UNIT III

PHASE CHANGE HEAT TRANSFER AND HEAT EXCHANGERS

3 Condensation and Boiling

Heat energy is being converted into electrical energy with the help of water as a working fluid. Water is first converted into steam when heated in a heat exchanger and then the exhaust steam coming out of the steam turbine/engine is condensed in a condenser so that the condensate (water) is recycled again for power generation. Therefore, the condensation and boiling processes involve heat transfer with change of phase. When a fluid changes its phase, the magnitude of its properties like density, viscosity, thermal conductivity, specific heat capacity, etc., change appreciably and the processes taking place are greatly influenced by them. Thus, the condensation and boiling processes must be well understood for an effective design of different types of heat exchangers being used in thermal and nuclear power plants, and in process cooling and heating systems.

3.1 Condensation-Filmwise and Dropwise

Condensation is the process of transition from a vapour to the liquid or solid state. The process is accompanied by liberation of heat energy due to the change 10 phase. When a vapour comes 10 contact with a surface maintained at a temperature lower than the saturation temperature of the vapour corresponding to the pressure at which it exists, the vapour condenses on the surface and the heat energy thus released has to be removed. The efficiency of the condensing unit is determined by the mode of condensation that takes place:

Filmwise - the condensing vapour forms a continuous film covering the entire surface,

Dropwise - the vapour condenses into small liquid droplets of various sizes. The dropwise condensation has a much higher rate of heat transfer than filmwise condensation because the condensate in dropwise condensation gets removed at a faster rate leading to better heat transfer between the vapour and the bare surface. .

It is therefore desirable to maintain a condition of dropwise condensation 1D commercial application. Dropwise condensation can only occur either on highly polished surfaces or on surfaces contaminated with certain chemicals. Filmwise condensation is expected to occur in most instances because the formation of dropwise condensation IS greatly influenced by the presence of non-condensable gases, the nature and composition of surfaces and the velocity of vapour past the surface.

5.2.3. Filmwise Condensation Mechanism on a Vertical Plane Surface--Assumption

Let us consider a plane vertical surface at a constant temperature, T_s on which a pure vapour at saturation temperature, T_g ($T_g > T_s$) is condensing. The coordinates are: X-axis along the plane surface wit~ its origin at the top edge and Y-axis is normal to the plane surface as shown in Fig. 11.1. The condensing liquid would wet the solid surface, spread out and form a continuous film over the entire condensing surface. It is further assumed that

(i) the continuous film of liquid will flow downward (positive X-axis) under the action of gravity and its thickness would increase as more and more vapour condenses at the liquid - vapour interface,

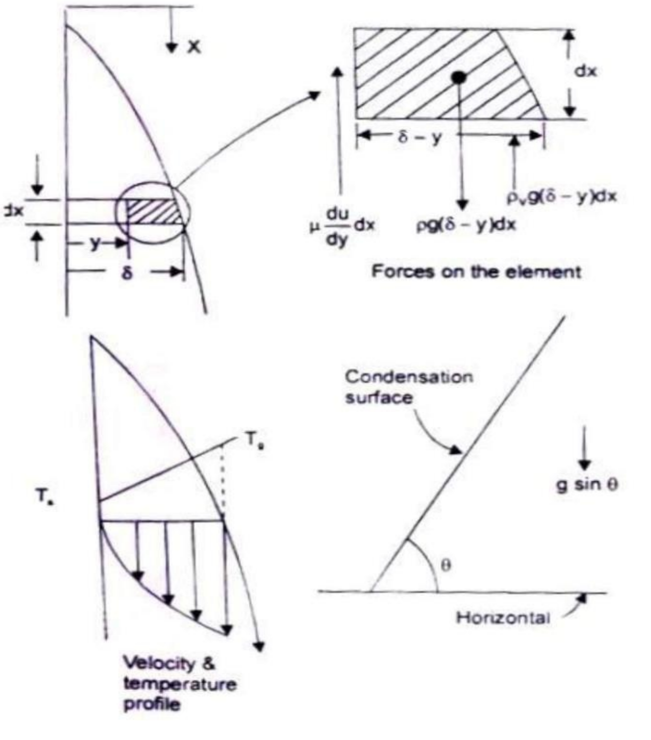

Fig. 5.11 Filmwise condensation on a vertical and Inclined surface

(ii) the continuous film so formed would offer a thermal resistance between the vapour and the surface and would reduce the heat transfer rates,

(iii) the flow in the film would be laminar,

(iv) there would be no shear stress exerted at the liquid vapour interface,

(v) the temperature profile would be linear, and

(vi) the weight of the liquid film would be balanced by the viscous shear in the liquid film and the buoyant force due to the displaced vapour.

5.2.4. An Expression for the Liquid Film Thickness and the Heat Transfer Coefficient in Laminar Filmwise Condensation on a Vertical Plate

We choose a small element, as shown in Fig. 11.1 and by making a force balance, we write

$$\rho g(\delta - y)dx = \mu(du/dy)dx + \rho_v g(\delta - y)dx \qquad (5.44)$$

where □ is the density of the liquid, □$_v$ is the density of vapour, □ is the viscosity of the liquid, Ii is the thickness of the liquid film at any x, and du/dy is the velocity gradient at x.

Since the no-slip condition requires u = 0 at y = 0, by integration we get:

$$u = (\square - \square_v)g(\square y - y^2/2) / \square \qquad (5.45)$$

And the mass flow rate of condensate through any x position of the film would be

$$\dot{m} = \int_0^\delta \rho u \; dy = \int_0^\delta \left[\rho(\rho - \rho_v)(g/\mu)\left(\delta y - y^2/2\right)\right]dy$$

$$= \rho(\rho - \rho_v)\; g\delta^3/3\mu \qquad (5.46)$$

The rate of heat transfer at the wall in the area dx is, for unit width,

$$\dot{Q} = -kA\left(dt/dy\right)_{y=0} = k(dx \times 1)\left(T_g - T_s\right)/\delta \; ,$$

(temperature distribution is linear)

Since the thickness of the film increases in the positive X-direction, an additional mass of vapour will condense between x and x + dx, i.e.,

$$\frac{d}{dx}\left(\frac{\rho(\rho-\rho_v)g\delta^3}{3\mu}\right)dx = \frac{d}{d\delta}\left(\frac{\rho(\rho-\rho_v)g\delta^3}{3\mu}\right)\frac{d\delta}{dx}dx$$

$$= \frac{\rho(\rho-\rho_v)g\delta^2 d\delta}{\mu}$$

This additional mass of condensing vapour will release heat energy and that has to removed by conduction through the wall, or,

$$\therefore \quad \frac{\rho(\rho-\rho_v)g\delta^2 d\delta}{\mu} \times h_{fg} = k\ dx\left(T_g - T_s\right)/\delta \tag{5.47}$$

We can, therefore, determine the thickness, δ, of the liquid film by integrating Eq. (11.4) with the boundary condition: at $x = 0$, $\delta = 0$,

$$\text{or,}\ \ \delta = \left(\frac{4\mu kx\left(T_g - T_s\right)}{gh_{fg}\rho(\rho-\rho_v)}\right)^{0.25} \tag{5.48}$$

The rate of heat transfer is also related by the relation,

$$h\,dx\left(T_g - T_s\right) = k\ dx\left(T_g - T_s\right)/\delta;\ \text{or,}\ h = k/\delta$$

which can be expressed In dimensionless form in terms of Nusselt number,

$$Nu = hx/k = \left[\frac{\rho(\rho-\rho_v)gh_{fg}x^3}{4\mu k\left(T_g - T_s\right)}\right]^{0.25} \tag{5.49}$$

The average value of the heat transfer coefficient is obtained by integrating over the length of the plate:

$$\bar{h} = (1/L)\int_0^L h_x\ dx = (4/3)h_x = L$$

$$Nu_L = 0.943\left[\frac{\rho(\rho-\rho_v)gh_{fg}L^3}{k\mu\left(T_g - T_s\right)}\right]^{0.25} \tag{5.50}$$

The properties of the liquid in Eq. (5.50) and Eq. (5.49) should be evaluated at the mean temperature, $T = (T_g + T_s)/2$.

The above analysis is also applicable to a plane surface inclined at angle □ with the horizontal, If g is everywhere replaced by g . sin□□.

Thus:

Local $$Nu_x = 0.707\left[\frac{\rho(\rho-\rho_v)h_{fg}x^3 g\sin\ \theta}{\mu k(T_g - T_s)}\right]^{0.25}$$

and the average $$Nu_L = 0.943\left[\frac{\rho(\rho-\rho_v)h_{fg}L^3 g\sin\theta}{\mu k(T_g - T_s)}\right] \tag{5.51}$$

These relations should be used with caution for small values of □□because some of the assumptions made in deriving these relations become invalid; for example, when □ is equal to zero, (a horizontal surface) we would get an absurd result. But these equations are valid for condensation on the outside surface of vertical tubes as long as the curvature of the tube surface is not too great.

Solution: (a) Tube Homontal: The mean film temperature is (50 + 76) = 63°C, and the properties are:

$\rho = 980\ kg/m^3,\ \mu = 0.432 \times 10^{-3} Pa - s,\ k = 0.66\ W/mK$

$h_{fg} = 2320\ kJ/kg,\ \rho >> \rho_v$

$$h = 0.725\left[\left(\rho^2 h_{fg} k^3 g\right)/\ \mu D\left(T_g - T_s\right)\right]^{0.25}$$

$$= 0.725\left[(980)^2 \times 2320\times10^3 \times (0.66)^3 \times 9.81/\left(0.432\times10^{-3}\times0.015\times26\right)\right]^{0.25}$$

$= 10\ kW/m^2K$

(b) Tube Vertical: Eq (5.50) should be used if the film thickness is very small in comparison with the tube diameter.

The film thickness, $$\delta = \left[\left\{\left(4\mu kL\left(T_g - T_s\right)\right)\right\}/\left\{gh_{fg}\,\rho\left(\rho-\rho_v\right)\right\}\right]^{0.25}$$

$$= \left[\frac{(980)^2 \times 9.81 \times 2320 \times 10^3 \times (0.66)^3}{0.432 \times 10^{-3} \times 1.5 \times 26} \right]^{0.25}$$

= = 0.212 mm << 15.0 mm, the tube diameter.

Therefore, the average heat transfer coefficient would be

$$h_v = h_h / \left[0.768 (L/D)^{0.25} \right] = 10/2.429 = 4.11 \text{ kW/m}^2\text{K}$$

(Thus, the performance of horizontal tubes for filmwise laminar condensation is much better than vertical tubes and as such horizontal tubes are preferred.)

Example 3.1A square array of four hundred tubes, 1.5 cm outer diameter is used to condense steam at atmospheric pressure. The tube walls are maintained at 88°C by a coolant flowing inside the tubes. Calculate the amount of steam condensed per hour per unit length of the tubes.

Solution: The properties at the mean temperature (88 + 100)/2 = 94°C are:

□ = 963 kg/mJ, □ = 3.06 × 10^{-4} Pa-s, k = 0.678 W/mK,

$h_{fg} = 2255 \times 10^3$ J/kg

A square array of 400 tubes will have N = 20. From Eq (5.57),

$$h = 0.725 \left[\left(g \rho^2 k^3 h_{fg} \right) \right] / \left[N \mu D \left(T_g - T_s \right) \right]^{0.25}$$

$$= 0.725 \left(\frac{9.81 \times (963)^2 \times (0.678)^3 \times 2255 \times 10^3}{20 \times 0.000306 \times 0.015 \times 12} \right) = 6.328 \text{ kW/m}^2\text{K}$$

Surface area for 400 tubes = 400 × 3.142 × 0.015 × 1 (let L = 1)

= 18.852 m^2 per metre length of the tube

$\dot{Q}$ = hA (□T) = 6.328 × 18.852 × 12 = 1431.56 kW

$\dot{m} = \dot{Q} / h_{fg}$ = 1431.56 × 3600/2255 = 2285.4 kg/hr per metre length.

3 Condensation inside Tubes-Empirical Relation

The condensation of vapours flowing inside a cylindrical tube is of importance in chemical and petro-chemical industries. The average heat transfer coefficient for vapours condensing inside either horizontal or vertical tubes can be determined, within 20 percent accuracy, by the relations:

For $Re_g < 5 \times 10^4$, $Nu_d = 5.03 (Re_g)^{1/3} (Pr)^{1/3}$

For $Re_g > 5 \times 10^4$, $Nu_d = 0.0265 (Re_g)^{0.8} (Pr)^{1/3}$ (5.53)

where Re_g is the Reynolds number defined in terms of the mass velocity, or, Re_g = DG/□, G being the mass rate of flow per unit cross-sectional area.

4 Dropwise Condensation-Merits and Demerits

In dropwise condensation, the condensation is found to appear in the form of individual drops. These drops increase in size and combine with another drop until their size is great enough that their weight causes them to run off the surface and the condensing surface is exposed for the formation of a new drop. This phenomenon has been observed to occur either on highly polished surfaces or on surface coated/contaminated with certain fatty acids. The heat transfer coefficient in dropwise condensation is five to ten times higher than the filmwise condensation under similar conditions. It is therefore, desirable that conditions should be maintained for dropwise condensation in commercial applications. The presence of non-condensable gases, the nature and composition of the surface, the vapour velocity past the surface have great influence on the formation of drops on coated/contaminated surfaces and It is rather difficult to achieve dropwise condensation.

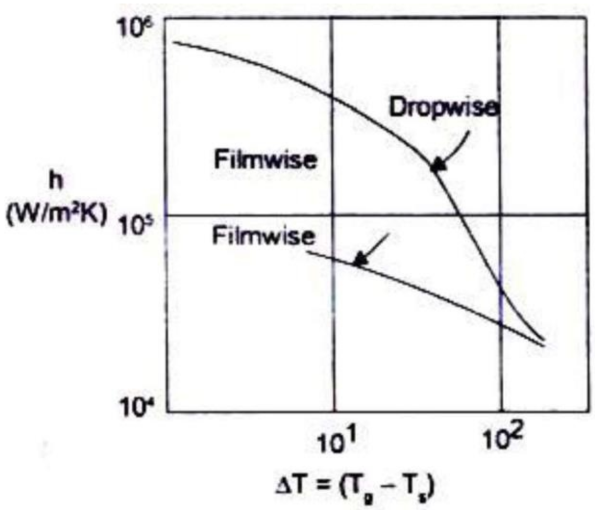

Fig. 5.12 Comparison of h for filmwise and dropwise condensation

Several theories have been proposed for the analysis of dropwise condensation. They do give explanations of the process but do not provide a relation to determine the heat transfer coefficient under various conditions. Fig 5.12 shows the comparison of heat transfer coefficient for filmwise and dropwise condensation.

5. Regimes of Boiling

Let us consider a heating surface (a wire or a flat plate) submerged In a pool of water which is at its saturation temperature. If the temperature of the heated surface exceeds the temperature of the liquid, heat energy will be transferred from the solid surface to the liquid. From Newton's law of cooling, we have

$$\dot{Q}/A = \dot{q} = h(T_w - T_s)$$

where $\dot{Q}/A$ is the heat flux, T_w is the temperature of the heated surface and T_s, is the temperature of the liquid, and the boiling process will start.

(i) Pool Boiling - Pool boiling occurs only when the temperature of the heated surface exceeds the saturation temperature of the liquid. The liquid above the hot surface is quiescent and its motion near the surface is due to free convection.

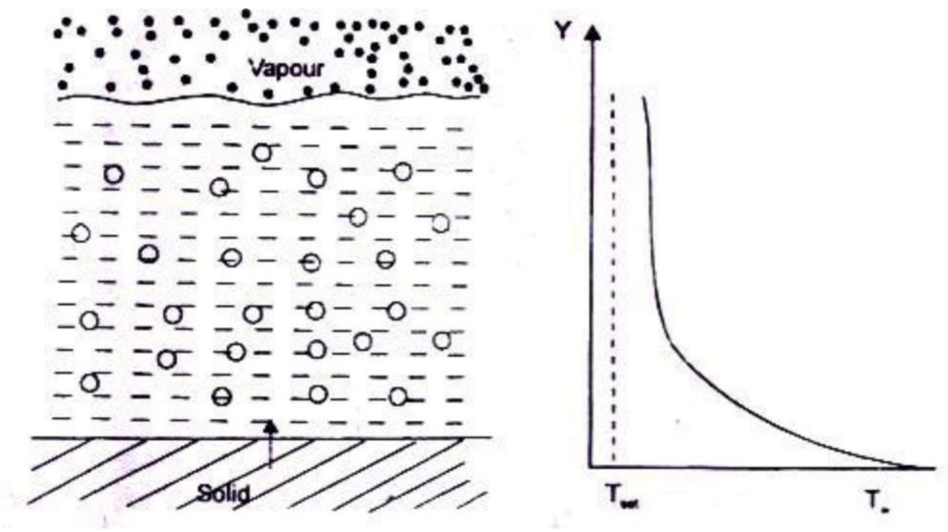

Fig. 5.13 Temperature distribution in pool boiling at liquid-vapour interface

Bubbles grow at the heated surface, get detached and move upward toward the free surface due to buoyancy effect. If the temperature of the liquid is lower than the saturation temperature, the process is called 'subcooled or local boiling'. If the temperature of the liquid is equal to the saturation temperature, the process is known as 'saturated or bulk boiling'. The temperature distribution in saturated pool boiling is shown in Fig5.13. When T_w exceeds T_s by a few degrees, the convection currents circulate in the superheated liquid and the evaporation takes place at the free surface of the liquid.

(ii) Nucleate Boiling - Fig. I 1.5 illustrates the different regimes of boiling where the heat flux $(\dot{Q}/A)$ is plotted against the temperature difference $(T_w - T_s)$. When the temperature T_w increases a little more, vapour bubbles are formed at a number of favoured spots on the heating surface. The vapour bubbles are initially small and condense before they reach the free surface. When the temperature is raised further, their number increases and they grow bigger and finally rise to the free surface. This phenomenon is called 'nucleate boiling'. It can be seen from the figure (5.14) that in nucleate boiling regime, the heat flux increases rapidly with increasing surface temperature. In the latter part of the nucleate boiling, (regime 3), heat transfer by evaporation is more important and predominating. The point A on the curve represents 'critical heat flux'.

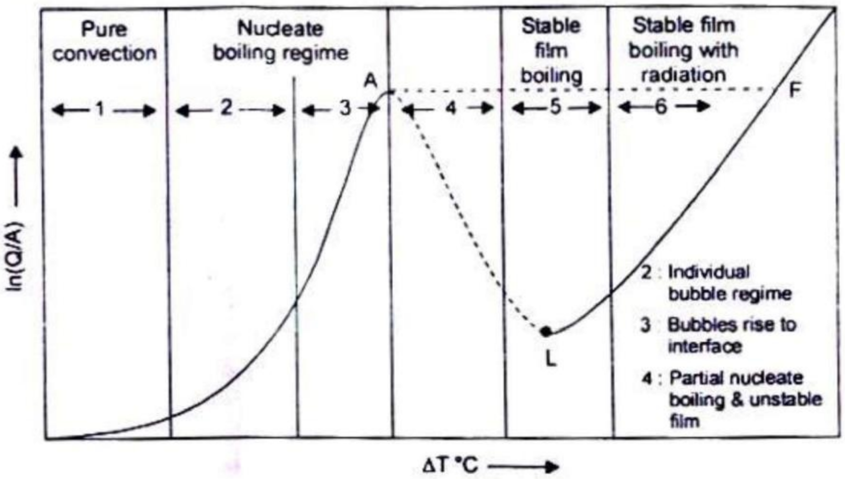

Fig. 5.14 Heat Flux - Temperature difference curve for boiling water heated by a wire (Nukiyama's boiling curve for saturated water at atmospheric pressure) (L is the Laidenfrost Point)

(iii) Film Bolling - When the excess temperature, □T = (T_W–T_S) increases beyond the point A, a vapour film forms and covers the entire heating surface. The heat transfer takes place through the vapour which is a poor conductor and this increased thermal resistance causes a drop in the heat flux. This phase is film boiling'. The transition from the nucleate boiling regime to the film boiling regime is not a sharp one and the vapour film under the action of circulating currents collapses and rapidly reforms. In regime 5, the film is stable and the heat flow rate is the lowest.

(iv) Critical Heat Flux and Burnout Point - For □T beyond 550°C (regime 6) the temperature of the heating metallic surface is very high and the heat transfer occurs predominantly by radiation, thereby, increasing the heat flux. And finally, a point is reached at which the heating surface melts - point F in Fig. 11.5. It can be observed from the boiling curve that the whole boiling process remains in the unstable state between A and F. Any increase in the heat flux beyond point A will cause a departure from the boiling curve and there would be a large increase in surface temperature.

6. Boiling Curve - Operating Constraints

The boiling curve, shown in Fig. 11.5, is based on the assumption that the temperature of the heated surface can be maintained at the desired value. In that case, it would be possible to operate the vapour producing system at the point of maximum flux with nucleate boiling. If the heat flux instead of the surface temperature, is the independent variable and it IS desired to operate the system at the point of maximum flux, it is just possible that a slight increase in the heat flux will increase the surface temperature substantially. And, the equilibrium will be established at point F. If the material of the heating element has its melting point temperature lower than the temperature at the equilibrium point F, the heating element will melt.

7 Factors Affecting Nucleate Boiling

Since high heat transfer rates and convection coefficients are associated with small values of the excess temperature, it is desirable that many engineering devices operate in the nucleate boiling regime. It is possible to get heat transfer coefficients in excess of 10^4 W/m^2 in nucleate boiling regime and these values are substantially larger than those normally obtained in convection processes with no phase change. The factors which affect the nucleate boiling are:

(a) Pressure - Pressure controls the rate of bubble growth and therefore affects the temperature difference causing the heat energy to flow. The maximum allowable heat flux for a boiling liquid first increases with pressure until critical pressure is reached and then decreases.

(b) Heating Surface Characteristics - The material of the heating element has a significant effect on the boiling heat transfer coefficient. Copper has a higher value than chromium, steel and zinc. Further, a rough surface gives a better heat transfer rate than a smooth or coated surface, because a rough surface gets wet more easily than a smooth one.

(c) Thermo-mechanical Properties of Liquids - A higher thermal conductivity of the liquid will cause higher heat transfer rates and the viscosity and surface tension will have a marked effect on the bubble size and their rate of formation which affects the rate of heat transfer.

(d) Mechanical Agitation - The rate of heat transfer will increase with the increasing degree of mechanical agitation. Forced convection increases mixing of bubbles and the rate of heat transfer.