Chapter 10: Boiling and Condensation[1]

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

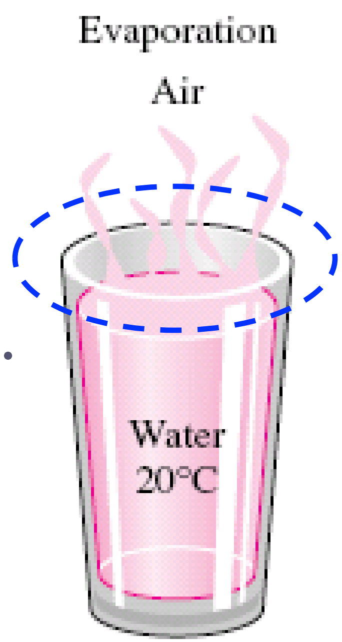
Objectives

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

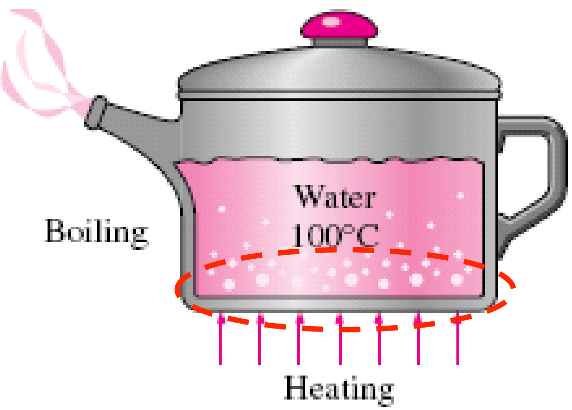
- Differentiate between evaporation and boiling, and gain familiarity with different types of boiling,
- Develop a good understanding of the boiling curve, and the different boiling regimes corresponding to different regions of the boiling curve,
- Calculate the heat flux and its critical value associated 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.



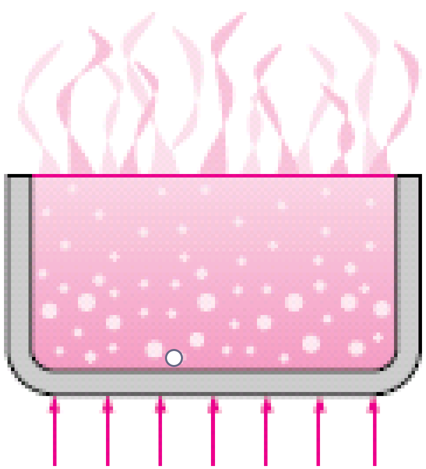
- **Boiling** occurs at the *solid–liquid interface* when a liquid is brought into contact with a surface maintained at a temperature sufficiently above the saturation temperature of the liquid



Classification of boiling

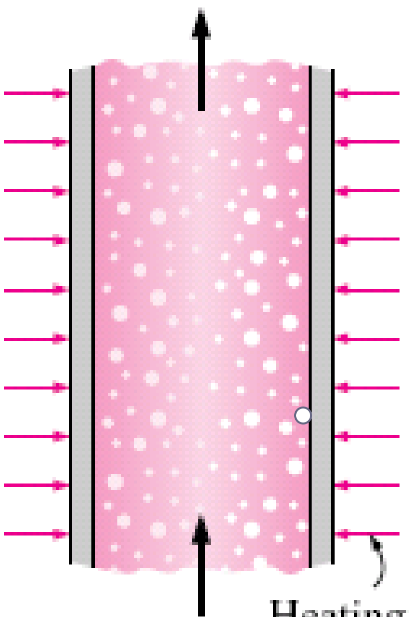
Pool Boiling

- 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.



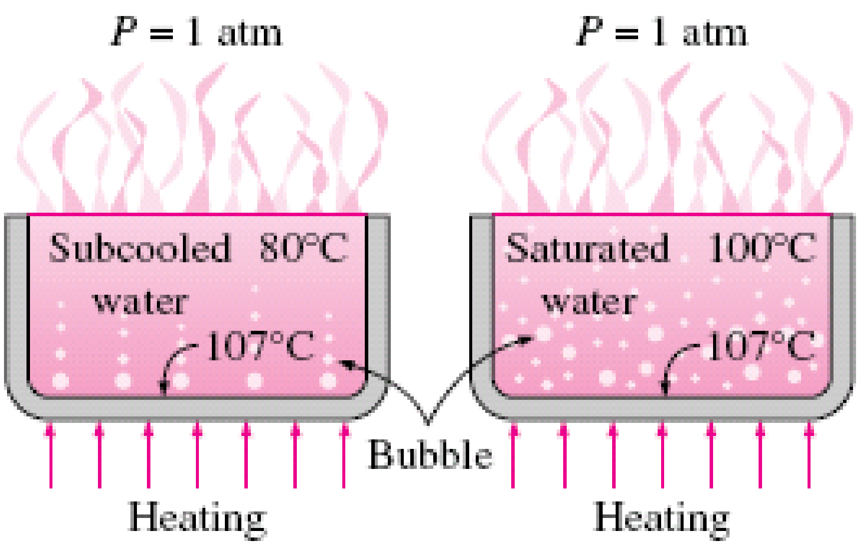
Classification of boiling

Subcooled Boiling

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

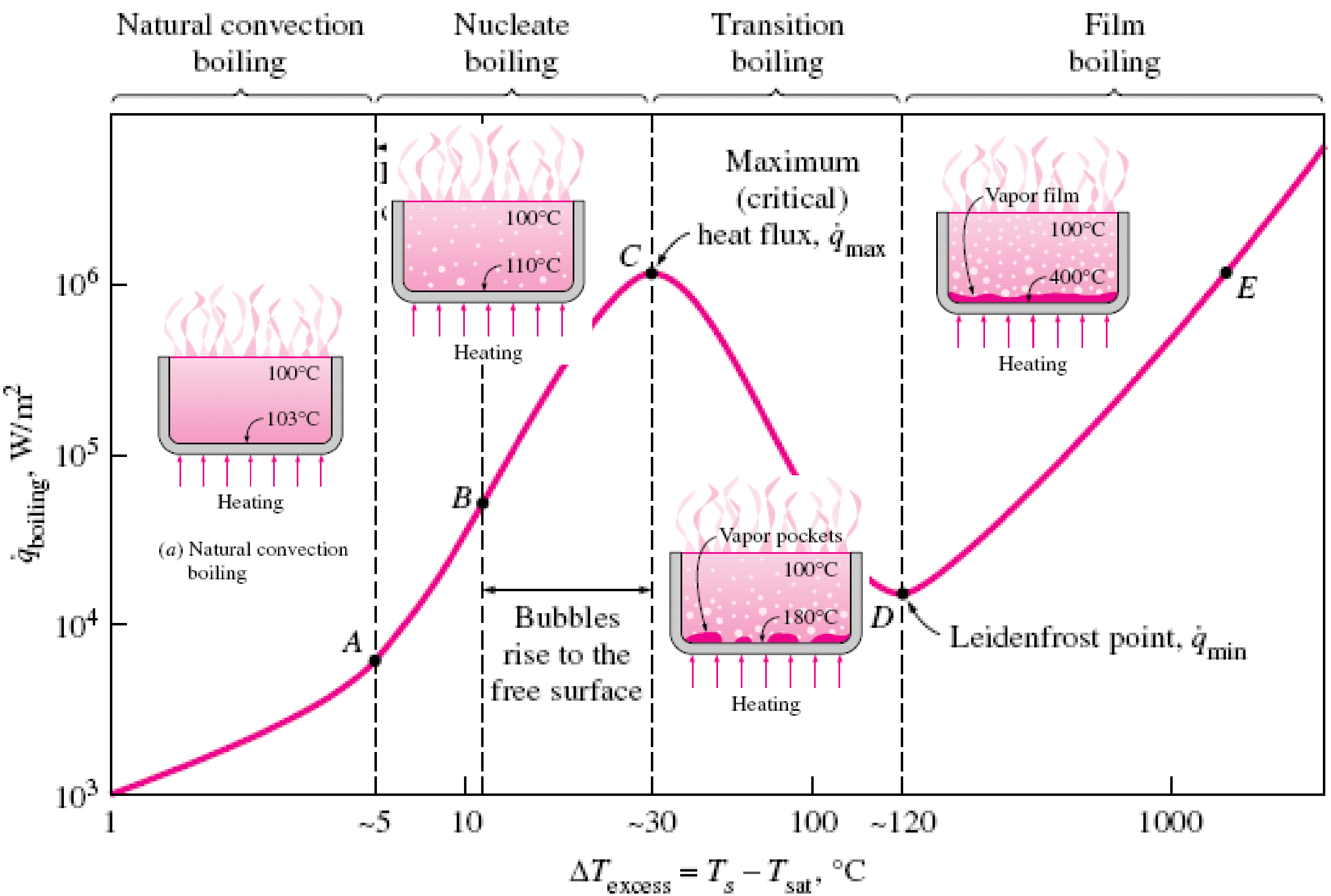
Saturated Boiling

- 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)

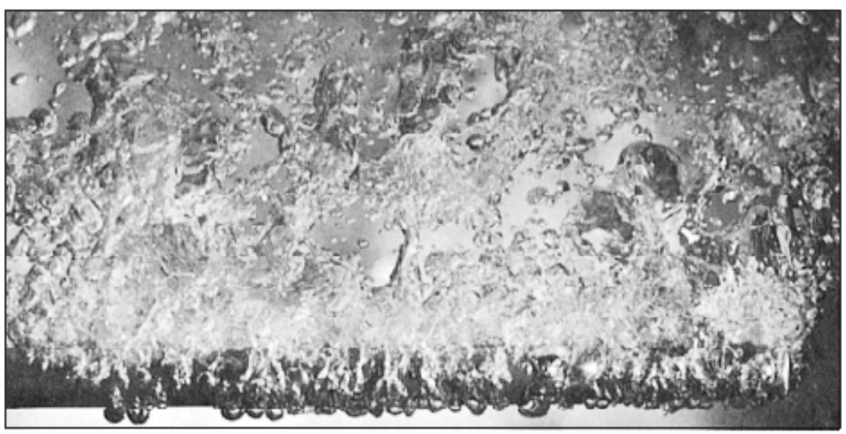
- Bubbles do not form on the heating surface until the liquid is heated a few degrees above the saturation temperature (about 2 to 6°C for water)

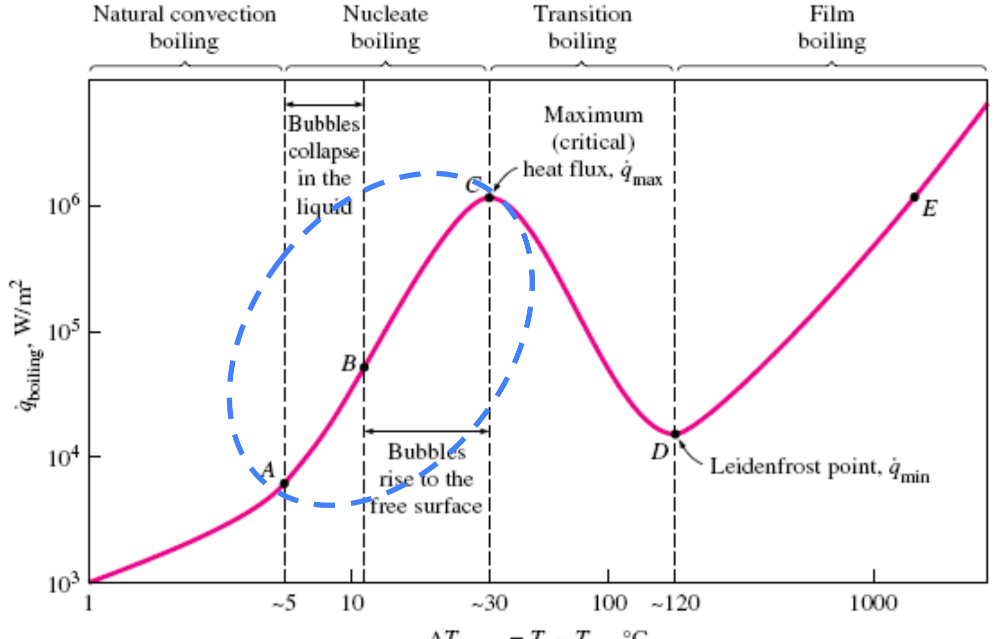
  → the liquid is slightly *superheated* in this case (*metastable* state).

- 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



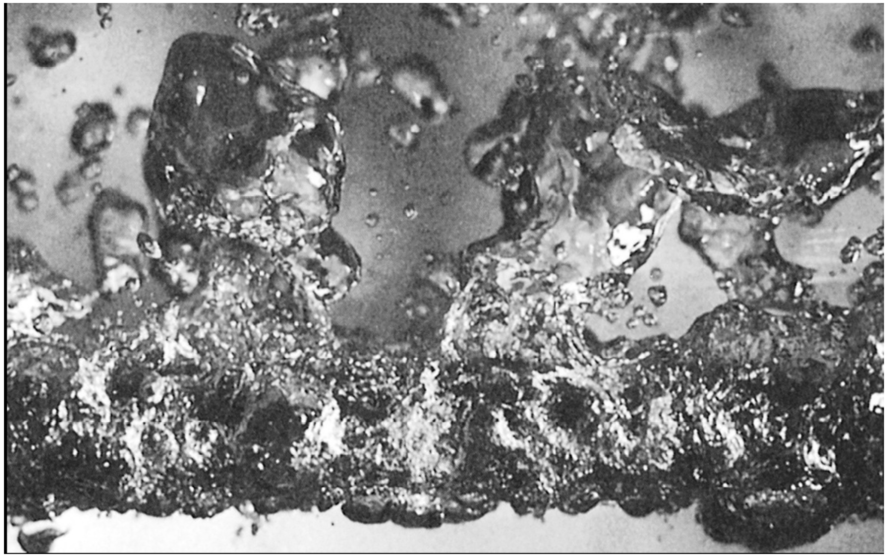


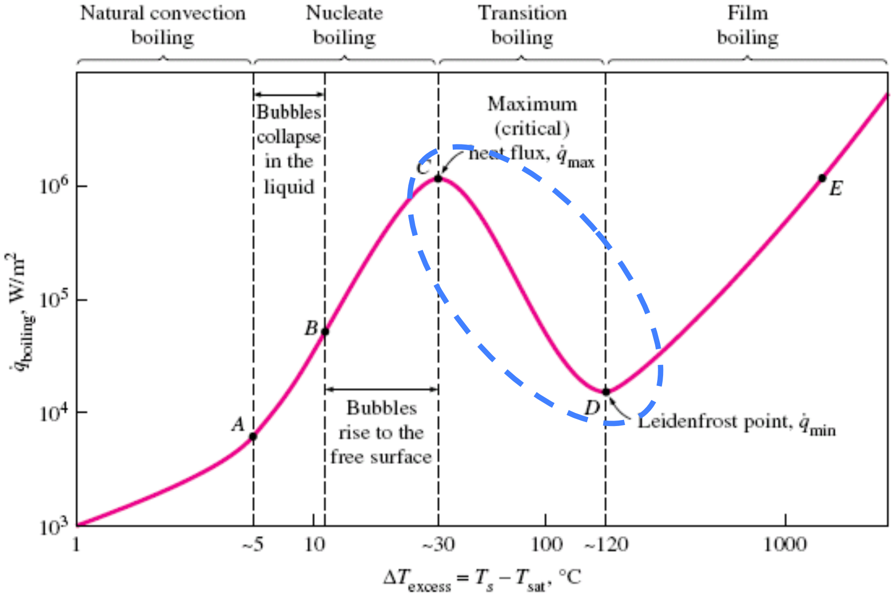
Nucleate Boiling

- 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 ΔT_{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

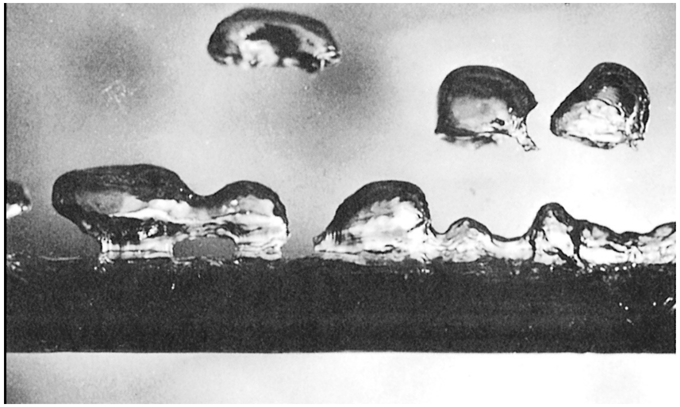
- When DT_{excess} is increased past point *C*, the heat flux decreases.
- This is because a large fraction of the heater surface is covered by a vapor film, which acts as an insulation.
- In the transition boiling regime, both nucleate and film boiling partially occur.

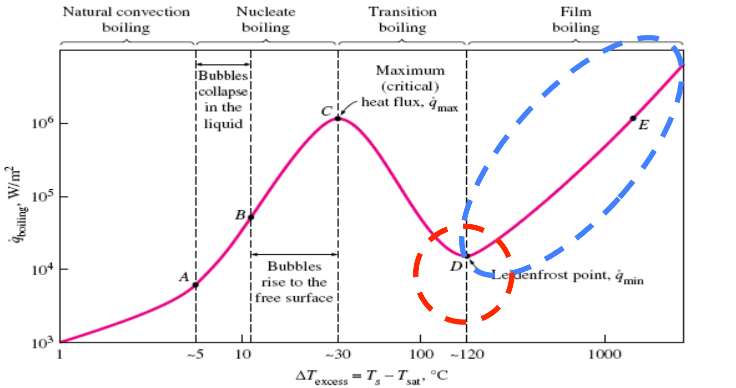




Film Boiling

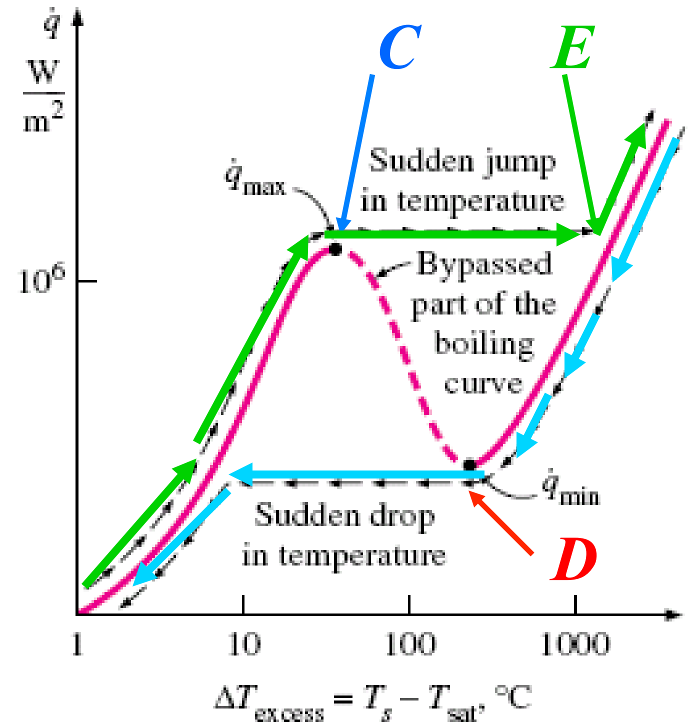
- Beyond Point *D* the heater surface is completely covered by a continuous stable vapor film.
- Point *D*, where the heat flux reaches a minimum is called the **Leidenfrost point**.
- The presence of a vapor film between the heater surface and the liquid is responsible for the low heat transfer rates 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:

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

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:

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

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

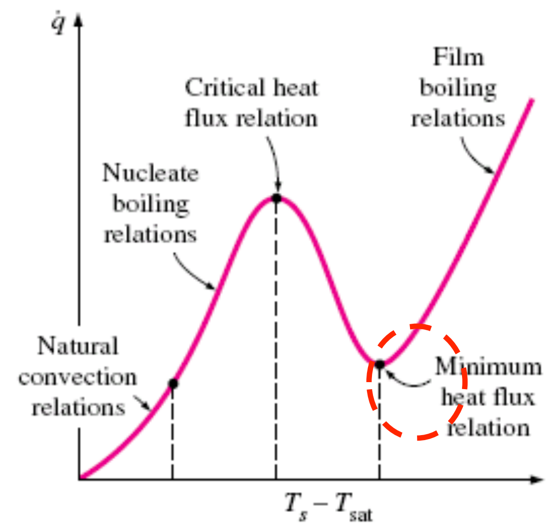
- The CHF is independent of the fluid–heating surface combination, as well as the viscosity, thermal conductivity, and the specific heat of the liquid.
- The CHF increases with pressure up to about one-third of the critical pressure, and then starts to decrease and becomes zero at the critical pressure.
- 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

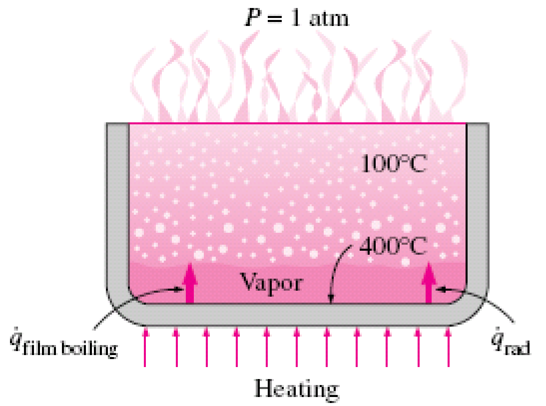
- Minimum heat flux, which occurs at the Leidenfrost point, is of practical interest since it represents the lower limit for the heat flux in the film boiling regime.
- Zuber derived the following expression for the minimum heat flux for a *large horizontal plate*

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

- 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

$$\overline{Nu}_D = \frac{\bar{h}_{conv} D}{k_v} = C\left[\frac{g(\rho_l - \rho_v)h'_{fg}D^3}{\nu_v k_v (T_s - T_{sat})}\right]^{1/4}, \quad \begin{array}{c} C = 0.62\,(cylinder) \\ C = 0.67\,(sphere) \end{array}$$

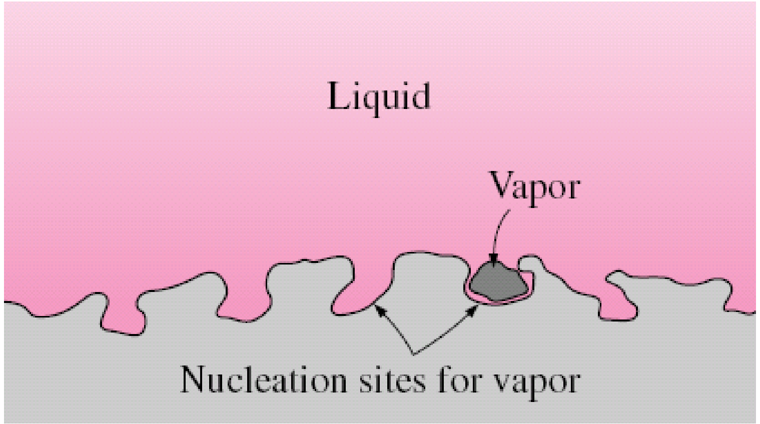
- At high surface temperatures (typically above 300°C), heat transfer across the vapor film by *radiation* becomes significant and needs to be considered.

- The two mechanisms of heat transfer (radiation and convection) adversely affect each other, causing the total heat transfer to be less than their sum.

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

Enhancement of Heat Transfer in Pool Boiling

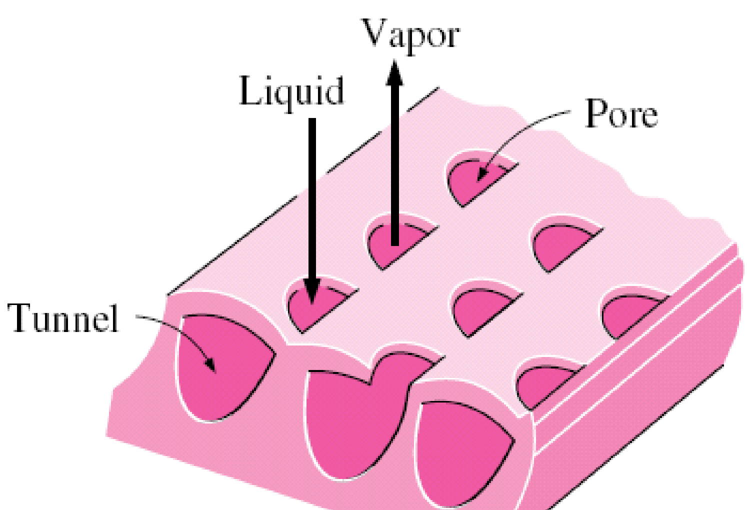
- The rate of heat transfer in the nucleate boiling regime strongly depends on the number of active nucleation sites on the surface, and the rate of bubble formation at each site.
- Therefore, modification that enhances *nucleation* on the heating surface will also enhance *heat transfer* in nucleate 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.



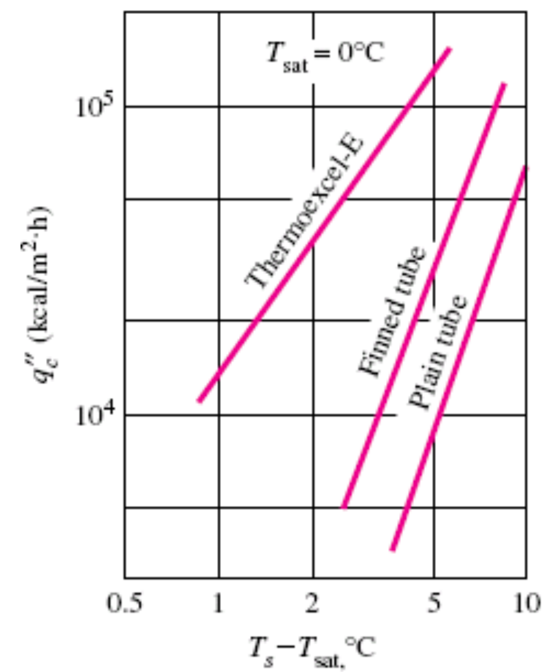
Enhancement of Heat Transfer in Pool Boiling

- 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.

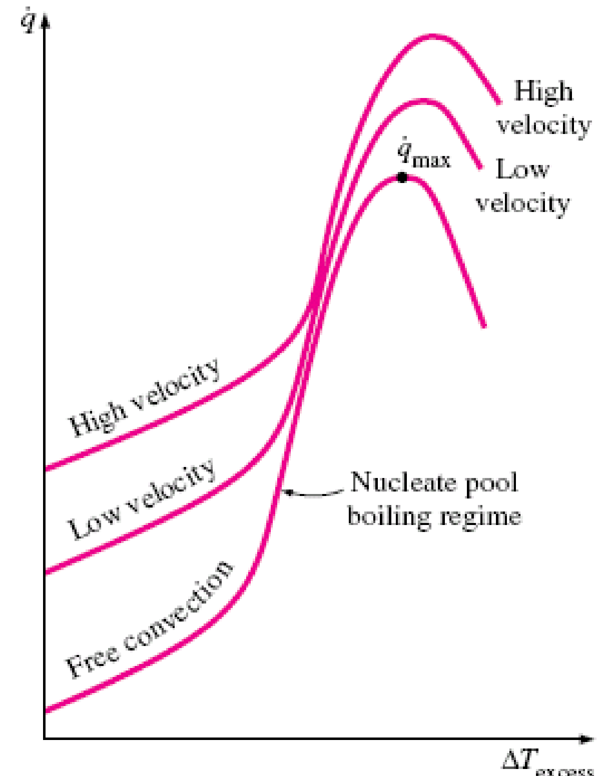


Thermoexcel-E



External Forced Convection Boiling (Flow Boiling)

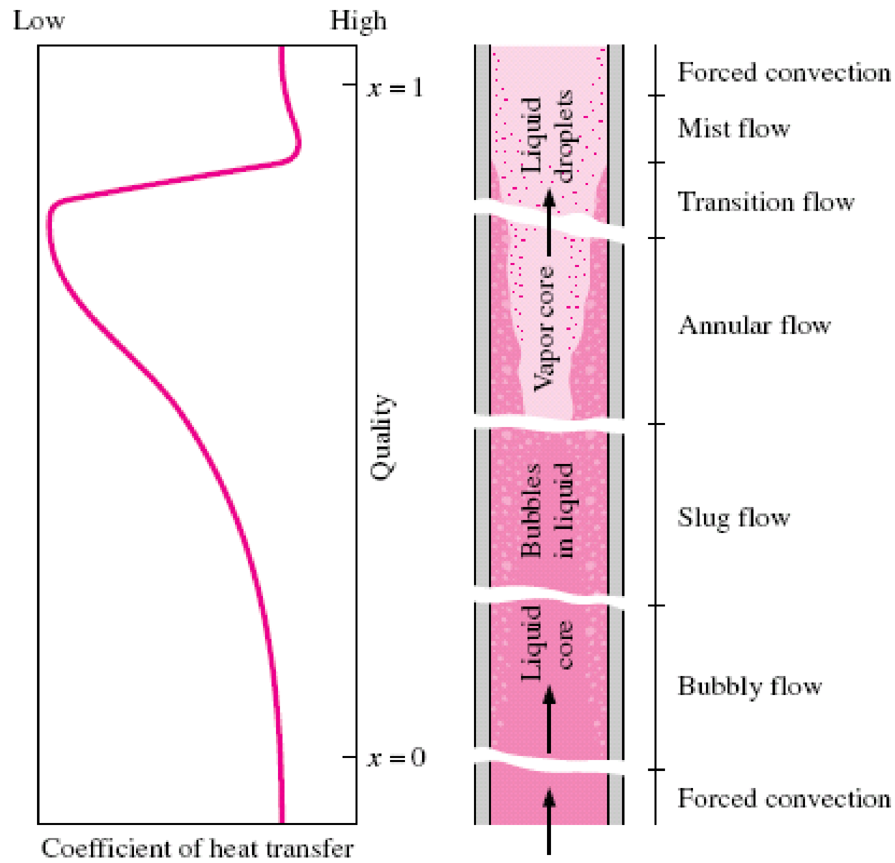
- In **flow boiling**, the fluid is forced to move by an external source such as a pump as it undergoes a phase-change process.
- The boiling in this case exhibits the combined effects of convection and pool boiling.
- Flow boiling is classified as either *external* and *internal flow boiling.*
- *External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

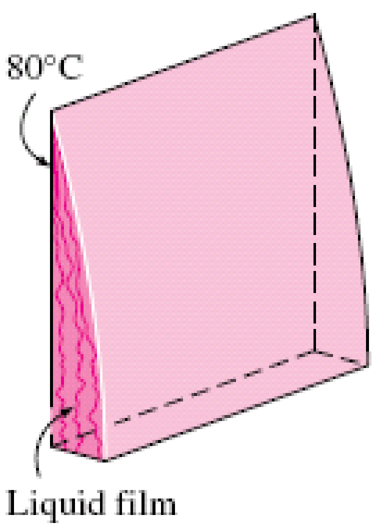
Condensation

- 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

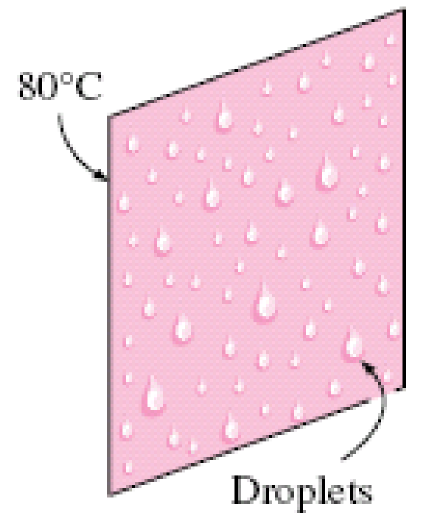
Film condensation

- 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.



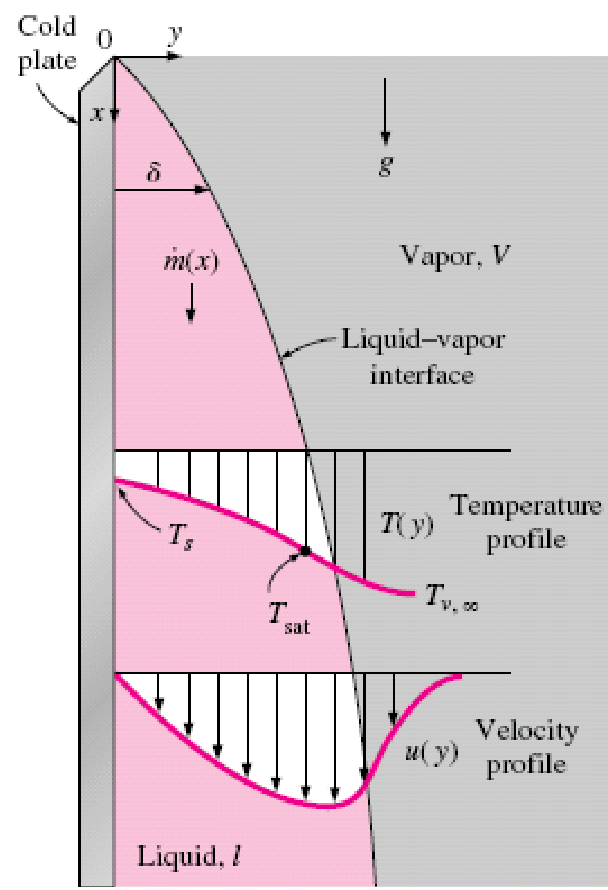
Dropwise condensation

- 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.
- d *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.

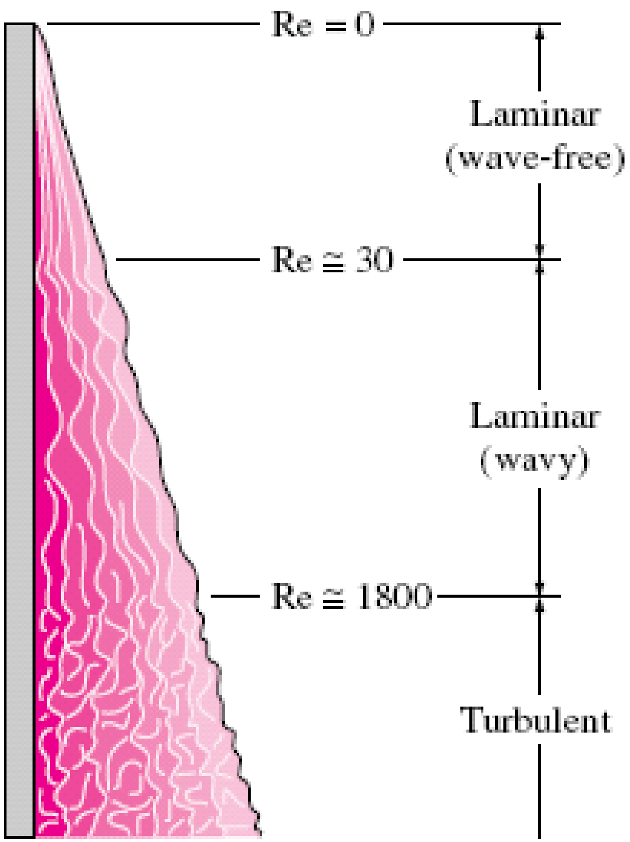


Vertical Plate — Flow Regimes

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

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

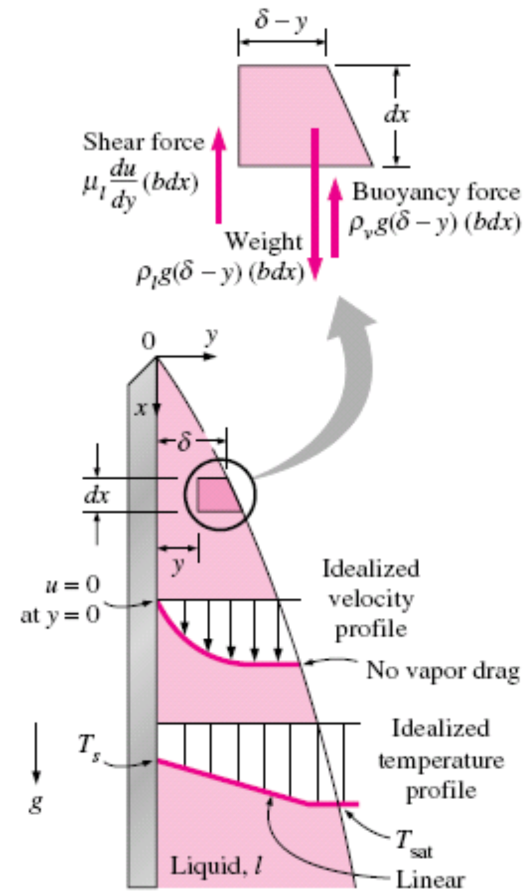
- Three prime flow regimes:
  - Re<30 — Laminar (wave-free),
  - 30<Re<1800 — Wavy-laminar,
  - Re>1800 — Turbulent.
- The Reynolds number increases in the flow direction.



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

- Netwon'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 ($u=u(y)$)

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

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

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

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

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

whose derivative with respect to x is

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

Thermal Considerations

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

$$dq = h_{fg}\,d\dot{m} = k_l\left(b\,dx\right)\frac{T_{sat} - T_s}{\delta}$$

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

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

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

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

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

- 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

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

- 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

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

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

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

- The average heat transfer coefficient over the entire plate is

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

- The Nusselt number:

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

- 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

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

- Flow regimes are:
  - Laminar wave-free flow: $\mathrm{Re}_\delta \leq 30$
  - Laminar wavy flow: $30 < \mathrm{Re}_\delta \leq 1800$
  - Turbulent flow: $\mathrm{Re}_\delta > 1800$

Turbulent Flow on Vertical Plates

- Laminar waver-free regime:

$$\bar{N}u_L = \frac{\bar{h}_L(v_l^2/g)^{1/3}}{k_l} = 1.46\,\mathrm{Re}_\delta^{-1/3}, \qquad \mathrm{Re}_\delta \le 30 \qquad \textbf{(10-38)}$$

- Laminar wavy regime:

$$\bar{N}u_L = \frac{\bar{h}_L(v_l^2/g)^{1/3}}{k_l} = \frac{\mathrm{Re}_\delta}{1.08\,\mathrm{Re}_\delta^{1.22} - 5.2}, \qquad 30 < \mathrm{Re}_\delta \le 1800 \qquad \textbf{(10-39)}$$

- Turbulent regime:

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

$\mathrm{Re}_\delta > 1800$ and $\mathrm{Pr}_l \ge 1$

Nondimensionalized Heat Transfer Coefficients

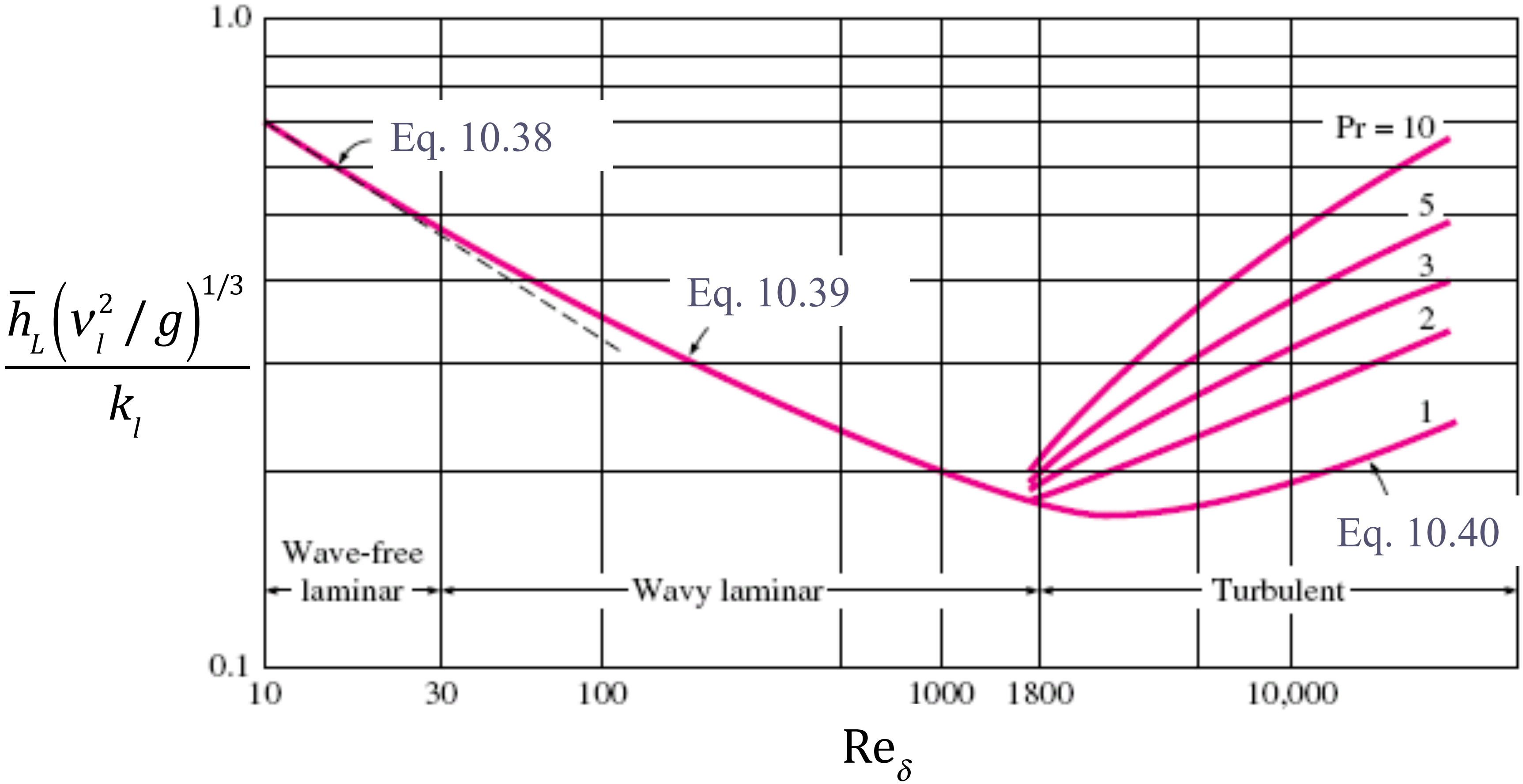
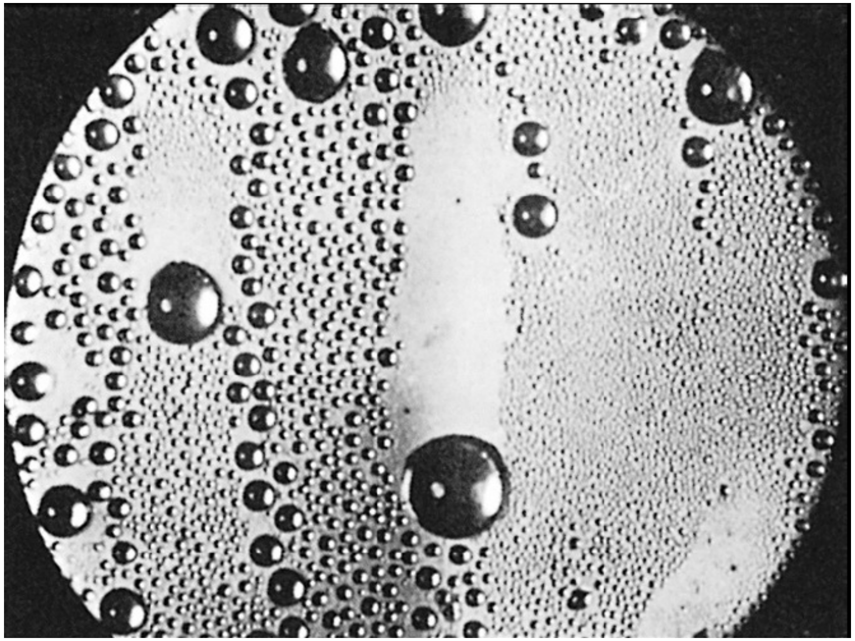


Figure 10.13: Modified Nusselt number for condensation on a vertical plate.

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:*

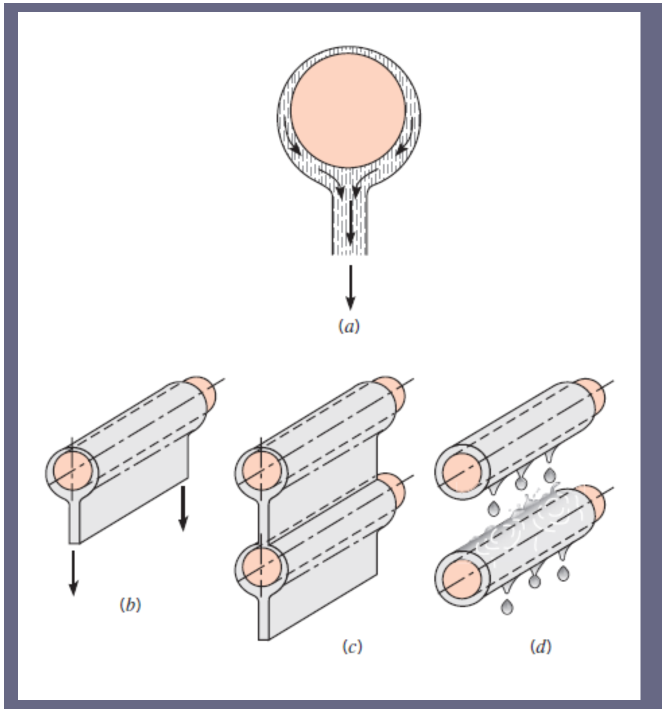
$$hdropwise = \begin{cases} 51{,}104 + 2044 T_{sat} & 22\,^{0}C < T_{sat} < 100\ ^{0}C \\ 255{,}310 & T_{sat} > 100\,^{0}C \end{cases}$$

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

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

Film Condensation on Radial Systems

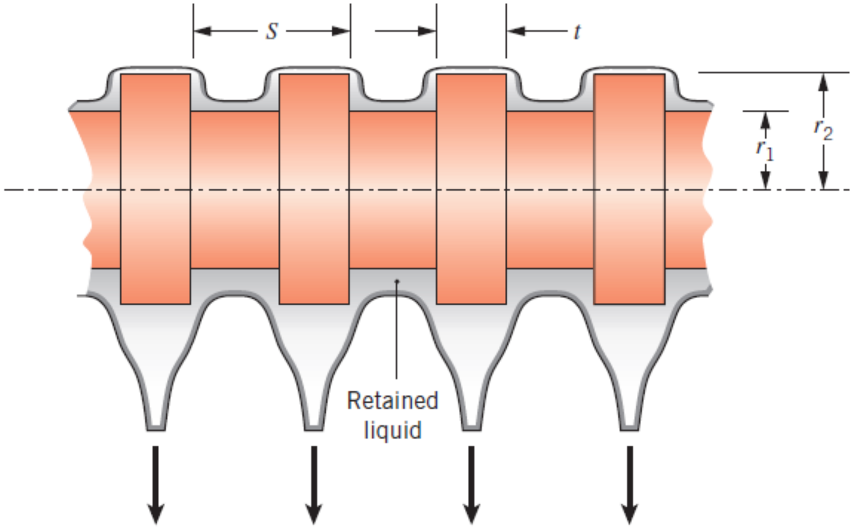


- A single smooth tube or smooth sphere:

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

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 tubed (uft) is given by Eq. 48-49