

14. Heat Pipes

Gaugler proposed the principle of heat pipe in 1942 for refrigeration engineering applications. However, his principle was not effectively applied and was not widely known until 1963, when Grover and his colleagues developed heat pipes for space applications and described the concept with the name of “Heat Pipe”. Since then, heat pipe has been undergone considerable research and development for a wide range of applications, including the cooling of electronic systems.

- Definition

Heat pipe is a high performance heat transmission device with an extremely high thermal conductivity. It utilizes latent heat of evaporation of the working fluid to transmit heat from one point to another along the length of the heat pipe with a very small temperature gradient. Due to the heat pipe technique depends on two phase flow heat transfer, so that we should devote apart to the concept of Boiling and Condensation heat transfer.

- Boiling and Condensation Heat Transfer

Boiling and condensation heat transfer in two-phase flow (Liquid-vapor phase) is important in many engineering processes and systems. Due to increased heat transfer rates compared to single-phase heat transfer.

The virtually isothermal heat transfer associated with boiling and condensation processes makes their inclusion in power and refrigeration processes highly advantageous from a thermodynamic efficiency point of view.

The convection coefficient for both boiling and condensation could depend on the difference between the surface and saturation temperatures, $\Delta T = T_s - T_{sat}$, the body force arising from the liquid – vapor density difference, $(\rho_L - \rho_v)g$, the surface tension σ , the latent heat h_{fg} , characteristic length, and the thermophysical properties of the liquid or vapor (ρ, cp, μ, k), so that.

$$h = h\{\Delta T, (\rho_L - \rho_v)g, \sigma, h_{fg}, L, \rho, c_p, \mu, k\} \quad (14.1)$$

Applications of this type of heat transfer include boiling and condensation heat transfer to cool electronic components in computers such as in heat pipe method which mainly depend on both phenomena as shown in Figure14.1, the use of compact evaporators and condensers for thermal control of aircraft avionics and spacecraft environments and immersion cooling for electronics.

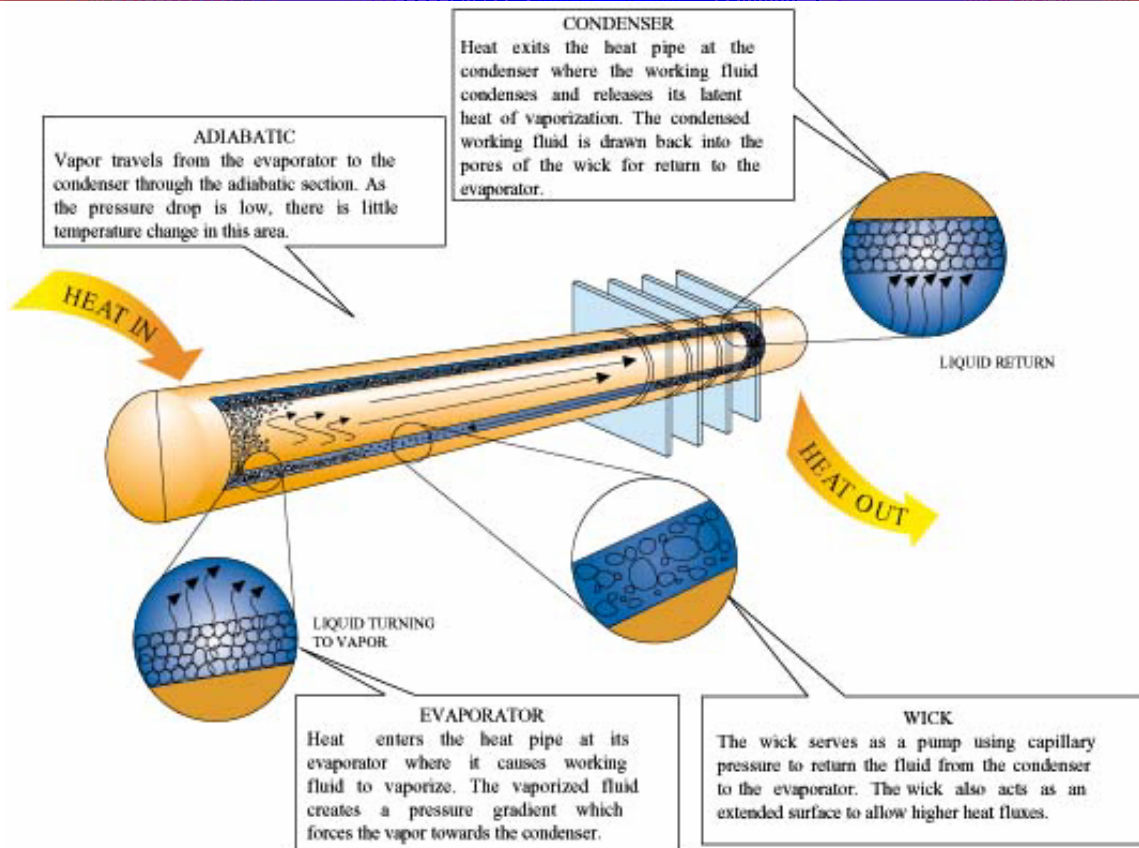


Figure 14.1 Heat pipe configuration in horizontal position

14.1 Boiling Heat Transfer

When a surface maintained at a temperature above the saturation temperature of the liquid is exposed to a liquid as shown in Figure 14.2. This liquid starts to evaporate; this phenomenon is known as boiling. Heat transferred from the solid surface to the liquid is

$$q'' = \bar{h}(T_s - T_{sat}) = \bar{h}\Delta T \quad (14.2)$$

The total evaporation rate may be then determined from the relation

$$m\dot{=} = \frac{q}{h_{fg}} \quad (14.3)$$

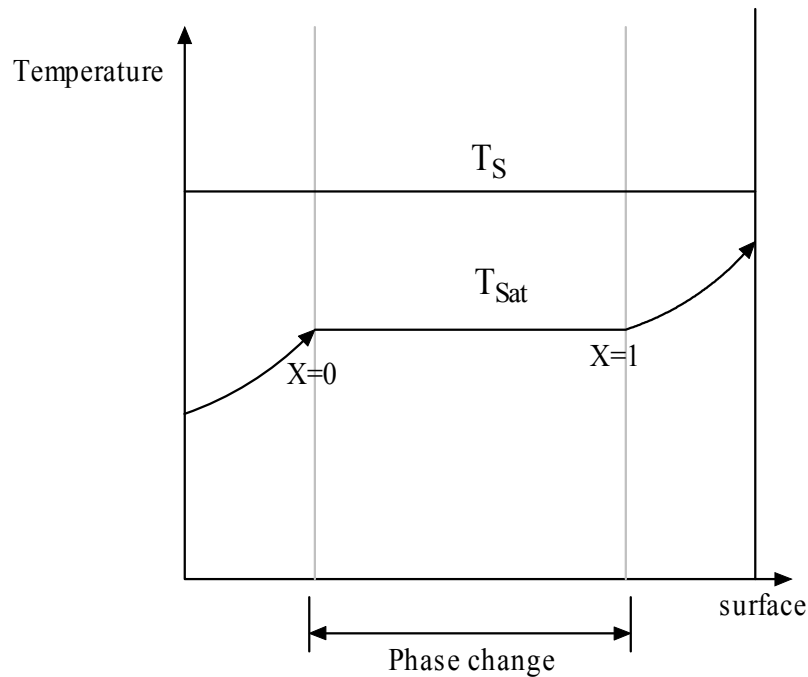


Figure 14.2 Surface –Temperature diagram for boiling heat transfer

The boiling may be divided into pool or flow boiling. If the heated surface is submerged below a free surface of liquid, this process is referred to as pool boiling. While if the fluid is moving by external means through duct or external flow over heated surfaces, this process is referred to as flow boiling. Also the boiling may be classified into subcooled or saturated boiling. In subcooled boiling, the temperature of the liquid is below the saturation temperature and the bubbles formed at the surface are condensed back when they travel up through the liquid (no liquid evaporation). While in saturated boiling the temperature of the liquid nearly equal the saturation temperature and the bubbles formed at the surface are then propelled by buoyancy forces, eventually escaping from a free surface(liquid is evaporated). In the foregoing lecture, we will consider saturated boiling only due to its importance.

14.1.1 Pool Boiling

The nature of the pool boiling process varies considerably depending on the conditions at which boiling occurs. The level of heat flux, the thermophysical properties of the liquid and vapor, the surface material and finish, and the physical size of the heated surface all may have an effect on the boiling process. The regimes of pool boiling are most easily understood by the boiling curve.

14.1.1.1 Pool Boiling Curve

In 1943, Nukiyama was the first to identify different regimes of pool boiling. He boiled saturated water on a horizontal wire. Then drew the heat flux versus temperature difference developing the so called pool boiling curve as shown in Figure14.3,

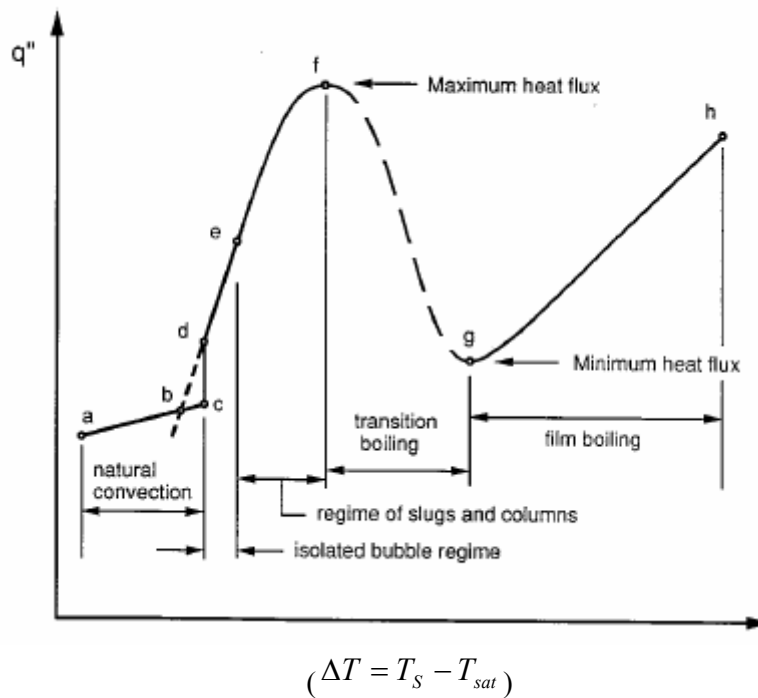


Figure14.3 Pool boiling curve: heat flux versus Temperature difference

14.1.1.2 Modes of Pool Boiling

The boiling curve in Figure14.3.has been divided into five regimes. Different boiling regimes are:

- Natural convection boiling:

When the surface temperature is a few degrees above the saturation temperature (T_{sat}), no nucleation sites may be active and heat may be transferred from the surface to the ambient liquid by natural convection alone as shown in Figure14.4. And q'' increases slowly with $T_s - T_{sat}$.

- Nucleate boiling:

The superheat becomes large enough to start nucleation at some of the cavities on the surface. This onset of nucleate boiling (ONB) condition occurs at point c. Once nucleate boiling is started as shown in Figure14.4. Any further increase in surface temperature causes the system operating point to move upward along section d - f of the boiling curve. This portion of the curve corresponds to the nucleate boiling regime. The nucleate boiling regime is divided into two distinct regimes namely:

1. Isolated bubble regime: it is the segment d-e of the boiling curve, where the flow becomes bubbly.

2. Slugs and columns regime: it is the segment $e-f$ of the curve, where vapor is being produced so rapidly that bubbles merging together form columns of vapor slugs that rise upward in the liquid pool toward its free surface.

A peak heat flux is reached if the surface temperature within the nucleate boiling regime is increased to a certain value. The peak value of heat flux is called the critical heat flux (CHF), designated as point f in the pool boiling curve.

- Transition boiling:

It is the segment $f-g$ of the boiling curve, where bubble formation is now so rapid that a vapor film or blanket begins to form on the surface as shown in Figure14.4. At any point on the surface, conditions may oscillate between film and nucleate pool boiling.

- Film boiling:

It start from point g this point also known as the Leidenfrost point (minimum heat flux); where the surface is hot enough to sustain a stable vapor film on the surface for an indefinite period of time. The entire surface then becomes blanketed with a vapor film as shown in Figure14.4. Within the film boiling regime; the heat flux increases as the superheat increases. This trend is a consequence of the increased conduction and/or convection transport due to the increased driving temperature difference across the vapor film. Radiation transport across the vapor layer may also become important at higher wall temperatures.

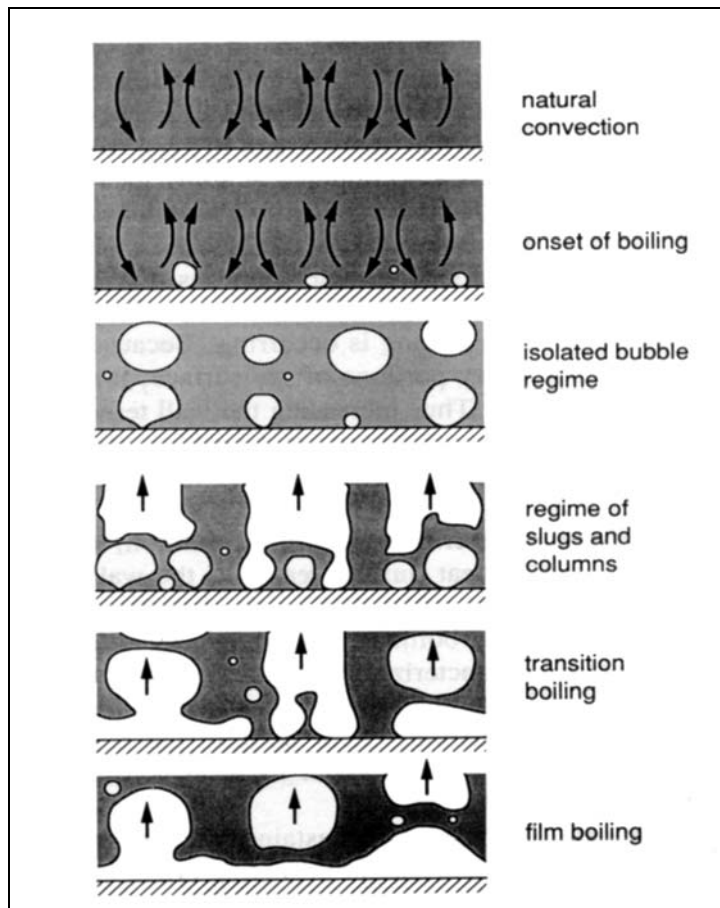


Figure14.4 Pool boiling regimes

14.1.1.3 Correlations of Pool Boiling

- nucleate pool boiling:

Commonly used correlation for nucleate boiling heat transfer developed by Rohsenow (1962) is

$$q'' = \mu_L h_{fg} \left[\frac{g(\rho_L - \rho_v)}{\sigma} \right]^{1/2} \left[\frac{c_{p,L} \Delta T}{C_{sf} h_{fg} Pr_L^s} \right]^3 \quad (14.4)$$

All properties are evaluated at saturated temperature (T_{sat})

Where the exponent $s = 1$ for water

$= 1.7$ for other liquids

And the values of C_{sf} in this correlation vary with the type of solid surface and the fluid type in the system and these values are shown in table 14.1.

Table14.1 Selected values for C_{sf} for use with Equation14.4

Fluid-Surfaces combination	C_{sf}
Water-copper	
Scored	0.0068
Polished	0.013
Water-stainless steel	
Chemically etched	0.013
Mechanically polished	0.013
Ground and polished	0.006
Water-brass	0.006
Water-nickel	0.006
Water-platinum	0.013
n-Pentane-copper	
Polished	0.0154
Lapped	0.0049
Benzene-chromium	0.0101
Ethyl alcohol- chromium	0.0027

- Critical heat flux q''_{max} :

According to Zuber (1959) for flat horizontal surfaces, the predicted maximum heat flux q''_{max} is

$$q''_{max} = \frac{\pi}{24} h_{fg} \rho_v \left[\frac{\sigma g(\rho_L - \rho_v)}{\rho_v^2} \right]^{1/4} \left[\frac{\rho_L + \rho_v}{\rho_L} \right]^{1/2} \quad (14.5)$$

All properties are evaluated at saturated temperature (T_{sat})

Other geometries are treated by Lienhard et al. (1973) and Lienhard and Dhir (1973b). An alternative model has been proposed by Haramura and Katto (1983).

- Minimum heat flux q''_{min} :

The transition regime is of small practical interest, as it may be obtained only by controlling the surface heater temperature so that no suitable theory has been obtained for this regime.

The minimum heat flux corresponds approximately to the lowest heat flux which will sustain stable film boiling. The following relation for the minimum heat flux q''_{min} derived by Zuber (1959) and Berenson (1961), from a large horizontal heated surface plate

$$q_{\min}'' = 0.09 \rho_v h_{fg} \left[\frac{g \sigma (\rho_L - \rho_v)}{(\rho_L + \rho_v)^2} \right]^{1/4} \quad (14.6)$$

Properties of vapor are evaluated at $T_f = (T_{sat} + T_s)/2$, while ρ_L and h_{fg} at T_{sat}

This correlation is accurate to approximately 50% for most fluids at moderate pressures but poorer at higher pressures. Similar result has been obtained for horizontal cylinders.

- Film pool boiling:

In film boiling, transport of heat across the vapor film from the wall to the interface may occur by boiling and radiation heat transfer. So that the total heat transfer coefficient may be calculated from the empirical relation

$$h = h_b \left(\frac{h_b}{h} \right)^{1/3} + h_r \quad (14.7)$$

Where Bromley, suggests the following relation for calculation of heat transfer coefficient in the stable film boiling region h_b on horizontal tube is

$$h_b = 0.62 \left[\frac{k_v^3 \rho_v g (\rho_L - \rho_v) (h_{fg} + 0.4 c_{pv} \Delta T)}{d \mu_v \Delta T} \right]^{1/4} \quad (14.8)$$

All properties evaluated as for the minimum heat flux calculation.

And the radiation heat transfer h_r is expressed as:

$$h_r = \frac{q_{rad}''}{T_s - T_{sat}} = \frac{5.67 \times 10^{-8} \varepsilon (T_s^4 - T_{sat}^4)}{T_s - T_{sat}} \quad (14.9)$$

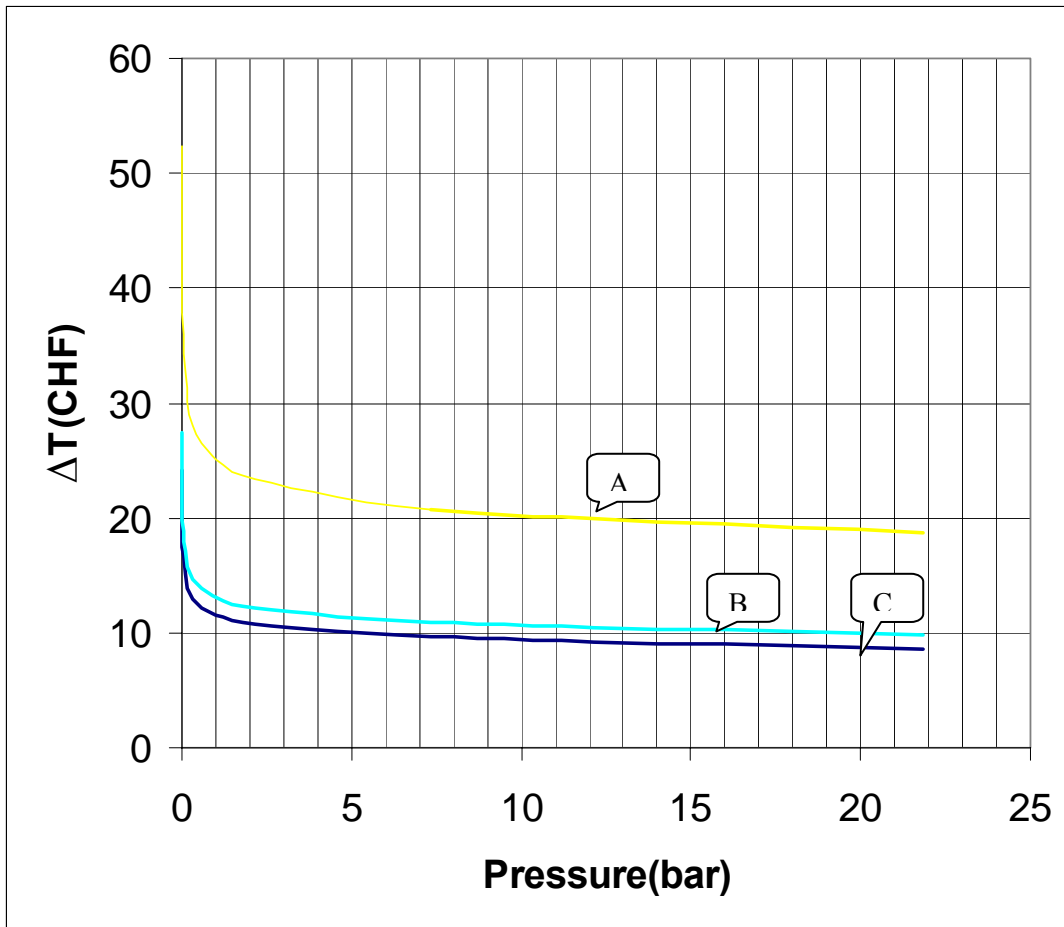
Where: ε is the emissivity of the solid surface.

Note that at low surface temperatures, radiation effects are negligible. At higher temperatures radiation effects must also be included. If the working fluid is water the radiation effect becomes significant at elevated surface temperature $T_s \geq 250^\circ\text{C}$ (arbitrary).

Table 14.2 Approximate temperature difference for different regimes in water boiling curve

Pool boiling regime	$\Delta T = T_s - T_{sat}$
Natural convection boiling	$0^\circ\text{C} < \Delta T \leq 5^\circ\text{C}$
Nucleate boiling	$5^\circ\text{C} \leq \Delta T \leq \Delta T_{CHF}$
Transition boiling	$\Delta T_{CHF} \leq \Delta T \leq 120^\circ\text{C}$
Film boiling	$120^\circ\text{C} \leq \Delta T$

The critical temperature difference ΔT_{CHF} for various surface metals with water as working fluid at different pressures is shown in Figure 14.5.



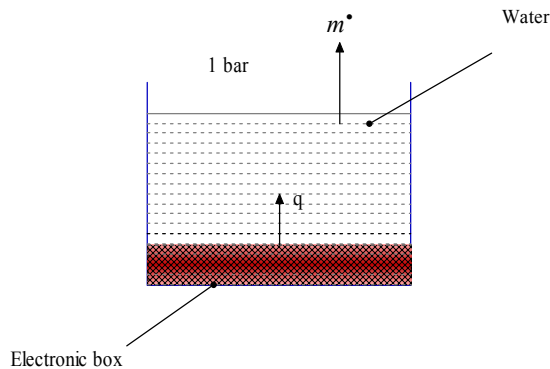
A	Copper polished-water
B	Copper scored-water
C	Brass-water

Figure 14.5 Pressure versus critical temperature difference ΔT_{CHF} for water

Example 14.1: An electronic box 0.02 m^2 upper surface area is immersed in bottom of a copper pan which contains water. The surface temperature of the box is $105 \text{ }^\circ\text{C}$ under steady state boiling condition. Estimate

- 1- The boiling heat transfer to boil water in this pan.
- 2- evaporation rate

Schematic:



Assumptions

- 1- Steady state conditions
- 2- Water at uniform temperature $T_{sat}=100$ °C due to the water exposed to atmospheric pressure
- 3- Pan surface of copper Polished
- 4- Negligible losses from heater to surrounding

Solution

$$\Delta T = T_s - T_{sat} = 105 - 100 = 5 \text{ }^\circ\text{C}$$

According to the boiling curve of Figure 14.5: For water-copper surface at nearly 1 bar it shows that $\Delta T_{CHF} = 25$ °C which is greater the temperature difference in our problem, thus nucleate boiling will occur.

Recall Equation 14.4

$$q'' = \mu_L h_{fg} \left[\frac{g(\rho_L - \rho_v)}{\sigma} \right]^{1/2} \left[\frac{c_{p,L} \Delta T}{C_{sf} h_{fg} Pr_L^s} \right]^3$$

$s=1$ for water, $C_{sf}=0.013$ for surface of copper Polished and The properties of the water are evaluated at $T_{sat} = 100$ °C.

The boiling heat flux is:

$$q'' = 279 \times 10^{-6} \times 2557 \times 10^3 \left[\frac{9.8(957.3)}{58.9 \times 10^{-3}} \right]^{1/2} \left[\frac{4.217 \times 10^3 \times 5}{0.013 \times 2557 \times 10^3 \times 1.76} \right]^3 = 17.1 \text{ kW/m}^2$$

The boiling heat transfer is

$$q = q'' \times A = 17.1 \times 10^3 \times 0.02 = 342 \text{ W}$$

Under steady state conditions the evaporation rate is:

$$m \cdot = \frac{q}{h_{fg}} = \frac{342}{2557 \times 10^3} = 0.1337 \times 10^{-3} \text{ kg/s}$$

14.1.2 Flow Boiling

Flow boiling may be experienced for either external or internal flows. Flow boiling in tubes is the most complex convective process encountered in applications such as evaporators; the flow is either horizontal or vertically upward, bubble growth and separation are strongly influenced by the flow velocity. Figure 14.6 schematically depicts a typical low-flux vaporization process in a horizontal tube. In this example liquid enters as subcooled liquid and leaves as superheated vapor. As indicated in Figure 14.6, the flow undergoes transitions in the boiling regime and the two-phase flow regime as it proceeds down the tubes. The regimes encountered depend on the entrance conditions and the thermal boundary conditions at the tube wall.

At low quality(x) the vaporization process is dominated by nucleate boiling, with convective effects being relatively weak. As the quality increases, the flow quickly enters the annular film flow regime in

which convective evaporation of the annular liquid film is the dominant heat transfer mechanism. Often the conditions are such that liquid droplets are often entrained in the core vapor flow during annular flow evaporation. Eventually, the annular film evaporates away, leaving the wall dry. Mist-flow evaporation of entrained liquid droplets continues in the post-dry out regime until only vapor remains.

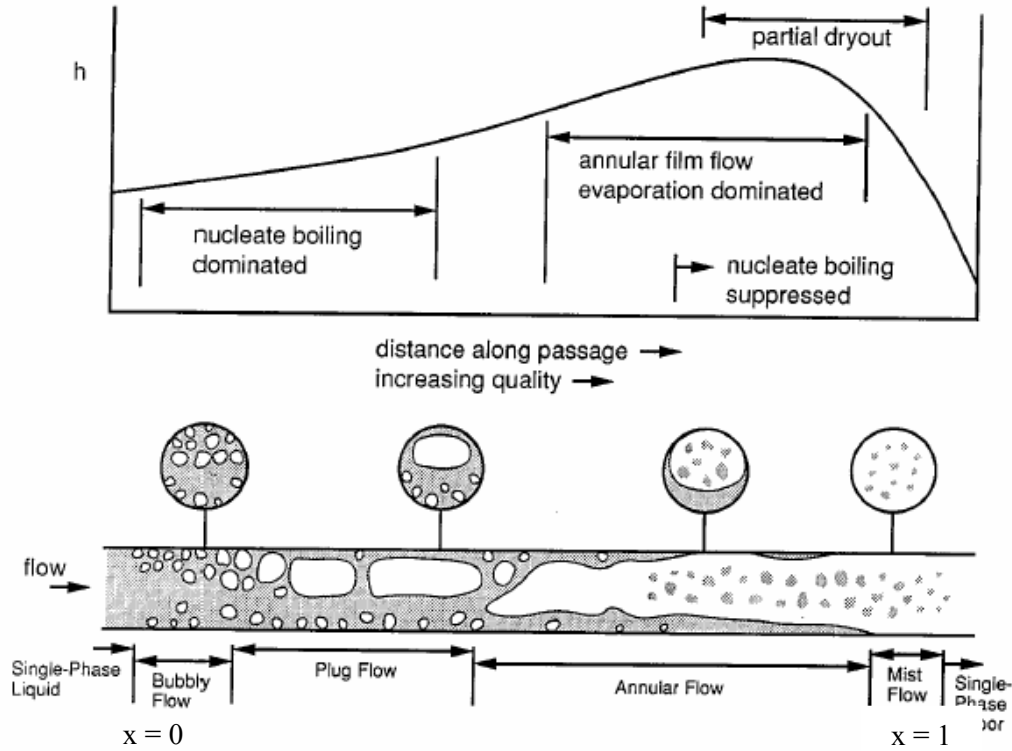


Figure 14.6 Qualitative variation of the heat transfer coefficient h and flow regime with quality for internal convective boiling in a horizontal tube at moderate wall superheat

The heat flux in the subcooled region is due to both forced convection and boiling heat transfer mechanisms, hence:

$$q''_{total} = q''_{boiling} + q''_{forced\ convection} \quad (14.10)$$

Where

$$q''_{boiling} = \mu_L h_{fg} \left[\frac{g(\rho_L - \rho_v)}{\sigma} \right]^{1/2} \left[\frac{c_{p,L} \Delta T}{C_{sf} h_{fg} Pr_L^s} \right]^3 \quad (14.10\ a)$$

And

$$q''_{forced\ convection} = 0.019 Re_d^{0.8} Pr^{0.4} \quad (14.10\ b)$$

Once saturation boiling conditions are reached, A fully developed boiling is eventually encountered which is independent of the flow velocity or forced convection effects, and the following relation for boiling water inside tubes can be used

$$q'' = 2.253(\Delta T)^{3.96} \text{ W/m}^2 \quad \text{for } 0.2 < P \leq 0.7 \text{ MN/m}^2 \quad (14.11)$$

$$q'' = 283.2(P)^{4/3} (\Delta T)^3 \text{ W/m}^2 \quad \text{for } 0.7 < P \leq 14 \text{ MN/m}^2 \quad (14.12)$$

Where P is the pressure in MN/ m²

14.2 Condensation

Condensation occurs when a surface is exposed to a liquid if it is maintained at a temperature below the saturation temperature of the liquid as shown in Figure 14.7. Also the heat transferred from the fluid to solid surface is

$$q'' = \bar{h}(T_s - T_{sat}) = \bar{h}\Delta T \quad (14.13)$$

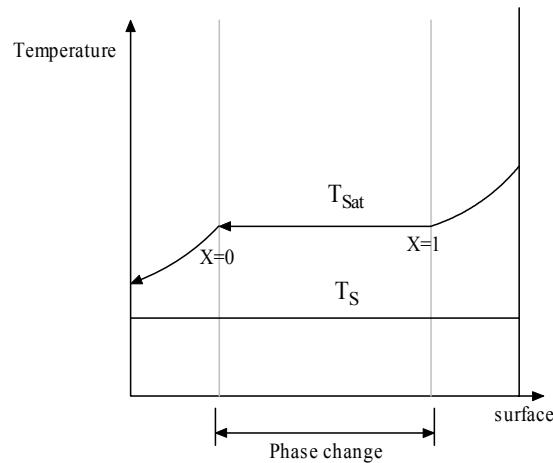


Figure 14.7 Surface –Temperature diagram for condensation heat transfer

As in the case of boiling, surface tension effects, surface wetting characteristics, and phase stability also play important roles in condensation processes.

Condensation on external surfaces may occur either as dropwise condensation or as film condensation, depending on the surface conditions. In fact, heat transfer rates in dropwise condensation are as much as 10 times higher than in film condensation.

14.2.1 Dropwise Condensation

Dropwise condensation occurs on new surfaces but it is extremely difficult to be maintained since most surfaces become wetted after exposure to a condensing vapor over an extended period of time. More than 90% of the surface is covered by drops, ranging from a few micrometers in diameter to agglomerations visible to the naked eye as shown in Figure 14.8. When droplets become large enough, they are generally removed from the surface by the action of gravity or drag forces resulting from the motion of the surrounding gas. As the drops roll or fall from the surface they merge with droplets in their path as shown in Figure 14.9.

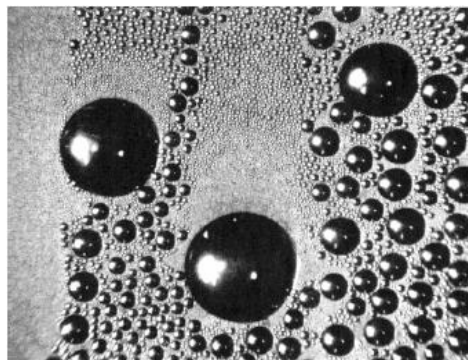


Figure 14.8 Typical photograph of dropwise condensation provided by Professor Borivoje B. Mikic

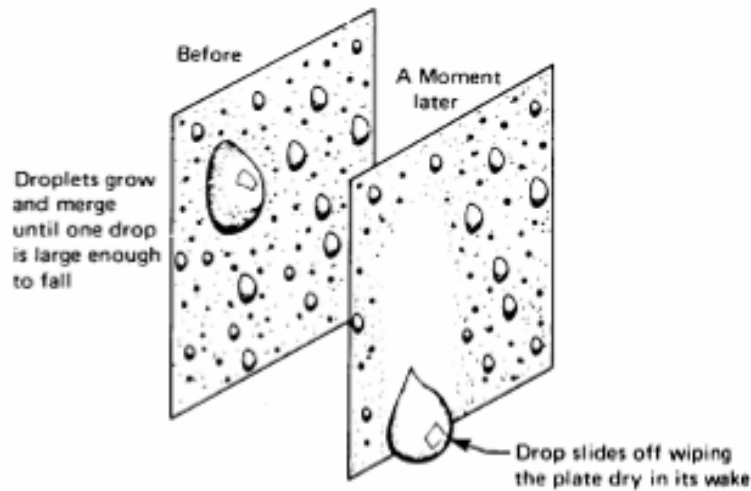


Figure 14.9 Process of liquid removal during dropwise condensation

14.2.2 External Film Condensation

In film condensation the liquid phase fully wets a cold surface in contact with a vapor near saturation conditions; the conversion of vapor to liquid will take the form of film condensation. As the name implies, the condensation takes place at the interface of a liquid film covering the solid surface as shown in Figure 14.10.

Because the latent heat of vaporization must be removed at the interface to sustain the process, the rate of condensation is directly linked to the rate at which heat is transported across the liquid film from the interface to the surface

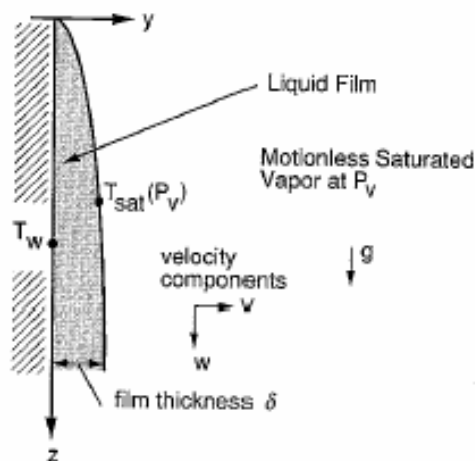


Figure 14.10 System model for falling-film condensation

Because the liquid film flows down the surface by gravity, this situation is sometimes referred to as falling-film condensation. Although it is desirable to achieve drop wise condensation in industrial application, it is often difficult to maintain this condition. Thus most of the common designs are based on film condensation.

14.2.2.1 Laminar Film Condensation on a Vertical Plate ($Re < 1800$)

The Reynolds number defined as

$$\text{Re} = \frac{4m^{\bullet}}{\mu_L p_w} \quad (14.14)$$

Where m^{\bullet} is the condensation rate and p_w is the wetted perimeter (width for vertical plate and πd for vertical tube)

In its simplest form, the classic Nusselt analysis incorporates the following idealizations:

- (1) Laminar flow,
- (2) Constant thermophysical properties,
- (3) That subcooling of liquid is negligible in the energy balance,
- (4) That inertia effects are negligible in the momentum balance,
- (5) The vapor is stationary and exerts no drag,
- (6) The liquid-vapor interface is smooth, and
- (7) The heat transfer across film is only by conduction (convection is neglected).

Under the above assumptions, the following relation for the local heat transfer coefficient h_x can be obtained

$$h_x = \left[\frac{g \rho_L (\rho_L - \rho_v) k_L^3 h_{fg}}{4(T_{sat} - T_s) \mu_L x} \right]^{1/4} \quad (14.15)$$

From

$$\bar{h}_L = \frac{1}{L} \int_0^L h_x dx$$

Then the average heat transfer coefficient over the entire length is then:

$$\bar{h}_L = 0.943 \left[\frac{g \rho_L (\rho_L - \rho_v) k_L^3 h_{fg}}{(T_{sat} - T_s) \mu_L L} \right]^{1/4} \quad (14.16)$$

Liquid properties are evaluated at $T_f = (T_s + T_{sat})/2$ while ρ_v and h_{fg} at T_{sat}

The analysis is identical for an inclined surface, except that the gravitational acceleration g is replaced by $g \cos \theta$ where θ is the angle between the vertical and the surface.

The total heat transfer to the surface may be obtained from the Newton's law of cooling:

$$q = \bar{h}_L A (T_{sat} - T_s) \quad (14.17)$$

The total condensation rate may be then determined from the relation

$$m^{\bullet} = \frac{q}{h_{fg}} \quad (14.18)$$

14.2.2.2 Turbulent Film Condensation on a Vertical and Inclined Plates and Cylinders ($\text{Re} > 1800$)

The suggested relation for turbulent film condensation on a vertical and inclined plates and cylinders is

$$\bar{h} = 0.0077 \left[\frac{g\rho_L(\rho_L - \rho_v)k_L^3}{\mu_L^2} \right]^{1/3} \text{Re}^{0.4} \quad (14.19)$$

14.2.2.3 Film Condensation on Radial Systems

The average heat transfer coefficient for laminar film condensation on the outer surface of a sphere and horizontal tubes ($\text{Re} < 1800$) is

$$\bar{h}_d = C \left[\frac{g\rho_L(\rho_L - \rho_v)k_L^3 h'_{fg}}{\mu_L(T_{sat} - T_s)d} \right]^{1/4} \quad (14.20)$$

The properties are evaluated as in Equation 14.16.

Where $C = 0.826$ for the sphere and 0.729 for the tube and h'_{fg} is the modified latent heat of the form $h'_{fg} = h_{fg} + 0.68C_{p,L}(T_{sat} - T_s)$

For tube bank: put $d = N_T d$ where N_T is the number of tube in vertical direction and $C = 0.729$

14.2.2.4 Film Condensation inside Horizontal Tubes

Schematically in Figure 14.11 depicts a typical condensation process in a horizontal round tube. Superheated vapor enters the tube and at the exit end the liquid is subcooled. At a point some distance downstream of the entrance, vapor begins to condense on the interior walls of the tube. The location at which this occurs is at or slightly before the bulk flow reaches the equilibrium saturation condition.

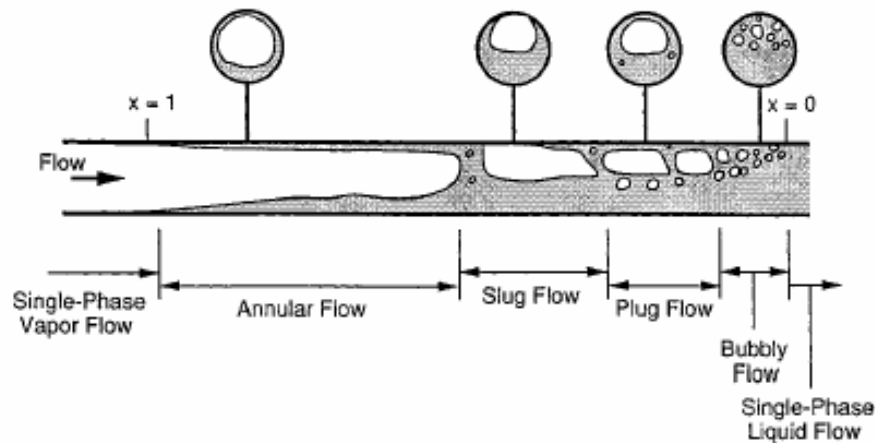


Figure 14.11 Typical condensation process in a horizontal tube

Condensation of refrigerants at low vapor velocities inside horizontal tubes can be modeled using the following equation:

$$\bar{h}_d = 0.555 \left[\frac{g\rho_L(\rho_L - \rho_v)k_L^3 h'_{fg}}{\mu_L(T_{sat} - T_s)d_i} \right]^{1/4} \quad (14.21)$$

For this case

$$h'_{fg} = h_{fg} + \frac{3}{8} C_{p,L} (T_{sat} - T_s)$$