Module 7: Condensation

Lecture 28: Study of Condensation Heat Transfer

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7.2 Introduction

Condensation occurs whenever a vapour comes into contact with a surface at a temperature lower than the saturation temperature corresponding to its vapour pressure.

The nature of condensation depends upon whether the liquid thus formed wets or does not wet the solid surface

If the liquid wets the surface, the condensate flows on the surface in the form of a lm and the process is called ***film condensation***. If on the otherhand, the liquid does not wet the solid surface, the condensate collects in the form of droplets, which either grow in size or coalesce with neighbouring droplets and eventually roll of the surface under the influence of gravity. This process is called ***drop condensation***.

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7.2 Film condensation on a vertical surface

Let us consider heat transfer during laminar film condensation on a vertical plate. The problem was first solved by Nusselt by making certain assumptions

Statement of the problem :-

Calculate the heat transfer coeffiecient during laminar film condensation of a pure, stationary and saturated vapour at a temperature T_{sat} on an isothermal vertical plate at temperature T_w.

The following assumptions are made :-

1. The fluid properties are constant

2. The liquid vapour interface is at the saturation temperature Tsat, i.e., there is no thermal resistance at the liquid vapour interface.

3. Momentum effects in the film are negligible, i.e., there is a static balance of forces.

4.The vapour exerts no shear stress at the liquid vapour interface. Vapour is quiescent $\text{i.e., } u_\infty = 0$

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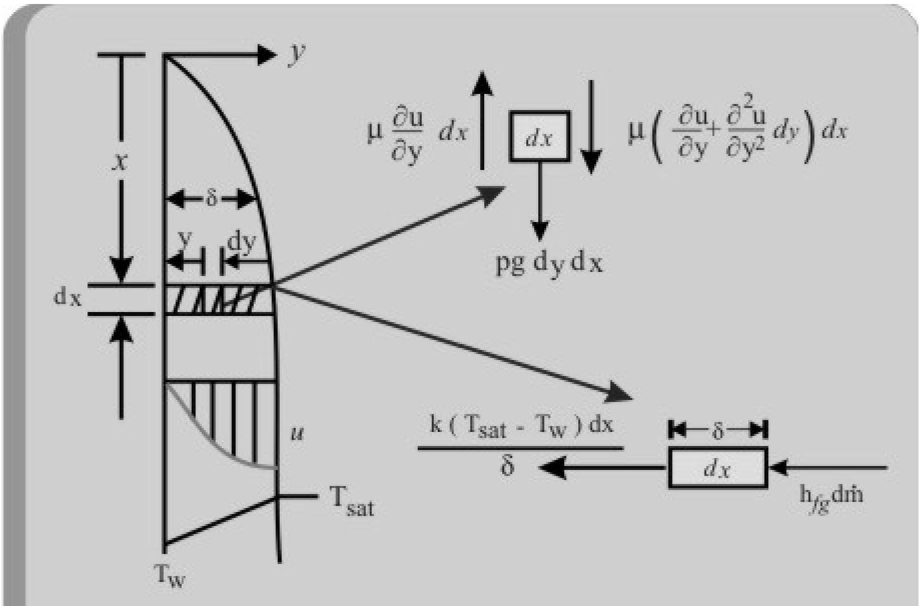

Figure 7.1 Analysis of film condensation on a vertical flat surface

5. The temperature distribution in the film is linear **.**

6. Enthalpy changes associated with sub-cooling of the liquid are negligible.

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Consider the diffrential control volume *dy dx* inside the liquid film. Using assumption **(3)** and neglecting any pressure variations within the liquid in the vertical direction, a force balance on the control volume in the vertical direction yields:

Net shear force exerted on the vertical faces of the control volume + weight of the liquid inside control volume in the downward direction = 0

Continuity and momentum equation for the liquid film

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \qquad \textbf{(7.1)}$$

$$\rho\left[u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y}\right] = -\frac{\partial p}{\partial x} + \mu\frac{\partial^2 u}{\partial y^2} + \rho g \qquad \textbf{(7.2)}$$

y direction momentum-equation is neglected since **v** is trivial.

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad \Rightarrow \quad \frac{\partial u}{\partial x} = 0$$

within vapour

$$\frac{\partial p}{\partial x} = -(-\rho_v\, g) = \rho_v\, g$$

Invoking the above **considerations in (7.2)**

$$\rho\left[u\,\frac{\partial u}{\partial x}+v\,\frac{\partial u}{\partial y}\right]=\mu\,\frac{\partial^2 u}{\partial y^2}+\rho g-\rho_v\,g$$

$$\text{or,}\quad \mu\,\frac{\partial^2 u}{\partial y^2}-(\rho_v-\rho)\,g=0$$

$$\text{or,}\quad \mu\,\frac{\partial^2 u}{\partial y^2}+(\rho-\rho_v)\,g=0$$

$$\text{or,}\quad \frac{d^2 u}{dy^2}=-\frac{g}{\mu}\,(\rho-\rho_v)$$

Since

$$\rho >> \rho_v,\quad \frac{d^2 u}{dy^2}=-\frac{g\rho}{\mu}$$

$$\text{or,}\quad \frac{du}{dy}=-\frac{\rho g\ y}{\mu}+A$$

$$\text{or,}\quad u=-\frac{\rho g}{\mu}\,\frac{y^2}{2}+Ay+B$$

Boundary conditions: @ $y=0,\ u=0$ will lead to $B=0$

At the interface, shear stress $=0$ (assumption (3))

Boundary conditions: @ $y=\delta,\ \frac{\partial u}{\partial y}=0 \quad\Rightarrow\quad A=\frac{\rho g\delta}{\mu}$

$$u=-\frac{g\rho}{\mu}\,\frac{y^2}{2}+\frac{\rho g\delta}{\mu}\,y$$

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$$u = \frac{\rho g}{\mu}\left[y\delta - \frac{y^2}{2}\right]$$

The mass flow rate of the liquid at a distance x from the top edge is given by

$$\dot{m} = \rho \int_o^\delta u \, dy$$

Or

$$\dot{m} = \frac{\rho^2 g}{\mu} \int_o^\delta \left(y\,\delta - \frac{y^2}{2}\right) dy = \frac{\rho^2 g \delta^3}{3\mu} \qquad \textbf{(7.3)}$$

Hence

$$d\dot{m} = \frac{\rho^2 g}{\mu}\delta^2 \, d\delta \qquad \textbf{(7.4)}$$

Equation (7.4) indicates that as we proceed from a section **x** to a section $x + dx$ and the condensate film thickness increases from $\delta \; to \; (\delta + d\delta)$, the mass flow rate increases by $\rho^2 g \delta^2 \, d\delta / \mu . k$. In other words, the condensation rate from $x \text{ to } x + dx$ is $\rho^2 \; g\delta^2 \; d\delta / \; \mu$ and the rate at which energy enters the condensate is $h_{fg} \; \rho^2 \; g\delta^2 \; d\delta / \; \mu$, where h_{fg} is the latent heat of vaporization.
Using assumption (6) and applying energy balance on a control volume of dimensions δ, dx we have Heat released due to condensation at the liquid vapour interface = Rate of heat transfer at the wall

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Heat released due to condensation at the liquid vapour interface = Rate of heat transfer at the wall

$$\frac{h_{fg}\,\rho^2\,g\delta^2}{\mu}\,d\delta = -\left\{-k\,\left(\frac{\partial T}{\partial y}\right)_{y=0}\right\}\,dx = \frac{k\,(T_{sat}-T_w)}{\delta}\,dx$$

$$\delta^3\,d\delta = \frac{k\,(T_{sat}-T_w)\,\mu}{h_{fg}\,\rho^2\,g}\,dx \qquad \textbf{(7.5)}$$

Integrating (7.5) and using the boundary condition that the lm has zero thickness at the top edge, i.e., $x=0, \delta=0$, we have the equation for film thickness.

$$\delta = \left[\frac{4\,k\,\mu(T_{sat}-T_w)\,x}{h_{fg}\,\rho^2\,g}\right]^{1/4} \qquad \textbf{(7.6)}$$

The local heat transfer coeffiecient is defined by the expression.

$$h = \frac{\text{Heat flux at wall}}{(T_{sat}-T_w)} \quad = \quad \frac{k(T_{sat}-T_w)}{\delta(T_{sat}-T_w)} = \frac{k}{\delta} \qquad \textbf{(7.7)}$$

Substituting **(7.6) into (7.7)**

$$h = \left[\frac{h_{fg}\,\rho^2\,g\,k^3}{4\,(T_{sat}-T_w)\,\mu\,x}\right]^{1/4} \qquad \textbf{(7.8)}$$

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The average heat transfer coeffecient over a length ***L*** of the plate will be

$$\begin{aligned}\bar{h} &= \frac{1}{L}\int_o^L h dx \\ \bar{h} &= \frac{1}{L}\left\{\frac{h_{fg}\,\rho^2\,g\,k^3}{4\,(T_{sat}-T_w)\,\mu}\right\}^{1/4}\int_o^L \frac{dx}{x^{1/4}} \\ \bar{h} &= \frac{4}{3}\left\{\frac{h_{fg}\,\rho^2\,g\,k^3}{4\,(T_{sat}-T_w)\,\mu\,L}\right\}^{1/4} \\ \bar{h} &= 0.943\left[\frac{h_{fg}\,\rho^2\,g\,k^3}{(T_{sat}-T_w)\,\mu\,L}\right]^{1/4}\end{aligned} \qquad \textbf{(7.9)}$$

Thus the average heat transfer coffiecient over a length is seen to be 4/3 times the local coeffiecient at x=l The expression **(7.9)** has been found to underpredict experimental results for laminar lm condensation by approximately 20 percent, **it is a practice to use a value of 1.13 in place of the coeffiecient 0.943**

Using (7.9), the average heat flux and condensation flux for a length L can be calculated. These are given by the expression $\bar{h}\,(T_{sat}-T_w)$ and $\bar{h}\,(T_{sat}-T_w)/\,h_{fg}$ respectively.

It should be noted that the results are valid for laminar flows. The existence of a laminar flow depends upon the value of the film Reynolds number which is defined by the expression

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$$Re = 4\dot{m}/\mu \qquad \textbf{(7.10)}$$

Where $\dot{m}$ is the condensate rate for unit width of the plate. The result of average heat transfer coeffiecient given in **(7.9)** can also be expressed in terms of the film Reynolds number.

$$\frac{\bar{h}}{k}\left(\frac{\nu^2}{g}\right)^{1/3} = 1.46\,(Re_L)^{1/3} \qquad \textbf{(7.11)}$$

Where Re_L is the film Reynolds number at $x = L$.

It has been observed experimentally that when the value of the film **Reynolds number is greater than 30,** there are ripples on the lm surface which increase the value of heat transfer coeffiecient Kutateladze has proposed that the value of the local heat transfer coefficient be multiplied by $0.8(Re/4)^{0.11}$ to account for the rippling effect. Using this correction, it can be shown that the average heat transfer coefficient is given by

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$$\frac{\bar{h}}{k}\left(\frac{\nu^2}{g}\right)^{1/3} = \frac{Re_L}{1.08\ Re_L^{1.22} - 5.2} \qquad \textbf{(7.12)}$$

Equation (7.12) is valid for the range of $30 < Re_L < 1600$ When the value of 1600 is exceeded, the condensate flow becomes turbulent. Heat transfer during the turbulent lm condensation has been investigated by many researchers**.**

The following equation valid for Prandtl numbers around unity has been obtained by Labuntsov.

$$\frac{\bar{h}}{k}\left(\frac{\nu^2}{g}\right)^{1/3} = \frac{Re_L}{8750 + 58\ (Pr)^{-1/2}\ (Re_L^{3/4} - 253)} \qquad \textbf{(7.13)}$$

A large number of experimental data is available on laminar and turbulent film condensation, the fluid most commonly used being steam.

It is observed that for fluids with Prandtl numbers greater than one, the experimental data taken in the absence of non-condensable gases agree reasonably well with the results obtained from equation **(7.11), (7.12) and (7.13).** Liquid properties are evaluated at the mean temperature $(T_{sat} + T_w)/2.$ However, with low Prandtl number fluids (liquid metals like mercury and sodium vapour), experimental investigations have yielded heat transfer coefficients which are considerably lower than the results of the above equations. It can be shown that this is due to the fact that assumption **(2)** is not valid for these cases and that the liquid vapour interface may be at a temperature much below the saturation temperature.The preceding equation for condensation on a vertical plate are also applicable in the case of condensation taking place on a vertical tube of length **L** as long as the diameter of the tube is large relative to film thickness.

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Heat transfer during laminar film condensation of a stationary, saturated and pure vapour on the outside of a horizontal tube and a vertical bank of horizontal tubes has also been
calculated by Nusselt.

The film grows symmetrically in thickness as the angle increases. For a single tube, the average heat transfer coeffiecient is given by.

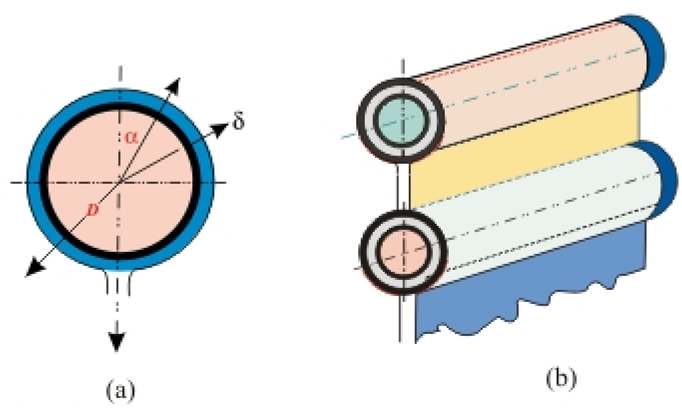

Figure 7.2: Film condensation (a) a sphere (b) a vertical tier of horizontal tubes with a continuous condensate sheet

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$$\bar{h}_D = 0.725\left[\frac{h_{fg}\,\rho^2\,g\,k^3}{\mu\,(T_{sat}-T_w)^D}\right]^{1/4} \qquad \textbf{(7.14)}$$

where **D** is the diameter.
The result can also be expressed in terms of the film Reynolds number as

$$\frac{\bar{h}_D}{k}\left(\frac{\nu^2}{g}\right)^{1/3} = 1.51(Re_D)^{-1/5} \qquad \textbf{(7.15)}$$

Re_D is the film Reynolds number at the bottom of the tube $(\alpha = \pi)$. **Equation (7.14)** is also valid for film condensation on the inside of horizontal tube if the lm does not separate and the tube is short in length. If the tube is long, a pool of liquid collects at the bottom. For a vertical bank of N horizontal tubes, it is assumed that the condensate from one tube drips completely and smoothly on the tube below, the average heat transfer coeffiecient

For a vertical **bank of N horizontal tubes**, it is assumed that the condensate from one tube drips completely and smoothly on the tube below, the average heat transfer coeffiecient **7 is given be**

$$\bar{h}_D = 0.725\left[\frac{h_{fg}\,\rho^2\,g\,k^3}{N(T_{sat}-T_w)\,\mu D}\right]^{1/4} \qquad \textbf{(7.16)}$$

Experiments generally yield heat transfer coeffiecients higher than those arrived at by using **(7.16)**. This is attributed to the fact that some splashing occurs when the condensate drips from one tube to the next. Consequently lm thickness smaller than the values ascribed by theory are obtained Based on his experience, Kern has suggested that the average coefficients for N tubes be taken $N^{-1/6}$ times the value of a single tube.

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Film condensation in horizontal tubes

Condensers used for refrigeration and airconditioning systems generally involve vapour condensation inside horizontal tubes. Conditions within the tube are complicated and depend strongly on the velocity of the vapour flowing through the tube. If this velocity is small, condensation occurs in the manner depicted by **Fig. (7.3a and (7.3b)**. The condensate flow is from the upper portion of the tube to the bottom, from whence it flows in a longitudinal direction with the vapour. For low vapour velocities such that

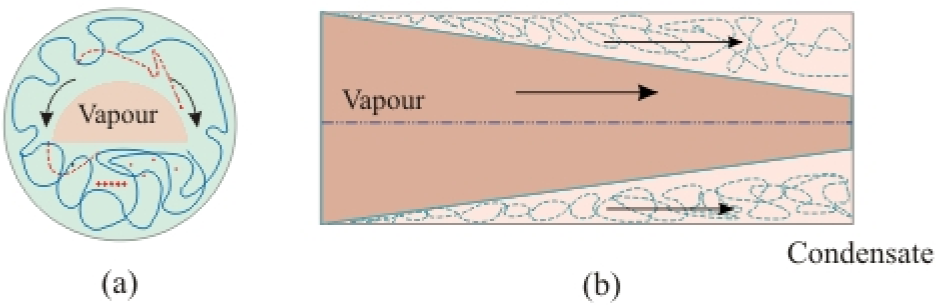

Figure 7.3:
Film condensation in a horizontal tube (a) cross section of condensate flow forlow vapor velocities (b) longitudinal section of condensate flow for large vapor velocities

$$Re_{vi} = \left(\frac{\rho_v\ u_{m,v} D}{\mu_v}\right)_i \quad < \quad 35000, \qquad \text{(where } i \text{ is for tube inlet)}$$

$$h_D = 0.555\left[\frac{h'_{fg}\ \rho^2\ g\ k^3}{\mu\ (T_{sat} - T_w)\ D}\right]^{1/4} \qquad \textbf{(7.17)}$$

Hence

$$h'_{fg} = h_{fg} + \frac{3}{8} c_{p,l}\ (T_{sat} - T_w) \qquad \textbf{(7.18)}$$

Typically, heat transfer coeffiecients for dropwise condensation are an order of magnitude larger than those of film condensation. Data are available for condensation of steam on copper surface.
Near atmospheric pressure, the data are correlated by an expression of the form.

$$\bar{h} = 51,104\ +\ 2044\ T_w \qquad 22^oC < T_w < 100^oC \qquad \textbf{(7.19)}$$

$$\bar{h} = 25510 \qquad 100^oC < T_w \qquad \textbf{(7.20)}$$

Where h has a unit of $W/_{m^2 k}$