- The main process in steam power plant is the conversion of heat energy into electrical energy.

- However, the entire process comprises of numerous steps designed to work together efficiently.

- The different arrangements are:
  - Coal and ash handling arrangement
  - Steam generating plant
    - Heat exchanger
  - Steam turbine
  - Alternator
  - Feedwater
  - Cooling arrangement
    - Heat exchanger

Heat Exchangers

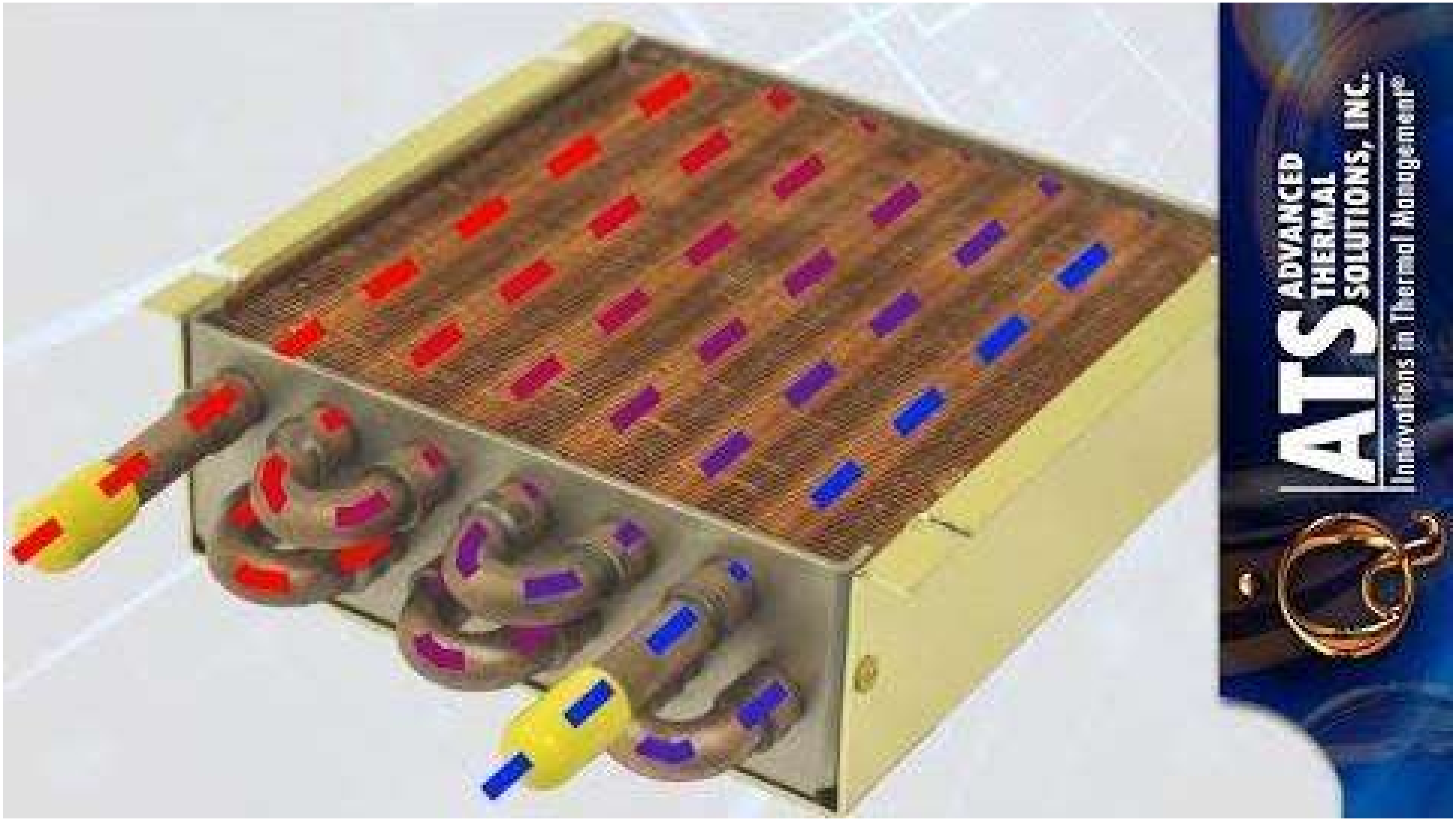
Learning Outcomes

- Classification of heat exchangers

- Heat exchanger design methods
  - ✓ Overall heat transfer coefficient
  - ✓ LMTD method
  - ✓ ε-NTU method

- Heat exchangers pressure drop

- Typical heat exchanger designs
  - ✓ Double pipe
  - ✓ Shell and tube

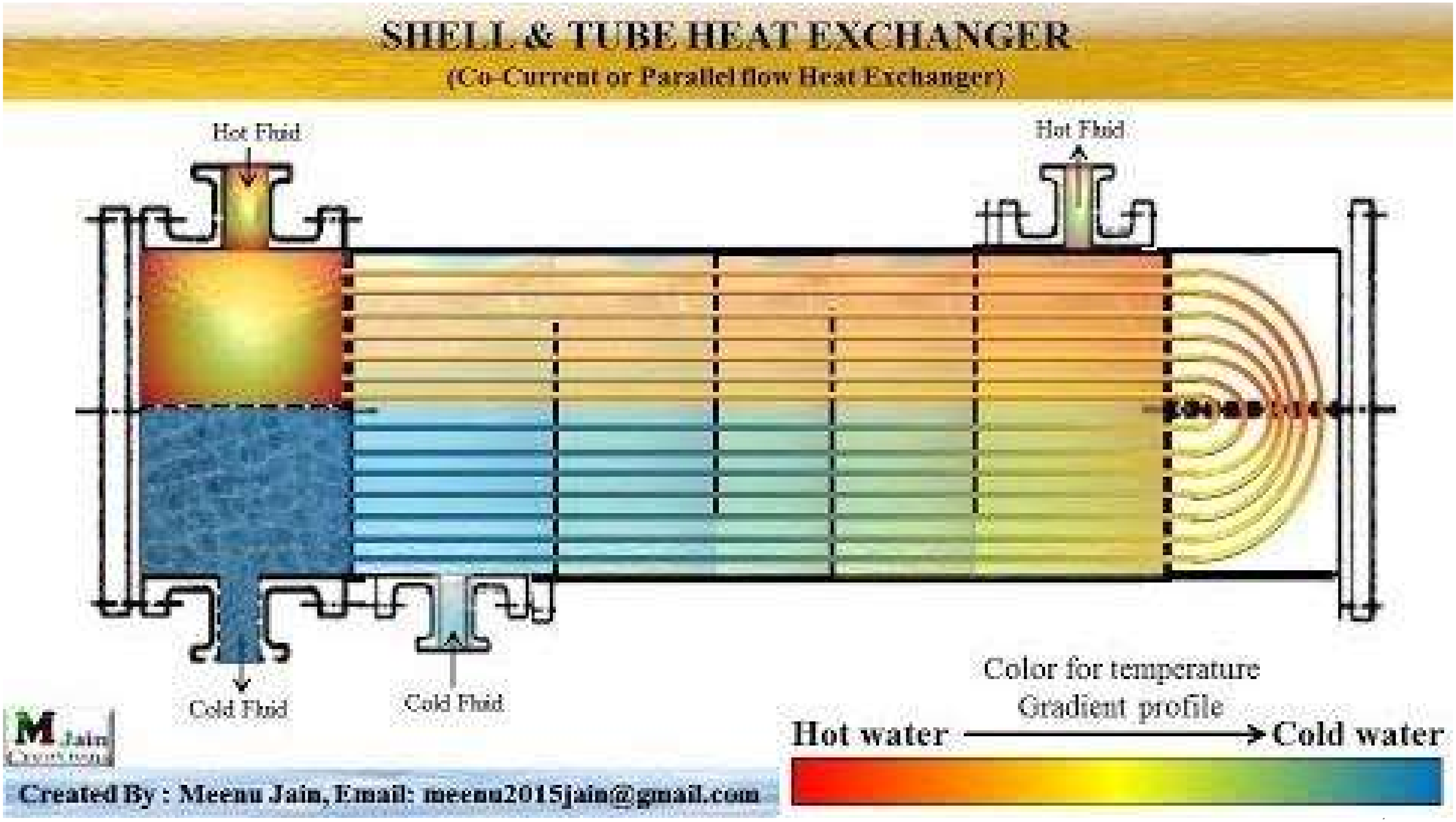


https://www.youtube.com/watch?v=WYJ9BsCrifQ

Classification

- Different ways to classify heat exchangers
  - ✓ Direct vs. indirect contact
  - ✓ Single phase vs. two phase
  - ✓ Geometry
    - ➢ Shell and tube
    - ➢ Plate and frame
    - ➢ Compact
    - ➢ Double pipe

  - ✓ Flow arrangements
    - ➢ Parallel flow
    - ➢ Counter flow
    - ➢ Cross flow



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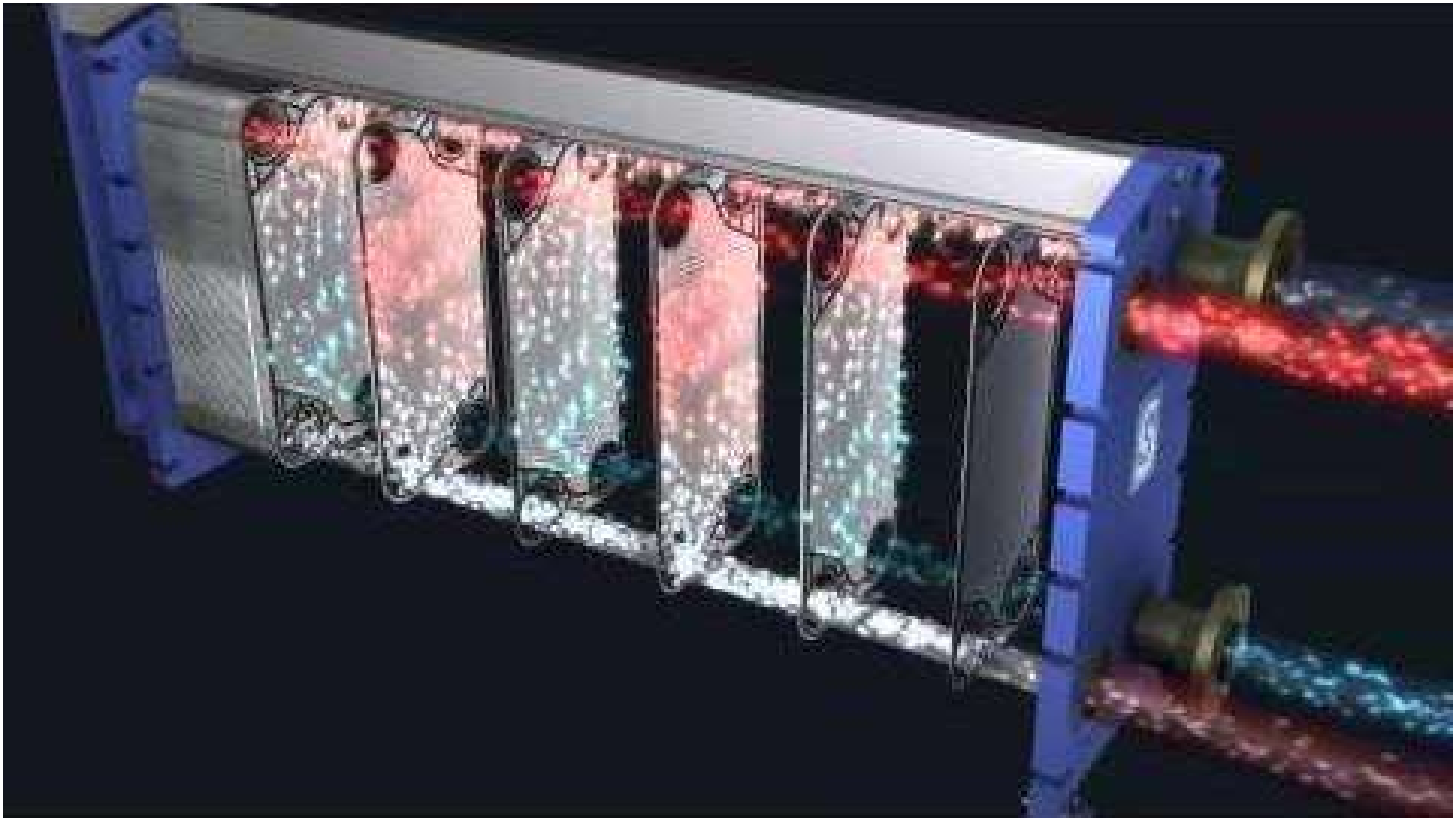

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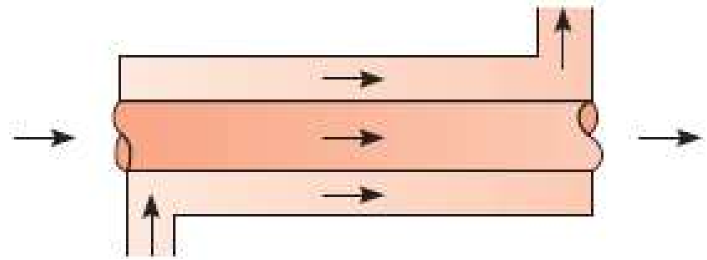
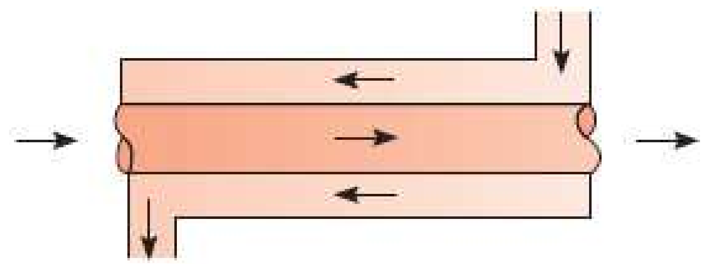
Classification

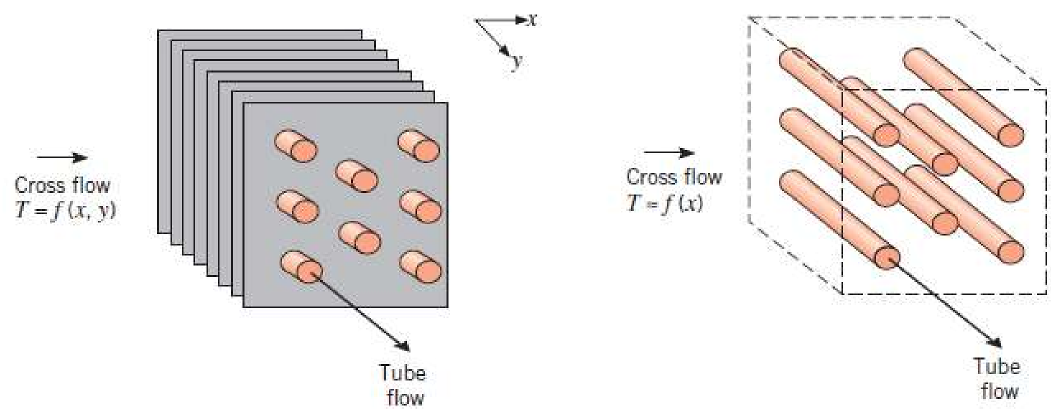
- Flow arrangement
  - ✓ Parallel flow



  - ✓ Counter flow

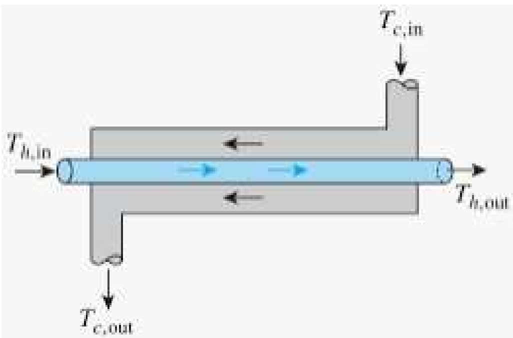


  - ✓ Cross flow



Important Heat Exchanger Types

Double pipe heat exchanger



Shell and tube heat exchanger

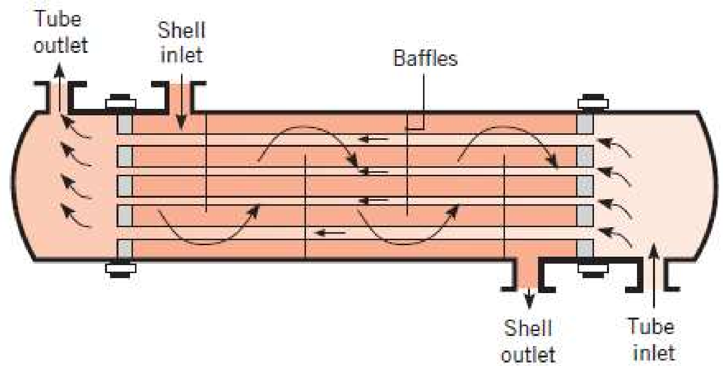
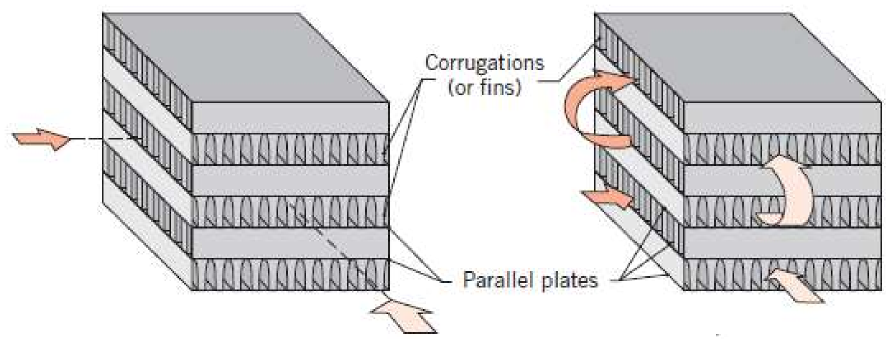
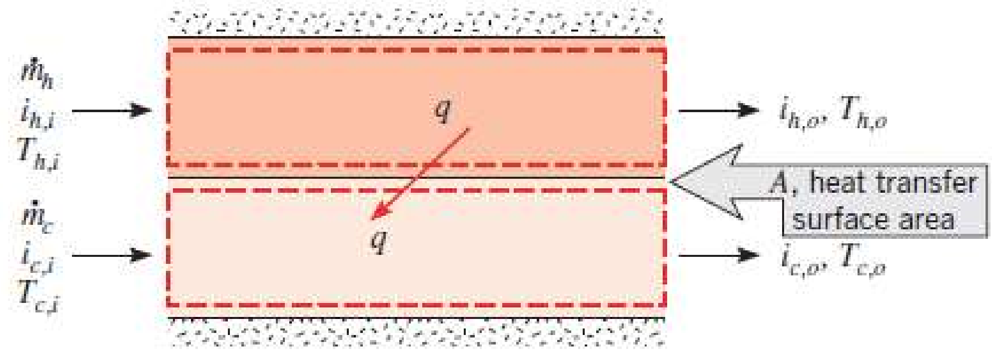


Plate heat exchanger

Compact heat exchanger



Energy Balance



- ✓ Cold side

$$Q_c = (\dot{m}c_p)_c (T_{c,o} - T_{c,i})$$

- ✓ Hot side

$$Q_h = (\dot{m}c_p)_h (T_{h,i} - T_{h,o})$$

- ✓ Heat transfer rate

$$Q = UA\Delta T_m$$

where U: overall heat transfer coefficient
A: total surface area for heat exchange
ΔT_m: some mean temperature difference

$$\Delta T_m = f(T_{h,i}, T_{h,o}, T_{c,i}, T_{c,o})$$

Overall Heat Transfer Coefficient

$$\frac{1}{UA} = \frac{1}{(\eta_o hA)_i} + \frac{1}{Sk_w} + \frac{1}{(\eta_o hA)_o}$$

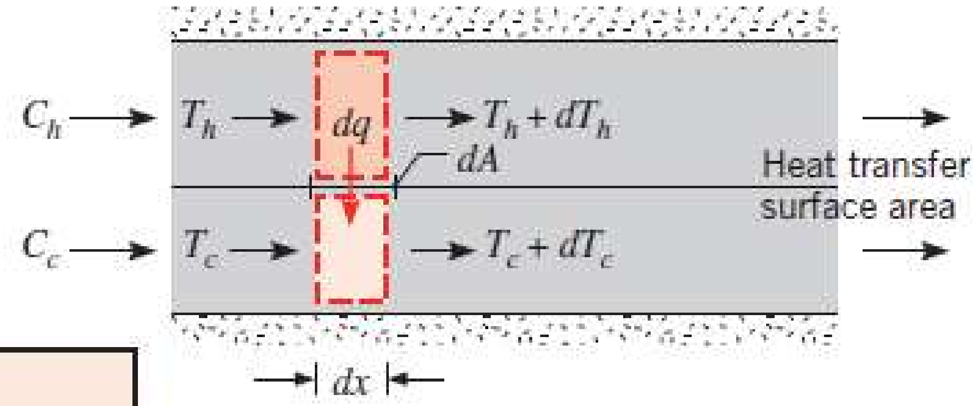
where η_o: surface efficiency
h: heat transfer coefficient
S: shape factor
k_w: wall conductivity

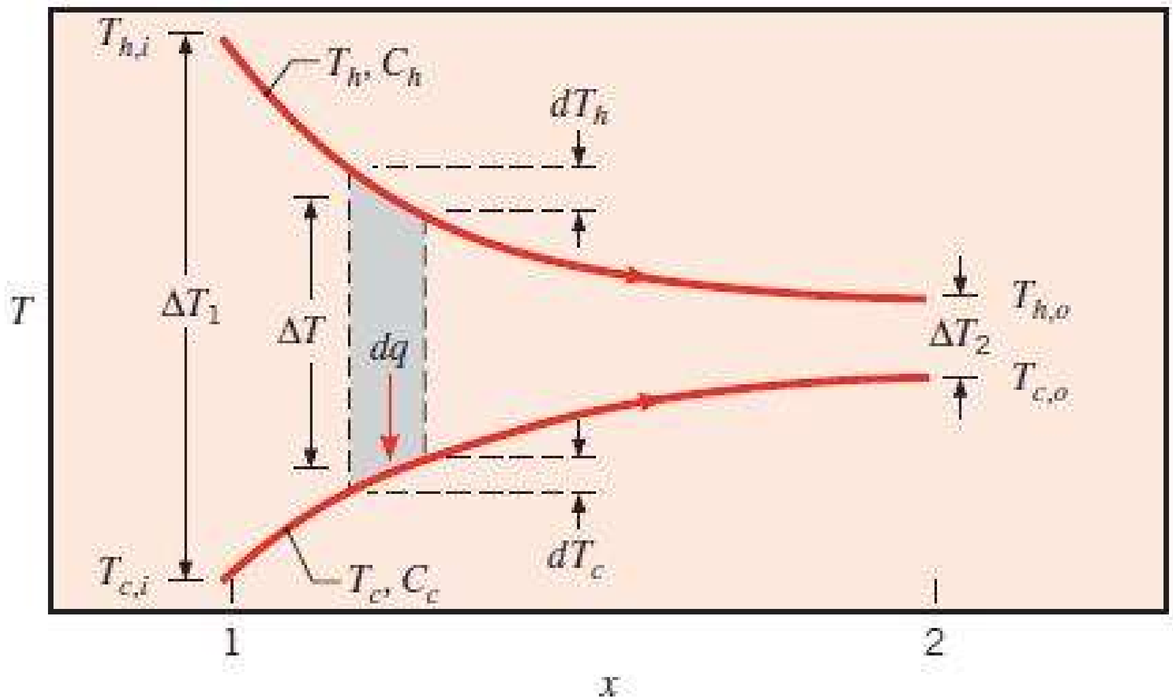
For surface with fins: $\eta_o = 1 - \frac{A_f}{A}\left(1 - \eta_f\right)$

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

Note: $UA = U_o A_o = U_i A_i$

Parallel Flow Heat Exchanger



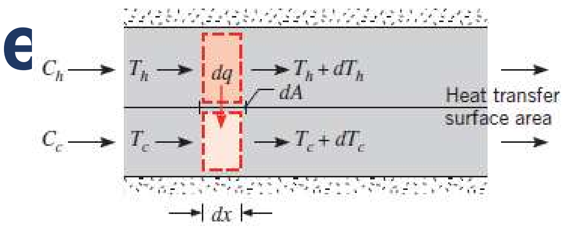


Parallel Flow Heat Exchanger

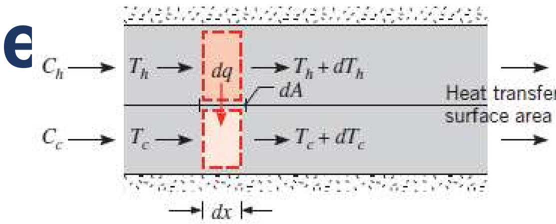
Assumptions:

1. The heat exchanger is insulated from its surroundings, in which case the only heat exchange is between the hot and cold fluids.
2. Axial conduction along the tubes is negligible.
3. Potential and kinetic energy changes are negligible.
4. The fluid specific heats are constant.
5. The overall heat transfer coefficient is constant.

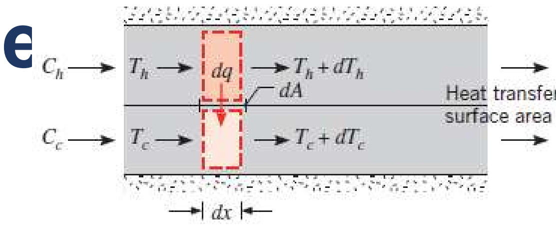
Parallel Flow Heat Exchange



Parallel Flow Heat Exchange

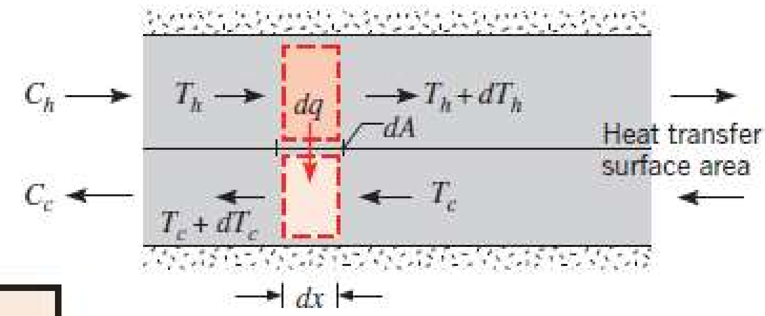


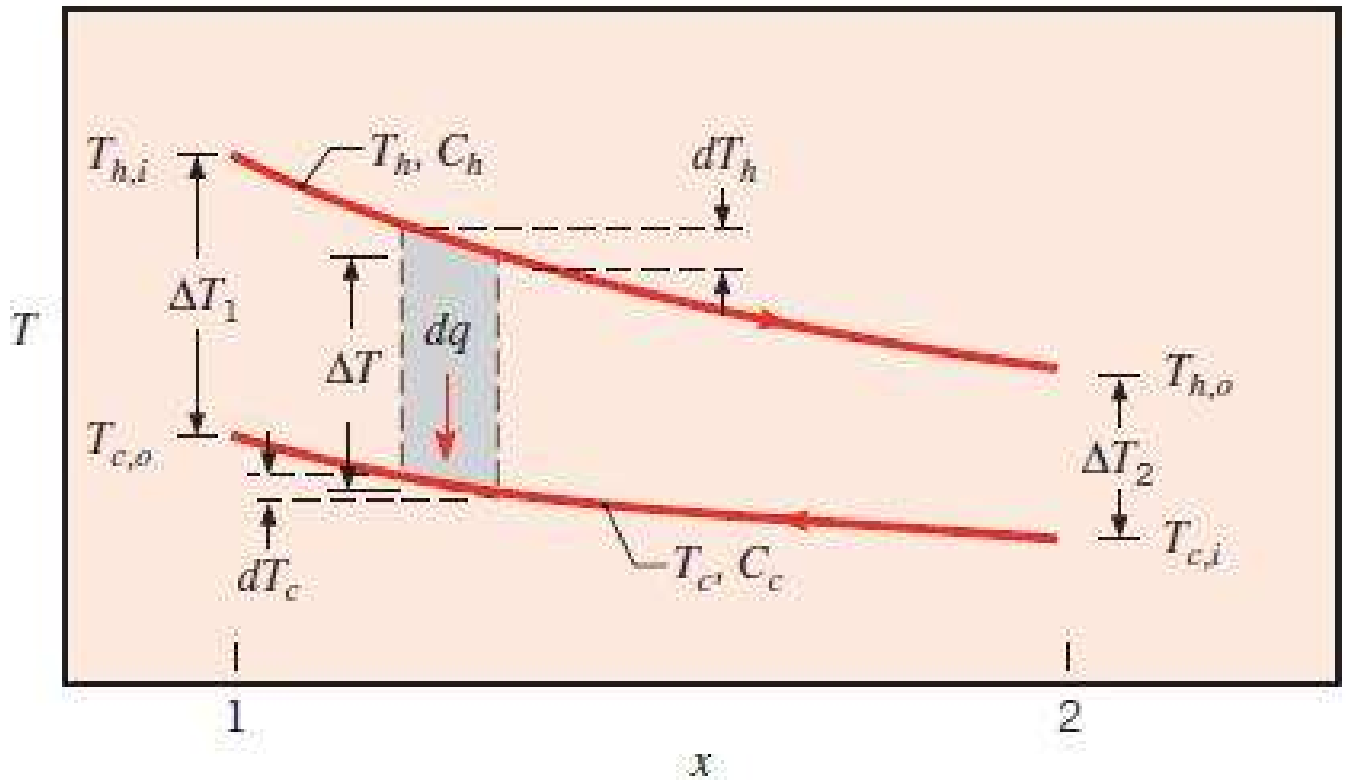
Parallel Flow Heat Exchange



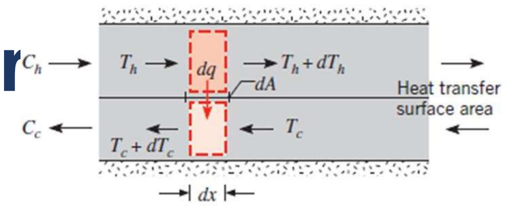
Counter Flow Heat Exchanger

$$\begin{bmatrix} \Delta T_1 \equiv T_{h,1} - T_{c,1} = T_{h,i} - T_{c,o} \\ \Delta T_2 \equiv T_{h,2} - T_{c,2} = T_{h,o} - T_{c,i} \end{bmatrix}$$



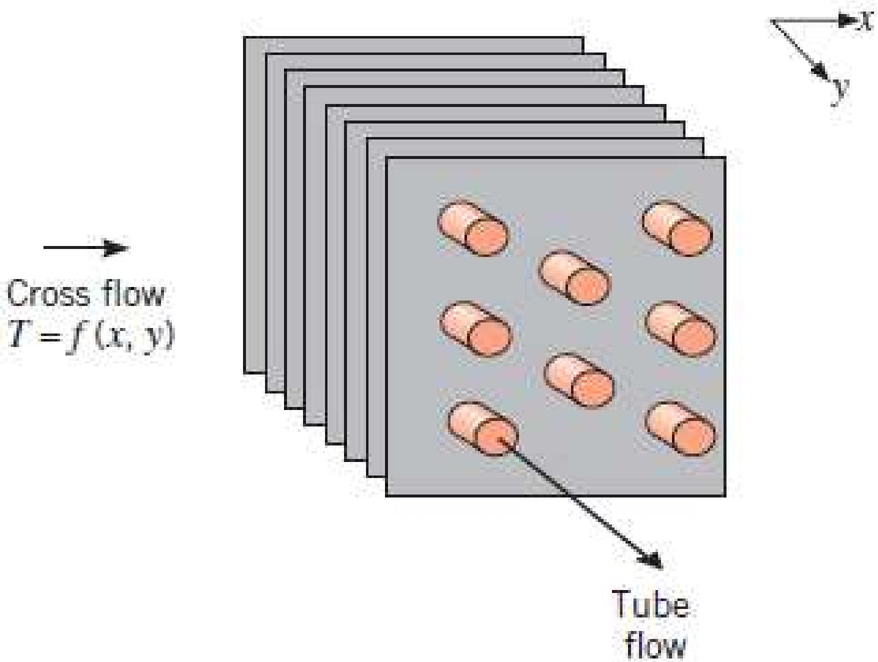


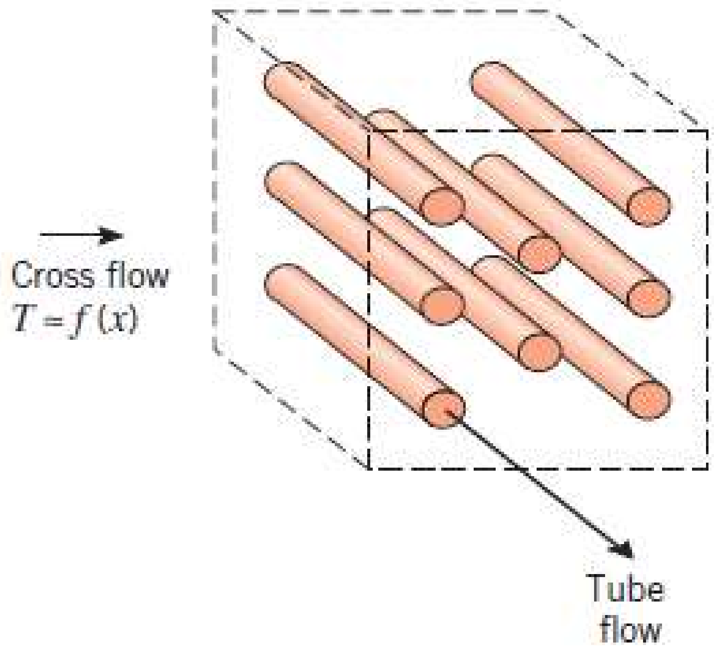
Counter Flow Heat Exchanger



- In contrast to the parallel-flow exchanger, this configuration provides for heat transfer between the hotter portions of the two fluids at one end, as well as between the colder portions at the other.
- The change in the temperature difference, $\Delta T = T_h - T_c$, with respect to x is nowhere as large as it is for the inlet region of the parallel-flow exchanger.
- Note that the outlet temperature of the cold fluid may now exceed the outlet temperature of the hot fluid.

Cross Flow Heat Exchanger





Cross Flow Heat Exchanger

Cross Flow Heat Exchanger

Cross Flow Heat Exchanger

Design Methods

- LMTD method (**L**og **M**ean **T**emperature **D**ifference)

  ✓ For parallel flow or counter flow

$$\Delta T_{LMTD} = \frac{\Delta T_2 - \Delta T_1}{\ln \dfrac{\Delta T_2}{\Delta T_1}}$$

where ΔT_1, ΔT_2 : temperature difference at each end of the heat exchanger

  ✓ For cross flow $\quad Q = FUA\Delta T_{LMTD}$

where F is correction factor and ΔT_{LMTD} is based upon the counter flow configuration

  ✓ Note: In *LMTD* method we assume that all of the temperatures are known

Design Methods

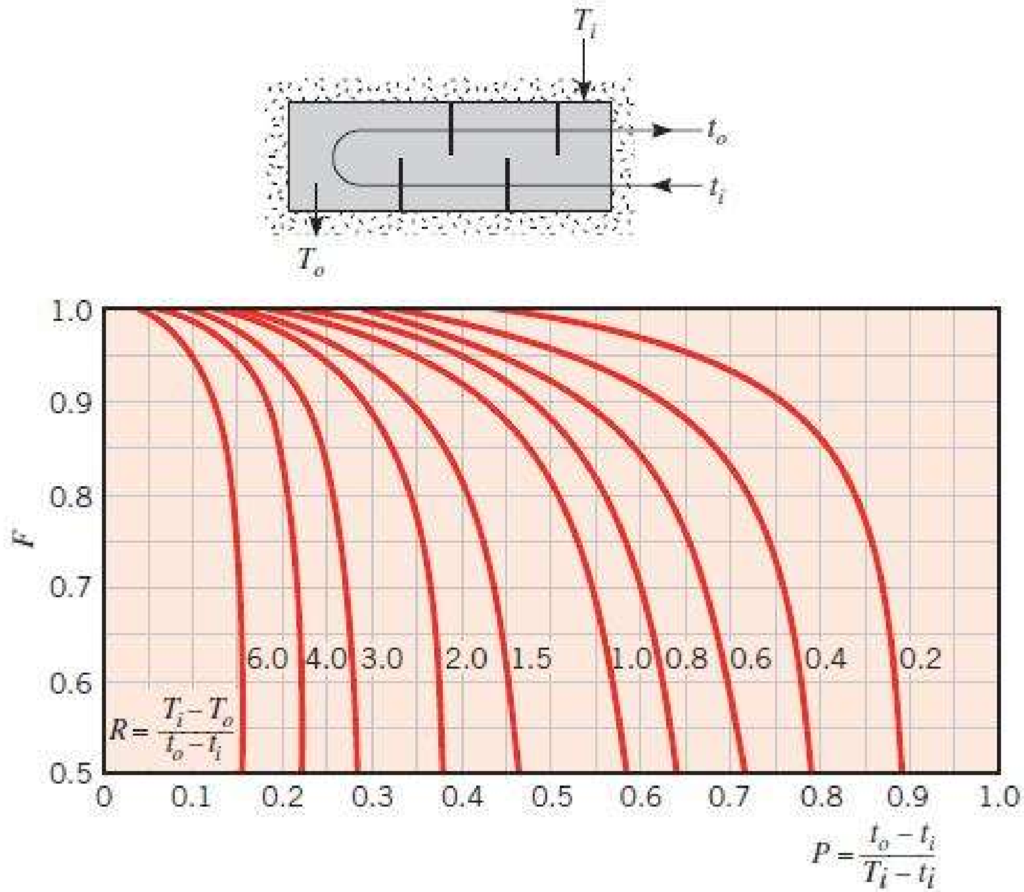
- LMTD method (**L**og **M**ean **T**emperature **D**ifference)
  - ✓ In this figure:

$$T_{h,i} = T_i$$

$$T_{h,o} = T_o$$

$$T_{c,i} = t_i$$

$$T_{c,o} = t_o$$



Correction factor for a shell-and-tube heat exchanger with one shell and any multiple of two tube passes (two, four, etc. tube passes).

Design Methods

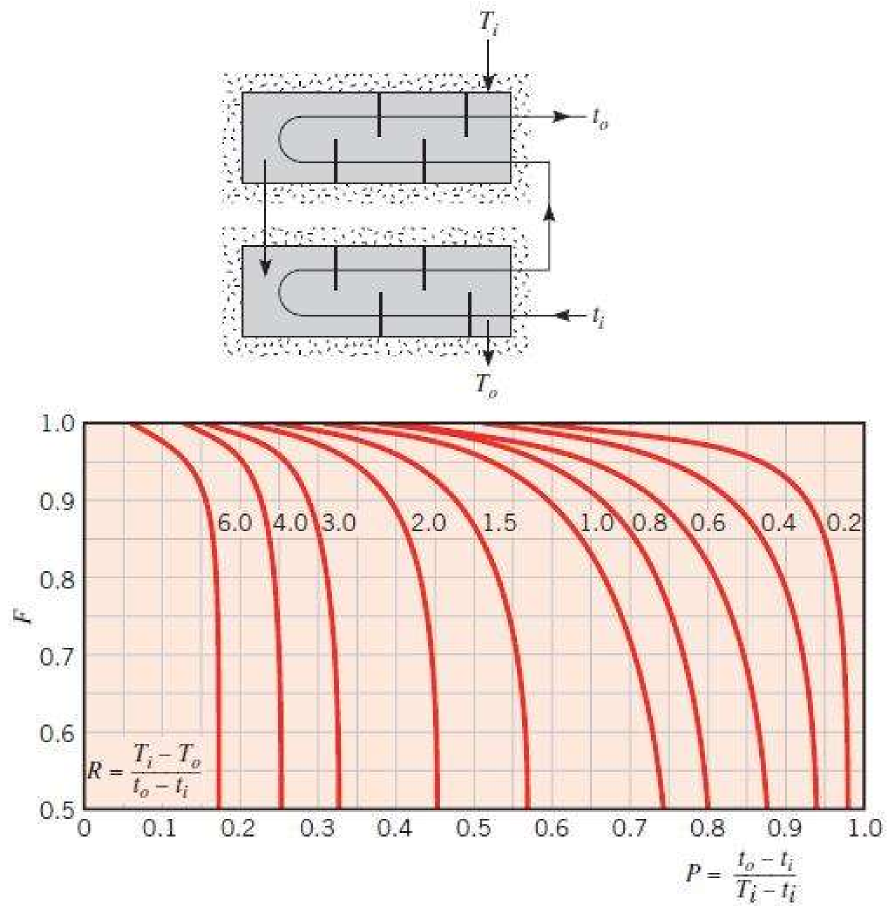
- LMTD method (**L**og **M**ean **T**emperature **D**ifference)
  - ✓ In this figure:

$$T_{h,i} = T_i$$

$$T_{h,o} = T_o$$

$$T_{c,i} = t_i$$

$$T_{c,o} = t_o$$



Correction factor for a shell-and-tube heat exchanger with two shell passes and any multiple of four tube passes (four, eight, etc. tube passes).

Design Methods

- LMTD method (**L**og **M**ean **T**emperature **D**ifference)

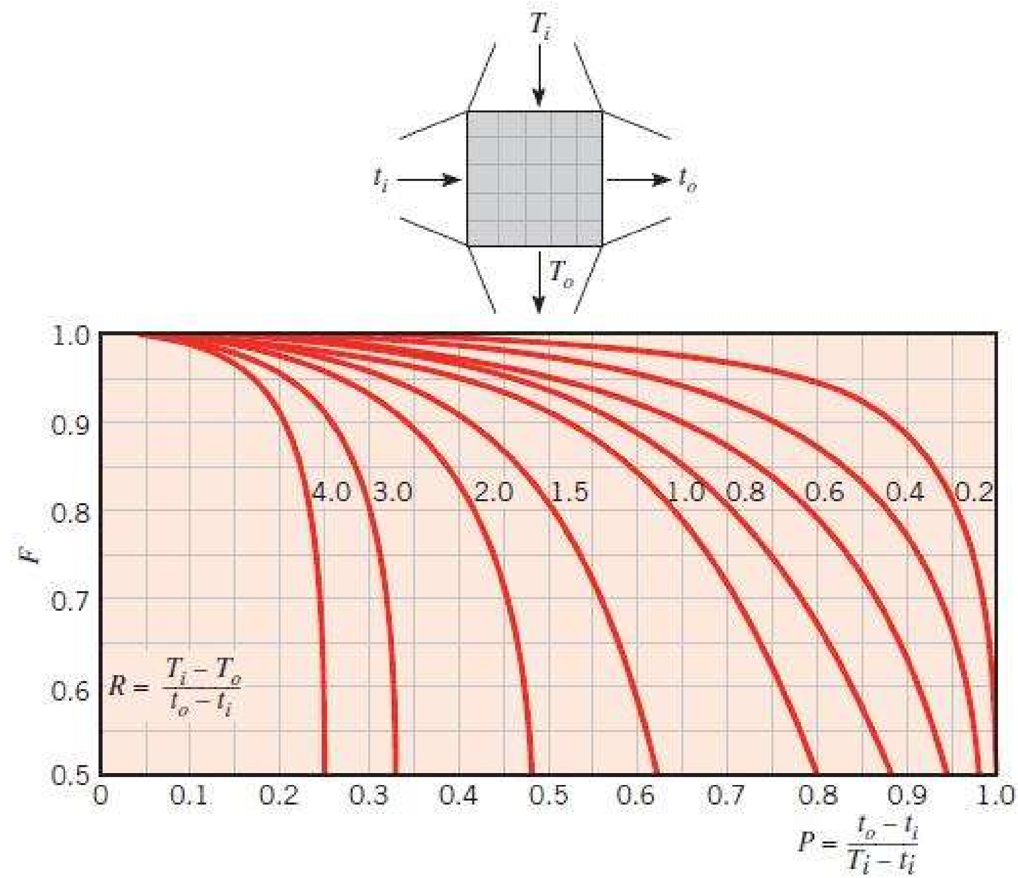
  ✓ In this figure:

  $$T_{h,i} = T_i$$

  $$T_{h,o} = T_o$$

  $$T_{c,i} = t_i$$

  $$T_{c,o} = t_o$$



Correction factor for a single-pass, cross-flow heat exchanger with both fluids unmixed.

Design Methods

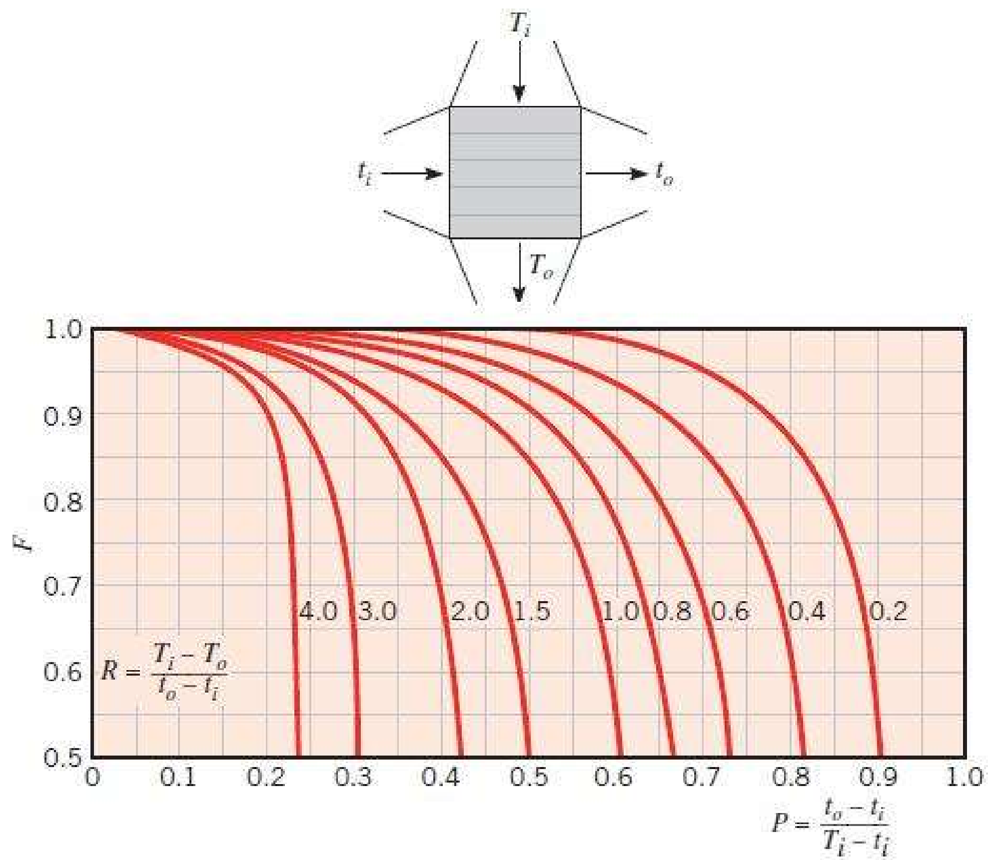
- LMTD method (**L**og **M**ean **T**emperature **D**ifference)
  - ✓ In this figure:

$$T_{h,i} = T_i$$

$$T_{h,o} = T_o$$

$$T_{c,i} = t_i$$

$$T_{c,o} = t_o$$



Correction factor for a single-pass, cross-flow heat exchanger with one fluid mixed and the other unmixed.

Problem (a)

An industrial boiler consists of tubes inside of which flow hot combustion gases. Water boils on the exterior of the tubes. When installed, the clean boiler has an overall heat transfer coefficient of 300 W/m²·K. Based on experience, it is anticipated that the fouling factors on the inner and outer surfaces will increase linearly with time as $R''_{f,i} = a_i t$ and $R''_{f,o} = a_o t$ where $a_i = 2.5 \times 10^{-11}$ m²·K/W·s and $a_o = 1.0 \times 10^{-11}$ m²·K/W·s for the inner and outer tube surfaces, respectively. If the boiler is to be cleaned when the overall heat transfer coefficient is reduced from its initial value by 25%, how long after installation should the first cleaning be scheduled? The boiler operates continuously between cleanings.

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Example 5.1

Oil flows in a heat exchanger with a mass flow rate of 20 kg/s and is to be cooled from $T_{h,i}$=120 °C to $T_{h,o}$=60 °C with water as a coolant flowing at a rate of 15 kg/s and an inlet temperature $T_{c,i}$=10 °C. If the total heat transfer coefficient is U=1100 W/m^{2}K determine the heat transfer area required for:

a) Parallel flow
b) Counter flow

$c_{p,oil}$=2000 J/kgK, $c_{p,water}$=4000 J/kgK

Oil flows in a heat exchanger with a mass flow rate of 20 (kg/s) and is to be cooled from $T_{h,i}$=120 (°C) to $T_{h,o}$=60 (°C) with water as a coolant flowing at a rate of 15 (kg/s) and an inlet temperature $T_{c,i}$=10 (°C). If the total heat transfer coefficient is U=1100 (W/m^2K) determine the heat transfer area required for:

a) Parallel flow

b) Counter flow

$c_{p,oil}$=2000 (J/kgK), $c_{p,water}$=4000 (J/kgK)

Oil flows in a heat exchanger with a mass flow rate of 20 (kg/s) and is to be cooled from $T_{h,i}$=120 (°C) to $T_{h,o}$=60 (°C) with water as a coolant flowing at a rate of 15 (kg/s) and an inlet temperature $T_{c,i}$=10 (°C). If the total heat transfer coefficient is U=1100 (W/m^{2}K) determine the heat transfer area required for:

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a) Parallel flow

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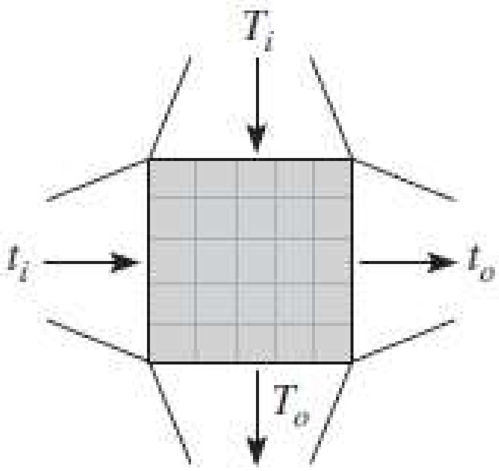
$c_{p,oil}$=2000 (J/kgK), $c_{p,water}$=4000 (J/kgK)

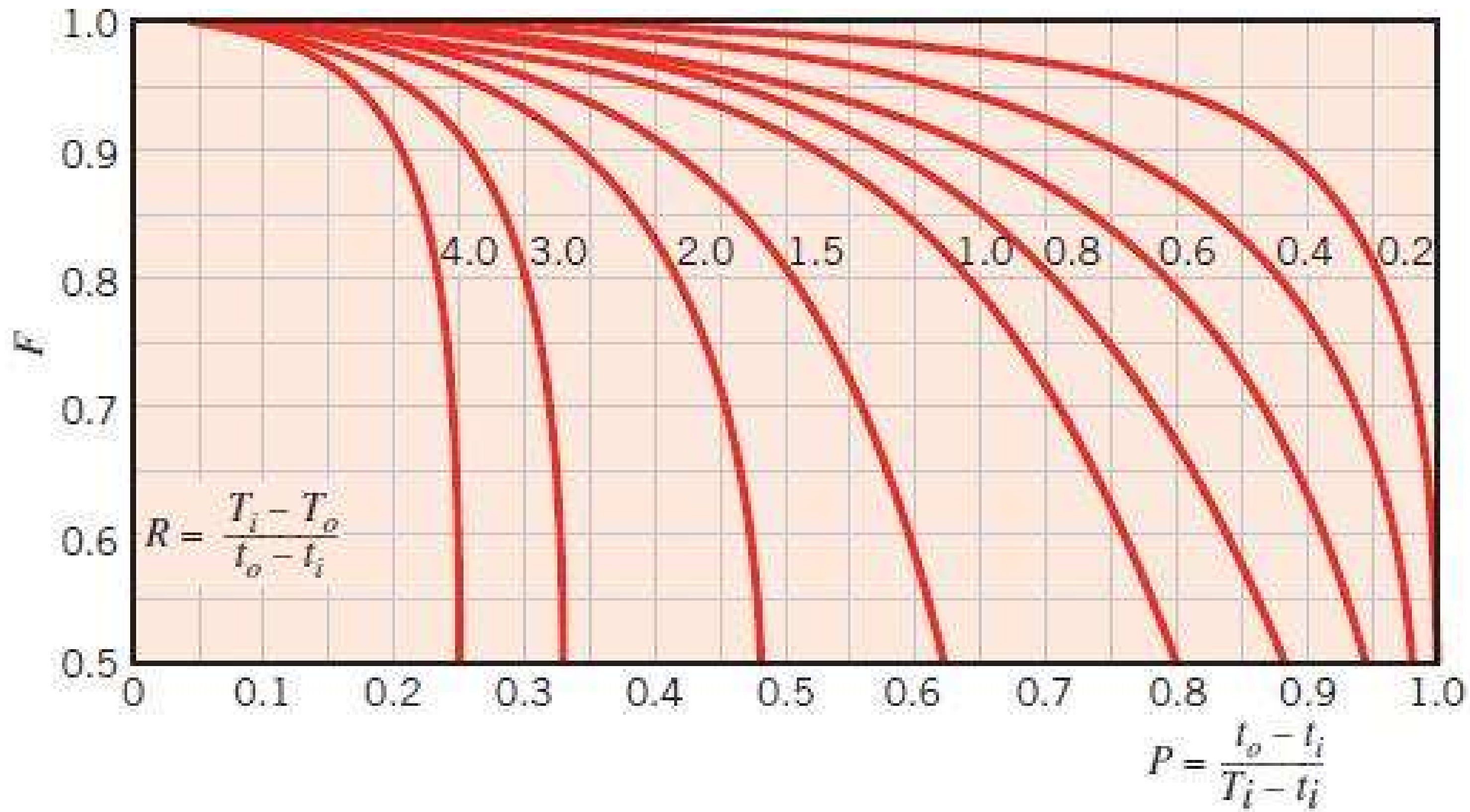
Problem (b)

A finned-tube, cross-flow heat exchanger is to use the exhaust of a gas turbine to heat pressurized water. Laboratory measurements are performed on a prototype version of the exchanger, which has a surface area of 8 m^2, to determine the overall heat transfer coefficient as a function of operating conditions. Measurements made under particular conditions, for which $\dot{m}_h$=1.5 kg/s, $T_{h,i}$ = 325°C, $\dot{m}_c$=0.5 kg/s, and $T_{c,i}$ = 25°C, reveal a water outlet temperature of $T_{c,o}$ = 125°C. What is the overall heat transfer coefficient of the exchanger using LMTD method?

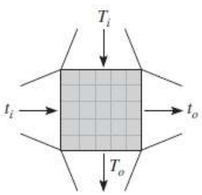
Problem (b)



T_i
t_i
t_o
T_o



1.0
0.9
0.8
0.7
0.6
0.5
F
4.0
3.0
2.0
1.5
1.0
0.8
0.6
0.4
0.2
$R = \frac{T_i - T_o}{t_o - t_i}$
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
$P = \frac{t_o - t_i}{T_i - t_i}$



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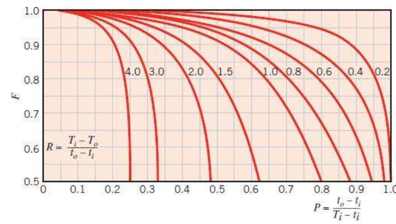

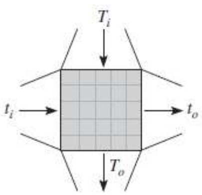

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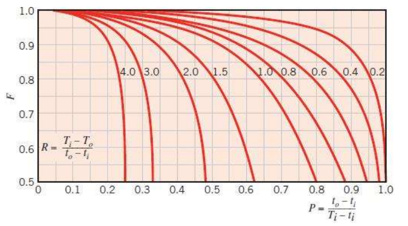

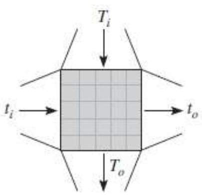

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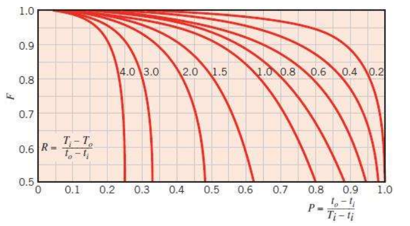
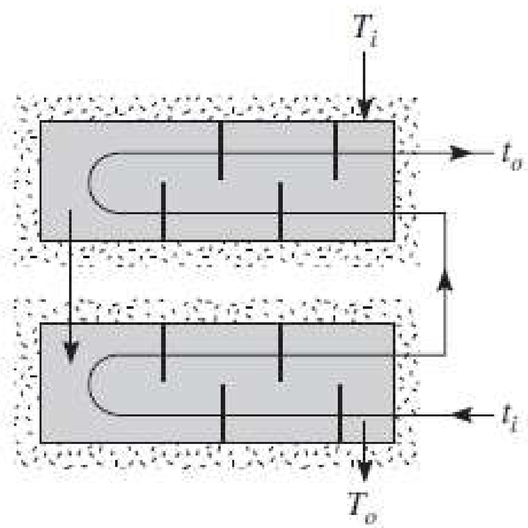


Problem (c)

Water at a rate of 45,500 kg/h is heated from 80 to 150°C in a heat exchanger having two shell passes and eight tube passes with a total surface area of 925 m^2. Hot exhaust gases having approximately the same thermophysical properties as air enter at 350°C and exit at 175°C. Determine the overall heat transfer coefficient.



https://www.godrej.com/p/heavy-engineering/Heat-Exchangers-Refineries/Breechlock-Heat-Exchanger

Problem (c)

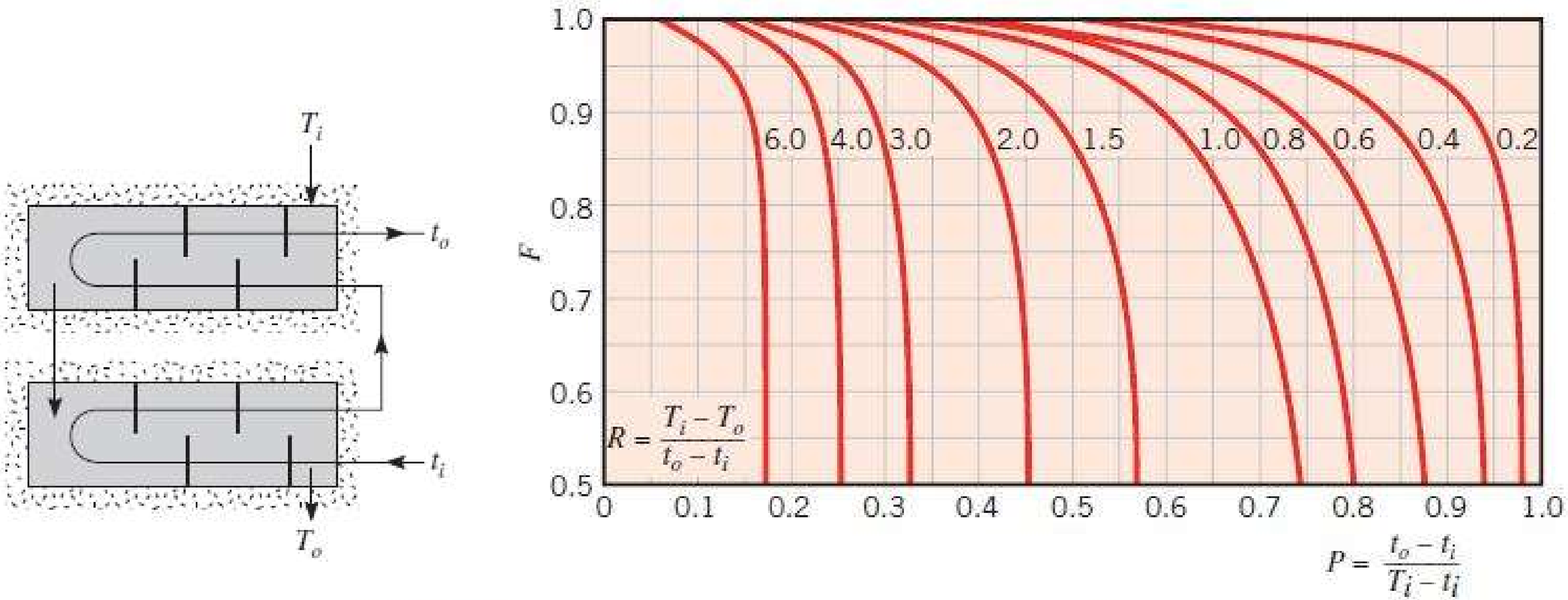
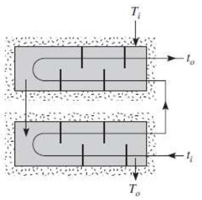


FIGURE 11S.2 Correction factor for a shell-and-tube heat exchanger with two shell passes and any multiple of four tube passes (four, eight, etc. tube passes).



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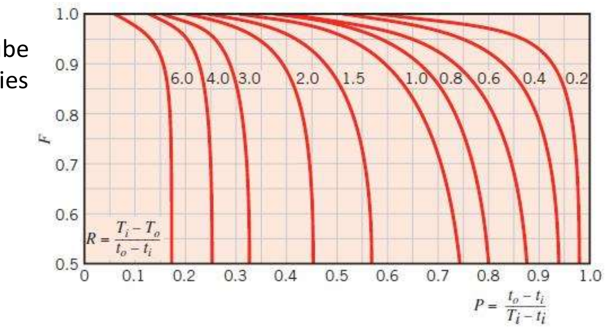
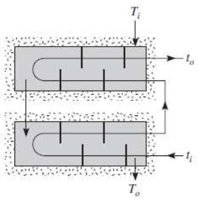
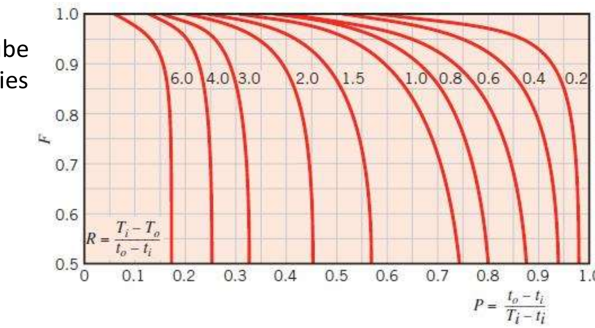


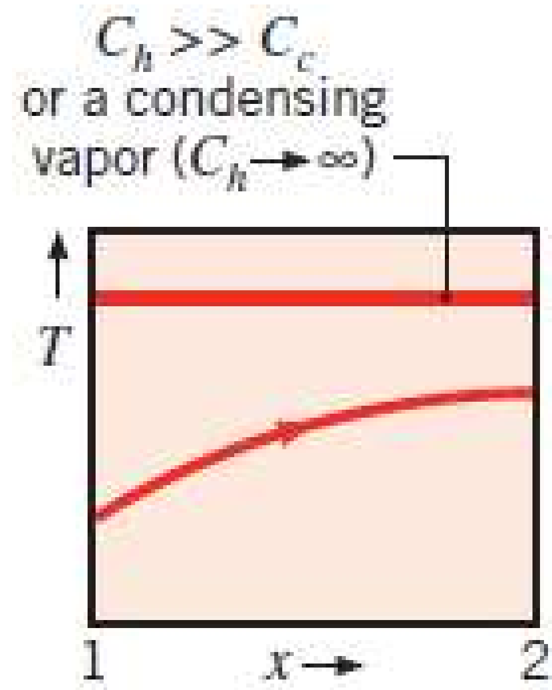


Water at a rate of 45,500 kg/h is heated from 80 to 150°C in a heat exchanger having two shell passes and eight tube passes with a total surface area of 925 m^2. Hot exhaust gases having approximately the same thermophysical properties as air enter at 350°C and exit at 175°C. Determine the overall heat transfer coefficient.



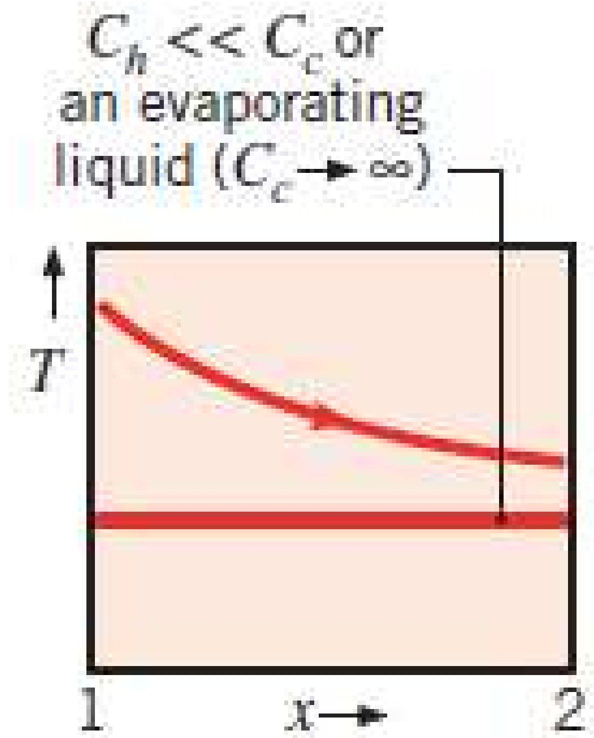
Special Operating Conditions

- It is useful to note certain special conditions under which heat exchangers may be operated.
- Some heat exchanger temperature distributions have the hot fluid heat capacity rate, $C_h \equiv m_h c_{p,h}$, being much larger than that of the cold fluid, $C_c \equiv m_c c_{p,c}$.
- 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, $C_h \rightarrow \infty$.



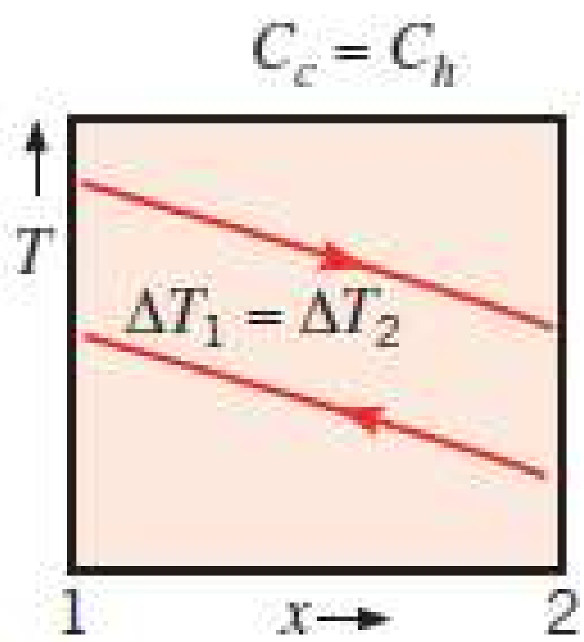
Special Operating Conditions

- In an evaporator or a boiler, it is the cold fluid that experiences a change in phase and remains at a nearly uniform temperature $C_c \rightarrow \infty$.
- The same effect is achieved without phase change if $C_h \ll C_c$.
- There instance also characterize an internal tube flow (or single stream heat exchanger) exchanging heat with a surface at constant temperature or an external fluid at constant temperature.



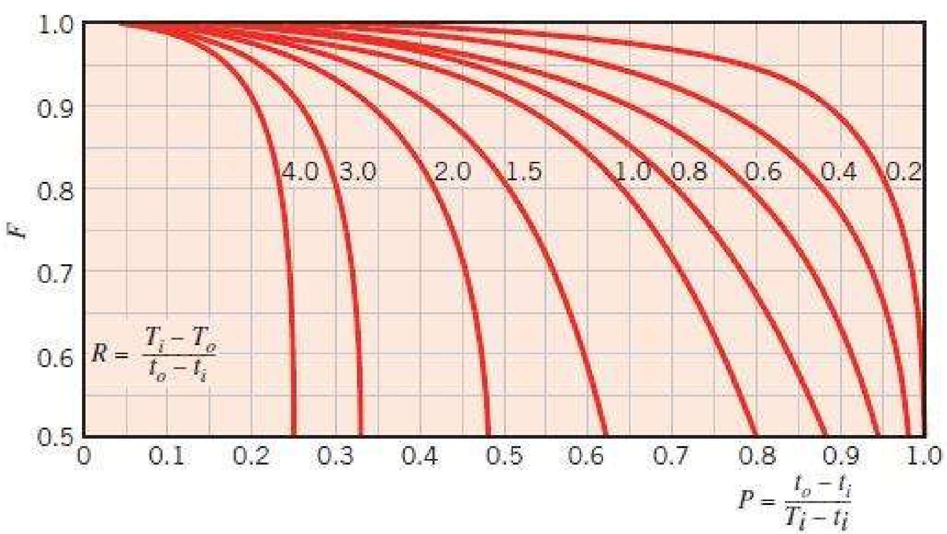
Special Operating Conditions

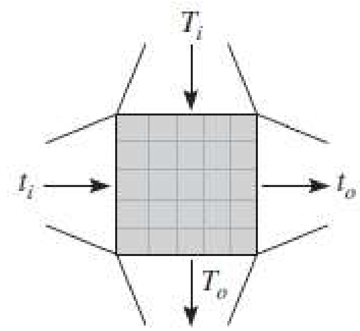
- The third special case involves a counterflow heat exchanger for which the heat capacity rates are equal (***C_h = C_c***). The temperature difference ΔT must then be constant throughout the exchanger, in which case **$\Delta T_1 = \Delta T_2 = \Delta T_{lmtd}$**.

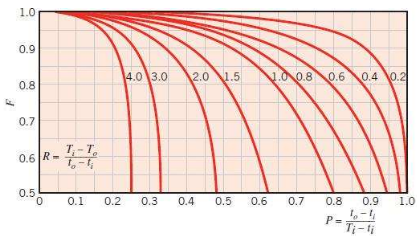


Example 5.2

The efficiency of a gas turbine is to be improved by increasing the air intake temperature to 210 (°C). A cross flow heat exchanger is designed to use exhaust gases to heat the air. The flow rates are $m_c=m_h=10$ (kg/s) and it may be assumed that the heat transfer coefficients are equal $h_c=h_h=150$ (W/m^{2}K) due to the construction. If the incoming air is $T_{c,i}=25$ (°C) and the exhaust gases are at 425 (°C). Determine the surface area required. Assume $c_{p,c}=c_{p,h}=1000$ (J/kgK), and both flows are unmixed.







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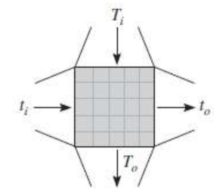
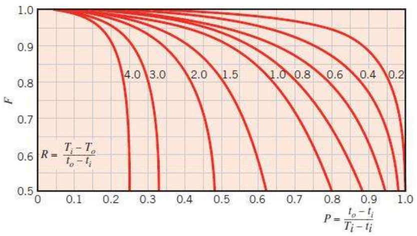
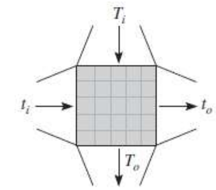




The efficiency of a gas turbine is to be improved by increasing the air intake temperature to 210 (°C). A cross flow heat exchanger is designed to use exhaust gases to heat the air. The flow rates are $m_c = m_h = 10$ (kg/s) and it may be assumed that the heat transfer coefficients are equal $h_c = h_h = 150$ (W/m^2K) due to the construction. If the incoming air is $T_{c,i} = 25$ (°C) and the exhaust gases are at 425 (°C). Determine the surface area required. Assume $c_{p,c} = c_{p,h} = 1000$ (J/kgK), and both flows are unmixed.



Design Methods

- Different heat exchanger problems:
  - ✓ Type 1: m_c, m_h are known

    $T_{h,i}$, $T_{h,o}$, $T_{c,i}$, $T_{c,o}$ are known

    $A = ?$

    Appropriate method is ***LMTD*** method.
  - ✓ Type 2: U and A are known

    $T_{h,i}$, $T_{c,i}$ are known

    $T_{h,o}$ and $T_{c,o} = ?$

    Appropriate method is ***ε-NTU*** method.

Design Methods

- ε-NTU method (**E**ffectiveness-**N**umber of **T**ransfer **U**nit)

$$C = \dot{m}C_p$$

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

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

✓ Now we may define the heat transfer rate as:

$$q_{max} = C_{min}(T_{H,in} - T_{C,in})$$

✓ 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:

$$\varepsilon = \frac{Q}{q_{max}}$$

✓ If the cold fluid is the minimum fluid:

$$\varepsilon = \frac{C_h(T_{H,in} - T_{H,out})}{C_{min}(T_{H,in} - T_{C,in})}$$

✓ If the hot fluid is the minimum fluid:

$$\varepsilon = \frac{C_c(T_{C,out} - T_{C,in})}{C_{min}(T_{H,in} - T_{C,in})}$$

✓ By definition the effectiveness must be in the range $0 \le \varepsilon \le 1$. 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

$$q_{max} = \varepsilon C_{min}(T_{H,in} - T_{C,in})$$

Design Methods

- ε-NTU method (**E**ffectiveness-**N**umber of **T**ransfer **U**nit)
  - ✓ For any heat exchanger it can be shown that

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

  - ✓ NTU:

$$NTU = \frac{UA}{C_{min}}$$

  - ✓ Capacity ratio:

$$C_r = \frac{C_{min}}{C_{max}}$$

  - ✓ Note: The ε-NTU method can be used for the first type of problems as well. NTU is known as a function of effectiveness and capacity ratio:

$$NTU = f(\varepsilon, C_r)$$

- **Effectiveness-Number of Transfer Unit Relations**

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- **Effectiveness-Number of Transfer Unit Relations**

Design Methods

- ε-NTU method (**E**ffectiveness-**N**umber of **T**ransfer **U**nit)

  ✓ Double pipe parallel flow:

$$\varepsilon = \frac{1 - exp[-NTU(1 + C_r)]}{1 + C_r}$$

$$NTU = \frac{-\ln[1 - \epsilon(1 + C_r)]}{1 + C_r}$$

  ✓ Double pipe counter flow:

$$\varepsilon = \frac{1 - exp[-NTU(1 - C_r)]}{1 - C_r exp[-NTU(1 - C_r)]}, \qquad C_r < 1$$

$$\varepsilon = \frac{NTU}{1 + NTU}, \qquad C_r = 1$$

$$NTU = \frac{1}{C_r - 1} \ln\left(\frac{\varepsilon - 1}{\varepsilon C_r - 1}\right), \qquad C_r < 1$$

$$NTU = \frac{\varepsilon}{1 - \varepsilon}, \qquad C_r = 1$$

All exchangers ($C_r = 0$)

$$\varepsilon = 1 - \exp(-\mathrm{NTU})$$

Design Methods

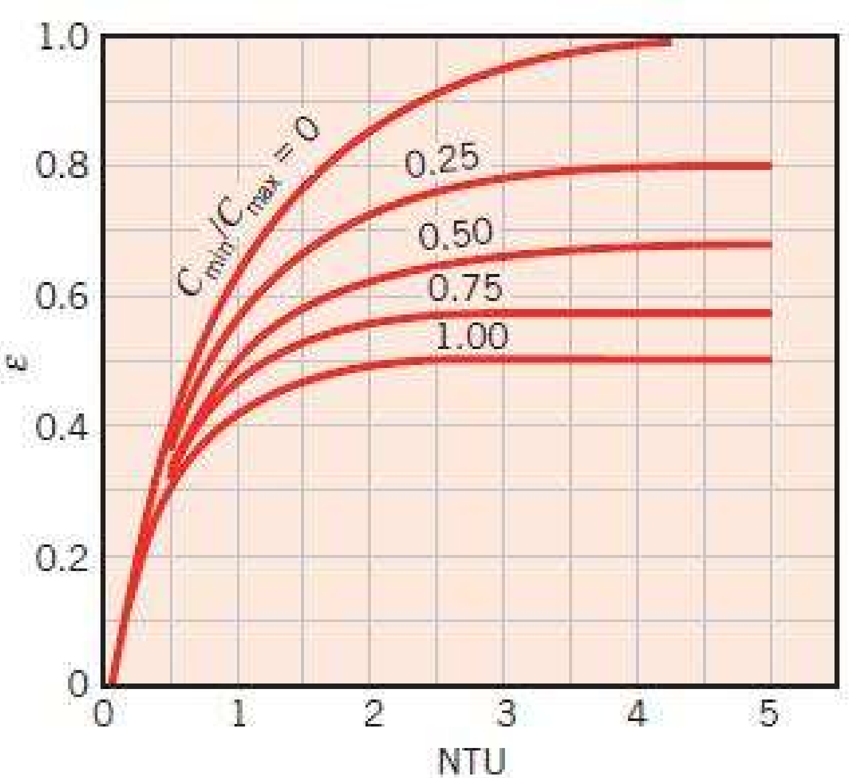
- ε-NTU method (**E**ffectiveness-**N**umber of **T**ransfer **U**nit)

Shell-and-tube	
One shell pass (2, 4, ... tube passes)	$(\mathrm{NTU})_1 = -(1 + C_r^2)^{-1/2} \ln\left(\frac{E-1}{E+1}\right)$ (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 $\varepsilon_1 = \frac{F-1}{F-C_r} \quad F = \left(\frac{\varepsilon C_r - 1}{\varepsilon - 1}\right)^{1/n} \quad \mathrm{NTU} = n(\mathrm{NTU})_1$ (11.31b, c, d)
Cross-flow (single pass)	
C_{max} (mixed), C_{min} (unmixed)	$\mathrm{NTU} = -\ln\left[1 + \left(\frac{1}{C_r}\right)\ln(1 - \varepsilon C_r)\right]$ (11.33b)
C_{min} (mixed), C_{max} (unmixed)	$\mathrm{NTU} = -\left(\frac{1}{C_r}\right)\ln[C_r \ln(1 - \varepsilon) + 1]$ (11.34b)

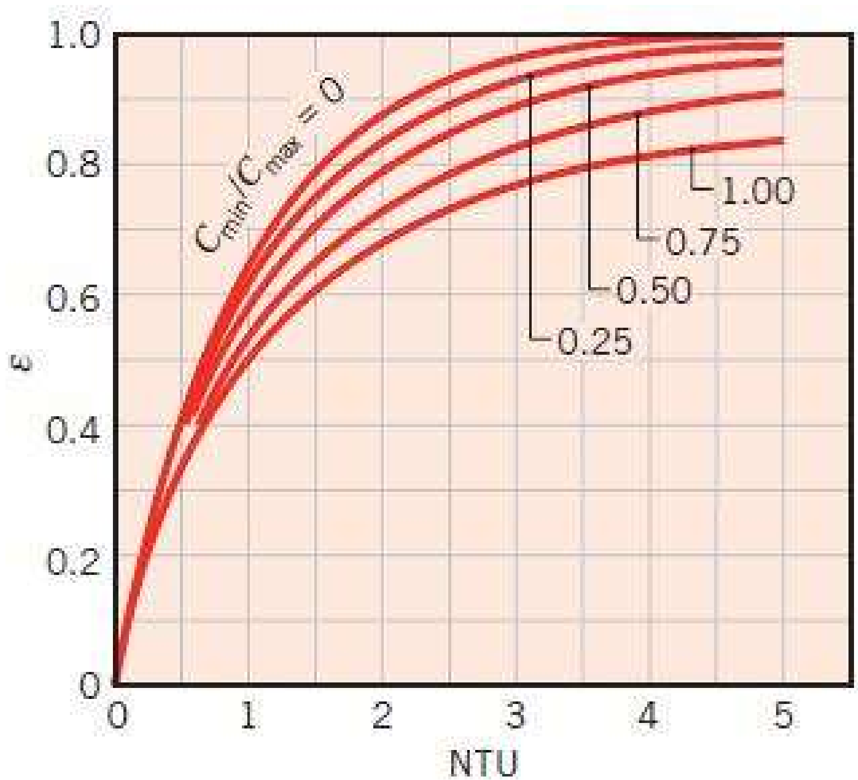
Design Methods

- ε-NTU method (**E**ffectiveness-**N**umber of **T**ransfer **U**nit)

Effectiveness of a parallel-flow heat exchanger



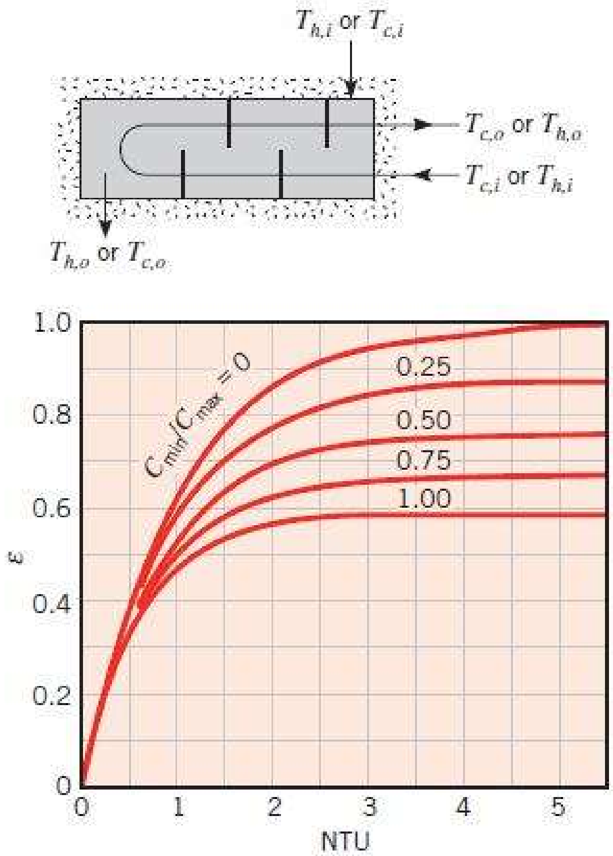
Effectiveness of a counterflow heat exchanger



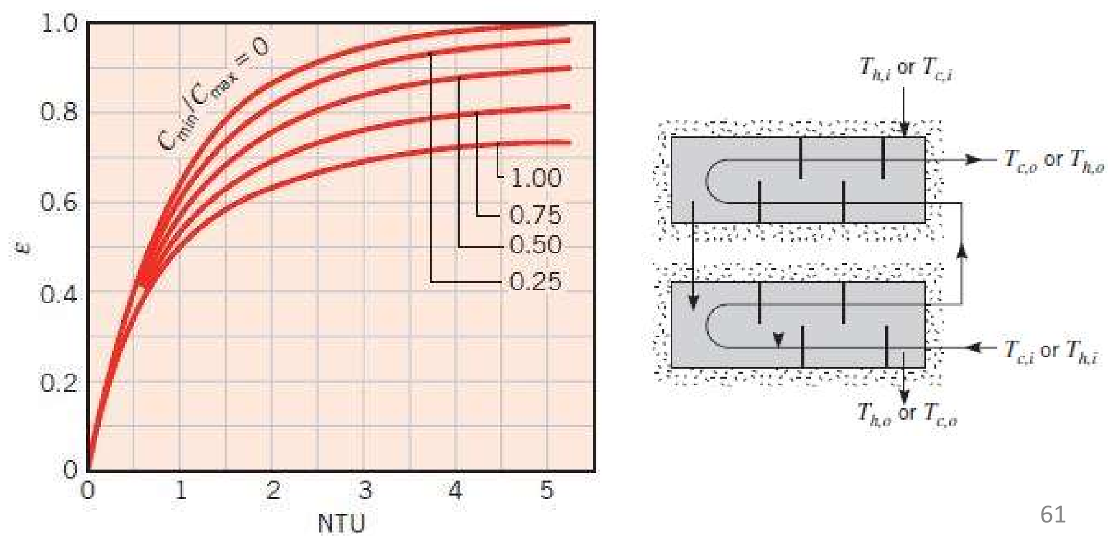
Design Methods

- ε-NTU method (**E**ffectiveness-**N**umber of **T**ransfer **U**nit)

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



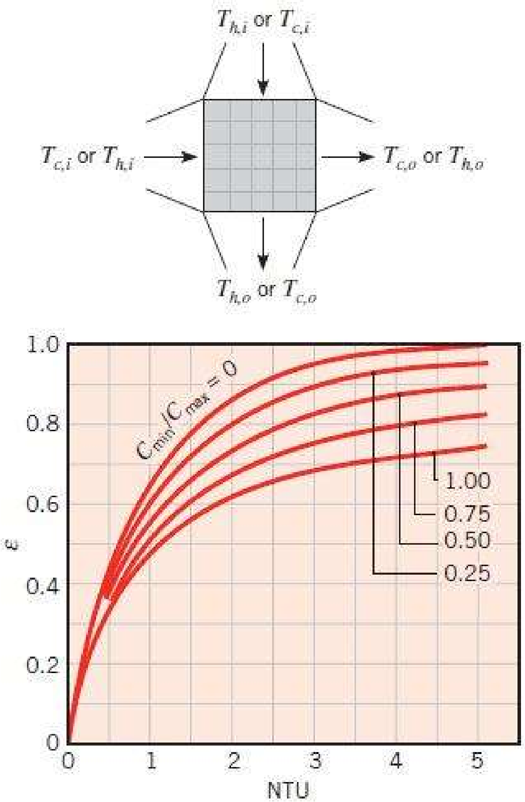
Effectiveness of a shell-and-tube heat exchanger with two shell passes and any multiple of four tube passes (four, eight, etc. tube passes)



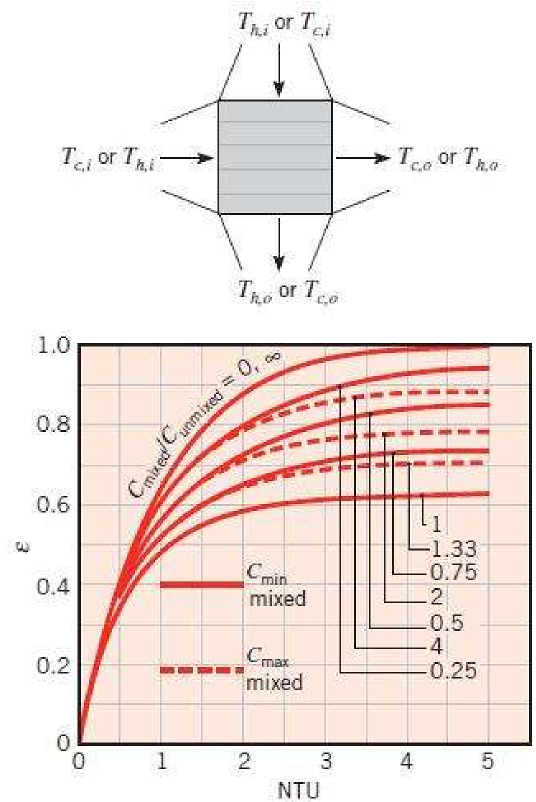
Design Methods

- ε-NTU method (**E**ffectiveness-**N**umber of **T**ransfer **U**nit)

Effectiveness of a single-pass, cross-flow heat exchanger with both fluids unmixed



Effectiveness of a single-pass, cross-flow heat exchanger with one fluid mixed and the other unmixed



Resolve Example 5.1 (use ε-NTU)

Oil flows in a heat exchanger with a mass flow rate of 20 kg/s and is to be cooled from $T_{h,i}$=120 °C to $T_{h,o}$=60 °C with water as a coolant flowing at a rate of 15 kg/s and an inlet temperature $T_{c,i}$=10 °C. If the total heat transfer coefficient is U=1100 W/m^{2}K determine the heat transfer area required for:

a) Parallel flow
b) Counter flow

$c_{p,oil}$=2000 J/kgK, $c_{p,water}$=4000 J/kgK

Oil flows in a heat exchanger with a mass flow rate of 20 (kg/s) and is to be cooled from $T_{h,i}$=120 (°C) to $T_{h,o}$=60 (°C) with water as a coolant flowing at a rate of 15 (kg/s) and an inlet temperature $T_{c,i}$=10 (°C). If the total heat transfer coefficient is U=1100 (W/m^2K) determine the heat transfer area required for:

a) Parallel flow

b) Counter flow

$c_{p,oil}$=2000 (J/kgK), $c_{p,water}$=4000 (J/kgK)

Oil flows in a heat exchanger with a mass flow rate of 20 (kg/s) and is to be cooled from $T_{h,i}$=120 (°C) to $T_{h,o}$=60 (°C) with water as a coolant flowing at a rate of 15 (kg/s) and an inlet temperature $T_{c,i}$=10 (°C). If the total heat transfer coefficient is U=1100 (W/m^{2}K) determine the heat transfer area required for:

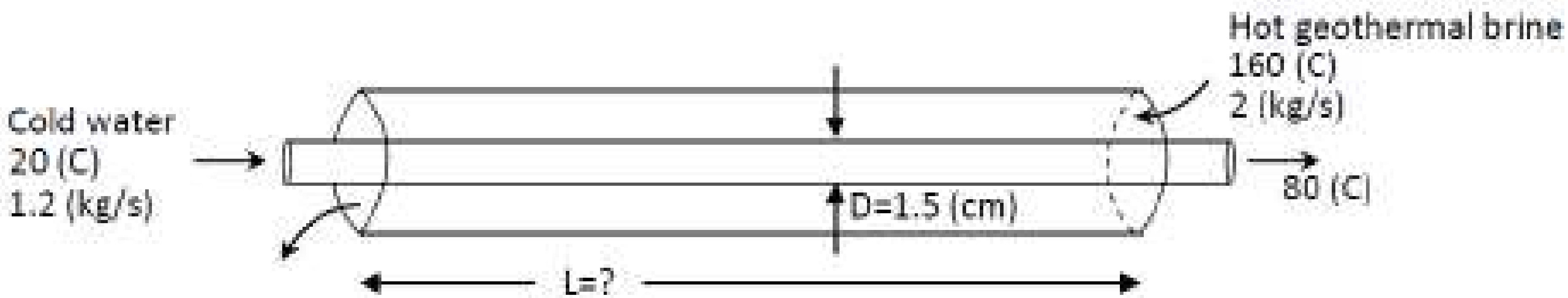
a) Parallel flow

b) Counter flow

$c_{p,oil}$=2000 (J/kgK), $c_{p,water}$=4000 (J/kgK)

Example 5.3

A counter flow double pipe heat exchanger is to heat water from 20 (°C) to 80 (°C) at a rate of 1.2 (kg/s). The heating is to be accomplished by geothermal water available at 160 (°C) at a mass flow rate of 2 (kg/s). The inner tube is thin walled and has a diameter of 1.5 (cm). The overall heat transfer coefficient of the heat exchanger is 640 (W/m^2K). Using the ε-NTU method determine the length of the heat exchanger required to achieve the desired heating. Use $c_{p,h}$=4.31 (kJ/kgK) and $c_{p,c}$=4.18 (kJ/kgK).



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Problem (d)

A shell-and-tube exchanger (two shells, four tube passes) is used to heat 10,000 kg/h of pressurized water from 35 to 120°C with 5000 kg/h pressurized water entering the exchanger at 300°C. If the overall heat transfer coefficient is 1500 $W/m^2 \cdot K$, determine the required heat exchanger area.

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Heat Exchanger Pressure Drop

- Importance of pressure drop in heat exchangers
  - ✓ Determining the required pumping power
  - ✓ Its effects on the heat exchanger heat transfer

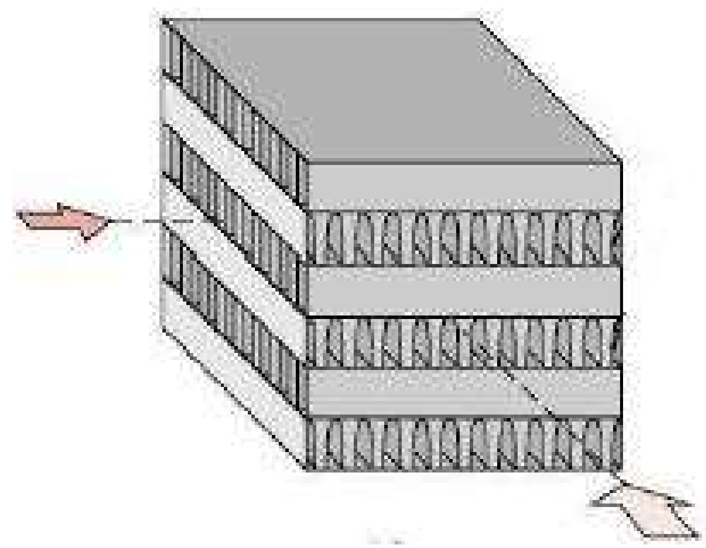
- Pressure drop is affected by:
  - ✓ Type of flow (laminar or turbulent)
  - ✓ Passage geometry
  - ✓ …

Heat Exchanger Pressure Drop

- Major contributions to the pressure drop in a plate-fin heat exchanger:

  ✓ Pressure drop in heat exchanger core
    - Friction losses
    - Momentum effects (changes in density)
    - Gravity effects, etc.

  ✓ Pressure drop in flow distribution devices
    - Inlet/outlet headers
    - Manifolds
    - Tanks
    - Nozzles
    - Ducting, etc.



Heat Exchanger Pressure Drop

- Core pressure drop:
  - ✓ Friction pressure drop

$$\Delta p_c = \frac{4fL}{D_h}\frac{1}{2}\frac{G^2}{\rho_m}$$

where f is Fanning friction factor, G is the mass flux and ρ_m is the mean density

$$G = \frac{\dot{m}}{A} \qquad v_m = \frac{v_i + v_o}{2} \rightarrow \frac{1}{\rho_m} = \frac{1}{2}\left(\frac{1}{\rho_i} + \frac{1}{\rho_o}\right)$$

  - ✓ Acceleration/deceleration pressure drop due to change in fluid density

$$\Delta p_a = G^2\left(\frac{1}{\rho_e} - \frac{1}{\rho_i}\right)$$

Heat Exchanger Pressure Drop

- Entrance pressure drop:

$$\Delta p_i = \left(1 - \sigma_i^2 + K_c\right)\frac{1}{2}\frac{G^2}{\rho_i}$$

where σ is the passage contraction ratio, G is the mass flux and K_c is the contraction *K* factor:

$$K_c \approx 0.42(1 - \sigma^2)^2$$

- Exit pressure drop:

$$\Delta p_e = -(1 - \sigma_e^2 + K_e)\frac{1}{2}\frac{G^2}{\rho_e}$$

where K_e is the expansion K factor:

$$K_e = (1 - \sigma)^2$$

Heat Exchanger Pressure Drop

- Total pressure drop:

$$\Delta p = \Delta p_i + \Delta p_c + \Delta p_a + \Delta p_e$$

$$\Delta p = \frac{G^2}{2\rho_i}\left[\left(1-\sigma_i^2+K_c\right)+f\frac{4L}{D_h}\left(\frac{\rho_i}{\rho_e}\right)+2\left(\frac{\rho_i}{\rho_e}-1\right)-(1-\sigma_e^2-K_e)\left(\frac{\rho_i}{\rho_e}\right)\right]$$

- The pumping power:

$$\dot{W}_p = \frac{1}{\eta_p}\frac{\dot{m}}{\rho}\Delta P$$

where η_p is the pump efficiency.

Heat Exchanger Pressure Drop

- Major contributions to the pressure drop in plate heat exchangers:
  - ✓ Inlet and outlet manifolds and ports
  - ✓ Core (plate passages)
    - o Friction pressure drop
    - o Momentum pressure drop (change in density)
  - ✓ Elevation change pressure drop for vertical flow exchangers

Heat Exchanger Pressure Drop

- Major contributions to the pressure drop in plate heat exchangers:

$$\Delta P = 1.5 n_p \left(\frac{1}{2}\frac{G_p^2}{\rho_i}\right) + 4f\frac{L}{D_e}\left(\frac{1}{2}\frac{G^2}{\rho_m}\right) + \left(\frac{1}{\rho_o} - \frac{1}{\rho_i}\right)G^2 \pm \rho_m g L$$

where G_p is the mass flux in port

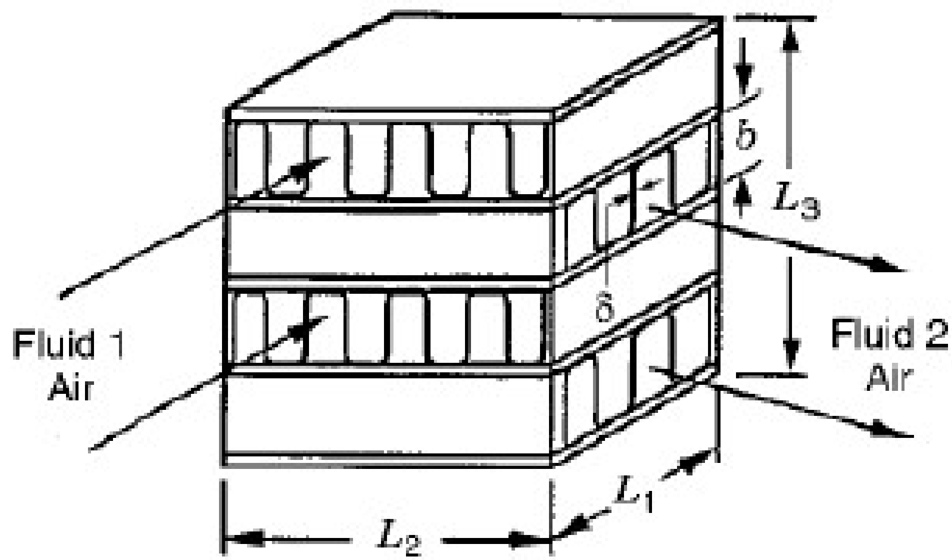
$$G_p = \frac{\dot{m}}{\left(\frac{\pi}{4}\right) D_p^2}$$

n_p is the number of passes on a given fluid side
D_e is passage equivalent diameter (usually twice the plate spacing)
ρ_i is the fluid density at inlet
ρ_o is the fluid density at outlet

Example 5.4

A gas to air single pass cross flow plate-fin heat exchanger has overall dimensions of 0.30 m × 0.60 m × 0.90 m and employs strip fins on the air side. The following information is provided for the air side. Determine the air side pressure drop (fluid 2 in the figure).

D_h=0.002383 (m)
f=0.0683
Q_i=0.6 (m^3/s)
A=0.1177 (m^2)
R=287.041 (J/kg.K)
P_i=110 (kPa)
T_i=4 (°C)
T_o=194.5 (°C)
σ=0.437



Example 5.4

Example 5.4

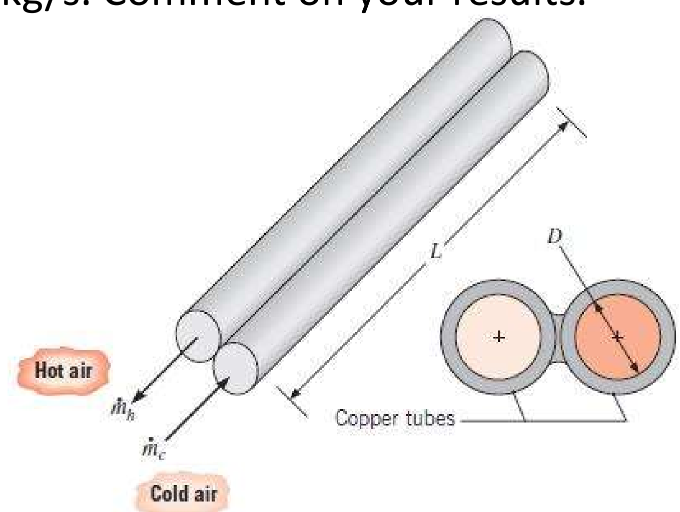
Example 5.4

Example 5.4

Example (e)

A twin-tube, counterflow heat exchanger operates with balanced flow rates of 0.003 kg/s for the hot and cold airstreams. The cold stream enters at 280 K and must be heated to 340 K using hot air at 360 K. The average pressure of the airstreams is 1 atm and the maximum allowable pressure drop for the cold air is 10 kPa. The tube walls may be assumed to act as fins, each with an efficiency of 100%.

(a) Determine the tube diameter D and length L that satisfy the prescribed heat transfer and pressure drop requirements.
(b) For the diameter D and length L found in part (a), generate plots of the cold stream outlet temperature, the heat transfer rate, and pressure drop as a function of balanced flow rates in the range from 0.002 to 0.004 kg/s. Comment on your results.



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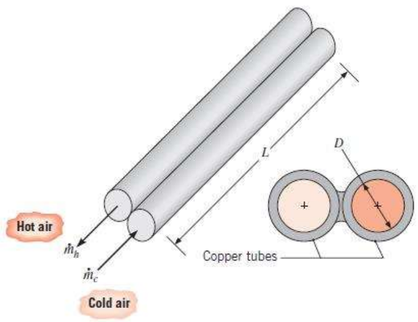

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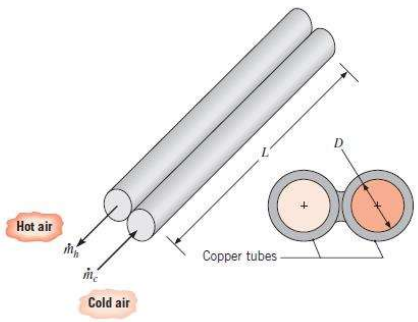
A twin-tube, counterflow heat exchanger operates with balanced flow rates of 0.003 kg/s for the hot and cold airstreams. The cold stream enters at 280 K and must be heated to 340 K using hot air at 360 K. The average pressure of the airstreams is 1 atm and the maximum allowable pressure drop for the cold air is 10 kPa. The tube walls may be assumed to act as fins, each with an efficiency of 100%.

(a) Determine the tube diameter D and length L that satisfy the prescribed heat transfer and pressure drop requirements.

(b) For the diameter D and length L found in part (a), generate plots of the cold stream outlet temperature, the heat transfer rate, and pressure drop as a function of balanced flow rates in the range from 0.002 to 0.004 kg/s. Comment on your results.



A twin-tube, counterflow heat exchanger operates with balanced flow rates of 0.003 kg/s for the hot and cold airstreams. The cold stream enters at 280 K and must be heated to 340 K using hot air at 360 K. The average pressure of the airstreams is 1 atm and the maximum allowable pressure drop for the cold air is 10 kPa. The tube walls may be assumed to act as fins, each with an efficiency of 100%.

(a) Determine the tube diameter D and length L that satisfy the prescribed heat transfer and pressure drop requirements.
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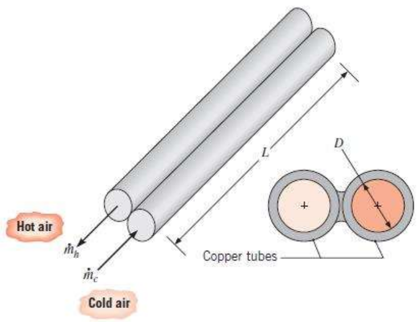

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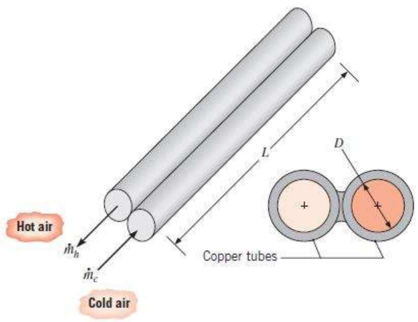

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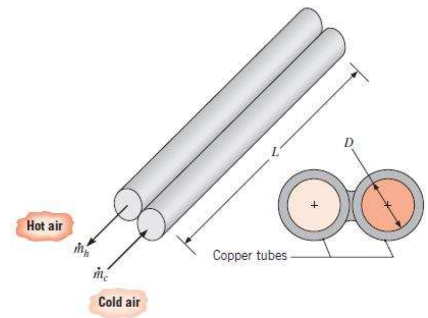

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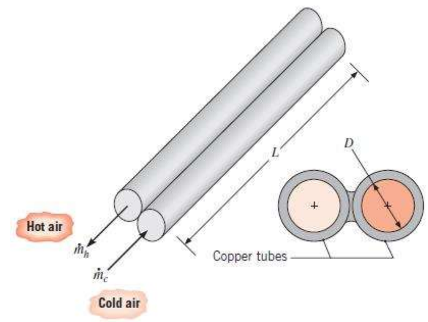
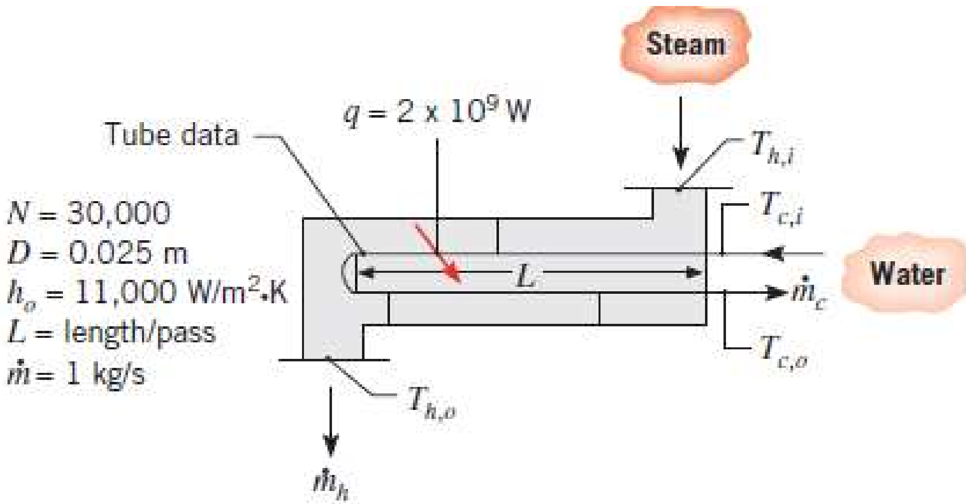
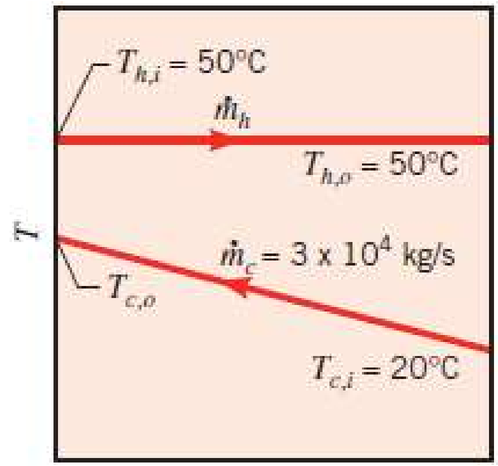
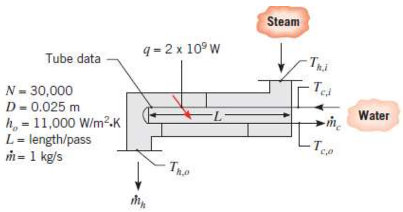


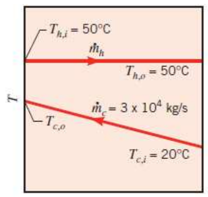
Example (g)

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/m^2·K. The heat transfer rate that must be effected by the exchanger is $q = 2 \times 10^9$ 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?









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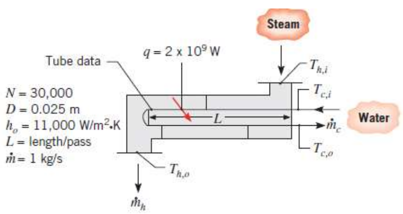

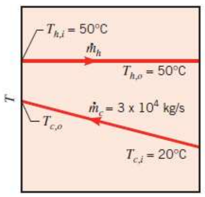

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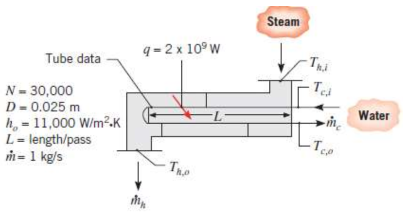

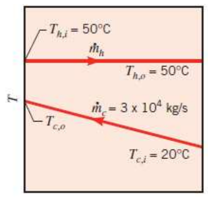

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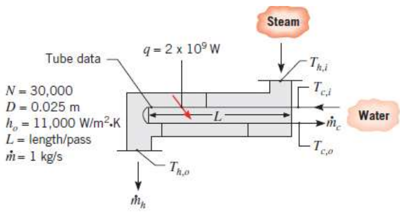

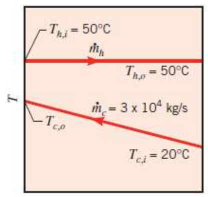

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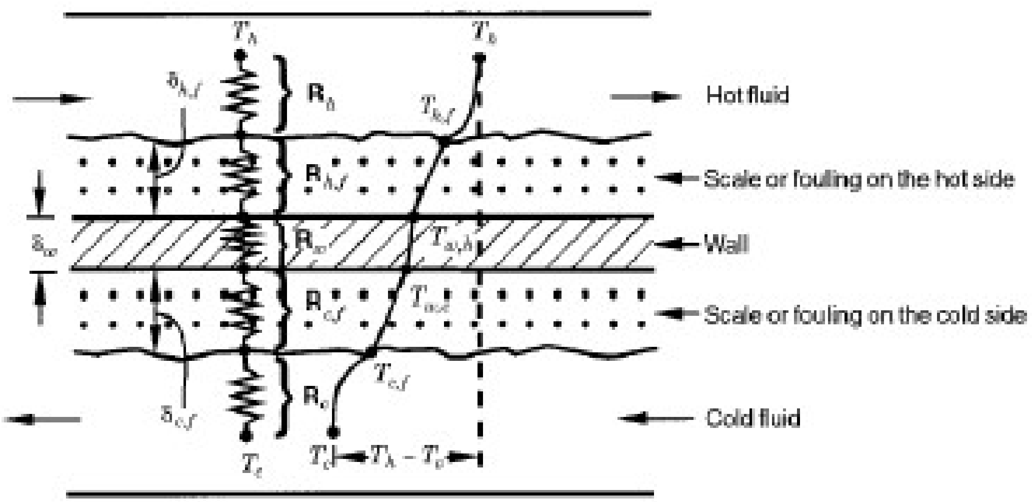
https://www.youtube.com/watch?v=vELQeiQ67Fo

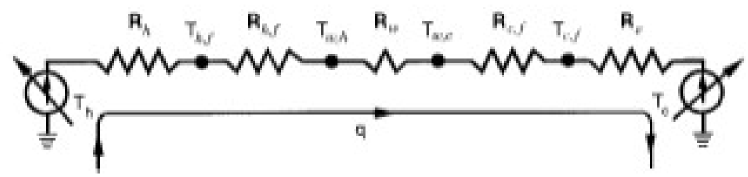
Fouling of Heat Exchangers

- Drawbacks:
  - ✓ Increase hydraulic resistance
    - Increase in the roughness
    - Decrease in flow area
  - ✓ Decrease in thermal performance
    - Increase thermal resistance
    - Less mass flow rate and heat transfer capacity
  - ✓ Corrosion
- So heat exchangers should be cleaned on a regular basis.

Fouling of Heat Exchangers

- Effects on heat transfer:
  - ✓ Fouling increases the thermal resistance of the heat exchanger and decreases the thermal efficiency.
  - ✓ In the case of fouling on both sides, there are two more thermal resistances in the thermal circuit.





Fouling of Heat Exchangers

- Overall heat transfer coefficient after fouling:

$$\frac{1}{UA} = \frac{1}{\eta_i h_i A_i} + R_{f,i} + \frac{1}{Sk_w} + R_{f,o} + \frac{1}{\eta_o h_o A_o}$$

where the fouling resistance for a plane wall is:

$$R_f = \frac{t_f}{k_f A_w}$$

and for a tube: (fouling outside the tube)

$$R_f = \frac{\ln\left(d_f/d_c\right)}{2\pi k_f L}$$

for a tube: (fouling inside the tube)

$$R_f = \frac{\ln\left(d_c/d_f\right)}{2\pi k_f L}$$

Fouling of Heat Exchangers

Table 3 - TEMA Design Fouling Resistances R_f
for a Number of Industrial Fluids

Fluid	$R''_f = R_f A \; [m^2 K/kW]$
Engine Oil	0.176
Fuel Oil no.2	0.352
Fuel Oil no.6	0.881
Quench Oil	0.705
Refrigerants	0.176
Hydraulic FLuids	0.176
Ammonia Liquids	0.176
Ethylene Glycol Solutions	0.352
Exhaust Gases	1.761
Natural Gas Flue Gases	0.881
Coal Flue Gases	1.761

Fouling of Heat Exchangers

- Design for fouling:
  - ✓ Fouling resistance
    - o Prescribing a fouling resistance (fouling factor) on each side of the surface where fouling is anticipated.
  - ✓ Cleanliness Factor
    - o Cleanliness factor (CF) relates the fouled overall heat transfer coefficient to the clean coefficient.
  - ✓ Percentage over surface
    - o Some added heat transfer surface area is considered initially.

- We examine the “percentage over surface” method in this course.

Example 5.5

A double pipe heat exchanger is used to condense steam at a rate of 120 kg/h at 45 °C. Cooling water (seawater) enters through the inner tube at a rate of 1.2 kg/s at 15 °C. The tube outer diameter is 25.4 mm and its inner diameter is 22.1 mm. If the overall heat transfer coefficient based on outer area is 3600 $W/m^2.K$ when the exchanger is clean, determine:

a) Outlet temperature of the cold fluid

b) ΔT_{LMTD}

c) Heat exchanger length

d) Fouling resistances on both sides for fouling thickness of 1 mm. For this part assume $k_{f,i}$=15 W/m.K and $k_{f,o}$=10.5 W/m.K

e) Overall heat transfer rate after 1(mm) of fouling on both sides

f) Over-surface percentage required to meet the desired thermal performance after fouling

In this problem assume: h_{fg}=2392 kJ/kg for steam at 45 °C and $c_{p,c}$=4180 J/kg.K for water at 15 °C.

A double pipe heat exchanger is used to condense steam at a rate of 120 kg/h at 45 °C. Cooling water (seawater) enters through the inner tube at a rate of 1.2 kg/s at 15 °C. The tube outer diameter is 25.4 mm and its inner diameter is 22.1 mm. If the overall heat transfer coefficient based on outer area is 3600 W/m².K when the exchanger is clean, determine:

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Typical Heat Exchanger Design

- Double pipe heat exchangers:
  - ✓ Overall heat transfer coefficient:

$$\frac{1}{UA} = \frac{1}{h_i(2\pi r_i L)} + \frac{\ln(r_o/r_i)}{2\pi k_w L} + \frac{1}{h_o(2\pi r_o L)}$$

where:

r_i: inner pipe inner radius

r_o: inner pipe outer radius

h_i: pipe heat transfer coefficient

h_o: annulus heat transfer coefficient

k_w: inner wall conductivity

  - ✓ Pressure drop for each fluid

$$\Delta p = \left[\sum \frac{4fL}{D_h} + \sum K\right]\frac{1}{2}\rho V^2$$

Typical Heat Exchanger Design

- Double pipe heat exchangers:
  - ✓ Heat transfer coefficient for internal flow forced convection: (for example for $0.6 < Pr < 160$, $Re > 10000$)

$$Nu = 0.023 Re^{0.8} Pr^{1/3}$$

where:

Nu: Nusselt number

Re: Reynolds number based on hydraulic diameter

Pr: Prandtl number

$$Nu = \frac{hD_h}{k_f}$$

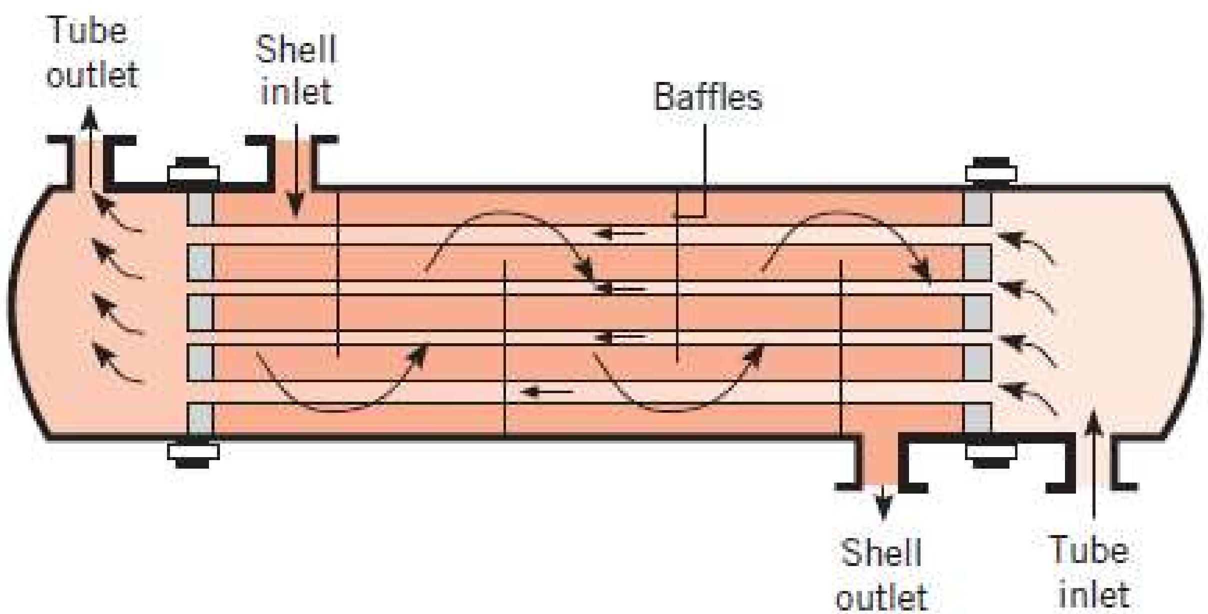
where:

h: heat transfer coefficient

k_f: fluid thermal conductivity

Typical Heat Exchanger Design

- Shell and tube heat exchangers:



✓ Flow types:
  - Parallel flow
  - Counter flow
  - Cross flow (or a combination of these due to baffles)

✓ Design process is according to TEMA standards.

Typical Heat Exchanger Design

- Shell and tube heat exchangers:
  - ✓ Overall heat transfer coefficient:

$$\frac{1}{UA} = \frac{1}{h_i A_i} + R_w + \frac{1}{h_o A_o}$$

where:
A_i: inner area of tubes
A_o: outer are of tubes
h_i: heat transfer coefficient at tube side
h_o: heat transfer coefficient at shell side

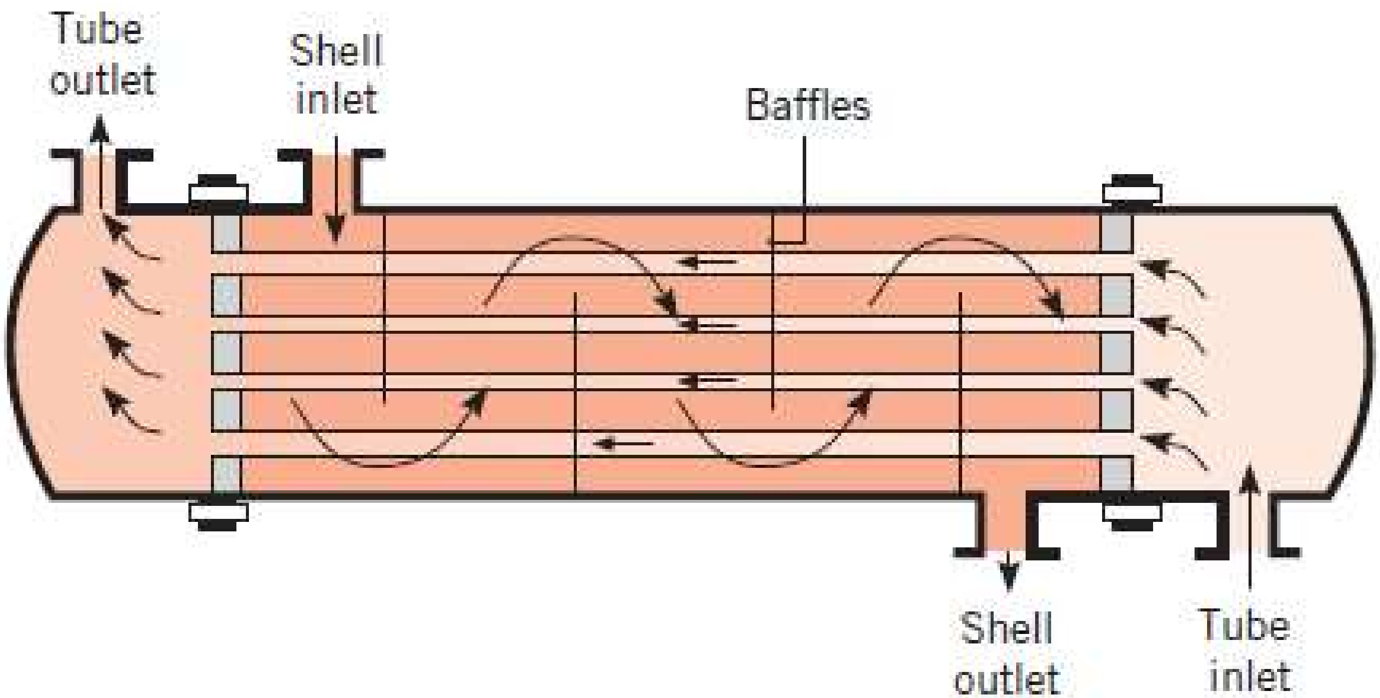
  - ✓ Heat transfer surface area:

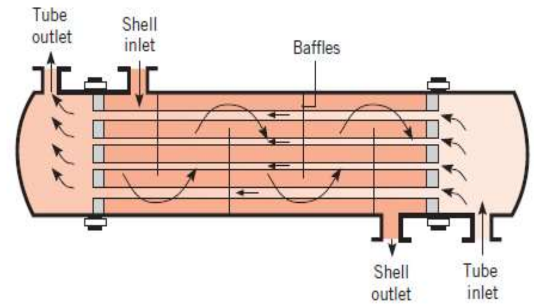
$$A_o = \pi d_o N_t L$$

where:
d_o: outer diameter of tubes
N_t: number of tubes
L: length of tubes

Typical Heat Exchanger Design

Analysis




Tube
outlet
Shell
inlet
Baffles
Shell
outlet
Tube
inlet

Typical Heat Exchanger Design

- Shell and tube heat exchangers:
  - ✓ Number of tubes:

$$N_t = CTP \frac{\pi D_h^2 1}{4 CL P_t^2}$$

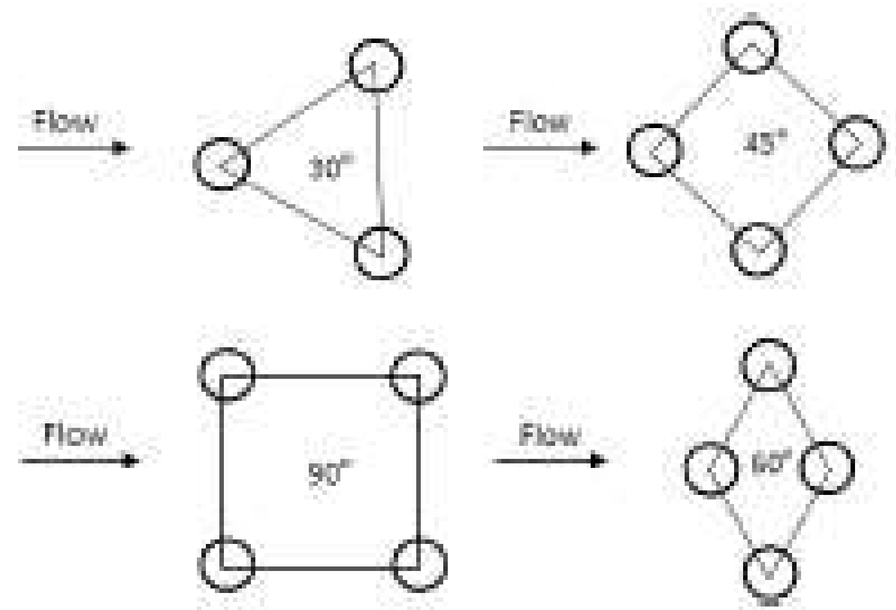
where:

CTP (tube coverage in the shell) = 0.93 (for one tube pass), 0.9 (for two tube passes), 0.85 (three tube passes)

CL (tube layout constant) = 1 (for 45° and 90°), 0.87 (for 30° and 60°)

P_t: tube pitch

D_s: shell diameter



Typical Heat Exchanger Design

- Shell and tube heat exchangers:
  - ✓ Solving for shell diameter we have:

$$D_s = 0.637\sqrt{\frac{CL}{CTP}}\left[\frac{A_o P_t^2}{d_o L}\right]^{1/2}$$

  - ✓ Shell side heat transfer coefficient:

$$Nu = 0.36 Re_{D_e}^{0.55} Pr^{1/3}$$

  - ✓ The effective diameter:
    - For square tube arrangement:

$$D_e = \frac{4(P_t^2 - \pi d_o^2/4)}{\pi d_o}$$

    - For triangular tube arrangement:

$$D_e = \frac{8\left(\sqrt{3}P_t^2/4 - \pi d_o^2/8\right)}{\pi d_o}$$

Example 5.6

A heat exchanger is to be designed to heat raw water by the use of condensed water at 67 ^oC and 0.2 bar (c_p=4179 J/kg.K), which will flow in the shell side with a mass flow rate of 50000 kg/h. The heat will be transferred to 30000 kg/h of city water coming from a supply at 17 ^oC (c_p=4184 J/kg.K). A parallel flow single pass-single pass shell and tube heat exchanger is preferable. A maximum coolant velocity of 1.5 m/s is suggested to prevent erosion. A maximum tube length of 5 (m) is required because of space limitations. The tube material is carbon steel (k=60 W/m.K). Raw water will flow inside of straight tubes with ID=16 mm and OD=19 mm. Tubes are laid out on a square pitch with a pitch ratio of 1.25 (P_t=1.25×OD). Water outlet temperature should not be less than 40 ^oC. Perform the preliminary analysis. Shell side and tube side heat transfer coefficients can be assumed as 5000 W/m^2.K and 4000 W/m^2.K.

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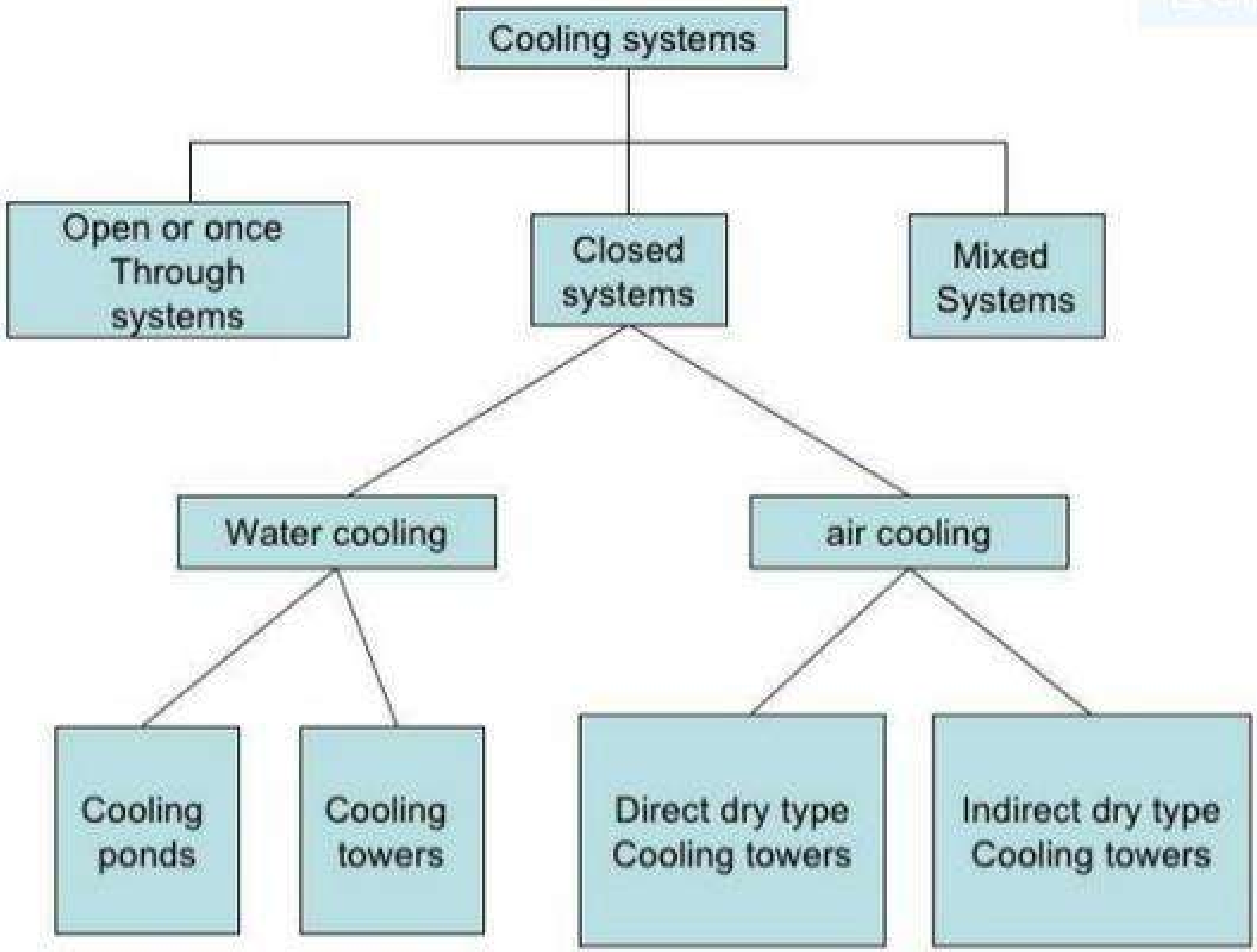
Cooling Equipment



https://www.slideshare.net/ssuser846488/ppt-for-power-plant

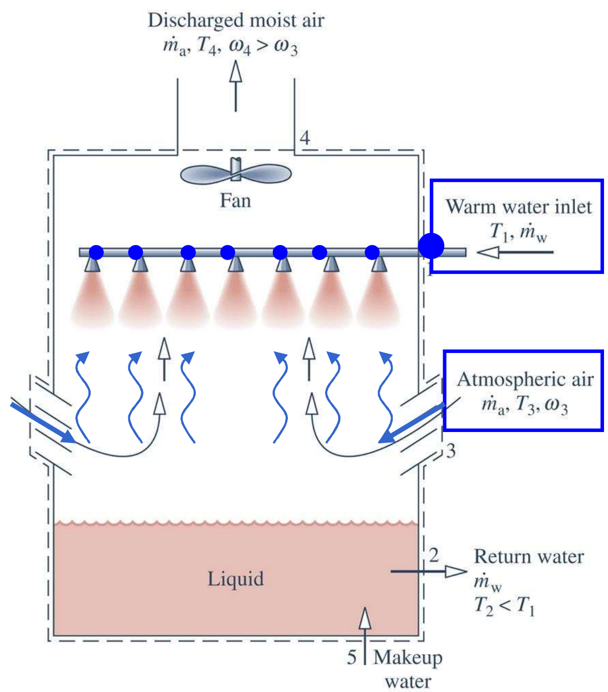
Introduction

Components

- Air inlet – serves as an entry point for air
- Drift eliminators – designed to capture droplets in the air stream
- Louvers – to sustain air flow and retain water within the cooling unit
- Nozzles – used to spray water
- Fans – maintain airflow within the tower

Cooling Towers (1 of 3)

►Moist air principles also play a role in the analysis of ***cooling towers*** such as shown in the figure.



►Major events occurring within the control volume enclosing the tower include the following:

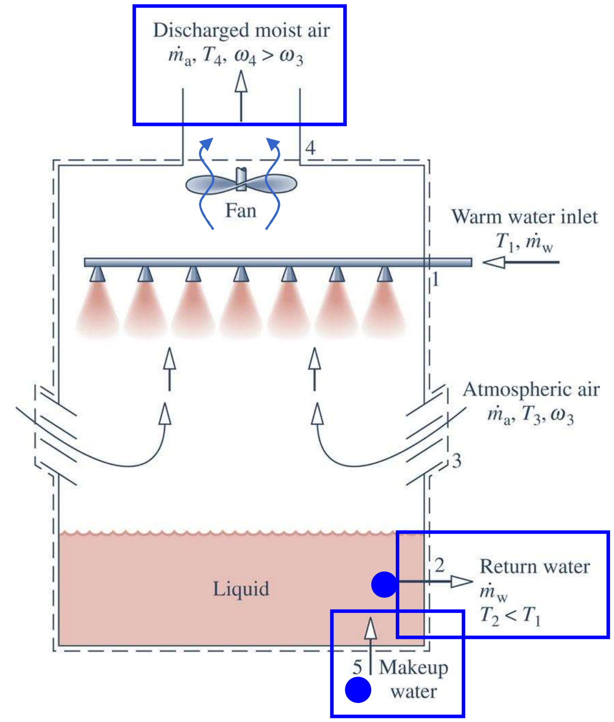
►The warm water to be cooled enters at **1** and is sprayed from the top of the tower.

►Atmospheric air enters at **3** and flows counter to the falling water.

Cooling Towers (2 of 3)

► As the liquid water and moist air interact within the tower, a fraction of the liquid evaporates, resulting in

- Liquid water that exits the tower at **2** with a lower temperature than the water entering at **1**, which is **the objective**.
- Moist air that exits the tower at **4** with a greater humidity ratio than the air entering at **3**.



► Since some of the incoming water has evaporated, an equivalent amount of makeup water is added at **5** so that the return mass flow rate equals the mass flow rate entering at **1**.

Cooling Towers (3 of 3)

►**Mass rate balances**. To evaluate the mass flow rate of the makeup water, apply mass rate balances to the control volume at steady state to get

$$\dot{m}_{a3} = \dot{m}_{a4} \qquad \text{(dry air)}$$

$$\cancel{\dot{m}_{w1}} + \dot{m}_5 + \dot{m}_{v3} = \cancel{\dot{m}_{w2}} + \dot{m}_{v4} \qquad \text{(water)}$$

$$\dot{m}_5 = \dot{m}_{v4} - \dot{m}_{v3}$$

With $\dot{m}_{v3} = \omega_3 \dot{m}_a$ and $\dot{m}_{v4} = \omega_4 \dot{m}_a$, where $\dot{m}_a$ is the common mass flow rate of the dry air, this becomes

$$\dot{m}_5 = \dot{m}_a(\omega_4 - \omega_3)$$

►**Energy rate balance**.

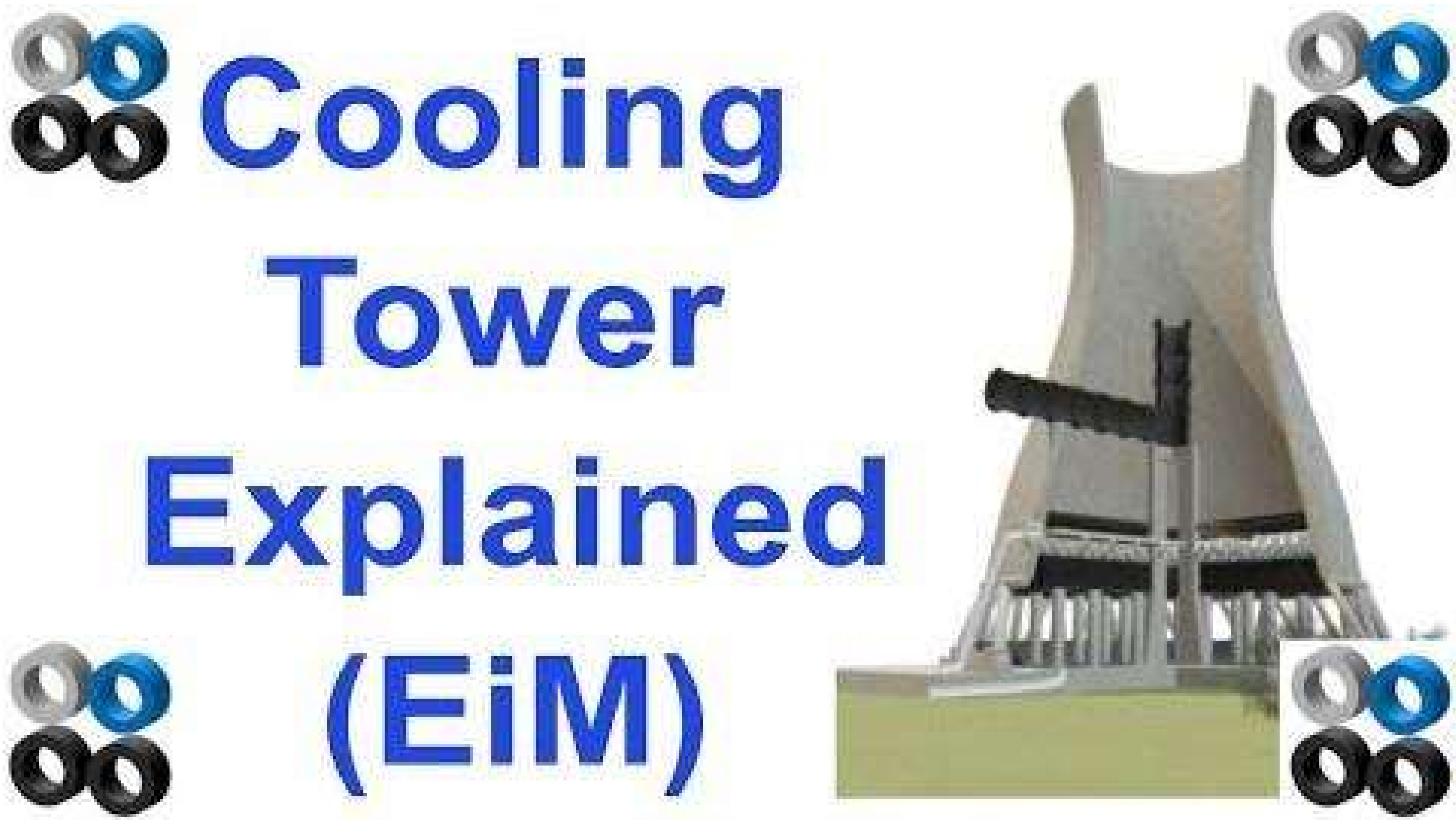
Draft System

Draft system

- Defined as the circulation of air due to pressure difference
- Comes in two main types:
  - Natural draft
  - Mechanical draft

Draft System

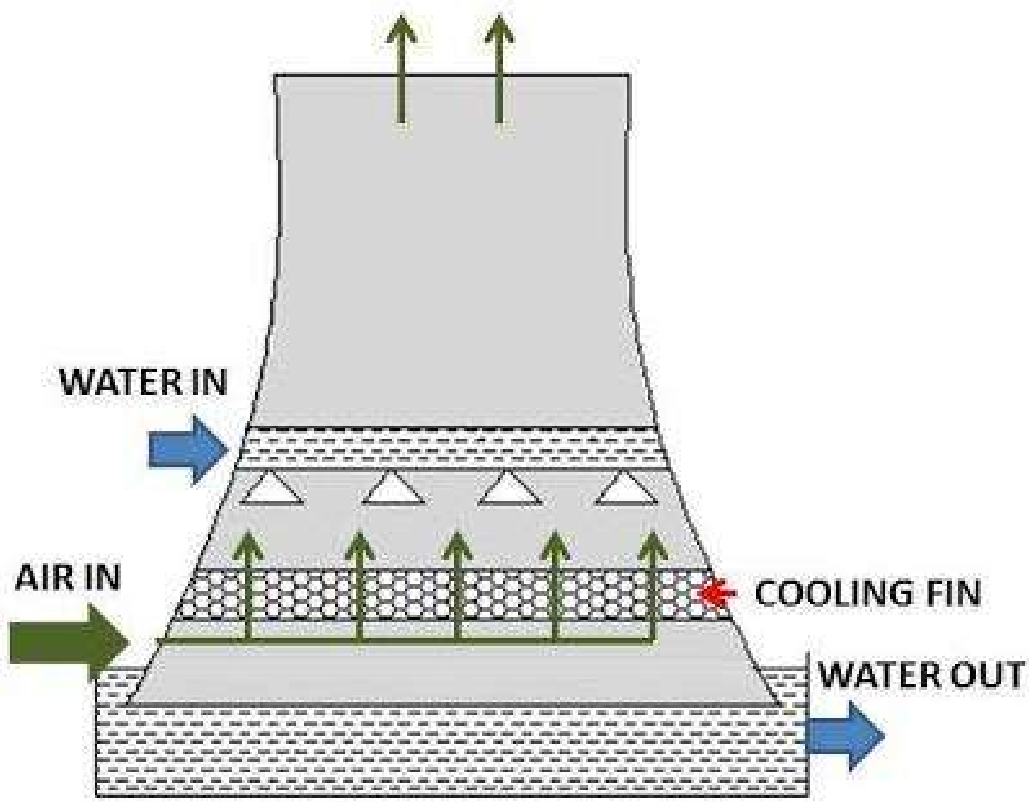
Natural draft



https://www.youtube.com/watch?v=5zQTtfDOLL0

Draft System

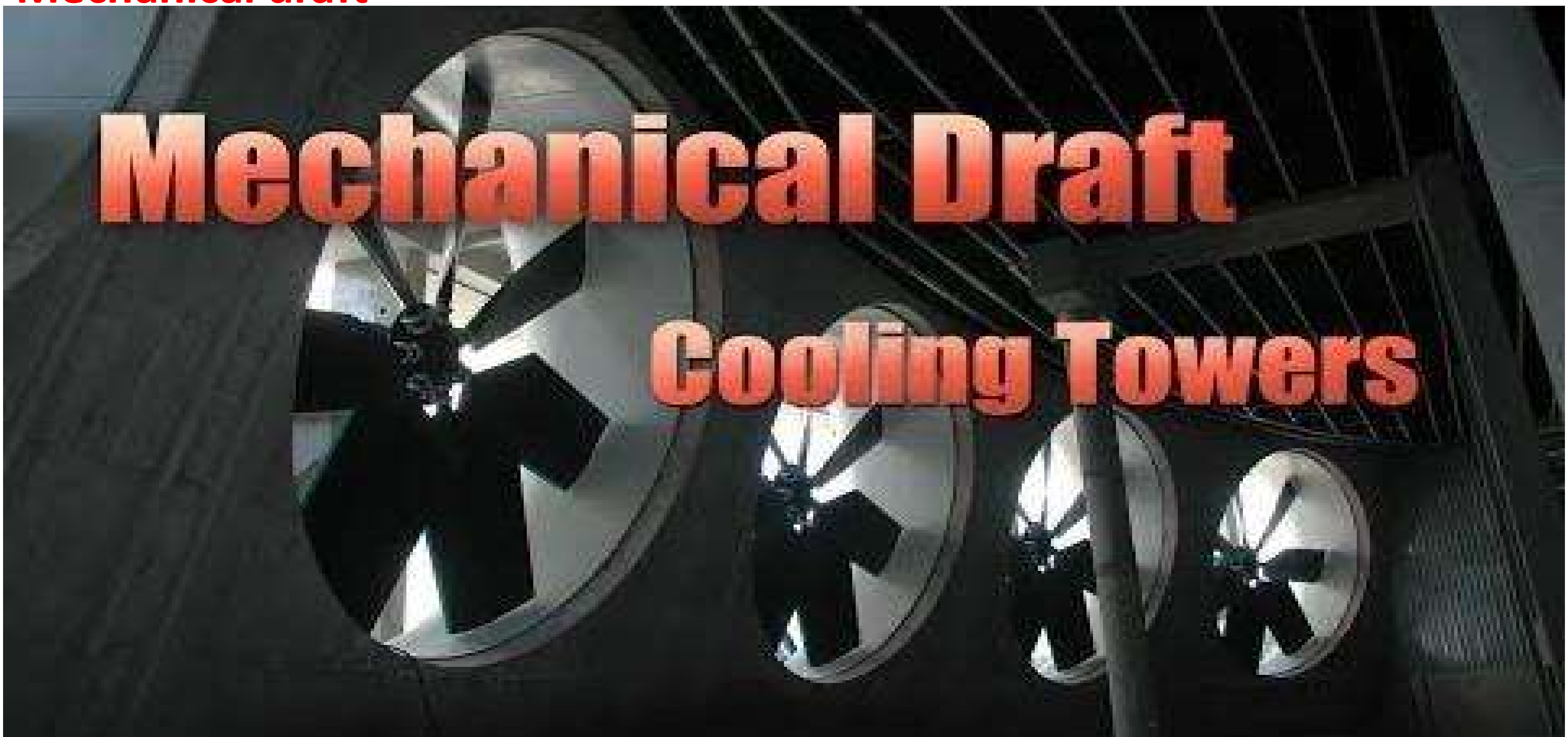
Natural draft cooling towers



https://www.mech4study.com/2017/12/what-is-cooling-tower-what-are-main-types-of-cooling-tower.html

Draft System

Mechanical draft



https://www.youtube.com/watch?v=jkbxsVemGec

Draft System

Mechanical draft cooling towers

- Large fans are used to maintain air circulation
- Heat transfer occurs as water falls over the surfaces
- Effective cooling rate depends on many parameters

Draft System

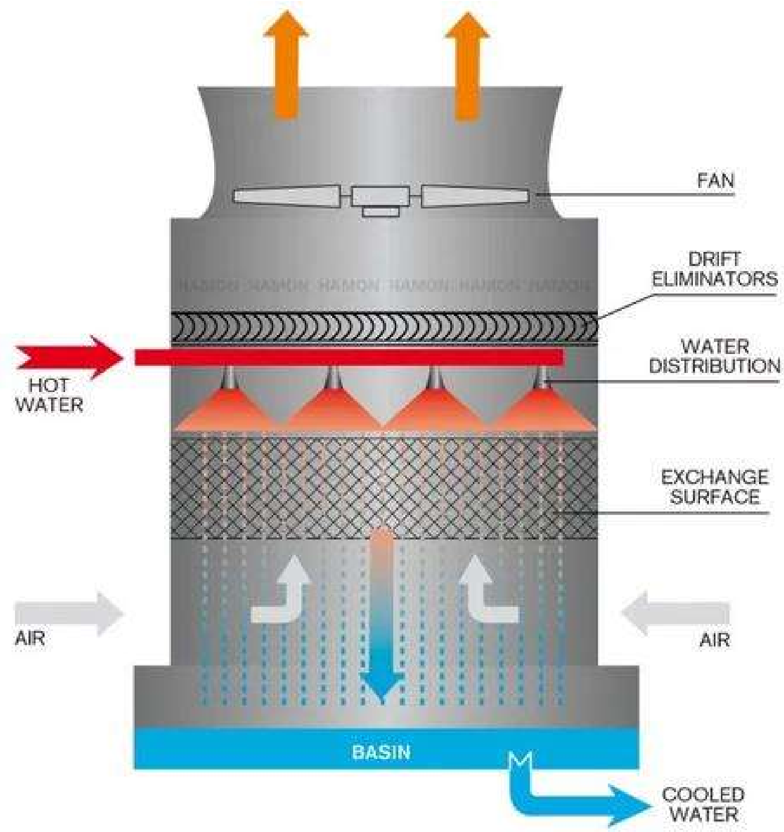
Induced draft cooling towers

- There are basically two types:
  - Cross flow
  - Counter flow
- Its main advantage is having less recirculation compared to forced draft towers
- Major disadvantage is that fans and motor required to drive mechanism need to be weather-proofed

Draft System

Induced draft cooling towers: counter flow

- Hot water enters from the top
- Air enters from the bottom and exits at the top
- Uses forced and induced draft fans

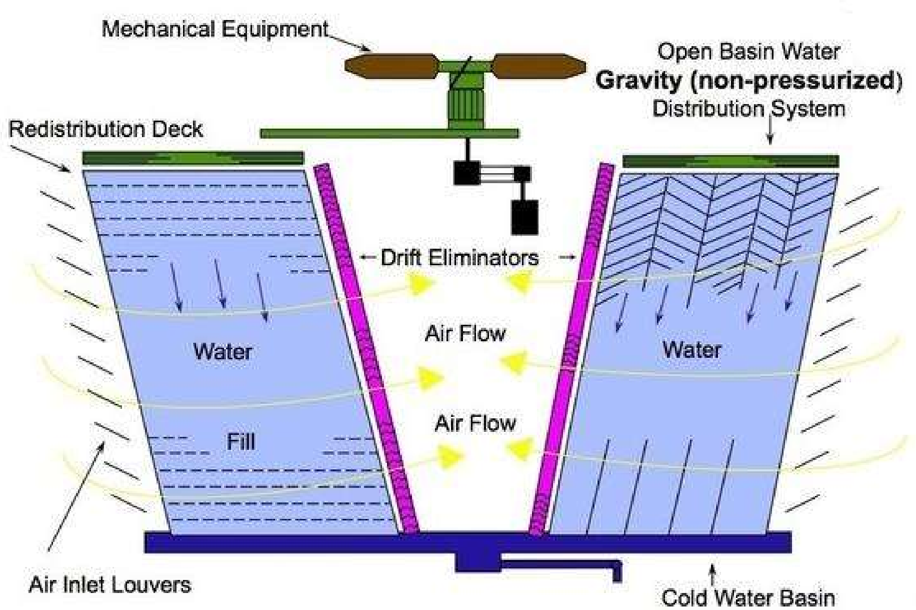


https://talkintrashwithuhn.com/2016/04/11/cooling-tower-winter-operation-at-toronto-western-hospitalooling-tower-winter-operation-at-toronto-western-hospital/

Draft System

Induced draft cooling towers: cross flow

- Water enters from the top and passes over fill
- Air enters on one side or on opposite sides
- Induced draft fan draws air across fill



https://www.quora.com/How-many-types-of-cooling-towers-are-there

Draft System

Forced draft cooling towers

- Centrifugal fan draws in air at the inlet
- Suited for high air resistance since fans operate relatively silent

https://www.quora.com/How-many-types-of-cooling-towers-are-there