

Chapter 10: Boiling and Condensation¹

¹Based on lecture by Yoav Peles, Mech. Aero. Nuc. Eng., RPI.

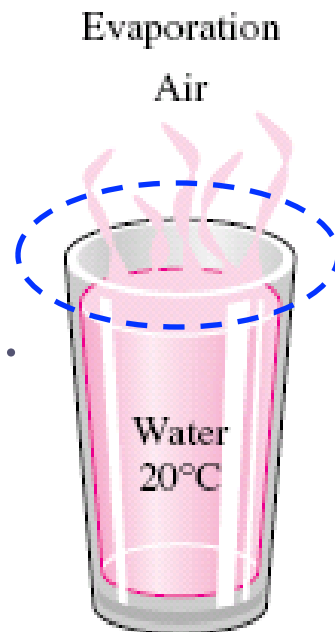
Objectives

When you finish studying this chapter, you should be able to:

- Differentiate between evaporation and boiling, and gain familiarity with different types of boiling,
- Develop a good understanding of the boiling curve, and the different boiling regimes corresponding to different regions of the boiling curve,
- Calculate the heat flux and its critical value associated with nucleate boiling, and examine the methods of boiling heat transfer enhancement,
- Derive a relation for the heat transfer coefficient in laminar film condensation over a vertical plate,
- Calculate the heat flux associated with condensation on inclined and horizontal plates, vertical and horizontal cylinders or spheres, and tube bundles,
- Examine dropwise condensation and understand the uncertainties associated with them.

Boiling Heat Transfer

- **Evaporation** occurs at the *liquid–vapor interface* when the vapor pressure is less than the saturation pressure of the liquid at a given temperature.



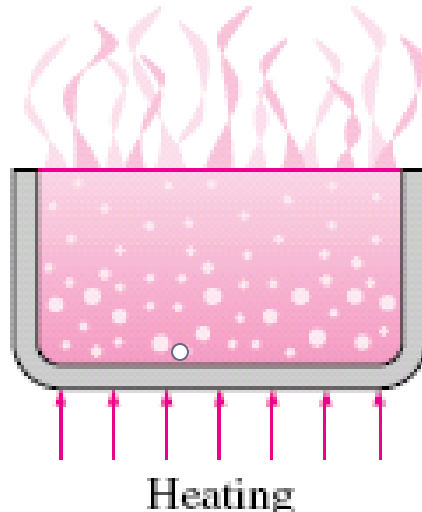
- **Boiling** occurs at the *solid–liquid interface* when a liquid is brought into contact with a surface maintained at a temperature sufficiently above the saturation temperature of the liquid



Classification of boiling

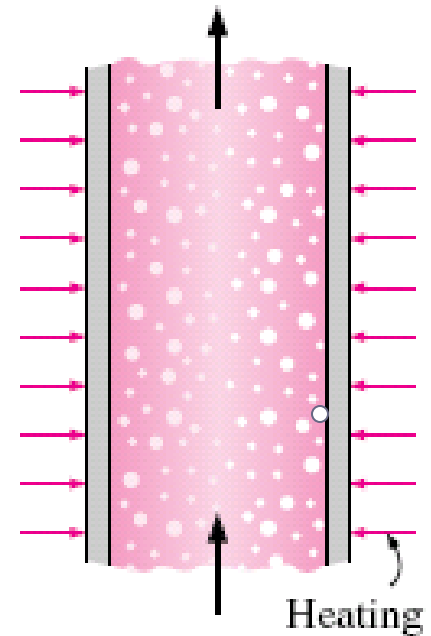
Pool Boiling

- Boiling is called **pool boiling** in the absence of bulk fluid flow.
- Any motion of the fluid is due to natural convection currents and the motion of the bubbles under the influence of buoyancy.



Flow Boiling

- Boiling is called **flow boiling** in the presence of bulk fluid flow.
- In flow boiling, the fluid is forced to move in a heated pipe or over a surface by external means such as a pump.



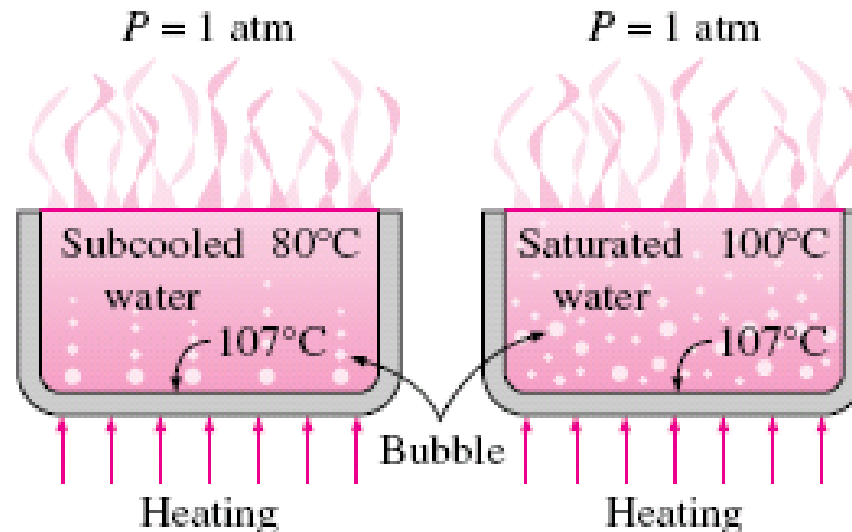
Classification of boiling

Subcooled Boiling

- When the temperature of the main body of the liquid is below the saturation temperature.

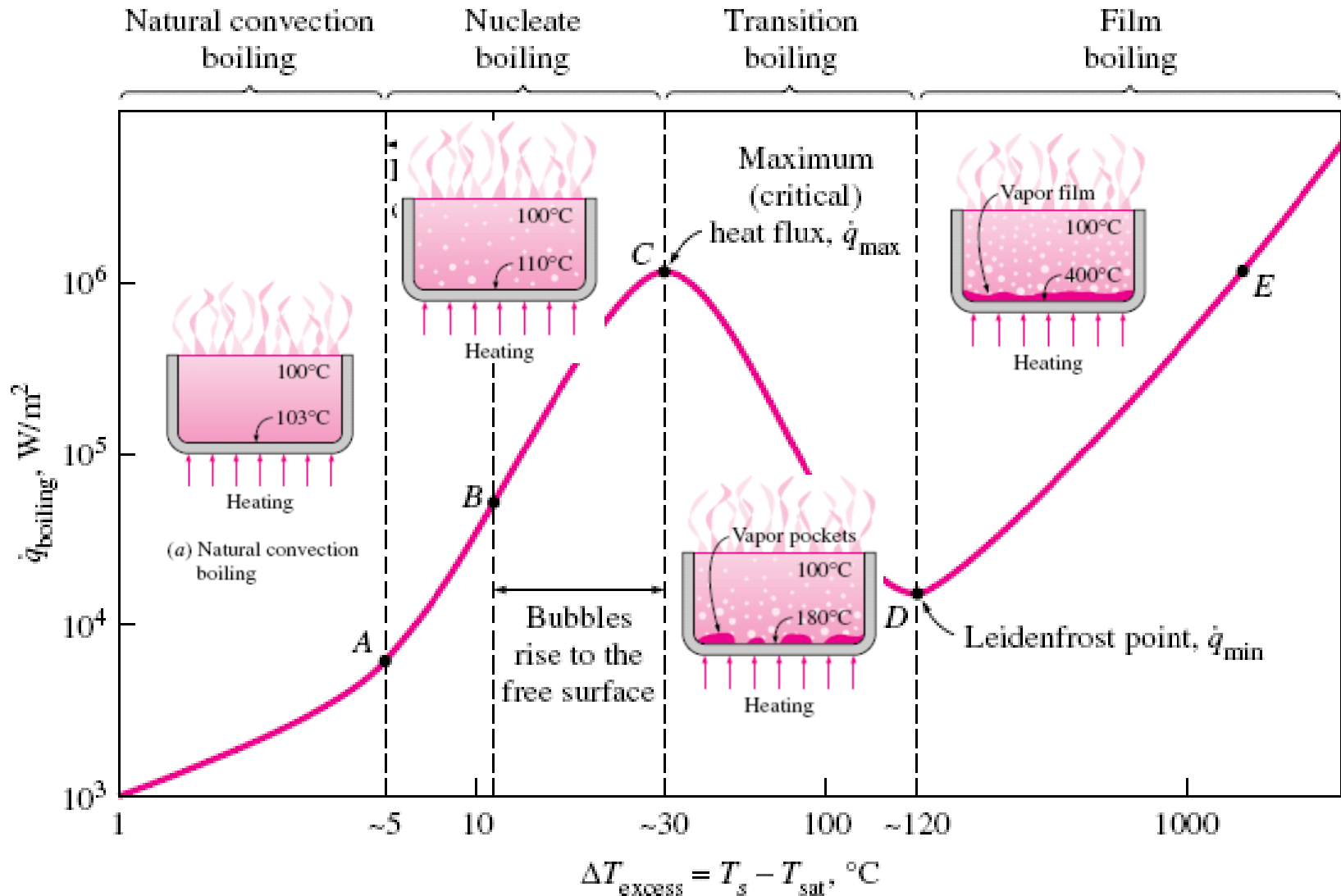
Saturated Boiling

- When the temperature of the liquid is equal to the saturation temperature.



Pool Boiling

Boiling takes different forms, depending on the $DT_{excess} = T_s - T_{sat}$



Natural Convection (to Point *A* on the Boiling Curve)

- Bubbles do not form on the heating surface until the liquid is heated a few degrees above the saturation temperature (about 2 to 6°C for water)

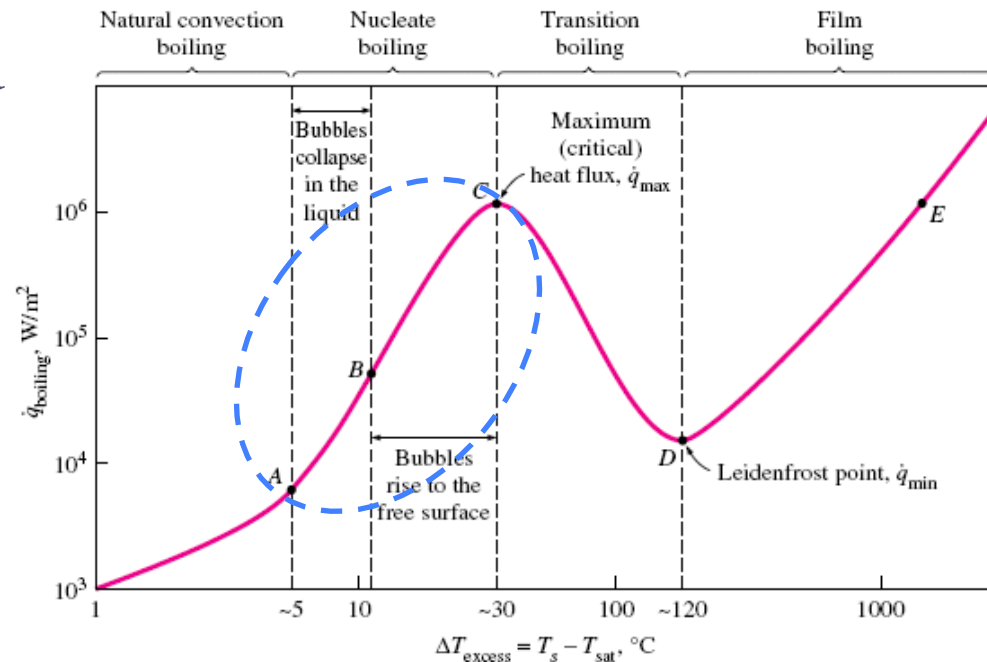
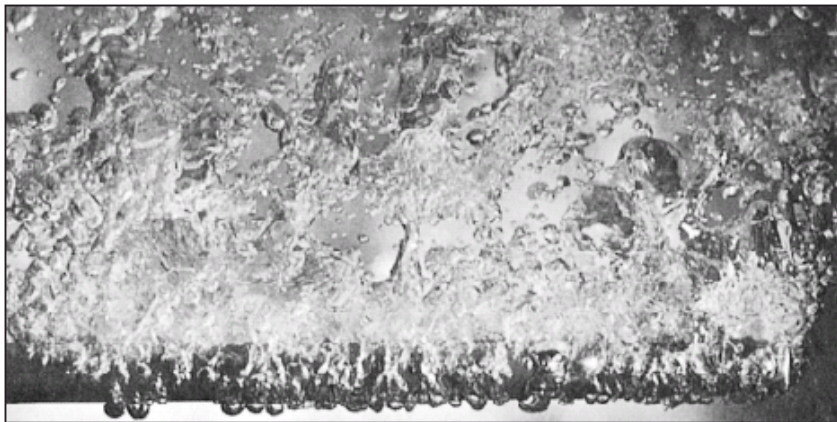


the liquid is slightly *superheated* in this case (*metastable* state).

- The **fluid motion** in this mode of boiling is governed by **natural convection** currents.
- **Heat transfer** from the heating surface to the fluid is by **natural convection**.

Nucleate Boiling

- The bubbles form at an **increasing rate** at an increasing number of nucleation sites as we move along the boiling curve **toward point C**.
- **Region A–B** — *isolated bubbles*.
- **Region B–C** — numerous *continuous columns of vapor* in the liquid

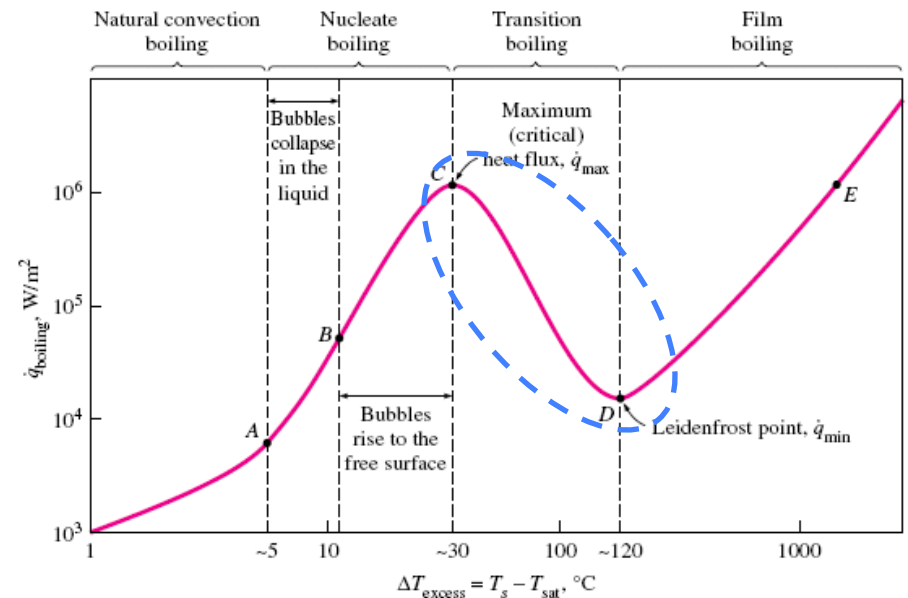
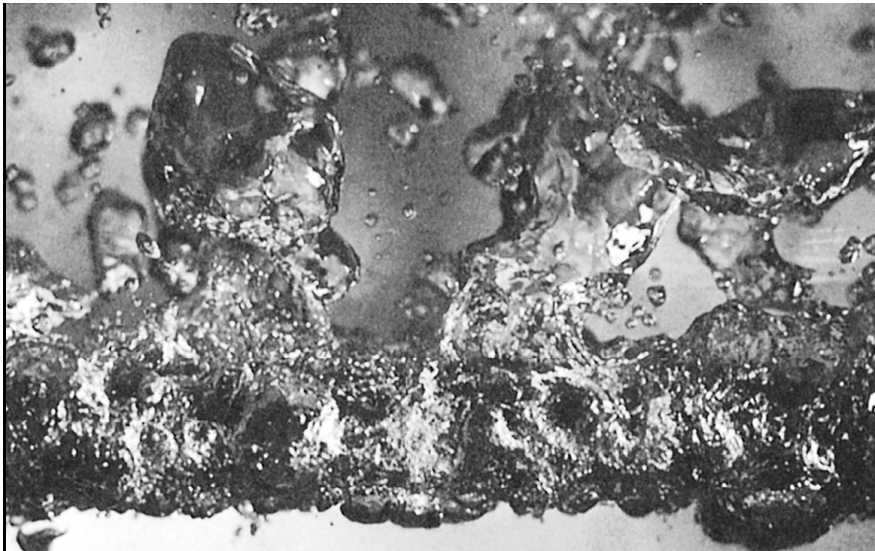


Nucleate Boiling

- In **region A–B** the stirring and agitation caused by the entrainment of the liquid to the heater surface is primarily responsible for the increased heat transfer coefficient.
- In **region A–B** the large heat fluxes obtainable in this region are caused by the combined effect of liquid entrainment and evaporation.
- **After point B** the heat flux increases at a lower rate with increasing DT_{excess} , and reaches a maximum at **point C**.
- The heat flux at this point is called the **critical** (or **maximum**) **heat flux**, and is of prime engineering importance.

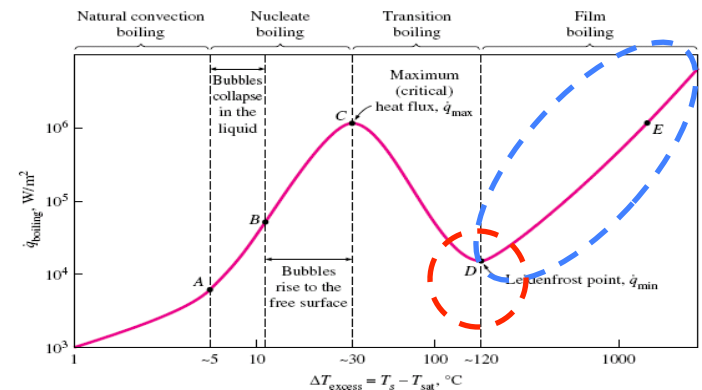
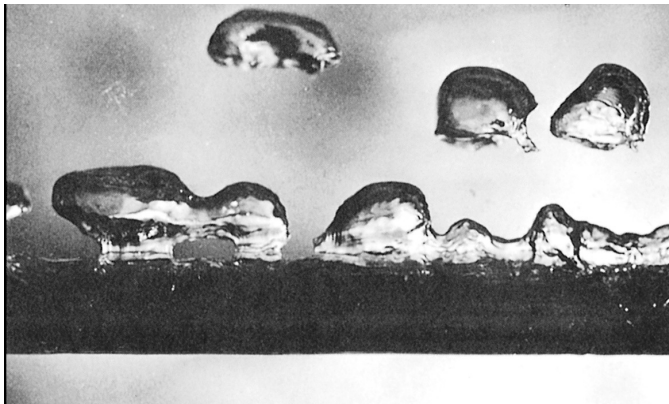
Transition Boiling

- When DT_{excess} is increased past point C , the heat flux decreases.
- This is because a large fraction of the **heater surface is covered by a vapor film**, which acts as an insulation.
- In the transition boiling regime, both nucleate and film boiling partially occur.



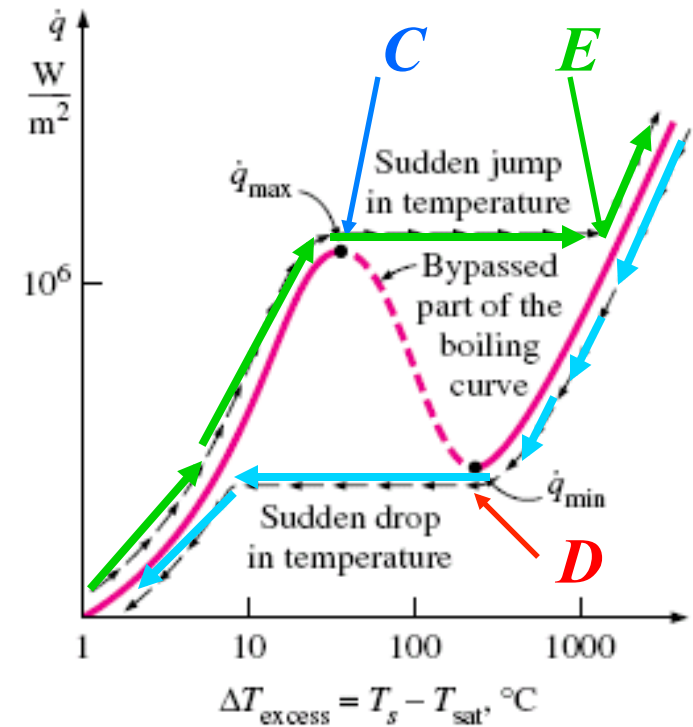
Film Boiling

- Beyond **Point D** the heater surface is completely covered by a continuous stable vapor film.
- **Point D**, where the heat flux reaches a minimum is called the **Leidenfrost point**.
- The presence of a vapor film between the heater surface and the liquid is responsible for the low heat transfer rates in the film boiling region.
- The heat transfer rate increases with increasing excess temperature due to radiation to the liquid.



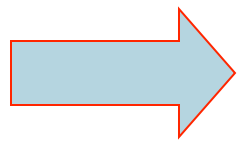
Burnout Phenomenon

- A typical boiling process does not follow the boiling curve beyond **point C**.
- When the power applied to the heated surface exceeded the value at **point C** even slightly, the surface temperature increased suddenly to **point E**.
- When the power is reduced gradually starting from **point E** the cooling curve follows Fig. 10–8 with a sudden drop in excess temperature when **point D** is reached.



Heat Transfer Correlations in Pool Boiling

- Boiling regimes differ considerably in their character



different heat transfer relations need to be used for different boiling regimes.

- In the *natural convection boiling* regime heat transfer rates can be accurately determined using natural convection relations.

Heat Transfer Correlations in Pool Boiling — Nucleate Boiling

- No general theoretical relations for heat transfer in the nucleate boiling regime is available.
- Experimental based correlations are used.
- The rate of heat transfer strongly depends on the nature of nucleation and the type and the condition of the heated surface.
- A widely used correlation proposed in 1952 by

Rohsenow:

$$q_s'' = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma} \right]^{1/2} \left(\frac{c_{p,l} \Delta T_e}{C_{s,f} h_{fg} \text{Pr}_l^n} \right)^3$$

Heat Transfer Correlations in Pool Boiling — Nucleate Boiling

- The values in Rohsenow equation can be used for *any geometry* since it is found that the rate of heat transfer during nucleate boiling is essentially independent of the geometry and orientation of the heated surface.
- The correlation is applicable to *clean* and relatively *smooth* surfaces.
- Error for the heat transfer rate for a given excess temperature: 100%.
- Error for the excess temperature for a given heat transfer rate for the heat transfer rate and by 30%.

Critical Heat Flux (CHF)

- The *maximum* (or *critical*) *heat flux* in nucleate pool boiling was determined theoretically by S. S. Kutateladze in Russia in 1948 and N. Zuber in the United States in 1958 to be:

$$q''_{\max} = Ch_{fg}\rho_v \left[\frac{\sigma g (\rho_l - \rho_v)}{\rho_v^2} \right]^{1/4}$$

C is a constant whose value depends on the heater geometry, but generally is about 0.15.

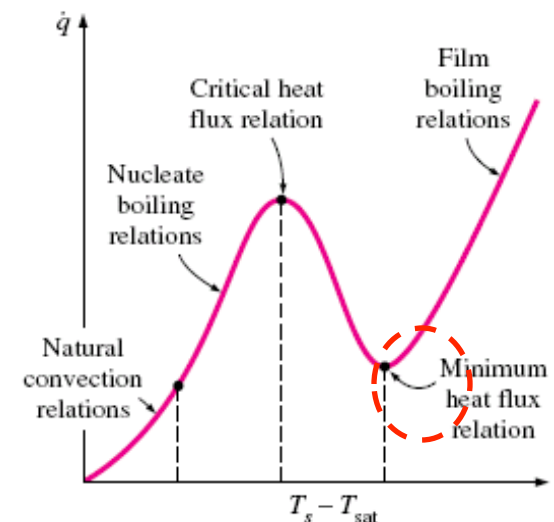
- The CHF is independent of the fluid–heating surface combination, as well as the viscosity, thermal conductivity, and the specific heat of the liquid.
- The CHF increases with pressure up to about one-third of the critical pressure, and then starts to decrease and becomes zero at the critical pressure.
- The CHF is proportional to h_{fg} , and large maximum heat fluxes can be obtained using fluids with a large enthalpy of vaporization, such as water.

Minimum Heat Flux

- **Minimum heat flux**, which occurs at the **Leidenfrost point**, is of practical interest since it represents the lower limit for the heat flux in the film boiling regime.
- Zuber derived the following expression for the minimum heat flux for a *large horizontal plate*

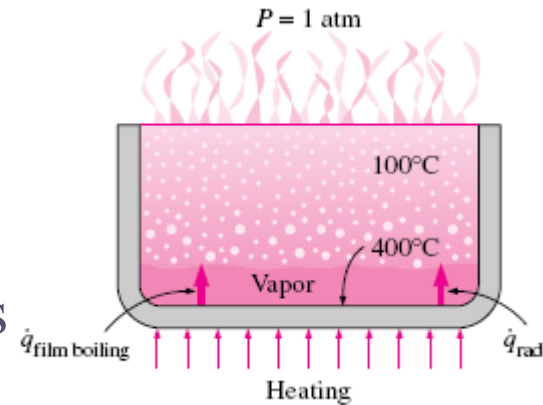
$$q''_{\min} = 0.09 \rho_v h_{fg} \left[\frac{\sigma g (\rho_l - \rho_v)}{(\rho_l + \rho_v)^2} \right]^{1/4}$$

- the relation above can be in error by 50% or more.



Film Boiling

- The Nusselt number for film boiling on a *horizontal cylinder or sphere* of diameter D is



$$\bar{Nu}_D = \frac{\bar{h}_{conv} D}{k_v} = C \left[\frac{g(\rho_l - \rho_v) h'_{fg} D^3}{\nu_v k_v (T_s - T_{sat})} \right]^{1/4}, \quad C = 0.62 \text{ (cylinder)}$$

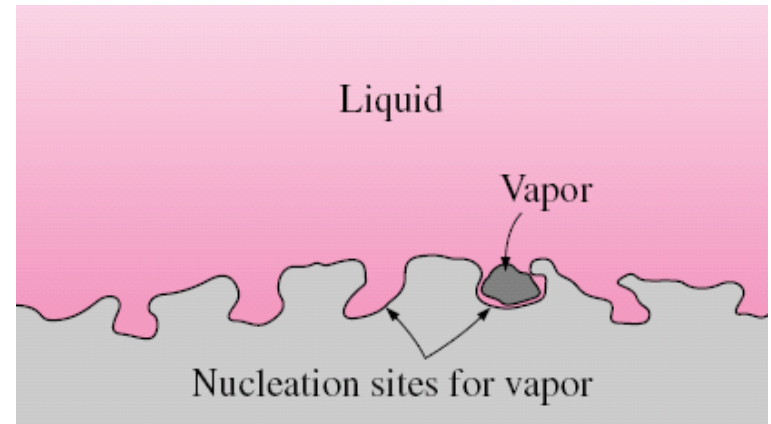
$$C = 0.67 \text{ (sphere)}$$

- At high surface temperatures (typically above 300°C), heat transfer across the vapor film by *radiation* becomes significant and needs to be considered.
- The two mechanisms of heat transfer (radiation and convection) adversely affect each other, causing the total heat transfer to be less than their sum.

$$\bar{h}^{4/3} = \bar{h}_{conv}^{4/3} + \bar{h}_{rad} \bar{h}^{1/3}, \quad \bar{h}_{rad} = \frac{\epsilon \sigma (T_s^4 - T_{sat}^4)}{T_s - T_{sat}}$$

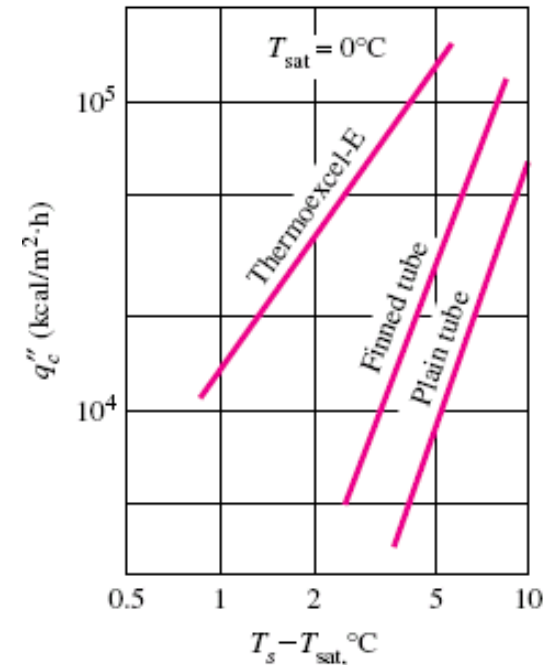
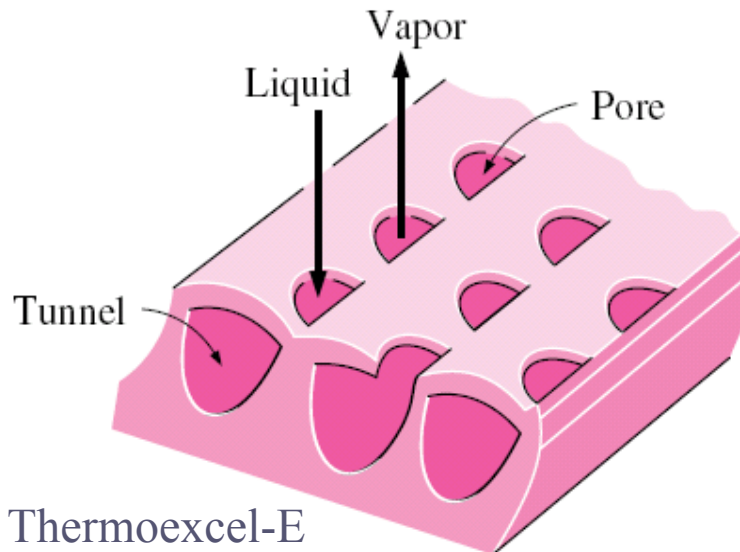
Enhancement of Heat Transfer in Pool Boiling

- The **rate of heat transfer** in the nucleate boiling regime strongly depends on the number of active **nucleation sites** on the surface, and the **rate of bubble formation** at each site.
- Therefore, modification that **enhances nucleation** on the heating surface will also **enhance heat transfer** in nucleate boiling.
- **Irregularities** on the heating surface, including roughness and dirt, serve as additional **nucleation sites** during boiling.
- The effect of **surface roughness** is observed to **decay with time**.



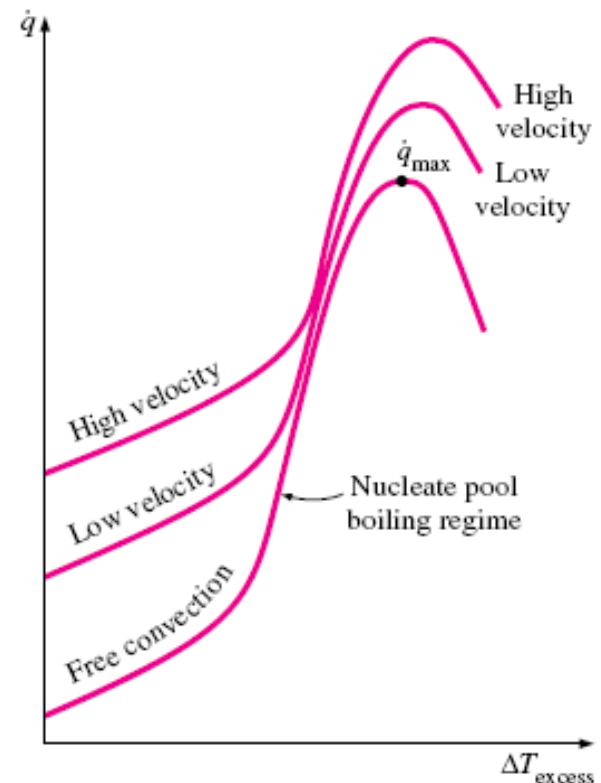
Enhancement of Heat Transfer in Pool Boiling

- Surfaces that provide enhanced heat transfer in nucleate boiling *permanently* are being manufactured and are available in the market.
- **Heat transfer** can be **enhanced** by a factor of up to 10 during nucleate boiling, and the **critical heat flux** by a factor of 3.



External Forced Convection Boiling (Flow Boiling)

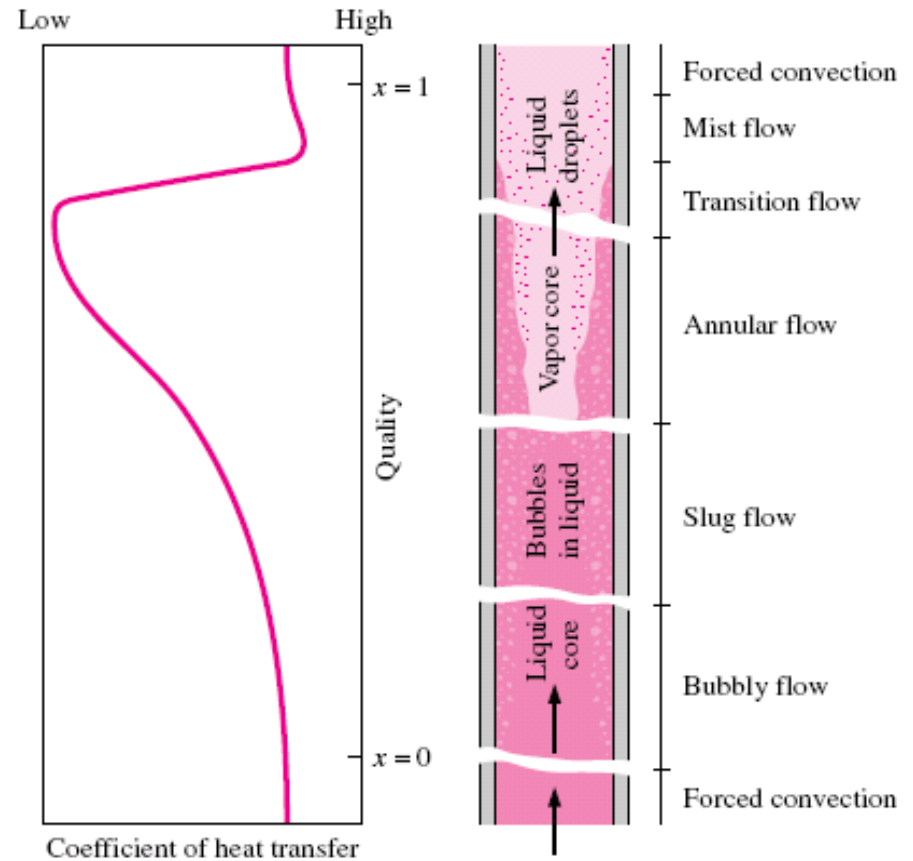
- In **flow boiling**, the fluid is forced to move by an external source such as a pump as it undergoes a phase-change process.
- The boiling in this case exhibits the combined effects of convection and pool boiling.
- Flow boiling is classified as either *external* and *internal flow boiling*.
- *External flow* — the higher the velocity, the higher the nucleate boiling heat flux and the critical heat flux.



Flow Boiling — Internal Flow

- The two-phase flow in a tube exhibits different flow boiling regimes, depending on the relative amounts of the liquid and the vapor phases.
- Typical flow regimes:
 - Liquid single-phase flow,
 - Bubbly flow,
 - Slug flow,
 - Annular flow,
 - Mist flow,
 - Vapor single-phase flow.

Axial position in the tube



Flow Boiling — Internal Flow

- **Liquid single-phase flow**
 - In the inlet region the liquid is subcooled and heat transfer to the liquid is by *forced convection* (assuming no subcooled boiling).
- **Bubbly flow**
 - Individual bubbles
 - Low mass qualities
- **Slug flow**
 - Bubbles coalesce into slugs of vapor.
 - Moderate mass qualities
- **Annular flow**
 - Core of the flow consists of vapor only, and liquid adjacent to the walls.
 - Very high heat transfer coefficients
- **Mist flow**
 - a sharp decrease in the heat transfer coefficient
- **Vapor single-phase flow**
 - The liquid phase is completely evaporated and vapor is superheated.

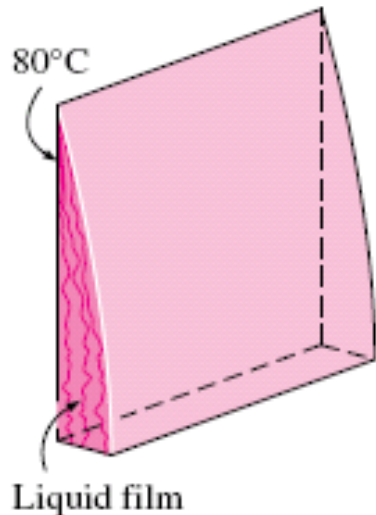
Condensation

- Condensation occurs when the temperature of a vapor is reduced *below* its saturation temperature.
- Only condensation on solid surfaces is considered in this chapter.
- Two forms of condensation:
 - *Film condensation*,
 - *Dropwise condensation*.

Condensation: Physical Mechanisms

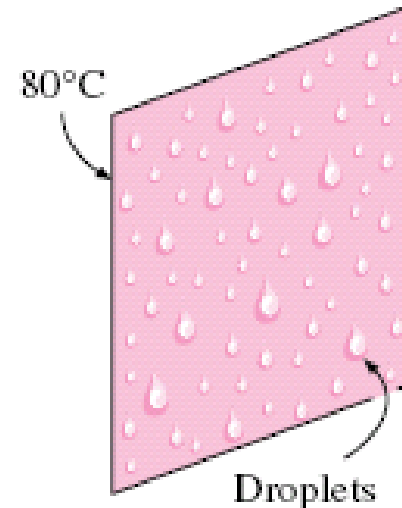
Film condensation

- The condensate wets the surface and forms a liquid film.
- The surface is blanketed by a liquid film which serves as a *resistance* to heat transfer.



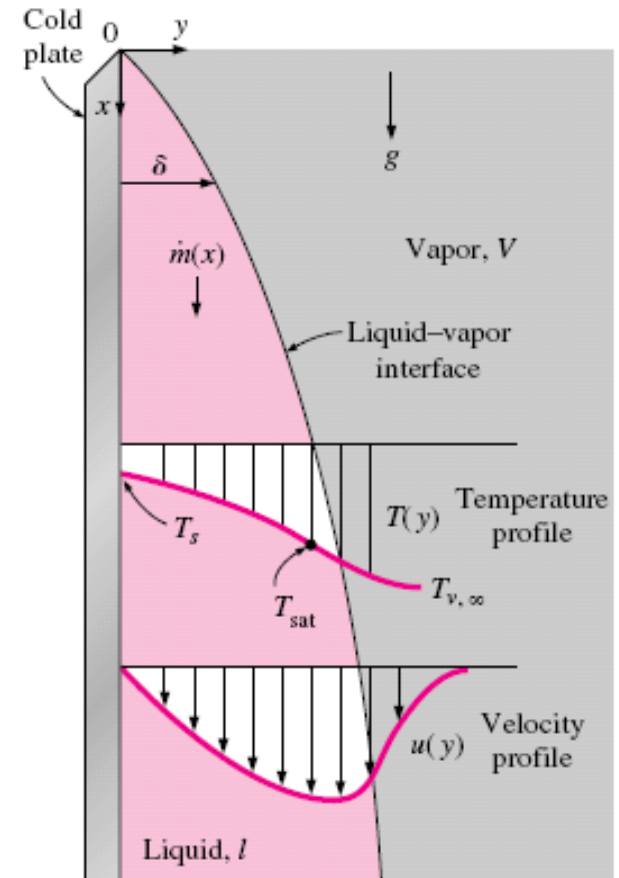
Dropwise condensation

- The condensed vapor forms droplets on the surface.
- The droplets slide down when they reach a certain size.
- No liquid film to resist heat transfer.
- As a result, heat transfer rates that are more than 10 times larger than with film condensation can be achieved.



Film Condensation on a Vertical Plate

- liquid film starts forming at the top of the plate and flows downward under the influence of gravity.
- δ increases in the flow direction x
- Heat in the amount h_{fg} is released during condensation and is transferred through the film to the plate surface.
- T_s must be below the saturation temperature for condensation.
- The temperature of the condensate is T_{sat} at the interface and decreases gradually to T_s at the wall.

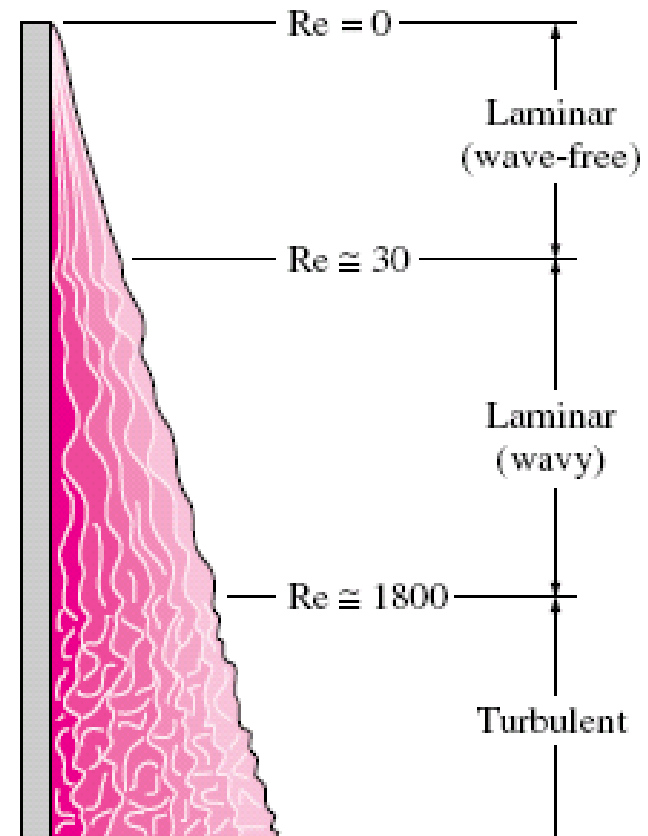


Vertical Plate — Flow Regimes

- The dimensionless parameter controlling the transition between regimes is the **Reynolds number** defined as:

$$Re_x = \frac{\overbrace{(4\delta)}^{\text{hydraulic diameter } (D_h)} \rho_l u_m}{\mu_l}$$

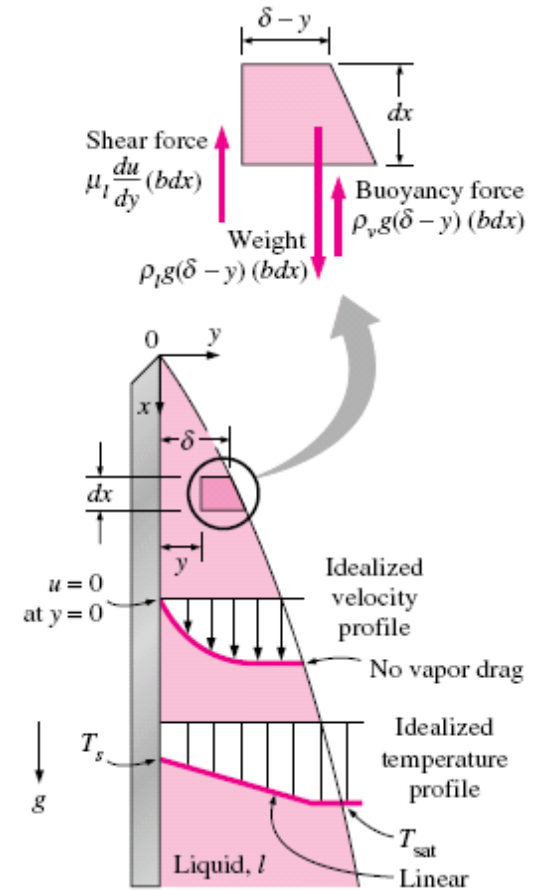
- Three prime flow regimes:
 - $Re < 30$ — **Laminar** (wave-free),
 - $30 < Re < 1800$ — **Wavy-laminar**,
 - $Re > 1800$ — **Turbulent**.
- The **Reynolds number** increases in the flow direction.



Heat Transfer Correlations for Film Condensation — Vertical wall

Assumptions:

1. Both the plate and the vapor are maintained at *constant temperatures* of T_s and T_{sat} , respectively, and the temperature across the liquid film varies *linearly*.
2. Heat transfer across the liquid film is by pure *conduction*.
3. The velocity of the vapor is low (or zero) so that it exerts *no drag* on the condensate (no viscous shear on the liquid–vapor interface).
4. The flow of the condensate is *laminar* ($Re < 30$) and the properties of the liquid are constant.
5. The acceleration of the condensate layer is negligible.



Height L and width b

Hydrodynamics

- Newton's second law of motion

Weight = Viscous shear force + Buoyancy force

or

$$\rho_l g (\delta - y) (bdx) = \mu_l \frac{du}{dy} (bdx) + \rho_v g (\delta - y) (bdx)$$

- Canceling the plate width b and solving for du/dy

$$\frac{du}{dy} = \frac{g(\rho_l - \rho_v)(\delta - y)}{\mu_l}$$

- Integrating from $y=0$ ($u=0$) to y ($u=u(y)$)

$$u(y) = \frac{g(\rho_l - \rho_v)\delta^2}{\mu_l} \left(\frac{y}{\delta} - \frac{1}{2} \left(\frac{y}{\delta} \right)^2 \right) \quad (10-18)$$

- The mass flow rate of the condensate at a location x is determined from

$$\dot{m}(x) = \int_A \rho_l u(y) dA = b \int_{y=0}^{\delta(x)} \rho_l u(y) dy = \Gamma(x) \quad (10-19)$$

Substituting $u(y)$ from Eq. 10–18, we get

$$\Gamma(x) = \frac{g\rho_l(\rho_l - \rho_v)\delta^3}{3\mu_l} \quad (10-20)$$

whose derivative with respect to x is

$$\frac{d\Gamma}{dx} = \frac{g\rho_l(\rho_l - \rho_v)\delta^2}{\mu_l} \frac{d\delta}{dx} \quad (10-25)$$

Thermal Considerations

- The rate of heat transfer from the vapor to the plate through the liquid film

$$dq = h_{fg} d\dot{m} = k_l (b dx) \frac{T_{sat} - T_s}{\delta}$$
$$\rightarrow \frac{1}{b} \frac{d\dot{m}}{dx} = \frac{d\Gamma}{dx} = \frac{k_l}{h_{fg}} \frac{T_{sat} - T_s}{\delta} \quad (10-24)$$

- Equating Eqs. 10–24 and 10–25 and separating the variables give

$$\delta^3 d\delta = \frac{\mu_l k_l (T_{sat} - T_s)}{g \rho_l (\rho_l - \rho_v) h_{fg}} dx$$

- Integrating from $x=0$ ($d=0$) to x ($d=d(x)$), the liquid film thickness at x is determined to be

$$\delta(x) = \left[\frac{4\mu_l k_l (T_{sat} - T_s) x}{g \rho_l (\rho_l - \rho_v) h_{fg}} \right]^{1/4} \quad (10-26)$$

- Rohsenow recommended using the modified latent heat

- Rohsenow recommended using the modified latent heat

$$h'_{fg} = h_{fg} + 0.68c_{p,l}(T_{sat} - T_s) = h_{fg}(1 + 0.68Ja)$$

Where the **Jacob** number is defined as

$$Ja = \frac{c_p(T_s - T_{sat})}{h_{fg}}$$

- Thus

$$\delta(x) = \left[\frac{4\mu_l k_l (T_{sat} - T_s) x}{g\rho_l(\rho_l - \rho_v)h'_{fg}} \right]^{1/4} \quad (10-26)$$

- Since the heat transfer across the liquid film is assumed to be by pure conduction, the heat transfer coefficient can be expressed through **Newton's law of cooling** and **Fourier law** as

$$\dot{q}_x = h_x (T_{sat} - T_s) = k_l \frac{T_{sat} - T_s}{\delta} \quad \rightarrow \quad h_x = \frac{k_l}{\delta} \quad (10-29)$$

- Substituting $d(x)$ from Eq. 10-26, the local heat transfer coefficient is determined to be

$$h_x = \left[\frac{g \rho_l (\rho_l - \rho_v) h'_{fg} k_l^3}{4 \mu_l (T_{sat} - T_s) x} \right]^{1/4} \quad (10-30)$$

- The average heat transfer coefficient over the entire plate is

$$\bar{h}_L = \frac{1}{L} \int_0^L h_x dx = \frac{4}{3} h_{x=L} = 0.943 \left[\frac{g \rho_l (\rho_l - \rho_v) h'_{fg} k_l^3}{\mu_l (T_{sat} - T_s) L} \right]^{1/4} \quad (10-32)$$

- The Nusselt number:

$$\bar{Nu}_L = \frac{\bar{h}_L L}{k_l} = 0.943 \left[\frac{g \rho_l (\rho_l - \rho_v) h'_{fg} L^3}{\mu_l k_l (T_{sat} - T_s)} \right]^{1/4} \quad (10-32)$$

- The liquid properties are evaluated at the film temperature $T_f = (T_{sat} + T_s)/2$.
- The vapor density and latent heat of vaporization h'_{fg} are evaluated at T_{sat} .

Turbulent Flow on Vertical Plates

- Define the Reynolds number

$$\text{Re}_\delta \equiv \frac{4\Gamma}{\mu_l} = \frac{4\dot{m}}{b\mu_l} = \frac{4\rho_l u_m \delta}{\mu_l} \quad (10-36)$$

- Flow regimes are:

– Laminar wave-free flow: $\text{Re}_\delta \leq 30$

– Laminar wavy flow: $30 < \text{Re}_\delta \leq 1800$

– Turbulent flow: $\text{Re}_\delta > 1800$

Turbulent Flow on Vertical Plates

- Laminar waver-free regime:

$$\bar{Nu}_L = \frac{\bar{h}_L (v_l^2 / g)^{1/3}}{k_l} = 1.46 \text{Re}_\delta^{-1/3}, \quad \text{Re}_\delta \leq 30 \quad (10-38)$$

- Laminar wavy regime:

$$\bar{Nu}_L = \frac{\bar{h}_L (v_l^2 / g)^{1/3}}{k_l} = \frac{\text{Re}_\delta}{1.08 \text{Re}_\delta^{1.22} - 5.2}, \quad 30 < \text{Re}_\delta \leq 1800 \quad (10-39)$$

- Turbulent regime:

$$\bar{Nu}_L = \frac{\bar{h}_L (v_l^2 / g)^{1/3}}{k_l} = \frac{\text{Re}_\delta}{8750 + 58 \text{Pr}_l^{-0.5} (\text{Re}_\delta^{0.75} - 253)}, \quad (10-39)$$

$$\text{Re}_\delta > 1800 \text{ and } \text{Pr}_l \geq 1$$

Nondimensionalized Heat Transfer Coefficients

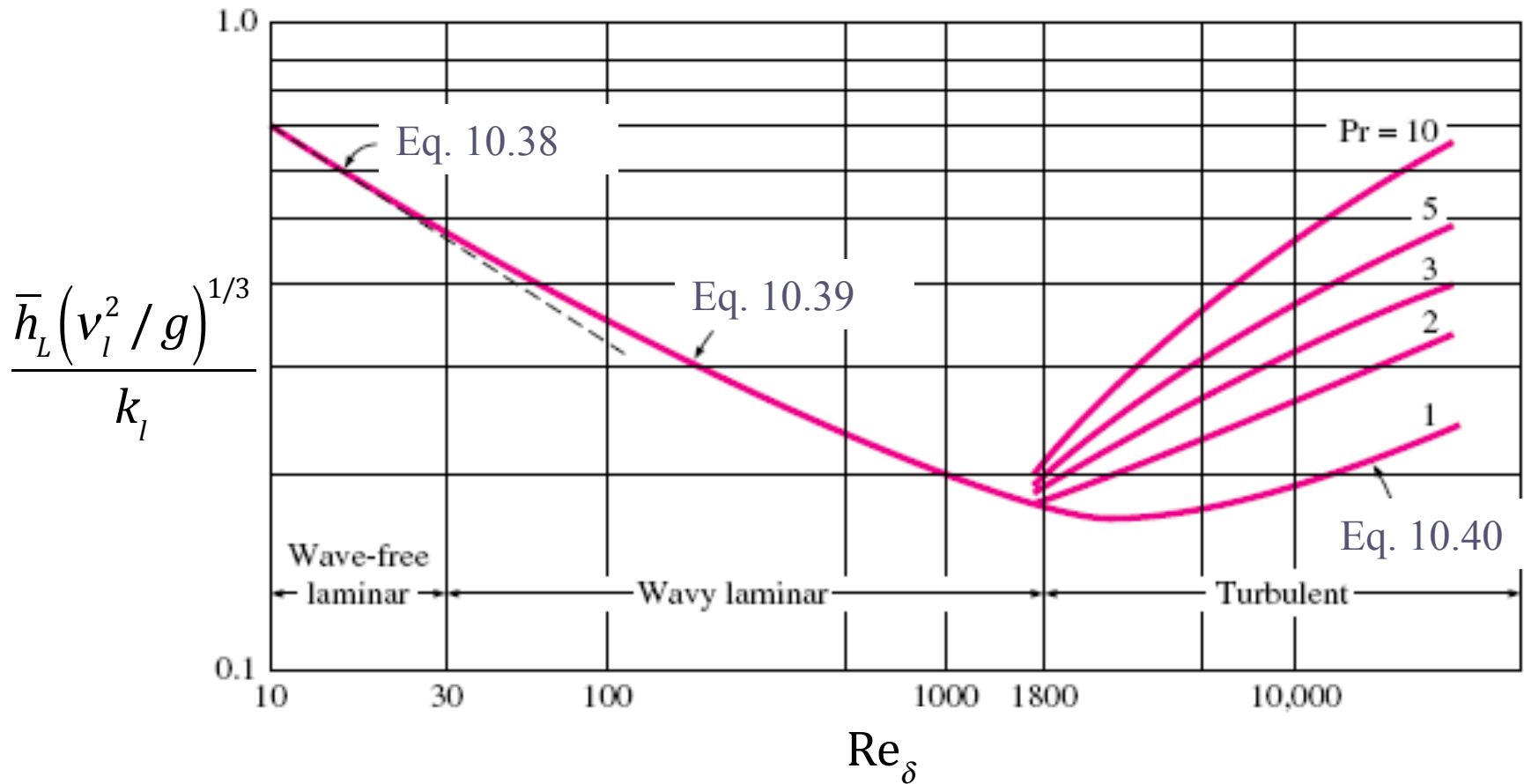
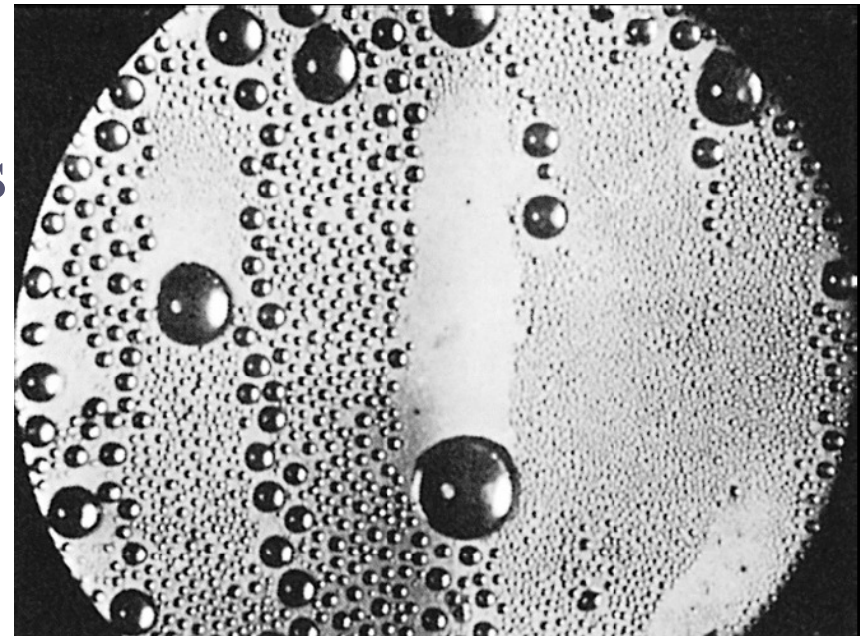


Figure 10.13: Modified Nusselt number for condensation on a vertical plate.

Dropwise Condensation

- One of the **most effective mechanisms** of **heat transfer**, and extremely large heat transfer coefficients can be achieved.
- **Small droplets** grow as a result of continued condensation, coalesce into large droplets, and **slide down** when they reach a certain size.
- **Large heat transfer** coefficients enable designers to achieve a specified heat transfer rate with a **smaller surface area**.



Dropwise Condensation

- The **challenge** in dropwise condensation is not to achieve it, but rather, to *sustain* it for prolonged periods of time.
- Dropwise condensation has been studied experimentally for a number of surface–fluid combinations.
- Griffith (1983) recommends these simple correlations for dropwise condensation of *steam* on *copper surfaces*:

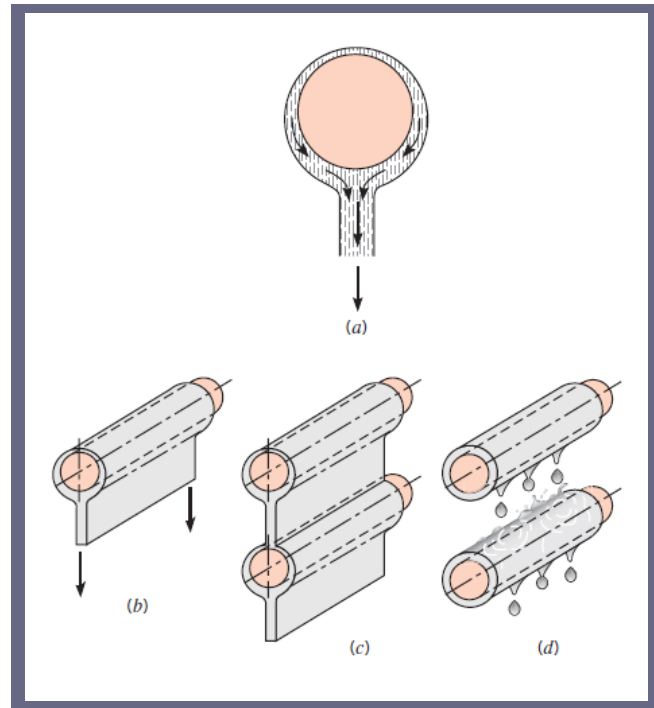
$$h_{dropwise} = \begin{cases} 51,104 + 2044T_{sat} & 22^{\circ}\text{C} < T_{sat} < 100^{\circ}\text{C} \\ 255,310 & T_{sat} > 100^{\circ}\text{C} \end{cases}$$

Wavy Laminar Flow on Vertical Plates

- The **waves** at the liquid–vapor interface tend to **increase heat transfer**.
- Knowledge is based on experimental studies.
- The **increase** in **heat transfer** due to the **wave** effect is, on average, about **20 percent**, but it can exceed 50 percent.
- Based on his experimental studies, Kutateladze (1963) recommended the following relation

$$h_{avg,wavy} = \frac{Re k_l}{1.08 Re^{1.22} - 5.2} \left(\frac{g}{v_l^2} \right)^{1/3} \quad ; \quad \rho_v = \rho_l$$

Film Condensation on Radial Systems

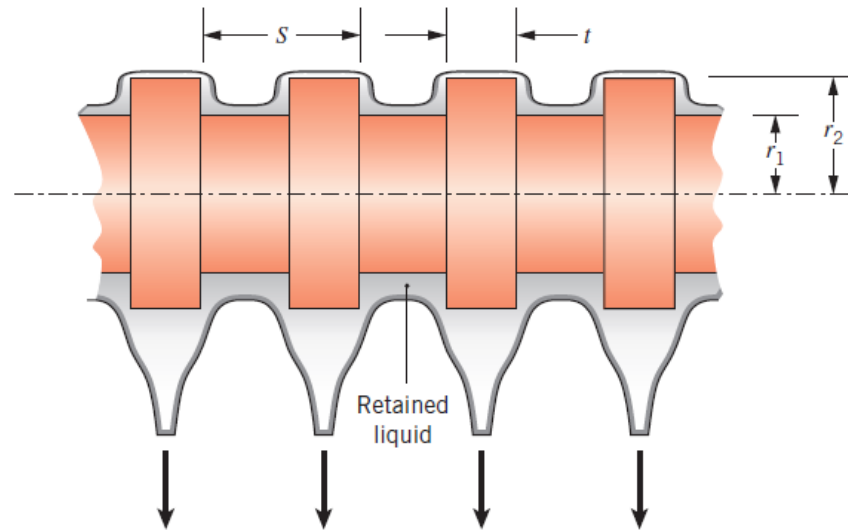


- A single smooth tube or smooth sphere:

$$\bar{Nu}_D = \frac{\bar{h}_D D}{k_l} = C \left[\frac{\rho_l g (\rho_l - \rho_v) h'_{fg} D^3}{\mu_l k_l (T_{sat} - T_s)} \right]^{1/4} \quad (10.46)$$

Tube: $C = 0.729$ Sphere: $C = 0.826$

- A single finned or ribbed tube :



- Small fins or ribs lead to curvature of the liquid surface, inducing circulation in the liquid driven by surface tension, enhancing heat transfer rates.

The **minimum enhancement** associated with the finned tube (ft) relative to the unfinned tube (uft) is given by Eq. 48-49