Heat and Mass Transfer, 3rd Edition
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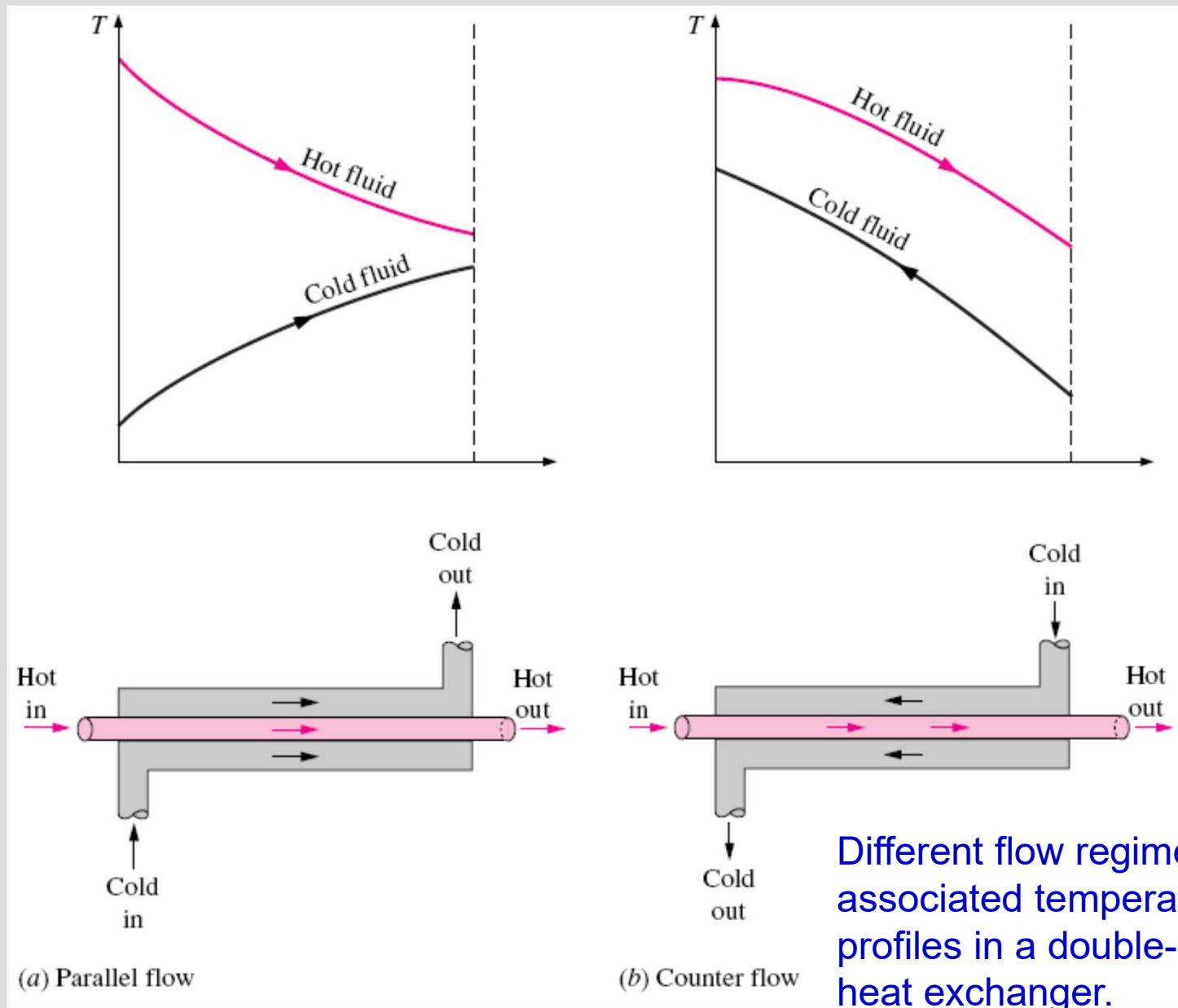**McGraw-Hill, New York, 2007**

Chapter 11
HEAT EXCHANGERS

Mehmet Kanoglu

Objectives

- Recognize numerous types of heat exchangers, and classify them

- Develop an awareness of fouling on surfaces, and determine the overall heat transfer coefficient for a heat exchanger

- Perform a general energy analysis on heat exchangers

- Obtain a relation for the logarithmic mean temperature difference for use in the LMTD method, and modify it for different types of heat exchangers using the correction factor

- Develop relations for effectiveness, and analyze heat exchangers when outlet temperatures are not known using the effectiveness-NTU method

- Know the primary considerations in the selection of heat exchangers.

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TYPES OF HEAT EXCHANGERS

(a) Parallel flow

(b) Counter flow

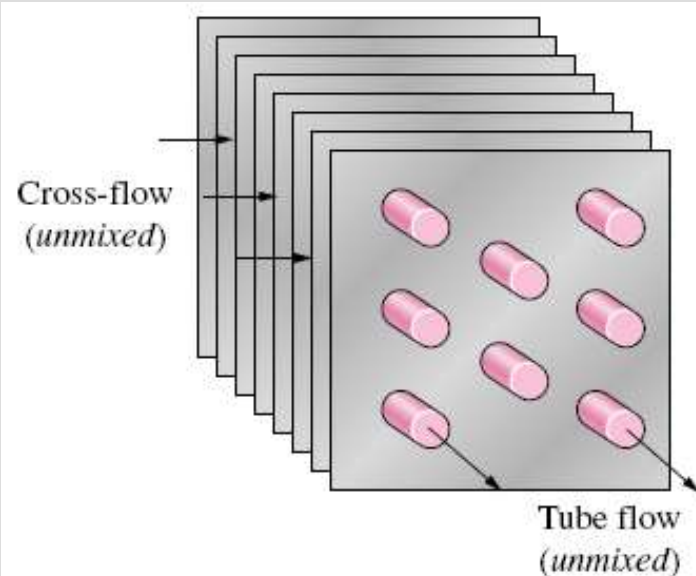Different flow regimes and associated temperature profiles in a double-pipe heat exchanger.

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Compact heat exchanger: It has a large heat transfer surface area per unit volume (e.g., car radiator, human lung). A heat exchanger with the *area density β > 700 m^2/m^3* is classified as being compact.
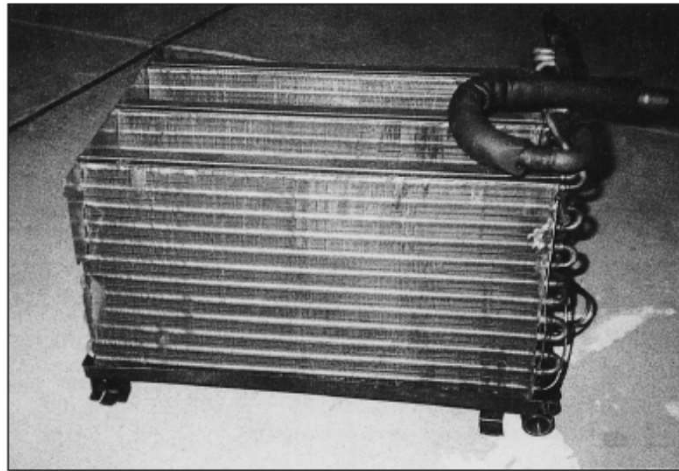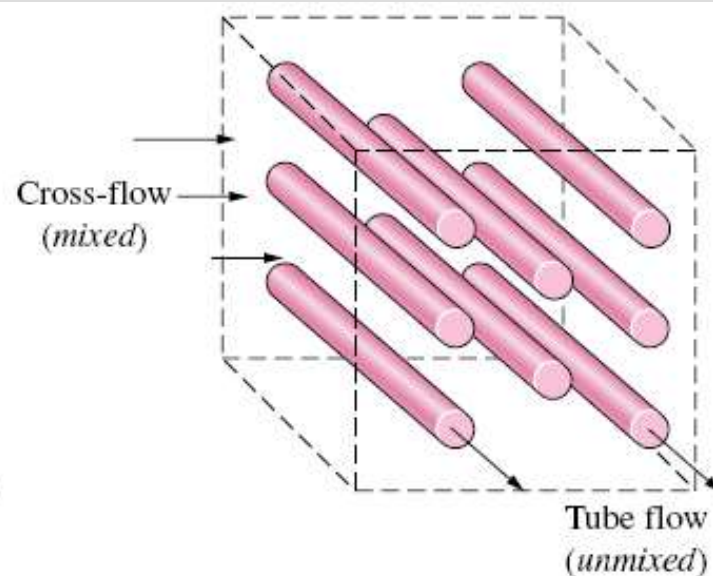
Cross-flow: In compact heat exchangers, the two fluids usually move *perpendicular* to each other. The cross-flow is further classified as *unmixed* and *mixed flow.*

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

Different flow configurations in cross-flow heat exchangers.

Cross-flow
(*unmixed*)

Cross-flow
(*mixed*)

Tube flow
(*unmixed*)

Tube flow
(*unmixed*)

(*a*) Both fluids unmixed

(*b*) One fluid mixed, one fluid unmixed

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Shell-and-tube heat exchanger: The most common type of heat exchanger in industrial applications.

They 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.
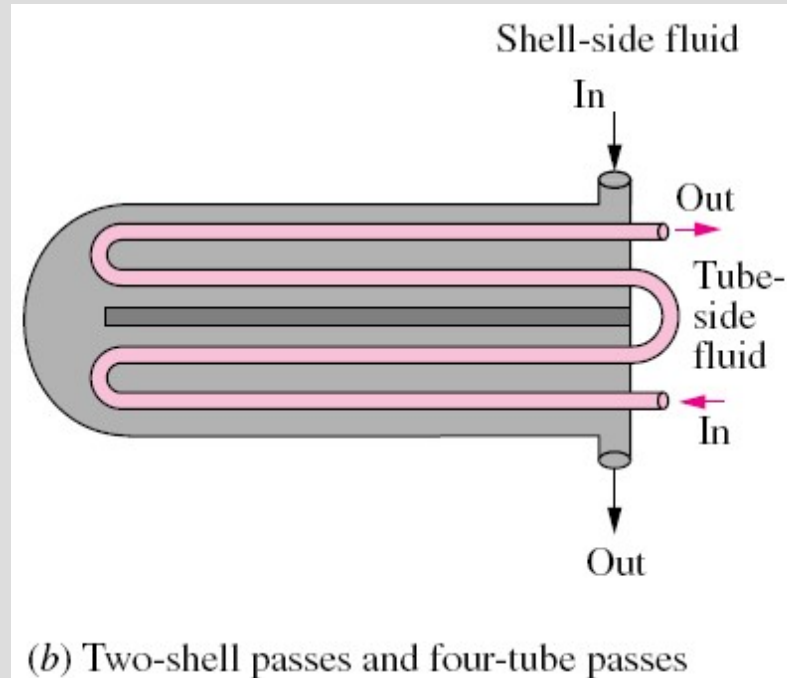
Shell-and-tube heat exchangers are further classified according to the number of shell and tube passes involved.
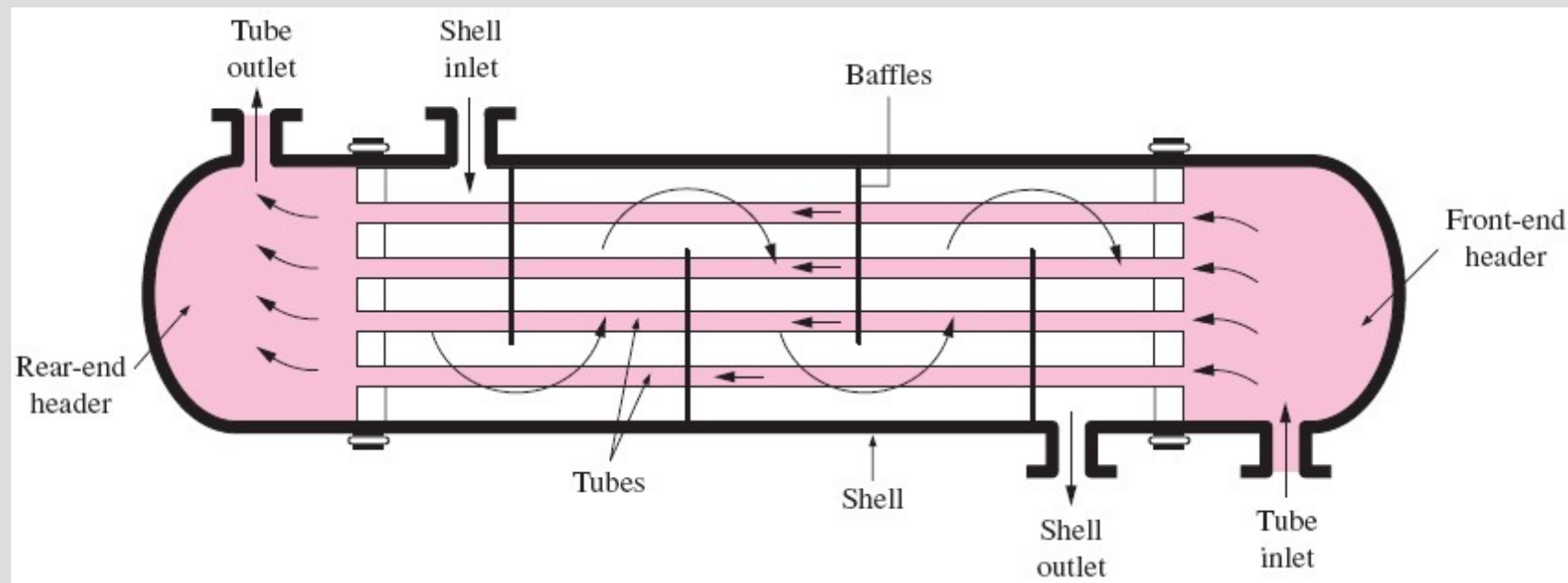
The schematic of a shell-and-tube heat exchanger (one-shell pass and one-tube pass).

(a) One-shell pass and two-tube passes

(b) Two-shell passes and four-tube passes
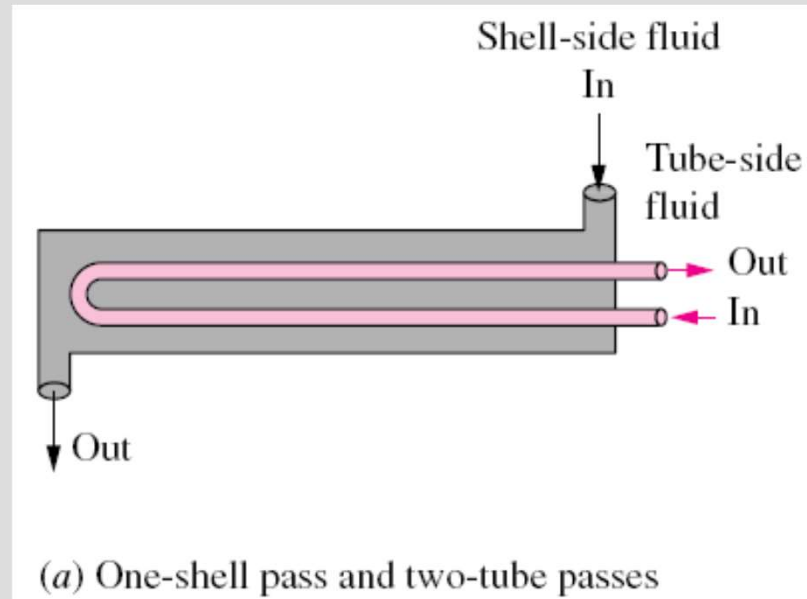
Multipass flow arrangements in shell-and-tube heat exchangers.

Regenerative heat exchanger: Involves the alternate passage of the hot and cold fluid streams through the same flow area.

Dynamic-type regenerator: Involves a rotating drum and continuous flow of the hot and cold fluid through different portions of the drum so that any portion of the drum passes periodically through the hot stream, storing heat, and then through the cold stream, rejecting this stored heat.

Condenser: One of the fluids is cooled and condenses as it flows through the heat exchanger.

Boiler: One of the fluids absorbs heat and vaporizes.

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THE OVERALL HEAT TRANSFER COEFFICIENT

- A heat exchanger typically involves two flowing fluids separated by a solid wall.

- Heat is first transferred from the hot fluid to the wall by *convection,* through the wall by *conduction,* and from the wall to the cold fluid again by *convection.*

- Any radiation effects are usually included in the convection heat transfer coefficients.
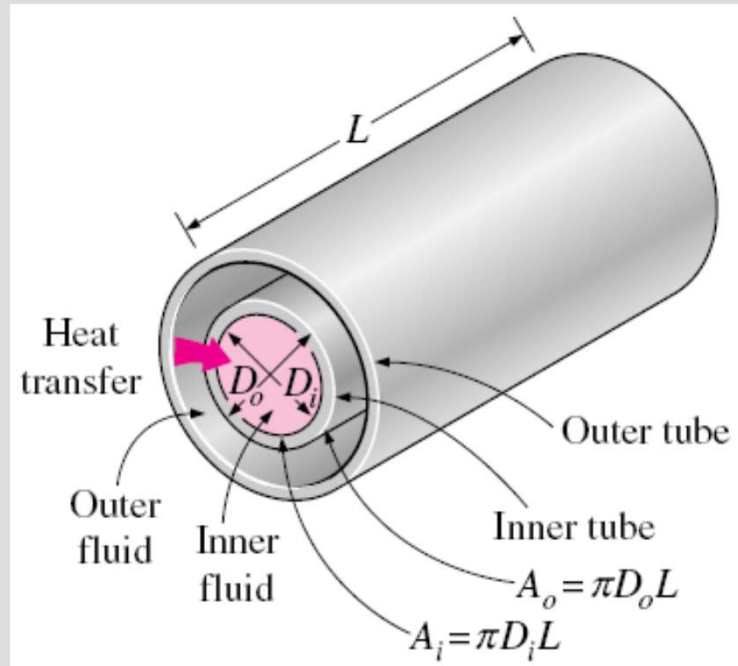
$$R_{wall} = \frac{\ln (D_o/D_i)}{2\pi kL}$$

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

$$A_i = \pi D_i L \text{ and } A_o = \pi D_o L$$

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

Heat transfer, Outer tube, Inner tube, Outer fluid, Inner fluid

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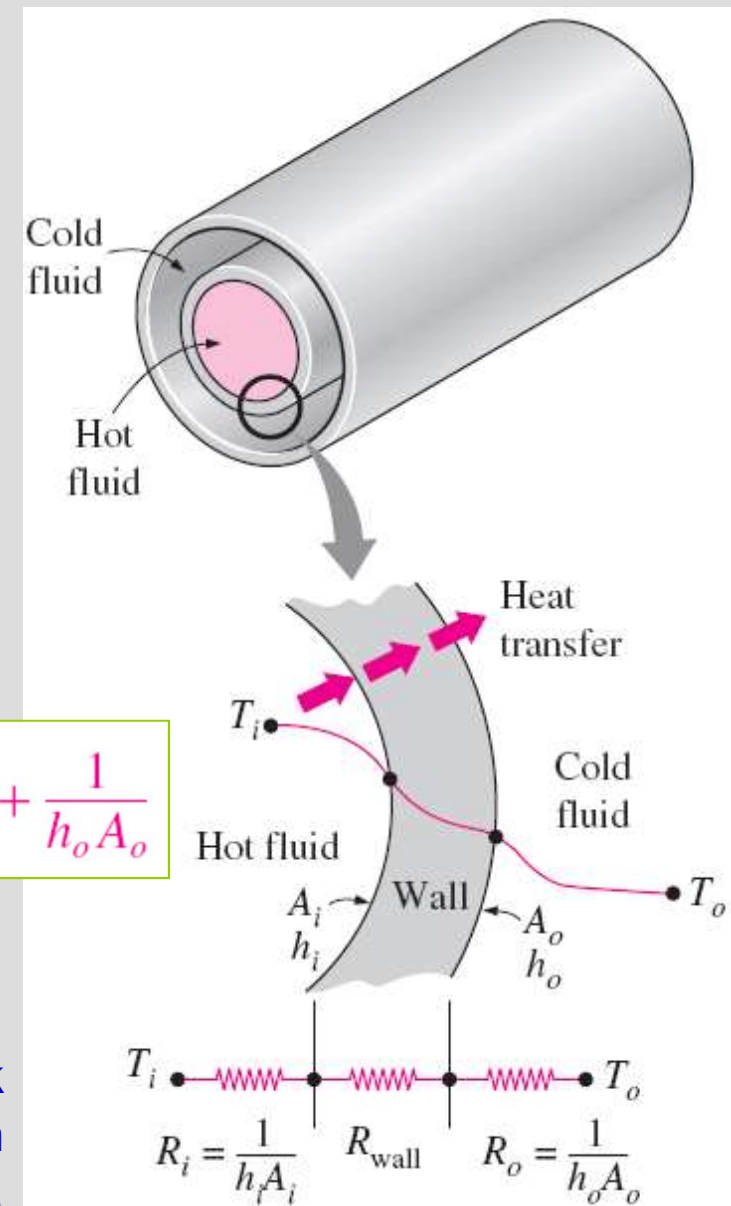
The two heat transfer surface areas associated with a double-pipe heat exchanger (for thin tubes, $D_i \approx D_o$ and thus $A_i \approx A_o$).

$$\dot{Q} = \frac{\Delta T}{R} = UA\Delta T = U_i A_i \Delta T = U_o A_o \Delta T$$

U the overall heat transfer coefficient, $W/m^2 \cdot °C$

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

$U_i A_i = U_o A_o$, but $U_i \neq U_o$ unless $A_i = A_o$

When $R_{wall} \approx 0$ $A_i \approx A_o \approx A_s$

$$\frac{1}{U} \approx \frac{1}{h_i} + \frac{1}{h_o} \qquad U \approx U_i \approx U_o$$

The overall heat transfer coefficient U is dominated by the *smaller* convection coefficient. When one of the convection coefficients is *much smaller* than the other (say, $h_i \ll h_o$), we have $1/h_i \gg 1/h_o$, and thus $U \approx h_i$. This situation arises frequently when one of the fluids is a gas and the other is a liquid. In such cases, fins are commonly used on the gas side to enhance the product UA and thus the heat transfer on that side.

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The overall heat transfer coefficient ranges from about 10 W/m²·°C for gas-to-gas heat exchangers to about 10,000 W/m²·°C for heat exchangers that involve phase changes.

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

$$A_s = A_{\text{total}} = A_{\text{fin}} + A_{\text{unfinned}}$$

For short fins of high thermal conductivity, we can use this total area in the convection resistance relation $R_{\text{conv}} = 1/hA_s$.

$$A_s = A_{\text{unfinned}} + \eta_{\text{fin}} A_{\text{fin}}$$

To account for fin efficiency

Representative values of the overall heat transfer coefficients in heat exchangers

Type of heat exchanger	U, W/m² · °C*
Water-to-water	850–1700
Water-to-oil	100–350
Water-to-gasoline or kerosene	300–1000
Feedwater heaters	1000–8500
Steam-to-light fuel oil	200–400
Steam-to-heavy fuel oil	50–200
Steam condenser	1000–6000
Freon condenser (water cooled)	300–1000
Ammonia condenser (water cooled)	800–1400
Alcohol condensers (water cooled)	250–700
Gas-to-gas	10–40
Water-to-air in finned tubes (water in tubes)	30–60[†]
	400–850[†]
Steam-to-air in finned tubes (steam in tubes)	30–300[†]
	400–4000[‡]

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. This is represented by a **fouling factor R_f**.

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

The fouling factor increases with the *operating temperature and* the length of service and decreases with the *velocity* of the fluids.
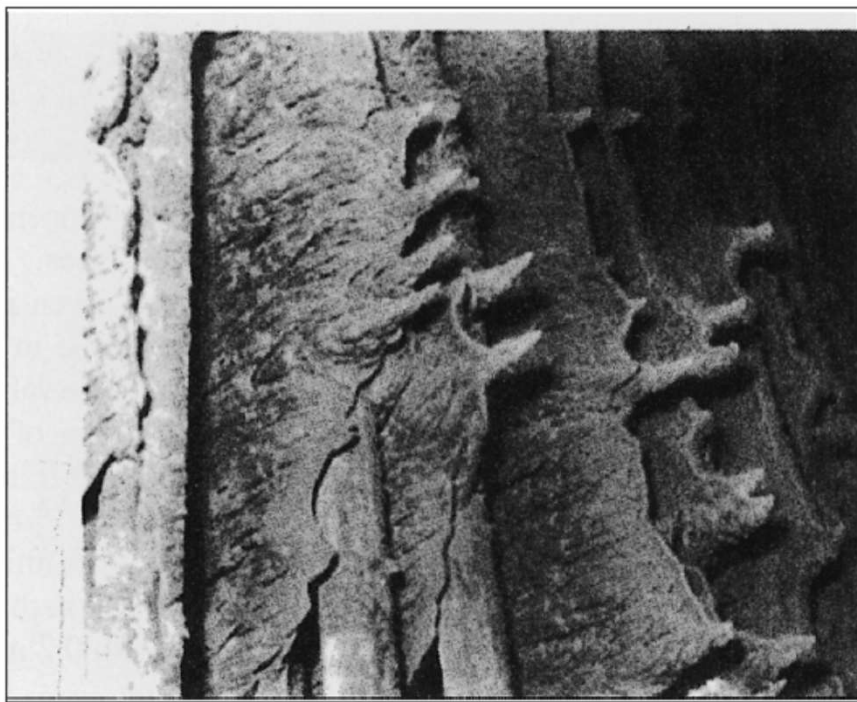
Precipitation fouling of ash particles on superheater tubes.

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

Fluid	R_f, m$^2 \cdot$ °C/W
Distilled water, sea-water, river water, boiler feedwater:	
Below 50°C	0.0001
Above 50°C	0.0002
Fuel oil	0.0009
Steam (oil-free)	0.0001
Refrigerants (liquid)	0.0002
Refrigerants (vapor)	0.0004
Alcohol vapors	0.0001
Air	0.0004

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Scaling/fouling

ANALYSIS OF HEAT EXCHANGERS

An engineer often finds himself or herself in a position

1. to *select a heat exchanger* that will achieve a *specified temperature change* in a fluid stream of known mass flow rate - the *log mean temperature difference* (or LMTD) method.

2. to *predict the outlet temperatures* of the hot and cold fluid streams in a *specified heat exchanger* - the *effectiveness–NTU* method.

The rate of heat transfer in heat exchanger (HE is insulated):

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

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

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

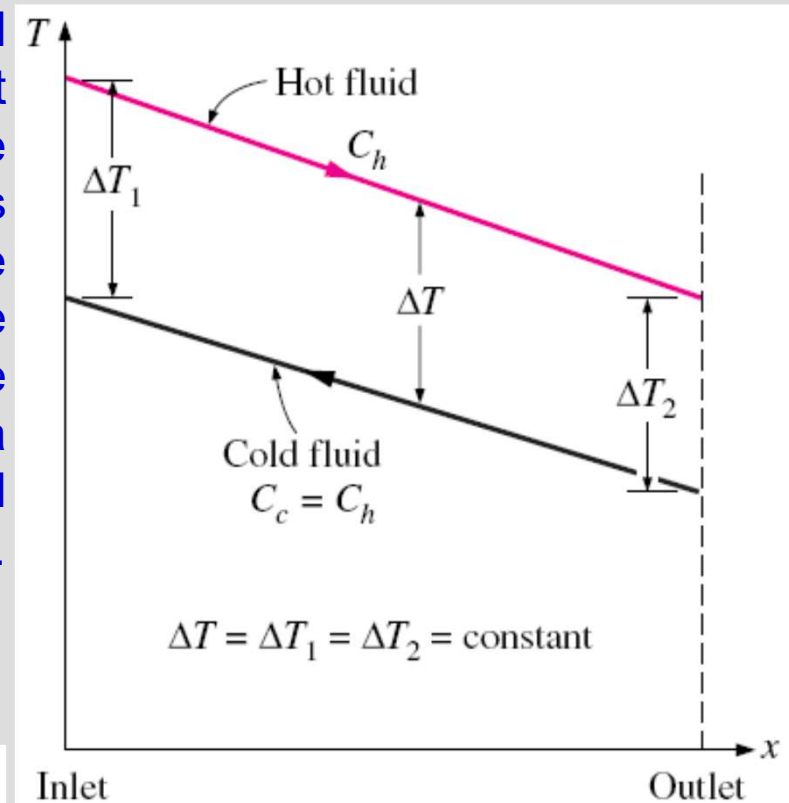c_{pc}, c_{ph} = specific heats

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

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

$$C_h = \dot{m}_h c_{ph} \quad \text{and} \quad C_c = \dot{m}_c c_{pc}$$

heat capacity rate

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

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

$\Delta T = \Delta T_1 = \Delta T_2 = \text{constant}$

m is the rate of evaporation or condensation of the fluid $\dot{Q} = \dot{m}h_{fg}$

h_{fg} is the enthalpy of vaporization of the fluid at the specified temperature or pressure.

The heat capacity rate of a fluid during a phase-change process must approach infinity since the temperature change is practically zero.

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

(b) Boiler ($C_c \to \infty$)
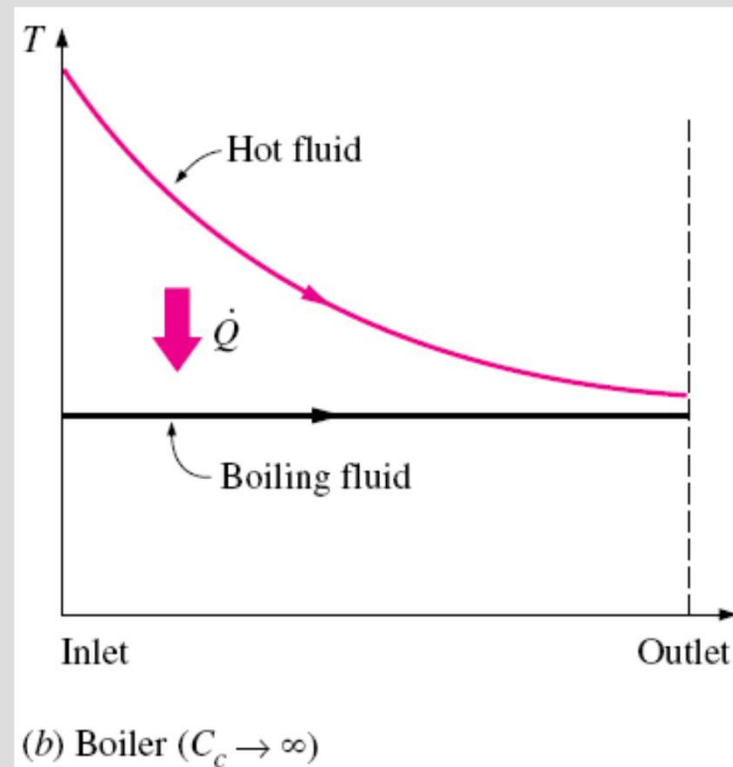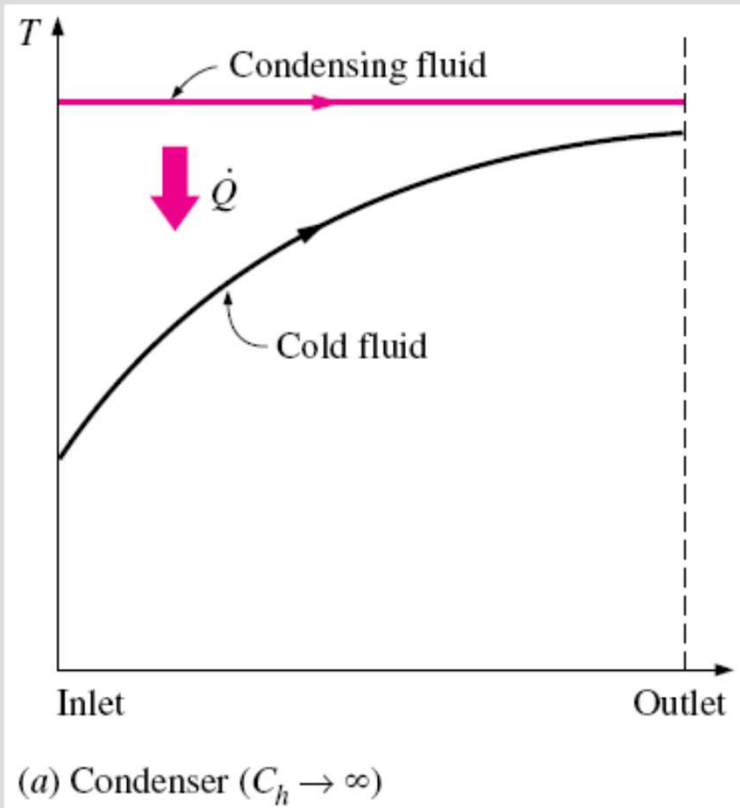
Variation of fluid temperatures in a heat exchanger when one of the fluids condenses or boils.

$$\dot{Q} = UA_s \, \Delta T_m$$

ΔT_m an appropriate average temperature difference between the two fluids (logarithmic)

THE LOG MEAN TEMPERATURE DIFFERENCE METHOD

$$\delta\dot{Q} = -\dot{m}_h c_{ph}\, dT_h \qquad \delta\dot{Q} = \dot{m}_c c_{pc}\, dT_c$$

$$dT_h = -\frac{\delta\dot{Q}}{\dot{m}_h c_{ph}} \qquad dT_c = \frac{\delta\dot{Q}}{\dot{m}_c c_{pc}}$$

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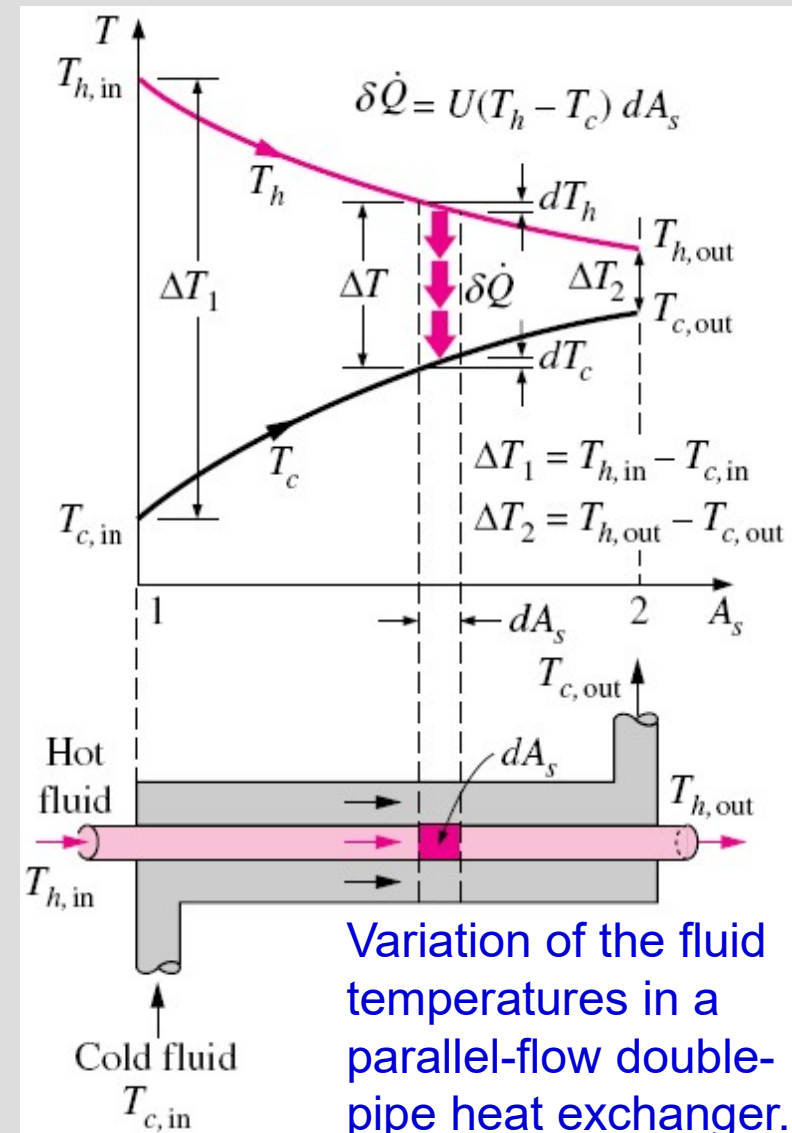
$$\delta\dot{Q} = U(T_h - T_c)\, dA_s$$

$$\frac{d(T_h - T_c)}{T_h - T_c} = -U\, dA_s\left(\frac{1}{\dot{m}_h c_{ph}} + \frac{1}{\dot{m}_c c_{pc}}\right)$$
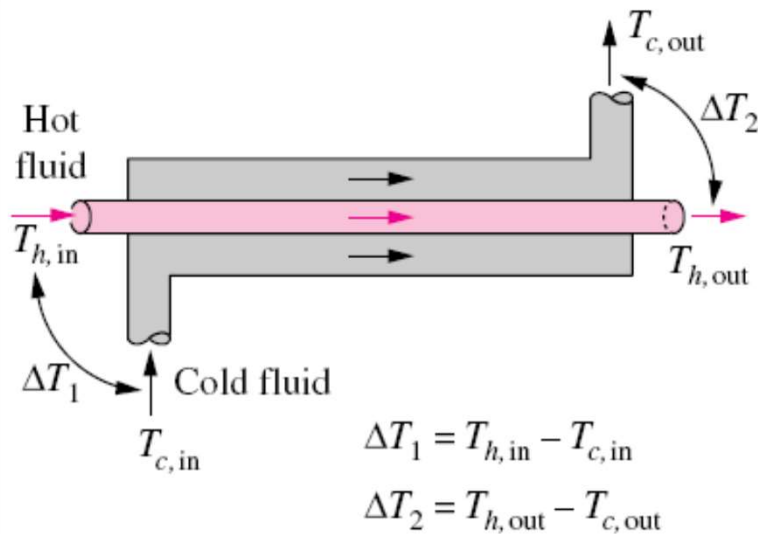
$$\ln\frac{T_{h,\,out} - T_{c,\,out}}{T_{h,\,in} - T_{c,\,in}} = -UA_s\left(\frac{1}{\dot{m}_h c_{ph}} + \frac{1}{\dot{m}_c c_{pc}}\right)$$
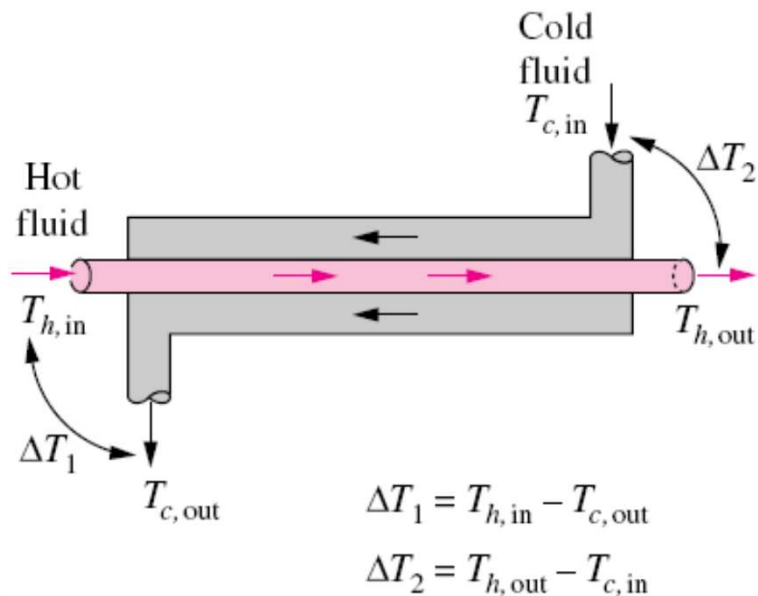
$$\dot{Q} = UA_s\, \Delta T_{lm}$$

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

log mean temperature difference

$$\delta\dot{Q} = U(T_h - T_c)\, dA_s$$

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

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

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

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

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

(a) Parallel-flow heat exchangers

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

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

(b) Counter-flow heat exchangers

The arithmetic mean temperature difference

$$\Delta T_{am} = \tfrac{1}{2}(\Delta T_1 + \Delta T_2)$$

The logarithmic mean temperature difference ΔT_{lm} is an *exact* representation of the *average temperature difference* between the hot and cold fluids.

Note that ΔT_{lm} is always less than ΔT_{am}. Therefore, using ΔT_{am} in calculations instead of ΔT_{lm} will overestimate the rate of heat transfer in a heat exchanger between the two fluids.

When ΔT_1 differs from ΔT_2 by no more than 40 percent, the error in using the arithmetic mean temperature difference is less than 1 percent. But the error increases to undesirable levels when ΔT_1 differs from ΔT_2 by greater amounts.

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

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Counter-Flow Heat Exchangers

In the limiting case, the cold fluid will be heated to the inlet temperature of the hot fluid. However, the outlet temperature of the cold fluid can *never* exceed the inlet temperature of the hot fluid.
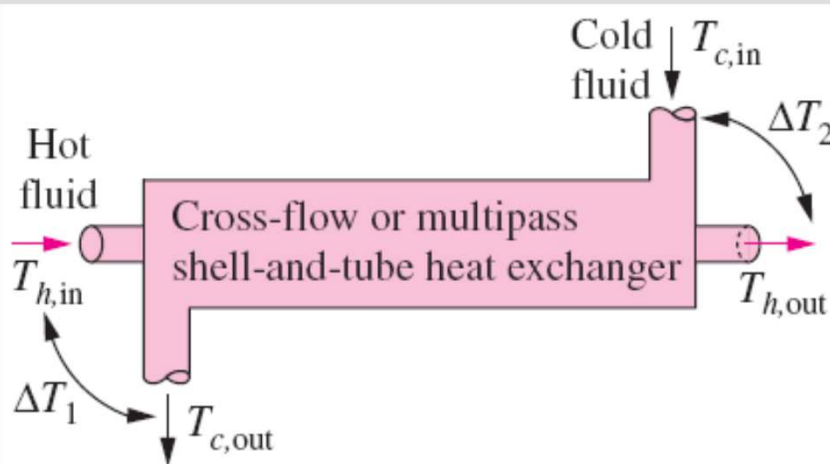
For specified inlet and outlet temperatures, ΔT_{lm} a counter-flow heat exchanger is always greater than that for a parallel-flow heat exchanger.

That is, $\Delta T_{lm, CF} > \Delta T_{lm, PF}$, and thus a smaller surface area (and thus a smaller heat exchanger) is needed to achieve a specified heat transfer rate in a counter-flow heat exchanger.
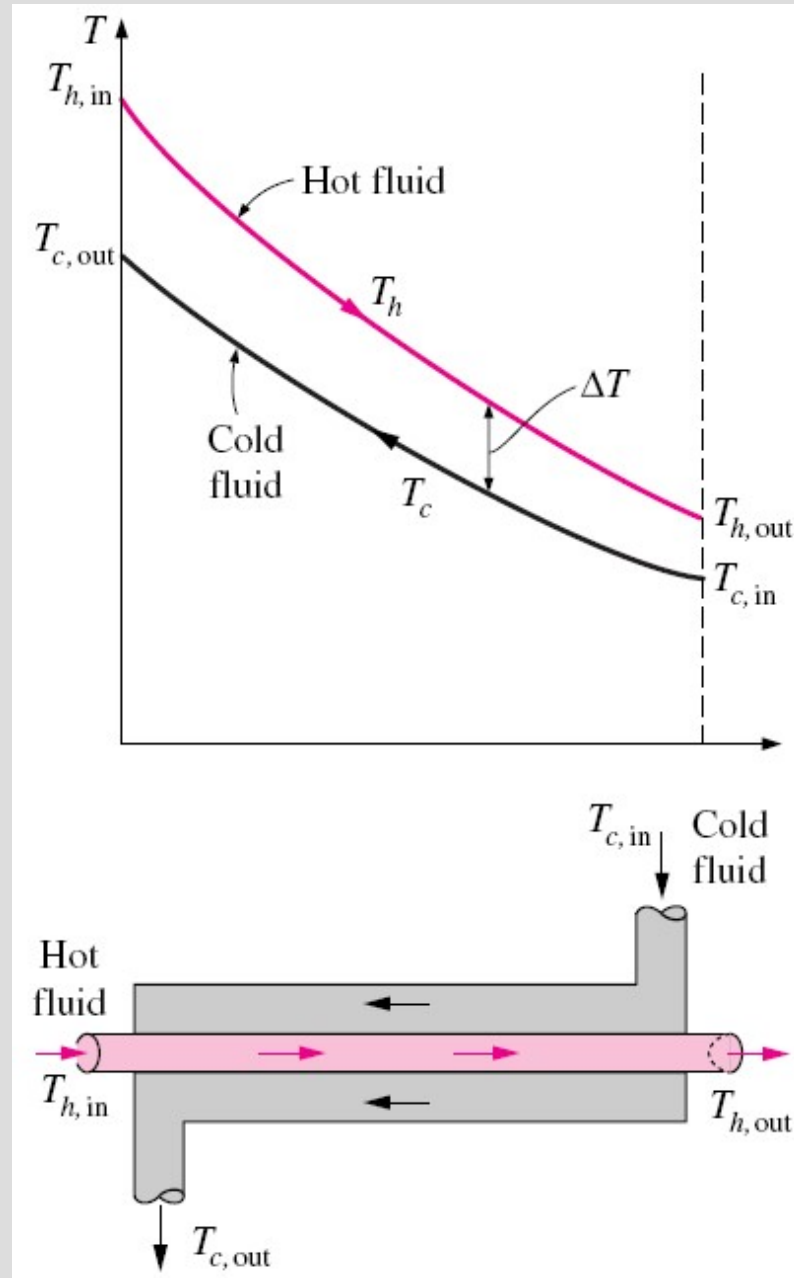
The variation of the fluid temperatures in a counter-flow double-pipe heat exchanger.

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

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

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

F for common cross-flow and shell-and-tube heat exchanger configurations is given in the figure versus two temperature ratios *P* and *R* defined as

Hot fluid $T_{h,\text{in}}$ ... Cross-flow or multipass shell-and-tube heat exchanger ... $T_{h,\text{out}}$

Cold fluid $T_{c,\text{in}}$ ... ΔT_2

ΔT_1 ... $T_{c,\text{out}}$

$$P = \frac{t_2 - t_1}{T_1 - t_1} \qquad R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{(\dot{m}c_p)_{\text{tube side}}}{(\dot{m}c_p)_{\text{shell side}}}$$
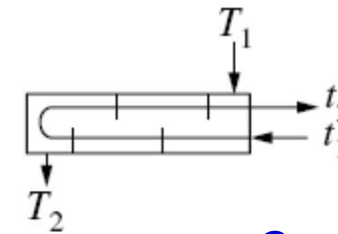
1 and 2 *inlet* and *outlet*
T and t *shell-* and *tube-side* temperatures

F = 1 for a condenser or boiler

Heat transfer rate:

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

where

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

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

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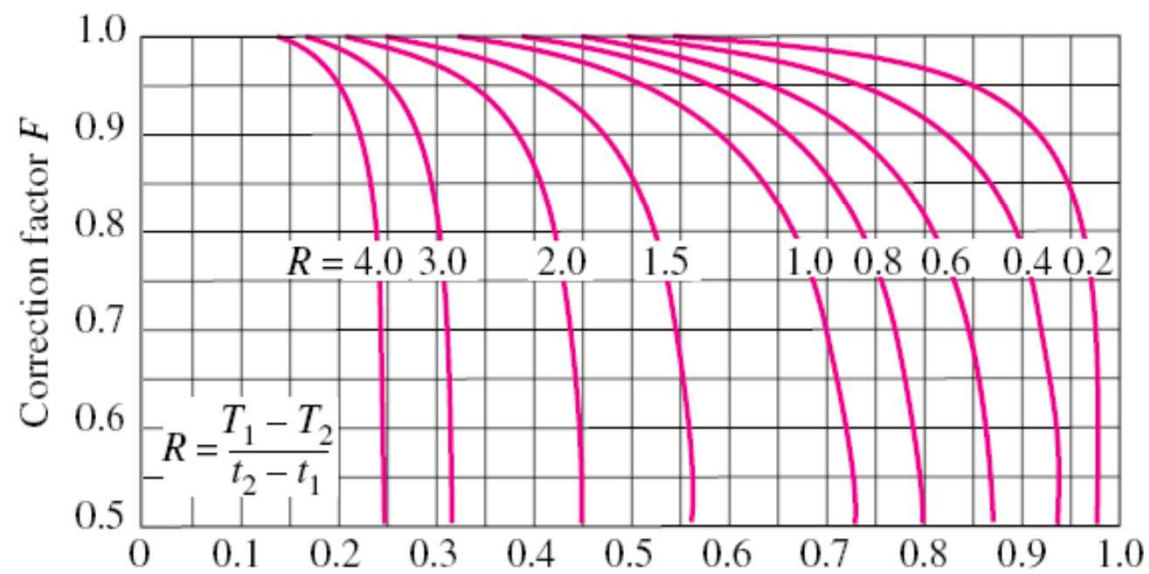
and

$$F = \ldots \text{ (Fig. 22–18)}$$

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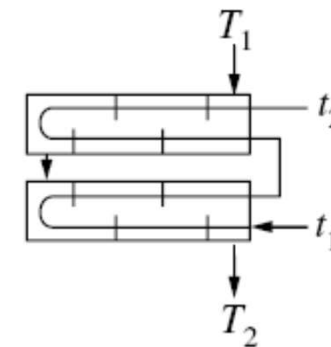
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Correction factor F charts for common shell-and-tube heat exchangers.

$R = \dfrac{T_1 - T_2}{t_2 - t_1}$

$P = \dfrac{t_2 - t_1}{T_1 - t_1}$
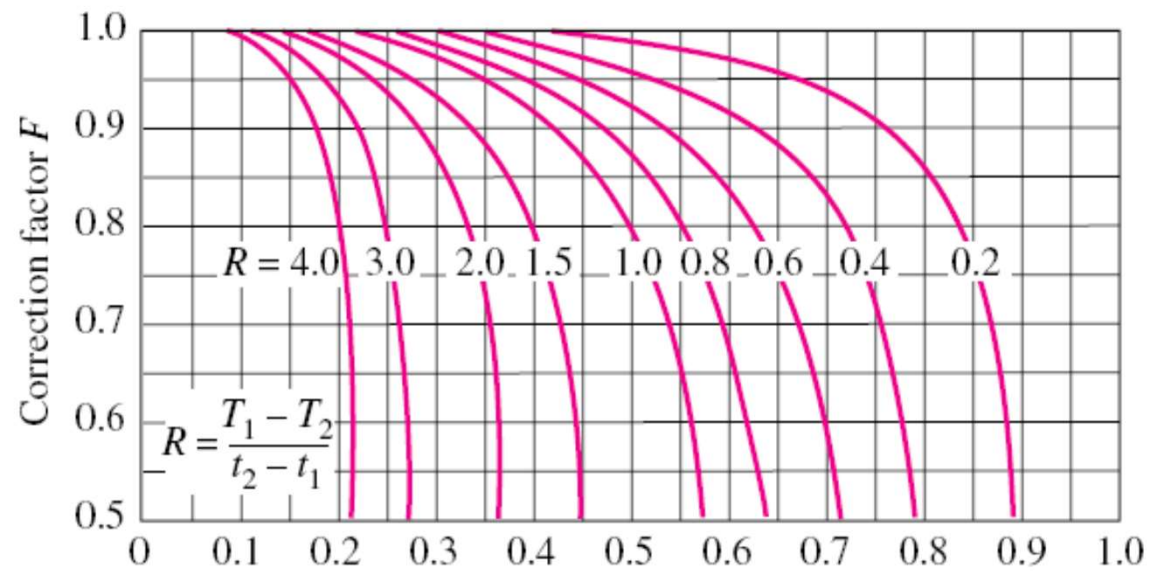
(a) One-shell pass and 2, 4, 6, etc. (any multiple of 2), tube passes

$R = 4.0$ 3.0 2.0 1.5 1.0 0.8 0.6 0.4 0.2

$R = \dfrac{T_1 - T_2}{t_2 - t_1}$

$P = \dfrac{t_2 - t_1}{T_1 - t_1}$

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

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$$R = \frac{T_1 - T_2}{t_2 - t_1}$$
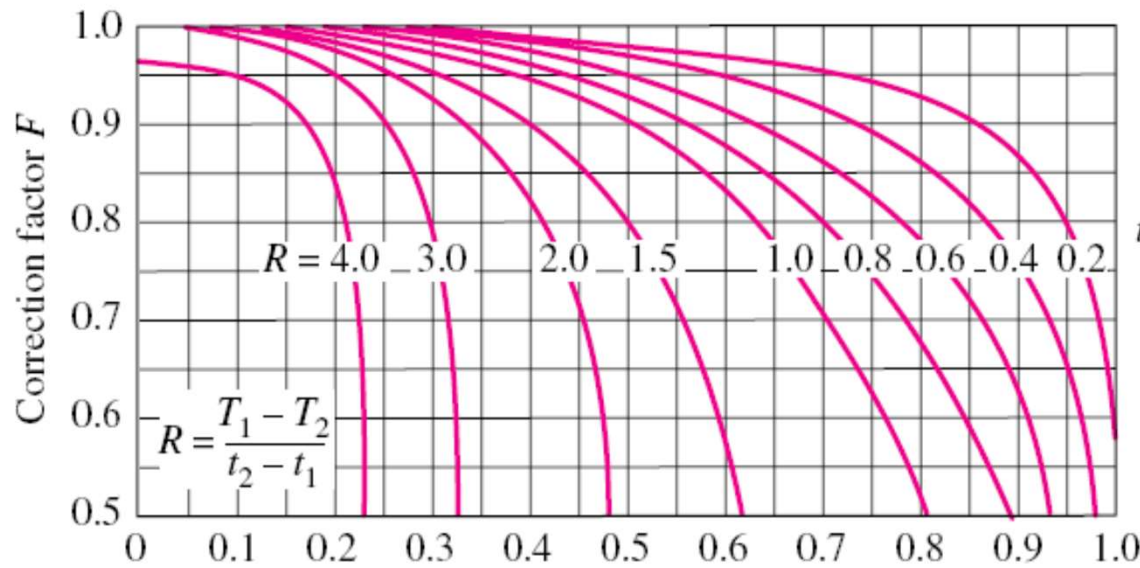
$$P = \frac{t_2 - t_1}{T_1 - t_1}$$

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

$$R = \frac{T_1 - T_2}{t_2 - t_1}$$

$$P = \frac{t_2 - t_1}{T_1 - t_1}$$

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

Correction factor F charts for common cross-flow heat exchangers.
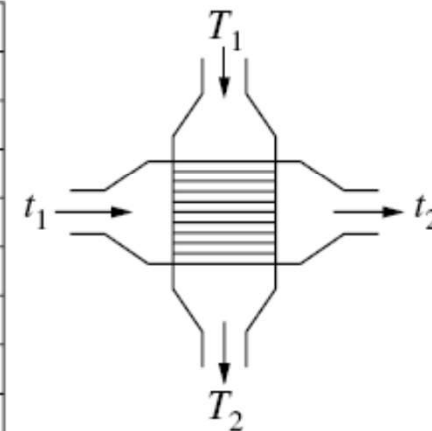
The LMTD method is very suitable for determining the *size* of a heat exchanger to realize prescribed outlet temperatures when the mass flow rates and the inlet and outlet temperatures of the hot and cold fluids are specified.

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

1. Select the type of heat exchanger suitable for the application.

2. Determine any unknown inlet or outlet temperature and the heat transfer rate using an energy balance.

3. Calculate the log mean temperature difference ΔT_{lm} and the correction factor F, if necessary.

4. Obtain (select or calculate) the value of the overall heat transfer coefficient U.

5. Calculate the heat transfer surface area A_s .

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

11–44 A stream of hydrocarbon ($c_p = 2.2$ kJ/kg · K) is cooled at a rate of 720 kg/h from 150°C to 40°C in the tube side of a double-pipe counter-flow heat exchanger. Water ($c_p = 4.18$ kJ/kg · K) enters the heat exchanger at 10°C at a rate of 540 kg/h. The outside diameter of the inner tube is 2.5 cm, and its length is 6.0 m. Calculate the overall heat transfer coefficient.

Analysis The rate of heat transfer is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{HC} = (720/3600 \text{ kg/s})(2.2 \text{ kJ/kg.°C})(150°C - 40°C) = 48.4 \text{ kW}$$

The outlet temperature of water is

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_w$$

$$48.4 \text{ kW} = (540/3600 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(T_{w,out} - 10°C)$$

$$T_{w,out} = 87.2 °C$$

The logarithmic mean temperature difference is

$$\Delta T_1 = T_{h,in} - T_{c,out} = 150°C - 87.2°C = 62.8°C$$

$$\Delta T_2 = T_{h,out} - T_{c,in} = 40°C - 10°C = 30°C$$

and $\quad \Delta T_{lm} = \dfrac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \dfrac{62.8 - 30}{\ln(62.8/30)} = 44.4°C$

The overall heat transfer coefficient is determined from

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

$$48.4 \text{ kW} = U(\pi \times 0.025 \times 6.0)(44.4°C)$$

$$U = \mathbf{2.31 \, kW/m^2 \cdot K}$$