Heat Transfer

Condensation

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Introduction

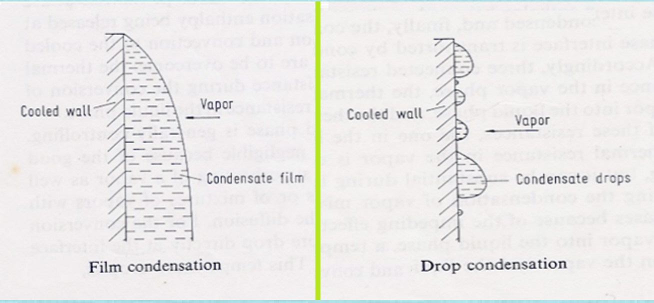
Heat Transfer in Condensation

There are different resistances to heat transfer in condensation:

- vapour phase resistance (convection and diffusion)
- inter phase resistance
- liquid phase resistance

The liquid phase resistance is controlling in most of the cases.

Types of Condensation



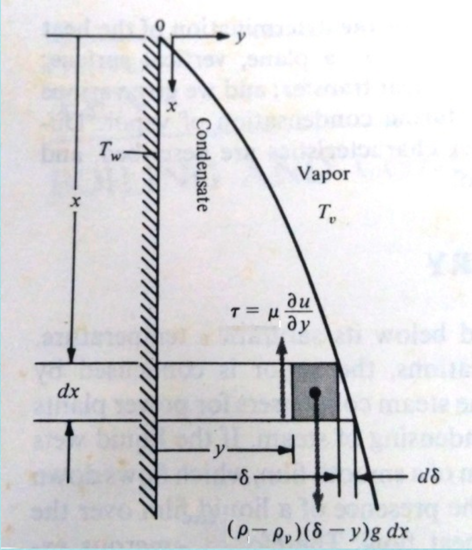
- **Film condensation** This is the most common type where the liquid film is formed which falls under gravity and always wets the surface.
- **Drop wise condensation** In this case wall is not wetted completely hence it depends on the wetting behaviour of the surface.

Nusselt Theory of Laminar Film Condensation

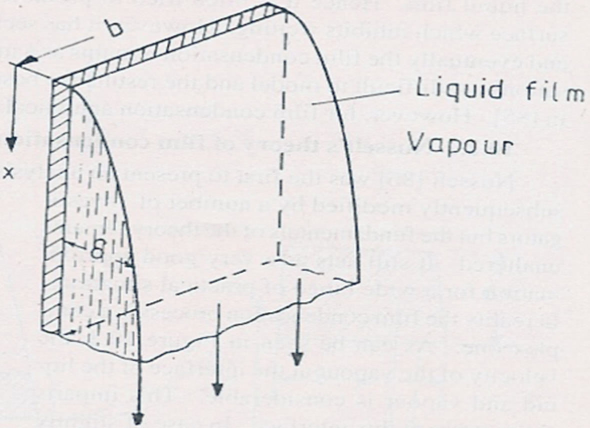
Nusselt made the following assumptions, in deriving the expression for heat transfer coefficient for condensation:

- The vapour is at rest and does not offer any shear at the interface.
- The flow of liquid is only by gravity and viscosity (no shear at the interface).
- The acceleration of liquid film (inertia) is negligible.
- Heat transfer is by pure conduction across the film.
- Properties are constant across the film and there is no interface resistance.
- The vapour is at saturation temperature.

Condensation on Vertical Surfaces



Condensation on Vertical Surfaces (contd..)



The condensate film flowing down a vertical cold flat plate

By force balance, shear force acting upward due to flow of fluid in the downward direction is equal to the net gravitational force.

$$\tau\, dx = (\rho - \rho_v)(\delta - y) dx\, g$$

Using Newton's law of viscosity,

$$\mu \frac{du}{dy}\, dx = (\rho - \rho_v)(\delta - y) g\, dx$$

At the wall surface, the liquid velocity is zero:

$$u = 0 \qquad \text{at } y = 0$$

Using this, and integrating the above, we get

$$u(y) = \frac{g(\rho - \rho_v)}{\mu}\left(\delta y - \frac{1}{2}y^2\right)$$

where δ is the film thickness at any x.

By definition, mass flow rate = density $\times$ volumetric flow rate. Therefore, for a unit width of plate, mass flow rate $m(x)$ through the position x is

$$m(x) = \int_0^{\delta} \rho\, u dy$$

We know,

$$u(y) = \frac{g(\rho - \rho_v)}{\mu}\left(\delta y - \frac{1}{2}y^2\right)$$

Therefore,

$$\begin{aligned} m(x) &= \int_0^{\delta} \rho \frac{g(\rho - \rho_v)}{\mu}\left(\delta y - \frac{1}{2}y^2\right) dy \\ &= \frac{\rho g(\rho - \rho_v)}{\mu}\left[\frac{\delta y^2}{2} - \frac{1}{6}y^3\right]_0^{\delta} \\ &= \frac{\rho g(\rho - \rho_v)}{\mu}\left[\frac{\delta \cdot \delta^2}{2} - \frac{1}{6}\delta^3\right] = \frac{\rho g(\rho - \rho_v)}{\mu}\left[\frac{\delta^3}{2} - \frac{\delta^3}{6}\right] \end{aligned}$$

$$\Longrightarrow \qquad m(x) = \frac{\rho g(\rho - \rho_v)\delta^3}{3\mu}$$

Differentiating the above, with respect to δ, we get

$$\frac{dm}{d\delta} = \frac{3\rho g(\rho - \rho_v)\delta^2}{3\mu} = \frac{\rho g(\rho - \rho_v)\delta^2}{\mu} \tag{1}$$

The rate of heat transfer (dQ_{conden}) associated with rate of condensation dm is given by

$$dQ_{\text{conden}} = \lambda\, dm$$

For a film of unit width, and thickness dx, the rate of transfer by conduction(dQ_{conduct}), through the film is given by

$$dQ_{\text{conduct}} = k\frac{T_v - T_w}{\delta}dx$$

Since, rate of condensation of vapor is equal to the rate of conduction inside the condensate film, we get

$$\begin{aligned} dQ &= dQ_{\text{conden}} = dQ_{\text{conduct}} \\ \implies \quad \lambda\, dm &= k\frac{T_v - T_w}{\delta} dx \\ \frac{dm}{dx} &= k\frac{T_v - T_w}{\delta} \end{aligned} \tag{2}$$

Eqn.(2) / Eqn.(1) gives

$$\frac{d\delta}{dx} = \frac{\mu k (T_v - T_w)}{g\rho(\rho - \rho_v)\lambda} \frac{1}{\delta^3} \tag{3}$$

Integrating Eqn.(3), with the condition $\delta = 0$ for $x = 0$ gives,

$$\delta(x) = \left[\frac{4\mu k (T_v - T_w) x}{g\rho(\rho - \rho_v)\lambda}\right]^{1/4} \tag{4}$$

Since, we have established the relation for the thickness of the condensate layer, the local heat transfer coefficient h_x, for condensation is determined from the definition:

$$h_x(T_v - T_w) = k\frac{T_v - T_w}{\delta(x)}$$

$$h_x = \frac{k}{\delta(x)}$$

Using Eqn.(4) in above, we get

$$h_x = \left[\frac{g\rho(\rho - \rho_v)\lambda k^3}{4\mu(T_v - T_w)x}\right]^{1/4} \tag{5}$$

From the above expression, we could note that the local heat transfer coefficient h_x varies with the distance as $x^{-1/4}$. The average heat transfer coefficient h_m is given by

$$h_m = \frac{1}{L}\int_0^L h_x\, dx = \frac{4}{3}h_x\Big|_L \tag{5}$$

Using Eqn.(4) in (5) we get,

$$\boxed{h_m = 0.943 \left[\frac{g\rho(\rho - \rho_v)\lambda k^3}{\mu(T_v - T_w)L}\right]^{1/4}}$$

In the above equation ρ, μ and k are properties of liquid.
For using the above equation for inclined plates, g is replaced by $g\cos\theta$, where θ is the angle between the vertical and the surface. However, it must be used with caution for larger values of θ and does not apply if $\theta = 90^\circ$. The expression may be used for condensation on the inner or outer surface a vertical tube of radius R, if $R \gg \delta$.
The physical properties in the above equation including λ are evaluated at the film temperature T_f

$$T_f = \frac{T_w + T_v}{2}$$

Condensation on a Horizontal Tube

$$\boxed{h_m = 0.729\left[\frac{g\rho(\rho-\rho_v)\lambda k^3}{\mu(T_v - T_w)D}\right]^{1/4}}$$

where D is the outside diameter of tube.

For condensation over a sphere of diameter D, the coefficient 0.729 is replaced with 0.826.

Comparing equations of horizontal and vertical condensation, we get

$$\frac{h_{m,\text{vert}}}{h_{m,\text{horiz}}} = 1.30 \left(\frac{D}{L}\right)^{1/4}$$

For $L = 2.87\ D$,

$$\frac{h_{m,\text{horiz}}}{h_{m,\text{vert}}} = 1$$

For $L = 100D$,

$$\frac{h_{m,\text{horiz}}}{h_{m,\text{vert}}} = 2.44$$

With this consideration, horizontal tube arrangements are generally preferred to vertical tube arrangements in condenser design.

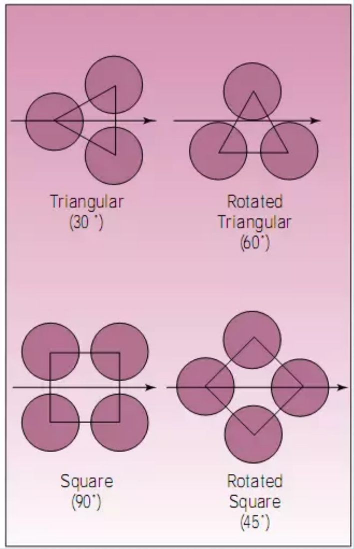
Condensation over Horizontal Tube Banks

For a vertical tier of N horizontal tubes, the relation between heat transfer coefficient for the one tube (h_D) and the average heat transfer coefficient over the N tubes ($h_{D,N}$) is given as

$$h_{D,N} = \frac{h_D}{N^{1/4}}$$

Such an arrangement is often used in condenser design.
A reduction in the heat transfer coefficient with increasing N may be attributed to an increase in the average film thickness for each successive tube due to accumulation of drip from the upper tubes. Obviously it is advantageous to stagger the tubes as the accumulation of drip from the upper rows is at least partially offset by the splashing effects, i.e., by the agitation caused by the drip as it falls from one tube to another. That is, instead of square pitch arrangement of tubes, triangular or rotated square pitch arrangement would be better.

Tubes Arrangement



Condensation Inside Horizontal Tube

$$\boxed{h_m = 0.555\left[\frac{g\rho(\rho-\rho_v)\lambda k^3}{\mu(T_v-T_w)D}\right]^{1/4}}$$

where D is the inside diameter of tube.

Horizontal Condensers are Better—Why?

Regardless of whether it is in the form of film or droplets, the condensate provides a resistance to heat transfer between the vapor and cold surface. Because this resistance increase with condensate thickness—which increases in the flow direction of condensate (i.e., downwards by gravity), it is desirable to use short vertical surfaces or horizontal cylinders.

Design Consideration of Condensers

Dropwise condensation is superior to filmwise condensation. In dropwise condensation most of the heat transfer is through drops of diameters of less than 100-μm and heat transfer rates are more than an order of magnitude larger than those with filmwise condensation.

To promote dropwise condensation, it is common practice to use surface coatings to inhibit wetting. Slicones, Teflon, waxes and fatty acids are often used for this purpose. However, such coatings gradually lose their effectiveness due to oxidation, fouling, or erosion, and filmwise condensation eventually occurs. For this reason, condenser design calculations are often based on the assumption of filmwise condensation.