

Chapter 9

Heat Exchangers

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Introduction

The process of heat exchange between two fluids that are at different temperatures and separated by a solid wall occurs in many engineering applications.

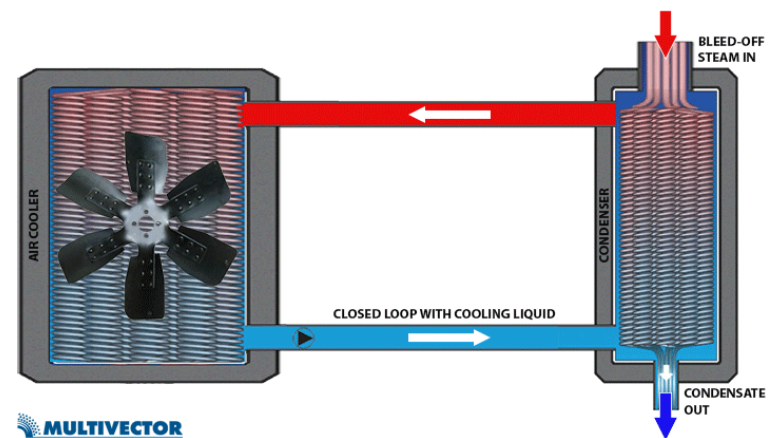
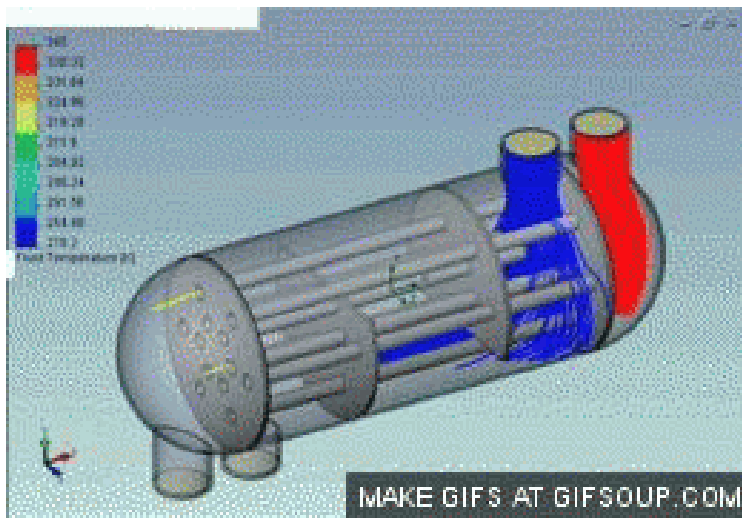
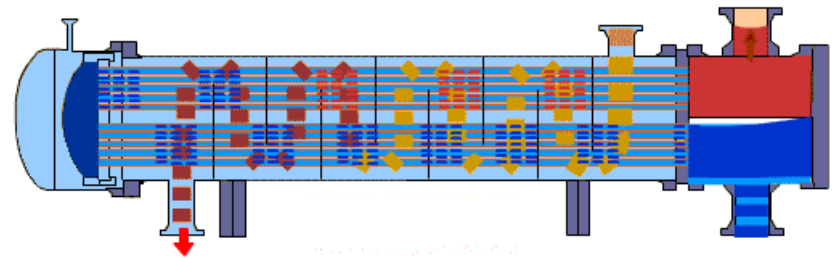
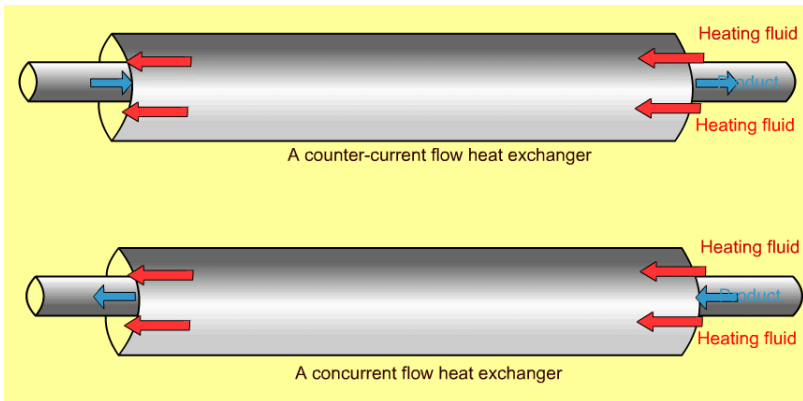
The device used to implement this exchange is termed a **heat exchanger**, and specific applications may be found in **space heating and air conditioning, power production, waste heat recovery, and chemical processing.**

Objectives

After completion of this lecture and tutorials on heat exchangers, student will be able to:

- ✓ Understand what a heat exchanger is and its uses
- ✓ Differentiate various types of heat exchangers
- ✓ Use the methods available for analysis of exchangers
 - [The LMTD Method](#)
 - [The \$\epsilon\$ – NTU Method](#)
- ✓ Sizing of new heat exchanger
- ✓ Performance analysis of an existing heat exchanger.

Applications





Heat Exchanger Types

Heat exchangers are typically classified according to:

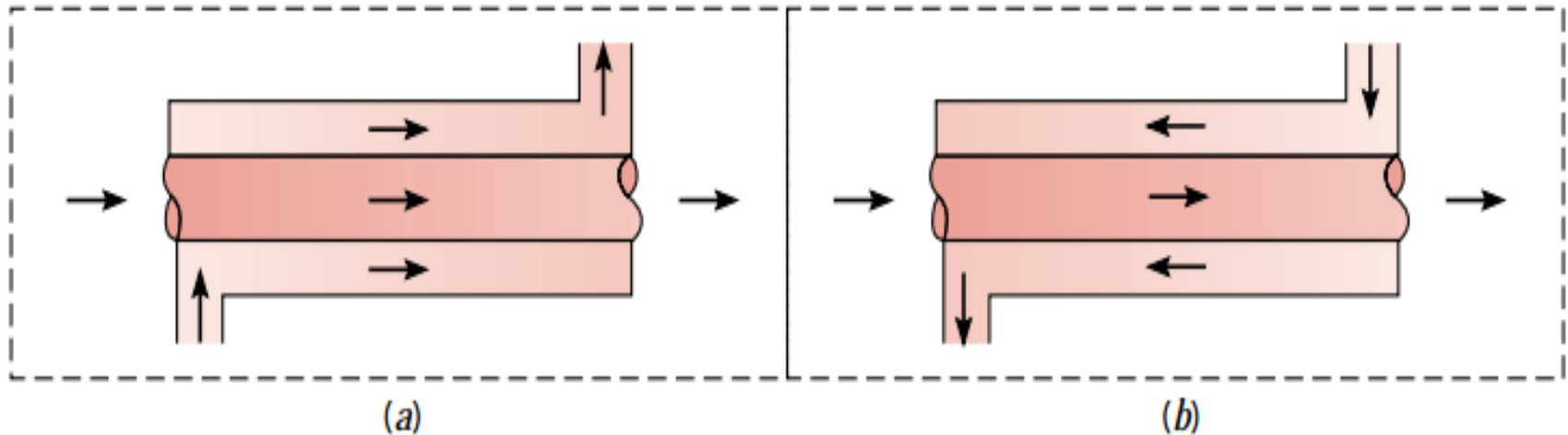
- **Flow arrangement and**
- **Type of construction.**

Basic Heat Exchanger Types:

- 1. Double Tube Heat Exchangers** (Parallel and Counter Types)
- 2. Cross – Flow Heat Exchangers**
- 3. Shell and Tube Heat Exchangers**
- 4. Compact Heat Exchangers**

Double Tube Heat Exchangers

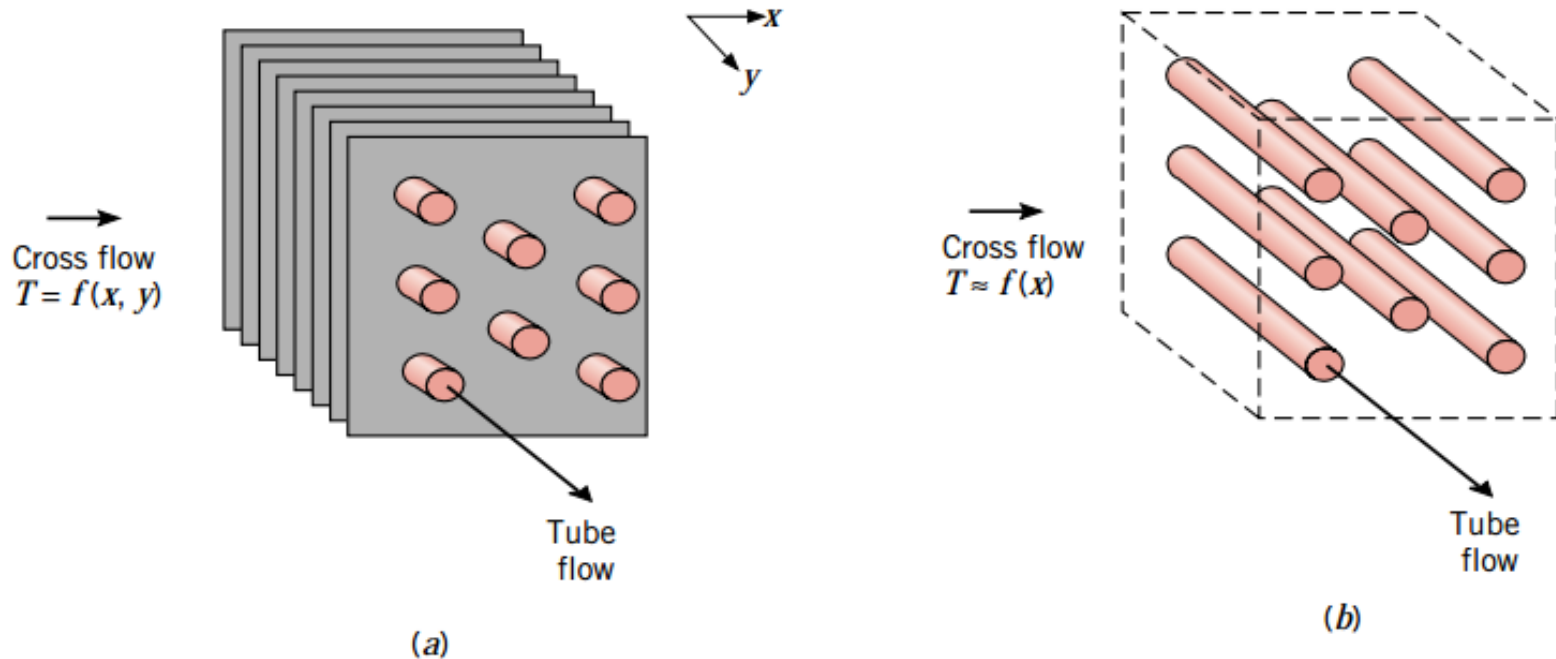
The simplest heat exchanger is one for which the hot and cold fluids move in the same or opposite directions in a concentric tube(or double-pipe) construction.



Concentric (double) tube heat exchangers. (a) Parallel flow. (b) Counter flow.

Cross - Flow Heat Exchangers

Alternatively, the fluids may move in cross flow(perpendicular to each other), as shown by the finned and unfinned tubular heat exchangers.



Cross-flow heat exchangers. (a) Finned with both fluids unmixed. (b) Unfinned with one fluid mixed and the other unmixed.

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The two configurations are typically differentiated by an idealization that treats fluid motion over the tubes as **unmixed** or **mixed**.

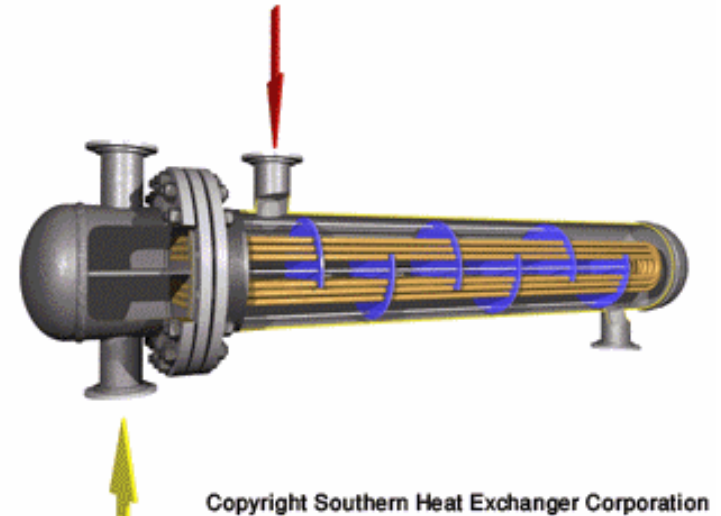
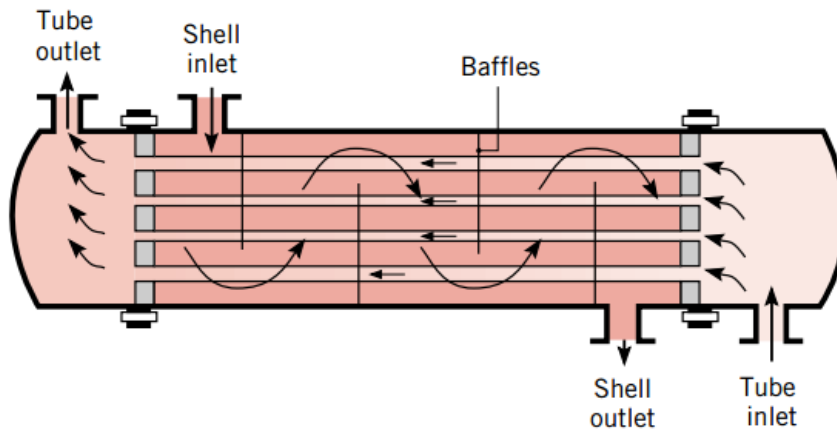
- ❑ The fluid is said to be unmixed because the fins inhibit motion in a direction (y) that is transverse to the main-flow direction (x). In this case the fluid temperature varies with x and y . [Fig (a)]
- ❑ In contrast, for the unfinned tube bundle of fluid motion, hence mixing, in the transverse direction is possible, and temperature variations are primarily in the main flow direction. [Fig (b)]

Since the tube flow is unmixed, both fluids are unmixed in the finned exchanger, while one fluid is mixed and the other unmixed in the unfinned exchanger.

The nature of the mixing condition can significantly influence heat exchanger performance.

Shell and Tube Heat Exchangers

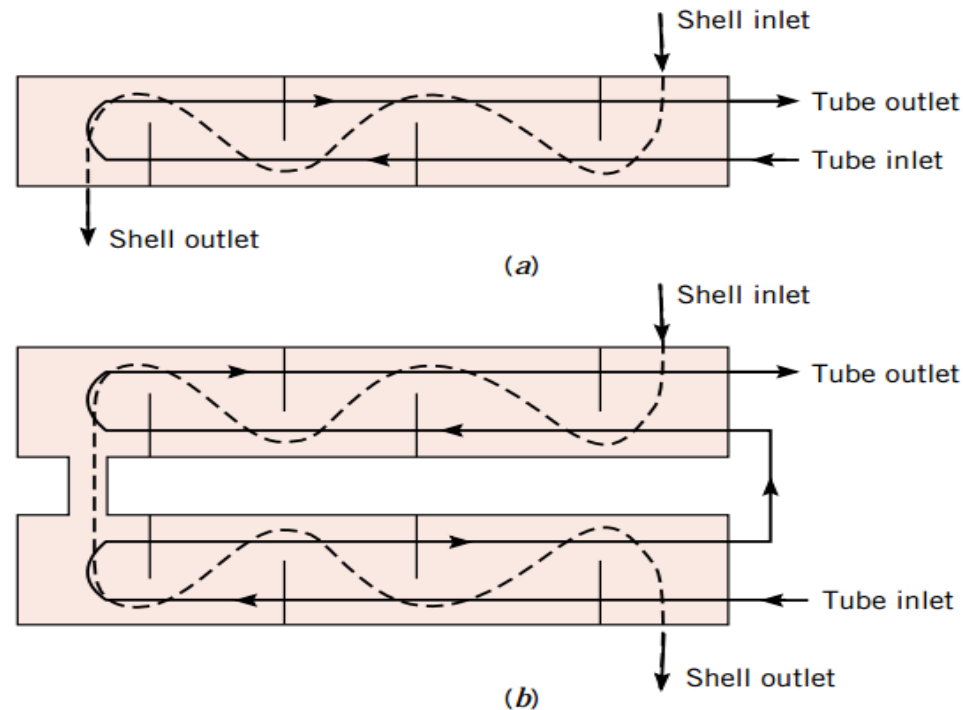
Another common configuration is the shell-and-tube heat exchanger. Specific forms differ according to the number of shell-and-tube passes, and the simplest form, which involves single tube and shell passes, is shown in Figure.



Shell-and-tube heat exchanger with one shell pass and one tube pass (cross-counter flow mode of operation).

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Baffles are usually installed to **increase the convection coefficient** of the shell-side fluid by **inducing turbulence** and a cross-flow velocity component. In addition, the baffles physically **support the tubes**, reducing **flow-induced tube vibration**.



*Shell-and-tube heat exchangers. (a) One shell pass and two tube passes.
(b) Two shell passes and four tube passes.*

Compact Heat Exchangers

A special and important class of heat exchangers is used to achieve a very large ($\geq 400 \text{ m}^2/\text{m}^3$ for liquids and $\geq 700 \text{ m}^2/\text{m}^3$ for gases) heat transfer surface area per unit volume.

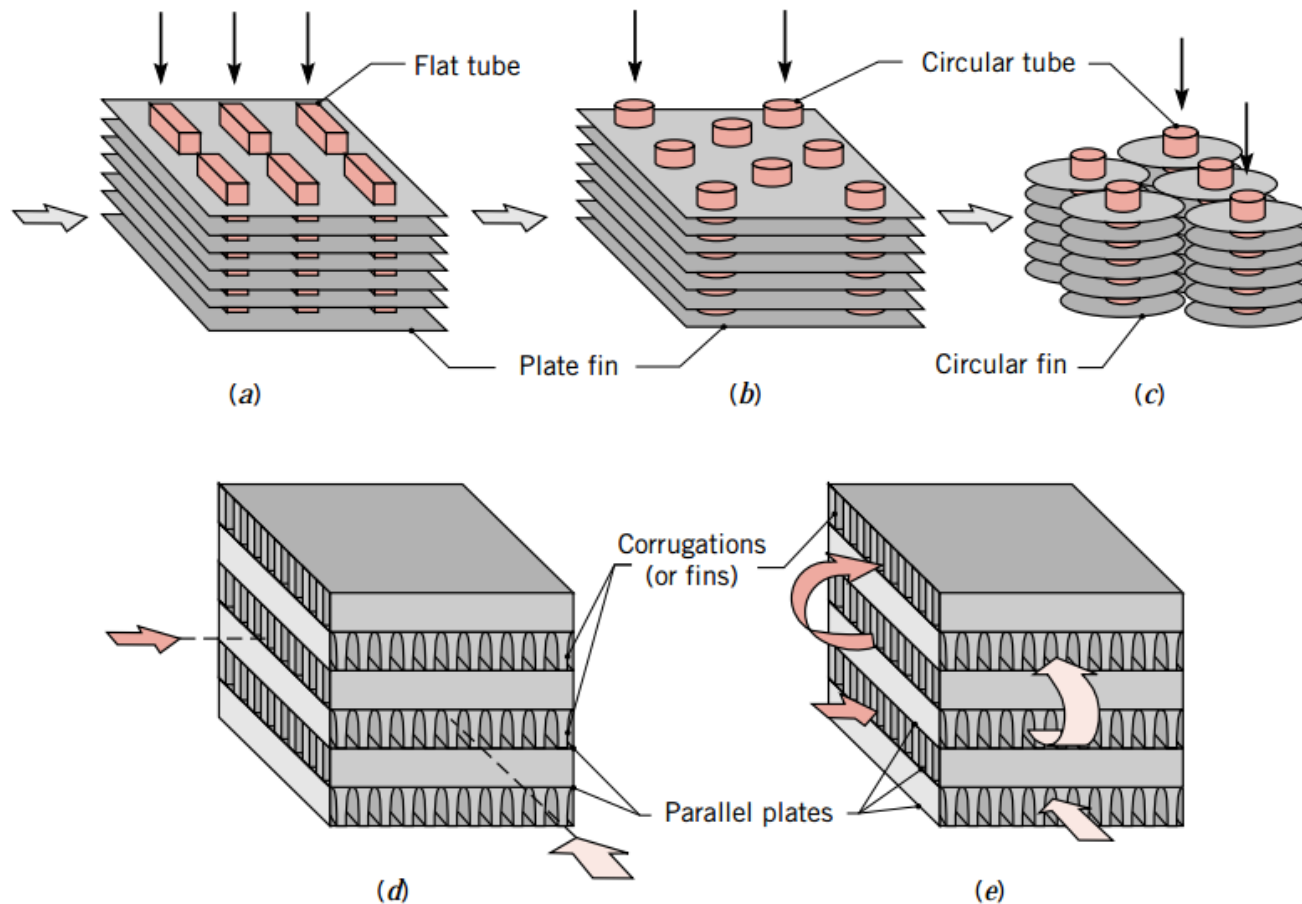
Termed **compact heat exchangers**, these devices have dense arrays of finned tubes or plates and are typically used when at least one of the fluids is a gas, and is hence characterized by a small convection coefficient.

The tubes may be **flat** or **circular**, as in Figures, and the fins may be plate or circular.

Parallel-plate heat exchangers may be finned or corrugated and may be used in **single-pass** or **multi-pass modes** of operation.

Flow passages associated with compact heat exchangers are typically small ($D_h = 5 \text{ mm}$), and the flow is usually laminar.

Cont'd ...



Compact heat exchanger cores:

(a) *Fin-tube (flat tubes, continuous plate fins).* (b) *Fin-tube (circular tubes, continuous plate fins).* (c) *Fin-tube (circular tubes, circular fins).* (d) *Plate-fin (single pass).* (e) *Plate-fin (multi-pass).*

Overall Heat Transfer Coefficient

An essential, and often the most uncertain, part of any heat exchanger analysis is **determination of the overall heat transfer coefficient**. that this coefficient is defined in terms of the total thermal resistance to heat transfer between two fluids.

The coefficient was determined by accounting for conduction and convection resistances between fluids separated by composite plane and cylindrical walls, respectively.

During normal heat exchanger operation, surfaces are often subject to fouling by fluid impurities, rust formation, or other reactions between the fluid and the wall material. The subsequent deposition of a film or scale on the surface can greatly increase the resistance to heat transfer between the fluids.

This effect can be treated by introducing an additional thermal resistance, termed the **fouling factor, R_f** . Its value depends on the operating temperature, fluid velocity, and length of service of the heat exchanger.

Cont'd ...

Accordingly, with inclusion of surface fouling and fin (extended surface) effects, the overall heat transfer coefficient may be expressed as:

$$\frac{1}{UA} = \frac{1}{U_c A_c} = \frac{1}{U_h A_h}$$
$$= \frac{1}{(\eta_o h A)_c} + \frac{R'_{f,c}}{(\eta_o A)_c} + R_w + \frac{R'_{f,h}}{(\eta_o A)_h} + \frac{1}{(\eta_o h A)_h}$$

where c and h refer to the cold and hot fluids, respectively.

Representative Fouling Factors

Fluid	R'_f (m ² · K/W)
Seawater and treated boiler feedwater (below 50°C)	0.0001
Seawater and treated boiler feedwater (above 50°C)	0.0002
River water (below 50°C)	0.0002–0.001
Fuel oil	0.0009
Refrigerating liquids	0.0002
Steam (nonoil bearing)	0.0001

Cont'd ...

The quantity η_o is termed the overall surface efficiency or temperature effectiveness of a finned surface. It is defined such that, for the hot or cold surface without fouling, the heat transfer rate is:

$$q = \eta_o h A (T_b - T_\infty)$$

$$\eta_o = 1 - \frac{A_f}{A} (1 - \eta_f)$$

$$\eta_f = \frac{\tanh (mL)}{mL}$$

$m = (2h/kt)^{1/2}$ and t is the fin thickness.

Cont'd ...

The wall conduction term in the overall heat transfer coefficient equation may often be neglected, since a thin wall of large thermal conductivity is generally used. Also, one of the convection coefficients is often much smaller than the other and hence dominates determination of the overall coefficient.

Representative Values of the Overall Heat Transfer Coefficient

Fluid Combination	U (W/m ² · K)
Water to water	850–1700
Water to oil	110–350
Steam condenser (water in tubes)	1000–6000
Ammonia condenser (water in tubes)	800–1400
Alcohol condenser (water in tubes)	250–700
Finned-tube heat exchanger (water in tubes, air in cross flow)	25–50

Precipitation fouling of ash particles on superheater tubes
(from *Steam, Its Generation, and Use*, Babcock and Wilcox Co., 1978).



Cont'd ...

For the unfinned, tubular heat exchangers:

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

The overall heat transfer coefficient may be determined from knowledge of the hot and cold fluid convection coefficients and fouling factors and from appropriate geometric parameters. For unfinned surfaces, the convection coefficients may be estimated from correlations presented for External Flow and Internal Flow. For standard fin configurations, the coefficients may be obtained from results compiled by Kays and London.

Heat Exchanger Analysis

1. LMTD Method

- It is a simple matter to use the log mean temperature difference (LMTD) method of heat exchanger analysis when the fluid inlet temperatures are known and the outlet temperatures are specified or readily determined from the energy balance expressions. The value of T_{lm} for the exchanger may then be determined.

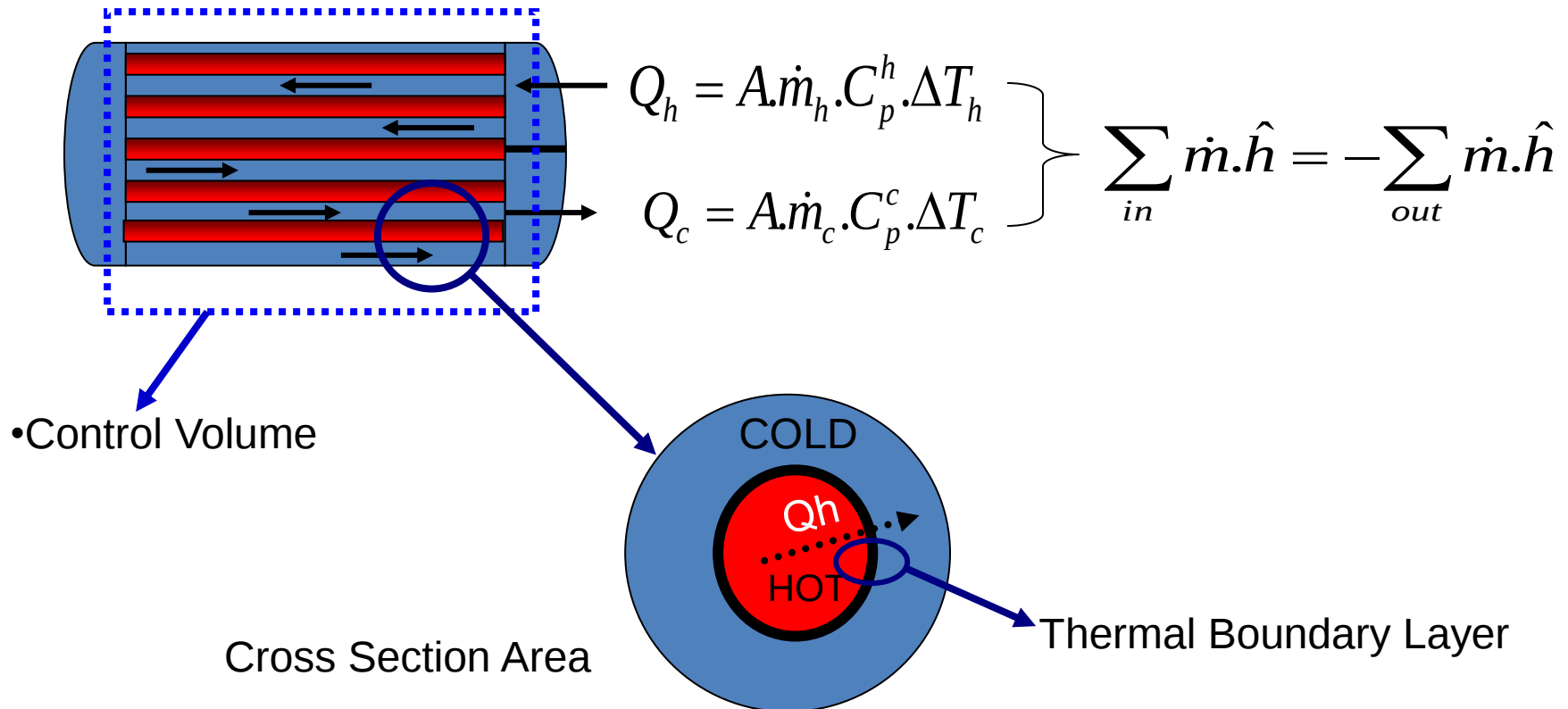
2. ϵ – NTU Method

- If only the inlet temperatures are known, use of the LMTD method requires a cumbersome iterative procedure. It is therefore preferable to use an alternative approach termed the effectiveness–NTU (or NTU) method.

Principle of a Heat Exchanger

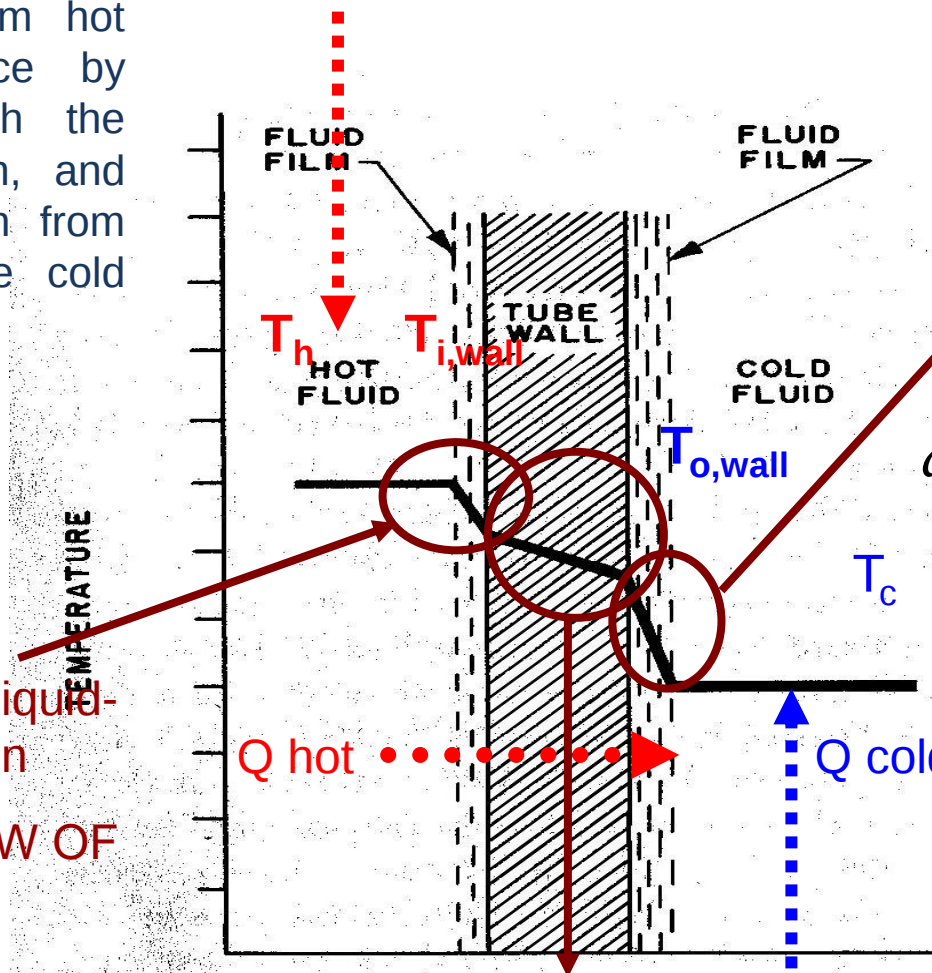
- First Law of Thermodynamics: “Energy is conserved.”

$$\cancel{\frac{dE}{dt}}^0 = \left(\sum_{in} \dot{m} \cdot \hat{h}_{in} - \sum_{out} \dot{m} \cdot \hat{h}_{out} \right) + \cancel{q}^0 + \cancel{\dot{w}_s}^0 + \cancel{\dot{e}_{generated}}^0$$



Temperature Gradient - Hot to Cold across Pipe Wall

Energy moves from hot fluid to a surface by convection, through the wall by conduction, and then by convection from the surface to the cold fluid.



Region I : Hot Liquid-Solid Convection

NEWTON'S LAW OF COOLING

$$dq_x = h_h \cdot (T_h - T_{iw}) \cdot dA$$

Region III: Solid – Cold Liquid Convection

NEWTON'S LAW OF COOLING

$$dq_x = h_c \cdot (T_{ow} - T_c) \cdot dA$$

Region II : Conduction Across Copper Wall
FOURIER'S LAW

$$dq_x = -k \cdot \frac{dT}{dr}$$

The Overall Heat Transfer Coefficient

Region I : Hot Liquid –
Solid Convection

$$q_x = h_{hot} \cdot (T_h - T_{iw}) \cdot A \rightarrow T_h - T_{iw} = \frac{q_x}{h_h \cdot A_i}$$

Region II :
Conduction Across
Copper Wall

$$q_x = \frac{k_{copper} \cdot 2\pi L}{\ln \frac{r_o}{r_i}} \rightarrow T_{o,wall} - T_{i,wall} = \frac{q_x \cdot \ln \left(\frac{r_o}{r_i} \right)}{k_{copper} \cdot 2\pi L}$$

Region III : Solid –
Cold Liquid Convection

$$q_x = h_c (T_{o,wall} - T_c) A_o \rightarrow T_{o,wall} - T_c = \frac{q_x}{h_c \cdot A_o}$$

$$T_h - T_c = \frac{q_x}{R_1 + R_2 + R_3}$$

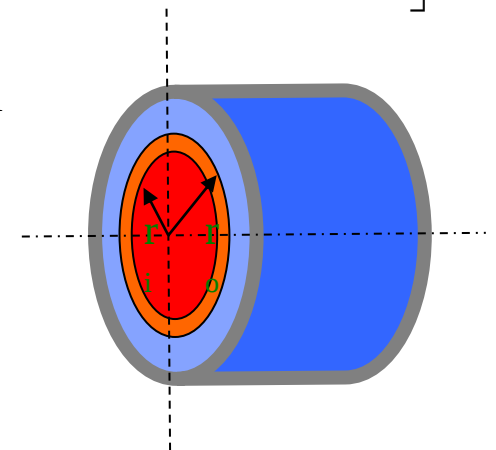
$$q_x = U \cdot A \cdot (T_h - T_c)$$

$$T_h - T_c = q_x \left[\frac{1}{h_h \cdot A_i} + \frac{\ln \left(\frac{r_o}{r_i} \right)}{k_{copper} \cdot 2\pi L} + \frac{1}{h_c \cdot A_o} \right]$$

$$U = \frac{1}{A \cdot \Sigma R}$$

$$U = \left[\frac{r_o}{h_{hot} \cdot r_i} + \frac{r_o \cdot \ln \left(\frac{r_o}{r_i} \right)}{k_{copper} \cdot r_i} + \frac{1}{h_{cold}} \right]^{-1}$$

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Use of The Log Mean Temperature Difference

$$\left. \begin{array}{l} \text{Hot Stream : } dq_h = \dot{m}_h \cdot C_p^h \cdot dT_h \\ \text{Cold Stream: } dq_c = \dot{m}_c \cdot C_p^c \cdot dT_c \end{array} \right\} \begin{array}{l} d(\Delta T) = dT_h - dT_c \\ \Delta T = T_h - T_c \end{array} \rightarrow d(\Delta T) = \left(\frac{dq_h}{\dot{m}_h \cdot C_p^h} - \frac{dq_c}{\dot{m}_c \cdot C_p^c} \right)$$

$$\left. \begin{array}{l} dq = -dq_{hot} = dq_{cold} \\ -dq = -U \cdot \Delta T \cdot dA \end{array} \right\} d(\Delta T) = -U \cdot \Delta T \cdot dA \left(\frac{1}{\dot{m}_h \cdot C_p^h} + \frac{1}{\dot{m}_c \cdot C_p^c} \right)$$

$$\int_{\Delta T_1}^{\Delta T_2} \frac{d(\Delta T)}{\Delta T} = -U \cdot \left(\frac{\Delta T_h}{q_h} + \frac{\Delta T_c}{q_c} \right) \cdot \int_{A_1}^{A_2} dA$$

$$\int_{\Delta T_1}^{\Delta T_2} \frac{d(\Delta T)}{\Delta T} = -U \cdot \left(\frac{1}{\dot{m}_h \cdot C_p^h} + \frac{1}{\dot{m}_c \cdot C_p^c} \right) \cdot \int_{A_1}^{A_2} dA$$

$$\ln \left(\frac{\Delta T_2}{\Delta T_1} \right) = -\frac{U \cdot A}{q} (\Delta T_h + \Delta T_c) = -\frac{U \cdot A}{q} [(T_h^{in} - T_h^{out}) - (T_c^{in} - T_c^{out})]$$

$$q = U \cdot A \frac{\Delta T_2 - \Delta T_1}{\ln \left(\frac{\Delta T_2}{\Delta T_1} \right)}$$

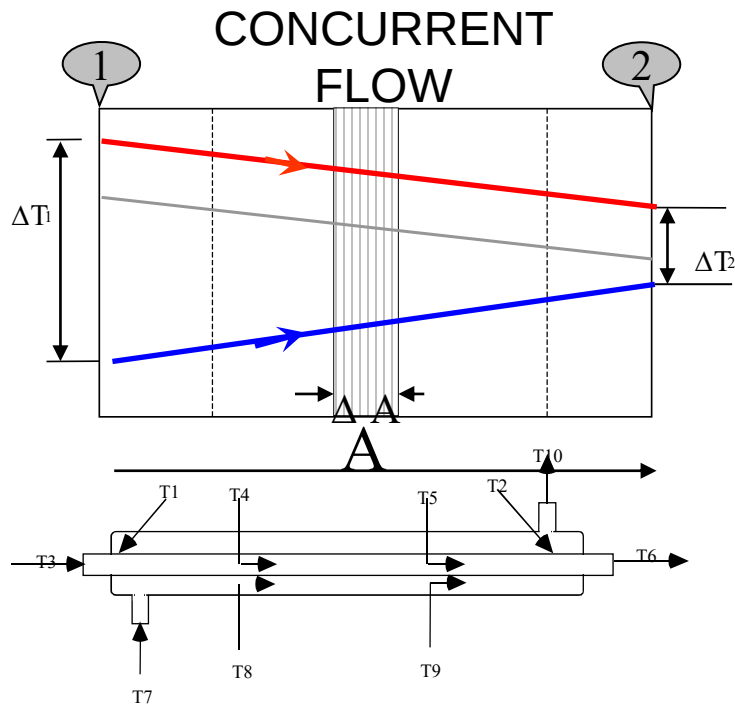
Log Mean Temperature

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$$\Delta T_{Ln} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)}$$



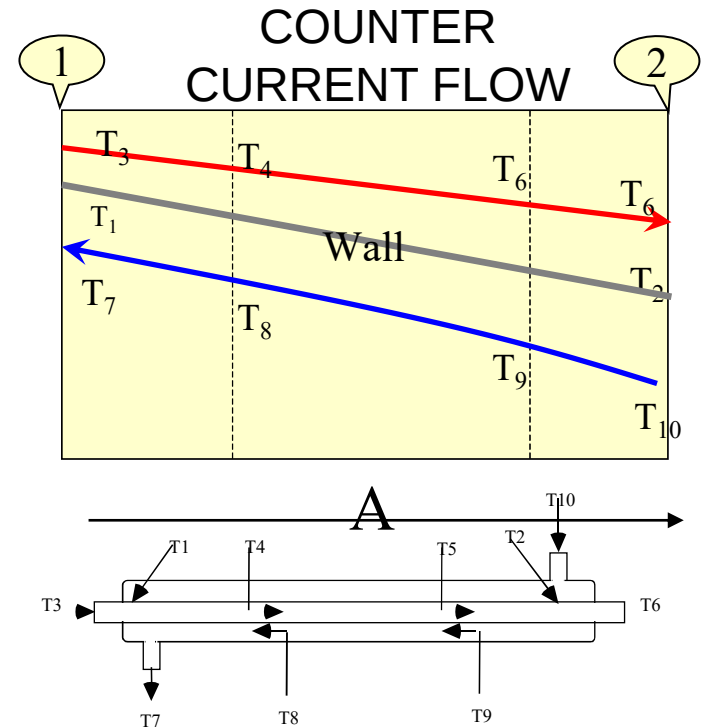
$$U = \frac{\dot{m}_h \cdot \dot{C}_p^h \cdot (T_3 - T_6)}{A \cdot \Delta T_{Ln}} = \frac{\dot{m}_c \cdot \dot{C}_p^c \cdot (T_7 - T_{10})}{A \cdot \Delta T_{Ln}}$$



Parallel Flow

$$\Delta T_1 = T_h^{in} - T_c^{in} = T_3 - T_7$$

$$\Delta T_2 = T_h^{out} - T_c^{out} = T_6 - T_{10}$$



Counter - Current Flow

$$\Delta T_1 = T_h^{in} - T_c^{out} = T_3 - T_7$$

$$\Delta T_2 = T_h^{out} - T_c^{in} = T_6 - T_{10}$$

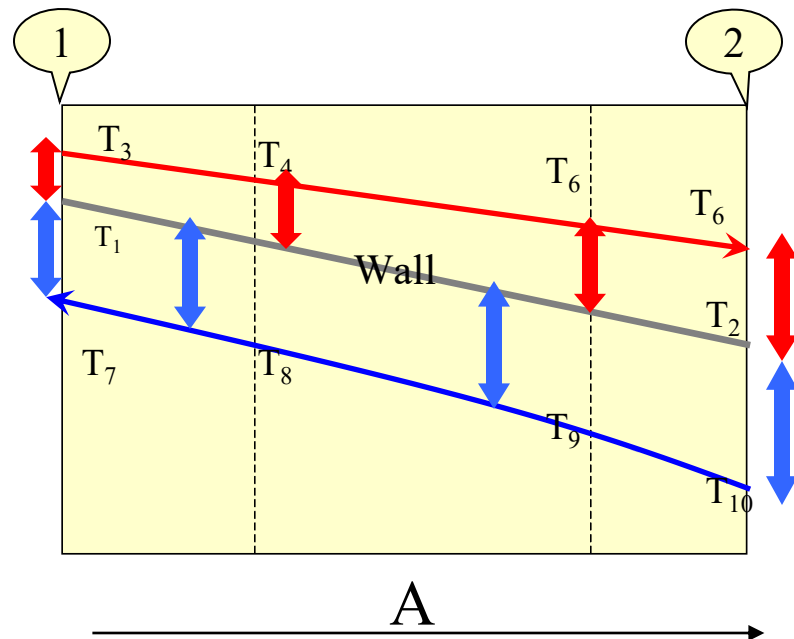
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$$q = h_h A_i \Delta T_{lm}$$

$$\Delta T_{lm} = \frac{(T_3 - T_1) - (T_6 - T_2)}{\ln \frac{(T_3 - T_1)}{(T_6 - T_2)}}$$

$$q = h_c A_o \Delta T_{lm}$$

$$\Delta T_{lm} = \frac{(T_1 - T_7) - (T_2 - T_{10})}{\ln \frac{(T_1 - T_7)}{(T_2 - T_{10})}}$$



Effectiveness – NTU Method

To define the effectiveness of a heat exchanger, we must first determine the maximum possible heat transfer rate, $q_{\max}$, for the exchanger.

This heat transfer rate could, in principle, be achieved in a counter flow heat exchanger of infinite length.

In such an exchanger, one of the fluids would experience the maximum possible temperature difference, $T_{h,i} - T_{c,i}$. To illustrate this point, consider a situation for which $C_c < C_h$, in which case, $|dT_c| > |dT_h|$. The cold fluid would then experience the larger temperature change, and since $L \rightarrow \infty$, it would be heated to the inlet temperature of the hot fluid ($T_{c,o} = T_{h,i}$).

Accordingly:

$$C_c < C_h: \quad q_{\max} = C_c(T_{h,i} - T_{c,i})$$

Cont'd ...

Similarly, if $C_h < C_c$, the hot fluid would experience the larger temperature change and would be cooled to the inlet temperature of the cold fluid ($T_{c,o} = T_{h,i}$).

From the foregoing results we are then prompted to write the general expression:

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

where $C_{\min}$ is equal to C_c or C_h , whichever is smaller.

It is now logical to define the effectiveness, ε , as the ratio of the actual heat transfer rate for a heat exchanger to the maximum possible heat transfer rate:

$$\varepsilon \equiv \frac{q}{q_{\max}}$$

$$\varepsilon = \frac{C_h(T_{h,i} - T_{h,o})}{C_{\min}(T_{h,i} - T_{c,i})}$$

$$\varepsilon = \frac{C_c(T_{c,o} - T_{c,i})}{C_{\min}(T_{h,i} - T_{c,i})}$$

Cont'd ...

By definition the effectiveness, which is dimensionless, must be in the range $0 \leq \varepsilon \leq 1$. It is useful because, if ε , $T_{h,i}$, and $T_{c,i}$ are known, the actual heat transfer rate may readily be determined from the expression.

$$q = \varepsilon C_{\min}(T_{h,i} - T_{c,i})$$

For any heat exchanger it can be shown that:

$$\varepsilon = f\left(\text{NTU}, \frac{C_{\min}}{C_{\max}}\right)$$

where $C_{\min}/C_{\max}$ is equal to C_c/C_h or C_h/C_c , depending on the relative magnitudes of the hot and cold fluid heat capacity rates. The number of transfer units (NTU) is a dimensionless parameter that is widely used for heat exchanger analysis and is defined as:

$$\text{NTU} \equiv \frac{UA}{C_{\min}}$$

Effectiveness – NTU Relations

To determine a specific form of the effectiveness–NTU relation, consider a parallel-flow heat exchanger for which $C_{\min} = C_h$. we then obtain:

$$\varepsilon = \frac{T_{h,i} - T_{h,o}}{T_{h,i} - T_{c,i}}$$

$$\frac{C_{\min}}{C_{\max}} = \frac{\dot{m}_h c_{p,h}}{\dot{m}_c c_{p,c}} = \frac{T_{c,o} - T_{c,i}}{T_{h,i} - T_{h,o}}$$

$$\ln \left(\frac{T_{h,o} - T_{c,o}}{T_{h,i} - T_{c,i}} \right) = -\frac{UA}{C_{\min}} \left(1 + \frac{C_{\min}}{C_{\max}} \right)$$

$$\frac{T_{h,o} - T_{c,o}}{T_{h,i} - T_{c,i}} = \exp \left[-NTU \left(1 + \frac{C_{\min}}{C_{\max}} \right) \right] \dots *$$

$$\frac{T_{h,o} - T_{c,o}}{T_{h,i} - T_{c,i}} = \frac{T_{h,o} - T_{h,i} + T_{h,i} - T_{c,o}}{T_{h,i} - T_{c,i}}$$

Cont'd ...

$$\frac{T_{h,o} - T_{c,o}}{T_{h,i} - T_{c,i}} = \frac{(T_{h,o} - T_{h,i}) + (T_{h,i} - T_{c,i}) - (C_{\min}/C_{\max})(T_{h,i} - T_{h,o})}{T_{h,i} - T_{c,i}}$$

$$\frac{T_{h,o} - T_{c,o}}{T_{h,i} - T_{c,i}} = -\varepsilon + 1 - \left(\frac{C_{\min}}{C_{\max}}\right)\varepsilon = 1 - \varepsilon \left(1 + \frac{C_{\min}}{C_{\max}}\right)$$

Substituting the above expression into Equation * and solving for ε , we obtain for the parallel-flow heat exchanger:

$$\varepsilon = \frac{1 - \exp \{-NTU[1 + (C_{\min}/C_{\max})]\}}{1 + (C_{\min}/C_{\max})}$$

Since precisely the same result may be obtained for $C_{\min} = C_c$, the above equation applies for any parallel-flow heat exchanger, irrespective of whether the minimum heat capacity rate is associated with the hot or cold fluid.

Cont'd ...

Similar expressions have been developed for a variety of heat exchangers, and representative results are summarized in Table below: where C_r is the heat capacity ratio $C_r = C_{\min}/C_{\max}$.

Heat Exchanger Effectiveness Relations [5]

Flow Arrangement	Relation	
Concentric tube		
Parallel flow	$\varepsilon = \frac{1 - \exp[-NTU(1 + C_r)]}{1 + C_r}$	(11.28a)
Counterflow	$\varepsilon = \frac{1 - \exp[-NTU(1 - C_r)]}{1 - C_r \exp[-NTU(1 - C_r)]} \quad (C_r < 1)$	
	$\varepsilon = \frac{NTU}{1 + NTU} \quad (C_r = 1)$	(11.29a)
Shell-and-tube		
One shell pass (2, 4, ... tube passes)	$\varepsilon_1 = 2 \left\{ 1 + C_r + (1 + C_r^2)^{1/2} \times \frac{1 + \exp[-(NTU)_1(1 + C_r^2)^{1/2}]}{1 - \exp[-(NTU)_1(1 + C_r^2)^{1/2}]} \right\}^{-1}$	(11.30a)
n Shell passes (2n, 4n, ... tube passes)	$\varepsilon = \left[\left(\frac{1 - \varepsilon_1 C_r}{1 - \varepsilon_1} \right)^n - 1 \right] \left[\left(\frac{1 - \varepsilon_1 C_r}{1 - \varepsilon_1} \right)^n - C_r \right]^{-1}$	(11.31a)
Cross-flow (single pass)		
Both fluids unmixed	$\varepsilon = 1 - \exp \left[\left(\frac{1}{C_r} \right) (NTU)^{0.22} \{ \exp[-C_r(NTU)^{0.78}] - 1 \} \right]$	(11.32)
$C_{\max}$ (mixed), $C_{\min}$ (unmixed)	$\varepsilon = \left(\frac{1}{C_r} \right) (1 - \exp \{ -C_r [1 - \exp(-NTU)] \})$	(11.33a)
$C_{\min}$ (mixed), $C_{\max}$ (unmixed)	$\varepsilon = 1 - \exp(-C_r^{-1} \{ 1 - \exp[-C_r(NTU)] \})$	(11.34a)
All exchangers ($C_r = 0$)	$\varepsilon = 1 - \exp(-NTU)$	(11.35a)

Cont'd ...

In heat exchanger design calculations, it is more convenient to work with ε –NTU relations of the form:

$$\text{NTU} = f\left(\varepsilon, \frac{C_{\min}}{C_{\max}}\right)$$

For a shell-and-tube heat exchanger with multiple shell passes, it is assumed that the total NTU is equally distributed between shell passes of the same arrangement, $\text{NTU} = n(\text{NTU})_1$.

Finally, this result would be multiplied by n to obtain the NTU for the entire exchanger, as indicated in Equation 11.31d in table below.

Cont'd ...

Heat Exchanger NTU Relations

Flow Arrangement	Relation	
Concentric tube		
Parallel flow	$NTU = - \frac{\ln [1 - \varepsilon(1 + C_r)]}{1 + C_r}$	(11.28b)
Counterflow	$NTU = \frac{1}{C_r - 1} \ln \left(\frac{\varepsilon - 1}{\varepsilon C_r - 1} \right) \quad (C_r < 1)$	
	$NTU = \frac{\varepsilon}{1 - \varepsilon} \quad (C_r = 1)$	(11.29b)
Shell-and-tube		
One shell pass (2, 4, . . . tube passes)	$(NTU)_1 = - (1 + C_r^2)^{-1/2} \ln \left(\frac{E - 1}{E + 1} \right)$	(11.30b)
	$E = \frac{2/\varepsilon_1 - (1 + C_r)}{(1 + C_r^2)^{1/2}}$	(11.30c)
n Shell passes ($2n, 4n, . . .$ tube passes)	Use Equations 11.30b and 11.30c with $\varepsilon_1 = \frac{F - 1}{F - C_r} \quad F = \left(\frac{\varepsilon C_r - 1}{\varepsilon - 1} \right)^{1/n} \quad NTU = n(NTU)_1 \quad (11.31b, c, d)$	
Cross-flow (single pass)		
$C_{\max}$ (mixed), $C_{\min}$ (unmixed)	$NTU = - \ln \left[1 + \left(\frac{1}{C_r} \right) \ln(1 - \varepsilon C_r) \right]$	(11.33b)
$C_{\min}$ (mixed), $C_{\max}$ (unmixed)	$NTU = - \left(\frac{1}{C_r} \right) \ln [C_r \ln(1 - \varepsilon) + 1]$	(11.34b)
All exchangers ($C_r = 0$)	$NTU = - \ln(1 - \varepsilon)$	(11.35b)

Cont'd ...

The foregoing expressions are represented graphically in Figures 11.10 through 11.15. When using Figure 11.13, the abscissa corresponds to the total number of transfer units, $NTU = n(NTU)_1$.

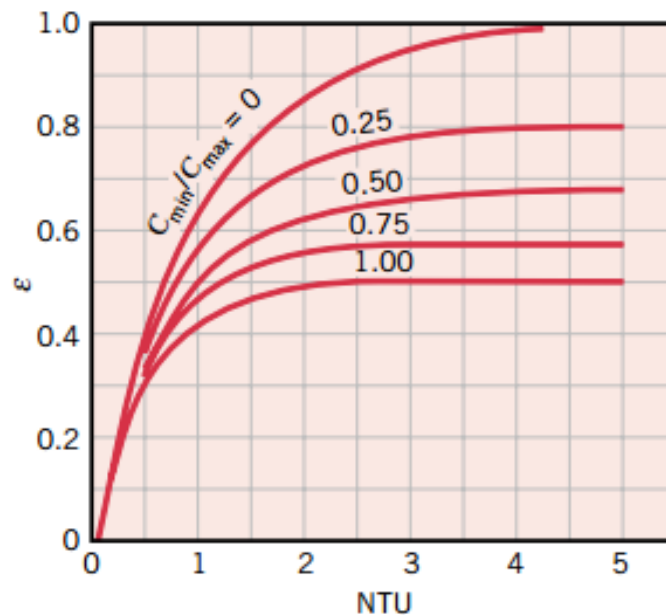


FIGURE 11.10 Effectiveness of a parallel-flow heat exchanger (Equation 11.28).

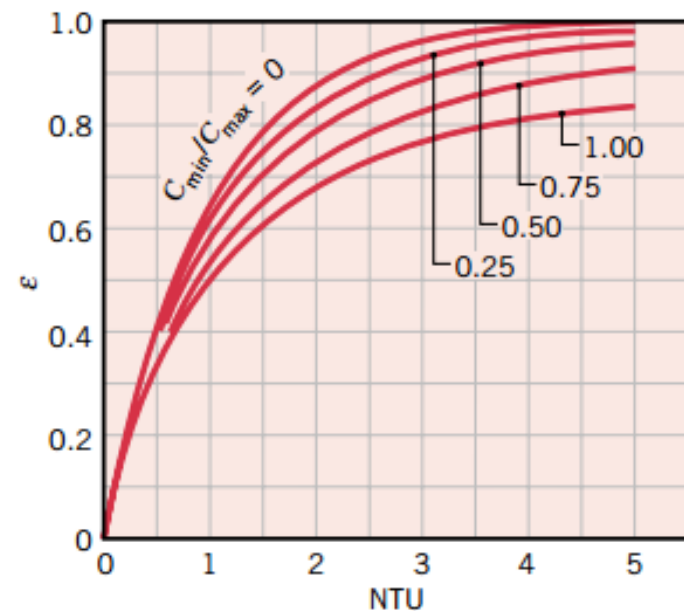


FIGURE 11.11 Effectiveness of a counterflow heat exchanger (Equation 11.29).

Cont'd ...

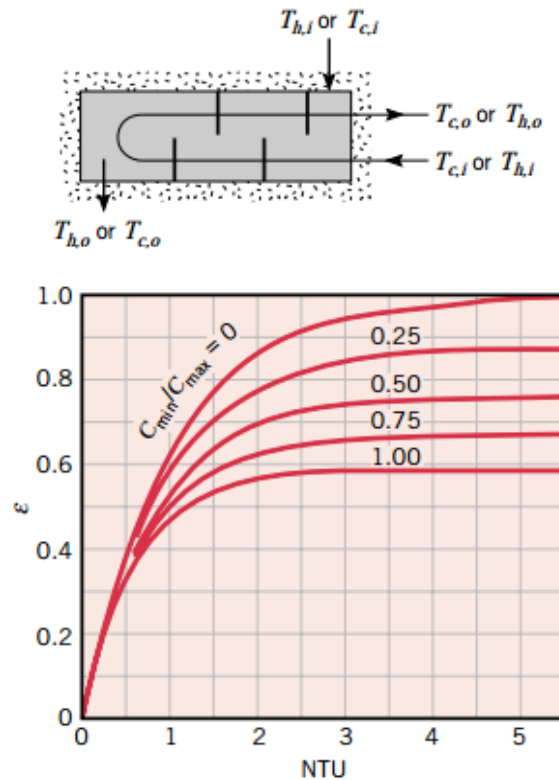


FIGURE 11.12 Effectiveness of a shell-and-tube heat exchanger with one shell and any multiple of two tube passes (two, four, etc. tube passes) (Equation 11.30).

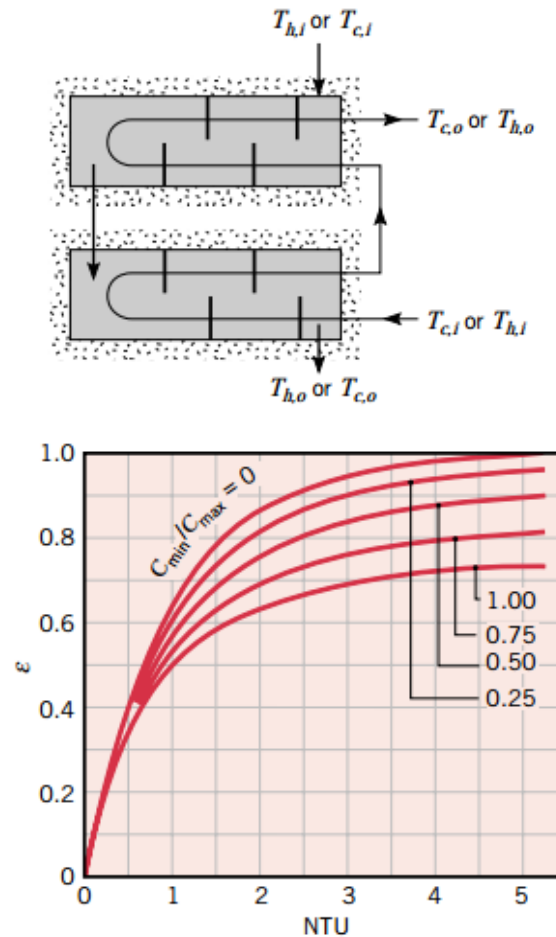


FIGURE 11.13 Effectiveness of a shell-and-tube heat exchanger with two shell passes and any multiple of four tube passes (four, eight, etc. tube passes) (Equation 11.31 with $n = 2$).

Cont'd ...

For Figure 11.15 the solid curves correspond to $C_{\min}$ mixed and $C_{\max}$ unmixed, while the dashed curves correspond to $C_{\min}$ unmixed and $C_{\max}$ mixed.

Note:

- ✓ Note that for $C_r = 0$, all heat exchangers have the same effectiveness, which may be computed from Equation 11.35a. Moreover, if $NTU \leq 0.25$, all heat exchangers have approximately the same effectiveness, regardless of the value of C_r , and ε may again be computed from Equation 11.35a.
- ✓ More generally, for $C_r > 0$ and $NTU \geq 0.25$, the counter flow exchanger is the most effective.
- ✓ For any exchanger, maximum and minimum values of the effectiveness are associated with $C_r = 0$ and $C_r = 1$, respectively

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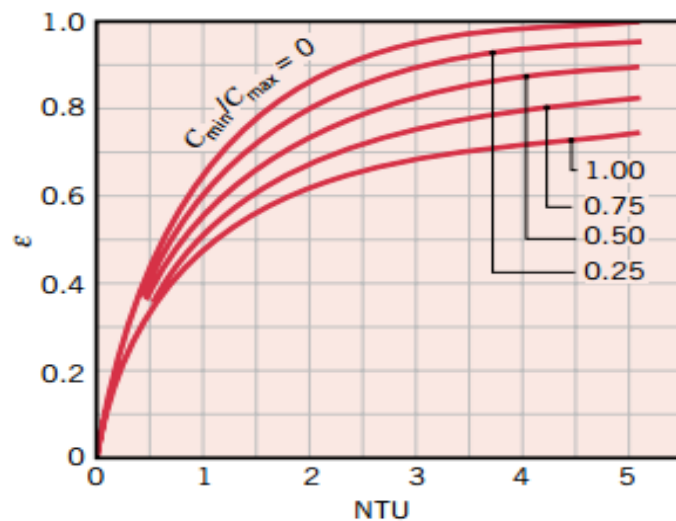
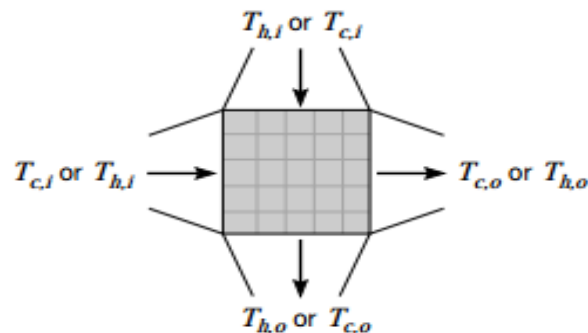


FIGURE 11.14 Effectiveness of a single-pass, cross-flow heat exchanger with both fluids unmixed (Equation 11.32).

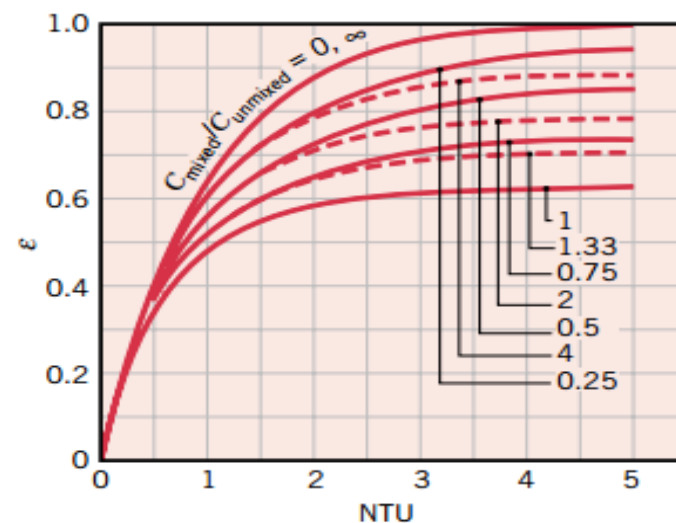
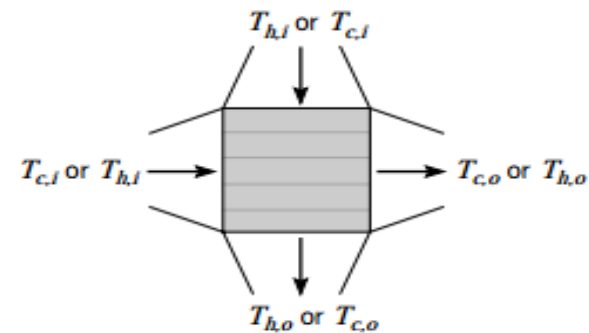


FIGURE 11.15 Effectiveness of a single-pass, cross-flow heat exchanger with one fluid mixed and the other unmixed (Equations 11.33, 11.34).

Cont'd ...

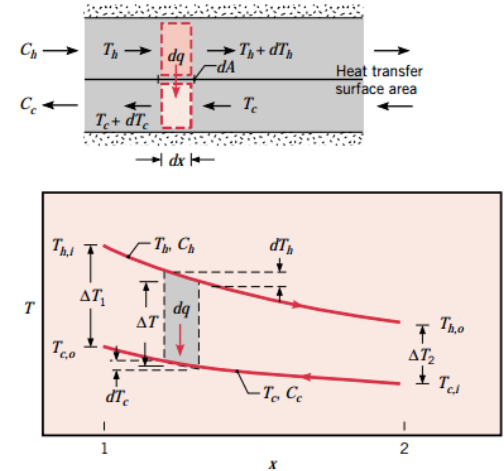
We also note that both the LMTD and NTU methods approach heat exchanger analysis from a global perspective and provide no information concerning conditions within the exchanger. Although flow and temperature variations within a heat exchanger may be determined using commercial CFD (computational fluid dynamic) computer codes, simpler numerical procedures may be adopted.

Special Operating Conditions

It is useful to note certain special conditions under which heat exchangers may be operated. Other than fig shown on the top right corner.

When the hot fluid has a heat capacity rate, $C_h = m_h c_{p,h}$, which is much larger than that of the cold fluid, $C_c = m_c c_{p,c}$. For this case the temperature of the hot fluid remains approximately constant throughout the heat exchanger, while the temperature of the cold fluid increases. The same condition is achieved if the hot fluid is a condensing vapor. Condensation occurs at constant temperature, and, for all practical purposes, $C_h \rightarrow \infty$.

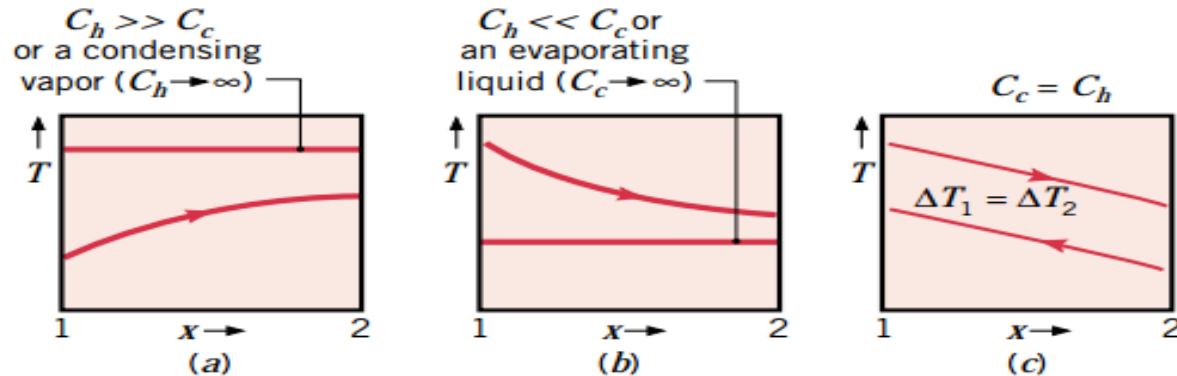
Conversely, in an evaporator or a boiler, it is the cold fluid that experiences a change in phase and remains at a nearly uniform temperature ($C_c \rightarrow \infty$).



Temperature distributions for a counterflow heat exchanger.

Cont'd ...

The third special case involves a counter flow heat exchanger for which the heat capacity rates are equal ($C_h = C_c$). The temperature difference ΔT must then be a constant throughout the exchanger, in which case $\Delta T_1 = \Delta T_2 = \Delta T_{lm}$.



Special heat exchanger conditions. (a) $C_h \gg C_c$ or a condensing vapor. (b) An evaporating liquid or $C_h \ll C_c$. (c) A counterflow heat exchanger with equivalent fluid heat capacities ($C_h = C_c$).

Although flow conditions are more complicated in multi-pass and cross-flow heat exchangers, Equations may still be used if modifications are made to the definition of the log mean temperature difference.

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

The log mean temperature difference T_{lm} relation developed earlier is limited to parallel-flow and counter-flow heat exchangers only. Similar relations are also developed for cross-flow and multi-pass 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:

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

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.

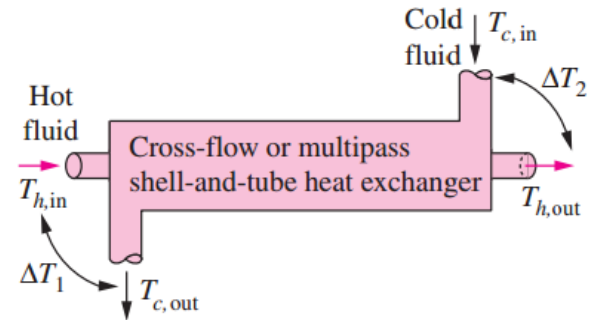
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The correction factor is less than unity for a cross-flow and multi-pass shell and-tube heat exchanger. That is, $F < 1$. The limiting value of $F = 1$ corresponds to the counter-flow heat exchanger. Thus, the correction factor F for a heat exchanger is a measure of deviation of the T_{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 Figure 13–18 versus two temperature ratios P and R defined as:

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

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



Heat transfer rate:

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

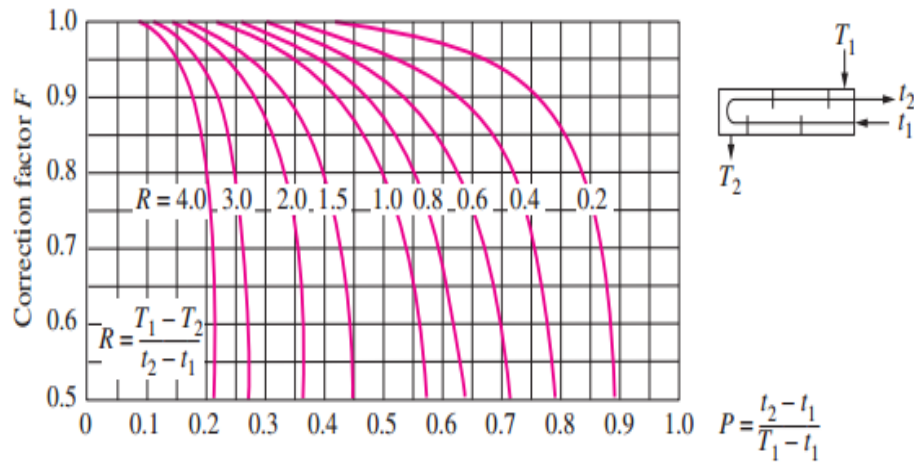
where

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

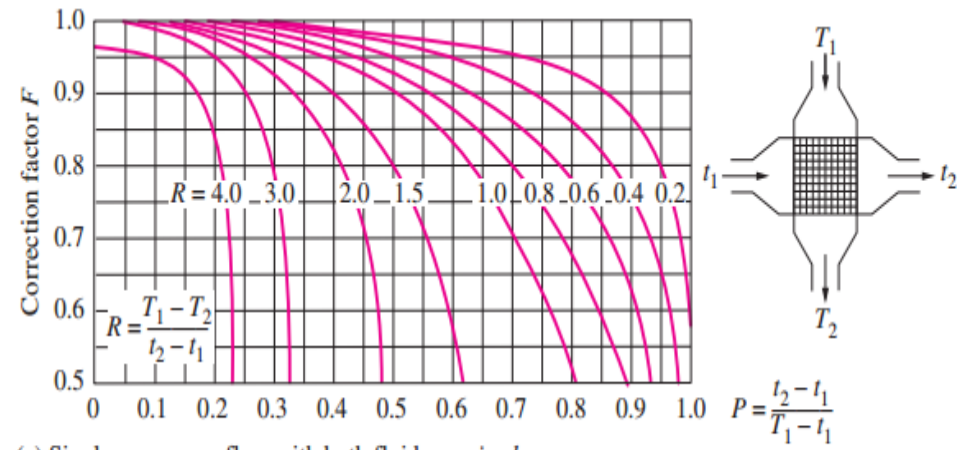
$$\Delta T_1 = T_{h,in} - T_{c,out}$$

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

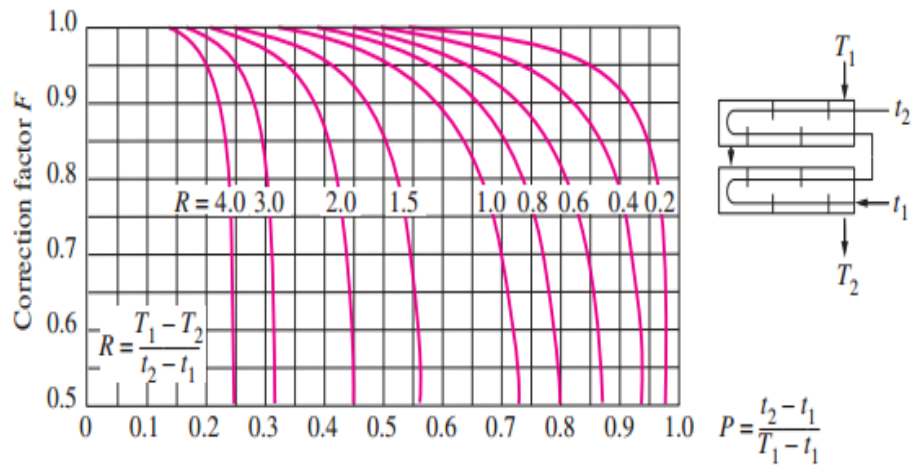
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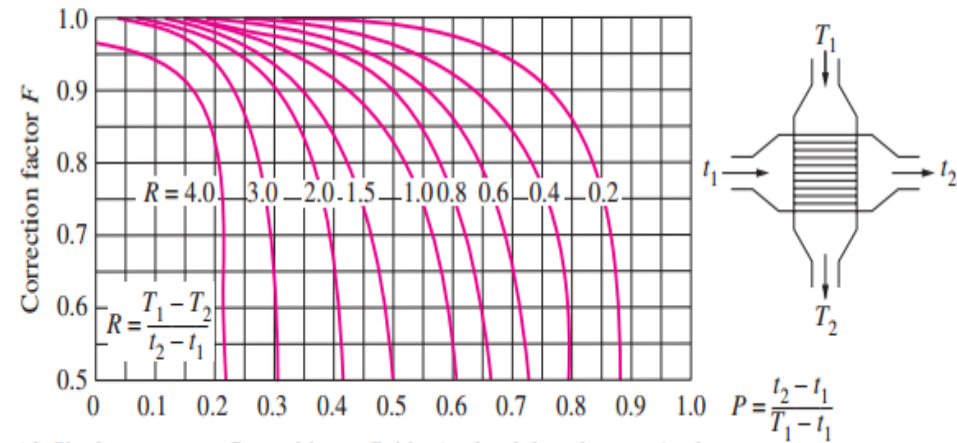
(a) One-shell pass and 2, 4, 6, etc. (any multiple of 2), tube passes



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



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



(d) Single-pass cross-flow with one fluid *mixed* and the other *unmixed*

Heat Exchanger Design and Performance Checking Procedures

Two general types of heat exchanger problems are commonly encountered by the practicing engineer:

1. Heat Exchanger Design Problem

Specifications:

- ❑ The fluid inlet temperatures and flow rates, as well as a desired hot or cold fluid outlet temperature, are prescribed.

Design Output:

- ❑ The design problem is then one of specifying a specific heat exchanger type and determining its size—that is, the heat transfer surface area, A required to achieve the desired outlet temperature.

Note:

The design problem is commonly encountered when a heat exchanger is to be custom-built for a specific application.

Performance Assessment of Heat Exchangers

2. Heat Exchanger Performance Problem

- ❑ In a heat exchanger performance calculation, an existing heat exchanger is analyzed to determine the heat transfer rate and the fluid outlet temperatures for prescribed flow rates and inlet temperatures.
- ❑ The performance calculation is commonly associated with the use of off-the-shelf heat exchanger types and sizes available from a vendor.

Selection of Heat Exchangers

Heat transfer enhancement in heat exchangers is usually accompanied by increased pressure drop, and thus higher pumping power. Therefore, any gain from the enhancement in heat transfer should be weighed against the cost of the accompanying pressure drop.

Engineers in industry often find themselves in a position to select heat exchangers to accomplish certain heat transfer tasks. **The parameters to be taken under consideration in selection are:**

- Heat Transfer Rate
- Cost
- Pumping Power
- Size and Weight
- Type Materials

Other Considerations

- Being leak-tight is an important consideration when toxic or expensive fluids
- Ease of servicing, low maintenance cost, and safety and reliability

Thank You

Questions please ... ?

Problems

Problem 1 – Cooling by Double Pipe Heat Exchanger (LMTD App.)

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

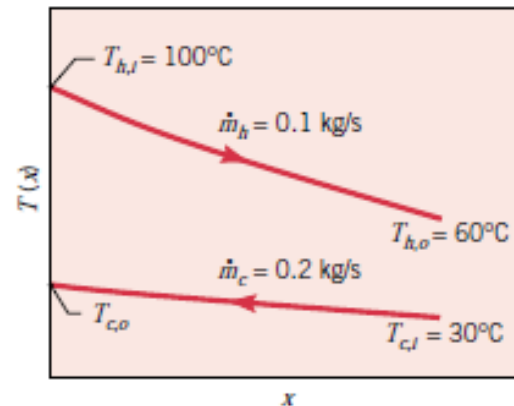
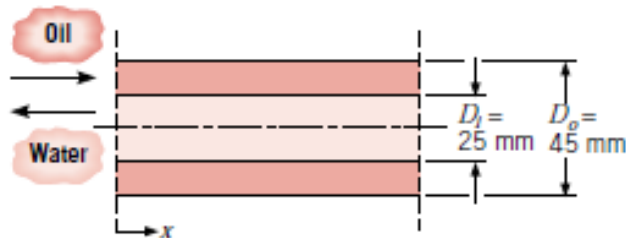
	Hot fluid	Cold fluid
Heat capacity rate, kW/K	6	3
Inlet temperature, °C	60	30
Outlet temperature, °C	—	54

- (a) Determine the outlet temperature of the hot fluid.
- (b) Is the heat exchanger operating in counter flow 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.
- (e) What would be the effectiveness of this exchanger if its length were made very large?

Problem 2 – Counter Flow, Double Pipe Heat Exchanger (LMTD App.)

A counter flow, concentric tube heat exchanger is used to cool the lubricating oil for a large industrial gas turbine engine. The flow rate of cooling water through the inner tube ($D_i = 25$ mm) is 0.2 kg/s, while the flow rate of oil through the outer annulus ($D_o = 45$ mm) is 0.1 kg/s. The oil and water enter at temperatures of 100 and 30°C, respectively. How long must the tube be made if the outlet temperature of the oil is to be 60°C?

Schematic:



Properties: Table A.5, unused engine oil ($\bar{T}_h = 80^\circ\text{C} = 353$ K): $c_p = 2131$ J/kg $\cdot$ K, $\mu = 3.25 \times 10^{-2}$ N $\cdot$ s/m², $k = 0.138$ W/m $\cdot$ K. Table A.6, water ($\bar{T}_c \approx 35^\circ\text{C}$): $c_p = 4178$ J/kg $\cdot$ K, $\mu = 725 \times 10^{-6}$ N $\cdot$ s/m², $k = 0.625$ W/m $\cdot$ K, $Pr = 4.85$.

Problem 3 – Chilling of Milk (ϵ – NTU Approach)

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. Ground water at 10°C counter flow concentric tube heat exchanger is available at a flow rate of $0.72 \text{ m}^3/\text{h}$. The density and specific heat of milk are $1030 \text{ kg}/\text{m}^3$ and $3860 \text{ J}/\text{Kg K}$, respectively.

(Safe-to-store temperature of milk is $\leq 13^{\circ}\text{C}$)

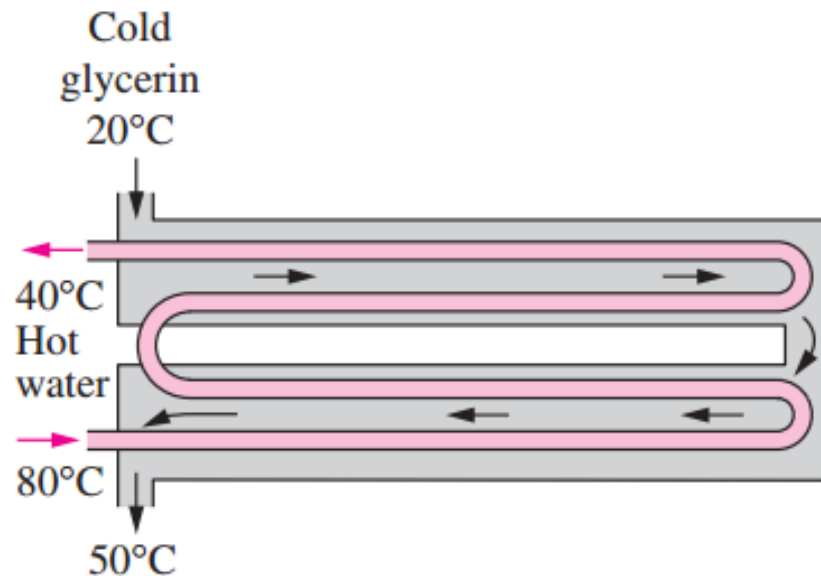
- (a) Determine the length of the exchanger if the inner pipe has a 50-mm diameter and the overall heat transfer coefficient is $U = 1000 \text{ W}/\text{m}^2 \text{ K}$.
- (b) Determine the outlet temperature of the water.

PROPERTIES: *Table A-6*, Water ($\bar{T}_c = 287 \text{ K}$, assume $T_{c,o} = 18^{\circ}\text{C}$): $\rho = 1000 \text{ kg}/\text{m}^3$

$c_p = 4187 \text{ J}/\text{kg} \cdot \text{K}$; Milk (given): $\rho = 1030 \text{ kg}/\text{m}^3$, $c_p = 3860 \text{ J}/\text{kg} \cdot \text{K}$.

Problem 4 - Heating by Multi-Pass Heat Exchanger

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 . The total length of the tubes in the heat exchanger is 60 m. The convection heat transfer coefficient is $25 \text{ W/m}^2 \cdot ^{\circ}\text{C}$ on the glycerin (shell) side and $160 \text{ W/m}^2 \cdot ^{\circ}\text{C}$ on the water (tube) side. Determine the rate of heat transfer in the heat exchanger.



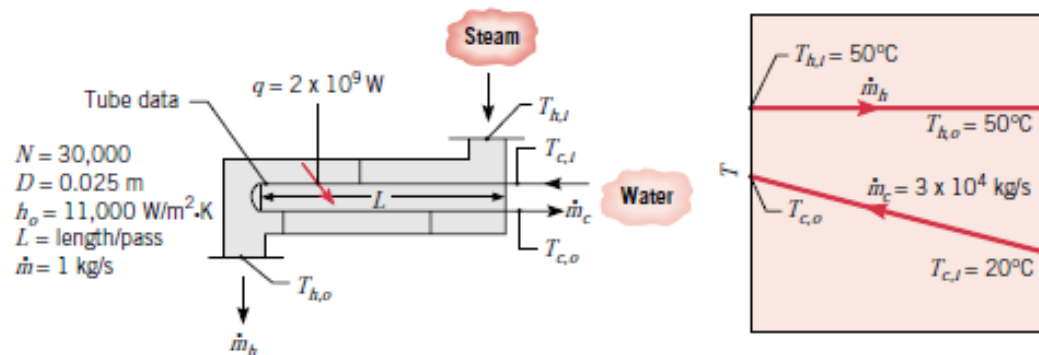
Problem 5 – Cooling by Cross-flow H.E (ϵ – NTU Approach)

A cross-flow heat exchanger consists of a bundle of 32 tubes in a 0.6 m^2 duct. Hot water at 150°C and a mean velocity of 0.5 m/s enters the tubes having inner and outer diameters of 10.2 and 12.5 mm . Atmospheric air at 10°C enters the exchanger with a volumetric flow rate of $1.0 \text{ m}^3/\text{s}$. The convection heat transfer coefficient on the tube outer surfaces is $400 \text{ W/m}^2 \text{ K}$. Estimate the fluid outlet temperatures.

PROPERTIES: *Table A-4, Air* ($T_{c,i} = 10^\circ\text{C} = 283 \text{ K}$, 1 atm): $\rho = 1.2407 \text{ kg/m}^3$; *Table A-4, Air* (assume $T_{c,o} \approx 40^\circ\text{C}$, $\bar{T}_c = (10 + 40)^\circ\text{C}/2 = 298 \text{ K}$, 1 atm): $c_p = 1007 \text{ J/kg}\cdot\text{K}$; *Table A-6, Water* (assume $T_{h,o} \approx 140^\circ\text{C}$, $\bar{T}_h = (140 + 150)^\circ\text{C}/2 = 418 \text{ K}$): $\rho = 1/v_f = 1/1.0850 \times 10^{-3} \text{ m}^3/\text{kg}$, $c_p = 4297 \text{ J/kg}\cdot\text{K}$, $\mu_f = 188 \times 10^{-6} \text{ N}\cdot\text{s/m}^2$, $k_f = 0.688 \text{ W/m}\cdot\text{K}$, $\text{Pr}_f = 1.18$.

Exercise – Condensation by Shell and Tube H.E

The condenser of a large steam power plant is a heat exchanger in which steam is condensed to liquid water. Assume the condenser to be a *shell-and-tube* heat exchanger consisting of a single shell and 30,000 tubes, each executing two passes. The tubes are of thin wall construction with $D = 25$ mm, and steam condenses on their outer surface with an associated convection coefficient of $h_o = 11,000$ W/m² K. The heat transfer rate that must be effected by the exchanger is $q = 2(10)^9$ W, and this is accomplished by passing cooling water through the tubes at a rate of $3(10)^4$ kg/s (the flow rate per tube is therefore 1 kg/s). The water enters at 20°C, while the steam condenses at 50°C. What is the temperature of the cooling water emerging from the condenser? What is the required tube length L per pass?



Good Luck!
