

and boiling in this case is called the *bulk* or *saturated* boiling. Next, we consider different boiling regimes in detail.

10-2 ■ POOL BOILING

So far we presented some general discussions on boiling. Now we turn our attention to the physical mechanisms involved in *pool boiling*, that is, the boiling of stationary fluids. In pool boiling, the fluid is not forced to flow by a mover such as a pump, and any motion of the fluid is due to natural convection currents and the motion of the bubbles under the influence of buoyancy.

As a familiar example of pool boiling, consider the boiling of tap water in a pan on top of a stove. The water will initially be at about 15°C , far below the saturation temperature of 100°C at standard atmospheric pressure. At the early stages of boiling, you will not notice anything significant except some bubbles that stick to the surface of the pan. These bubbles are caused by the release of air molecules dissolved in liquid water and should not be confused with vapor bubbles. As the water temperature rises, you will notice chunks of liquid water rolling up and down as a result of natural convection currents, followed by the first vapor bubbles forming at the bottom surface of the pan. These bubbles get smaller as they detach from the surface and start rising, and eventually collapse in the cooler water above. This is *subcooled boiling* since the bulk of the liquid water has not reached saturation temperature yet. The intensity of bubble formation increases as the water temperature rises further, and you will notice waves of vapor bubbles coming from the bottom and rising to the top when the water temperature reaches the saturation temperature (100°C at standard atmospheric conditions). This full scale boiling is the *saturated boiling*.

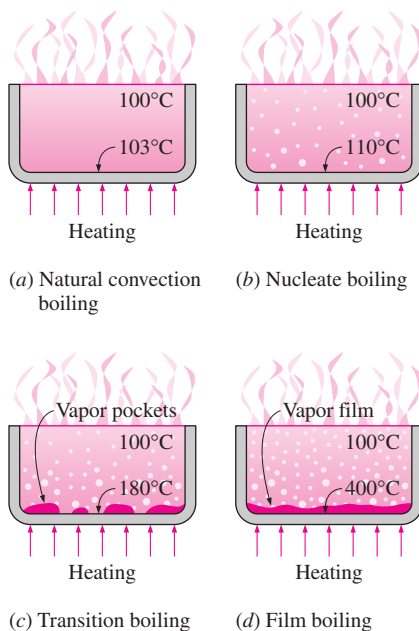
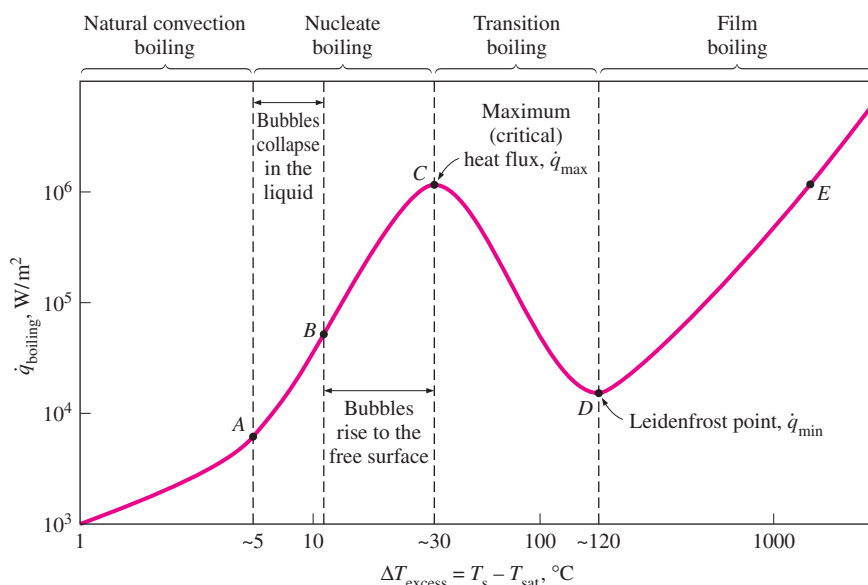


FIGURE 10-5
Different boiling regimes
in pool boiling.

Boiling Regimes and the Boiling Curve

Boiling is probably the most familiar form of heat transfer, yet it remains to be the least understood form. After hundreds of papers written on the subject, we still do not fully understand the process of bubble formation and we must still rely on empirical or semi-empirical relations to predict the rate of boiling heat transfer.

The pioneering work on boiling was done in 1934 by S. Nukiyama, who used electrically heated nichrome and platinum wires immersed in liquids in his experiments. Nukiyama noticed that boiling takes different forms, depending on the value of the excess temperature ΔT_{excess} . Four different boiling regimes are observed: *natural convection boiling*, *nucleate boiling*, *transition boiling*, and *film boiling* (Fig. 10-5). These regimes are illustrated on the **boiling curve** in Figure 10-6, which is a plot of boiling heat flux versus the excess temperature. Although the boiling curve given in this figure is for water, the general shape of the boiling curve remains the same for different fluids. The specific shape of the curve depends on the fluid–heating surface material combination and the fluid pressure, but it is practically independent of the geometry of the heating surface. We will describe each boiling regime in detail.

**FIGURE 10–6**

Typical boiling curve for water at 1 atm pressure.

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

We learned in thermodynamics that a pure substance at a specified pressure starts boiling when it reaches the saturation temperature at that pressure. But in practice we do not see any bubbles forming on the heating surface until the liquid is heated a few degrees above the saturation temperature (about 2 to 6°C for water). Therefore, the liquid is slightly *superheated* in this case (a *metastable* condition) and evaporates when it rises to the free surface. The fluid motion in this mode of boiling is governed by natural convection currents, and heat transfer from the heating surface to the fluid is by natural convection.

Nucleate Boiling (between Points A and C)

The first bubbles start forming at point A of the boiling curve at various preferential sites on the heating surface. The bubbles form at an increasing rate at an increasing number of nucleation sites as we move along the boiling curve toward point C.

The nucleate boiling regime can be separated into two distinct regions. In region A–B, *isolated bubbles* are formed at various preferential nucleation sites on the heated surface. But these bubbles are dissipated in the liquid shortly after they separate from the surface. The space vacated by the rising bubbles is filled by the liquid in the vicinity of the heater surface, and the process is repeated. The stirring and agitation caused by the entrainment of the liquid to the heater surface is primarily responsible for the increased heat transfer coefficient and heat flux in this region of nucleate boiling.

In region B–C, the heater temperature is further increased, and bubbles form at such great rates at such a large number of nucleation sites that they form numerous *continuous columns of vapor* in the liquid. These bubbles move all the way up to the free surface, where they break up and release their vapor content. The large heat fluxes obtainable in this region are caused by the combined effect of liquid entrainment and evaporation.

At large values of ΔT_{excess} , the rate of evaporation at the heater surface reaches such high values that a large fraction of the heater surface is covered by bubbles, making it difficult for the liquid to reach the heater surface and wet it. Consequently, 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**, $\dot{q}_{\text{max}}$. For water, the critical heat flux exceeds 1 MW/m^2 .

Nucleate boiling is the most desirable boiling regime in practice because high heat transfer rates can be achieved in this regime with relatively small values of ΔT_{excess} , typically under 30°C for water. The photographs in Figure 10–7 show the nature of bubble formation and bubble motion associated with nucleate, transition, and film boiling.

Transition Boiling (between Points C and D on the Boiling Curve)

As the heater temperature and thus the ΔT_{excess} is increased past point C, the heat flux decreases, as shown in Figure 10–6. This is because a large fraction of the heater surface is covered by a vapor film, which acts as an insulation due to the low thermal conductivity of the vapor relative to that of the liquid. In the transition boiling regime, both nucleate and film boiling partially occur. Nucleate boiling at point C is completely replaced by film boiling at point D. Operation in the transition boiling regime, which is also called the *unstable film boiling regime*, is avoided in practice. For water, transition boiling occurs over the excess temperature range from about 30°C to about 120°C .

Film Boiling (beyond Point D)

In this region 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**, in honor of J. C. Leidenfrost, who observed in 1756 that liquid droplets on a very hot surface jump around and slowly boil away. 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 as a result of heat transfer from the heated surface to the liquid through the vapor film by radiation, which becomes significant at high temperatures.

A typical boiling process will not follow the boiling curve beyond point C, as Nukiyama has observed during his experiments. Nukiyama noticed, with surprise, that when the power applied to the nichrome wire immersed in water exceeded $\dot{q}_{\text{max}}$ even slightly, the wire temperature increased suddenly to the melting point of the wire and *burnout* occurred beyond his control. When he repeated the experiments with platinum wire, which has a much higher melting point, he was able to avoid burnout and maintain heat fluxes higher than $\dot{q}_{\text{max}}$. When he gradually reduced power, he obtained the cooling curve shown in Figure 10–8 with a sudden drop in excess temperature when $\dot{q}_{\text{min}}$ is reached. Note that the boiling process cannot follow the transition boiling part of the boiling curve past point C unless the power applied is reduced suddenly.

The *burnout phenomenon* in boiling can be explained as follows: In order to move beyond point C where $\dot{q}_{\text{max}}$ occurs, we must increase the heater surface temperature T_s . To increase T_s , however, we must increase the heat flux. But

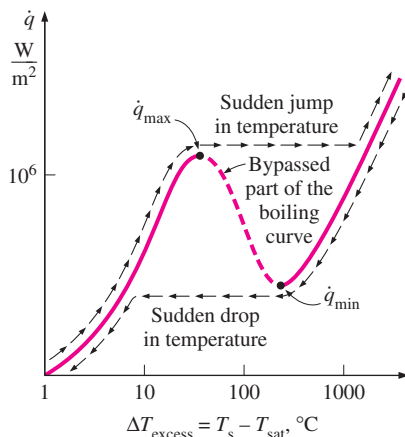
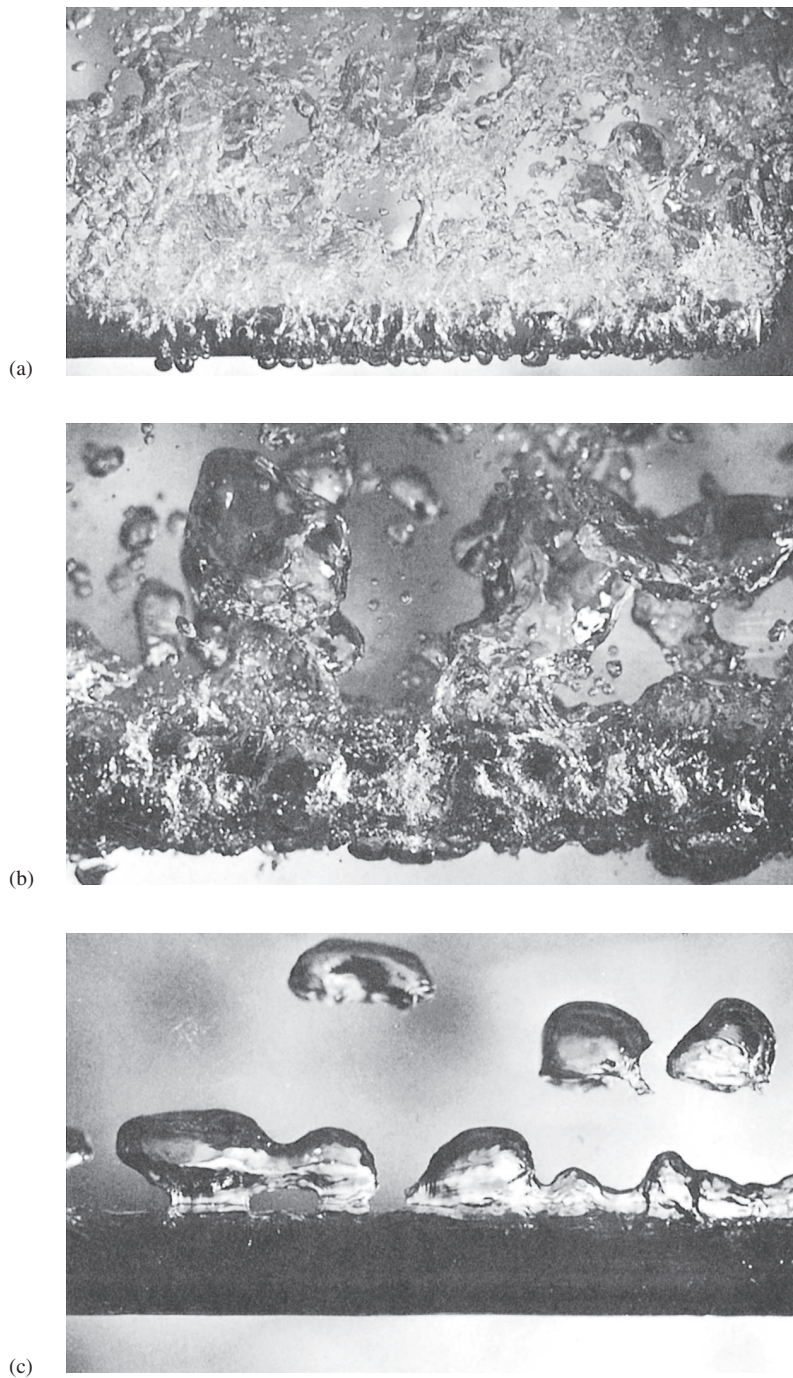


FIGURE 10–8

The actual boiling curve obtained with heated platinum wire in water as the heat flux is increased and then decreased.

**FIGURE 10-7**

Various boiling regimes during boiling of methanol on a horizontal 1-cm-diameter steam-heated copper tube: (a) nucleate boiling, (b) transition boiling, and (c) film boiling (from J. W. Westwater and J. G. Santangelo, University of Illinois at Champaign-Urbana).

the fluid cannot receive this increased energy at an excess temperature just beyond point *C*. Therefore, the heater surface ends up absorbing the increased energy, causing the heater surface temperature T_s to rise. But the fluid can receive even less energy at this increased excess temperature, causing the heater surface temperature T_s to rise even further. This continues until the surface

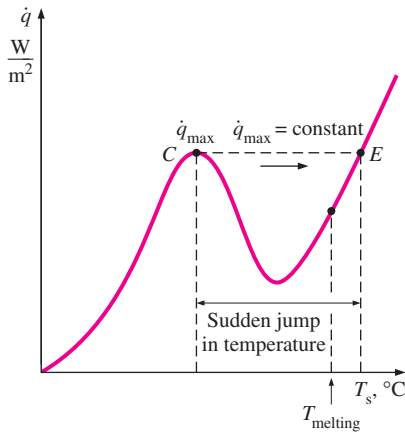


FIGURE 10-9

An attempt to increase the boiling heat flux beyond the *critical* value often causes the temperature of the heating element to jump suddenly to a value that is above the melting point, resulting in *burnout*.

temperature reaches a point at which it no longer rises and the heat supplied can be transferred to the fluid steadily. This is point *E* on the boiling curve, which corresponds to very high surface temperatures. Therefore, any attempt to increase the heat flux beyond $\dot{q}_{\max}$ will cause the operation point on the boiling curve to jump suddenly from point *C* to point *E*. However, surface temperature that corresponds to point *E* is beyond the melting point of most heater materials, and *burnout* occurs. Therefore, point *C* on the boiling curve is also called the **burnout point**, and the heat flux at this point the **burnout heat flux** (Fig. 10–9).

Most boiling heat transfer equipment in practice operate slightly below $\dot{q}_{\max}$ to avoid any disastrous burnout. However, in cryogenic applications involving fluids with very low boiling points such as oxygen and nitrogen, point *E* usually falls below the melting point of the heater materials, and steady film boiling can be used in those cases without any danger of burnout.

Heat Transfer Correlations in Pool Boiling

Boiling regimes discussed above differ considerably in their character, and thus different heat transfer relations need to be used for different boiling regimes. In the *natural convection boiling* regime, boiling is governed by natural convection currents, and heat transfer rates in this case can be determined accurately using natural convection relations presented in Chapter 9.

Nucleate Boiling

In the *nucleate boiling* regime, the rate of heat transfer strongly depends on the nature of nucleation (the number of active nucleation sites on the surface, the rate of bubble formation at each site, etc.), which is difficult to predict. The type and the condition of the heated surface also affect the heat transfer. These complications made it difficult to develop theoretical relations for heat transfer in the nucleate boiling regime, and people had to rely on relations based on experimental data. The most widely used correlation for the rate of heat transfer in the nucleate boiling regime was proposed in 1952 by Rohsenow, and expressed as

$$\dot{q}_{\text{nucleate}} = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma} \right]^{1/2} \left[\frac{C_p(T_s - T_{\text{sat}})}{C_{sf} h_{fg} \text{Pr}_l^n} \right]^3 \quad (10-2)$$

where

- $\dot{q}_{\text{nucleate}}$ = 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

It can be shown easily that using property values in the specified units in the Rohsenow equation produces the desired unit W/m^2 for the boiling heat flux, thus saving one from having to go through tedious unit manipulations (Fig. 10–10).

The surface tension at the vapor–liquid interface is given in Table 10–1 for water, and Table 10–2 for some other fluids. Experimentally determined values of the constant C_{sf} are given in Table 10–3 for various fluid–surface combinations. These values 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 fluid properties in Eq. 10–2 are to be evaluated at the saturation temperature T_{sat} .

The *condition* of the heater surface greatly affects heat transfer, and the Rohsenow equation given above is applicable to *clean* and relatively *smooth* surfaces. The results obtained using the Rohsenow equation can be in error by $\pm 100\%$ for the heat transfer rate for a given excess temperature and by $\pm 30\%$ for the excess temperature for a given heat transfer rate. Therefore, care should be exercised in the interpretation of the results.

Recall from thermodynamics that the enthalpy of vaporization h_{fg} of a pure substance decreases with increasing pressure (or temperature) and reaches zero at the critical point. Noting that h_{fg} appears in the denominator of the Rohsenow equation, we should see a significant rise in the rate of heat transfer at *high pressures* during nucleate boiling.

Peak Heat Flux

In the design of boiling heat transfer equipment, it is extremely important for the designer to have a knowledge of the maximum heat flux in order to avoid the danger of burnout. 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 using quite different approaches, and is expressed as (Fig. 10–11)

$$\dot{q}_{\text{max}} = C_{cr} h_{fg} [\sigma g \rho_v^2 (\rho_l - \rho_v)]^{1/4} \quad (10-3)$$

where C_{cr} is a constant whose value depends on the heater geometry. Exhaustive experimental studies by Lienhard and his coworkers indicated that the value of C_{cr} is about 0.15. Specific values of C_{cr} for different heater geometries are listed in Table 10–4. Note that the heaters are classified as being large or small based on the value of the parameter L^* .

Equation 10–3 will give the maximum heat flux in W/m^2 if the properties are used *in the units specified* earlier in their descriptions following Eq. 10–2. The maximum heat flux is independent of the fluid–heating surface combination, as well as the viscosity, thermal conductivity, and the specific heat of the liquid.

Note that ρ_v increases but σ and h_{fg} decrease with increasing pressure, and thus the change in $\dot{q}_{\text{max}}$ with pressure depends on which effect dominates. The experimental studies of Cichelli and Bonilla indicate that $\dot{q}_{\text{max}}$ increases with pressure up to about one-third of the critical pressure, and then starts to decrease and becomes zero at the critical pressure. Also note that $\dot{q}_{\text{max}}$ is proportional to h_{fg} , and large maximum heat fluxes can be obtained using fluids with a large enthalpy of vaporization, such as water.

$$\begin{aligned} \dot{q} &= \left(\frac{\text{kg}}{\text{m} \cdot \text{s}} \right) \left(\frac{\text{J}}{\text{kg}} \right) \\ &\times \left(\frac{\frac{\text{m kg}}{\text{s}^2 \text{m}^3}}{\frac{\text{N}}{\text{m}}} \right)^{1/2} \left(\frac{\frac{\text{J}}{\text{kg} \cdot ^\circ\text{C}}}{\frac{\text{J}}{\text{kg}}} \right)^3 \\ &= \frac{\text{W}}{\text{m}} \left(\frac{1}{\text{m}^2} \right)^{1/2} (1)^3 \\ &= \text{W/m}^2 \end{aligned}$$

FIGURE 10–10

Equation 10–2 gives the boiling heat flux in W/m^2 when the quantities are expressed in the units specified in their descriptions.

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 lbm/s^2 .

TABLE 10-2

Surface tension of some fluids (from Suryanarayana, Ref. 26; originally based on data from Jasper, Ref. 14)

Substance and Temp. Range	Surface Tension, σ , N/m* (T in °C)
Ammonia, -75 to -40°C:	$0.0264 + 0.000223T$
Benzene, 10 to 80°C:	$0.0315 - 0.000129T$
Butane, -70 to -20°C:	$0.0149 - 0.000121T$
Carbon dioxide, -30 to -20°C:	$0.0043 - 0.000160T$
Ethyl alcohol, 10 to 70°C:	$0.0241 - 0.000083T$
Mercury, 5 to 200°C:	$0.4906 - 0.000205T$
Methyl alcohol, 10 to 60°C:	$0.0240 - 0.000077T$
Pentane, 10 to 30°C:	$0.0183 - 0.000110T$
Propane, -90 to -10°C:	$0.0092 - 0.000087T$

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

TABLE 10-3

Values 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

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. Using the stability theory, Zuber derived the following expression for the minimum heat flux for a *large horizontal plate*,

$$\dot{q}_{\min} = 0.09\rho_v h_{fg} \left[\frac{\sigma g(\rho_l - \rho_v)}{(\rho_l + \rho_v)^2} \right]^{1/4} \quad (10-4)$$

where the constant 0.09 was determined by Berenson in 1961. He replaced the theoretically determined value of $\frac{\pi}{24}$ by 0.09 to match the experimental data better. Still, the relation above can be in error by 50 percent or more.

TABLE 10-4

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

Film Boiling

Using an analysis similar to Nusselt's theory on filmwise condensation presented in the next section, Bromley developed a theory for the prediction of heat flux for stable *film boiling* on the outside of a horizontal cylinder. The heat flux for film boiling on a *horizontal cylinder* or *sphere* of diameter D is given by

$$\dot{q}_{\text{film}} = C_{\text{film}} \left[\frac{g k_v^3 \rho_v (\rho_l - \rho_v) [h_{fg} + 0.4 C_{pv} (T_s - T_{\text{sat}})]^{1/4}}{\mu_v D (T_s - T_{\text{sat}})} \right] (T_s - T_{\text{sat}}) \quad (10-5)$$

where k_v is the thermal conductivity of the vapor in $\text{W/m} \cdot ^\circ\text{C}$ and

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

Other properties are as listed before in connection with Eq. 10-2. We used a modified latent heat of vaporization in Eq. 10-5 to account for the heat transfer associated with the superheating of the vapor.

The *vapor* properties are to be evaluated at the *film temperature*, given as $T_f = (T_s + T_{\text{sat}})/2$, which is the *average temperature* of the vapor film. The liquid properties and h_{fg} are to be evaluated at the saturation temperature at the specified pressure. Again, this relation will give the film boiling heat flux in W/m^2 if the properties are used *in the units specified* earlier in their descriptions following Eq. 10-2.

At high surface temperatures (typically above 300°C), heat transfer across the vapor film by *radiation* becomes significant and needs to be considered (Fig. 10-12). Treating the vapor film as a transparent medium sandwiched between two large parallel plates and approximating the liquid as a blackbody, *radiation heat transfer* can be determined from

$$\dot{q}_{\text{rad}} = \varepsilon \sigma (T_s^4 - T_{\text{sat}}^4) \quad (10-6)$$

where ε is the emissivity of the heating surface and $\sigma = 5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4$ is the Stefan–Boltzman constant. Note that the temperature in this case *must* be expressed in K, not $^\circ\text{C}$, and that surface tension and the Stefan–Boltzman constant share the same symbol.

You may be tempted to simply add the convection and radiation heat transfers to determine the total heat transfer during film boiling. However, these two mechanisms of heat transfer adversely affect each other, causing the total heat transfer to be less than their sum. For example, the radiation heat transfer from the surface to the liquid enhances the rate of evaporation, and thus the thickness of the vapor film, which impedes convection heat transfer. For $\dot{q}_{\text{rad}} < \dot{q}_{\text{film}}$, Bromley determined that the relation

$$\dot{q}_{\text{total}} = \dot{q}_{\text{film}} + \frac{3}{4} \dot{q}_{\text{rad}} \quad (10-7)$$

correlates experimental data well.

Operation in the *transition boiling* regime is normally avoided in the design of heat transfer equipment, and thus no major attempt has been made to develop general correlations for boiling heat transfer in this regime.

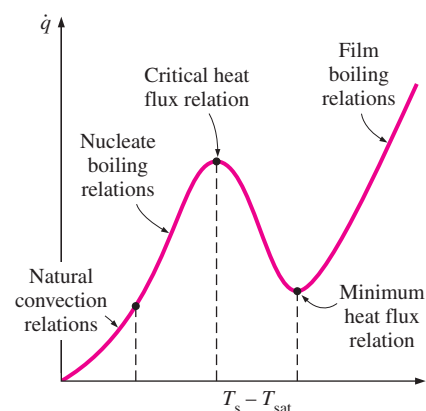


FIGURE 10-11

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

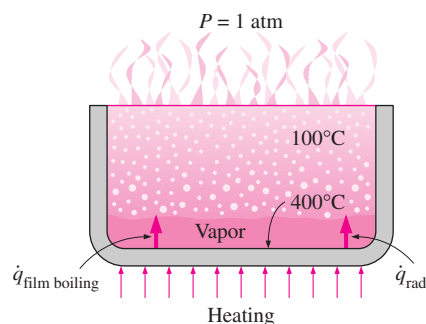


FIGURE 10-12

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

Note that the gravitational acceleration g , whose value is approximately 9.81 m/s^2 at sea level, appears in all of the relations above for boiling heat transfer. The effects of low and high gravity (as encountered in aerospace applications and turbomachinery) are studied experimentally. The studies confirm that the critical heat flux and heat flux in film boiling are proportional to $g^{1/4}$. However, they indicate that heat flux in nucleate boiling is practically independent of gravity g , instead of being proportional to $g^{1/2}$, as dictated by Eq. 10–2.

Enhancement of Heat Transfer in Pool Boiling

The pool boiling heat transfer relations given above apply to smooth surfaces. Below we will discuss some methods to enhance heat transfer in pool boiling.

We pointed out earlier that 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, any modification that will enhance *nucleation* on the heating surface will also enhance *heat transfer* in nucleate boiling. It is observed that *irregularities* on the heating surface, including roughness and dirt, serve as additional nucleation sites during boiling, as shown in Figure 10–13. For example, the first bubbles in a pan filled with water are most likely to form at the *scratches* at the bottom surface. These scratches act like “nests” for the bubbles to form and thus increase the rate of bubble formation. Berensen has shown that heat flux in the nucleate boiling regime can be increased by a factor of 10 by *roughening* the heating surface. However, these high heat transfer rates cannot be sustained for long since the effect of surface roughness is observed to decay with time, and the heat flux to drop eventually to values encountered on smooth surfaces. The effect of surface roughness is negligible on the critical heat flux and the heat flux in film boiling.

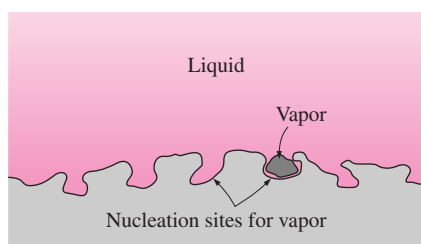


FIGURE 10–13

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

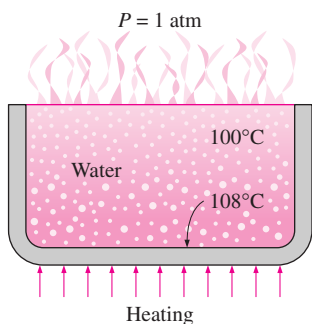


FIGURE 10–15

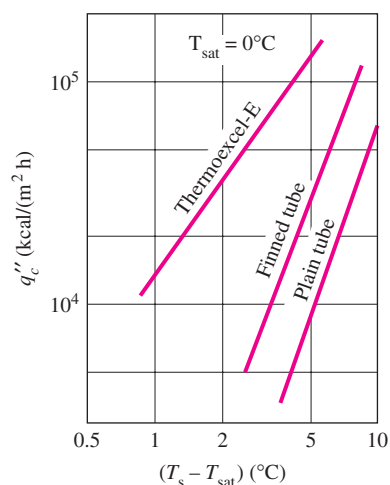
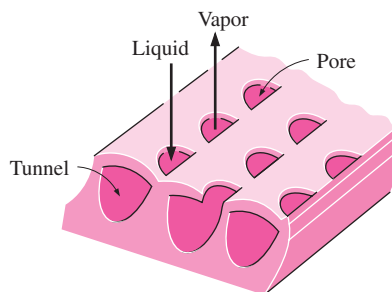
Schematic for Example 10–1.

Surfaces that provide enhanced heat transfer in nucleate boiling *permanently* are being manufactured and are available in the market. Enhancement in nucleation and thus heat transfer in such special surfaces is achieved either by *coating* the surface with a thin layer (much less than 1 mm) of very porous material or by *forming cavities* on the surface mechanically to facilitate continuous vapor formation. Such surfaces are reported to enhance heat transfer in the nucleate boiling regime by a factor of up to 10, and the critical heat flux by a factor of 3. The enhancement provided by one such material prepared by machine roughening, the thermoexcel-E, is shown in Figure 10–14. The use of finned surfaces is also known to enhance nucleate boiling heat transfer and the critical heat flux.

Boiling heat transfer can also be enhanced by other techniques such as *mechanical agitation* and *surface vibration*. These techniques are not practical, however, because of the complications involved.

EXAMPLE 10–1 Nucleate Boiling of Water in a Pan

Water is to be boiled at atmospheric pressure in a mechanically polished stainless steel pan placed on top of a heating unit, as shown in Figure 10–15. The inner surface of the bottom of the pan is maintained at 108°C . If the diameter of the bottom of the pan is 30 cm, determine (a) the rate of heat transfer to the water and (b) the rate of evaporation of water.

**FIGURE 10-14**

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

SOLUTION Water is boiled at 1 atm pressure on a stainless steel surface. The rate of heat transfer to the water and the rate of evaporation of water are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat losses from the heater and the pan are negligible.

Properties The properties of water at the saturation temperature of 100°C are $\sigma = 0.0589$ N/m (Table 10-1) and, from Table A-9,

$$\begin{aligned}\rho_l &= 957.9 \text{ kg/m}^3 & h_{fg} &= 2257.0 \times 10^3 \text{ J/kg} \\ \rho_v &= 0.6 \text{ kg/m}^3 & \mu_l &= 0.282 \times 10^{-3} \text{ kg} \cdot \text{m/s} \\ \text{Pr}_l &= 1.75 & C_{pl} &= 4217 \text{ J/kg} \cdot ^\circ\text{C}\end{aligned}$$

Also, $C_{sf} = 0.0130$ and $n = 1.0$ for the boiling of water on a mechanically polished stainless steel surface (Table 10-3). Note that we expressed the properties in units specified under Eq. 10-2 in connection with their definitions in order to avoid unit manipulations.

Analysis (a) The excess temperature in this case is $\Delta T = T_s - T_{\text{sat}} = 108 - 100 = 8^\circ\text{C}$ which is relatively low (less than 30°C). Therefore, nucleate boiling will occur. The heat flux in this case can be determined from the Rohsenow relation to be

$$\begin{aligned}\dot{q}_{\text{nucleate}} &= \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma} \right]^{1/2} \left[\frac{C_{pl}(T_s - T_{\text{sat}})}{C_{sf} h_{fg} \text{Pr}_l^n} \right]^3 \\ &= (0.282 \times 10^{-3})(2257 \times 10^3) \left[\frac{9.81 \times (957.9 - 0.6)}{0.0589} \right]^{1/2} \\ &\quad \times \left(\frac{4217(108 - 100)}{0.0130(2257 \times 10^3)1.75} \right)^3 \\ &= 7.20 \times 10^4 \text{ W/m}^2\end{aligned}$$

The surface area of the bottom of the pan is

$$A = \pi D^2/4 = \pi(0.3 \text{ m})^2/4 = 0.07069 \text{ m}^2$$

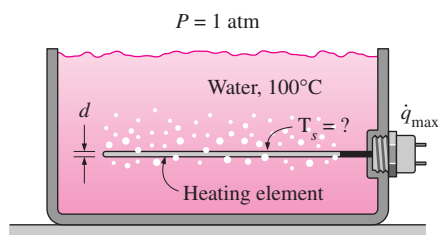


FIGURE 10-16
Schematic for Example 10-2.

Then the rate of heat transfer during nucleate boiling becomes

$$\dot{Q}_{\text{boiling}} = A\dot{q}_{\text{nucleate}} = (0.07069 \text{ m}^2)(7.20 \times 10^4 \text{ W/m}^2) = \mathbf{5093 \text{ W}}$$

(b) The rate of evaporation of water is determined from

$$\dot{m}_{\text{evaporation}} = \frac{\dot{Q}_{\text{boiling}}}{h_{fg}} = \frac{5093 \text{ J/s}}{2257 \times 10^3 \text{ J/kg}} = \mathbf{2.26 \times 10^{-3} \text{ kg/s}}$$

That is, water in the pan will boil at a rate of more than 2 grams per second.

EXAMPLE 10-2 Peak Heat Flux in Nucleate Boiling

Water in a tank is to be boiled at sea level by a 1-cm-diameter nickel plated steel heating element equipped with electrical resistance wires inside, as shown in Figure 10-16. Determine the maximum heat flux that can be attained in the nucleate boiling regime and the surface temperature of the heater surface in that case.

SOLUTION Water is boiled at 1 atm pressure on a nickel plated steel surface. The maximum (critical) heat flux and the surface temperature are to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat losses from the boiler are negligible.

Properties The properties of water at the saturation temperature of 100°C are $\sigma = 0.0589 \text{ N/m}$ (Table 10-1) and, from Table A-9,

$$\begin{aligned} \rho_l &= 957.9 \text{ kg/m}^3 & h_{fg} &= 2257 \times 10^3 \text{ J/kg} \\ \rho_v &= 0.6 \text{ kg/m}^3 & \mu_l &= 0.282 \times 10^{-3} \text{ kg} \cdot \text{m/s} \\ \text{Pr}_l &= 1.75 & C_{pl} &= 4217 \text{ J/kg} \cdot ^\circ\text{C} \end{aligned}$$

Also, $C_{sf} = 0.0060$ and $n = 1.0$ for the boiling of water on a nickel plated surface (Table 10-3). Note that we expressed the properties in units specified under Eqs. 10-2 and 10-3 in connection with their definitions in order to avoid unit manipulations.

Analysis The heating element in this case can be considered to be a short cylinder whose characteristic dimension is its radius. That is, $L = r = 0.005 \text{ m}$. The dimensionless parameter L^* and the constant C_{cr} are determined from Table 10-4 to be

$$L^* = L \left(\frac{g(\rho_l - \rho_v)}{\sigma} \right)^{1/2} = (0.005) \left(\frac{(9.81)(957.8 - 0.6)}{0.0589} \right)^{1/2} = 2.00 > 1.2$$

which corresponds to $C_{cr} = 0.12$.

Then the maximum or critical heat flux is determined from Eq. 10-3 to be

$$\begin{aligned} \dot{q}_{\text{max}} &= C_{cr} h_{fg} [\sigma g \rho_v^2 (\rho_l - \rho_v)]^{1/4} \\ &= 0.12(2257 \times 10^3) [0.0589 \times 9.8 \times (0.6)^2 (957.9 - 0.6)]^{1/4} \\ &= \mathbf{1.02 \times 10^6 \text{ W/m}^2} \end{aligned}$$

The Rohsenow relation, which gives the nucleate boiling heat flux for a specified surface temperature, can also be used to determine the surface temperature when the heat flux is given. Substituting the maximum heat flux into Eq. 10–2 together with other properties gives

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

$$1,017,200 = (0.282 \times 10^{-3})(2257 \times 10^3) \left[\frac{9.81(957.9 - 0.6)}{0.0589} \right]^{1/2}$$

$$\left[\frac{4217(T_s - 100)}{0.0130(2257 \times 10^3) 1.75} \right]$$

$$T_s = 119^\circ\text{C}$$

Discussion Note that heat fluxes on the order of 1 MW/m² can be obtained in nucleate boiling with a temperature difference of less than 20°C.

EXAMPLE 10–3 Film Boiling of Water on a Heating Element

Water is boiled at atmospheric pressure by a horizontal polished copper heating element of diameter $D = 5$ mm and emissivity $\varepsilon = 0.05$ immersed in water, as shown in Figure 10–17. If the surface temperature of the heating wire is 350°C, determine the rate of heat transfer from the wire to the water per unit length of the wire.

SOLUTION Water is boiled at 1 atm by a horizontal polished copper heating element. The rate of heat transfer to the water per unit length of the heater is to be determined.

Assumptions 1 Steady operating conditions exist. 2 Heat losses from the boiler are negligible.

Properties The properties of water at the saturation temperature of 100°C are $h_{fg} = 2257 \times 10^3$ J/kg and $\rho_l = 957.9$ kg/m³ (Table A-9). The properties of vapor at the film temperature of $T_f = (T_{\text{sat}} + T_s)/2 = (100 + 350)/2 = 225^\circ\text{C} = 498$ K (which is sufficiently close to 500 K) are, from Table A-16,

$$\begin{aligned} \rho_v &= 0.441 \text{ kg/m}^3 & C_{pv} &= 1977 \text{ J/kg} \cdot ^\circ\text{C} \\ \mu_v &= 1.73 \times 10^{-5} \text{ kg/m} \cdot \text{s} & k_v &= 0.0357 \text{ W/m} \cdot ^\circ\text{C} \end{aligned}$$

Note that we expressed the properties in units that will cancel each other in boiling heat transfer relations. Also note that we used vapor properties at 1 atm pressure from Table A-16 instead of the properties of saturated vapor from Table A-9 at 250°C since the latter are at the saturation pressure of 4.0 MPa.

Analysis The excess temperature in this case is $\Delta T = T_s - T_{\text{sat}} = 350 - 100 = 250^\circ\text{C}$, which is much larger than 30°C for water. Therefore, film boiling will occur. The film boiling heat flux in this case can be determined from Eq. 10–5 to be

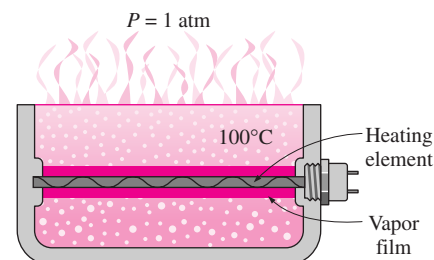


FIGURE 10–17
Schematic for Example 10–3.

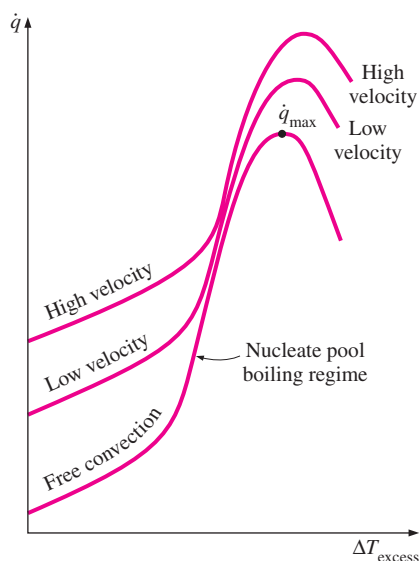


FIGURE 10-18

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

$$\begin{aligned} \dot{q}_{\text{film}} &= 0.62 \left[\frac{g k_v^3 \rho_v (\rho_l - \rho_v) [h_{fg} + 0.4 C_{pv} (T_s - T_{\text{sat}})]}{\mu_v D (T_s - T_{\text{sat}})} \right]^{1/4} (T_s - T_{\text{sat}}) \\ &= 0.62 \left[\frac{9.81 (0.0357)^3 (0.441) (957.9 - 0.441)}{[(2257 \times 10^3 + 0.4 \times 1977(250))]} \right]^{1/4} \times 250 \\ &= 5.93 \times 10^4 \text{ W/m}^2 \end{aligned}$$

The radiation heat flux is determined from Eq. 10-6 to be

$$\begin{aligned} \dot{q}_{\text{rad}} &= \varepsilon \sigma (T_s^4 - T_{\text{sat}}^4) \\ &= (0.05)(5.67 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4)[(250 + 273 \text{ K})^4 - (100 + 273 \text{ K})^4] \\ &= 157 \text{ W/m}^2 \end{aligned}$$

Note that heat transfer by radiation is negligible in this case because of the low emissivity of the surface and the relatively low surface temperature of the heating element. Then the total heat flux becomes (Eq. 10-7)

$$\dot{q}_{\text{total}} = \dot{q}_{\text{film}} + \frac{3}{4} \dot{q}_{\text{rad}} = 5.93 \times 10^4 + \frac{3}{4} \times 157 = 5.94 \times 10^4 \text{ W/m}^2$$

Finally, the rate of heat transfer from the heating element to the water is determined by multiplying the heat flux by the heat transfer surface area,

$$\begin{aligned} \dot{Q}_{\text{total}} &= A \dot{q}_{\text{total}} = (\pi D L) \dot{q}_{\text{total}} \\ &= (\pi \times 0.005 \text{ m} \times 1 \text{ m})(5.94 \times 10^4 \text{ W/m}^2) \\ &= \mathbf{933 \text{ W}} \end{aligned}$$

Discussion Note that the 5-mm-diameter copper heating element will consume about 1 kW of electric power per unit length in steady operation in the film boiling regime. This energy is transferred to the water through the vapor film that forms around the wire.

10-3 ■ FLOW BOILING

The *pool boiling* we considered so far involves a pool of seemingly motionless liquid, with vapor bubbles rising to the top as a result of buoyancy effects. 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. The flow boiling is also classified as either *external* and *internal flow boiling* depending on whether the fluid is forced to flow over a heated surface or inside a heated tube.

External flow boiling over a plate or cylinder is similar to pool boiling, but the added motion increases both the nucleate boiling heat flux and the critical heat flux considerably, as shown in Figure 10-18. Note that the higher the velocity, the higher the nucleate boiling heat flux and the critical heat flux. In experiments with water, critical heat flux values as high as 35 MW/m² have been obtained (compare this to the pool boiling value of 1.3 MW/m² at 1 atm pressure) by increasing the fluid velocity.

Internal flow boiling is much more complicated in nature because there is no free surface for the vapor to escape, and thus both the liquid and the vapor are forced to flow together. The two-phase flow in a tube exhibits different flow boiling regimes, depending on the relative amounts of the liquid and the vapor phases. This complicates the analysis even further.

The different stages encountered in flow boiling in a heated tube are illustrated in Figure 10–19 together with the variation of the heat transfer coefficient along the tube. Initially, the liquid is subcooled and heat transfer to the liquid is by *forced convection*. Then bubbles start forming on the inner surfaces of the tube, and the detached bubbles are drafted into the mainstream. This gives the fluid flow a bubbly appearance, and thus the name *bubbly flow regime*. As the fluid is heated further, the bubbles grow in size and eventually coalesce into slugs of vapor. Up to half of the volume in the tube in this *slug-flow regime* is occupied by vapor. After a while the core of the flow consists of vapor only, and the liquid is confined only in the annular space between the vapor core and the tube walls. This is the *annular-flow regime*, and very high heat transfer coefficients are realized in this regime. As the heating continues, the annular liquid layer gets thinner and thinner, and eventually dry spots start to appear on the inner surfaces of the tube. The appearance of dry spots is accompanied by a sharp decrease in the heat transfer coefficient. This *transition regime* continues until the inner surface of the tube is completely dry. Any liquid at this moment is in the form of droplets suspended in the vapor core, which resembles a mist, and we have a *mist-flow regime* until all the liquid droplets are vaporized. At the end of the mist-flow regime we have saturated vapor, which becomes superheated with any further heat transfer.

Note that the tube contains a liquid before the bubbly flow regime and a vapor after the mist-flow regime. Heat transfer in those two cases can be determined using the appropriate relations for single-phase convection heat transfer. Many correlations are proposed for the determination of heat transfer

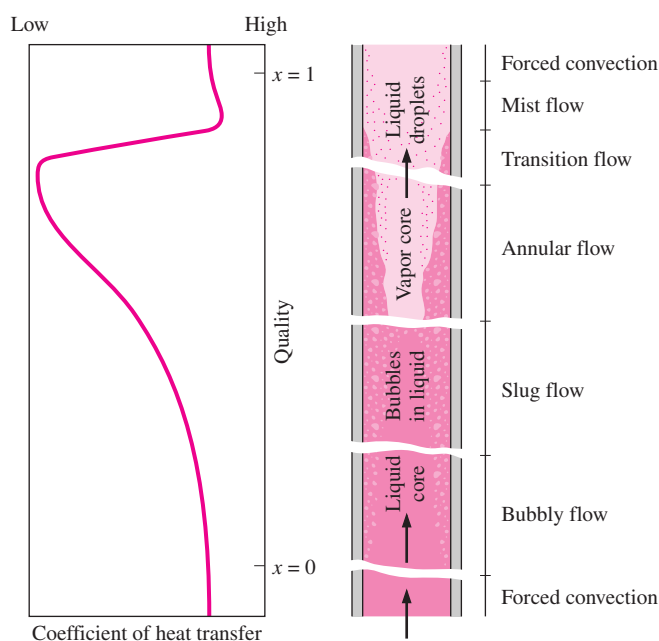


FIGURE 10–19
Different flow regimes encountered in flow boiling in a tube under forced convection.

in the two-phase flow (bubbly flow, slug-flow, annular-flow, and mist-flow) cases, but they are beyond the scope of this introductory text. A crude estimate for heat flux in flow boiling can be obtained by simply adding the forced convection and pool boiling heat fluxes.

10-4 ■ CONDENSATION HEAT TRANSFER

Condensation occurs when the temperature of a vapor is reduced *below* its saturation temperature T_{sat} . This is usually done by bringing the vapor into contact with a solid surface whose temperature T_s is *below* the saturation temperature T_{sat} of the vapor. But condensation can also occur on the free surface of a liquid or even in a gas when the temperature of the liquid or the gas to which the vapor is exposed is below T_{sat} . In the latter case, the liquid droplets suspended in the gas form a fog. In this chapter, we will consider condensation on solid surfaces only.

Two distinct forms of condensation are observed: *film condensation* and *dropwise condensation*. In **film condensation**, the condensate wets the surface and forms a liquid film on the surface that slides down under the influence of gravity. The thickness of the liquid film increases in the flow direction as more vapor condenses on the film. This is how condensation normally occurs in practice. In **dropwise condensation**, the condensed vapor forms droplets on the surface instead of a continuous film, and the surface is covered by countless droplets of varying diameters (Fig. 10–20).

In film condensation, the surface is blanketed by a liquid film of increasing thickness, and this “liquid wall” between solid surface and the vapor serves as a *resistance* to heat transfer. The heat of vaporization h_{fg} released as the vapor condenses must pass through this resistance before it can reach the solid surface and be transferred to the medium on the other side. In dropwise condensation, however, the droplets slide down when they reach a certain size, clearing the surface and exposing it to vapor. There is no liquid film in this case to resist heat transfer. As a result, heat transfer rates that are more than 10 times larger than those associated with film condensation can be achieved with dropwise condensation. Therefore, dropwise condensation is the preferred mode of condensation in heat transfer applications, and people have long tried to achieve sustained dropwise condensation by using various vapor additives and surface coatings. These attempts have not been very successful, however, since the dropwise condensation achieved did not last long and converted to film condensation after some time. Therefore, it is common practice to be conservative and assume film condensation in the design of heat transfer equipment.

10-5 ■ FILM CONDENSATION

We now consider film condensation on a vertical plate, as shown in Figure 10–21. The liquid film starts forming at the top of the plate and flows downward under the influence of gravity. The thickness of the film δ *increases* in the flow direction x because of continued condensation at the liquid–vapor interface. Heat in the amount h_{fg} (the latent heat of vaporization) is released during condensation and is *transferred* through the film to the plate surface at temperature T_s . Note that T_s must be below the saturation temperature T_{sat} of the vapor for condensation to occur.

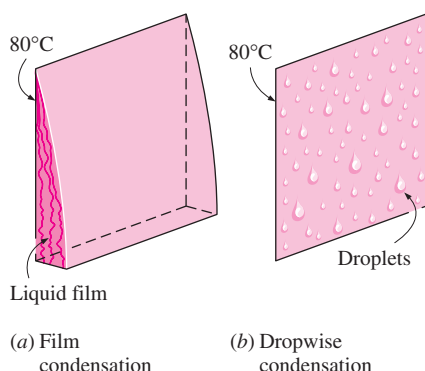


FIGURE 10-20

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.

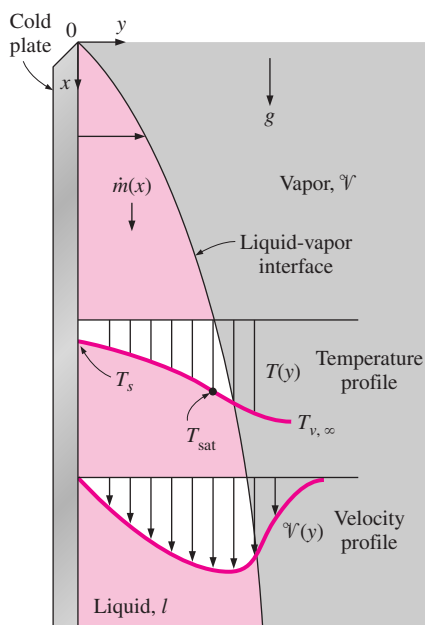


FIGURE 10-21

Film condensation on a vertical plate.