Chapter 22

Design Principles and Industrial Applications

INTRODUCTION

Current design practices for some heat exchangers usually fall into the category of state-of-the-art and pure empiricism. Past experience with similar applications is commonly used as the sole basis for the design procedure. The vendor maintains proprietary files on past exchanger installations; these files are periodically revised and expanded as new orders are evaluated. In designing a new exchanger, the files are consulted for similar applications and old designs are heavily relied on.

By contrast, the engineering profession in general, and the chemical engineering profession in particular, has developed well-defined procedures for most "standard" heat exchangers (e.g., double pipe, tube and bundle, boilers, etc.). These techniques, tested and refined for nearly a century, are routinely used by today's engineers.

The purpose of this chapter is to introduce the reader to some heat exchange process design principles and industrial applications. Such an introduction, however sketchy, can provide the reader with a better understanding of the major engineering aspects of a heat exchanger, including some of the operational, economics, controls and instrumentation for safety requirements, and any potential environmental factors associated with the unit. No attempt is made in the sections that follow to provide extensive coverage of this topic; only general procedures and concepts are presented and discussed.

Chapter contents include:

General design procedures

Process schematics

Purchasing a heat exchanger

Applications

The last section contains 24 Illustrative Examples that are concerned with heat exchanger design principles and/or industrial applications.

Heat Transfer Applications for the Practicing Engineer. Louis Theodore

GENERAL DESIGN PROCEDURES

There are usually five conceptual steps to be considered with the design of any equipment and they naturally apply to heat exchangers. These are:

1. Identification of the parameters that must be specified.
2. Application of the fundamentals underlying theoretical equations or concepts.
3. Enumeration, explanation, and application of simplifying assumptions.
4. Possible use of correction factors for non-ideal behavior.
5. Identification of other factors that must be considered for adequate equipment specification.

Calculation procedures for most of these have been presented earlier in this Part. Since design calculations are generally based on the maximum throughput capacity for the heat exchanger, these calculations are never completely accurate. It is usually necessary to apply reasonable safety factors when setting the final design. Safety factors vary widely and are a strong function of the accuracy of the data involved, calculational procedures, and past experience. Attempting to justify these is a difficult task.[1,2]

Unlike many of the problems encountered and solved by the engineer, there is no absolutely correct solution to a design problem; however, there is usually a *better* solution. Many alternative exchanger designs when properly implemented will function satisfactorily, but one alternative will usually prove to be economically more efficient and/or attractive than the others.

Overall and componential material balances have already been described in rather extensive detail in Part One. Material balances may be based on mass, moles, or volume, usually on a rate (time rate of change) basis. However, the material balance calculation should be based on mass or volume rates since both play an important role in equipment sizing calculations.

Some design calculations in the chemical process industry today include transient effects that can account for process upsets, startups, shutdowns, and so on. The describing equations for these time-varying (unsteady-state) systems are differential. The equations usually take the form of a first-order derivative with respect to time, where time is the independent variable. However, design calculations for almost all heat exchangers assume steady-state conditions, with the ultimate design based on worst-case or maximum flow conditions. This greatly simplifies these calculations since the describing equations are no longer differential, but rather algebraic.

The safe operation of the exchanger requires that the controls keep the system operating within a safe operating envelope. The envelope is based on many of the design, process, and (if applicable) regulatory constraints. The control system should also be designed to vary one or more of the process variables to maintain the appropriate conditions for the exchanger. These variations are often programmed into the system based on past experience with the specific unit. The operational parameters that may vary include the flow rates, temperatures, and system pressure.

The control system may be subjected to (extensive) analysis on operational problems and items that could go wrong. A hazard and operational (*HAZOP*) analysis (see Chapter 24) can be conducted on the system to examine and identify all possible failure mechanisms. It is important that all of the failure mechanisms have appropriate response reactions by the control system. Several of the failure mechanisms that must be addressed within the appropriate control system response are excess (excursion) or minimal temperature, excessive or subnormal flow rate, equipment failure, component failure, and broken circuits.

In the case of a heat exchanger design, data on similar existing units are normally available and economic estimates and/or process feasibility are determined from these data. It should be pointed out again that most heat exchangers in real practice are designed by duplicating or *mimicking* similar existing systems. Simple algebraic correlations that are based on past experience are the rule rather than the exception. This stark reality is often disappointing and depressing to students and novice engineers involved in design.

PROCESS SCHEMATICS

To the practicing engineer, particularly the process engineer, the process flow sheet is the key instrument for defining, refining, and documenting a process unit. The process flow diagram is the authorized process blueprint and the framework for specifications used in equipment designation and design; it is the single, authoritative document employed to define, construct, and operate the unit and/or process.[3]

Beyond equipment symbols and process stream flow lines, there are several essential constituents contributing to a detailed process flow sheet. These include equipment identification numbers and names, temperature and pressure designations, utility designations, flow rates for each process stream, and a material balance table pertaining to process flow lines. The process flow diagram may also contain additional information such as major instrumentation and physical properties of the process streams. When properly assembled and employed, a process schematic provides a coherent picture of the process. It can point out some deficiencies in the process that may have been overlooked earlier in the study. But basically, the flow sheet symbolically and pictorially represents the interrelation between the various flow streams and the exchanger (or any other equipment), and permits easy calculations of material and energy balances.

There are a number of symbols that are universally employed to represent equipment, equipment parts, valves, piping, and so on. Some of these are depicted in the schematic in Figure 22.1. Although there are a significant number of these symbols, only a few are needed for even the most complex heat exchanger unit. These symbols obviously reduce, and in some instances replace, detailed written descriptions of the unit or process. Note that many of the symbols are pictorial, which helps in better describing process components and information.

The degree of sophistication and details of an exchanger flow sheet usually vary with time. The flow sheet may initially consist of a simple free-hand block diagram with limited information that includes only the equipment; later versions may include

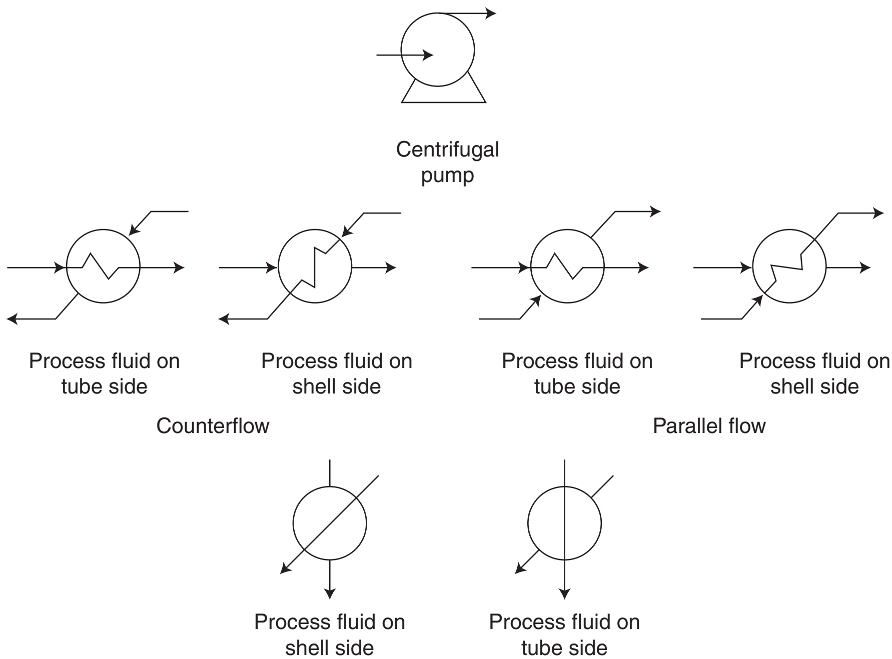

Figure 22.1 Selected flow sheet symbols.

line drawings with pertinent process data such as overall and componential flow rates, temperatures, pressures, and instrumentation. During the later stages, the flow sheet can consist of a highly detailed P&I (piping and instrumentation) diagram; this aspect of the design procedure is beyond the scope of this text; the reader is referred to the literature(4) for information on P&I diagrams.

In summary, industrial plant flow sheets are the international language of the engineer, particularly the chemical engineer. Chemical engineers conceptually view a (chemical) plant as consisting of a series of interrelated building blocks that are defined as *units* or *unit operations* (the heat exchanger is one such unit). The plant essentially ties together the various pieces of equipment that make up the process. Flow schematics follow the successive steps of a process by indicating where the pieces of equipment are located and the material streams entering and leaving each unit.

PURCHASING A HEAT EXCHANGER

Prior to the purchase of a heat exchanger, experience has shown that the following points should be emphasized:

1. Refrain from purchasing any heat exchanger without reviewing *certified independent test data* on its performance under a similar application.

Request the manufacturer to provide performance information and design specifications.

2. In the event that sufficient performance data are unavailable, request that the equipment supplier provide a small pilot model for evaluation under existing conditions.
3. Prepare a good set of specifications. Include a *strong performance guarantee* from the manufacturer to ensure that the heat exchanger will meet all design criteria and specific process conditions.
4. Closely review the overall process, other equipment, and economic fundamentals.
5. Make a careful material balance study.
6. Refrain from purchasing any heat exchanger until *firm* installation cost estimates have been added to the cost. *Escalating installation costs are the rule rather than the exception*.
7. Give operation and maintenance costs high priority on the list of exchanger selection factors.
8. Refrain from purchasing any heat exchanger until a solid commitment from the vendor(s) is obtained. Make every effort to ensure that the exchanger is compatible with the (plant) process.
9. The specification should include written assurance of *prompt* technical assistance from the supplier. This, together with a completely understandable operating manual (with parts list, full schematics, consistent units and notations, etc.), is essential and is too often forgotten in the rush to get the heat exchanger operating.
10. Schedules can be critical. In such cases, delivery guarantees should be obtained from the manufacturers and penalties identified.
11. The heat exchanger should be of fail-safe design with built-in indicators to show when performance is deteriorating.
12. Perhaps most importantly, withhold 10–15% of the purchase price until satisfactory operation is clearly demonstrated.

The usual design, procurement, installation, and/or startup problems can be further compounded by any one or a combination of the following:

1. Unfamiliarity of process engineers with heat exchangers.
2. New suppliers, frequently with unproven heat exchanger equipment.
3. Lack of industry standards with some designs.
4. Compliance schedules that are too tight.
5. Vague specifications.
6. Weak guarantees.
7. Unreliable delivery schedules.
8. Process reliability problems.

Proper selection of a particular heat exchanger for a specific application can be extremely difficult and complicated. The final choice in heat exchanger selection is usually dictated by that unit capable of achieving the aforementioned design criteria and required process conditions at the lowest uniform annual cost (amortized capital investment plus operation and maintenance costs—see also Chapter 27, Part Four).

In order to compare specific exchanger alternatives, knowledge of the particular application and site is also essential. A preliminary screening, however, may be performed by reviewing the advantages and disadvantages of each type or class of unit. However, there are many other situations where knowledge of the capabilities of the various options, combined with common sense, will simplify the selection process.

APPLICATIONS

The 24 Illustrative Examples to follow serve the function of merging many of the design procedures and industrial applications presented earlier in the text (including this chapter) into a more complete package.

ILLUSTRATIVE EXAMPLE 22.1

List four construction considerations/measures that should be followed during the design of a process/plant heat exchanger.

***SOLUTION*:**

1. Keep hot steam lines away from workers.
2. Insulate the entire exchanger so it is not hot to the touch.
3. Make sure all the flows entering the exchanger are turned off when it is being serviced.
4. Without compromising efficiency, allow adequate space inside the exchanger so that it can be easily cleaned. ■

ILLUSTRATIVE EXAMPLE 22.2

Discuss some overall mechanical considerations in heat exchanger design.

***SOLUTION*:** Choosing the most economical exchanger requires consideration of several mechanical specifications, but it is of prime importance to use standard designs. This narrows the choice but has the advantage of simplifying the job. The only exception is usually special designs. ■

ILLUSTRATIVE EXAMPLE 22.3

Discuss the difference between simulation and design.

***SOLUTION*:** Both simulation and design approaches are examined in this text.

Two factors should be considered in simulation (predicting performance):

1. Will the existing exchanger(s) transfer the duty required? This involves the calculation of all temperatures and heat transfer rates.
2. Will the pressure drop be within acceptable limits?

Thus, a simulation (or rating) problem involves analyzing the performance of an existing heat exchanger of known length and area.

In design, the exchanger does not exist. The duty or heat transfer rate is usually specified. The size and configuration of the exchanger must be specified subject to a number of other constraints. Thus, when the problem involves calculating the heat exchanger area needed to accomplish a set temperature change or a specific duty, it is termed a *design* problem. ■

ILLUSTRATIVE EXAMPLE 22.4

Discuss some key design specifications for combustion devices.

***SOLUTION*:** A key design specification for any combustion device, including heat exchangers, boilers, and incinerators, is the operating temperature. Materials of construction must withstand the operating temperature without experiencing any damage for safe and efficient plant operation. In addition, the volumetric flow rate is a strong function of temperature (see Chapter 5) with gaseous applications and plays an important role in properly sizing combustion/reactor equipment. ■

ILLUSTRATIVE EXAMPLE 22.5

List factor(s) that need to be considered when a heat exchanger is selected as an off-the-shelf unit or designed specifically for a particular application.

***SOLUTION*:**

1. Heat transfer rate requirements such as $\dot{Q}$.
2. Temperature limitations (safety and materials of construction).
3. Capital costs.
4. Operation costs.
5. Physical size.
6. Space limitations.
7. Pressure drop constraints (one large tube or many little tubes). ■

ILLUSTRATIVE EXAMPLE 22.6

It is desired to evaporate 1000 lb/h of 60°F water at 1 atm at a power plant. Utility superheated steam at 40 atm and 1000°F is available, but since this steam is to be used elsewhere in the plant, it cannot drop below 20 atm and 600°F. What mass flowrate of the utility steam is required? Assume that there is no heat loss in the evaporator.

From the steam tables:

$$P = 40\ \text{atm} \qquad T = 1000°\text{F} \qquad h = 1572\ \text{Btu/lb}$$
$$P = 20\ \text{atm} \qquad T = 600°\text{F} \qquad h = 1316\ \text{Btu/lb}$$

For saturated steam:

$$P = 1\ \text{atm} \qquad h = 1151\ \text{Btu/lb}$$

For saturated water:

$$T = 60°\text{F} \qquad h = 28.1\ \text{Btu/lb}$$

***SOLUTION*:** A detailed flow diagram of the process is provided in Figure 22.2. Assuming the process to be steady-state and noting that there is no heat loss or shaft work, the energy balance on a rate basis is

$$\dot{Q} = \Delta\dot{h} = 0$$

This equation indicates that the sum of the enthalpy changes for the two streams must equal zero.

The change in enthalpy for the vaporization of the water stream is

$$\begin{aligned}\Delta\dot{h}_{\text{vaporization}} &= (1000\ \text{lb/h})(1151 - 28.1\ \text{Btu/lb}) \\ &= 1.123 \times 10^6\ \text{Btu/h}\end{aligned}$$

The change of enthalpy for the cooling of the superheated steam may now be determined. Since the mass flowrate of the steam is unknown, its Δh must be expressed in terms of this mass flowrate, which is represented in Figure 22.2 as $\dot{m}$ lb/h.

$$\Delta\dot{h}_{\text{cooling}} = \dot{m}(1572 - 1316) = (256)\dot{m}\ \text{Btu/h}$$

Since the overall $\Delta\dot{H}$ is zero, the enthalpy changes of the two streams must total zero. Thus,

$$\begin{aligned}\Delta\dot{h}_{\text{vaporization}} + \Delta\dot{h}_{\text{cooling}} &= 0 \\ 1.123 \times 10^6 &= (256)\dot{m} \\ \dot{m} &= 4387\ \text{lb/h}\end{aligned}$$

■

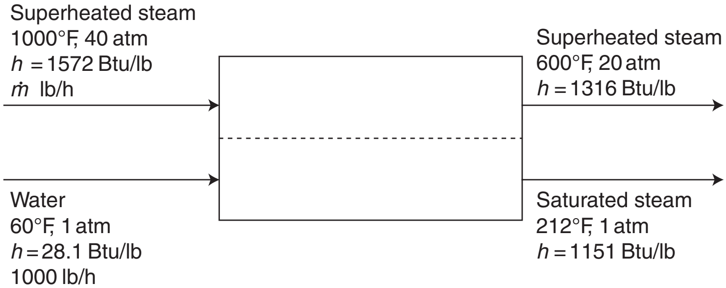

Figure 22.2 Flow diagram for Illustrative Example 22.6.

ILLUSTRATIVE EXAMPLE 22.7

Determine the total flowrate of cooling water required for the services listed below if a cooling tower system supplies the water at 90°F with a return temperature of 115°F. How much fresh water makeup is required if 5% of the return water is sent to "blowdown?" Note that the cooling water heat capacity is $1.00\ \text{Btu}/(\text{lb}\cdot{}^\circ\text{F})$, the heat of vaporization at cooling tower operating conditions is 1030 Btu/lb, and the density of water at cooling tower operating conditions is $62.0\ \text{lb}/\text{ft}^3$. Process data is provided in Table 22.1.

***SOLUTION*:** The required cooling water flowrate, q_{CW}, is given by the following equation:

$$q_{\text{CW}} = \dot{Q}_{\text{HL}}/[(\Delta T)(c_p)(\rho)]$$

where $\dot{Q}_{\text{HL}}$ = heat load, Btu/min

ΔT = change in temperature = 115°F − 90°F = 25°F

c_p = heat capacity = $1.00\ \text{Btu}/(\text{lb}\cdot{}^\circ\text{F})$

ρ = density of water = $(62.0\ \text{lb}/\text{ft}^3)(0.1337\ \text{ft}^3/\text{gal}) = 8.289\ \text{lb}/\text{gal}$

The heat load is

$$\begin{aligned}\dot{Q}_{\text{HL}} &= (12 + 6 + 23.5 + 17 + 31.5)(10^6\ \text{Btu/h})/60\ \text{min/h} \\ &= 1{,}500{,}000\ \text{Btu/min}\end{aligned}$$

Thus,

$$q_{\text{CW}} = \frac{1{,}500{,}000\ \text{Btu/min}}{(25^\circ\text{F})(1.00\ \text{Btu/lb}\cdot{}^\circ\text{F})(8.289\ \text{lb/gal})} = 7250\ \text{gpm}$$

The blow-down flow, q_{BD}, is given by the following:

$$q_{\text{BD}} = (\text{BDR})(q_{\text{CW}})$$

where BDR is the blow-down rate = 5% = 0.05. Thus,

$$q_{\text{BD}} = (0.05)(7250\ \text{gpm}) = 362.5\ \text{gpm}$$

■

Table 22.1 Data for Illustrative Example 22.7

Process unit	Heat duty, Btu/h	Required temperature, °F
1	12,000,000	250
2	6,000,000	200–276
3	23,500,000	130–175
4	17,000,000	300
5	31,500,000	150–225

ILLUSTRATIVE EXAMPLE 22.8

Determine how many pounds per hour of steam are required for the following situation if steam is provided at 500 psig, and if steam is provided at both 500 and 75 psig pressures. The plant has the heating requirements given in Table 22.2. Also note the properties of saturated steam in Table 22.3.

***SOLUTION*:** The total required flowrate of 500 psig steam, $\dot{m}_{\mathrm{BT}}$, is given by:

$$\dot{m}_{\mathrm{BT}} = \dot{m}_{\mathrm{B1}} + \dot{m}_{\mathrm{B2}} + \dot{m}_{\mathrm{B3}} + \dot{m}_{\mathrm{B4}}$$

For the above equation:

$$\dot{m}_{\mathrm{B1}} \text{ (mass flowrate of 500 psig steam through unit 1)} = \dot{Q}_1/h_{\mathrm{vap}} = 13{,}320\,\mathrm{lb/h}$$

$$\dot{m}_{\mathrm{B2}} \text{ (mass flowrate of 500 psig steam through unit 2)} = \dot{Q}_2/h_{\mathrm{vap}} = 10{,}655\,\mathrm{lb/h}$$

$$\dot{m}_{\mathrm{B3}} \text{ (mass flowrate of 500 psig steam through unit 3)} = \dot{Q}_3/h_{\mathrm{vap}} = 15{,}980\,\mathrm{lb/h}$$

$$\dot{m}_{\mathrm{B4}} \text{ (mass flowrate of 500 psig steam through unit 4)} = \dot{Q}_4/h_{\mathrm{vap}} = 26{,}635\,\mathrm{lb/h}$$

Thus,

$$\dot{m}_{BT} = \Sigma \dot{m}_{Bi} = 66{,}590\,\mathrm{lb/h}$$

■

ILLUSTRATIVE EXAMPLE 22.9

A feed stream to a distillation column processes of 28,830 gals/day of a light oil, and is fed to the column through a 6-inch I.D. pipeline. The temperature of the feed is usually 27°C. You are asked to explore the possibility of using either a countercurrent or a parallel (co-current) double pipe heat exchanger as an auxiliary heater for the feed stream. The double-pipe exchanger is to heat the oil from 23.5°C to 27°C. Water at 8406 gals/day and 93°C is available for heating the oil. The necessary physical properties are provided in Table 22.4. The inner diameter of the pipe forming the annular region is 20.3 cm and the outer surface of that pipe is well insulated. For a counter-current flow arrangement, calculate the following using the SI system:

1. The lowest temperature the exiting heating water could reach. Is there sufficient energy available from the water to raise the oil temperature? (If there is not enough energy, there is no point in designing the exchanger.)

Table 22.2 Process Data for Illustrative Example 22.8

Process unit	Unit heat duty ($\dot{Q}$), Btu/h	Required temperature, °F
1	10,000,000	250
2	8,000,000	450
3	12,000,000	400
4	20,000,000	300

Table 22.3 Steam Data

Pressure provided, psig	Saturation temperature, °F	Enthalpy of vaporization (h_{vap}), Btu/lb
75	320	894
500	470	751

2. The log mean temperature difference.
3. The overall heat transfer coefficient, U, for a new clean exchanger based on the inside area is 35.4 W/m$^2\cdot$K. Correct the U for the fouling of the heat exchanger. Use a fouling factor, R_f, of quenching oil, which is 0.0007 m$^2\cdot$K/W.
4. The length of the double pipe heat exchanger.
5. The effectiveness of the exchanger and the NTU.

***SOLUTION*:** Convert oil properties to SI units:

$$\rho_{\text{oil}} = 53\ \text{lb/ft}^3 = (53)(16.0185) = 849\ \text{kg/m}^3$$
$$c_{\text{oil}} = c_{p,\text{oil}} = 0.46\ \text{Btu/lb}\cdot{}^\circ\text{F} = (0.46)(4186.7) = 1926\ \text{J/kg}\cdot{}^\circ\text{C}$$
$$\mu_{\text{oil}} = 150\ \text{cP} = 0.15\ \text{kg/m}\cdot\text{s}$$
$$\nu_{\text{oil}} = \mu_{\text{oil}}/\rho_{\text{oil}} = 0.15/849 = 1.767\times10^{-4}\ \text{m}^2/\text{s}$$
$$k_{\text{oil}} = 0.11\ \text{Btu/h}\cdot\text{ft}\cdot{}^\circ\text{F} = (0.11)(1.7303) = 0.19\ \text{W/m}\cdot{}^\circ\text{C}$$
$$q_{\text{oil}} = \text{volumetric flow rate of oil} = (28{,}830)(4.381\times10^{-8}) = 0.001263\ \text{m}^3/\text{s}$$
$$\dot{m}_{\text{oil}} = \text{mass flow rate of oil} = \rho_{\text{oil}}q_{1,\text{oil}} = (849)(0.001263) = 1.072\ \text{kg/s}$$

Also convert water properties to SI units:

$$\rho_w = 964\ \text{kg/m}^3$$
$$c_w = c_{p,w} = 4204\ \text{J/kg}\cdot{}^\circ\text{C}$$
$$\begin{aligned}\mu_w &= 0.7\ \text{lb/ft}\cdot\text{h}\\ &= 0.0001944\ \text{lb/ft}\cdot\text{s}\\ &= (0.0001944)(1.4881)\\ &= 2.89\times10^{-4}\ \text{kg/m}\cdot\text{s}\end{aligned}$$
$$\nu_w = \mu_w/\rho_w = 3\times10^{-7}\ \text{m}^2/\text{s}$$
$$k_w = 0.678\ \text{W/m}\cdot{}^\circ\text{C}$$

Table 22.4 Property Data for Illustrative Example 22.9

Oil (27°C)	Water (93°C)	Pipe
$\rho = 53\ \text{lb/ft}^3$	$\rho = 964\ \text{kg/m}^3$	ID = 6.0 in
$c_p = 0.46\ \text{Btu/lb}\cdot{}^\circ\text{F}$	$c_p = 4204\ \text{J/kg}\cdot{}^\circ\text{C}$	OD = 168 mm
$\mu = 150\ \text{cP}$	$\mu = 0.7\ \text{lb/h}\cdot\text{ft}$	$k = 45\ \text{W/m}\cdot{}^\circ\text{C}$
$k = 0.11\ \text{Btu/h}\cdot\text{ft}\cdot{}^\circ\text{F}$	$k = 0.678\ \text{W/m}\cdot{}^\circ\text{C}$	

In addition,

$$
\begin{aligned}
q_w &= (8406)(4.38 \times 10^{-8}) = 3.683 \times 10^{-4}\,\mathrm{m^3/s} \\
\dot{m}_w &= \text{mass flow rate of water} \\
&= \rho_w q_w \\
&= (964)(3.683 \times 10^{-4}) \\
&= 0.355\,\mathrm{kg/s}
\end{aligned}
$$

Organize the above properties in tabular form (see Table 22.5).
Calculate the duty of the exchanger. Use the information on the oil side:

$$\dot{Q} = \dot{m}_{\text{oil}} c_{\text{oil}} (t_2 - t_1) = (2064.7)(27 - 23.5) = 7227.2\,\mathrm{W}$$

1. Determine the lowest possible temperature of the water. This corresponds to the inlet temperature of the oil:

$$T_{2,\min} = t_1 = 23.5^\circ\mathrm{C}$$

2. Check the maximum $\dot{Q}$ to be given off by the hot water:

$$\dot{Q}_{w,\max} = \dot{m}_w c_w (T_1 - T_{2,\min}) = (1492.4)(93 - 23.5) = 1.037 \times 10^5\,\mathrm{W}$$

Since $\dot{Q} = 7227.2\,\mathrm{W}$, there is enough energy in the water to heat the oil.
Calculate the exit temperature, T_2, of the water:

$$
\begin{aligned}
\dot{Q} &= 7227.2 = (1492.4)(T_1 - T_2) = (1492.4)(93 - T_2) \\
T_2 &= 88.16^\circ\mathrm{C}
\end{aligned}
$$

Calculate the log mean temperature difference:

$$
\begin{aligned}
\Delta T_1 \text{ (inlet)} &= 88.16 - 23.5 = 64.66^\circ\mathrm{C} \\
\Delta T_2 \text{ (outlet)} &= 93 - 27 = 66^\circ\mathrm{C}
\end{aligned}
$$

$$\Delta T_{\text{lm}} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\dfrac{\Delta T_1}{\Delta T_2}\right)} = \frac{(64.66 - 66)}{\ln\left(\dfrac{64.66}{66}\right)} = 65.33^\circ\mathrm{C}$$

Table 22.5 Key Properties and Data for Illustrative Example 22.9

	Stream			
Parameter	1	2	1	2
Type	Oil (being heated)		Water (being cooled)	
Side	Tube		Shell (annulus)	
ρ, $\mathrm{kg/m^3}$	849		964	
c, $\mathrm{J/kg \cdot K}$	1926		4204	
ν, $\mathrm{m^2/s}$	1.767×10^{-4}		3×10^{-7}	
$\dot{m}$, kg/s	1.072		0.355	
t, T, °C	23.5	27	93	
Capacitance rate, $C = \dot{m}c$, W/K	2064.7		1492.4	

3. Include the effects of fouling in the overall heat transfer coefficient U; use a fouling resistance, $R_f = 0.0007\ \text{m}^2 \cdot \text{K/W}$:

$$\frac{1}{U_{\text{dirty}}} = \frac{1}{U_{\text{clean}}} + R_f$$
$$= \frac{1}{35.4} + 0.0007 = 0.02822 + 0.0007 = 0.02892$$
$$U_{\text{dirty}} = 34.6\ \text{W/m}^2 \cdot {}^\circ\text{C (based on the inside area)}$$

4. Calculate the required heat transfer area and the length of the heat exchanger:

$$\dot{Q} = UA\Delta T_{\text{lm}}$$
$$7227.2 = (34.6)(A)(65.33)$$
$$A = 3.197\ \text{m}^2 = \pi D_1 L = \pi(0.1524)L$$
$$L = 6.68\ \text{m}$$

5. Calculate the maximum amount of heat absorbed by the oil. The maximum amount of heat is absorbed when the oil is heated to the maximum possible temperature, which is 93°C:

$$\dot{Q}_{\text{max,oil}} = (2064.7)(93 - 23.5) = 1.435 \times 10^5\ \text{W}$$

Select the $\dot{Q}_{\text{max}}$ for calculating the effectiveness. The values of $\dot{Q}_{\text{max,oil}}$ and $\dot{Q}_{\text{max},w}$ should be compared. The lower value is $\dot{Q}_{\text{max}}$.

$$\dot{Q}_{\text{max}} = \dot{Q}_{\text{max},w} = 1.037 \times 10^5\ \text{W}$$

The effectiveness, ε, is therefore:

$$\varepsilon = \dot{Q}/\dot{Q}_{\text{max}}$$
$$= 7227.2/(1.037 \times 10^5) = 0.0696 = 6.96\%$$

Calculate the number of transfer units, NTU, and the ratio of $(\dot{m}c)_{\text{max}}/(\dot{m}c)_{\text{min}}$:

$$\text{NTU} = U_{\text{dirty}}A/(\dot{m}c)_{\text{min}}$$
$$= (34.6)(3.197)/1492.4$$
$$= 0.0741$$

where the ratio of $(\dot{m}c)_{\text{max}}/(\dot{m}c)_{\text{min}} = \text{C}_{\text{max}}/\text{C}_{\text{min}} = 2064.7/1492.4 = 1.383$. ■

ILLUSTRATIVE EXAMPLE 22.10

Refer to Illustrative Example 22.9. How would the answer to Parts 2 and 4 be affected if the flow were parallel?

***SOLUTION*:** If the flow was cocurrent (parallel) in Illustrative Example 22.9,

$$\Delta T_1 = 93 - 23.5 = 69.5^\circ\text{C}$$
$$\Delta T_2 = 88.16 - 27 = 61.16^\circ\text{C}$$
$$\Delta T_{\text{lm}} = \frac{69.5 - 61.16}{\ln\left(\dfrac{69.5}{61.66}\right)} = 65.24^\circ\text{C}$$

and

$$\dot{Q} = UA\Delta T_{\text{lm}}$$
$$7227.2 = (34.6)A_h(65.24)$$
$$A = 3.2\,\text{m}^2 = \pi(0.1524)L$$
$$L = 6.69\,\text{m}$$

The tube length would increase slightly. ■

ILLUSTRATIVE EXAMPLE 22.11

How would the results of Illustrative Example 22.9 be affected if this flow were co-current?

***SOLUTION*:** Since co-current flow is the equivalent of parallel flow, the answer to the previous example applies. ■

ILLUSTRATIVE EXAMPLE 22.12

A brine solution at 10°F is heated in a food processing plant by flowing through a heated pipe. The pipe surface is maintained at 80°F. The pipe surface area for heat transfer is 2.5 ft^2. The brine solution (with a density of 62.4 lb/ft^3 and a heat capacity of 0.99 Btu/lb · °F) flows at a rate of 20 lb/min. The overall heat transfer coefficient varies linearly with the temperature approach, with values of 150 Btu/h · ft^2 · °F at the brine solution entrance (where the brine temperature is 10°F) and 140 Btu/h · ft · °F at the brine solution exit. Determine:

1. the temperature approach at the brine inlet side
2. the exit temperature of the brine solution.

***SOLUTION*:** Calculate the temperature approach at the pipe entrance:

$$\Delta T_1 = T - t_1 = 80 - 10 = 70^\circ\text{F} = 70/1.8 = 38.9^\circ\text{C}$$

Note that ΔT_2 $(T - t_2)$ cannot be calculated since t_2 is not known. Apply an energy balance to the brine solution across the full length of the pipe:

$$\dot{Q} = \dot{m}c(t_2 - t_1) = \dot{m}c(\Delta T_1 - \Delta T_2)$$
$$= (1200)(0.99)(70 - \Delta T_2) = 1188(70 - \Delta T_2)$$

Express the equation for the LMTD:

$$\Delta T_{\rm lm} = (70 - \Delta T_2)/\ln(70/\Delta T_2)$$

Also express the equation for the heat transfer rate:

$$\dot{Q} = (A)\frac{U_2(\Delta T_1) - U_1(\Delta T_2)}{\ln\left(\dfrac{U_2\Delta T_1}{U_1\Delta T_2}\right)} \tag{14.38}$$

$$= (2.5)\frac{140(70) - 150(\Delta T_2)}{\ln\left(\dfrac{(140)(70)}{(150)(\Delta T_2)}\right)}$$

Note that U varies linearly with ΔT.

Combine the above two equations and eliminate $\dot{Q}$:

$$1188(70 - \Delta T_2) = (2.5)\frac{(140)(70) - (150)(\Delta T_2)}{\ln\left(\dfrac{(140)(70)}{(150)(\Delta T_2)}\right)}$$

This equation is non-linear with one unknown (ΔT_2). This equation may be solved by trial-and-error. (Note that $0 \leq \Delta T_2 \leq 70°$F). Solution gives

$$\Delta T_2 \approx 51.6°\text{F} \approx 28.7°\text{C}$$

The temperature of the brine solution may now be calculated:

$$\Delta T_2 = 51.6 = T - t_2 = 80 - t_2$$
$$t_2 = 80 - 51.6 = 28.4°\text{F}$$
$$= (28.4 - 32)/1.8 = -2°\text{C}$$

■

ILLUSTRATIVE EXAMPLE 22.13

Refer to Illustrative Example 22.12. Calculate:

1. the rate of heat transfer, $\dot{Q}$
2. the log mean temperature difference.

***SOLUTION*:** The heat transfer rate, $\dot{Q}$, is

$$\dot{Q} = 1188(70 - 51.6) = 21{,}860\ \text{Btu/h}$$
$$= 21{,}860/3.412 = 6407\ \text{W}$$

and the LMTD is

$$\Delta T_{\rm lm} = (70 - 51.6)/\ln(70/51.6) = 60.3°\text{F}$$
$$= 60.3/1.8 = 33.5°\text{C}$$

■

ILLUSTRATIVE EXAMPLE 22.14

Describe the various types of heat exchangers that might be employed with a distillation unit.

***SOLUTION*:** See Figure 22.3. The five types include:

Standard exchanger
Heater
Condensor
Chiller
Reboiler ■

ILLUSTRATIVE EXAMPLE 22.15

Discuss industrial applications involving exchangers employed in parallel and series.(4)

***SOLUTION*:** In plants where large numbers of exchangers are used, certain size standards (total number of tubes, pass arrangements, baffle spacing) are established for 1–2 exchangers

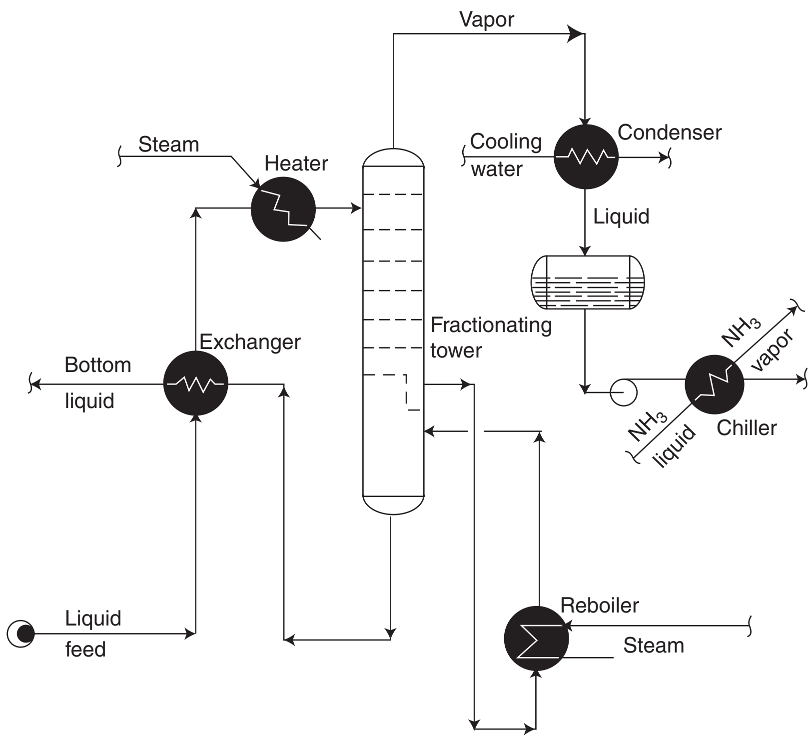

Figure 22.3 Heat exchanger use in a distillation unit.

so that a majority of future services can be fulfilled by an arrangement of a number of standard exchangers in series or parallel. Although this can sometimes cause a problem because of the impossibility of utilizing the equipment most efficiently, it has the advantage of reducing the type and number of replacement parts, tubes, tools, etc. When an exchanger has become obsolete in these plants, it is customary to find a number of exchangers of similar size available for other uses. If the tube bundle is simply retubed, the exchanger will frequently be as serviceable as when new. When two exchangers are connected in series on *both* the shell and tube sides, they form a temperature arrangement which has been shown to be similar to the 2–4 exchanger. When a temperature cross involves a correction factor for an arrangement which approximates true counterflow more closely than that possible in a 1–2 exchanger, it can be met by a series arrangement of a number of 1–2 exchangers. The 2–4, 3–6, 4–8, etc., arrangements are all based upon shells and channels being connected in series. Any arrangement which is an even multiple of two shell passes such as 2–4, 4–8, etc., may be employed by using a number of 1–2 exchangers or half as many 2–4 exchangers. ■

ILLUSTRATIVE EXAMPLE 22.16

Walas[(5)] has provided some simple “rules of thumb” for selecting and designing heat exchangers. Provide an outline of his suggestions.

***SOLUTION*:**

1. Take true countercurrent flow in a shell-and-tube exchanger as the basis for comparison.
2. Standard tubes are $\frac{3}{4}$-in OD, 1-in triangular spacing, 16 ft long; a shell 1 ft in diameter accommodates 100 ft^2; 2 ft diameter, 450 ft^2; 3 ft diameter, 1100 ft^2.
3. Tube side is for corrosive, fouling, scaling, and high-pressure fluids.
4. Shell side is for viscous and condensing fluids.
5. Pressure drops are approximately 1.5 psi for boiling liquids and 3–9 psi for other services.
6. Minimum temperature-approach is 20°F with normal coolants, 10°F or less with refrigerants.
7. Water inlet-temperature is 90°F, maximum outlet 120°F.
8. For estimating, use the following heat-transfer coefficients (Btu/h · ft^2 · °F): water-to-liquid, 150; condensers, 150; liquid-to-liquid, 50; liquid-to-gas, 5.0; gas-to-gas, 5.0; reboiler, 200. For the maximum flux in reboilers, use 10,000.
9. Double-pipe exchangers are competitive at duties requiring 100–200 ft^2.
10. Compact (plate or finned-tube) exchangers have 350 ft^2 of surface area/ft^3 of volume and about four times the heat transfer per cubic foot of shell-and-tube units.
11. Plate-and-frame exchangers are suited for sanitary (environmental) service and, in stainless steel, are 25–50% cheaper than shell and tube units.
12. For air coolers: tubes are 0.75–1.00 in OD; total finned surface is 15–20 ft^2/ft^2 of bare surface; the overall heat-transfer coefficient, $U = 80–100$ Btu/h · ft^2 bare surface · °F; fan power-input is 2–5 hp/million Btu · h; the approach is 50°F or more.
13. For fired heaters: radiant rate is 12,000 Btu/h · ft^2; convection rate, 4000 Btu/h · ft^2; cold oil-tube velocity, 6 ft/s; thermal efficiency, 70–75%; flue gas temperature,

250–350°F above feed inlet; stack-gas temperature 650–950°F. Approximately equal heat transfer occurs in both sections. ■

The following information applies to Illustrative Examples 22.17–22.22 below. A 1–2 shell and tube heat exchanger is employed to cool 1000 lb/h of hot process fluid. The hot process fluid is liquid water entering the unit at 212°F and is located on the shell side. The cold process fluid is liquid water, flowing at 2000 lb/h, entering at 35°F, and is located on the tube side. The tubes are $\frac{3}{4}$-inch OD, 18 BWG, 16 ft long and on a $\frac{15}{16}$-inch triangular pitch. The insulated shell is 15.25-inch ID and 25% percent segmental baffles are spaced 12 inches apart. The allowable fouling "factor" is 0.003 $\text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$ and the allowable pressure drop is 10 psi on both sides. Select the correct answers.

ILLUSTRATIVE EXAMPLE 22.17

What happens when the insulation is removed from the outside of the shell? Select the correct answer.

- **(a)** The hot fluid gains heat from the environment and the cold fluid exit temperature decreases.
- **(b)** The hot fluid gains heat from the environment and the cold fluid exit temperature increases.
- **(c)** The hot fluid loses heat to the environment and the cold fluid exit temperature decreases.
- **(d)** The hot fluid loses heat to the environment and the cold fluid exit temperature increases.
- **(e)** The hot fluid loses heat to the environment, but the cold fluid exit temperature remains the same.

***SOLUTION*:** Since insulation protects against heat loss, the hot fluid loses heat to the environment and the cold fluid exit temperature decreases. Therefore, the correct answer is (**c**). ■

ILLUSTRATIVE EXAMPLE 22.18

What happens when the baffle spacing is decreased to 3.5 inches? Select the correct answer.

- **(a)** The pressure drop on the tube side increases.
- **(b)** The cold fluid flow becomes more turbulent.
- **(c)** The pressure drop on the shell side decreases.
- **(d)** The hot fluid experiences a phase change.
- **(e)** The pressure drop on the shell side increases.

***SOLUTION*:** When baffles are moved closer, the fluid takes a more "torturous" path and a larger number of sudden changes in direction, resulting in a greater pressure drop. Therefore, the answer is (**e**). ■

ILLUSTRATIVE EXAMPLE 22.19

Which process and/or design scenario is more acceptable?

1. $R_f = 0.0001$ (consistent units) and $\Delta P_{total} = 4$ psi
2. $R_f = 0.0032$ (consistent units) and $\Delta P_{total} = 12$ psi

Select the correct answer.

(a) Scenario (1)
(b) Scenario (2)
(c) Neither scenario
(d) Both scenarios
(e) Not enough information provided

***SOLUTION*:** Scenario (**2**) is marginally acceptable because the total pressure drop is only 20% higher than allowed and the fouling factor varies by less than 7%. However, the fouling factor in scenario (**1**) is an order of magnitude lower than allowed. Therefore, the correct answer is (**a**). ■

ILLUSTRATIVE EXAMPLE 22.20

What happens if the hot process fluid is replaced by an equal amount of steam? Select the correct answer.

(a) The exit temperature of the cold fluid decreases.
(b) The exit temperature of the cold fluid increases.
(c) There is less heat transfer to the cold fluid.
(d) There is no effect.
(e) Not enough information provided.

***SOLUTION*:** The temperature of the steam is higher and, as the steam condenses, there is a larger temperature driving force resulting in a greater heat transfer. As a result, the exit temperature of the cold fluid must increase. Therefore, the correct answer is (**b**). ■

ILLUSTRATIVE EXAMPLE 22.21

How would the design have changed if the original hot process fluid had been 1000 lb/h of steam? Select the correct answer.

(a) The number of tube passes would be increased.
(b) A shorter heat exchanger is needed.
(c) There is less heat transfer to the cold fluid.
(d) A longer heat exchanger is needed.
(e) No change is needed.

***SOLUTION*:** Using steam provides a greater temperature difference driving force and would require less heat transfer area, resulting in the design of a shorter heat exchanger. Therefore, the correct answer is (**b**). ■

ILLUSTRATIVE EXAMPLE 22.22

Is the design of a shell and tube heat exchanger affected by the direction of flow of the process fluids? Select the correct answer.

- **(a)** Yes, because the heat transfer area is affected by the log-mean temperature difference, which depends on the direction of flow.
- **(b)** Yes, because the heat transfer area is a function of fouling, which is dependent on the direction of flow.
- **(c)** No, because the direction of flow only affects the number of tube passes.
- **(d)** No, there is no effect.
- **(e)** It depends on numerous other factors.

***SOLUTION*:** Yes, because the heat transfer area is affected by the log-mean temperature difference, which depends on the direction of flow. Countercurrent flow results in a larger temperature driving force. Therefore the correct answer is (**a**). ■

ILLUSTRATIVE EXAMPLE 22.23

Heating or cooling of liquids in batch processes is used in a number of commercial applications. Some reasons for using a batch rather than a continuous heat transfer operation include: the liquid is not continuously available, the liquid cleaning and regeneration is a significant part of the operation, or batch operation is simpler and cheaper.

For example, vapor degreasers are widely used for cleaning metal parts. A degreaser consists of a tank partially filled with a solvent. The tank is equipped with a heating coil to heat the solvent close to its boiling point. The vapor of the solvent occupies the remaining volume of the tank, forming the "solvent vapor zone." When a metal part is placed in the solvent vapor zone, the solvent condenses on the metal part and then drips off, taking contaminants with it. For ease of use, vapor degreasers are often open to the atmosphere. This makes it easier to introduce and remove the metal parts. It has been a common practice to use a halogenated hydrocarbon for such cleaning since they are excellent solvents, volatile, and non-flammable; however, they can be toxic and the open tank of a degreaser can be a significant source of solvent emissions or volatile organic components (VOCs).

When an agitated liquid batch, initially at T_i, is heated, the temperature, T, of the liquid at any time, t, can be assumed to be uniform if the liquid is well stirred. Three different types of calculations involving the batch heating of liquids are outlined in Table 22.6.

The assumptions in these analyses include:

1. constant properties for the duration of the process,
2. constant overall heat transfer coefficient, U, for the process,
3. a constant temperature, T_∞, of the heating medium,

Table 22.6 Batch Heating of Liquids

Case	Description	Calculate
Design	It is desired to heat the liquid to a specified temperature within a given time.	The surface area required.
Simulation	It is desired to heat the liquid for a specified time; the heat transfer surface area is known.	The final temperature, T.
Time	The heat transfer surface area and the final temperature are known.	The batch time, t.

4. a vigorously stirred unit (and therefore a uniform temperature),
5. negligible heat losses, and
6. no phase change.

The describing equation for these batch systems can be developed. Consider a liquid batch that has a volume, V, density, ρ, and heat capacity, c. The initial temperature is T_i, and the temperature at any time, t, is T. The energy balance on the liquid in the agitated tank simplifies to

$$UA(T_\infty - T) = \rho V c_p \frac{dT}{dt} \tag{1}$$

An excess temperature, θ, can be defined as

$$\theta = T_\infty - T; \quad \theta_i = T_\infty - T_i, \quad T_i = \text{initial temprature} \tag{2}$$

Substituting this expression into Equation (1) and integrating with respect to t yields

$$\frac{\theta}{\theta_i} = e^{-t/\tau} \tag{3}$$

where τ_t is the thermal time constant that is given by

$$\tau = \rho V c_p / UA \tag{4}$$

Design a degreaser employing the development provided above; i.e., calculate the heating surface area requirement for a vapor degreaser using 1,1,1-trichloroethane, $C1_3C\text{-}CH_3$ (TCA) as the solvent. The degreaser tank has a rectangular cross section (6 ft × 3 ft) and a height of 5 ft. Liquid TCA at 18°C is poured into the tank to a height of 1 ft. The solvent vapor zone will occupy the remaining 4 ft. At time $t = 0$, saturated steam is passed into the heating coil. The steam temperature is 100°C. The overall heat transfer coefficient, U, from the steam to the liquid TCA, based on the outside area of the coil, is 200 Btu/h · ft^2 · °F. It is desired to heat the liquid TCA from 18°C to its boiling point (74°C) without evaporation in 180 seconds. The approximate properties of TCA are: density, $\rho = 87.4$ lb/ft^3, heat capacity, $c_p = 0.23$ Btu/lb · °F, and viscosity = 0.56 cP. The design should include:

1. the required surface area of the heating coil
2. the total heat added to the liquid TCA.

***SOLUTION*:** Calculate the initial and final excess temperatures:

$$\theta_i = T_\infty - T_i = 100 - 18 = 82°\text{C} = 147.6°\text{F}$$
$$\theta_f = T_\infty - T_f = 100 - 74 = 26°\text{C} = 46.8°\text{F};\ T_f = \text{final temperature}$$

Calculate t/τ_t and the thermal time constant:

$$\begin{aligned} t/\tau &= \ln(\theta_i/\theta_f) \\ &= \ln[(8.2)/(2.6)] \\ &= 1.149 \\ \tau &= t/1.149 \\ &= 180/1.149 \\ &= 156.7\ \text{s} \end{aligned} \tag{3}$$

The required heating area is given by

$$\begin{aligned} A &= \rho V c_p / U\tau \\ &= (87.4)(18)(0.23)/(3600)(200)(156.7) \\ &= 41.6\ \text{ft}^3 \end{aligned}$$

This is the required area of the outside surface of the heating oil. Convert the final and initial temperatures to °F:

$$\begin{aligned} T_f &= 18°\text{C} = 46.8°\text{F} \\ T &= 74°\text{C} = 147.6°\text{F} \end{aligned}$$

The total amount of heat added is then:

$$\begin{aligned} Q &= \rho V c_p (T_f - T_i) \\ &= (87.4)(18)(0.23)(147.6 - 46.8) \\ &= 36{,}469\ \text{Btu} \end{aligned}$$

■

ILLUSTRATIVE EXAMPLE 22.24

62,000 lb/h of pure ethyl alcohol ($h_{\text{vap}} = 365$ Btu/lb; specific gravity, s = 0.79) at 2.0 psig is to be condensed by water ($\rho = 62.5$ lb/ft^3) entering at 85°F and exiting at 120°F. A 1–2 horizontal condenser consists of 700(N_t) one-inch outside diameter, 14 BWG tubes, 16-inch long on a 1.25-inch triangular pitch. There are four (n) tube passes. Assume that the flow is countercurrent and that the alcohol is on the shell side. A fouling factor of 0.003 Btu/h · ft^2 · °F is recommended. The cooling water inside film coefficient has been previously determined to be 862.4 Btu/h · ft^2 · °F. Neglecting any pressure drop considerations, determine if the exchanger is suitable. Additional information is provided in Table 22.7.

Table 22.7 Exchanger Information for Illustrative Example 22.24

Shell	Tubes
ID = 39 inch	Pitch = 1.25 inch
Baffle spacing = 39 inch	Flow area/tube = 0.546 in^2
Flow area = 2.11 ft^2	ID = 0.834 inch
Condensation temperature, $t_c = 173°$F	Wall thickness = 0.083 inch

For film condensation on horizontal tubes (see also Chapter 12), the average heat transfer coefficient is to be calculated from the equation (consistent units):

$$\bar{h} = 1.51\left[\frac{k_f \rho_f^2 g \mu_f}{\mu_f^2 4G'}\right]^{1/3}$$

where k_f = thermal conductivity
ρ_f = fluid density
g = acceleration due to gravity
μ_f = viscosity of the fluid
G' = loading, $G' = (\dot{m}_{\text{alcohol}}/LN_t^{2/3})$
$\dot{m}_{\text{alcohol}}$ = condensate mass flow
L = tube length
N_t = number of tubes in bundle
V = velocity

For a clean (unused) tube, the overall heat transfer coefficient can be calculated from the following equation:

$$U_C = U_{\text{clean}} = \frac{h_{io}h_o}{h_{io} + h_o}$$

where $U_C = U_{\text{clean}}$ = heat transfer coefficient of a clean (new/unused) tube
h_{io} = corrected inside heat transfer coefficient to the outside diameter, with $h_{io} = h_i(D_i/D_o)$

The dirt fouling factor may be calculated from:

$$R_D = R_{\text{dirty}} = \frac{U_c - U}{U_c U} \tag{14.32}$$

where $R_D = R_{\text{dirty}}$ = dirt (fouling) factor
U = heat transfer coefficient calculated from the actual heat transfer

***SOLUTION*:** Calculate heat loss from the alcohol:

$$\dot{Q}_{\text{alcohol}} = \dot{m}_{\text{alcohol}} h_{\text{vap}}$$
$$\dot{Q}_{\text{alcohol}} = (62{,}000\ \text{lb/h})(365\ \text{Btu/lb}) = 22{,}630{,}000\ \text{Btu/h}$$

Also, calculate the water mass flow rate. Since the heat lost from alcohol equals the heat gained by water:

$$\dot{m}_{H_2O} = \frac{\dot{Q}_{H_2O}}{c_p \Delta t}$$
$$= \frac{22{,}630{,}000\ \text{Btu/h}}{(1.0\ \text{Btu/lb}\cdot{}^\circ\text{F})(120^\circ\text{F} - 85^\circ\text{F})} = 646{,}571\ \text{lb/h}$$

Calculate the log-mean temperature difference:

$$\text{LMTD} = \frac{88^\circ\text{F} - 53^\circ\text{F}}{\ln(88^\circ\text{F}/53^\circ\text{F})} = 69^\circ\text{F}$$

Determined the total flow area for both the shell side and the tube side. The flow area for the shell side is given:

$$a_s = 2.11\ \text{ft}^2$$

The total flow area for the tube side can be calculated from the following equation:

$$a_t = \frac{N_t a_t}{144(n)}$$
$$= \frac{(700\ \text{tubes})(0.546\ \text{in}^2/\text{tube})}{(144\ \text{in}^2/\text{ft}^2)(4\ \text{passes})}$$
$$= 0.664\ \text{ft}^2$$

Use the above two areas to calculate the mass velocity of the flow in the shell side and the tube side:

$$G_s = \frac{\dot{m}_{\text{alcohol}}}{a_s}$$
$$= \frac{62{,}000\ \text{lb/h}}{(2.11\ \text{ft}^2)}$$
$$= 29{,}384\ \text{lb/h}\cdot\text{ft}^2$$
$$G_t = \frac{\dot{m}_{H_2O}}{a_t}$$
$$= \frac{646{,}571\ \text{lb/h}}{0.664\ \text{ft}^2}$$
$$= 973{,}752\ \text{lb/h}\cdot\text{ft}^2$$

Calculate the velocity of water in the tubes:

$$V_{H_2O} = \frac{G_t}{3600\rho}$$
$$= \frac{973{,}752\ \text{lb/h}\cdot\text{ft}^2}{(3600\ \text{s/h})(62.4\ \text{lb/ft}^3)}$$
$$= 4.33\ \text{ft/s}$$

Also, calculate loading G':

$$G' = \frac{\dot{m}_{\text{alcohol}}}{LN_t^{2/3}} = \frac{62{,}000\ \text{lb/h}}{(16\ \text{ft})(700\ \text{tubes})^{2/3}} = 49.15\ \text{lb/h} \cdot \text{ft}$$

Determine the heat transfer coefficient for the shell side:

$$\bar{h} = 1.51\left[\frac{k_f^3 \rho_f^2 g \mu_f}{\mu_f^2 4G'}\right]^{1/3}$$

For the alcohol:

$$k_f = 0.105\ \text{Btu/h} \cdot \text{ft} \cdot {}^\circ\text{F}$$
$$\mu_f = (0.55)(2.42) = 1.331\ \text{lb/ft} \cdot \text{h}$$
$$s_f = 0.79$$
$$\rho_f = (0.79)(62.5\ \text{lb/ft}^3) = 49.375\ \text{lb/ft}^3$$

Substituting,

$$\bar{h} = 1.51\left[\frac{(0.105\ \text{Btu/h} \cdot \text{ft} \cdot {}^\circ\text{F})^3(49.375\ \text{lb/ft}^3)^2(32.4\ \text{ft/s}^2)(3600\ \text{s/h})^2}{(1.331\ \text{lb/ft} \cdot \text{h})(4)(49.15\ \text{lb/ft} \cdot \text{h})}\right]^{1/3}$$
$$\bar{h} = h_o = 249.8\ \text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$$

In this case, the average heat transfer coefficient of the film is the outside heat transfer coefficient of the tube bundle. Use this result to calculate the overall heat transfer coefficient for a new (clean) heat exchanger

$$U_C = \frac{h_{io} h_o}{h_{io} + h_o} = \frac{(249.8\ \text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F})(862.4\ \text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F})}{(249.8\ \text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}) + (862.4\ \text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F})} = 193\ \text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$$

Calculate the design (D) overall heat transfer coefficient:

$$A = (700\ \text{tubes})(16\ \text{ft})(0.2618\ \text{ft}) = 2932\ \text{ft}^2$$
$$U_D = \frac{\dot{Q}}{A\Delta T}$$

Substituting

$$U_D = \frac{22{,}630{,}000\ \text{Btu/h}}{(2932\ \text{ft}^2)(69^\circ\text{F})}$$
$$= 111.86\ \text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}$$

Finally, calculate the dirt (d) factor and determine whether the heat exchanger is suitable for the process conditions using the equation provided:

$$R_D = \frac{U_C - U_D}{U_C U_D} \tag{14.32}$$
$$= \frac{(193\ \text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}) - (111.86\ \text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F})}{(193\ \text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F})(111.86\ \text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F})}$$
$$= 0.0038\ (\text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F})$$

Therefore, the exchanger as specified is unsuitable for these process conditions since the fouling factor is above the recommended value. Cleaning is recommended. ■

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