Condensation

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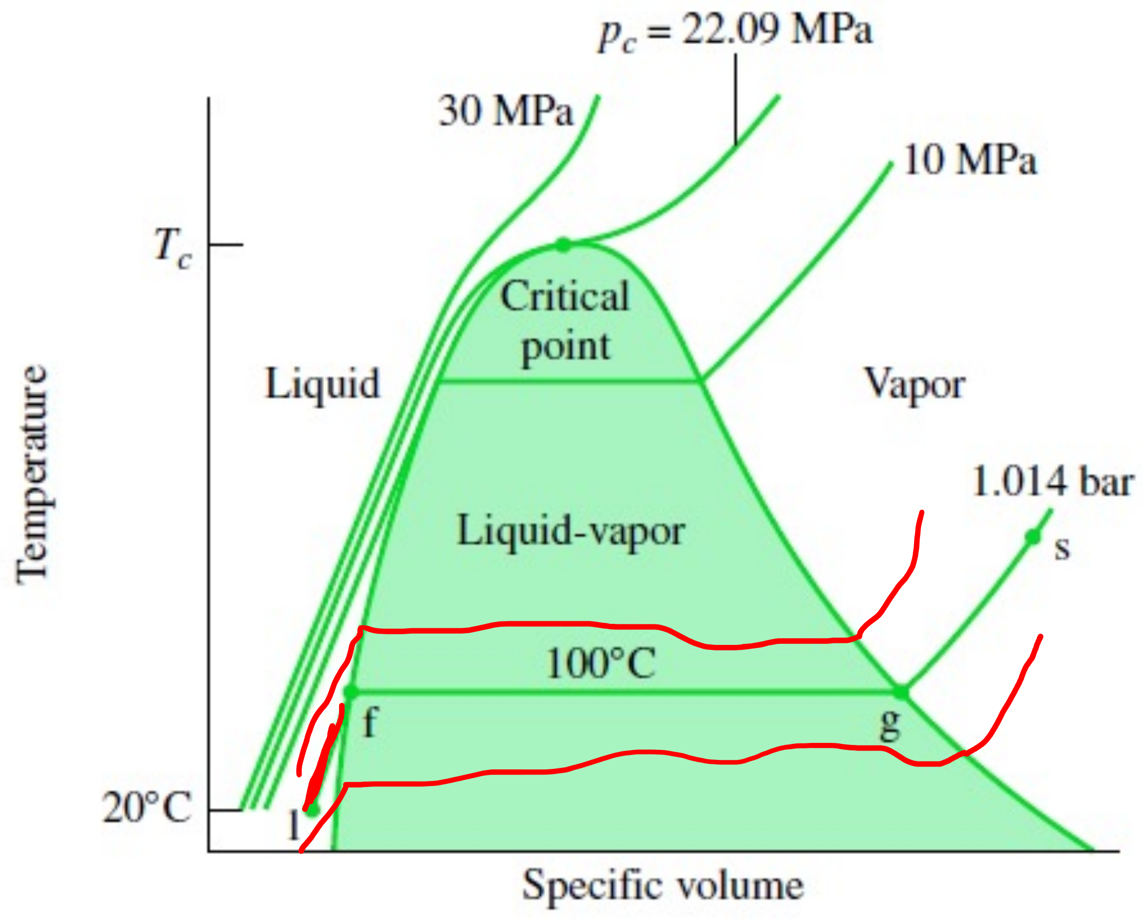
Condensation: Physical Mechanisms

- Kondensasi adalah proses kebalikan dari pendidihan / boiling.
- Kondensasi terjadi ketika suhu uap berkurang di bawah suhu saturasinya.
- Dalam peralatan industri, proses biasanya dihasilkan dari kontak antara uap dan permukaan yang dingin (Gambar 10.9a, b).
- Energi laten uap dilepaskan, panas dipindahkan ke permukaan, dan kondensat terbentuk.
- Mode umum lainnya adalah kondensasi homogen (Gambar 10.9c), di mana uap mengembun sebagai tetesan tersuspensi dalam fase gas untuk membentuk kabut, dan kondensasi kontak langsung (Gambar 10.9d), yang terjadi ketika uap dibawa ke dalam kontak dengan cairan dingin.



Sketsa diagram temperatur – volume spesifik untuk air yang menunjukkan cairan, dua fasa cairan-uap dan daerah uap

1-f = subcooled liquid karena temperatur pada keadaan ini kurang dari temperatur jenuh/saturasi pada tekanan yang diberikan

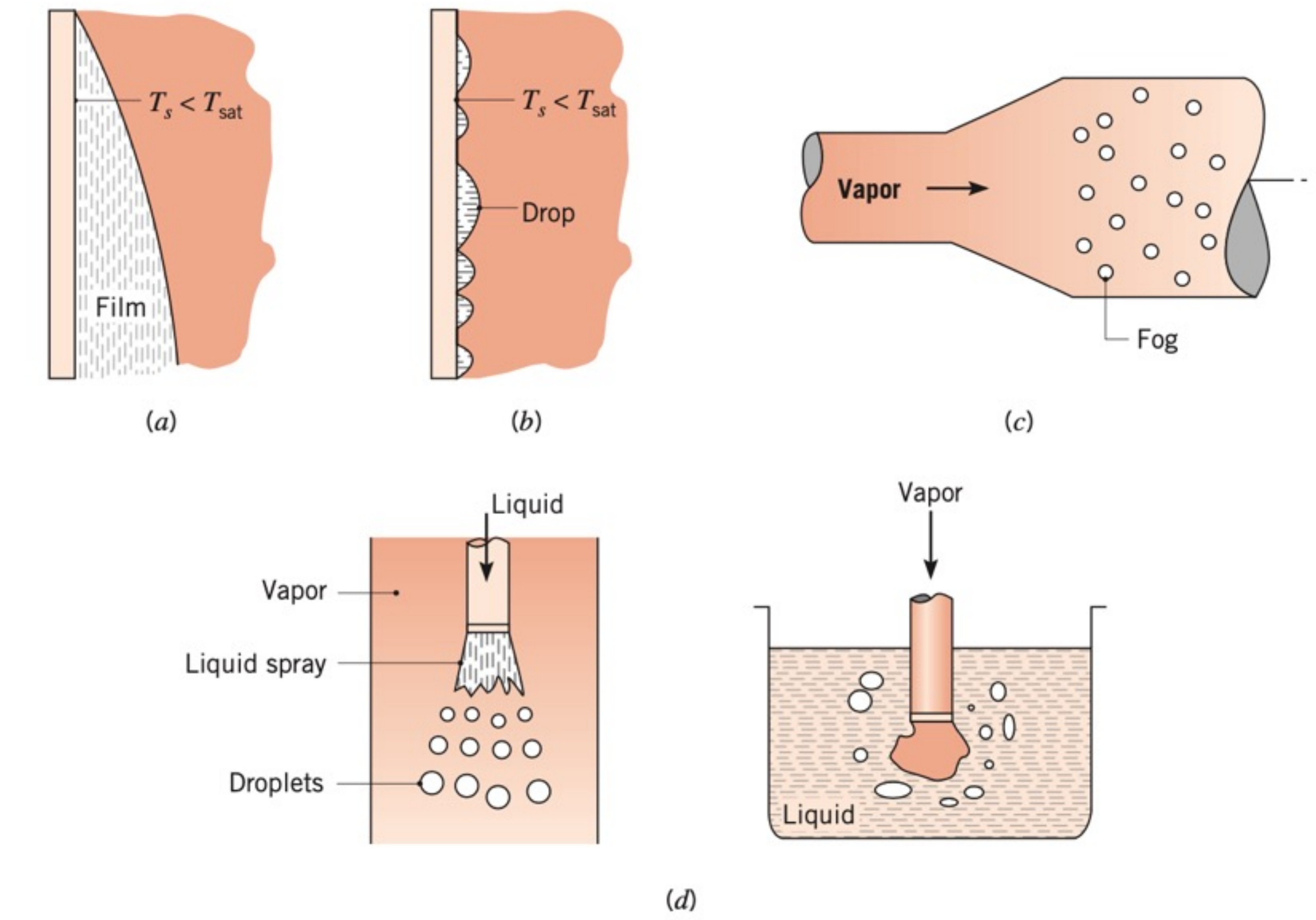
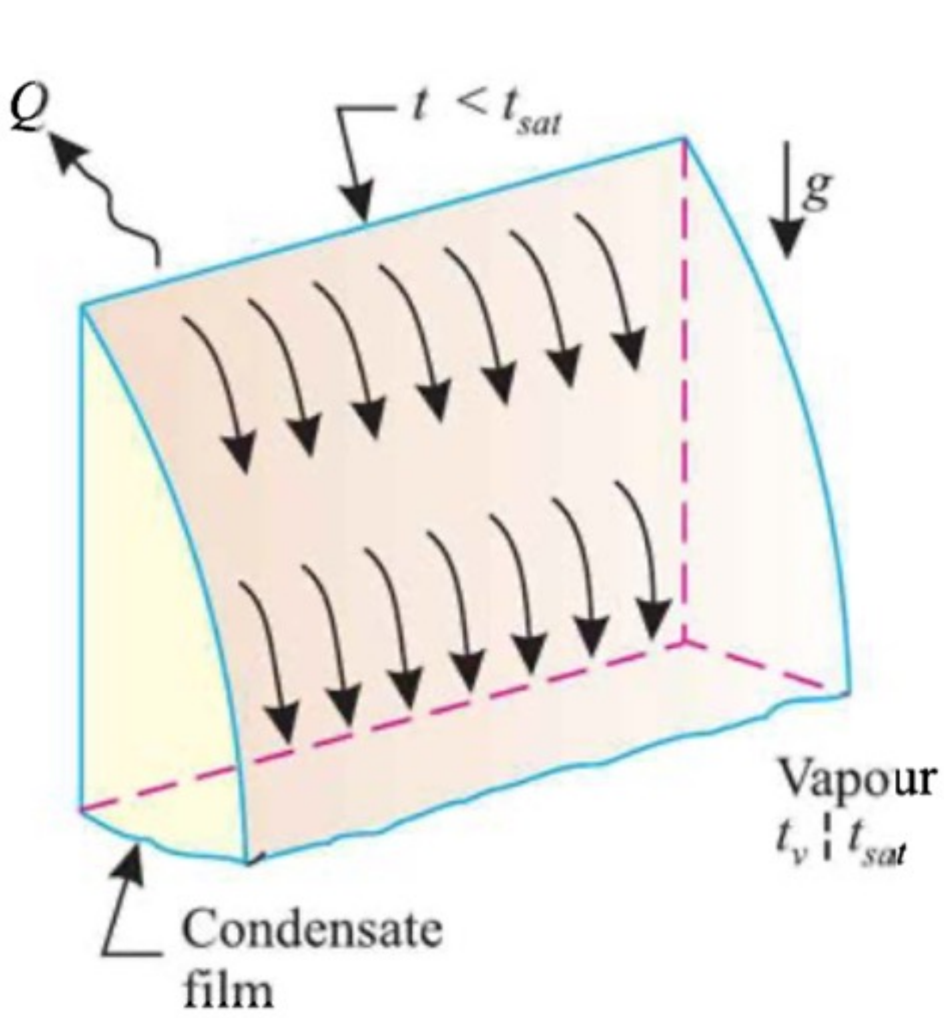
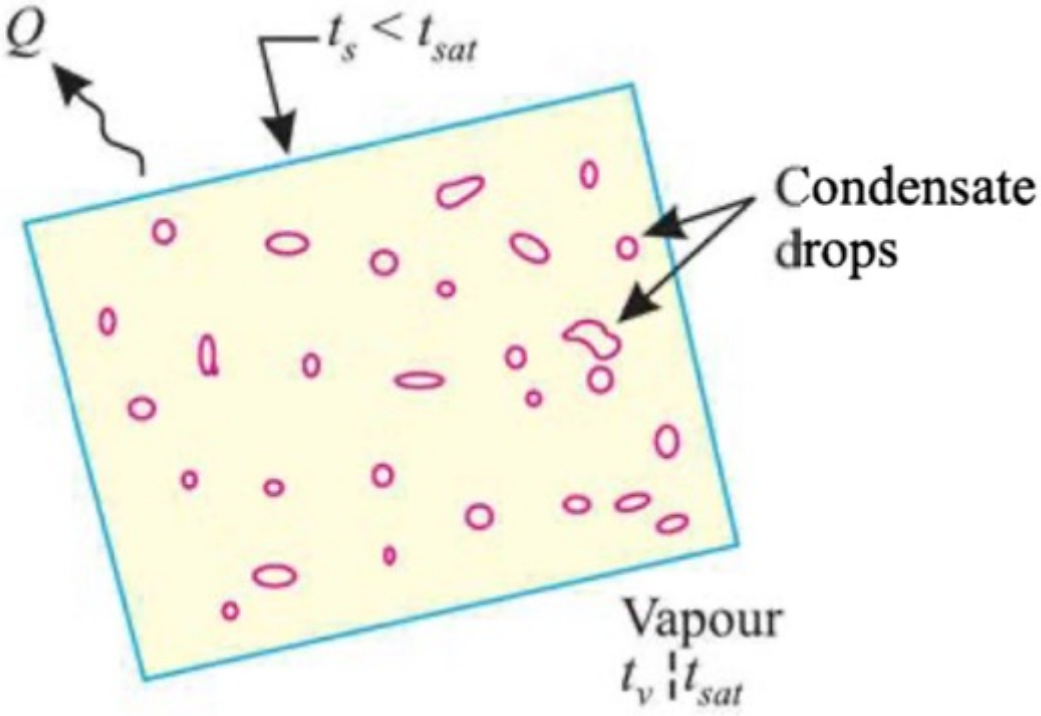


FIGURE 10.9 Modes of condensation. (*a*) Film. (*b*) Dropwise condensation on a surface. (*c*) Homogeneous condensation or fog formation resulting from increased pressure due to expansion. (*d*) Direct contact condensation.



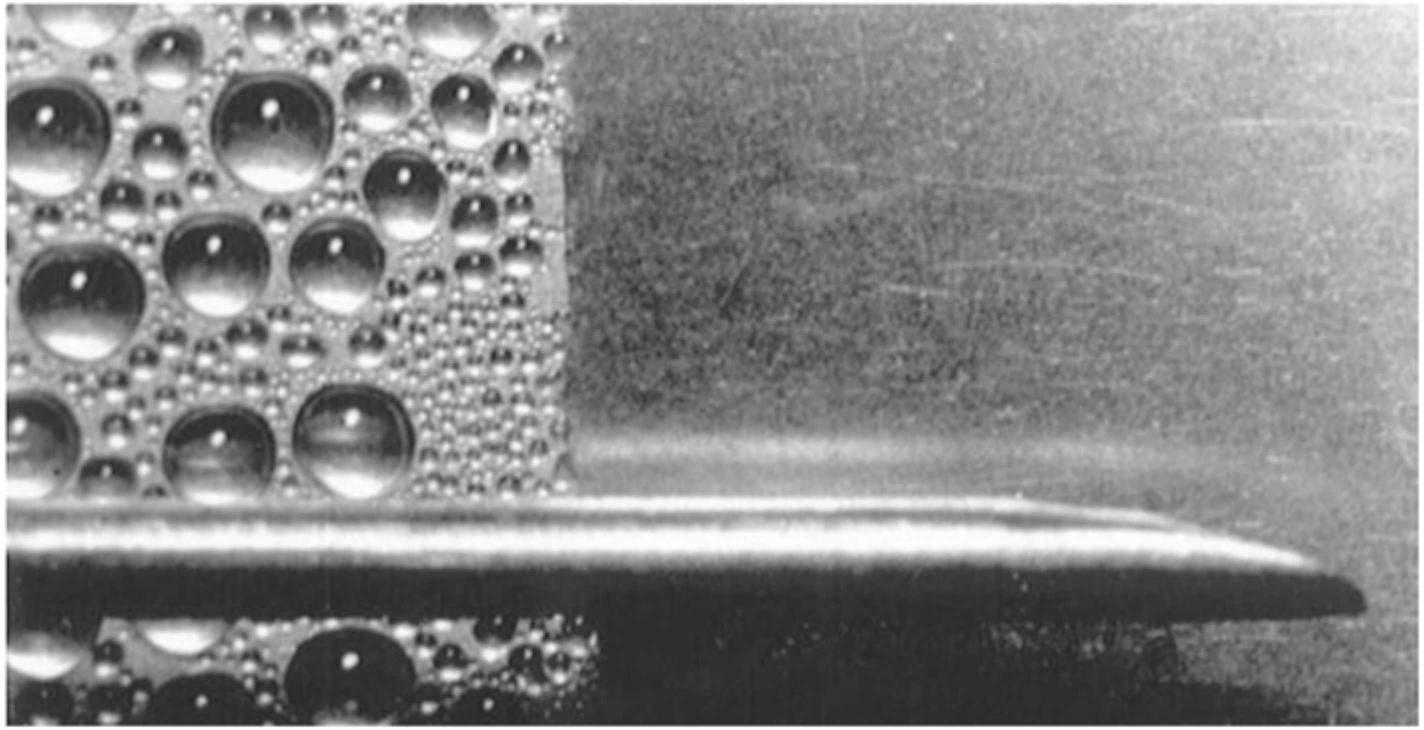
(*i*) Film condensation



(*ii*) Dropwise condensation

- Seperti ditunjukkan pada Gambar 10.9a, b, kondensasi dapat terjadi dalam salah satu dari dua cara, tergantung pada kondisi permukaan.
- Bentuk kondensasi yang dominan adalah di mana film cair menutupi seluruh permukaan kondensasi, dan di bawah aksi gravitasi film mengalir terus menerus dari permukaan. Kondensasi film umumnya merupakan karakteristik permukaan yang bersih dan tidak terkontaminasi.
- Namun, jika permukaan dilapisi dengan zat yang menghambat pembasahan, adalah mungkin untuk mempertahankan kondensasi tetes demi tetes. Tetesan terbentuk di retakan, lubang, dan rongga di permukaan dan dapat tumbuh dan menyatu melalui kondensasi lanjutan. Biasanya, lebih dari 90% permukaan ditutupi oleh tetesan, mulai dari diameter beberapa mikrometer hingga yang terlihat dengan mata telanjang. Tetesan mengalir dari permukaan karena aksi gravitasi.

- Film dan kondensasi tetesan uap pada permukaan tembaga vertikal ditunjukkan pada Gambar 10.10.
- Lapisan tipis plat tembaga diaplikasikan ke bagian kiri permukaan untuk memperlihatkan kondensasi tetes demi tetes. Sebuah probe termokopel dengan diameter 1 mm memanjang terlihat pada gambar.

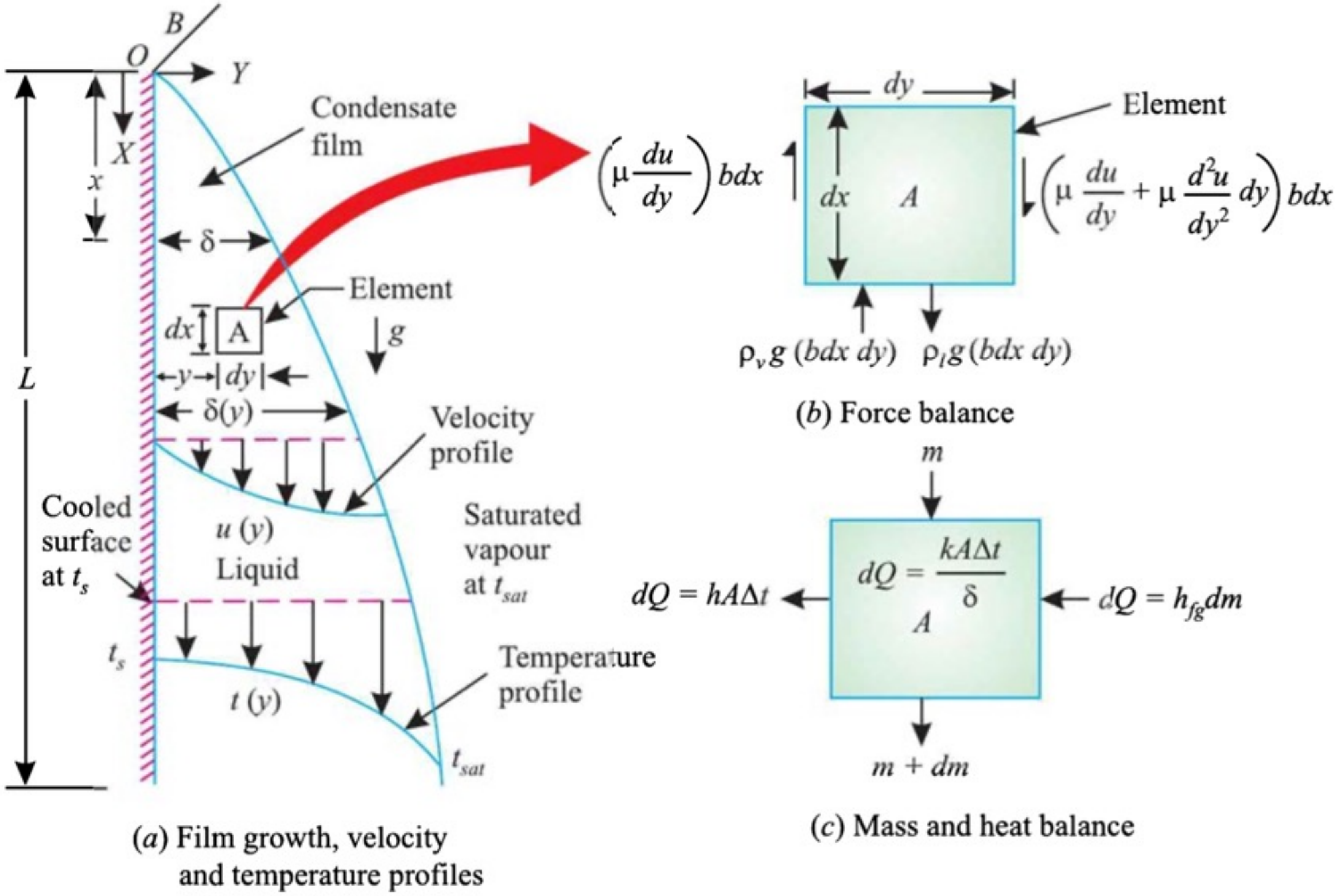


(*a*) (*b*)

FIGURE 10.10 Condensation on a vertical surface. (*a*) Dropwise. (*b*) Film. (Photograph courtesy of Professor J. W. Westwater, University of Illinois at Champaign-Urbana.)

- Terlepas dari apakah itu dalam bentuk film atau tetesan, kondensat memberikan ketahanan terhadap perpindahan panas antara uap dan permukaan. Karena resistensi ini meningkat dengan ketebalan kondensat, yang meningkat dalam arah aliran, perlu untuk menggunakan permukaan vertikal pendek atau silinder horizontal dalam situasi yang melibatkan kondensasi film.
- Oleh karena itu, sebagian besar kondensor terdiri dari bundel tabung horizontal yang melaluinya cairan pendingin mengalir dan di sekitarnya uap yang akan dikondensasikan disirkulasikan.
- Dalam hal mempertahankan laju kondensasi dan perpindahan panas yang tinggi, pembentukan tetesan lebih unggul daripada pembentukan film.
- Dalam kondensasi tetes, sebagian besar perpindahan panas adalah melalui tetesan dengan diameter kurang dari 100 μm, dan laju transfer yang lebih dari urutan besarnya lebih besar daripada yang terkait dengan kondensasi film dapat dicapai.
- Oleh karena itu, praktik umum untuk menggunakan pelapis permukaan yang menghambat pembasahan, dan karenanya merangsang kondensasi tetes demi tetes. Silikon, Teflon, dan bermacam-macam lilin dan asam lemak sering digunakan untuk tujuan ini.

Kondensasi Laminar pada Plat Vertikal



l = tinggi plat vertical
b = lebar
δ = tebal film pada jarak x

ρ_l = Density of liquid film,
ρ_v = Density of vapour,
h_{fg} = Latent heat of condensation,
k = Conductivity of liquid film,
μ = Absolute viscosity of liquid film,
t_s = Surface temperature, and

t_{sat} = temperatur saturasi/jenuh dari uap pada tekanan yang berlaku.

Analisa dari Nusselt terhadap kondensasi film dengan asumsi sebagai berikut:

- Film dari cairan mengalir karena pengaruh gravitasi.
- Aliran kondensat adalah laminar dan sifat-sifat fluida adalah konstan.
- Lapisan film berada pada kontak termal dengan permukaan yang dingin oleh karena itu temperatur pada bagian dalam film sama dengan temperatur permukaan (t_s). Temperatur liquid-vapor interface sama dengan temperatur saturasi/jenuh (t_{sat}) pada tekanan yang berlaku.
- Gaya geser viscous dan gaya gravitasi diasumsikan bekerja pada fluida, jadi gaya normal viscos dan gaya inersia diabaikan.
- Tegangan geser pada liquid-vapor interface diabaikan. Hal ini berarti tidak ada gradien kecepatan pada liquid-vapor interface.
- Perpindahan kalor pada lapisan kondensat adalah murni konduksi dan distribusi temperatur adalah linier.
- Uap kondensasi sepenuhnya bersih dan bebas dari gas, udara dan kotoran-kotoran yang tidak terkondensasi.
- Radiasi antara lapisan film uap dan cairan; komponen horizontal dari kecepatan pada setiap titik pada lapisan film cairan; kurva lapisan film adalah dianggap sangat kecil.

(*a*) Velocity distribution :

To find an expression for velocity distribution u as a function of distance y from the wall surface, let us consider the equilibrium between gravity and viscous forces on an elementary volume ($b\,dx\,dy$) of the liquid film.

Gravitational force on the element = $\rho_l g\,(b\,dx\,dy) - \rho_v g\,(b\,dx\,dy)$ ...(*i*)

Viscous shear force on the element

$$= \mu \frac{du}{dy}(b\,dx) - \left(\mu \frac{du}{dy} + \mu \frac{d^2u}{dy^2}dy\right)(b\,dx) \qquad \text{...}(ii)$$

Equating (*i*) and (*ii*), we get

$$\rho_l g\,(b\,dx\,dy) - \rho_v g\,(b\,dx\,dy) = \mu \frac{du}{dy}(b\,dx) - \left(\mu \frac{du}{dy} + \mu \frac{d^2u}{dy^2}dy\right)(b\,dx)$$

$$\frac{d^2u}{dy^2} = -\frac{(\rho_l - \rho_v)\,g}{\mu} \qquad \text{...(9.16)}$$

Upon integration, we have

$$\frac{du}{dy} = -\frac{(\rho_l - \rho_v)\,g}{\mu}\,y + C_1$$

Integrating again, we get

$$u = -\frac{(\rho_l - \rho_v)\,(y^2/2)\,g}{\mu} + C_1 y + C_2$$

The relevant boundary conditions are:

At $y = 0$, $u = 0$

At $y = \delta$, $\frac{du}{dy} = 0$

Using these boundary conditions, we get the following values of C_1 and C_2 :

$$C_1 = \frac{(\rho_l - \rho_v)\, g\, \delta}{\mu}, \text{ and } C_2 = 0$$

Substituting the values of C_1 and C_2 we get the velocity profile

$$u = \frac{(\rho_l - \rho_v)\, g}{\mu}\left[\delta\, y - \frac{y^2}{2}\right] \quad \text{...(9.17)}$$

or,

$$u = \frac{(\rho_l - \rho_v)\, g \cdot \delta^2}{\mu}\left[\frac{y}{\delta} - \frac{1}{2}\left(\frac{y}{\delta}\right)^2\right] \quad \text{...(9.18)}$$

Equation (9.18) is the *required* ***velocity profile***.

The mean flow velocity u_{mean} of the liquid film at a distance y is given by

$$u_m = \frac{1}{\delta}\int_0^\delta u\, dy$$

$$= \frac{1}{\delta}\int_0^\delta \frac{(\rho_l - \rho_v)\, g \cdot \delta^2}{\mu}\left[\frac{y}{\delta} - \frac{1}{2}\left(\frac{y}{\delta}\right)^2\right] dy$$

or,

$$u_m = \frac{(\rho_l - \rho_v)\, g \cdot \delta^2}{3\,\mu} \quad \text{...(9.19)}$$

(b) Mass flow rate :

The mass flow rate of condensate through any x position of the film is given by :

Mass flow rate (m) = Mean flow velocity (u_m) × flow area × density

or, $$m = \frac{(\rho_l - \rho_v)\, g \cdot \delta^2}{3\,\mu} \times b\,.\,\delta \times \rho_l = \frac{\rho_l\,(\rho_l - \rho_v)\, g\,.\, b\,.\delta^3}{3\,\mu} \quad \text{...(9.20)}$$

The mass flow is thus a function of x; this is so because the film thickness δ is essentially dependent upon x.

As the flow proceeds from x to $(x + dx)$ the film grows from δ to $(\delta + d\delta)$ because of additional condensate. The mass of condensate added between x and $(x + dx)$ can be worked out by differentiating eqn. (9.20) with respect to x (or δ).

$$dm = \frac{d}{dx}\left[\frac{\rho_l\,(\rho_l - \rho_v)\, g\,.\, b\,.\delta^3}{3\,\mu}\right] \cdot dx$$

$$= \frac{d}{d\,\delta}\left[\frac{\rho_l\,(\rho_l - \rho_v)\, g\,.\, b\,.\delta^3}{3\,\mu}\right] \frac{d\delta}{dx}\, dx$$

or, $$dm = \left[\frac{\rho_l\,(\rho_l - \rho_v)\, g\,.\, b\,.\delta^2}{\mu}\right] d\delta \quad \text{...(9.21)}$$

(*c*) **Heat flux :**

The heat flow rate into the film (dQ) equals the rate of energy release due to condensation at the surface. Thus,

$$dQ = h_{fg} \cdot dm = h_{fg} \left[\frac{\rho_l (\rho_l - \rho_v) g \cdot b \delta^2}{\mu} \right] d\delta \quad ...(9.22)$$

According to our assumption the heat transfer across the condensate layer is by pure conduction, hence

$$dQ = \frac{k (b\ dx)}{\delta} (t_{sat} - t_s) \quad ...(9.23)$$

Combining eqns. (9.22) and (9.23), we have

$$\frac{h_{fg}\ \rho_l (\rho_l - \rho_v) g . b \delta^2}{\mu} \cdot d\delta = \frac{k (b\ dx)}{\delta} (t_{sat} - t_s)$$

or, $$\delta^3 . d\delta = \frac{k \mu}{\rho_l (\rho_l - l_v) g\ h_{fg}} (t_{sat} - t_s)\ dx$$

Integrating the above equation, we get

$$\frac{\delta^4}{4} = \frac{k \mu}{\rho_l (\rho_l - \rho_v) g\ h_{fg}} (t_{sat} - t_s) x + C_1$$

Substitution of the boundary condition : $\delta = 0$ at $x = 0$ yields $C_1 = 0$. Hence

$$\delta = \left[\frac{4 k \mu (t_{sat} - t_s) x}{\rho_l (\rho_l - \rho_v) g\ h_{fg}} \right]^{1/4} \quad ...(9.24)$$

The equation (9.24) depicts that the heat film thickness increases as the *fourth root* of the distance down the surface; the increase is rather rapid at the upper end of the vertical surface and slows thereafter.

(*d*) Film heat transfer coefficient :

According to Nusselt assumption the heat flow from the vapour to the surface is by conduction through the liquid film. Thus

$$dQ = \frac{k\,(b\,dx)}{\delta}(t_{sat} - t_s) \qquad ...(i)$$

The heat flow can also be expressed as

$$dQ = h_x (b\,dx)\,(t_{sat} - t_s) \qquad ...(ii)$$

where h_x is the local heat transfer coefficient.

From (*i*) and (*ii*), we get

$$\frac{k\,(b\,dx)}{\delta}(t_{sat} - t_s) = h_x\,(b\,dx)\,(t_{sat} - t_s)$$

or,

$$h_x = \frac{k}{\delta} \qquad ...(9.25)$$

Equation (9.25) depicts that at a definite point on the heat transfer surface, the film coefficient h_x is *directly* proportional to *thermal conductivity* k and *inversely* proportional to *thickness of film* δ at that point.

Substituting the value of δ from equation (9.24), we get

$$h_x = \left[\frac{\rho_l\,(\rho_l - \rho_v)\,k^3\,g\,h_{fg}}{4\,\mu x\,(t_{sat} - t_s)}\right]^{1/4} \qquad ...(9.26)$$

Local heat transfer coefficient at the lower end of the plate, *i.e.*, $x = L$

$$h_L = \left[\frac{k^3\,\rho^2\,g\,h_{fg}}{4\,\mu\,L(t_{sat} - t_s)}\right]^{1/4} \qquad ...(9.27)$$

Evidently the *rate of condensation heat transfer is higher at the upper end of the plate than that at the lower end.*

The average value of heat transfer can be obtained by integrating the local value of coefficient [Eqn. (9.26)] as follows :

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

$$= \frac{1}{L}\int_0^L \left[\frac{\rho_l (\rho_l - \rho_v) k^3 g h_{fg}}{4 \mu x (t_{sat} - t_s)}\right]^{1/4} dx$$

$$= \frac{1}{L}\left[\frac{\rho_l (\rho_l - \rho_v) k^3 g h_{fg}}{4 \mu (t_{sat} - t_s)}\right]^{1/4} \int_0^L x^{-1/4} \, dx$$

$$= \frac{1}{L}\left[\frac{\rho_l (\rho_l - \rho_v) k^3 g h_{fg}}{4 \mu (t_{sat} - t_s)}\right]^{1/4} \left[\frac{x^{(-1/4+1)}}{-1/4+1}\right]_0^L$$

or,
$$\bar{h} = \frac{4}{3}\left[\frac{\rho_l (\rho_l - \rho_v) k^3 g h_{fg}}{4 \mu L (t_{sat} - t_s)}\right]^{1/4} \quad ...(9.28)$$

$$\bar{h} = \frac{4}{3} h_L \left(= \frac{4}{3} \times \frac{k}{\delta_L}\right)$$

where h_L is the local heat transfer coefficient *at the lower edge of the plate*.

This shows that the average heat transfer coefficient is $\frac{4}{3}$ times the local heat transfer coefficient at the trailing edge of plate.

Equation 9.28 is usually written in the form

$$\bar{h} = 0.943 \left[\frac{\rho_l (\rho_l - \rho_v) k^3 g h_{fg}}{\mu L (t_{sat} - t_s)} \right]^{1/4} \quad ...(9.29)$$

The Nusselt solution derived above is an approximate one because experimental results have shown that it yields results which are approximately *20 percent* lower than the measured values. McAdams proposed to use a value of 1.13 in place of coefficient 0.943. Hence

$$\bar{h} = 1.13 \left[\frac{\rho_l (\rho_l - \rho_v) k^3 g h_{fg}}{\mu L (t_{sat} - t_s)} \right]^{1/4} \quad ...(9.30)$$

While using above equation, it may be noted that, all liquid properties are to be evaluated at the temperature $\left(\frac{t_{sat} + t_s}{2} \right)$ and h_{fg} should be evaluated at t_{sat}.

The total heat transfer to the surface,

$$Q = h A_s (t_{sat} - t_s) \quad ...(9.31)$$

The total condensation rate,

$$m = \frac{Q}{h_{fg}} = \frac{h A_s (t_{sat} - t_s)}{h_{fg}} \quad ...(9.32)$$

Figure 9.8. shows the variation of film thickness and film coefficient with plate height [graphical representation of eqns. (9.24), (9.26) and (9.28)]. The film thickness increases with the increase of plate height. Heat transfer rate decreases with the increase of plate height since thermal resistance increases with the film thickness.

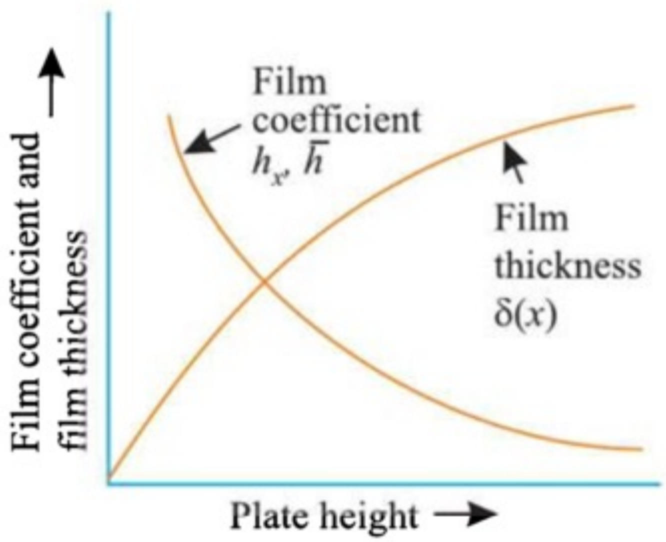


Fig. 9.8. Film thickness and film coefficient vs. plate height.

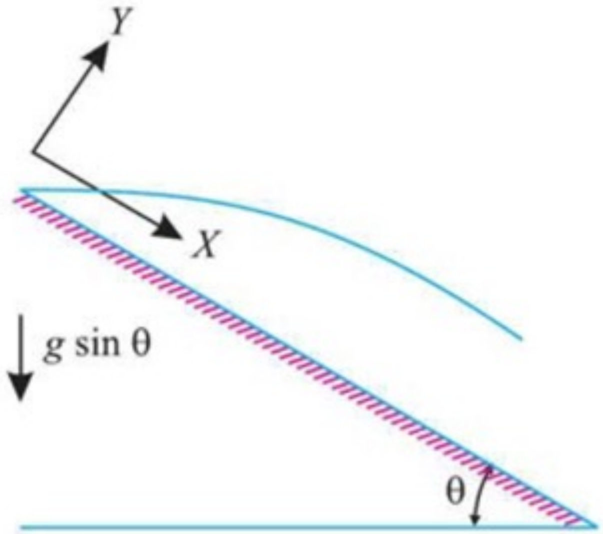


Fig. 9.9. Condensation on an inclined surface.

(*e*) Inclined flat plate surface

For inclined flat surfaces, the gravitational acceleration g in equation (9.30) is replaced by $g \sin \theta$ where θ is the angle between the surface and horizontal (Refer Fig. 9.9). The Eqn. (9.30) is modified as :

$$h_{inclined} = 1.13 \left[\frac{\rho_l (\rho_l - \rho_v) k^3 (g \sin \theta) h_{fg}}{\mu L (t_{sat} - t_s)} \right]^{1/4} \quad ...(9.33)$$

or, $$h_{inclined} = h_{vertical} \times (\sin \theta)^{1/4} \quad ...(9.34)$$

Equation (9.34) is applicable only for cases where θ is *small*; is not at all applicable for horizontal plate.

9.3.3. TURBULENT FILM CONDENSATION

When the plate on which condensation occurs is *quite long* or when the *liquid film is vigorous enough*, the condensate flow may become turbulent. The *turbulent results in higher heat transfer rates because heat is now transferred not only by condensation but also by eddy diffusion*. The transition criterion may be expressed in terms of Reynolds number defined as,

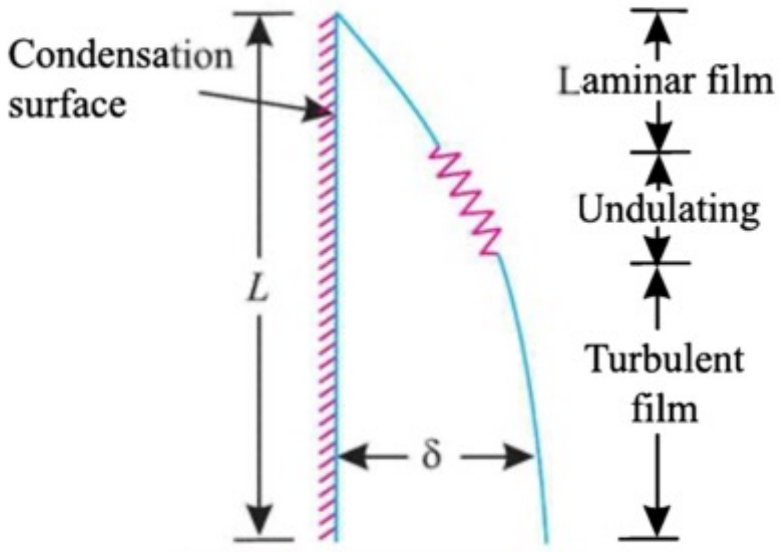


Fig. 9.10. Regions of film condensation on a vertical surface.

$$Re = \frac{\rho_l \; u_m \; D_h}{\mu_l}$$

where, D_h = Hydraulic diameter

$$= 4 \times \frac{\text{cross-sectional area of fluid flow}}{\text{wetted perimeter}} = \frac{4A}{P}, \text{ and}$$

u_m = Mean or average velocity of flow.

$$Re = \frac{\rho_l \, \mu_m \times 4\, A_c}{P \times \mu_l} = \frac{4\; m}{P\; \mu_l} \qquad ...(9.35)$$

where, $m = \rho A \; u_m$

For a vertical plate of unit depth, $P = 1$, the Reynolds number is sometimes expressed in terms of the mass flow rate per unit depth of plate Γ, so that

$$Re = \frac{4\,\Gamma}{\mu_l} \qquad ...(9.36)$$

with $\Gamma = 0$, at the top of the plate and Γ increasing with x.

The Reynolds number may also be related to heat transfer coefficient as follows:

$$Q = \bar{h} \; A_s (t_{sat} - t_s) = \dot{m} \; h_{fg}$$

or,

$$\dot{m} = \frac{Q}{h_{fg}} = \frac{\bar{h} \; A_s \; (t_{sat} - t_s)}{h_{fg}}$$

or,

$$Re = \frac{4\bar{h} \; A_s \; (t_{sat} - t_s)}{h_{fg} \; P\mu_l} \quad ...(9.37)$$

For the plate, $A = L \times B$ and $P = B$, where L and B are height and width of plate, respectively. Thus,

$$Re = \frac{4\bar{h} \; L \, (t_{sat} - t_s)}{h_{fg} \, \mu_l} \quad ...(9.38)$$

When the value of *Re* exceeds 1800 (approximately), the turbulence will appear in the liquid film. For $Re > 1800$, the following correlation is used:

$$\bar{h} \; (= h_{turb}) = 0.0077 \left[\frac{\rho_l \; (\rho_l - \rho_v) k^3 \; g}{\mu_l^2} \right]^{1/3} (R_l)^{0.4} \quad ...(9.39)$$

Kondensasi Film pada Pipa Horisontal

Nusselt's analysis for laminar filmwise condensation on horizontal tubes leads to the following relations:

$$\bar{h} = 0.0725 \left[\frac{\rho_l (\rho_l - \rho_v) k^3 \, g \, h_{fg}}{\mu_l (t_{sat} - t_s) D} \right]^{1/4} \quad \text{...(9.40)}$$

... For *single horizonal tube.*

$$\bar{h} = 0.0725 \left[\frac{\rho_l (\rho_l - \rho_v) k^3 \, g \, h_{fg}}{N \mu_l (t_{sat} - t_s) D} \right]^{1/4} \quad \text{...(9.41)}$$

...For *horizontal tube bank* with N tubes placed directly over one another in the vertical direction.

where, D = Outer diameter of the tube.

Kondensasi Film dalam Pipa Horisontal

Condensation of vapour inside the tubes finds several engineering applications such as condensers used in refrigeration and air-conditioning systems and several chemical and petrochemical industries. The phenomena insides tubes are very complicated *because the overall flow rate of vapour strongly affects the heat transfer rate and also the rate of condensation on the walls.*

Chato (1962) has recommended the following correlation for low velocities inside horizontal tubes (condensation of refrigerants):

$$\bar{h} = 0.555 \left[\frac{\rho_l (\rho_l - \rho_v) k^3 g h'_{fg}}{\mu_l D (t_{sat} - t_s)} \right]^{1/4} \quad ...(9.42)$$

where, $$h'_{fg} = h_{fg} + \frac{3}{8} c_{pl} (t_{sat} - t_s) \quad ...(9.43)$$

Equation (9.43) is restricted to low vapour Reynolds number such that

$$Re_v = \left(\frac{\rho_v u_{m,v} D}{\mu_v} \right) < 3500$$

where Re_v is evaluated at inlet conditions to the tubes.

Example 9.8. *Saturated steam at $t_{sat.}$ = 90°C (p = 70.14 k Pa) condenses on the outer surface of a 1.5 m long 2.5 m OD vertical tube maintained at a uniform temperature T_∞ = 70°C. Assuming film condensation, calculate :*

(i) The local transfer coefficient at the bottom of the tube, and

(ii) The average heat transfer coefficient over the entire length of the tube.

Properties of water at 80° C are : $\rho_l = 974\ kg/m^3$, $k_t = 0.668$ W/m K, $\mu_l = 0.335 \times 10^3$ kg/ms, h_{fg} = 2309 kJ/kg, $\rho_v << \rho_l$. **(AMIE Winter, 1998)**

Solution. *Given :* t_{sat} = 90°C (p = 70.14 kPa); L = 1.5 m;

D = 2.5 cm = 0.025 m; t_s = 70°C.

Properties of water at $80°\text{C}\left(t_f = \frac{90 + 70}{2} = 80°\text{C}\right)$; $\rho_l = 974\ \text{kg/m}^3$;

k = 0.668 W/m K; $\mu = 0.335 \times 10^3$ kg/ms; h_{fg} = 2309 kJ/kg ($\rho_v << \rho_l$)

(*i*) The local heat transfer coefficient, h_x :

With usual notations, the local heat transfer coefficient for film condensation is given as :

$$h_x = \left[\frac{\rho_l \, (\rho_l - \rho_v) \, k^3 \, g \, h_{fg}}{4 \, \mu x \, (t_{sat} - t_s)} \right]^{1/4} \qquad \text{...[Eqn. (9.26)]}$$

∴ Local heat transfer coefficient at the bottom of the tube, $x = 1.5$ m, is

$$h_L (= h_{1.5}) = \left[\frac{(974)^2 \times (0.668)^3 \times 9.81 \, (2309 \times 10^3)}{4 \times 0.335 \times 10^{-3} \times 1.5 \, (90 - 70)} \right]^{1/4} \qquad (\text{as } \rho_v << \rho_l)$$

$$= \left[\frac{6.4053 \times 10^{15})}{40.2} \right]^{1/4} = \mathbf{3552.9 \ W/m^2 \, °C} \qquad \textbf{(Ans.)}$$

(*ii*) Average heat transfer coefficient, $\bar{h}$:

$$\bar{h} = \frac{4}{3} h_L = \frac{4}{3} \times 3552.9 = \mathbf{4737.2 \ W/m^2 \, °C} \qquad \textbf{(Ans.)}$$

Example 9.9. *Saturated steam at 120°C condenses on a 2 cm OD vertical tube which is 20 cm long. The tube wall is maintained at a temperature of 119°C. Calculate the average heat transfer coefficient and the thickness of the condensate film at the base of the tube. Assume Nusselt's solution is valid. Given :*

$$p_{sat} = 1.985 \text{ bar}; \rho_w = 943 \text{ kg/m}^3 ; h_{fg} = 2202.2 \text{ kJ/kg};$$
$$k_w = 0.686 \text{ W/m K}; \mu = 237.3 \times 10^{-6} \text{ Ns/m}^2.$$

(AMIE Summer, 1997)

Solution. From Nusselt's solution, we have

$$\delta = \left[\frac{4k\mu\,(t_{sat.} - t_s)\,x}{\rho_l\,(\rho_l - \rho_v)\,g\,h_{fg}}\right]^{1/4} \quad \text{...[Eqn. (9.24)]}$$

or,
$$\delta_L = \left[\frac{4 \times 0.686 \times 237.3 \times 10^{-6} \times (120 - 119) \times 0.2}{(943)^2 \times 9.81 \times 2202.2 \times 10^3}\right]^{1/4},$$

neglecting ρ_v in comparison to ρ_l (or ρ_w)

$$= \left[\frac{0.0001302}{1.92 \times 10^{13}}\right]^{1/4} = 5.1 \times 10^{-5} \text{ m} \quad \text{or} \quad \mathbf{0.051 \text{ mm (Ans.)}}$$

Now,
$$h_L = \frac{k}{\delta_L} = \frac{0.686}{0.051 \times 10^{-3}} \simeq 13451$$

∴ Average heat transfer coefficient,

$$\bar{h} = \frac{4}{3}\,h_L = \frac{4}{3} \times 13451 = \mathbf{17934.67 \text{ W/m}^2\text{K}} \quad \text{(Ans.)}$$

Example 9.10. *A vertical cooling fin approximating a flat plate 40 cm in height is exposed to saturated steam at atmospheric pressure ($t_{sat.}$ = 100°C, h_{fg} = 2257 kJ/kg). The fin is maintained at a temperature of 90°C. Estimate the following :*

(i) *Thickness of the film at the bottom of the fin;*

(ii) *Overall heat transfer coefficient; and*

(iii) *Heat transfer rate after incorporating McAdam's correction.*

The relevant fluid properties are :

$$\rho_l = 965.3 \text{ kg/m}^3$$
$$k_l = 0.68 \text{ W/m°C}$$
$$\mu_l = 3.153 \times 10^{-4} \text{ N s/m}^2$$

The following relations may be used :

$$\delta_x = \left[\frac{4k_l\mu_l\,(t_{sat.} - t_g)x}{gh_{fg}\,\rho_l\,(\rho_l - \rho_v)}\right]^{1/4}$$

$$\bar{h} = \frac{4}{3}\frac{k}{\delta_L}$$

(AMIE, Summer, 2001)

Solution. *Given* : L = 60 cm = 0.6 m; $t_{sat.}$ = 100°C; h_{fg} = 2257 kJ/kg;
t_s = 90°C; ρ_l = 965.3 kg/m^3; k_l = 0.68 W/m°C;
μ_l = 3.153 × 10^{-4} N s/m^2

(*i*) Thickness of film at the bottom edge of the fin, δ_L :

$$\delta_x = \left[\frac{4k_l\mu_l\,(t_{sat.} - t_s)x}{gh_{fg}\,\rho_l\,(\rho_l - \rho_v)}\right]^{1/4} \quad \text{...(Given)}$$

or,

$$\delta_L = \left[\frac{4k_l\mu_l\,(t_{sat.} - t_s)\,L}{gh_{fg}\,\rho_l^2}\right]^{1/4}, \text{ as } \rho_l >> \rho_v$$

$$= \left[\frac{4 \times 0.68 \times 3.153 \times 10^{-4}\,(100 - 90) \times 0.4}{9.81 \times 2257 \times 10^3 \times (965.3)^2}\right]^{1/4} = \left[\frac{34.305 \times 10^{-4}}{2.063 \times 10^{13}}\right]^{1/4}$$

$= 0.0001136$ m $=$ **0.1136 mm** **(Ans.)**

(*ii*) Overall heat transfer coefficient, $\bar{h}$:

$$\bar{h} = \frac{4}{3}\frac{k_l}{\delta_L} = \frac{4}{3} \times \frac{0.68}{0.0001136} = \mathbf{7981.22\ W/m^2{}^\circ C} \quad \textbf{(Ans.)}$$

(*iii*) Heat transfer rate with McAdam's correction :

With McAdam's correction, the value of $\bar{h}$ is 20 percent higher. Hence heat transfer rate after incorporating McAdam's correction for unit width, is :

$$Q = 1.2 \times 7981.22 \times (0.4 \times 1) \times (100 - 90)$$

$= 38309.8$ W/m or **38.3098 kW per m width** **(Ans.)**

Example 9.11. *A vertical plate 500 mm high and maintained at 30°C is exposed to saturated steam at atmospheric pressure. Calculate the following:*

(i) The rate of heat transfer, and

(ii) The condensate rate per hour per metre of the plate width for film condensation.

The properties of water film at the mean temperature are:

$\rho = 980.3$ kg/m^3; $k = 66.4 \times 10^{-2}$ W/m°C; $\mu = 434 \times 10^{-6}$ kg/ms and $h_{fg} = 2257$ kJ/hg.

Assume vapour density is small compared to that of the condensate.

Solution. *Given*: $L = 500$ mm $= 0.5$ m; $B = 1$m; $t_s = 30$°C.

(*i*) The rate of heat transfer per metre width, *Q*:

$$\bar{h} = 0.943 \left[\frac{\rho_l (\rho_l - \rho_v) k^3 g h_{fg}}{\mu L (t_{sat.} - t_s)} \right]^{1/4} \quad \text{...[Eqn. (9.29)]}$$

$$= 0.943 \left[\frac{\rho_l^2 k^3 g h_{fg}}{\mu L (t_{sat.} - t_s)} \right]^{1/4} \quad \text{...neglecting } \rho_v [(\rho_v << \rho_l \text{ ...given})]$$

or,

$$\bar{h} = 0.943 \left[\frac{(980.3)^2 \times (66.4 \times 10^{-2})^3 \times 9.81 \times (2257 \times 10^3)}{434 \times 10^{-6} \times 0.5 (100 - 30)} \right]^{1/4}$$

$$= 0.943 \left[\frac{6.229 \times 10^{12})}{0.0152} \right]^{1/4} = 4242.8 \text{ W/m}^2 \text{°C}$$

$$\therefore \quad Q = \bar{h}\, A\, (t_{sat} - t_s) = h \times (L \times B)\, (t_{sat} - t_s)$$

$$= 4242.8 \times (0.5 \times 1)\,(100 - 30) = 148498 \text{ W}$$

$$= \frac{148498 \times 3600}{1000} = \mathbf{534.59 \times 10^3 \text{ kJ/h}}$$

(*ii*) The condensate rate per meter width, *m*:

$$m = \frac{Q}{h_{fg}} = \frac{534.59 \times 10^3}{2257} = \mathbf{236.86 \text{ kg/h}} \text{ (Ans.)}$$

Example 9.12. *A vertical plate 350 mm high and 420 mm wide, at 40°C, is exposed to saturated steam at 1 atm. Calculate the following:*

(i) The film thickness at the bottom of the plate;

(ii) The maximum velocity at the bottom of the plate;

(iii) The total heat flux to the plate.

Assume vapour density is small compared to that of the condensate.

Solution. *Given*: $t_s = 40°C$; $t_{sat} = 100°C$, $L = 350$ mm $= 0.35$ m, $B = 420$ mm $= 0.42$ m.

The properties will be evaluated at the film temperature, *i.e.*, the average of t_{sat} and t_s;

$$t_f = \frac{100 + 40}{2} = 70°C; \text{ further } h_{fg} \text{ is evaluated at } 100°C.$$

The properties at 70°C are:

$\rho_l = 977.8$ kg/m^3; $\mu = 0.4 \times 10^{-3}$ kg/ms; $k = 0.667$ W/m°C and $h_{fg} = 2257$ kJ/kg.

(*i*) The film thickness at the bottom of the plate, δ:

$$\delta = \left[\frac{4k\,\mu\,(t_{sat} - t_s)\,x}{g\,\rho_l\,(\rho_l - \rho_v)\,h_{fg}}\right]^{1/4} \qquad \text{...[Eqn. (9.24)]}$$

$$= \left[\frac{4k\,\mu\,(t_{sat} - t_s)\,x}{g \cdot \rho_l^2\,h_{fg}}\right]^{1/4} \qquad \text{Neglecting } \rho_v,\ \rho_v << \rho_l \qquad \text{...(given)}$$

or, $$\delta = \left[\frac{4 \times 0.667 \times 0.4 \times 10^{-3}\,(100 - 40) \times 0.35}{9.81 \times (977.8)^2 \times 2257 \times 10^3}\right]^{1/4} = 1.8 \times 10^{-4} \text{ m} = \mathbf{0.18\ mm}$$

($\because$ $x = l = 0.35$ m in this case)

(*ii*) The maximum velocity at the bottom of the plate, u_{max} :

$$u = \frac{(\rho_l - \rho_v)\, g}{\mu} \left(\delta\, y - \frac{y^2}{2} \right) \qquad \text{...[Eqn. (9.17)]}$$

$$= \frac{\rho_l\, g}{\mu} \left(\delta y - \frac{y^2}{2} \right) \qquad \text{...neglecting } \rho_v$$

At $y = \delta$, $u = u_{max}$., therefore,

$$u_{max} = \frac{\rho_l\, g\, \delta^2}{2\mu} = \frac{977.8 \times 9.81 \times (1.8 \times 10^{-4})^2}{2 \times 0.4 \times 10^{-3}} = \mathbf{0.338\ m/s} \qquad \textbf{(Ans.)}$$

(*iii*) The total heat flux to the plate, Q:

$$\bar{h} = 0.943 \left[\frac{\rho_l\, (\rho_l - \rho_v)\, k^3\, g\, h_{fg}}{\mu\, L\, (t_{sat} - t_s)} \right]^{1/4} \qquad \text{...[Eqn. (9.29)]}$$

$$= 0.943 \left[\frac{\rho_l^2 \; k^3 \; g \; h_{fg}}{\mu \; L \; (t_{sat} - t_s)} \right]^{1/4} \qquad \text{...neglecting } \rho_v$$

or, $$\bar{h} = 0.943 \left[\frac{(977.8)^2 \times (0.667)^3 \times 9.81 \times 2257 \times 10^3}{0.4 \times 10^{-3} \times 0.35 \; (100 - 40)} \right]^{1/4}$$

$$= 0.943 \left[\frac{6.282 \times 10^{12}}{8.4 \times 10^{-3}} \right]^{1/4} = 4931.35 \text{ W/m°C}$$

The total heat flux is given by

$$Q = \bar{h} \, A \, (t_{sat} - t_s) = \bar{h} \times (L \times B) \, (t_{sat} - t_s)$$

$$= 4931.35 \times 0.35 \times 0.42 \times (100 - 40)$$

$= 43494$ W or **43.494 kW** **(Ans.)**

Example 9.13. *Vertical flat plate in the form of fin is 600 m in height and is exposed to steam at atmospheric pressure. If surface of the plate is maintained at 60°C, calculate the following:*

(i) *The film thickness at the trailing edge of the film,*

(ii) *The overall heat transfer coefficient,*

(iii) *The heat transfer rate, and*

(iv) *The condensate mass flow rate.*

Assume laminar flow conditions and unit width of the plate.

Solution. *Given* : L = 600 mm = 0.6 m; t_s = 100°C;

The properties of vapour at atmospheric pressure are:

$$t_{sat} = 100°C, \; h_{fg} = 2257 \text{ kJ/kg}; \; \rho_v = 0.596 \text{ kg/m}^3.$$

The properties of saturated vapour at the mean film temperature $t_f = \dfrac{100 + 60}{2} = 80°C$ are:

$$\rho_l = 971.8 \text{ kg/m}^3, \; k = 67.413 \times 10^{-2} \text{ W/m°C}, \; \mu = 355.3 \times 10^{-6} \text{ Ns/m}^2 \text{ or kg/ms}$$

(*i*) The film thickness at the trailing edge of the plate, δ (at $x = L = 0.6$ m):

$$\delta = \left[\frac{4\,k\,\mu\,(t_{sat} - t_s)\,x}{\rho_l\,(\rho_l - \rho_v)\,g\,h_{fg}}\right]^{1/4} \qquad \text{...[Eqn. (9.24)]}$$

$$\delta_L = \left[\frac{4 \times 67.413 \times 10^{-2} \times 355.3 \times 10^{-6}\ (100 - 60) \times 0.6}{971.8\ (971.8 - 0.596) \times 9.81 \times (2257 \times 10^3)}\right]^{1/4}$$

or,

$$\delta_L = \frac{0.02299}{2.08972 \times 10^{13}} = 1.82 \times 10^{-4}\ \text{m} = \mathbf{0.182\ mm} \qquad \textbf{(Ans.)}$$

(*ii*) The overall heat transfer coefficient, $\bar{h}$:

$$\bar{h} = \frac{4}{3} h_L = \frac{4}{3} \frac{k}{\delta_L} = \frac{4}{3} \times \frac{67.413 \times 10^{-2}}{1.82 \times 10^{-4}} = 4938.68\ \text{W/m}^2\,°\text{C}$$

Using McAdam's correction which is 20% higher than Nusselt's result, we have

$$\bar{h} = 4938.68 \times 1.2$$

$$= \mathbf{5926.4\ W/m^2°C} \qquad \textbf{(Ans.)}$$

(*iii*) The heat transfer rate, Q :

$$Q = \bar{h}\ A_s\,(t_{sat} - t_s) = h \times (L \times B)\,(t_{sat} - t_s)$$

$$= 5926.4 \times (0.6 \times 1)\,(100 - 60) = \mathbf{142233.6\ W}$$

(*iv*) The condensate mass flow rate, *m* :

$$m = \frac{Q}{h_{fg}} \quad \text{...[Eqn. (9.32)]}$$

$$= \frac{142233.6}{2257 \times 10^3} = 0.063 \text{ kg/s or } \mathbf{226.8 \text{ kg/h}} \quad \textbf{(Ans.)}$$

Let us check whether the flow is laminar or not.

$$Re = \frac{4m}{\mu B} \quad \text{...[Eqn. (9.35)]}$$

$$= \frac{4 \times 0.063}{355.3 \times 10^{-6} \times 1} = 709.26 < 1800$$

This shows that the assumption of laminar flow is correct.

Example 9.14. *A vertical tube of 60 mm outside diameter and 1.2 m long is exposed to steam at atmospheric pressure. The outer surface of the tube is maintained at a temperature of 50°C by circulating cold water through the tube. Calculate the following:*

(i) The rate of heat transfer to the coolant, and

(ii) The rate of condensation of steam.

Solution. *Given* : $D = 60$ mm $= 0.06$ m, $L = 1.2$ m, $t_s = 50°C$

Assuming the condensation film is laminar and noncondensable gases in steam are absent;

The mean film temperature $t_f = \dfrac{100 + 50}{2} = 75°C$

The thermo-physical properties of water at 75°C are:

$$\rho_l = 975 \text{ kg/m}^3,\ \mu_l = 375 \times 10^{-6} \text{ Ns/m}^2,\ k = 0.67 \text{ W/m°C}.$$

The properties of saturated vapour at $t_{sat} = 100°C$ are :

$$\rho_v = 0.596 \text{ kg/m}^3,\ h_{fg} = 2257 \text{ kJ/kg}.$$

(*i*) The rate of heat transfer, Q:

For laminar condensation on a vertical surface

$$\bar{h} = 1.13 \left[\frac{\rho_l (\rho_l - \rho_v)\, k^3\, g\, h_{fg}}{\mu\, L\, (t_{sat} - t_s)} \right]^{1/4} \qquad \text{...[Eqn. (9.30)]}$$

or,
$$\bar{h} = 1.13 \left[\frac{975\,(975 - 0.596) \times (0.67)^3 \times 9.81 \times (2257 \times 10^3)}{375 \times 10^{-6} \times 1.2 \times (100 - 50)} \right]^{1/4}$$

$$= 4627.3 \text{ W/m}^2\text{°C}$$

$$Q = \bar{h}\, A_s\, (t_{sat} - t_s) = \bar{h}\, (\pi DL)\, (t_{sat} - t_s)$$

$$= 4627.3 \times (\pi \times 0.06 \times 1.2)\,(100 - 50) = 52333.5$$

$$= \mathbf{52.333\ kW\ (Ans.)}$$

(*ii*) The rate of condensation of steam, m :

The condensation rate is given by

$$m = \frac{Q}{h_{fg}} = \frac{52333.5}{2257 \times 10^3} = 0.0232 \text{ kg/s} = \mathbf{83.52\ kg/h\ (Ans.)}$$

Let us check the assumption of laminar film condensation by calculating *Re*.

$$Re = \frac{4m}{P\,\mu_l} \qquad \text{...[Eqn. (9.35)]}$$

or,
$$Re = \frac{4 \times 0.0232}{\pi D \times 375 \times 10^{-6}} = \frac{4 \times 0.0232}{\pi \times 0.06 \times 375 \times 10^{-6}} = 1312.85$$

Since, Re (= 1312.85) < 1800, hence the flow is *laminar*.

Example 9.15. *A horizontal tube of outer diameter 20 mm is exposed to dry steam at 100°C. The tube surface temperature is maintained at 84°C by circulating water through it. Calculate the rate of formation of condensate per metre length of the tube.*

Solution. *Given*: D = 20 mm = 0.02 m, t_s = 84°C; t_{sat} = 100°C

The mean film temperature $t_f = \dfrac{100 + 84}{2} = 92°C$

The properties of saturated liquid at 92°C are:

$\rho_l = 963.4 \text{ kg/m}^3,\ \mu_l = 306 \times 10^{-6} \text{ Ns/m}^2;\ k = 0.677 \text{ W/m°C}$

The properties of saturated vapour at t_{sat} = 100°C are:

$$\rho_v = 0.596 \text{ kg/m}^3,\ h_{fg} = 2257 \text{ kJ/kg}$$

Rate of formation of condensate per metre length of the tube, *m*:

The average heat transfer coefficient is given by

$$\bar{h} = 0.725 \left[\frac{\rho_l (\rho_l - \rho_v) k^3 g h_{fg}}{\mu_l (t_{sat} - t_s) D} \right]^{1/4} \qquad \text{...[Eqn. (9.40)]}$$

or,

$$\bar{h} = 0.725 \left[\frac{(963.4)(963.4 - 0.596) \times (0.677)^3 \times 9.81 \times (2257 \times 10^3)}{306 \times 10^{-6} (100 - 84) \times 0.02} \right]^{1/4}$$

$$= 11579.7 \text{ W/m}^2\text{°C}$$

The heat transfer per unit length is

$$\frac{Q}{L} = \bar{h} \times \pi D \times (t_{sat} - t_s)$$

$$= 11579.7 \times \pi \times 0.02 \times (100 - 84) = 11641.2 \text{ W}$$

Rate of formation of condensate per metre length of the tube,

$$\frac{m}{L} = \frac{Q/L}{h_{fg}} = \frac{11641.2}{2257 \times 10^3} = 5.157 \times 10^{-3} \text{ kg/s} = \mathbf{18.56 \text{ kg/h}} \qquad \textbf{(Ans.)}$$

Example 9.16. *A steam condenser consisting of a square array of 625 horizontal tubes, each 6mm in diameter, is installed at the exhaust hood of a steam turbine. The tubes are exposed to saturated steam at a pressure of 15 kPa. If the tube surface temperature is maintained at 25°C, calculate:*

(i) The heat transfer coefficient, and

(ii) The rate at which steam is condensed per unit length of the tubes.

Assume film condensation on the tubes and absence of non-condensable gases.

Solution. *Given:* $D = 6 \text{ mm} = 0.006 \text{ m}, t_s = 25°\text{C}.$

Corresponding to 15 kPa pressure, the properties of vapour (from the table) are:

$$t_{sat} = 54°\text{C}, \rho_v = 0.098 \text{ kg/m}^3, h_{fg} = 2373 \text{ kJ/kg}.$$

The properties of saturated water at film temperature $t_f = \dfrac{54 + 25}{2} = 39.5°\text{C}$ are:

$$\rho_l = 992 \text{ kg/m}^3; \mu = 663 \times 10^{-6} \text{ Ns/m}^2; k = 0.631 \text{ W/m°C}$$

Since the tubes are arranged in square array, therefore, the number of horizontal tubes in vertical column is : $N = \sqrt{625} = 25$

(*i*) The heat transfer coefficient, $\bar{h}$:

The average heat transfer coefficient for steam condensing on bank of horizontal tubes is given by

$$\bar{h} = 0.725 \left[\frac{\rho_l (\rho_l - \rho_v) k^3 g h_{fg}}{N \mu_l (t_{sat} - t_s) D} \right]^{1/4} \quad \text{...[Eqn. (9.41)]}$$

or,

$$\bar{h} = 0.725 \left[\frac{992 (992 - 0.098) \times (0.631)^3 \times 9.81 \times (2373 \times 10^3)}{25 \times 663 \times 10^{-6} (54 - 25) \times 0.006} \right]^{1/4}$$

$$= 0.725 \left(\frac{5.7548 \times 10^{12}}{2.884 \times 10^{-3}} \right)^{1/4} = \mathbf{4845.6 \ W/m^2 {}^\circ C \ (Ans.)}$$

(*ii*) The rate at which steam is condensed per unit length, *m* :

The rate of condensation for the single tube of the array per metre length is

$$m_1 = \frac{Q}{h_{fg}} = \frac{\bar{h} \pi D (t_{sat} - t_s)}{h_{fg}}$$

$$= \frac{4845.6 \times \pi \times 0.006 (54 - 25)}{2373 \times 10^3} = 1.116 \times 10^{-3} \text{ kg/s.m}$$

The rate of condensation for the complete array is

$$m = 625 \times m_1 = 625 \times 1.116 \times 10^{-3} = \mathbf{0.6975 \ kg/s.m \ (Ans.)}$$

Example 9.17. *A 750 mm square plate, maintained at 28°C is exposed to steam at 8.132 kPa. Calculate the following:*

(i) The film thickness, local heat transfer coefficient and mean flow velocity of condensate at 400 mm from the top of the plate,

(ii) The average heat transfer coefficient and total heat transfer from the entire plate,

(iii) Total steam condensation rate, and

(iv) The heat transfer coefficient if the plate is inclined at 25° with the horizontal plane.

Solution. *Given*: $L = B = 750$ mm $= 0.75$ m, $t_s = 28$°C, $x = 400$ mm $= 0.4$ m

Assume laminar flow film condensation.

Properties of saturated vapour at 8.132 kPa (or 0.08132 bar) are:

$$t_{sat} = 42°\text{C};\ \rho_v = 0.0561\ \text{kg/m}^3;\ h_{fg} = 2402\ \text{kJ/kg}$$

The mean film temperature $t_f = \dfrac{42 + 28}{2} = 35°\text{C}$

The properties of saturated water at 35°C are:

$$\rho_l = 993.95\ \text{kg/m}^3;\ k = 62.53 \times 10^{-2}\ \text{W/m°C},\ \mu = 728.15 \times 10^{-6}\ \text{kg/ms}.$$

(*i*) δ_x, h_x, u_m at 400 mm from the top of the plate:

The *film thickness* at a distance x from the top edge of the plate is given by:

$$\delta = \left[\frac{4\ k\ \mu\ (t_{sat} - t_s)\ x}{\rho_l\ (\rho_l - \rho_v)\ g\ h_{fg}}\right]^{1/4} \qquad \text{...[Eqn. (9.24)]}$$

or, $$\delta = \left[\frac{4 \times 62.53 \times 10^{-2} \times 728.15 \times 10^{-6}\ (42 - 28) \times x}{993.95\ (993.95 - 0.0561) \times 9.81 \times (2402 \times 10^3)}\right]^{1/4} = 1.819 \times 10^{-4}\ (x)^{1/4}$$

At $x = 0.4$ m,

$$\delta x = 1.819 \times 10^{-4} \times (0.4)^{1/4} \simeq 1.45 \times 10^{-4} \text{ m} \simeq \mathbf{0.145\ mm} \quad \textbf{(Ans.)}$$

At $x = L = 0.75$ m,

$$\delta_L = 1.819 \times 10^{-4}\ (0.75)^{1/4} = 1.69 \times 10^{-4} \text{ m} = 0.169 \text{ mm}$$

The local *heat transfer coefficient,*

$$h_x = \frac{k}{\delta_x} = \frac{62.53 \times 10^{-2}}{1.45 \times 10^{-4}} = \mathbf{4312.41\ W/m^2 {}^\circ C} \quad \textbf{(Ans.)}$$

The *mean flow velocity* of condensate,

$$u_m = \frac{(\rho_l - \rho_v)\ g\ .\ \delta^2}{3\ \mu} \qquad \text{...[Eqn. (9.19)]}$$

or, $$u_m = \left[\frac{(993.95 - 0.0561) \times 9.81 \times (1.45 \times 10^{-4})^2}{3 \times 728.15 \times 10^{-6}}\right] = \mathbf{0.0938\ m/s}$$

(*ii*) Average heat transfer coefficient ($\bar{h}$):

$$\bar{h} = h_L = \frac{4}{3} \cdot \frac{k}{\delta_L} = \frac{4}{3} \times \frac{62.53 \times 10^{-2}}{1.69 \times 10^{-4}} = 4933.33 \text{ W/m}^2\text{°C}$$

(where, δ_L = the film thickness at the bottom of the plate)

Using McAdams correction,

$$\bar{h} = 1.2 \times 4933.33 = \mathbf{5920 \text{ W/m}^2\text{°C (Ans.)}}$$

The total heat transfer from the entire plate, *Q*:

$$Q = \bar{h}\ A_s (t_{sat} - t_s) = \bar{h}\ (L \times B)(t_{sat} - t_s)$$
$$= 5920 \times (0.75 \times 0.75)(42 - 28) = \mathbf{46620 \text{ W (Ans.)}}$$

(*iii*) Total steam condensation rate, *m*:

$$m = \frac{Q}{h_{fg}} \qquad \text{...[Eqn. (9.32)]}$$

or, $$m = \frac{46620}{2402 \times 10^3} = 0.0194 \text{ kg/s or } \mathbf{69.87 \text{ kg/h}} \qquad \mathbf{(Ans.)}$$

(*iv*) The heat transfer coefficient when the plate is inclined 25° with the horizontal, $h_{inclined}$:

$$h_{inclined} = h_{vertical} \times (\sin\theta)^{1/4} \qquad \text{...[Eqn. (9.34)]}$$
$$= 5920 \times (\sin 25°)^{1/4} = \mathbf{4773.2 \text{ W/m}^2\text{°C (Ans.)}}$$

Let us check the type of flow [using Eqn. (9.35)],

$$Re = \frac{4m}{\mu\ B} = \frac{4 \times 0.0194}{728.15 \times 10^{-6} \times 0.75} = 142 < 1800$$

Hence assumption is correct.

Example 9.18. *A vertical plate 3.2 m high maintained at 54°C, is exposed to saturated steam at atmospheric pressure. Calculate the heat transfer rate per unit width.*

Solution. *Given*: $L = 3.2$ m; $B = 1$ m, $t_s = 54°C$; $t_{sat} = 100°C$.

Heat transfer rate per unit width :

In order to determine whether the condensate film is laminar or turbulent; the Reynolds number must be checked.

The mean film temperature $t_f = \dfrac{100 + 54}{2} = 77°C$

The properties of the condensate at 77°C are :

$$\mu_l = 365 \times 10^{-6} \text{ Ns/m}^2; \qquad k = 668 \times 10^{-3} \text{ W/m°C};$$

$$\rho_l = \frac{1}{1.027 \times 10^{-3}} = 973.7 \text{ kg/m}^3.$$

The properties of saturated vapour at $t_{sat} = 100°C$

$$\rho_v = 0.596 \text{ kg/m}^3; h_{fg} = 2257 \text{ kJ/kg}.$$

Assuming the flow to be turbulent the relevant equations are :

$$Re = \frac{4\, \bar{h}\, L\, (t_{sat} - t_s)}{h_{fg} \cdot \mu_l} \qquad \text{...[Eqn. (9.38)]}$$

and,

$$\bar{h} = 0.0077 \left[\frac{\rho_l\, (\rho_l - \rho_v)\, k^2 g}{\mu_l^2} \right]^{1/3} (Re)^{0.4} \qquad \text{...[Eqn. (9.39)]}$$

Eliminating $\bar{h}$ from these equations, we get the condition that the flow will be turbulent if

$$0.00296\left[\frac{\rho_l\ (\rho_l - \rho_v)\ k^3 g\ (t_{sat} - t_s)^3\ L^3}{\mu_l^5\ (h_{fg})^3}\right]^{5/9} > 1800$$

Substituting the values, we get

$$0.00296\left[\frac{973.7\ (973.7 - 0.596)\ (668 \times 10^{-3})^3 \times 9.81 \times (100 - 54)^3 \times (3.2)^3}{(365 \times 10^{-6})^5\ (2257 \times 10^3)^3}\right]^{5/9}$$

or, $$0.00296\left[\frac{8.837 \times 10^{12}}{74.48}\right]^{5/9} = 4144.8 > 1800$$

Thus the film is turbulent as assumed and $Re = 4144.8$

$$\therefore \qquad \bar{h} = 0.007\left[\frac{973.7\ (973.7 - 0.596) \times (668 \times 10^{-3})^3 \times 9.81}{(365 \times 10^{-6})^2}\right]^{1/3} \times (4144.8)^{0.4}$$

$$= 0.0077 \times (2.0797 \times 10^{13})^{1/3} \times 27.99 = 5866.62\ \text{W/m}^2\ °\text{C}$$

Heat transfer rate per unit width,

$$Q = \bar{h}\ A_s\ (t_{sat} - t_s)$$

$$= 5866.62 \times (3.2 \times 1)\ (100 - 54) = 863566\ \text{W/m} = \mathbf{863.566\ kW/m\ (Ans.)}$$

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