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CHAPTER EIGHT

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 transfer in a heat exchanger usually involves convection in each fluid and conduction through the wall separating the two fluids.

8.1 Heat Exchanger Types

Heat exchangers are typically classified according to flow arrangement and type of construction. The simplest type of heat exchanger consists of two concentric pipes of differentdiameters, as shown in Figure (8.1), called the double-pipe heatexchanger.

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.

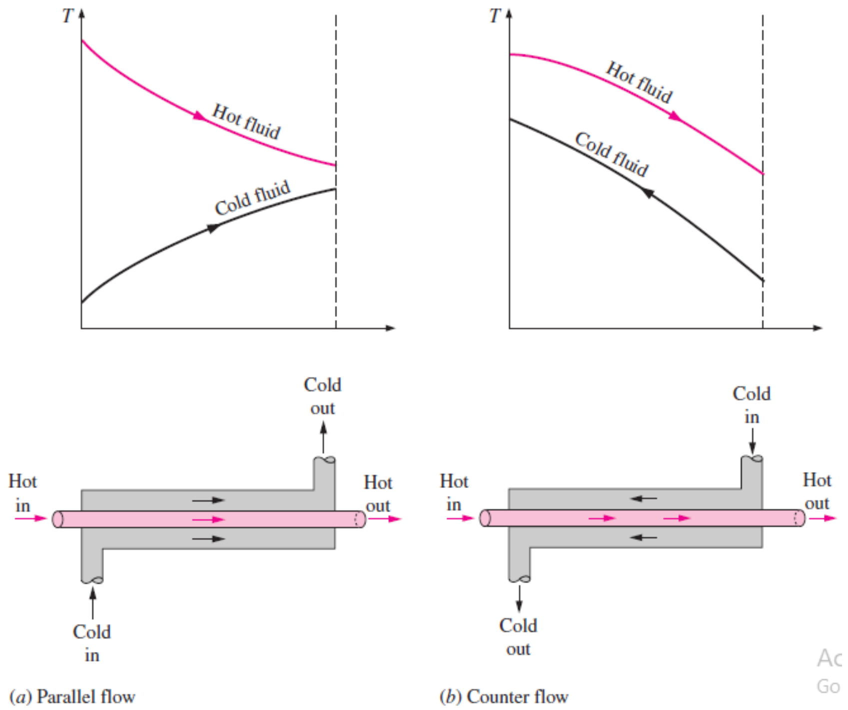


Figure (8.1) Different flow regimes and associated temperature profiles in a double-pipe heat exchanger.

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. In compact heat exchangers, the two fluids usually move perpendicular to each other, and such flow configuration is called cross-flow. commonly used in applications with strict limitations on the weight and volume of heat exchangers. The cross-flow is further classified as unmixed and mixed flow, depending on the flow configuration, as shown in Figure (8.2).

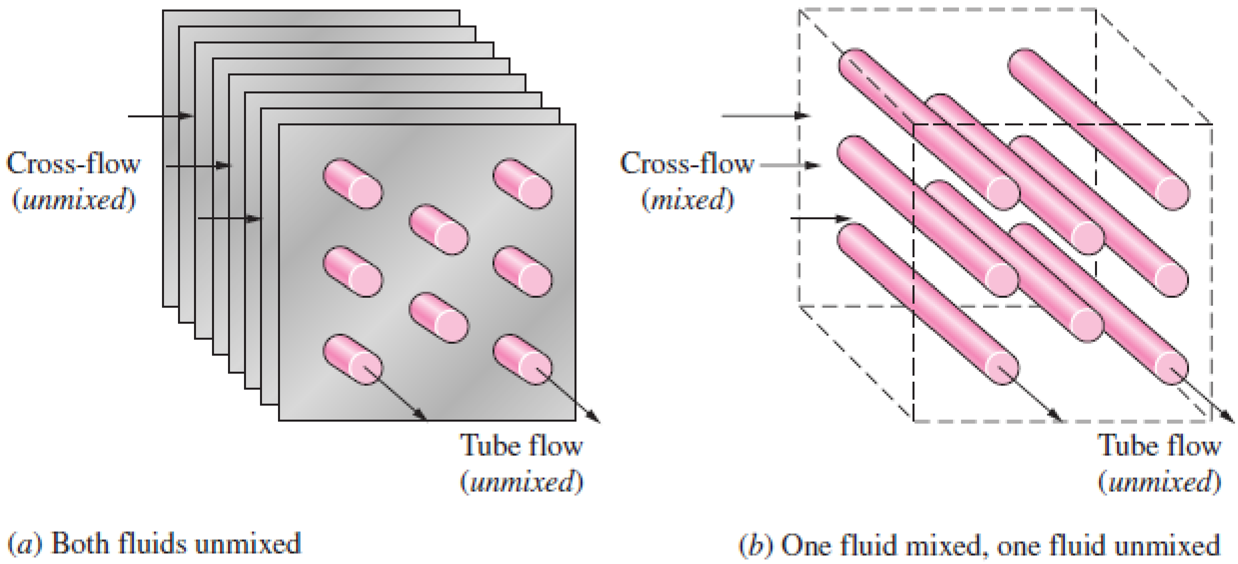


Figure (8.2) Different flow configurations in cross-flow heat exchangers.

Perhaps the most common type of heat exchanger in industrial applications is the shell-and-tube heat exchanger, shown in Figure (8.3). 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. Despite their widespread use, shelland-tube heat exchangers are not suitable for use in automotive and aircraft applications because of their relatively large size and weight.

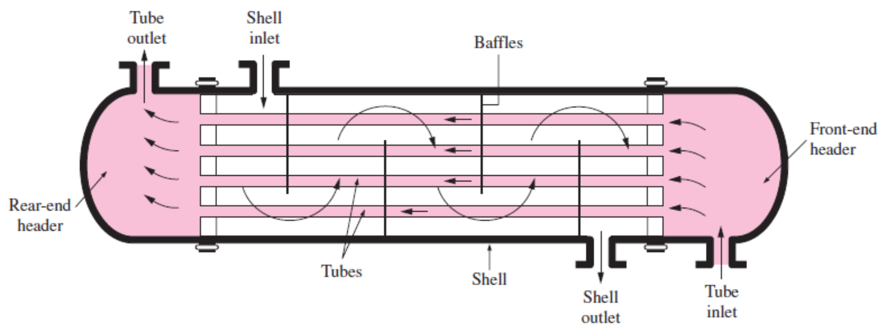


Figure (8.3) The schematic of a shell-and-tube heat exchanger (one-shell pass and one-tube pass).

Shell-and-tube heat exchangers are further classified according to the number of shell and tube passes involved shown in Figure (8.4).

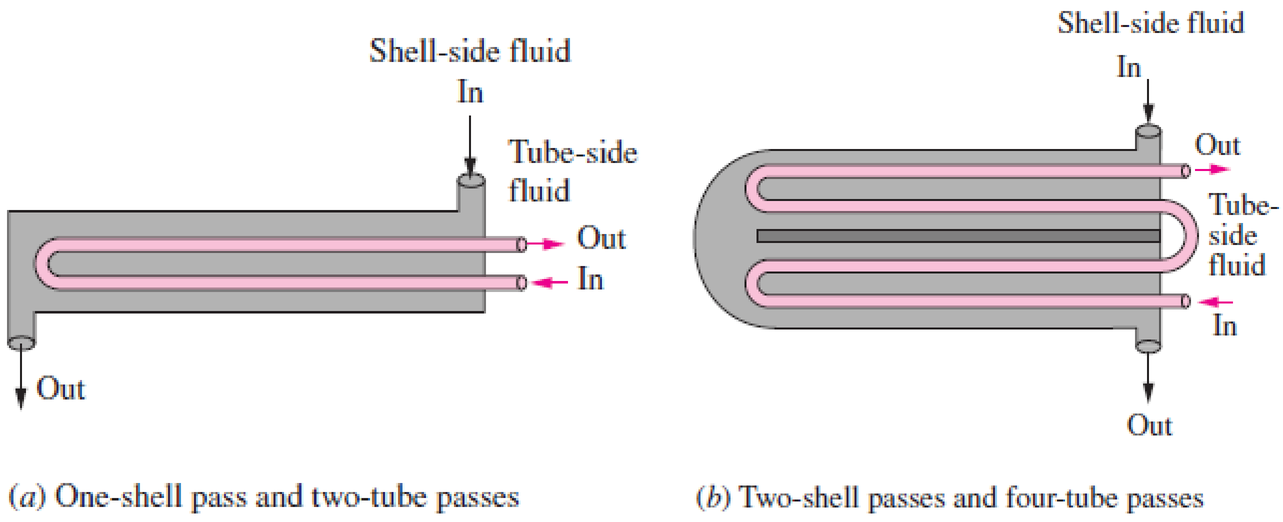


Figure (8.4) Multipass flow arrangements in shelland tube heat exchangers.

8.2 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. The thermal resistance network associated with this heat transfer process involves two convection and one conduction resistances, as shown in Figure (8.5).

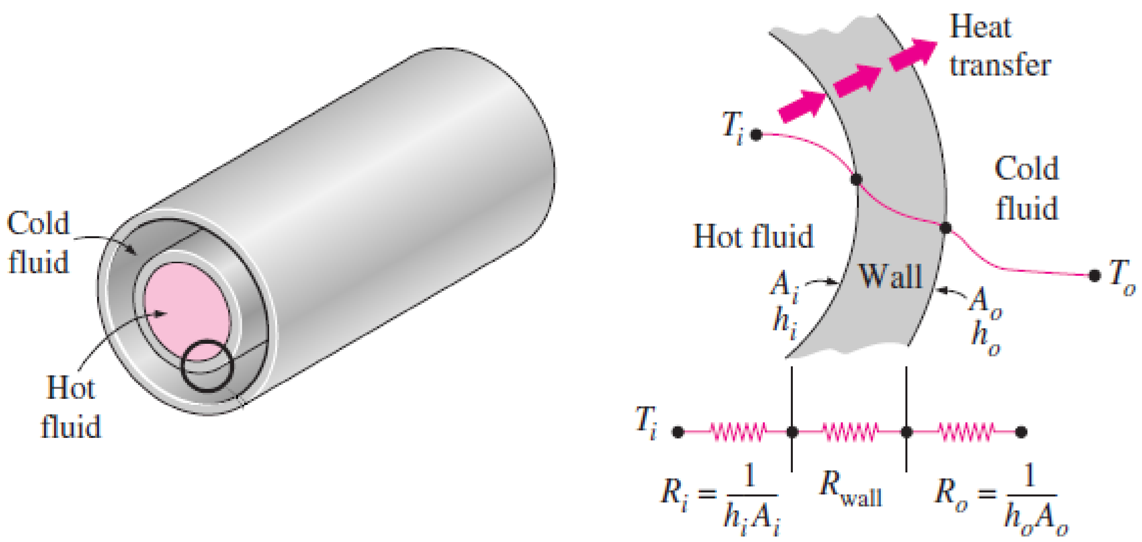


Figure (8.5) Thermal resistance network associated with heat transfer in a double-pipe heat exchanger.

For a double-pipe heat exchanger shown in Figure (8.6) the total thermal resistance becomes

$$R = R_{\text{total}} = R_i + R_{\text{wall}} + R_o = \frac{1}{h_i A_i} + \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{1}{h_o A_o} \qquad (8.1)$$

Where

A_i: is the area of the inner surface of the wall that separates the two fluids ($A_i = \pi D_i L$).

A_o: is the area of the outer surface of the wall ($A_0 = \pi D_0 L$).

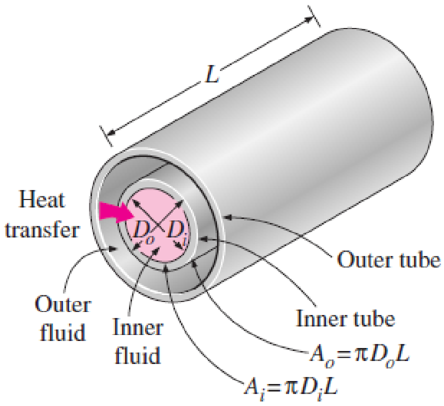


Figure (8.6) The two heat transfer surface areas associated with a double-pipe heat exchanger.

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

$$Q = \frac{\Delta T}{R} = UA\,\Delta T = U_i A_i\,\Delta T = U_o A_o\,\Delta T \qquad (8.2)$$

where U: is the overall heat transfer coefficient (W/m^2.°C).

$$\frac{1}{UA_s} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o} = R = \frac{1}{h_i A_i} + R_{\text{wall}} + \frac{1}{h_o A_o} \qquad (8.3)$$

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 ($R_{wall} \approx 0$)

$$R_{wall} = \frac{\ln(D_o / D_i)}{2\pi k L} \qquad (8.4)$$

The inner and outer surfaces of the tube are almost identical ($A_i \approx A_o \approx A_s$). Then the overall heat transfer coefficient simplifies to:

$$\frac{1}{U} \approx \frac{1}{h_i} + \frac{1}{h_o} \qquad (8.5)$$

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 overall heat transfer coefficient relation given above 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 shell-and-tube heat exchanger, it can be expressed as

$$\frac{1}{UA_s} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o} = R = \frac{1}{h_i A_i} + \frac{R_{f,i}}{A_i} + \frac{\ln(D_o/D_i)}{2\pi k L} + \frac{R_{f,o}}{A_o} + \frac{1}{h_o A_o} \qquad (8.6)$$

where $R_{f,i}$ and $R_{f,0}$ are the fouling factors at inner and outer surfaces respectively.

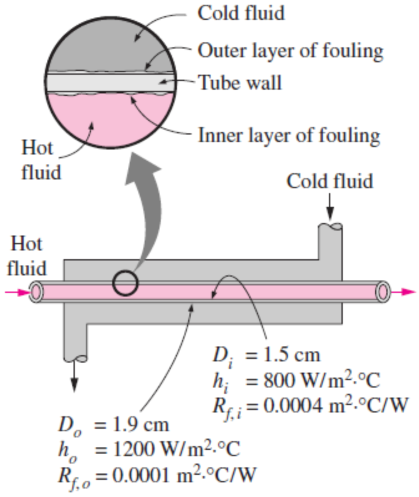
Example (8.1): A double-pipe (shell-and-tube) heat exchanger is constructed of a stainless steel ($k = 15.1\ W/m \cdot °C$) inner tube of inner diameter ($D_i = 1.5\ cm$) and outer diameter ($D_o = 1.9\ cm$) and an outer shell of inner diameter (3.2 cm). The convection heat transfer coefficient is given to be ($h_i = 800\ W/m^2.°C$) on the inner surface of the tube and ($h_o = 1200\ W/m^2.°C$) on the outer surface. For a fouling factor of ($R_{f,i} = 0.0004\ m^2.°C/W$) on the tube side and ($R_{f,0} = 0.0001\ m^2.°C/W$) 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:

(a)

$$R = \frac{1}{UA_s} = \frac{1}{U_iA_i} = \frac{1}{U_oA_o} = \frac{1}{h_iA_i} + \frac{R_{f,i}}{A_i} + \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{R_{f,o}}{A_o} + \frac{1}{h_oA_o}$$

$$A_i = \pi D_i L = \pi(0.015\ \text{m})(1\ \text{m}) = 0.0471\ \text{m}^2$$
$$A_o = \pi D_o L = \pi(0.019\ \text{m})(1\ \text{m}) = 0.0597\ \text{m}^2$$



$$R = \frac{1}{800 * 0.0471} + \frac{0.0004}{0.0471} + \frac{\ln(1.9/1.5)}{2\pi * 15.1 * 1} + \frac{1}{1200 * 0.0597} + \frac{0.0001}{0.0597}$$

$$R = 0.0532\ °C/W$$

(b)

$$U_i = \frac{1}{RA_i} = \frac{1}{0.0532 * 0.0471} = 399\ W/m^2.°C$$

$$U_0 = \frac{1}{RA_0} = \frac{1}{0.0532 * 0.0597} = 315 W/m^2.°C$$

8.3 Analysis of Heat Exchangers

How to select a heat exchanger that will achieve a specified temperature change in a fluid stream of known mass flow rate?

How to predict the outlet temperatures of the hot and cold fluid streams in a specified heat exchanger?

There are two methods used in the analysis of heat exchangers:

1- Log mean temperature difference (or LMTD) method is best suited for the first task.
2- Effectiveness–NTU (Number of Transfer Units) method.

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.

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.

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,

$$q = \dot{m}_c C_{pc}(T_{c,out} - T_{c,in}) \qquad (8.7)$$

and

$$q = \dot{m}_h C_{ph}(T_{h,in} - T_{h,out}) \qquad (8.8)$$

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

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

$$C_h = \dot{m}_h C_{ph} \qquad \text{and} \qquad C_c = \dot{m}_c C_{pc} \qquad (8.9)$$

With the definition of the heat capacity rate above,

$$q = \dot{m}_c C_{pc}(T_{c,out} - T_{c,in}) \quad \text{and} \quad q = \dot{m}_h C_{ph}(T_{h,in} - T_{h,out})$$

Note that the only time the temperature rise of a cold fluid (T_{co}-T_{ci}) is equal to the temperature drop of the hot fluid (T_{hi}-T_{ho}) is when the heat capacity rates of the two fluids are equal to each other Figure (8.7).

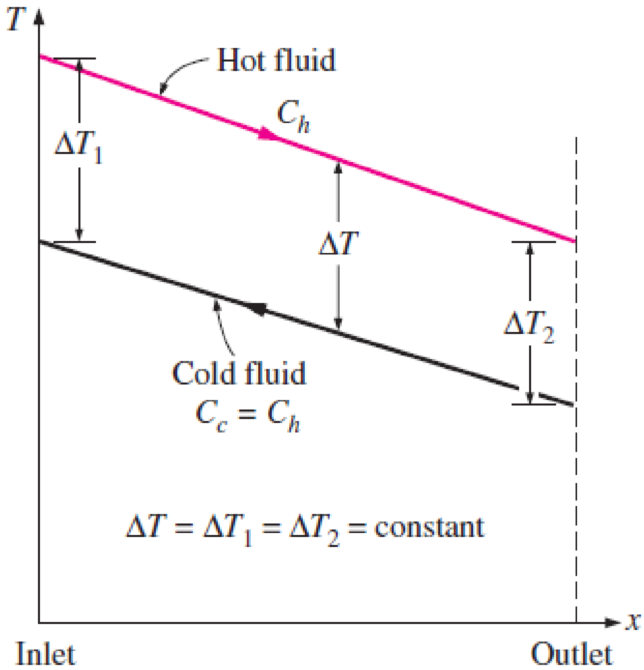


Figure (8.7) Two fluids that have the same mass flow rate and the same specific heat experience the same temperature change in a well-insulated heat exchanger

8.4 The Log Mean Temperature Difference Method (LMTD)

Log Mean Temperature Difference Method 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.

When ΔT_{lm}, mass flow rates, and overall heat transfer coefficient are available, the heat transfer surface area of the heat exchanger can be determined from

$$q = UA_s \Delta T_{lm} \qquad (8.10)$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} \qquad (8.11)$$

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 Figure (8.8).

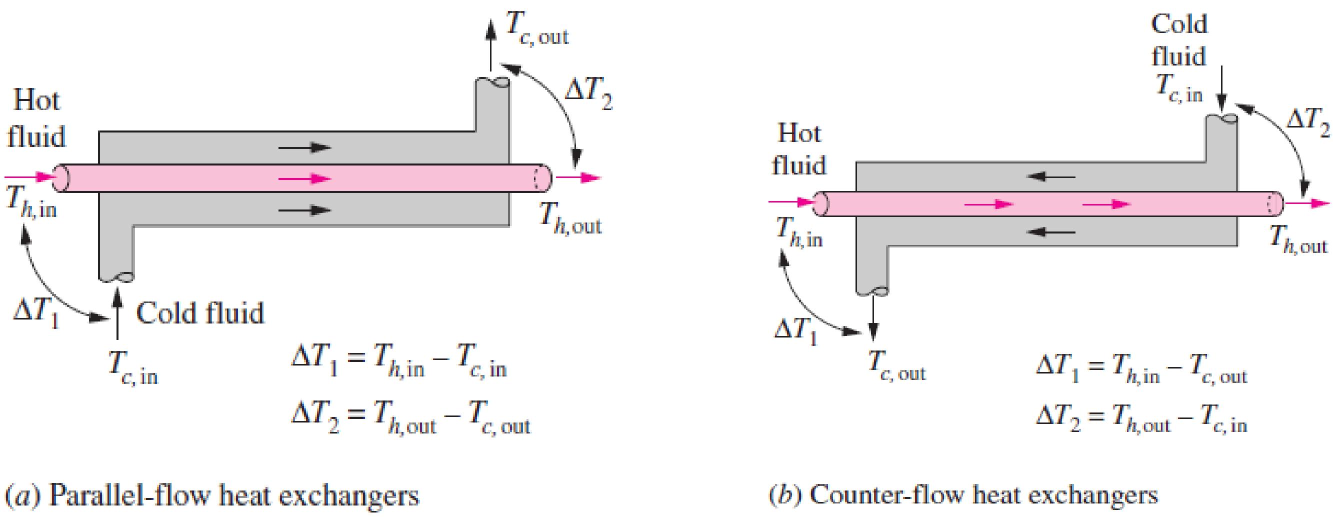


Figure (8.8) The ΔT_1 and ΔT_2 expressions in parallel-flow and counter-flow heat exchangers.

The procedure to be followed by the selection process is:

- Select the type of heat exchanger suitable for the application.
- Determine any unknown inlet or outlet temperature and the heat transfer rate using an energy balance.

- Calculate the log mean temperature difference (ΔT_{lm}) and the correction factor F, if necessary.
- Obtain (select or calculate) the value of the overall heat transfer coefficient (U).
- 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).

8.5 The Effectiveness

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

$$\varepsilon = \frac{q}{q_{max}} = \frac{\text{Actual heat transfer rate}}{\text{Maximum possible heat transfer rate}} \qquad (8.12)$$

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

$$q = C_c(T_{c,out} - T_{c,in}) = C_h(T_{h,in} - T_{h,out}) \qquad (8.13)$$

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 the hot fluids, respectively.

The maximum possible heat transfer rate in a heat exchanger is

$$q_{max} = C_{min}(T_{h,\,in} - T_{c,\,in}) \qquad (8.14)$$

where C_{min} is the smaller of C_c and C_h.

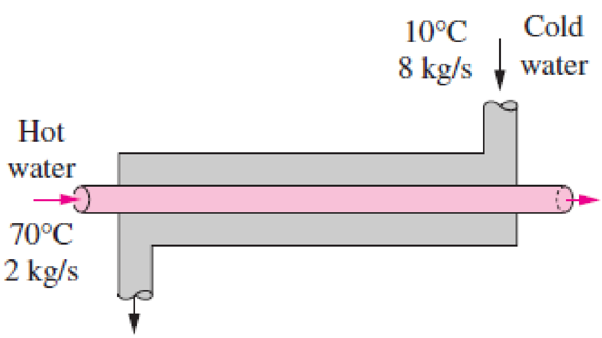
Example (8.2): Cold water enters a counter-flow heat exchanger at (10 °C) at a rate of (8 kg/s), where it is heated by a hot water stream that enters the heat exchanger at (70 °C) at a rate of (2 kg/s). Assuming the specific heat of water to remain constant at (C_p = 4.18 kJ/kg· °C), determine the maximum heat transfer rate and the outlet temperatures of the cold and the hot water streams for this limiting case.

Solution:

$$C_h = \dot{m}_h C_{ph} = 2 * 4.180 = 8.360\ KW/°C$$

$$C_c = \dot{m}_c C_{pc} = 8 * 4.180 = 33.4\ KW/°C$$

$$C_{min} = C_h = 8.360\ KW/°C$$



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

$$q_{max} = 3.86 * (70 - 10)$$

$$q_{max} = 502\ KW$$

$$q = C_c (T_{c,out} - T_{c,in})$$

$$502 = 33.4(T_{c,out} - 10)$$

$$T_{c,out} = 25\ °C$$

$$q = C_h (T_{h,in} - T_{h,out})$$

$$502 = 8.36 * (70 - T_{h,out})$$

$$T_{h,out} = 10\ °C$$

8.6 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

$$\Delta T_{lm} = F\,\Delta T_{lm,\,CF} \qquad (8.15)$$

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 correction factor is less than unity for a cross-flow and multipass shell and tube heat exchanger ($F < 1$).

$\Delta T_{lm,CF}$: 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

$$\Delta T_{lm,\,CF} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} \qquad (8.16)$$

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

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

the heat transfer rate for cross-flow and multipass shell-and-tube heat exchangers that shown in Figure (8.9) is determined from

$$q = UA_s F\,\Delta T_{lm,\,CF} \qquad (8.17)$$

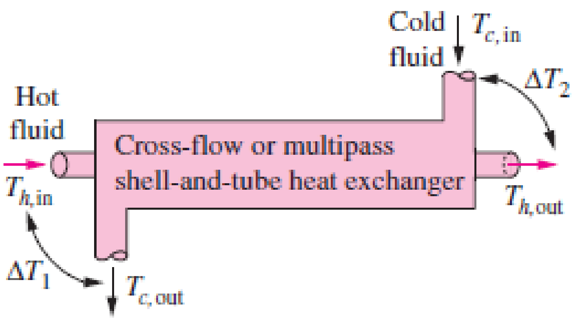


Figure (8.9) The determination of the heat transfer rate for cross-flow and multipass shell-and-tube heat exchangers using the correction factor.

The correction factor (F) for common cross-flow and shell-and-tube heat exchanger configurations is given in Figure (8.10) versus two temperature ratios (P) and (R) defined as

$$P = \frac{t_2 - t_1}{T_1 - t_1} \quad (8.18)$$

$$R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{(\dot{m}C_p)_{\text{tube side}}}{(\dot{m}C_p)_{\text{shell side}}} \quad (8.19)$$

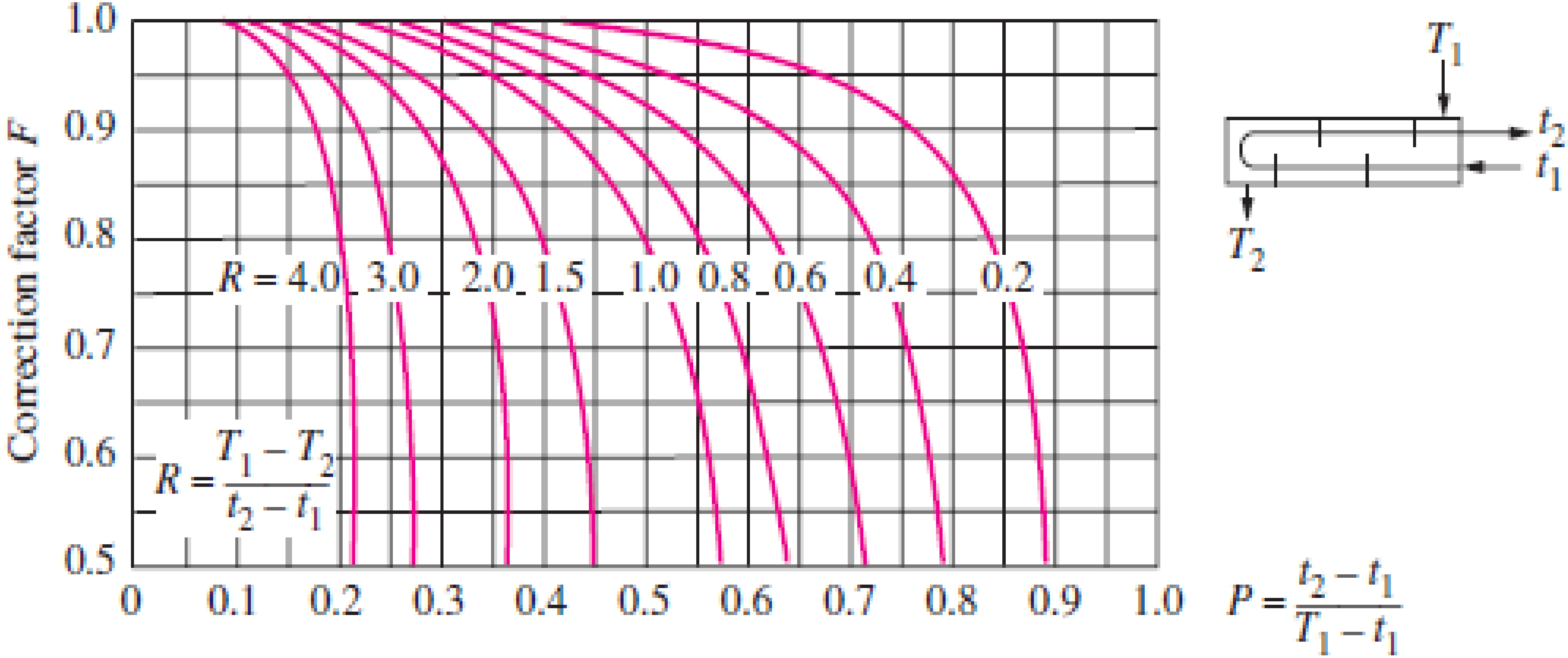
where the subscripts (1) and (2) represent the inlet and outlet, respectively.

Note that

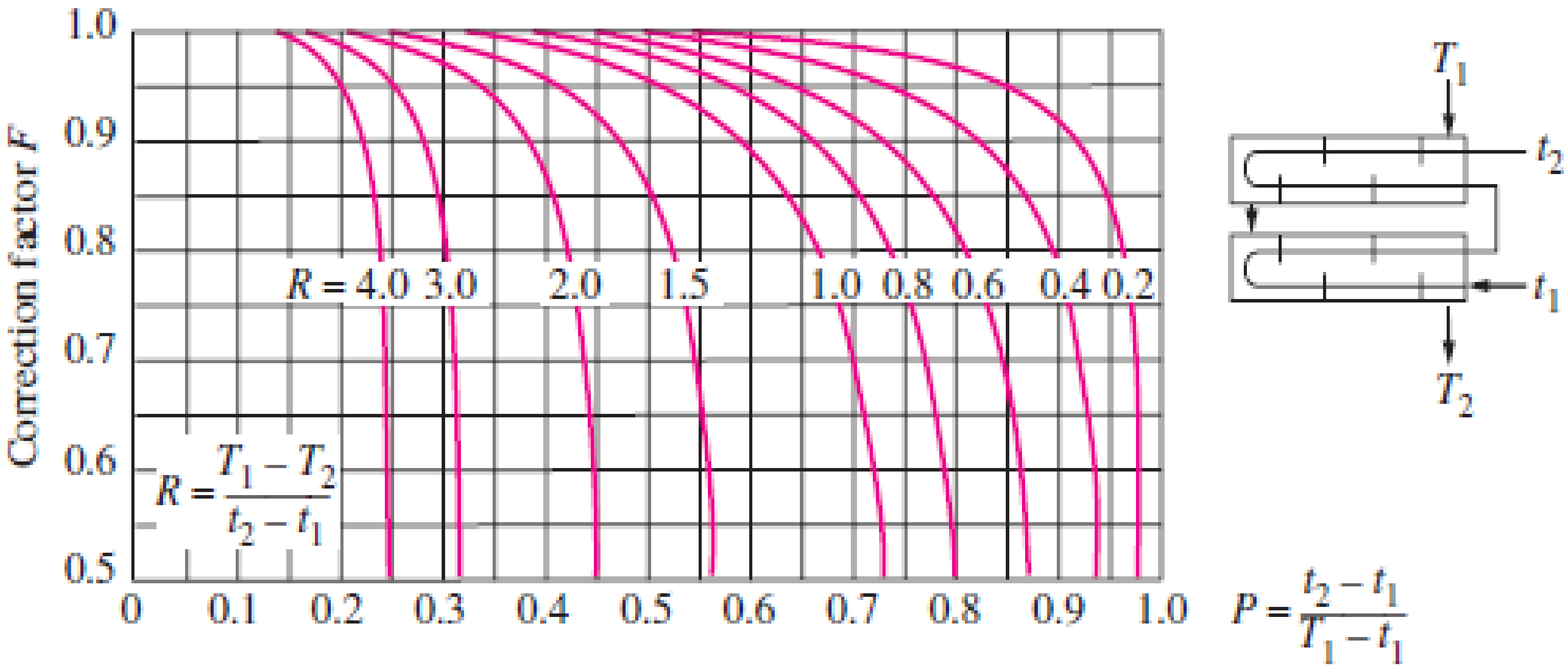
1- For a shell-and-tube heat exchanger, (T) and (t) represent the shell- and tube-side temperatures, respectively, as shown in the correction factor charts.

2- The value of P ranges from 0 to 1 ($0 < P < 1$) and the value of R ranges from 0 to infinity ($0 < R < \infty$).

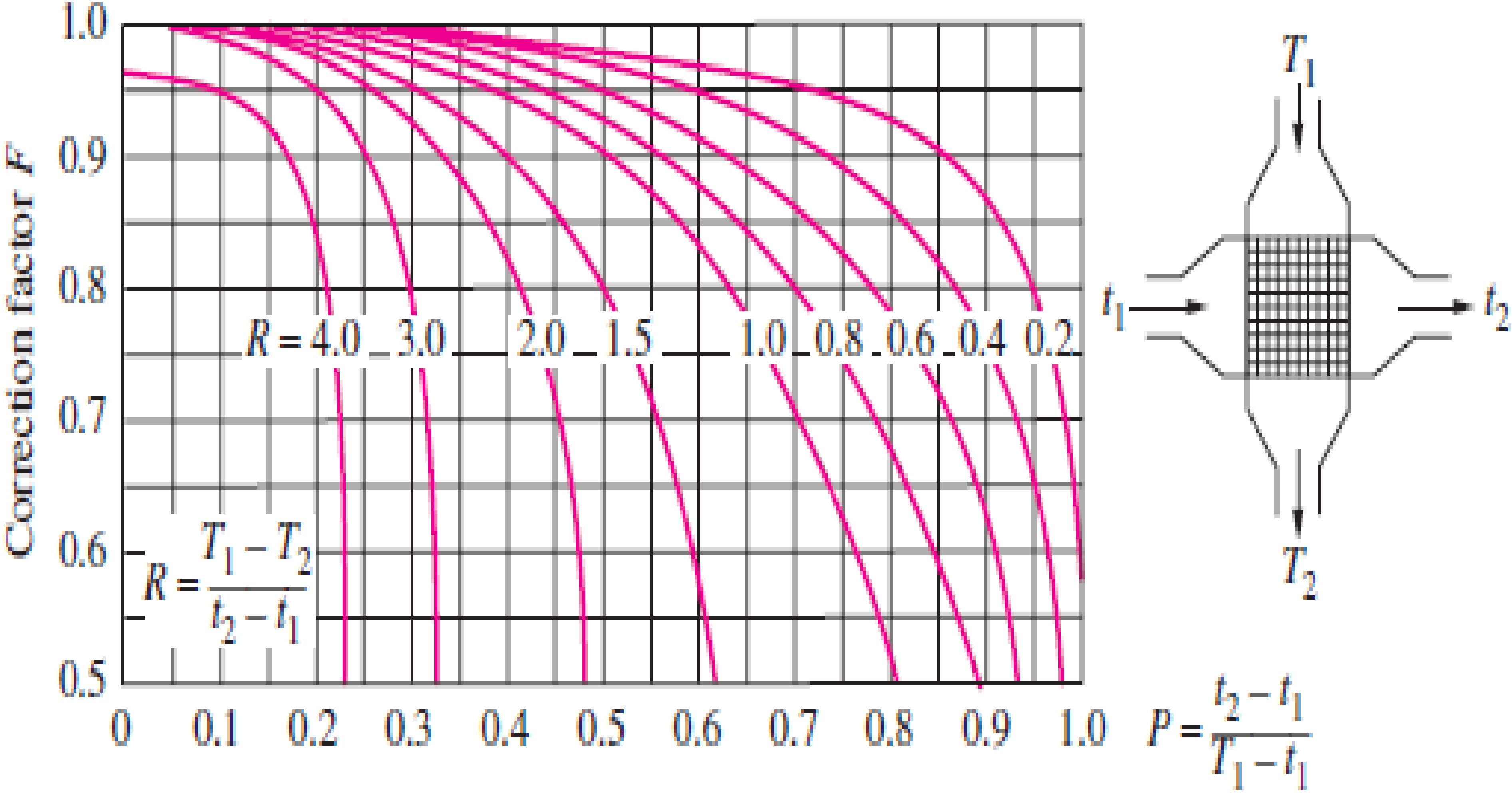
($R = 0$) corresponding to the phase-change (condensation or boiling) on the shell-side and ($R \rightarrow \infty$) to phase-change on the tube side. The correction factor is ($F = 1$) for both of these limiting cases. Therefore, the correction factor for a condenser or boiler is ($F = 1$), regardless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



(*a*) One-shell pass and 2, 4, 6, etc. (any multiple of 2), tube passes



(*b*) Two-shell passes and 4, 8, 12, etc. (any multiple of 4), tube passes



(c) Single-pass cross-flow with both fluids *unmixed*

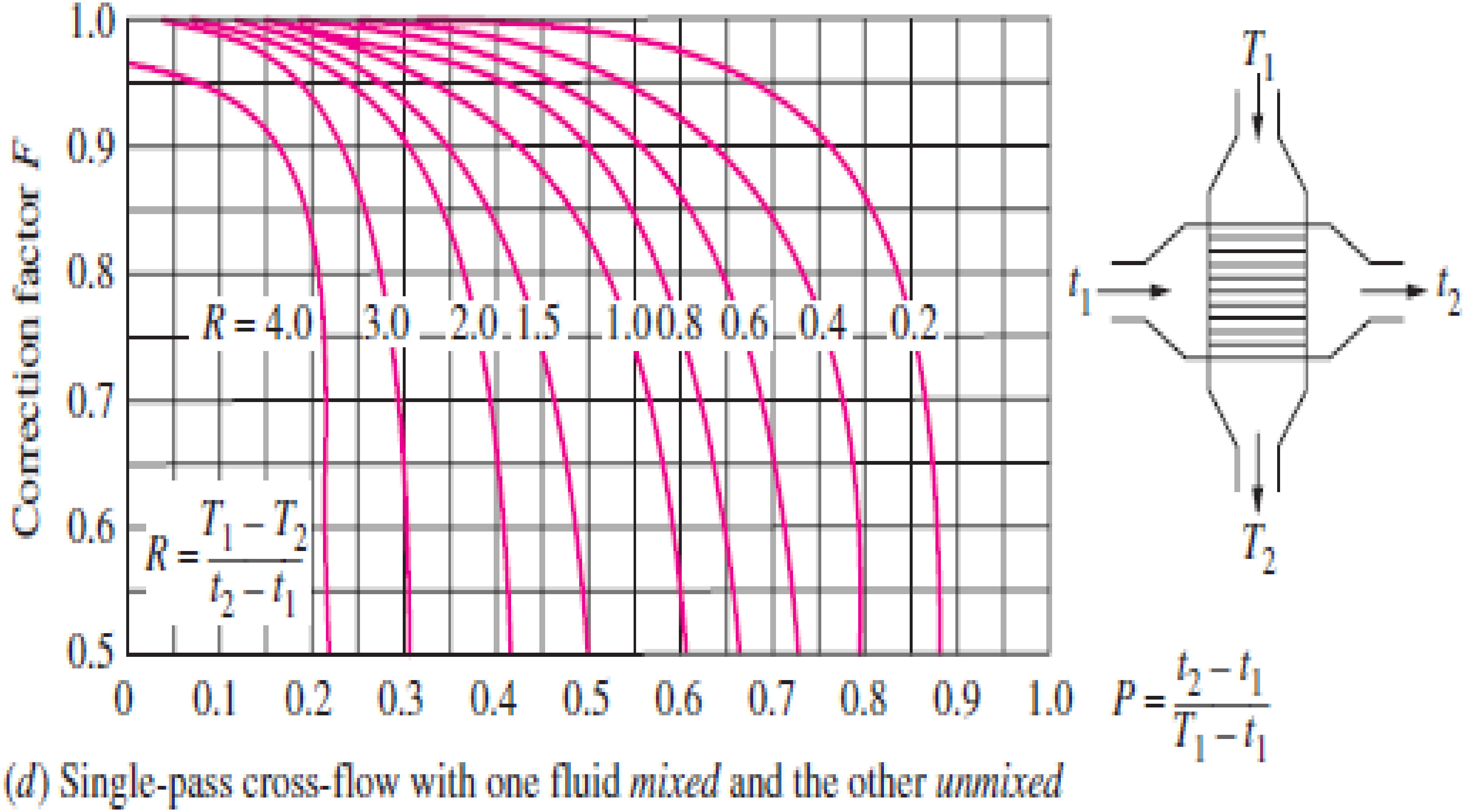
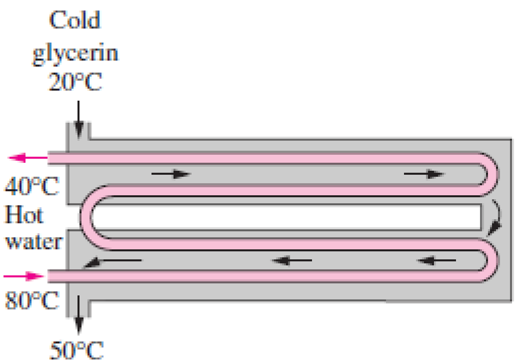


Figure (8.10) Correction factor *F* charts for common shell-and-tube and cross-flow heat exchangers

Example (8.3): A 2-shell passes and 4-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) as shown in Figure below. The total length of the tubes in the heat exchanger is (60 m). The convection heat transfer coefficient is (25 W/m^2.°C) on the glycerin (shell) side and (160 W/m^2.°C) on the water (tube) side. Determine the rate of heat transfer in the heat exchanger (*a*) before any fouling occurs and (*b*) after fouling with a fouling factor of (0.0006 m^2.°C/ W) occurs on the outer surfaces of the tubes.



Solution:

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

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

$$\Delta T_1 = T_{h,\text{in}} - T_{c,\text{out}} = (80 - 50)°\text{C} = 30°\text{C}$$
$$\Delta T_2 = T_{h,\text{out}} - T_{c,\text{in}} = (40 - 20)°\text{C} = 20°\text{C}$$

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

$$\left.\begin{aligned} P &= \frac{t_2 - t_1}{T_1 - t_1} = \frac{40-80}{20-80} = 0.67 \\ R &= \frac{T_1 - T_2}{t_2 - t_1} = \frac{20-50}{40-80} = 0.75 \end{aligned}\right\} F = 0.91 \qquad \text{(Fig.(8.10) b)}$$

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

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o}} = \frac{1}{\frac{1}{160} + \frac{1}{25}}$$

$$U = 21.6 \text{ W/m}^2 \cdot {}^\circ\text{C}$$

$$q = 21.6 * 3.77 * 0.91 * 24.7 = 1830\ W$$

(*b*) When there is fouling on one of the surfaces, the overall heat transfer coefficient *U* is

$$U = \frac{1}{\frac{1}{h_i} + \frac{1}{h_o} + R_f} = \frac{1}{\frac{1}{160} + \frac{1}{25} + 0.0006}$$

$$U = 21.3 \text{ W/m}^2 \cdot {}^\circ\text{C}$$

$$q = 21.3 * 3.77 * 0.91 * 24.7 = 1805\ W$$

8.7 The Effectiveness–NTU Method

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 (*A*) 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.

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, defined as

$$\varepsilon = \frac{q}{q_{\max}} = \frac{Actual\ \ heat\ \ transfer\ \ rate}{Maximum\ \ possible\ heat\ \ transfer\ \ rate}$$

The determination of (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 (q) can be determined from

$$q = \varepsilon q_{\max} = \varepsilon C_{\min}(T_{h,in} - T_{c,in}) \qquad (8.20)$$

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. The effectiveness (ε) relation for the double-pipe parallel-flow heat exchanger can be expressed as

$$\varepsilon_{\text{parallel flow}} = \frac{1 - \exp\left[-\frac{UA_s}{C_{\min}}\left(1 + \frac{C_{\min}}{C_{\max}}\right)\right]}{1 + \frac{C_{\min}}{C_{\max}}} \qquad (8.21)$$

where

C_{min}: is the smaller heat capacity ratio.

C_{max}: is the larger one, and it makes no difference whether Cmin 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

$$\text{NTU} = \frac{UA_s}{C_{\min}} = \frac{UA_s}{(\dot{m}C_p)_{\min}} \qquad (8.22)$$

Note that (NTU) is proportional to (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

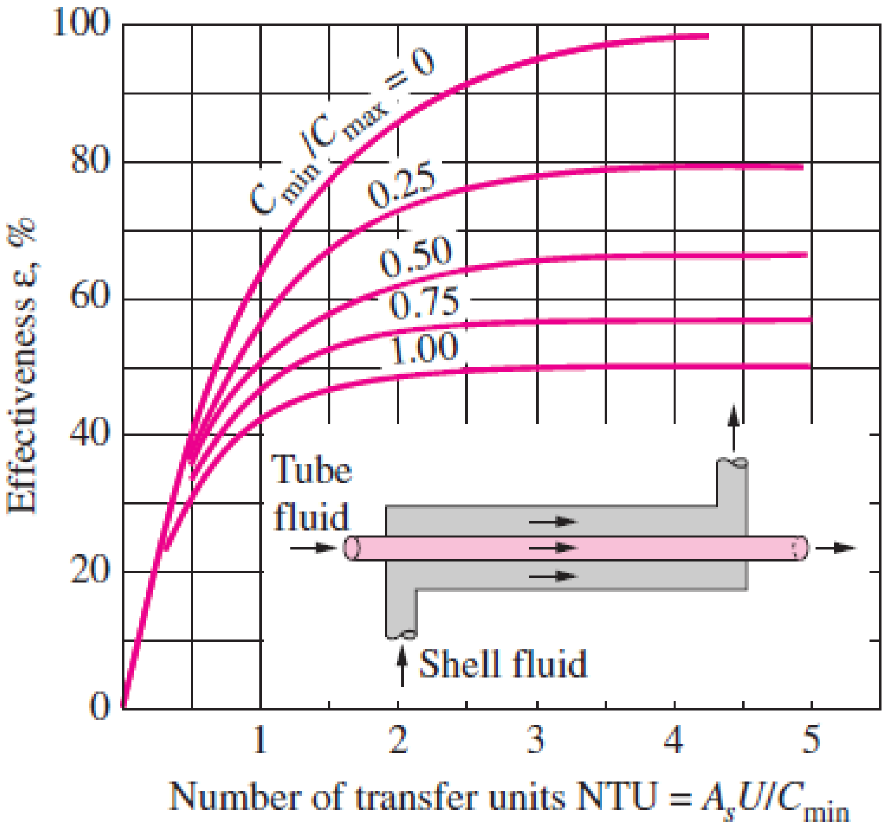
$$c = \frac{C_{min}}{C_{max}} \qquad (8.23)$$

Effectiveness relations have been developed for a large number of heat exchangers, and the results are given in Table (8.1). The effectivenesses of some common types of heat exchangers are also plotted in Figure (8.11).

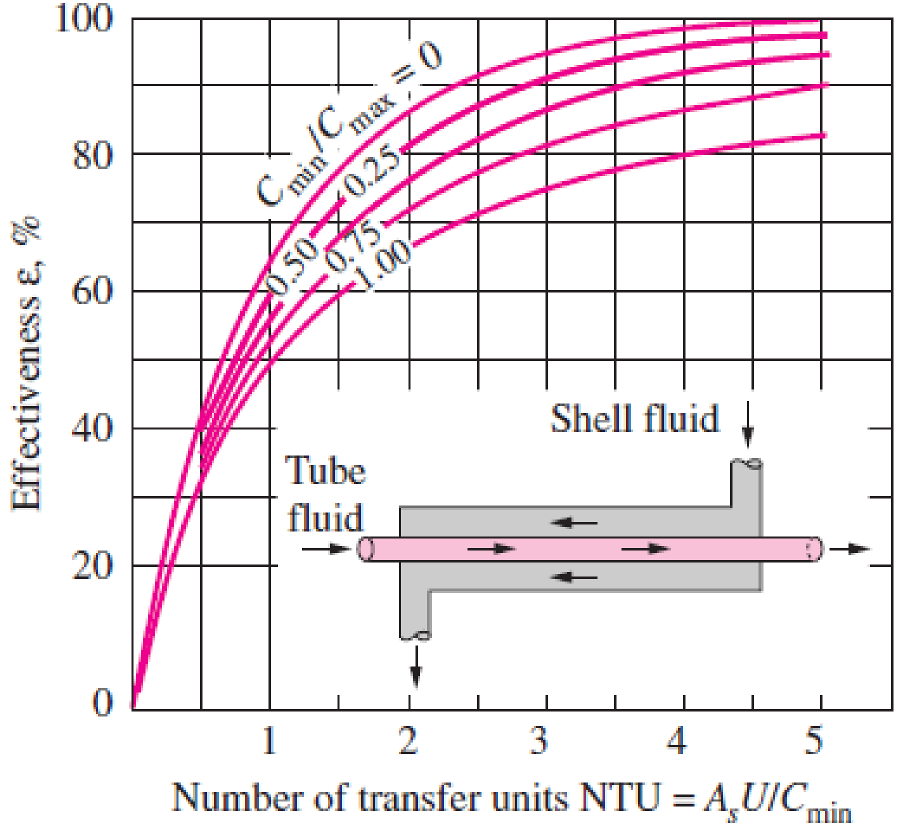
Table (8.1) Effectiveness relations for heat exchangers: where

$$NTU = UA_s / C_{min} \ \ and \ \ c = \frac{C_{min}}{C_{max}} = (\dot{m}\,Cp)_{min}/(\dot{m}Cp)_{max}$$

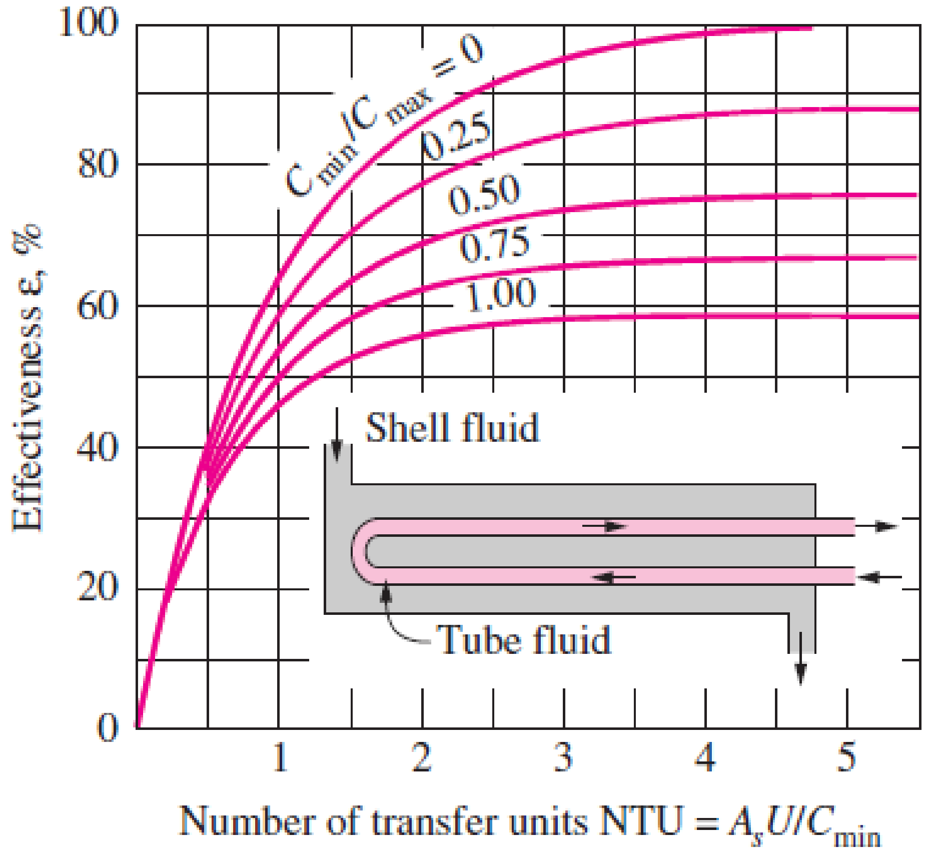
Heat exchanger type	Effectiveness relation
1 *Double pipe:*	
Parallel-flow	$\varepsilon = \dfrac{1 - \exp[-NTU(1+c)]}{1+c}$
Counter-flow	$\varepsilon = \dfrac{1 - \exp[-NTU(1-c)]}{1 - c\exp[-NTU(1-c)]}$
2 *Shell and tube:*	
One-shell pass 2, 4, . . . tube passes	$\varepsilon = 2\left\{1 + c + \sqrt{1+c^2}\,\dfrac{1+\exp[-NTU\sqrt{1+c^2}]}{1-\exp[-NTU\sqrt{1+c^2}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



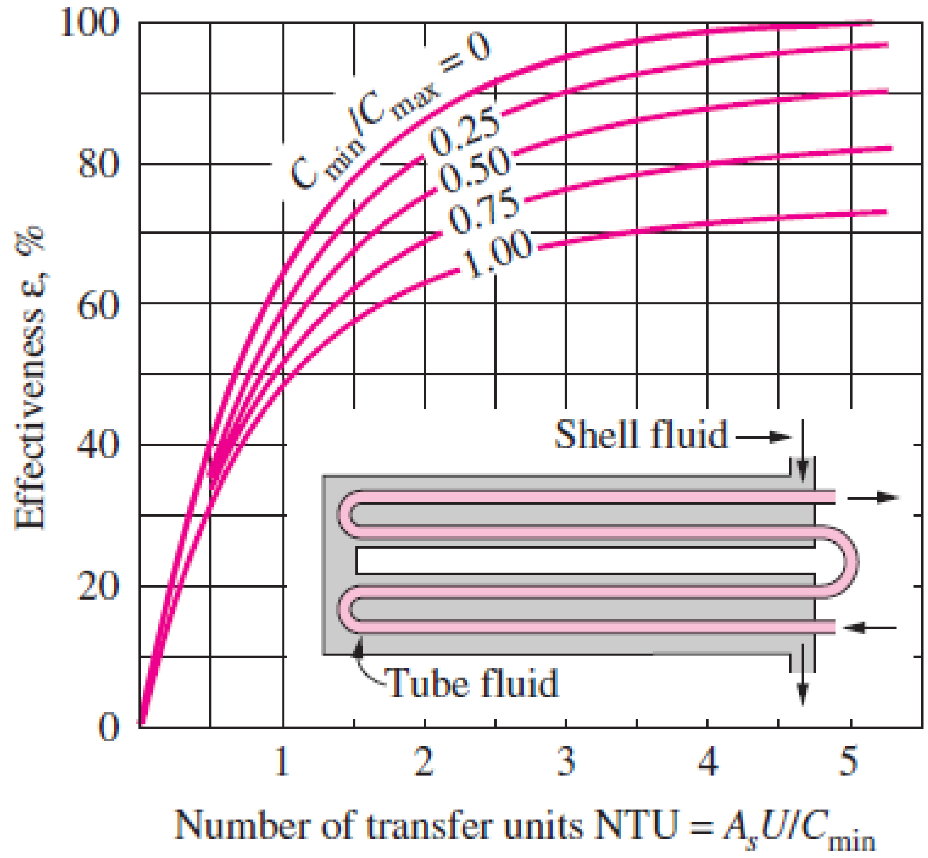
(*a*) Parallel-flow



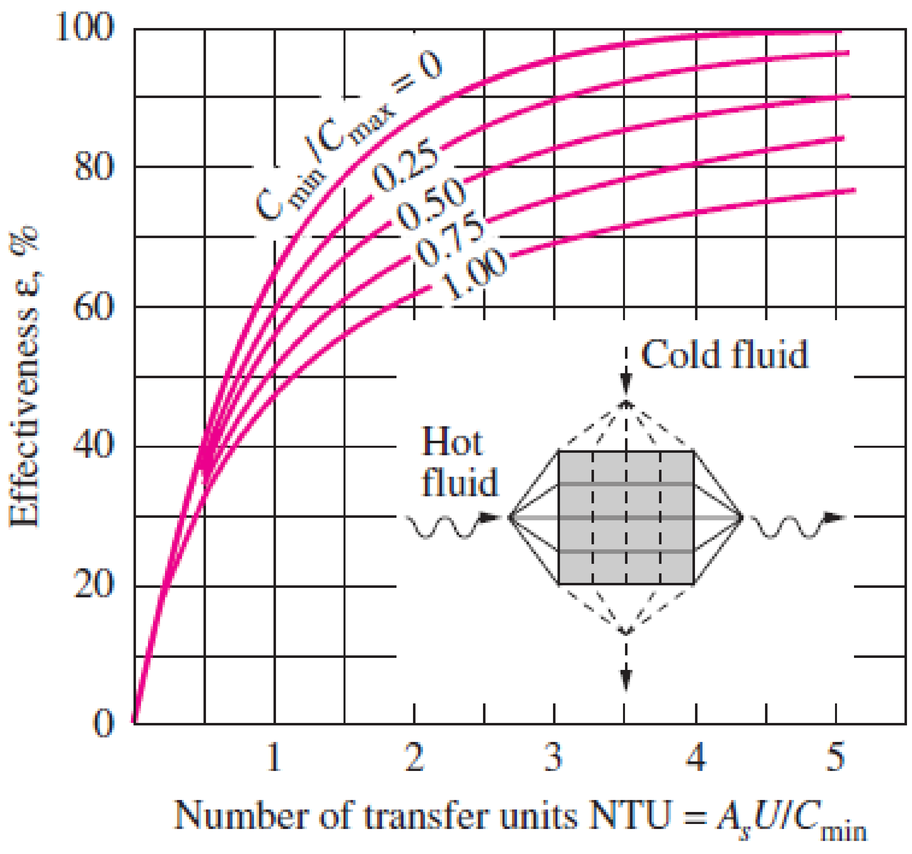
(*b*) Counter-flow



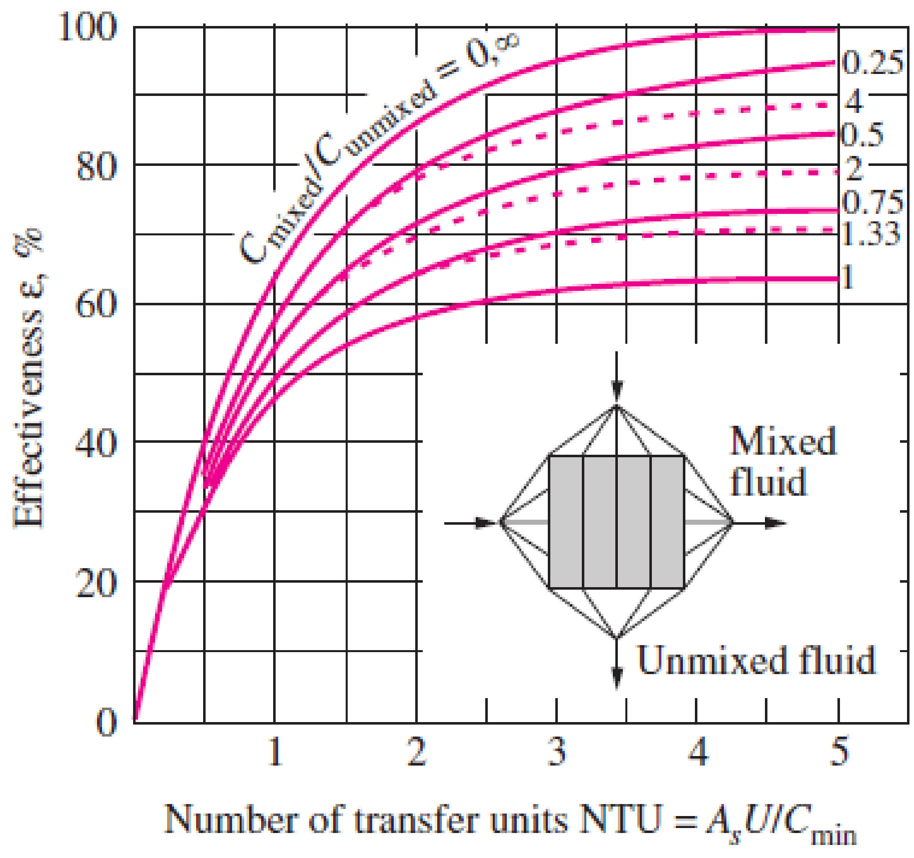
(*c*) One-shell pass and 2, 4, 6, … tube passes



(*d*) Two-shell passes and 4, 8, 12, … tube passes



(*e*) Cross-flow with both fluids unmixed



(*f*) Cross-flow with one fluid mixed and the other unmixed

Figure (8.11) Effectiveness for heat exchangers when the effectiveness is known the relations in Table (8.2) give the NTU directly.

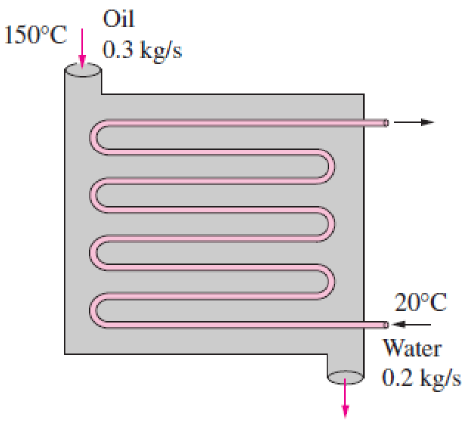
Table (8.2) NTU relations for heat exchangers: where

$$NTU = UA_s / C_{min} \text{ and } c = \frac{C_{min}}{C_{max}} = (\dot{m}\, Cp)_{min} / (\dot{m} Cp)_{max}$$

Heat exchanger type	NTU relation
1 *Double-pipe:* Parallel-flow	$NTU = -\frac{\ln[1 - \varepsilon(1 + c)]}{1 + c}$
Counter-flow	$NTU = \frac{1}{c - 1} \ln\left(\frac{\varepsilon - 1}{\varepsilon c - 1}\right)$
2 *Shell and tube:* One-shell pass 2, 4, . . . tube passes	$NTU = -\frac{1}{\sqrt{1 + c^2}} \ln\left(\frac{2/\varepsilon - 1 - c - \sqrt{1 + c^2}}{2/\varepsilon - 1 - c + \sqrt{1 + c^2}}\right)$
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* $c = 0$	$NTU = -\ln(1 - \varepsilon)$

Example (8.4): Hot oil is to be cooled by water in a 1-shell-pass and 8-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 W/m^2.°C). 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:



$$C_h = \dot{m}_h C_{ph} = 0.3 * 2.13 = 0.639\ kW/°C$$

$$C_c = \dot{m}_c C_{pc} = 0.2 * 4.18 = 0.836\ kW/°C$$

$$C_{min} = C_h = 0.639\ kW/°C$$

$$c = \frac{C_{min}}{C_{max}} = \frac{0.639}{0.836} = 0.764$$

$$q_{max} = C_{min}(T_{h,in} - T_{c,in})$$

$$q_{max} = 0.639 * (150 - 20) = 83.1\ kW$$

$$A_s = n(\pi D L) = 8\pi * 0.014 * 5 = 1.76\ m^2$$

$$NTU = \frac{UA_s}{C_{min}} = \frac{310 * 1.76}{639} = 0.853$$

$$\varepsilon = 2\left\{1 + c + \sqrt{1 + c^2}\,\frac{1 + \exp[-NTU\sqrt{1 + c^2}}{1 - \exp[-NTU\sqrt{1 + c^2}}\right\}^{-1}$$

$$\varepsilon = 2\left\{1 + 0.764 + \sqrt{1 + 0.764^2}\,\frac{1 + \exp[-0.853 * \sqrt{1 + 0.764^2}}{1 - \exp[-0.853 * \sqrt{1 + 0.764^2}}\right\}^{-1}$$

$$\varepsilon = 0.46$$

$$q = \varepsilon q_{max} = 0.46 * 83.1 = 39.1\ KW$$

$$q = C_c(T_{c,out} - T_{c,in})$$

$39.1 = 0.836(T_{c,out} - 20)$

$T_{c,out} = 66.8\ ℃$

$q = C_h(T_{h,in} - T_{h,out})$

$39.1 = 0.639(150 - T_{h,out})$

$T_{h,out} = 88.8\ ℃$

Home Work (9)

1- The properties and flow rates for the hot and cold fluids of a heat exchanger are shown in the following table. Which fluid limits the heat transfer rate of the exchanger?

	Hot fluid	Cold fluid
Density, kg/m^3	997	1247
Specific heat, $J/kg \cdot K$	4179	2564
Thermal conductivity, $W/m \cdot K$	0.613	0.287
Viscosity, $N \cdot s/m^2$	8.55×10^{-4}	1.68×10^{-4}
Flow rate, m^3/h	14	16

2- Consider a concentric tube heat exchanger with an area of (50 m^2) operating under the following conditions:

	Hot fluid	Cold fluid
Heat capacity rate, kW/K	6	3
Inlet temperature, °C	60	30
Outlet temperature, °C	—	54

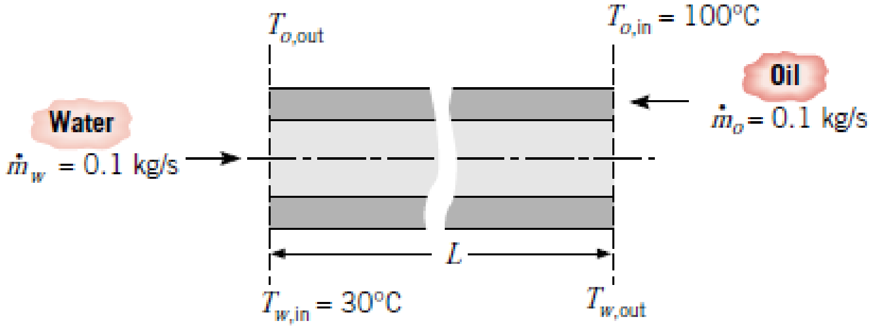
(a) Determine the outlet temperature of the hot fluid.

(b) Is the heat exchanger operating in counterflow or parallel flow, or can't you tell from the available information?

(c) Calculate the overall heat transfer coefficient.

(d) Calculate the effectiveness of this exchanger.

3- A concentric tube heat exchanger for cooling lubricating oil is comprised of a thin-walled inner tube of (25 mm) diameter carrying water and an outer tube of (45 mm) diameter carrying the oil. The exchanger operates in counter-flow with an overall heat transfer coefficient of (60 W/m^2.K) and the tabulated average properties.



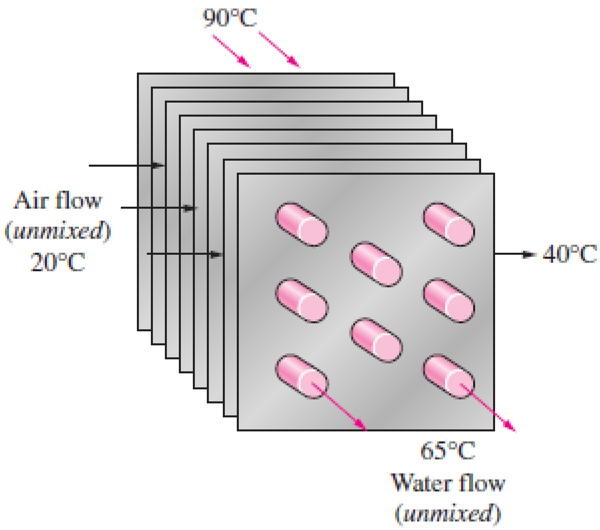
Properties	Water	Oil
ρ (kg/m^3)	1000	800
c_p (J/kg · K)	4200	1900
ν (m^2/s)	7×10^{-7}	1×10^{-5}
k (W/m · K)	0.64	0.134
Pr	4.7	140

(a) If the outlet temperature of the oil is (60 °C), determine the total heat transfer and the outlet temperature of the water.

(b) Determine the length required for the heat exchanger.

Home Work (10)

1- 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 as shown in Figure below. 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. Take (C_p = 4.195 kJ/kg · °C).



2- A counterflow, concentric tube heat exchanger used for engine cooling has been in service for an extended period of time. The heat transfer surface area of the exchanger is ($5\ m^2$), and the *design value* of the overall convection coefficient is ($38\ W/m^2.°C$). During a test run, engine oil flowing at (0.1 kg/s) is cooled from (110 °C) to (66 °C) by water supplied at a temperature of (25 °C) and a flow rate of (0.2 kg/s). Determine the fouling factor. Take (C_p of oil is 2166 J/kg.K) and (C_p of water is 4178 J/kg.K).

3- In a dairy operation, milk at a flow rate of (250 liter/hour) and a cow-body temperature of (38.6 °C) must be chilled to a safe-to-store temperature of (13 °C) or less. Ground water at (10 °C) is available at a flow rate of ($0.72\ m^3/h$). The density and specific heat of milk are ($1030\ kg/m^3$) and (3860 J/kg.K) , respectively while density and specific heat of water are ($1000\ kg/m^3$) and (4187 J/kg.K) , respectively. (a) Determine the (UA) product of a counterflow heat exchanger required for the chilling process. Determine the length of the exchanger if the inner pipe has a (50 mm) diameter and the overall heat transfer coefficient is ($U = 1000\ W/m^2.K$) . (b) Determine the outlet temperature of the water.