfer surface area becomes

$$\text{NTU} = \frac{UA_s}{C_{\min}} \longrightarrow A_s = \frac{\text{NTU } C_{\min}}{U} = \frac{(0.651)(5020 \text{ W/K})}{640 \text{ W/m}^2\cdot\text{K}} = 5.11 \text{ m}^2$$

To provide this much heat transfer surface area, the length of the tube must be

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{5.11 \text{ m}^2}{\pi(0.015 \text{ m})} = \mathbf{108 \text{ m}}$$

Discussion This problem was solved in Example 22-4 using the LMTD method. Note that we obtained practically the same result in a systematic and straightforward manner using the effectiveness-NTU method.

EXAMPLE 22–9 Cooling Hot Oil by Water in a Multipass Heat Exchanger

Hot oil is to be cooled by water in a one-shell-pass and eight-tube-passes heat exchanger. The tubes are thin-walled and are made of copper with an internal diameter of 1.4 cm. The length of each tube pass in the heat exchanger is 5 m, and the overall heat transfer coefficient is $310 \text{ W/m}^2\cdot\text{K}$. Water flows through the tubes at a rate of 0.2 kg/s , and the oil through the shell at a rate of 0.3 kg/s . The water and the oil enter at temperatures of 20°C and 150°C , respectively. Determine the rate of heat transfer in the heat exchanger and the outlet temperatures of the water and the oil.

SOLUTION Hot oil is to be cooled by water in a heat exchanger. The mass flow rates and the inlet temperatures are given. The rate of heat transfer and the outlet temperatures are to be determined.

Assumptions 1 Steady operating conditions exist. 2 The heat exchanger is well insulated so that heat loss to the surroundings is negligible. 3 The thickness of the tube is negligible since it is thin-walled. 4 Changes in the kinetic and potential energies of fluid streams are negligible. 5 The overall heat transfer coefficient is constant and uniform.

Properties We take the specific heats of water and oil to be 4.18 and $2.13 \text{ kJ/kg}\cdot^\circ\text{C}$, respectively.

Analysis The schematic of the heat exchanger is given in Fig. 22–31. The outlet temperatures are not specified, and they cannot be determined from an energy balance. The use of the LMTD method in this case will involve tedious iterations, and thus the ε -NTU method is indicated. The first step in the ε -NTU method is to determine the heat capacity rates of the hot and cold fluids and identify the smaller one:

$$C_h = \dot{m}_h c_{ph} = (0.3 \text{ kg/s})(2.13 \text{ kJ/kg}\cdot^\circ\text{C}) = 0.639 \text{ kW/K}$$

$$C_c = \dot{m}_c c_{pc} = (0.2 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot^\circ\text{C}) = 0.836 \text{ kW/K}$$

Therefore,

$$C_{\min} = C_h = 0.639 \text{ kW/K} \text{ and } c = \frac{C_{\min}}{C_{\max}} = \frac{0.639}{0.836} = 0.764$$

Then the maximum heat transfer rate is determined from Eq. 22–32 to be

$$\dot{Q}_{\max} = C_{\min}(T_{h,\text{in}} - T_{c,\text{in}}) = (0.639 \text{ kW/K})(150 - 20)^\circ\text{C} = 83.1 \text{ kW}$$

That is, the maximum possible heat transfer rate in this heat exchanger is 83.1 kW . The heat transfer surface area is

$$A_s = n(\pi DL) = 8\pi(0.014 \text{ m})(5 \text{ m}) = 1.76 \text{ m}^2$$

Then the NTU of this heat exchanger becomes

$$\text{NTU} = \frac{UA_s}{C_{\min}} = \frac{(310 \text{ W/m}^2\cdot\text{K})(1.76 \text{ m}^2)}{639 \text{ W/K}} = 0.854$$

The effectiveness of this heat exchanger corresponding to $c = 0.764$ and $\text{NTU} = 0.854$ is determined from Fig. 22–27c to be

$$\varepsilon = 0.47$$

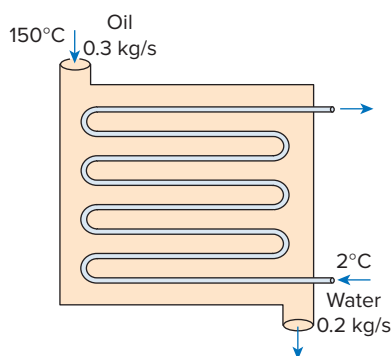


FIGURE 22–31

Schematic for Example 22–9.

We could also determine the effectiveness from the third relation in Table 22–4 more accurately but with more labor. Then the actual rate of heat transfer becomes

$$\dot{Q} = \varepsilon \dot{Q}_{\max} = (0.47)(83.1 \text{ kW}) = \mathbf{39.1 \text{ kW}}$$

Finally, the outlet temperatures of the cold and the hot fluid streams are determined to be

$$\begin{aligned} \dot{Q} &= C_c(T_{c, \text{out}} - T_{c, \text{in}}) \longrightarrow T_{c, \text{out}} = T_{c, \text{in}} + \frac{\dot{Q}}{C_c} \\ &= 20^\circ\text{C} + \frac{39.1 \text{ kW}}{0.836 \text{ kW/K}} = \mathbf{66.8^\circ\text{C}} \end{aligned}$$

$$\begin{aligned} \dot{Q} &= C_h(T_{h, \text{in}} - T_{h, \text{out}}) \longrightarrow T_{h, \text{out}} = T_{h, \text{in}} - \frac{\dot{Q}}{C_h} \\ &= 150^\circ\text{C} - \frac{39.1 \text{ kW}}{0.639 \text{ kW/K}} = \mathbf{88.8^\circ\text{C}} \end{aligned}$$

Therefore, the temperature of the cooling water will rise from 20°C to 66.8°C as it cools the hot oil from 150°C to 88.8°C in this heat exchanger.

SUMMARY

Heat exchangers are devices that allow the exchange of heat between two fluids without allowing them to mix with each other. Heat exchangers are manufactured in a variety of types, the simplest being the *double-pipe* heat exchanger. In a *parallel-flow* type, both the hot and cold fluids enter the heat exchanger at the same end and move in the same direction, whereas in a *counter-flow* type, the hot and cold fluids enter the heat exchanger at opposite ends and flow in opposite directions. In *compact* heat exchangers, the two fluids move perpendicular to each other, and such a flow configuration is called *cross-flow*. Other common types of heat exchangers in industrial applications are the *plate* and the *shell-and-tube* heat exchangers.

Heat transfer in a heat exchanger usually involves convection in each fluid and conduction through the wall separating the two fluids. In the analysis of heat exchangers, it is convenient to work with an *overall heat transfer coefficient* U or a *total thermal resistance* R , expressed as

$$\frac{1}{UA_s} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o} = R = \frac{1}{h_i A_i} + R_{\text{wall}} + \frac{1}{h_o A_o}$$

where the subscripts i and o stand for the inner and outer surfaces of the wall that separates the two fluids, respectively. When the wall thickness of the tube is small and the thermal

conductivity of the tube material is high, the relation simplifies to

$$\frac{1}{U} \approx \frac{1}{h_i} + \frac{1}{h_o}$$

where $U \approx U_i \approx U_o$. The effects of fouling on both the inner and the outer surfaces of the tubes of a heat exchanger can be accounted for by

$$\begin{aligned} \frac{1}{UA_s} &= \frac{1}{U_i A_i} = \frac{1}{U_o A_o} = R \\ &= \frac{1}{h_i A_i} + \frac{R_{f,i}}{A_i} + \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{R_{f,o}}{A_o} + \frac{1}{h_o A_o} \end{aligned}$$

where $A_i = \pi D_i L$ and $A_o = \pi D_o L$ are the areas of the inner and outer surfaces and $R_{f,i}$ and $R_{f,o}$ are the fouling factors at those surfaces.

In a well-insulated heat exchanger, the rate of heat transfer from the hot fluid is equal to the rate of heat transfer to the cold one. That is

$$\dot{Q} = \dot{m}_c c_{pc}(T_{c, \text{out}} - T_{c, \text{in}}) = C_c(T_{c, \text{out}} - T_{c, \text{in}})$$

and

$$\dot{Q} = \dot{m}_h c_{ph}(T_{h, \text{in}} - T_{h, \text{out}}) = C_h(T_{h, \text{in}} - T_{h, \text{out}})$$

where the subscripts c and h stand for the cold and hot fluids, respectively, and the product of the mass flow rate and the specific heat of a fluid $\dot{m}c_p$ is called the *heat capacity rate*.

Of the two methods used in the analysis of heat exchangers, the *log mean temperature difference* (or LMTD) method is best suited for determining the size of a heat exchanger when all the inlet and the outlet temperatures are known. The *effectiveness-NTU* method is best suited to predict the outlet temperatures of the hot and cold fluid streams in a specified heat exchanger. In the LMTD method, the rate of heat transfer is determined from

$$\dot{Q} = UA_s \Delta T_{lm}$$

where

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln (\Delta T_1 / \Delta T_2)}$$

is the *log mean temperature difference*, which is the suitable form of the average temperature difference for use in the analysis of heat exchangers. Here ΔT_1 and ΔT_2 represent

the temperature differences between the two fluids at the two ends (inlet and outlet) of the heat exchanger. For cross-flow and multipass shell-and-tube heat exchangers, the logarithmic mean temperature difference is related to the counter-flow one $\Delta T_{lm, CF}$ as

$$\Delta T_{lm} = F \Delta T_{lm, CF}$$

where F is the *correction factor*, which depends on the geometry of the heat exchanger and the inlet and outlet temperatures of the hot and cold fluid streams.

The *effectiveness* of a heat exchanger is defined as

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} = \frac{\text{Actual heat transfer rate}}{\text{Maximum possible heat transfer rate}}$$

where

$$\dot{Q}_{\max} = C_{\min}(T_{h, in} - T_{c, in})$$

and $C_{\min}$ is the smaller of $C_h = \dot{m}_h c_{ph}$ and $C_c = \dot{m}_c c_{pc}$. The effectiveness of heat exchangers can be determined from effectiveness relations or charts.

REFERENCES AND SUGGESTED READING

1. N. Afgan and E. U. Schlunder. *Heat Exchanger: Design and Theory Sourcebook*. Washington, DC: McGraw-Hill/Scripta, 1974.
2. R. A. Bowman, A. C. Mueller, and W. M. Nagle. "Mean Temperature Difference in Design." *Transactions of the ASME* 62, 1940, pp. 283–289.
3. A. P. Fraas. *Heat Exchanger Design*, 2nd ed. New York: Wiley, 1989.
4. K. A. Gardner. "Variable Heat Transfer Rate Correction in Multipass Exchangers, Shell Side Film Controlling." *Transactions of the ASME* 67 (1945), pp. 31–38.
5. W. M. Kays and A. L. London. *Compact Heat Exchangers*, 3rd ed. New York: McGraw-Hill, 1984.
6. W. M. Kays and H. C. Perkins. In *Handbook of Heat Transfer*, W. M. Rohsenow and J. P. Hartnett, eds. New York: McGraw-Hill, 1972, Chap. 7.
7. Q. Li, G. Flamant, X. Yuan, P. Neveu, and L. Luo. "Compact Heat Exchangers: A Review and Future Applications for a New Generation of High Temperature Solar Receivers." *Renewable and Sustainable Energy Reviews*, 15(2011), pp. 4855–4875.
8. A. C. Mueller. "Heat Exchangers." In *Handbook of Heat Transfer*, W. M. Rohsenow and J. P. Hartnett, eds. New York: McGraw-Hill, 1972, Chap. 18.
9. M. N. Özisik. *Heat Transfer—A Basic Approach*. New York: McGraw-Hill, 1985.
10. E. U. Schlunder. *Heat Exchanger Design Handbook*. Washington, DC: Hemisphere, 1982.
11. *Standards of Tubular Exchanger Manufacturers Association*, 9th ed. New York: Tubular Exchanger Manufacturers Association, 2007.
12. R. A. Stevens, J. Fernandes, and J. R. Woolf. "Mean Temperature Difference in One, Two, and Three Pass Crossflow Heat Exchangers." *Transactions of the ASME* 79 (1957), pp. 287–297.
13. J. Taborek, G. F. Hewitt, and N. Afgan. *Heat Exchangers: Theory and Practice*. New York: Hemisphere, 1983.
14. G. Walker. *Industrial Heat Exchangers*. Washington, DC: Hemisphere, 1982.

15 Heat Exchanger Networks I – Heat Transfer Equipment

Many types of heat transfer equipment are used in the process industries. By far, the most commonly used type is the shell-and-tube heat exchanger. Two of the more common types of shell-and-tube heat exchanger are illustrated in Figure 15.1. The first, shown in Figure 15.1a, shows the fluid flowing through the tubes in a *single pass*. Figure 15.1b shows a design in which the fluid flows in *two passes* through the tube-side. The two-pass arrangement can be constructed using “U” tubes to turn the tube-side fluid, rather than relying on the heat exchanger head to reverse the flow, as shown in Figure 15.1b. The fluid flowing in the shell is made to flow repeatedly across the outside of the tubes through the use of baffles. Figure 15.2a shows an ideal shell-side flow pattern using *segmented* baffles with combinations of ideal cross-flow and ideal axial flow. Cross-flow gives both higher rates of heat transfer and higher pressure drops than axial flow. In practice, the flow pattern is not ideal, as illustrated in Figure 15.2a, but leakage occurs through the tube-to-baffle clearance, as illustrated in Figure 15.2b. Also, bypassing occurs between the tube bundle and the shell and is a function of the shell-to-baffle clearance, as illustrated in Figure 15.2b. Sealing devices are usually included in the design to minimize bypassing. Both leakage and bypassing act to reduce the rate of heat transfer on the shell-side. Baffle designs other than segmented baffles are available, giving different flow patterns on the shell-side. There are many other shell-and-tube designs than those illustrated in Figure 15.1^{1–4}.

It should be noted that in Figure 15.1a, the hot fluid flows vertically down. This is normal practice since a hot liquid will become more dense as it is cooled, and therefore less

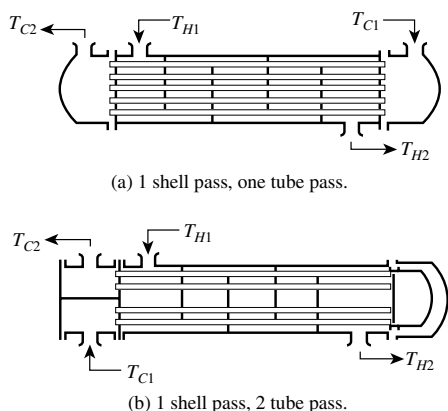


Figure 15.1 Shell-and-tube heat exchangers.

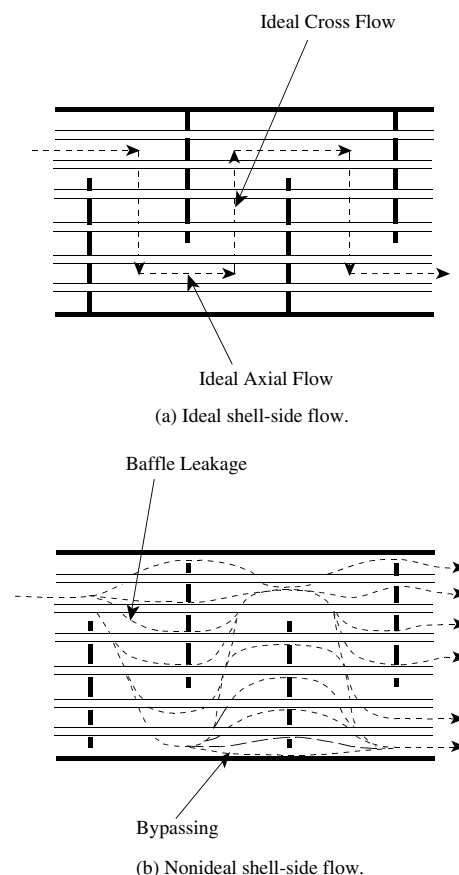


Figure 15.2 Shell-side flow patterns.

buoyant, and would tend to naturally flow downwards as a result of the buoyancy forces. Also, if some condensation of the vapor were occurring, this would also tend to flow naturally downwards. Similarly, in Figure 15.1b, the cold fluid on the tube-side of the heat exchanger flows upwards. This is because a cold liquid being heated up would become less dense and therefore more buoyant, and would tend to naturally flow upwards as a result of the buoyancy forces. Alternatively, if a liquid were being partially vaporized, any vapor would tend to naturally flow upwards.

Consider first the resistance to heat transfer across the wall of the tubes.

15.1 OVERALL HEAT TRANSFER COEFFICIENTS

Figure 15.3 illustrates the resistance to heat transfer across the wall of the tube. There are five resistances to heat transfer. Each can be characterized by a *heat transfer coefficient*.

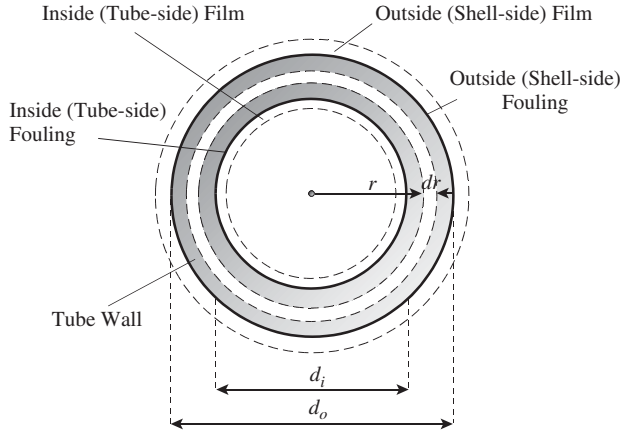


Figure 15.3 Resistance to heat transfer across the tube.

1. *Shell-side film coefficient.* The heat transfer through the resistance created by the fluid on the outside (shell-side) of the tubes is given by:

$$Q = h_S A_O \Delta T_S \quad (15.1)$$

where Q = heat transferred per unit time ($\text{J}\cdot\text{s}^{-1} = \text{W}$)
 h_S = film heat transfer coefficient on the outside (shell-side) of the tubes ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$)
 A_O = heat transfer area outside (shell-side) of the tubes (m^2)
 ΔT_S = temperature difference across the outside (shell-side) film (K)

2. *Shell-side fouling coefficient.* Heat transfer is usually impeded by surface deposits on the heat transfer surface (*fouling*). The material deposited as fouling usually has a low thermal conductivity. Fouling is time dependent and depends on the fluid velocity, temperature and many other factors. Fouling is difficult to predict and allowances are usually based on experience. Design is based on an assumed value to be expected after a reasonable period of time before the exchanger is cleaned.

The heat transfer through the resistance created by the outside (shell-side) fouling is quantified by a *fouling coefficient* given by:

$$Q = h_{SF} A_O \Delta T_{SF} \quad (15.2)$$

where h_{SF} = outside (shell-side) fouling coefficient ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$)
 ΔT_{SF} = temperature difference across the outside (shell-side) fouling resistance (K)

3. *Tube wall coefficient.* Heat transfer across the tube wall is described by the Fourier Equation¹:

$$Q = -kA \frac{dT}{dr} \quad (15.3)$$

where k = thermal conductivity of the tube wall material ($\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$)

r = radial distance (m)

A = heat transfer area at radial distance r (m)

Consider an incremental thickness of tube wall dr with radius r as illustrated in Figure 15.3.

$$A = 2\pi rL \quad (15.4)$$

where L = tube length (m)

Substituting Equation 15.4 into Equation 15.3 and integrating:

$$-\frac{Q}{2\pi kL} \int_{r_I}^{r_O} \frac{dr}{r} = \int_{T_I}^{T_O} dT \quad (15.5)$$

where r_O = outside tube radius (m)

r_I = inside tube radius (m)

T_O = outside surface temperature of the tube ($^{\circ}\text{C}$)

T_I = inside surface temperature of the tube ($^{\circ}\text{C}$)

Integrating Equation 15.5 gives:

$$-\frac{Q}{2\pi kL} \ln\left(\frac{r_O}{r_I}\right) = T_O - T_I \quad (15.6)$$

Thus:

$$Q = \frac{2\pi kL}{\ln\left(\frac{d_O}{d_I}\right)} \Delta T_W \quad (15.7)$$

where d_O, d_I = outside and inside tube diameters (m)

ΔT_W = temperature difference across the wall (K)

4. *Tube-side fouling coefficient.* The heat transfer through the resistance created by the inside (tube-side) fouling is given by:

$$Q = h_{TF} A_I \Delta T_{TF} \quad (15.8)$$

where h_{TF} = inside (tube-side) fouling coefficient ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$)

A_I = inside (tube-side) heat transfer area of tubes (m^2)

ΔT_{TF} = temperature difference across the tube-side fouling resistance (K)

5. *Tube-side film coefficient.* The heat transfer through the resistance created by the fluid on the inside (tube-side) of the tubes is given by:

$$Q = h_T A_I \Delta T_T \quad (15.9)$$

where h_T = inside (tube-side) film heat transfer coefficient ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$)

ΔT_T = temperature difference across the inside (tube-side) film (K)

The five resistances can be added. If ΔT represents the temperature difference between the bulk temperature of

the fluid on the outside and inside of the tubes, then the temperature differences across the individual resistances can be added to give:

$$\begin{aligned}\Delta T &= \Delta T_S + \Delta T_{SF} + \Delta T_W + \Delta T_{TF} + \Delta T_T \\ &= \frac{Q}{h_S A_O} + \frac{Q}{h_{SF} A_O} + \frac{Q}{2\pi kL} \ln \left(\frac{d_O}{d_I} \right) \\ &\quad + \frac{Q}{h_{TF} A_I} + \frac{Q}{h_T A_I}\end{aligned}\quad (15.10)$$

Rearranging Equation 15.10 gives:

$$\begin{aligned}\Delta T &= \frac{Q}{A_O} \left[\frac{1}{h_S} + \frac{1}{h_{SF}} + \frac{d_O}{2k} \ln \left(\frac{d_O}{d_I} \right) \right. \\ &\quad \left. + \frac{d_O}{d_I} \cdot \frac{1}{h_{TF}} + \frac{d_O}{d_I} \cdot \frac{1}{h_T} \right]\end{aligned}\quad (15.11)$$

If the overall heat transfer is written as:

$$\Delta T = \frac{Q}{A_O} \frac{1}{U} \quad (15.12)$$

where U = overall heat transfer coefficient based on the outside area of the tube ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$), then comparing Equations 15.11 and 15.12:

$$\frac{1}{U} = \frac{1}{h_S} + \frac{1}{h_{SF}} + \frac{d_O}{2k} \ln \left(\frac{d_O}{d_I} \right) + \frac{d_O}{d_I} \cdot \frac{1}{h_{TF}} + \frac{d_O}{d_I} \cdot \frac{1}{h_T} \quad (15.13)$$

Table 15.1 lists typical values for the film transfer coefficients¹⁻⁴.

Table 15.2 gives typical values of fouling coefficients¹⁻⁴.

Fouling is often quoted as a *fouling factor*. This is simply the reciprocal of the fouling coefficient.

Table 15.1 Typical values for film transfer coefficients.

	h_S or h_T ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$)
<i>No change of state</i>	
Water	2000–6000
Gases	10–500
Organic liquid (low viscosity)	1000–3000
Organic liquid (high viscosity)	100–1000
<i>Condensing</i>	
Steam	5000–15,000
Organic (low viscosity)	1000–2500
Organic (high viscosity)	500–1000
Ammonia	3000–6000
<i>Evaporation</i>	
Water	2000–10,000
Organic (low viscosity)	500–2000
Organic (high viscosity)	100–500
Ammonia	1000–2500

Table 15.2 Typical values of fouling coefficients.

	h_{SF} or h_{TF} ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$)
<i>Water</i>	
Distilled	11,000
Boiler feedwater	6000–11,000
Town water	2000–5000
Well water	1000–3000
Clear river	2000–6000
Good-quality cooling water	3000–6000
Poor-quality cooling water	1000–2000
Sea	6000–11,000
Boiler blowdown	3000
<i>Steam</i>	
Good quality	20,000
Contaminated	5000–11,000
<i>Liquids</i>	
Aqueous salt solutions	3000–6000
Organic (low viscosity)	3000–11,000
Organic (high viscosity)	1000–3000
Machinery oil	6000
Fuel oils	1000
Tars	500–1000
Vegetable oils	2000
<i>Gases</i>	
Air	2000–4000
Organic vapor	5000–11,000

As pointed out previously, fouling is dependent on time, fluid velocity, temperature and other factors. Table 15.3 illustrates this with typical fouling coefficients for crude oil as a function of fluid velocity and temperature.

It is also interesting to consider the heat transfer coefficient of the tube wall relative to the film and fouling coefficients. Table 15.4 tabulates the coefficients for a variety of materials for some standard tube sizes. It should be noted that the sizes in Table 15.4 are not universally standard.

From Table 15.4, the heat transfer coefficient for the tube wall in many cases is so high that its contribution to the overall heat transfer coefficient can be neglected.

15.2 HEAT TRANSFER COEFFICIENTS AND PRESSURE DROPS FOR SHELL-AND-TUBE HEAT EXCHANGERS

Calculation of the overall heat transfer coefficient from Equation 15.13 requires knowledge of the film transfer coefficients. Although Table 15.1 presents typical values,

Table 15.3 Fouling coefficient for crude oil as a function of temperature and fluid velocity ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$)¹.

Below 90°C		90 to 150°C		150 to 260°C		Over 260°C	
Less than 1 m·s ⁻¹	Greater than 1 m·s ⁻¹	Less than 1 m·s ⁻¹	Greater than 1 m·s ⁻¹	Less than 1 m·s ⁻¹	Greater than 1 m·s ⁻¹	Less than 1 m·s ⁻¹	Greater than 1 m·s ⁻¹
2000	3000	2000	3000	1000	3000	1000	2000

Table 15.4 Tube wall coefficients based on the outer diameter for a variety of metals at 100°C.

Metal	$k(\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1})$	$\frac{2k}{d_O \ln d_O/d_I}(\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1})$			
		$d_O = 20 \text{ mm}$		$d_O = 25 \text{ mm}$	
		$d_I = 16.8 \text{ mm}$	$d_I = 16 \text{ mm}$	$d_I = 21 \text{ mm}$	$d_I = 19.8 \text{ mm}$
Copper	378	216,800	169,400	173,400	129,700
Aluminum	206	118,200	92,300	94,500	70,700
Nickel	61	35,000	27,300	28,000	20,900
Cupro-nickel	45	25,800	20,200	20,600	15,400
Steel	45	25,800	20,200	20,600	15,400
Monel	30	17,200	13,400	13,800	10,300
Stainless steel	16	9200	7200	7300	5500
Titanium	16	9200	7200	7300	5500

the actual values depend on the velocities (flowrates) of the fluids, fluid physical properties and exchanger geometry. A method is needed that will allow an estimate of the film transfer coefficients for both the tube-side and the shell-side. In addition to heat transfer coefficients, it is also necessary to be able to estimate the pressure drops.

At the conceptual stage for heat exchanger network synthesis, the calculation of heat transfer coefficients and pressure drops should depend as little as possible on the detailed geometry. However, some assumptions must be made regarding the geometry.

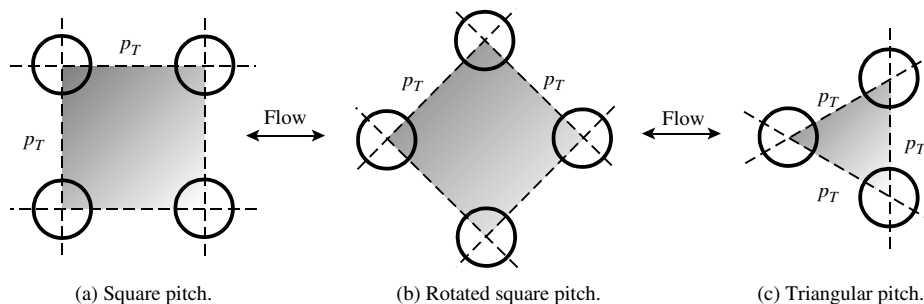
1. *Tube diameter.* Standard sizes are used, but standards are not universal. Common sizes are $d_O = 20 \text{ mm}$ with $d_I = 16 \text{ mm}$ and $d_O = 25 \text{ mm}$ with $d_I = 19.8 \text{ mm}$.

2. *Tube length.* Standard tube lengths are preferred, but again, standards are not universal. However, in principle,

any length of tube can be used. The working length of the tube is slightly shorter than its end-to-end length from the length taken up by the tube sheets on which it is mounted. An allowance of 0.05 m for two tube sheets is a reasonable assumption in the preliminary design stage. The choice of tube length is a degree of freedom at the discretion of the designer. The same heat transfer area can be made available either with a small number of long tubes in a small diameter but long shell or a large number of short tubes in a large diameter but short shell. The ratio of tube length to shell diameter is usually in the range 5 to 10.

3. *Tube pitch.* Tube pitch (p_T) is the center-to-center distance between adjacent tubes and is usually $1.25d_O$.

4. *Tube configuration.* Tubes can be arranged in either a *square* or *triangular configuration*, as illustrated in Figure 15.4. A square configuration is used for fouling

**Figure 15.4** Tube configurations.

fluids to provide *lanes* between the rows for mechanical cleaning. Cleaning lanes should be continuous through the entire tube bundle. A triangular configuration is restricted to nonfouling fluids because it is more difficult to clean mechanically. However, for a given tube pitch, a triangular configuration allows a greater number of tubes in a given shell diameter. A square configuration will normally be a conservative assumption in conceptual design. For a square pitch, each tube is contained in an area of p_T^2 . For a triangular pitch, each triangular pitch with sides p_T having an area of $0.5p_T^2 \sin 60^\circ$ contains half a tube. Thus, a single tube in a triangular pitch is contained in an area of $p_T^2 \sin 60^\circ = 0.866p_T^2$. This means that for the same pitch, a triangular configuration can allow a greater number of tubes than a square configuration in the same size of shell. The assumption of a square configuration will be conservative.

5. Baffle cut. Baffles are used to direct the fluid stream across the tubes. The *baffle cut* is the height of the segment removed to form the baffle as a fraction of the baffle disc diameter. Baffle cuts from 0.15 to 0.45 are used. In conceptual design, a value of 0.25 can be assumed.

Simple models are developed in Appendix C in which heat transfer coefficient and pressure drop are both related to velocity⁵. It is then possible to derive a correlation between the heat transfer coefficient, pressure drop and the surface area by using velocity as the bridge between the two⁵.

For the tube-side:

$$h_T = K_{hT} v_T^{0.8} \quad (15.14)$$

$$\Delta P_T = K_{PT} \frac{A d_I^2}{4 F_I d_O} v_T^{2.8} + 1.25 N_{PT} \rho v_T^2 \quad (15.15)$$

$$\Delta P_T = K_{PT1} A h_T^{3.5} + K_{PT2} h_T^{2.5} \quad (15.16)$$

where $K_{hT} = C \left(\frac{k}{d_I} \right) Pr^{1/3} \left(\frac{d_I \rho}{\mu} \right)^{0.8} \quad (15.17)$

$$K_{PT} = 0.092 \rho^{0.8} \mu^{0.2} d_I^{-1.2} \quad (15.18)$$

$$K_{PT1} = \frac{0.023 \rho^{0.8} \mu^{0.2} d_I^{0.8}}{F_I d_O} \left[\frac{1}{K_{hT}} \right]^{3.5} \quad (15.19)$$

$$K_{PT2} = 1.25 N_{TP} \rho \left[\frac{1}{K_{hT}} \right]^{2.5} \quad (15.20)$$

h_T = tube-side film transfer coefficient

v_T = fluid velocity for the tube-side

ΔP_T = tube-side pressure drop

N_{PT} = number of tube passes

L = tube length

ρ = fluid density

A = heat transfer area

C = constant (0.021 for gases, 0.023 for nonviscous liquids, 0.027 for viscous liquids)

k = fluid thermal conductivity

d_I = inside(tube-side)diameter

Pr = Prandtl Number

$$= \frac{C_P \mu}{k}$$

C_P = fluid heat capacity

μ = fluid viscosity at the bulk fluid temperature

F_I = inside(tube-side)volumetric flowrate

d_O = outside(shell-side)diameter

For the shell-side:

$$h_S = K_{hS} v_S^{0.64} \quad (15.21)$$

$$\Delta P_S = 2 K_{PS1} v_S^{1.83} + K_{PS2} A v_S^{2.83} - K_{PS3} v_S^{1.83} + K_{PS4} A v_S^3 \quad (15.22)$$

$$\Delta P_S = K_{S1} h_S^{2.86} + K_{S2} A h_S^{4.42} + K_{S3} A h_S^{4.69} \quad (15.23)$$

where

$$K_{hS} = \frac{0.24 F_{hI} F_{hV} F_{hB} F_{hL} \rho^{0.64} C_P^{\frac{1}{3}} k^{\frac{2}{3}}}{\mu^{0.307} d_O^{0.36}} \quad (15.24)$$

$$K_{PS1} = \frac{1.298 F_{Pb} D_S (1 - B_C) \rho^{0.83} \mu^{0.17}}{p_T d_O^{0.17}} \quad (15.25)$$

$$K_{PS2} = \frac{0.5261 F_{Pb} F_{PL} p_C (1 - 2 B_C) (p_T - d_O) \rho^{0.83} \mu^{0.17}}{d_O^{1.17} F_O} \quad (15.26)$$

$$K_{PS3} = \frac{2.596 F_{Pb} F_{PL} (1 - 2 B_C) D_S \rho^{0.83} \mu^{0.17}}{p_T d_O^{0.17}} \quad (15.27)$$

$$K_{PS4} = \frac{0.2026 F_{PL} p_C p_T (p_T - d_O) \rho}{d_O F_O} \left(\frac{2}{D_S} + \frac{0.6 B_C}{p_T} \right) \quad (15.28)$$

$$K_{S1} = \frac{2 K_{PS1} - K_{PS3}}{K_{hS}^{2.86}} \quad (15.29)$$

$$K_{S2} = \frac{K_{PS2}}{K_{hS}^{4.42}} \quad (15.30)$$

$$K_{S3} = \frac{K_{PS4}}{K_{hS}^{4.69}} \quad (15.31)$$

where h_S = shell-side film transfer coefficient

v_S = fluid velocity for the shell-side

$$= \frac{F_O D_S}{p_T} (p_T - d_O) L_B \quad (15.32)$$

F_O = outside (tube-side) volumetric flowrate

B_C = baffle cut

D_S = shell diameter

p_C = pitch configuration factor ($p_C = 1$ for square pitch, $p_C = 0.866$ for triangular pitch)

- p_T = tube pitch (i.e. center-to-center distance between adjacent tubes)
 L_B = distance between baffles
 F_{hn} = correction factor to allow for the effect of the number of tube rows crossed (see Appendix C)
 F_{hw} = the window correction factor (see Appendix C)
 F_{hb} = the bypass stream correction factor (see Appendix C)
 F_{hL} = the leakage correction factor (see Appendix C)
 F_{Pb} = bypass correction factor for pressure drop to allow for flow between the tube bundle and the shell wall and is a function of the shell to bundle clearance (see Appendix C)
 F_{PL} = leakage correction factor to allow for leakage through the tube-to-baffle clearance and the baffle-to-shell clearance (see Appendix C)

The shell-side correction factors F_{hn} , F_{hw} , F_{hb} , F_{hL} , F_{Pb} and F_{PL} are adjustable parameters that characterize the flow pattern on the shell-side. The values depend on the design of the heat exchanger, the fluids being heated and cooled and the fouled condition on the shell-side. For preliminary design in the clean condition, reasonable values are $F_{hn} = F_{hw} = 1$, $F_{hb} = F_{hL} = 0.8$, $F_{Pb} = 0.8$ and $F_{PL} = 0.5$. As the shell-side fouls, bypassing and leakage will decrease, and the factors will tend towards 1 (but not necessarily reach 1). Choosing the most appropriate values is at the discretion of the designer.

Before these equations can be applied, a number of issues need to be considered:

1. *Shell diameter.* The equations for the shell-side feature the shell diameter, which is unknown at the conceptual design stage. This can be eliminated by considering the heat transfer area.

$$A = N_T \pi d_O L \quad (15.33)$$

where A = heat transfer area based on the tube outside surface (m^2)

N_T = total number of tubes (–)

d_O = tube outside diameter (m)

L = tube length (m)

For a square pitch, each tube is contained in an area of p_T^2 . The number of tubes can then be approximated as:

$$N_T = \frac{\pi D_S^2}{4 p_T^2} \quad (15.34)$$

For a triangular pitch, each tube is contained in an area of $0.866 p_T^2$. The number of tubes can then be generalized as:

$$N_T = \frac{\pi D_S^2}{4 p_C p_T^2} \quad (15.35)$$

where p_C = pitch configuration factor
 = 1 for square pitch
 = 0.866 for triangular pitch

In practice, the number of tubes in a given shell diameter will be less than that predicted by Equation 15.35 because of the mechanical features such as pass partition plates, not accounted for. Combining Equations 15.33 and 15.35 gives:

$$A = \frac{\pi^2 d_O D_S^2 L}{4 p_C p_T^2} \quad (15.36)$$

Equation 15.36 can be rearranged to give:

$$D_S = \left(\frac{4 p_C p_T^2 A}{\pi^2 d_O L} \right)^{\frac{1}{2}} \quad (15.37)$$

Equation 15.37 can be used to approximate the shell diameter if the tube length L is specified. This might be preferred, as standard tube lengths will tend to be used. Alternatively, the tube length to shell diameter (L/D_S) can be specified. If this is specified, then Equation 15.36 can be rearranged to give:

$$D_S = \left(\frac{4 p_C p_T^2 A}{\pi^2 d_O (L/D_S)} \right)^{\frac{1}{3}} \quad (15.38)$$

The ratio of tube length to shell diameter is usually in the range 5 to 10.

2. *Fluid velocity.* Equations 15.14, 15.15, 15.21 and 15.22 require knowledge of the fluid velocity. Liquid velocity on the tube-side is usually of the order of 1 to 3 $\text{m}\cdot\text{s}^{-1}$. On the shell-side, liquid velocity is usually of the order of 0.5 to 2 $\text{m}\cdot\text{s}^{-1}$. Gas velocity on both the tube-side and the shell-side typically vary in the range 5 to 70 $\text{m}\cdot\text{s}^{-1}$. The higher the pressure, the lower the gas velocity.

High velocities will result in high heat transfer coefficients and will tend to reduce fouling. However, high velocities also result in a high pressure drop. Also, if solids are present in the fluids, then high velocities can result in erosion. High velocities on the shell-side for liquids (typically greater than 3 $\text{m}\cdot\text{s}^{-1}$) can also induce vibration within the heat exchanger that can cause long-term damage to the heat exchanger.

3. *Pressure drop.* Rather than specify a fluid velocity, it is often preferred to specify the pressure drop across the heat exchanger. In retrofit situations, where a new heat exchanger is to be installed in an existing plant, the allowable pressure drop is often highly constrained. This is because in a retrofit situation, it is often desirable to avoid

the installation of new pumps. In the absence of a specific retrofit constraint for pressure drop, the value for liquids is normally in the range 0.35 to 0.7 bar. For low-viscosity liquids (less than $1 \text{ mN}\cdot\text{s}\cdot\text{m}^{-2}$), the pressure drop is likely to be less than 0.35 bar. For gases, the maximum allowable pressure drop varies typically between 1 bar for high-pressure gases (10 bar and above) down to 0.01 bar for gases under vacuum conditions. For gases, the cost associated with the pressure drop is more sensitive to aspects of the design outside of the immediate considerations for the heat exchanger design than is the case for liquids.

4. Fouling. Fouling is the accumulation of undesired material at the heat transfer surfaces. It is a transient process that begins with a clean heat transfer surface and continues until the point where the surface becomes fouled to the point where the heat exchanger can no longer be used effectively. At this point, the heat exchanger must be taken out of service and cleaned. Cleaning can be accomplished by mechanical or chemical processes. Mechanical cleaning involves the use of high-pressure water jets and so on. Chemical cleaning exploits cleaning fluids that react with and/or dissolve surface deposits. This usually involves creating a cleaning circuit involving the heat exchanger and a pump in which the cleaning fluid is recirculated through the heat exchanger at a relatively high velocity.

In some instances (e.g. cooling water circuits), chemicals can be added to the heat transfer fluids to inhibit fouling. The chemicals added to inhibit fouling and the means of cleaning depend on the nature of the fouling. Fouling can be classified as:

- *particulate*, in which solid particles suspended in the process stream are carried to the heat transfer surface and accumulate;
- *scaling*, in which solid material is precipitated from solution on the heat transfer surface through inverse solubility (e.g. calcium carbonate deposit from hardness in water);
- *crystallization*, in which solid material is crystallized from solution as a result of the change in temperature and/or the presence of nucleation sites at the surface;
- *freezing*, in which material is decreased in temperature locally to below its freezing point;
- *chemical reaction*, which results from reactions at the heat transfer surface (e.g. polymerization and cracking reactions);
- *corrosion*, in which the heat transfer surface is exposed to a corrosive fluid that reacts to produce byproducts of corrosion on the surface;
- *biological fouling* or *biofouling*, in which a layer of microorganisms grows on the heat transfer surface producing *slime*.

The rate at which fouling occurs depends on many factors, but the two most important are temperature and fluid

velocity. Fouling can be removed from the surface by high surface shear stress created by high fluid velocity. Indeed, for certain types of fouling, it has been demonstrated that for a given surface temperature, there is a threshold velocity above which fouling will not occur.

As pointed out in Section 15.1, fouling adds additional resistances to heat transfer on both the tube-side and the shell-side. If one of the fluids in the heat exchanger is significantly more fouling than the other, the fouling fluid is normally placed on the tube-side. If this fouling fluid placed on the tube-side has a particularly high tendency to foul (e.g. crude oil), then designing for a low pressure drop can be a false economy. As the fluid fouls the tube-side, not only will the overall heat transfer coefficient decrease significantly but the pressure drop will also increase significantly as a direct result of the fouling. If, for this highly fouling fluid, the exchanger had been designed for an initial high pressure drop on the tube-side, the resulting high velocity would not only have increased the initial rate of heat transfer, but also decreased the rate of deterioration of the rate of heat transfer and the rate of increase of pressure drop. An initial design for a tube-side pressure drop of 0.7 bar for a highly fouling fluid (e.g. crude oil) might result in a fouled pressure drop of say 4 to 5 bar after a year of operation. Had the initial design been for, say, 2.5 bar, this could have led to a lower pressure drop after a year of operation and an overall better performance.

In addition to the use of antifouling chemical agents to mitigate the effect of fouling on the tube-side, *twisted tubes* can be used rather than plain tubes. These have surface irregularities. Plain tubes can also be fitted on the inside with *tube inserts*. Twisted tubes and tube inserts promote additional turbulence and pressure drop and reduce the surface temperature of the tube to mitigate fouling. Tube inserts will be dealt with in more detail later.

The effect of fouling on the shell-side is more complex. On the shell-side, the heat transfer and pressure drop are also affected by the fouling changing the flow pattern. Bypassing and leakage reduces both the shell-side heat transfer coefficient and pressure drop. Fouling will tend to block the clearances, increasing the amount of cross-flow. This will have the effect of increasing the shell-side heat transfer coefficient and pressure drop. This increase will be countered by film resistances of the heat transfer surfaces. Initially, the fouling might even increase the heat transfer coefficient by promoting better cross-flow, only to decrease it later as the film resistance increases.

The segmented baffles normally used in shell-and-tube heat exchangers are not the best arrangement for fouling fluids placed on the shell-side, as stagnant zones are created by the flow pattern. Rather than use segmented baffles, helical flow baffles can be used, which induce a spiral flow pattern along the exchanger on the shell-side, eliminating the stagnant zones.

5. *Standard dimensions.* Applying the equations described here will result in certain specifications for heat transfer area, tube length and shell diameter. In practice, standard tube lengths, shell diameters and tube layouts are used. The preliminary design would then need to be adjusted to meet standard size and layout specifications.

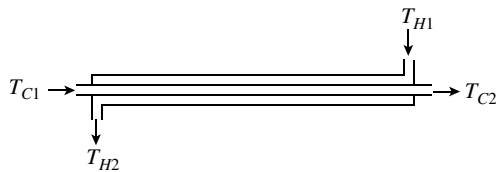
15.3 TEMPERATURE DIFFERENCES IN SHELL-AND-TUBE HEAT EXCHANGERS

Consider the heat exchange process in Figure 15.5a. The heat transfer arrangement shows a *concentric pipe* heat exchanger. The flow of the fluids in such a device is truly countercurrent. To understand the heat transfer, assume that it is a steady state process in which all of the fluid properties and overall heat transfer coefficient U are constant. It is also assumed that there is no phase change and that there is no heat loss from the system. The heat transfer process is shown in Figure 15.5b in a plot of heat transfer versus temperature. The slope of the ΔT line in Figure 15.5b is given by:

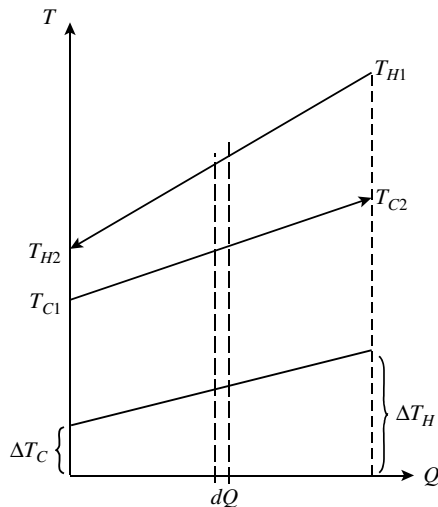
$$\frac{d(\Delta T)}{dQ} = \frac{\Delta T_H - \Delta T_C}{Q} \quad (15.39)$$

Applying Equation 15.39 across the differential element in Figure 15.5b:

$$dQ = U \cdot dA \cdot \Delta T \quad (15.40)$$



(a) Countercurrent heat exchange.



(b) Heat transfer versus temperature.

Figure 15.5 Heat exchange temperature difference.

Combining Equations 15.39 and 15.40 gives:

$$\frac{\Delta T_H - \Delta T_C}{Q} = \frac{d(\Delta T)}{U \cdot dA \cdot \Delta T} \quad (15.41)$$

Rearranging and integrating gives:

$$\int_0^A dA = \frac{Q}{\Delta T_H - \Delta T_C} \cdot \frac{1}{U} \int_{\Delta T_C}^{\Delta T_H} \frac{d(\Delta T)}{\Delta T} \quad (15.42)$$

Thus:

$$A = \frac{Q}{\Delta T_H - \Delta T_C} \cdot \frac{1}{U} \ln \frac{\Delta T_H}{\Delta T_C} \quad (15.43)$$

$$Q = UA \left[\frac{\Delta T_H - \Delta T_C}{\ln \frac{\Delta T_H}{\Delta T_C}} \right] \quad (15.44)$$

$$= UA \Delta T_{LM} \quad (15.45)$$

where ΔT_{LM} = logarithmic mean temperature difference

Note that this result would have been obtained if

- the fluid positions inside and outside of the tube in Figure 15.5b had been reversed;
- the direction of one of the fluids had been reversed giving cocurrent flow;
- either fluid had been isothermal.

If both fluids had been isothermal, then the logarithmic and average temperature differences would be equal and $\Delta T_H = \Delta T_C = \Delta T$. It should also be noted that:

$$\frac{\Delta T_H - \Delta T_C}{\ln \frac{\Delta T_H}{\Delta T_C}} = \frac{\Delta T_C - \Delta T_H}{\ln \frac{\Delta T_C}{\Delta T_H}} \quad (15.46)$$

Although the result in Equation 15.45 applies to both countercurrent and cocurrent flow, in practice, cocurrent flow is almost never used as, given fixed fluid inlet and outlet temperatures, the logarithmic mean temperature difference for countercurrent flow is always larger. This in turn leads to smaller surface area requirements. Also, as shown in Figure 15.6a for countercurrent flow, the final temperature of the hot fluid can be lower than the final temperature of the cold fluid (sometimes known as *temperature cross*), whereas in Figure 15.6b, it is clear that there can never be a temperature cross.

For a given heat duty and overall heat transfer coefficient, the 1–1 design (1 shell pass – 1 tube pass), as illustrated in Figure 15.7a, offers the lowest requirement for surface area for shell-and-tube heat exchangers. Many flow arrangements other than the 1–1 design exist, the most common of which is the 1–2 design (1 shell pass – 2 tube passes), as illustrated in Figure 15.7b. Because the flow arrangement involves part countercurrent and part cocurrent flow, the effective temperature difference for heat exchange is

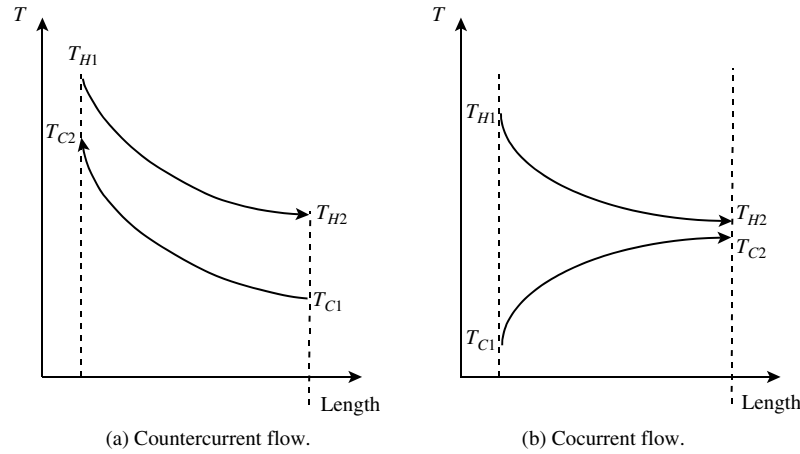


Figure 15.6 Fluid temperatures can never cross in a cocurrent heat exchanger.

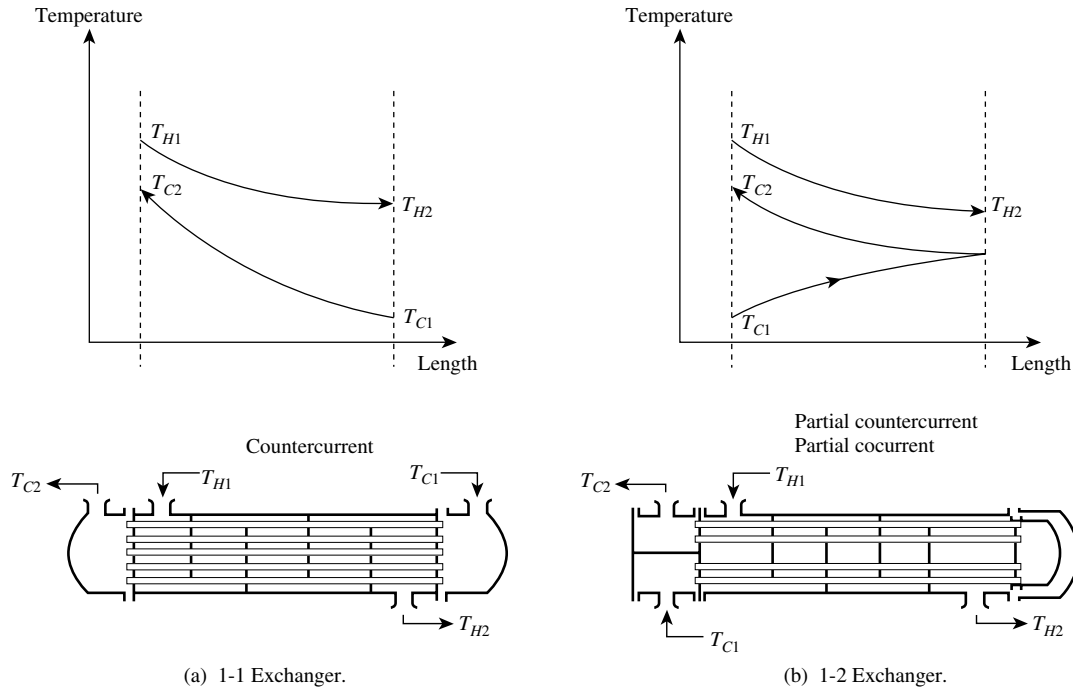


Figure 15.7 1-1 shells approach pure countercurrent flow, whereas 1-2 shells exhibit partial countercurrent and partial cocurrent flow.

reduced compared to a pure countercurrent device. This is accounted for, in design, by the introduction of the F_T factor into the basic heat exchanger design equation⁶:

$$Q = UA\Delta T_{LM} F_T \quad \text{where} \quad 0 < F_T < 1 \quad (15.47)$$

Thus, for a given exchanger duty and overall heat transfer coefficient, the 1-2 design needs a larger area than the 1-1 design. However, the 1-2 design offers many practical advantages. These include, in particular, allowance for thermal expansion, easy mechanical cleaning and good heat transfer coefficients on the tube-side (due to higher velocity).

The F_T correction factor is usually correlated in terms of two dimensionless ratios, the ratio of the two heat capacity

flowrates (R) and the thermal effectiveness of the exchanger (P)⁶:

$$F_T = f(R, P) \quad (15.48)$$

where

$$R = CP_C/CP_H = (T_{H1} - T_{H2})/(T_{C2} - T_{C1}) \quad (15.49)$$

and

$$P = (T_{C2} - T_{C1})/(T_{H1} - T_{C1}) \quad (15.50)$$

Note therefore that F_T depends only on the inlet and outlet temperatures of the streams in a 1-2 heat exchanger. An expression can be developed for the F_T in a 1-2 heat exchanger. The proof is lengthy and can be found elsewhere^{1,6}.

For $R \neq 1$:

$$F_T = \frac{\sqrt{R^2 + 1} \ln \left[\frac{(1 - P)}{(1 - RP)} \right]}{(R - 1) \ln \left[\frac{(2 - P(R + 1 - \sqrt{R^2 + 1}))}{(2 - P(R + 1 + \sqrt{R^2 + 1}))} \right]} \quad (15.51)$$

For $R = 1$:

$$F_T = \frac{\left[\frac{\sqrt{2}P}{(1 - P)} \right]}{\ln \left[\frac{(2 - P(2 - \sqrt{2}))}{(2 - P(2 + \sqrt{2}))} \right]} \quad (15.52)$$

The shape of this function is illustrated in Figure 15.8. It can be seen that the slope of the F_T curve for a given R becomes very steep and approaches an asymptote as the thermal effectiveness P increases.

Three basic situations can be encountered when using 1–2 exchangers (Figure 15.8):

1. The final temperature of the hot stream is higher than the final temperature of the cold, as illustrated in Figure 15.8a. This is a so-called *temperature approach*. This situation is straightforward to design for, since it can always be accommodated in a single 1–2 shell.
2. The final temperature of the hot stream is slightly lower than the final temperature of the cold stream, as illustrated in Figure 15.8b. This is a so-called *temperature cross*. This situation is usually straightforward to design for, provided the temperature cross is small, because again it can probably be accommodated in a single shell. However, the decrease in F_T increases the heat transfer area requirements significantly.
3. As the amount of temperature cross increases, however, problems are encountered as illustrated in Figure 15.8c. The F_T decreases significantly, causing a dramatic increase in the heat transfer area requirements. Local reversal of heat flow may also be encountered, which is wasteful in heat transfer area. The design may even become infeasible. Thus, for a given R , the design of the heat exchanger becomes less and less efficient as the asymptote for the F_T curve is approached.

The maximum temperature cross that can be tolerated is often set by rules of thumb, for example, $F_T > 0.75^1$. It is important to avoid low values of F_T because:

1. Low values of F_T indicate inefficient use of the heat transfer area.
2. Any violation of the simplifying assumptions used in the approach tends to have a particularly significant effect in areas of the F_T chart where slopes are particularly steep.
3. Any uncertainties or inaccuracies in design data also have a more significant effect when slopes are steep.

Consequently, to be confident in a design, those parts of the F_T chart where slopes are steep should be avoided, irrespective of $F_T > 0.75^7$. A simple method to achieve this is based upon the fact that for any value of R there is a maximum asymptotic value for P , say P_{max} , which is given as F_T tends to $-\infty$, and is given by⁷:

$$P_{max} = \frac{2}{R + 1 + \sqrt{R^2 + 1}} \quad (15.53)$$

Equation 15.53 is derived in Appendix D. Practical designs will be limited to some fraction of P_{max} , that is⁷:

$$P = X_P P_{max}, 0 < X_P < 1 \quad (15.54)$$

where X_P is a constant defined by the designer.

A line of constant X_P is compared with a line of constant F_T in Figure 15.9⁸. It can be seen that the line of constant X_P avoids the regions of steep slope.

Situations are often encountered where the design is infeasible in a single 1–2 shell, because the F_T is too low or the F_T slope too large. If this happens, either different types of shell or multiple shell arrangements must be considered^{1–3}. Here, consideration will be restricted to multiple shell arrangements of the 1–2 type. By using two 1–2 shells in series (Figure 15.10), the temperature cross in each individual shell is reduced below that for a single 1–2 shell for the same duty. The profiles shown in Figure 15.10 could in principle be achieved either by two 1–2 shells in series or by a single 2–4 shell.

For a number of 1–2 shells in series, a transformation can be developed based on the fact that for N_{SHELLS} in series, each shell pass has the same value of F_T , which also equals the F_T across all N_{SHELLS} passes⁶. Also, all values of P of each shell pass (P_{1-2}) are equal, but not equal to the value of P across all N_{SHELLS} (P_{N-2N}). Of course, R is constant across all shells and the overall design.

For $R \neq 1$ ⁶:

$$P_{N-2N} = \frac{1 - \left[\frac{1 - P_{1-2}R}{1 - P_{1-2}} \right]^{N_{SHELLS}}}{R - \left[\frac{1 - P_{1-2}R}{1 - P_{1-2}} \right]^{N_{SHELLS}}} \quad (15.55)$$

For $R = 1$ ⁶:

$$P_{N-2N} = \frac{P_{1-2}N_{SHELLS}}{P_{1-2}N_{SHELLS} - P_{1-2} + 1} \quad (15.56)$$

Thus, given an overall value of P_{N-2N} for the duty involving N_{SHELLS} , Equations 15.55 and 15.56 allow the value of P_{1-2} to be calculated for each shell. To do this, first define a variable Z :

$$Z = \left[\frac{1 - P_{1-2}R}{1 - P_{1-2}} \right]^{N_{SHELLS}} \quad (15.57)$$

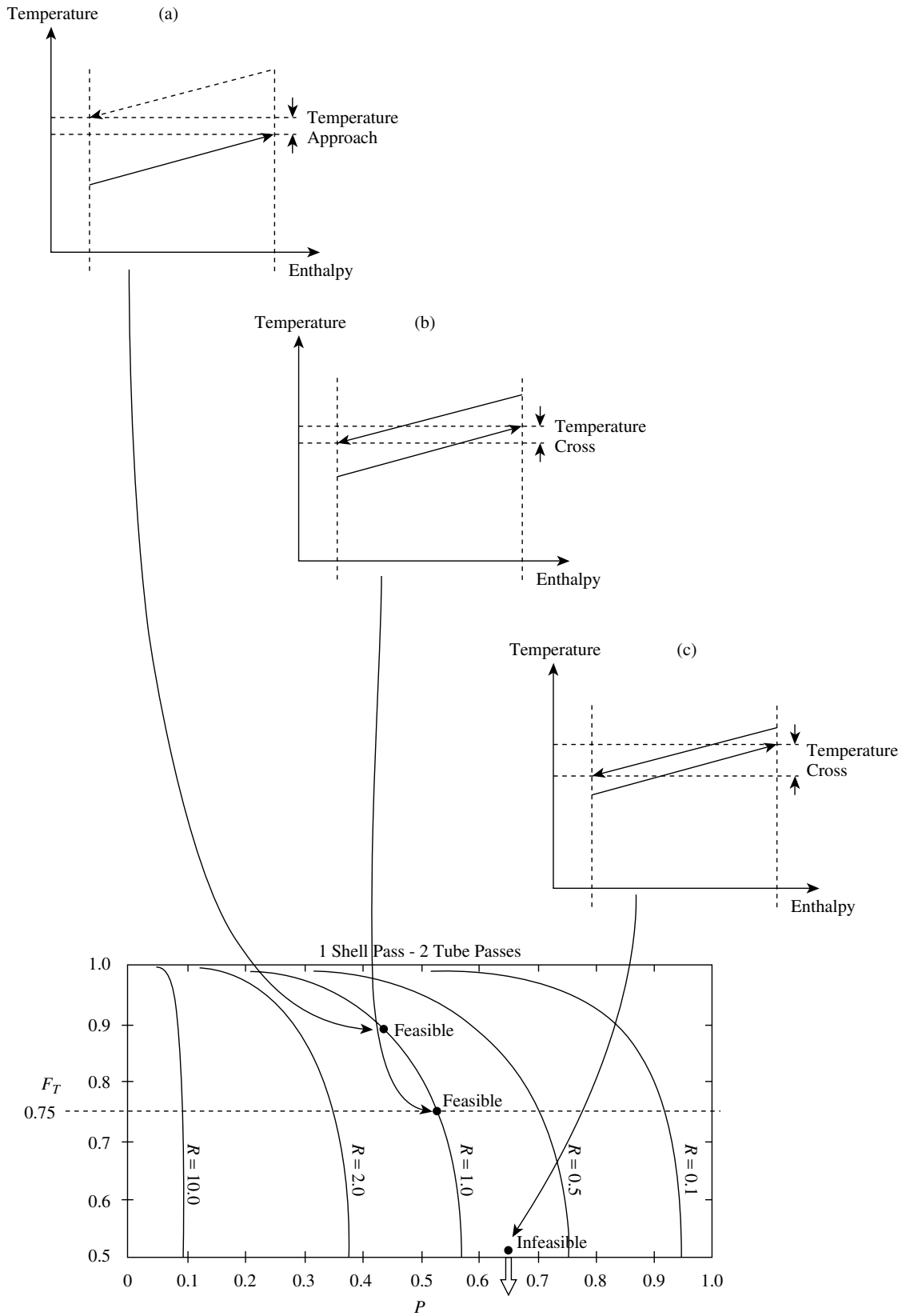


Figure 15.8 Designs with a temperature approach or small temperature cross can be accommodated in a single 1-2 shell, whereas designs with a large temperature cross become infeasible. (From Ahmad S, Linnhoff B and Smith R, 1988, *Trans ASME J Heat Transfer*, 110: 304, reproduced by permission of the American Society of Mechanical Engineers).

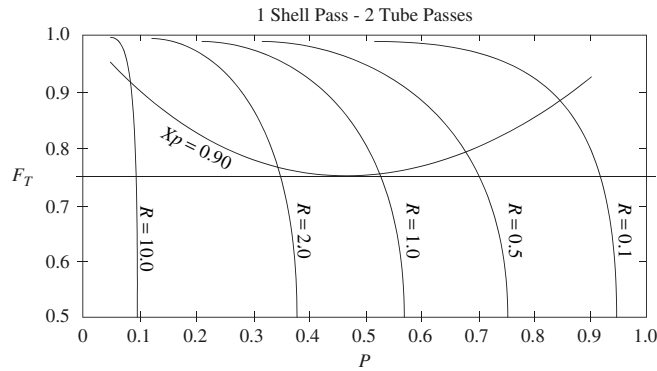


Figure 15.9 The X_P parameter avoids steep slopes on the F_T curves, whereas minimum F_T does not. (From Ahmad S, Linnhoff B and Smith R, 1990, *Computers and Chem Eng*, 7: 751, reproduced by permission of Elsevier Ltd).

Substituting Z into Equation 15.55 and rearranging gives:

$$Z = \frac{1 - P_{N-2N}R}{1 - P_{N-2N}} \quad (15.58)$$

Thus, starting with the overall P_{N-2N} for the duty, Z is first calculated from Equation 15.58. Then P_{1-2} is calculated for each shell by inverting Equation 15.57 and substituting the value of Z :

$$P_{1-2} = \frac{Z^{1/N_{SHELLS}} - 1}{Z^{1/N_{SHELLS}} - R} \quad \text{for } R \neq 1 \quad (15.59)$$

For $R = 1$, a simple rearrangement of Equation 15.56 gives:

$$P_{1-2} = \frac{P_{N-2N}}{P_{N-2N} - N_{SHELLS} P_{N-2N} + N_{SHELLS}} \quad (15.60)$$

Traditionally, the designer would approach a design for an individual unit by trial and error. Starting by assuming one shell, the F_T can be evaluated. If the F_T is not acceptable, then the number of shells in series is progressively increased until a satisfactory value of F_T is obtained for each shell. For a number of 1–2 shells in series:

$$F_{1-2} < F_{2-4} < F_{3-6} < F_{4-8} \dots < 1 \quad (15.61)$$

Adopting the design criterion given by Equation 15.54 as the basis, any need for trial and error can be eliminated since an explicit expression for the number of shells for a given unit is derived in Appendix E⁷.

$R \neq 1$:

$$N_{SHELLS} = \frac{\ln \left(\frac{1 - RP}{1 - P} \right)}{\ln W} \quad (15.62)$$

where

$$W = \frac{R + 1 + \sqrt{R^2 + 1} - 2RX_P}{R + 1 + \sqrt{R^2 + 1} - 2X_P} \quad (15.63)$$

$R = 1$:

$$N_{SHELLS} = \frac{\left(\frac{P}{1 - P} \right) \left(1 + \frac{\sqrt{2}}{2} - X_P \right)}{X_P} \quad (15.64)$$

X_P is chosen to satisfy the minimum allowable F_T (for example, for $F_{Tmin} > 0.75$, $X_P = 0.9$ is used). Once the real (noninteger) number of shells is calculated from Equation 15.62 or 15.64, this is rounded up to the next largest number to obtain the number of shells. Generally,

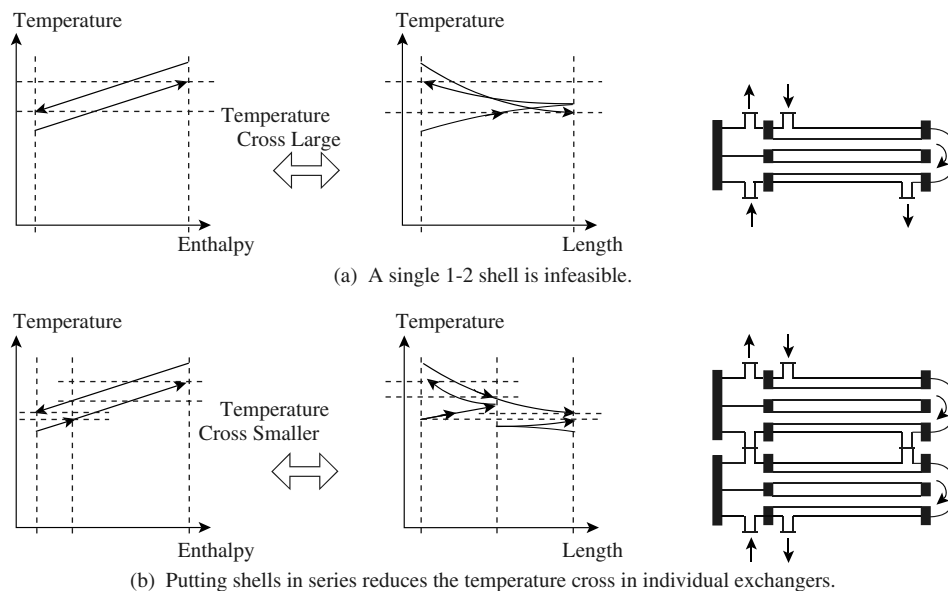


Figure 15.10 A large overall temperature cross requires shells in series to reduce the cross in individual exchangers. (From Ahmad S, Linnhoff B and Smith R, 1988, *Trans ASME J Heat Transfer*, 110: 304, reproduced by permission of the American Society of Mechanical Engineers).

the smaller the number of shells for a given overall duty, the cheaper will be the design. The higher the value of X_P chosen, the larger will be the number of shells, but the safer the design. Thus, a compromise is required. A value of $X_P = 0.9$ is reasonable for conceptual design.

In addition to the F_T limiting the temperature cross in each shell and dividing the overall duty into a number of shells, there is a maximum physical size that can be fabricated in a single shell. For shell-and-tube heat exchangers with removable tube bundles, the maximum size of shell is around 1000 m². However, a significantly lower figure might well be preferred for maintenance and cleaning purposes. Fixed bundle heat exchangers can be much larger, typically up to 4500 m².

Example 15.1 A hot stream is to be cooled from 300 to 100°C by exchange with a cold stream being heated from 60 to 200°C in a single unit. 1–2 shell-and-tube heat exchangers are to be used subject to $X_P = 0.9$. The duty for the exchanger is 3.5 MW and the overall heat transfer coefficient is estimated to be 100 W·m⁻²·K⁻¹. Calculate:

- number of shells required
- P_{1-2} for each shell
- F_T for the shells in series
- heat transfer area

Solution

$$\begin{aligned}
 \text{a.} \quad R &= \frac{T_{H1} - T_{H2}}{T_{C2} - T_{C1}} \\
 &= \frac{300 - 100}{200 - 60} \\
 &= 1.4286 \\
 P_{N-2N} &= \frac{T_{C2} - T_{C1}}{T_{H1} - T_{C1}} \\
 &= \frac{200 - 60}{300 - 60} \\
 &= 0.5833 \\
 W &= \frac{R + 1 + \sqrt{R^2 + 1} - 2RX_P}{R + 1 + \sqrt{R^2 + 1} - 2X_P} \\
 &= 0.6748 \\
 N_{SHELLS} &= \frac{\ln \left[\frac{(1 - RP_{N-2N})}{(1 - P_{N-2N})} \right]}{\ln W} \\
 &= 2.33
 \end{aligned}$$

Thus, the unit requires three shells.

$$\begin{aligned}
 \text{b.} \quad Z &= \frac{1 - P_{N-2N}R}{1 - P_{N-2N}} \\
 &= \frac{1 - 0.5833 \times 1.4286}{1 - 0.5833} \\
 &= 0.4 \\
 P_{1-2} &= \frac{Z^{1/N_{SHELLS}} - 1}{Z^{1/N_{SHELLS}} - R}
 \end{aligned}$$

$$\begin{aligned}
 &= \frac{(0.4)^{\frac{1}{3}} - 1}{(0.4)^{\frac{1}{3}} - 1.4286} \\
 &= 0.3805
 \end{aligned}$$

$$\text{c.} \quad F_T = \frac{\sqrt{R^2 + 1} \ln \left[\frac{(1 - P)}{(1 - RP)} \right]}{(R - 1) \ln \left[\frac{[2 - P(R + 1 - \sqrt{R^2 + 1})]}{[2 - P(R + 1 + \sqrt{R^2 + 1})]} \right]}$$

Substituting $R = 1.4286$ and $P = 0.3805$:

$$\begin{aligned}
 F_T &= 0.86 \\
 \text{d.} \quad A &= \frac{Q}{U \Delta T_{LM} F_T} \\
 &= \frac{3.5 \times 10^6}{100 \times 65.48 \times 0.86} \\
 &= 619 \text{ m}^2
 \end{aligned}$$

15.4 ALLOCATION OF FLUIDS IN SHELL-AND-TUBE HEAT EXCHANGERS

In a new design, the allocation of streams to the tube-side or shell-side will be unknown at this stage. The issues to be considered for the allocation include:

a. *Materials of construction.* If expensive materials of construction are required for one of the fluids because of its corrosive nature or high temperature, then this fluid should normally be allocated to the tube-side. This can reduce the cost of expensive materials of construction.

b. *Fouling.* The fluid that has the greatest tendency to foul the heat transfer surfaces should be allocated to the tube-side. This will give better control over the design fluid velocity, and the higher allowable velocity in the tubes will reduce fouling. Stagnant zones and zones with low velocity can occur on the shell-side, leading to accelerated fouling. Also, the tubes are easier to clean than the shell-side.

c. *Operating pressures.* The stream with the higher pressure should be allocated to the tube-side. The smaller the diameter of a tube, the thinner the wall needed to contain the same pressure. The tubes are therefore more effective at containing high pressure than the shell.

d. *Pressure drop.* For the same pressure drop, higher heat transfer coefficients will be obtained on the tube-side than the shell-side. The fluid with the lowest allowable pressure drop should normally be allocated to the tube-side.

e. *Viscosity.* Generally, a higher heat transfer coefficient will be obtained by allocating the more viscous material to the shell-side, provided the flow is turbulent. The critical Reynolds number for turbulent flow on the shell-side is in the region of 200.

f. *Stream flowrates.* Allocating the fluid with the lower flowrate to the shell-side will normally allow a higher heat transfer coefficient to be obtained for that fluid and give the most economic design.

g. *Fluid temperatures.* Placing the hotter fluid in the tubes will reduce the shell surface temperature, and hence the need for lagging to reduce heat loss and might be desirable for safety reasons.

These general guidelines can contradict each other. If this is the case, some compromise must be made.

Example 15.2 A crude oil stream is to be preheated by recovering heat from a kerosene product in a shell-and-tube heat exchanger. The flowrates, temperatures and physical properties (at the mean temperatures) are given in Table 15.5.

Table 15.5 Data for a heat recovery problem.

	Kerosene	Crude oil
Flowrate ($\text{m}^3 \cdot \text{s}^{-1}$)	0.037	0.103
Initial temperature ($^{\circ}\text{C}$)	200	35
Final temperature ($^{\circ}\text{C}$)	95	75
Density ($\text{kg} \cdot \text{m}^{-3}$)	730	830
Heat capacity ($\text{J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$)	2470	2050
Viscosity ($\text{N} \cdot \text{s} \cdot \text{m}^{-2}$)	4.0×10^{-4}	3.6×10^{-3}
Thermal conductivity ($\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$)	0.132	0.133

The exchanger type to be used is 1 shell-pass, 2 tube-passes. Because crude oil will have a greater tendency to foul than kerosene, crude oil will be allocated to the tube-side. Assume the fouling coefficient for kerosene on the shell-side is $5000 \text{ W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$ and for the crude oil on the tube-side is $2000 \text{ W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$.

Steel tubes will be used and the following assumptions can be made regarding the heat exchanger geometry:

$$\begin{aligned} d_o &= 20 \text{ mm} \\ d_i &= 16 \text{ mm} \\ p_T &= 1.25d_o \\ p_C &= 1(\text{square pitch}) \\ B_C &= 0.25 \\ L/D_S &= 7 \end{aligned}$$

- Calculate the number of shells required and the F_T based on $X_P = 0.9$.
- Estimate the overall heat transfer coefficient, heat transfer area and pressure drops for the tube-side and shell-side assuming a fluid velocity of $1 \text{ m} \cdot \text{s}^{-1}$ on both the tube-side and shell-side. Assume the flow on the shell-side is characterized by the clean condition, that is, $F_{hm} = F_{hw} = 1$, $F_{hb} = F_{hL} = 0.8$, $F_{pb} = 0.8$ and $F_{PL} = 0.5$.

- Estimate the overall heat transfer coefficient and heat transfer area for a tube-side pressure drop of 0.3 bar and a shell-side pressure drop of 0.6 bar. Again, assume the flow on the shell-side is characterized by the clean condition.
- Repeat the calculation in Part c above but assuming the flow is characterized by the fouled condition. Assume $F_{hm} = F_{hw} = F_{hb} = F_{hL} = F_{pb} = F_{PL} = 1$.

Solution

- The calculations can be conveniently carried out in a spreadsheet.

$$R = \frac{T_{H1} - T_{H2}}{T_{C2} - T_{C1}} = \frac{200 - 95}{75 - 35} = 2.625$$

$$P = \frac{T_{C2} - T_{C1}}{T_{H1} - T_{C1}} = \frac{75 - 35}{200 - 35} = 0.2424$$

$$W = \frac{R + 1 + \sqrt{R^2 + 1} - 2RX_P}{R + 1 + \sqrt{R^2 + 1} - 2X_P} = 0.3688$$

$$N_{SHELLS} = \frac{\ln \left[\frac{1 - RP}{1 - P} \right]}{\ln W} = 0.74$$

Thus, the unit requires 1 shell.

$$F_T = \frac{\sqrt{R^2 + 1} \ln \left[\frac{(1 - P)}{(1 - RP)} \right]}{(R - 1) \ln \left\{ \frac{[2 - P(R + 1 - \sqrt{R^2 + 1})]}{[2 - P(R + 1 + \sqrt{R^2 + 1})]} \right\}}$$

Substituting $R = 2.625$ and $P = 0.2424$:

$$F_T = 0.90$$

$$\begin{aligned} \text{b. } K_{hT} &= C \left[\frac{k}{d_i} \right] Pr^{\frac{1}{3}} \left[\frac{d_i \rho}{\mu} \right]^{0.8} \\ &= 0.023 \left[\frac{0.133}{0.016} \right] \left[\frac{2050 \times 3.6 \times 10^{-3}}{0.133} \right]^{\frac{1}{3}} \\ &\quad \times \left[\frac{0.016 \times 830}{3.6 \times 10^{-3}} \right]^{0.8} \\ &= 520.4 \end{aligned}$$

For an assumed tube-side velocity of $1 \text{ m} \cdot \text{s}^{-1}$:

$$\begin{aligned} h_T &= K_{hT} v_T^{0.8} \\ &= 520.4 \times 1^{0.8} \\ &= 520.4 \text{ W} \cdot \text{m}^{-2} \cdot \text{K}^{-1} \end{aligned}$$

$$\begin{aligned} K_{hS} &= \frac{0.24 F_{hm} F_{hw} F_{hb} F_{hL} \rho^{0.64} C_P^{\frac{1}{3}} k^{\frac{2}{3}}}{\mu^{0.307} d_o^{0.36}} \\ &\quad \frac{0.24 \times 1 \times 1 \times 0.8 \times 0.8}{\times (730)^{0.64} \times (2470)^{\frac{1}{3}} \times (0.132)^{2/3}} \\ &= \frac{(4 \times 10^{-4})^{0.307} \times (0.02)^{0.36}}{1653} \end{aligned}$$

For an assumed shell-side velocity of $1 \text{ m}\cdot\text{s}^{-1}$:

$$\begin{aligned} h_S &= K_{hS} v_S^{0.64} \\ &= 1653 \times 1^{0.64} \\ &= 1653 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \end{aligned}$$

Taking the wall coefficient from Table 15.4, the overall heat transfer coefficient can be estimated from:

$$\begin{aligned} \frac{1}{U} &= \frac{1}{h_S} + \frac{1}{h_{SF}} + \frac{1}{h_W} + \frac{d_o}{d_i} \frac{1}{h_{TF}} + \frac{d_o}{d_i} \frac{1}{h_T} \\ &= \frac{1}{1653} + \frac{1}{5000} + \frac{1}{20,200} \\ &\quad + \frac{0.02}{0.016} \left(\frac{1}{2000} + \frac{1}{520.4} \right) \\ U &= 257.6 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \\ \Delta T_{LM} &= \frac{(200 - 75) - (95 - 35)}{\ln \left[\frac{200 - 75}{95 - 35} \right]} \\ &= 88.6^\circ\text{C} \\ A &= \frac{Q}{U \Delta T_{LM} F_T} \\ &= \frac{0.037 \times 730 \times 2470 \times (200 - 95)}{257.4 \times 88.6 \times 0.9} \\ &= 341 \text{ m}^2 \end{aligned}$$

For the tube-side pressure drop:

$$\begin{aligned} K_{PT} &= 0.092 \rho^{0.8} \mu^{0.2} d_T^{-1.2} \\ &= 0.092 \times (830)^{0.8} \times (3.6 \times 10^{-3})^{0.2} \times (0.016)^{-1.2} \\ &= 923.4 \end{aligned}$$

$$\begin{aligned} \Delta P_T &= K_{PT} \frac{A d_T^2}{4 F_T d_O} v_T^{2.8} + 1.25 N_{PT} \rho v_T^2 \\ &= \frac{923.4 \times 341 \times (0.016)^2 \times 1^{2.8}}{4 \times 0.103 \times 0.02} + 1.25 \times 2 \times 830 \times 1^2 \\ &= 11,900 \text{ N}\cdot\text{m}^{-2} \end{aligned}$$

For the shell-side pressure drop, first estimate the shell diameter:

$$\begin{aligned} D_S &= \left(\frac{4 p_C p_T^2 A}{\pi^2 d_O (L/D_S)} \right)^{\frac{1}{3}} \\ &= \left(\frac{4 \times 1 \times 0.025^2 \times 341}{\pi^2 \times 0.02 \times 7} \right)^{\frac{1}{3}} \\ &= 0.85 \text{ m} \\ K_{PS1} &= \frac{1.298 F_{pb} D_S (1 - B_C) \rho^{0.83} \mu^{0.17}}{p_T d_O^{0.17}} \\ &= \frac{1.298 \times 0.8 \times 0.85 \times (1 - 0.25)}{0.025 \times (0.02)^{0.17}} \\ &= 3246 \end{aligned}$$

$$\begin{aligned} K_{PS2} &= \frac{0.5261 F_{pb} F_{PL} p_C (1 - 2 B_C) (p_T - d_O) \rho^{0.83} \mu^{0.17}}{d_O^{1.17} F_I} \\ &= \frac{0.5261 \times 0.8 \times 0.5 \times 1 \times (1 - 2 \times 0.25)}{\times (0.025 - 0.02) \times (730)^{0.83} \times (4 \times 10^{-4})^{0.17}} \\ &= \frac{(0.02)^{1.17} \times 0.037}{87.01} \end{aligned}$$

$$\begin{aligned} K_{PS3} &= \frac{2.596 F_{pb} F_{PL} (1 - 2 B_C) D_S \rho^{0.83} \mu^{0.17}}{p_T d_O^{0.17}} \\ &= \frac{2.596 \times 0.8 \times 0.5 \times (1 - 2 \times 0.25) \times 0.85}{\times (730)^{0.83} \times (4 \times 10^{-4})^{0.17}} \\ &= \frac{0.025 \times (0.02)^{0.17}}{2164} \end{aligned}$$

$$\begin{aligned} K_{PS4} &= \frac{0.2026 F_{PL} p_C p_T (p_T - d_O) \rho}{d_O F_O} \left[\frac{2}{D_S} + \frac{0.6 B_C}{p_T} \right] \\ &= \frac{0.2026 \times 0.5 \times 1 \times 0.025 (0.025 - 0.02) \times 730}{0.02 \times 0.037} \\ &\quad \times \left[\frac{2}{0.85} + \frac{0.6 \times 0.25}{0.025} \right] \\ &= 104.3 \end{aligned}$$

$$\begin{aligned} \Delta P_S &= 2 K_{PS1} v_S^{1.83} + K_{PS2} A v_S^{2.83} - K_{PS3} v_S^{1.83} + K_{PS4} A v_S^3 \\ &= 2 \times 3246 \times 1^{1.83} + 87.01 \times 341 \times 1^{2.83} - 2164 \\ &\quad \times 1^{1.83} + 104.3 \times 341 \times 1^3 \\ &= 69,600 \text{ N}\cdot\text{m}^{-2} \end{aligned}$$

The shell-side pressure drop is high. The fluid velocity on the shell-side could be decreased and the calculation repeated until an acceptable pressure drop is obtained. However, the approach allows the pressure drop to be specified directly.

c. The tube-side and shell-side pressure drops are now fixed to be:

$$\begin{aligned} \Delta P_T &= 30,000 \text{ N}\cdot\text{m}^{-2} \\ 30,000 &= K_{PT1} A h_T^{3.5} + K_{PT2} h_T^{2.5} \end{aligned} \quad (15.65)$$

$$\begin{aligned} \Delta P_S &= 60,000 \text{ N}\cdot\text{m}^{-2} \\ 60,000 &= K_{S1} h_s^{2.86} + K_{S2} A h_s^{4.42} + K_{S3} A h_s^{4.69} \end{aligned} \quad (15.66)$$

Thus, h_T and h_S can be varied by trial and error to satisfy these two equations. At each new value of h_T and h_S , the U and A need to be calculated for fixed Q , ΔT_{LM} and F_T . Also, for each iteration, the new shell diameter needs to be calculated in order to update K_{PS1} , K_{PS3} , K_{PS4} , K_{S1} and K_{S3} .

For flow characterized on the shell-side by the clean condition:

$$\begin{aligned} F_{hm} &= F_{hw} = 1 \\ F_{hb} &= F_{hL} = 0.8 \\ F_{pb} &= 0.8, F_{PL} = 0.5 \end{aligned}$$

The calculation is iterative and can be conveniently carried out using a solver in a spreadsheet to satisfy the Equations 15.65 and 15.66 simultaneously. To reach the two variables simultaneously, the objective can be set up such that the difference between the

left- and right-hand sides of Equations 15.65 and 15.66 are given the values, say *Objective 1* and *Objective 2* respectively. Then the spreadsheet solver is used to search for:

$$(\text{Objective 1})^2 + (\text{Objective 2})^2 = 0 \text{ (within a tolerance)}$$

The result is:

$$\begin{aligned}\Delta P_T &= 30,000 \text{ N}\cdot\text{m}^{-2} \\ h_T &= 721 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \\ \Delta P_S &= 60,000 \text{ N}\cdot\text{m}^{-2} \\ h_S &= 1662 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \\ U &= 311.5 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \\ A &= 282 \text{ m}^2 \\ D_S &= 0.8 \text{ m}\end{aligned}$$

Also, given the specified pressure drop, the corresponding fluid velocities can be calculated from:

$$\begin{aligned}v_T &= \left(\frac{h_T}{K_{ht}} \right)^{\frac{1}{0.8}} \\ &= 1.5 \text{ m}\cdot\text{s}^{-1} \\ v_S &= \left(\frac{h_S}{K_{hs}} \right)^{\frac{1}{0.64}} \\ &= 1.0 \text{ m}\cdot\text{s}^{-1}\end{aligned}$$

d. Fouling will tend to reduce *bypass* and leakage on the shell-side. As an extreme case, all shell-side factors can be assumed to be 1.0 to illustrate the trend that fouling will have. As in Part C above, h_T and h_S are varied simultaneously to satisfy the pressure drop specifications. The result is:

$$\begin{aligned}\Delta P_T &= 30,000 \text{ N}\cdot\text{m}^{-2} \\ h_T &= 731 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \\ \Delta P_S &= 60,000 \text{ N}\cdot\text{m}^{-2} \\ h_S &= 2226 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \\ U &= 329.7 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \\ A &= 267 \text{ m}^2 \\ D_S &= 0.78 \text{ m}\end{aligned}$$

The effect of the fouling on the shell-side flow is to increase the cross-flow and increase the overall heat transfer coefficient for a fixed pressure drop (assuming the same fouling coefficients in both cases).

Given the uncertainties associated with the calculations, especially those on the shell-side, a sensible design basis for the heat transfer area specification would be the shell-side flow characterized by the clean condition. Of course, the fouling coefficients for the shell-side and tube-side should be included to account for the surface fouling resistance.

This approach provides a preliminary specification for the heat exchanger. The actual heat exchanger will be restricted to standard tube lengths, tube layout and shell size. The preliminary design would then be adjusted up to meet standard size and layout specifications.

15.5 EXTENDED SURFACE TUBES

In situations in which the film transfer coefficient on the outside of the tubes in a heat exchanger is much lower than the inside, the outside becomes the controlling coefficient. Considering Equation 15.13 for the overall heat transfer coefficient, if the outside film is controlling, then no matter what is done to increase the inside coefficient, it will have little effect on the overall heat transfer coefficient. This will happen when heating or cooling viscous liquids and gases. In such circumstances, a more compact unit will result if a greater surface area is presented to the controlling fluid by the use of extended-surface tubes. Extended surfaces increase the rate of heat transfer per unit length of tube and the resulting exchanger can be much smaller and cheaper than the corresponding plain-tube exchanger. The external tube surface can be extended in one of two general ways:

- *Integrally formed tubes*, in which the extended surface is produced by cold-forming fins onto the surface of the tube by extrusion of the parent tube.
- *Nonintegrally formed fins*, in which the surface is extended by attaching pieces of metal in the form of longitudinal or transverse strips, wire or spines by welding, brazing, grooving and peening, or shrink fitting the extended surface to the tube. The method of fixing the extended surface to the parent tube might create a resistance to heat transfer.

By far the most common design of extended surface tube uses “high” transverse fins, as illustrated in Figure 15.11a. High transverse fins can increase the surface area by a factor of up to 16 relative to plain tubes. Such tubes are commonly used in heat exchangers for cooling duties where hot fluid is passed through the inside of horizontal tubes with ambient air from a fan flowing vertically across the outside. These *air-cooled heat exchangers* or *fin-fan exchangers* are illustrated in Figures 15.11b and 15.11c. The ambient air can be drawn across the tubes in an *induced draft* arrangement, as shown in Figure 15.11b, or driven across the tubes in a *forced draft* arrangement, as shown in Figure 15.11c. The enhancement in the heat transfer depends on the dimensions of the tube, the dimensions of the fins, the number of fins per unit length, bundle arrangement (square or triangular pitch), tube and fin materials of construction, as well as the fluid velocities, physical properties, temperatures and fouling characteristics. Enhancements in the outside film transfer coefficients for such exchangers are of the order 50 to 70% when compared with plain tubes. Typical outside film transfer coefficients would usually be of the order of $60 \text{ W}\cdot\text{m}^{-2}\cdot^\circ\text{C}$. Air-cooled heat exchangers of the type in Figure 15.11 are used extensively for cooling utility, particularly when cooling water is scarce.

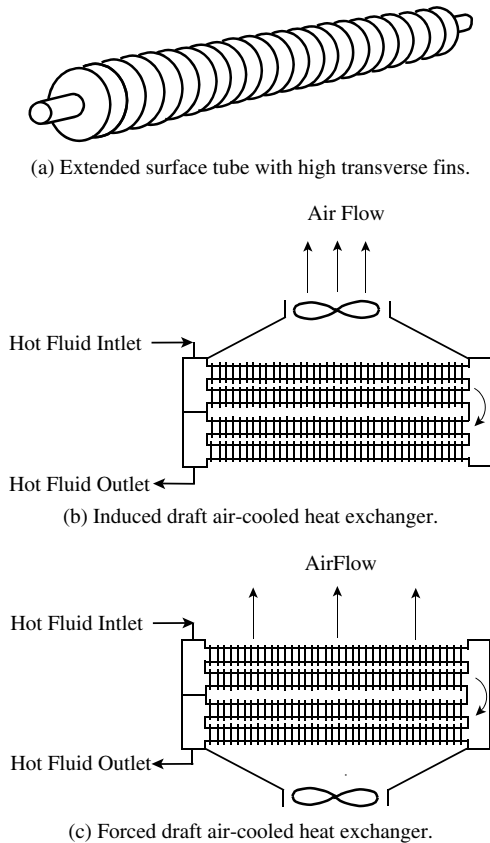


Figure 15.11 Extended surface heat transfer.

Extended surfaces created from integrally formed “low” transverse fins can be used in conventional shell-and-tube heat exchangers to enhance the outside film transfer coefficient. Low transverse fins can increase the surface area by a factor of around 2.5 relative to plain tubes.

Longitudinal fins can also be used, but their application is restricted to small heat exchangers in the form of a concentric pipe heat exchanger, similar to the schematic in Figure 15.5a. In this arrangement, the inner tube would be the extended surface tube with the fins in the annular space to enhance the heat transfer. Longitudinal fins can increase the surface area by a factor of 14 to 20 relative to plain tubes.

15.6 RETROFIT OF HEAT EXCHANGERS

In retrofit situations, existing exchangers might be subjected to changes in flowrate. For example, a debottlenecking project might require an increase in throughput. If this is the case, then the change in the heat transfer coefficients can be approximated by:

For the tube-side coefficient

$$\frac{h_{T2}}{h_{T1}} = \left(\frac{F_{I2}}{F_{I1}} \right)^{0.8} \quad (15.67)$$

where h_{T1}, h_{T2} = tube-side film transfer coefficients for the initial and new flowrates

F_{I1}, F_{I2} = tube-side volumetric flowrates for the initial and new flowrates

For the shell-side coefficient

$$\frac{h_{S2}}{h_{S1}} = \left(\frac{F_{O2}}{F_{O1}} \right)^{0.64} \quad (15.68)$$

where h_{S1}, h_{S2} = shell-side film transfer coefficients for the initial and new flowrates

F_{O1}, F_{O2} = shell-side volumetric flowrates for the initial and new flowrates

The change in pressure drop can be approximated by:

For the tube-side pressure drop

$$\frac{\Delta P_{T2}}{\Delta P_{T1}} = \left(\frac{F_{I2}}{F_{I1}} \right)^{1.9} \quad (15.69)$$

where $\Delta P_{T1}, \Delta P_{T2}$ = tube-side pressure drop for the initial and new flowrates

For the shell-side pressure drop

$$\frac{\Delta P_{S2}}{\Delta P_{S1}} = \left(\frac{F_{O2}}{F_{O1}} \right)^{2.9} \quad (15.70)$$

where $\Delta P_{S1}, \Delta P_{S2}$ = shell-side pressure drop for the initial and new flowrates.

Although an increase in flowrate can result in increased film transfer coefficients, an increase in flowrate is also usually accompanied by an increase in heat transfer duty. In turn, this might lead to a need for increased heat transfer area.

For example, an increase in the process throughput might result in an increase in the heat duty, and hence an increase in the heat transfer area, if the temperatures are unchanged. More generally, additional heat transfer area might be required as a result of increased heat duty, operation under reduced temperature differences, operation under reduced heat transfer coefficients or increased fouling.

If shell-and-tube heat exchangers are being used and additional heat transfer area is required, it might be possible to install a new tube bundle into existing shells if the additional area requirement is small. This might be possible if the tube pitch can be decreased or the pitch configuration changed from a square to a triangular pitch. If a significant amount of additional area is required, then the existing area can be supplemented by adding a new shell (or more than one shell if there is a large area requirement). New heat exchanger shells can be added to an existing match in one of two ways:

a. *Series.* When new exchanger shells are added in series to an existing match, then the full flowrate goes through

the existing exchangers of the match. The pressure drop and heat transfer coefficients of the existing exchangers of the match will only change significantly if the changes lead to significant changes in the operating conditions of the existing match (e.g. increase in flowrate in a debottlenecking study). The addition of new exchangers in series to the existing match will lead to an increase in the overall pressure drop across the match. This might be important if the pump (or compressor) is close to its maximum capacity.

- b. *Parallel*. If new exchanger shells are added in parallel to an existing match, then the flowrate through the existing exchangers is decreased. This will decrease the flowrate and pressure drop through the existing exchangers of the match. However, it will also decrease the heat transfer coefficient, decreasing its performance. The addition of new exchangers will therefore leave the pressure difference largely unchanged. The pressure drop across the match will be the largest between that of the existing exchangers under the new conditions and the new exchangers installed in parallel.

Increasing the heat transfer area, either by replacing tube bundles or by adding new shells, is likely to be expensive. Rather than install additional heat transfer area to cater for the new operational requirements, the overall heat transfer coefficient of the heat exchanger can be increased. It might be possible to modify the tube-side flow pattern to increase the number of tube passes, thereby increasing the tube-side film transfer coefficient. On the shell-side, it might be possible to modify the baffle arrangement to increase the shell-side film transfer coefficient, for example, by decreasing the baffle spacing. Heat transfer enhancement can also be considered. Changing the heat transfer surface from a plain surface to a finned, ribbed or nonuniform surface can increase the rate of heat transfer of the tube surfaces. This can be applied to the inside or outside surface of the tube, or both, but requires the tubes to be changed. Alternatively, devices can be inserted into the inside of existing plain tubes to enhance the inside heat transfer coefficient, at the expense of increased pressure drop. Insertion devices include^{8,9}:

- *Twisted tapes*. These consist of a thin strip of twisted metal with the same width as the tube inside diameter that is slid into the tube. The flow is caused to spiral along the tube length.
- *Static mixers*. Static mixers are manufactured from thin metal strips with the same width as the tube internal diameter. In contrast with twisted tapes, the helix is not continuous. The flow is caused to twist through 180° and then twist in the reverse direction through 180°, and so on. The continuous splitting, reorientation and recombination of the fluid cause the enhancement as it flows through the insert.
- *Coiled wire*. The coil is manufactured by tightly wrapping wire onto a rod such that the outside diameter of the coil is slightly larger than the inside diameter of the tube. This ensures that when it is fitted into the tube there is no movement when in service. The coil functions as nonintegral tube roughness.
- *Mesh inserts*. Mesh inserts are manufactured from a matrix of thin wire filaments. The matrix consists of small loops offset in a helical arrangement and supported by two thicker central wires twisted together. The diameter of the wire mesh is slightly larger than the tube diameter to give close contact with the tube wall. The mesh causes a swirling and random mixing from the flow and is usually restricted to laminar flow applications.

If the new increased heat duty and new log mean temperature difference are fixed, then the additional surface area above the existing area using a plain tube surface is given by⁹:

$$Q = U(A_{\text{existing}} + \Delta A)\Delta T_{LM} F_T \quad (15.71)$$

where

Q = heat transfer duty

U = overall heat transfer coefficient

A_{existing} = existing heat transfer area

ΔA = additional area requirement

ΔT_{LM} = logarithmic mean temperature difference

F_T = logarithmic mean temperature difference correction factor

If the same duty is to be serviced by enhanced heat transfer without additional area:

$$Q = U_e A_{\text{existing}} \Delta T_{LM} F_T \quad (15.72)$$

where U_e = enhanced overall heat transfer coefficient

Thus, for heat transfer enhancement to solve the problem without installation of new area:

$$U_e A_{\text{existing}} \geq U(A_{\text{existing}} + \Delta A) \quad (15.73)$$

Rearranging this equation gives:

$$\frac{U_e}{U} \geq \frac{A_{\text{existing}} + \Delta A}{A_{\text{existing}}} \quad (15.74)$$

The overall heat transfer coefficient is given by:

$$\frac{1}{U} = \frac{1}{h_O} + \frac{1}{h_W} + \frac{d_O}{d_I} \frac{1}{h_I} \quad (15.75)$$

where

$$\frac{1}{h_O} = \frac{1}{h_S} + \frac{1}{h_{SF}}$$

$$\frac{1}{h_I} = \frac{1}{h_T} + \frac{1}{h_{TF}}$$

U = overall heat transfer coefficient

h_O, h_I = heat transfer coefficients for the outside and inside of plain tubes including the fouling resistance

h_W = heat transfer coefficients for the tube wall

d_O, d_I = outside and inside tube diameters

h_S, h_T = film heat transfer coefficients for the shell-side and tube-side of plain tubes

h_{SF}, h_{TF} = fouling coefficients for the shell-side and tube-side of plain tubes

If heat transfer enhancement is only for the inside of the tubes using inserts, then:

$$\frac{1}{U_e} = \frac{1}{h_O} + \frac{1}{h_W} + \frac{d_O}{d_I} \frac{1}{h_{Ie}} \quad (15.76)$$

where $\frac{1}{h_{Ie}} = \frac{1}{h_{Te}} + \frac{1}{h_{TFe}}$

U_e = enhanced overall heat transfer coefficient

h_{Ie} = enhanced film transfer coefficient for the inside of tubes including the fouling resistance

h_{Te} = enhanced film heat transfer coefficients for the tube-side

h_{TFe} = enhanced fouling resistance for the tube-side.

Dividing Equation 15.75 by 15.76 gives:

$$\frac{U_e}{U} = \frac{h_{Ie}(d_O h_O h_W + d_O h_O h_I + d_I h_W h_I)}{h_I(d_O h_O h_W + d_O h_O h_{Ie} + d_I h_W h_{Ie})} \quad (15.77)$$

Combining Equations 15.74 and 15.77 gives:

$$\frac{h_{Ie}}{h_I} \geq \frac{d_O h_O h_W (A_{existing} + \Delta A)}{d_O h_O h_W A_{existing} - \Delta h_I (d_O h_O + d_I h_W)} \quad (15.78)$$

Equation 15.78 gives the criterion for heat transfer enhancement to cater to the new duty without increasing the heat transfer area. If it is assumed that the resistance to heat transfer across the tube wall is negligible (i.e. h_W goes to infinity) and the difference between the inside and outside diameters is negligible (i.e. $d_O = d_I$), then Equation 15.78 simplifies to⁹:

$$\frac{h_{Ie}}{h_I} \geq \frac{h_O (A_{existing} + \Delta A)}{h_O A_{existing} - h_I \Delta A} \quad (15.79)$$

The enhancement ratio (h_{Ie}/h_I) for tube inserts can be correlated as a function of Reynolds number based on the internal diameter of the tube $Re = \rho V d_i / \mu$ ¹⁰. Different tube inserts can be compared on the basis of plots of enhancement ratio (h_{Te}/h_T) versus Reynolds number.

The major disadvantage in using heat transfer enhancement is that it increases the pressure drop. In retrofit, this can be important, as the pumps driving the flow might be limited in their capacity to meet the required increase in pressure drop.

Example 15.3 Suppose the data in Table 15.5 now relate to the performance of an existing heat exchanger. The details of the

exchanger geometry (which uses steel tubes) are:

$$d_O = 20 \text{ mm}$$

$$d_I = 16 \text{ mm}$$

$$p_T = 25 \text{ mm (square pitch)}$$

$$B_C = 0.25$$

$$L = 5.95 \text{ (after subtracting tube sheet thickness)}$$

$$D_S = 0.84 \text{ m (internal diameter)}$$

$$N_T = 824$$

$$N_{TP} = 2$$

This exchanger is required to operate under new conditions that call for an increase in flowrate of 30% for both fluids. The temperatures at the inlet of the exchanger are unchanged, but the temperatures at the exit must be maintained at their current values. The pumps feeding the exchanger have spare capacity and therefore some increase in the pressure drop of both fluids is acceptable.

- Estimate the increase in the overall heat transfer coefficient and the resulting outlet temperatures for the new conditions.
- Can the increase in heat duty be accommodated by keeping the existing tube bundle and using tube inserts? A number of inserts are available. Their thermal performance is given in Table 15.6 and friction factor performance given in Table 15.7 as a function of the Reynolds number based on the internal tube diameter, d_I ¹⁰. The value of H in Tables 15.6 and 15.7 relates to the axial distance between a 180° twist in the device¹⁰.

Table 15.6 Thermal performance of tube insert devices.

Type of insert	Enhancement ratio h_{Ie}/h_I
Coiled wire H/d_I 5.5	$4.191 Re^{-0.1}$
Coiled wire H/d_I 2.8	$5.309 Re^{-0.1}$
Coiled wire H/d_I 1.12	$7.313 Re^{-0.1}$
Twisted tape H/d_I 8.5	$4.687 Re^{-0.1203}$
Twisted tape H/d_I 3.6	$15.88 Re^{-0.2206}$

Table 15.7 Friction factor performance of tube insert devices.

Type of insert	Enhancement ratio c_{fe}/c_f
Coiled wire Hd_I 5.5	$4.948 Re^{-0.0953}$
Coiled wire Hd_I 2.8	$5.465 Re^{-0.0345}$
Coiled wire Hd_I 1.12	$5.685 Re^{0.0432}$
Twisted tape Hd_I 8.5	$4.765 Re^{-0.1166}$
Twisted tape Hd_I 3.6	$12.65 Re^{-0.1787}$

Solution

- Existing heat duty

$$= 0.037 \times 730 \times 2470 \times (200 - 95)$$

$$= 7.01 \times 10^6 \text{ W}$$

From Example 15.2, $\Delta T_{LM} = 88.6^\circ\text{C}$. Since the F_T is based on temperatures only and the temperatures are the same as in Example 15.2, the existing $F_T = 0.90$ as calculated in Example 15.2.

$$\begin{aligned} A &= \pi d_o L N_T \\ &= \pi \times 0.02 \times 5.95 \times 824 \\ &= 308 \text{ m}^2 \\ U &= \frac{Q}{A \Delta T_{LM} F_T} \\ &= \frac{7.01 \times 10^6}{308 \times 88.6 \times 0.90} \\ &= 285 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \end{aligned}$$

While it is possible to calculate the existing overall heat transfer coefficient from the operating data, it is not possible to calculate the individual film transfer coefficients. The individual film transfer coefficients can be combined in any number of ways to add up to an overall value of $285 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$. However, the film transfer coefficients can be estimated from the correlations in Appendix C. Given that the tube-side correlations are much more reliable than the shell-side correlations, the best way to determine the individual coefficients is to calculate the coefficient for the tube-side and allocate the shell-side coefficient to add up to $U = 285 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$. Thus, to calculate the tube-side film transfer coefficient, K_{hT} must first be determined.

$$\begin{aligned} K_{hT} &= C \left[\frac{k}{d_i} \right] Pr^{\frac{1}{3}} \left[\frac{d_i \rho}{\mu} \right]^{0.8} \\ &= 0.023 \left[\frac{0.133}{0.016} \right] \left[\frac{2050 \times 3.6 \times 10^{-3}}{0.133} \right]^{\frac{1}{3}} \left[\frac{0.016 \times 830}{3.6 \times 10^{-3}} \right]^{0.8} \\ &= 520.4 \\ v_T &= \frac{F_I}{\frac{\pi d_i^2}{4} \frac{N_T}{N_{TP}}} \\ &= \frac{0.103}{\frac{\pi \times 0.016^2}{4} \frac{824}{2}} \\ &= 1.24 \text{ m}\cdot\text{s}^{-1} \\ h_T &= K_{hT} V_T^{0.8} \\ &= 520.4 \times 1.24^{0.8} \\ &= 618 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \end{aligned}$$

Assume fouling coefficients:

$$\begin{aligned} h_{TF} &= 2000 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \\ h_{SF} &= 5000 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \\ \frac{1}{U} &= \frac{1}{h_S} + \frac{1}{h_{SF}} + \frac{1}{h_W} + \frac{d_o}{d_i} \frac{1}{h_{TF}} + \frac{d_o}{d_i} \frac{1}{h_T} \\ \frac{1}{285} &= \frac{1}{h_S} + \frac{1}{5000} + \frac{1}{20,200} + \frac{0.02}{0.016} \left(\frac{1}{2000} + \frac{1}{618} \right) \\ h_S &= 1635 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \end{aligned}$$

The change in the tube-side and shell-side film transfer coefficients as a result of a 30% increase in both flowrates can be estimated from Equations 15.67 and 15.68:

$$\begin{aligned} \frac{h_{T2}}{h_{T1}} &= \left(\frac{F_{I2}}{F_{I1}} \right)^{0.8} \\ h_{T2} &= 618 \times 1.3^{0.8} \\ &= 762 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \\ \frac{h_{S2}}{h_{S1}} &= \left(\frac{F_{O2}}{F_{O1}} \right)^{0.64} \\ h_{S2} &= 1635 \times 1.3^{0.64} \\ &= 1934 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \end{aligned}$$

If the fouling coefficients are assumed to be unchanged, the new overall heat transfer coefficient is given by:

$$\begin{aligned} \frac{1}{U} &= \frac{1}{1934} + \frac{1}{5000} + \frac{1}{20,200} + \frac{0.02}{0.016} \left(\frac{1}{2000} + \frac{1}{762} \right) \\ U &= 330 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \end{aligned}$$

At the increased flowrate condition, the outlet temperatures and heat duty for the existing area are unknown.

$$\begin{aligned} Q &= 0.037 \times 1.3 \times 730 \times 2470(200 - T_{H2}) \\ &= 0.103 \times 1.3 \times 830 \times 2050(T_{C2} - 35) \\ A &= \frac{Q}{U \Delta T_{LM} F_T} \end{aligned}$$

Thus, the outlet temperature of the hot or cold stream can be assumed. This specifies the heat duty and hence the outlet temperature of the other stream. In this case, assume the outlet temperature of the hot stream and calculate the final temperature of the cold stream from:

$$T_{C2} = T_{C1} + \frac{Q}{F_I \rho C_p}$$

This allows ΔT_{LM} and F_T to be calculated. Given the value of U , this allows the area to be calculated. The outlet temperature of the hot stream can then be varied by trial and error until the calculated area equals the actual area of 308 m^2 .

$$\begin{aligned} \text{Try } T_{H2} &= 100^\circ\text{C} \\ Q &= 8.67 \times 10^6 \text{ W} \\ T_{C2} &= 73.1^\circ\text{C} \\ \Delta T_{LM} &= 92.5^\circ\text{C} \\ F_T &= 0.92 \\ A &= 309 \text{ m}^2 \end{aligned}$$

This is close; an exact value is given by further trial and error to give:

$$\begin{aligned} T_{H2} &= 100.2^\circ\text{C} \\ Q &= 8.66 \times 10^6 \text{ W} \\ T_{C2} &= 73^\circ\text{C} \end{aligned}$$

$$\Delta T_{LM} = 92.7^\circ\text{C}$$

$$F_T = 0.92$$

$$A = 308 \text{ m}^2$$

b. Maintaining the existing temperatures, the new duty is given by:

$$Q = 0.037 \times 1.3 \times 730 \times 2470(200-95)$$

$$= 9.11 \times 10^6 \text{ W}$$

$$A = \frac{Q}{U \cdot \Delta T_{LM} \cdot F_T}$$

$$= \frac{9.11 \times 10^6}{330 \times 88.6 \times 0.90}$$

$$= 346 \text{ m}^2$$

$$\Delta A = 346 - 308$$

$$= 38 \text{ m}^2$$

Calculate the outside and inside coefficients including the fouling:

$$\frac{1}{h_O} = \frac{1}{h_S} + \frac{1}{h_{SF}}$$

$$= \frac{1}{1934} + \frac{1}{5000}$$

$$h_O = 1395 \text{ W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$$

$$\frac{1}{h_I} = \frac{1}{h_T} + \frac{1}{h_{TF}}$$

$$= \frac{1}{762} + \frac{1}{2000}$$

$$h_I = 552 \text{ W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$$

The heat transfer enhancement ratio can now be calculated from Equation 15.78:

$$\frac{h_{Ie}}{h_I} = \frac{d_O h_O h_W (A_{existing} + \Delta A)}{d_O h_O h_W A_{existing} - \Delta A h_I (d_O h_O + d_I h_W)}$$

$$= \frac{0.02 \times 1395 \times 20,200(308 + 38)}{0.02 \times 1395 \times 20,200 \times 308 - 38 \times 552 (0.02 \times 1395 + 0.016 \times 20,200)}$$

$$= 1.17$$

For the purpose of comparison, now neglect the wall resistance and difference between the inside and outside tube diameters. From Equation 15.79:

$$\frac{h_{Ie}}{h_I} = \frac{h_O (A_{existing} + \Delta A)}{h_O A_{existing} - h_I \Delta A}$$

$$= \frac{1395(308 + 38)}{1395 \times 308 - 552 \times 38}$$

$$= 1.18$$

For most applications, Equation 15.79 is accurate enough.

The enhancement ratio for a given device can be correlated as a function of the Reynolds Number⁹:

$$Re = \frac{\rho d_I v_T}{\mu}$$

$$= \frac{830 \times 0.016 \times 1.24 \times 1.3}{3.6 \times 10^{-3}}$$

$$= 5946$$

Table 15.8 shows the enhancement ratios for various tube inserts.

Table 15.8 Enhancement ratios for heat transfer of various tube inserts.

Type of insert	Enhancement ratio
Coiled wire H/d _I 5.5	4.191 $Re^{-0.1} = 1.76$
Coiled wire H/d _I 2.8	5.309 $Re^{-0.1} = 2.23$
Coiled wire H/d _I 1.12	7.313 $Re^{-0.1} = 3.07$
Twisted tape H/d _I 8.5	4.687 $Re^{-0.1203} = 1.65$
Twisted tape H/d _I 3.6	15.88 $Re^{-0.2206} = 2.33$

Thus, all of the tube inserts will potentially provide the required enhancement without the need for extra heat transfer area. However, it would be preferred to have the enhancement with minimum increase in pressure drop. Table 15.9 shows the increase in friction factor for the various inserts.

Table 15.9 Enhancement ratios for friction factor of various tube inserts.

Type of insert	Friction factor ratio
Coiled wire H/d _I 5.5	4.948 $Re^{-0.0953} = 2.16$
Coiled wire H/d _I 2.8	5.465 $Re^{-0.0345} = 4.05$
Coiled wire H/d _I 1.12	5.685 $Re^{0.0432} = 8.28$
Twisted tape H/d _I 8.5	4.765 $Re^{-0.1166} = 1.73$
Twisted tape H/d _I 3.6	12.654 $Re^{-0.1787} = 2.68$

On the basis of heat transfer enhancement and pressure drop considerations, Twisted Tape H/d_I 8.5 would be chosen. However, a detailed examination of capital cost might cause this to be revised.

15.7 CONDENSERS

The construction of shell-and-tube condensers is very similar to shell-and-tube heat exchangers for duties that do not involve a change of phase. Condensers can be horizontally or vertically oriented with the condensation on the tube-side or the shell-side. The magnitude of the condensing film coefficient for a given quantity of vapor condensation on a given surface is significantly different depending on the orientation of the condenser. The condensation normally takes place on the shell-side of horizontal exchangers and the tube-side of vertical

exchangers. Horizontal shell-side condensation is normally preferred, as the condensing film transfer coefficients are higher. Condensation on the tube-side of horizontal condensers is normally restricted to the use of condensing steam as a heating medium.

Condensation can take place by one of two mechanisms:

- film-wise condensation*, in which the condensing vapor wets the surface of the tube forming a continuous film
- drop-wise condensation*, in which droplets of condensation do not wet the surface and after growing, fall from the tube to expose fresh condensing surface without forming a continuous film.

Although drop-wise condensation can produce much higher condensing film transfer coefficients, it is unpredictable, and the design is carried out on the basis of film-wise condensation.

The basic equations describing film-wise condensation were developed by Nusselt¹¹. The derivation of the equations has been given by Kern¹ and others. A number of assumptions are made in the derivation:

- the liquid film flows smoothly and steadily by gravitational forces;
- the liquid film is in laminar flow;
- no noncondensable gases are present in the vapor phase;
- no vapor shear force acts on the liquid–vapor interface;
- momentum terms are negligible;
- temperature distribution in the condensate film is linear;
- the only heat transferred across the liquid film is the latent heat of condensation released at the liquid–vapor interface (transfer of sensible heat in the liquid film is negligible);
- temperature of the liquid–vapor interface is equal to the saturation temperature;
- physical properties of the liquid film are constant.

For condensation outside of a horizontal tube¹:

$$h_C = 0.725 \left(\frac{k_L^3 \rho_L^2 \Delta H_{VAP} g}{d_O \mu_L \Delta T} \right)^{\frac{1}{4}} \quad (15.80)$$

where h_C = condensing film coefficient (W·m⁻²·K⁻¹)
 k_L = thermal conductivity of the liquid (W·m⁻¹·K⁻¹)
 ρ_L = density of the liquid (kg·m⁻³)
 ΔH_{VAP} = latent heat (J·kg⁻¹)
 g = gravitational constant (9.81 m·s⁻²)
 d_O = outside diameter of tube (m)
 μ_L = viscosity of the liquid (N·s·m⁻² or kg·m⁻¹·s⁻¹)
 ΔT = temperature difference across the condensate film (K)

The analysis was later modified to include some of the factors neglected by Nusselt^{12,13}. One of these is the effect of buoyancy forces acting on the liquid film. This results in the ρ_L^2 term in Equation 15.80 being replaced by $\rho_L(\rho_L - \rho_V)$. Such buoyancy forces are usually only important close to the critical point. In most cases, the two most important factors that cause a significant deviation from Equation 15.80 are the presence of vapor shear forces and noncondensable gases in the vapor. Vapor shear forces act to increase the heat transfer coefficient, whereas noncondensable gases act to decrease it.

Since the ΔT across the film is unknown, it is best eliminated from Equation 15.80. By definition of the condensing film coefficient:

$$m \Delta H_{VAP} = h_C \pi d_O L N_T \Delta T \quad (15.81)$$

where m = flowrate of condensate (kg·s⁻¹)

L = tube length (m)

N_T = number of tubes (–)

Substituting Equation 15.81 into Equation 15.80 and rearranging gives:

$$h_C = 0.954 k_L \left(\frac{\rho_L^2 L g N_T}{\mu_L m} \right)^{\frac{1}{3}} \quad (15.82)$$

For the shell-side of a horizontal tube bundle, dripping of condensate over successive rows acts to decrease the condensing coefficient. This can be accounted for by multiplying the condensing coefficient for a single tube by an empirical correction involving the number of tubes in a vertical row. However, in a tube bundle, the number of tubes in the vertical rows varies according to the position in the bundle. A simple empirical correction is¹⁴:

$$h_C = 0.954 k_L \left(\frac{\rho_L^2 L g N_T}{\mu_L m} \right)^{\frac{1}{3}} N_R^{-\frac{1}{6}} \quad (15.83)$$

N_R = number of tubes in a vertical row (–)

$$\approx 0.78 \frac{D_S}{p_T}$$

D_S = shell diameter (m)

p_T = vertical tube pitch (m)

For condensation inside horizontal tubes, the Nusselt Equation can be applied with a correction for the reduction in condensing coefficient caused by the accumulation of condensation. The correction usually applied is 0.8. No correction for the number of tubes is required. Thus, for condensation inside horizontal tubes:

$$h_C = 0.763 k_L \left(\frac{\rho_L^2 L g N_T}{\mu_L m} \right)^{\frac{1}{3}} \quad (15.84)$$

The Nusselt Equations apply to laminar flow of the condensing film. For horizontal condensation the equations

are applicable for:

$$Re = \frac{2m}{\mu_L LN_T} < 2000 \quad (15.85)$$

For condensation on a vertical surface, the Nusselt Equation takes the form¹:

$$h_C = 0.943 \left(\frac{k_L^3 \rho_L^2 \Delta H_{VAP} g}{L \mu_L \Delta T} \right)^{\frac{1}{4}} \quad (15.86)$$

where L = length of the condensing surface (m)

ΔT = temperature difference across the condensate film (K)

Combining Equation 15.81 for condensation on the outside of a tube with Equation 15.86 gives:

$$h_C = 1.35 k_L \left(\frac{\rho_L^2 d_O g N_T}{\mu_L m} \right)^{\frac{1}{3}} \quad (15.87)$$

For condensation on the inside of the vertical tubes:

$$m \Delta H_{VAP} = h_C \pi d_I L N_T \Delta T \quad (15.88)$$

Combining Equations 15.86 and 15.88 gives the Nusselt Equation for condensation inside vertical tubes:

$$h_C = 1.35 k_L \left(\frac{\rho_L^2 d_I g N_T}{\mu_L m} \right)^{\frac{1}{3}} \quad (15.89)$$

For vertical condensation, the Nusselt Equations are valid for a laminar film according to:

$$Re = \frac{4m}{\pi \mu_L d_I N_T} < 2000 \quad (15.90)$$

In the above equations, the film thickness and hence the condensing coefficient varies across the surface. The correlations give an average coefficient applicable to the entire surface. The condensing coefficients are independent of shell-side geometry (e.g. baffle cut, distance etc.). The Nusselt Equations give reasonably good agreement with experimental data for laminar flow of the condensate film in the absence of vapor shear forces and noncondensable gases in the vapor. In the absence of noncondensable gases, the Nusselt Equations will tend to give a conservative prediction of the condensing coefficient. Vapor shear and turbulence in the film can lead to considerably higher values than those predicted by the Nusselt Equations.

For a simple total condenser involving isothermal condensation:

$$Q = m \Delta H_{VAP} = UA \Delta T_{LM} \quad (15.91)$$

where U is defined by Equation 15.13. Note that if the condensing fluid is pure and therefore isothermal, no F_T correction factor is required if multipass exchangers are used, that is, $F_T = 1$.

If the heat exchange involves desuperheating as well as condensation, then the exchanger can be divided into zones with linear temperature–enthalpy profiles in each zone. Figure 15.12a illustrates desuperheating and condensation on the shell-side of a horizontal condenser. The total heat transfer area is the sum of the values for each zone:

$$A = A_{DS} + A_{CN} = \frac{Q_{DS}}{U_{DS} \Delta T_{LM,DS}} + \frac{Q_{CN}}{U_{CN} \Delta T_{LM,CN}} \quad (15.92)$$

where

A = total heat transfer area

A_{DS} = heat transfer area for the desuperheating zone

A_{CN} = heat transfer area for the condensing zone

Q_{DS} = heat transfer duty for the desuperheating zone

Q_{CN} = heat transfer duty for the condensing zone

U_{DS} = overall heat transfer coefficient for the desuperheating zone

U_{CN} = overall heat transfer coefficient for the condensing zone

$\Delta T_{LM,DS}$ = logarithmic mean temperature difference for the desuperheating zone

$\Delta T_{LM,CN}$ = logarithmic mean temperature difference for the condensing zone

To calculate the condensing heat transfer coefficient requires the length of the condensing zone L to be specified. Thus, a value of L must be estimated before the calculation can be made. For the value of L to be correct, it must comply with:

$$L = \frac{A_{CN}}{A} \times \text{tube length} \quad (15.93)$$

The value of L is then varied until there is agreement with Equation 15.93.

It might also be necessary to subcool the condensate. As with desuperheating, if subcooling is required, the heat exchanger can be divided into zones. Figure 15.12b illustrates subcooling on the shell-side of a vertical condenser. The subcooling arrangement in Figure 15.12b is achieved by using a loop seal to create a partially submerged tube bundle¹. For subcooling, the heat transfer area is given by:

$$A = A_{CN} + A_{SC} = \frac{Q_{CN}}{U_{CN} \Delta T_{LM,CN}} + \frac{Q_{SC}}{U_{SC} \Delta T_{LM,SC}} \quad (15.94)$$

where

A_{SC} = heat transfer area for the subcooling zone

Q_{SC} = heat transfer duty for the subcooling zone

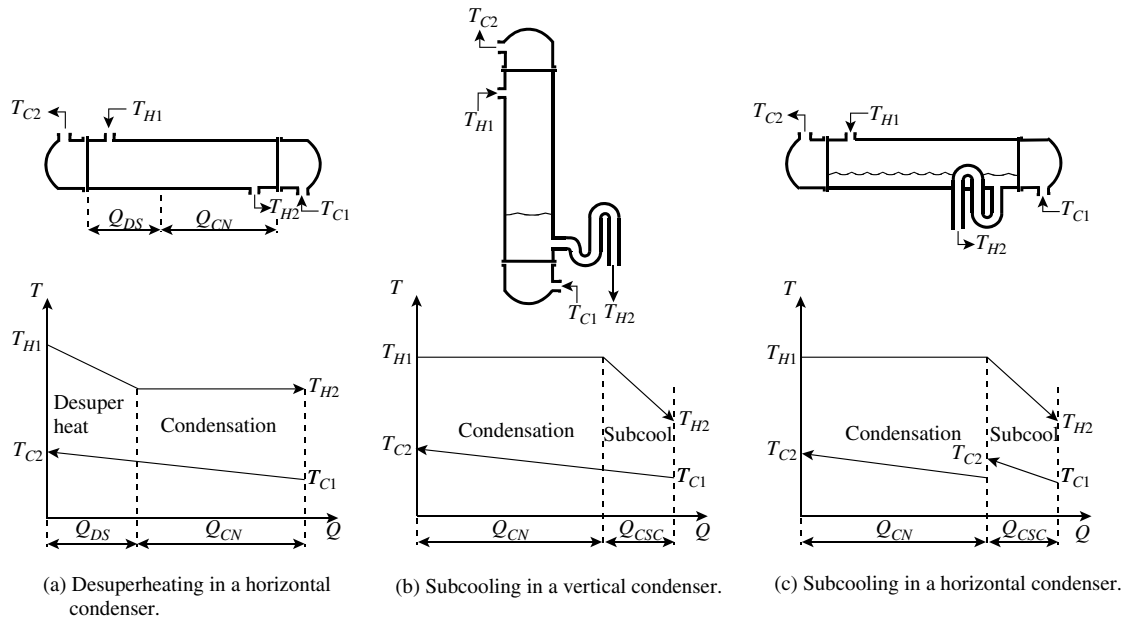


Figure 15.12 Condensation with desuperheating and subcooling.

U_{SC} = overall heat transfer coefficient for the subcooling zone

$\Delta T_{LM,SC}$ = logarithmic mean temperature difference for the subcooling zone

To calculate the condensing heat transfer coefficient again requires the length of the condensing zone L to be specified. Thus, a value of L must be estimated and adjusted until it complies with Equation 15.93.

Figure 15.12c illustrates subcooling on the shell-side of a horizontal condenser. The subcooling arrangement in Figure 15.12c is again achieved by using a loop seal to create a partially submerged tube bundle¹. Rather than use a loop seal, a dam baffle can be used to partially submerge the bundle¹. Figure 15.12c shows the zones this time represented in parallel, rather than the series arrangements in Figures 15.12a and 15.12b. Calculation of the condensing heat transfer coefficient for a horizontal exchanger requires the number of tubes in the condensing zone $N_{T,CN}$ to be specified. Thus, a value of $N_{T,CN}$ must be estimated before the calculation can be made. For the value of $N_{T,CN}$ to be correct, it must comply with:

$$N_{T,CN} = \frac{A_{CN}}{A} \times \text{number of tubes} \quad (15.95)$$

The value of $N_{T,CN}$ is then varied until there is agreement with Equation 15.95.

For multicomponent condensation, the condensation will not be isothermal, leading to a nonlinear temperature–enthalpy profile for the condensation. If this is the case, then the exchanger can be divided into a number of zones with the temperature–enthalpy profiles linearized in each zone. Each zone is then modeled separately and zones summed to obtain the overall area requirement¹.

Particular care needs to be adopted if a vapor to be condensed has noncondensable gases present. Here the vapor diffuses through the gas to the cold surface where it condenses. But as the condensation proceeds, the concentration of the noncondensable gas increases, which increases the diffusional resistance and decreases the condensing coefficient. To take this into account requires complex models, which is outside the scope of this text.

Pressure drop during condensation results essentially from the vapor flow. As condensation proceeds, the vapor flowrate decreases. The equations described previously for pressure drop in shell-and-tube heat exchangers are only applicable under constant flow conditions. Again the exchanger can be divided into zones. However, in preliminary design, a reasonable estimate of the pressure drop can usually be obtained by basing the calculation on the mean of the inlet and outlet vapor flowrates.

Example 15.3 A flowrate of $0.1 \text{ kmol}\cdot\text{s}^{-1}$ of essentially pure acetone vapor from the overhead of a distillation column is to be condensed without any condensate subcooling. The condensation is to take place on the shell-side of a horizontal shell-and-tube heat exchanger against cooling water flowing in two passes on the tube-side. The operating pressure of the condenser is 1.52 bar. At this pressure, the acetone condenses at 67°C . The cooling water can be assumed to be at 25°C and to be returned to the cooling tower at 35°C . The condenser can be assumed to be steel with tubes with 20-mm outside diameter and 2-mm wall thickness. The tube pitch can be assumed to be $1.25d_o$ and a square configuration. The ratio of tube length to shell diameter can be assumed to be 5. The physical property data are given in Table 15.10. The properties of acetone are at 67°C . Although the average film temperature will be lower than this, the value of $k(\rho^2/\mu)^{1/3}$ tends not to be very sensitive to temperature. The molar mass of acetone can be assumed to be $58 \text{ kg}\cdot\text{kmol}^{-1}$. Assume the fouling coefficients to

be $11,000 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$ and $5000 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$ for the shell-side and tube-side respectively. For an allowable tube-side velocity of $2 \text{ m}\cdot\text{s}^{-1}$, estimate the heat transfer area.

Table 15.10 Physical property data for acetone and water.

Property	Acetone (67°C)	Water (30°C)
Density ($\text{kg}\cdot\text{m}^{-3}$)	736	996
Heat capacity ($\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$)	2320	4180
Viscosity ($\text{N}\cdot\text{s}\cdot\text{m}^{-2}$)	0.213×10^{-3}	0.797×10^{-3}
Thermal conductivity ($\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$)	0.137	0.618
Heat of vaporization ($\text{J}\cdot\text{kg}^{-1}$)	494,000	—

Solution

$$\text{Flowrate of acetone} = 0.1 \times 58$$

$$= 5.8 \text{ kg}\cdot\text{s}^{-1}$$

$$\text{Duty on condenser} = 5.8 \times 494,000$$

$$= 2.865 \times 10^6 \text{ W}$$

$$\text{Flowrate of cooling water} = \frac{2.865 \times 10^6}{4180(35 - 25)}$$

$$= 68.54 \text{ kg}\cdot\text{s}^{-1}$$

$$= 0.0688 \text{ m}^3\cdot\text{s}^{-1}$$

To determine the condensing film coefficient using Equation 15.83 requires the number of tubes to be known.

$$N_T = \frac{\pi D_S^2}{4 p_C p_T^2}$$

$$\text{where } D_S = \left(\frac{4 p_C p_T^2 A}{\pi^2 d_O (L/D_S)} \right)^{\frac{1}{3}}$$

Thus, the solution must be iterative as the heat transfer area A is unknown. An initial estimate of A (say 100 m^2) is required, giving:

$$D_S = 0.633 \text{ m}$$

$$N_T = 503$$

$$h_C = 0.954 k_L \left(\frac{\rho_L^2 L g N_T}{\mu_L m} \right)^{\frac{1}{3}} N_R^{-\frac{1}{6}}$$

$$= 0.954 k_L \left(\frac{\rho_L^2 (L/D_S) D_S g N_T}{\mu_L m} \right)^{\frac{1}{3}} \left(0.78 \frac{D_S}{p_T} \right)^{-\frac{1}{6}}$$

$$= 0.954 \times 0.137 \left(\frac{736^2 \times 5 \times 0.633 \times 9.81 \times 503}{2.13 \times 10^{-4} \times 5.8} \right)^{\frac{1}{3}}$$

$$\times \left(0.78 \frac{0.633}{0.02 \times 1.25} \right)^{-\frac{1}{6}}$$

$$= 1510 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$$

The tube-side heat transfer coefficient is given by:

$$\begin{aligned} K_{hT} &= C \left[\frac{k}{d_I} \right] Pr^{\frac{1}{3}} \left[\frac{d_I \rho}{\mu} \right]^{0.8} \\ &= 0.023 \left[\frac{0.618}{0.016} \right] \left[\frac{4180 \times 7.97 \times 10^{-4}}{0.618} \right]^{\frac{1}{3}} \\ &\quad \times \left[\frac{0.016 \times 996}{7.97 \times 10^{-4}} \right]^{0.8} \\ &= 4298 \end{aligned}$$

For a tube-side velocity of $2 \text{ m}\cdot\text{s}^{-1}$:

$$\begin{aligned} h_T &= K_{hT} V_T^{0.8} \\ &= 4298 \times 2^{0.8} \\ &= 7483 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \end{aligned}$$

Now the overall heat transfer coefficient can be estimated.

$$\begin{aligned} \frac{1}{U} &= \frac{1}{h_C} + \frac{1}{h_{SF}} + \frac{d_O}{2k} \ln \frac{d_O}{d_I} + \frac{d_O}{d_I} \frac{1}{h_{TF}} + \frac{d_O}{d_I} \frac{1}{h_T} \\ &= \frac{1}{1510} + \frac{1}{11,000} + \frac{1}{20,200} + \frac{0.02}{0.016} \left[\frac{1}{5000} + \frac{1}{7483} \right] \end{aligned}$$

$$U = 820 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$$

$$\Delta T_{LM} = \frac{(67 - 35) + (67 - 25)}{\ln \left[\frac{67 - 35}{67 - 25} \right]} = 36.8 \text{ K}$$

Now the duty can be calculated from:

$$\begin{aligned} Q &= UA \Delta T_{LM} \\ &= 820 \times 100 \times 36.8 \\ &= 3.01 \times 10^6 \text{ W} \end{aligned}$$

This does not agree with the specified duty of $2.865 \times 10^6 \text{ W}$. To make the heat duty balanced requires the heat transfer area A to be adjusted by trial and error. At each value of A , the number of tubes and h_C must be calculated. This can be readily done using a spreadsheet solver. The result is:

$$Q = 2.865 \times 10^6 \text{ W}$$

$$h_C = 1491 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$$

$$h_T = 7483 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$$

$$U = 814 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$$

$$A = 96 \text{ m}^2$$

$$D_S = 0.623 \text{ m}$$

$$N_T = 488$$

Rather than specify the tube-side velocity, the tube-side pressure drop could have been specified (e.g. $\Delta P_T = 30,000 \text{ N}\cdot\text{m}^{-2}$). Had this been the case, then the calculation would have required Equation 15.16 to be solved simultaneously with the above equations by varying A and h_T simultaneously, similar to the solution of Example 15.1c.

15.8 REBOILERS AND VAPORIZERS

Reboilers are required for distillation columns to vaporize a fraction of the bottom product, as discussed in Chapter 9. It may also be the case that a liquid needs to be vaporized for other purposes, for example, a liquid feed needs to be vaporized before entering a reactor. The discussion here will focus on reboiling a distillation column, but the same principles apply to other types of vaporizers.

Three common designs of the reboiler are illustrated in Figure 15.13. The first shown in Figure 15.13a is a *kettle reboiler*. Vaporization takes place on the outside of tubes immersed in a pool of liquid. The bottom product is taken from an overflow from the liquid pool and there is no recirculation between the reboiler and the column. In some designs, the tube bundle can be installed in the base of the column as an internal reboiler. The kettle reboiler incorporates a volume above the liquid pool and tube bundle for vapor and liquid disengagement. The shell diameter is typically 40% greater than the bundle diameter to allow for this. The second type of reboiler shown in Figure 15.13b, the *horizontal thermosyphon*, also features vaporization on the outside of the tubes. However, in this case, there is recirculation around the base of the column. A mixture of vapor and liquid leaves the reboiler and enters the base of the column where it separates. The third type of reboiler, the *vertical thermosyphon*, is illustrated in Figure 15.13c. Again, there is a recirculation around the base of the column, but this time the vaporization takes place inside the tubes. Boiling on the tube-side will normally be carried out in a 1–1 exchanger.

The three reboilers in Figure 15.13 are shown under *natural circulation*. The flow of liquid from the column to the reboiler is created by the difference in hydrostatic head between the column of liquid feeding the reboiler and the vapor–liquid mixture created by the reboiler.

The amount of liquid vaporized in the reboiler should not be more than 80%, otherwise this will tend to lead to excessive fouling of the reboiler. For kettle reboilers, there is no recirculation. But for thermosyphon reboilers, a *recirculation ratio* can be defined as:

$$\text{Recirculation ratio} = \frac{\text{flowrate of liquid at reboiler outlet}}{\text{flowrate of vapor at reboiler outlet}}$$

This usually lies between 0.25 and 6. The greater the value of recirculation ratio, the less fouling there is in the reboiler. Lower values tend to be used in horizontal thermosyphons and higher values (greater than 4) used in vertical thermosyphons. The recirculation ratio is a degree of freedom at the discretion of the designer. This should be fixed later when the detailed design is carried out.

The kettle reboiler has the advantage that it is equivalent to a theoretical stage for the distillation but is relatively expensive due to the extra volume required for vapor disengagement. Also, the liquid has a high residence time in the heating zone, which can be a problem if the material is prone to thermal decomposition. If the reboiler must operate at a high pressure, then the large diameter of the kettle shell is a disadvantage. A large diameter cylindrical shell requires a thicker wall to withstand a given pressure than a small diameter shell. The horizontal and vertical thermosyphons both have the disadvantage of not providing a theoretical stage for the distillation. But the thermosyphon reboilers are less prone to fouling than kettle reboilers and have a lower residence time in the heating zone. Thermosyphon reboilers require additional height inside the column shell than kettle reboilers to allow for vapor disengagement as the vapor–liquid mixture enters the column. Vertical thermosyphon reboilers require the column to be at a higher elevation than kettle and horizontal thermosyphons. Horizontal thermosyphons tend to be preferred to vertical ones if the heat transfer area is large, as horizontal arrangements are easier to maintain. Although thermosyphon reboilers can be used under vacuum conditions, care must be exercised, as the effect of pressure on the boiling point of the fluid entering the reboiler must be considered. When reboiling multicomponent systems, the vaporization can take place over a range of temperature. The forced flow in a thermosyphon can give a higher mean temperature difference than a kettle for the same percentage of vaporization, as the kettle boiling temperature is uniform. Generally the heat flux (heat transferred per unit area) and heat transfer coefficients are in the order

kettle < horizontal thermosyphon < vertical thermosyphon

Given these arguments, it is not surprising that the most common design of reboiler is the vertical thermosyphon.

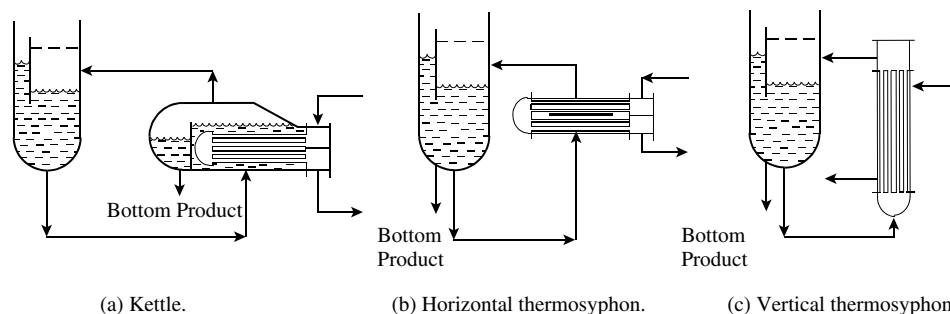


Figure 15.13 Reboiler designs.

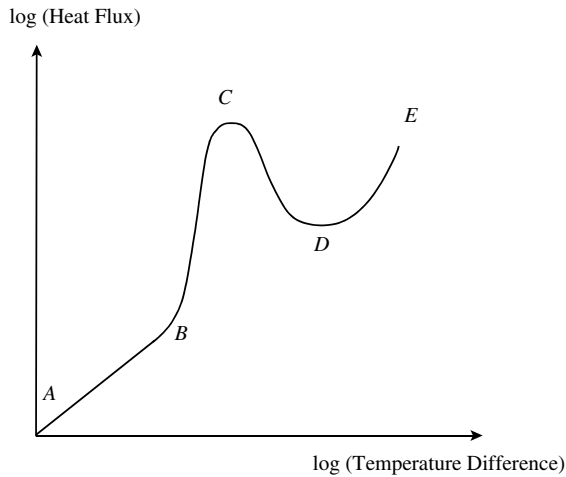


Figure 15.14 Heat transfer characteristics of boiling.

The basic characteristics of the boiling process are illustrated in Figure 15.14. This shows a plot of heat flux versus the temperature difference between the vaporizing surface and the bulk liquid plotted on logarithmic axes. Initially, between Points A and B in Figure 15.14, heat transfer is by natural convection. Superheated liquid rises to the liquid surface where evaporation takes place. As the temperature difference increases beyond Point B in Figure 15.14, nucleate boiling occurs in which vapor bubbles are formed at the heating surface and released from the surface. Nucleate boiling, as its name implies, depends on the presence of nuclei. In boiling, the nuclei are preexisting inclusions of noncondensable gas or vapor in cavities on the heat transfer surface. It thus depends on the character of the heat transfer surface.

Point C in Figure 15.14 is termed the *critical heat flux* or *maximum boiling flux* or *peak boiling flux* as bubbles coalesce on the surface creating a vapor blanket. Critical heat flux occurs because insufficient liquid is able to reach the heat transfer surface due to the rate at which vapor is leaving. Beyond Point D, the surface is dry and entirely blanketed by vapor and heat is transferred by conduction and radiation.

Reboilers are designed to operate below the peak flux, as beyond it either the heat flux would be lower, or much higher temperature difference would be required. Design is normally restricted to have a heat flux less than 70% of the critical flux. Preliminary design of kettle and horizontal thermosyphon reboilers can be based on *pool boiling*. In pool boiling, the heating surface is surrounded by a large body of fluid in which the fluid motion is only induced by natural convection currents and the motion of bubbles. A simple correlation for nucleate boiling that can be used for the preliminary design of kettle and horizontal thermosyphon reboilers is that due to Palen¹⁵:

$$h_{NB} = 0.182 P_C^{0.67} q^{0.7} \left(\frac{P}{P_C} \right)^{0.17} \quad (15.96)$$

where h_{NB} = nucleate boiling coefficient ($\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$)

P_C = liquid critical pressure (bar)

P = operating pressure (bar)

q = heat flux ($\text{W} \cdot \text{m}^{-2}$)

$= h_{NB} (T_w - T_{SAT})$

T_w = wall temperature of the heating surface ($^{\circ}\text{C}$)

T_{SAT} = saturation temperature of the boiling liquid ($^{\circ}\text{C}$).

This equation applies to vaporization of single components, but can be used for close boiling mixtures without too much error. Coefficients for wide boiling mixtures will be overestimated.

Mostinski¹⁶ gives a correlation for the estimation of the critical heat flux for single tubes:

$$q_{C1} = 3.67 \times 10^4 P_C \left(\frac{P}{P_C} \right)^{0.35} \left[1 - \frac{P}{P_C} \right]^{0.9} \quad (15.97)$$

where q_{C1} = critical heat flux for single tube ($\text{W} \cdot \text{m}^{-2}$)

This can be corrected for tube bundles by¹⁵:

$$q_C = q_{C1} \phi_B \quad (15.98)$$

where q_C = critical heat flux for the tube bundle ($\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$)

$$\phi_B = \frac{3.1\pi D_B L}{A}$$

D_B = tube bundle diameter (m)

L = tube length (m)

A = heat transfer area (m^2)

The design of vertical thermosyphon reboilers requires iterative calculations in which the exchanger needs to be divided into zones. The energy and pressure balances need to be performed simultaneously. Frank and Prickett¹⁷ performed a range of detailed simulations and presented the results graphically. This can be used as the basis of preliminary design.

The graphical data can be correlated as:

For *Aqueous Solutions*

$$q = 384.52 \Delta T + 130.07 \Delta T^2 - 2.4204 \Delta T^3 \quad (15.99)$$

where q = heat flux ($\text{W} \cdot \text{m}^{-2}$)

ΔT = mean temperature difference between the heat transfer surface and the bulk fluid (K)

For *Organic Liquids*

$$q = -100.98 + 1705.9 \Delta T + 26.37 \Delta T^2 - 0.288 \Delta T^3 - 5902.8 T_R \Delta T + 6031.3 T_R^2 \Delta T \quad (15.100)$$

where T_R = reduced temperature of fluid

$= T/T_C$

T = temperature (K)

T_C = critical temperature (K)

For the simulations, the heating was assumed to be supplied using saturated steam on the shell-side with a combined condensation and fouling coefficient of $5700 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$. The fouling coefficient on the tube-side was assumed to be $5700 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$.

If these values are appropriate, then the overall heat transfer coefficient can be calculated from:

$$U = \frac{q}{\Delta T} \quad (15.101)$$

The correlation should be used with caution outside the range $0.6 < T_R < 0.8$ and should not be used below a pressure of 0.3 bar. When dealing with a clean, nondegrading material, the process fouling coefficient should be increased to around $11,000 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$, but should be reduced to 1400 to $1900 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$ for material that has a tendency to polymerize¹⁷. If a shell-side coefficient of process fouling coefficient different from $5700 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$ is required, the corrected overall heat transfer coefficient can be calculated from¹⁷:

$$\frac{1}{U'} = \frac{1}{U} - \frac{1}{5700} + \frac{1}{h_S} - \frac{1}{5700} + \frac{1}{h_{TF}} \quad (15.102)$$

where U' = corrected overall heat transfer coefficient ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$)

h_S = required shell-side coefficient including fouling ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$)

h_{TF} = required process fouling coefficient ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$)

The mean temperature difference should be less than 35 to 55°C . This will avoid excessive fouling and excessive vaporization per pass (i.e. low recirculation ratio), leading to poor heat transfer in the upper parts of the tubes as heat transfer to a liquid annulus is replaced by heat transfer to a mist.

Great caution should be exercised regarding correlations for boiling: they are notoriously unreliable. Unlike other heat transfer phenomena, the goal of reliable prediction of boiling rates has proved to be elusive. Many correlations other than those given here are available in the literature. Their predictions of boiling coefficients for the same duty can differ by an order of magnitude. Even the predictions from detailed simulations should be treated with great caution.

Rather than use natural circulation, as in the designs in Figure 15.13, the liquid can be fed to the reboiler by a pump using *forced circulation*. Vaporization in forced convection reboilers is usually limited to be less than 1 to 5%. In some cases, it might be desirable to suppress boiling inside the heat exchanger completely by installing a control valve at the exchanger outlet to increase the pressure in the exchanger and suppress boiling. The liquid leaving the exchanger will then partially vaporize as the liquid pressure is decreased across the valve. Suppression of boiling inside the heat exchanger might be desirable if

it leads to excessive fouling. In preliminary design, forced convection reboilers can be based on the assumption that heat transfer is by forced convection only. This will give a conservative design if some boiling is allowed in the exchanger and will overestimate the heat transfer area. High tube velocities of the order of 3 to $5 \text{ m}\cdot\text{s}^{-1}$ or higher are used to reduce fouling.

If forced-convective boiling is to be carried out, the boiling can take place either on the shell-side or the tube-side. The designs for shell-side boiling are essentially the same as any 1–1 or 1–2 exchanger. Boiling on the tube-side will normally be carried out in a 1–1 exchanger.

Forced convection suppresses nucleate boiling but introduces a significant component of forced convection heat transfer. In fact, in most practical situations, including natural circulation, both forced convection and nucleate boiling are important. In forced-convective boiling, the boiling heat transfer coefficient can be estimated by a combination of convective and nucleate boiling heat transfer with the nucleate boiling component reduced by a suppression factor. There is significant uncertainty associated with the prediction of such components. Because the amount of vapor changes through the exchanger, the exchanger needs to be divided into zones and the correlations applied in each zone.

Natural circulation will lead to cheaper designs than forced circulation, but forced circulation can lead to lower fouling than the corresponding natural circulation designs.

Finally, if vaporization of a liquid is required for applications other than distillation, then the same principles and methods can be applied. The one distinctive difference is that with all designs, with the exception of kettle designs, a vapor–liquid separation device will be required at the vaporizer outlet, as illustrated in Figure 15.15.

Example 15.4 A reboiler is required to supply $0.1 \text{ kmol}\cdot\text{s}^{-1}$ of vapor to a distillation column. The column bottom product is almost pure butane. The column operates with a pressure at the bottom of the column of 19.25 bar. At this pressure, the butane vaporizes at a temperature of 112°C . The vaporization can be assumed to be essentially isothermal and is to be carried out using steam with a condensing temperature of 140°C . The heat of vaporization for butane is $233,000 \text{ J}\cdot\text{kg}^{-1}$, its critical pressure 38 bar, critical temperature 425.2 K and molar mass $58 \text{ kg}\cdot\text{kmol}^{-1}$. Steel tubes with 30 mm outside diameter, 2 mm wall thickness and length 3.95 m are to be used. The thermal conductivity of the tube wall can be taken to be $45 \text{ W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$. The film coefficient (including fouling) for the condensing steam can be assumed to be $5700 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$. Estimate the heat transfer area for

- kettle reboiler
- vertical thermosyphon reboiler

Solution

- Heat load = $0.1 \times 58 \times 233,000$
 $= 1.351 \times 10^6 \text{ W}$

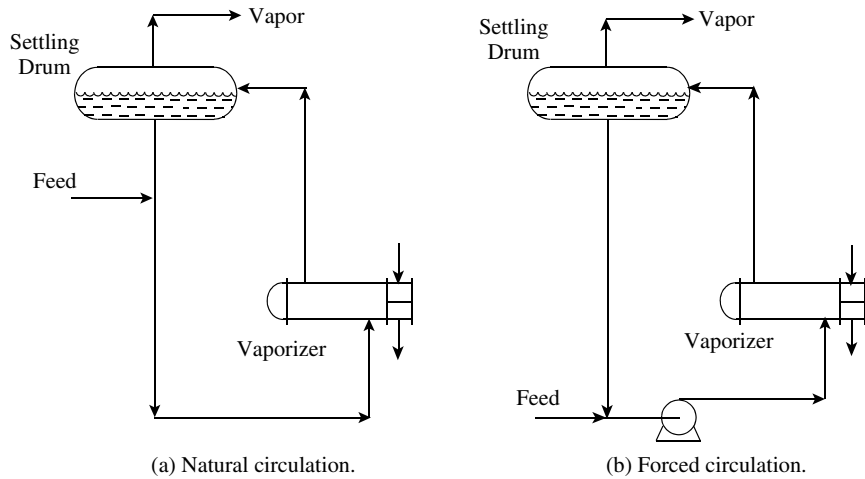


Figure 15.15 Process vaporizer arrangements.

The boiling film coefficient for a kettle reboiler can be estimated from the correlation for pool boiling. Equation 15.96 gives one such method due to Palen¹⁵. However, the correlation requires the heat flux to be known, and therefore the heat transfer area to be known. Hence the calculation will need to be iterative. An initial estimate of the overall heat transfer coefficient of $2000 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$ gives:

$$\begin{aligned}
 A &= \frac{Q}{U \Delta T} \\
 &= \frac{1.351 \times 10^6}{2000 \times (140 - 112)} \\
 &= 24.1 \text{ m}^2 \\
 q &= \frac{Q}{A} \\
 &= \frac{1.351 \times 10^6}{24.1} \\
 &= 56,100 \text{ W}\cdot\text{m}^{-2}
 \end{aligned}$$

Now the boiling film coefficient can be calculated:

$$\begin{aligned}
 h_{NB} &= 0.182 P_C^{0.67} q^{0.7} \left(\frac{P}{P_C} \right)^{0.17} \\
 &= 0.182 \times 38^{0.67} \times 56,100^{0.7} \left(\frac{19.25}{38} \right)^{0.17} \\
 &= 3914 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}
 \end{aligned}$$

The overall heat transfer coefficient can be calculated from:

$$\begin{aligned}
 \frac{1}{U} &= \frac{1}{h_{NB}} + \frac{1}{h_{SF}} + \frac{d_O}{2k} \ln \frac{d_O}{d_I} + \frac{d_O}{d_I} \frac{1}{h_{TF}} + \frac{d_O}{d_I} \frac{1}{h_T} \\
 &= \frac{1}{3914} + \frac{1}{5700} + \frac{0.03}{2 \times 45} \ln \frac{0.03}{0.026} + \frac{0.03}{0.026} \left(\frac{1}{5700} \right) \\
 U &= 1468 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \\
 Q &= UA \Delta T \\
 &= 1468 \times 24.1 \times 28 \\
 &= 0.991 \times 10^6 \text{ W}
 \end{aligned}$$

The calculated heat duty does not agree with the specified duty of $1.351 \times 10^6 \text{ W}$. To make the duty balance requires the heat transfer area to be adjusted by trial and error. At each value of A , a new heat flux and boiling film coefficient is calculated. This allows the new overall heat transfer coefficient to be calculated, and so on, until the calculated heat duty agrees with a specified duty. This can be readily done using a spreadsheet solver. The result is:

$$\begin{aligned}
 Q &= 1.351 \times 10^6 \text{ W} \\
 h_{NB} &= 2883 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \\
 U &= 1295 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1} \\
 A &= 37.3 \text{ m}^2 \\
 q &= 36,250 \text{ W}\cdot\text{m}^{-2}
 \end{aligned}$$

This heat flux must be checked to see if it is below the critical heat flux. The critical heat flux for a single tube is given by the Mostinski Equation (Equation 15.97):

$$\begin{aligned}
 q_{C1} &= 3.67 \times 10^4 P_C \left(\frac{P}{P_C} \right)^{0.35} \left[1 - \frac{P}{P_C} \right]^{0.9} \\
 &= 5.82 \times 10^5 \text{ W}\cdot\text{m}^{-2}
 \end{aligned}$$

To correct this for the bundle requires the bundle diameter to be calculated from Equation 15.37:

$$D_B = \left(\frac{4 P_C P_T^2 A}{\pi d_O L} \right)^{\frac{1}{2}}$$

Assume a pitch of $1.25 d_O$ and square configuration with a tube length of 3.95 m:

$$\begin{aligned}
 D_B &= \left(\frac{4 \times 1 \times (0.03 \times 1.25)^2 \times 37.3}{\pi \times 0.03 \times 3.95} \right)^{\frac{1}{2}} \\
 &= 0.75 \text{ m} \\
 \phi_B &= \frac{3.1 \pi D_B L}{A} \\
 &= \frac{3.1 \times \pi \times 0.75 \times 3.95}{37.3} \\
 &= 0.77
 \end{aligned}$$

$$\begin{aligned}
 q_C &= q_{C1} \phi_B \\
 &= 5.82 \times 10^5 \times 0.77 \\
 &= 4.48 \times 10^5 \text{ W} \cdot \text{m}^{-2}
 \end{aligned}$$

The flux is well below the maximum predicted by the Mostinski Equation.

It should be noted that the shell diameter will be around 40% greater than that of the tube bundle to allow for vapor disengagement.

b. For a vertical thermosyphon reboiler, the heat flux can be approximated using Equation 15.100 for organic liquids.

$$\begin{aligned}
 T_R &= \frac{T}{T_C} \\
 &= \frac{112 + 273.15}{425.2} \\
 &= 0.91
 \end{aligned}$$

This is outside the range over which the original data was correlated and hence the results should be treated with caution.

$$\begin{aligned}
 q &= -100.98 + 1705.9 \times 28 + 26.37 \times 28^2 - 0.288 \times 28^3 \\
 &\quad - 5902.8 \times 0.91 \times 28 + 6031.3 \times 0.91^2 \times 28 \\
 &= 51,460 \text{ W} \cdot \text{m}^{-2} \\
 U &= \frac{q}{\Delta T} \\
 &= \frac{51,460}{28} \\
 &= 1838 \text{ W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}
 \end{aligned}$$

This does not need to be corrected using Equation 15.102 as the steam condensing film coefficient and process fouling coefficient agree with the assumptions on which the correlation is based.

$$\begin{aligned}
 A &= \frac{Q}{q} \\
 &= \frac{1.351 \times 10^6}{51,460} \\
 &= 26.3 \text{ m}^2
 \end{aligned}$$

On the basis of these calculations, the vertical thermosyphon would appear to be the best option. However, the correlations to predict boiling have an extremely low reliability. Predicted minor variations in heat transfer area should not be used to choose options, as the predictions for boiling are so unreliable.

15.9 OTHER TYPES OF HEAT EXCHANGE EQUIPMENT

While the shell-and-tube heat exchanger is the most common type of heat exchanger in the process industries, it does have some significant limitations:

a. The flow is not truly countercurrent. Even the 1–1 exchanger is not truly countercurrent as the flow on the shell-side is partially cross-flow. This means that the

shell-and-tube exchanger is mostly limited to transfer heat with a minimum temperature difference of 10°C. Designs can in some cases achieve a smaller temperature difference (perhaps down to 5°C), but care is needed in the application. Some heat transfer duties demand very small temperature differences.

b. The *area density* (heat transfer area per unit of volume of exchanger) is relatively low. A conventional shell-and-tube heat exchanger has an area density of the order of 100 m²·m^{−3}. Other designs of the heat exchanger can achieve area densities of 300 m²·m^{−3} to 700 m²·m^{−3} and in some designs greater than 1000 m²·m^{−3}.

The most important alternatives to shell-and-tube designs are:

1. *Gasketed plate heat exchanger*. After the shell-and-tube heat exchanger, probably the most commonly used alternative is the *gasketed plate heat exchanger*. This consists of a series of parallel plates with gaskets between the plates to provide a fluid seal. The plates are corrugated both to increase the turbulence (and hence the film transfer coefficients) and to give mechanical rigidity. The enhancement of heat transfer coefficients from the corrugations is a particular advantage when heating and cooling more viscous materials. The turbulence promoted by the corrugations also helps to reduce fouling relative to plain surfaces. The plates are held together and compressed in a frame by the use of lateral bolts. The basic arrangement is illustrated in Figure 15.16. Each corrugated plate is provided with four ports. The gasket arrangement around the four ports forces the hot and cold fluids to flow down alternate channels. This provides countercurrent flow, allowing temperature differences between hot and cold fluids down to around 1°C. Most applications for gasketed plate heat exchangers are for liquid–liquid duties, but can also be applied to condensing and evaporating duties. The limitations of the gasket seals mean that applications are normally restricted to be between −30 and 200°C with pressures up to 20 bar.

A gasketed plate heat exchanger is usually significantly cheaper than a shell-and-tube heat exchanger for the same duty. This is especially the case if the shell-and-tube heat exchanger must be fabricated in more expensive materials such as stainless steel.

Another advantage in retrofit is that an existing frame can often accommodate additional plates if a higher capacity is required. For the same flowrate, an increase in the number of plates decreases the flowrate through the channels and therefore the heat transfer coefficients. However, if the flowrate increases, the number of plates can be increased to accommodate a higher duty (at the expense of increased pressure drop).

The flow arrangement shown in Figure 15.16 involves a single pass for each fluid. More complex flow arrangements can also be used.

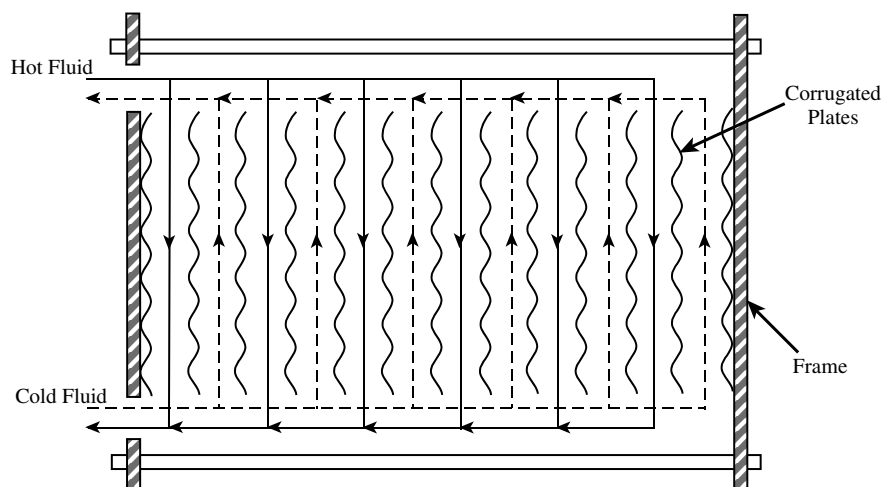


Figure 15.16 Plate-and-frame heat exchanger.

2. *Welded plate heat exchangers.* The limitations of the gaskets in gasketed plate heat exchangers can be overcome by welding the plates together. This eliminates both the gaskets and the frame from the design. Elimination of the gaskets extends the range of application to a wider range of temperatures and pressures. The highest pressures for welded plate heat exchangers can be achieved by mounting the plates within a shell. Welded plate heat exchangers can achieve an area density of up to $300 \text{ m}^2 \cdot \text{m}^{-3}$. The operating temperature range varies between -200 and 900°C . Pressures up to 300 bar can be accommodated.

The costs for this form of exchanger are higher than those for gasketed plate heat exchangers. An important limitation is that they can only be cleaned chemically and not mechanically.

3. *Plate-fin heat exchangers.* Another type of plate heat exchanger is the *plate-fin heat exchanger*. This is illustrated in Figure 15.17. The plate-fin heat exchanger consists of a series of flat plates, between which is a matrix formed

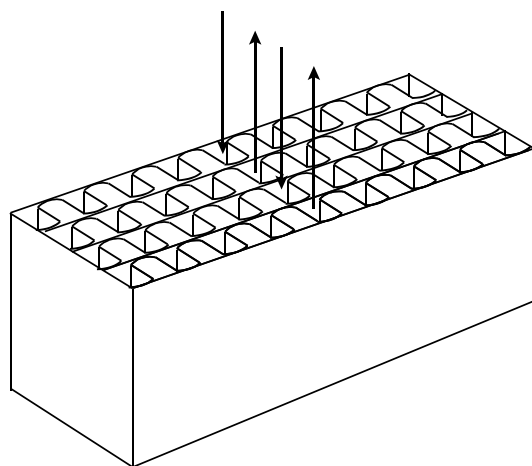


Figure 15.17 Plate-fin heat exchanger.

from corrugated metal that provides a large extended heat transfer area. The components are bonded together either by brazing or diffusion bonding. Many different surface geometries are available to promote heat transfer. The space between the plates and the surface geometry chosen for each fluid are important degrees of freedom in the design of such units. The area density of such units is typically in the range 850 to $1500 \text{ m}^2 \cdot \text{m}^{-3}$. The operating range for such units depends both on the bonding technique used and the material of construction. Aluminum-brazed plate-fin heat exchangers are used for cryogenic applications, but can also be used up to temperatures of around 100°C . Stainless steel plate-fin heat exchangers are able to operate up to 650°C and titanium units up to 550°C . Aluminum-brazed units can operate up to 100 bar, stainless steel units up to 50 bar and higher. Higher pressures require diffusion-bonded units. Plate-fin heat exchangers not only have the advantages of high area density and high heat transfer coefficients but also have a number of other advantages that make them overwhelmingly attractive for certain applications. Temperature differences of 1°C or less can be tolerated in such units. Also, the units can be designed to handle multiple streams through the use of complex header arrangements. This allows, in effect, for a heat exchanger network to be accommodated within a single unit.

4. *Spiral heat exchangers.* Spiral heat exchangers can be thought of as plate heat exchangers in which the plates are formed into a spiral, as illustrated in Figure 15.18. The channels are closed by gasketed end-plates. The hot fluid enters at the center of the unit and flows from the inside outwards. The cold fluid enters at the periphery and flows towards the center in a countercurrent flow arrangement. The gap between the plates can be adjusted to suit the application. Operating temperatures up to 400°C and operating pressures up to 20 bar are possible. The units have a low tendency to foul, but are easily cleaned by removing the end-plates. Again, true countercurrent flow

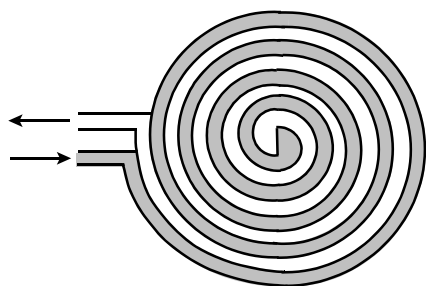


Figure 15.18 Spiral heat exchanger.

is possible and therefore lower temperature driving forces can be tolerated. Spiral heat exchangers are suited to small heat transfer duties that have a tendency to foul.

15.10 FIRED HEATERS

In some situations, process heat needs to be supplied:

- with a high heat duty (e.g. a very large reboiler)
- at a high temperature (e.g. at a temperature above which heat can be supplied by steam)
- with a high heat flux (e.g. heat of reaction in situations where a short residence time in the reactor is required)

In such cases, radiant heat transfer is used from the combustion of fuel in a *fired heater* or *furnace*. Sometimes the function is to purely provide heat; sometimes the fired heater is also a reactor and provides heat of reaction. The special case of steam generation in a fired heater (a steam boiler) will be dealt with in Chapter 23. Fired heater designs vary according to the function, heating duty, type of fuel and the method of introducing combustion air. However, process furnaces have a number of features in common. A simple design is illustrated in Figure 15.19. The chamber where combustion takes place, the *radiant section*

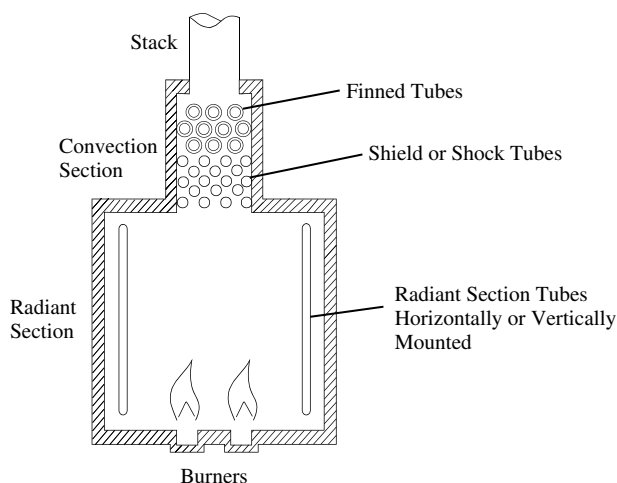


Figure 15.19 A typical furnace arrangement.

or *firebox*, is refractory lined and heat is transferred to tubes mounted in the chamber through which passes the fluid to be heated. Heat transfer in the radiant section is mainly by radiation, with a small contribution (less than 10%) by convection. The tubes are usually mounted around the walls of the radiant section and can be mounted vertically or horizontally. The burners can be mounted in the base or the walls of the radiant zone. The fuel for process-fired heaters is usually gaseous or liquid. The shape of the radiant section can be cylindrical or rectangular.

After the flue gas leaves the combustion chamber, most furnace designs extract further heat from the flue gas in horizontal banks of tubes in a *convection section*, before the flue gas is vented to the atmosphere. The temperature of the flue gases at the exit of the radiant section is usually in the range 700 to 900°C. The first few rows of tubes at the exit of the radiant section are plain tubes, known as *shock tubes* or *shield tubes*. These tubes need to be robust enough to be able to withstand high temperatures and receive significant radiant heat from the radiant section. Heat transfer to the shock tubes is both by radiation and by convection. After the shock tubes, the hot flue gases flow across banks of tubes that usually have extended surfaces to increase the rate of heat transfer to the flue gas. The heat transferred in the radiant section will usually be between 50 and 70% of the total heat transferred.

Heat input to the fired heater is from three sources:

- net heat of combustion (Q_{COMB})
- sensible heat of combustion air (Q_{AIR})
- sensible heat of the fuel, together with heat from atomizing steam if it is used for heavy fuel oil combustion (Q_{FUEL})

The heat output from the fired heater is to four sinks:

- heat transferred to the radiant section tubes (Q_{RADIANT})
- heat transferred in the convection section (Q_{CONV})
- casing losses (Q_{CASING})
- sensible heat of exit flue gas (Q_{LOSS})

Overall, there must be an energy balance such that:

$$Q_{\text{COMB}} = Q_{\text{RADIANT}} + Q_{\text{CONV}} + Q_{\text{CASING}} + Q_{\text{LOSS}} - Q_{\text{AIR}} - Q_{\text{FUEL}} \quad (15.103)$$

The split between the radiant and convection section heat varies according to the design. Casing losses are usually between 1 and 3% of the heat release from combustion. The heat loss from the stack is constrained by the desire to avoid any condensation of water vapor in the convection section. If there is any sulfur present in the fuel, then the condensate will be corrosive. The temperature at which the flue gas starts to condense is the acid dew point. For sulfur-bearing fuels, the temperature of the flue gas is normally

kept above 150 to 160°C. For combustion of sulfur-free gaseous fuels, the temperature can be decreased to below 100°C.

Three methods are used to produce airflow through the fired heater:

- Natural Draft.** In natural draft, the pressure in the furnace is maintained slightly below atmospheric by a flow of air created from the difference in density between the hot flue gas in the stack and the ambient air. The stack must therefore be high enough to provide adequate draft.
- Forced Draft.** Forced draft uses a fan to create the airflow by blowing air into the furnace. The furnace is slightly above atmospheric pressure. The stack height is now only required to provide adequate gas dispersion.
- Induced Draft.** Induced draft uses a fan between the furnace and the stack, leading to a pressure in the furnace slightly below atmospheric. Again, the stack height is only required to provide adequate gas dispersion.

Natural draft is a low capital cost option, but the draft requirements tend to lead to high stack temperature with a low efficiency for the furnace. Forced draft and induced draft are higher capital cost options than natural draft but tend to lead to higher furnace efficiencies as the stack temperature can be lowered significantly.

The preliminary specification of fired heaters is largely based on heat duty and furnace efficiency. A simple model can be developed on the basis of the concept of the *theoretical flame temperature* or *adiabatic combustion temperature*. Theoretical flame temperature is the temperature attained when a fuel is burnt in air or oxygen without loss or gain of heat. A combustion process is the reaction of carbon, hydrogen, sulfur and nitrogen in the fuel with oxygen to produce carbon dioxide, carbon monoxide, water, sulfur dioxide, sulfur trioxide and oxides of nitrogen. Depending on the fuel being solid, liquid or gaseous, the carbon, hydrogen, sulfur and nitrogen in the fuel can be in their elemental forms or as compounds. Nitrogen in combustion air or elemental nitrogen in a gaseous fuel can react with oxygen to form oxides of nitrogen at high temperatures. However, the formation of oxides of nitrogen in this way can be neglected when calculating the theoretical flame temperature, as only small quantities are formed. Also, in a well-designed and operated combustion device, there should be no carbon monoxide formed, but all carbon should be oxidized to carbon dioxide. Any sulfur present in the fuel can be assumed to react to form sulfur dioxide. In practice, some formation of sulfur trioxide will occur, but measurements on combustion systems indicate that sulfur trioxide is only formed in small quantities and less than what would be predicted by thermodynamic equilibrium with the sulfur dioxide. Thus, it can be assumed that all sulfur in the fuel reacts to sulfur dioxide.

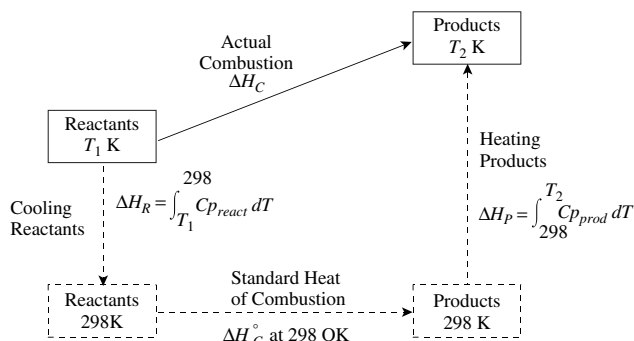


Figure 15.20 The enthalpy change from initial to final state in a combustion process is independent of path.

To calculate the heat release from combustion and the temperature of the products of combustion, the thermodynamic path shown in Figure 15.20 can be followed¹⁸. The actual combustion process goes from reactants at temperature T_1 to products at temperature T_2 . However, it is more convenient to follow the alternative path from reactants at temperature T_1 that are initially cooled (or heated) to standard temperature of 298 K. The combustion reactions are then carried out at a constant temperature of 298 K. Standard heats of combustion are available for this. The products of combustion are then heated from 298 K to the final temperature of T_2 . The actual heat of combustion is given by¹⁸:

$$\Delta H_{\text{COMB}} = \Delta H_R + \Delta H_{\text{COMB}}^O + \Delta H_P \quad (15.104)$$

where ΔH_{COMB} = heat of combustion ($\text{J}\cdot\text{kmol}^{-1}$)

ΔH_R = heat to bring reactants from their initial temperature to standard temperature ($\text{J}\cdot\text{kmol}^{-1}$)

$$= \int_{T_1}^{298} C_{p_{\text{react}}} dT$$

ΔH_{COMB}^O = standard heat of combustion at 298 K ($\text{J}\cdot\text{kmol}^{-1}$)

ΔH_P = heat to bring products from standard temperature to the final temperature ($\text{J}\cdot\text{kmol}^{-1}$)

$$= \int_{298}^{T_2} C_{p_{\text{prod}}} dT$$

$C_{p_{\text{react}}}$ = heat capacity of reactants ($\text{J}\cdot\text{kmol}^{-1}\cdot\text{K}^{-1}$)

$C_{p_{\text{prod}}}$ = heat capacity of products ($\text{J}\cdot\text{kmol}^{-1}\cdot\text{K}^{-1}$)

For adiabatic combustion, $\Delta H_{\text{COMB}} = 0$ and Equation 15.89 becomes:

$$\Delta H_P = -\Delta H_R - \Delta H_{\text{COMB}}^O \quad (15.105)$$

Standard heats of combustion are widely available. Table 15.11 provides a list of some combustion reactions.

When using data for standard heats of combustion, care should be taken regarding the initial state of the reactants and the final state of the products. If these do not correspond with the conditions for the actual combustion, then errors can arise. Note in Table 15.11, the final state of water was

Table 15.11 Standard heats of combustion.

Reaction	ΔH_{COMB}^O at 298 K (MJ·kmol ⁻¹)
C(sol) + O ₂ → CO ₂	-393.5
S(sol) + O ₂ → SO ₂	-297.1
CO + $\frac{1}{2}$ O ₂ → CO ₂	-283.0
H ₂ + $\frac{1}{2}$ O ₂ → H ₂ O(vap)	-241.8
CH ₄ (vap) + 2O ₂ → CO ₂ + 2H ₂ O(vap)	-802.8
C ₂ H ₆ (vap) + 3 $\frac{1}{2}$ O ₂ → 2CO ₂ + 3H ₂ O(vap)	-1428.7
C ₃ H ₈ (vap) + 5O ₂ → 3CO ₂ + 4H ₂ O(vap)	-2043.2
n-C ₄ H ₁₀ (vap) + 6 $\frac{1}{2}$ O ₂ → 4CO ₂ + 5H ₂ O(vap)	-2657.7
i-C ₄ H ₁₀ (vap) + 6 $\frac{1}{2}$ O ₂ → 4CO ₂ + 5H ₂ O(vap)	-2649.1

taken in all cases to be vapor rather than liquid. This relates to the condition of the combustion products as they would normally be in practice. Combustion data for fuels are given in terms of different conditions for the water that is formed. When the water vapor remains in the vapor phase, the heat of combustion is known as the *net calorific value* or *lower heating value*. When the water condenses, it is known as the *gross calorific value* or *higher heating value*. Data can obviously be converted from one to the other from knowledge of how many molecules of water are formed and the heat of vaporization of water. When dealing with solid fuels, extra care needs to be taken as data can be quoted “as fired” or on a “dry ash-free basis”.

Data for some typical gaseous and liquid fuels are given in Tables 15.12 and 15.13.

One further problem needs to be considered before a calculation can be attempted. The heat balance as illustrated in Figure 15.20 involves extremely large temperature changes for the combustion process. This means that the

Table 15.13 Data for some typical liquid fuels.

Fuel	Composition (% by mass)				Calorific value (MJ·kg ⁻¹)	
	C	H	S	O + N + Ash	Gross	Net
Light fuel oil	85.6	11.7	2.5	0.2	43.5	41.1
Medium fuel oil	85.6	11.5	2.6	0.3	43.1	40.8
Heavy fuel oil	85.4	11.4	2.8	0.4	42.9	40.5

Table 15.12 Data for some typical natural gases.

Gas	Composition (% by volume)								Calorific value (MJ·m ⁻³)	
	CO ₂	N ₂	CH ₄	C ₂ H ₆	C ₃ H ₈	C ₁₀ H ₁₄	C ₅ H ₁₂	C ₅ H ₁₄	Gross	Net
North Sea	0.2	1.5	94.4	3.0	0.5	0.2	0.1	0.1	38.62	34.82
Groningen	0.9	15.0	81.8	2.7	0.4	0.1	0.1	–	33.28	30.00
Algeria	0.2	5.5	83.8	7.1	2.1	0.9	0.4	–	39.1	

heat capacity will vary significantly, especially for the products of combustion. The variation in heat capacity can be expressed as a polynomial in temperature, as discussed in Chapter 6. However, with combustion processes, care should be taken that the data have been correlated over a wide range of temperature. Thus, for example¹⁸:

$$C_P = \alpha_0 + \alpha_1 T + \alpha_2 T^2 + \alpha_3 T^3 \quad (15.106)$$

where C_P = heat capacity (J·kmol⁻¹·K⁻¹ or kJ·kmol⁻¹·K⁻¹)

$\alpha_0, \alpha_1, \alpha_2, \alpha_3$ = constants

T = absolute temperature (K)

Thus,

$$\begin{aligned} \Delta H &= \int_{T_1}^{T_2} C_P dT \\ &= \int_{T_1}^{T_2} (\alpha_0 + \alpha_1 T + \alpha_2 T^2 + \alpha_3 T^3) dT \\ &= \left[\alpha_0 T + \frac{\alpha_1 T^2}{2} + \frac{\alpha_2 T^3}{3} + \frac{\alpha_3 T^4}{4} \right]_{T_1}^{T_2} \quad (15.107) \end{aligned}$$

where ΔH = enthalpy change from T_1 to T_2 (J·kmol⁻¹, kJ·kmol⁻¹)

Equation 15.92 and Table 15.14 can be used to calculate the enthalpy changes, for example, in a spreadsheet.

Table 15.14 Heat capacity constants.

Component	C_P (kJ·kmol ⁻¹ ·K ⁻¹)			
	α_0	$\alpha_1 \times 10^2$	$\alpha_2 \times 10^5$	$\alpha_3 \times 10^9$
O ₂	25.4767	1.5202	-0.7155	1.3117
N ₂	28.9015	-0.1571	0.8081	-2.8726
H ₂ O	32.2384	0.1923	1.0555	-3.5952
CO ₂	22.2570	5.9808	-3.5010	7.4693
SO ₂	25.7781	5.7945	-3.8112	8.6122
CO	28.3111	0.1675	0.5372	-2.2219
H ₂	29.1066	-0.1916	0.4004	-0.8704
CH ₄	19.8873	5.0242	1.2686	-11.0113
C ₂ H ₆	6.8998	17.2664	-6.4058	7.2850
C ₃ H ₈	-4.0444	30.4757	-15.7214	31.7359
n-C ₄ H ₁₀	3.9565	37.1495	-18.3382	35.0016
i-C ₄ H ₁₀	-7.9131	41.6000	-23.0065	49.9067

Table 15.15 Mean heat capacity data.

$T(^{\circ}\text{C})$	$\overline{C_P}$ (kJ·kmol ⁻¹ ·K ⁻¹)											
	O ₂	N ₂	H ₂ O	CO ₂	SO ₂	CO	H ₂	CH ₄	C ₂ H ₆	C ₃ H ₈	<i>n</i> -C ₄ H ₁₀	<i>i</i> -C ₄ H ₁₀
25	29.41	29.07	33.65	37.17	39.89	29.12	28.87	35.69	52.86	73.65	99.30	96.95
100	29.82	29.18	33.94	38.65	41.24	29.40	28.88	37.76	57.87	81.65	109.20	107.56
200	30.33	29.34	34.36	40.47	42.87	29.64	28.92	40.51	64.22	91.59	121.56	120.68
300	30.81	29.54	34.82	42.12	44.33	29.89	28.98	43.25	70.20	100.75	132.99	132.70
400	31.26	29.76	35.31	43.61	45.61	30.16	29.05	45.97	75.84	109.19	143.55	143.69
500	31.67	30.00	35.83	44.96	46.74	30.44	29.14	48.64	81.13	116.95	153.29	153.73
600	32.05	30.26	36.38	46.17	47.74	30.73	29.25	51.26	86.08	124.07	162.27	162.88
700	32.41	30.53	36.94	47.25	48.60	31.02	29.37	53.80	90.72	130.61	170.53	171.23
800	32.73	30.80	37.52	48.23	49.34	31.31	29.51	56.24	95.05	136.61	178.14	178.84
900	33.04	31.09	38.10	49.10	49.99	31.59	29.65	58.58	99.08	142.12	185.14	185.80
1000	33.32	31.36	38.68	49.88	50.54	31.88	29.80	60.79	102.82	147.19	191.59	192.18
1100	33.58	31.64	39.27	50.58	51.02	32.15	29.97	62.86	106.28	151.86	197.53	198.05
1200	33.82	31.90	39.84	51.21	51.43	32.40	30.14	64.77	109.48	156.19	203.03	203.49
1300	34.04	32.15	40.39	51.78	51.79	32.64	30.32	66.51	112.42	160.22	208.12	208.58
1400	34.25	32.39	40.93	52.31	52.12	32.86	30.50	68.05	115.12	163.99	212.88	213.38
1500	34.44	32.60	41.44	52.80	52.42	33.06	30.69	69.39	117.58	167.57	217.34	217.98
1600	34.63	32.78	41.93	53.27	52.70	33.23	30.87	70.50	119.83	170.98	221.56	222.44
1700	34.81	32.93	42.37	53.73	52.99	33.37	31.06	71.37	121.86	174.28	225.59	226.84
1800	34.97	33.05	42.78	54.18	53.29	33.48	31.25	71.98	123.69	177.53	229.49	231.27
1900	35.14	33.13	43.13	54.65	53.62	33.55	31.44	72.32	125.33	180.76	233.31	235.78
2000	35.30	33.16	43.44	55.14	53.99	33.58	31.62	72.37	126.79	184.02	237.09	240.47
2100	35.46	33.14	43.69	55.66	54.41	33.60	31.80	72.11	128.08	187.36	240.91	245.39
2200	35.62	33.07	43.87	56.22	54.89	33.51	31.98	71.52	129.22	190.84	244.79	250.63

Alternatively, the heat capacity data can be used to derive mean heat capacities. The mean heat capacity can be defined as¹⁸:

$$\overline{C_P} = \frac{\int_{T_1}^{T_2} C_P dT}{T_2 - T_1} \quad (15.108)$$

where $\overline{C_P}$ = mean heat capacity between temperatures T_1 and T_2 (J·kmol⁻¹·K⁻¹ or kJ·kmol⁻¹·K⁻¹)

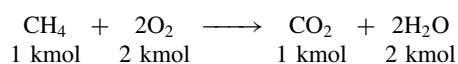
Table 15.15 presents mean heat capacity data between 25°C and a given temperature for a range of temperatures.

Example 15.5 A gas, which can be considered to be pure methane, is to be used as fuel in a furnace. Both the fuel gas and combustion air are both at 25°C. Calculate the theoretical flame temperature if the methane is burnt in:

- its stoichiometric ratio in dry air
- 15% excess dry air
- 15% excess air with a relative humidity of 60%

Solution

- Stoichiometric air requirements are defined by



2 kmol oxygen are required per kmol of methane burnt. If air is assumed to be 21% of oxygen and nitrogen is assumed inert,

then combustion products are:

$$\text{O}_2 = 2 \text{ kmol}$$

$$\text{N}_2 = \frac{2 \times 0.79}{0.21} = 7.52 \text{ kmol}$$

$$\text{H}_2\text{O} = 2 \text{ kmol}$$

$$\text{CO}_2 = 1 \text{ kmol}$$

Since the fuel and combustion air are at the standard temperature of 25°C, $\Delta H_R = 0$. To calculate ΔH_p , start by estimating the theoretical flame temperature to be 2000°C. From:

$$\overline{C_{P\text{O}_2}} = 35.30 \text{ kJ·kmol}^{-1}\cdot\text{K}^{-1}$$

$$\overline{C_{P\text{N}_2}} = 33.16 \text{ kJ·kmol}^{-1}\cdot\text{K}^{-1}$$

$$\overline{C_{P\text{H}_2\text{O}}} = 43.44 \text{ kJ·kmol}^{-1}\cdot\text{K}^{-1}$$

$$\overline{C_{P\text{CO}_2}} = 55.14 \text{ kJ·kmol}^{-1}\cdot\text{K}^{-1}$$

$$\begin{aligned} \Delta H_p &= (7.52 \times 33.16 + 2 \times 43.44 + 1 \times 55.14)(T - 25) \\ &= 391.38(T - 25) \end{aligned}$$

From Table 15.11, $\Delta H_{\text{COMB}}^{\circ} = 802.8 \times 10^3 \text{ kJ·kmol}^{-1}$. Thus, from an energy balance:

$$\Delta H_p = -\Delta H_R - \Delta H_{\text{COMB}}^{\circ}$$

$$391.38(T_{\text{TFT}} - 25) = 0 - (-802.8 \times 10^3)$$

$$T_{\text{TFT}} = 2076^{\circ}\text{C}$$

This agrees well with the initial estimate. Had there been significant disagreement, then revised mean heat capacities would need to be taken and the calculation repeated.

b. If 15% excess air is used, then combustion products are:

$$\text{O}_2 = 2 \times 0.15 = 0.3 \text{ kmol}$$

$$\text{N}_2 = \frac{2 \times 0.79}{0.21} \times 1.15 = 8.65 \text{ kmol}$$

$$\text{H}_2\text{O} = 2 \text{ kmol}$$

$$\text{CO}_2 = 1 \text{ kmol}$$

Again, estimate the theoretical flame temperature to be 2000°C. The mean heat capacities are as before and the heat balance is now:

$$\begin{aligned} \Delta H_P &= (0.3 \times 35.3 + 8.65 \times 33.16 + 2 \\ &\quad \times 43.44 + 1 \times 55.14)(T_{TFT} - 25) \\ &= 439.44(T_{TFT} - 25) \end{aligned}$$

$$\Delta H_P = \Delta H_R - \Delta H_{COMB}^o$$

$$439.44(T_{TFT} - 25) = 0 - (-802.8 \times 10^3)$$

$$T_{TFT} = 1852^\circ\text{C}$$

c. So far the combustion air has been assumed to be dry. If the combustion air is humid, then the water vapor will act as another inert in the combustion. The relative humidity of air is the percentage relative to saturation.

Saturated vapor pressure of water at 25°C (from steam tables)

$$= 0.03166 \text{ bar}$$

For 60% relative humidity, the partial pressure of water vapor is given by:

$$\begin{aligned} p_{\text{H}_2\text{O}} &= 0.03166 \times 0.6 \\ &= 0.0190 \text{ bar} \end{aligned}$$

Thus, the mole fraction of water vapor in the combustion air for a pressure of 1 atm (1.013 bar) is given by:

$$\begin{aligned} y_{\text{H}_2\text{O}} &= \frac{p_{\text{H}_2\text{O}}}{P} \\ &= \frac{0.0190}{1.013} \\ &= 0.0188 \end{aligned}$$

The combustion air for 15% excess

$$\begin{aligned} &= (2 + 7.52)1.15 \\ &= 10.95 \text{ kmol} \end{aligned}$$

Water from combustion air

$$\begin{aligned} &= 10.95 \times \frac{y_{\text{H}_2\text{O}}}{1 - y_{\text{H}_2\text{O}}} \\ &= 10.95 \times \frac{0.0188}{1 - 0.0188} \\ &= 0.21 \text{ kmol} \end{aligned}$$

The total water vapor in the combustion products

$$\begin{aligned} &= 2 + 0.21 \\ &= 2.21 \text{ kmol} \end{aligned}$$

Again, estimate the theoretical flame temperature to be 2000°C. The mean heat capacities are as before and the heat balance is:

$$\begin{aligned} \Delta H_P &= (0.3 \times 35.3 + 8.65 \times 33.16 + 2.21 \\ &\quad \times 43.44 + 1 \times 55.14) \times (T_{TFT} - 25) \\ &= 448.57(T_{TFT} - 25) \end{aligned}$$

$$448.57(T_{TFT} - 25) = 0 - (-802.8 \times 10^3)$$

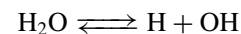
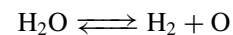
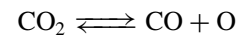
$$T_{TFT} = 1815^\circ\text{C}$$

Again, iterate with revised heat capacities for greater accuracy.

It can be seen that excess air and humidity in the combustion air both act to reduce the theoretical flame temperature. However, the excess air has the more significant effect. In some combustion processes, steam is injected into the combustion process to decrease the flame temperature to decrease NO_x formation. This will be discussed later in Chapter 25.

In these calculations, the fuel and combustion air were both at the standard temperature of 25°C. If the temperature of either had been below 25°C, then ΔH_R would have acted to decrease the theoretical flame temperature. If either had been above 25°C, the effect would have been to increase the theoretical flame temperature. One energy conservation technique sometimes used in furnace design is to use waste heat to preheat the combustion air. This has the effect of increasing the theoretical flame temperature, and as will be seen later, increases the fuel efficiency.

It should be emphasized that the theoretical and real flame temperatures will be significantly different. The real flame temperature will be lower than the theoretical flame temperature because, in practice, heat is lost from the flame (mainly due to radiation). Also, part of the heat released provides heat for a variety of endothermic dissociation reactions, which occur at high temperatures, such as:



However, as the temperature of the flue gas decreases, as heat is extracted, the dissociation reactions reverse and the heat is released. Thus, although theoretical flame temperature does not reflect the true flame temperature, it does provide a convenient reference to indicate how much heat is actually released by combustion as the flue gas is cooled. Figure 15.21 shows the flue gas starting from the theoretical flame temperature. This is cooled

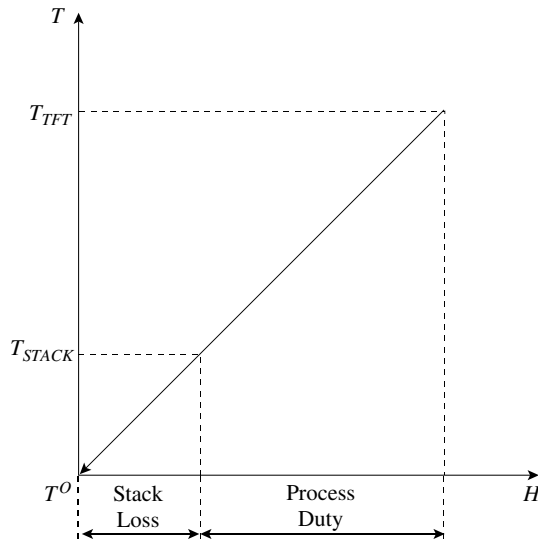


Figure 15.21 The flue gas profile for a fired heater.

in the radiant and convection sections to provide the process duty. The temperature at the exit of the convection section is the stack temperature. The cooling from the stack temperature to the ambient temperature is the *stack loss*. The temperatures in the radiant section (above 700 to 900°C) are thus not accurately represented by the profile based on the theoretical flame temperature. The temperature differences between the sources of radiant heat and the furnace tubes in the radiant zone of the furnace cannot be represented accurately by simple models, and requires a detailed simulation model. Such a model must account for the burner design, the absorption and emission of radiant heat for the surfaces in the radiant zone of the particular geometry, and so on. Fortunately, this does not usually present a problem in using the simple model represented in Figure 15.21 for preliminary design, since temperature differences in the radiant zone are very large anyway. Also, as the flue gas is cooled and passes through the convection section of the furnace, the temperatures are more representative of what they would be in practice. However, it should be emphasized again that the simple model in Figure 15.21 does allow the correct heat duty to be represented. Furnace efficiency can be defined as:

$$\text{Furnace efficiency} = \frac{\text{Heat to process}}{\text{Heat released by fuel}} \quad (15.109)$$

The profile shown in Figure 15.21 represents the furnace efficiency, if the casing heat losses are neglected. Making this assumption, the process duty plus the stack loss represents the heat released by the fuel.

Example 15.6 For the three cases in Example 15.5, determine the furnace efficiency for an assumed stack temperature of 100°C.

Solution

- a. For dry, stoichiometric air $T_{TFT} = 2076^\circ\text{C}$:

$$\begin{aligned} \text{Furnace efficiency} &= \frac{\text{Heat to process}}{\text{Heat released by fuel}} \\ &= \frac{391.38(2076 - 100)}{802.8 \times 10^3} \\ &= 0.963 \end{aligned}$$

- b. For dry air in 15% excess, $T_{TFT} = 1852^\circ\text{C}$:

$$\begin{aligned} \text{Furnace efficiency} &= \frac{439.44(1852 - 100)}{802.8 \times 10^3} \\ &= 0.96 \end{aligned}$$

- c. For 15% excess air with 60% relative humidity $T_{TFT} = 1815^\circ\text{C}$:

$$\begin{aligned} \text{Furnace efficiency} &= \frac{448.57(1815 - 100)}{802.8 \times 10^3} \\ &= 0.958 \end{aligned}$$

For a given stack temperature, the higher the theoretical flame temperature, the higher the furnace efficiency. However there is a minimum excess air required to ensure that the combustion is itself efficient.

All combustion processes work with an excess of air or oxygen to ensure complete combustion of the fuel. Excess air typically ranges between 5 and 25% depending on the fuel, burner design and furnace design. Typical excess air is 10 to 15% for gaseous fuels, 15 to 20% for liquid fuels and 20 to 25% for solid fuels. Natural draft furnaces normally work with a higher excess air than forced draft designs. The excess oxygen in the flue gas should normally be around 3%. However, in large furnaces, the excess oxygen might be significantly less (e.g. 2% for fuel oil, 1% for natural gas).

As excess air is reduced, theoretical flame temperature increases. This has the effect of reducing the stack loss and increasing the thermal efficiency of the furnace for a given process heating duty. Alternatively, if the combustion air is preheated (e.g. by heat recovery), then again the theoretical flame temperature increases, reducing the stack loss.

Although, higher flame temperatures reduce the fuel consumption for a given process heating duty, there is one significant disadvantage. Higher flame temperatures increase the formation of oxides of nitrogen, which are environmentally harmful.

Obviously, the lower the stack temperature, the higher the furnace efficiency. As already noted, it is desirable to avoid condensation in the convection section and the stack. If there is any sulfur in the fuel, the condensate will be corrosive. There is thus a practical minimum to which a flue gas can be cooled without condensation causing corrosion in the stack, known as the *acid dew point*. If there is any sulfur in the fuel, the stack temperature is normally kept above 150 to 160°C. Natural gas can normally be cooled to

a significantly lower temperature (100°C or less), because it is usually treated to remove sulfur down to less than 1 ppm.

In preliminary design, the heat duty and furnace efficiency are the prime considerations. However, if the tube area needs to be specified, a preliminary estimate can be obtained from an assumed flux. In the radiant section, this usually lies in the range of 45,000 W·m⁻² to 65,000 W·m⁻² of tube surface, with a value of around 55,000 W·m⁻² most often used. The heat flux is particularly important if a reaction is being carried out in the furnace tubes. Overall heat transfer coefficients in the convection section are in the range 20 to 50 W·m⁻²·K⁻¹.

15.11 HEAT TRANSFER EQUIPMENT – SUMMARY

Of the many types of heat transfer equipment used in the process industries, the shell-and-tube heat exchanger is by far the most common. A number of different flow arrangements are possible with shell-and-tube heat exchangers, but the one shell pass–one tube pass and the one shell pass–two tube pass arrangements are the most common.

Resistance to heat transfer across the tube wall for shell-and-tube heat exchangers is made up of five individual resistances to heat transfer:

- shell-side film coefficient
- shell-side fouling coefficient
- tube wall coefficient
- tube-side fouling coefficient
- tube-side film coefficient

These are combined to give an overall heat transfer coefficient in which one of the individual coefficients might be controlling.

At the conceptual stage for the design of shell-and-tube heat exchangers, the calculation of heat transfer coefficients and pressure drops should depend as little as possible on the detailed geometry. Simple models are available in which both heat transfer coefficients and pressure drops can be related to the fluid velocity. It is then possible to derive a correlation between the heat transfer coefficient, pressure drop and heat transfer surface area by using the velocity as a bridge between the two. The 1–1 heat exchanger is designed as a countercurrent device. The 1–2 heat exchanger has an element of cocurrent flow as well as the countercurrent flow. This must be corrected by the introduction of a correction to the logarithmic mean temperature difference.

In retrofit situations, existing heat exchangers might be subjected to changes in flowrate, heat transfer duty, temperature differences or fouling characteristics. Heat transfer coefficients and pressure drops can be approximated from

their original values from the change in flowrate. Increased duties can be accommodated by changing the tube bundle to incorporate a greater heat transfer area in the same shell, or by the use of tube insert enhancement devices.

Shell-and-tube heat exchangers are also used extensively for condensing duties. Condensers can be horizontally or vertically mounted with the condensation on the tube-side or the shell-side. Condensation normally takes place on the shell-side of horizontal exchangers and the tube-side of vertical exchangers.

Reboilers are required on distillation columns to vaporize a fraction of the bottom product. Three common designs of reboiler are used: the kettle reboiler, the horizontal and vertical thermosyphon reboilers. The preliminary design of kettle and horizontal thermosyphon reboilers can be based on correlations for pool boiling. The design of vertical thermosyphon reboilers requires the hydraulic and thermal design to be carried out simultaneously. Preliminary design of such units can be based on correlations derived from a large number of detailed designs. Great care must be exercised in the preliminary design of reboilers as the predictions of the correlations are extremely unreliable.

While the shell-and-tube heat exchanger is the most commonly used in the process industries, it has the disadvantages that the flow is not truly countercurrent, which limits the minimum temperature difference that can be accommodated, and the area density is relatively low. Commonly used alternatives for shell-and-tube heat exchangers are:

- Gasketed plate heat exchangers
- Welded plate heat exchangers
- Plate-fin heat exchangers
- Spiral heat exchangers

Some heat transfer operations demand a high heat duty, high temperature and/or high heat flux. In such cases, radiant heat transfer is used from the combustion of fuel in a fired heater. Fired heater designs vary according to the function, heating duty, type of fuel and method of introducing combustion air. In conceptual design, the heat duty in the various sections of the furnace is more important than the heat transfer area, as the cost in preliminary design is based on the heat duty. The theoretical flame temperature provides a simple model to allow the heat duty to be determined.

15.12 EXERCISES

1. A distillation operation separating a low-viscosity hydrocarbon mixture requires three shell-and-tube heat exchangers. The liquid feed is to be preheated to saturated liquid by heat recovery from another low-viscosity hydrocarbon stream. The reboiler is to be a vertical thermosyphon using steam heating.

The condenser is to be a horizontal exchanger with the condensing stream on the shell-side and the cooling serviced by cooling water. What would be the expected rank order of the three heat exchangers in terms of their overall heat transfer coefficient?

2. For the three heat exchangers from Exercise 1, make a first estimate of the order of magnitude of the overall heat coefficients from tabulated values of film transfer coefficients and fouling coefficients. Neglect the resistance from the tube walls.
3. Under what circumstances might a thermosyphon reboiler be orientated vertically or horizontally?
4. Under what circumstances might a distillation condenser be orientated vertically or horizontally?
5. The demethanizer distillation column of an ethylene process works at extremely low temperatures. The feed is cooled with extremely small temperature differences of the order of 1°C to minimize the refrigeration costs associated with the cooling. What type of heat exchanger would you expect to be used for this duty?
6. Gasketed plate heat exchangers are commonly used in food processing. What advantages does the design offer in such applications?
7. A hot stream is to be cooled from 210 to 80°C by heating a cold stream from 60 to 150°C with a duty of 1.7 MW. A 1–1 shell-and-tube heat exchanger is to be used and the overall heat transfer coefficient has been estimated to be $120\text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$. Calculate the heat transfer area of the unit.
8. Instead of using a 1–1 design in Example 7, a 1–2 design is to be used subject to $X_P = 0.9$. Assume that the overall heat transfer coefficient is unchanged. (In practice, it would be expected to increase). Calculate
 - a. the number of shells required
 - b. P_{1-2} for each shell
 - c. F_T for the shells in series
 - d. the heat transfer area
9. Liquid *n*-butanol at 115° is to be cooled to 45°C against cooling water between 25 and 35°C . The flowrate of *n*-butanol is $10\text{ kg}\cdot\text{s}^{-1}$. A 1–1 shell-and-tube heat exchanger is to be used. The cooling water has a greater fouling tendency than the *n*-butanol and will have a higher flowrate; hence the cooling water will be allocated to the tube-side and the *n*-butanol to the shell-side. Physical property data for the fluids are given in Table 15.16.

Table 15.16 Physical property data for *n*-butanol and water.

	<i>n</i> -Butanol	Water
Density ($\text{kg}\cdot\text{m}^{-3}$)	712	993
Heat capacity ($\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$)	3200	4190
Viscosity ($\text{N}\cdot\text{s}\cdot\text{m}^{-2}$)	4.0×10^{-4}	8.0×10^{-4}
Thermal conductivity ($\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$)	0.127	0.616

Assume that the fouling coefficient for *n*-butanol is $10,000\text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$ and that for cooling water is $3,000\text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$. Steel tubes are to be used and the following assumptions can be made regarding the heat exchanger

geometry:

$$d_O = 20\text{ mm}$$

$$d_I = 16\text{ mm}$$

$$p_T = 1.25 d_O$$

$$p_C = 0.866(\text{triangular pitch})$$

$$B_C = 0.25$$

$$L/D_S = 7$$

Estimate the overall heat transfer coefficient, heat transfer area and pressure drops for the tube-side:

- a. for a fluid velocity of $1\text{ m}\cdot\text{s}^{-1}$ on both tube-side and the shell-side
- b. for a tube-side and shell-side pressure drop of 0.6 bar. Assume the shell-side is characterized by:

$$F_{hn} = F_{hw} = 1, F_{hb} = F_{hL} = 0.8, F_{pb} = 0.8 \text{ and } F_{PL} = 0.5$$

10. Table 15.17 gives estimates of the film transfer coefficients of an existing shell-and-tube heat exchanger, assuming the tube-wall resistance to be negligible.

Table 15.17 Estimated film transfer coefficients for an existing heat exchanger.

	Tube-side	Shell-side
Film transfer coefficient ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$)	500	750
Fouling coefficient ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$)	1000	2000

The flowrate on the tube-side is to be increased by 20% with the shell-side flowrate fixed.

- a. Estimate the change in the tube-side film transfer coefficient.
- b. The tube-side pressure drop before the increase is 0.7 bar. Estimate the change in the tube-side pressure drop.
- c. The data for an existing heat exchanger are given in Table 15.18.

Table 15.18 Data for existing heat exchanger.

	Tube-side	Shell-side
Flowrate ($\text{kg}\cdot\text{s}^{-1}$)	7.7	11.0
Initial temperature ($^\circ\text{C}$)	90	30
Final temperature ($^\circ\text{C}$)	40	50
Heat capacity ($\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$)	2400	4180

Estimate the change in the overall heat transfer coefficient and the new outlet temperatures for a 20% increase in the tube-side flow.

11. A condenser is required to condense a flowrate of $7\text{ kg}\cdot\text{s}^{-1}$ of isopropanol. The condensation takes place isothermally at 83°C without subcooling of the condensate. The cooling is provided by cooling water between 25 and 35°C . The condenser can be assumed to be steel with 20 mm tubes

with 2 mm wall thickness. The tube pitch can be assumed to be $1.25d_o$ with a square configuration. The length to shell diameter can be assumed to be 5. The physical property data are given in Table 15.19.

Table 15.19 Physical property data for isopropanol and water.

Property	Isopropanol (83°C)	Water (30°C)
Density ($\text{kg}\cdot\text{m}^{-3}$)	732	996
Liquid heat capacity ($\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$)	3370	4180
Viscosity ($\text{N}\cdot\text{s}\cdot\text{m}^{-2}$)	0.502×10^{-3}	0.797×10^{-3}
Thermal conductivity ($\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$)	0.131	0.618
Heat of vaporization ($\text{J}\cdot\text{kg}^{-1}$)	678,000	–

Assume the fouling coefficients to be $10,000 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$ and $5000 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$ for isopropanol and cooling water respectively. For an assumed cooling water velocity of $1 \text{ m}\cdot\text{s}^{-1}$, estimate the heat transfer area for:

- a horizontal condenser with shell-side condensation.
 - a vertical condenser with tube-side condensation.
- For the case of vertical tube-side condensation from Exercise 11, the condensate is to be subcooled to 45°C . By dividing the condenser into two zones for condensation and subcooling, estimate the heat transfer area.
 - A reboiler of a distillation column is required to supply $10 \text{ kg}\cdot\text{s}^{-1}$ of toluene vapor. The column operating pressure at the bottom of the column is 1.6 bar. At this pressure, the toluene vaporizes at 127°C and can be assumed to be isothermal. Steam at 160°C is to be used for the vaporization. The latent heat of vaporization of toluene is $344,000 \text{ J}\cdot\text{kg}^{-1}$, the critical pressure is 40.5 bar and critical temperature is 594 K. Steel tubes with 30 mm outside diameter, 2 mm wall thickness and length 3.95 m are to be used. The film coefficient (including fouling) for the condensing steam can be assumed to be $5700 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$. Estimate the heat transfer area for:
 - kettle reboiler.
 - vertical thermosyphon reboiler.
 - The purge gas from a petrochemical process is at 25°C and contains a mole fraction of methane of 0.6, the balance being hydrogen. This purge gas is to be burnt in a furnace to provide heat to a process with a cold stream pinch temperature of 150°C ($\Delta T_{\min} = 50^\circ\text{C}$). Ambient temperature is 10°C .
 - Calculate the theoretical flame temperature if 15% excess air is used in the combustion. Standard heats of combustion are given in Table 15.11 and mean molar heat capacities in Table 15.15.
 - Calculate the furnace efficiency.

- Suggest ways in which the furnace efficiency could be improved.

REFERENCES

- Kern DQ (1950) *Process Heat Transfer*, McGraw-Hill.
- Hewitt GF (1992) *Handbook of Heat Exchangers Design*, Begell House Inc.
- Hewitt GF, Shires GL and Bott TR (1994) *Process Heat Transfer*, CRC Press Inc.
- Sinnott RK (1996) *Chemical Engineering, Volume 6 Chemical Engineering Design*, Butterworth Heinemann.
- Nie X-R (1998) *Optimisation Strategies for Heat Exchanger Network Design Considering Pressure Drop Aspects*, PhD Thesis, UMIST, UK.
- Bowman RA, Mueller AC and Nagle WM (1940) Mean Temperature Differences in Design, *Trans ASME*, **62**: 283.
- Ahmad S, Linnhoff B and Smith R (1988) Design of Multi-pass Heat Exchangers: an Alternative Approach, *Trans ASME J Heat Transfer*, **110**: 304.
- Ahmad S, Linnhoff B and Smith R (1990) Cost Optimum Heat Exchanger Networks II Targets and Design for Detailed Capital Cost Models, *Comp Chem Eng*, **7**: 751.
- Zhu XX, Zangir M and Klemes J (2000) Heat Transfer Enhancement for Heat Exchanger Network Retrofit, *Heat Transfer Eng*, **21**: 7.
- Polley GT, Reyes Athie CM and Gough M (1992) Use of Heat Transfer Enhancement in Process Integration, *Heat Recovery Syst CHP*, **12**: 191.
- Nusselt W (1916) Die Oberflächenkondensation des Wasserdampfes, *Z Ver Deut Ing*, **60**: 541, 569.
- Rosenow WM (1956) Heat Transfer and Temperature Distribution in Laminar Film Condensation, *Trans ASME*, **78**: 1645.
- Rosenow WM, Webber JH and Ling AT (1956) Effect of Velocity on Laminar and Turbulent Film Condensation, *Trans ASME*, **78**: 1645.
- Mueller AC in Hewitt GF (1992) *Handbook of Heat Exchanger Design*, Begell House Inc.
- Palen JW in Hewitt GF (1992) *Handbook of Heat Exchanger Design*, Begell House Inc.
- Mostinski IL (1963) Calculation of Boiling Heat Transfer Coefficients, Based on the Law of Corresponding States, *Br Chem Eng*, **8**: 580.
- Frank O and Prickett RD (1973) Design of Vertical Thermosyphon Reboilers, *Chem Eng*, **3**: 107.
- Hougen OA, Watson KM and Ragatz RA (1954) *Chemical Process Principles Part I Material and Energy Balances*, 2nd Edition, John Wiley.

16 Heat Exchanger Networks II – Energy Targets

The design philosophy started at the heart of the onion with the reactor and moved out to the next layer, the separation and recycle system (Figure 1.7). Acceptance of the major processing steps (reactors, separators and recycles) in the inner two layers of the onion fixes the material and energy balance. Thus, the heating and cooling duties for the next layer of the onion, the heat recovery system, are known. However, completing the design of the heat exchanger network is not necessary in order to assess the completed design. *Targets* can be set for the heat exchanger network to assess the performance of the complete process design without actually having to carry out the network design. These targets allow both energy and capital cost for the heat exchanger network to be assessed. Moreover, the targets allow the designer to suggest process changes for the reactor and separation and recycle systems to improve the targets for energy and capital cost of the heat exchanger network.

Using targets for the heat exchanger network, rather than designs, allows many design options for the overall process to be screened quickly and conveniently. Screening many design options by completed designs is usually simply not practical in terms of the time and effort required. First consider the details of how to set energy targets. Capital cost targets will be considered in the next chapter. In later chapters, energy targets will be used to suggest design improvements to the reaction, separation and recycle systems.

16.1 COMPOSITE CURVES

The analysis of the heat exchanger network first identifies sources of heat (termed *hot streams*) and sinks (termed *cold streams*) from the material and energy balance. Consider first a very simple problem with just one hot stream (heat source) and one cold stream (heat sink). The initial temperature (termed *supply temperature*), final temperature (termed *target temperature*) and enthalpy change of both streams are given in Table 16.1.

Steam is available at 180°C and cooling water at 20°C. Clearly, it is possible to heat the cold stream using steam and cool the hot stream, in Table 16.1, using cooling water. However, this would incur excessive energy cost. It is also incompatible with the goals of sustainable industrial activity, which call for use of the minimum energy consumption. Instead, it is preferable to try to

Table 16.1 Two-stream heat recovery problem.

Stream	Type	Supply temperature T_S (°C)	Target temperature T_T (°C)	ΔH (MW)
1	Cold	40	110	14
2	Hot	160	40	–12

recover the heat between process streams, if this is possible. The scope for heat recovery can be determined by plotting both streams from Table 16.1 on temperature-enthalpy axes. For feasible heat exchange between the two streams, the hot stream must at all points be hotter than the cold stream. Figure 16.1a shows the temperature-enthalpy plot for this problem with a minimum temperature difference (ΔT_{min}) of 10°C. The region of overlap between the two streams in Figure 16.1a determines the amount of heat recovery possible (for $\Delta T_{min} = 10^\circ\text{C}$). For this problem, the heat recovery (Q_{REC}) is 11 MW. The part of the cold stream that extends beyond the start of the hot stream in Figure 16.1a cannot be heated by recovery and requires steam. This is the minimum hot utility or *energy target* (Q_{Hmin}), which for this problem is 3 MW. The part of the hot stream that extends beyond the start of the cold stream in Figure 16.1a cannot be cooled by heat recovery and requires cooling water. This is the minimum cold utility (Q_{Cmin}), which for this problem is 1 MW. Also shown at the bottom of Figure 16.1a is the arrangement of heat exchangers corresponding with the temperature-enthalpy plot.

The temperatures or enthalpy change for the streams (and hence their slope) cannot be changed, but the relative position of the two streams can be changed by moving them horizontally relative to each other. This is possible since the reference enthalpy for the hot stream can be changed independently from the reference enthalpy for the cold stream. Figure 16.1b shows the same two streams moved to a different relative position such that ΔT_{min} is now 20°C. The amount of overlap between the streams is reduced (and hence heat recovery is reduced) to 10 MW. A greater amount of the cold stream now extends beyond the start of the hot stream, and hence the amount of steam is increased to 4 MW. Also, more of the hot stream extends beyond the start of the cold stream, increasing the cooling water demand to 2 MW. Thus, the approach of plotting a hot and a cold stream on the same temperature-enthalpy axes can determine hot and cold utility for a given value of ΔT_{min} .

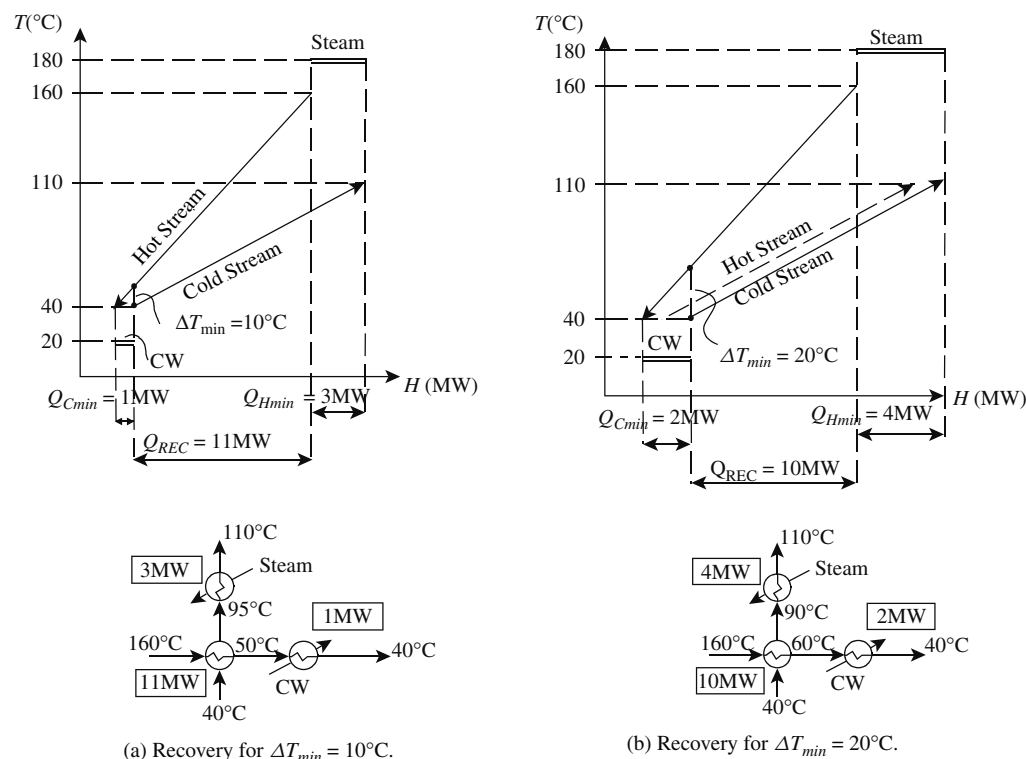


Figure 16.1 A simple heat recovery problem with one hot stream and one cold stream.

The importance of ΔT_{min} is that it sets the relative location of the hot and cold streams in this two-stream problem, and therefore the amount of heat recovery. Setting the value of ΔT_{min} or Q_{Hmin} or Q_{Cmin} sets the relative location and the amount of heat recovery.

Consider now the extension of this approach to several hot streams and cold streams. Figure 16.2 shows a simple flowsheet. Flowrates, temperatures and heat duties for each stream are shown. Two of the streams in Figure 16.2

are sources of heat (hot streams) and two are sinks for heat (cold streams). Assuming that the heat capacities are constant, the data for the hot and cold streams can be extracted as given in Table 16.2. Note that the heat capacities (CP) are total heat capacities, being the product of mass flowrate and specific heat capacity ($CP = m \cdot C_p$). Had the heat capacities varied significantly, the nonlinear temperature-enthalpy behavior could have been represented by a series of linear segments (see Chapter 19).

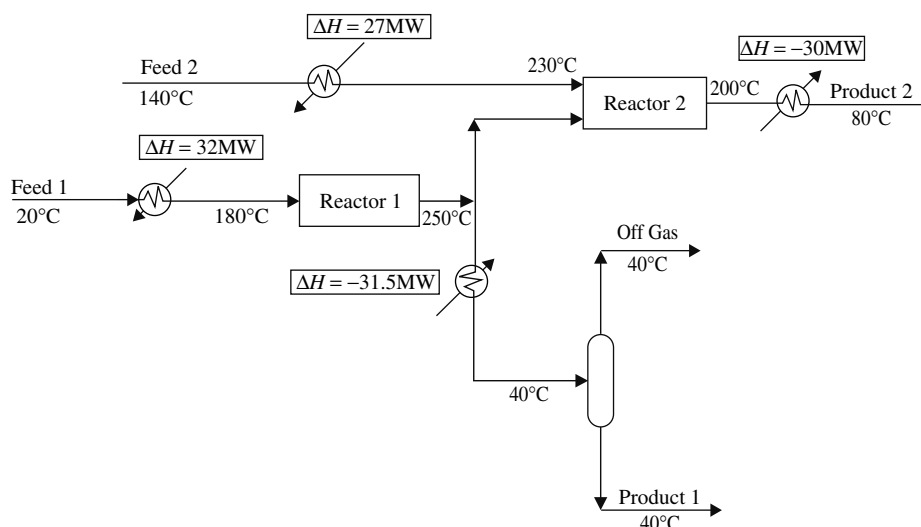


Figure 16.2 A simple flowsheet with two hot streams and two cold streams.

Table 16.2 Heat exchange stream data for the flowsheet Figure 16.2.

Stream	Type	Supply temperature T_S (°C)	Target temperature T_T (°C)	ΔH (MW)	Heat capacity flowrate CP (MW·K ⁻¹)
1. Reactor 1 feed	Cold	20	180	32.0	0.2
2. Reactor 1 product	Hot	250	40	-31.5	0.15
3. Reactor 2 feed	Cold	140	230	27.0	0.3
4. Reactor 2 product	Hot	200	80	-30.0	0.25

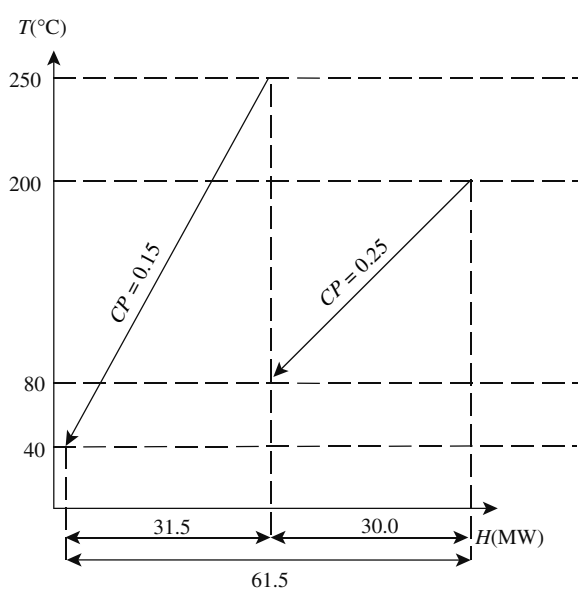
Instead of dealing with individual streams as given in Table 16.1, an overview of the process is needed. Figure 16.3a shows the two hot streams individually on temperature-enthalpy axes. How these hot streams behave overall can be quantified by combining them in the given temperature ranges^{1,2,3}. The temperature ranges in question are defined by changes in the overall rate of change of enthalpy with temperature. If heat capacities are constant, then changes will occur only when streams start or finish. Thus, in Figure 16.3, the temperature axis is divided into ranges defined by the supply and target temperatures of the streams.

Within each temperature range, the streams are combined to produce a composite hot stream. This composite hot stream has a CP in any temperature range that is the sum of the individual streams. Also, in any temperature range, the enthalpy change of the composite stream is the sum of the enthalpy changes of the individual streams. Figure 16.3b

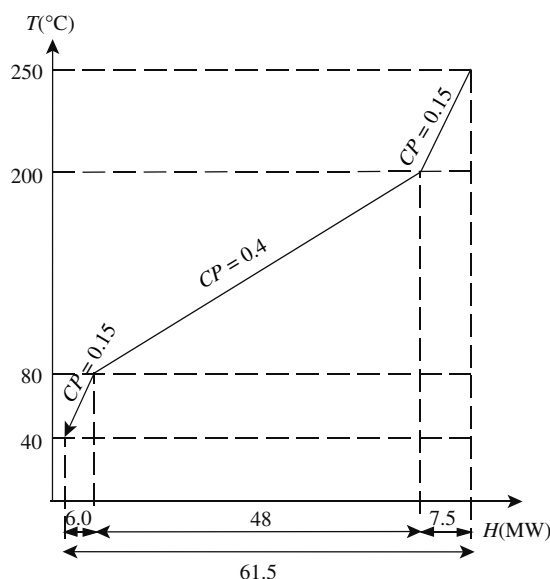
shows the *composite curve* of the hot streams^{1,2,3}. The composite hot stream is a single stream that is equivalent to the individual hot streams in terms of temperature and enthalpy. Similarly, the composite curve of the cold streams for the problem can be produced, as shown in Figure 16.4. Again, the composite cold stream is a single stream that is equivalent to the individual cold streams in terms of temperature and enthalpy.

The composite hot and cold curves can now be plotted on the same axes, as in Figure 16.5. Plotting the composite hot and cold curves is analogous to plotting the single hot and cold streams in Figure 16.1. The composite curves in Figure 16.5a are set to have a minimum temperature difference (ΔT_{min}) of 10°C. Where the curves overlap in Figure 16.5a, heat can be recovered vertically from the hot streams that comprise the hot composite curve into the cold streams that comprise the cold composite curve. The way in which the composite curves are constructed (i.e. monotonically decreasing hot composite curve and monotonically increasing cold composite curve) allows maximum overlap between the curves and hence maximum heat recovery. Maximizing the energy recovery thereby minimizes the external requirements for heating and cooling duties and minimizes the energy consumption. In this problem, for $\Delta T_{min} = 10^\circ\text{C}$, the maximum heat recovery (Q_{REC}) is 51.5 MW.

Where the cold composite curve extends beyond the start of the hot composite curve in Figure 16.5a, heat recovery is not possible, and the cold composite must be supplied with external hot utility such as steam. This represents the target for hot utility (Q_{Hmin}). For this problem, with $\Delta T_{min} = 10^\circ\text{C}$, $Q_{Hmin} = 7.5$ MW. Where the hot composite curve



(a) The hot streams plotted separately.



(b) The composite hot stream.

Figure 16.3 The hot streams can be combined to obtain a composite stream.

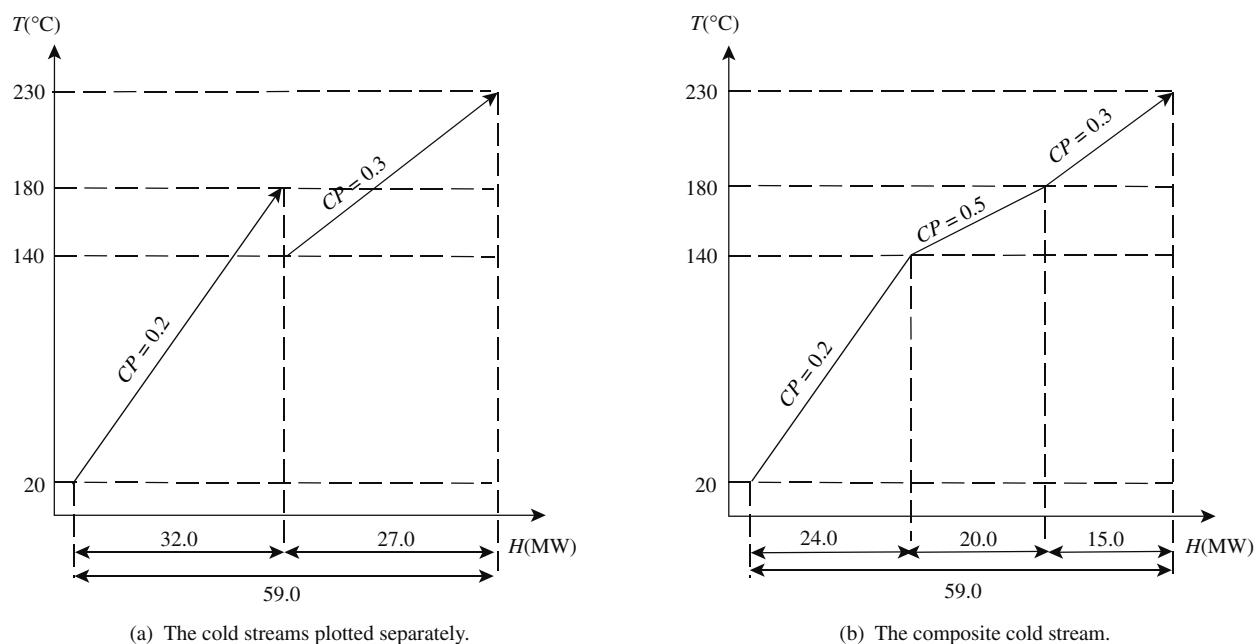


Figure 16.4 The cold streams can be combined to obtain a composite cold stream.

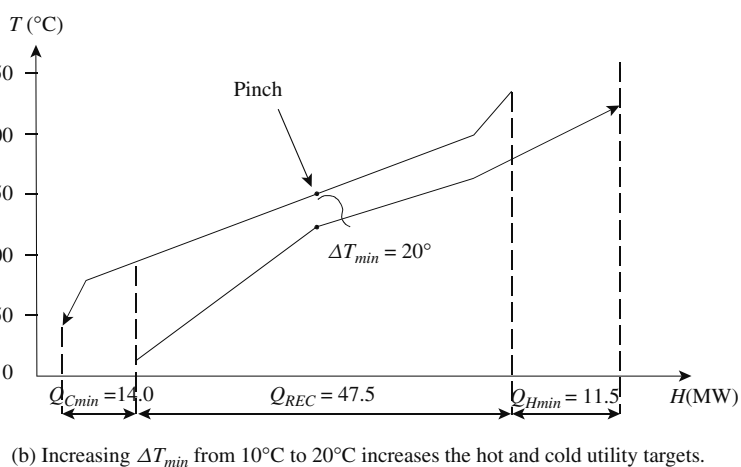
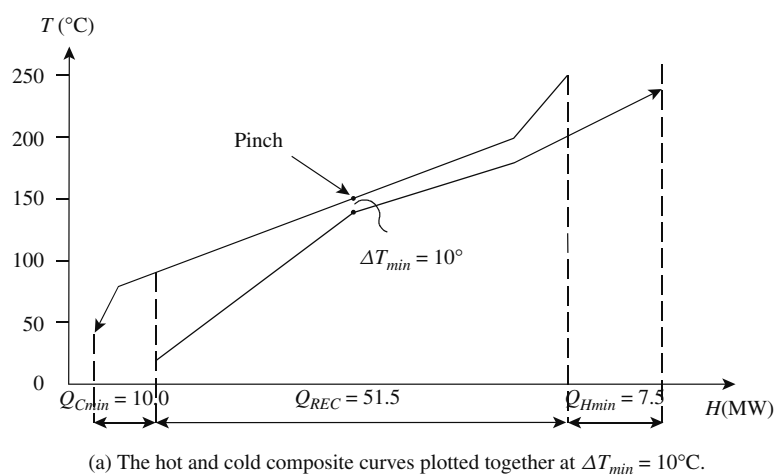


Figure 16.5 Plotting the hot and cold composite curves together allows the targets for hot and cold utility to be obtained.

extends beyond the start of the cold composite curve in Figure 16.5a, heat recovery is again not possible and the hot composite curve must be supplied with external cold utility such as cooling water. This represents the target for cold utility (Q_{Cmin}). For this problem, setting $\Delta T_{min} = 10^\circ\text{C}$ gives $Q_{Cmin} = 10.0$ MW.

Specifying the hot utility or cold utility heat duty or ΔT_{min} fixes the relative position of the two curves. As with the simple problem in Figure 16.1, the relative position of the two curves is a degree of freedom⁴. Again, the relative position of the two curves can be changed by moving them horizontally relative to each other. Clearly, to consider heat recovery from hot streams into cold streams, the hot composite curve must be in a position such that it is always above the cold composite curve for feasible heat transfer. Thereafter, the relative position of the curves can be chosen. Figure 16.5b shows the curves with $\Delta T_{min} = 20^\circ\text{C}$. The hot and cold utility targets are now increased to 11.5 MW and 14 MW respectively.

Figure 16.6 illustrates what happens to the cost of the system as the relative position of the composite curves is changed over a range of values of ΔT_{min} . When the curves just touch, there is no driving force for heat transfer at one point in the process, which would require infinite heat transfer area and hence infinite capital cost. As the energy target (and hence ΔT_{min} between the curves) is increased, the capital cost decreases. This results from increased temperature differences throughout the process, decreasing the heat transfer area. On the other hand, the energy cost increases as ΔT_{min} increases. There is a trade-off between energy and capital cost and an economic amount of energy recovery. Later, it will be shown how this trade-off can be carried out using energy and capital cost targets.

However, care should be taken not to ignore practical constraints when setting ΔT_{min} . To achieve a small ΔT_{min} in a design requires heat exchangers that exhibit pure countercurrent flow. With shell-and-tube heat exchangers

this is not possible, even if single-shell pass and single-tube pass designs are used, because the shell-side stream takes periodic cross-flow. Consequently, operating with a ΔT_{min} less than 10°C should be avoided, unless under special circumstances⁵. A smaller value of 5°C or less can be achieved with plate heat exchangers, and the value can go as low as 1 to 2°C with plate-fin designs⁵. It should be noted, however, that such constraints only apply to the exchangers that operate around the point of closest approach between the composite curves. Additional constraints apply if vaporization or condensation is occurring at the point of closest approach (see Chapter 15).

16.2 THE HEAT RECOVERY PINCH

As discussed above, the correct setting for the composite curves is determined by an economic trade-off between energy and capital, corresponding to an economic minimum temperature difference between the curves, ΔT_{min} . Accepting for the moment that the correct economic ΔT_{min} is known, this fixes the relative position of the composite curves and hence the energy target. The value of ΔT_{min} and its location between the composite curves have important implications for design, if the energy target is to be achieved in the design of a heat exchanger network. The ΔT_{min} is normally observed at only one point between the hot and the cold composite curves, called the *heat recovery pinch*^{3,6-8}. The pinch point has a special significance.

The trade-off between energy and capital in the composite curves suggests that, “on average”, individual exchangers should have a temperature difference no smaller than ΔT_{min} . A good initialization in heat exchanger network design is to assume that no individual heat exchanger has a temperature difference smaller than the ΔT_{min} between the composite curves.

With this rule in mind, divide the process at the pinch as shown in Figure 16.7a. Above the pinch (in temperature terms) the process is in heat balance with the minimum hot utility, Q_{Hmin} . Heat is received from hot utility and no heat is rejected. The process acts as a heat sink. Below the pinch (in temperature terms), the process is in heat balance with the minimum cold utility, Q_{Cmin} . No heat is received but heat is rejected to cold utility. The process acts as a heat source.

Consider now the possibility of transferring heat between these two systems. Figure 16.7b shows that it is possible to transfer heat from hot streams above the pinch to cold streams below it. The pinch temperature for hot streams for the problem is 150°C and for cold streams 140°C . Transfer of heat from above the pinch to below, as shown in Figure 16.7b, means transfer of heat from hot streams with a temperature of 150°C or greater into cold streams with a temperature of 140°C or less. This is clearly possible. By contrast, Figure 16.7c shows that heat transfer from hot streams below the pinch to cold streams

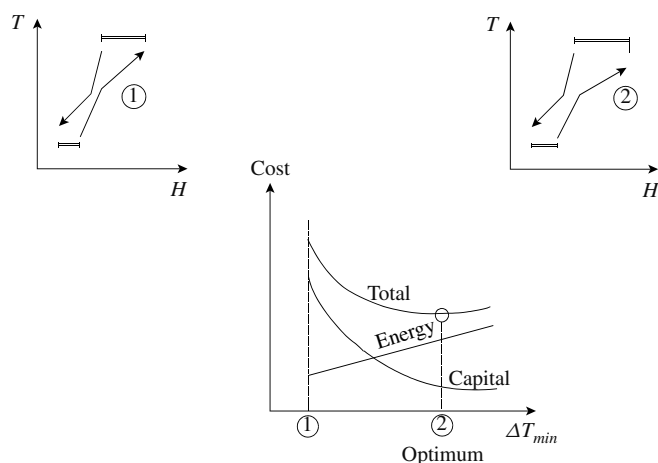


Figure 16.6 The correct setting for ΔT_{min} is fixed economic trade-offs.

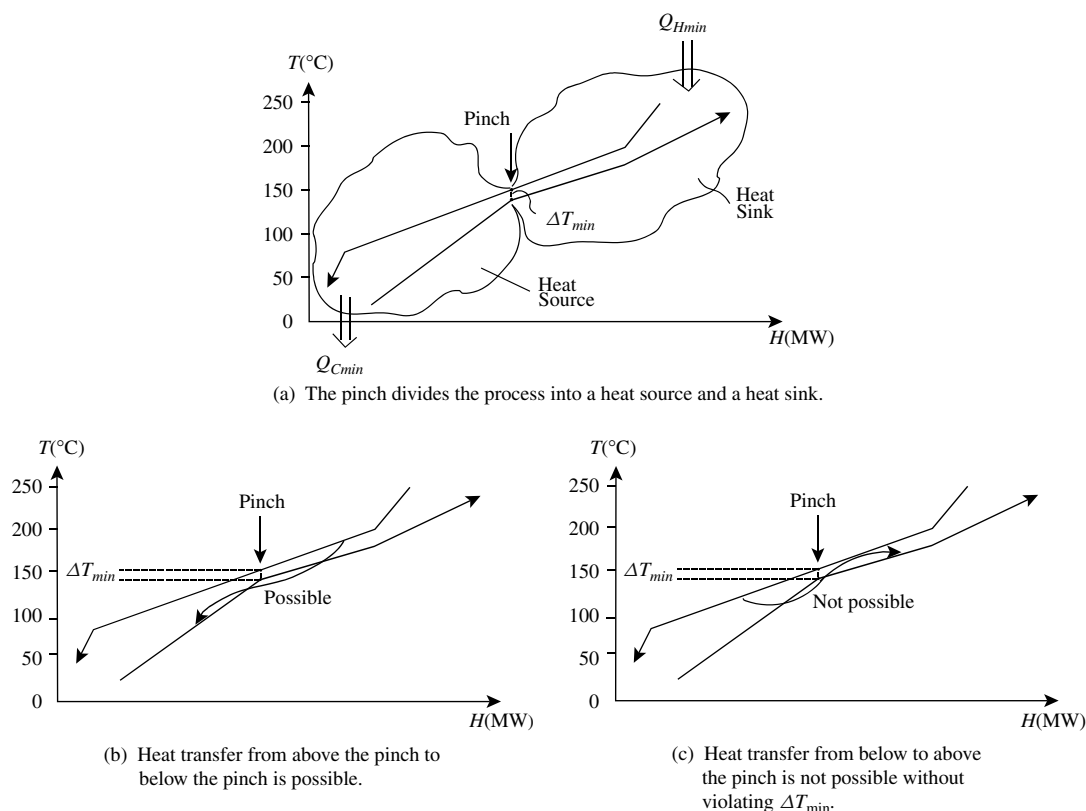


Figure 16.7 The composite curves set the energy target and the location of the pinch.

above is not possible. Such transfer requires heat being transferred from hot streams with a temperature of 150°C or less into cold streams with a temperature of 140°C or greater. This is clearly not possible (without violating the $\Delta T_{\min}$ constraint).

If an amount of heat XP is transferred from the system above the pinch to the system below the pinch, as in Figure 16.8a, this will create a deficit of heat XP above the pinch and an additional surplus of heat XP below the pinch. The only way this can be corrected is by importing an extra XP amount of heat from hot utility^{3,4} and exporting an extra XP amount of heat to cold utility^{3,4}.

Analogous effects are caused by the inappropriate use of utilities. Utilities are appropriate if they are necessary to satisfy the enthalpy imbalance in that part of the process. Above the pinch, hot utility (in this case, steam) is needed to satisfy the enthalpy imbalance. Figure 16.8b illustrates what happens if inappropriate use of utilities is made. If cooling to cold utility XP is used to cool hot streams above the pinch, this creates an enthalpy imbalance in the system above the pinch. To satisfy the enthalpy imbalance above the pinch, an import of $(Q_{H\min} + XP)$ heat from hot utility is required. Overall, $(Q_{C\min} + XP)$ of cold utility is used^{3,4}.

Another inappropriate use of utilities involves heating of some of the cold streams below the pinch by hot utility (steam in this case). Below the pinch, cold utility is needed to satisfy the enthalpy imbalance. Figure 16.8c illustrates

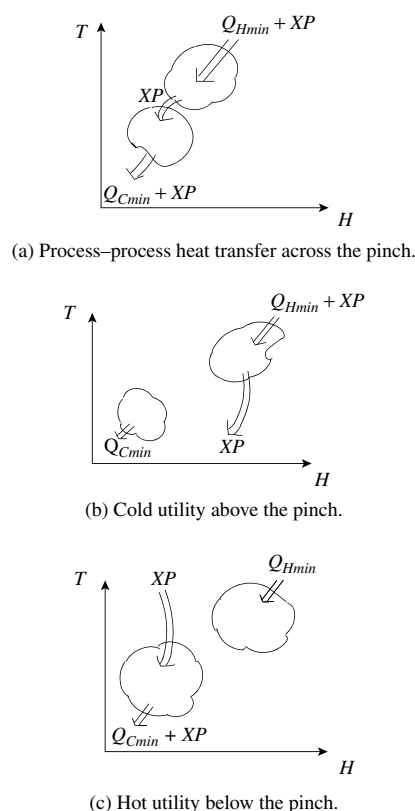
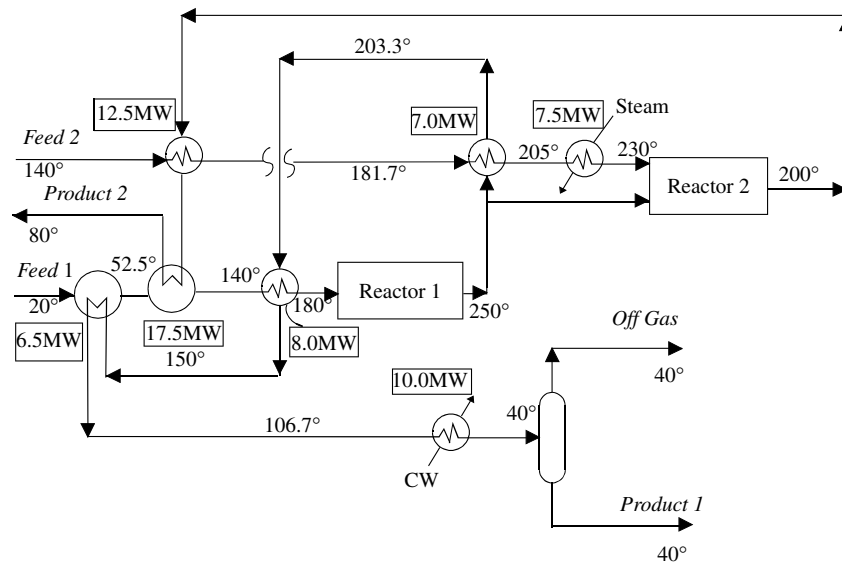
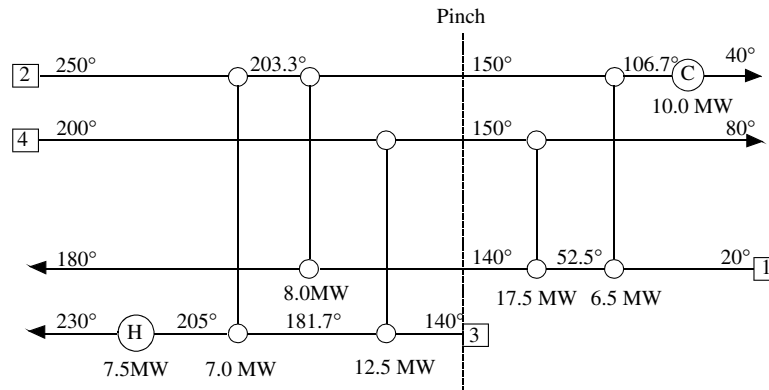


Figure 16.8 Three forms of cross pinch heat transfer.



(a)



(b)

Figure 16.9 A design that achieves the energy target.

what happens if an amount of heat XP from hot utility is used below the pinch. Q_{Hmin} must still be supplied above the pinch to satisfy the enthalpy imbalance above the pinch. Overall, $(Q_{Hmin} + XP)$ of steam is used and $(Q_{Hmin} + XP)$ of cooling water^{3,4}.

In other words, to achieve the energy target set by the composite curves, the designer must not transfer heat across the pinch by^{3,4}:

- Process-to-process heat transfer
- Inappropriate use of utilities

These rules are both necessary and sufficient to ensure that the target is achieved, providing that the initialization rule is adhered to that no individual heat exchanger should have a temperature difference smaller than ΔT_{min} .

Figure 16.9a shows a design corresponding to the flowsheet in Figure 16.2, which achieves the target of $Q_{Hmin} = 7.5$ MW and $Q_{Cmin} = 10$ MW for $\Delta T_{min} = 10^\circ\text{C}$.

Figure 16.9b shows an alternative representation of the flowsheet in Figure 16.9a, known as the *grid diagram*⁹. The grid diagram shows only heat transfer operations. Hot streams are at the top running left to right. Cold streams are at the bottom running right to left. A heat exchange match is represented by a vertical line joining two circles on the two streams being matched. An exchanger using hot utility is represented by a circle with an “H”. An exchanger using cold utility is represented by a circle with a “C”. The importance of the grid diagram is clear in Figure 16.9b, since the pinch, and how it divides the process into two parts, is easily accommodated. Dividing the process into two parts on a conventional diagram such as shown in Figure 16.9a is both difficult and extremely cumbersome.

Details of how the design in Figure 16.9 was developed are explained in Chapter 18. For now, simply take note that the targets set by the composite curves are achievable in design, providing that the pinch is recognized and there

is no transfer of heat across the pinch, either process-to-process or through inappropriate use of utilities. However, the insight of the pinch is needed to analyze some of the important decisions still to be made before the network design is addressed.

16.3 THRESHOLD PROBLEMS

Not all problems have a pinch to divide the process into two parts⁴. Consider the composite curves in Figure 16.10a. At this setting, both steam and cooling water are required. As the composite curves are moved closer together, both the steam and cooling water requirements decrease until the setting shown in Figure 16.10b. At this setting, the composite curves are in alignment at the hot end, indicating that there is no longer a demand for hot utility. Moving the curves closer together as shown in Figure 16.10c, decreases the cold utility demand at the cold end but opens up a demand for cold utility at the hot end corresponding with the decrease at the cold end. In other words, as the curves are moved closer together, beyond the setting in Figure 16.10b, the utility demand is constant. The setting shown in Figure 16.10b marks a threshold, and problems that exhibit this feature are known as *threshold problems*⁴. In some threshold problems, the hot utility requirement disappears as in Figure 16.10. In others, the cold utility disappears as shown in Figure 16.11.

Considering the capital-energy trade-off for threshold problems, there are two possible outcomes as shown in

Figure 16.12. Below the threshold ΔT_{min} , energy costs are constant, since utility demand is constant. Figure 16.12a shows a situation where the optimum occurs at the threshold ΔT_{min} . Figure 16.12b shows a situation where the optimum occurs above the threshold ΔT_{min} . The flat profile of energy costs below the threshold ΔT_{min} means that the optimum can never occur below the threshold value. It can only be at or above the threshold value.

In a situation, as shown in Figure 16.12a, with the optimum ΔT_{min} at the threshold, there is no pinch. On the other hand, in a situation as shown in Figure 16.12b with the optimum above the threshold value, there is a demand for both utilities and there is a pinch.

It is interesting to note that threshold problems are quite common in practice and although they do not have a process pinch, pinches are introduced into the design when multiple utilities are added. Figure 16.13a shows composite curves similar to the composite curves from Figure 16.10 but with two levels of cold utility used instead of one. In this case, the second cold utility is steam generation. The introduction of this second utility causes a pinch. This is known as a *utility pinch* since it is caused by the introduction of an additional utility⁴.

Figure 16.13b shows composite curves similar to those from Figure 16.11, but with two levels of steam used. Again, the introduction of a second steam level causes a utility pinch.

In design, the same rules must be obeyed around a utility pinch as a process pinch. Heat should not be transferred across it by process-to-process transfer and there should

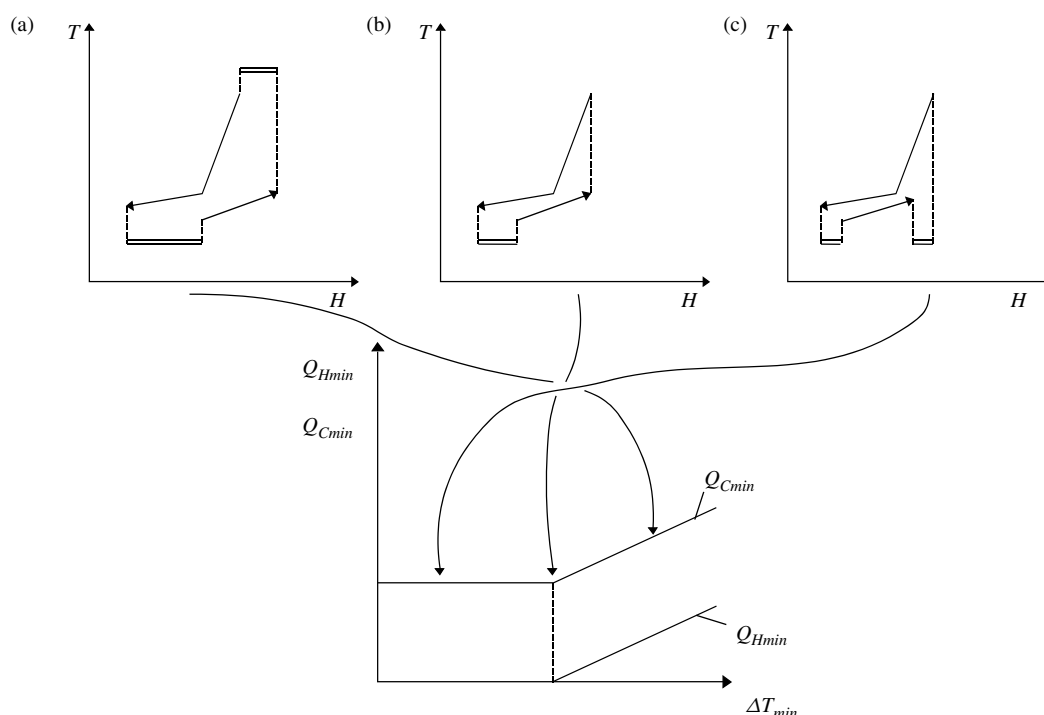


Figure 16.10 As ΔT_{min} is varied, some problems require only cold utility below a threshold value.

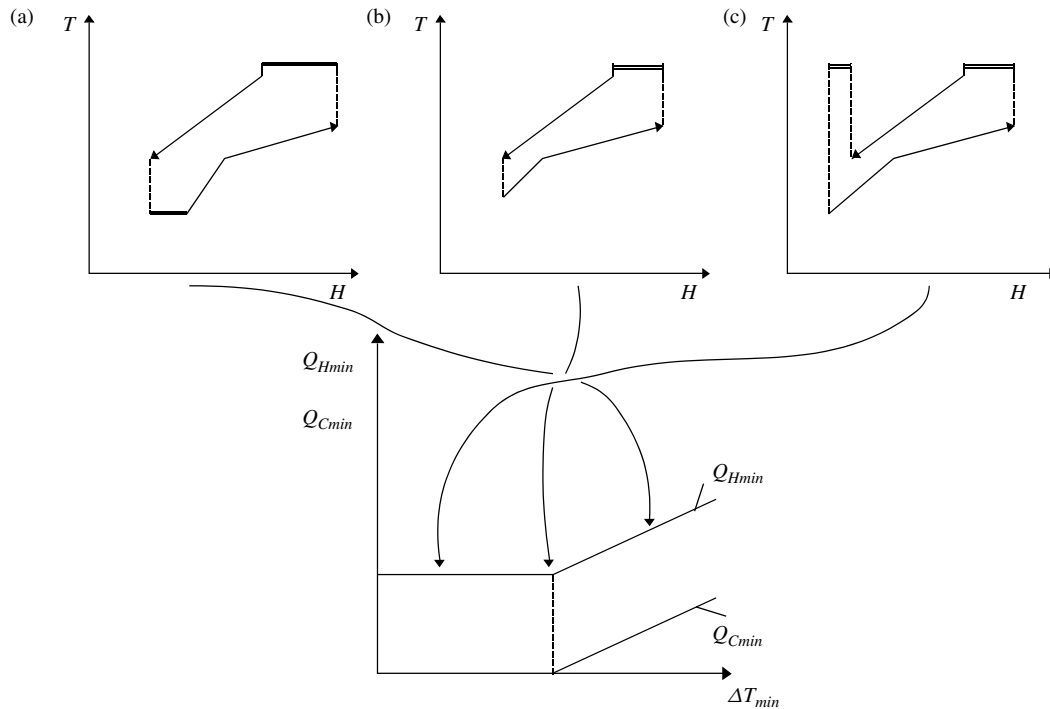
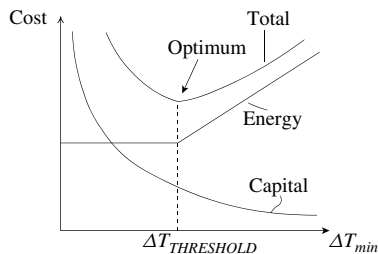
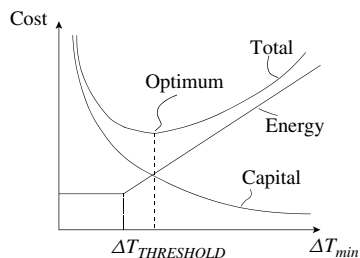


Figure 16.11 In some threshold problems, only hot utility is required below the threshold value of ΔT_{min} .



(a) The capital - energy trade-off can lead to an optimum at threshold ΔT_{min}



(b) The capital - energy trade-off can lead to an optimum above $\Delta T_{THRESHOLD}$

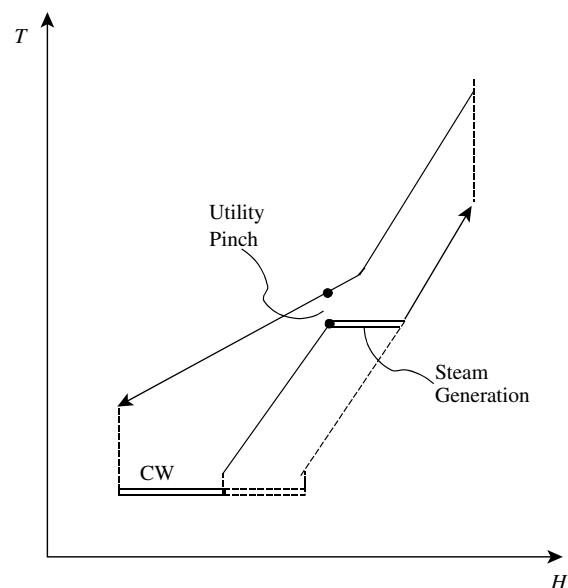
Figure 16.12 The optimum setting of the capital/energy trade off for threshold problems.

be no inappropriate use of utilities. In Figure 16.13a, this means that the only utility to be used above the utility pinch is steam generation and only cooling water below. In Figure 16.13b, this means that the only utility to be used above the utility pinch is high-pressure steam and only low-pressure steam below.

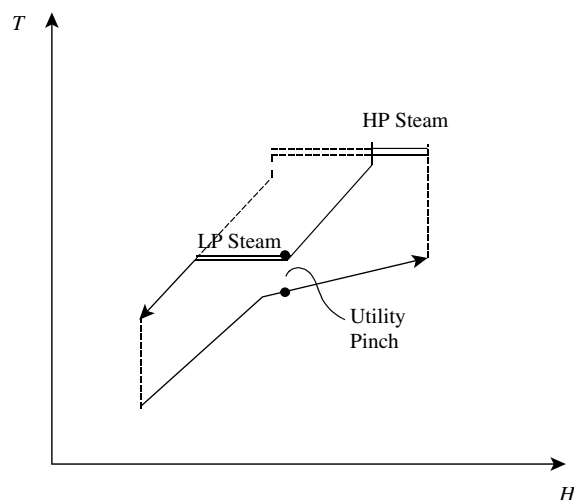
16.4 THE PROBLEM TABLE ALGORITHM

Although composite curves can be used to set energy targets, they are inconvenient since they are based on a graphical construction. A method of calculating energy targets directly without the necessity of graphical construction can be developed^{1,9}. The process is first divided into temperature intervals in the same way as was done for construction of the composite curves. Figure 16.14a shows that it is not possible to recover all of the heat in each temperature interval since temperature driving forces are not feasible throughout the interval. Some heat recovery is possible, but all of the heat cannot be recovered. The amount that can be recovered depends on the relative slopes of the two curves in the temperature interval. This problem can be overcome if, purely for the purposes of construction, the hot composite is shifted to be $\Delta T_{min}/2$ colder than it is in practice and that the cold composite is shifted to be $\Delta T_{min}/2$ hotter than it is in practice as shown in Figure 16.14b. The *shifted composite curves* now touch at the pinch. Carrying out a heat balance between the shifted composite curves within a *shifted temperature interval* shows that heat transfer is feasible throughout each shifted temperature interval, since hot streams in practice are actually $\Delta T_{min}/2$ hotter and cold streams $\Delta T_{min}/2$ colder. Within each shifted interval, the hot streams are in reality hotter than the cold streams by ΔT_{min} .

It is important to note that shifting the curves vertically does not alter the horizontal overlap between the curves. It therefore does not alter the amount by which the



(a)



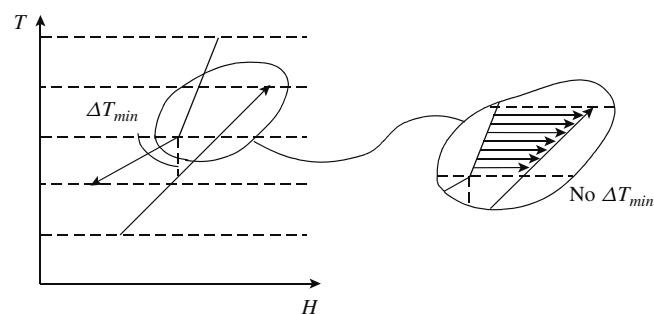
(b)

Figure 16.13 Threshold problems are turned into pinched problem when additional utilities are added.

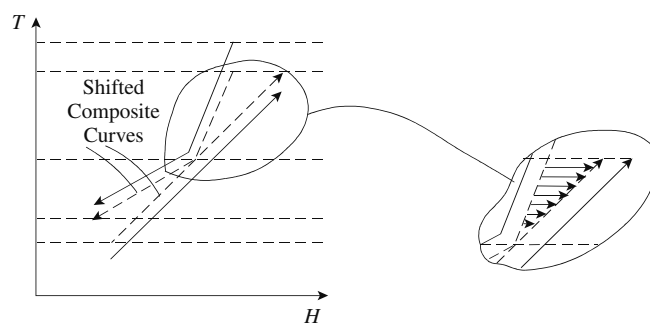
cold composite curve extends beyond the start of the hot composite curve at the hot end of the problem. Also, it does not alter the amount by which the hot composite curve extends beyond the start of the cold composite curve at the cold end. The shift simply removes the problem of ensuring temperature feasibility within temperature intervals.

This shifting technique can be used to develop a strategy to calculate the energy targets without having to construct composite curves^{1,9}:

1. Set up shifted temperature intervals from the stream supply and target temperatures by subtracting $\Delta T_{min}/2$ from the hot streams and adding $\Delta T_{min}/2$ to the cold streams (as in Figure 16.14b).



(a) Driving forces not feasible within each interval.



(b) Heat transfer within temperature intervals now feasible.

Figure 16.14 Shifting the composite curves in temperature allows complete heat recovery within temperature intervals.

2. In each shifted temperature interval, calculate a simple energy balance from:

$$\Delta H_i = \left[\sum_{\text{Cold streams}} CP_C - \sum_{\text{Hot streams}} CP_H \right] \Delta T_i \quad (16.1)$$

where ΔH_i is the heat balance for shifted temperature interval i and ΔT_i is the temperature difference across it. If the cold streams dominate the hot streams in a temperature interval, then the interval has a net deficit of heat, and ΔH is positive. If hot streams dominate cold streams, the interval has a net surplus of heat, and ΔH is negative. This is consistent with standard thermodynamic convention, for example, for an exothermic reaction, ΔH is negative. If no hot utility is used, this is equivalent to constructing the shifted composite curves shown in Figure 16.15a.

3. The overlap in the shifted curves as shown in Figure 16.15a means that heat transfer is infeasible. At some point, this overlap is a maximum. This maximum overlap is added as hot utility to correct the overlap. The shifted curves now touch at the pinch as shown in Figure 16.15b. Since the shifted curves just touch, the actual curves are separated by ΔT_{min} at this point, Figure 16.15b.

This basic approach can be developed into a formal algorithm known as the *problem table algorithm*⁹. The

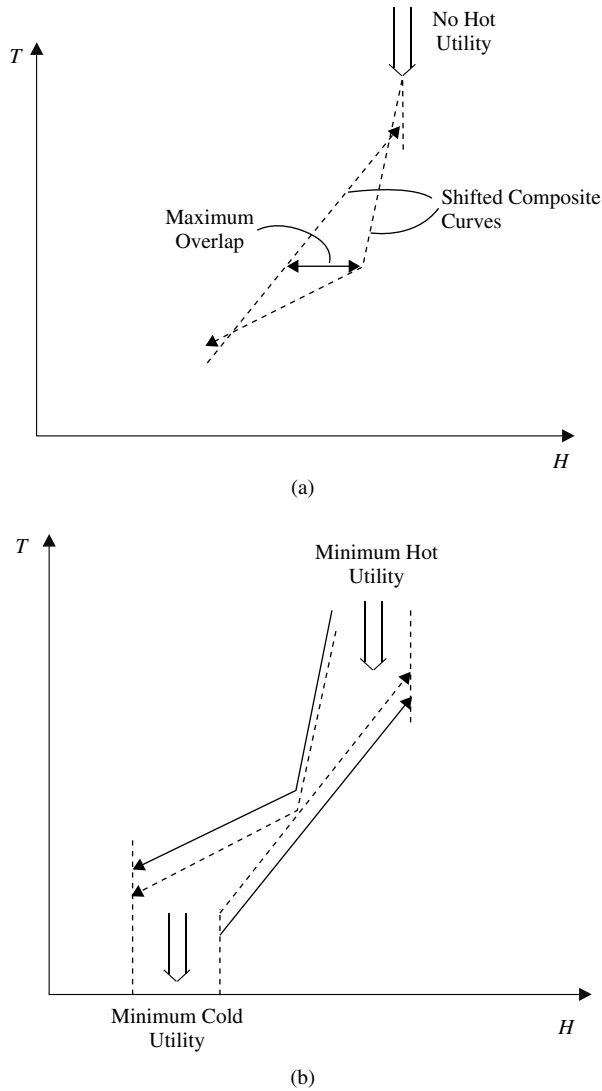


Figure 16.15 The utility target can be determined from the maximum overlap between the shifted composite curves.

algorithm will be explained using the data from Figure 16.2 for $\Delta T_{min} = 10^\circ\text{C}$.

First determine the shifted temperature intervals (T^*) from actual supply and target temperatures. Hot streams are shifted down in temperature by $\Delta T_{min}/2$ and cold streams up by $\Delta T_{min}/2$ as detailed in Table 16.3.

The stream population is shown in Figure 16.16 with a vertical temperature scale. The interval temperatures

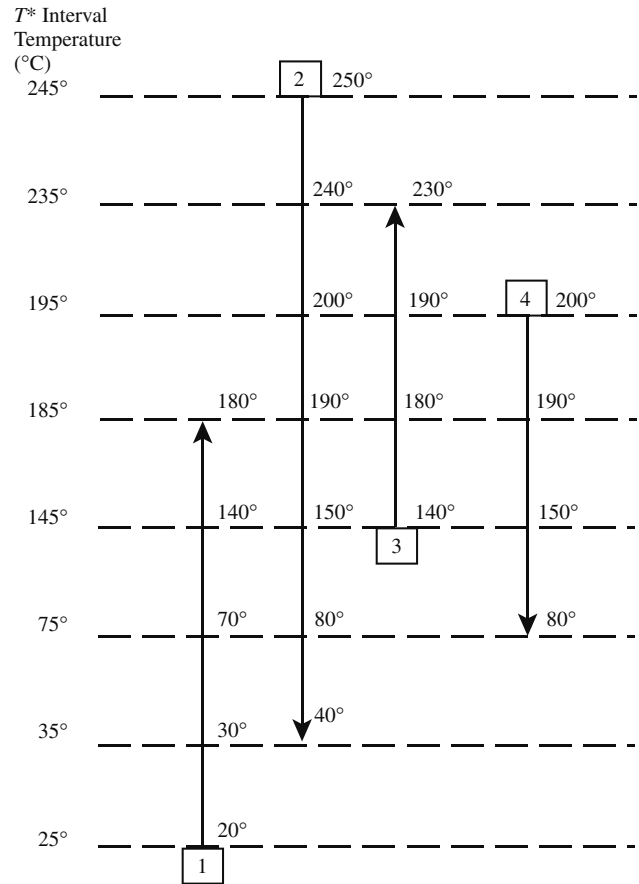


Figure 16.16 The stream population for the data from Figure 16.2.

Interval Temperature	Stream Population	$\Delta T_{INTERNAL}$ ($^\circ\text{C}$)	$\Sigma CP_C - \Sigma CP_H$ MW·K ⁻¹	$\Delta H_{INTERNAL}$ (MW)	Surplus/Deficit
245°	2				
235°		10	-0.15	-1.5	Surplus
195°	2, 4	40	0.15	6.0	Deficit
185°		10	-0.1	-1.0	Surplus
145°	2, 3, 4	40	0.1	4.0	Deficit
75°	1, 3, 4	70	-0.2	-14.0	Surplus
35°	1, 3, 4	40	0.05	2.0	Deficit
25°	1	10	0.2	2.0	Deficit

Figure 16.17 The temperature interval heat balances.

Table 16.3 Shifted temperatures for the data from Table 16.2.

Stream	Type	T_S	T_T	T_S^*	T_T^*
1	Cold	20	180	25	185
2	Hot	250	40	245	35
3	Cold	140	230	145	235
4	Hot	200	80	195	75

shown in Figure 16.16 are set to $\Delta T_{min}/2$ below hot stream temperatures and $\Delta T_{min}/2$ above cold stream temperatures.

Next, a heat balance is carried out within each shifted temperature interval according to Equation 16.1. The result is given in Figure 16.17, in which some of the shifted intervals are seen to have a surplus of heat and some have a

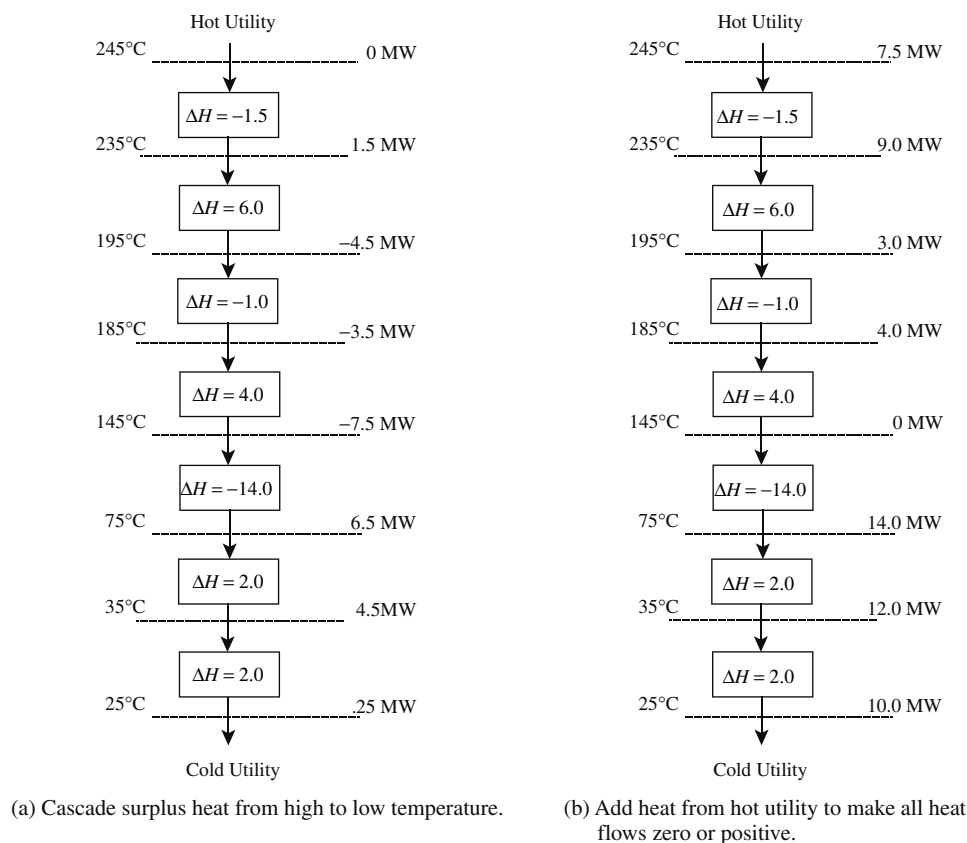


Figure 16.18 The problem table cascade.

deficit. The heat balance within each shifted interval allows maximum heat recovery within each interval. However, recovery must also be allowed between intervals.

Now, *cascade* any surplus heat down the temperature scale from interval to interval. This is possible because any excess heat available from the hot streams in an interval is hot enough to supply a deficit in the cold streams in the next interval down. Figure 16.18 shows the cascade for the problem. First, assume no heat is supplied to the first interval from hot utility, Figure 16.18a. The first interval has a surplus of 1.5 MW, which is cascaded to the next interval. This second interval has a deficit of 6 MW, which leaves the heat cascaded from this interval to be -4.5 MW. In the third interval, the process has a surplus of 1 MW, which leaves -3.5 MW, to be cascaded to the next interval, and so on.

Some of the heat flows in Figure 16.18a are negative, which is infeasible. Heat cannot be transferred up the temperature scale. To make the cascade feasible, sufficient heat must be added from hot utility to make the heat flows to be at least zero. The smallest amount of heat needed from hot utility is the largest negative heat flow from Figure 16.18a, that is 7.5 MW. In Figure 16.18b, 7.5 MW is added from hot utility to the first interval. This does not change the heat balance within each interval, but increases

all of the heat flows between intervals by 7.5 MW, giving one heat flow of just zero at an interval temperature of 145°C.

More than 7.5 MW could be added from hot utility to the first interval, but the objective is to find the minimum hot and cold utility, hence only the minimum is added. Thus from Figure 16.18b, $Q_{Hmin} = 7.5$ MW and $Q_{Cmin} = 10$ MW. This corresponds with the values obtained from the composite curves in Figure 16.5. One further important piece of information can be deduced from the cascade in Figure 16.18b. The point where the heat flow goes to zero at $T^* = 145^\circ\text{C}$ corresponds to the pinch. Thus, the actual hot and cold stream pinch temperatures are 150°C and 140°C respectively. Again, this agrees with the result from the composite curves in Figure 16.5.

The initial setting for the heat cascade in Figure 16.18a corresponds to the setting for the shifted composite curves in Figure 16.15a where there is an overlap. The setting of the heat cascade for zero or positive heat flows in Figure 16.18b corresponds to the shifted composite curve setting in Figure 16.15b.

The composite curves are useful in providing conceptual understanding of the process but the problem table algorithm is a more convenient calculation tool.

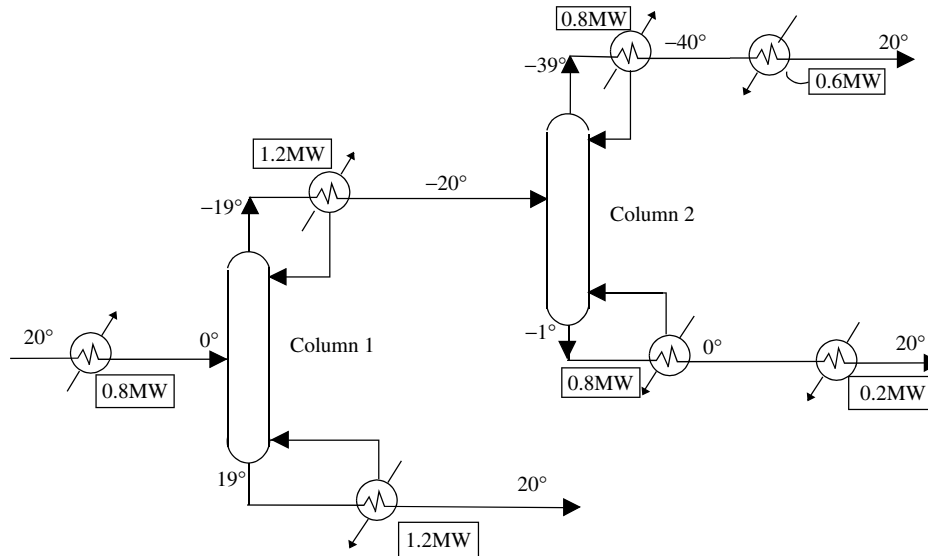


Figure 16.19 A low-temperature distillation process.

Example 16.1 The flowsheet for a low-temperature distillation process is shown in Figure 16.19. Calculate the minimum hot and cold utility requirements and the location of the pinch assuming $\Delta T_{min} = 5^\circ\text{C}$.

Solution First extract the stream data from the flowsheet. This is given in Table 16.4 below.

Next, calculate the shifted interval temperatures. Hot stream temperatures are shifted down by 2.5°C and cold stream temperatures shifted up by 2.5°C , as shown in Table 16.5.

Now carry out a heat balance within each shifted temperature interval as shown in Figure 16.20.

Finally, the heat cascade is shown in Figure 16.21. Figure 16.21a shows the cascade with zero hot utility. This leads to negative heat flows, the largest of which is -1.84 MW . Adding 1.84 MW from hot utility as shown in Figure 16.21b gives $Q_{Hmin} = 1.84\text{ MW}$, $Q_{Cmin} = 1.84\text{ MW}$, hot stream pinch temperature $= -19^\circ\text{C}$ and cold stream pinch temperature $= -24^\circ\text{C}$.

Table 16.4 Stream data for low-temperature distillation process.

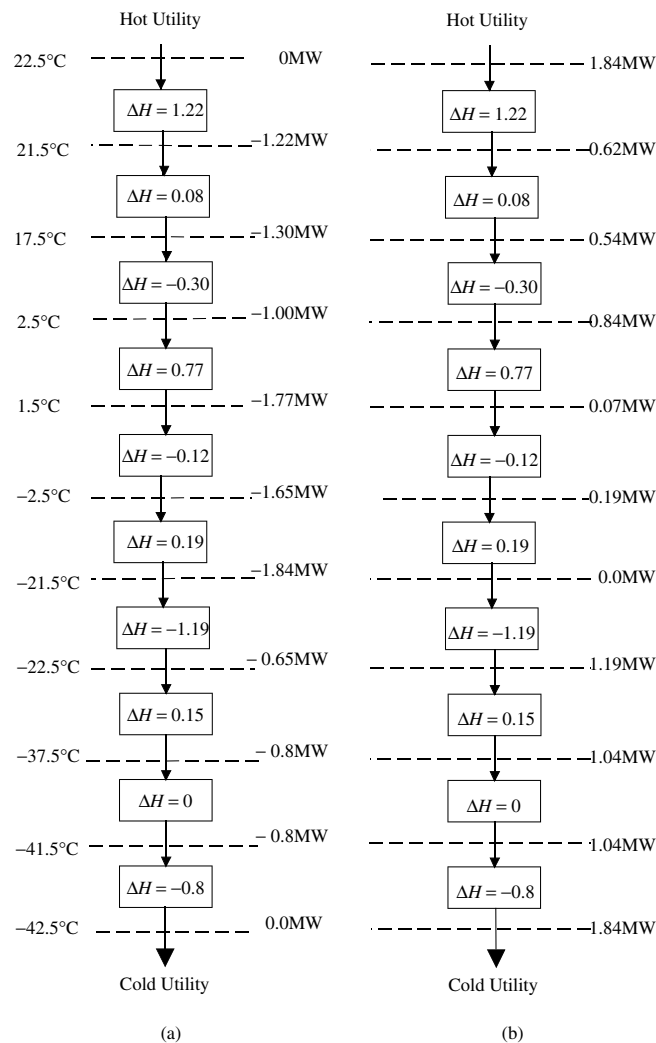
Stream	Type	Supply temperature $T_S(^{\circ}\text{C})$	Target temperature $T_T(^{\circ}\text{C})$	ΔH (MW)	CP (MW·K ⁻¹)
1. Feed to column 1	Hot	20	0	-0.8	0.04
2. Column 1 condenser	Hot	-19	-20	-1.2	1.2
3. Column 2 condenser	Hot	-39	-40	-0.8	0.8
4. Column 1 reboiler	Cold	19	20	1.2	1.2
5. Column 2 reboiler	Cold	-1	0	0.8	0.8
6. Column 2 bottoms	Cold	0	20	0.2	0.01
7. Column 2 overheads	Cold	-40	20	0.6	0.01

Interval Temperature	Stream Population	$\Delta T_{INTERVAL}$	ΣCP_C $-\Sigma CP_H$	$\Delta H_{INTERVAL}$	Surplus/ Deficit	
22.5°C						
21.5°C			1	1.22	1.22	Deficit
17.5°C			4	0.02	0.08	Deficit
2.5°C			15	-0.02	-0.30	Surplus
1.5°C			1	0.77	0.77	Deficit
-2.5°C			4	-0.03	-0.12	Surplus
-21.5°C			19	0.01	0.19	Deficit
-22.5°C			1	-1.19	-1.19	Surplus
-37.5°C			15	0.01	0.15	Deficit
-41.5°C			4	0	0	-----
-42.5°C			1	-0.8	-0.8	Surplus

Figure 16.20 Temperature interval heat balances for Example 16.1.

Table 16.5 Shifted temperatures for the data in Table 16.4.

Stream	Type	T_S	T_T	T_S^*	T_T^*
1	Hot	20	0	17.5	-2.5
2	Hot	-19	-20	-21.5	-22.5
3	Hot	-39	-40	-41.5	-42.5
4	Cold	19	20	21.5	22.5
5	Cold	-1	0	1.5	2.5
6	Cold	0	20	2.5	22.5
7	Cold	-40	20	-37.5	22.5

**Figure 16.21** The problem table cascade for Example 16.1.

16.5 NONGLOBAL MINIMUM TEMPERATURE DIFFERENCES

So far, it has been assumed that the minimum temperature difference for a heat exchanger network applies globally between all streams in the network. However, there are occasions when nonglobal minimum temperature differences might be required. For example, suppose a heat

exchanger network is servicing some streams that are liquid and some that are gaseous. For the liquid–liquid heat transfer matches, a value of perhaps $\Delta T_{min} = 10^\circ\text{C}$ is appropriate. But for the gas–gas matches, a larger temperature minimum temperature difference is required, say $\Delta T_{min} = 20^\circ\text{C}$. How can this be accommodated in the targeting?

When carrying out the problem table algorithm, the temperatures were shifted according to $\Delta T_{min}/2$ being added to the cold streams and subtracted from the hot streams. This value of $\Delta T_{min}/2$ can be considered to be a *contribution* to the overall ΔT_{min} between the hot and the cold streams. Rather than making the ΔT_{min} contribution equal for all streams, it could be made stream-specific:

$$T_{H,i}^* = T_{H,i} - \Delta T_{min,cont,i}$$

$$T_{C,j}^* = T_{C,j} + \Delta T_{min,cont,j}$$

where $T_{H,i}^*$, $T_{H,i}$ are the shifted and actual temperatures for Hot Stream i , $T_{C,j}^*$, $T_{C,j}$ are the shifted and actual temperatures for Cold Stream j , and $\Delta T_{min,cont,i}$ and $\Delta T_{min,cont,j}$ are the contributions to ΔT_{min} for Hot Stream i and Cold Stream j . Thus, for the above example, if the ΔT_{min} contribution for liquid streams is taken to be 5°C and for gas streams 10°C , then a liquid–liquid match would lead to $\Delta T_{min} = 10^\circ\text{C}$, a gas–gas match would lead to $\Delta T_{min} = 20^\circ\text{C}$ and a liquid–gas match would lead to $\Delta T_{min} = 15^\circ\text{C}$ ⁴. To include this in the problem table algorithm is straightforward. All that needs to be done is that the appropriate ΔT_{min} contribution is to be allocated to each stream and then that ΔT_{min} contribution is subtracted in the case of hot streams and added in the case of cold streams. This would lead to different interval temperatures compared with a global minimum temperature difference. The remainder of the problem table algorithm would be the same. Once the interval temperatures based on ΔT_{min} contributions have been established, the interval heat balances can be performed and the cascade set up in the same way as for a global ΔT_{min} .

From the point of view of the composite curves, the location of the pinch and the ΔT_{min} at the pinch would depend on which kind of streams were located in the region of the point of closest approach between the composite curves. If only liquid streams were present around the point of closest approach of the composite curves, then in the above example, $\Delta T_{min} = 10^\circ\text{C}$ will apply. If there were only gas streams in the region around the point of closest approach, then in the above example, $\Delta T_{min} = 20^\circ\text{C}$ would apply. If there was a mixture of liquid and gas streams at the point of closest approach, then $\Delta T_{min} = 15^\circ\text{C}$ would apply.

16.6 PROCESS CONSTRAINTS

So far it has been assumed that any hot stream could, in principle, be matched with any cold stream providing there is feasible temperature difference between them. Often,

practical constraints prevent this. For example, it might be the case that if two streams are matched in a heat exchanger and a leak develops, such that the two streams come into contact, this might produce an unacceptably hazardous situation. If this were the case, then no doubt a constraint would be imposed to prevent the two streams being matched. Another reason for a constraint might be that two streams are expected to be geographically very distant from each other, leading to unacceptably long pipe runs. Potential control and start-up problems might also call for constraints. There are many reasons why constraints might be imposed.

One common reason for imposing constraints results from *areas of integrity*¹⁰. A process is often normally designed to have logically identifiable sections or areas. An example might be the “reaction area” and “separation area” of the process. These areas might need to be kept separate for reasons such as start-up, shutdown, operational flexibility, safety, and so on. The areas are often made operationally independent by use of intermediate storage between the areas. Such independent areas are generally described as areas of integrity and impose constraints on the ability to transfer heat. Clearly, to maintain operational independence, two areas cannot be dependent on each other for heating and cooling by recovery.

The question now is, given that there are often constraints to deal with, how to evaluate the effect of these constraints on the system performance? The problem table algorithm cannot be used directly if constraints are imposed. However, often the effect of constraints on the energy performance can be evaluated using the problem table algorithm, together with a little common sense. The following example illustrates how¹⁰.

Example 16.2 A process is to be divided into two operationally independent areas of integrity, Area A and Area B. The stream data for the two areas are given in Table 16.6¹⁰.

Calculate the penalty in utility consumption to maintain the two areas of integrity for $\Delta T_{min} = 20^\circ\text{C}$.

Solution To identify the penalty, first calculate the utility consumption of the two areas separate from each other as shown in Figure 16.22a. Next, combine all of the streams from both areas and again calculate the utility consumption, Figure 16.22b. Figure 16.23a shows the problem table cascade for Area A, the cascade for Area B is shown in Figure 16.23b, and that for Areas A and B combined in Figure 16.23c.

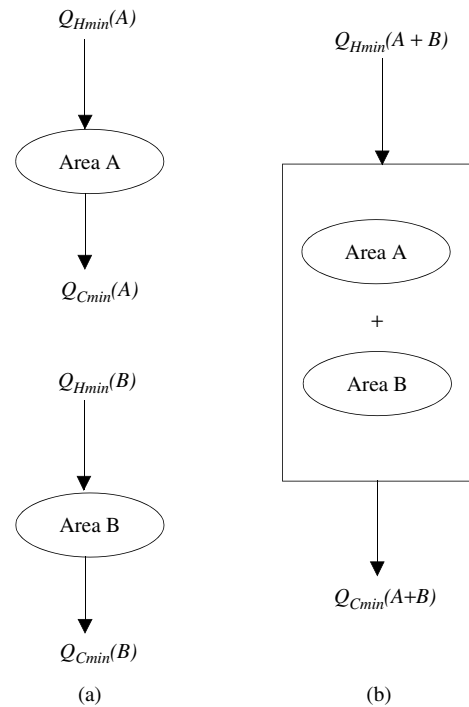


Figure 16.22 The areas of integrity can be targeted separately and then the combination targeted.

With Areas A and B separate, the total hot utility consumption is $(1400 + 0) = 1400$ kW and the total cold utility consumption is $(0 + 1350) = 1350$ kW. With Areas A and B combined, the total utility consumption is 950 kW of hot utility and 900 kW of cold utility.

The penalty for maintaining the areas of integrity is thus $(1400 - 950) = 450$ kW of hot utility and $(1350 - 900) = 450$ kW of cold utility.

Having quantified the penalty as a result of imposing constraints, the designer can exercise judgment as to whether it is acceptable or too expensive. If it is too expensive, there is a choice between two options:

- Reject the constraints and operate the process as a single system.
- Find a way to overcome the constraint. This is often possible by using a heat transfer fluid. The simplest option is via the existing utility system. For example, rather than have a direct match between two streams,

Table 16.6 Stream data for heat recovery between two areas of integrity.

Area A				Area B			
Stream	T_S (°C)	T_T (°C)	CP (kW·K ⁻¹)	Stream	T_S (°C)	T_T (°C)	CP (kW·K ⁻¹)
1	190	110	2.5	3	140	50	20.0
2	90	170	20.0	4	30	120	5.0

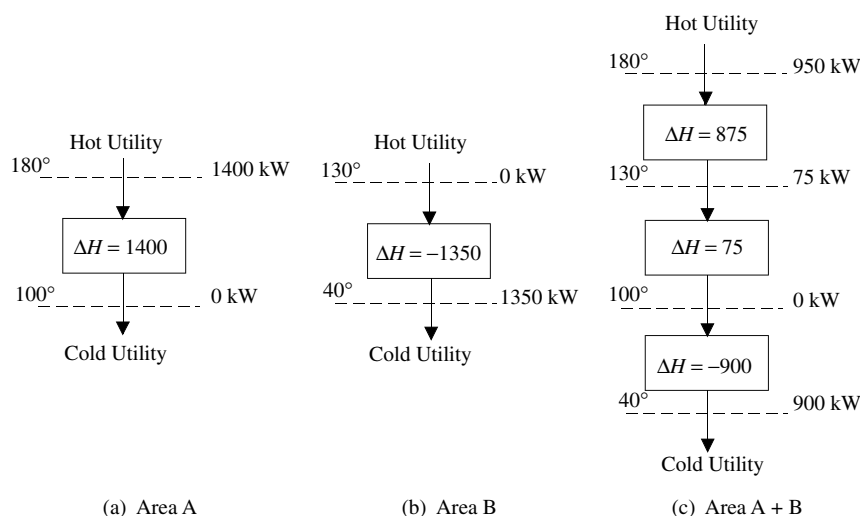


Figure 16.23 Problem table cascade for the separate and combined areas of integrity.

it might be possible for the heat source to generate steam to be fed into the steam mains and the heat sink to use steam from the same mains. The utility system then acts as a buffer between the heat sources and sinks. Another possibility might be to use a heat transfer fluid such as hot oil that circulates between the two streams being matched. To maintain operational independence, a standby heater and cooler supplied by utilities can be provided in the hot oil circuit, so that if either the heat source or sink is not operational, utilities could substitute heat recovery for short periods.

Many constraints can be evaluated by scoping out the problem with different boundaries. In Example 16.2, the sets of streams that were constrained to be separate were collected to be within each boundary for targeting. Comparing the targets for the streams within each boundary with that for all the streams put together allows the penalty of the constraint to be evaluated. The approach is more widely applicable than just areas of integrity. Whenever a stream, or set of streams, is to be maintained separate from any other set of streams, the same approach can be used. However, this approach of scoping out the problem with different boundaries has limitations in the evaluation of constraints. More complex constraints require linear programming to obtain the energy target^{11,12}.

16.7 UTILITY SELECTION

After maximizing heat recovery in the heat exchanger network, those heating and cooling duties not serviced by heat recovery must be provided by external utilities. The most common hot utility is steam. It is usually available at several levels. High-temperature heating duties require furnace flue gas or a hot oil circuit. Cold utilities might be refrigeration, cooling water, air-cooling, furnace

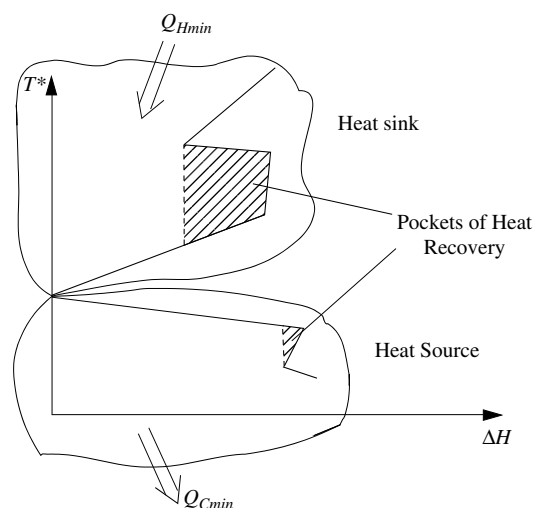


Figure 16.24 The grand composite curve shows the utility requirements both in enthalpy and temperature terms.

air preheating, boiler feedwater preheating or even steam generation if heat needs to be rejected at high temperatures.

Although the composite curves can be used to set energy targets, they are not a suitable tool for the selection of utilities. The *grand composite curve* is a more appropriate tool for understanding the interface between the process and the utility system^{4,13,14}. It will also be shown in later chapters to be a useful tool to study the interaction between heat-integrated reactors and separators and the rest of the process.

The grand composite curve is obtained by plotting the problem table cascade. A typical grand composite curve is shown in Figure 16.24. It shows the heat flow through the process against temperature. It should be noted that the temperature plotted here is *shifted temperature* (T^*) and not actual temperature. Hot streams are represented $\Delta T_{min}/2$ colder and cold streams $\Delta T_{min}/2$ hotter than they

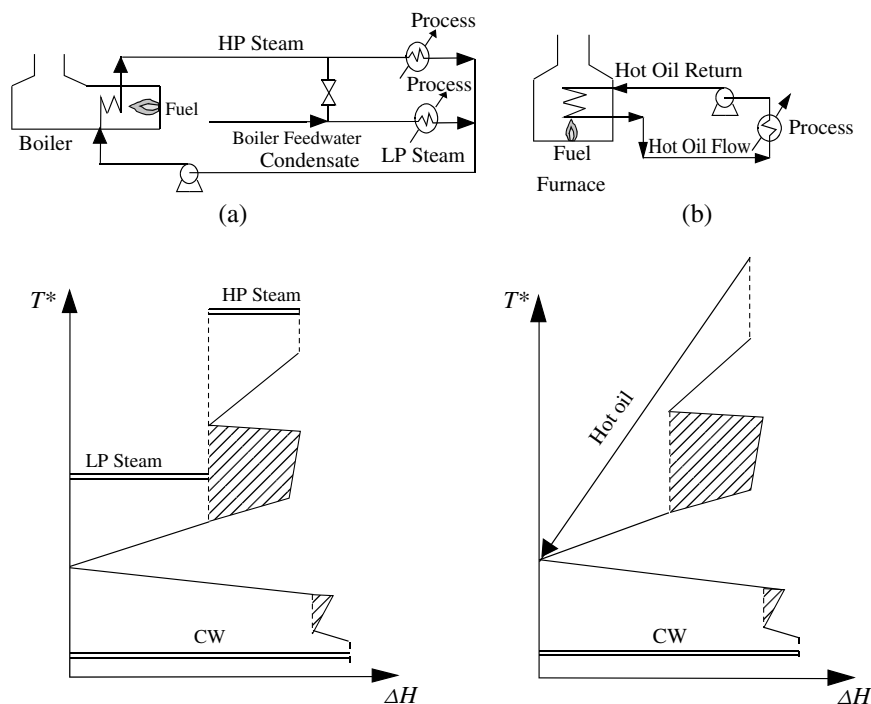


Figure 16.25 The grand composite curve allows alternative utility mixes to be evaluated.

are in practice. Thus, an allowance for ΔT_{min} is built into the construction.

The point of zero heat flow in the grand composite curve in Figure 16.24 is the heat recovery pinch. The open “jaws” at the top and bottom are Q_{Hmin} and Q_{Cmin} respectively. Thus, the heat sink above the pinch and heat source below the pinch can be identified as shown in Figure 16.24. The shaded areas in Figure 16.24, known as *pockets*, represent areas of additional process-to-process heat transfer. Remember that the profile of the grand composite curve represents residual heating and cooling demands after recovering heat within the shifted temperature intervals in the problem table algorithm. In the pockets in Figure 16.24, a local surplus of heat in the process is used at temperature differences in excess of ΔT_{min} to satisfy a local deficit. This reflects the cascading of excess heat from high-temperature intervals to lower temperature intervals in the problem table algorithm.

Figure 16.25a shows the same grand composite curve with two levels of steam used as hot utility. Figure 16.25b shows again the same grand composite curve but with hot oil used as hot utility.

Example 16.3 The problem table cascade for the process in Figure 16.2 is given in Figure 16.18. Using the grand composite curve:

- For two levels of steam at saturation conditions and temperatures of 240°C and 180°C, determine the loads on the two steam levels that maximizes the use of the lower pressure steam.

- Instead of using steam, a hot oil circuit is to be used with a supply temperature of 280°C and $C_P = 2.1 \text{ kJ}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$. Calculate the minimum flowrate of hot oil.

Solution

- For $\Delta T_{min} = 10^\circ\text{C}$, the two steam levels are plotted on the grand composite curve at temperatures of 235°C and 175°C. Figure 16.26a shows the loads that maximize the use of the lower pressure steam. Calculate the load on the low-pressure steam by interpolation of the cascade heat flows. At $T^* = 175^\circ\text{C}$:

$$\begin{aligned} \text{Load of } 180^\circ\text{C steam} &= \frac{75 - 145}{185 - 145} \times 4 \\ &= 3 \text{ MW} \end{aligned}$$

$$\begin{aligned} \text{Load of } 240^\circ\text{C steam} &= 7.5 - 3 \\ &= 4.5 \text{ MW} \end{aligned}$$

- Figure 16.26b shows the grand composite curve with hot oil providing the hot utility requirements. If the minimum flowrate is required, then this corresponds to the steepest slope and minimum return temperature. For this problem, the minimum return temperature for the hot oil is pinch temperature ($T^* = 145^\circ\text{C}$, $T = 150^\circ\text{C}$ for hot streams). Thus,

$$\begin{aligned} \text{Minimum flowrate} &= 7.5 \times 10^3 \times \frac{1}{2.1} \times \frac{1}{(280 - 150)} \\ &= 27.5 \text{ kg}\cdot\text{s}^{-1} \end{aligned}$$

In other problems, the shape of the grand composite curve away from the pinch could have limited the flowrate.

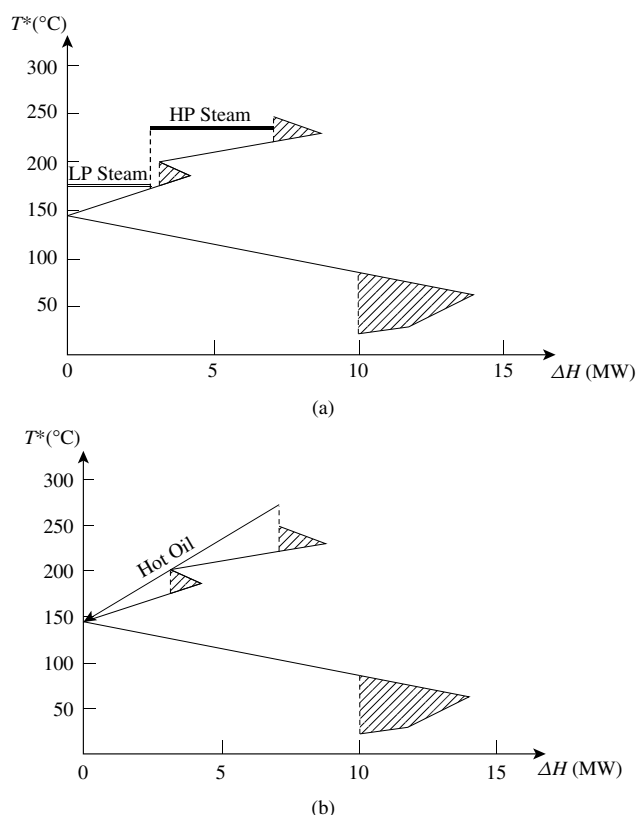


Figure 16.26 Alternative utility mixes for the process in Figure 16.2.

16.8 FURNACES

When hot utility needs to be at a high temperature and/or provide high heat fluxes, radiant heat transfer is used from combustion of fuel in a furnace. Furnace designs vary

according to the function of the furnace, heating duty, type of fuel and the method of introducing combustion air (see Chapter 15). Sometimes the function is to purely provide heat, sometimes the furnace is also a reactor and provides heat of reaction. However, process furnaces have a number of features in common. In the chamber in which combustion takes place, heat is transferred mainly by radiation to tubes around the walls of the chamber, through which passes the fluid to be heated. After the flue gas leaves the combustion chamber, most furnace designs extract further heat from the flue gas in a convection section before the flue gas is vented to atmosphere.

Figure 16.27 shows a grand composite curve with a flue gas matched against it to provide hot utility¹⁵. The flue gas starts at its theoretical flame temperature (see Chapter 15) shifted for ΔT_{min} on the grand composite curve and presents a sloping profile because it is giving up sensible heat. Theoretical flame temperature is the temperature attained when a fuel is burnt in air or oxygen without loss or gain of heat, as explained in Chapter 15.

In Figure 16.27, the flue gas is cooled to pinch temperature before being released to atmosphere. The heat released from the flue gas between pinch temperature and ambient is the stack loss. Thus in Figure 16.27, for a given grand composite curve and theoretical flame temperature, the heat from fuel and stack loss can be determined.

All combustion processes work with an excess of air or oxygen to ensure complete combustion of the fuel. Excess air typically ranges between 5 and 20% depending on the fuel, burner design and furnace design. As excess air is reduced, theoretical flame temperature increases as shown in Figure 16.28. This has the effect of reducing the stack loss and increasing the thermal efficiency of the furnace for a given process heating duty. Alternatively, if

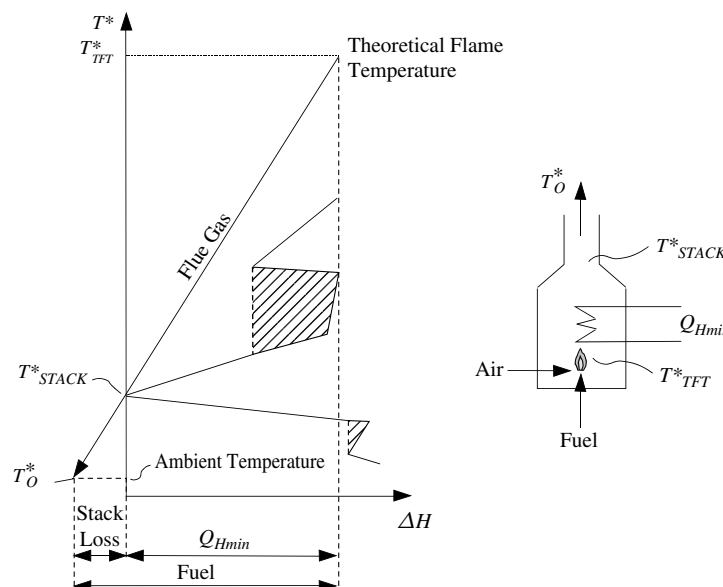


Figure 16.27 Simple furnace model.

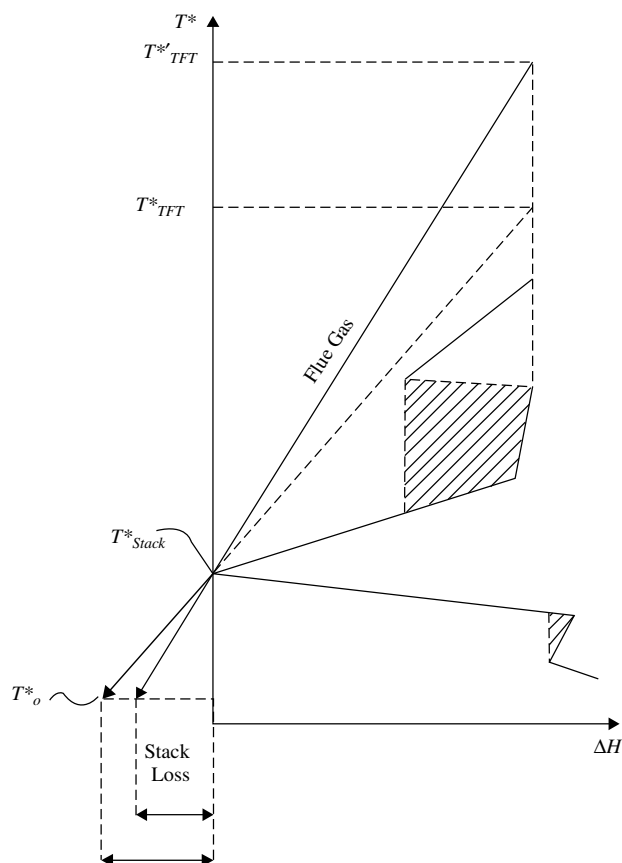


Figure 16.28 Increasing the theoretical flame temperature by reducing excess air or preheating combustion air reduces the stack loss.

the combustion air is preheated (say by heat recovery), then again the theoretical flame temperature increases as shown in Figure 16.28, reducing the stack loss.

Although, the higher flame temperatures in Figure 16.28 reduce the fuel consumption for a given process heating duty, there is one significant disadvantage. Higher flame temperatures increase the formation of oxides of nitrogen, which are environmentally harmful. This point will be returned to in Chapter 25.

In Figures 16.27 and 16.28, the flue gas is capable of being cooled to pinch temperature before being released to atmosphere. This is not always the case. Figure 16.29a shows a situation in which the flue gas is released to atmosphere above pinch temperature for practical reasons. There is a practical minimum, the acid dew point, to which a flue gas can be cooled without condensation causing corrosion in the stack (see Chapter 15). The minimum stack temperature in Figure 16.29a is fixed by acid dew point. Another case is shown in Figure 16.29b where the process away from the pinch limits the slope of the flue gas line and hence the stack loss.

Example 16.4 The process in Figure 16.2 is to have its hot utility supplied by a furnace. The theoretical flame temperature for combustion is 1800°C and the acid dew point for the flue gas is 160°C . Ambient temperature is 10°C . Assume $\Delta T_{\min} = 10^{\circ}\text{C}$ for process-to-process heat transfer but $\Delta T_{\min} = 30^{\circ}\text{C}$ for flue gas to process heat transfer. A high value for $\Delta T_{\min}$ for flue gas to process heat transfer has been chosen because of poor heat-transfer coefficients in the convection bank of the furnace. Calculate the fuel required, stack loss and furnace efficiency.

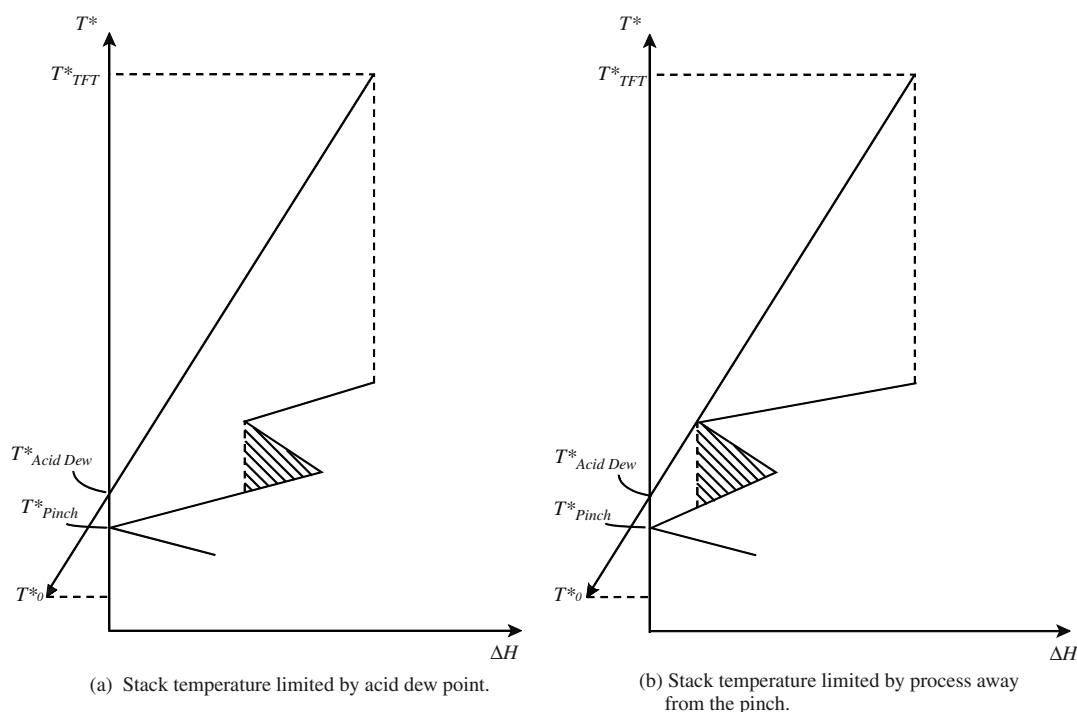


Figure 16.29 Furnace stack temperature can be limited by other factors than pinch temperature.

Solution The first problem is that a different value of ΔT_{min} is required for different matches. The problem table algorithm is easily adapted to accommodate this. This is achieved by assigning ΔT_{min} contributions to streams. If the process streams are assigned a contribution of 5°C and flue gas a contribution of 25°C , then a process/process match has a ΔT_{min} of $(5 + 5) = 10^\circ\text{C}$ and a process/flue gas match has a ΔT_{min} of $(5 + 25) = 30^\circ\text{C}$. When setting up the interval temperatures in the problem table algorithm, the interval boundaries are now set at hot stream temperatures minus their ΔT_{min} contribution, rather than half the global ΔT_{min} . Similarly, boundaries are now set on the basis of cold stream temperatures plus their ΔT_{min} contribution.

Figure 16.30 shows the grand composite curve plotted from the problem table cascade in Figure 16.18b. The starting point for the flue gas is an actual temperature of 1800°C , which corresponds to a shifted temperature of $(1800 - 25) = 1775^\circ\text{C}$ on the grand composite curve. The flue gas profile is not restricted above the pinch and can be cooled to pinch temperature corresponding with a shifted temperature of 145°C before venting to atmosphere. The actual stack temperature is thus $(145 + 25) = 170^\circ\text{C}$. This is just above the acid dew point of 160°C . Now calculate the fuel consumption.

$$Q_{Hmin} = 7.5 \text{ MW}$$

$$CP_{FLUE\ GAS} = \frac{7.5}{1775 - 145}$$

$$= 0.0046 \text{ MW}\cdot\text{K}^{-1}$$

The fuel consumption is now calculated by taking the flue gas from theoretical flame temperature to ambient temperature:

$$\text{Fuel required} = 0.0046(1800 - 10)$$

$$= 8.23 \text{ MW}$$

$$\text{Stack loss} = 0.0046(170 - 10)$$

$$= 0.74 \text{ MW}$$

$$\text{Furnace efficiency} = \frac{Q_{Hmin}}{\text{Fuel Required}} \times 100$$

$$= \frac{7.5}{8.23} \times 100$$

$$= 91\%$$

16.9 COGENERATION (COMBINED HEAT AND POWER GENERATION)

More complex utility options are encountered when combined heat and power generation (or *cogeneration*) is exploited. Here the heat rejected by a heat engine such as a steam turbine, gas turbine or diesel engine is used as hot utility.

Fundamentally, there are two possible ways to integrate a heat engine exhaust¹⁴. In Figure 16.31, the process is

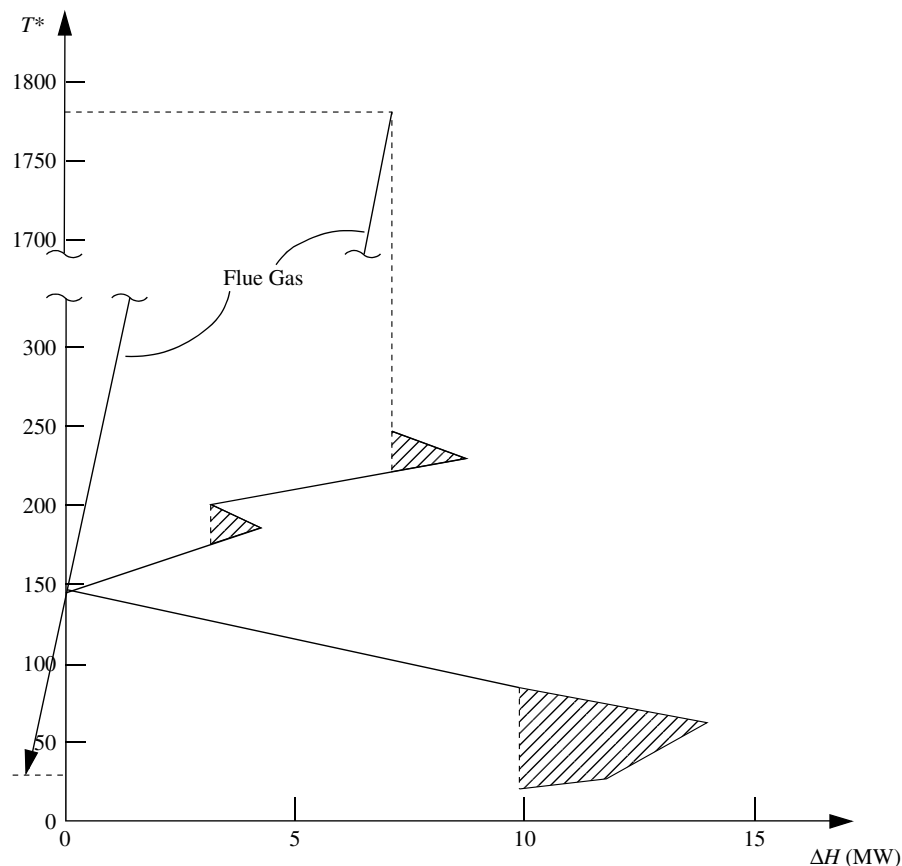


Figure 16.30 Flue gas matched against the grand composite curve of the process in Figure 16.2.

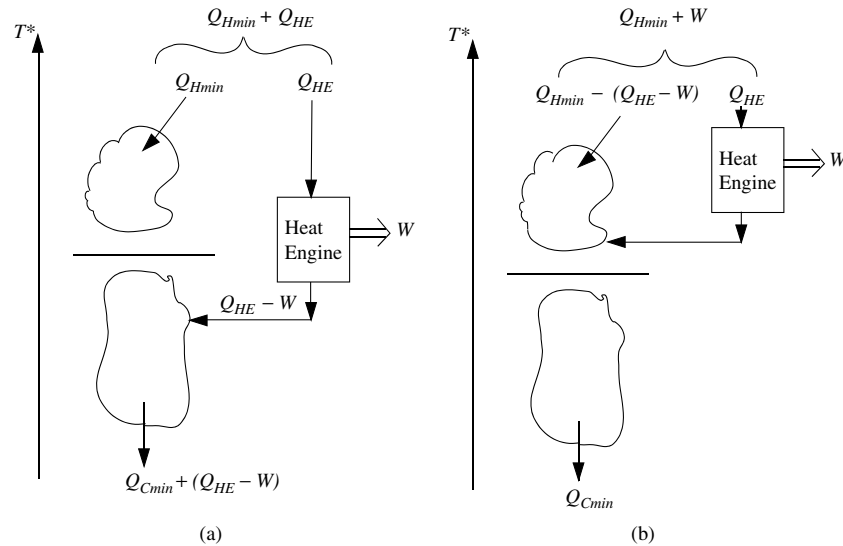


Figure 16.31 Heat engine exhaust can be integrated either across or not across the pinch.

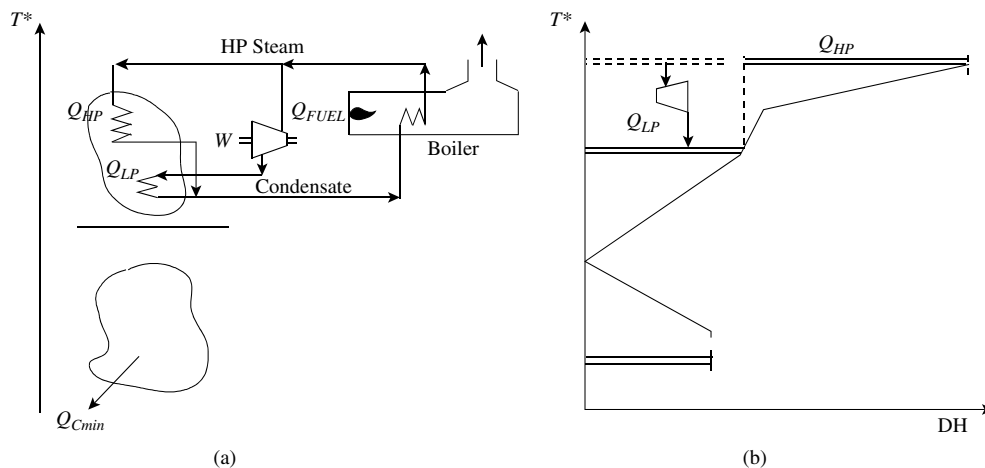


Figure 16.32 Integration of a steam turbine with the process.

represented as a heat sink and heat source separated by the pinch. Integration of the heat engine across the pinch as shown in Figure 16.31a is counterproductive. The process still requires Q_{Hmin} and the heat engine performs no better than operated stand-alone. There is no saving by integrating a heat engine across the pinch¹⁴.

Figure 16.31b shows the heat engine integrated above the pinch. In rejecting heat above the pinch, it is rejecting heat into the part of the process, which is overall a heat sink. In so doing, it is exploiting the temperature difference that exists between the utility source and the process sink, producing power at high efficiency. The net effect in Figure 16.31b is the import of extra energy W from heat sources to produce W power. Owing to the process and heat engine acting as one system, apparently conversion of heat to power at 100% efficiency is achieved¹⁴.

Now consider the two most commonly used heat engines (steam and gas turbines) in more detail to see whether

they achieve this in practice. To make a quantitative assessment of any combined heat and power scheme, the grand composite curve should be used and the heat engine exhaust treated like any other utility.

Figure 16.32 shows a steam turbine integrated with the process above the pinch. A steam turbine is conceptually like a centrifugal compressor, but working in reverse. Steam is expanded from high to low pressure in the machine, producing power. In Figure 16.32 heat Q_{HP} is taken into the process from high-pressure steam. The balance of the hot utility demand Q_{LP} is taken from the steam turbine exhaust. In Figure 16.32a, heat Q_{FUEL} is taken into the boiler from fuel. An overall energy balance gives:

$$Q_{FUEL} = Q_{HP} + Q_{LP} + W + Q_{LOSS} \quad (16.2)$$

The process requires $(Q_{HP} + Q_{LP})$ to satisfy its enthalpy imbalance above the pinch. If there were no losses from

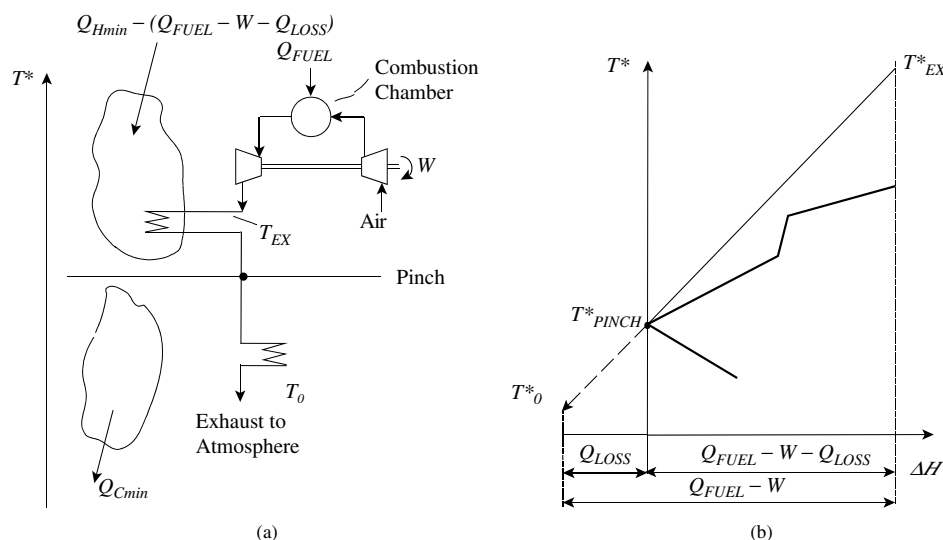


Figure 16.33 A gas turbine exhaust matched against the process (same as a flue gas).

the boiler, then fuel W would be converted to power W at 100% efficiency. However, the boiler losses Q_{LOSS} reduce this to below 100% conversion. In practice, in addition to the boiler losses, there can also be significant losses from the steam distribution system. Figure 16.32b shows how the grand composite curve can be used to size steam turbine cycles¹⁴. Steam turbines will be dealt with in more detail in Chapter 23.

Figure 16.33 shows a schematic of a simple gas turbine. The machine is essentially a rotary compressor mounted on the same shaft as a turbine. Air enters the compressor where it is compressed before entering a combustion chamber. Here the combustion of fuel increases its temperature. The mixture of air and combustion gases is expanded in the turbine. The input of energy to the combustion chamber allows enough power to be developed in the turbine to both drive the compressor and provide useful power. The performance of the machine is specified in terms of the power output, airflow rate through the machine, efficiency of conversion of heat to power and the temperature of the exhaust. Gas turbines are normally used only for relatively large-scale applications, and will be dealt with in more detail in Chapter 23.

Figure 16.33 shows a gas turbine matched against the grand composite curve¹⁴. As with the steam turbine, if there was no stack loss to atmosphere (i.e. if Q_{LOSS} was zero), then W heat would be turned into W power. The stack losses in Figure 16.33 reduce the efficiency of conversion of heat to power. The overall efficiency of conversion of heat to power depends on the turbine exhaust profile, the pinch temperature and the shape of the process grand composite.

Example 16.5 The stream data for a heat recovery problem are given in Table 16.7 below.

Table 16.7 Stream data for Example 16.5.

Stream No.	Type	T_S (°C)	T_T (°C)	Heat capacity flowrate (MW·K ⁻¹)
1	Hot	450	50	0.25
2	Hot	50	40	1.5
3	Cold	30	400	0.22
4	Cold	30	400	0.05
5	Cold	120	121	22.0

A problem table analysis for $\Delta T_{min} = 20^\circ\text{C}$ results in the heat cascade given in Table 16.8.

Table 16.8 Problem table cascade for Example 16.5.

T^* (°C)	Cascade heat flow (MW)
440	21.9
410	29.4
131	23.82
130	1.8
40	0
30	15

The process also has a requirement for 7 MW of power. Two alternative cogeneration schemes are to be compared economically.

- A steam turbine with its exhaust saturated at 150°C used for process heating is one of the options to be considered. Superheated steam is generated in the central boiler house at 41 bar with a temperature of 300°C . This superheated steam can be expanded in a single-stage turbine with an isentropic

- b. The exhaust from the gas turbine is primarily air with a small amount of combustion gases. Hence, the CP of the exhaust can be approximated to be that of the airflow. Assuming C_p for air = $1.03 \text{ kJ}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$.

$$\begin{aligned} CP_{EX} &= 97 \times 1.03 \\ &= 100 \text{ kW}\cdot\text{K}^{-1} \end{aligned}$$

The gas turbine option is shown in Figure 16.34b.

$$\begin{aligned} Q_{EX} &= CP_{EX}(T_{EX} - T_0) \\ &= 0.1 \times (400 - 10) \\ &= 39 \text{ MW} \end{aligned}$$

$$\begin{aligned} Q_{FUEL} &= \frac{Q_{EX}}{(1 - \eta_{GT})} \\ &= \frac{39}{(1 - 0.3)} \\ &= 55.71 \text{ MW} \end{aligned}$$

$$\begin{aligned} W &= Q_{FUEL} - Q_{EX} \\ &= 16.71 \text{ MW} \end{aligned}$$

- c. Steam turbine economics:

$$\begin{aligned} \text{Cost of fuel} &= (21.9 + 3.96) \times \frac{3.2 \times 10^{-3}}{0.8} \\ &= \$0.10 \text{ s}^{-1} \end{aligned}$$

$$\begin{aligned} \text{Cost of imported electricity} &= (7 - 3.96) \times 19.2 \times 10^{-3} \\ &= \$0.06 \text{ s}^{-1} \end{aligned}$$

$$\text{Net cost} = \$0.16 \text{ s}^{-1}$$

Gas turbine economics:

$$\begin{aligned} \text{Cost of fuel} &= 55.71 \times 4.5 \times 10^{-3} \\ &= \$0.25 \text{ s}^{-1} \end{aligned}$$

$$\begin{aligned} \text{Electricity credit} &= (16.71 - 7) \times 14.4 \times 10^{-3} \\ &= \$0.14 \text{ s}^{-1} \end{aligned}$$

$$\text{Net cost} = \$0.11 \text{ s}^{-1}$$

Hence the gas turbine is the most profitable in terms of energy costs. However, this is only part of the story since the capital cost of a gas turbine installation is likely to be significantly higher than that of a steam turbine installation.

Example 16.6 The problem table cascade for a process is given in Table 16.9 below for $\Delta T_{min} = 10^\circ\text{C}$.

It is proposed to provide process cooling by steam generation from boiler feedwater with a temperature of 100°C .

Table 16.9 Problem table cascade.

Interval temperature ($^\circ\text{C}$)	Heat flow (MW)
495	3.6
455	9.2
415	10.8
305	4.2
285	0
215	16.8
195	17.6
185	16.6
125	16.6
95	21.1
85	18.1

- Determine how much steam can be generated at a saturation temperature of 230°C .
- Determine how much steam can be generated with a saturation temperature of 230°C and superheated to the maximum temperature possible against the process.
- Calculate how much power can be generated from the superheated steam from Part *b*, assuming a single-stage condensing steam turbine is to be used with an isentropic efficiency of 85%. Cooling water is available at 20°C and is to be returned to the cooling tower at 30°C .

Solution

- a. Heat available for steam generation at 235°C interval temperature
- $$= 12.0 \text{ MW}$$

From steam tables, the latent heat of water at a saturated temperature of 230°C is $1812 \text{ kJ}\cdot\text{kg}^{-1}$.

$$\begin{aligned} \text{Steam production} &= \frac{12.0 \times 10^3}{1812} \\ &= 6.62 \text{ kg}\cdot\text{s}^{-1} \end{aligned}$$

Taking the heat capacity of water to be $4.3 \text{ kJ}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$, heat duty on boiler feedwater preheating

$$\begin{aligned} &= 6.62 \times 4.3 \times 10^{-3} (230 - 100) \\ &= 3.70 \text{ MW} \end{aligned}$$

The profile of steam generation is shown against the grand composite curve in Figure 16.35a. The process can support both boiler feedwater preheat and steam generation.

- b. Maximum superheat temperature

$$\begin{aligned} &= 285^\circ\text{C interval} \\ &= 280^\circ\text{C actual} \end{aligned}$$

The profile is shown against the grand composite curve in Figure 16.35b.

From steam tables, enthalpy of superheated steam at 280°C and 28 bar

$$= 2947 \text{ kJ}\cdot\text{kg}^{-1}$$

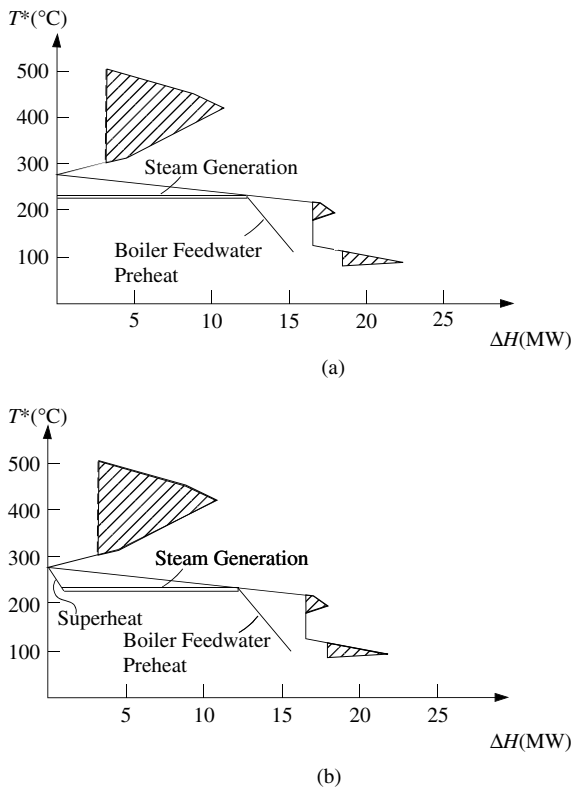


Figure 16.35 Alternative cold utilities for Example 16.6.

and enthalpy of saturated water at 230°C and 28 bar

$$= 991 \text{ kJ} \cdot \text{kg}^{-1}$$

$$\begin{aligned} \text{Steam production} &= \frac{12.0 \times 10^3}{(2947 - 991)} \\ &= 6.13 \text{ kg} \cdot \text{s}^{-1} \end{aligned}$$

- c. In a condensing turbine, the exhaust from the turbine is condensed under vacuum against cooling water. The lower the condensing temperature, the greater the power generation. The lowest condensing temperature for this problem is cooling water temperature plus ΔT_{min} , that is, $30 + 10 = 40^\circ\text{C}$. From steam tables, inlet conditions at $T_1 = 280^\circ\text{C}$ and $P_1 = 28 \text{ bar}$ are:

$$H_1 = 2947 \text{ kJ} \cdot \text{kg}^{-1}$$

$$S_1 = 6.488 \text{ kJ} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$$

Turbine outlet conditions for isentropic expansion to 40°C from steam tables are:

$$P_2 = 0.074 \text{ bar}$$

For $S_2 = 6.488 \text{ kJ} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$, the wetness fraction (X) and outlet enthalpy H_2 can be calculated as shown in Example 14.4.

$$X = 0.23$$

$$H_2 = 2020 \text{ kJ} \cdot \text{kg}^{-1}$$

For a single-stage expansion with isentropic efficiency of 85%:

$$\begin{aligned} H_2' &= 2947 - 0.85(2947 - 2020) \\ &= 2159 \text{ kJ} \cdot \text{kg}^{-1} \end{aligned}$$

The power generation (W) is given by:

$$\begin{aligned} W &= 6.13(2947 - 2159) \times 10^{-3} \\ &= 4.8 \text{ MW} \end{aligned}$$

The wetness fraction for the real expansion is given by:

$$\begin{aligned} H_2' &= 2159 = X'H_L + (1 - X')H_V \\ &= 167.5X' + 2574(1 - X') \\ X' &= 0.17 \end{aligned}$$

This wetness fraction is possibly too high, since high levels of wetness can cause damage to the turbine. To allow a lower wetness fraction of say $X = 0.15$, the outlet pressure of the turbine must be raised to 0.2 bar, corresponding to a condensing temperature of 60°C . However, in so doing, the power generation decreases to 4.2 MW.

16.10 INTEGRATION OF HEAT PUMPS

A heat pump is a device that takes in low-temperature heat and upgrades it to a higher temperature to provide process heat. A schematic of a simple vapor compression heat pump is shown in Figure 16.36. In Figure 16.36, the heat pump absorbs heat at a low temperature in the evaporator, consumes power when the working fluid is compressed and rejects heat at a higher temperature in the condenser. The condensed working fluid is expanded and partially vaporizes. The cycle then repeats. The working fluid is usually a pure component, which means that the evaporation and condensation take place isothermally. When considering the integration of a heat pump with the process, there are appropriate and inappropriate ways to integrate heat pumps.

There are two fundamental ways in which a heat pump can be integrated with the process; across and not across the pinch¹⁴. Integration not across (above) the pinch is illustrated in Figure 16.37a. This arrangement imports W

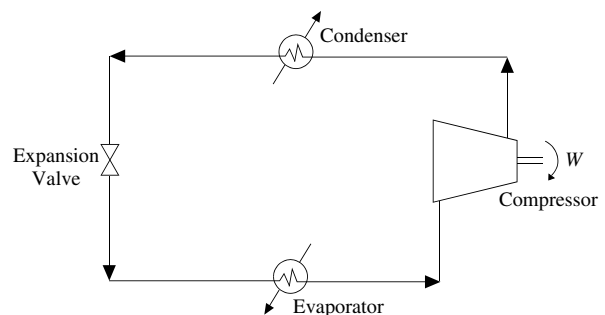


Figure 16.36 Schematic of a simple vapor compression heat pump.

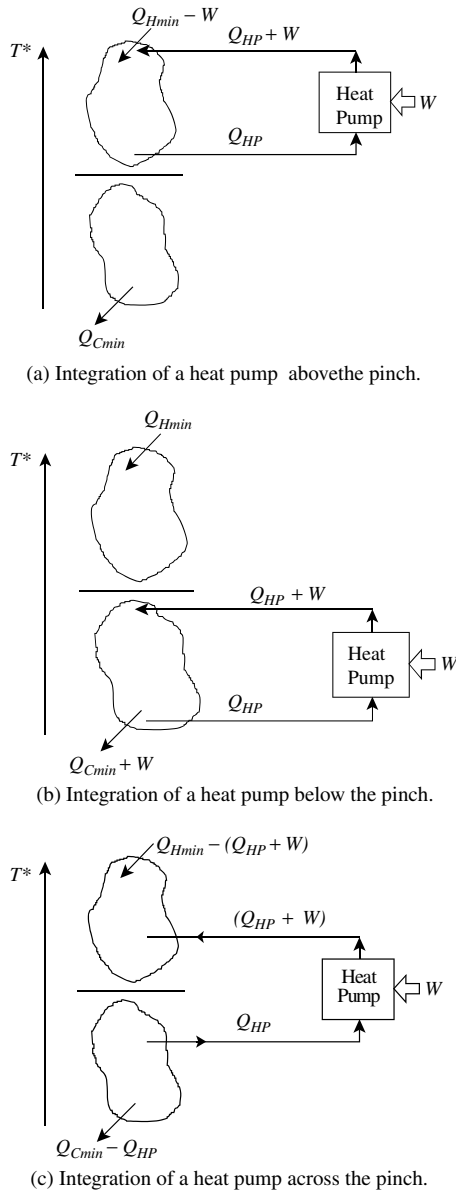


Figure 16.37 Integration of heat pump with the process.

power and saves W hot utility. In other words, the system converts power into heat, which is not normally worthwhile economically. Another integration not across (below) the pinch is shown in Figure 16.37b. The result is worse economically. Power is turned into waste heat¹⁴.

Integration across the pinch is illustrated in Figure 16.37c. This arrangement brings a genuine saving. It also makes overall sense since heat is pumped from the part of the process that is overall a heat source to the part that is overall a heat sink.

Figure 16.38 shows a heat pump appropriately integrated against a process. Figure 16.38a shows the overall balance. Figure 16.38b illustrates how the grand composite curve can be used to size the heat pump. How the heat pump performs determines its coefficient of performance. The coefficient of performance for a heat pump can generally be defined as the useful energy delivered to the process divided by the power expended to produce this useful energy. From Figure 16.38a:

$$COP_{HP} = \frac{Q_{HP} + W}{W} \quad (16.3)$$

where COP_{HP} is the heat pump coefficient of performance, Q_{HP} the heat absorbed at low temperature and W the power consumed.

For any given type of heat pump, a higher COP_{HP} leads to better economics. Having a better COP and hence better economics means working across a small temperature lift with the heat pump. The smaller the temperature lift, the better the COP_{HP} . For most applications, a temperature lift greater than 25°C is rarely economic. Attractive heat pump application normally requires a lift much less than 25°C .

Using the grand composite curve, the loads and temperatures of the cooling and heating duties and hence the COP_{HP} of integrated heat pumps can be readily assessed.

Thus, the appropriate placement of heat pumps is that they should be placed across the pinch¹⁴. Note that the principle needs careful interpretation if there are utility pinches. In such circumstances, heat pump placement above the process pinch or below it can be economic, providing

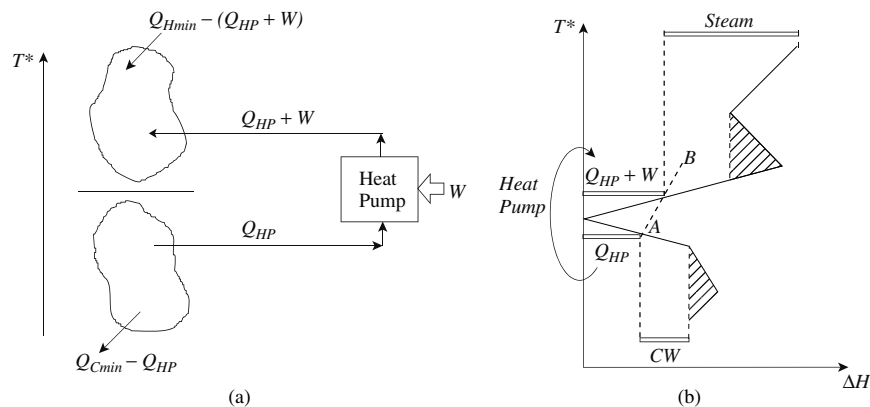


Figure 16.38 The grand composite curve allows heat pump cycles to be sized.

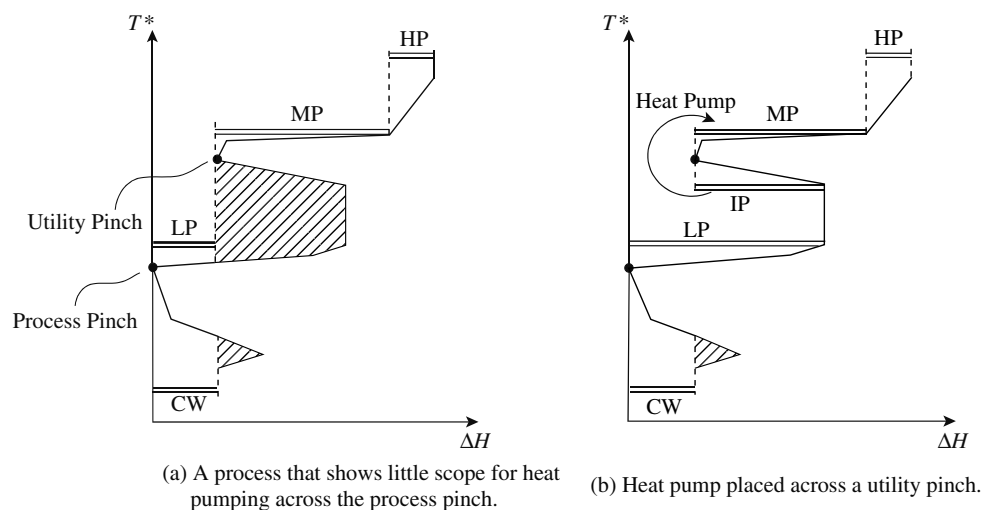


Figure 16.39 Heat pumping can be applied across utility pinches as well as the process pinch.

the heat pump is placed across a utility pinch. Figure 16.39a shows a process that does not show a significant potential for heat pumping across the process pinch. The heat source just below the process pinch is small. The heat sink just above the process pinch is also small. Significant heat pumping across the process pinch can only be accomplished across a large temperature difference in this case, which will result in a poor COP_{HP} . However, Figure 16.39b shows another possible heat pump arrangement. A heat source in the pocket of the grand composite curve is pumped through a relatively small temperature difference to replace medium-pressure (MP) steam. To compensate for taking heat from within the pocket of the grand composite curve, additional low-pressure (LP) steam is used for process heating to maintain the heat balance. Thus, the heat pump results in a saving in MP steam for a sacrifice of extra LP steam usage and the power required for the heat pump. This might be economic if there is a large cost difference between MP and LP steam. Although the heat pump does not operate across the process pinch, it does not violate the principle of the appropriate placement of heat pumps. The heat pump in Figure 16.39b operates across a utility pinch.

16.11 HEAT EXCHANGER NETWORK ENERGY TARGETS – SUMMARY

The energy targets for the process can be set without having to design the heat exchanger network and utility system. These energy targets can be calculated directly from the material and energy balance. Thus, energy costs can be established without design for the outer layers of the process onion. Using the grand composite curve, different utility scenarios can be screened quickly and conveniently, including cogeneration and heat pumps.

16.12 EXERCISES

1. A heat recovery problem consists of two streams given in Table 16.10:

Table 16.10 Stream data for Exercise 1.

Stream	Type	Supply temperature (°C)	Target temperature (°C)	Enthalpy change (MW)
1	Hot	100	40	12
2	Cold	10	150	7

Steam is available at 180°C and cooling water at 20°C.

- a. For a minimum permissible temperature difference (ΔT_{min}) of 10°C, calculate the minimum hot and cold utility requirements.
 - b. What are the hot and cold stream pinch temperatures?
 - c. If the steam is supplied as the exhaust from a steam turbine, is the heat engine appropriately placed?
 - d. If the (ΔT_{min}) is increased to 20°C, what will happen to the utility requirements?
2. The stream data for a process are given in Table 16.11.

Table 16.11 Stream data for Exercise 2.

Stream No.	Type	T_S (°C)	T_T (°C)	Heat duty (MW)
1	Hot	160	40	3.6
2	Hot	50	50	5.0
3	Hot	140	110	1.5
4	Cold	160	160	5.0
5	Cold	60	150	1.8

- a. Sketch the composite curves for $\Delta T_{min} = 10^\circ\text{C}$.
- b. From the composite curves, determine the target for hot and cold utility for $\Delta T_{min} 10^\circ\text{C}$.

3. The problem table cascade for a process is given in the Table 16.12 for $\Delta T_{min} = 10^\circ\text{C}$.

Table 16.12 Problem table cascade for Exercise 3.

Interval temperature ($^\circ\text{C}$)	Cascade heat flow (MW)
360	9.0
310	7.6
230	8.0
210	4.8
190	5.8
190	0
170	1.0
150	7.6
60	3.0

The minimum permissible temperature differences for various options are given in Table 16.13.

Table 16.13 ΔT_{min} for different matches.

Type of match	ΔT_{min} ($^\circ\text{C}$)
Process/process	10
Process/steam	10
Process/flue gas (furnace or gas turbine)	50

- a. Calculate the amount of fuel required to satisfy the heating requirements for a furnace used to supply hot utility for a theoretical flame temperature of 1800°C and acid dew point of 150°C .
- b. If the furnace from Part *a* is used in conjunction with saturated steam at 250°C for heating, such that the heat duty for the steam is maximized, what is the heat duty on the steam and the furnace heat duty?
4. The stream data for a process involving an exothermic chemical reaction are given in the Table 16.14.

Table 16.14 Stream data for Exercise 4.

Stream No.	Type	Enthalpy Change (kW)	T_S ($^\circ\text{C}$)	T_T ($^\circ\text{C}$)
1	Hot	-7000	377	375
2	Hot	-3600	376	180
3	Hot	-2400	180	70
4	Cold	2400	60	160
5	Cold	200	20	130
6	Cold	200	160	260

A problem table analysis of the data indicates that it is a threshold problem requiring only cold utility. The threshold value of ΔT_{min} is 117°C , corresponding with a cold utility duty of 10,100 kW. It is proposed to use steam generation as cold utility for which $\Delta T_{min} = 10^\circ\text{C}$.

- a. Assume that saturated boiler feedwater is available and that the steam generated is saturated, in order to calculate how

much steam can be generated by the process at a pressure of 41 bar. The temperature of saturated steam at this pressure is 252°C and the latent heat is $1706 \text{ kJ}\cdot\text{kg}^{-1}$.

- b. If the steam is superheated to a temperature of 350°C , calculate how much steam can be generated at 41 bar. Assume the heat capacity of steam is $4.0 \text{ kJ}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$.
- c. What would happen if the steam in Part *b* was generated from boiler feedwater at 100°C with a heat capacity of $4.2 \text{ kJ}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$. How would the steam generation be calculated under these circumstances?
5. The problem table cascade for a process is given in Table 16.15 for $\Delta T_{min} = 10^\circ\text{C}$.

Table 16.15 Problem table cascade for Exercise 5.

Interval temperature ($^\circ\text{C}$)	Heat flow (MW)
495	3.6
455	9.2
415	10.8
305	4.2
285	0
215	16.8
195	17.6
185	16.6
125	16.6
95	21.1
85	18.1

It is proposed to provide process cooling by steam generation from boiler feedwater with a temperature of 80°C .

- a. Determine how much saturated steam can be generated at a temperature of 230°C . The latent heat of steam under these conditions is $1812 \text{ kJ}\cdot\text{kg}^{-1}$. The heat capacity of water can be taken as $4.2 \text{ kJ}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$.
- b. Determine how much steam can be generated with a saturation temperature of 230°C and superheated to the maximum temperature possible against the process. The heat capacity of steam can be taken as $3.45 \text{ kJ}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$.
- c. Calculate how much power can be generated from the superheated steam from Part *b*, assuming the exhaust steam is saturated at a pressure of 4 bar.
6. The problem table cascade for a process is given in Table 16.16 for $\Delta T_{min} = 20^\circ\text{C}$.

Table 16.16 Problem table cascade for Exercise 6.

Interval temperature ($^\circ\text{C}$)	Heat flow (MW)
440	21.9
410	29.4
130	23.82
130	1.8
100	0
95	15

The process also has a requirement for 7 MW of power. Three alternative utility schemes are to be compared economically:

- A steam turbine with its exhaust saturated at 150°C used for process heating is one of the options to be considered. Steam is raised in the central boiler house at 41 bar with an enthalpy of 3137 kJ·kg⁻¹. The enthalpy of saturated steam at 150°C is 2747 kJ·kg⁻¹ and its latent heat is 2115 kJ·kg⁻¹. Calculate the maximum power generation possible by matching the exhaust steam against the process.
- A second possible scheme uses a gas turbine with an exhaust temperature of 400°C and heat capacity flowrate of 0.1 MW·K⁻¹. Calculate the power generation if the turbine converts heat to power with an efficiency of 30%. Ambient temperature is 10°C.
- A third possible scheme integrates a heat pump with the process. The power required by the heat pump is given by

$$W = \frac{Q_H}{0.6} \frac{T_H - T_C}{T_H}$$

where Q_H is the heat rejected by the heat pump at T_H (K). Heat is absorbed into the heat pump at T_C (K). Calculate the power required by the heat pump.

- The cost utilities are given in the Table 16.17.

Table 16.17 Utility costs for Exercise 6.

Utility	Cost (\$·MW ⁻¹)
Fuel for gas turbine	0.0042
Fuel for steam generation	0.0030
Imported power	0.018
Credit for exported power	0.014
Cooling water	0.00018

The overall efficiency of steam generation and distribution is 60%. Which scheme is most cost-effective, the steam turbine, the gas turbine or the heat pump?

- The Table 16.18 represents a problem table cascade ($\Delta T_{min} = 20^\circ\text{C}$).

Table 16.18 Problem table cascade for Exercise 7.

Interval temperature (°C)	Heat flow (kW)
160	1000
150	0
130	1100
110	1400
100	900
80	1300
40	1400
10	1800
-10	1900
-30	2200

The following utilities are available:

- MP steam at 200°C
- LP steam at 107°C raised from boiler feed water at 60°C
- Cooling water (20 to 40°C)
- Refrigeration at 0°C
- Refrigeration at -40°C

For matches between process and refrigeration, $\Delta T_{min} = 10^\circ\text{C}$.

Draw the process grand composite curve and set the targets for the utilities. Below the pinch use of higher temperature, cold utilities should be maximized. For boiler feedwater, the specific heat capacity is 4.2 kJ·kg⁻¹·K⁻¹ and the latent heat of vaporization is 2238 kJ·kg⁻¹.

REFERENCES

- Hohman EC (1971) *Optimum Networks of Heat Exchange*, PhD Thesis, University of Southern California.
- Huang F and Elshout RV (1976) Optimizing the Heat Recover of Crude Units, *Chem Eng Prog*, **72**: 68.
- Linnhoff B, Mason DR and Wardle I (1979) Understanding Heat Exchanger Networks, *Comp Chem Eng*, **3**: 295.
- Linnhoff B, Townsend DW, Boland D, Hewitt GF, Thomas BEA, Guy AR and Marsland RH (1982) *A User Guide on Process Integration for the Efficient Use of Energy*, IChemE, UK.
- Polley GT (1993) Heat Exchanger Design and Process Integration, *Chem Eng*, **8**: 16.
- Umeda T, Itoh J and Shiroko K (1978) Heat Exchange System Synthesis, *Chem Eng Prog*, **74**: 70.
- Umeda T, Harada T and Shiroko K (1979) A Thermodynamic Approach to the Synthesis of Heat Integration Systems in Chemical Processes, *Comp Chem Eng*, **3**: 273.
- Umeda T, Niida K and Shiroko K (1979) A Thermodynamic Approach to Heat Integration in Distillation Systems, *AIChE J*, **25**: 423.
- Linnhoff B and Flower JR (1978) Synthesis of Heat Exchanger Networks, *AIChE J*, **24**: 633.
- Ahmad S and Hui DCW (1991) Heat Recovery Between Areas of Integrity, *Comp Chem Eng*, **15**: 809.
- Cerda J, Westerberg AW, Mason D and Linnhoff B (1983) Minimum Utility Usage in Heat Exchanger Network Synthesis – A Transportation Problem, *Chem Eng Sci*, **38**: 373.
- Papoulias SA and Grossmann IE (1983) A Structural Optimization Approach in Process Synthesis – II Heat Recovery Networks, *Comp Chem Eng*, **7**: 707.
- Itoh J, Shiroko K and Umeda T (1982) Extensive Application of the T-Q Diagram to Heat Integrated System Synthesis, *International Conference on Proceedings Systems Engineering (PSE-82)*, Kyoto, 92.
- Townsend DW and Linnhoff B (1983) Heat and Power Networks in Process Design, *AIChE J*, **29**: 742.
- Linnhoff B and de Leur J (1988) Appropriate Placement of Furnaces in the Integrated Process, *IChemE Symp Ser*, **109**: 259.

18 Heat Exchanger Networks IV – Network Design

Having explored the targets for the heat exchanger network, it now remains to develop the design of the heat exchanger network.

18.1 THE PINCH DESIGN METHOD

The capital-energy trade-off in the heat exchanger networks was discussed in Chapter 16. Varying ΔT_{min} , as shown in Figure 16.6, changed the relative position of the process composite curves. As ΔT_{min} is changed from a small to a large value, the capital cost decreases but the energy cost increases. When the two costs are combined to obtain a total cost, the optimum point in the capital-energy trade-off is identified, corresponding with an optimum value of ΔT_{min} (Figure 16.6). As pointed out in Chapter 16, the trade-off between energy and capital suggests that individual exchangers should have a temperature difference no smaller than the ΔT_{min} between the composite curves.

It was suggested in Chapter 16 that a good initialization would be to assume that no individual exchanger should have a temperature difference smaller than ΔT_{min} . Having made this assumption, two rules were deduced in Chapter 16. If the energy target set by the composite curves (or the problem table algorithm) is to be achieved, the design must not transfer heat across the pinch by:

- Process-to-process heat transfer
- Inappropriate use of utilities

These rules are necessary for the design to achieve the energy target, given that no individual exchanger should have a temperature difference smaller than ΔT_{min} . To comply with these two rules, the process should be divided at the pinch. As pointed out in Chapter 16, this is most clearly done by representing the stream data in the grid diagram. Figure 18.1 shows the stream data from Table 16.2 in grid form with the pinch marked. Above the pinch, steam can be used (up to Q_{Hmin}), and below the pinch cooling water can be used (up to Q_{Cmin}). But what strategy should be adopted for the design? A number of simple criteria can be developed to help¹.

1. *Start at the pinch.* The pinch is the most constrained region of the problem. At the pinch, ΔT_{min} exists between all hot and cold streams. As a result, the number of feasible matches in this region is severely restricted. Quite often there

are essential matches to be made. If such matches are not made, the result will be either use of temperature differences smaller than ΔT_{min} or excessive use of utilities resulting from heat transfer across the pinch. If the design was started away from the pinch at the hot end or cold end of the problem, then initial matches are likely to need follow-up matches that violate the pinch or the ΔT_{min} criterion as the pinch is approached. Putting the argument the other way around, if the design is started at the pinch, then initial decisions are made in the most constrained part of the problem. This is much less likely to lead to difficulties later.

2. *The CP inequality for individual matches.* Figure 18.2a shows the temperature profiles for an individual exchanger at the pinch, above the pinch^{1,2}. Moving away from the pinch, temperature differences must increase. Figure 18.2a shows a match between a hot stream and a cold stream that has a *CP* smaller than the hot stream. At the pinch, the match starts with a temperature difference equal to ΔT_{min} . The relative slopes of the temperature–enthalpy profiles of the two streams mean that the temperature differences become smaller moving away from the pinch, which is infeasible. On the other hand, Figure 18.2b shows a match involving the same hot stream but with a cold stream that has a larger *CP*. The relative slopes of the temperature–enthalpy profiles now cause the temperature differences to become larger moving away from the pinch, which is feasible. Thus, starting with ΔT_{min} at the pinch, for temperature differences to increase moving away from the pinch^{1,2}:

$$CP_H \leq CP_C \text{ (above pinch)} \quad (18.1)$$

Figure 18.3 shows the situation below the pinch at the pinch. If a cold stream is matched with a hot stream with smaller *CP*, as shown in Figure 18.3a (i.e. a steeper slope), then the temperature differences become smaller (which is infeasible). If the same cold stream is matched with a hot stream with a larger *CP* (i.e. a less steep slope), as shown in Figure 18.3b, then temperature differences become larger, which is feasible. Thus, starting with ΔT_{min} at the pinch, for temperature differences to increase moving away from the pinch^{1,2}:

$$CP_H \geq CP_C \text{ (below pinch)} \quad (18.2)$$

Note that the *CP* inequalities given by Equations 18.1 and 18.2 only apply at the pinch and when both ends of the match are at pinch conditions.

3. *The CP-table.* Identification of the essential matches in the region of the pinch is clarified by use of the *CP-table*^{1,2}.

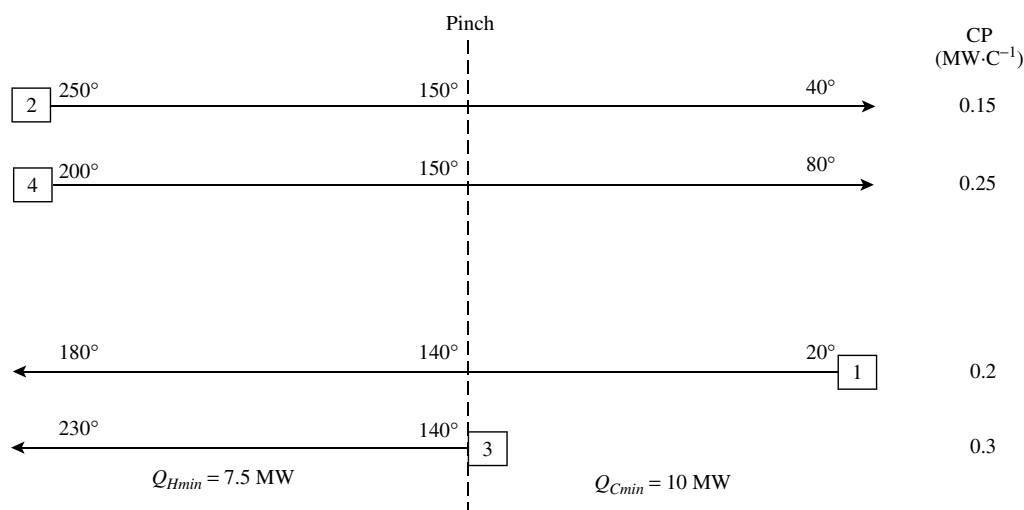


Figure 18.1 The grid diagram for the data from the Table 16.2.

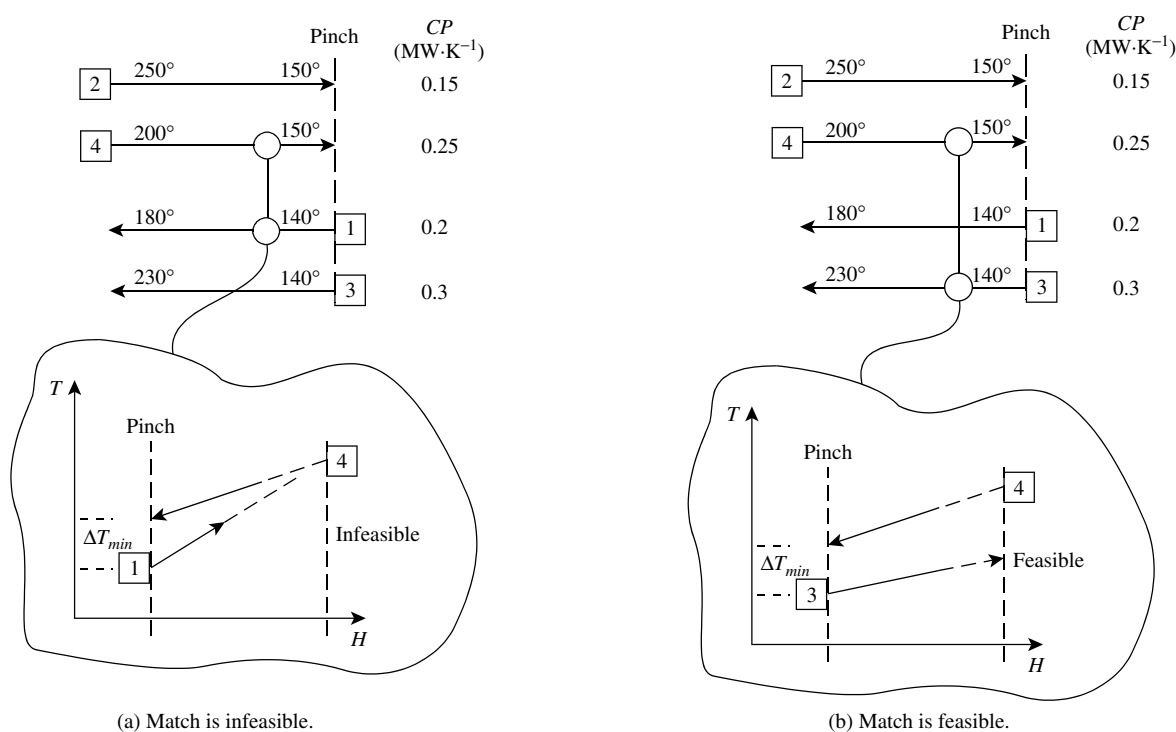


Figure 18.2 Criteria for the pinch matches above the pinch.

In the CP-table, the CP values of the hot and cold streams for the streams at the pinch are listed in descending order.

Figure 18.4a shows the grid diagram with CP-table for the design above the pinch. Cold utility must not be used above the pinch, which means that hot streams must be cooled to pinch temperature by heat recovery. Hot utility can be used, if necessary, on the cold streams above the pinch. Thus, it is essential to match hot streams above the pinch with a cold partner. In addition, if the hot stream is at pinch conditions, the cold stream it is to be matched with must also be at pinch conditions, otherwise the ΔT_{min}

constraint will be violated. Figure 18.4a shows a feasible design arrangement above the pinch that does not use temperature differences smaller than ΔT_{min} . Note again that the CP inequality only applies when a match is made between two streams that are both at the pinch. Away from the pinch, temperature differences increase, and it is no longer essential to obey the CP inequalities.

Figure 18.4b shows the grid diagram with CP-table for the design below the pinch. Hot utility must not be used below the pinch, which means that cold streams must be heated to pinch temperature by heat recovery. Cold utility

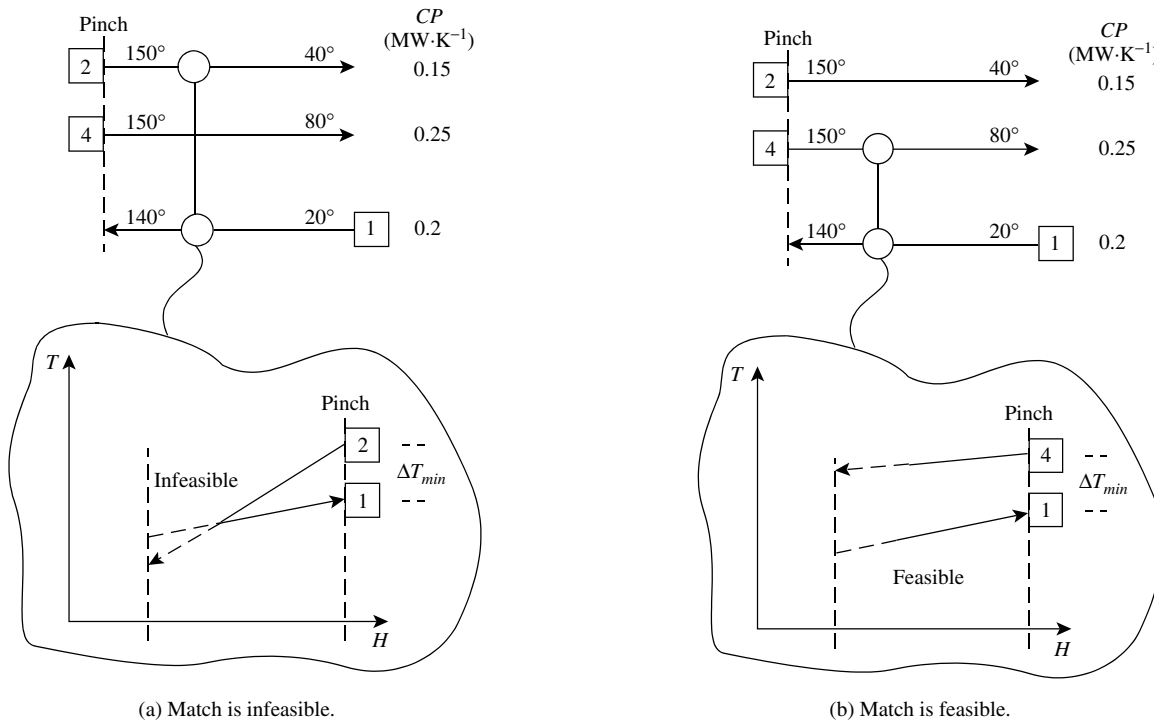


Figure 18.3 Criteria for pinch matches below the pinch.

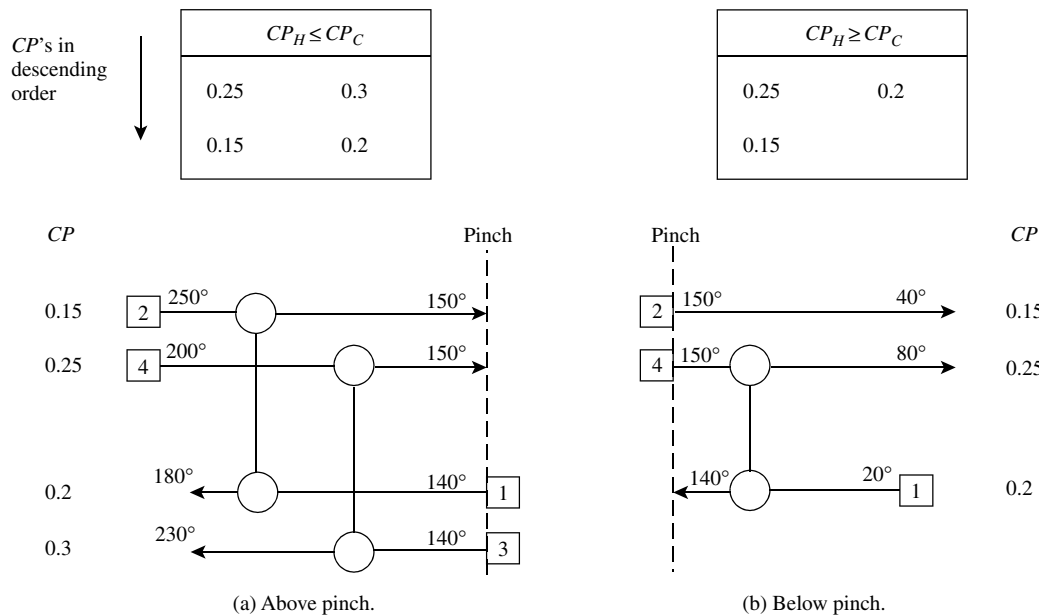


Figure 18.4 The CP table for the designs above and below the pinch for the problem from Table 16.2.

can be used, if necessary, on the hot streams below the pinch. Thus, it is essential to match cold streams below the pinch with a hot partner. In addition, if the cold stream is at pinch conditions, the hot stream it is to be matched with must also be at pinch conditions, otherwise the ΔT_{min} constraint will be violated. Figure 18.4b shows a design arrangement below the pinch that does not use temperature differences smaller than ΔT_{min} .

Having decided that some essential matches need to be made around the pinch, the next question is how big should the matches be?

4. The “tick-off” heuristic. Once the matches around the pinch have been chosen to satisfy the criteria for minimum energy, the design should be continued in such a manner as to keep capital costs to a minimum. One important criterion

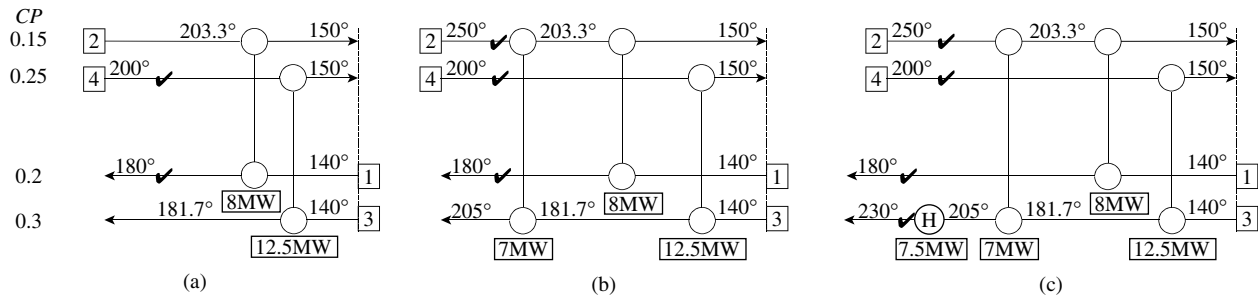


Figure 18.5 Sizing the units above the pinch using the tick-off heuristic.

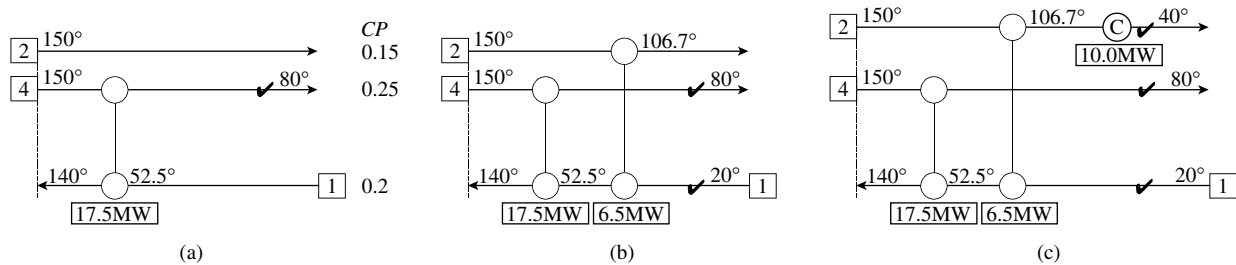


Figure 18.6 Sizing the units below the pinch using the tick-off heuristic.

in the capital cost is the number of units (there are others, of course, which shall be addressed later). Keeping the number of units to a minimum can be achieved using the *tick-off heuristic*. To tick off a stream, individual units are made as large as possible, that is, the smaller of the two heat duties on the streams being matched.

Figure 18.5a shows the matches around the pinch from Figure 18.4a with their duties maximized to tick off streams. It should be emphasized that the tick-off heuristic is only a heuristic and can occasionally penalize the design. Methods will be developed later, which allow such penalties to be identified as the design proceeds.

The design in Figure 18.5a can now be completed by satisfying the heating and cooling duties away from the pinch. Cooling water must not be used above the pinch. Therefore, if there are hot streams above the pinch for which the pinch matches do not satisfy the duties, additional process-to-process heat recovery is required. Figure 18.5b shows an additional match to satisfy the residual cooling of the hot streams above the pinch. Again, the duty on the unit is maximized. Finally, above the pinch, the residual heating duty on the cold streams must be satisfied. Since there are no hot streams left above the pinch, hot utility must be used as shown in Figure 18.5c.

Turning now to the cold end design, Figure 18.6a shows the pinch design with the streams ticked off. If there are any cold streams below the pinch for which the pinch matches do not satisfy the duties, then additional process-to-process heat recovery is required, since hot utility must not be used. Figure 18.6b shows an additional match to satisfy the residual heating of the cold streams below the pinch. Again, the duty on the unit is maximized. Finally, below

the pinch, the residual cooling duty on the hot streams must be satisfied. Since there are no cold streams left below the pinch, cold utility must be used (Figure 18.6c).

The final design shown in Figure 18.7 amalgamates the hot end design from Figure 18.5c and cold end design from Figure 18.6c. The duty on hot utility of 7.5 MW agrees with Q_{Hmin} and the duty on cold utility of 10.0 MW agrees with Q_{Cmin} predicted by the composite curves and the problem table algorithm.

Note one further point from Figure 18.7. The number of units is 7 in total (including the heater and cooler). Referring back to Example 17.1, the target for the minimum number of units was predicted to be 7. It therefore appears that there was something in the procedure that naturally steered the design to achieve the target for the minimum number of units.

It is in fact the tick-off heuristic that steered the design toward the minimum number of units^{1,2}. The target for the minimum number of units was given by Equation 17.2:

$$N_{UNITS} = S - 1 \quad (17.2)$$

Before any matches are placed, the target indicates that the number of units required is equal to the number of streams (including utilities) minus one. The tick-off heuristic satisfied the heat duty on one stream every time one of the units was used. The stream that has been ticked off is no longer part of the remaining design problem. The tick-off heuristic ensures that having placed a unit (and used up one of the available units), a stream is removed from the problem. Thus, Equation 17.2 is satisfied if every match satisfies the heat duty on a stream or a utility.

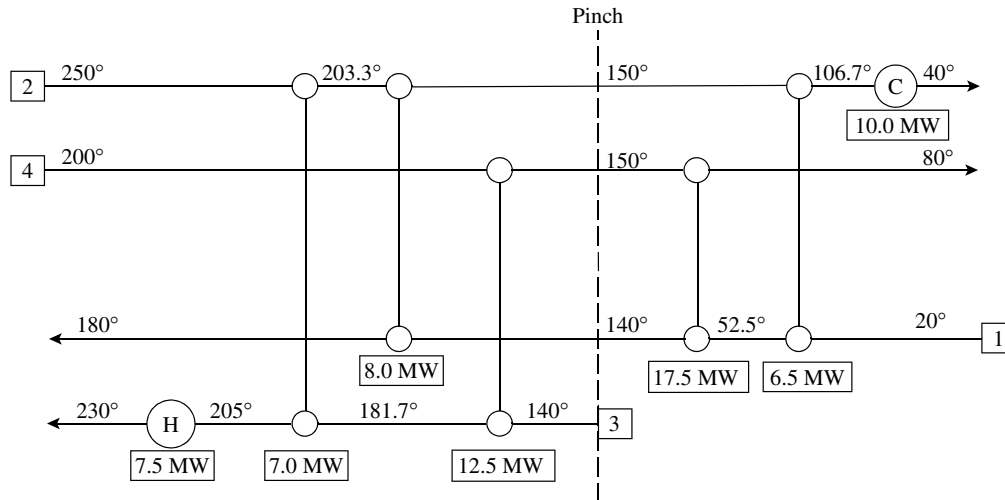


Figure 18.7 The completed design for the data from Table 16.2.

This design procedure is known as the *pinch design method* and can be summarized in five steps¹.

- Divide the problem at the pinch into separate problems.
- The design for the separate problems is started at the pinch, moving away.
- Temperature feasibility requires constraints on the CP values to be satisfied for matches between the streams at the pinch.
- The loads on individual units are determined using the tick-off heuristic to minimize the number of units. Occasionally, the heuristic causes problems.
- Away from the pinch, there is usually more freedom in the choice of matches. In this case, the designer can discriminate on the basis of operability, plant layout and so on.

Example 18.1 The process stream data for a heat recovery network problem are given in Table 18.1.

Table 18.1 Stream data for Example 18.1.

Stream No.	Type	Supply temperature (°C)	Target temperature (°C)	Heat capacity flowrate (MW·K ⁻¹)
1	Hot	400	60	0.3
2	Hot	210	40	0.5
3	Cold	20	160	0.4
4	Cold	100	300	0.6

A problem table analysis on this data reveals that the minimum hot utility requirement for the process is 15 MW and the minimum cold utility requirement is 26 MW for a minimum allowable temperature difference of 20°C. The analysis also reveals that the pinch is located at a temperature of 120°C for hot streams and 100°C for cold streams. Design a heat exchanger network for maximum energy recovery featuring the minimum number of units.

Solution Figure 18.8a shows the hot end design with the CP-table. Above the pinch, adjacent to the pinch, $CP_H \leq CP_C$. The duty on the units has been maximized according to the tick-off heuristic.

Figure 18.8b shows the cold end design with the CP-table. Below the pinch, adjacent to the pinch, $CP_H \geq CP_C$. Again the duty on units has been maximized according to the tick-off heuristic.

The completed design is shown in Figure 18.8c. The minimum number of units for this problem is given by

$$\begin{aligned}
 N_{\text{UNITS}} &= (S - 1)_{\text{ABOVE PINCH}} + (S - 1)_{\text{BELOW PINCH}} \\
 &= (5 - 1) + (4 - 1) \\
 &= 7
 \end{aligned}$$

The design in Figure 18.8 is seen to achieve the minimum number of units target.

The pinch design method, as discussed so far, has assumed the same $\Delta T_{\min}$ applied between all stream matches. In Chapter 16, it was discussed how the basic targeting methods for the composite curves and the problem table algorithm can be modified to allow stream-specific values of $\Delta T_{\min}$. The example was quoted in which liquid streams were required to have a $\Delta T_{\min}$ contribution of 5°C and gas streams a $\Delta T_{\min}$ contribution of 10°C. For liquid–liquid matches, this would lead to a $\Delta T_{\min} = 10^\circ\text{C}$. For gas–gas matches, this would lead to a $\Delta T_{\min} = 20^\circ\text{C}$. For liquid–gas matches, it will lead to a $\Delta T_{\min} = 15^\circ\text{C}$ ². Modifying the problem table and the composite curves to account for these stream-specific values of $\Delta T_{\min}$ is straightforward. But how is the pinch design method modified to take account of such $\Delta T_{\min}$ contributions? Figure 18.9 illustrates the approach. Suppose the interval pinch temperature from the problem table is 120°C. This would correspond with hot stream pinch temperatures of 125°C and 130°C for hot streams with $\Delta T_{\min}$ contributions of 5°C and 10°C respectively. For

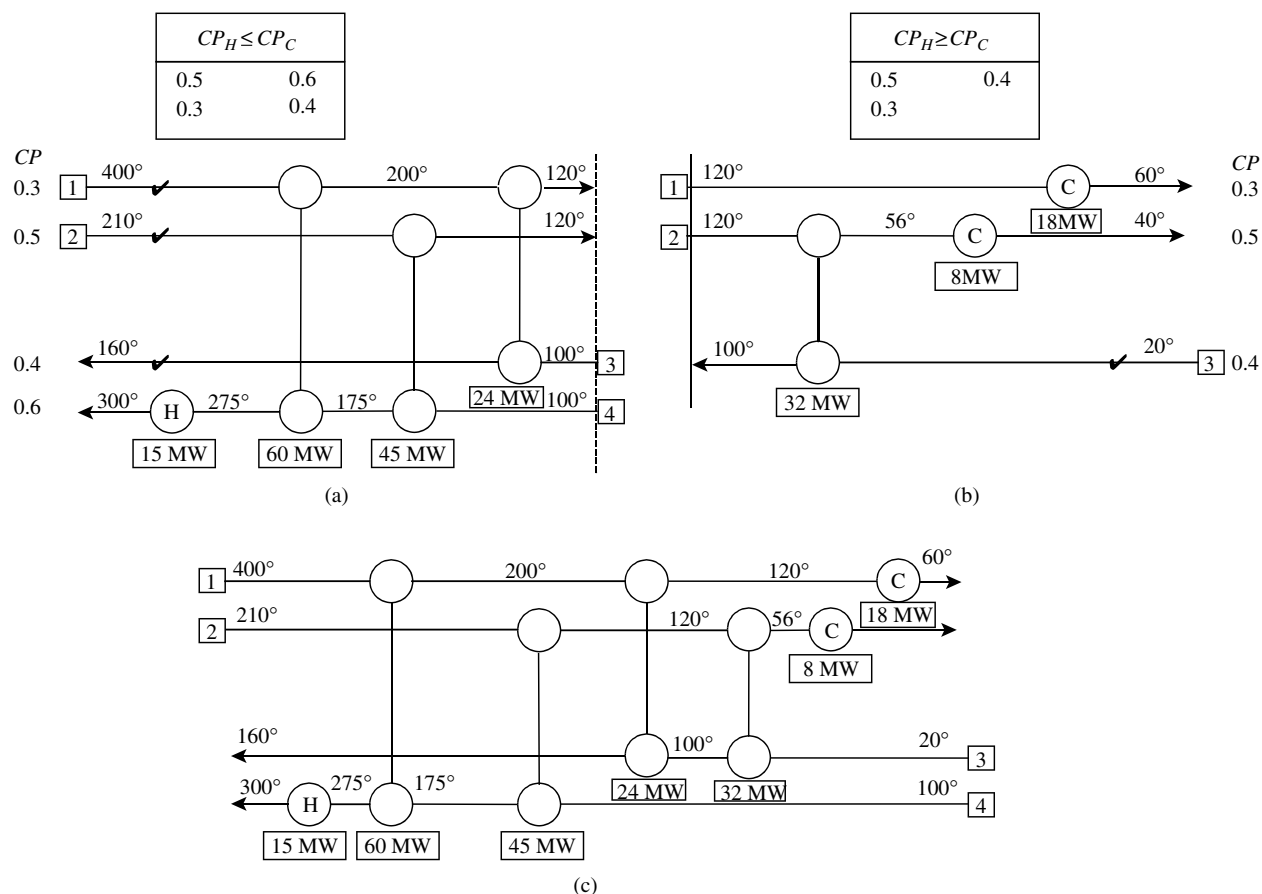


Figure 18.8 Maximum energy recovery design for Example 18.1.

an interval pinch temperature of 120°C, the corresponding cold stream pinch temperatures would be 115°C and 110°C for cold streams with ΔT_{min} contributions of 5°C and 10°C respectively. The stream grid is set up as shown in Figure 18.9 with the pinch matches being made and given their appropriate value of ΔT_{min} depending on the streams being matched. The constraints regarding the CP inequalities apply in this case, just as the case with a global value of ΔT_{min} .

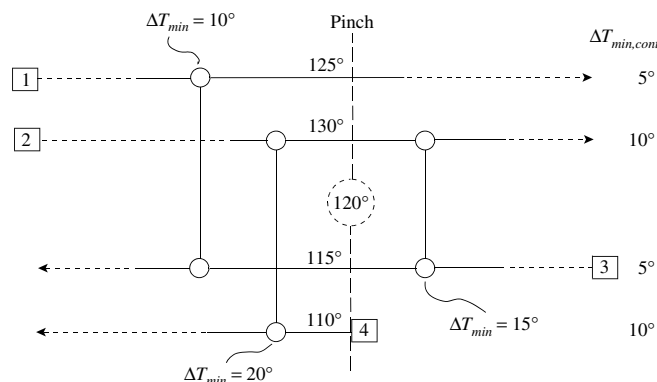


Figure 18.9 The design grid for a problem stream-specific with ΔT_{min} contributions.

18.2 DESIGN FOR THRESHOLD PROBLEMS

In Section 16.3, it was discussed that some problems, known as threshold problems, do not have a pinch. They need either hot utility or cold utility but not both. How should the approach be modified to deal with the design of threshold problems?

The philosophy in the pinch design method was to start the design where it was most constrained. If the design is pinched, the problem is most constrained at the pinch. If there is no pinch, where is the design most constrained? Figure 18.10a shows one typical threshold problem that requires no hot utility, just cold utility. The most constrained part of this problem is the no-utility end². This is where temperature differences are smallest and there may be constraints, as shown in Figure 18.10b, where the target temperatures on some of the hot streams can only be satisfied by specific matches. Also, if individual matches are required to have a temperature difference no smaller than the threshold ΔT_{min} , the CP inequalities described in the pinch design method must be applied. For the most part, problems similar to those in Figure 18.10a are treated as one half of a pinched problem.

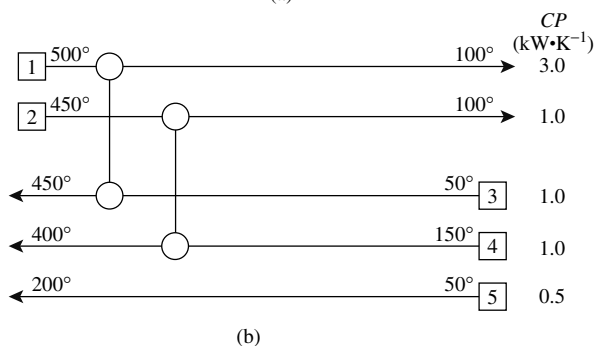
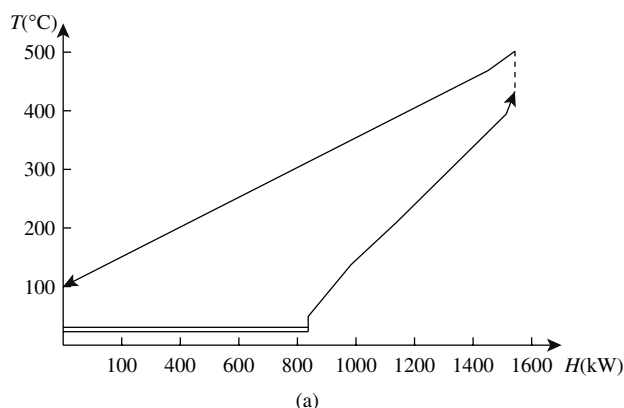


Figure 18.10 Even though threshold problems have large driving forces, there are still often essential matches to be made, especially at the no-utility end.

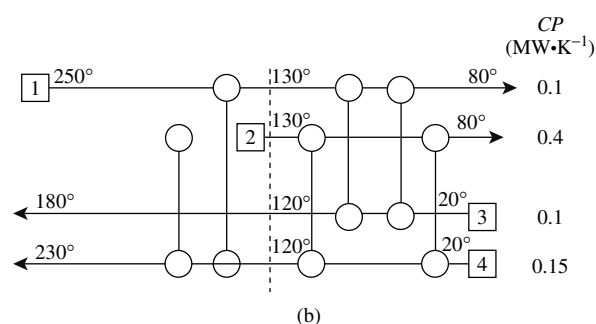
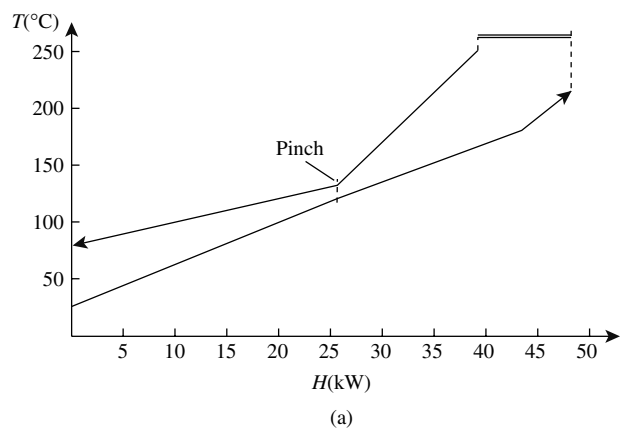


Figure 18.11 Some threshold problems must be treated as pinched problems requiring essential matches at both the no utility end and the pinch.

Figure 18.11 shows another threshold problem that requires only hot utility. This problem is different in characteristic from the one in Figure 18.10. Now the minimum temperature difference is in the middle of the problem causing a pseudo-pinch. The best strategy to deal with this type of threshold problem is to treat it as a pinched problem. In Figure 18.11, the problem is divided into two parts at the pseudo-pinch, and the pinch design method followed. The only complication in applying the pinch design method for such problems is that one half of the problem (the cold end in Figure 18.11) will not feature the flexibility offered by matching against utility.

18.3 STREAM SPLITTING

The pinch design method developed earlier followed several rules and guidelines to allow design for minimum utility (or maximum energy recovery) in the minimum number of units. Occasionally, it appears not to be possible to create the appropriate matches because one or other of the design criteria cannot be satisfied.

Consider Figure 18.12a that shows the above-pinch part of a design. Cold utility must not be used above the pinch, which means that all hot streams must be cooled to pinch temperature by heat recovery. There are three hot streams

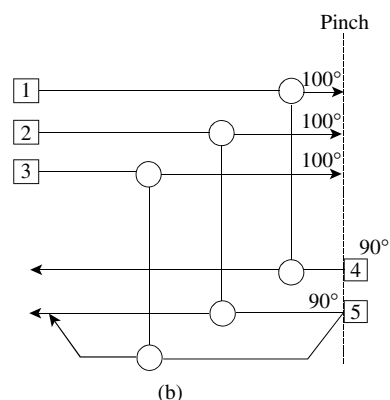
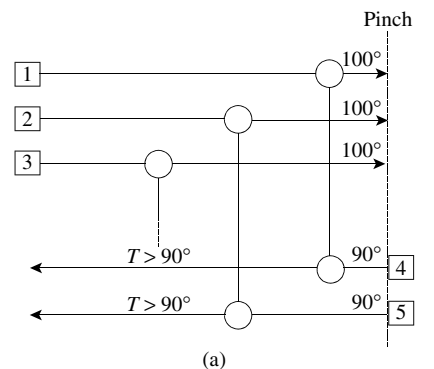


Figure 18.12 If the number of hot streams at the pinch, above the pinch, is greater than the number of cold streams, then stream splitting of the cold streams is required.

and two cold streams in Figure 18.12a. Thus, regardless of the CP values of the streams, one of the hot streams cannot be cooled to pinch temperature without some violation of the ΔT_{min} constraint. The problem can only be resolved by splitting a cold stream into two parallel branches as shown in Figure 18.12b. Now each hot stream has a cold partner with which to match, capable of cooling it to pinch temperature. Thus, in addition to the CP inequality criteria introduced earlier, there is a stream number criterion above the pinch such that^{1,2}:

$$S_H \leq S_C \text{ (above pinch)} \quad (18.3)$$

where S_H = number of hot streams at the pinch (including branches)

S_C = number of cold streams at the pinch (including branches)

If there had been a greater number of cold streams than hot streams in the design above the pinch, this would not have created a problem, since hot utility can be used above the pinch.

By contrast, now consider part of a design below the pinch, as shown in Figure 18.13a. Here hot utility must not be used, which means that all cold streams must be heated to pinch temperature by heat recovery. There are now three

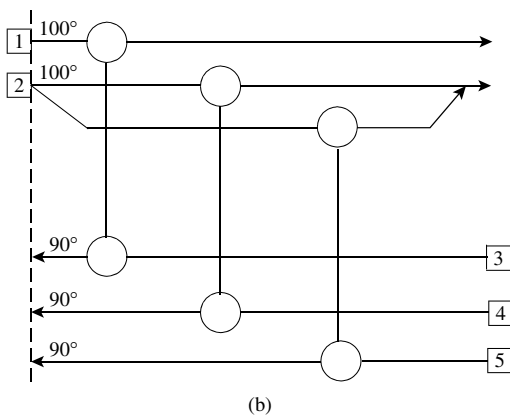
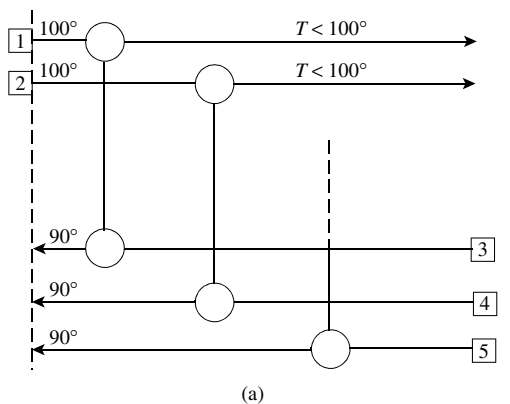


Figure 18.13 If the number of cold streams below the pinch, at the pinch, is greater than the pinch number of hot streams, then stream splitting of the hot stream is required.

cold streams and two hot streams in Figure 18.13a. Again, regardless of CP values, one of the cold streams cannot be heated to pinch temperature without some violation of the ΔT_{min} constraint. The problem can only be resolved by splitting a hot stream into two parallel branches, as shown in Figure 18.13b. Now each cold stream has a hot partner with which to match and capable of heating it to pinch temperature. Thus there is a stream number criterion below the pinch, such that^{1,2}:

$$S_H \geq S_C \text{ (below pinch)} \quad (18.4)$$

Had there been more hot streams than cold below the pinch, this would not have created a problem since cold utility can be used below the pinch.

It is not only the number of streams that creates the need to split streams at the pinch. Sometimes the CP inequality criteria, Equations 18.1 and 18.2, cannot be met at the pinch without a stream split. Consider the above-pinch part of a problem in Figure 18.14a. The number of hot streams is less than the number of cold streams, and hence Equation 18.3 is satisfied. However, the CP inequality, Equation 18.1, must be satisfied. Neither of the two cold streams has a large enough CP . The hot stream can be made smaller by splitting it into two parallel branches (Figure 18.14b).

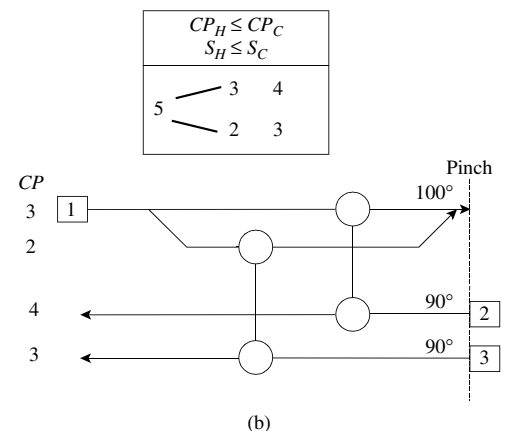
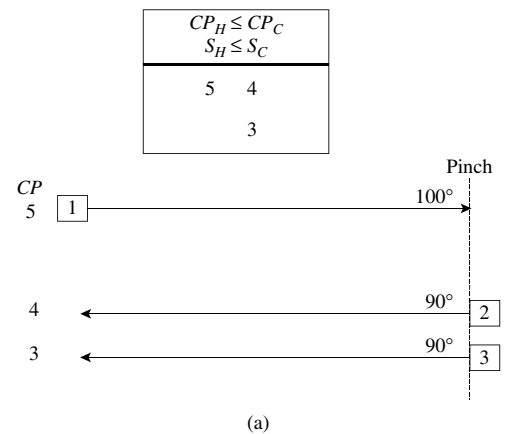


Figure 18.14 The CP inequality rules can necessitate stream splitting above the pinch.

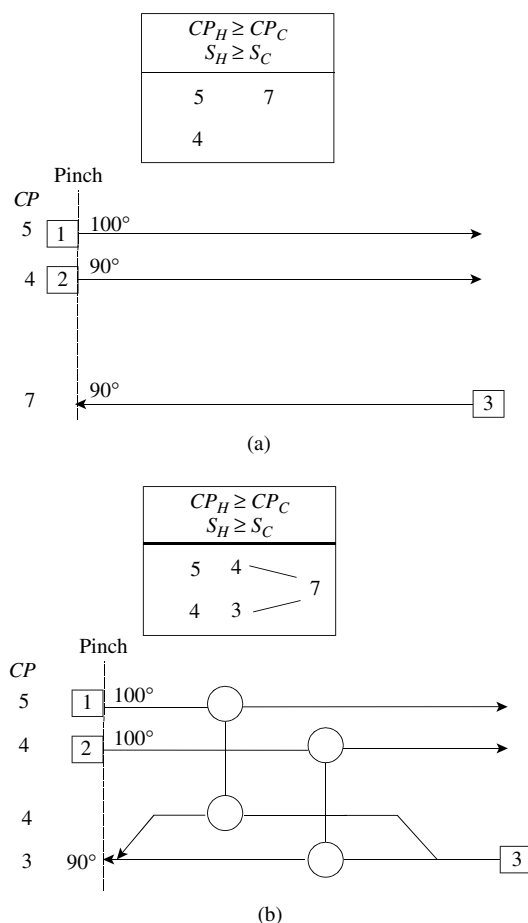


Figure 18.15 The CP in equality rules can necessitate stream splitting below the pinch.

Figure 18.15a shows the below-pinch part of a problem. The number of hot streams is greater than the number of cold streams, and hence Equation 18.4 is satisfied. However, neither of the two hot streams has a large enough CP to satisfy the CP inequality, Equation 18.2. The cold stream can be made smaller by splitting it into two parallel branches (Figure 18.14b).

Clearly, in designs different from those in Figures 18.14 and 18.15, when streams are split to satisfy the CP inequality, this might create a problem with the number of streams at the pinch such that Equations 18.3 and 18.4 are no longer satisfied. This would then require further stream splits to satisfy the stream number criterion. Figures 18.16a and 18.16b present algorithms for the overall approach^{1,2}.

One further important point needs to be made regarding stream splitting. In Figure 18.14, the hot stream is split into two branches with CP values of 3 and 2 to satisfy the CP inequality criteria. However, a different split could have been chosen. For example, the split could have been into branch CP values of 4 and 1, or 2.5 and 2.5, or 2 and 3 (or any setting between 4 and 1, and 2 and 3). Each of these would also have satisfied the CP inequalities. Thus,

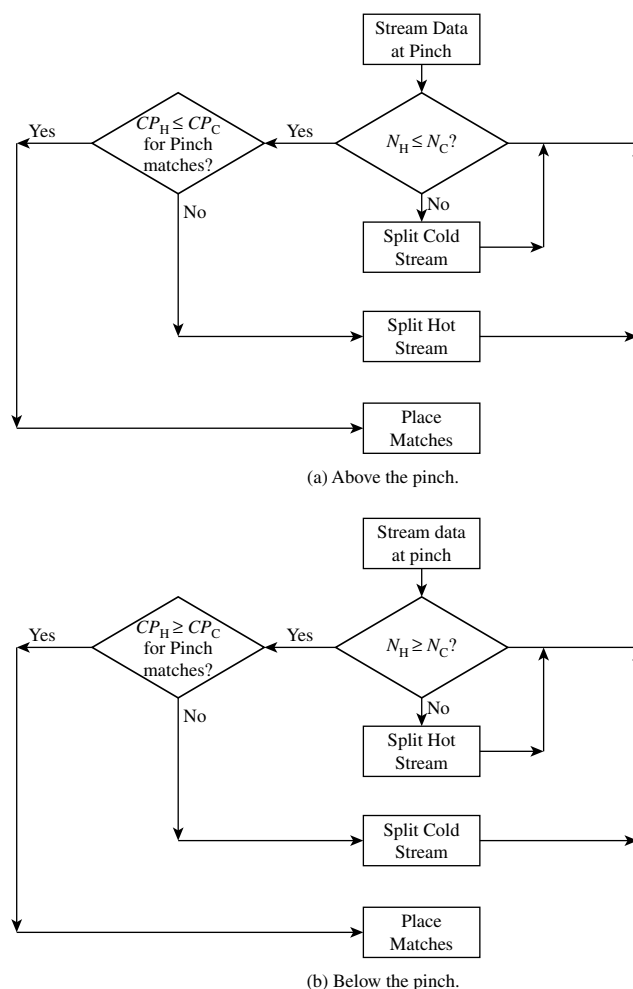


Figure 18.16 Stream splitting algorithms.

there is a degree of freedom in the design to choose the branch flowrates. By fixing the heat duties on the two units in Figure 18.14b and changing the branch flowrates, the temperature differences across each unit are changed. The best choice can only be made by sizing and costing the various units in the completed network for different branch flowrates. This is an important degree of freedom when the network is optimized. Similar arguments could be made regarding the cold end design in Figure 18.15b.

Example 18.2 A problem table analysis for part of a high-temperature process reveals that for $\Delta T_{min} = 20^\circ\text{C}$ the process requires 9.2 MW of hot utility, 6.4 MW of cold utility and the pinch is located at 520°C for hot streams and 500°C for cold streams. The process stream data are given in Table 18.2. Design a heat exchanger network for maximum energy recovery that features the minimum number of units.

Solution Figure 18.17a shows the stream grid with the CP-tables for the above- and below-pinch designs. Following the algorithms in Figure 18.16, a hot stream must be split above the pinch to satisfy the CP inequality, as shown in Figure 18.17b.

Table 18.2 Stream data.

Stream No.	Type	Supply Temp. (°C)	Target Temp. (°C)	Heat Capacity Flowrate (MW·K ⁻¹)
1	Hot	720	320	0.045
2	Hot	520	220	0.04
3	Cold	300	900	0.043
4	Cold	200	550	0.02

Thereafter the design is straightforward, and the final design is shown in Figure 18.17c.

The target for the minimum number of units is given by:

$$\begin{aligned}
 N_{UNITS} &= (S - 1)_{ABOVE\ PINCH} + (S - 1)_{BELOW\ PINCH} \\
 &= (4 - 1) + (5 - 1) \\
 &= 7
 \end{aligned}$$

The design in Figure 18.17c is seen to achieve the minimum units target.

18.4 DESIGN FOR MULTIPLE PINCHES

In Chapter 16, it was discussed as to how the use of multiple utilities could give rise to multiple pinches. For example, the design for the process in Figure 16.2 could have used either a single level of hot utility or two steam levels (Figure 16.26a). The targeting indicated that instead of using 7.5 MW of high-pressure steam at 240°C, 3 MW of this could be substituted with low-pressure steam at 180°C. Where the low-pressure steam touches the grand composite curve in Figure 16.26a results in a utility pinch. Figure 18.18a shows the grid diagram when two steam levels are used with the utility pinch dividing the process into three parts.

Following the pinch rules, heat should not be transferred across either the process pinch or the utility pinch by process-to-process heat exchange. Also, there must be no inappropriate use of utilities. This means that above the utility pinch shown in Figure 18.18a, high-pressure steam should be used and no low-pressure steam or cooling water. Between the utility pinch, and the process pinch low-pressure steam should be used and no high-pressure steam or cooling water. Below the process pinch in Figure 18.18 only cooling water should be used. The appropriate utility streams have been included with the process streams in Figure 18.18.

The network can now be designed using the pinch design method^{1,2}. The philosophy of the pinch design method is to start at the pinch, and move away. At the pinch, the rules for the CP inequality and the number of streams must be obeyed. Above the utility pinch in Figure 18.18, and below the process pinch in Figure 18.18 there is clearly no problem in applying this philosophy. However, between the two pinches there is a problem, since designing away from both pinches could lead to a clash.

More careful examination of Figure 18.18a reveals that, between the two pinches, one is more constrained than the other. Below the utility pinch, $CP_H \geq CP_C$ is required and low-pressure steam is available as a hot stream with an extremely large CP . In fact, if steam is assumed to condense or vaporize isothermally, it will have a CP that is infinite. Thus, following the philosophy of starting the design in the most constrained region, the design between the pinches in Figure 18.18a should be started at the most constrained pinch, which is the process pinch.

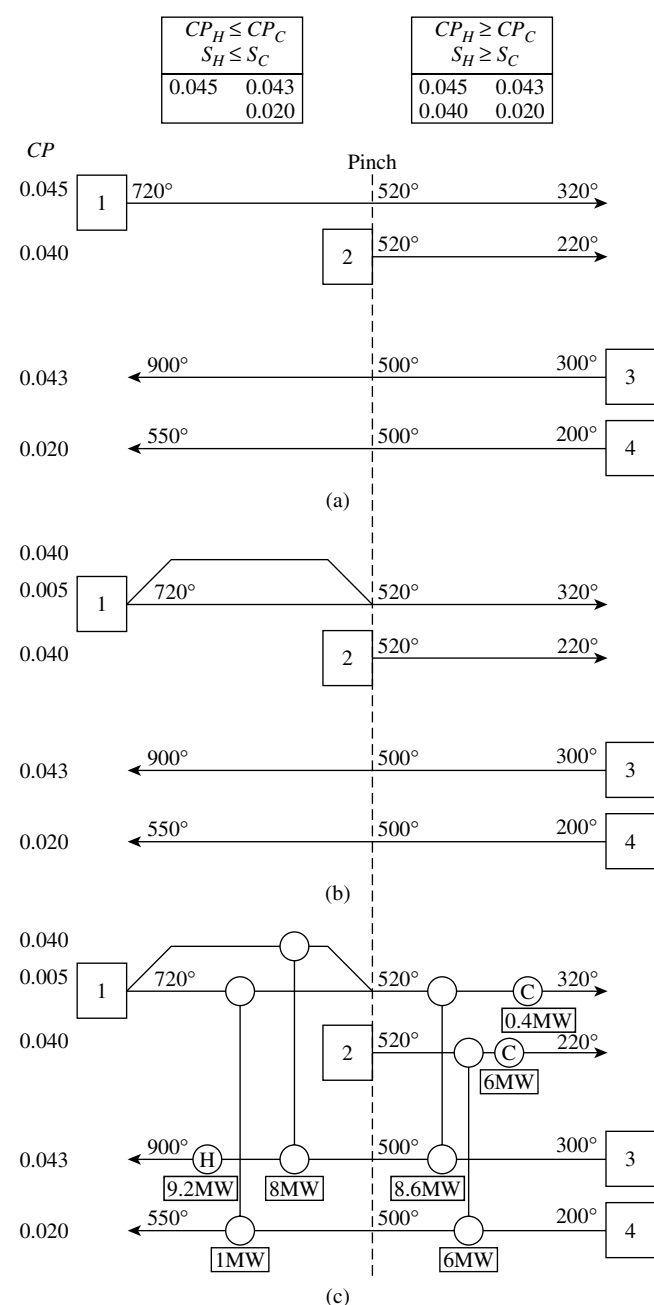


Figure 18.17 Maximum energy recovery design for Example 18.2.

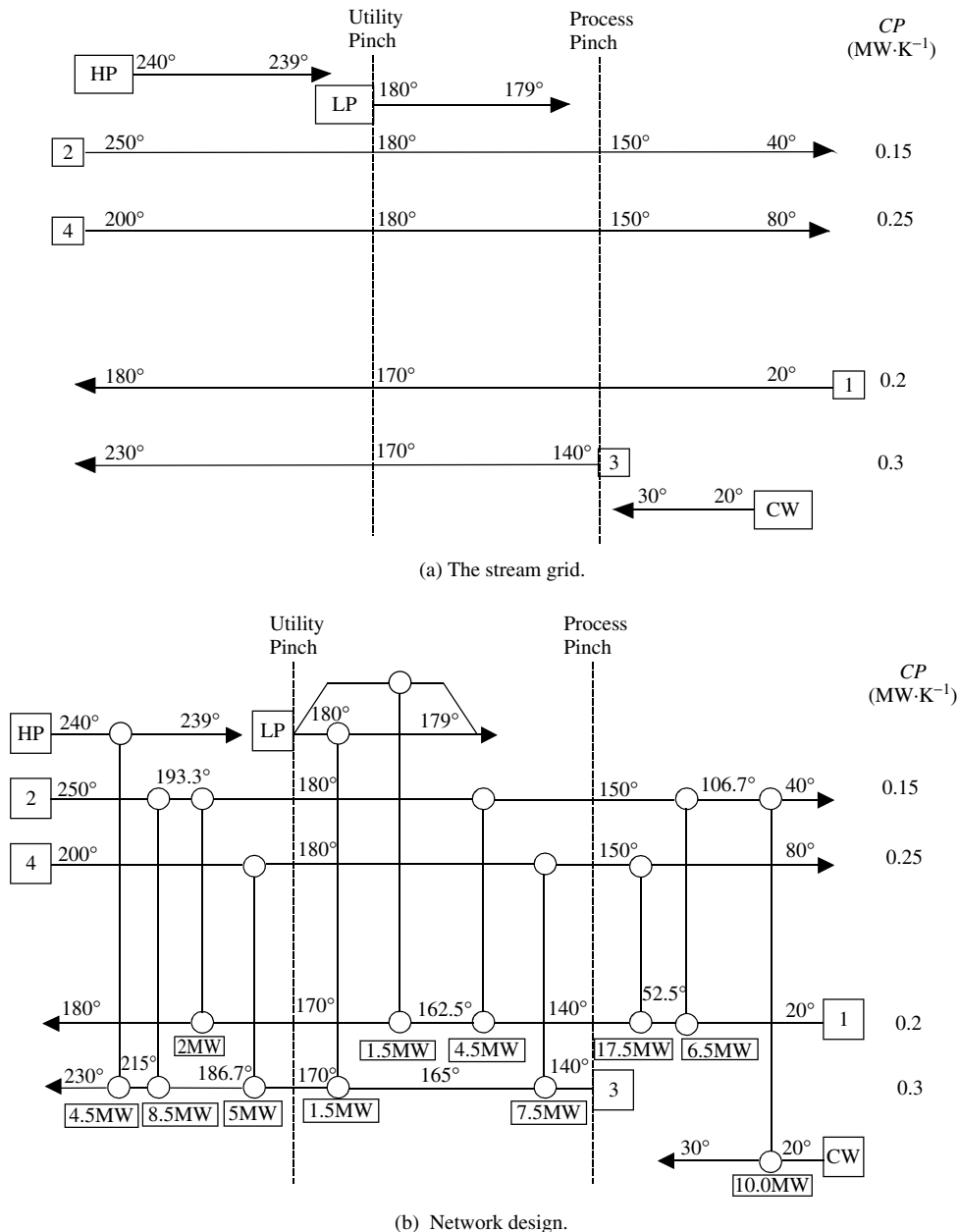


Figure 18.18 Network design for the process from Figure 16.2 using two steam levels.

Following this approach, the design is straightforward and the final design is shown in Figure 18.18b. It achieves the target set in Example 16.3 and in the minimum number of units. Remember that, in this case, to calculate the minimum number of units, the stream count must be performed separately in the three parts of the problem. Note that the stream split on the low-pressure steam in Figure 18.18b is not strictly necessary, but is made for practical reasons. Without the stream split, steam would have to partially condense in one unit and the steam-condensate mixture transferred to the next unit. The stream split allows two conventional steam heaters on low-pressure steam to be used. It is clear from Figure 18.18b that the

use of two steam levels has increased the complexity of the design considerably. However, the complexity of the design can be reduced later when the structure is subjected to optimization. The optimization can remove units that are uneconomic.

It is rare for there to be two process pinches in a problem. Multiple pinches usually arise from the introduction of additional utilities causing utility pinches. However, cases such as that shown in Figure 18.19 are not uncommon, where there is strictly speaking, only one pinch (one place where ΔT_{min} occurs), but there is a near-pinch. This near-pinch is a point in the process where the temperature difference becomes small enough to be effectively another pinch, even

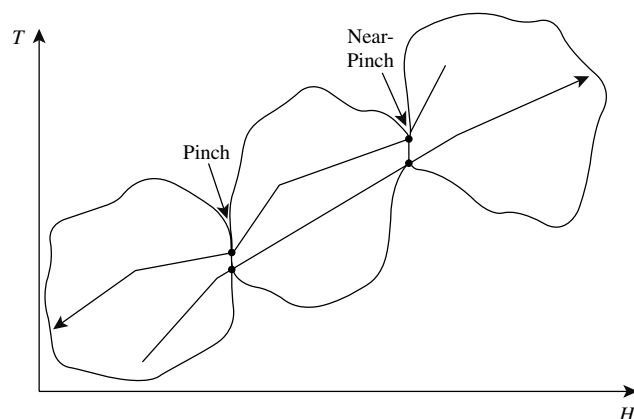


Figure 18.19 A near-pinch might require the design to be treated as if it had multiple pinches.

though the temperature difference is slightly larger than ΔT_{min} . Because the region around the near-pinch will be almost as constrained as the pinch, the best strategy is often to treat the near-pinch as if it was another pinch and divide the problem into three parts as shown in Figure 18.19. The initial design would therefore avoid heat transfer across the near-pinch and the process pinch and use hot utility only above the near-pinch and cold utility only below the pinch. The designer can then exploit the small amount of freedom around the near-pinch. Some heat transfer across the near-pinch is possible without causing an energy penalty. Exploiting this freedom allows the design to be simplified slightly.

Example 18.3 The stream data for a process are given in Table 18.3:

Table 18.3 Stream data.

Stream No.	Type	Supply temperature (°C)	Target temperature (°C)	Heat capacity flowrate (MW·K ⁻¹)
1	Hot	635	155	0.044
2	Cold	10	615	0.023
3	Cold	85	250	0.020
4	Cold	250	615	0.020

It has been decided to integrate a gas turbine (GT) exhaust with the process. The exhaust temperature of the GT is 400°C with $CP = 0.05 \text{ MW} \cdot \text{K}^{-1}$. Ambient temperature is 10°C.

- Calculate the problem table cascade for $\Delta T_{min} = 20^\circ\text{C}$.
- Saturated steam is to be generated by the process at a high-pressure level of 250°C and low-pressure level of 140°C, each from saturated boiler feedwater. The generation of the higher-pressure steam is to be maximized. How much steam can be generated at the two levels assuming boiler feedwater and final steam condition are both saturated?
- Design a network for maximum energy recovery for $\Delta T_{min} = 20^\circ\text{C}$ that generates steam at these two levels.
- What is the residual cooling demand?

Solution

- The problem table cascade is shown in Table 18.4 for $\Delta T_{min} = 20^\circ\text{C}$.

Table 18.4 Problem table cascade.

Interval temperature (°C)	Cascade heat flow (MW)
625	0
390	0.235
260	6.865
145	12.73
95	13.08
20	15.105
0	16.105

- For high-pressure steam, $T^* = 260^\circ\text{C}$, for low-pressure steam, $T^* = 150^\circ\text{C}$. Figure 18.20 shows the grand composite curve plotted from the problem table cascade. The two levels of steam generation are shown.

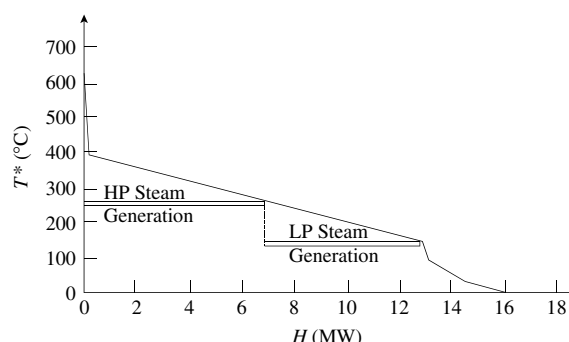


Figure 18.20 Grand composite curve for Example 18.3 showing two levels of steam generation.

Duty on high-pressure steam generation

$$= 6.865 \text{ MW}$$

By interpolation from the problem table cascade, heat flow at $T^* = 150^\circ\text{C}$

$$= 6.865 + \frac{(260 - 150)}{(260 - 145)} \times (12.73 - 6.865)$$

$$= 12.475 \text{ MW}$$

Duty on low-pressure steam generation

$$= 12.475 - 6.865$$

$$= 5.61 \text{ MW}$$

- The use of two levels of steam generation in Figure 18.20 creates two utility pinches. Thus, the stream grid needs to be divided into three parts. Figure 18.21 shows the final design, which achieves the targets set for both high-pressure (HP) and low-pressure (LP) steam generation. In Figure 18.21 above the HP pinch, the CP of Stream 1 and the GT stream are too large

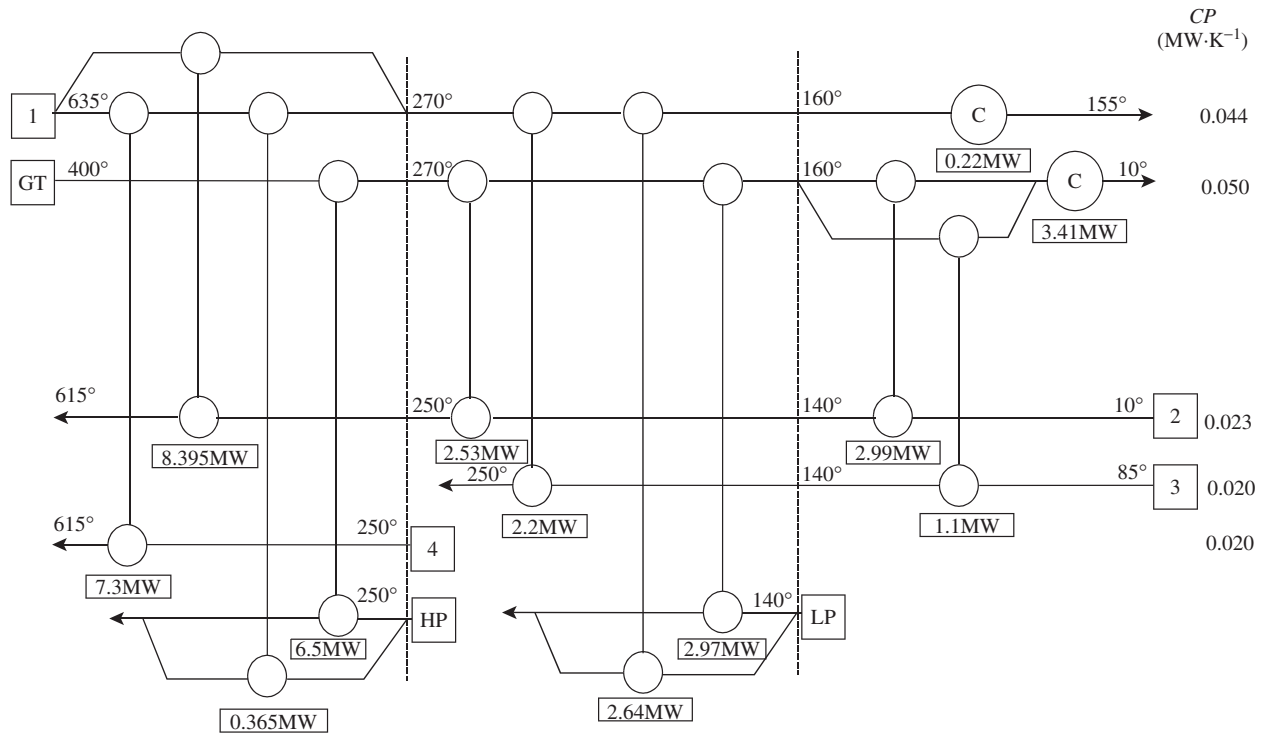


Figure 18.21 Network design for Example 18.3.

to match directly against Streams 3 and 4. This is overcome in Figure 18.21 by splitting Stream 1 and exploiting the infinite CP of the HP steam generation. Between the pinches, the design is started below the HP pinch and developed toward the LP pinch, where the infinite CP of the LP steam generation can be exploited to satisfy the CP inequalities. Below the LP pinch, although the CP inequalities can be satisfied by direct matches, the heat duty on Stream 1 is small compared with the other streams. If Stream 1 did not exist below the LP pinch, then this would call for the GT stream to be split. This has been done in Figure 18.21 because of the small duty on Stream 1. Although the steam generation for both high-pressure and low-pressure steam are shown with steam splits in Figure 18.21, in practice these units would be individual steam generators, each fed with boiler feedwater. Also, the stream split for the GT exhaust could be accommodated by splitting the gas turbine flow into two ducts, or by placing two sets of tubes in the same GT exhaust located in parallel. The design in Figure 18.21 has significant scope for simplification, but at the penalty of reduced energy efficiency. Such trade-offs are at the discretion of the designer.

- d. There is a cooling demand of 0.22 MW on Stream 1 that needs to be satisfied by cold utility and cooling demand of 3.41 MW required by the GT exhaust. The cooling of the GT exhaust is satisfied by simply venting it to atmosphere after heat recovery has been completed.

18.5 REMAINING PROBLEM ANALYSIS

The considerations addressed so far in network design have been restricted to those of energy performance and

number of units. In addition, the problems have all been straightforward to design for maximum energy recovery in the minimum number of units by ticking-off streams. Not all problems are so straightforward. Also, heat transfer area, number of shells when using 1–2 shells, capital cost and so on should be considered when placing matches. Here, a more sophisticated approach is needed³.

When a match is placed, the duty needs to be chosen with some quantitative assessment of the match in the context of the whole network, without having to complete the network. This can be done by exploiting the powers of targeting using a technique known as *remaining problem analysis*.

Consider first design for minimum energy in a more complex problem than has so far been addressed. If a problem table analysis is performed on the stream data, Q_{Hmin} and Q_{Cmin} can be calculated. When the network is designed and a match placed, it would be useful to assess whether there will be any energy penalty caused by some feature of the match without having to complete the design. Whether there will be a penalty can be determined by performing a problem table analysis on the *remaining problem*. The problem table analysis is simply repeated on the stream data, leaving out those parts of the hot and cold stream satisfied by the match. One of the two results would then occur:

1. The algorithm may calculate Q_{Hmin} and Q_{Cmin} to be unchanged. In this case, the designer knows that the match will not penalize the design in terms of increased utility usage.

2. The algorithm may calculate an increase in Q_{Hmin} and Q_{Cmin} . This means that the match is transferring heat across the pinch or that there is some feature of the design that will cause cross-pinch heat transfer if the design was completed. If the match is not transferring heat across the pinch directly, then the increase in utility will result from the match being too big as a result of the tick-off heuristic.

The remaining problem analysis technique can be applied to any feature of the network that can be targeted, such as a minimum area. In Chapter 17, the approach to targeting for heat transfer area (Equation 17.6) was based on vertical heat transfer from the hot composite curve to the cold composite curve. If heat transfer coefficients do not vary significantly, this model predicts the minimum area requirements adequately for most purposes³. Thus, if heat transfer coefficients do not vary significantly, then the matches created in the design should come as close as possible to the conditions that would correspond with vertical transfer between the composite curves. Remaining problem analysis can be used to approach the area target, as closely as a practical design permits, using a minimum (or near-minimum) number of units. Suppose a match is placed, then its area requirement can be calculated. A remaining problem analysis can be carried out by calculating the area target for the stream data, leaving out those parts of the data satisfied by the match. The area of the match is now added to the area target for the remaining problem. Subtraction of the original area target for the whole-stream data $A_{NETWORK}$ gives the area penalty incurred.

If heat transfer coefficients vary significantly, then the vertical heat transfer model adopted in Equation 17.6 predicts a network area that is higher than the true minimum, as illustrated in Figure 17.4. Under these circumstances, a careful pattern of nonvertical matching is required to approach the minimum network area. However, the remaining problem analysis approach can still be used to steer the design toward a minimum area under these circumstances. When heat transfer coefficients vary significantly, the minimum network area can be predicted using linear programming^{4,5}. The remaining problem analysis approach can then be applied using these more sophisticated area targeting methods. Under such circumstances, the design is likely to be difficult to steer toward the minimum area, and an automated design method based on the optimization of a superstructure can be used, as will be discussed later.

Targets for number of shells, capital cost and total cost also can be set. Thus, remaining problem analysis can be used on these design parameters also.

Example 18.4 The stream data for a process are given in Table 18.5 below:

Steam is available condensing between 180 and 179°C and cooling water between 20 and 30°C. All film transfer coefficients are $200 \text{ W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$. For $\Delta T_{min} = 10^\circ\text{C}$, the minimum hot

Table 18.5 Stream data.

Stream		Supply temp. (°C)	Target temp. (°C)	Heat capacity flowrate (MW·K ⁻¹)
No.	Type			
1	Hot	150	50	0.2
2	Hot	170	40	0.1
3	Cold	50	120	0.3
4	Cold	80	110	0.5

and cold utility duties are 7 MW and 4 MW respectively, corresponding with a pinch at 90°C on the hot streams and 80°C on the cold streams.

- Develop a maximum energy recovery design above the pinch that comes close to the area target in the minimum number of units.
- Develop a maximum energy recovery design below the pinch that comes as close as possible to the minimum number of units.

Solution

- The area target for the above-pinch problem shown in Figure 18.22 is 8859 m². If the design is started at the pinch

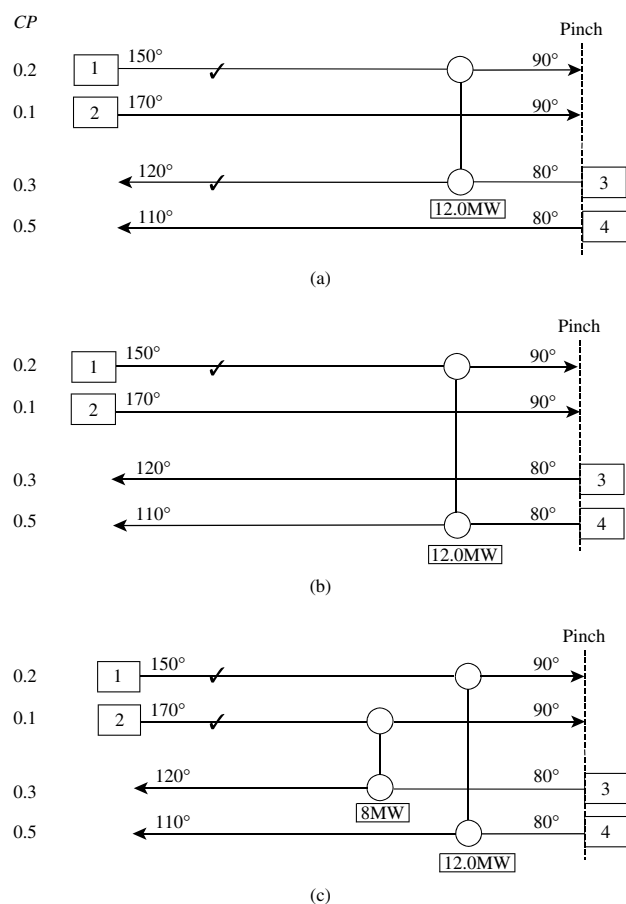


Figure 18.22 Above the pinch design for Example 18.4.

with Stream 1, then Figure 18.22a shows a feasible match that obeys the *CP* inequality. Maximizing its duty to 12 MW allows two streams to be ticked off simultaneously. This results from a coincidence in the stream data, the duties for Streams 1 and 3 being equal above the pinch. The area of the match is 6592 m² and the target for the remaining problem above the pinch is 3419 m², giving a total of 10,011 m². Thus, the match in Figure 18.22a causes the overall target to be exceeded by 1152 m² (13%). This does not seem to be a good match.

Figure 18.22b shows an alternative match for Stream 1 that also obeys the *CP* inequality. The tick-off heuristic also fixes its duty to be 12 MW. The area for this match is 5087 m², and the target for the remaining problem above the pinch is 3788 m², giving a total of 8,875 m². Thus, the match in Figure 18.22b causes the overall target to be exceeded by 16 m² (0.2%). This seems to be a better match and therefore is accepted.

Placing the next match above the pinch as shown in Figure 18.22c also allows the *CP* inequality to be obeyed. The area for both matches in Figure 18.22c is 7856 m² and the target for the remaining problem is 1020 m², giving a total of 8876 m². Accepting both matches causes the overall area target to be exceeded by 17 m² (0.2%). This seems to be reasonable, and both matches are accepted. No further process-to-process matches are possible, and it remains to place hot utility.

b. The cold utility target for the problem shown in Figure 18.23 is 4 MW. If the design is started at the pinch with Stream 3, then Stream 3 must be split to satisfy the *CP* inequality (Figure 18.23a). Matching one of the branches against Stream

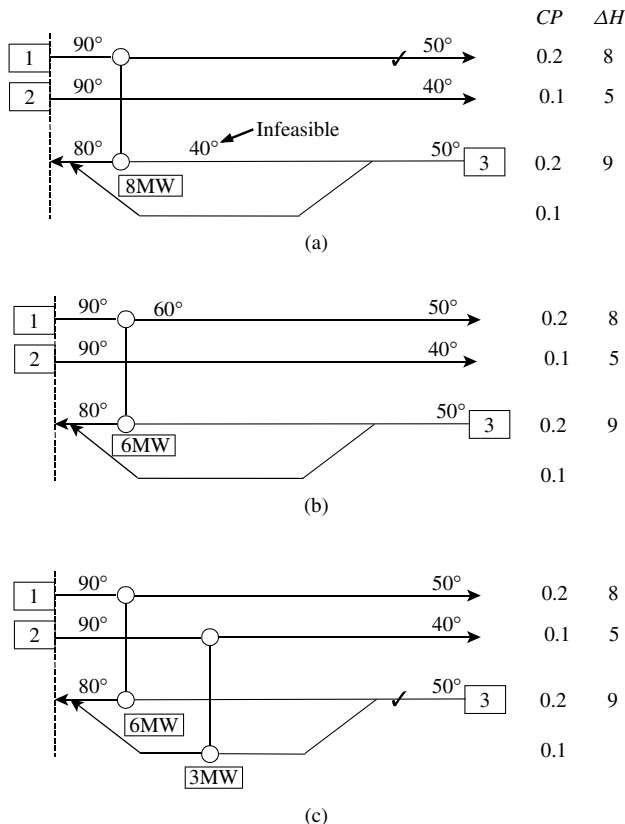


Figure 18.23 Below the pinch design for Example 18.4.

1 and ticking off Stream 1 results in a duty of 8 MW. This is a case in which the tick-off heuristic has caused problems. The match is infeasible, because the temperature difference between the streams at the cold end of the match is infeasible. Its duty must be reduced to 6 MW to be feasible without either stream being ticked off (Figure 18.23b).

Figure 18.23c shows an additional match placed on the other branch for Stream 3 with its duty maximized to 3 MW to tick off Stream 3. No further process-to-process matches are possible, and it remains to place cold utility.

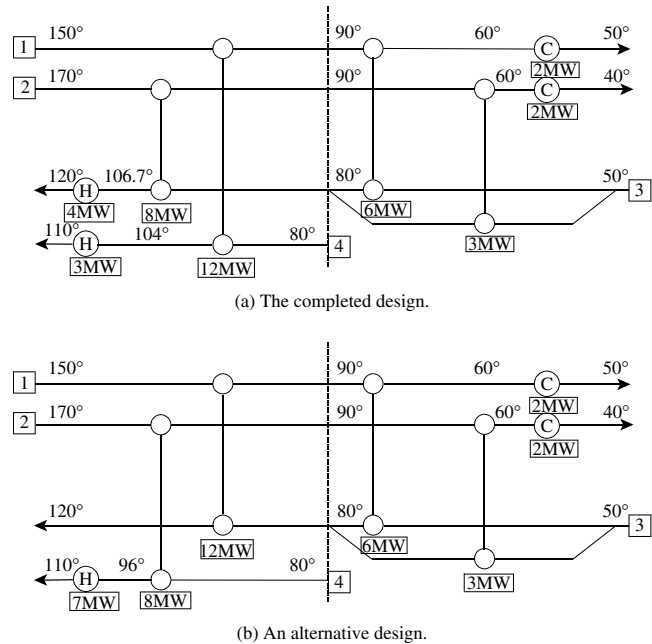


Figure 18.24 Alternative designs for Example 18.4.

Figure 18.24a shows the complete design, achieving maximum energy recovery in one more unit than the target minimum, due to the inability to tick off streams below the pinch. If the match in Figure 18.22a had been accepted and the design completed, then the design in Figure 18.24b would have been obtained. This achieves the target for the minimum number of units of 7 (at the expense of excessive area). This results from the coincidence of data mentioned earlier in Figure 18.22a, which allowed two streams to be ticked off simultaneously. The result is that the design above the pinch uses one fewer unit than target, owing to the formation of two components above the pinch (see Section 17.1). The design below the pinch uses one more than target, and the net result is that the overall design achieves the target for the minimum number of units.

18.6 NETWORK OPTIMIZATION

The pinch design method creates a network structure based on the assumption that no heat exchanger should have a temperature difference smaller than ΔT_{min} . Having now created a structure for the heat exchanger network, the structure can now be subjected to continuous optimization. The constraint that no exchanger should have a temperature

difference smaller than ΔT_{min} can now be relaxed. The continuous optimization of heat exchanger networks is based on the redistribution of the exchanger duties. Some exchangers should perhaps be larger, some smaller and some perhaps removed from the design altogether. Exchangers are removed from the design if the optimization sets their duty to zero.

Given a network structure, it is possible to identify loops and paths for it, as in Section 17.1. Within the context of optimization, only those paths that connect two different utilities need to be considered. This could be a path from steam to cooling water or a path from high-pressure steam used as hot utility to low-pressure steam also used as hot utility. These paths between two different utilities will be termed to be *utility paths*. Loops and utility paths both provide degrees of freedom in the optimization^{1,2}.

Consider Figure 18.25a that shows the network design from Figure 18.7, but with a loop highlighted. Heat can be shifted around loops. Figure 18.25a shows the effect of shifting heat duty U around the loop. In this loop, heat duty U is simply moved from Unit E to Unit B . The change in heat duties around the loop maintains the network heat balance and the supply and target temperatures of the streams. However, the temperatures around the loop change and hence the temperature differences of the exchangers in the loop change in addition to their duties. The magnitude of U could be changed to different values and the network costed at each value to find the optimum setting for U . If the optimum setting for U turns out to be 6.5 MW (the original duty on E), then the duty on one of the exchangers is zero and should be removed from the design.

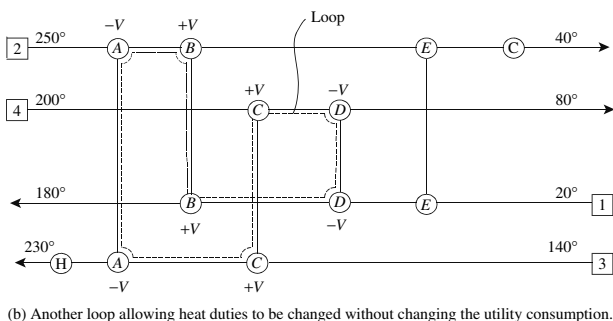
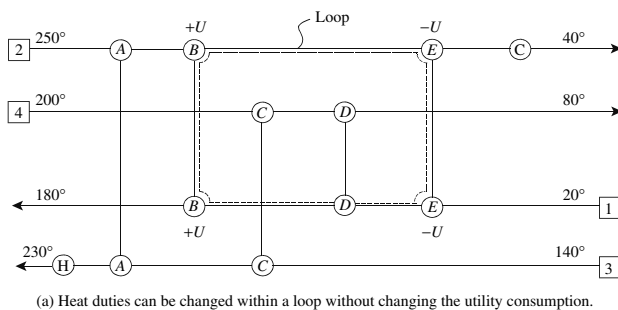


Figure 18.25 The loops that can be exploited for the optimization of the design for Figure 16.7.

Figure 18.25b shows the network with another loop marked. Figure 18.25b shows the effect of shifting heat duty V around the loop. Again the heat balance is maintained, but the temperatures as well as the duties around the loop change. As before, the value of V can be optimized by costing the network at different settings of V . If V is optimized to 7.0 MW (the original duty on A), then the duty on A becomes zero and this unit is removed from the design. Note again that once optimization is started, there is no longer a constraint to maintain temperature differences to be larger than ΔT_{min} .

Figure 18.26a shows the network with a utility path highlighted. Heat duty can be shifted along utility paths in a similar way to that for loops. Figure 18.26a shows the effect of shifting heat duty W along the path. This time the heat balance changes because the loads imported from hot utility and exported to cold utility both change by W . The supply and target temperatures are maintained. If W is optimized to 7.0 MW, this will result in Unit A being removed from the design. Different values of W can be taken and the network sized and costed at each value to find the optimal setting for W . Figure 18.26b shows other utility paths that can be exploited for optimization.

In fact, the optimization of the network requires that U , V , W , X , Y and Z in Figures 18.25 and 18.26 must be optimized simultaneously. Furthermore, stream splits may exist in the design, and variations of their branch flowrates can be superimposed on the exploitation of loops and paths in the optimization. During this optimization, the design is no longer constrained to have temperature differences larger than ΔT_{min} (although very small values in individual heat exchangers should be avoided for practical reasons). Also, pinches no longer divide the design into independent thermodynamic regions, and there is no longer any concern about cross-pinch heat transfer. The objective now is simply to minimize cost.

Thus, loops, utility paths and stream splits offer the degrees of freedom for manipulating the network cost. The objective function in new design is usually to minimize total cost, that is, combined operating and annualized capital cost. The annualization period chosen for the capital cost will have a direct influence on the optimization. A longer annualization period will lead to more energy-efficient designs.

In practice, rather than manipulate loops and paths explicitly, the optimization is normally formulated such that the individual duties on each match are varied in the multivariable optimization, subject to:

- the total enthalpy change on each stream being within a specified tolerance of the original stream data,
- nonnegative heat duty for each match,
- positive temperature difference for each exchanger to be greater than a practical minimum value for a given type of heat exchanger,

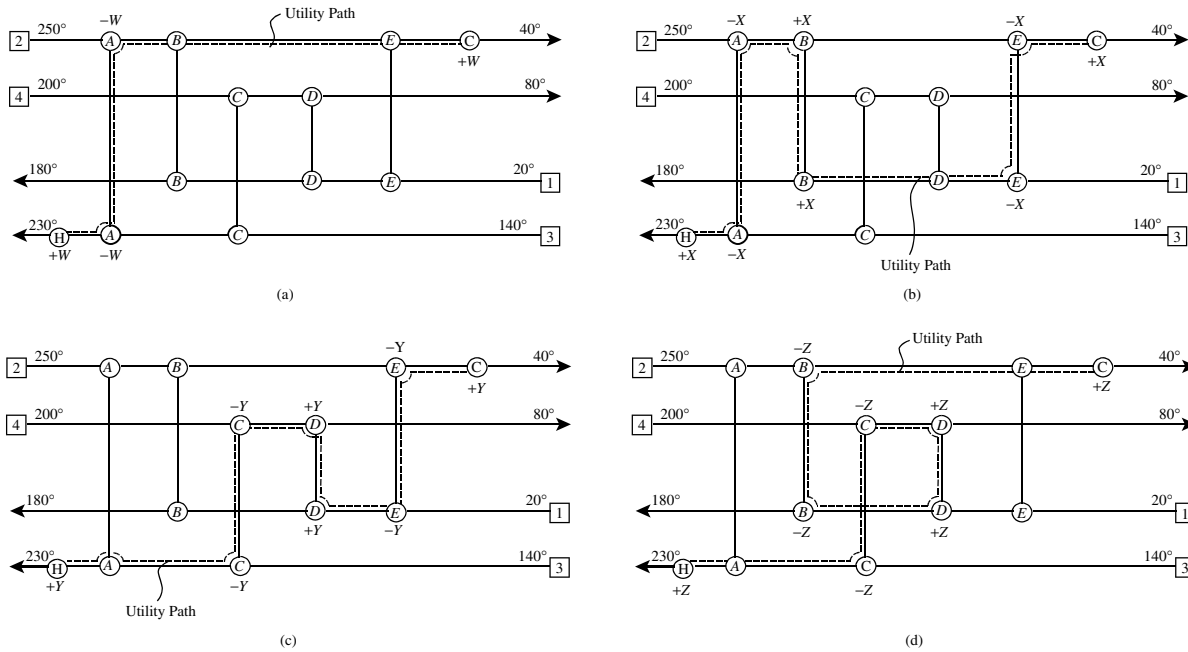


Figure 18.26 The utility paths that can be exploited for the optimization of the design from Figure 18.7.

- for stream splits, branch flowrates must be positive and above a practical minimum flowrate.

In a network, some of the duties on the matches will not be able to be varied because they are not in a loop or a utility path. This simplifies the optimization. The problem is one of multivariable nonlinear continuous optimization⁶.

If the network is optimized at fixed energy consumption, then only loops and stream splits are exploited. When energy consumption is allowed to vary, utility paths must also be included. As the network energy consumption increases, the overall capital cost tends to decrease and vice versa.

Example 18.5 Evolve the heat exchanger network in Figure 18.7 to simplify its structure.

- Remove the smallest heat recovery unit from the network by exploiting the degree of freedom in a loop.
- Recalculate the network temperatures and identify any violations of the $\Delta T_{min} = 10^\circ\text{C}$ constraint.
- Restore the original $\Delta T_{min} = 10^\circ\text{C}$ throughout the network by exploiting a utility path.

Solution

- Figure 18.27a shows the network from Figure 18.7 with a loop highlighted involving the unit with the smallest heat duty (6.5 MW). A heat duty of 6.5 MW has been shifted around the loop to adjust the smallest unit to a duty of zero. This will change the temperatures around the loop.
- Heat balances around the units for the new heat duties allow the new temperatures in the network to be calculated, as shown in Figure 18.27b. Also highlighted in Figure 18.27b, is a unit with an infeasible temperature difference. Not only is it less than

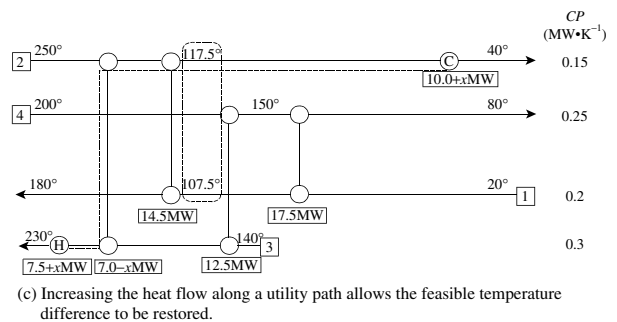
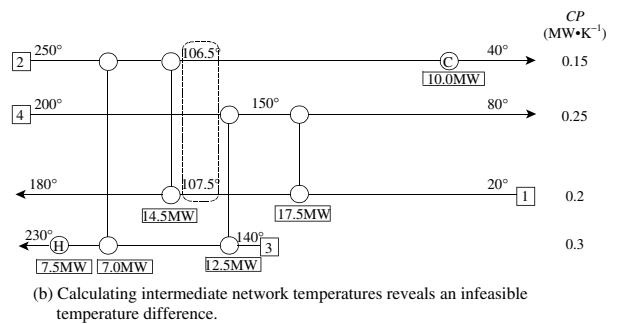
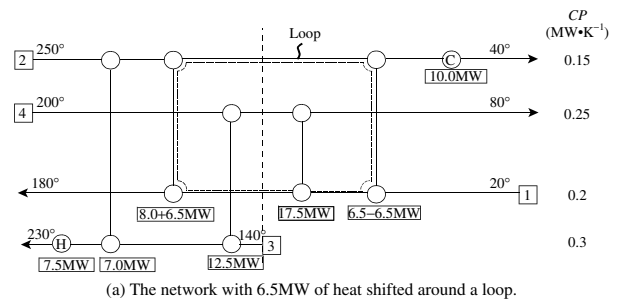


Figure 18.27 Evolution of a network to remove a unit.

the original ΔT_{min} of 10°C , it is actually negative. Removing a unit in this way will always create a temperature difference smaller than the original ΔT_{min} , if the hot and cold utility remain fixed. There is a minimum number of units to satisfy the problem to hot utility consumption of Q_{Hmin} and a cold utility consumption of Q_{Cmin} , subject to the ΔT_{min} constraint. If a unit is then removed, something must be violated. By its nature, shifting heat load around a loop does not change the energy consumption, but does change internal temperatures.

- c. Given the infeasible temperature difference in Figure 18.27b, this can be corrected by exploiting a utility path to change the temperatures in the network at the expense of increased energy consumption. Figure 18.27c shows the network with a utility path highlighted. The utility path allows one of the infeasible temperatures to be adjusted (Stream 2 in this case). If the original ΔT_{min} of 10°C is to be restored, then the intermediate temperature of Stream 2 needs to be adjusted to 117.5°C as shown in Figure 18.27c. The unknown is how much additional heat duty (x MW) needs to be shifted along the utility path to restore the temperature to 117.5°C . This can be determined by a simple heat balance around the cooler.

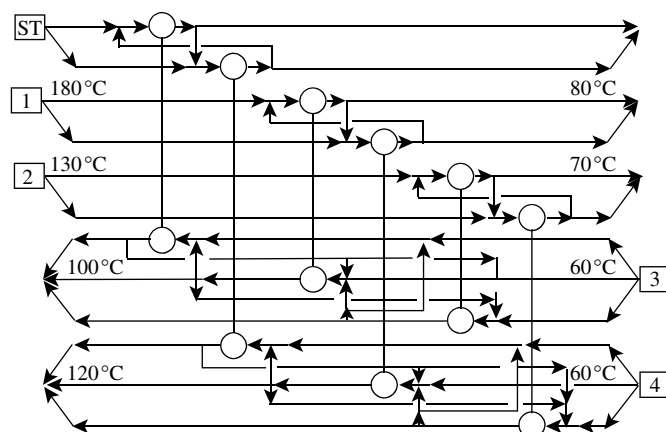
$$10.0 + x = 0.15 (117.5 - 40)$$

$$x = 1.6 \text{ MW}$$

Thus the hot and cold utility consumption both need to be increased by 1.6 MW to restore the ΔT_{min} to the original 10°C . In fact there is no justification to restore the ΔT_{min} back to the original 10°C . The amount of additional energy shifted along the utility path is a degree of freedom that should be set by cost optimization. However, the example illustrates how the degrees of freedom can be manipulated in network optimization.

18.7 THE SUPERSTRUCTURE APPROACH TO HEAT EXCHANGER NETWORK DESIGN

The approach to heat exchanger network design discussed so far was based on the creation of an irreducible structure.



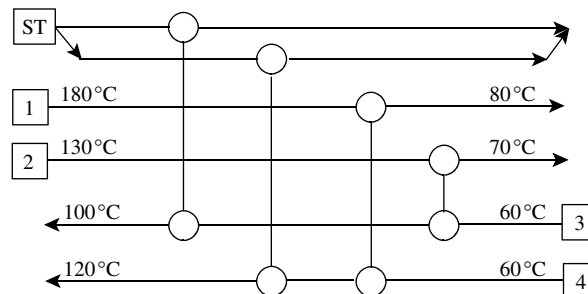
(a) A superstructure for part of a heat exchanger network.

No redundant features were included. However, after the structure was created, when the network was optimized, some of the features might be removed by the optimization. The scope for the optimization to remove features is a consequence of the assumptions made during the creation of the initial structure. However, no attempt was made to deliberately include redundant features.

An alternative approach is to create a superstructure that deliberately includes redundant features and then subject this to optimization. Redundant features are then removed by the optimization. Floudas, Ciric and Grossman⁷ showed how a heat exchanger network superstructure could be set up with all structural features included. Figure 18.28a shows such a superstructure for part of a heat exchanger network problem involving two hot streams, two cold streams and steam. All possibilities have been included within this superstructure. The basic idea is then to optimize the superstructure in order to remove the unnecessary features and minimize the cost, possibly leading to the design as shown in Figure 18.28b. While this looks simple in principle, the optimization required is a mixed integer nonlinear programming problem (MINLP, see Chapter 3)⁸. This is a difficult optimization problem with all of the issues associated with local optima.

One of the ways to avoid this problem is to simplify the superstructure to remove some of the structural options in Figure 18.28a⁹. This is done in Figure 18.29. This structure is created by splitting each hot stream into a number of branches equal to the number of cold streams and splitting each cold stream into a number of branches equal to the number of hot streams. In this way, a structure is created that allows each hot stream to be matched with each cold stream⁹.

One significant advantage of the simplified superstructure in Figure 18.29 is that each exchanger can be modeled by a linear equation if the supply and target temperatures of the streams are fixed. The area for each heat exchanger is



(b) The optimized design.

Figure 18.28 Heat exchanger network design from the optimization of a superstructure.

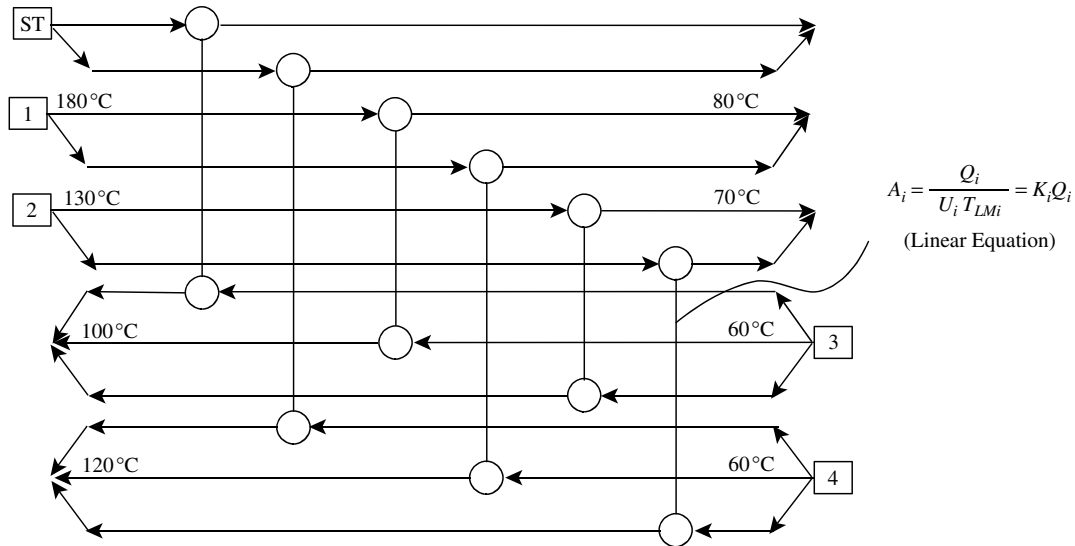


Figure 18.29 Starting with a simpler superstructure removes many structural options (some of which might be desirable, but makes the optimization linear.

given by:

$$A = \frac{Q}{U \Delta T_{LM}} \quad (18.5)$$

where A = heat transfer area
 Q = heat exchanger duty
 U = overall heat transfer coefficient
 ΔT_{LM} = logarithmic mean temperature difference

If U is assumed to be constant, despite changing conditions during the optimization, then because ΔT_{LM} is fixed for each match in Figure 18.29, Equation 18.5

becomes a linear equation in Q . If the capital cost of the heat exchangers is taken to be a linear function, then the optimization simplifies to be mixed integer linear programming (MILP) rather than MINLP. The problem with this is that the initial superstructure shown in Figure 18.29 features parallel configurations, as shown in 18.30a. If the linear optimization needs to retain all three matches on the stream, as shown in Figure 18.30a, then this cannot evolve to all of the series configurations, 1×2 parallel configurations, series-parallel configurations and parallel-series configurations as shown in Figure 18.30b.

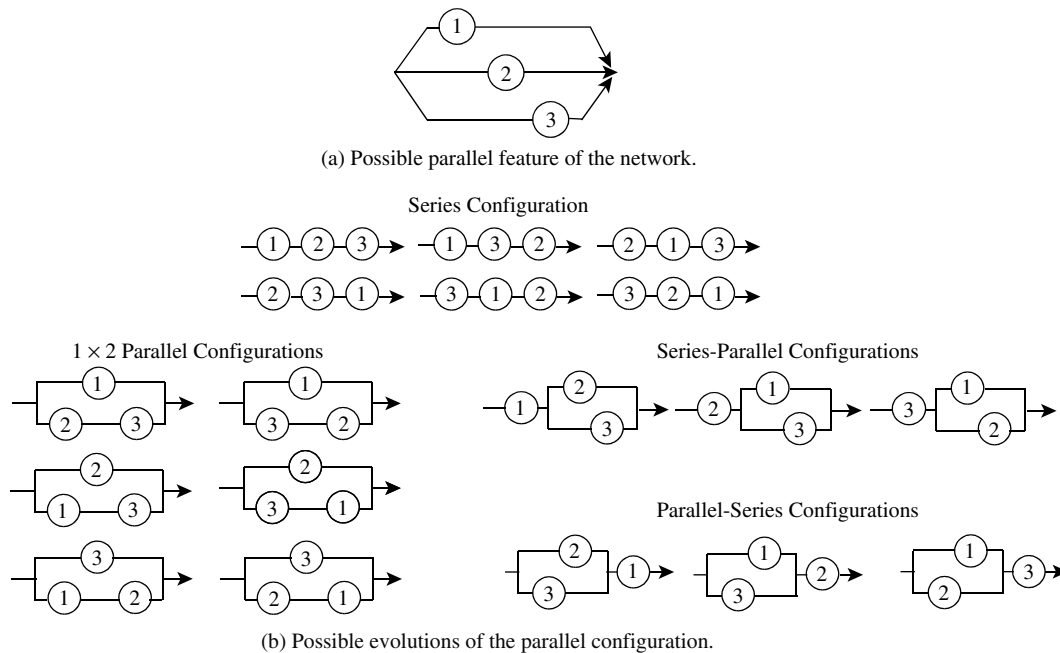


Figure 18.30 The simplified superstructure cannot be evolved to all structural options.

Another issue is how to divide the overall problem in order to set up the initial superstructure. The problem could be divided at the pinch, as done in the pinch design method, and a superstructure set up on each side of the pinch like the one in Figure 18.29. However, for large complex problems, this would not be comprehensive enough in terms of the number of structural options. The overall problem could therefore be divided into enthalpy intervals as shown in Figure 18.31 and a simplified superstructure created within each enthalpy interval¹⁰. Rather than dividing the composite curves into enthalpy intervals, as shown in Figure 18.31, with a superstructure within each enthalpy interval, enthalpy intervals can be merged into *blocks*¹¹. A superstructure is then created within each block rather than each enthalpy interval. This simplifies the complete superstructure for the whole network. This network superstructure could then be subjected to optimization using MILP.

Note that the ΔT_{min} for the problem must be fixed in order to remain an MILP problem. Fixing ΔT_{min} fixes the composite curves and the temperatures across each enthalpy interval or block. Unfortunately, this would not necessarily lead to the best network, as the initial superstructure was already simplified with many structural options missing. But this can be allowed for by first carrying out the

optimization on the basis of the simplified superstructure in Figure 18.29. If then the optimized superstructure featured structural options like the one in Figure 18.32a, then additional structural features could be added to give the structure in Figure 18.32b, which then can be subjected to MINLP optimization. This has the potential then to optimize to any parallel, series-parallel or parallel-series arrangement. This breaks the overall optimization down into two steps.

1. A simplified superstructure is first optimized using MILP.
2. If the solution from the simplified superstructure features parallel arrangements, such as that in Figure 18.32a, then additional features are added to the network, as shown in Figure 18.32b. This is then subjected to MINLP optimization to allow all the structural options in Figure 18.30 to be accessed.

This approach is one way to avoid the difficulties inherent with applying MINLP to a large complex superstructure. Another way might be to apply stochastic optimization¹². However, this has the disadvantage of being computationally very demanding for a reasonable size of heat exchanger network.

The major advantage of the superstructure approach to heat exchanger network design is that, in principle, it is capable

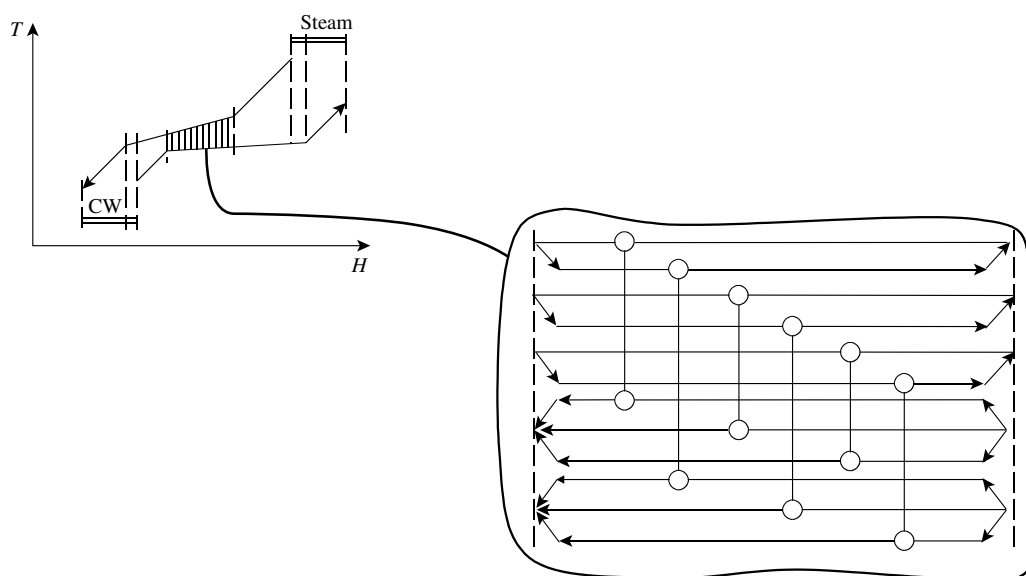


Figure 18.31 Create a simplified superstructure for each enthalpy interval.

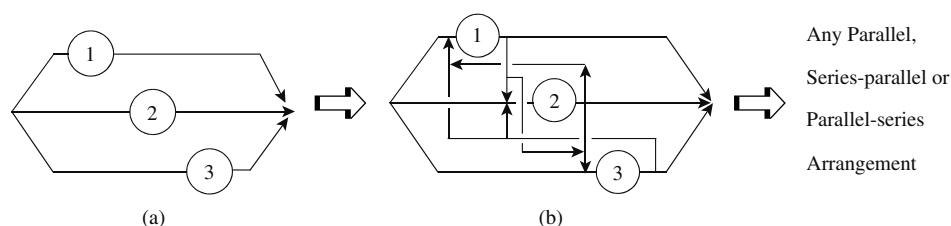


Figure 18.32 If the solution from the optimization of a simplified superstructure features parallel branches, then add additional structural features and re-optimize.

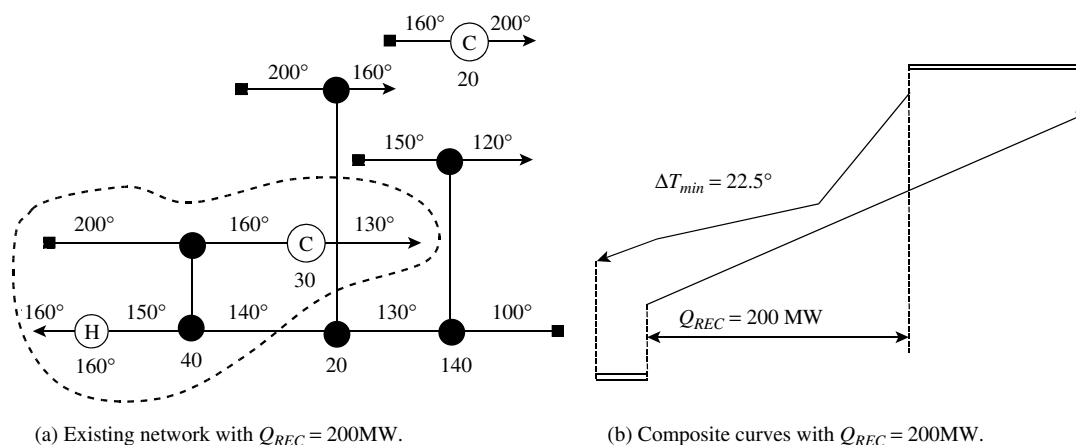


Figure 18.33 An existing heat exchanger network. (From Asante NDK and Zhu XX, 1997, *Trans IChemE*, 75A: 349, reproduced by permission of the Institution of Chemical Engineers.)

of designing large networks with complex constraints, mixed materials of construction, equipment types and so on. One major disadvantage of the approach is that the optimization is a difficult MINLP. Another significant disadvantage is that, because a computer carries out the optimization, the designer is removed from the decision-making.

18.8 RETROFIT OF HEAT EXCHANGER NETWORKS

So far, all considerations regarding the targeting and design of heat exchanger networks have related to new, or grassroots, design. It is often necessary to retrofit an existing heat exchanger network. The need to retrofit might arise from a desire to reduce the utility consumption of the existing network, need to increase the throughput, modification of the feed to the process or a modification to the product specification. All of these objectives might require heat duties within the network to be changed.

One approach to retrofit would be to try and evolve the network toward an ideal grassroots design. Following this approach, stream data would be targeted using the composite curves or the problem table to determine the location of the pinch for an assumed ΔT_{min} . Knowing the location of the pinch, the existing units in the network could then be located relative to the pinch. Any cross-pinch heat transfer could then be eliminated by disconnecting the units that are transferring heat across the pinch. These could be process-to-process, or the inappropriate use of utilities (e.g. use of steam below the pinch). The network could then be reconnected, with as many features of the existing network retained as possible¹³. This would no doubt improve the energy performance of an existing heat exchanger network, but it has a number of fundamental problems:

- Which ΔT_{min} should be used? As the value of ΔT_{min} changes, the location of the pinch changes and therefore

the assessment of which unit to transfer heat across the pinch also changes.

- Existing equipment is only reused in an ad hoc way.
- The retrofit is likely to involve a large number of modifications to the existing network.
- Constraints associated with the existing network are not readily included.

The problem with this approach is that it is attempting to change the network to a grassroots design rather than accepting the features that already exist. A better approach is to evolve the network from the existing structure in order to identify only the most critical, and therefore cost-effective, changes to the network structure¹⁴.

Suppose that it is desired to reduce the energy consumption of the existing heat exchanger network shown in Figure 18.33a. Figure 18.33b shows the composite curves for the basic stream data. The composite curves have been adjusted such that the overlap between the composite curves corresponds with the actual existing heat recovery duty of 200 MW. For the composite curves, this corresponds with a ΔT_{min} of 22.5°C. On the other hand, it can be seen that the existing heat exchanger network has minimum temperature differences of 20°C. First consider how the energy consumption of the existing network might be decreased without changing the network structure. As discussed previously, this is only possible by the exploitation of a utility path. The existing network in Figure 18.33a has only one degree of freedom that can be exploited for network optimization. This is highlighted within the bubble superimposed on the network. It involves a utility path between the heater and the cooler. The matches in the network outside of this bubble are all constrained by the heat duties on individual streams. The only way, therefore, that the utility consumption of the existing network can be decreased is by shifting heat load along the path between the heater and the cooler.

Figure 18.34a shows the network evolved to reduce the utility consumption to the point where the temperature

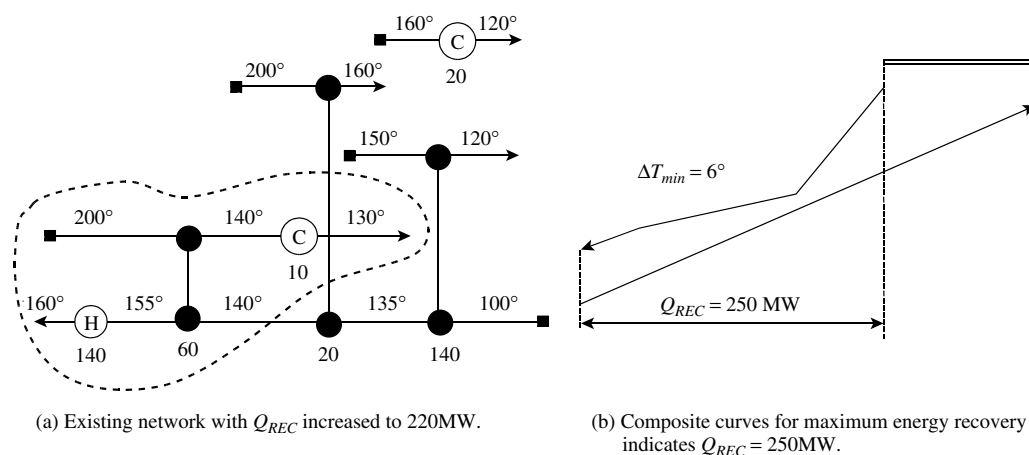


Figure 18.34 Maximizing the energy recovery of the existing heat exchanger network even down to $\Delta T = 0^\circ\text{C}$ gives an energy performance worse than the energy target. (From Asante NDK and Zhu XX, 1997, *Trans IChemE*, **75A**: 349, reproduced by permission of the Institution of Chemical Engineers.)

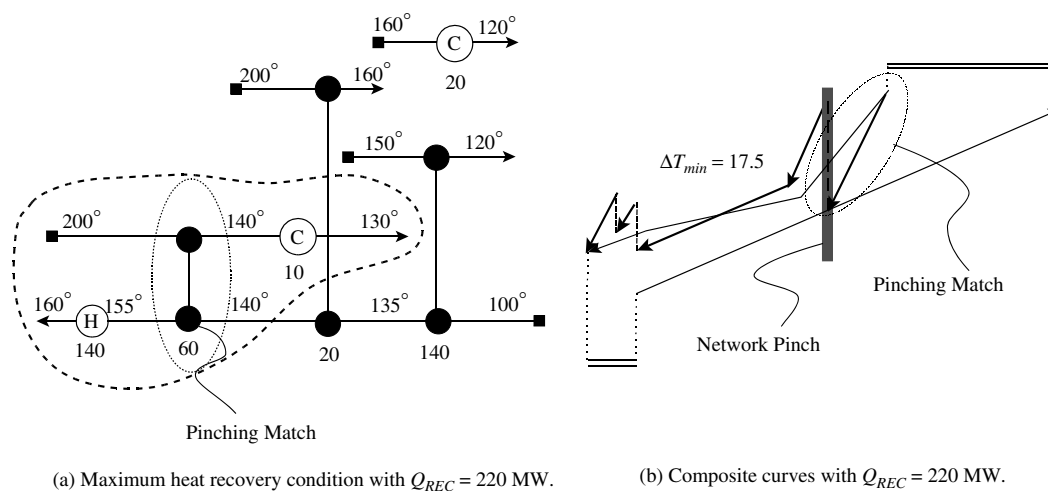


Figure 18.35 The network pinch limits the energy recovery in the existing heat exchanger network. (From Asante NDK and Zhu XX, 1997, *Trans IChemE*, **75A**: 349, reproduced by permission of the Institution of Chemical Engineers.)

difference in the existing network is now 0°C^{14} . This is not a limit that could be achieved in practice, but is taken here for the sake of illustration. For a minimum temperature difference of 0°C in the existing network, the energy recovery must increase to 220 MW. Compare the composite curves for increased heat recovery. These are shown in Figure 18.34b for their maximum overlap, which is 250 MW. At this setting, the composite curves still have a ΔT_{min} of 6°C . This reveals that the maximum heat recovery within the existing heat exchanger network structure is different from that theoretically possible from the composite curves. The difference is caused by the fact that the existing heat exchanger network structure is not appropriate for maximum energy recovery. How can the existing network structure be modified to improve its performance?

The existing heat exchanger network is shown again in Figure 18.35 with its recovery increased to an absolute

maximum. Figure 18.35a highlights what is limiting the existing heat exchanger network in terms of its heat recovery. One of the existing units features minimum temperature difference (0°C in this case for the sake of illustration). The composite curves are also shown in Figure 18.35b, but set to the same overall heat recovery load as that featuring the existing heat exchanger network at its limit (220 MW). Superimposed on the composite curves are the temperature profiles for the hot streams in each of the existing units. One of the matches limits the heat recovery, in this case the one featuring a temperature difference of 0°C . This match that limits the heat recovery is known as the *pinching match*¹⁴. The point at which this occurs is known as the *network pinch*¹⁴. The network pinch limits the heat recovery for the existing heat exchanger network.

In practice, if the network pinch is being identified in a design study, then a practical minimum temperature

difference of say 10°C or 20°C would be taken rather than the 0°C used here for the sake of illustration. The principle is exactly the same. One or more matches in the existing network feature ΔT_{min} when the network is pinched.

Consider now how the network pinch might be overcome. There are four ways in which the network pinch can be overcome and the performance of the existing heat exchanger network improved beyond that for the pinched condition¹⁴.

a. *Resequencing.* Figure 18.36 illustrates how *resequencing* can be used to overcome the network pinch. Resequencing moves the unit to a new location in the network, but between the same streams as the original match. Figure 18.36 shows a cold stream being heated by two hot streams. One of the hot stream profiles indicates that it is a pinching match and features a minimum temperature difference of 0°C (again for the sake of illustration). In Figure 18.36, the pinching match is adjacent to another hot stream that has a finite temperature difference for the heat exchange. If the position of the two hot streams is swapped by a simple resequence, as shown in Figure 18.36, then the pinching match no longer limits. This means that there is now new scope to reduce the energy consumption of the network by exploiting a utility path, as shown in Figure 18.36. If the utility path is exploited to its limit, then a new network pinch is created, but now at a lower utility consumption for the network.

b. *Repiping.* Repiping is very similar to resequencing. Like resequencing, repiping moves the unit to a new location in the network. However, in repiping, the unit can be moved to a location involving streams other than in the original location, rather than be restricted to operate between the same streams. Repiping is a more general case

than resequencing, but might not be practical for a variety of reasons, for example, materials of construction being unsuitable for other streams. The basic principle of repiping is the same as that for resequencing, but a distinction needs to be made for practical reasons.

c. *Inserting a new match.* Figure 18.37 shows how the network pinch can be overcome by inserting a new match. Again the principle is illustrated by two hot streams providing heat to a cold stream. One of the matches is pinched. If a new match is inserted such that the heat duty on the hot stream adjacent to the pinching match is decreased and replaced by the new match, then the position of the pinching match can be changed such that it is no longer pinching. This introduces scope to exploit the utility path to reduce the utility consumption of the network, until it is again pinched. The network is now pinched again, but at a lower utility consumption.

d. *Introduce additional stream splitting.* The fourth way to overcome the network pinch is by introducing additional stream splitting to the existing network, as illustrated in Figure 18.38. In this case, it can be seen that two matches are pinched simultaneously. By introducing a stream split, the cold stream profiles in the two pinched units are now such that one of the pinching matches is no longer pinched. This means that there is scope to exploit a utility path and reduce the energy consumption. This is being carried out to its limit in Figure 18.38 such that the network is again pinched, but at a lower utility consumption.

Again, it should be emphasized that in practice a finite practical ΔT_{min} should be used rather than 0°C . However, any assumption of ΔT_{min} to identify, and then overcome, the network pinch does not guarantee

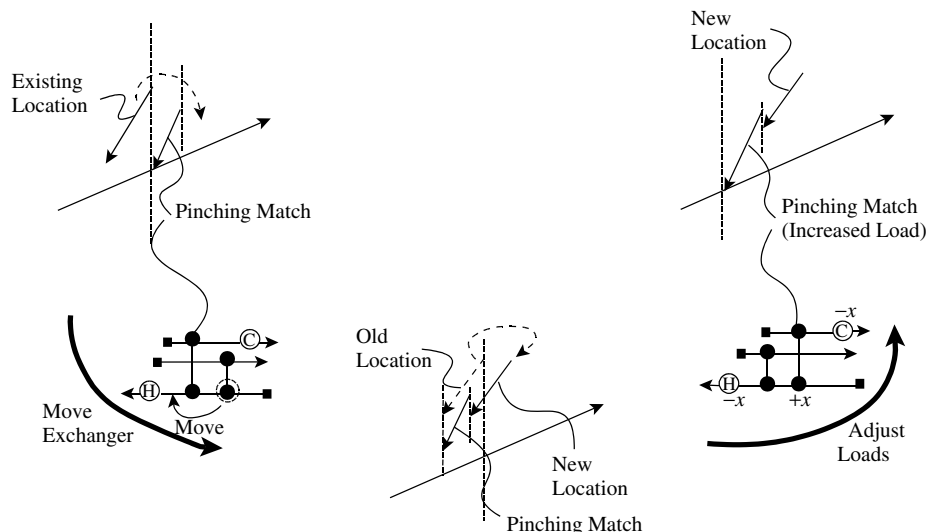


Figure 18.36 Resequencing a match can be used to overcome the network pinch. (From Asante NDK and Zhu XX, 1997, *Trans IChemE*, 75A: 349, reproduced by permission of the Institution of Chemical Engineers.)

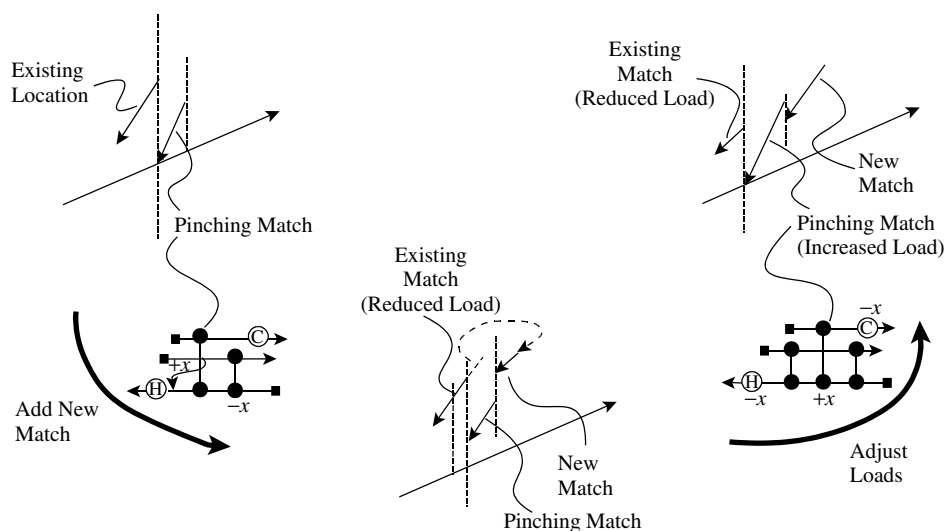


Figure 18.37 Adding a new heat exchanger can be used to overcome the heat network pinch. (From Asante NDK and Zhu XX, 1997, *Trans IChemE*, 75A: 349, reproduced by permission of the Institution of Chemical Engineers.)

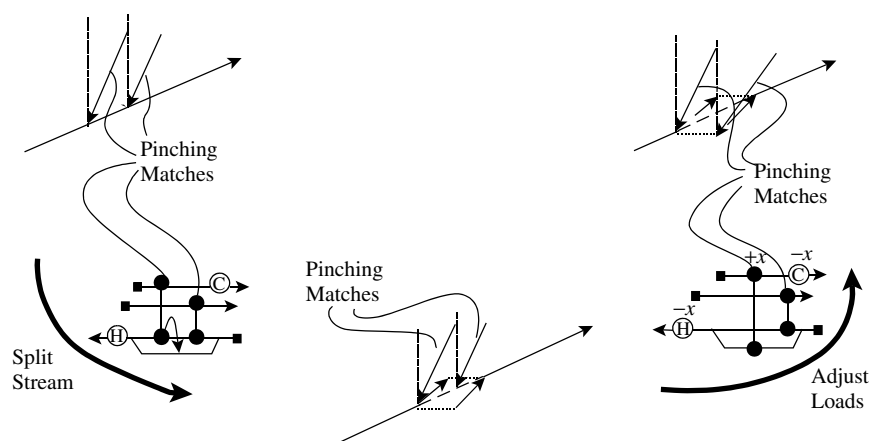


Figure 18.38 Changing the stream splitting arrangement can be used to overcome the network pinch. (From Asante NDK and Zhu XX, 1997, *Trans IChemE*, 75A: 349, reproduced by permission of the Institution of Chemical Engineers.)

an optimum retrofit. The modified network should be subjected to cost optimization in order to obtain the correct setting for the capital-energy trade-off. The optimization of a heat exchanger network with a fixed structure is a Nonlinear Programming (NLP) optimization, as discussed previously. When dealing with the optimization of existing heat exchanger networks, it needs to be recognized that there is existing heat transfer area in place, but new area (or enhanced heat transfer) needs to be installed in some parts of the network to improve the network performance. The existing heat transfer area has zero capital cost and only the new heat transfer area and pipework modifications need to be included in the capital costs for the optimization. Care is required when specifying the form of the capital cost correlation in retrofit. Designers often like to use a cost per unit area, such as:

$$\text{Capital Cost} = bA \quad (18.6)$$

where A = heat transfer area

b = cost coefficient

If the capital cost of new heat transfer area is expressed in the form of Equation 18.6, then this will lead to poor retrofit projects. The problem with Equation 18.6 is that the optimization is likely to spread the new heat transfer area in the network in many locations, without incurring a cost penalty associated with the many modifications that would result. To ensure that new heat transfer area is not spread around throughout the existing heat exchanger network, a capital cost correlation should be used that is of the form:

$$\text{Capital Cost} = a + bA^c \quad (18.7)$$

where a, b, c = cost coefficients

In Equation 18.7, the coefficient a is a threshold cost that is incurred even if a small amount of heat transfer area

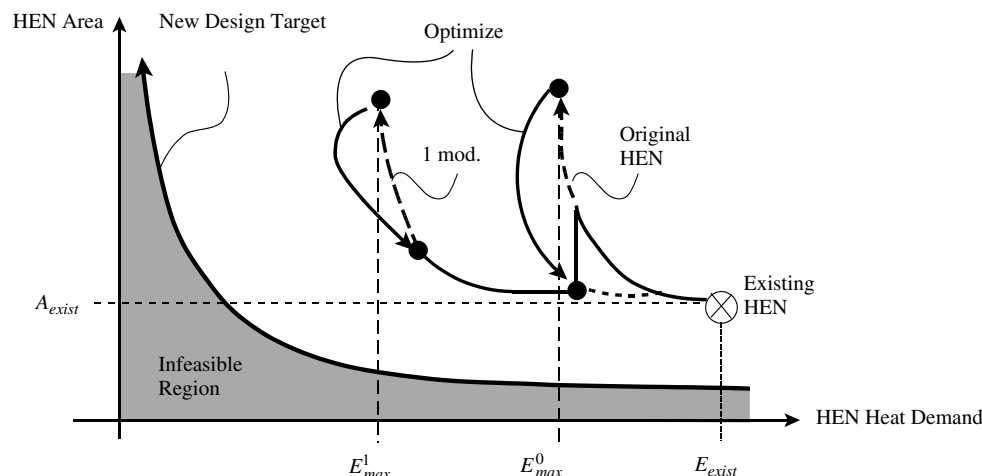


Figure 18.39 The heat exchanger network can be optimized after each modification. (From Asante NDK and Zhu XX, 1997, *Trans IChemE*, **75A**: 349, reproduced by permission of the Institution of Chemical Engineers.)

is installed. If a large threshold cost is used in the cost correlation, then this would lead an optimizer to attempt to concentrate any new heat transfer area required in as few locations as possible in the network. This, in turn, will lead to fewer modifications in the retrofit project to accommodate the new heat transfer area requirements.

It should be noted that a resequence or repipe does not involve zero capital cost, even though no new heat exchanger equipment might be purchased. The pipework modifications for a resequence or repipe might be very expensive. Also, equipment might need to be relocated. Methods for capital cost estimation for retrofit were discussed in Chapter 2.

Figure 18.39 shows the capital-energy trade-off on a plot of heat exchanger network area versus energy consumption. The new (grassroot) design target marks the infeasible region in Figure 18.39. Targeting methods for energy were discussed in Chapter 16 and targeting methods for network area were discussed in Chapter 17. The existing network performance cannot be better than the target, but it can be worse, as illustrated in Figure 18.39. If the existing network structure is retained, then there is a limit beyond which the network cannot be improved in terms of its energy performance. This is when the existing network structure is pinched at a practical minimum temperature difference for the heat exchangers. In order to approach this condition, it will require an additional heat transfer area. The more the energy consumption is decreased, the additional heat transfer area required will increase per unit energy reduction, as the network pinch is approached. For the existing heat exchanger network structure, the capital-energy trade-off can be optimized, as shown in Figure 18.39. Alternatively, the structure of the network could be modified, in which case, the limit for heat recovery for the modified network is now at a lower energy consumption, Figure 18.39. Again, the network has been modified on the basis of an assumed practical

minimum temperature difference that is not necessarily providing the correct setting for the capital-energy trade-off. Therefore, the modified network can be optimized to provide the optimized network for one structural modification. This procedure could be repeated for a second and third modification, and so on. In practice, retrofits are usually only economic for a small number of network modifications.

This approach to heat exchanger network retrofit based on the concept of the network pinch can be automated¹⁵. A superstructure can be created for network resequencing (or repiping) and the best resequence (or repipe) identified by optimizing the superstructure of resequences (or repipes) for an assumed practical ΔT_{min} ¹⁵. This can be formulated as an MILP optimization if the network is optimized for minimum energy consumption with a fixed ΔT_{min} . Inclusion of area calculations to estimate heat exchanger capital costs would make the optimization nonlinear. Similarly, a superstructure can be created for positioning new matches or stream split modifications in the existing network and the superstructure optimized for minimum energy cost for an assumed practical ΔT_{min} . The problem can again be formulated as an MILP problem¹⁵.

After each suggested modification has been identified, the network can be subjected to a detailed capital-energy trade-off requiring NLP optimization. Optimization of the capital-energy trade-off is illustrated in Figure 18.40. Structural modifications are first explored using MILP and then the correct setting of the capital-energy trade-off corrected using NLP. By decomposing the problem in this way, what is overall an MINLP problem, is carried out by MILP followed by NLP, which is a more robust approach to the optimization.

This approach to heat exchanger network retrofit allows modifications to be introduced one at a time. In this way, the designer has control over the complexity of the network retrofit. At each stage, a suggested modification can be

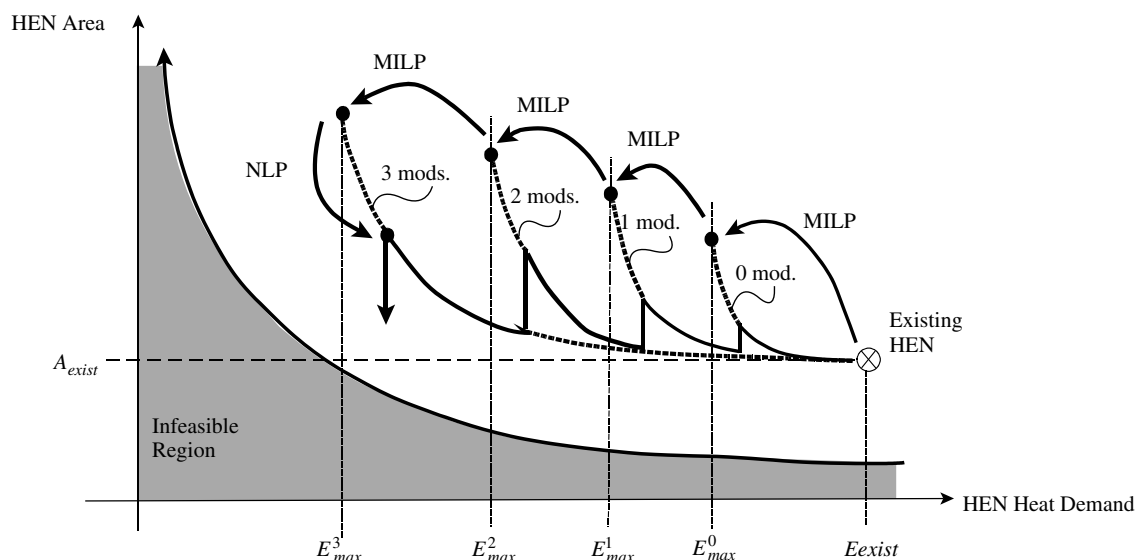


Figure 18.40 The retrofit can be proceed in a stepwise approach. (From Asante NDK and Zhu XX, 1997, *Trans IChemE*, **75A**: 349, reproduced by permission of the Institution of Chemical Engineers.)

evaluated in terms of its practicality and true costs. If the best structural option identified is considered to be impractical (e.g. the plant is too congested), then another modification can be suggested instead. For example, the optimization could be carried out to identify the best resequence, only to find that this is impractical. If this is the case, then the second best resequence could be taken, or the option of a new match chosen and so on. The designer has control and can assess each modification as is suggested by the optimization. Having identified an acceptable number of modifications, the modifications are subjected to a detailed capital-energy trade-off to assess their true economic performance.

The approach leads to simple and practical retrofit designs and has the major advantage that it allows the designer to assess modifications one at a time and to keep control over the complexity of the retrofit. Its disadvantage is that different combinations of modifications can be taken and there is no guarantee that this will lead to an optimum network retrofit. However, it is almost impossible to say that any retrofit is optimal or nonoptimal. The features of each retrofit are unique and it is difficult to formulate all the constraints for a retrofit in order to guarantee the very best retrofit.

18.9 ADDITION OF NEW HEAT TRANSFER AREA IN RETROFIT

Retrofit of heat exchanger networks requires resequencing and repiping of existing heat exchangers, installation of new heat exchange matches and changes to stream splitting arrangements. However, existing matches that have not been moved might require additional heat transfer area

as a result of changes in the operation. This might result from increased heat duty, operation under reduced temperature differences or operation under reduced heat transfer coefficients. Methods for the retrofit of heat exchangers have been discussed in Chapter 15. If shell-and-tube heat exchangers are being used and additional heat transfer area is required, it might be possible to install a new tube bundle into existing shells if the additional area requirement is small. If this is not feasible, then the existing area can be supplemented by adding a new shell (or more than one shell if there is a large area requirement) in:

- a. *Series*. The addition of new exchangers in series to the existing match will lead to an increase in the overall pressure drop across the match. This might be important if the pump (or compressor) is close to its maximum capacity.
- b. *Parallel*. The addition of new exchangers in parallel will leave the pressure difference largely unchanged. The pressure drop across the match will be the largest between that of the existing exchangers under the new conditions and new exchangers installed in parallel.

Also, as discussed in Chapter 15, rather than install additional heat transfer area to cater for the new operational requirements, heat transfer enhancement can be considered. Changes to the number of tube passes or the baffle arrangement might allow the heat transfer coefficient to be enhanced. Alternatively, tube inserts could be used. This was discussed in Chapter 15. The major disadvantage in using heat transfer enhancement is that it increases the pressure drop. In retrofit this can be important, as the pumps driving the flow might be limited in their capacity to meet the required increase in pressure drop.

Rather than shell-and-tube heat exchangers, the network might involve plate heat exchangers. Adding additional area to plate heat exchangers is generally more straightforward than shell-and-tube exchangers. Plate heat exchangers can often have their area increased by adding additional plates to the existing frame. If an additional frame needs to be added to provide the additional area, then the new frame can be added in series or parallel with the existing frame. Series arrangements increase the pressure drop. Parallel arrangements decrease the flowrate through each frame but decrease the heat transfer coefficient in the existing frame. The pressure drop across the match will be the largest between that of the existing exchangers under the new conditions and new exchangers installed in parallel.

18.10 HEAT EXCHANGER NETWORK DESIGN – SUMMARY

A good initialization for heat exchanger network design is to assume that no individual exchanger should have a temperature difference smaller than ΔT_{min} . Having decided that no exchanger should have a temperature difference smaller than ΔT_{min} two rules were deduced in Chapter 16. If the energy target set by the composite curves (or the problem table algorithm) is to be achieved, there must be no transfer heat across the pinch by:

- Process-to-process heat transfer
- Inappropriate use of utilities.

These rules are both necessary and sufficient for the design to achieve the energy target given that no individual exchanger should have a temperature difference smaller than ΔT_{min} .

The design of heat exchanger networks can be summarized in five steps.

1. Divide the problem at the pinch into separate problems.
2. The design for the separate problems is started at the pinch, moving away.
3. Temperature feasibility requires constraints on the CP 's to be satisfied for matches between the streams at the pinch.
4. The loads on individual units are determined using the tick-off heuristic to minimize the number of units. Occasionally, the heuristic causes problems.
5. Away from the pinch, there is usually more freedom in the choice of matches. In this case, the designer can discriminate on the basis of operability, plant layout and so on.

If the number of hot streams at the pinch above the pinch is greater than the number of cold streams, then

the cold streams must be split to satisfy the ΔT_{min} constraint. If the number of cold streams at the pinch below the pinch is greater than the number of hot streams, then the hot streams must be split to satisfy the ΔT_{min} constraints. If the CP inequalities for all streams at the pinch cannot be satisfied, this also can necessitate stream splitting.

If the problem involves more than one pinch, then between pinches the design should be started from the most constrained pinch.

Remaining problem analysis can be used to make a quantitative assessment of matches in the context of the whole network without having to complete the network.

Once the initial network structure has been defined, then loops, utility paths and stream splits offer the degrees of freedom for manipulating network cost in multivariable continuous optimization. When the design is optimized, any constraint that temperature differences should be larger than ΔT_{min} or that there should not be heat transfer across the pinch no longer applies. The objective is simply to design for minimum total cost.

For more complex network designs, especially those involving many constraints, mixed equipment specifications and so on, design methods based on the optimization of a superstructure can be used.

Network retrofit can be performed on the basis of the concept of the network pinch. Resequencing and repiping of heat exchangers, the introduction of new exchangers and additional stream splitting can all be used to overcome the network pinch. The identification of structural modifications to the network can be done on the basis of minimizing the energy consumption for a ΔT_{min} using MILP. Once structural modifications have been identified, these should be subjected to a detailed capital-energy trade-off using NLP optimization. New heat transfer area to existing matches can be added through the addition of new heat exchanger shells in series or parallel or through heat transfer enhancement to existing exchangers.

18.11 EXERCISES

1. The stream data for a process are given in the Table 18.6:

Table 18.6 Stream data for Exercise 1.

Stream		Supply temperature	Target temperature	Heat capacity flowrate
No.	Type	(°C)	(°C)	(MW·K ⁻¹)
1	Hot	200	100	0.4
2	Hot	200	100	0.2
3	Hot	150	60	1.2
4	Cold	50	140	1.1
5	Cold	80	120	2.4

- For a ΔT_{min} of 10°C, the pinch is at 90°C for hot streams and 80°C for cold streams. Design a heat exchanger network for maximum energy recovery that features the minimum number of units.
 - By relaxing the constraint of $\Delta T_{min} = 10^\circ\text{C}$, shift heat load through a utility path such that the network uses only hot utility and no cold utility at all.
2. The process stream data for a heat recovery network problem are given in Table 18.7

Table 18.7 Stream data for Exercise 2.

Stream	T_S (°C)	T_T (°C)	CP (MW·K ⁻¹)
1. Hot	300	80	0.15
2. Hot	200	40	0.225
3. Cold	40	180	0.2
4. Cold	140	280	0.3

- Determine the energy targets for hot and cold utility for $\Delta T_{min} = 20^\circ\text{C}$.
 - Design a heat exchanger network to achieve the energy targets in the minimum number of units.
 - Identify the scope to reduce the number of units by manipulating loops and utility paths by sacrificing energy consumption.
 - Design a network to realize this scope and restore $\Delta T_{min} = 20^\circ\text{C}$.
3. The heat recovery stream data for a process are given in Table 18.8.

Table 18.8 Stream data for Exercise 3.

Stream	T_S (°C)	T_T (°C)	CP (kW·K ⁻¹)
1. Hot	180	40	200
2. Hot	150	40	400
3. Cold	60	180	300
4. Cold	30	130	220

A problem table analysis for $\Delta T_{min} = 10^\circ\text{C}$ results in the heat recovery cascade given in Table 18.9.

Table 18.9 Heat recovery cascade for Exercise 3.

Interval temperature (°C)	Heat flow (MW)
185	6.0
175	3.0
145	0
135	3.0
65	8.6
35	20.0

- Design a heat recovery network in the minimum number of units. The number of units is below what might have been expected from the target for the number of units; why?

- Low-pressure saturated steam can be generated from saturated boiler feedwater at an interval temperature of 115°C. Determine how much low-pressure steam can be generated and design a network to achieve this duty.
 - The network design from Part *b* results in two steam generators. Evolve the network to remove one of these by suffering a penalty in steam generation, but maintain $\Delta T_{min} = 10^\circ\text{C}$.
4. The process stream data for a heat recover problem are given in Table 18.10.

Table 18.10 Stream data for Exercise 4.

Stream	T_S (°C)	T_T (°C)	CP (MW·K ⁻¹)
1. Cold	18	123	0.0933
2. Cold	118	193	0.1961
3. Cold	189	286	0.1796
4. Hot	159	77	0.2285
5. Hot	267	80	0.0204
6. Hot	343	90	0.0538

A problem table analysis reveals that $Q_{Hmin} = 13.95$ MW, $Q_{Cmin} = 8.18$ MW, hot stream pinch temperature is 159°C and cold stream pinch temperature is 149°C for $\Delta T_{min} = 10^\circ\text{C}$.

- Design a heat recovery network with the minimum number of units. (Hint: below the pinch it may be necessary to split a stream away from the pinch to achieve the minimum number of units).
 - Evolve the design to eliminate stream splits below the pinch, while maintaining the same number of units, by allowing a temperature difference slightly smaller than 10°C.
5. The data for a heat recovery problem are given in the Table 18.11.

Table 18.11 Stream data for Exercise 5.

Stream		T_S (°C)	T_T (°C)	Heat capacity flowrate (MW·K ⁻¹)
No.	Type			
1	Hot	120	65	0.5
2	Hot	80	50	0.3
3	Hot	135	110	0.29
4	Hot	220	95	0.02
5	Hot	135	105	0.26
6	Cold	65	90	0.15
7	Cold	75	200	0.14
8	Cold	30	210	0.1
9	Cold	60	140	0.05
Steam		250	–	–
Cooling water		15	–	–

The problem table cascade is given in Table 18.12 for $\Delta T_{min} = 20^\circ\text{C}$.

Given this data:

- Set out the stream grid.
- Design a maximum energy recovery network.

Table 18.12 Problem table cascade for Exercise 5.

Interval temperature (°C)	Heat flow (MW)
220	20.95
210	19.95
150	6.75
125	0
110	4.2
100	12
95	13.7
85	14.5
75	16.5
70	18.25
55	28.75
40	31.75

Table 18.15 Heat flow cascade for Exercise 7.

Interval temperature (°C)	Heat flow (kW)
349	1528
303	1758
273	1368
210	675
205	520
165	0
140	250
135	255
95	1095
85	1255
83	1283
35	851

Table 18.13 Stream data for Exercise 6.

Stream No.	Type	T_s (°C)	T_T (°C)	Stream heat duty (MW)
1	Hot	150	50	-20
2	Hot	170	40	-13
3	Cold	50	120	21
4	Cold	80	110	15

6. The stream data for a process are given in the Table 18.13. Steam is available between 180 and 179°C and cooling water between 20 and 40°C. For $\Delta T_{min} = 10^\circ\text{C}$, the minimum hot and cold utility duties are 7 MW and 4 MW respectively. The pinch is at 90°C on the hot streams and 80°C on the cold streams.
- Calculate the target for the minimum number of units for maximum energy recovery.
 - Develop two alternative maximum energy recovery designs, keeping units to a minimum.
 - Explain why the design below the pinch cannot achieve the target for the minimum number of units.
 - How many degrees of freedom are available for network optimization?

7. The stream data for a process are given in the Table 18.14:

Table 18.14 Stream data for Exercise 7.

Stream No.	Type	T_s (°C)	T_T (°C)	CP (kW·K ⁻¹)
1	Hot	170	88	23
2	Hot	278	90	2
3	Hot	354	100	5
4	Cold	30	135	9
5	Cold	130	205	20
6	Cold	200	298	18

A problem table heat cascade for $\Delta T_{min} = 10^\circ\text{C}$ is given the Table 18.15. Hot utility is to be provided by a hot oil circuit with a supply temperature of 400°C. Cooling water is available at 20°C.

- Calculate the minimum flowrate of hot oil if C_P for the hot oil is $2.1 \text{ kJ}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$, assuming ΔT_{min} process-to-process heat recovery is 10°C and process to hot oil to be 20°C .
 - Design a heat exchanger network for maximum energy recovery in the minimum number of units ensuring $\Delta T_{min} = 10^\circ\text{C}$ for all process-to-process heat exchangers throughout the network.
 - Suggest an alternative network design below the pinch to eliminate stream splits by accepting a violation of ΔT_{min} .
 - What tools could have been used to develop the design below the pinch more systematically?
8. The stream data for a process are given in Table 18.16

Table 18.16 Stream data for Exercise 8.

Stream No.	Type	T_s (°C)	T_T (°C)	Heat duty (MW)
1	Hot	150	30	7.2
2	Hot	40	40	10
3	Hot	130	100	3
4	Cold	150	150	10
5	Cold	50	140	3.6

- Sketch the composite curves for $\Delta T_{min} = 10^\circ\text{C}$.
 - Determine the target for hot and cold utility for $\Delta T_{min} = 10^\circ\text{C}$.
 - Design a maximum energy recovery network in the minimum number of units for $\Delta T_{min} = 10^\circ\text{C}$.
 - Can the number of units be reduced by evolution of the network?
9. The stream data for a process are given in Table 18.17. $\Delta T_{THRESHOLD}$ for the problem is 50°C and ΔT_{min} is 20°C . A problem table analysis on this data produces the cascade given in Table 18.18 for $\Delta T_{min} = 20^\circ\text{C}$.

Table 18.17 Stream data for exercise 9.

Stream		T_S (°C)	T_T (°C)	Heat capacity flowrate (MW·K ⁻¹)
No.	Type			
1	Hot	500	100	4
2	Cold	50	450	1
3	Cold	60	400	1
4	Cold	40	420	0.75

Table 18.18 Heat flow cascade for exercise 9.

Interval temperature (°C)	Heat flow (MW)
490	0
460	120
430	210
410	255
90	655
70	600
60	582
50	575

- Design a heat exchanger network for this problem that achieves maximum energy recovery in the minimum number of units.
- Determine how much steam at a condensing temperature of 180°C can be generated by this process
- Sketch the composite curves for the process showing maximum steam generation at 180°C.
- Design a network that achieves maximum energy recovery in the minimum number of units and which generates the maximum possible steam at 180°C.

REFERENCES

- Linnhoff B and Hindmarsh E (1983) The Pinch Design Method of Heat Exchanger Networks, *Chem Eng Sci*, **38**: 745.
- Linnhoff B, Townsend DW and Boland D, Hewitt GF, Thomas BEA, Guy AR and Marsland RH (1982) *A User Guide on Process Integration for the Efficient Use of Energy*, IChemE, Rugby, UK.
- Linnhoff B and Ahmad S (1990) Cost Optimum Heat Exchanger Networks: I. Minimum Energy and Capital Using Simple Models for Capital Cost, *Comp Chem Eng*, **14**: 729.
- Saboo AK, Morari M and Colberg RD (1986) RESHEX: An Interactive Software Package for the Synthesis and Analysis of Resilient Heat Exchanger Networks: II. Discussion of area Targeting and Network Synthesis Algorithms, *Comp Chem Eng*, **10**: 591.
- Ahmad S, Linnhoff B and Smith R (1990) Cost Optimum Heat Exchanger Networks: II. Targets and Design for Detailed Capital Cost Models, *Comp Chem Eng*, **14**: 751.
- Gunderson T and Naess L (1988) The Synthesis of Cost Optimal Heat Exchanger Networks: An Industrial Review of the State of the Art, *Comp Chem Eng*, **12**: 503.
- Floudas CA, Ciric AR and Grossmann IE (1986) Automatic Synthesis of Optimum Heat Exchanger Network Configurations, *AIChE J*, **32**: 276.
- Grossmann IE (1990) Mixed-Integer Non-Linear Programming Techniques for the Synthesis of Engineering Systems, *Res Eng Des*, **1**: 205.
- Grossmann IE and Sargent RWH (1978) Optimum Design of Heat Exchanger Networks, *Comp Chem Eng*, **2**: 1.
- Yee TF, Grossmann IE and Kravanja Z (1990) Simultaneous Optimization Models for Heat Integration – I. Area and Energy Targeting of Modeling of Multi-stream Exchangers, *Comp Chem Eng*, **14**: 1151.
- Zhu XX, O'Neill BK, Roach JR and Wood RM (1995) New Method for Heat Exchanger Network Synthesis Using Area Targeting Procedures, *Comp Chem Eng*, **19**: 197.
- Dolan WB, Cummings PT and Le Van MD (1990) Algorithm Efficiency of Simulated Annealing for Heat Exchanger Network Design, *Comp Chem Eng*, **14**: 1039.
- Tjoe TN and Linnhoff B (1986) Using Pinch Technology for Process Retrofit, *Chem Eng*, **April**: 47.
- Asante NDK and Zhu XX (1997) An Automated and Interactive Approach for Heat Exchanger Network Retrofit, *Trans IChemE*, **75**: 349.
- Zhu XX and Asante NDK (1999) Diagnosis and Optimization Approach for Heat Exchanger Network Retrofit, *AIChE J*, **45**: 1488.