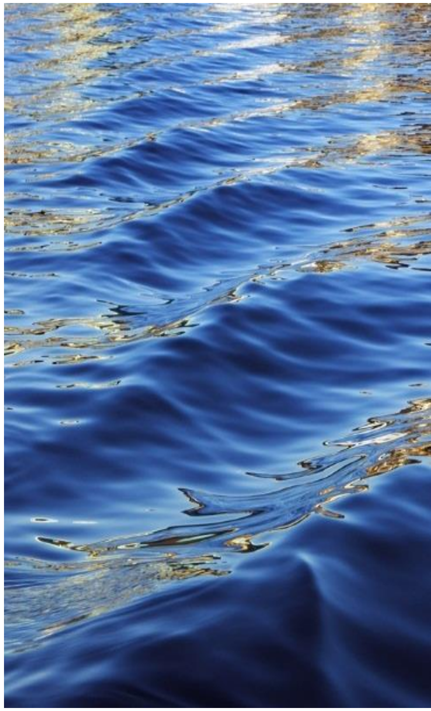
Environmental Thermal Engineering

Lecture Note #4

Professor Min Soo KIM

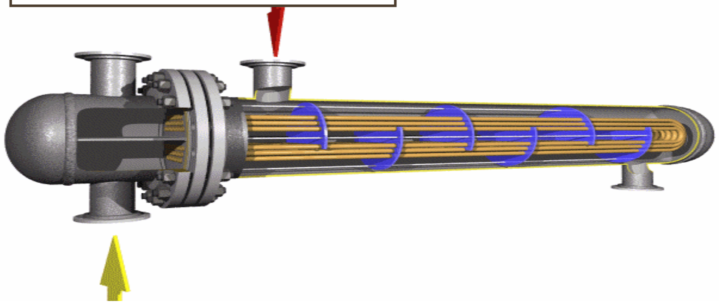


Heat Exchanger

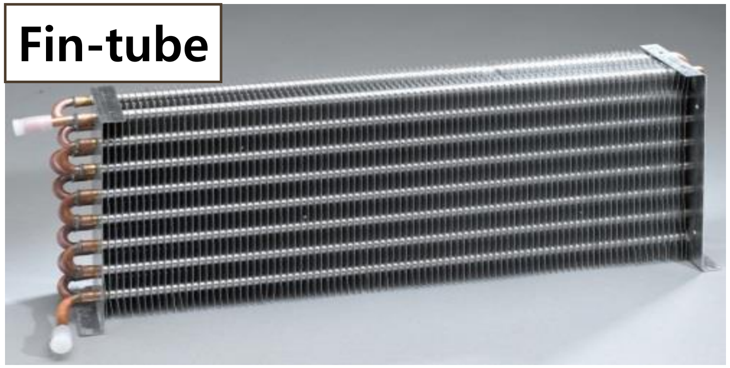
Heat Exchanger

- The process of heat exchange between two fluids that are at different temperatures and separated by a solid wall occurs in many engineering applications. The device used to implement this exchange is termed as a heat exchanger.

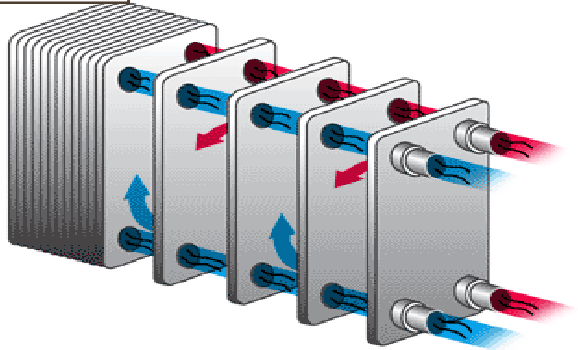
Shell and Tube



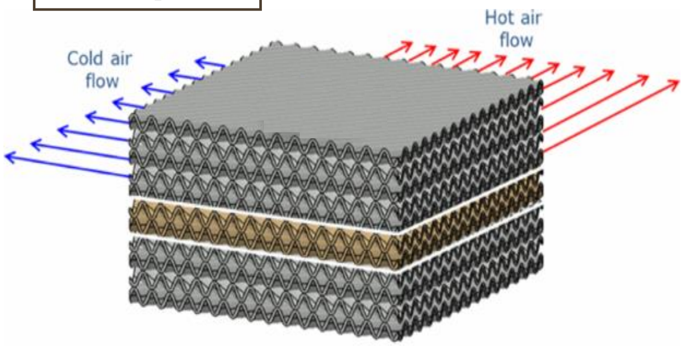
Fin-tube



Plate

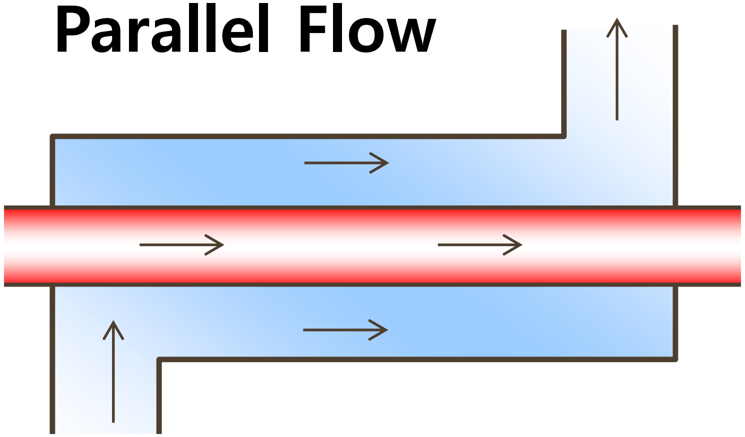


Compact

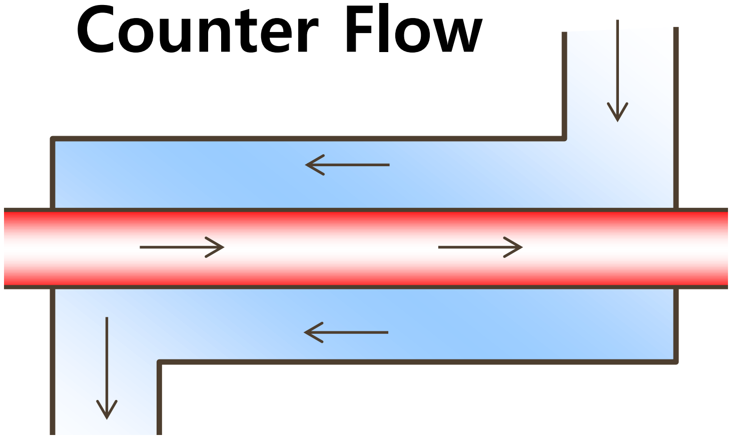


Heat Exchanger by Flow Arrangement

Parallel Flow

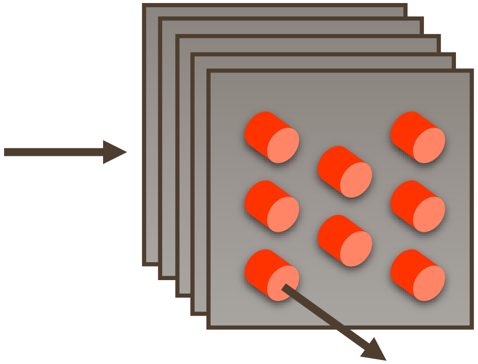


Counter Flow

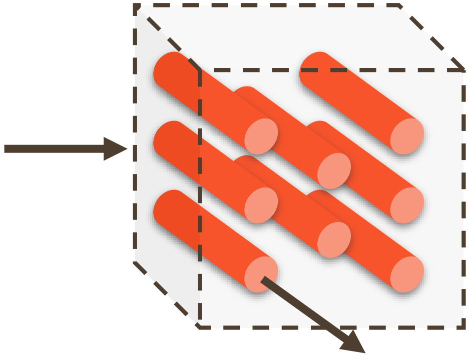


Cross Flow

(Unmixed)



(Mixed)



- **Intermediate efficiency** between parallel-flow and counter-flow heat exchanger
- Automobile radiators are designed as cross-flow design

Overall Heat Transfer Coefficient

Definition : Total thermal resistance of heat transfer between two fluids

$$\frac{1}{UA}=\frac{1}{U_cA_c}=\frac{1}{U_hA_h}=\frac{1}{(\eta_o hA)_c}+\frac{R''_{f,c}}{(\eta_o hA)_c}+R_w+\frac{R''_{f,h}}{(\eta_o hA)_h}+\frac{1}{(\eta_o hA)_h}$$

U : Overall Heat Transfer Coefficient

A : Heat Transfer Area

R_f : Fouling Factor

η : Fin Efficiency

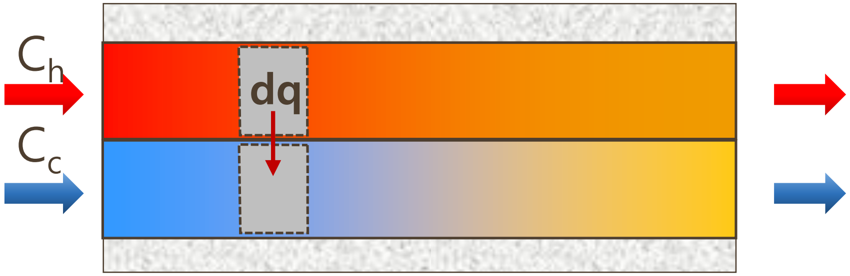
h : Heat Transfer Coefficient

R_w: Conduction Resistance

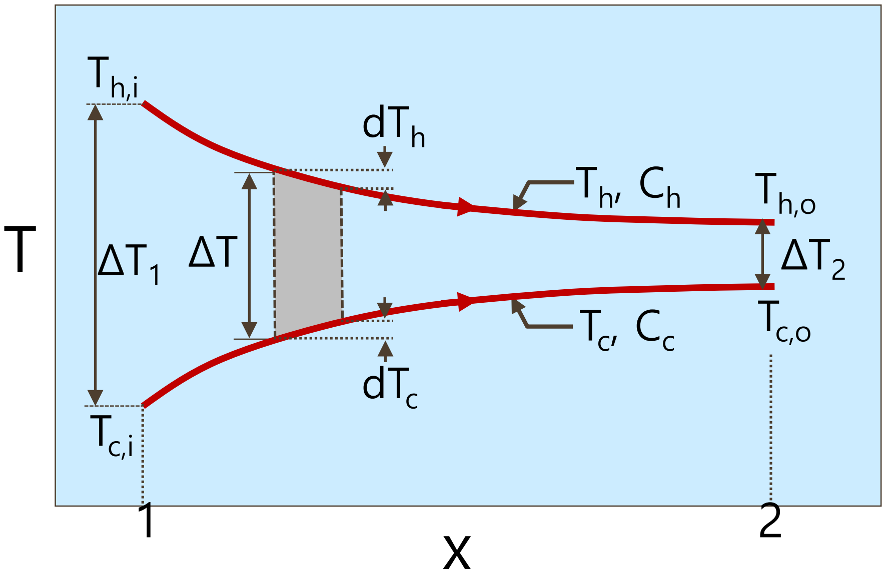
Log Mean Temperature Difference (LMTD)

❑ Parallel-flow

When the fluid temperatures are known



$$\Delta T_1 = T_{h,i} - T_{c,i}$$
$$\Delta T_2 = T_{h,o} - T_{c,o}$$



$$q = UA\Delta T_{LM} = UA\frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2 / \Delta T_1)}$$

FIGURE Temperature distribution for a parallel-flow heat exchanger

Log Mean Temperature Difference (LMTD)

❑ Parallel-flow (continued)

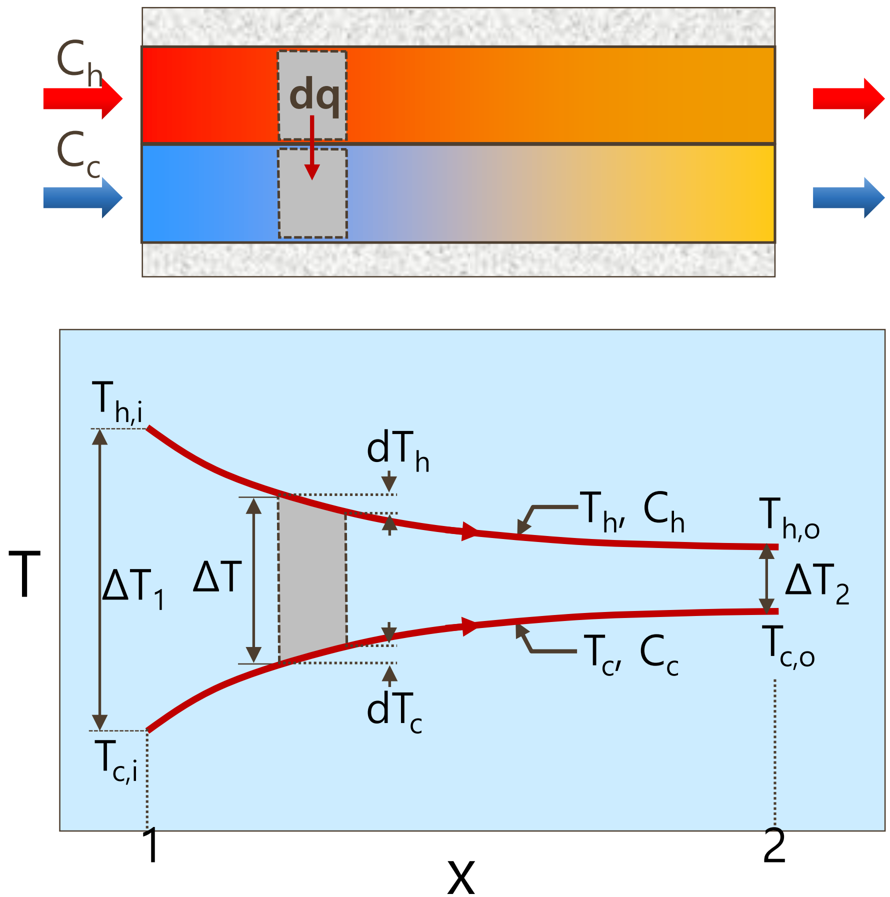


FIGURE Temperature distribution for a parallel-flow heat exchanger

$$dq = -\dot{m}_h c_{p,h} dT_h \equiv -C_h dT_h$$

$$dq = \dot{m}_c c_{p,c} dT_c \equiv C_c dT_c$$

$$dq = U \Delta T dA$$

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

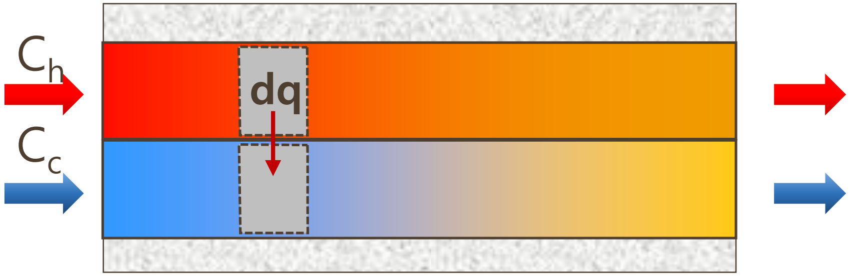
$$d(\Delta T) = -dq\left(\frac{1}{C_h} + \frac{1}{C_c}\right)$$

$$\int_1^2 \frac{d(\Delta T)}{\Delta T} = -U\left(\frac{1}{C_h} + \frac{1}{C_c}\right)\int_1^2 dA$$

$$\ln\left(\frac{\Delta T_2}{\Delta T_1}\right) = -UA\left(\frac{1}{C_h} + \frac{1}{C_c}\right)$$

Log Mean Temperature Difference (LMTD)

❑ Parallel-flow (continued)



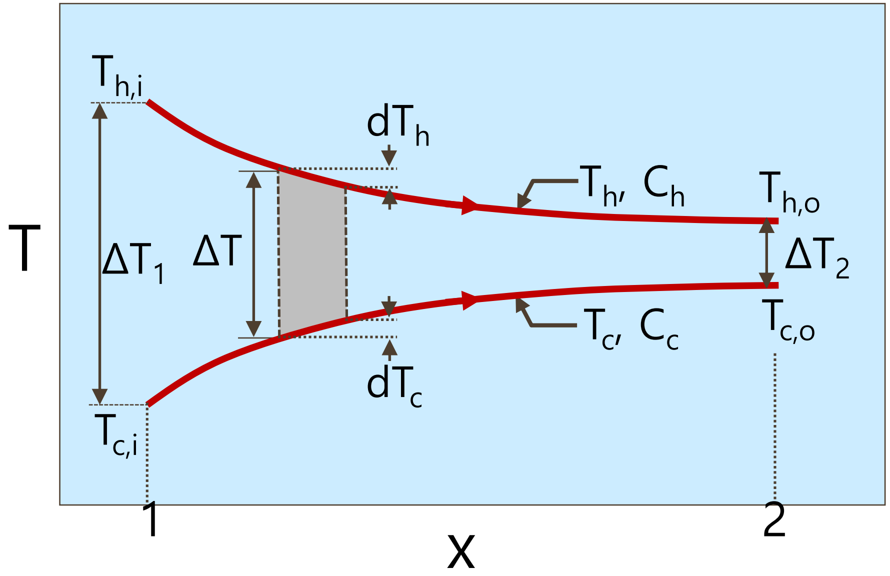


FIGURE Temperature distribution for a parallel-flow heat exchanger

$$d(\Delta T) = -dq\left(\frac{1}{C_h} + \frac{1}{C_c}\right)$$

$$\int_1^2 \frac{d(\Delta T)}{\Delta T} = -U\left(\frac{1}{C_h} + \frac{1}{C_c}\right)\int_1^2 dA$$

$$\ln\left(\frac{\Delta T_2}{\Delta T_1}\right) = -UA\left(\frac{1}{C_h} + \frac{1}{C_c}\right)$$

$$q = C_h(T_{h,i} - T_{h,o}) = C_c(T_{c,o} - T_{c,i})$$

$$\ln\left(\frac{\Delta T_2}{\Delta T_1}\right) = -UA\left(\frac{T_{h,i} - T_{h,o}}{q} + \frac{T_{c,o} - T_{c,i}}{q}\right)$$

$$= -\frac{UA}{q}\left[(T_{h,i} - T_{c,i}) - (T_{h,o} - T_{c,o})\right]$$

$$= -\frac{UA}{q}\left[\Delta T_1 - \Delta T_2\right]$$

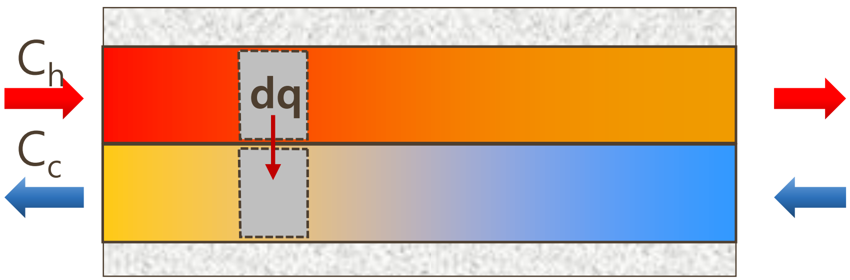
$$\boldsymbol{q = UA\frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2/\Delta T_1)} = UA\Delta T_{lm}}$$

Heat Exchanger

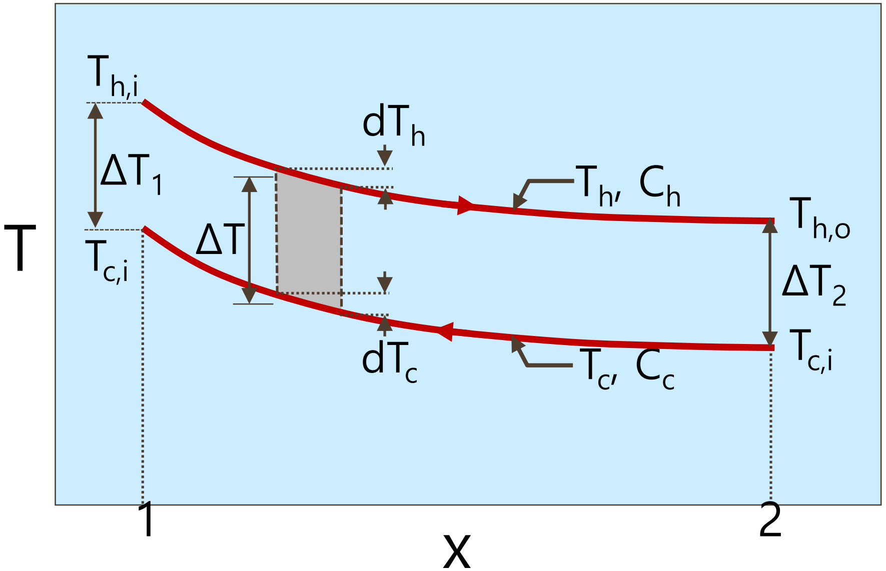
Log Mean Temperature Difference (LMTD)

❑ Counter-flow

When the fluid temperatures are known



$$\Delta T_1 = T_{h,i} - T_{c,o}$$
$$\Delta T_2 = T_{h,o} - T_{c,i}$$



$$q = UA\Delta T_{LM} = UA\frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2 / \Delta T_1)}$$

FIGURE Temperature distribution for a counter-flow heat exchanger

Log Mean Temperature Difference (LMTD)

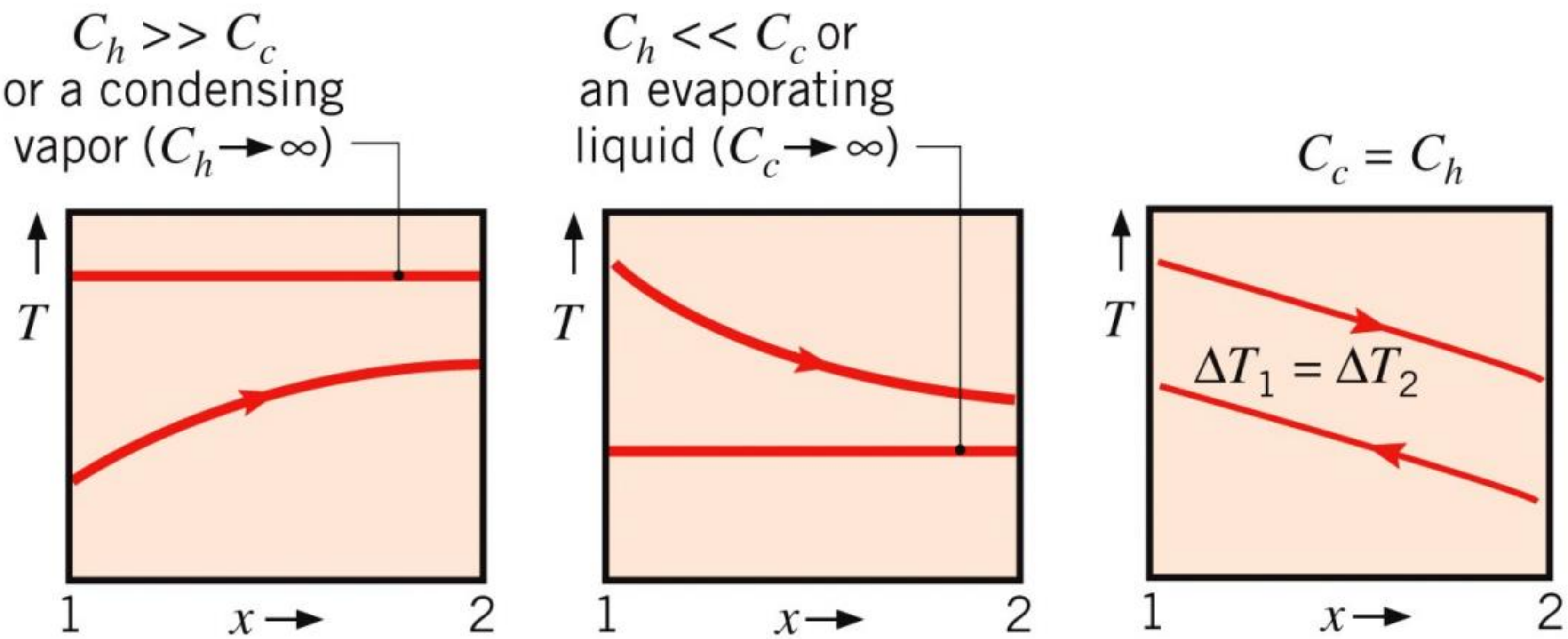


FIGURE Special heat exchanger conditions

ε-NTU method

❑ Definition

The heat exchanger **effectiveness** : $\varepsilon = \dfrac{\text{Actual Heat Transfer}}{\text{Maximum Possible Heat Transfer}} = \dfrac{q}{q_{max}}$

Number of heat transfer unit : $\mathrm{NTU} = \dfrac{UA}{C_{\min}}$

Capacity rate ratio : $C_r = \dfrac{C_{\min}}{C_{\max}}$

❑ Maximum Possible Heat Transfer

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

$C_c < C_h,\ C_c = C_{min}$

$C_h < C_c,\ C_h = C_{min}$

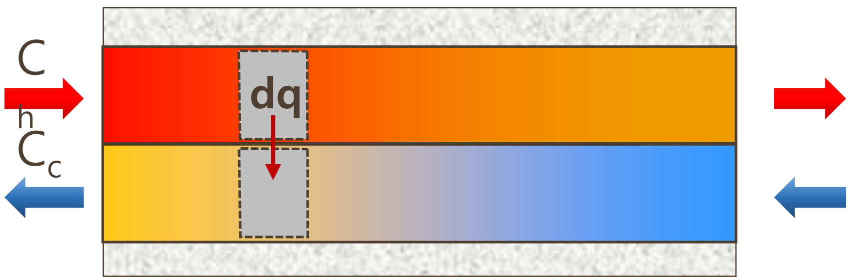
❑ Effectiveness

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

ε-NTU method

❑ Counter-flow

Considering Cold fluid to be minimum fluid



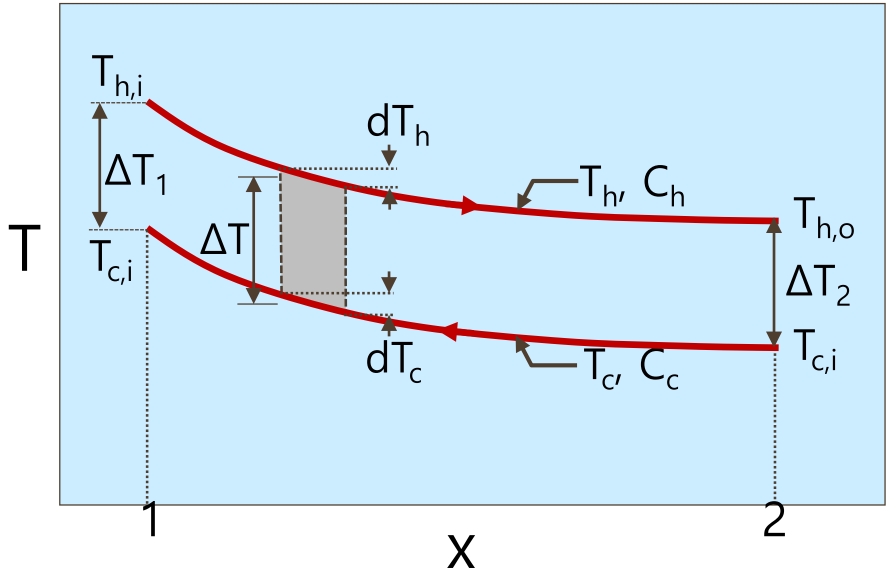


FIGURE Temperature distribution for a counter-flow heat exchanger

$$q = C_C\left(T_{c,o} - T_{c,i}\right)$$

$$= UA\,\frac{(T_{h,o} - T_{c,i}) - (T_{h,i} - T_{c,o})}{\ln\left[\left(T_{h,o} - T_{c,i}\right)/\left(T_{h,i} - T_{c,o}\right]\right.}$$

$$+$$

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

$$\Downarrow$$

$$\ln\left(\frac{\frac{1}{\varepsilon} - C_{min}}{\frac{1}{\varepsilon} - 1}\right) = \frac{UA}{C_{min}}\left(1 - \frac{C_{min}}{C_{max}}\right)$$

$$\Downarrow$$

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

(For counter flow)

Heat Exchanger

ε-NTU method

Table Heat exchanger effectiveness relations

$\left(C_r = \dfrac{C_{\min}}{C_{\max}}\right)$

Flow arrangement	Relation
Concentric tube	
Parallel flow	$\varepsilon = \dfrac{1 - \exp[-NTU(1 + C_r)]}{1 + C_r}$
Counter flow	$\varepsilon = \dfrac{1 - \exp[-\mathrm{NTU}(1 - C_r)]}{1 - C_r \exp[-\mathrm{NTU}(1 - C_r)]}$ $(@\ C_r < 1)$ $\quad \varepsilon = \dfrac{\mathrm{NTU}}{1 + \mathrm{NTU}}$ $(@\ C_r = 1)$
Shell and tube	
One shell pass (2, 4, … tube passes)	$\varepsilon_1 = 2\left\{1 + C_r + (1 + C_r^2)^{1/2} \times \dfrac{1 + \exp[-NTU(1 + C_r^2)^{1/2}]}{1 - \exp[-NTU(1 + C_r^2)^{1/2}]}\right\}^{-1}$
n shell passes (2n, 4n, … tube passes)	$\varepsilon = \left[\left(\dfrac{1 - \varepsilon_1 C_r}{1 - \varepsilon_1}\right)^n - 1\right]\left[\left(\dfrac{1 - \varepsilon_1 C_r}{1 - \varepsilon_1}\right)^n - C_r\right]^{-1}$
All exchangers (C_r=0)	$\varepsilon = 1 - \exp(-NTU)$

ε-NTU method

$\left(C_r = \frac{C_{\min}}{C_{\max}}\right)$

Table Heat exchanger NTU relations

Flow arrangement	Relation
Concentric tube	
Parallel flow	$\mathrm{NTU} = \frac{\ln[1-\varepsilon(1+C_r)]}{1+C_r}$
Counter flow	$\mathrm{NTU} = \frac{1}{C_r - 1}\ln\left(\frac{\varepsilon - 1}{\varepsilon C_r - 1}\right)$ (@ $C_r < 1$) $\mathrm{NTU} = \frac{\varepsilon}{1-\varepsilon}$ (@ $C_r = 1$)
Shell and tube	
One shell pass (2, 4, … tube passes)	$NTU = -(1+C_r^2)^{-1/2}\ln\left(\frac{E-1}{E+1}\right), E = \frac{2/\varepsilon_1 - (1+C_r)}{(1+C_r^2)^{1/2}}$
n shell passes (2n, 4n, … tube passes)	$\varepsilon = \frac{F-1}{F-C_r}, F = \left(\frac{\varepsilon C_r - 1}{\varepsilon - 1}\right)^{1/n}$
All exchangers (C_r=0)	$NTU = -\ln(1-\varepsilon)$

Evaporator & Condensor

A heat exchanger with phase change of working fluid

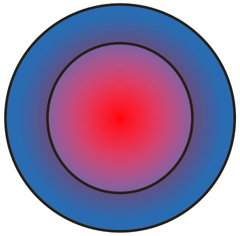


Table Some types of evaporators and condensers

Component	Refrigerant	Fluid
Condenser	Inside tubes	Gas outside Liquid outside*
	Outside tubes	Gas inside* Liquid inside
Evaporator	Inside tubes	Gas outside Liquid outside*
	Outside tubes	Gas inside* Liquid inside

* Seldomly used

Compact & Plate Heat Exchanger

Compact Heat Exchanger

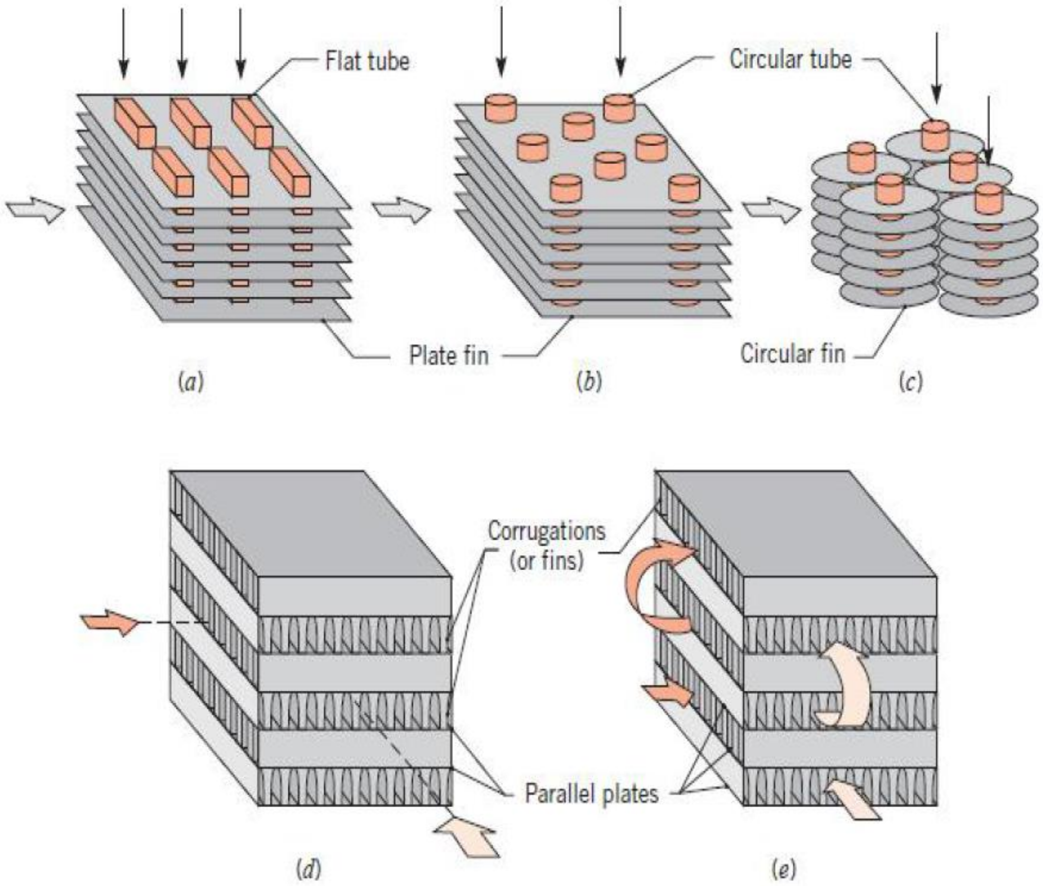
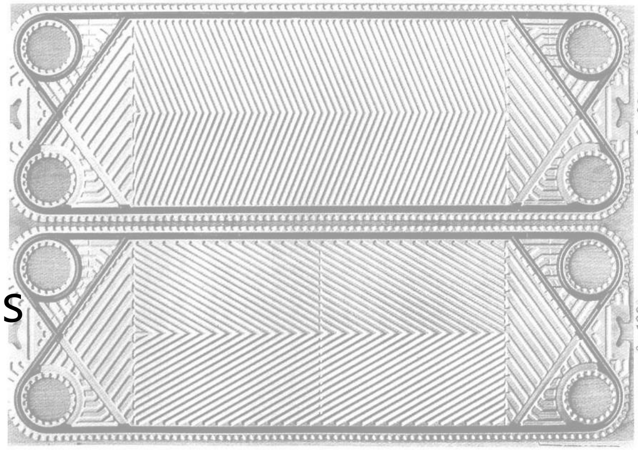


FIGURE Compact heat exchanger cores. (a) Fin-tube(flat tubes, continuous plate fins). (b) Fin-tube (circular tubes, continuous plate fins). (c) Fin-tube (circular tubes, circular fins). (d) Plate-fin (single pass). (e) Plate-fin (multi-passes)

Compact Heat Exchanger

1. Compact heat exchangers have a very large heat transfer surface area per unit volume (> 700 m^2/m^3)
2. Have dense arrays of finned tubes or plates.
3. Are usually used when at least one of the fluids is a gas.
4. Flow passages are typically small (D_h<5 mm)
5. Flow is usually laminar.

Plate Heat Exchanger



1. **<u>High NTU Values</u>**
   High heat transfer coefficients
   High recuperative efficiencies
   NTU of 6 can be achieved in a single pass

2. **<u>Flexibility</u>**
   The arrangements & the surface area can be easily altered.

3. **<u>Accessibility</u>**
   It is relatively easy to dismantle and access all working-surfaces for inspection and cleaning.

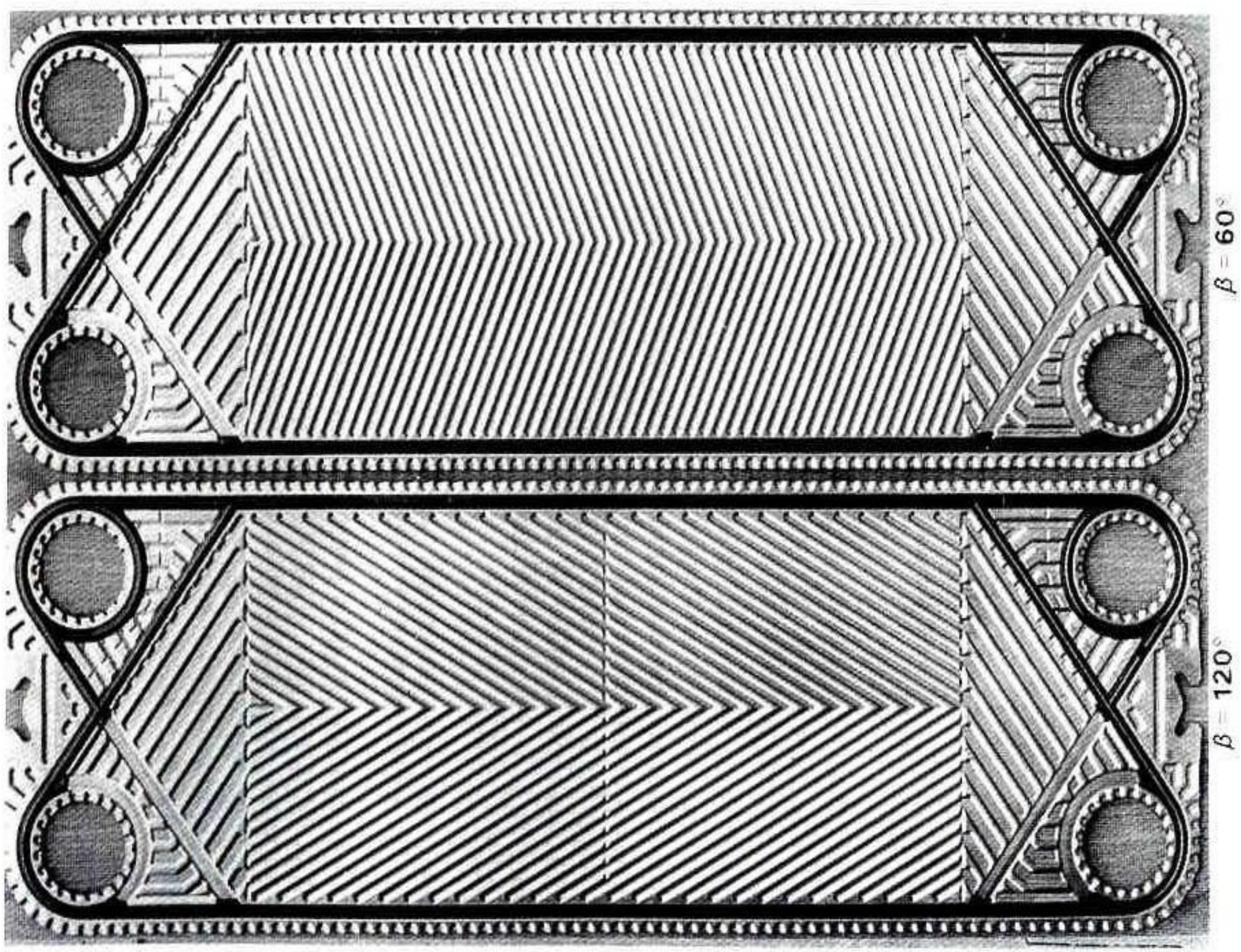
Plate Heat Exchanger

4. **Compactness**
   Liquid holdup compared with the surface area is large. Flow channels with narrow gaps lead to a compact construction.

5. **Multiple Duties**
   Use of special connector plates that act as intermediate headers allows a number of separate heat exchange duties to be housed in a single frame.

6. **Reduced Fouling**
   The surface shear stresses are very high, allowing a high fouling removal rate

Plate Heat Exchanger



β=60°

β=120°

FIGURE Typical plates of plate heat exchanger (Courtesy of Alfa Laval AB, Sweden)

Plate Heat Exchanger

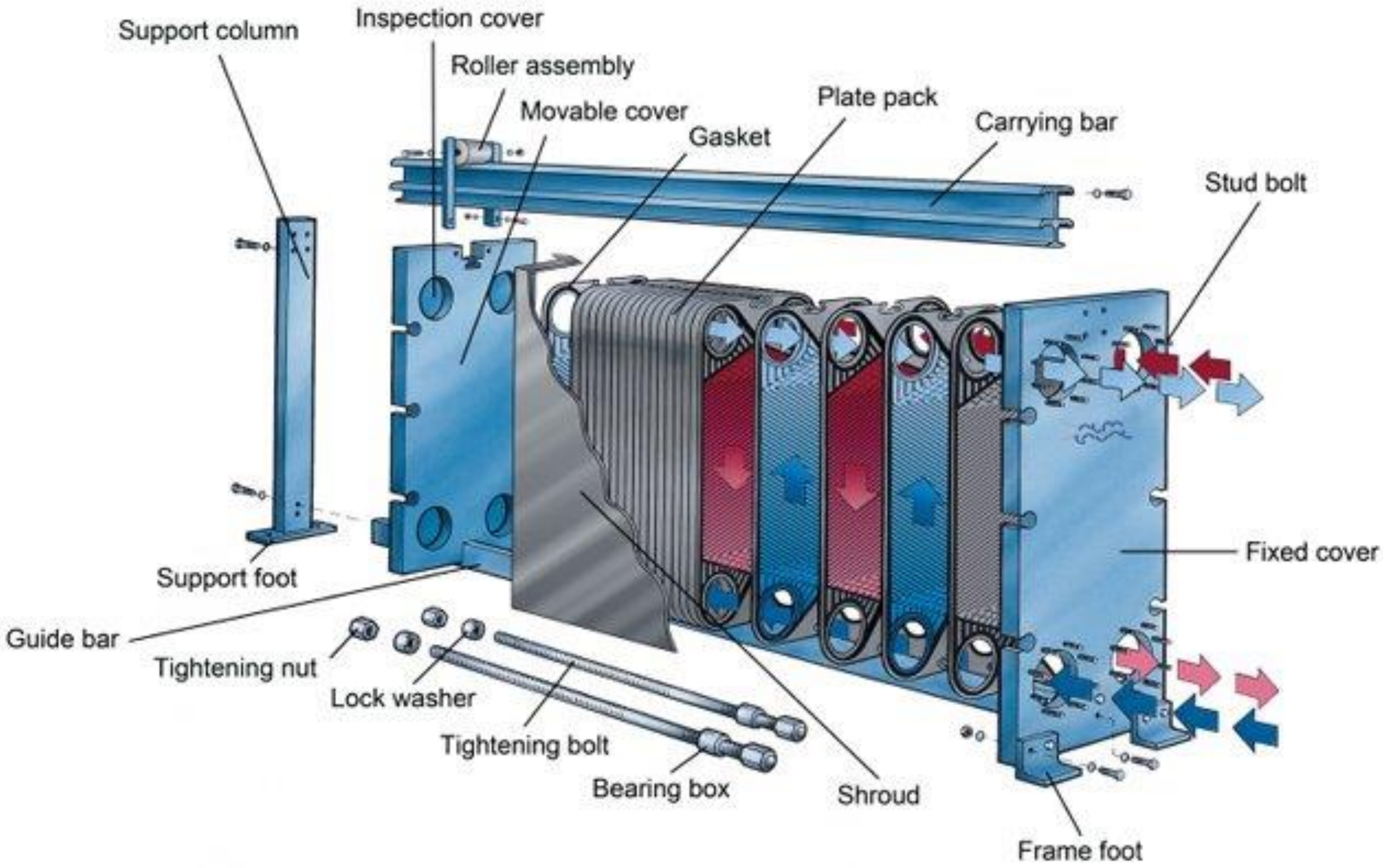


FIGURE Gasket type plate heat exchanger

Boiling

Pool Boiling

❑ Pool boiling

The liquid is stationary and its motion near the surface is due to free convection and mixing induced by bubble growth and detachment.

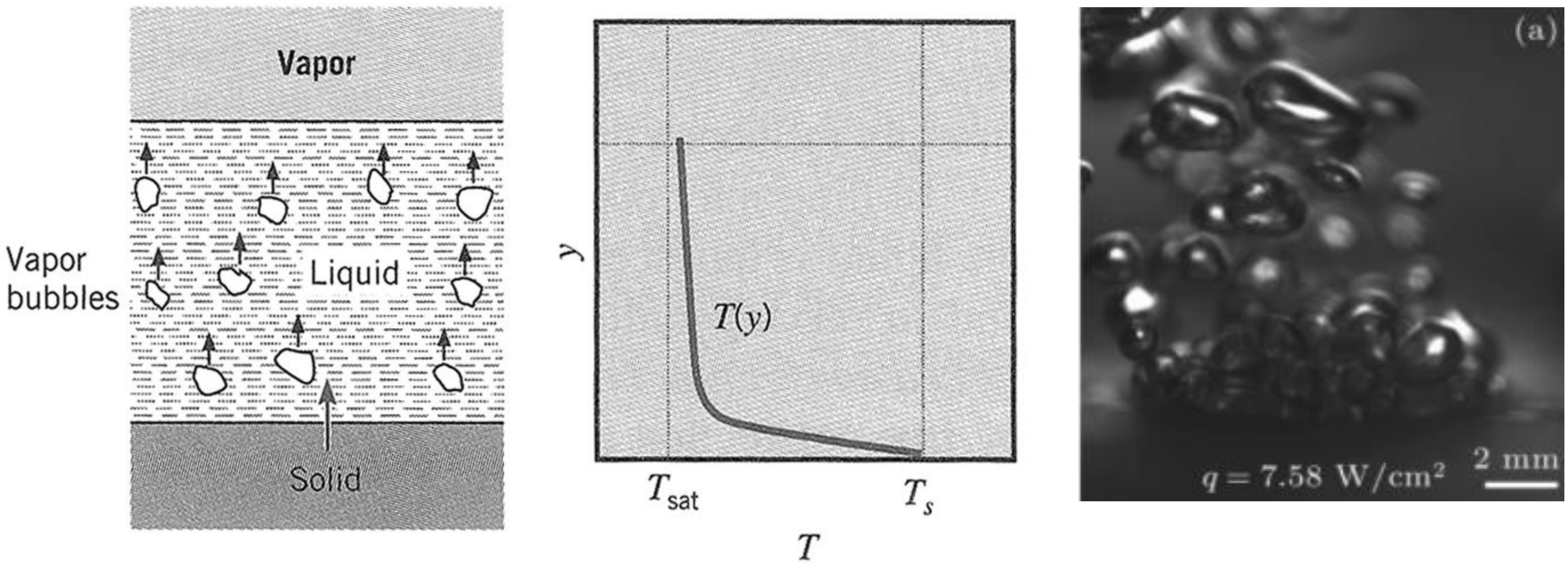


FIGURE Temperature distribution in saturated pool boiling with a liquid vapor interface

Boiling Curve

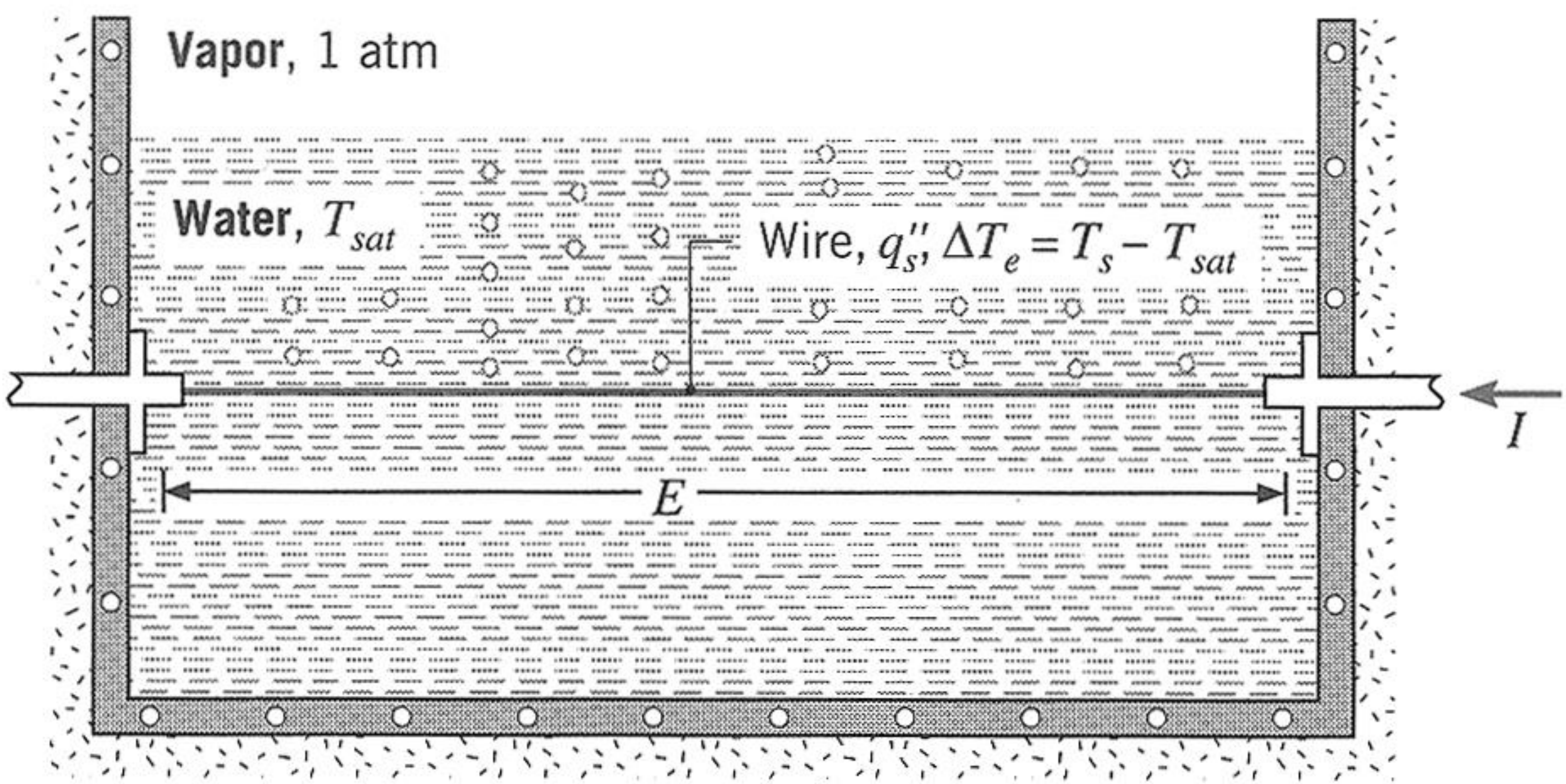


FIGURE Nukiyama's power-controlled heating apparatus demonstrating the boiling curve

Boiling Curve

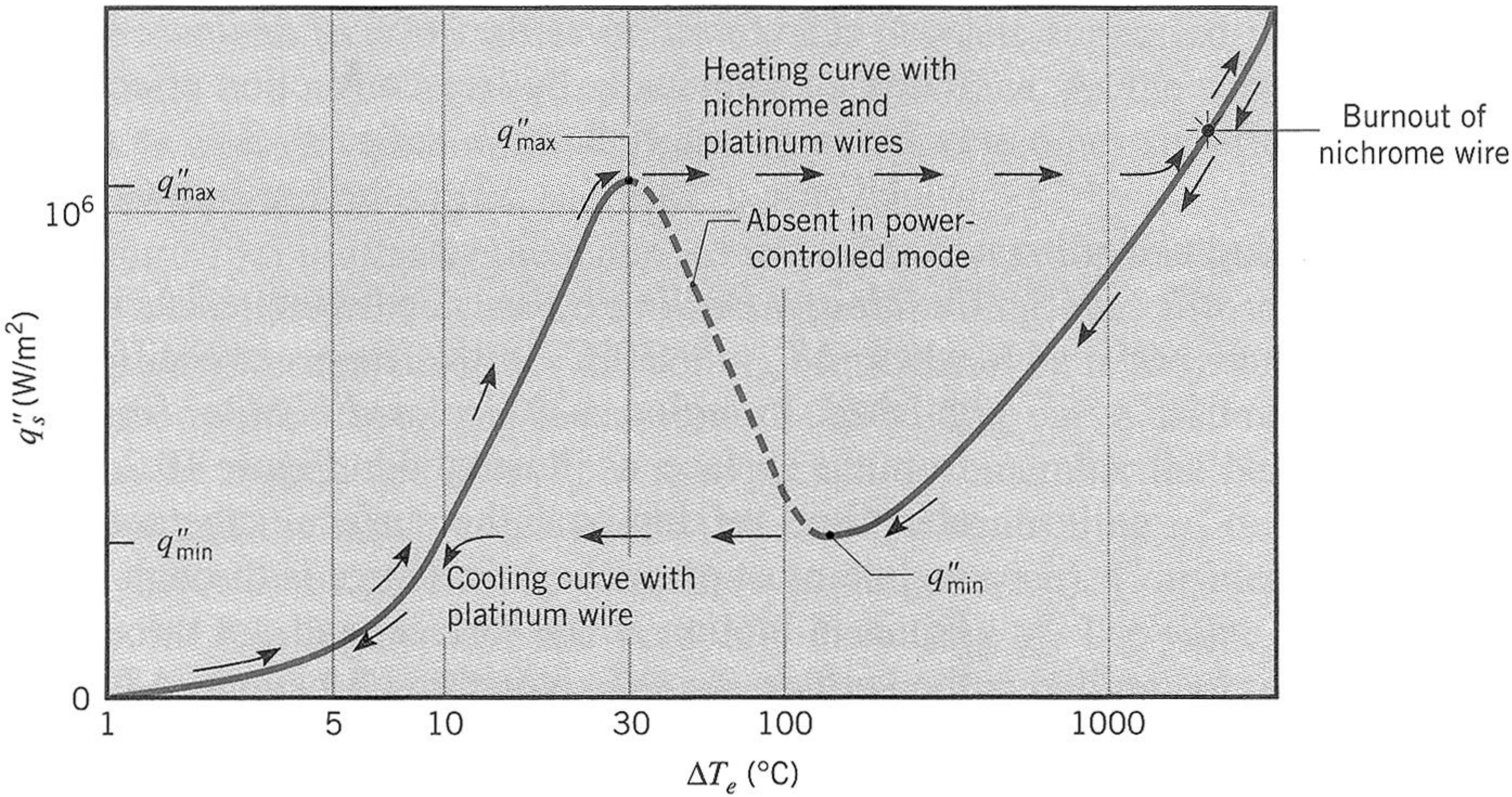


FIGURE Nukiyama's boiling curve for saturated water at 1 atm

Forced Convective Boiling

- Flow is due to a directed (bulk) motion of the fluid, as well as to buoyancy effects
- Conditions depend strongly on geometry, which may involve *external* flow over heated plate and cylinders or *internal* (duct) flow
- Internal forced convection boiling is commonly referred to as *two-phase flow* and is characterized by rapid changes from liquid to vapor in the flow direction

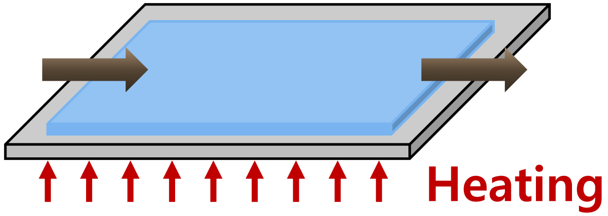


FIGURE. Internal convective boiling

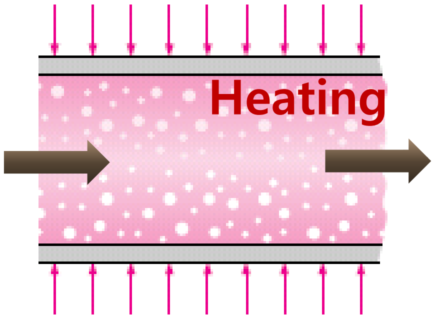


FIGURE. External convective boiling

External Forced Convection Boiling

- For external flow over a heated plate, the heat flux can be estimated by standard forced convection correlation up to the inception of nucleate boiling

- As the temperature of the heated plate is increased, nucleate boiling will occur, causing the heat flux to increase

- For a liquid of velocity V moving in cross flow over a cylinder of diameter D, Lienhard and Eichhorn have developed the following expressions of low- and high-velocity flows

Low velocity

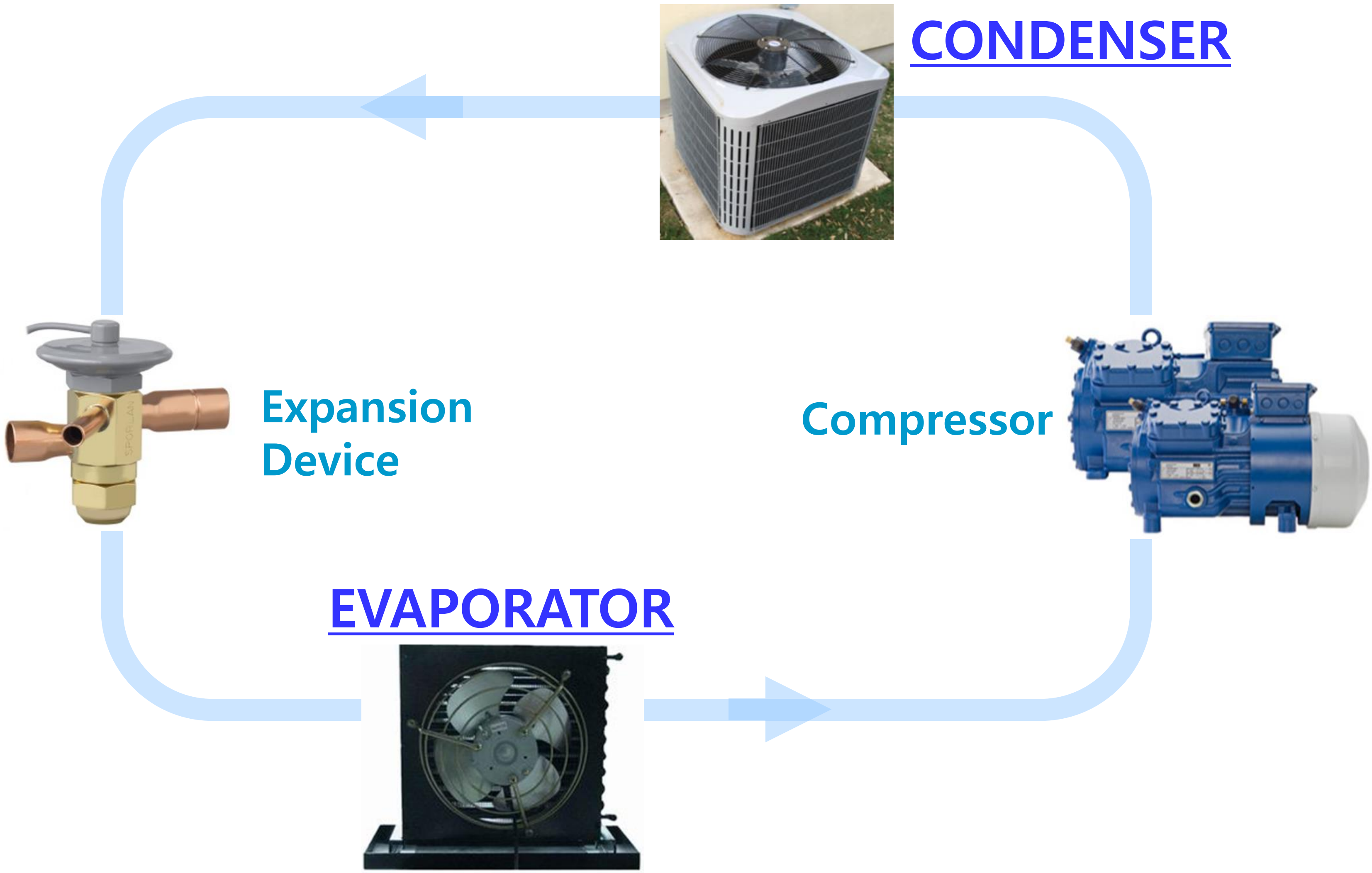
$$\frac{q''_{max}}{\rho_v h_{fg} V} = \frac{1}{\pi}\left[1 + \left(\frac{4}{We_D}\right)^{1/3}\right]$$

High Velocity

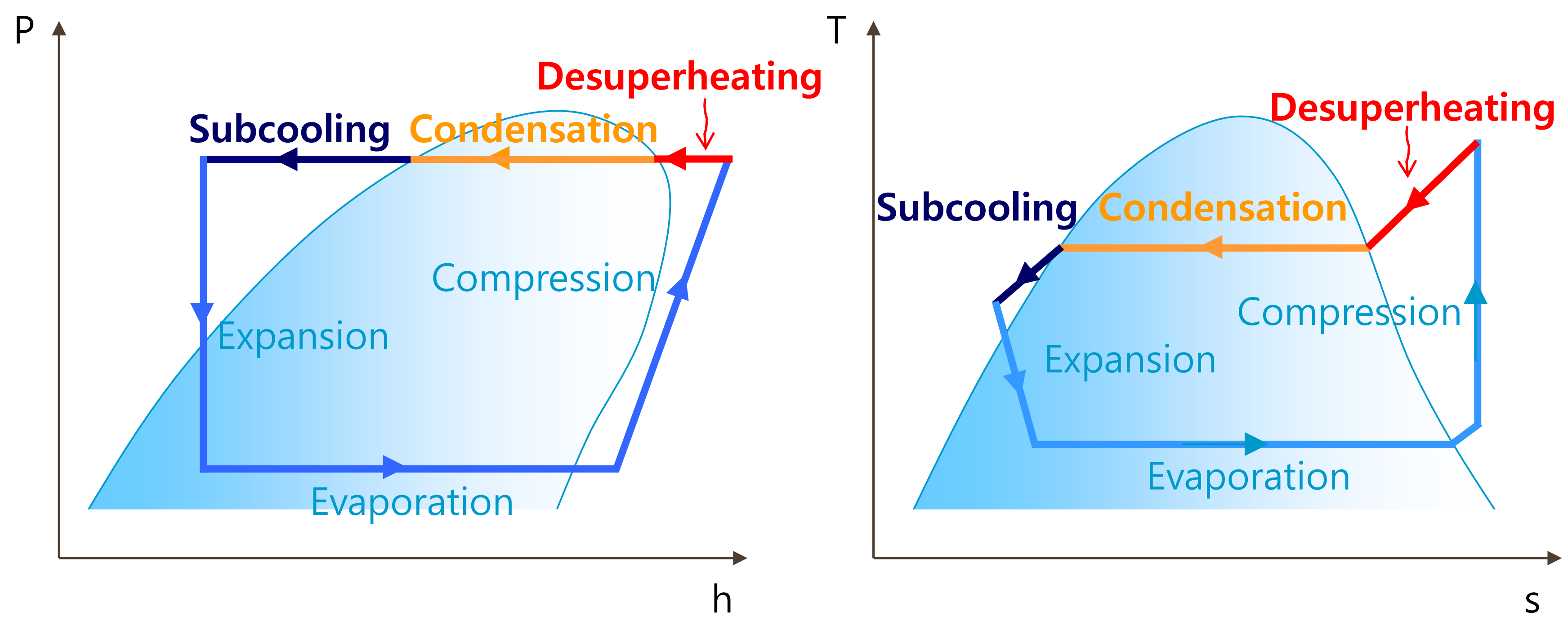
$$\frac{q''_{max}}{\rho_v h_{fg} V} = \frac{(\rho_l / \rho_v)^{3/4}}{169\pi} + \frac{(\rho_l / \rho_v)^{1/2}}{19.2\pi We_D}$$

Evaporation and Condensation

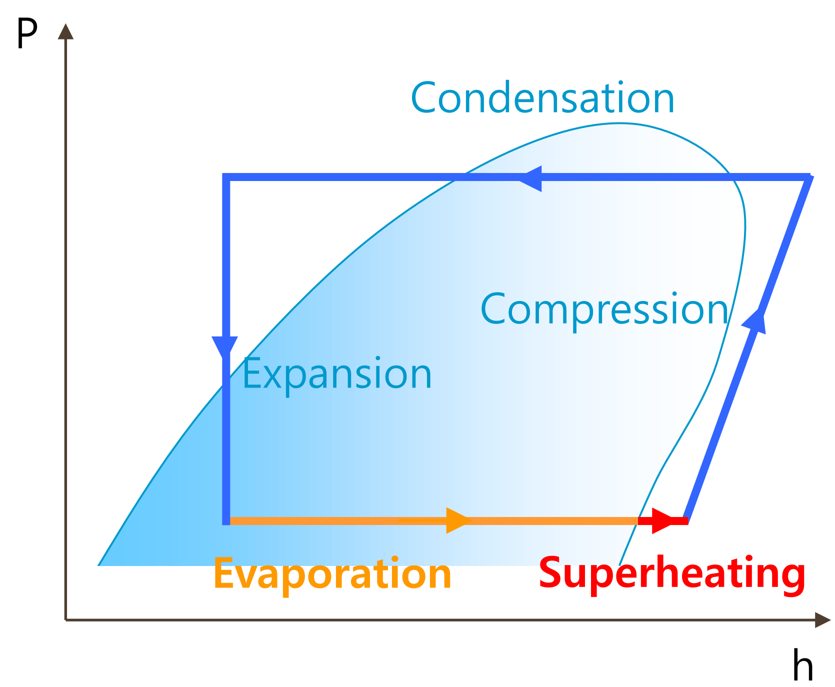
Refrigeration Cycle

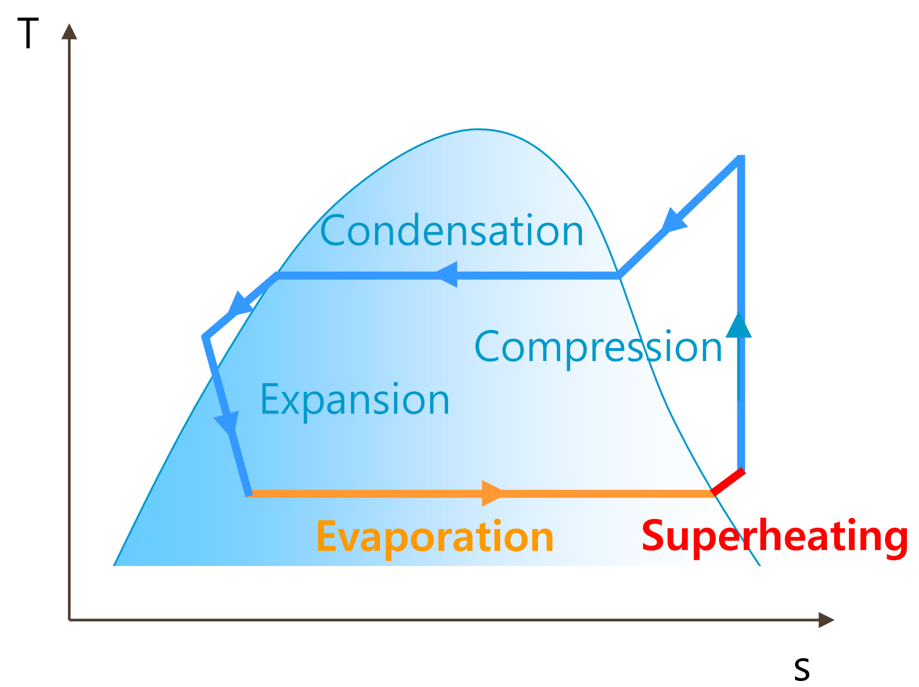


Condensation in Refrigeration Cycle



Evaporation in Refrigeration Cycle





Two-phase Flow

- It is associated with bubble formation at the inner surface of a heated tube through which a liquid is flowing.

- Bubble growth and separation are strongly influenced by the flow velocity.

- Hydrodynamic effects differ significantly from those corresponding to pool boiling.

- The process is complicated by the existence of different two-phase flow patterns that preclude the development of generalized theories.

Two-phase flow (Horizontal)

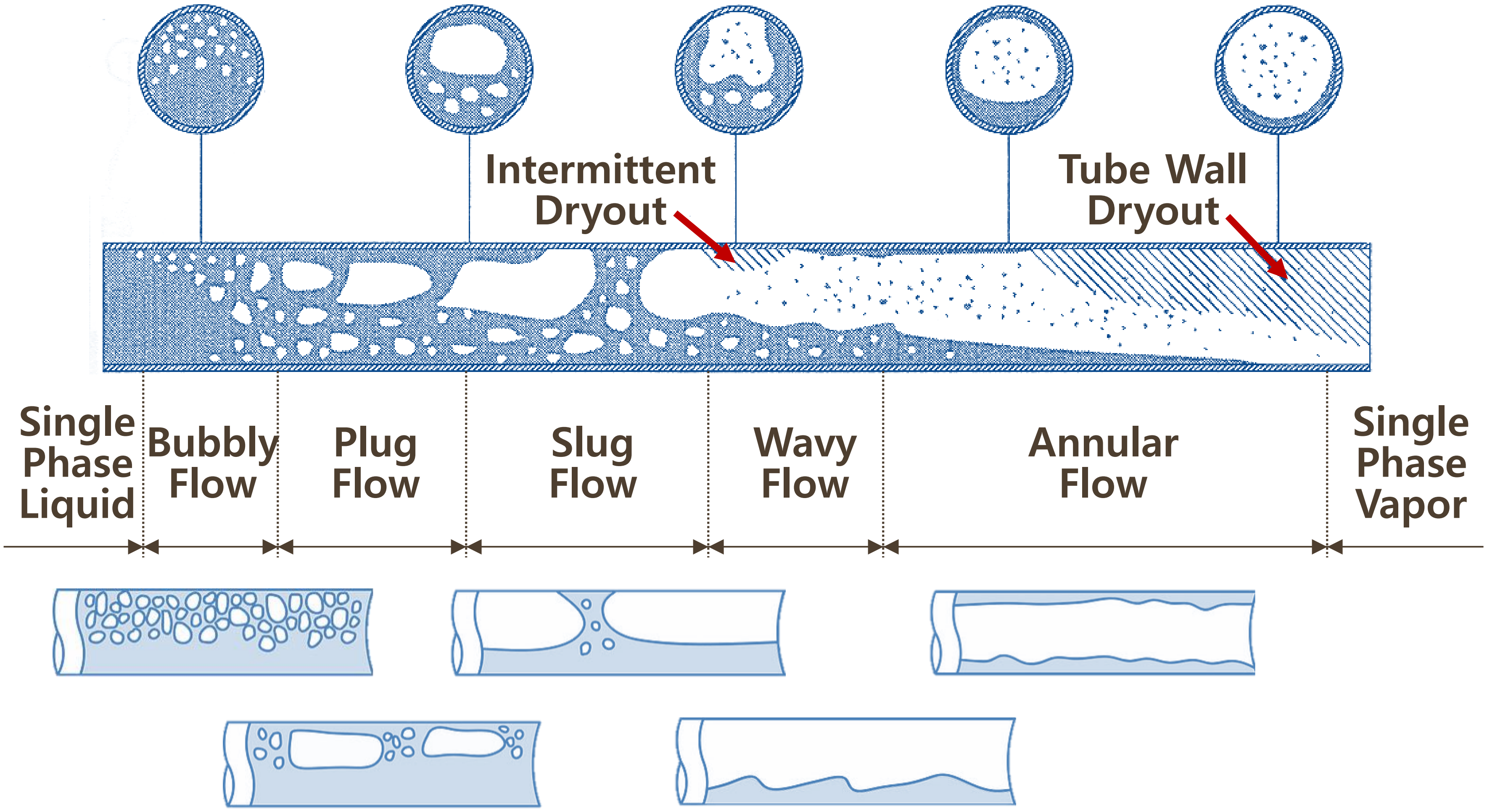


FIGURE Flow patterns in a uniformly heated horizontal tube

Two-phase flow (Vertical)

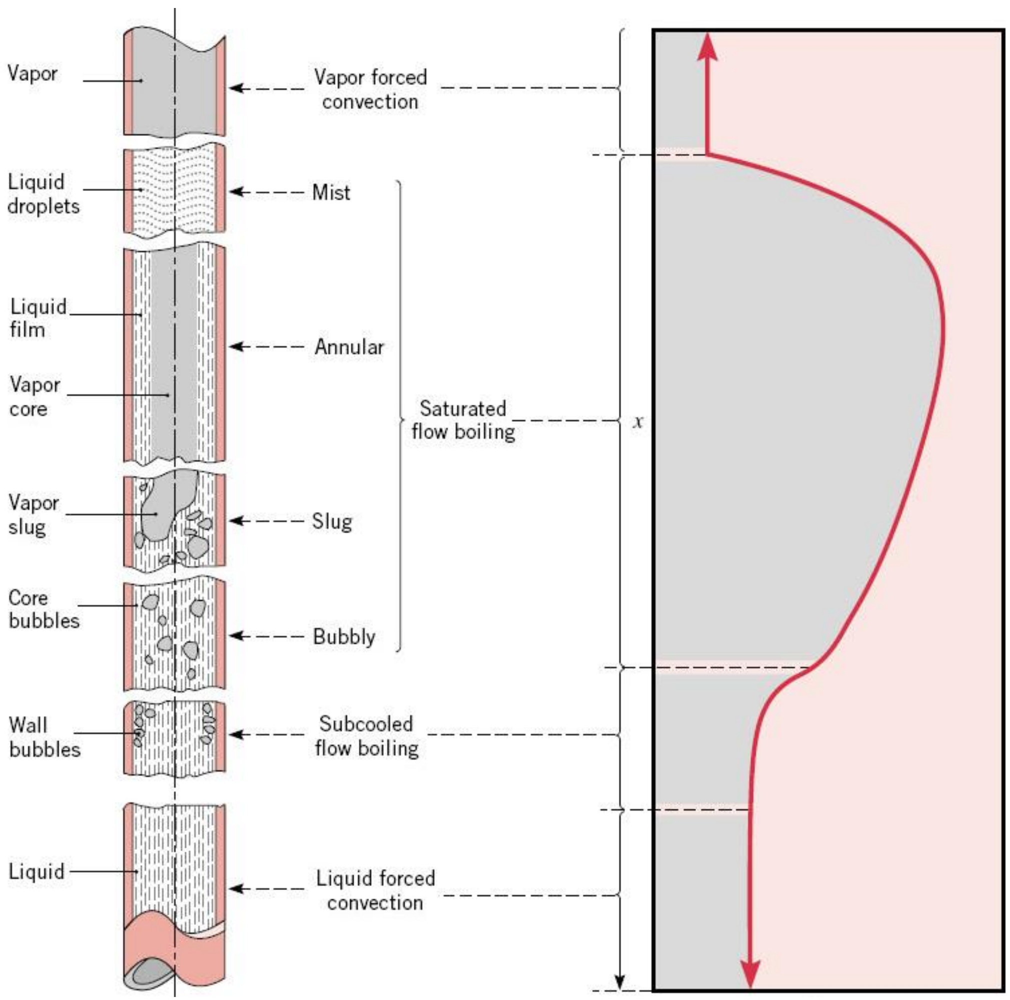


FIGURE Flow patterns in a uniformly heated vertical tube

Heat Transfer Coefficient

Definition :

$$h = \frac{q''}{T_{wi} - T_{sat}}$$

h : evaporation heat transfer coefficient, kW/m^2K

q" : applied heat flux, kW/m^2

T_{wi} : inner wall temperature, K

T_{sat} : saturation temperature of refrigerant, K

Evaporation

- In most refrigerating evaporators the refrigerant boils in the tubes and cools the fluid that passes over the outside of the tubes.

- Most evaporators are designed and controlled to bring the refrigerant to a small degree of superheat as it leaves the evaporator to protect the compressor downstream from the damaging effects of liquid.

- The medium transferring heat to the evaporator may be the air stream to be cooled (direct-expansion or DX coils) or may be water or brine, as in the case of chillers.

Evaporation and Condensation

Examples of Evaporators

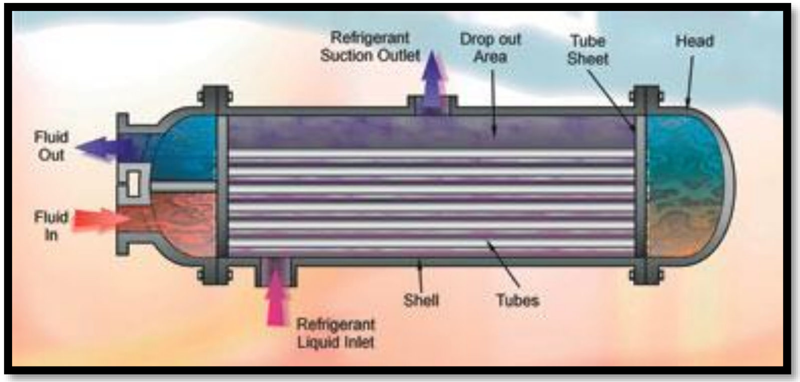


FIGURE Flooded evaporator

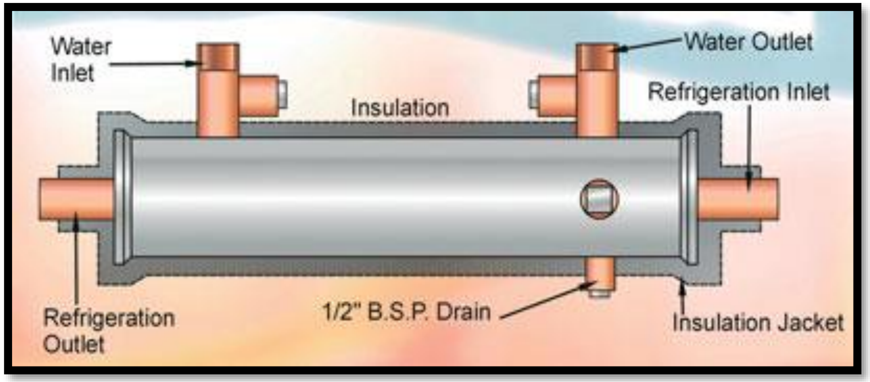
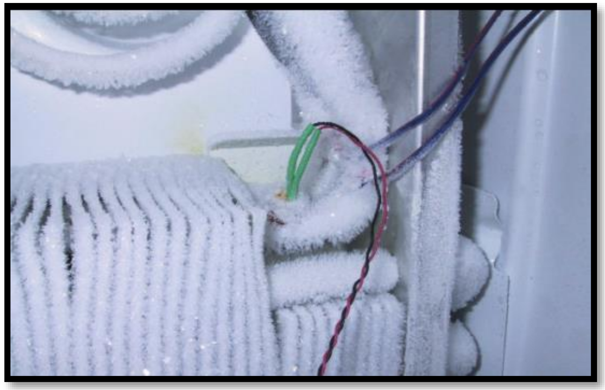


FIGURE Direct expansion (DX) evaporator

Frost

- When the surface temperature of an air-cooling evaporator falls below 0°C, FROST forms. It is detrimental to the operation of the refrigeration system for two reasons.
  1. Thick layers of frost acts as an insulation
  2. It can reduce the airflow rates



- Defrosting methods
  1. Hot-gas defrost – discharge hot gas from the compressor
  2. Water defrost - spray water directly over the coil.
  3. Electric defrost – utilize installed electrical heating elements

Why ICE is hexagonal?

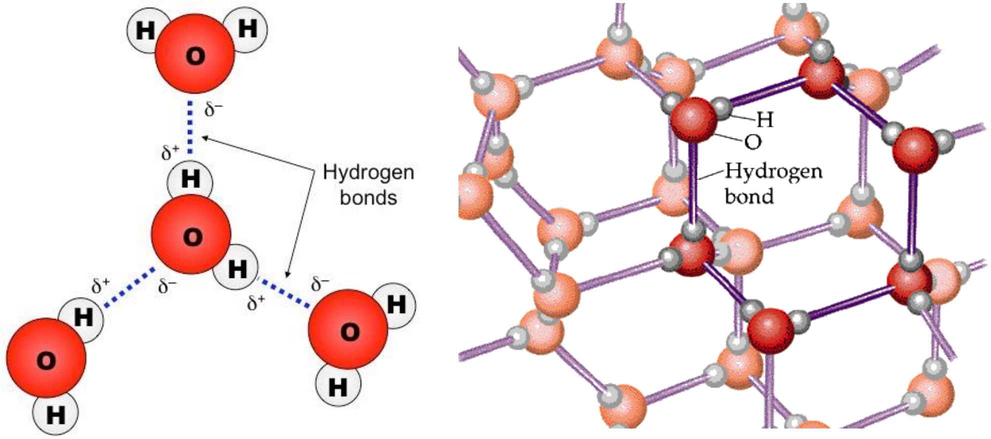


FIGURE Hydrogen bonds in Ice

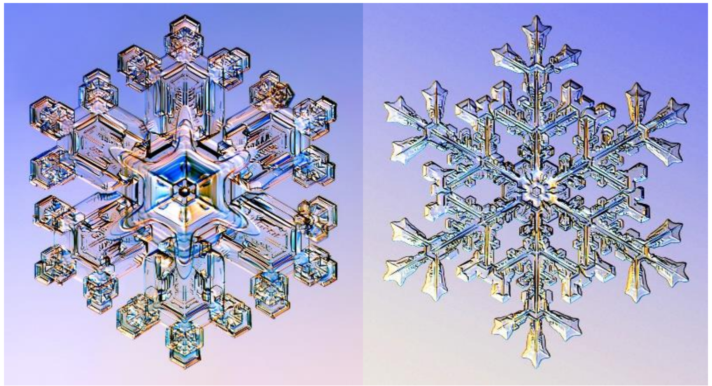


FIGURE Ice crystal

In Ice Ih, each water forms four hydrogen bonds with O—O distances of 2.76 Angstroms to the nearest oxygen neighbor. The O-O-O angles are 109 degrees, typical of tetrahedrally coordinated lattice structure.

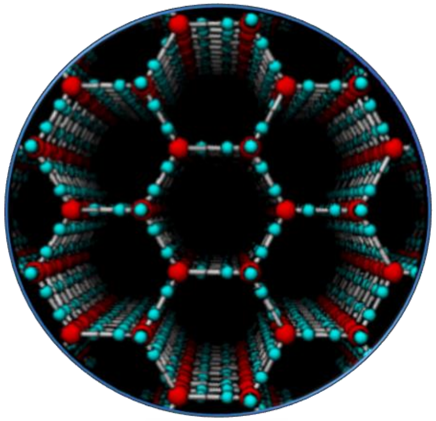


FIGURE Magnified ice(x 100mil.)

Heat Transfer Coefficient

Definition :

$$h = \frac{q''}{T_{sat} - T_{wi}}$$

h : evaporation heat transfer coefficient, kW/m^{2}K

q'' : applied heat flux, kW/m^2

T_{wi} : inner wall temperature, K

T_{sat} : saturation temperature of refrigerant, K

Condensation

- **Condensation occurs when the vapor temperature is decreased below its saturation temperature.**

- **In industrial equipment, the process commonly results from contact between the vapor and a cool *surface***

- **The latent heat is released, heat is transferred to the surface, and the condensate is formed**

Modes of Condensation

1. The contact between the vapor and a cool surface

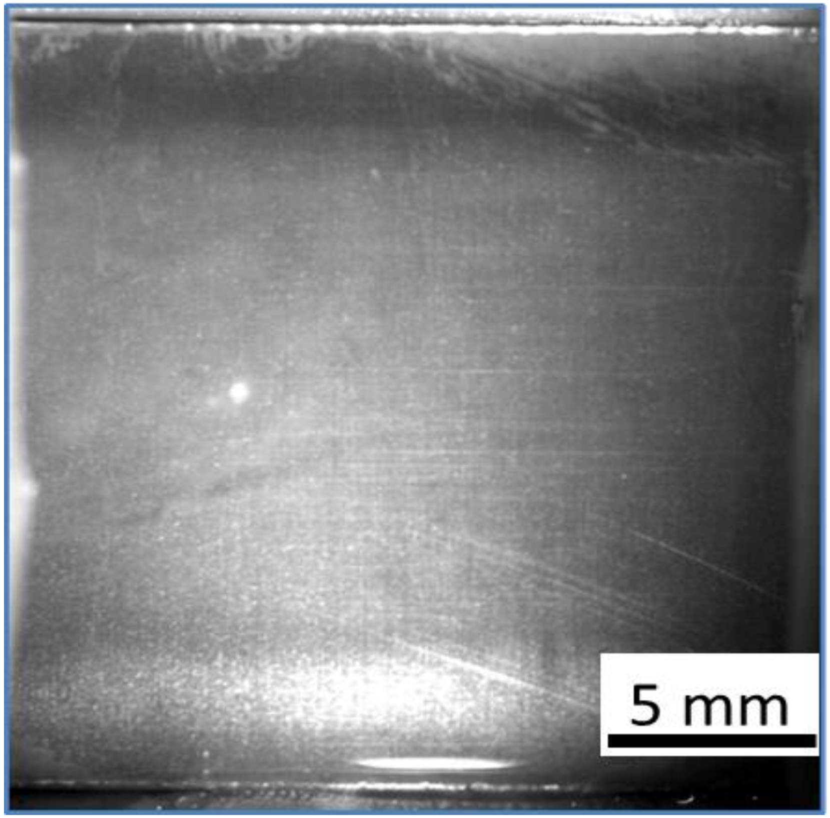


FIGURE Filmwise condensation

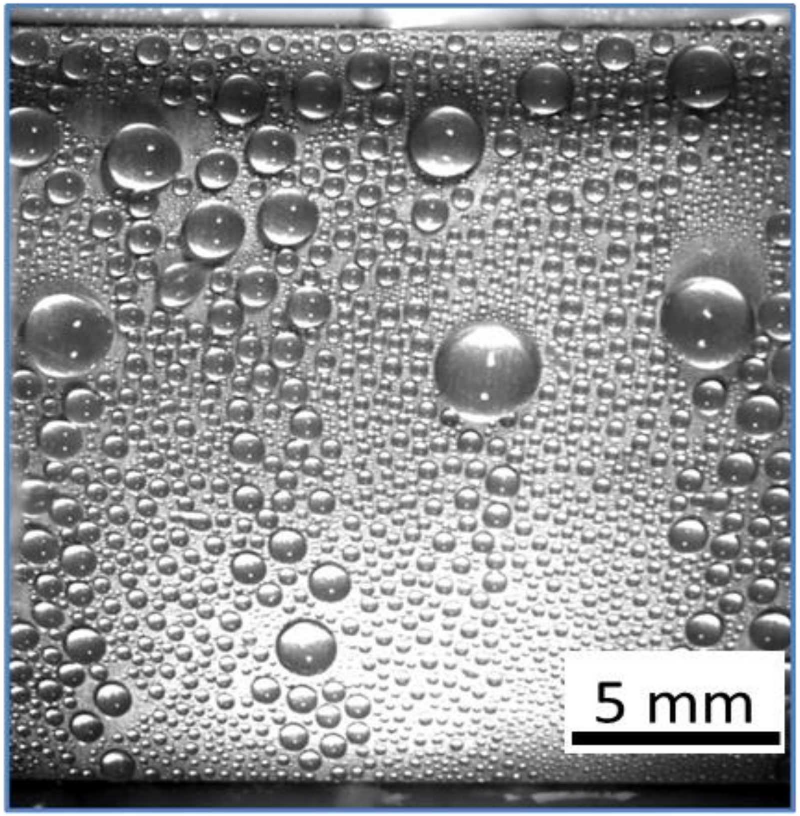


FIGURE Dropwise condensation

Modes of Condensation

2. **Vapor condenses out as droplets suspended in a gas phase to form a fog**

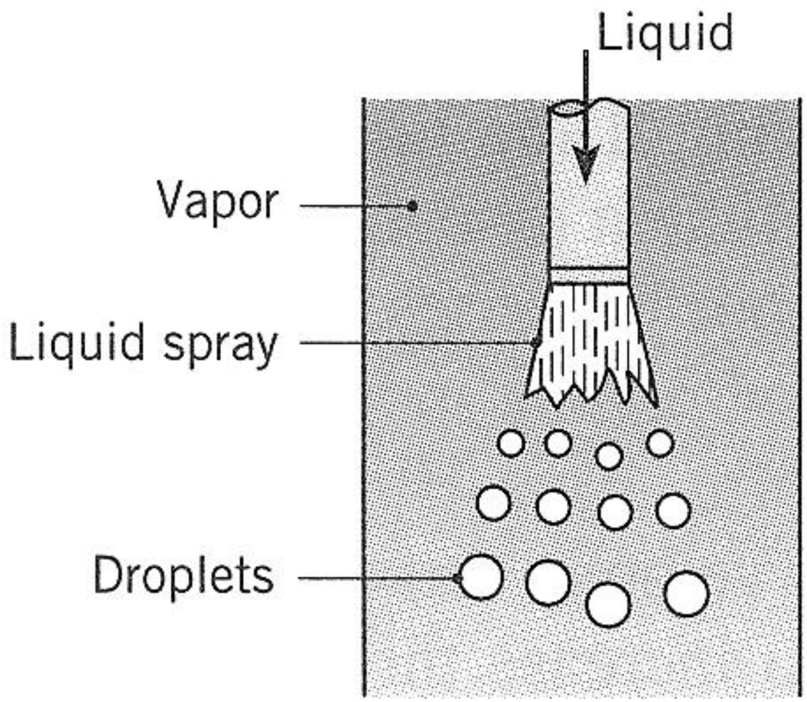
→ **Homogeneous condensation**

FIGURE cloud

Modes of Condensation

2. **Condensation occurs when vapor is brought into contact with a cold liquid**

→ **Direct Condensation**



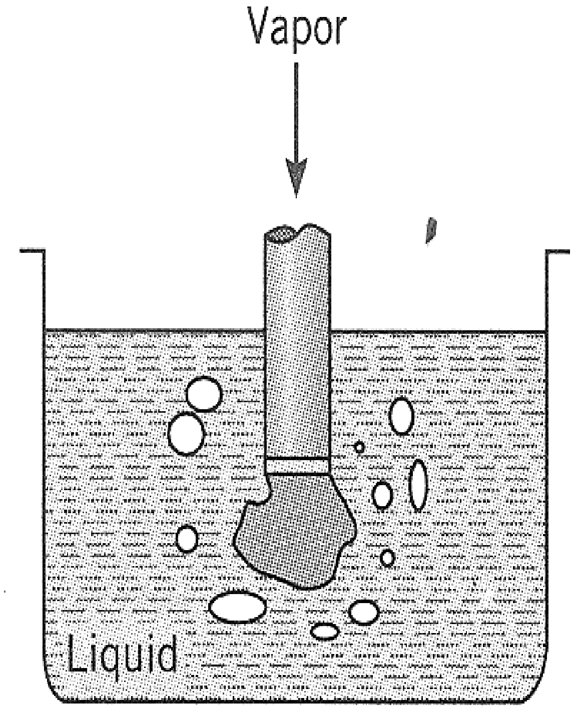


FIGURE Direct contact condensation

Modes of Condensation

The condensation heat transfer coefficient by Nusselt (1916)

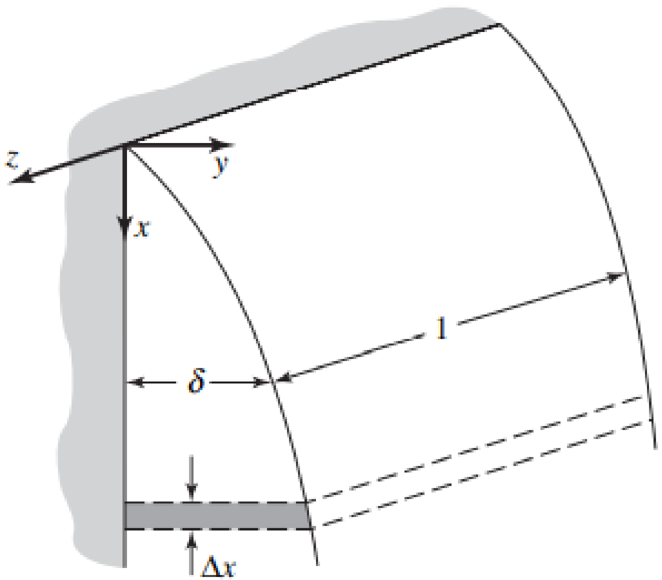


FIGURE Filmwise condensation on a vertical plane

Heat-transfer coefficient, h

$$h_x = \left\{\frac{\rho_L g k^3(\rho_L - \rho_v)\left[h_{fg} + \frac{3}{8}c_{pL}(T_{sat} - T_w)\right]}{4\mu(T_{sat} - T_w)x}\right\}^{\frac{1}{4}}$$

$$h = \frac{1}{L}\int_0^L h_x dx$$

$$h = 0.943\left\{\frac{\rho_L g k^3(\rho_L - \rho_v)\left[h_{fg} + \frac{3}{8}c_{pL}(T_{sat} - T_w)\right]}{L\mu(T_{sat} - T_w)}\right\}^{\frac{1}{4}}$$

Condensers

- **The fluid to which heat is rejected is usually either AIR or WATER.**

- **When the condenser is water-cooled, the water is sent to a cooling tower for ultimate rejection of the heat to the atmosphere.**

- **The water-cooled condenser is favored over the air-cooled condenser, where there is a long distance between the compressor and the point where heat is to be rejected.**

Non-condensables

- **Non-condensables such as air or nitrogen are collected in the condenser when they enter the refrigeration system.**

- **They reduce the efficiency of the system for two reasons.**

  (1) The total pressure in the condenser is elevated.

  → It requires more power for the compressor.

  (2) Instead of diffusing throughout the condenser, the non-condensables cling to the condenser tubes.

  → The condensing surface area is reduced, which also tends to raise the condensing pressure.

Condenser

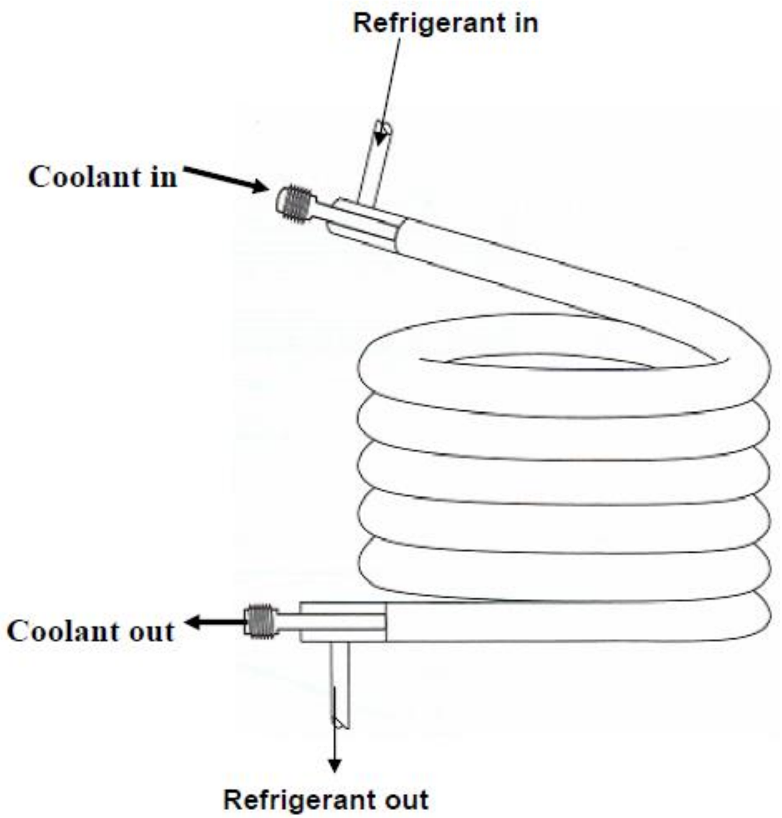


FIGURE Double coil condenser

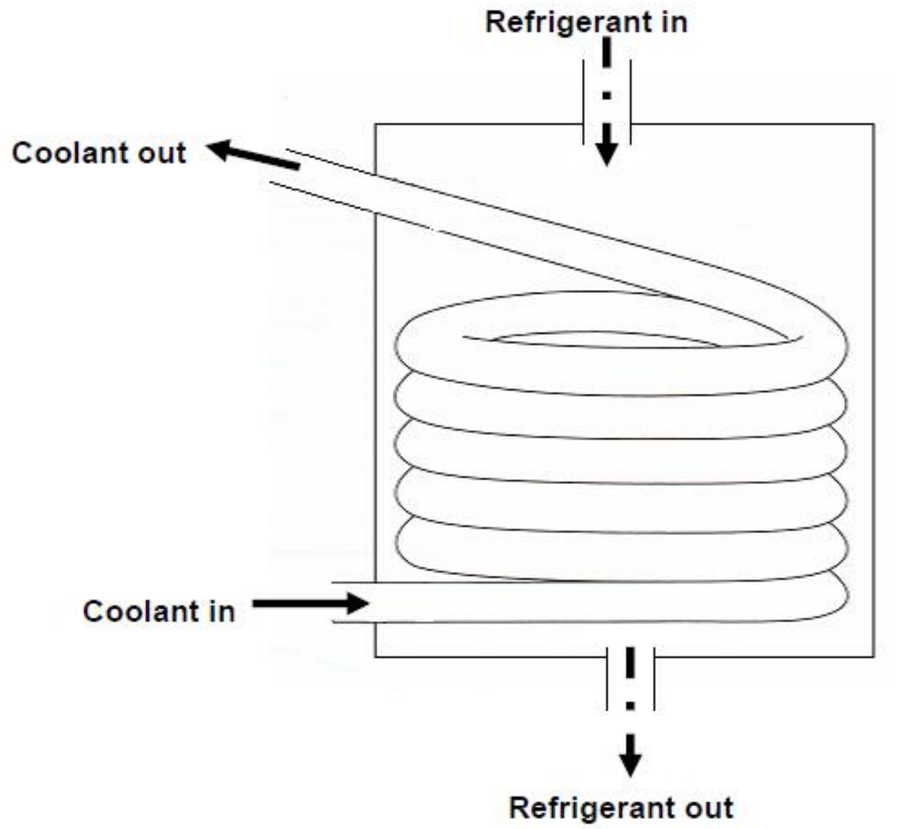
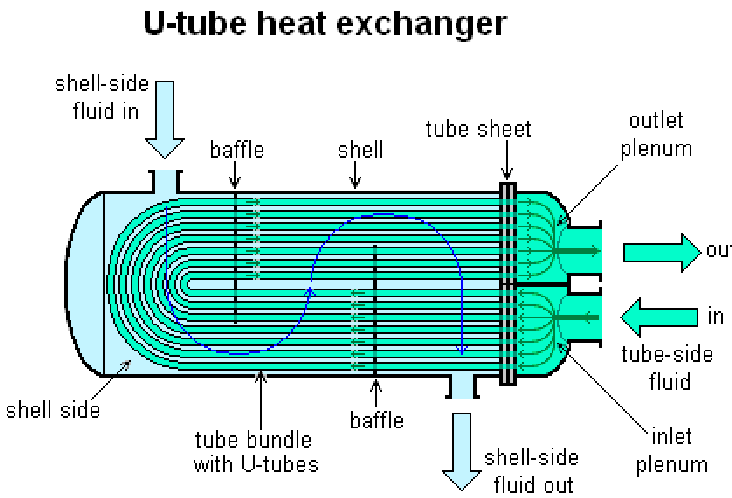
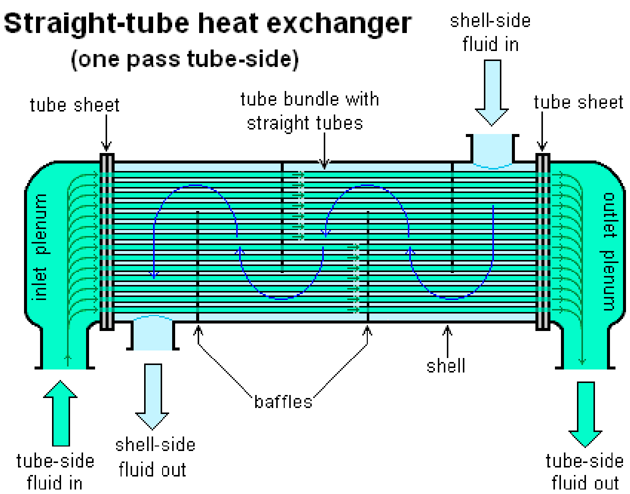
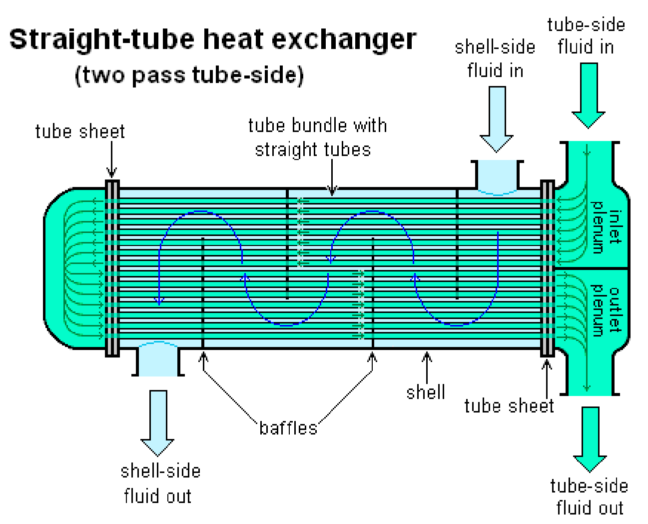


FIGURE Shell and coil condenser

Shell-and-Tube Heat Exchanger







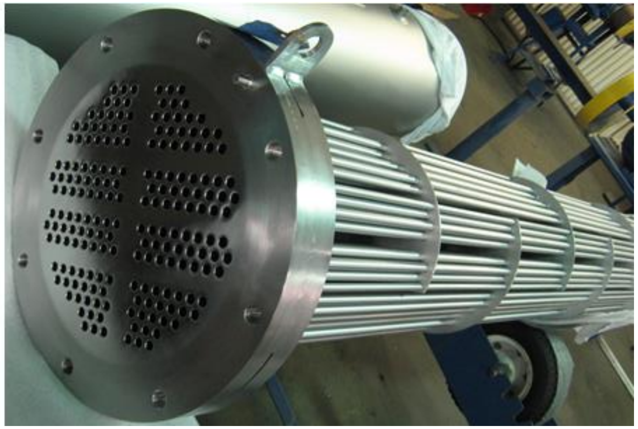


FIGURE Shell-and-tube water-cooled condensers.

Shell-and-Tube Heat Exchanger

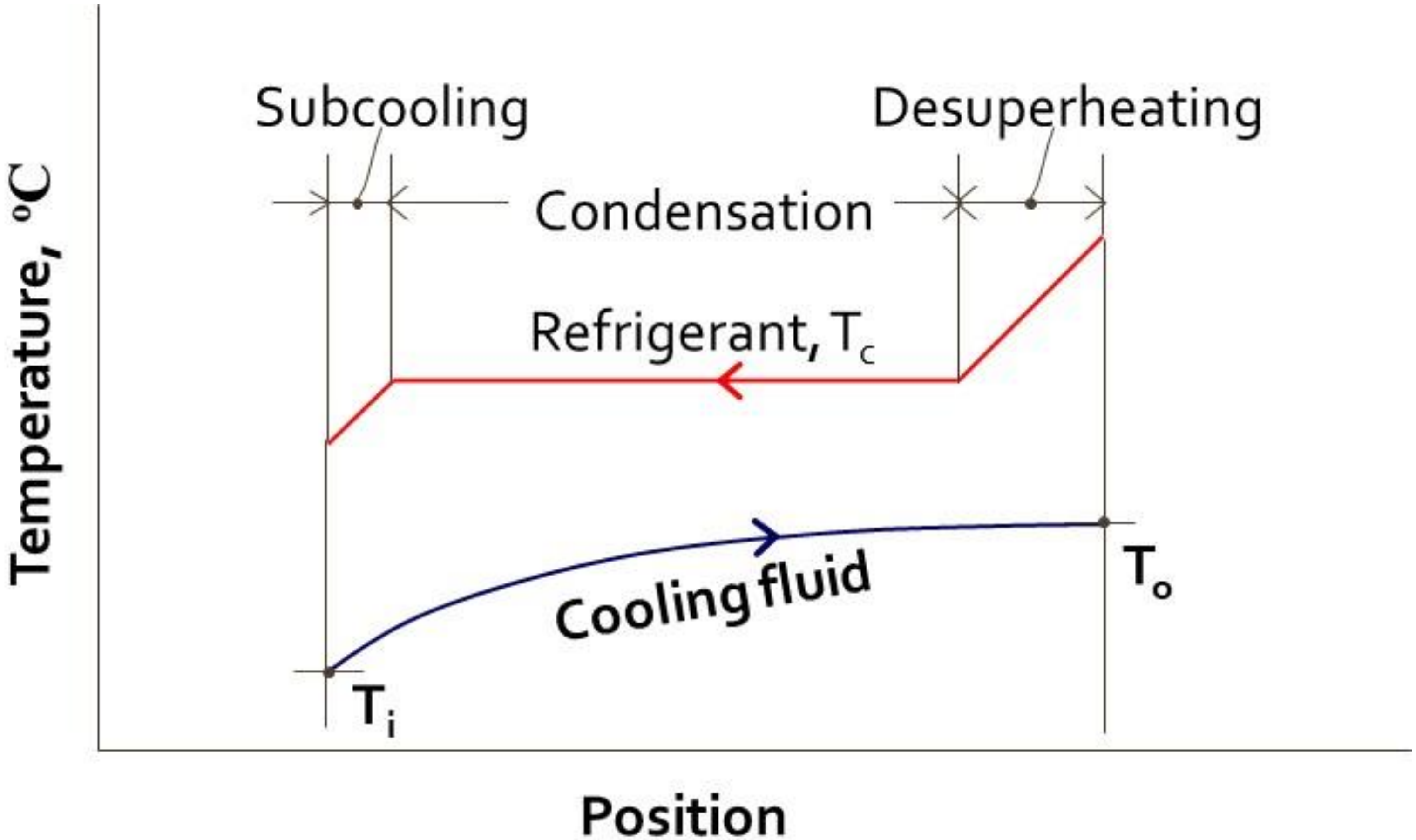


FIGURE Temperature distributions in a condenser

Micro-Fin Tube

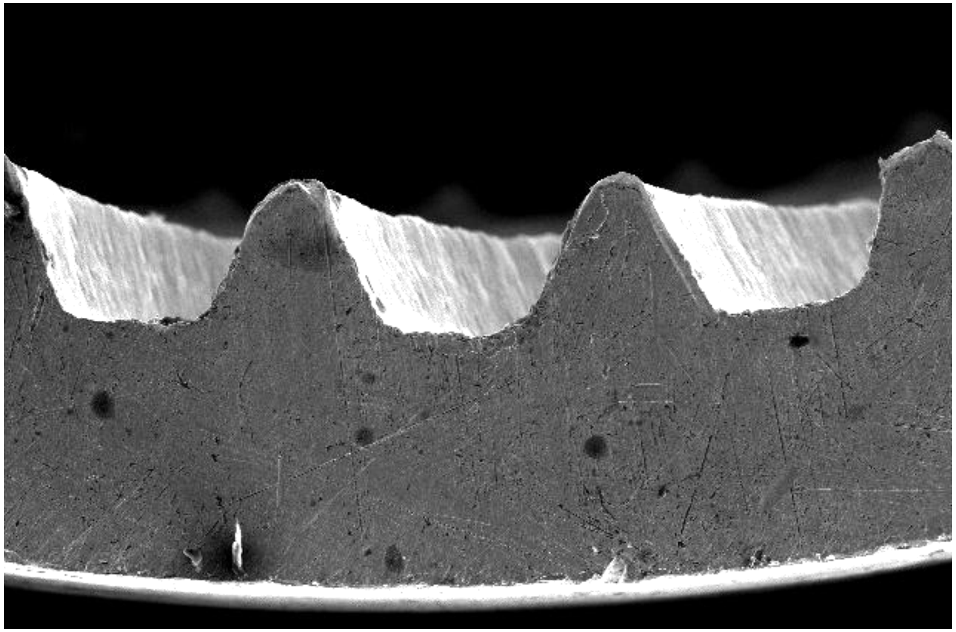


FIGURE Cross-sectional view of micro-fin tube

- ❑ **Reduced volume of the heat exchangers with enhanced surface**
- ❑ **Ensured a larger heat transfer when compared to equivalent smooth tubes**
- ❑ **Heat transfer coefficient ↑ vs. Pressure drop ↑**

Question and Answer Session