

UOIT
Faculty of Engineering and Applied Science
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ENGR3930U HEAT TRANSFER

Boiling and Condensation

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Objectives

- 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.

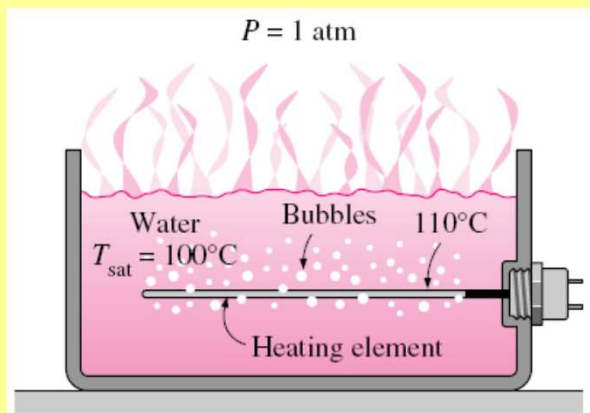


FIGURE 10-2

Boiling occurs when a liquid is brought into contact with a surface at a temperature above the saturation temperature of the liquid.



FIGURE 10-1

A liquid-to-vapor phase change process is called *evaporation* if it originates at a liquid–vapor interface and *boiling* if it occurs at a solid–liquid interface.

Classification of 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.
- 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.

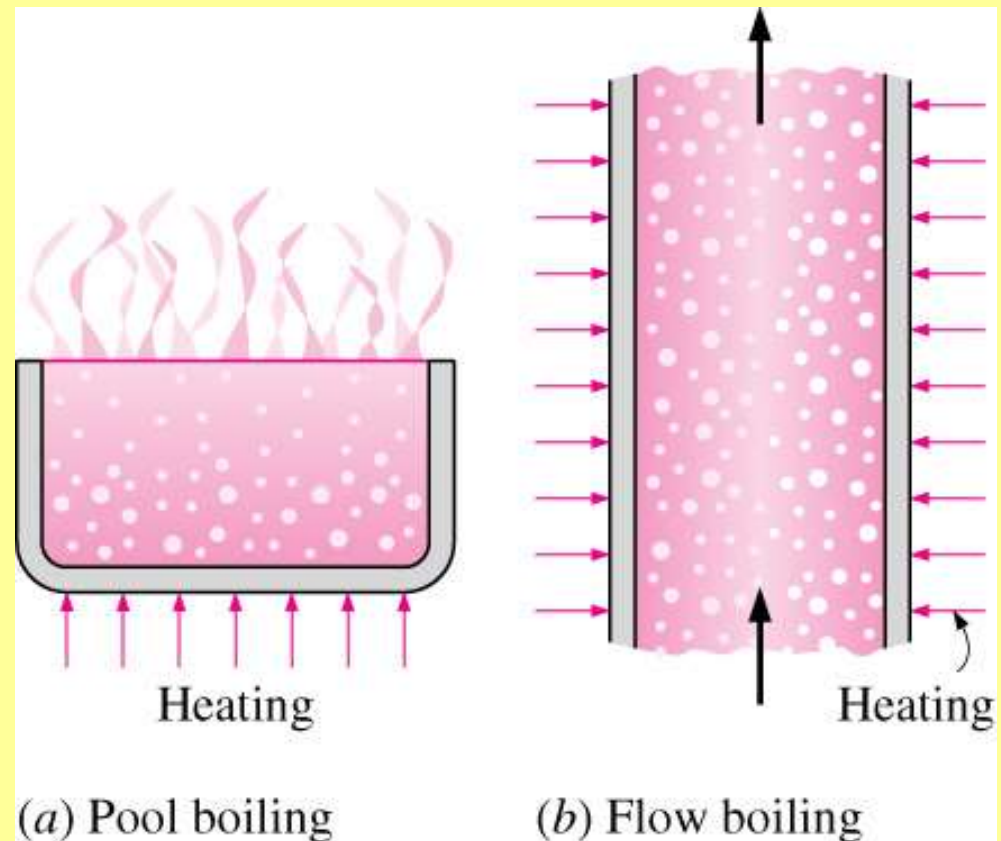


FIGURE 10-3

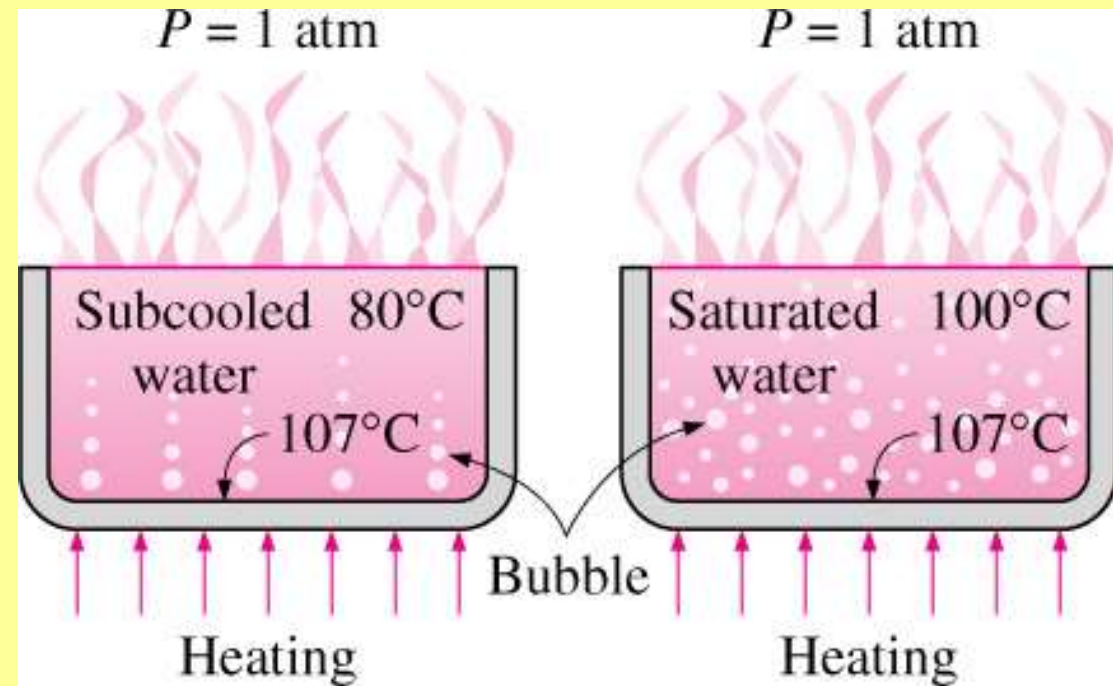
Classification of boiling on the basis of the presence of bulk fluid motion.

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.



(a) Subcooled boiling

(b) Saturated boiling

FIGURE 10-4

Classification of boiling
on the basis of the presence of
bulk liquid temperature.

Pool Boiling

$$\dot{q}_{\text{boiling}} = h(T_s - T_{\text{sat}}) = h\Delta T_{\text{excess}}$$

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

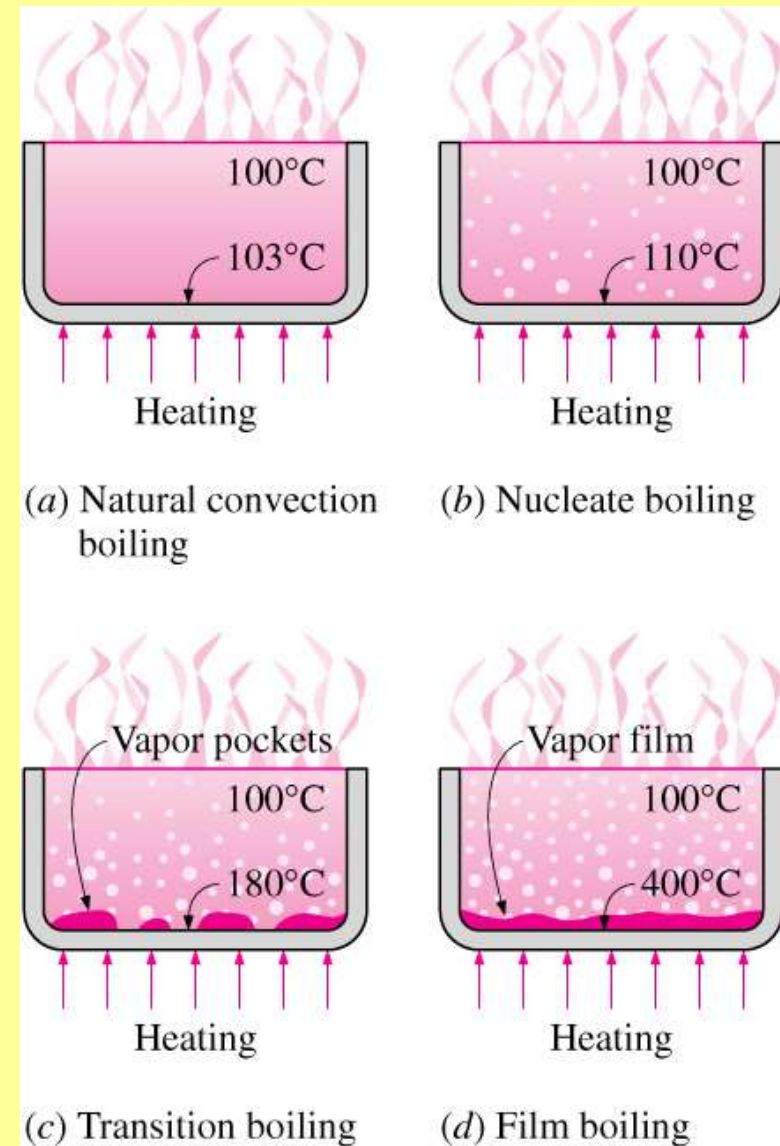
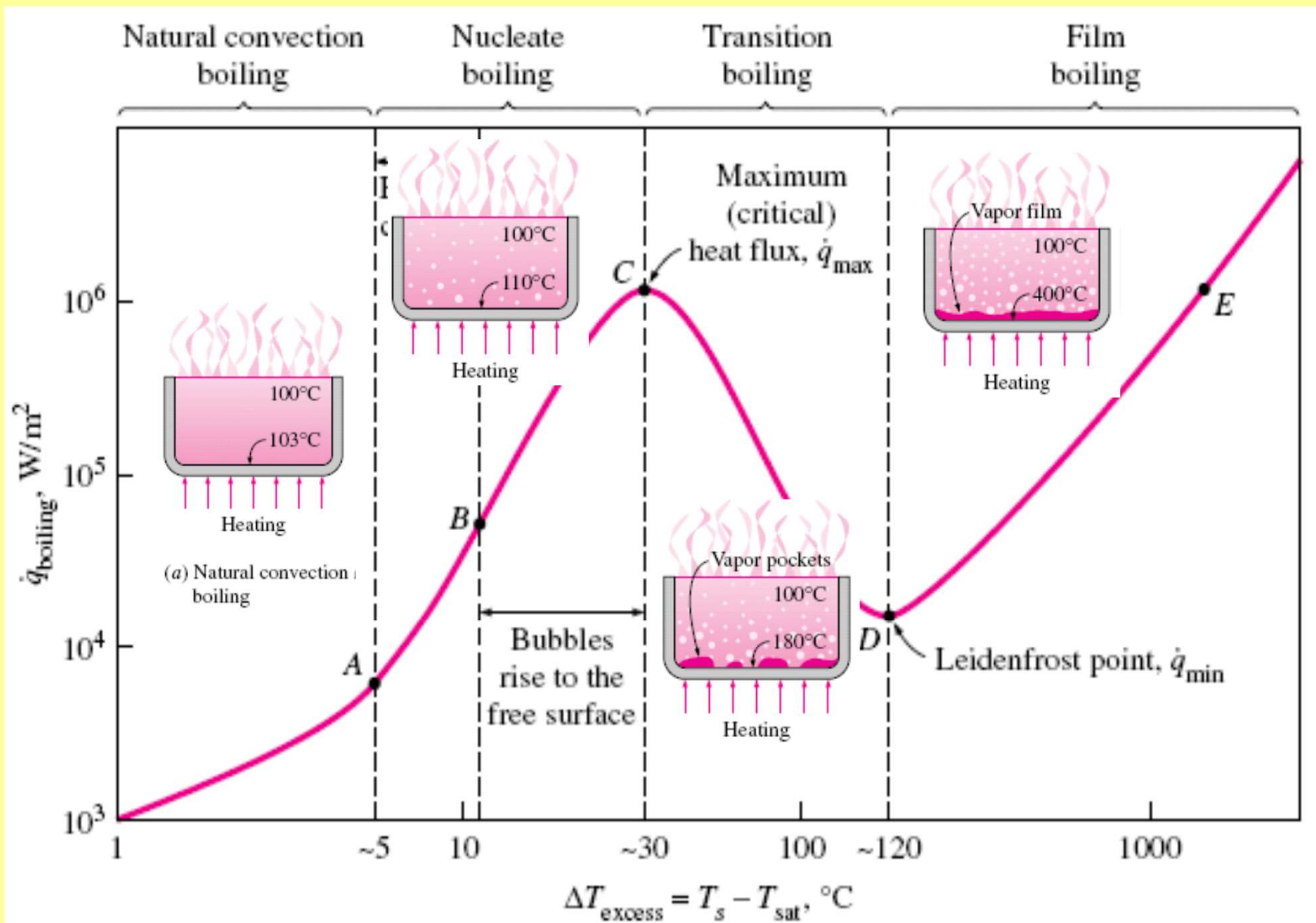


FIGURE 10-5

Different boiling regimes
in pool boiling.

Pool Boiling



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)

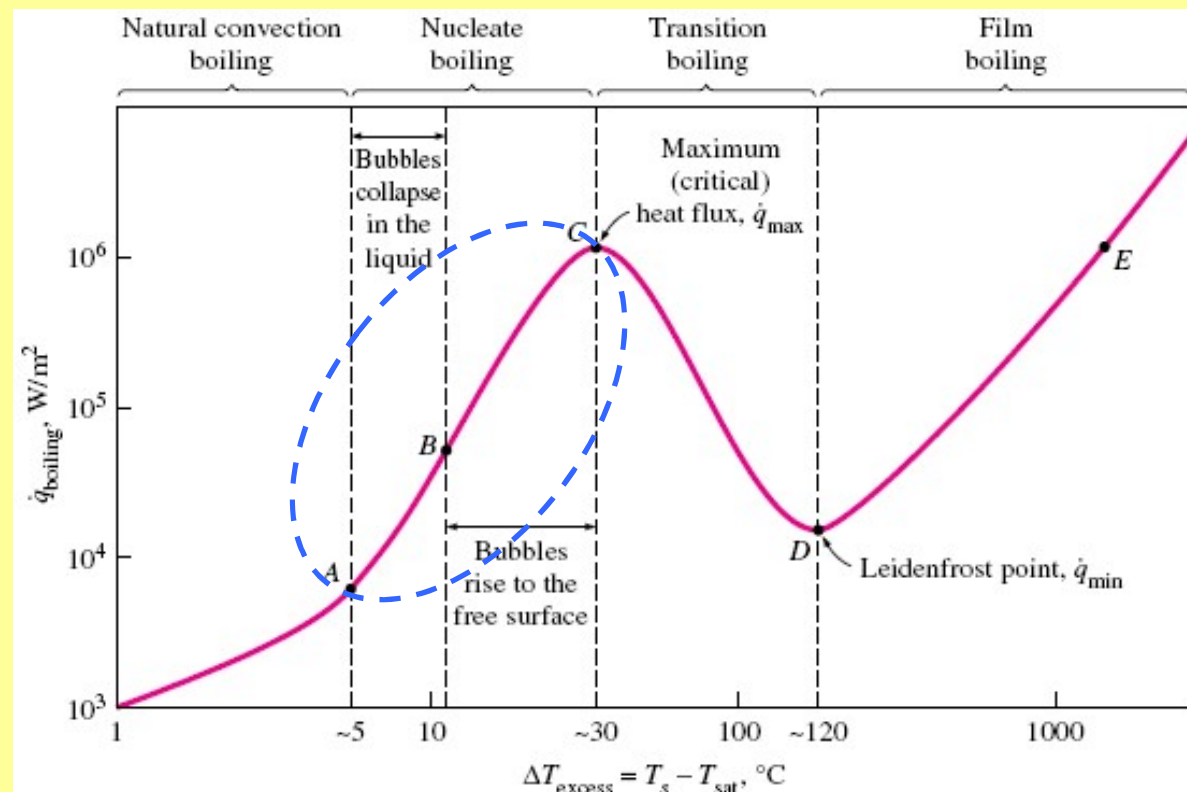
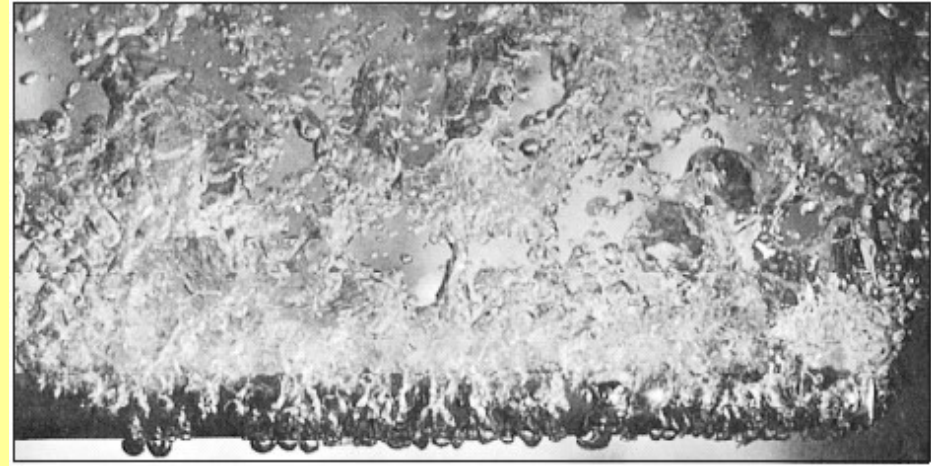


the liquid is slightly superheated in this case (metastable state).

- 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*.

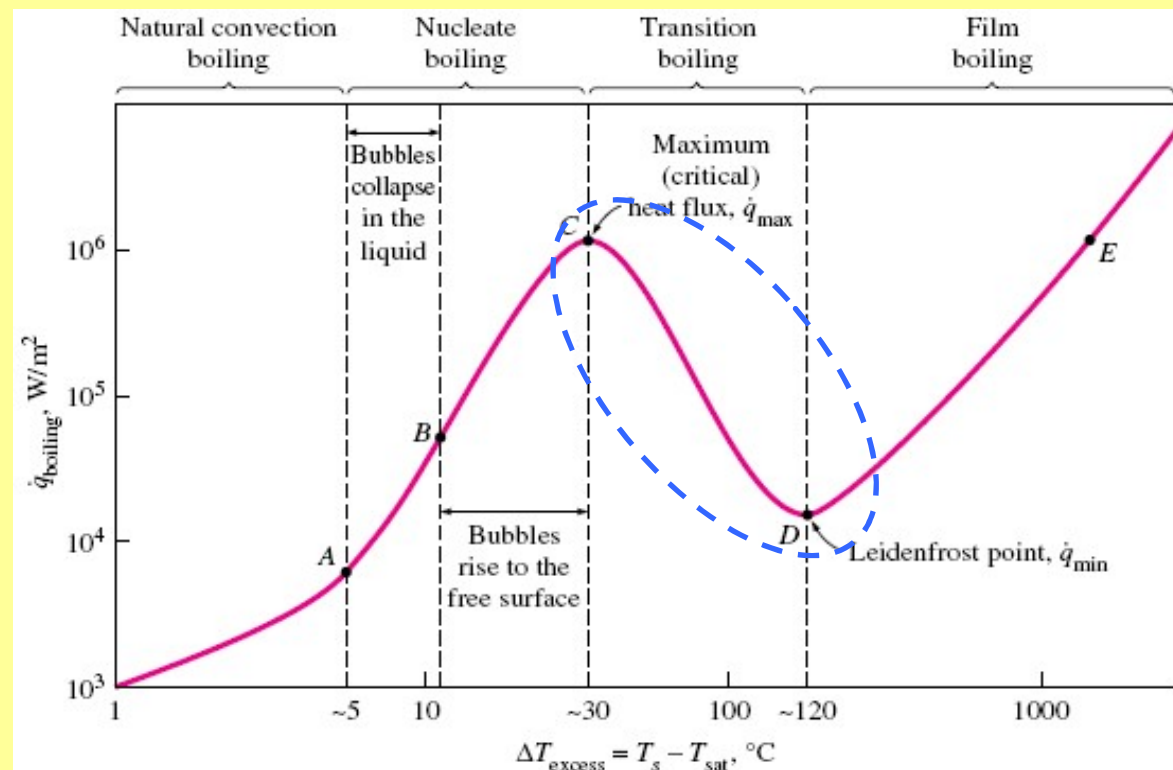
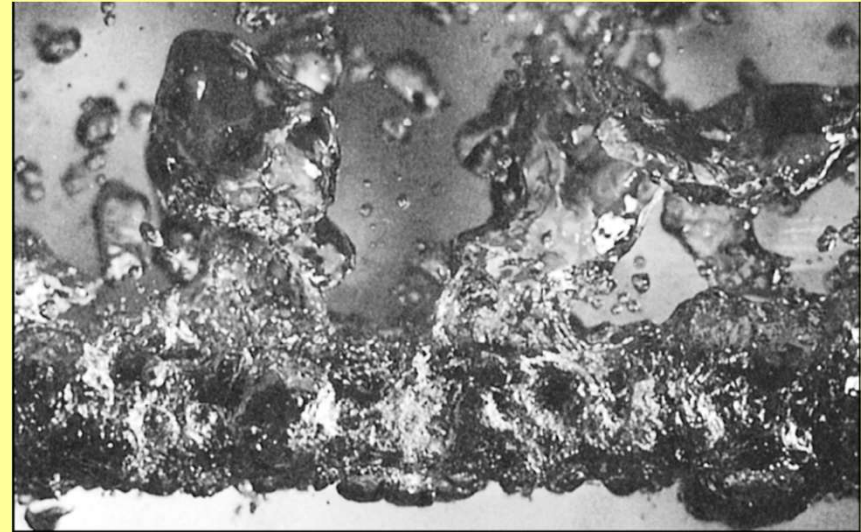


Nucleate Boiling

- 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.

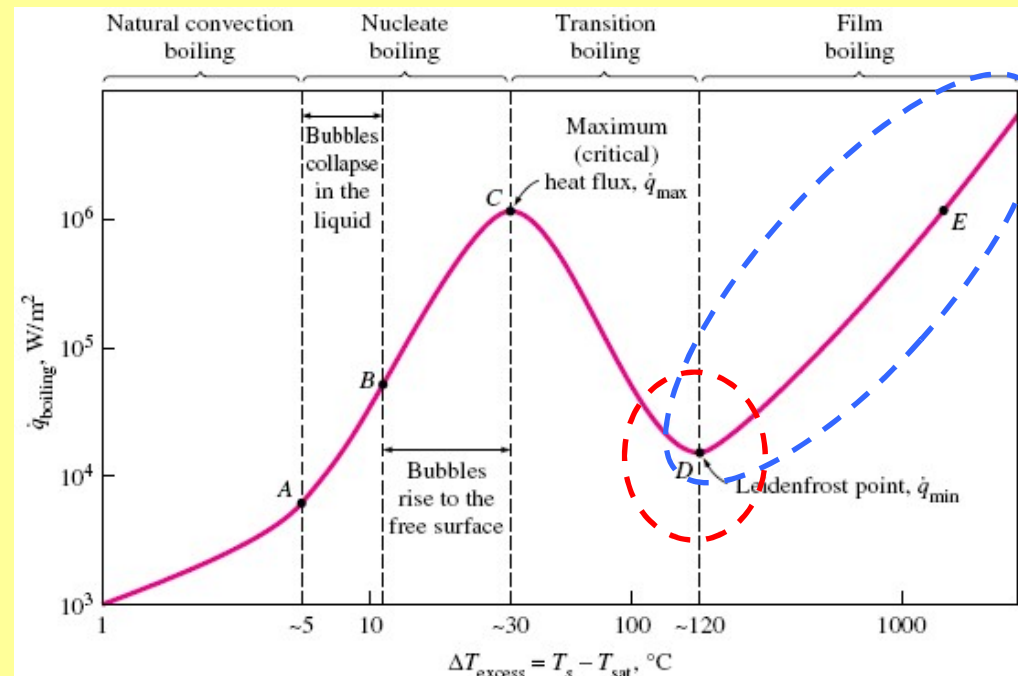
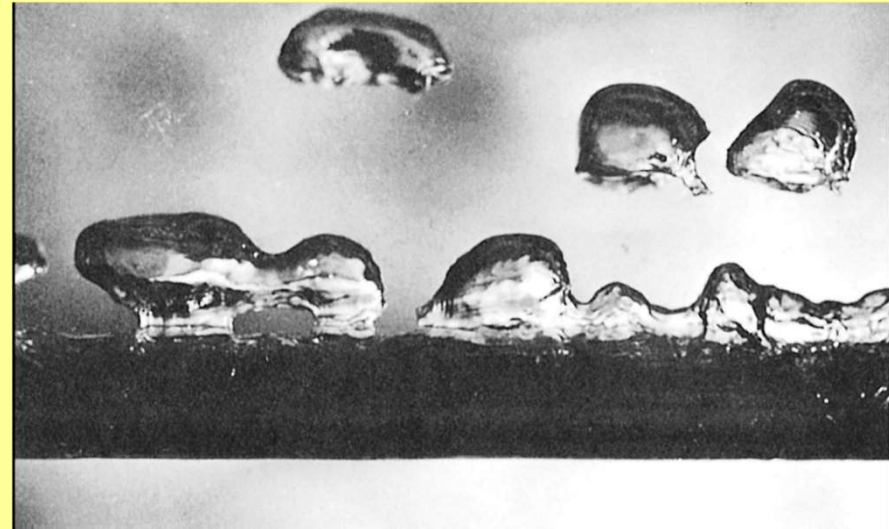
Transition Boiling

- When ΔT_{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.



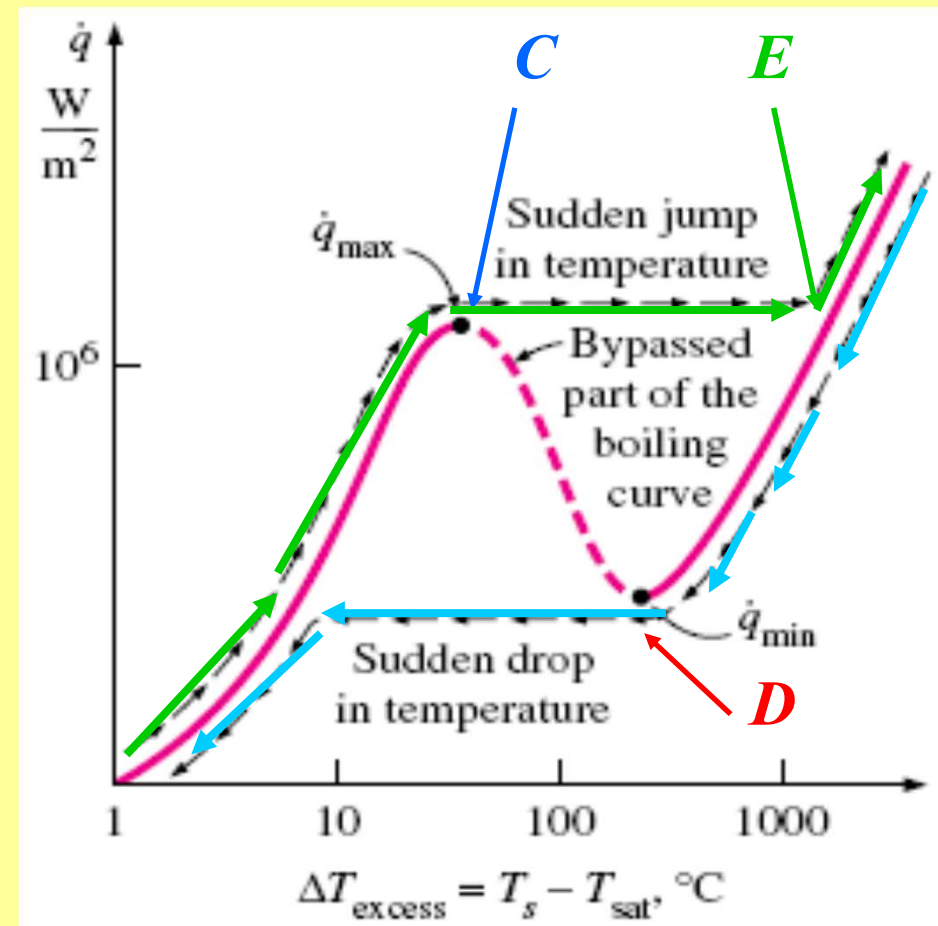
Film Boiling

- 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.
- 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.

- For nucleate boiling a widely used correlation proposed in 1952 by Rohsenow:

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma} \right]^{1/2} \left[\frac{c_{pl}(T_s - T_{\text{sat}})}{C_{sf} h_{fg} \text{Pr}_l^n} \right]^3$$

$\dot{q}_{\text{nucleate}}$ = nucleate boiling heat flux, W/m²

μ_l = viscosity of the liquid, kg/m · s

h_{fg} = enthalpy of vaporization, J/kg

g = gravitational acceleration, m/s²

ρ_l = density of the liquid, kg/m³

ρ_v = density of the vapor, kg/m³

σ = surface tension of liquid–vapor interface, N/m

c_{pl} = specific heat of the liquid, J/kg · °C

T_s = surface temperature of the heater, °C

T_{sat} = saturation temperature of the fluid, °C

C_{sf} = experimental constant that depends on surface–fluid combination

Pr_l = Prandtl number of the liquid

n = experimental constant that depends on the fluid

TABLE 10–1

Surface tension of liquid–vapor interface for water

$T, ^\circ\text{C}$	$\sigma, \text{N/m}^*$
0	0.0757
20	0.0727
40	0.0696
60	0.0662
80	0.0627
100	0.0589
120	0.0550
140	0.0509
160	0.0466
180	0.0422
200	0.0377
220	0.0331
240	0.0284
260	0.0237
280	0.0190
300	0.0144
320	0.0099
340	0.0056
360	0.0019
374	0.0

*Multiply by 0.06852 to convert to lbf/ft or by 2.2046 to convert to lbfm/s².

TABLE 10-3Values of the coefficient C_{sf} and n for various fluid–surface combinations

Fluid–Heating Surface Combination	C_{sf}	n
Water–copper (polished)	0.0130	1.0
Water–copper (scored)	0.0068	1.0
Water–stainless steel (mechanically polished)	0.0130	1.0
Water–stainless steel (ground and polished)	0.0060	1.0
Water–stainless steel (teflon pitted)	0.0058	1.0
Water–stainless steel (chemically etched)	0.0130	1.0
Water–brass	0.0060	1.0
Water–nickel	0.0060	1.0
Water–platinum	0.0130	1.0
<i>n</i> -Pentane–copper (polished)	0.0154	1.7
<i>n</i> -Pentane–chromium	0.0150	1.7
Benzene–chromium	0.1010	1.7
Ethyl alcohol–chromium	0.0027	1.7
Carbon tetrachloride–copper	0.0130	1.7
Isopropanol–copper	0.0025	1.7

TABLE 10-4Values of the coefficient C_{cr} for use in Eq. 10-3 for maximum heat flux (dimensionless parameter $L^* = L[g(\rho_l - \rho_v)/\sigma]^{1/2}$)

Heater Geometry	C_{cr}	Charac. Dimension of Heater, L	Range of L^*
Large horizontal flat heater	0.149	Width or diameter	$L^* > 27$
Small horizontal flat heater ¹	$18.9K_1$	Width or diameter	$9 < L^* < 20$
Large horizontal cylinder	0.12	Radius	$L^* > 1.2$
Small horizontal cylinder	$0.12L^{*-0.25}$	Radius	$0.15 < L^* < 1.2$
Large sphere	0.11	Radius	$L^* > 4.26$
Small sphere	$0.227L^{*-0.5}$	Radius	$0.15 < L^* < 4.26$

¹ $K_1 = \sigma/[g(\rho_l - \rho_v)A_{\text{heater}}]$

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:

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

C_{cr} 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*

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

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

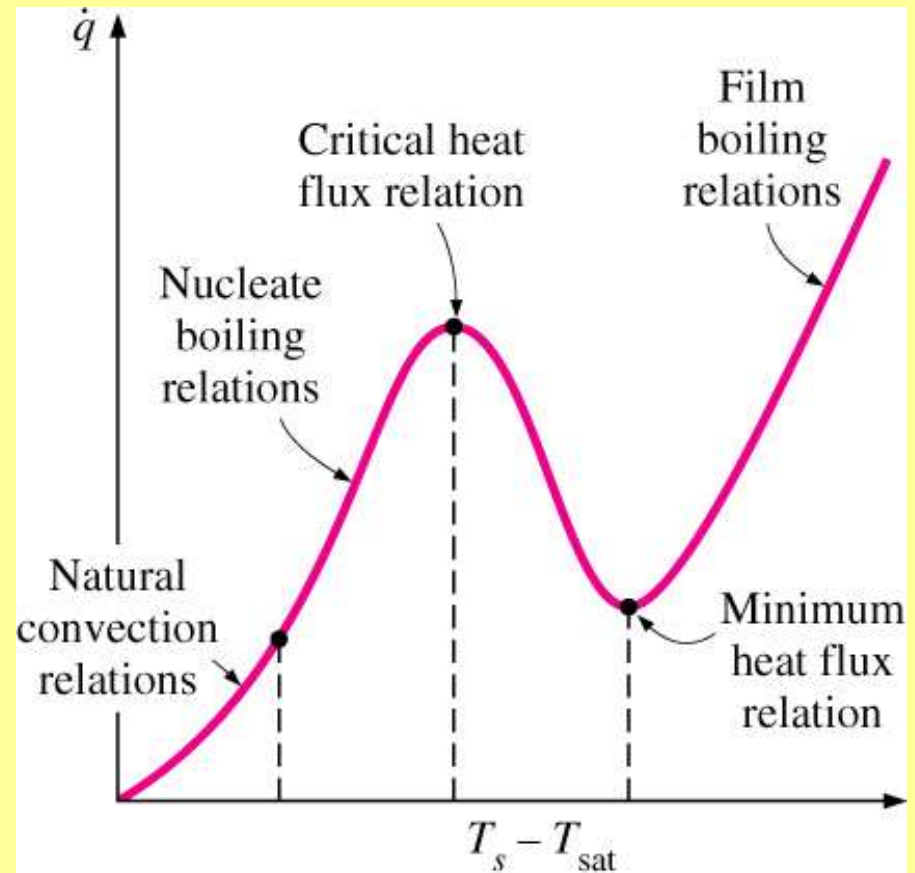


FIGURE 10–11

Different relations are used to determine the heat flux in different boiling regimes.

Film Boiling

The heat flux for film boiling on a *horizontal cylinder* or *sphere* of diameter D is given by

$$\dot{q}_{film} = C_{film} \left[\frac{g k_v^3 \rho_v (\rho_l - \rho_v) [h_{fg} + 0.4 C_{pv} (T_s - T_{sat})]}{\mu_v D (T_s - T_{sat})} \right]^{1/4} (T_s - T_{sat})$$

$$C_{film} = \begin{cases} 0.62 & \text{for horizontal cylinders} \\ 0.67 & \text{for spheres} \end{cases}$$

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

$$\dot{q}_{rad} = \varepsilon \sigma (T_s^4 - T_{sat}^4)$$

$$\dot{q}_{total} = \dot{q}_{film} + \frac{3}{4} \dot{q}_{rad}$$

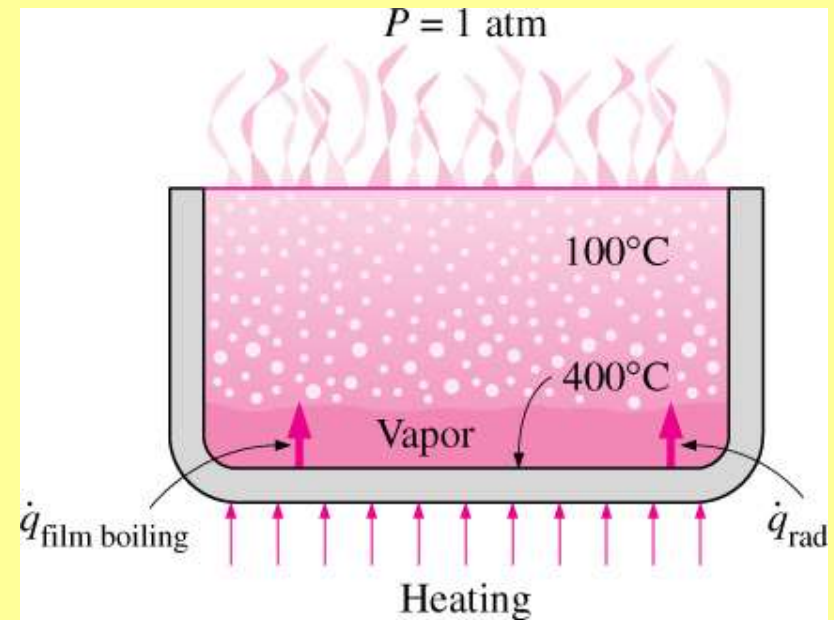


FIGURE 10-12

At high heater surface temperatures, radiation heat transfer becomes significant during film boiling.

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**.

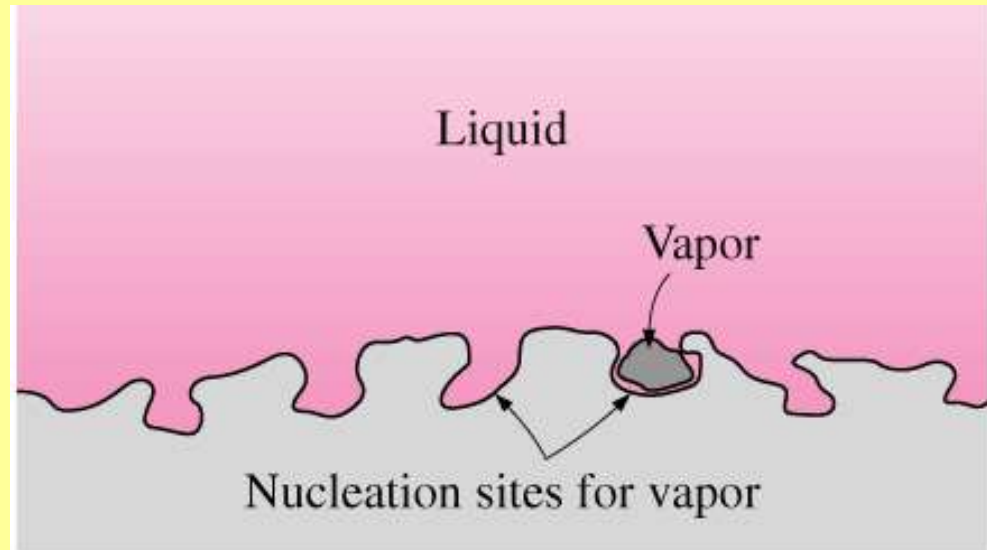


FIGURE 10-13

The cavities on a rough surface act as nucleation sites and enhance boiling heat transfer.

- 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.

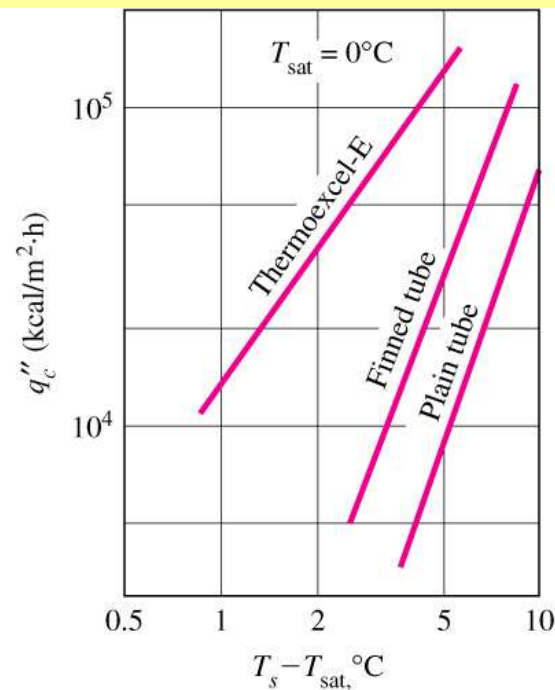
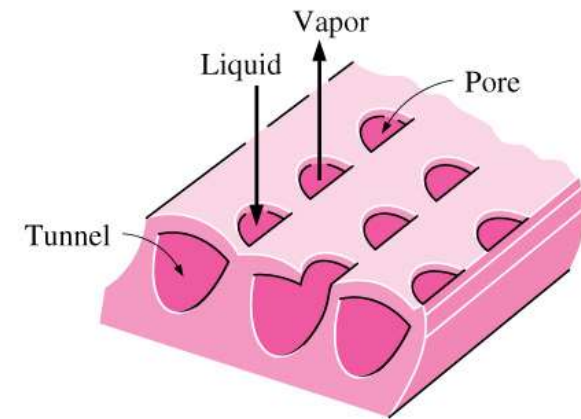


FIGURE 10-14

The enhancement of boiling heat transfer in Freon-12 by a mechanically roughened surface, thermoexcel-E.



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.

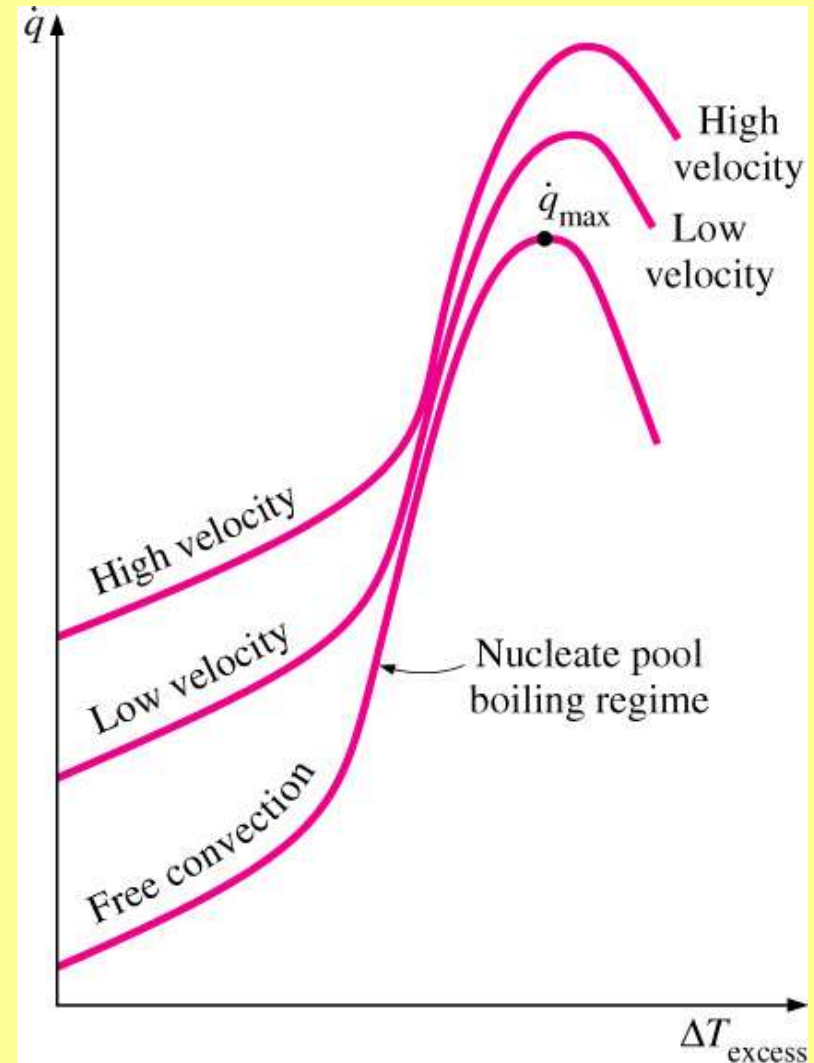


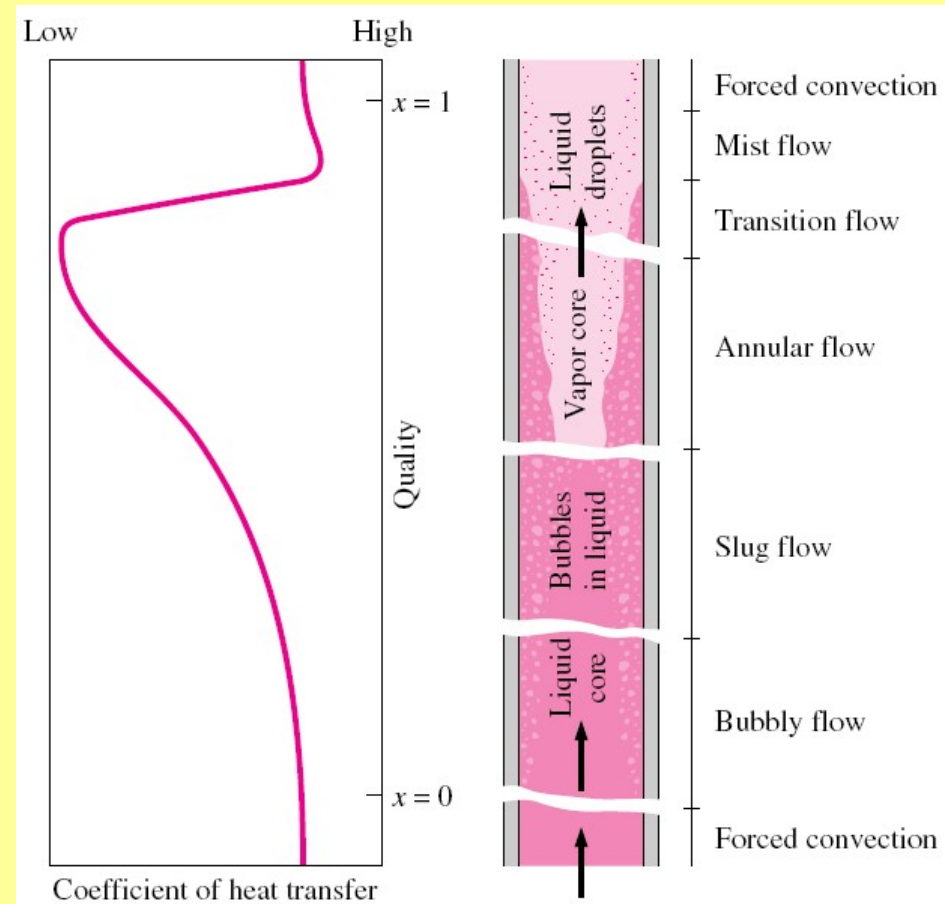
FIGURE 10-18

The effect of forced convection on external flow boiling for different flow velocities.

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.

10–13 Water is boiled at 90°C by a horizontal brass heating element of diameter 7 mm. Determine the surface temperature of the heater for the minimum heat flux case.

Properties The properties of water at the saturation temperature of 90°C are (Tables 10-1 and A-9)

$$\begin{aligned}\rho_l &= 965.3 \text{ kg/m}^3 & h_{fg} &= 2283 \times 10^3 \text{ J/kg} \\ \rho_v &= 0.4235 \text{ kg/m}^3 & \mu_l &= 0.315 \times 10^{-3} \text{ kg/m} \cdot \text{s} \\ \sigma &= 0.0608 \text{ N/m} & c_{pl} &= 4206 \text{ J/kg} \cdot ^\circ\text{C} \\ \text{Pr}_l &= 1.96\end{aligned}$$

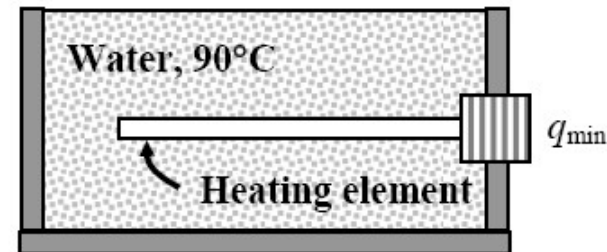
Also, $C_{sf} = 0.0060$ and $n = 1.0$ for the boiling of water on a brass heating (Table 10-3).

Analysis The minimum heat flux is determined from

$$\begin{aligned}\dot{q}_{\min} &= 0.09 \rho_v h_{fg} \left[\frac{\sigma g (\rho_l - \rho_v)}{(\rho_l + \rho_v)^2} \right]^{1/4} \\ &= 0.09 (0.4235) (2283 \times 10^3) \left[\frac{(0.0608)(9.81)(965.3 - 0.4235)}{(965.3 + 0.4235)^2} \right]^{1/4} = 13,715 \text{ W/m}^2\end{aligned}$$

The surface temperature can be determined from Rohsenow equation to be

$$\begin{aligned}\dot{q}_{\text{nucleate}} &= \mu_l h_{fg} \left[\frac{g (\rho_l - \rho_v)}{\sigma} \right]^{1/2} \left(\frac{c_{p,l} (T_s - T_{\text{sat}})}{C_{sf} h_{fg} \text{Pr}_l^n} \right)^3 \\ 13,715 \text{ W/m}^2 &= (0.315 \times 10^{-3}) (2283 \times 10^3) \left[\frac{9.81 (965.3 - 0.4235)}{0.0608} \right]^{1/2} \left(\frac{4206 (T_s - 90)}{0.0060 (2283 \times 10^3) 1.96} \right)^3 \\ T_s &= \mathbf{92.3^\circ\text{C}}\end{aligned}$$



Condensation occurs when the temperature of a vapor is reduced *below* its saturation temperature.

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.

Condensation

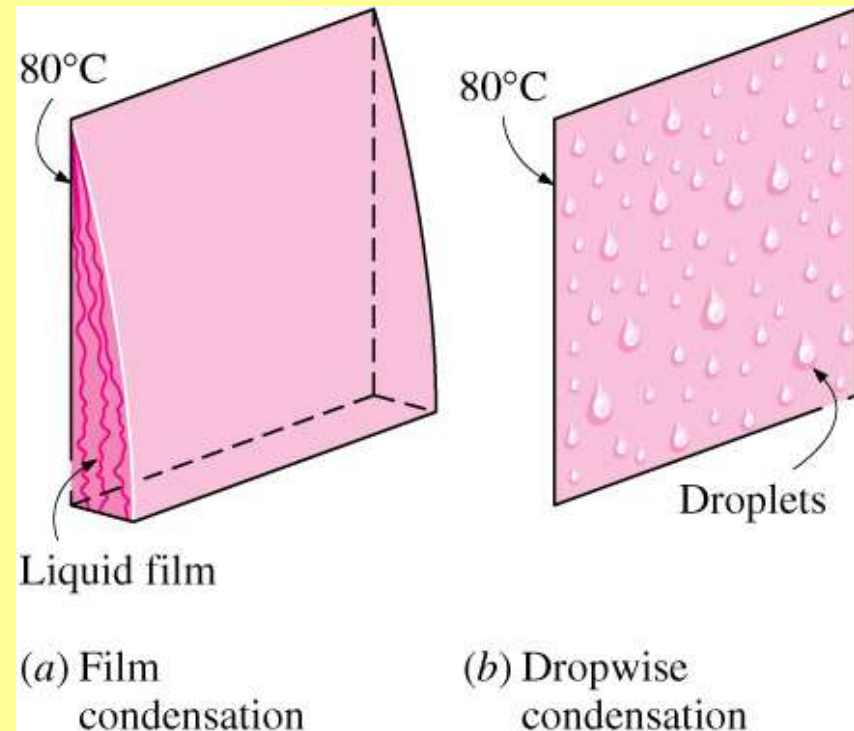


FIGURE 10–20

When a vapor is exposed to a surface at a temperature below T_{sat} , condensation in the form of a liquid film or individual droplets occurs on the surface.

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.

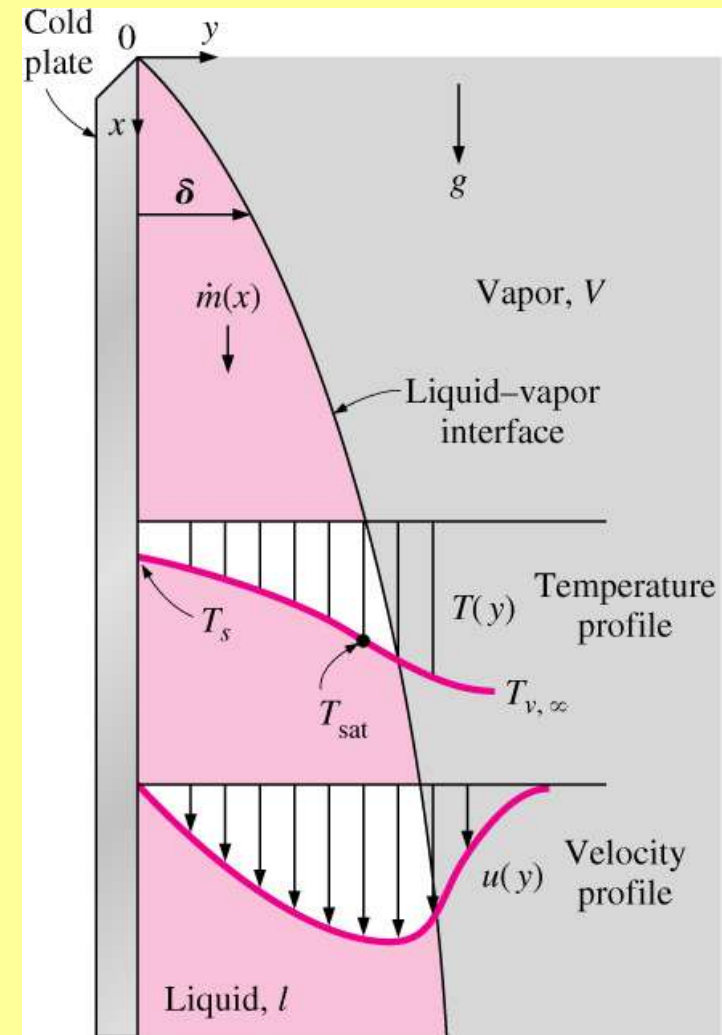


FIGURE 10-21

Film condensation on a vertical plate.

$$\text{Re} = \frac{D_h \rho_l V_l}{\mu_l} = \frac{4 A_c \rho_l V_l}{p \mu_l} = \frac{4 \rho_l V_l \delta}{\mu_l} = \frac{4 \dot{m}}{p \mu_l}$$

$D_h = 4A_c/p = 4\delta$ = hydraulic diameter of the condensate flow, m

p = wetted perimeter of the condensate, m

$A_c = p\delta$ = wetted perimeter \times film thickness, m^2 , cross-sectional area of the condensate flow at the lowest part of the flow

ρ_l = density of the liquid, kg/m^3

μ_l = viscosity of the liquid, $\text{kg}/\text{m} \cdot \text{s}$

V_l = average velocity of the condensate at the lowest part of the flow, m/s

$\dot{m} = \rho_l V_l A_c$ = mass flow rate of the condensate at the lowest part, kg/s

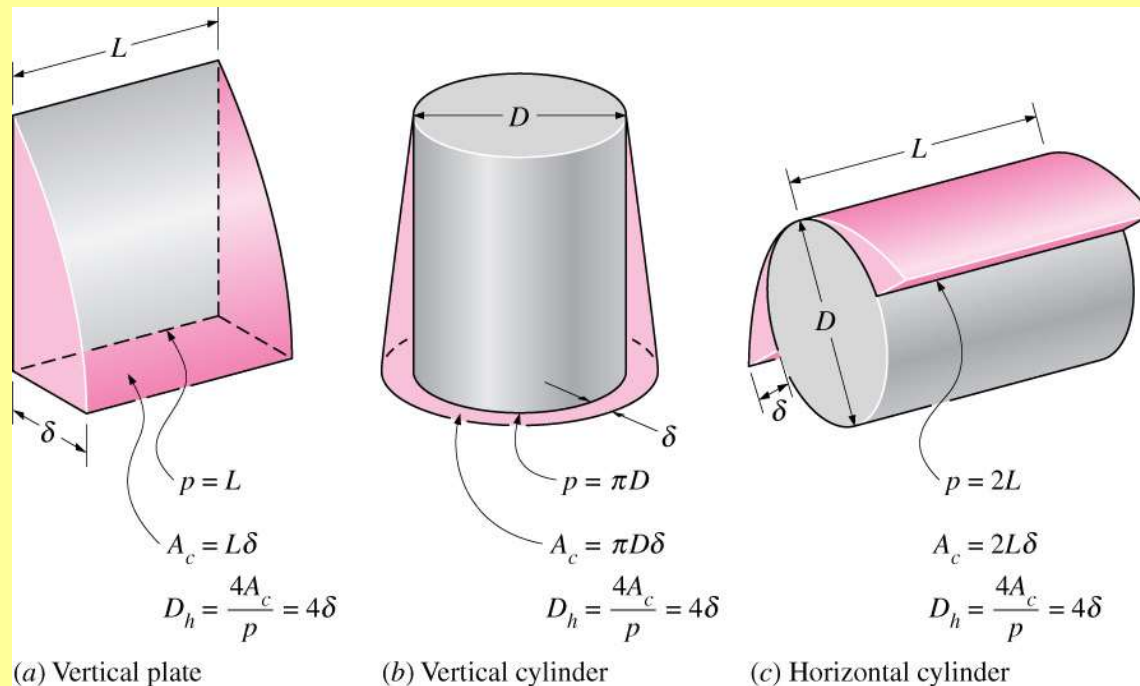


FIGURE 10-22

The wetted perimeter p , the condensate cross-sectional area A_c , and the hydraulic diameter D_h for some common geometries.

modified latent heat of vaporization h_{fg}^* , defined as

$$h_{fg}^* = h_{fg} + 0.68c_{pl}(T_{\text{sat}} - T_s) \quad (10-9a)$$

where c_{pl} is the specific heat of the liquid at the average film temperature.

We can have a similar argument for vapor that enters the condenser as **superheated vapor** at a temperature T_v instead of as saturated vapor. In this case the vapor must be cooled first to T_{sat} before it can condense, and this heat must be transferred to the wall as well. The amount of heat released as a unit mass of superheated vapor at a temperature T_v is cooled to T_{sat} is simply $c_{pv}(T_v - T_{\text{sat}})$, where c_{pv} is the specific heat of the vapor at the average temperature of $(T_v + T_{\text{sat}})/2$. The modified latent heat of vaporization in this case becomes

$$h_{fg}^* = h_{fg} + 0.68c_{pl}(T_{\text{sat}} - T_s) + c_{pv}(T_v - T_{\text{sat}}) \quad (10-9b)$$

With these considerations, the rate of heat transfer can be expressed as

$$\dot{Q}_{\text{conden}} = hA_s(T_{\text{sat}} - T_s) = \dot{m}h_{fg}^* \quad (10-10)$$

where A_s is the heat transfer area (the surface area on which condensation occurs). Solving for \dot{m} from the equation above and substituting it into Eq. 10-8 gives yet another relation for the Reynolds number,

$$\text{Re} = \frac{4\dot{Q}_{\text{conden}}}{p\mu_l h_{fg}^*} = \frac{4A_s h(T_{\text{sat}} - T_s)}{p\mu_l h_{fg}^*} \quad (10-11)$$

This relation is convenient to use to determine the Reynolds number when the condensation heat transfer coefficient or the rate of heat transfer is known.

The temperature of the liquid film varies from T_{sat} on the liquid–vapor interface to T_s at the wall surface. Therefore, the properties of the liquid should be evaluated at the *film temperature* $T_f = (T_{\text{sat}} + T_s)/2$, which is approximately the *average* temperature of the liquid. The h_{fg} , however, should be evaluated at T_{sat} since it is not affected by the subcooling of the liquid.

Vertical Plate — Flow Regimes

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

$$Re = \frac{D_h \rho_l V_l}{\mu_l} = \frac{4 A_c \rho_l V_l}{p \mu_l} = \frac{4 \rho_l V_l \delta}{\mu_l} = \frac{4 \dot{m}}{p \mu_l}$$

- 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.

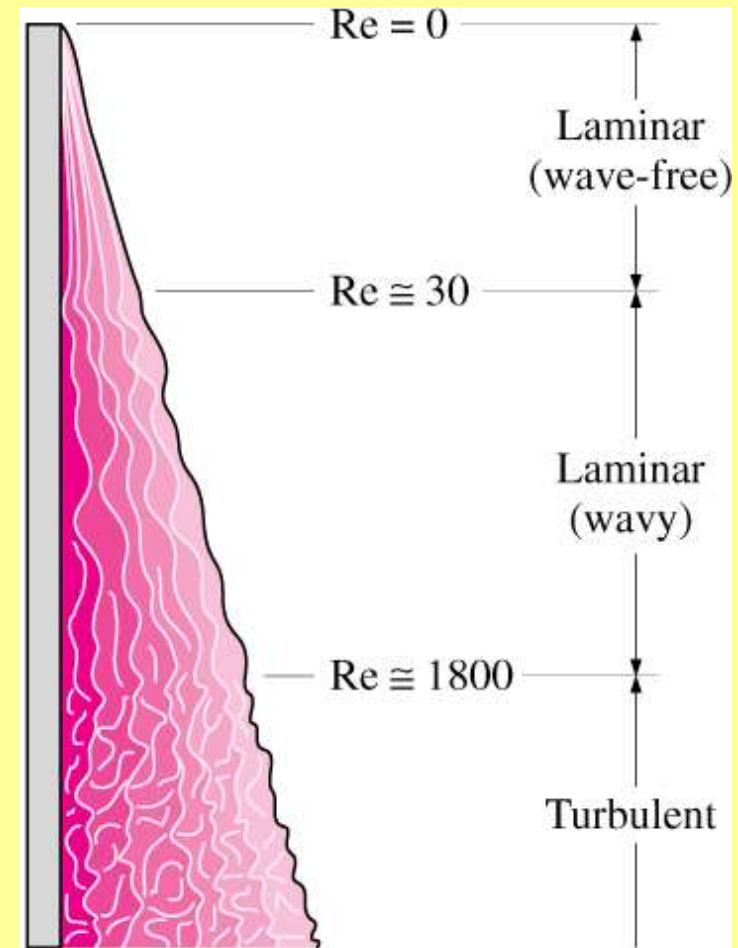


FIGURE 10-23

Flow regimes during film condensation on a vertical plate.

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.

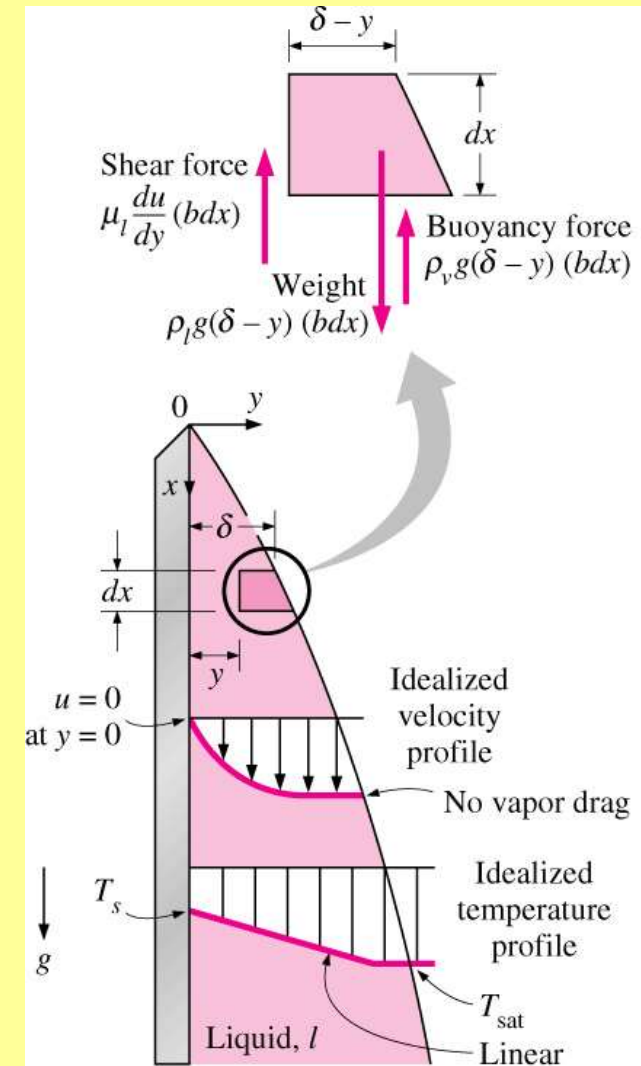


FIGURE 10-24

The volume element of condensate on a vertical plate considered in Nusselt's analysis.

The *average heat transfer coefficient* for laminar film condensation over a vertical flat plate of height L is

$$h_{\text{vert}} = 0.943 \left[\frac{g \rho_l (\rho_l - \rho_v) h_{fg}^* k_l^3}{\mu_l (T_{\text{sat}} - T_s) L} \right]^{1/4} \quad (\text{W/m}^2 \cdot ^\circ\text{C}), \quad 0 < \text{Re} < 30 \quad (10-22)$$

where

g = gravitational acceleration, m/s^2

ρ_l, ρ_v = densities of the liquid and vapor, respectively, kg/m^3

μ_l = viscosity of the liquid, $\text{kg/m} \cdot \text{s}$

$h_{fg}^* = h_{fg} + 0.68 c_{pl} (T_{\text{sat}} - T_s)$ = modified latent heat of vaporization, J/kg

k_l = thermal conductivity of the liquid, $\text{W/m} \cdot ^\circ\text{C}$

L = height of the vertical plate, m

T_s = surface temperature of the plate, $^\circ\text{C}$

T_{sat} = saturation temperature of the condensing fluid, $^\circ\text{C}$

At a given temperature, $\rho_v \ll \rho_l$ and thus $\rho_l - \rho_v \approx \rho_l$ except near the critical point of the substance. Using this approximation and substituting Eqs. 10-14 and 10-18 at $x = L$ into Eq. 10-8 by noting that $\delta_{x=L} = k_l / h_{x=L}$ and $h_{\text{vert}} = \frac{4}{3} h_{x=L}$ (Eqs. 10-19 and 10-21) give

$$\text{Re} \equiv \frac{4g\rho_l(\rho_l - \rho_v)\delta^3}{3\mu_l^2} = \frac{4g\rho_l^2}{3\mu_l^2} \left(\frac{k_l}{h_{x=L}} \right)^3 = \frac{4g}{3\nu_l^2} \left(\frac{k_l}{3h_{\text{vert}}/4} \right)^3 \quad (10-23)$$

Then the heat transfer coefficient h_{vert} in terms of Re becomes

$$h_{\text{vert}} \equiv 1.47 k_l \text{Re}^{-1/3} \left(\frac{g}{\nu_l^2} \right)^{1/3}, \quad \begin{matrix} 0 < \text{Re} < 30 \\ \rho_v \ll \rho_l \end{matrix} \quad (10-24)$$

The average heat transfer coefficient in wavy laminar condensate flow for $\rho_v \ll \rho_l$ and $30 < \text{Re} < 1800$

$$h_{\text{vert, wavy}} = \frac{\text{Re } k_l}{1.08 \text{ Re}^{1.22} - 5.2} \left(\frac{g}{\nu_l^2} \right)^{1/3}, \quad \begin{matrix} 30 < \text{Re} < 1800 \\ \rho_v \ll \rho_l \end{matrix}$$

A simpler alternative to the relation above proposed by Kutateladze (1963) is

$$h_{\text{vert, wavy}} = 0.8 \text{ Re}^{0.11} h_{\text{vert (smooth)}} \quad (10-26)$$

which relates the heat transfer coefficient in wavy laminar flow to that in wave-free laminar flow. McAdams (1954) went even further and suggested accounting for the increase in heat transfer in the wavy region by simply increasing the heat transfer coefficient determined from Eq. 10-22 for the laminar case by 20 percent. It is also suggested using Eq. 10-22 for the wavy region also, with the understanding that this is a conservative approach that provides a safety margin in thermal design. In this book we use Eq. 10-25.

A relation for the Reynolds number in the wavy laminar region can be determined by substituting the h relation in Eq. 10-25 into the Re relation in Eq. 10-11 and simplifying. It yields

$$\text{Re}_{\text{vert, wavy}} = \left[4.81 + \frac{3.70 L k_l (T_{\text{sat}} - T_s)}{\mu_l h_{fg}^*} \left(\frac{g}{\nu_l^2} \right)^{1/3} \right]^{0.820}, \quad \rho_v \ll \rho_l \quad (10-27)$$

Turbulent Flow on Vertical Plates

At a Reynolds number of about 1800, the condensate flow becomes turbulent. Several empirical relations of varying degrees of complexity are proposed for the heat transfer coefficient for turbulent flow. Again assuming $\rho_v \ll \rho_l$ for simplicity, Labuntsov (1957) proposed the following relation for the turbulent flow of condensate on *vertical plates*:

$$h_{\text{vert, turbulent}} = \frac{\text{Re } k_l}{8750 + 58 \text{Pr}^{-0.5} (\text{Re}^{0.75} - 253)} \left(\frac{g}{\nu_l^2} \right)^{1/3}, \quad \text{Re} > 1800, \quad \rho_v \ll \rho_l \quad (10-28)$$

The physical properties of the condensate are again to be evaluated at the film temperature $T_f = (T_{\text{sat}} + T_s)/2$. The Re relation in this case is obtained by substituting the h relation above into the Re relation in Eq. 10-11, which gives

$$\text{Re}_{\text{vert, turbulent}} = \left[\frac{0.0690 L k_l \text{Pr}^{0.5} (T_{\text{sat}} - T_s)}{\mu_l h_{fg}^*} \left(\frac{g}{\nu_l^2} \right)^{1/3} - 151 \text{Pr}^{0.5} + 253 \right]^{4/3} \quad (10-29)$$

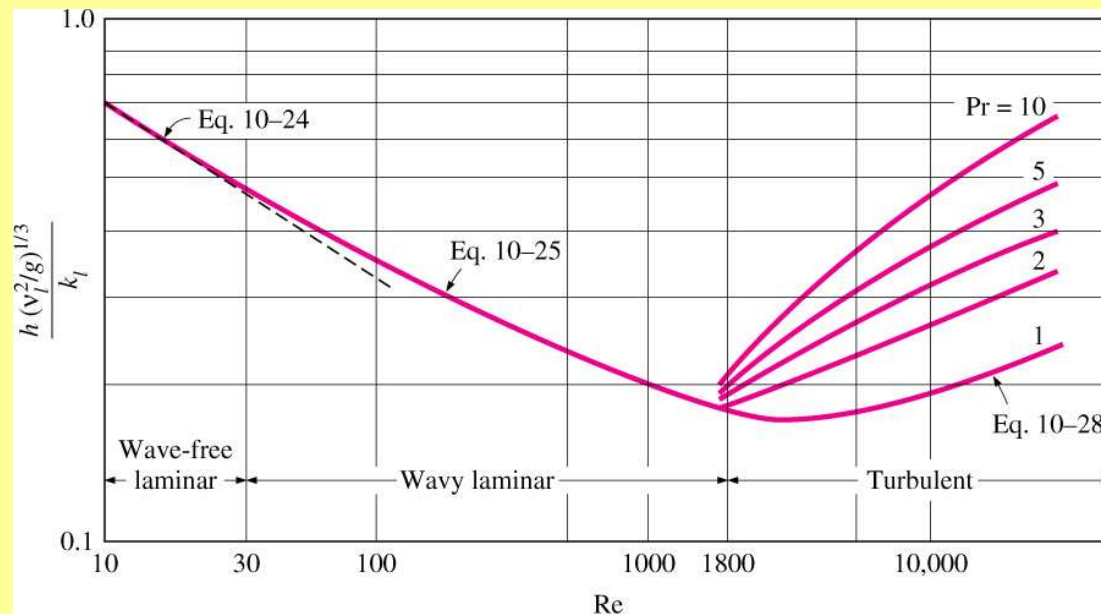


FIGURE 10-26

Nondimensionalized heat transfer coefficients for the wave-free laminar, wavy laminar, and turbulent flow of condensate on vertical plates.

Inclined Plates

$$h_{\text{inclined}} = h_{\text{vert}} (\cos \theta)^{1/4} \quad (\text{laminar})$$

Horizontal tubes and spheres

$$h_{\text{horiz}} = 0.729 \left[\frac{g \rho_l (\rho_l - \rho_v) h_{fg}^* k_l^3}{\mu_l (T_{\text{sat}} - T_s) D} \right]^{1/4}$$

Inclined Plates

$$\frac{h_{\text{vert}}}{h_{\text{horiz}}} = 1.29 \left(\frac{D}{L} \right)^{1/4}$$

Horizontal tube banks

$$h_{\text{horiz}, N \text{ tubes}} = 0.729 \left[\frac{g \rho_l (\rho_l - \rho_v) h_{fg}^* k_l^3}{\mu_l (T_{\text{sat}} - T_s) N D} \right]^{1/4} = \frac{1}{N^{1/4}} h_{\text{horiz}, 1 \text{ tube}}$$

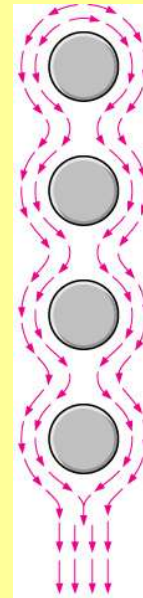


FIGURE 10-28
Film condensation on a vertical tier of horizontal tubes.

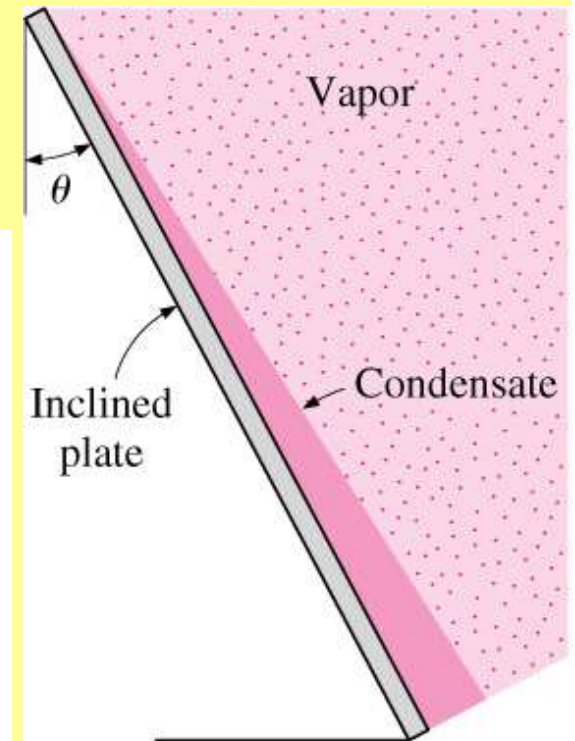


FIGURE 10-27
Film condensation on an inclined plate.

For *low vapor velocities*, Chato (1962) recommends this expression for condensation

$$h_{\text{internal}} = 0.555 \left[\frac{g \rho_l (\rho_l - \rho_v) k_l^3}{\mu_l (T_{\text{sat}} - T_s)} \left(h_{fg} + \frac{3}{8} c_{pl} (T_{\text{sat}} - T_s) \right) \right]^{1/4} \quad (10-34)$$

for

$$\text{Re}_{\text{vapor}} = \left(\frac{\rho_v V_v D}{\mu_v} \right)_{\text{inlet}} < 35,000 \quad (10-35)$$

Film condensation inside horizontal tubes

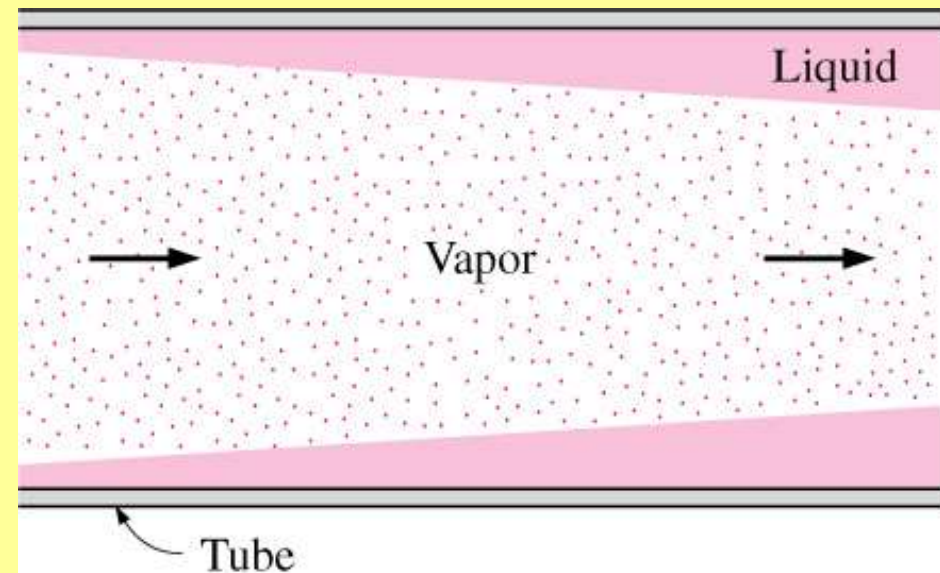


FIGURE 10-34

Condensate flow in a horizontal tube
with large vapor velocities.

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**.

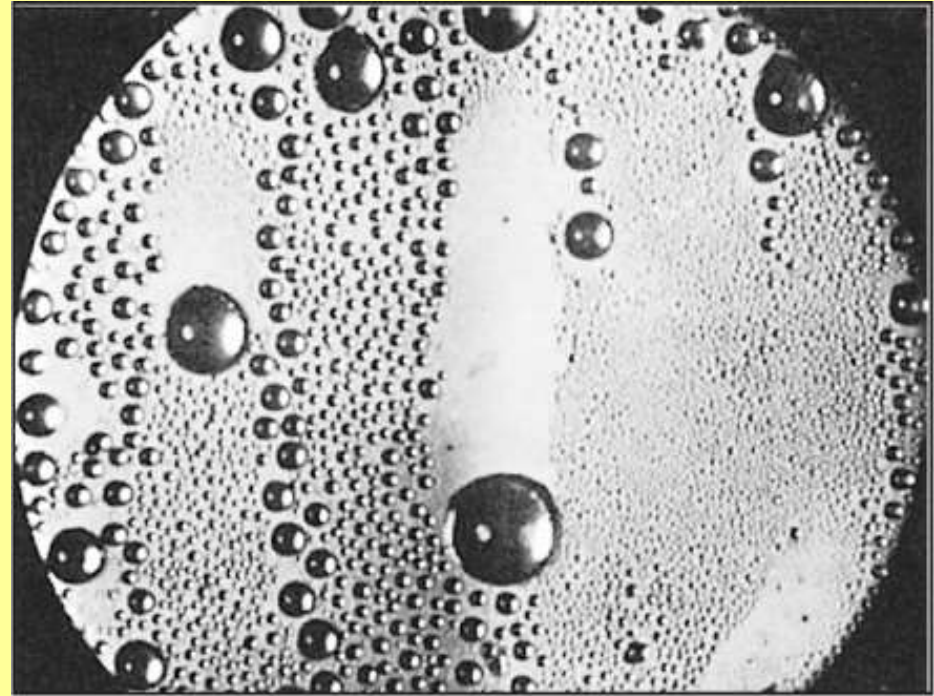


FIGURE 10–35

Dropwise condensation of steam on a vertical surface.

(From Hampson and Özişik.)

- 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*:

$$h_{\text{dropwise}} = \begin{cases} 51,104 + 2044T_{\text{sat}}, & 22^{\circ}\text{C} < T_{\text{sat}} < 100^{\circ}\text{C} \\ 255,310 & T_{\text{sat}} > 100^{\circ}\text{C} \end{cases} \quad \begin{matrix} (10-36) \\ (10-37) \end{matrix}$$

where T_{sat} is in $^{\circ}\text{C}$ and the heat transfer coefficient h_{dropwise} is in $\text{W}/\text{m}^2 \cdot ^{\circ}\text{C}$.

10–54 Saturated steam at 55°C is to be condensed at a rate of 10 kg/h on the outside of a 3-cm -outer-diameter vertical tube whose surface is maintained at 45°C by the cooling water. Determine the required tube length.

Properties The properties of water at the saturation temperature of 55°C are $h_{fg} = 2371 \times 10^3\text{ J/kg}$ and $\rho_v = 0.1045\text{ kg/m}^3$. The properties of liquid water at the film temperature of $T_f = (T_{\text{sat}} + T_s)/2 = (55 + 45)/2 = 50^\circ\text{C}$ are (Table A-9),

$$\rho_l = 988.1\text{ kg/m}^3$$

$$\mu_l = 0.547 \times 10^{-3}\text{ kg/m}\cdot\text{s}$$

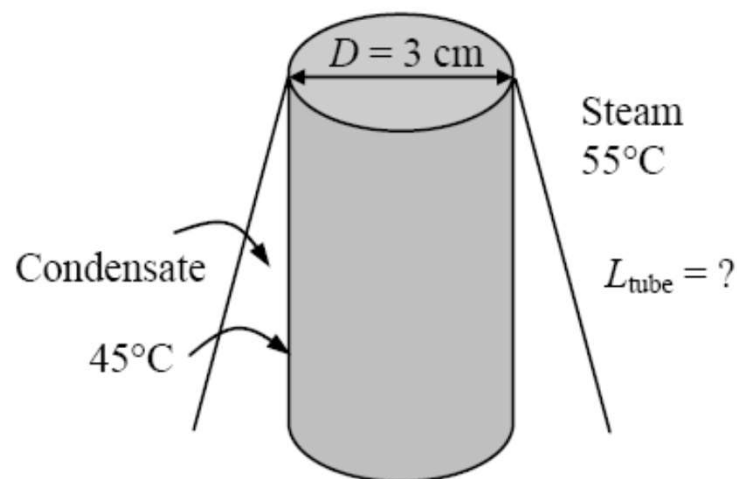
$$\nu_l = \mu_l / \rho_l = 0.554 \times 10^{-6}\text{ m}^2/\text{s}$$

$$c_{pl} = 4181\text{ J/kg}\cdot^\circ\text{C}$$

$$k_l = 0.644\text{ W/m}\cdot^\circ\text{C}$$

Analysis The modified latent heat of vaporization is

$$\begin{aligned} h_{fg}^* &= h_{fg} + 0.68c_{pl}(T_{\text{sat}} - T_s) \\ &= 2371 \times 10^3\text{ J/kg} + 0.68 \times 4181\text{ J/kg}\cdot^\circ\text{C}(55 - 45)^\circ\text{C} = 2399 \times 10^3\text{ J/kg} \end{aligned}$$



The Reynolds number is determined from its definition to be

$$\text{Re} = \frac{4\dot{m}}{p\mu_l} = \frac{4(10 / 3600 \text{ kg/s})}{\pi(0.03 \text{ m})(0.547 \times 10^{-3} \text{ kg/m} \cdot \text{s})} = 215.5$$

which is between 30 and 1800. Therefore the condensate flow is wavy laminar, and the condensation heat transfer coefficient is determined from

$$\begin{aligned} h &= h_{\text{vertical,wavy}} = \frac{\text{Re } k_l}{1.08 \text{Re}^{1.22} - 5.2} \left(\frac{g}{\nu_l^2} \right)^{1/3} \\ &= \frac{215.5 \times (0.644 \text{ W/m} \cdot ^\circ\text{C})}{1.08(215.5)^{1.22} - 5.2} \left(\frac{9.8 \text{ m/s}^2}{(0.554 \times 10^{-6} \text{ m}^2/\text{s})^2} \right)^{1/3} = 5644 \text{ W/m}^2 \cdot ^\circ\text{C} \end{aligned}$$

The rate of heat transfer during this condensation process is

$$\dot{Q} = \dot{m}h_{fg}^* = (10 / 3600 \text{ kg/s})(2399 \times 10^3 \text{ J/kg}) = 6,664 \text{ W}$$

Heat transfer can also be expressed as

$$\dot{Q} = hA_s(T_{\text{sat}} - T_s) = h(\pi DL)(T_{\text{sat}} - T_s)$$

Then the required length of the tube becomes

$$L = \frac{\dot{Q}}{h(\pi D)(T_{\text{sat}} - T_s)} = \frac{6664 \text{ W}}{(5644 \text{ W/m}^2 \cdot ^\circ\text{C})\pi(0.03 \text{ m})(55 - 45)^\circ\text{C}} = \mathbf{1.21 \text{ m}}$$

Concluding Points

- Boiling heat transfer
- Pool boiling
 - Boiling regimes and the boiling curve
- Flow boiling
- Condensation heat transfer
- Film condensation
- Film condensation inside horizontal tubes
- Dropwise condensation