HEAT EXCHANGERS

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Contents

Objectives

- Perform an energy balance over the heat exchanger
- Identify key variables affecting heat transfer and quantify the nature of any significant effects
- Develop appropriate models to describe heat transfer characteristics

Introduction to Heat Exchangers

What Are Heat Exchangers?

Heat exchangers are units designed to transfer heat from a hot flowing stream to a cold flowing stream.

Why Use Heat Exchangers?

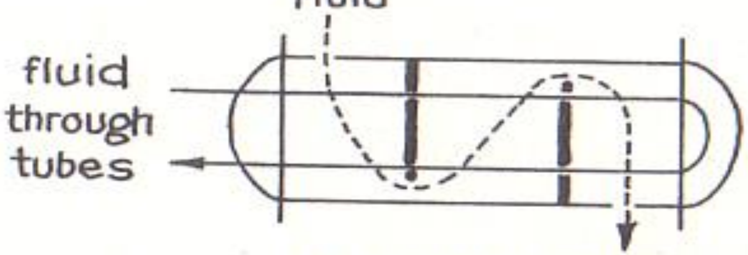
Heat exchangers and heat recovery is often used to improve process efficiency.

Types of Heat Exchangers

There are three broad categories:

- The **recuperator**, or through-the-wall non storing exchanger
- The **direct contact** non storing exchanger
- The **regenerator**, accumulator, or heat storage exchanger

Recuperators



SHELL AND TUBE EXCHANGER

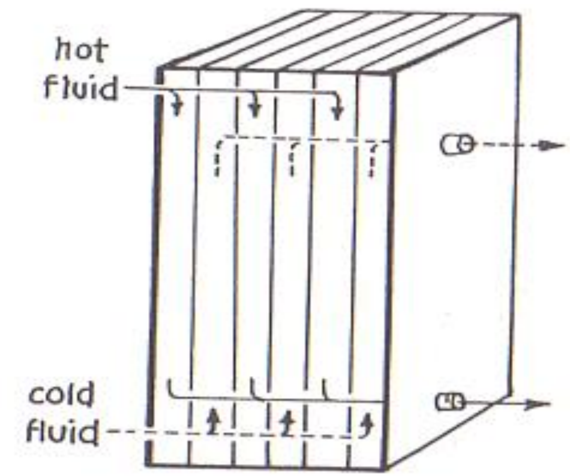
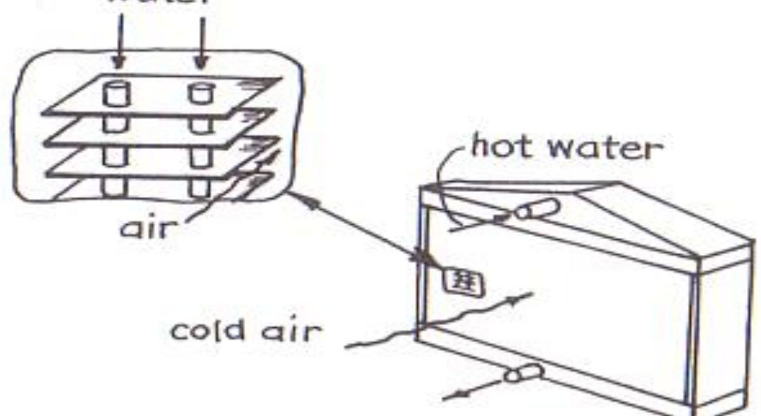
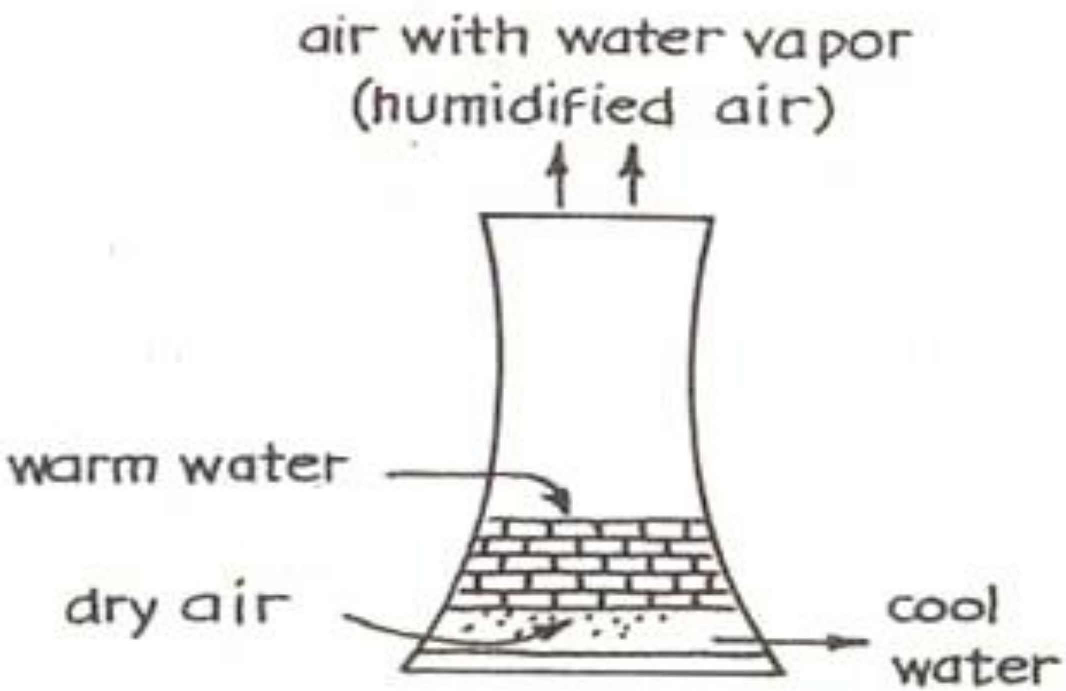


PLATE EXCHANGER



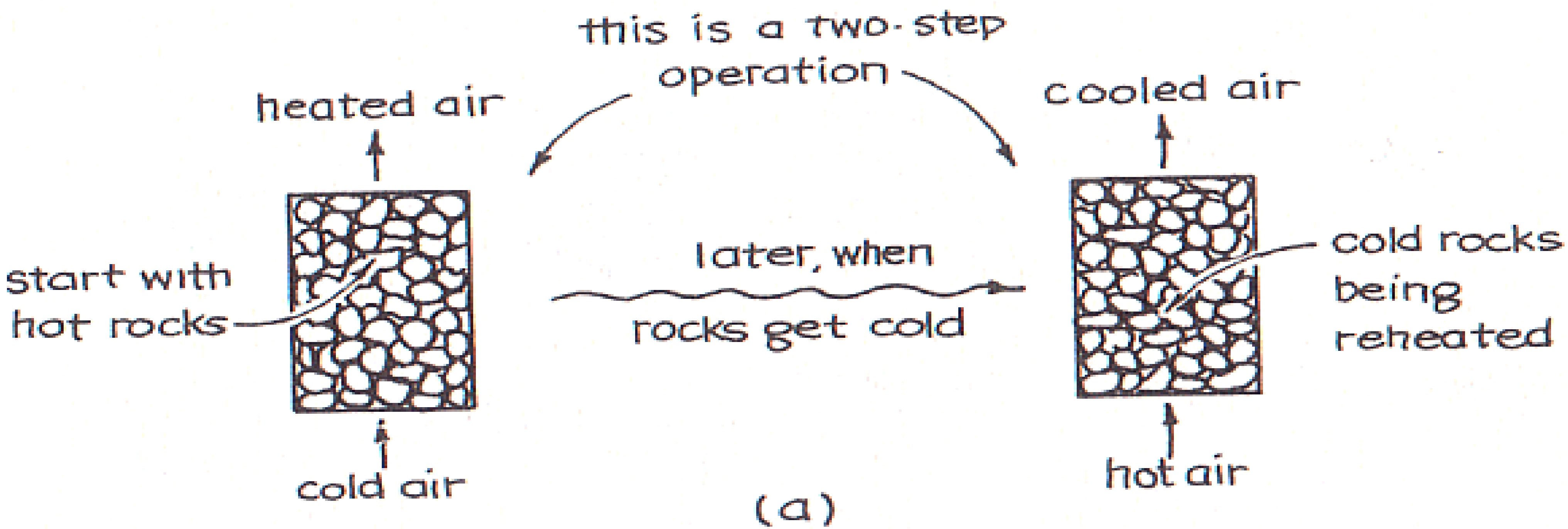
AUTOMOBILE RADIATOR

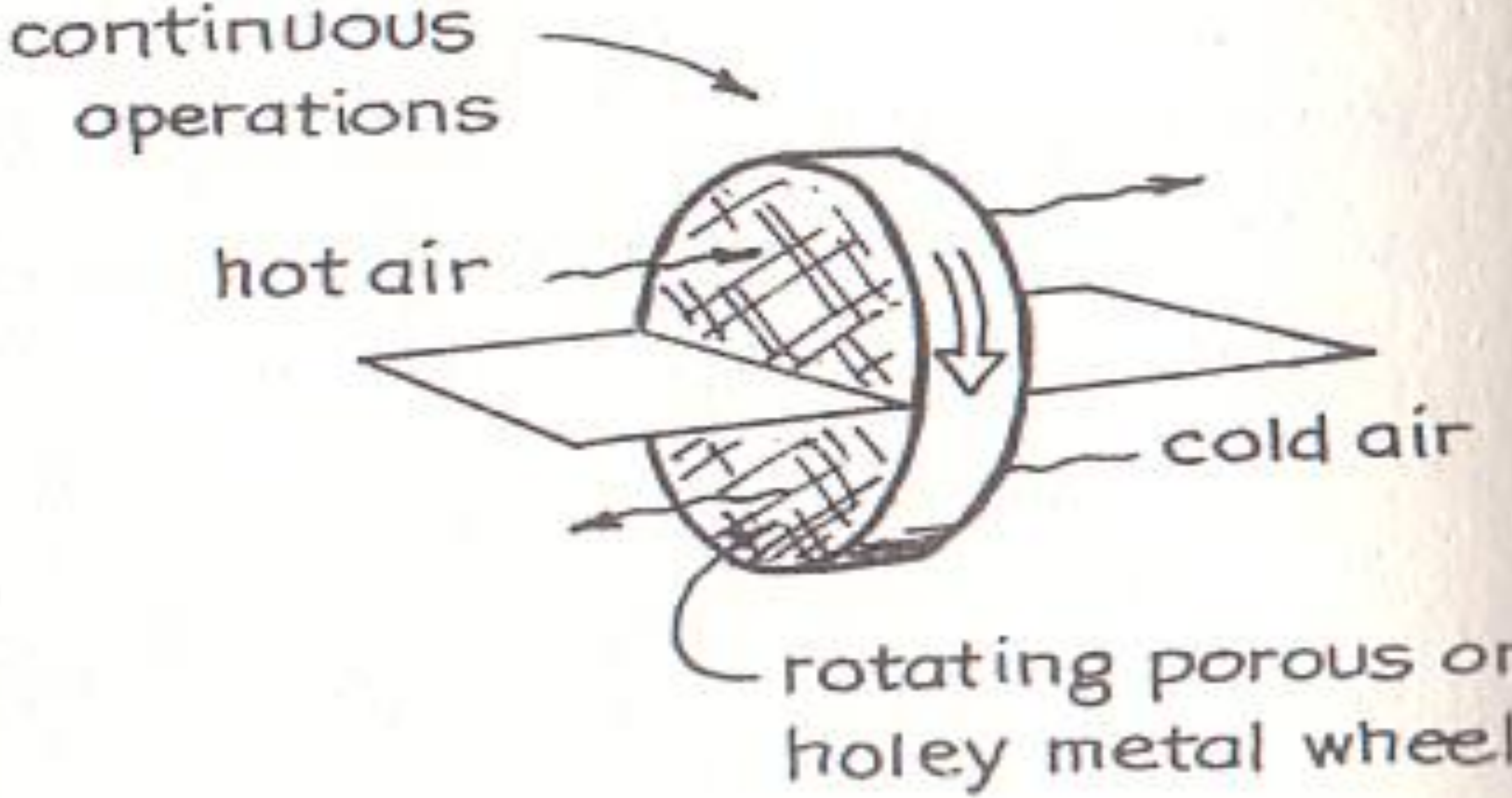
Direct Contact



COOLING TOWER
(to cool warm water
without refrigeration)

Regenerators



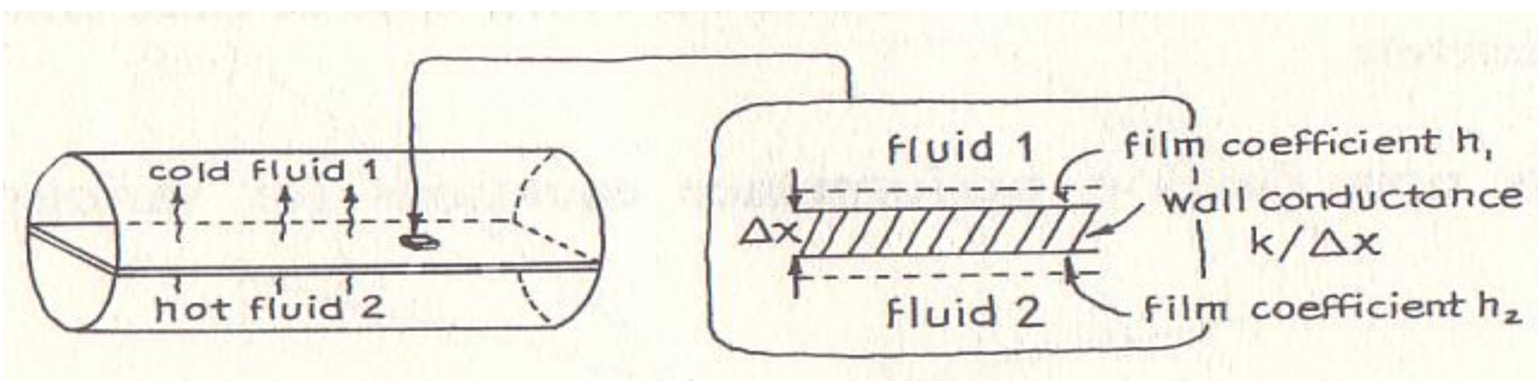

continuous operations
hot air
cold air
rotating porous or holey metal wheel

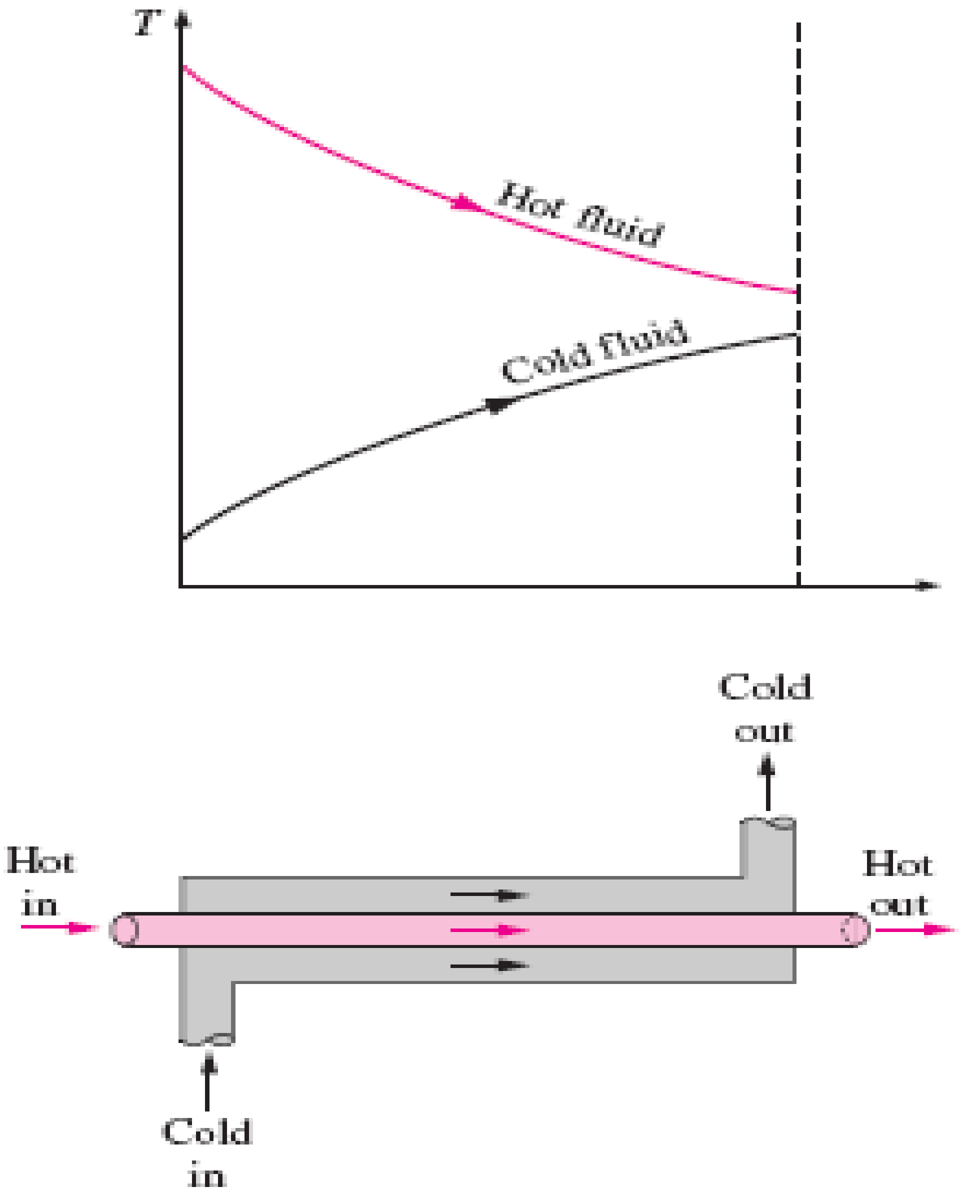
(b)

Heat Transfer Within a Heat Exchanger

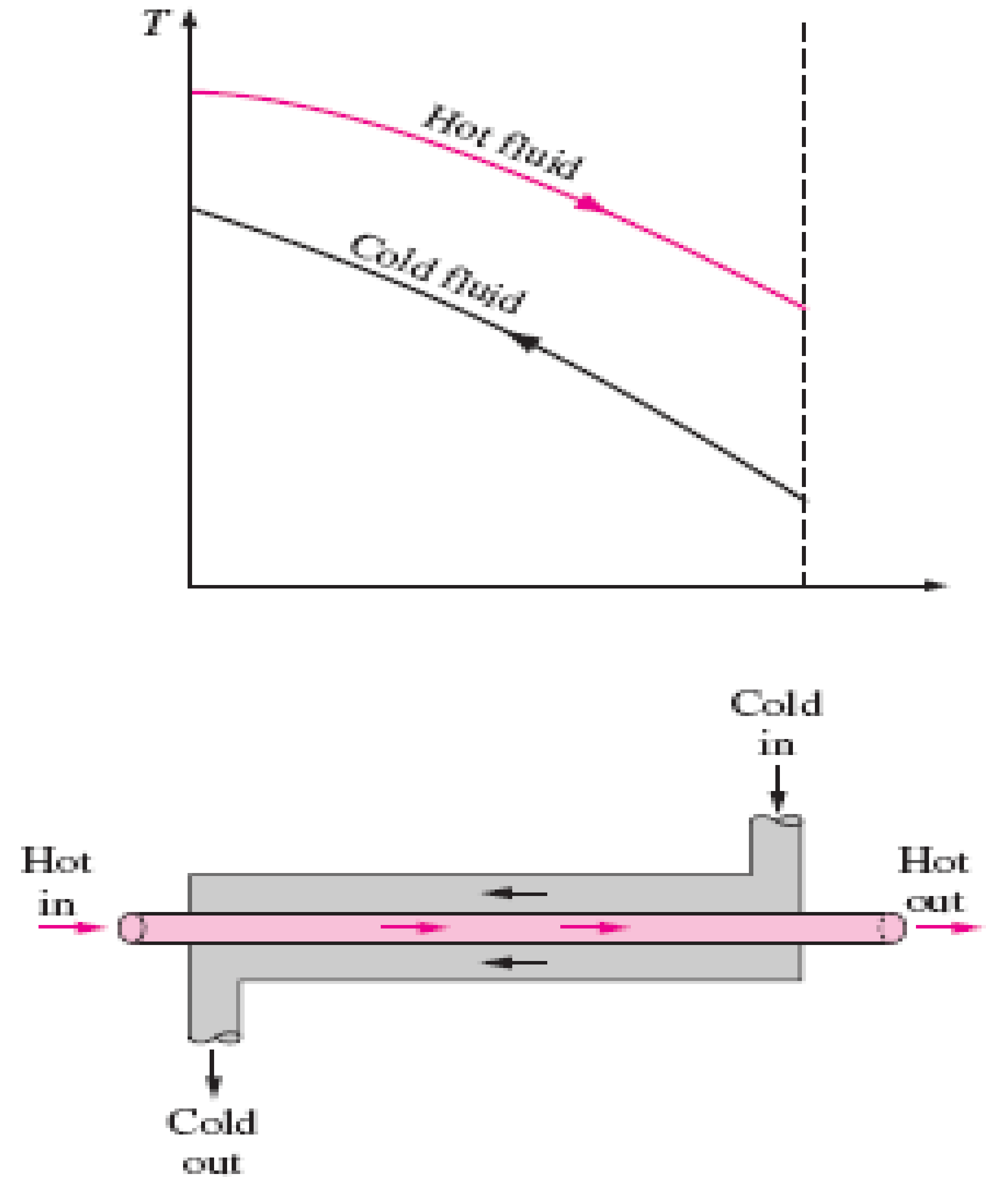
- **Heat transfer within a heat exchanger typically involves a combination of conduction and convection**
- **The overall heat transfer coefficient U accounts for the overall resistance to heat transfer from convection and conduction**

$$\frac{1}{UA} = \frac{1}{h_1 A_1} + \frac{1}{h_2 A_2} + \frac{\Delta x}{k A_{mean}}$$

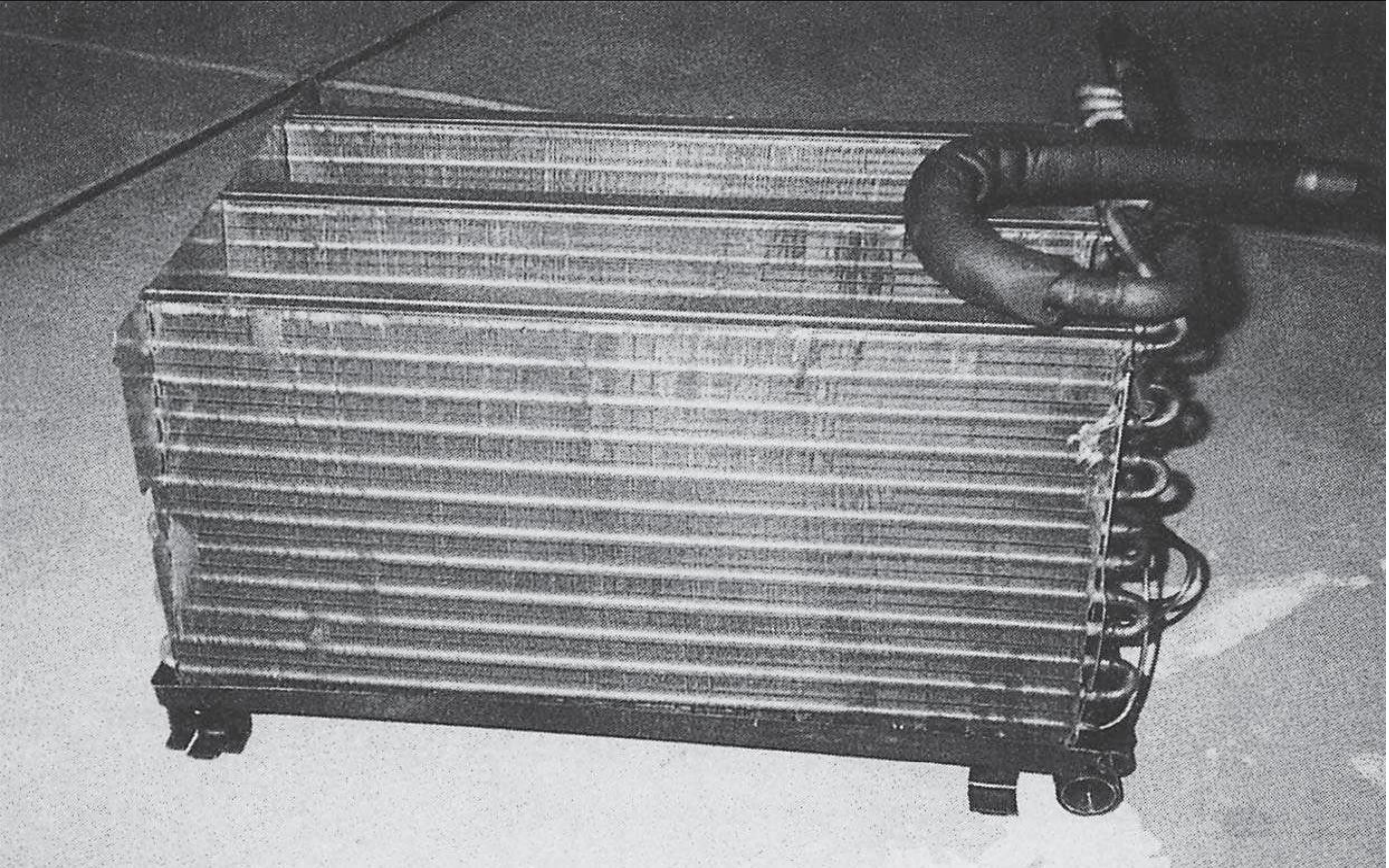


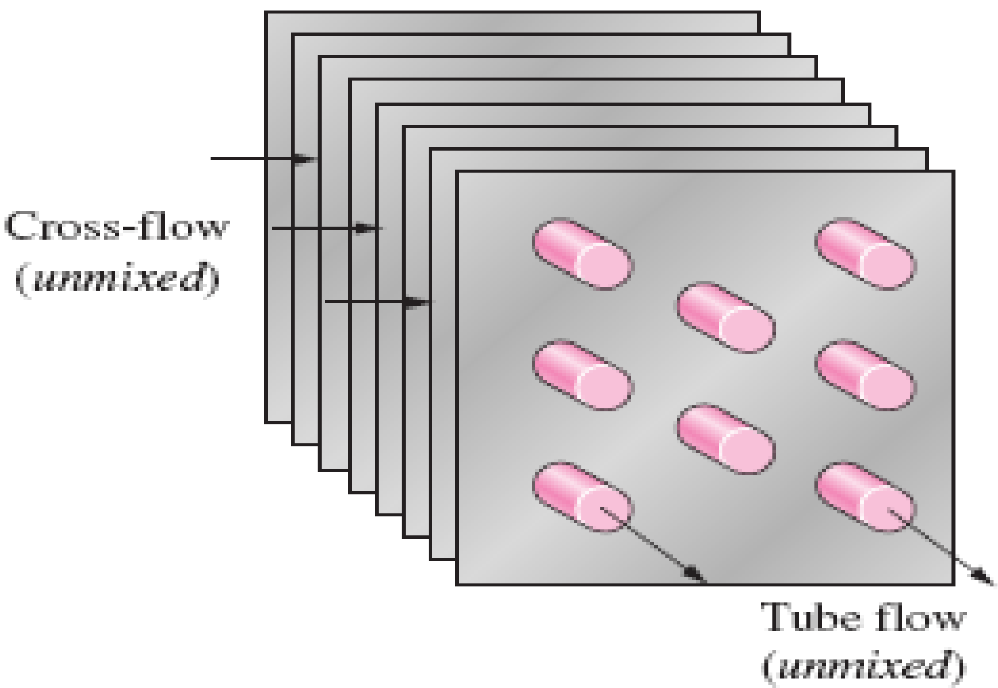


(a) Parallel flow

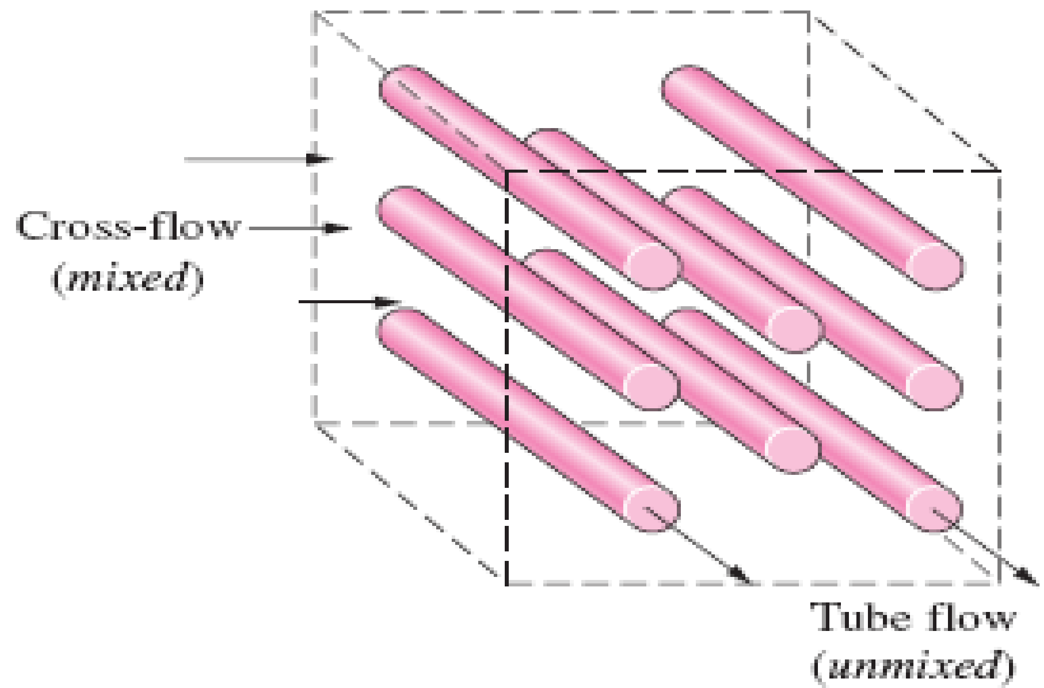


(b) Counter flow

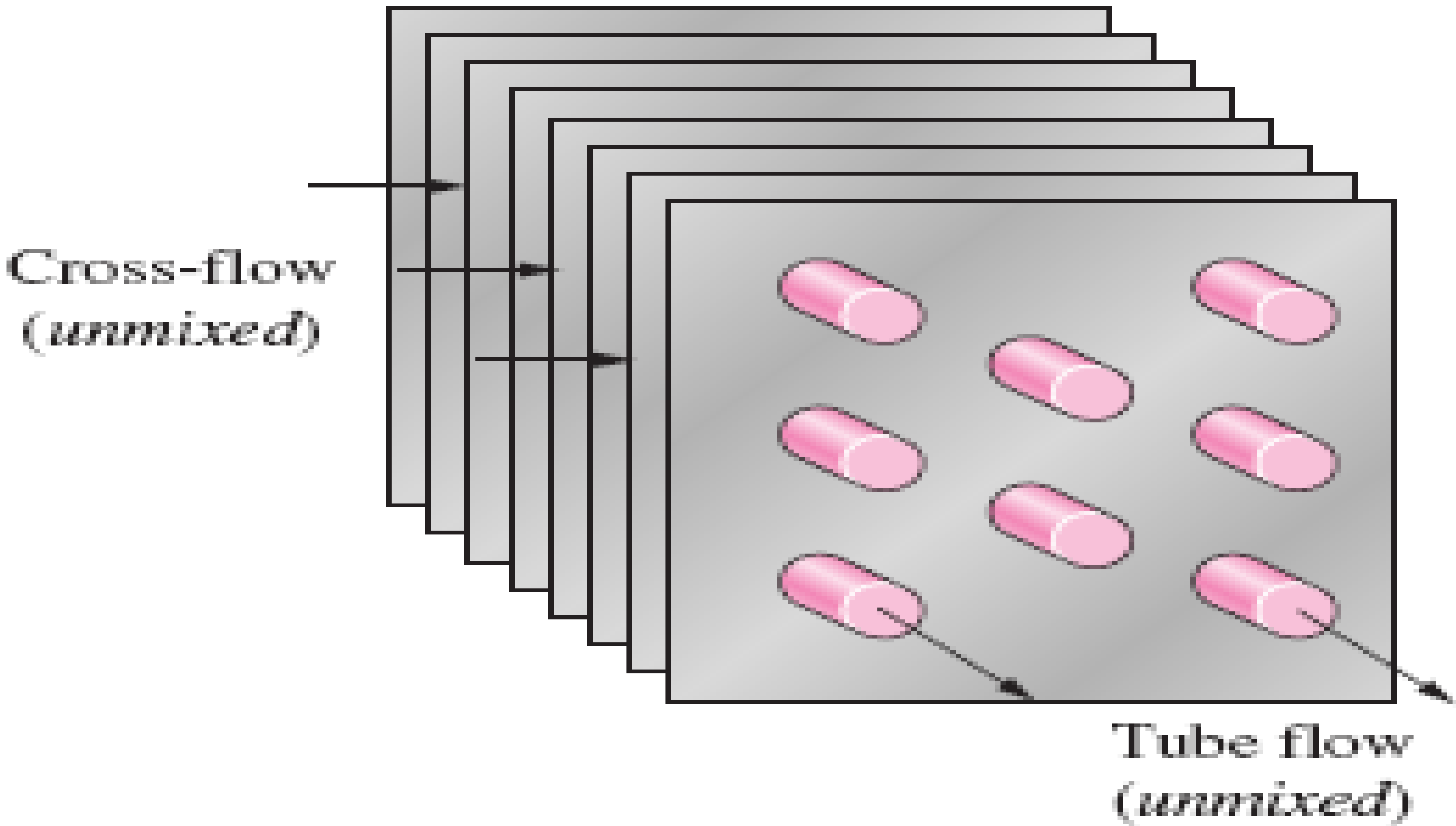




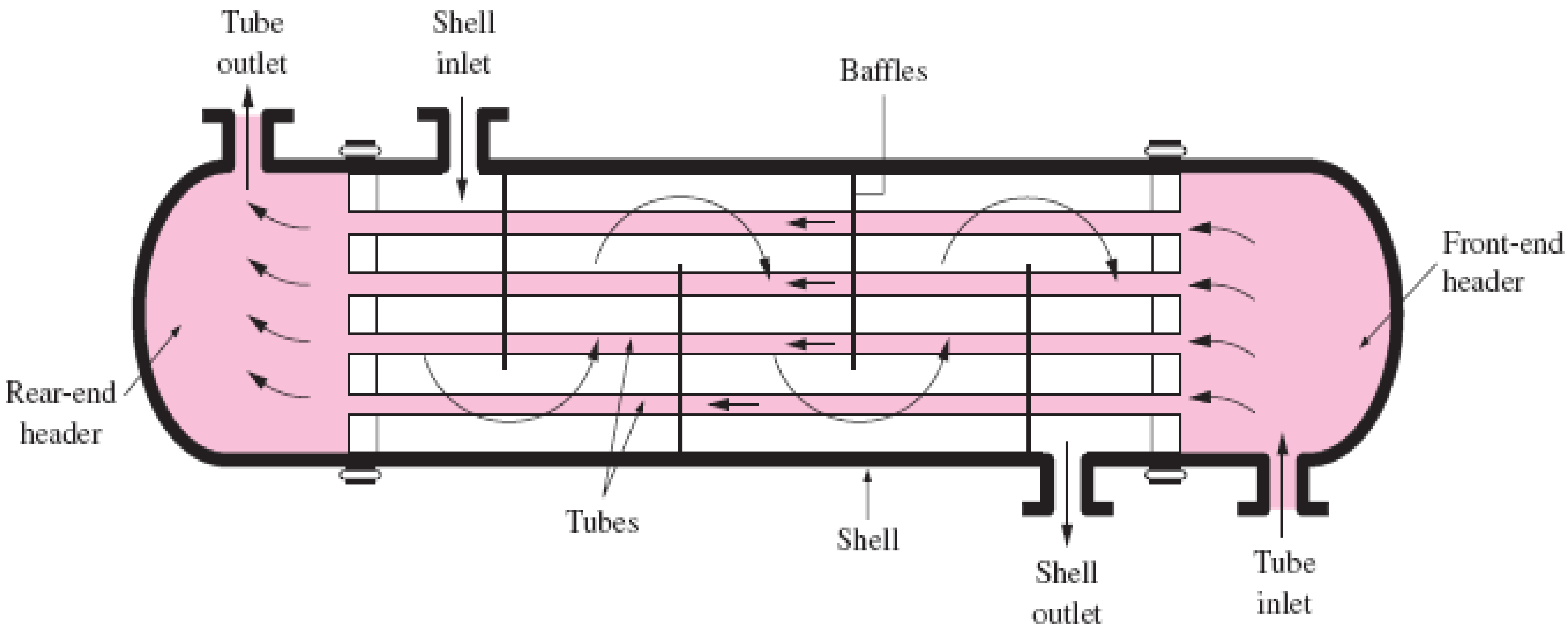
(*a*) Both fluids unmixed



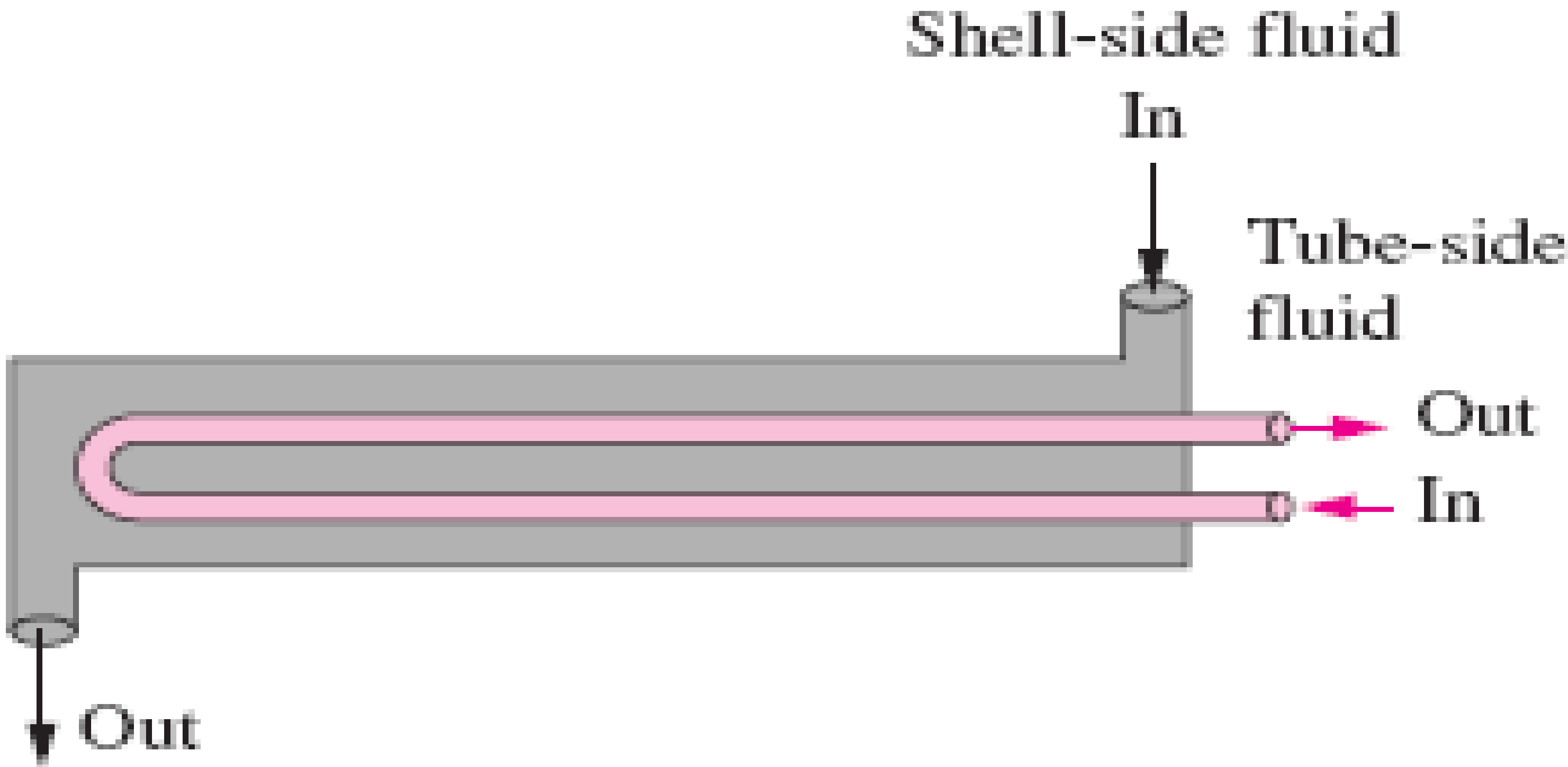
(*b*) One fluid mixed, one fluid unmixed



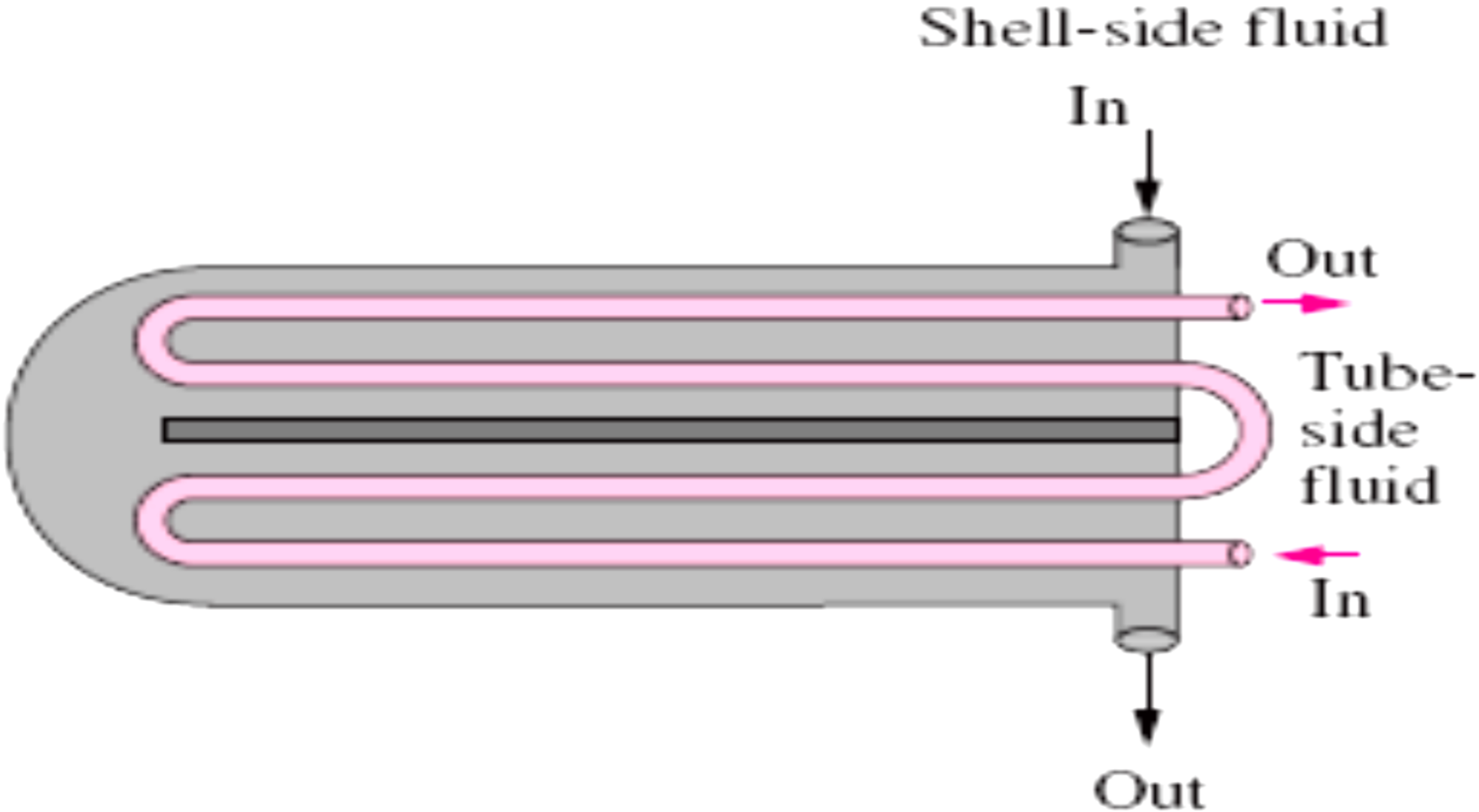
(*a*) Both fluids unmixed



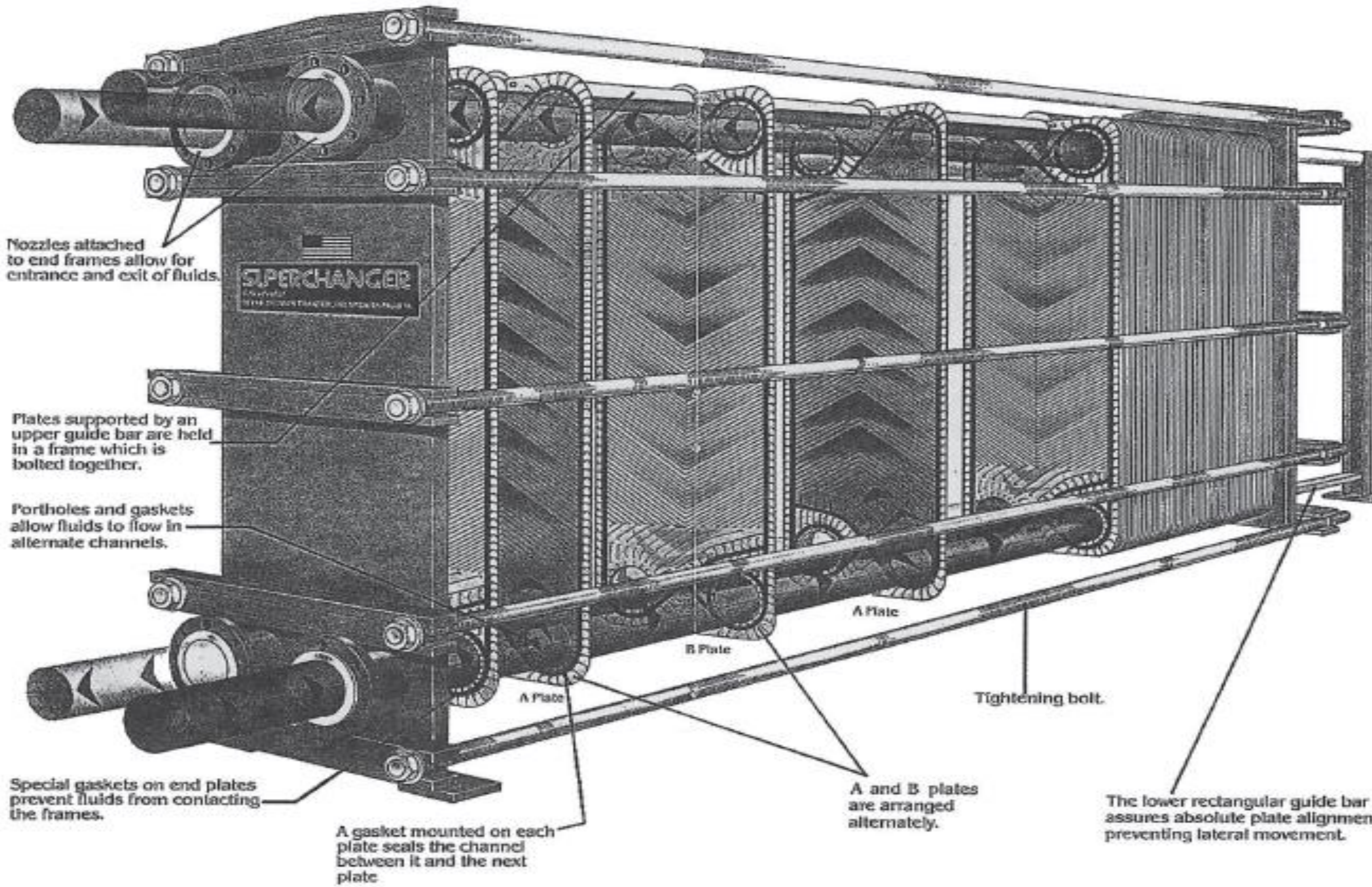
The schematic of a shell-and-tube heat exchanger (one-shell pass and one-tube pass).



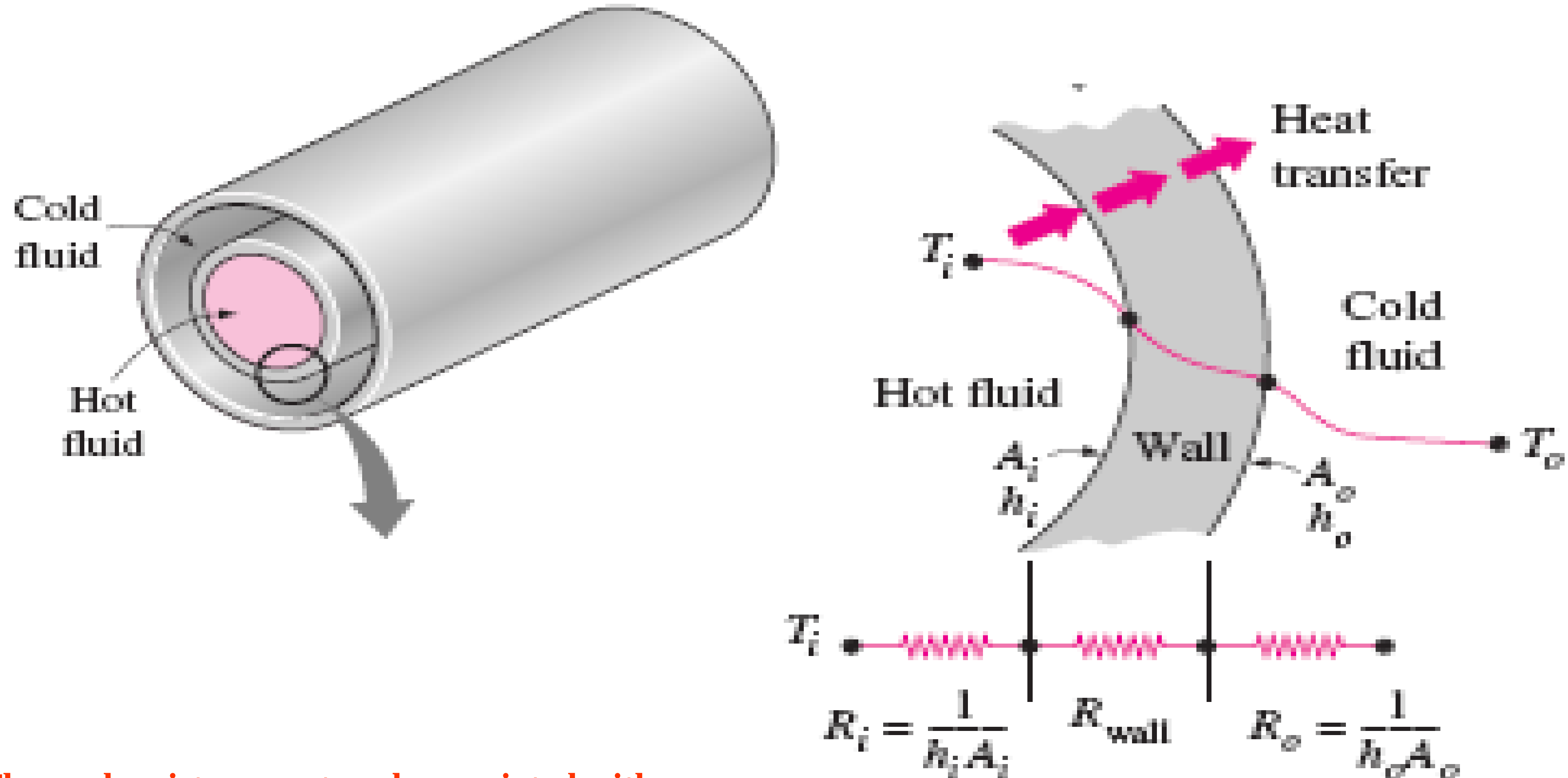
(a) One-shell pass and two-tube passes



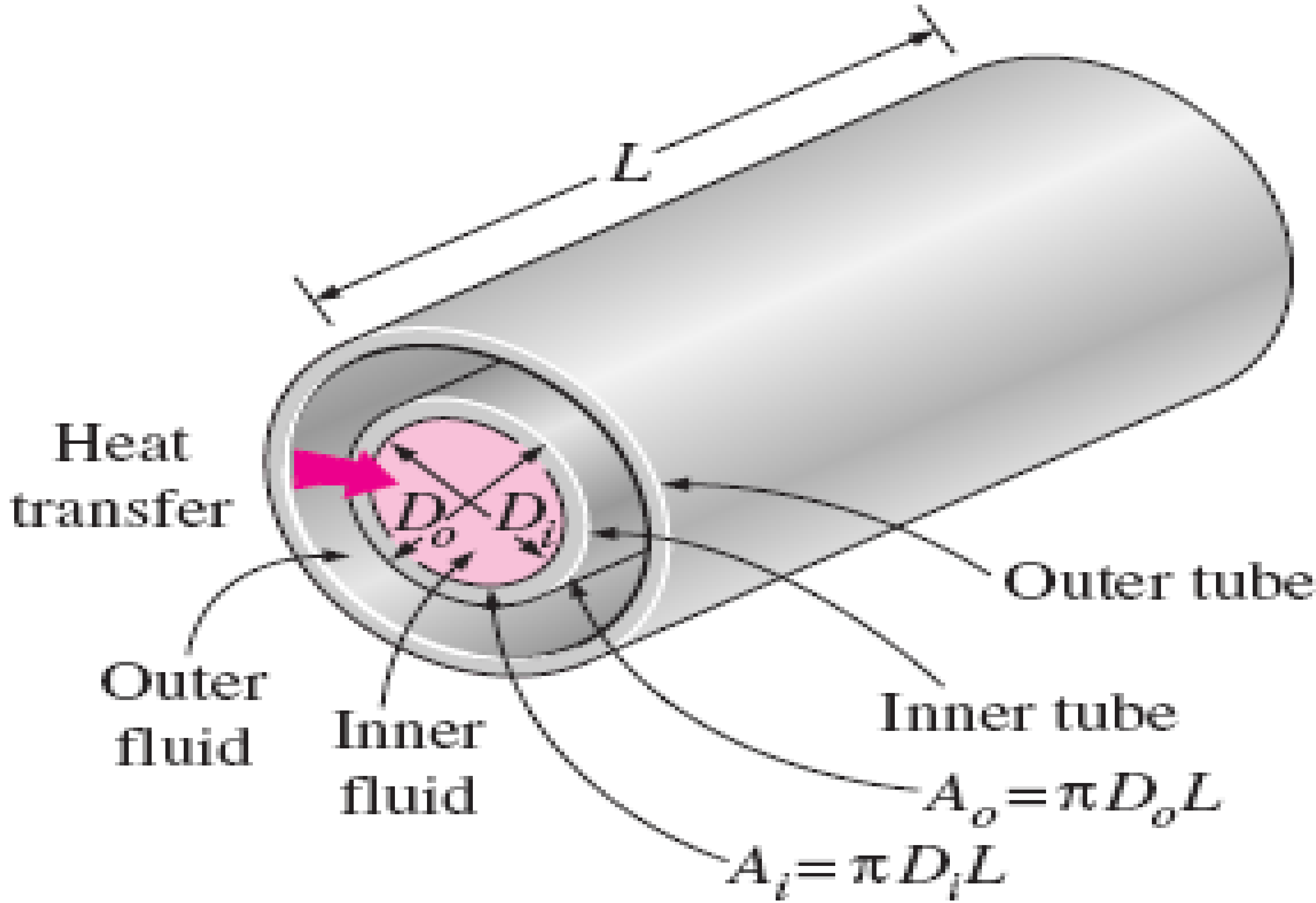
(*b*) Two-shell passes and four-tube passes



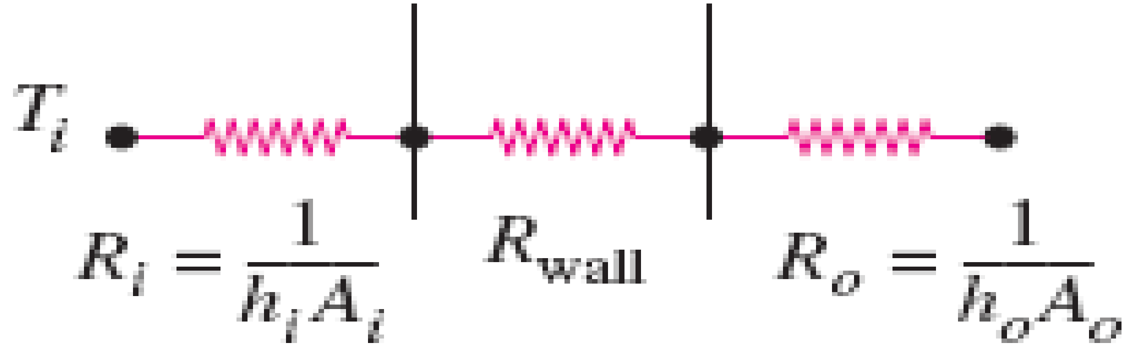
A plate-and-frame liquid-to-liquid heat exchanger



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



The two heat transfer surface areas associated with a double-pipe heat exchanger (for thin tubes, *Di* ≈ *Do* and thus *Ai* ≈ *Ao*).



The thermal resistance of the tube wall :

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

The total thermal resistance :

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

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

The rate of heat transfer between the two fluids as :

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

U is the overall heat transfer coefficient [W/m^2.^{o}C]

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

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 ($\boldsymbol{R_{\text{wall}} \approx 0}$) and the inner and outer surfaces of the tube are almost identical ($\boldsymbol{A_i \approx A_o \approx A_s}$).

Equation for the overall heat transfer coefficient simplifies to:

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

The overall heat transfer coefficient *U* dominated by the **smaller convection coefficient**, since the inverse of a large number is small.

- When one of the convection coefficients is *much smaller* than the other (say, $h_i << h_o$), we have $1/h_i >> 1/h_o$, and thus $U \approx h_i$.
- Therefore, the smaller heat transfer coefficient creates a ***bottleneck*** on the path of heat flow and seriously impedes heat transfer.
- This situation arises frequently when one of the fluids is a gas and the other is a liquid.
- In such cases, ***fins*** are commonly used on the gas side to enhance the product UA_s and thus the heat transfer on that side.

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

**Multiply the listed values by 0.176 to convert them to Btu/h ·ft2 ·°F.*

†Based on air-side surface area.

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

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

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

- For **short fins** of high thermal conductivity, we can use this total area in the convection resistance relation $\boldsymbol{R_{\text{conv}} = 1/hA_s}$ since the fins in this case will be very nearly isothermal.

- Otherwise, we should determine the effective surface area A_s from

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

Fouling Factor

- The performance of heat exchangers usually deteriorates with time as a result of accumulation of ***deposits*** on heat transfer surfaces.

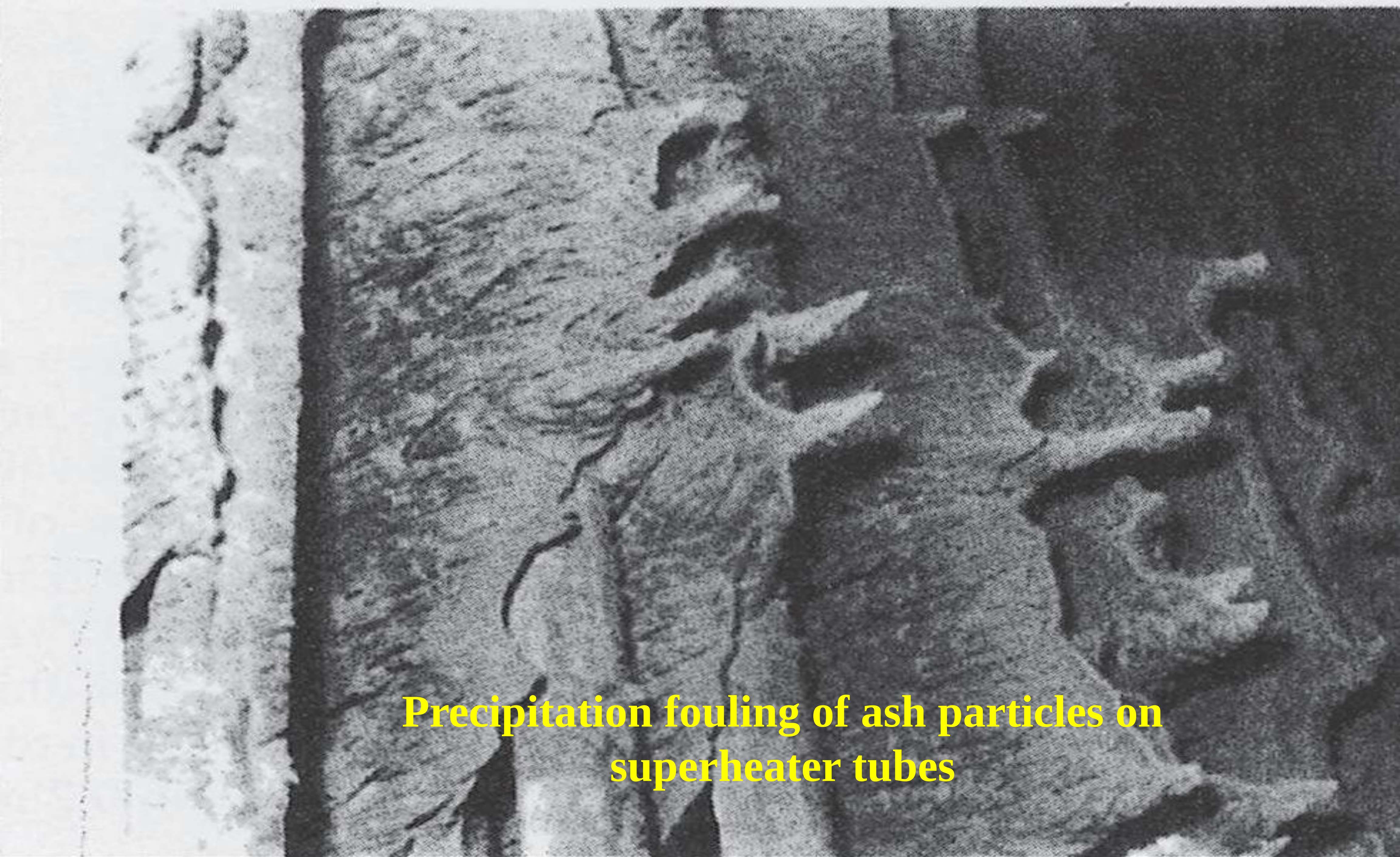
- The layer of deposits represents ***additional resistance*** to heat transfer and causes the rate of heat transfer in a heat exchanger to decrease.

- The net effect of these accumulations on heat transfer is represented by a **fouling factor** R_f, which is a measure of the ***thermal resistance*** introduced by fouling.

- The most common type of fouling is the ***precipitation*** of solid deposits in a fluid on the heat transfer surfaces.

- Another form of fouling, which is common in the chemical process industry, is *corrosion* and other ***chemical fouling***.

- Heat exchangers may also be fouled by the growth of algae in warm fluids (***biological fouling)***



Precipitation fouling of ash particles on superheater tubes

- The fouling factor depends on the ***operating temperature*** and the ***velocity*** of the fluids, as well as the length of service.

- Fouling increases with *increasing temperature* and *decreasing velocity*.

- For an unfinned shell-and-tube heat exchanger :

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

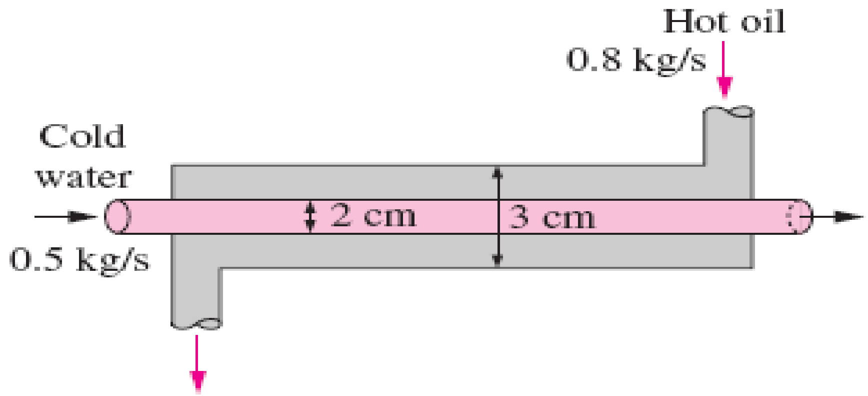
$\boldsymbol{R_{f,i}}$ and $\boldsymbol{R_{f,o}}$ are the fouling factors

Representative fouling factors (thermal resistance due to fouling for a unit surface area) (*Source:* Tubular Exchange Manufacturers Association.)

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

EXAMPLE : Overall Heat Transfer Coefficient of a Heat Exchanger

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



SOLUTION

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

Assumptions

1. The thermal resistance of the inner tube is negligible since the tube material is highly conductive and its thickness is negligible.

2. Both the oil and water flow are fully developed.

3. Properties of the oil and water are constant.

- The properties of water at 45°C are

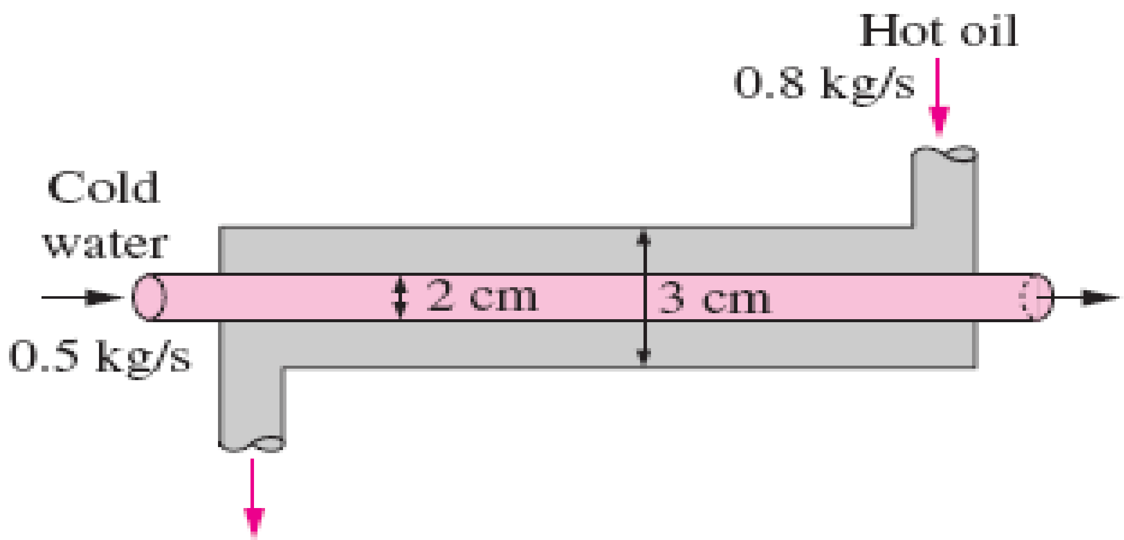
$\rho = 990 \text{ kg/m}^3$ $\qquad$ $\text{Pr} = 3.91$

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

- The properties of oil at 80°C are

$\rho = 852 \text{ kg/m}^3$ $\qquad$ $\text{Pr} = 490$

$k = 0.138 \text{ W/m} \cdot \text{°C}$ $\qquad$ $\nu = 37.5 \times 10^{-6} \text{ m}^2\text{/s}$



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

- The hydraulic diameter for a circular tube is the diameter of the tube itself, $D_h = D = 0.02$ m.
- The mean velocity of water in the tube and the Reynolds number are

$$\mathcal{V}_m = \frac{\dot{m}}{\rho A_c} = \frac{\dot{m}}{\rho(\frac{1}{4}\pi D^2)} = \frac{0.5 \text{ kg/s}}{(990 \text{ kg/m}^3)[\frac{1}{4}\pi(0.02 \text{ m})^2]} = 1.61 \text{ m/s}$$

$$\mathrm{Re} = \frac{\mathcal{V}_m D_h}{\nu} = \frac{(1.61\ \mathrm{m/s})(0.02\ \mathrm{m})}{0.602 \times 10^{-6}\ \mathrm{m^2/s}} = 53{,}490$$

- Therefore, the flow of water is turbulent.
- Assuming the flow to be fully developed, the Nusselt number can be determined from

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

$$h = \frac{k}{D_h}\mathrm{Nu} = \frac{0.637\ \mathrm{W/m \cdot {}^\circ C}}{0.02\ \mathrm{m}}(240.6) = 7663\ \mathrm{W/m^2 \cdot {}^\circ C}$$

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

$$\rho = 852 \text{ kg/m}^3 \qquad \nu = 37.5 \times 10^{-6} \text{ m}^2\text{/s}$$
$$k = 0.138 \text{ W/m} \cdot {}^\circ\text{C} \qquad \text{Pr} = 490$$

- The hydraulic diameter for the annular space is

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

The mean velocity and the Reynolds number in this case are

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

$$\text{Re} = \frac{\mathcal{V}_m D_h}{\nu} = \frac{(2.39 \text{ m/s})(0.01 \text{ m})}{37.5 \times 10^{-6} \text{ m}^2\text{/s}} = 637$$

which is less than 4000.
Therefore, the flow of oil is laminar.

Nusselt number for fully developed laminar flow in a circular annulus with one surface insulated and the other isothermal (Kays and Perkins, Ref. 8.)

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

- Assuming fully developed flow, the Nusselt number on the tube side of the annular space Nu_i corresponding to D_i / D_o = 0.02/0.03 = 0.667 can be determined from the table by interpolation, we find :

$$Nu = 5.45$$

- and

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

- Then the overall heat transfer coefficient for this heat exchanger becomes

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

Discussion

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

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

Calculating U using Log Mean Temperature

Hot Stream : $dq_h = \dot{m}_h.C_p^h.dT_h$

Cold Stream: $dq_c = \dot{m}_c.C_p^c.dT_c$

$$d(\Delta T) = dT_h - dT_c$$

$$d(\Delta T) = d(T_h - T_c)$$

$$\Delta T = T_h - T_c$$

$$\rightarrow d(\Delta T) = \left(\frac{dq_h}{m_h.C_p^h} - \frac{dq_c}{m_c.C_p^c}\right)$$

$$dq = -dq_{hot} = dq_{cold}$$

$$-dq = -U.\Delta T.dA$$

$$d(\Delta T) = -U.\Delta T.dA\left(\frac{1}{m_h.C_p^h} + \frac{1}{m_c.C_p^c}\right)$$

$$\int_{\Delta T_1}^{\Delta T_2} \frac{d(\Delta T)}{\Delta T} = -U.\left(\frac{1}{m_h.C_p^h} + \frac{1}{m_c.C_p^c}\right).\int_{A_1}^{A_2} dA$$

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

$$q = \dot{m}_h.C_p^h.\Delta T_h$$

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

Log Mean Temperature Difference

Calculating U using Log Mean Temperature

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

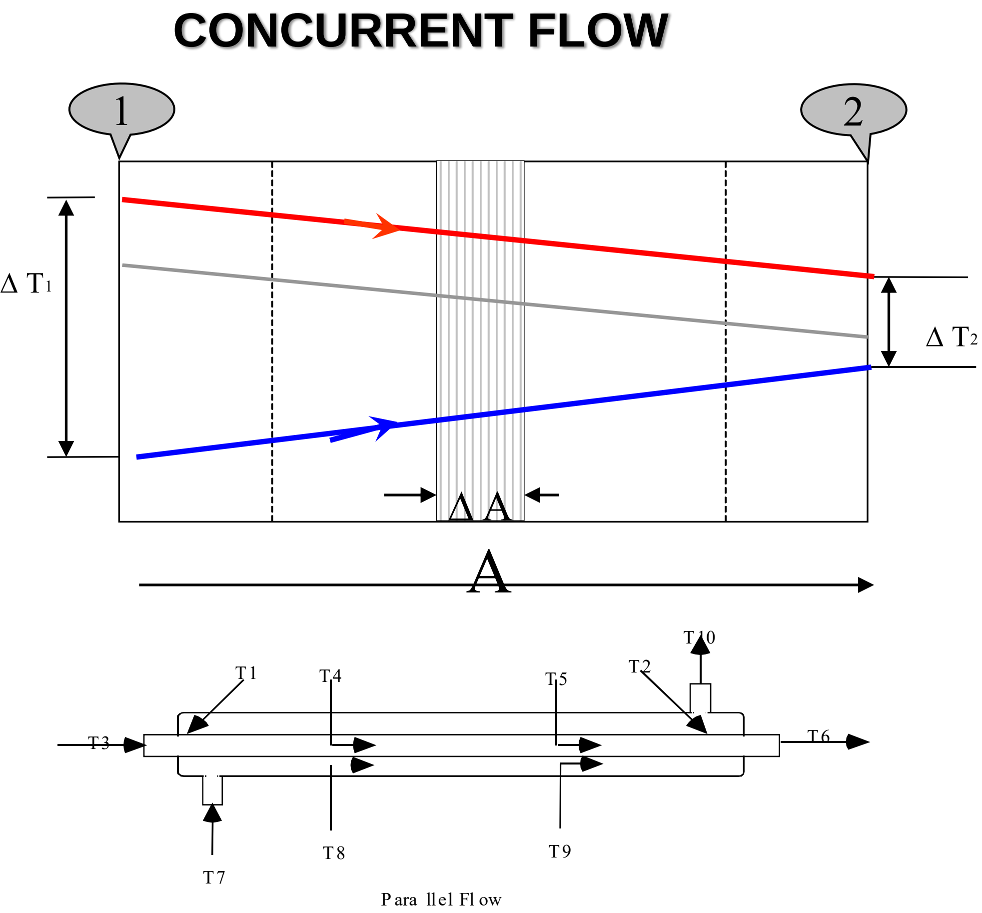
$$\ln\left(\frac{\Delta T_2}{\Delta T_1}\right) = -\frac{U.A}{q}\left[\left(T_h^{in} - T_c^{in}\right) - \left(T_h^{out} - T_c^{out}\right)\right]$$

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

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

Log Mean Temperature Difference

Log Mean Temperature Evaluation



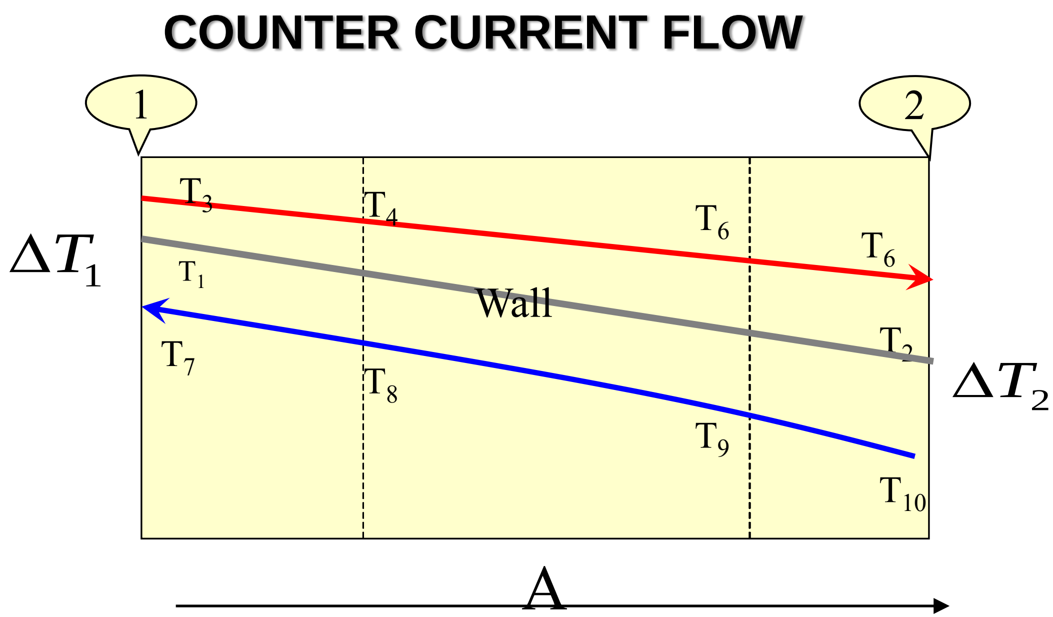
$$\Delta T_1 = T_h^{in} - T_c^{in} = T_3 - T_7$$

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

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

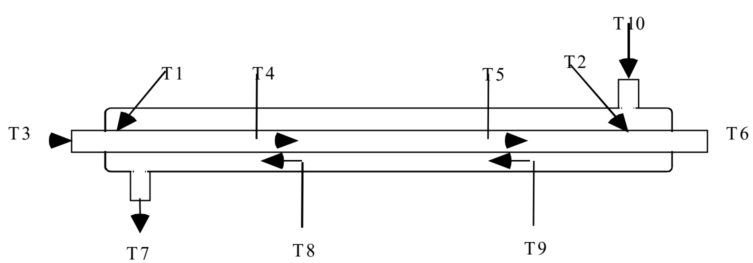
$$U = \frac{\dot{m}_h.\dot{C}_p^h.(T_3 - T_6)}{A.\Delta T_{Ln}} = \frac{\dot{m}_c.\dot{C}_p^c.(T_7 - T_{10})}{A.\Delta T_{Ln}}$$

Log Mean Temperature Evaluation



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

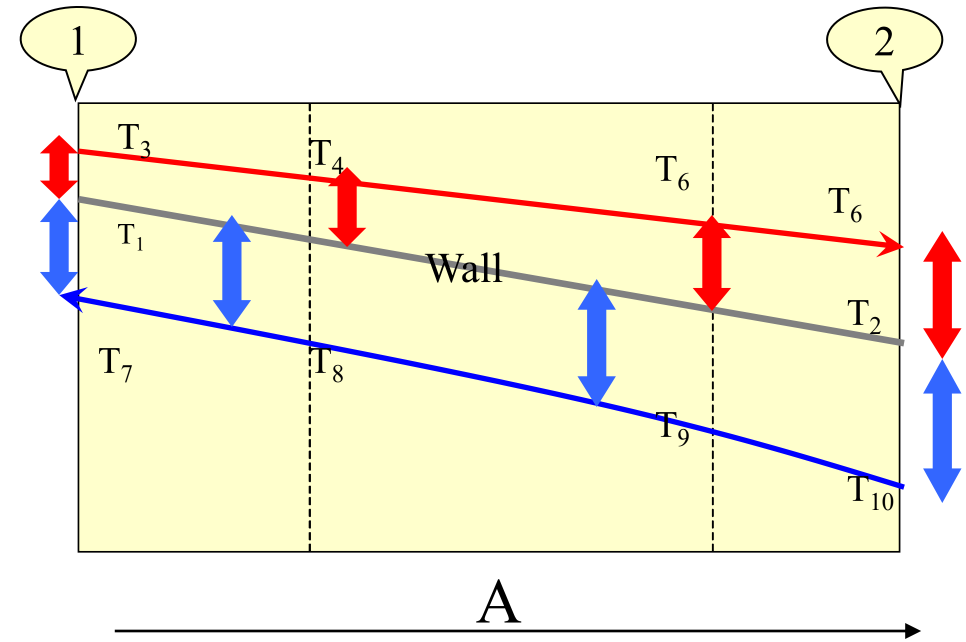
$$\Delta T_2 = T_h^{out} - T_c^{in} = T_6 - T_{10}$$



Counter - Current Flow

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

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



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

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

DIMENSIONLESS ANALYSIS TO CHARACTERIZE A HEAT EXCHANGER

$$\mathrm{Nu} = \mathrm{f}(\mathrm{Re}, \mathrm{Pr}, \mathrm{L/D}, \mu_b/\mu_o)$$

$$\frac{h.D}{k} \qquad \frac{v.D.\rho}{\mu} \qquad \frac{C_p.\mu}{k}$$

- Further Simplification: $$Nu = a.\mathrm{Re}^b\,.\mathrm{Pr}^c$$

$$Nu = \frac{D}{\delta}$$

Can Be Obtained from 2 set of experiments

One set, run for constant Pr

And second set, run for constant Re

h

$$q = \frac{k}{\delta} A(T_w - T)$$

•Empirical Correlation

- **For laminar flow**

Nu = 1.62 (Re*Pr*L/D)

- **For turbulent flow**

$$Nu = 0.026.\mathrm{Re}^{0.8}.\mathrm{Pr}^{1/3}.\left(\frac{\mu_b}{\mu_o}\right)^{0.14}$$

- Good To Predict within 20%
- Conditions: L/D > 10
  0.6 < Pr < 16,700
  Re > 20,000

The End
Terima kasih

REFERENCES

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