When a heated surface exceeds the saturation temperature of the surrounding coolant, boiling on the surface becomes possible. This is true whether the bulk fluid temperature is at or below the local saturation temperature. If the bulk fluid temperature is below the saturation temperature, boiling is referred to as "local" or "subcooled" boiling. If the bulk fluid temperature is equal to the saturation temperature, then "bulk" boiling is said to occur. Bubbles formed on the heated surface depart the surface and are transported by the bulk fluid, such that a condition of two-phase flow is said to exist. Depending on the degree of subcooling and the length of the heated channel, the bubbles may or may not condense and collapse prior to exiting the channel. In subcooled boiling this process results in further heating of the fluid toward the saturation temperature. In saturated or bulk boiling, bubbles can be transported along the entire length of the heated channel without collapsing.

Fundamental Concepts

In a discussion of two-phase flow, several fundamental quantities need defining:

**Phase Velocity**

\[ v_k = \text{velocity of phase } k, \text{ where } k \text{ may represent the liquid (} k = \ell \text{) or vapor (} k = g \text{) phase} \]

**Volume Fraction**

The volume fraction of phase \( k \) in a two-phase mixture is

\[ \alpha_k = \frac{V_k}{V} \]

where \( V_k \) is the volume occupied by phase \( k \), and \( V \) is the total two-phase volume. If the volume in question consists of the cross sectional area of a flow channel times a length segment \( \Delta z \), the volume fraction can be considered an area fraction

\[ \alpha_k = \frac{A_k}{A_x} \]

where \( A_x \) is the total cross section flow area. The vapor volume fraction \( \alpha_g \) is often referred to as the void fraction (\( \alpha \)) even though the vapor volume is filled with a low density gas and no true void exist.

Note: \( \alpha_v + \alpha_g = 1 \)

**Volumetric Flux** (Superficial Velocity)

The volumetric flux of phase \( k \) is defined to be the volumetric flow rate of phase \( k \) divided by the total flow area, i.e.

\[ f_k = \frac{v_k A_k}{A_x} = \alpha_k v_k \]

**Slip Ratio**

The slip ratio is defined as the ratio of the vapor and liquid phase velocities.

\[ S = \frac{v_g}{v_\ell} \]
If the liquid and vapor velocities are equal, then the slip ratio is one and the flow is said to be *Homogeneous*.

**Quality**

Three qualities are of particular interest in our analysis of two-phase systems, these include:

**Equilibrium Quality** \((x_e)\)

\[
x_e = \frac{h - h_f}{h_{fg}}
\]  

(5)

The equilibrium quality corresponds to the flow fraction of vapor only if thermodynamic equilibrium exists between the phases. Since the equilibrium quality is defined in terms of the fluid enthalpy, it can have values greater than one and less than zero. Under these conditions the equilibrium quality can be thought of as a measure of the degree of subcooling or superheat of the fluid, but can not be used to determine the fluid state.

**Flow Quality** \((x)\)

The flow quality is the true flow fraction of the vapor phase and is always between zero and one, regardless of the fluid's state. We define the flow quality as

\[
x = \frac{\dot{m}_g}{\dot{m}} = \frac{\dot{m}_g}{\dot{m}_g + \dot{m}_l}
\]  

(6)

The mass flow rate of the liquid phase is

\[
\dot{m}_l = \rho_l v_l A_x = \alpha \rho v_l A_x
\]  

(7)

Similarly for the vapor phase,

\[
\dot{m}_g = \rho_g v_g A_g = \alpha \rho v_g A_x
\]  

(8)

such that

\[
x = \frac{\dot{m}_g}{\dot{m}_g + \dot{m}_l} = \frac{(\alpha \rho v)_g}{(\alpha \rho v)_g + (\alpha \rho v)_l}
\]  

(9)

**Note**: The equilibrium quality has physical meaning only when between zero and one, even though values can be calculated outside of this range. The flow quality represents the true flow fraction of the vapor phase and can only have values between zero and one. Under saturation conditions, the equilibrium and flow quality are equivalent.

**Static Quality** \((x_s)\)

The static quality is the mass fraction of the vapor phase, and like the flow quality is always between zero and one. We define the static quality as
\[ x_s = \frac{M_g}{M} = \frac{M_g}{M_g + M_l} \quad (10) \]

The mass of the liquid phase is
\[ M_l = \rho_l V_l = \alpha_l \rho_l V \quad (11) \]

Similarly for the vapor phase,
\[ M_g = \rho_g V_g = \alpha_g \rho_g V \quad (12) \]

such that
\[ x_s = \frac{M_g}{M_g + M_l} = \frac{(\alpha \rho)_g}{(\alpha \rho)_g + (\alpha \rho)_l} \quad (13) \]

Note: The static quality and the flow quality are equivalent under Homogeneous Flow conditions.

**Void-Quality Relationships**

Mass flux is defined as the mass flow rate divided by the cross sectional flow area of the channel. In a two-phase system, the total system mass flow rate is the sum of the liquid and vapor mass flow rates such that
\[ G = \frac{\dot{m}_g + \dot{m}_l}{A_x} = (\alpha \rho)_g V + (\alpha \rho)_l V \quad (14) \]

From the definition of flow quality,
\[ Gx = \alpha_g \rho_g v_g = \alpha \rho_g v_g \quad (15) \]

and
\[ G(1-x) = \alpha_l \rho_l v_l = (1-\alpha) \rho_l v_l \quad (16) \]

Dividing these two equations and rearranging gives for the slip ratio
\[ S = \frac{x}{1-x} \frac{1-\alpha}{\alpha} \frac{\rho_l}{\rho_g} \quad (17) \]

or solving for void fraction
\[ \alpha = \frac{1}{1 + \frac{1-x}{x} \frac{\rho_g}{\rho_l} S} \quad (18) \]

Equation 18 is often referred to as the *Fundamental Void-Quality-Slip* relation.

In general, the quality and void fraction increase continuously along the channel, which implies the slip ratio also varies continuously along the channel.
The distribution of vapor in a boiling system affects both the heat transfer and the flow properties of the fluid. A number of flow patterns or “flow regimes” have been observed experimentally by viewing flow of liquid-vapor mixtures through transparent tubes. While the number and characteristics of specific flow regimes are somewhat subjective, four principal flow regimes are almost universally accepted. These patterns are illustrated in Figure 1 and include Bubbly Flow (a & b), Slug Flow (c), Churn or Churn-Turbulent Flow (d), and Annular Flow (e).

These flow regimes may be generally characterized as

**Bubbly Flow:** Individual dispersed bubbles transported in a continuous liquid phase.

**Slug Flow:** Large bullet shaped bubbles separated by liquid plugs.

**Churn Flow:** The vapor flows in a somewhat chaotic manner through the liquid, with the vapor generally concentrated in the center of the channel, and the liquid displaced toward the channel walls.

**Annular Flow:** The vapor forms a continuous core, with a liquid film flowing along the channel walls.

To predict the existence of a particular flow regime, or the transition from one flow regime to another, requires that the visually observed flow patterns be quantified in terms of measurable (or computed) quantities. This is normally accomplished through the use of *flow regime maps*. Two typical flow regime maps are given below. The Hewitt and Roberts map is valid for both air-water and steam-water systems. The Govier and Aziz map was obtained for air-water flows in 1 inch diameter tubes.

(From *Thermohydraulics of Two-Phase Systems for Industrial Design and Nuclear Engineering*, by J. M. Delhaye, M. Giot and M. L. Riethmuller)
Figure 2: Hewitt and Roberts Flow Regime Map (from Delhaye, Giot and Riethmuller)

Figure 3: Govier and Aziz Flow Regime Map for Air-Water (from Lahey and Moody)
Flow Boiling Regimes

A number of different heat transfer mechanisms are possible when a fluid is heated in a boiling channel. These different heat transfer mechanisms are often represented by use of a boiling curve. A boiling curve for low quality, high heat flux systems is illustrated below.

In Figure 4, the heat transfer rate is plotted as a function of the wall superheat (the difference between the wall temperature and the fluid saturation temperature). The curve is divided into 5 regions (a – e), each of which corresponds to a different heat transfer mechanism. These different heat transfer regions are discussed briefly below.

Region a: The minimum criteria for boiling, is that the temperature of the heated surface exceed the local saturation temperature, i.e. some degree of wall superheat is required for boiling to occur. In region a, wall superheat is insufficient to support bubble formation and growth. Heat transfer is by single-phase forced convection and is a strong function of fluid velocity (mass flux) and temperature.

Region b: Bubbles begin forming at nucleation sites on the heated surface. These nucleation sites are generally associated with pits or crevices on the heated surfaces in which non dissolved gases or vapor can accumulate allowing bubble formation. As the bubbles grow and depart the surface they carry latent heat, as well as generate increased turbulence and mixing which increases the heat transfer rate. Boiling under these conditions is referred to as nucleate boiling. In region b, heat transfer is a complicated mixture of single-phase forced convection and nucleate boiling. As a result, this region is often called the mixed boiling or partial nucleate boiling region. In general, as the wall temperature increases, the fraction of the wall surface subject to nucleate boiling increases, until bubble formation occupies the entire heated surface.

Region c: In region c, bubble density increases rapidly with increasing wall superheat. Heat transfer is dominated by local conditions in the vicinity of the wall generated by bubble growth and departure. These bubbles transport large amounts of latent heat from the surface at the fluid saturation temperature and greatly increase fluid turbulence and mixing in the vicinity of the wall. As a result, heat transfer becomes independent of bulk fluid conditions such as flow velocity and temperature. Heat transfer is said to be by fully developed nucleate boiling and is
characterized by substantial increases in heat transfer rate for moderate increases in wall
temperature.

The bubble density at the wall however can not be increased indefinitely. At point c*, the
bubble density becomes sufficiently high to impede liquid flow back to the surface. Bubbles
tend to coalesce, forming insulating vapor patches that reduce the heat transfer rate. Point c* is
called the point of Critical Heat Flux (CHF), and the critical heat flux mechanism is Departure
from Nucleate Boiling or DNB.

Region d: Further increases in wall superheat result in increasingly greater portions of the heated surface
covered by insulating vapor patches. The reduction in effective heat transfer area more than
compensates for the increase in wall temperature to reduce the overall heat transfer rate. This
region is referred to as the partial film or transition film boiling region.

Region e: A continuous vapor film completely blankets the heated surface. Heat transfer is by conduction
and convection through the superheated vapor layer with evaporation at the liquid/vapor
interface. Wall temperatures can become sufficiently high, such that radiative heat transfer
becomes important. This region is called the stable film boiling regime.

Steady state operation beyond the point of critical heat flux is only possible for wall temperature controlled
systems, where the heat input to the surface can be adjusted to maintain a given wall temperature. In
reactor systems, it is power and therefore heat flux which is controlled. In a heat flux controlled system, an
increase in the heat flux beyond the critical point results in Departure from Nucleate Boiling with an
associated increase in the wall temperature. This increase in the wall temperature causes more of the
heated surface to be blanketed by vapor, further increasing the wall temperature. The wall temperature
would then follow a transient path, “jumping” from c* to c’ where steady state operation at the new heat
flux would be possible, assuming the highly elevated wall temperatures were within the material limits of
the system. In reactors, these elevated wall temperatures can easily lead to fuel failure. Departure from
Nucleate Boiling is the dominant critical heat flux mechanism in Pressurized Water Reactors.

In low heat flux, high quality systems typical of Boiling Water Reactor operation, thermal-hydraulic
conditions within the core allow for the transition to annular flow. Vapor velocities and interfacial
turbulence are sufficiently high to suppress nucleation in the thin liquid film adjacent to the heated surface.
Heat is transferred by conduction and convection through the liquid film with evaporation at the
liquid/vapor interface. This heat transfer mechanism is referred to as forced convection vaporization and is
characterized by extremely high heat transfer coefficients. In fact, heat transfer coefficients can be so high,
that increased heat transfer rates can be achieved with decreasing wall temperatures. The boiling curve
under these conditions is illustrated in Figure 5.
Figure 5: Flow Boiling Curve for High Quality, Low Heat Flux Systems

Regions a - c are identical to those in Figure 4, with Region d the forced convection vaporization region. At point d*, the liquid film becomes so thin that dry patches can appear on the heated surface lowering the heat transfer rate. The critical heat flux under these conditions is not due to DNB but results from the complete evaporation or dryout of the liquid film flowing along the heated surface. The critical heat flux mechanism is then said to be dryout dominated. Further increases in wall temperature beyond the point of critical heat flux results in dryout of increasingly greater portions of the heated surface and a corresponding reduction in the heat transfer rate, eventually leading to complete evaporation of the liquid film. As in the DNB dominated system, steady state operation beyond the point of critical heat flux is only possible in wall temperature controlled systems.

Heat transfer in region f is due primarily to single-phase forced convection to a super heated vapor, combined with evaporation of entrained liquid droplets. Radiation can also become important at high wall temperatures. Temperature excursions following dryout are typically less severe than those following DNB, as some single-phase forced convection cooling is available from the vapor core. In reactor systems however, wall temperatures can still reach levels such that fuel failure is likely. Figure 6 illustrates the heat transfer and flow regimes which might be expected in a dryout controlled channel. The point at which critical heat flux is reached, regardless of the mechanism, has also been referred to as the boiling crisis.
Figure 6: Heat Transfer and Flow Regimes in a Boiling Channel (from Convective Boiling and Condensation, by J. G. Collier)
Heat Transfer in Boiling Channels

As illustrated in Figure 6, a number of different heat transfer regimes can occur simultaneously within a boiling channel. This is further illustrated in Figure 7 below.

Figure 7: Boiling Regimes in a Heated Channel

The position $z_n$ is called the Incipient Boiling Point or the Nucleation Point and is the position where the wall superheat is sufficient to support bubble growth. The position $z_B$ denotes the onset of fully developed nucleate boiling. Prior to boiling $z \in [0, z_n]$, heat transfer is by single phase forced convection. The heat flux is linear with wall temperature and given by Newton's Law of cooling

$$q^c(z) = h_c(T_{\text{wall}}(z) - T_{\text{w}}(z))$$  \hspace{1cm} (19)

where the convective heat transfer coefficient $h_c$ is generally correlated in the form

$$\text{Nu} = C \text{Re}^{0.8} \text{Pr}^m.$$ \hspace{1cm} (20)

Correlations of this type include the Dittus-Boelter Correlation for flow in conduits and annuli, and the Weisman correlation for flow parallel to rod bundles.

At the onset of boiling, the heat flux becomes nonlinear with respect to wall temperature. In the fully developed nucleate boiling regime ($z \geq z_B$), heat flux is usually correlated in the form
\[ \frac{q''(z)}{10^6} = \zeta \left( T_{co}(z) - T_{sat} \right)^m \]  
(21)

where:
\[ q''(z) \Rightarrow \text{Btu/hr-ft}^2 \]
\[ T \Rightarrow F \]

or solving for the clad surface temperature

\[ T_{co}(z) = T_{sat} + \zeta^{-1/m} \left( \frac{q''(z)}{10^6} \right)^{1/m} \]  
(22)

Two popular correlations of this type valid for both local and bulk boiling conditions in water are the Jens-Lottes correlation

\[ \zeta = \frac{\exp(4P/900)}{60^4} \]
\[ m = 4 \]

and the Thom correlation

\[ \zeta = \frac{\exp(2P/1260)}{72^2} \]
\[ m = 2 \]

where \( P \) is pressure in psia. Assuming the Jens-Lottes correlation, the outer clad temperature is

\[ T_{co}(z) = T_{sat} + 1.897 \text{Btu/hr-ft}^2 \exp(-P/900) \]  
(23)

A number of models exist for computing the wall temperature in the mixed boiling region \( z \in [z_n, z_B] \). One such model for subcooled boiling is that by Bergles and Rohsenow

\[ q^*(z) = q_{FC}^*(z) \left[ 1 + \left( \frac{q_{NB}^*(z)}{q_{FC}^*(z)} \right) \left( 1 - \frac{q_{NB}^*(z_n)}{q_{NB}^*(z)} \right) \right]^{1/2} \]  
(24)

where:
\[ q_{FC}^*(z) = h_c[T_{co}(z) - T_{w}(z)] \] is the heat flux associated with single phase forced convection
\[ q_{NB}^*(z) = \zeta \times 10^6 [T_{co}(z) - T_{sat}]^m \] is the heat flux associated with fully developed nucleate boiling

The Incipient Boiling Point is generally correlated in terms of a critical wall superheat. A commonly used correlation of this type is

\[ q''(z_n) = 15.6P^{1.156}[T_{co}(z_n) - T_{sat}]^{2.30/P^{0.6234}} \]  
(25)

where \( q^*(z_n) \) is the local heat flux at the Incipient Boiling Point (Btu/hr-ft²) and \( P \) is pressure (psia).

In order to apply Equations 19, 22 and 24 to an arbitrarily heated channel where both single-phase forced convection and nucleate boiling may occur, the transition points from single-phase forced convection to mixed boiling and from mixed boiling to fully developed nucleate boiling must be determined. These transition points are usually taken to
insure the wall temperature is continuous at the transition point. The following procedure can be used to determine the transition points and compute wall temperature in a boiling channel.

In the single phase forced convection region, the wall temperature at any point is given by

\[ T_{\text{co}}(z) = T_{\infty}(z) + \frac{q''(z)}{h_c} \]  

(26)

where \( h_c \) is the single-phase forced convection heat transfer coefficient and \( T_{\infty}(z) \) is the local fluid temperature. In a single channel with no mixing, the local fluid temperature can be obtained from the simple energy balance

\[ T_{\infty}(z) = T_{\infty}(0) + \frac{1}{mC_p} \int_0^z q''(z') \pi D dz' \]  

(27)

At the Incipient Boiling Point, we require the wall temperature in Equation 25 to satisfy Equation 26. The Incipient Boiling Point is then the solution of

\[ q''(z_n) = 15.6 P^{1.156} \left[ T_{\infty}(z_n) + \frac{q''(z_n)}{h_c} - T_{\text{sat}} \right]^{2.30/\mu^{0.0234}} \]  

(28)

where

\[ T_{\infty}(z_n) = T_{\infty}(0) + \frac{1}{mC_p} \int_0^{z_n} q''(z') \pi D dz' \]  

(29)

such that for a given mass flow rate, inlet conditions and heat flux profile, the only unknown in Equation 28 is \( z_n \). In general, Equation 28 is transcendental and must be solved iteratively. Once \( z_n \) is known,

\[ q''_{\text{NB}}(z_n) = \xi \times 10^6 [T_{\text{co}}(z_n) - T_{\text{sat}}]^{\mu} \]  

(30)

where

\[ T_{\text{co}}(z_n) = T_{\infty}(z_n) + \frac{q''(z_n)}{h_c} \]  

(31)

such that \( q''_{\text{NB}}(z_n) \) is a constant. The wall temperature at any location in the mixed boiling region is the solution of Equation 24,

\[ q''(z) = q''_{\text{FC}}(z) \left[ 1 + \left[ \frac{q''_{\text{NB}}(z)}{q''_{\text{FC}}(z)} \left( 1 - \frac{q''_{\text{NB}}(z_n)}{q''_{\text{NB}}(z)} \right) \right]^{1/2} \right]^{1/2} \]

where the only unknown at any location is \( T_{\text{co}}(z) \).
At the transition point between mixed and fully developed nucleate boiling, the wall temperature satisfies
\[ q^*(z_B) = \zeta \times 10^6[T_{co}(z_B) - T_{sat}]^m = q^*_{NB}(z_B) \]
such that at the transition point, Equation 24 reduces to
\[ q^*(z_B) = q_{FC}^*(z_B) \left[ 1 + \left( \frac{q^*(z_B)}{q_{FC}^*(z_B)} \left( 1 - \frac{q_{NB}^*(z_B)}{q^*(z_B)} \right) \right)^2 \right]^{1/2} \] (32)
where
\[ q_{FC}^*(z_B) = h_c[T_{co}(z_B) - T_{sat}] \] (33)
\[ T_{co}(z_B) = T_{sat} + \zeta^{-\frac{1}{m}} \left( \frac{q^*(z_B)}{10^6} \right)^\frac{1}{m} \] (34)
and
\[ T_{\infty}(z_B) = T_{\infty}(0) + \frac{1}{mC_p} \int_0^{z_B} q^*(z') \pi D dz' \] (35)

For a given heat flux profile and channel operating conditions, Equations 32, 33, 34 and 35 can be reduced to a single nonlinear equation in the boiling transition point \( z_B \).

No adequate criteria has been established to determine the transition from nucleate boiling to forced convection vaporization. However, a single correlation that is valid for both nucleate boiling and forced convection vaporization has been developed by Chen for saturated boiling conditions and extended to include subcooled boiling by others. According to the Chen correlation, the heat flux can be related to the wall (or clad) temperature by
\[ q''(z) = h_{lo}[T_w(z) - T_{\infty}(z)] + h_{2\theta}[T_w(z) - T_{sat}] \] (36)
where \( h_{lo} \) is a “liquid only” convective heat transfer coefficient and \( h_{2\theta} \) is a nucleate boiling heat transfer coefficient. The “liquid only” convective heat transfer coefficient is similar to the Dittus-Boelter correlation where the Reynolds number is computed based on the liquid mass flux
\[ h_{lo} = 0.023 \left[ \frac{G(1-x)D_e}{\mu} \right]^{0.8} \left( \frac{C_p\mu}{k} \right)^{0.4} \left( \frac{k}{D_e} \right)^{0.4} \] (37)
The Reynolds number factor \( F \) is an experimentally determined correction factor and is defined to be the ratio of the true two-phase Reynolds number to the single-phase, liquid only Reynolds number, i.e.
\[ F = \left[ \frac{Re_2}{Re_{lo}} \right]^{0.8} : \ Re_{lo} = \frac{G(1-x)D_e}{\mu} \]
The Reynolds number factor is illustrated in Figure 8 below and can be expressed analytically in terms of the turbulent Martinelli Parameter \( \chi_B \) as
\[ F = \begin{cases} 
1.0 & \frac{1}{X_n} \leq 0.10 \\
2.35 \left( \frac{1}{X_n} + 0.213 \right)^{0.736} & \frac{1}{X_n} > 0.10
\end{cases} \] \quad (38)

where

\[ \frac{1}{X_n} = \left( \frac{x}{1-x} \right)^{0.9} \left( \frac{\rho_f}{\rho_g} \right)^{0.5} \left( \frac{\mu_g}{\mu_f} \right)^{0.1} \] \quad (39)

The nucleate boiling coefficient has the form

\[ h_{2d} = 0.00122 \left[ \frac{h_f^{0.79} \mu_f^{0.45} \rho_f^{0.49} \eta_f^{0.25}}{\sigma^{0.5} \mu_f^{0.29} \mu_g^{0.24} \rho_g^{0.24} \rho_f \mu_f} \right]^{0.75} \left( \frac{h_{fg} F}{T_w \nu_{fg}} \right) \left( T_w - T_{sat} \right)^{0.99} S \] \quad (40)

and is in terms of an experimentally determined nucleate boiling suppression factor \( S \). The suppression factor is a measure of the true super heat in the liquid film and is defined as

\[ S = \left[ \frac{\Delta T_{ave}}{T_w - T_{sat}} \right]^{0.99} \] \quad (41)

The suppression factor is illustrated in Figure 9 and can be expressed analytically as

\[ S = 0.9622 - 0.5822 \times \tan^{-1} \left( \frac{\text{Re}_{2d}}{6.18 \times 10^4} \right) \] \quad (42)
Figure 8: Reynolds Number Factor

Figure 9: Suppression Factor
Example:

The hot channel in a PWR operates over a significant fraction of its length under nucleate boiling. Assuming that boiling occurs at the position of maximum heat flux, determine the clad temperature and the boiling heat transfer coefficient at this point.

**Problem Data**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
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<tr>
<td>Core Averaged Heat Flux</td>
<td>189,800 Btu/hr-ft²</td>
</tr>
<tr>
<td>Power Peaking Factor</td>
<td>2.5</td>
</tr>
<tr>
<td>System Pressure</td>
<td>2250 psia</td>
</tr>
</tbody>
</table>

**Clad Temperature**

Assuming the Jens-Lottes correlation for the boiling heat transfer coefficient, the clad temperature at any point along the boiling length is

$$ T_{co}(z) = T_{sat} + 1.897q''(z)^{1/4} \exp(-P/900). $$

If we denote the maximum heat flux by \( q''_0 \), then the clad temperature at this point is given by

$$ T_{co} = T_{sat} + 1.897q''_0^{1/4} \exp(-P/900) $$

At a system pressure of 2250 psia, the fluid saturation temperature is 652.67 F. The maximum heat flux in the channel is obtained from

\[
q''_0 = q''_{max} = Fq\overline{q''} = (2.5)(189,800) = 474,500 \text{ Btu/ft}^2\text{Hr} \\
T_{co} = 652.67 + 1.897(474,500)^{1/4} \exp(-2250/900) \\
= 656.8 F
\]

**Boiling Heat Transfer Coefficient**

Writing the heat flux in terms of Newton's Law of Cooling,

\[
q''(z) = h_c(T_{co}(z) - T_{sat}) \Rightarrow h_c = \frac{q''(z)}{T_{co}(z) - T_{sat}}
\]

At the position of maximum heat flux, the boiling heat transfer coefficient is then

\[
h_c = \frac{474,500}{656.8 - 652.67} = 114,891 \text{ Btu/ft}^2\text{Hr} - F
\]

Note the magnitude of the boiling heat transfer coefficient relative to a convective heat transfer coefficient.