

Heat transfer in boiling and condensation

In the previous chapter we have discussed about the convective heat transfer in which the homogeneous single phase system was considered. The heat transfer processes associated with the change of fluid phase have great importance in chemical process industries. In this chapter, we will focus our attention towards the phase change from liquid to vapour and vice-versa.

6.1 Heat transfer during boiling

The conversion of a liquid into a vapour is one of the important and obvious phenomena. It has been found that if water (say) is totally distilled and degassed so that it does not have any impurity or dissolved gases, it will undergo liquid to vapour phase change without the appearance of bubbles, when it is heated in a clean and smooth container. However, in normal situation, as can be understood, the presence of impurities, dissolved gases, and surface irregularities causes the appearance of vapour bubble on the heating surface, when the rate of heat input is significantly high.

The boiling may be in general of two types. The one in which the heating surface is submerged in a quiescent part of liquid, and the heat transfer occur by free convection and bubble agitation. The process is known as pool boiling. The pool boiling may further be divided into sub-cooled or local boiling and saturated or bulk boiling. If the temperature of the liquid is below the saturation temperature, the process is known as sub-cooled, or local, boiling. If the liquid is maintained at saturation temperature, the process is known as saturated or bulk boiling.

The other form of the boiling is known as forced convective boiling in which the boiling occurs simultaneously with fluid motion induced by externally imposed pressure difference. In this chapter, we will mostly consider the pool boiling.

As generally the bubbles are formed during boiling, we will first refresh the following basic information. Consider a spherical bubble of radius r in a liquid as shown in fig. 6.1

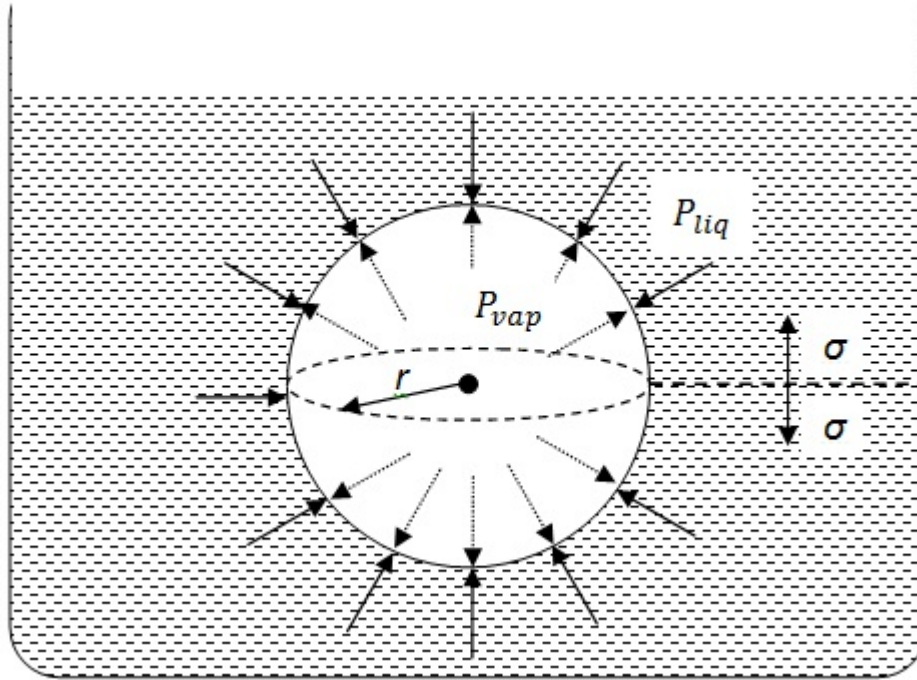


Fig. 6.1: Force balance on a submerge spherical bubble in a liquid

The pressure of vaporisation inside the bubble, P_{vap} , must exceed that in the surrounding liquid, P_{liq} , because of the surface tension (σ) acting on the liquid-vapour interface.

The force balance on the equatorial plane

$$\begin{aligned} \pi r^2 (P_{vap} - P_{liq}) &= 2\pi r \sigma \\ P_{vap} - P_{liq} &= \frac{2\sigma}{r} \end{aligned} \quad (6.1)$$

The eq. 6.1 shows that to create a bubble of small radius, it would be necessary to develop very large pressure in the vapour. In other word, a high degree super heat is necessary for the generation of a tiny bubble (or nucleus) in the bulk liquid. This is the reason, the bubble are usually formed at bits existing in the surface irregularities, where a bubble of finite initial radius may form, or where gasses dissolved in the system of the liquid come out of the solution.

6.2 Boiling of saturated liquid

In this section, we will discuss about the boiling curve which is a log-log plot between heat flux (q/A) or heat transfer coefficient (h) and excess temperature (ΔT). Excess temperature ($\Delta T = T_w - T_{sat}$) is the temperature difference between heating surface (T_w) and saturated temperature of the liquid (T_{sat}).

Figure 6.2 shows a typical representative pool boiling curves for water contained in a container where the water is heated by an immersed horizontal wire. Consider we are measuring the heat flux (thus, h) and the temperature difference (ΔT) between the boiling water (T_{sat}) and the wall

temperature of the heater wire (T_w). The temperature of the heater wire is gradually raised while measuring the heat flux between heated surface and boiling water until a large value of ΔT reaches. The corresponding plot is prepared at the log-log scale. The plot shows six different sections in the boiling curve shown in the fig.6.2.

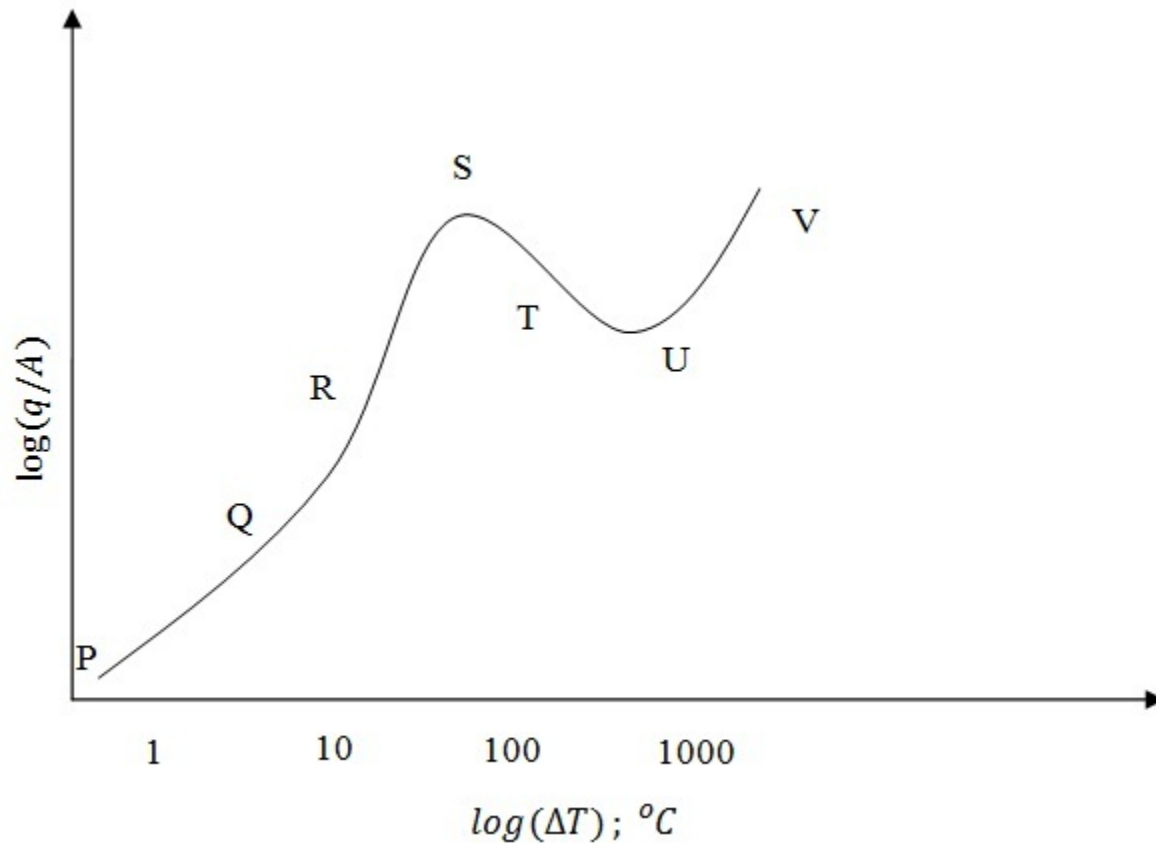


Fig. 6.2: Saturated water boiling curve

The different regimes of the boiling plot (fig. 6.2) have different mechanism. We will see those mechanisms in-brief in the following section.

Section PQ: In section PQ , initially when the wire temperature is slightly above the saturation temperature of the liquid, the liquid in contact with the heating surface get slightly superheated. The free convection of this heated fluid element is responsible for motion of the fluid, and it subsequently evaporated when it rises to the surface. This regime is called the interfacial evaporation regime.

Section QS: The section QS is composed of section QR and section RS . In QR section, bubbles begin to form on the surface of the wire and are dissipated in the liquid after detaching from the heating surface. If the excess temperature (further increases, bubbles form rapidly on the surface of the heating wire, and released from it, rise to the surface of the liquid, and are discharged into the top of the water surface (fig 6.3). This particular phenomenon is shown in section RS . Near the point S , the vapour bubbles rise as columns and bigger bubbles are formed. The vapour bubbles break and coalesce thus an intense motion of the liquid occurs which in-turn increases the heat

transfer coefficient or heat transfer flux to the liquid from the heating wire. The section QS is known as nucleate boiling.

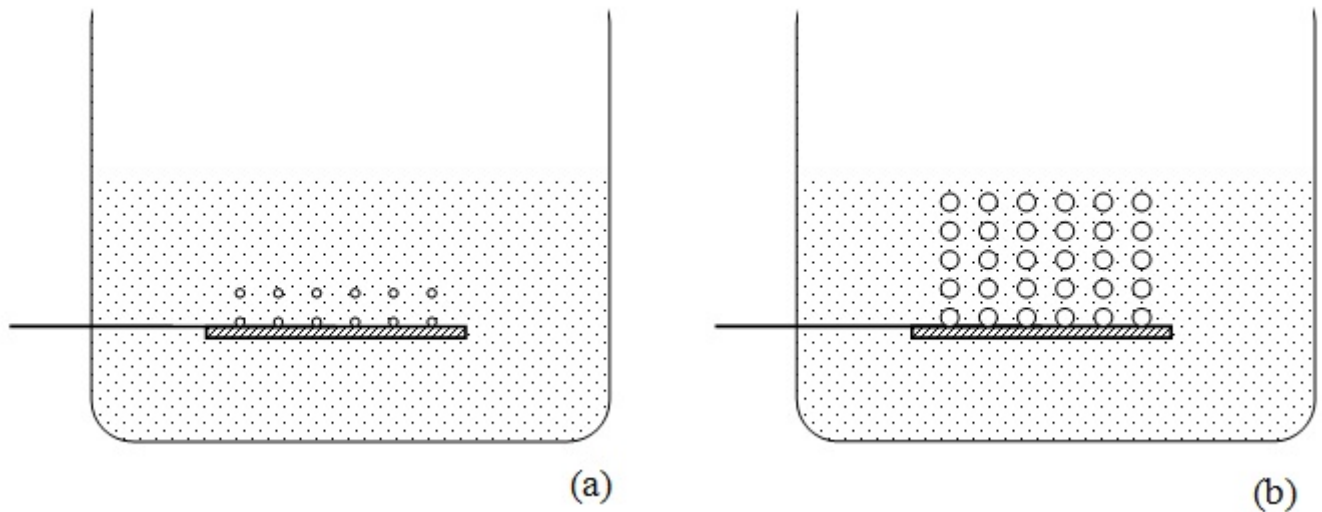


Fig. 6.3: (a) Formation of tiny bubbles, and (b) Grown up bubbles

Section ST : At the beginning of the section ST or at the end of the section, the maximum number of bubbles are generated from the heating surface. The bubbles almost occupy the full surface of the heating wire. Therefore, the agitation becomes highest as they discharge from the surface. Thus, maximum heat transfer coefficient is obtained at point S . However, once the population of the bubbles reaches to maximum, the nearby bubbles coalesce and eventually a film of vapour forms on the heating surface. This layer is highly unstable and it forms momentarily and breaks. This is known as transition boiling (nucleate to film). In this situation the vapour film (unstable) imparts a thermal resistance and thus the heat transfer coefficient reduces rapidly.

Section TU : If the excess temperature is further increased, the coalesced bubbles form so rapidly that they blanket the heating surface (stable vapour film) and prevent the inflow of fresh liquid from taking their place. The heat conducts only by the conduction through this stable vapour film. As a result the flux of heat transfer decreases continuously and reaches a minimum at point U . All the resistance to the heat transfer is imposed by this layer stable layer of vapour film.

Section UV : At very high excess temperature the heat transfer is facilitated by the radiation through the vapour film and thus the heat transfer coefficient start increasing. Infact the excess temperature in this regime is so high that the heating wire may get melted. This situation is known as boiling crises. The combine regime of ST , TU , and UV is known as film boiling regime.

At this stage it would be interesting to know the Leidenfrost phenomenon, which was observed by Leidenfrost in 1756. When water droplets fall on a very hot surface they dance and jump on the hot surface and reduces in size and eventually the droplets disappear. The mechanism is related to the film boiling of the water droplets. When water droplet drops on to the very hot surface, a film of vapour forms immediately between the droplet and the hot surface. The vapour film generated provide and up-thrust to the droplet. Therefore, the droplet moves up and when again the droplet comes in the contact of the hot surface, the vapour generated out of the water droplet and the phenomenon continues till it disappears.

The effectiveness of nucleate boiling depends primarily on the ease with which bubbles form and free themselves from the heating surface. The important factor in controlling the rate of bubble detachment is the interfacial tension between the liquid and the heating surface. If this interfacial tension is large the bubbles tends to spread along the surface and blocked the heat transfer area, rather than leaving the surface, to make room for other bubbles. The heat transfer coefficient obtained during the nucleation boiling is sensitive to the nature of the liquid, the type and condition of the heating surface, the composition and purity of the liquid, agitation, temperature and pressure.

Fact: Film boiling is not normally desired in commercial equipment because the heat transfer rate is low for such a large temperature drop.

6.2.1 Nucleation boiling

Rohsenow correlation may be used for calculating pool boiling heat transfer

$$q = \mu_l \lambda \left[\frac{g(\rho_l - \rho_v)}{\sigma} \right]^{1/2} \left[\frac{c_{pl} T_e}{C_{sf} \lambda (Pr_l)^n} \right]^3 \quad (6.2)$$

where,

q is the heat flux (W/m²)

μ_l is the liquid viscosity (Pa.s)

λ is the enthalpy of liquid vaporisation (J/kg)

ρ_l and ρ_v are the liquid and vapour density, respectively, (kg/m³)

c_{pl} is the specific heat of liquid (J/kg/°C)

σ is the surface tension (N/m)

T_e is the excess temperature of the boiling surface, $T_w - T_{sat}$ (K)

Pr_l is the liquid Prandtl number

C_{sf} and n are the constants and depend on the liquid and heating surface combination for boiling operation, for example,

Liquid-heating surface combination	C_{sf}	n
Water-Stainless steel	0.01	1.0
Water-Brass	0.013	1.0
Water-Copper	0.006	1.0
Benzene-Chromium	0.01	1.7

All the properties are to be evaluated at film temperature.

6.2.2 Maximum heat flux

Maximum heat flux corresponding to the point *S* in the fig.6.2 can be found by Leinhard correlation,

$$q_{max} = 0.149 \lambda \rho_v \left[\sigma g \frac{\rho_l - \rho_v}{\rho_v^2} \right]^{1/4} \quad (6.3)$$

The notations are same as for eq.6.2.

6.2.3 Film boiling

$$h = 0.62 \left[\frac{k_v^3 \rho_v (P_l - P_v) g (L + 0.4 C_{pv} T_e)}{d \mu_v T_e} \right]^{1/4} \quad (6.4)$$

where, k_v is the thermal conductivity of the vapour, μ_v is the viscosity of the vapour, d is the characteristic length (tube diameter or height of the vertical plate), other notations are same as for eq. 6.2.

If the surface temperature is high enough to consider the contribution of radiative heat transfer, the total heat transfer coefficient may be calculation by,

$$h_T = h \left(\frac{h}{h_T} \right)^{1/3} + h_r \quad (6.5)$$

where, h_r is the radiative heat transfer coefficient and is given in eq.6.4.

Upto this section, we have discussed about the boiling phenomenon where the liquid phase changes to vapour phase. In the subsequent sections, we will study the opposite phenomena of boiling that is condensation, where the vapour phase changes to the liquid phase.

6.3 Heat transfer during condensation

Condensation of vapours on the surfaces cooler than the condensing temperature of the vapour is an important phenomenon in chemical process industries like boiling phenomenon. It is quite clear that in condensation the phase changes from vapour to liquid. Consider a vertical flat plate which is exposed to a condensable vapour. If the temperature of the plate is below the saturation temperature of the vapour, condensate will form on the surface and flows down the plate due to gravity. It is to be noted that a liquid at its boiling point is a saturated liquid and the vapour in equilibrium with the saturated liquid is saturated vapour. A liquid or vapour above the saturation temperature is called superheated. If the non-condensable gases will present in the vapour the rate of condensation of the vapour will reduce significantly.

Condensation may be of two types, film condensation and dropwise condensation. If the liquid (condensate) wets the surface, a smooth film is formed and the process is called film type condensation. In this process, the surface is blocked by the film, which grows in thickness as it moves down the plate. A temperature gradient exists in the film and the film represents thermal resistance in the heat transfer. The latent heat is transferred through the wall to the cooling fluid on the other side of the wall. However, if the liquid does not wet the system, drops are formed on the surface in some random fashion. This process is called dropwise condensation. Some of the surface will always be free from the condensate drops (for a reasonable time period).

Now, with the help of the above discussion one can easily understand that the condensate film offers significant heat transfer resistance as compared to dropwise condensation. In dropwise condensation the surface is not fully covered by the liquid and exposed to the vapour for the condensation. Therefore, the heat transfer coefficient will be higher for dropwise condensation. Thus the dropwise condensation is preferred over the film condensation. However, the dropwise condensation is not practically easy to achieve. We have to put some coating on the surface or we have to add some additive to the vapour to have dropwise condensation. Practically, these techniques for dropwise condensation are not easy for the sustained dropwise condensation. Because of these reasons, in many instances we assume film condensation because the film condensation sustained on the surface and it is comparatively easy to quantify and analyse.

6.4 Film condensation on a vertical flat plate

Figure 6.4 shows a vertical wall very long in z-direction. The wall is exposed to a condensable vapour. The condensate film is assumed to be fully developed laminar flow with zero interfacial shear and constant liquid properties. It is also assumed that the vapour is saturated and the heat transfer through the condensate film occurs by condensation only and the temperature profile is assumed to be linear.

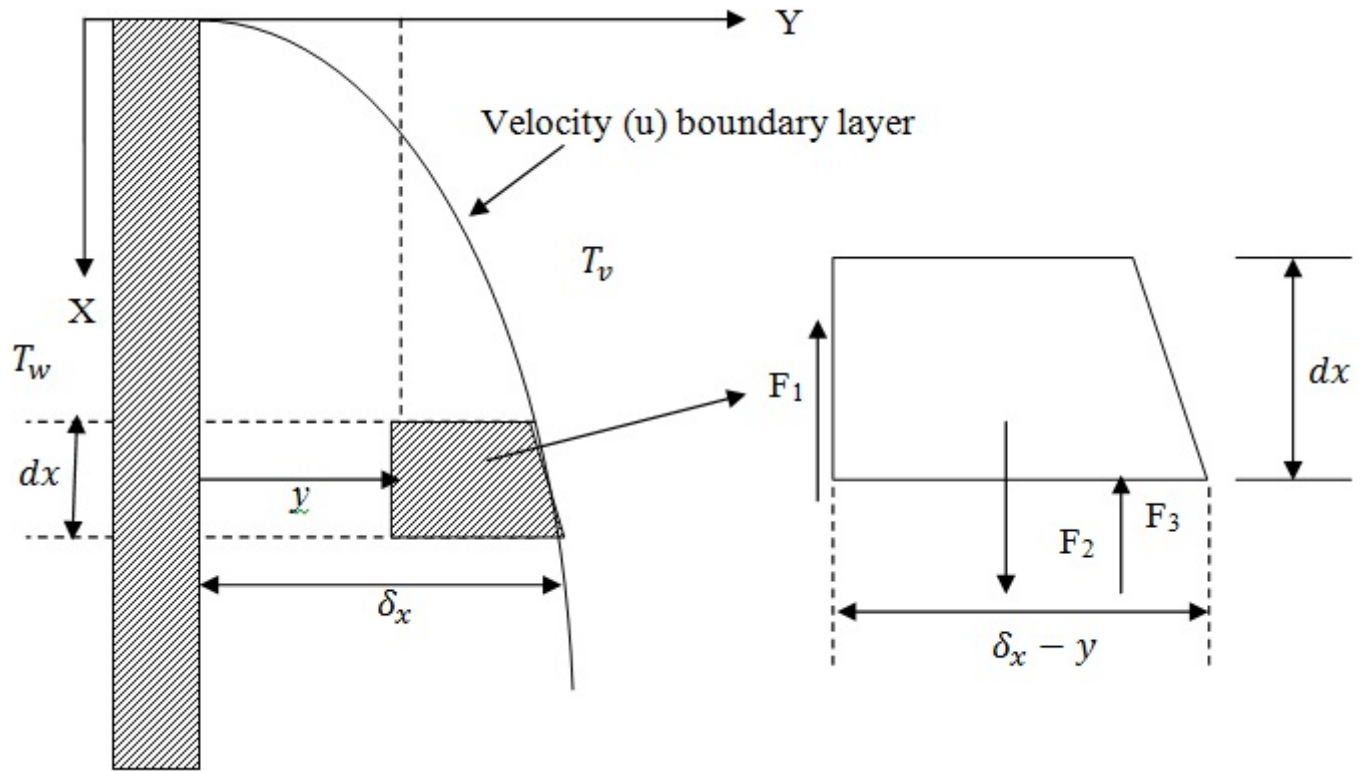


Fig. 6.4: Condensation of film in laminar flow

The wall temperature is maintained at temperature T_w and the vapour temperature at the edge of the film is the saturation temperature T_v . The condensate film thickness is represented by δ_x , a function of x . A fluid element of thickness dx was assumed with a unit width in the z -direction. The force balance on the element provides,

$$F_1 = F_2 - F_3$$

where, shear force $F_1 = \mu_l \frac{du}{dy} dx$; μ_l is the viscosity of the condensate (liquid). In the subsequent sections of this module, the subscripts l and v will represent liquid and vapour phase.

Gravity force, $F_2 = \rho_l g (\delta_x - y) dx$, and

Buoyancy force, $F_3 = \rho_v g (\delta_x - y) dx$

Thus,

$$\mu_l \frac{du}{dy} dx = \rho_l g (\delta_x - y) dx - \rho_v g (\delta_x - y) dx$$

On integrating for the following boundary condition, $u = 0$ at $y = 0$; no slip condition.

$$u = \frac{(\rho_l - \rho_v)g}{\mu_l} \left[y\delta_x - \frac{y^2}{2} \right] \quad (6.6)$$

Equation 6.6 shows the velocity profile in the condensate falling film.

The corresponding mass flow rate of the condensate for dy thickness and unit width of the film,

$$\begin{aligned} \dot{m} &= \int_0^{\delta} u(dy \cdot 1) \rho_l \\ \dot{m} &= \frac{\rho_l(\rho_l - \rho_v)g\delta_x^3}{3\mu_l} \end{aligned} \quad (6.7)$$

where dy is the length of the volume element at y distance. The rate of condensation for dx (over element surface) area exposed to the vapour can be obtained from the rate of heat transfer through this area.

$$= k_l \frac{T_v - T_w}{\delta_x} dx \cdot 1$$

The rate of heat transfer

The thermal conductivity of the liquid is represented by k_l . The above rate of heat transfer is due to the latent heat of condensation of the vapour. Thus,

$$d\dot{m} = k_l \frac{T_v - T_w}{\delta_x \lambda} dx \quad (6.8)$$

The specific latent heat of condensation is represented by λ . On solving eqs.6.7 and 6.8, for boundary layer conditions ($x = 0$; $\delta_x = 0$)

$$\delta_x = \left[\frac{4\mu_l k_l x (T_v - T_w)}{g\lambda\rho_l(\rho_l - \rho_v)} \right]^{1/4} \quad (6.9)$$

The eq. 6.9 gives the local condensate film thickness at any location x . If h is the film heat transfer coefficient for the condensate film, heat flux through the film at any location is,

$$h = k_l \left[\frac{4\mu_l k_l x (T_v - T_w)}{g\lambda\rho_l(\rho_l - \rho_v)} \right]^{-\frac{1}{4}} \quad (6.10a)$$

The local Nusselt number will be,

$$Nu_x = \frac{hx}{k_l} \left[\frac{g\lambda\rho_l(\rho_l-\rho_v)x^3}{4\mu_l k_l (T_v-T_w)} \right]^{1/4}$$

We can also calculate the average heat transfer coefficient along the length of the surface,

$$h_{av} = \bar{h} = \frac{1}{L} \int_0^L h dx = 0.943 \left[\frac{g\lambda\rho_l(\rho_l-\rho_v)k_l^3}{4\mu_l L (T_v-T_w)} \right]^{1/4} ; \quad \bar{h} = \frac{4}{3} h_{x=L} \quad (6.10b)$$

In eq. 6.10, the liquid properties can be taken at the mean film temperature $\left(\frac{T_v+T_w}{2} \right)$. The

equation 6.10 is applicable for $Pr > 0.5$ and $\frac{c_p \tau}{\lambda} \leq 1.0$

It can also be understood that at any location on the plate the liquid film temperature changes from T_v to T_w . It indicates that apart from latent heat some amount of sensible heat will also be removed. Thus, to take this into account and to further improve the accuracy of Nusselt's equation

(eq. 6.10), a modified latent heat term $\dot{\lambda} = \lambda(1 + 0.68 J_a)$ can be used in place of λ . The term J_a is called the Jacob number as is defined by eq. 6.11. All the properties are to be evaluated at film temperature.

$$J_a = c_{pl} \frac{T_v - T_w}{\lambda} \quad (6.11)$$

In the previous discussion we have not discussed about the ripples or turbulent condition of the condensate film as it grows while coming down from the vertical wall. The previous discussion was applicable only when the flow in the condensate film was 1-D and the velocity profile was half parabolic all along the length of the wall. However, if the rate of condensation is high or the height of the condensing wall is more, the thickness of the condensate film neither remains small nor the flow remains laminar.

The nature of the flow is determined by the film Reynolds number (Re_f). The local average liquid velocity in the film can be obtained by eq. 6.6.

$$\bar{u} = \frac{1}{\delta_x} \int_0^{\delta_x} u dy = \frac{(\rho_l - \rho_v)g \delta^2}{3\mu_l}$$

Now, the Re_f can be calculated by,

$$Re_f = \frac{D \bar{u} \rho}{\mu_l} = \frac{4 \dot{m}}{\mu_l} \quad (6.12)$$

where D is the hydraulic diameter of the condensate film. The hydraulic diameter can be calculated by the flow area ($\delta_x \cdot 1$) and wetted perimeter (unit breadth, thus 1). It has been found that, if

Case 1: $Re_f \leq 30$; the film remains laminar and the free surface of the film remains wave free.

Case 2: $30 < Re_f < 1600$; the film remains laminar but the waves and ripples appear on the surface.

Case 3: $Re_f \geq 1600$; the film becomes turbulent and surface becomes wavy.

The corresponding average heat transfer coefficient can be calculated by the following correlation,

$$\overline{Nu}(=Co) = \frac{\bar{h}\delta_x}{k_l} = \frac{\bar{h}}{k_l} \left[\frac{\mu_l^2}{\rho_l(\rho_l - \rho_v)g} \right]^{1/3} = 1.47(Re_f)^{-1/3} \quad : \text{for Case 1}$$

(It is same as eq. 6.10 if Re_f is taken at the bottom of the wall.)

$$\overline{Nu} = \frac{Re_f}{1.08(Re_f)^{1.22} - 5.2} \quad : \text{for Case 2}$$

$$\overline{Nu} = \frac{Re_f}{8750 + 58[(Re_f)^{0.75} - 253]Re_f^{-0.5}} \quad : \text{for Case 3}$$

The Nusselt number in case-1 is defined as Modified Nusselt number or condensation number (Co).

The above relations may also be used for condensation inside or outside of a vertical tube if the tube diameter is very large in comparison to condensate film thickness. Moreover, the relations are valid for the tilted surfaces also. If the surface make an angle " θ " from the vertical plane the " g " will be replaced by " $g \cdot \cos\theta$ " in the above equations