

## **Condensation Heat Transfer: Principles, Mechanisms, and Mathematical Modeling**

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### **Abstract**

The paper explores the process of condensation, where vapor transforms into liquid upon cooling, a critical mechanism for heat transfer in energy systems like electric power generation, refrigeration, and HVAC. It differentiates between film wise and dropwise condensation, noting that dropwise condensation generally allows for more efficient heat transfer. The study offers a thorough introduction to the principles of heat transfer through condensation, emphasizing its industrial significance and detailing the mechanisms involved, such as surface tension and fluid dynamics. A mathematical analysis is performed to model heat exchange rates and temperature distributions, considering various factors like vapor velocity and surface properties. The findings aim to improve understanding of heat transfer phenomena and provide equations useful for optimizing thermal systems in engineering contexts.

**Keywords:** Film Condensation; Drop wise Condensation; Heat Transfer; Thermodynamics; Fluid Dynamics; Cooling Systems

## **1. Overview of Condensation**

Condensation is a process where vapor is cooled and transforms into liquid, playing a crucial role in numerous applications such as thermal power plants, chemical refining, water harvesting, refrigeration, and electronics cooling. Enhancing the rates of condensation heat transfer in these applications can significantly improve process efficiencies. This phase-change phenomenon can occur in two distinct modes: filmwise condensation (FWC), characterized by the formation of a continuous liquid film, and dropwise condensation (DWC), which involves the creation of discrete droplets. The mode of condensation that occurs is primarily influenced by factors such as the thermophysical properties of the condensate, the operating conditions, and the characteristics of the condensing surface. In the case of FWC, the presence of the liquid film introduces additional thermal resistance to heat transfer, resulting in larger temperature gradients across the film, [1] and [2].

In DWC, droplets slide down when they reach a certain critical size, clearing the surface and exposing it to vapor, thereby eliminating the liquid film that typically resists heat transfer. This characteristic led Schmidt et al. to discover in 1920 that dropwise condensation exhibits heat transfer coefficients 5-7 times higher than those found in filmwise condensation. As a result, researchers worldwide have become interested in promoting DWC, recognizing its potential to enable the design of smaller and less expensive condensers, which could be a significant breakthrough in the field, [3].

In DWC, discrete droplets form at specific nucleation sites and grow through coalescence or direct condensation until they reach a critical radius. Once this radius is surpassed, the body forces, usually gravitational, surpass the surface tension that keeps the droplet anchored to the surface, leading to the droplets detaching from the surface, [4]. The departing droplets clear the surface through vapor shearing, allowing for the formation of new nucleation sites on the surface, which enables the process to repeat itself, [2], [5] and [6].

In dropwise condensation, the cooled surfaces typically promote “non-wetting”. Various techniques are employed to promote dropwise condensation, including treating or coating the surface with polymers, such as Teflon and organic promoters like fatty acids, or by implanting ions of low energy, [7], [8], [9], [10], [11] and [12]. Besides applying a promoter layer on the surface, another effective strategy for enhancing dropwise condensation involves the introduction of a promoting chemical into the condensing vapor. This innovative approach works by incorporating specific substances that can alter the surface dynamics of the condensing droplets as they form on a surface.

The promoting chemicals work by altering the surface tension at the interface between vapor and liquid droplets, which accelerates the growth rates of the droplets and promotes their detachment from the surface once they attain a certain size. Consequently, rather than creating a continuous liquid film typical of filmwise condensation, this method encourages the formation of discrete droplets.

The significance of promoting chemicals in this context lies in their ability to enhance the efficiency of heat transfer processes. By encouraging droplets to rapidly form, grow, and then roll off the surface, the overall thermal performance improves considerably compared to surfaces that rely solely on physical treatments or coatings.

Additionally, the choice of promoting chemical can be tailored for specific applications, such as utilizing hydrophobic agents to improve the overall wettability characteristics of a surface or deploying surfactants that can modify the interfacial properties. This versatility not only enhances the efficiency of condensation processes but also facilitates the design of systems that require effective heat management, such as in power generation or refrigeration applications.

In summary, the introduction of promoting chemicals into the condensing vapor represents a promising avenue for augmenting dropwise condensation, offering a highly effective means to improve thermal performance, control condensation behavior, and optimize energy efficiency in various industrial applications, [13] – [20]. Although these promoters and coatings are used, sustaining dropwise condensation over an extended period remains challenging, as the condensing vapor is likely to wear away the promoter layer eventually, [21]. The duration for which dropwise condensation needs to be sustained varies according to the specific application, typically ranging from approximately 100 hours to over 4 years, [22]. The wear-off of the promoter layer leads to a wetting

surface, causing a shift back to filmwise condensation. Additionally, other techniques to enhance dropwise condensation include implementing structural modifications to the surface, [20], [23] and [24].

Recent studies are exploring innovative approaches to enhance and sustain the phenomenon of dropwise condensation, a process that offers significant advantages in terms of heat transfer efficiency. Researchers are investigating various methods that involve the integration of surface coating technologies, modifications to the structural properties of surfaces, and the periodic injection of promoting substances in vapor form. These combined strategies aim to create environments conducive to efficient dropwise condensation, which is particularly desirable in various industrial applications.

One of the significant challenges in achieving sustained dropwise condensation arises when dealing with certain fluids, particularly those with surface tensions considerably lower than that of water. Fluorinated refrigerants and hydrocarbons are pertinent examples of such fluids. In these cases, the surface energies of the promoting surface and the fluid become comparable, making it difficult to maintain the dropwise condensation mode. This is because the surface tension of the drop must be significantly lower than the interaction forces at play, which can lead to a transition to filmwise condensation that is a less favorable condition that results in reduced efficiency.

For over 90 years, a concerted effort within the scientific community has focused on achieving and sustaining both internal and external dropwise condensation for various industrial applications, driving a vast array of research that has encompassed numerous theoretical explorations, the development of predictive models, and a wide array of experimental studies. This pursuit of understanding the underlying physics is crucial, as it lays the groundwork for improving heat exchanger systems, enhancing refrigeration technologies, and developing more effective thermal management applications.

This paper provides a comprehensive review of the existing literature on the theory and models related to both external and internal dropwise condensation. Furthermore, it examines experimental research findings that contribute to the understanding of this complex phenomenon. By digging deeper into the mechanics of dropwise condensation, the review aims to illuminate specific applications and scenarios where the benefits of utilizing dropwise condensation could be maximized. Enhanced performance in heat transfer systems could lead to energy savings, improved system efficiencies, and reduced environmental impacts, making this research particularly relevant in today's context of sustainable technological advancement, [25] – [36].

## 2. General Aspects

The condensation process is the opposite of boiling. It takes place when saturated vapor meets a surface with a temperature lower than the saturation temperature associated with the vapor pressure. During condensation, latent heat is released, resulting in heat transfer to the surface. The condensing liquid may cool below the dew point when in contact with the cold surface, which can lead to additional vapor condensing on either the exposed surface or the existing condensing liquid vapor. Below, Figure 1 illustrates the physical mechanisms involved in condensation [37] – [46].

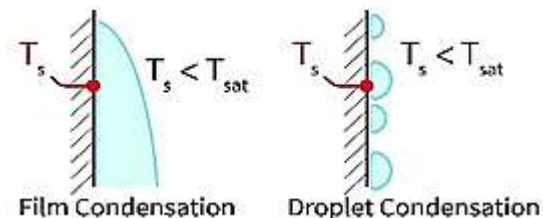


Figure 1: The Physical Mechanisms of Condensation

### 3. Forms of Condensation

Depending on the condition of the cold surface, condensation can occur in two possible ways: film or strip condensation and drop condensation [25], [27], [30] and [34].

#### 3.1 Film Condensation

If the condensed material tends to wet the surface and thus forms a liquid sheet, then the condensation process is known as film condensation. In this procedure, heat is transferred from the steam to the cold medium through a film of condensate formed on the surface. The liquid flows below the cooling surface under the action of gravity and the layer continuously grows in thickness due to the newly condensed vapors. The connected film gives thermal resistance and checks advanced (additional) heat transfer between the vapor and the surface.

Additionally, heat transfer from the steam to the cold surface occurs through the film formed on the surface. Heat is transferred from the steam to the condensate formed on the surface by convection and is further transferred from the condensing film to the cooling surface by conduction. This combined method of heat transfer by conduction and convection reduces heat transfer rates significantly (compared to drop condensation). This is why the heat transfer rates of film condensation are lower. It is less than that of drop condensation. Figure 2 below shows film condensation.

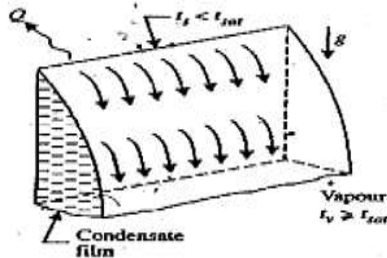


Figure 2: Film Condensation

#### 3.2 Dropwise Condensation

In dropwise condensation, vapor transforms into tiny droplets of liquid of varying sizes, which fall randomly to the surface below. These droplets emerge in the surface's cracks and pits, increasing in size as they move away or detach from the surface. They may collide with other droplets and quickly flow off the surface under the force of gravity, without creating a continuous film. Figure 3 illustrates the process of condensation with droplets on a vertical plate.

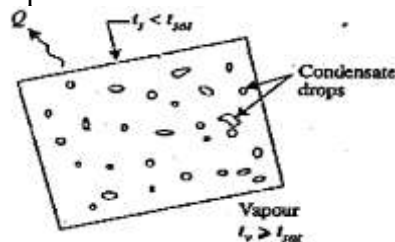


Figure 3: Condensation with Drops on a Vertical Plate

In this form of condensation, a significant amount of the solid surface area is directly in contact with vapor, lacking an insulating layer of the condensing liquid. As a result, this method achieves a higher heat transfer rate, reaching up to 750 kW/m<sup>2</sup>. Drop condensation occurs on highly polished surfaces or those contaminated with impurities like fatty acids and organic compounds. This type of condensation typically offers a heat transfer coefficient that is 5 to 10 times greater than that of film condensation. Although drop condensation is generally favored over film condensation, it poses challenges in execution or setup. This is largely because most surfaces become wet after extended exposure to condensation vapors. However, drop condensation can be effectively achieved under controlled conditions using enhanced condensate additives and various surface coatings.

#### 3.3 Laminar Film Condensation on a Vertical Plate

The analysis of film condensation on a vertical plate is based on the work of Nusselt (1916). In cases where the vapor velocity is not excessively high and the liquid layer is not overly thick, the flow of the condensate tends to be laminar. The thickness of the condensate film depends on both the rate of

vapor condensation and the rate at which the condensate is swept away from the surface. As illustrated in Figure 4, the film thickness on a vertical surface will progressively increase from the top to the bottom.

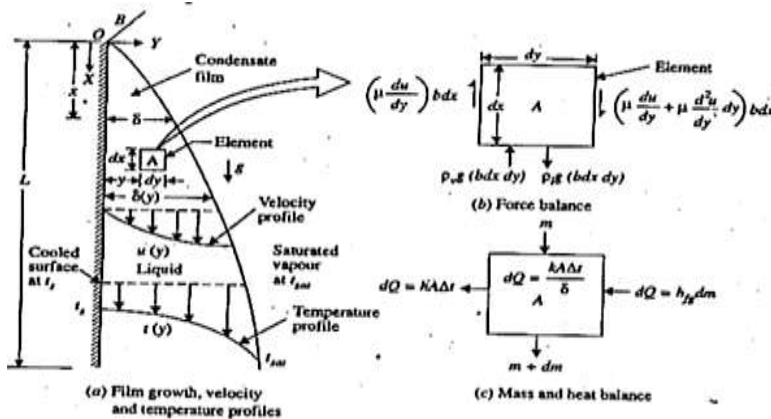


Figure 4: Laminated Condensation on a Vertical Flat Plate

### Nusselt Analysis of Film Condensation: Simplifying Assumptions

- i. The liquid film formed during condensation flows primarily due to the force of gravity.
- ii. The condensate flow is characterized as laminar, with fluid properties assumed to remain constant.
- iii. There is efficient thermal contact between the liquid film and the cooling surface, which allows us to assume that the temperature throughout the film matches the surface temperature ( $t_s$ ). Additionally, the temperature at the liquid-vapor interface is considered equivalent to the saturation temperature ( $t_{sat}$ ) corresponding to the existing pressure.
- iv. It is assumed that the only significant forces acting on the fluid are viscous shear and gravity. Consequently, the contributions from perpendicular viscous forces and inertial forces are neglected.
- v. The shear stress at the liquid-vapor interface is sufficiently small to be disregarded, indicating that there is no velocity gradient present at that interface.

$$\left[ i. e., \left( \frac{\partial u}{\partial y} \right)_{y=\delta} = 0 \right]$$

- vi. Heat transfer through the condensing layer occurs solely through conduction, resulting in a linear temperature distribution.
- vii. The condensed steam is entirely pure, devoid of gases, air, and any non-condensable impurities.
- viii. Radiation exchanges between the vapor and the liquid film are taken into account; the horizontal velocity component at any point within the fluid film is considered, while the curvature of the film is negligible and thus ignored.

Consider a film condensation process taking place on the surface of a flat, vertical plate, as illustrated in Figure 4. The coordinate system is also depicted in the figure. The origin 'o' is located at the top of the plate, with the x-axis aligned along the vertical surface, moving downward in the positive direction, and the y-axis perpendicular to it. The vertical plate has a height of l, a width of b, and  $\delta$  denotes the thickness of the film at a distance x from the origin. The thickness of the liquid film is initially zero at the upper edge of the plate and gradually increases due to ongoing condensation at the liquid-vapor interface, reaching its maximum at the bottom of the plate.

Let,  $\rho_l$  = density of the fluid film.

$\rho_v$  = vapor density.

$\rho_f g$  = latent heat of condensation.

$k$  = fluid film conductivity.

$\mu$  = absolute viscosity of the fluid film.

$t_s$  = surface temperature.

$t_{sat}$  = steam saturation temperature at prevailing pressure.

**(a) Velocity Distribution**

To find an expression for the velocity distribution  $u$  as a function of the distance  $y$  from the wall surface, let us consider an equilibrium between the forces of gravitation and viscosity over an initial volume ( $b dx dy$ ) of the fluid film,

$$\text{gravitational force on the element} = \rho_l g(b dx dy) - \rho_v g(b dx dy) \quad (i)$$

Viscous shear force on the element,

$$\text{Viscous shear force on the element} = \mu \frac{du}{dy}(b dx) - \left[ \mu \frac{du}{dy} + \mu \frac{d^2u}{dy^2} dy \right] (b dx) \quad (ii)$$

Equating equations (i) and (ii) we get equation (1),

$$\begin{aligned} \rho_l g(b dx dy) - \rho_v g(b dx dy) &= \mu \frac{du}{dy}(b dx) - \left[ \mu \frac{du}{dy} + \mu \frac{d^2u}{dy^2} dy \right] (b dx) \\ \frac{d^2u}{dy^2} &= \frac{-(\rho_l - \rho_v)g}{\mu} \end{aligned} \quad (1)$$

By integration we get,

$$\frac{du}{dy} = \frac{-(\rho_l - \rho_v)g}{\mu} y + c_1$$

Integrating again, we get,

$$u = \frac{-(\rho_l - \rho_v)(y^2/2)g}{\mu} + c_1 y + c_2$$

The boundary conditions are as follows:

When  $y = 0, u = 0$

When  $y = \delta, du/dy = 0$

Using these boundary conditions, we obtain the following values of  $c_1$  and  $c_2$ ,

$$c_1 = \frac{(\rho_l - \rho_v)g\delta}{\mu} \quad \text{and} \quad c_2 = 0$$

By substituting the values of  $c_1$  and  $c_2$ , we obtain the velocity profile.

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

$$\text{or} \quad u = \frac{(\rho_l - \rho_v)g \cdot \delta^2}{\mu} \left[ \frac{y}{\delta} - \frac{1}{2} \left( \frac{y}{\delta} \right)^2 \right] \quad (3)$$

Equation (3) is the profile of the required speed.

The average flow velocity  $u_{mean}$  of the liquid film at distance  $y$  is given by the equation,

$$\begin{aligned} u_m &= \frac{1}{\delta} \int_0^\delta u dy \\ &= \frac{1}{\delta} \int_0^\delta \frac{(\rho_l - \rho_v)g \cdot \delta^2}{\mu} \left[ \frac{y}{\delta} - \frac{1}{2} \left( \frac{y}{\delta} \right)^2 \right] dy \\ \text{or} \quad u_m &= \frac{(\rho_l - \rho_v)g \cdot \delta^2}{3\mu} \end{aligned} \quad (4)$$

**(b) Mass Flow Rate**

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

$$\text{mass flow rate (m)} = \text{mean flow velocity (} u_m \text{)} \times \text{flow area} \times \text{density}$$

$$\text{or} \quad m = \frac{(\rho_l - \rho_v)g \cdot \delta^2}{3\mu} \times b \cdot \delta \times \rho_l = \frac{\rho_l(\rho_l - \rho_v)g \cdot b \cdot \delta^2}{3\mu} \quad (5)$$

Therefore, the mass flow is a function of  $x$ ; this is because the film thickness  $\delta$  is fundamentally dependent on  $x$ .



As the flow goes from  $x$  to  $(x + \delta x)$  the film grows from  $\delta$  to  $(\delta + d\delta)$  due to the additional condensed matter. The mass of condensate added between  $x$  and  $(x + \delta x)$  can be calculated by differentiating equation (5) with respect to  $x$  (or  $\delta$ ).

$$\begin{aligned} dm &= \frac{d}{dx} \left[ \frac{\rho_l(\rho_l - \rho_v)g \cdot b \cdot \delta^3}{3\mu} \right] dx \\ &= \frac{d}{dx} \left[ \frac{\rho_l(\rho_l - \rho_v)g \cdot b \cdot \delta^3}{3\mu} \right] \frac{d\delta}{dx} dx \\ dm &= \left[ \frac{\rho_l(\rho_l - \rho_v)g \cdot b \cdot \delta^2}{\mu} \right] dx \quad (6) \end{aligned}$$

### (c) Heat Flux

The rate of heat flow through the film ( $dQ$ ) is equivalent to the rate of energy release due to condensation at the surface. thus,

$$dQ = h_{fg} \cdot dm = h_{fg} \left[ \frac{\rho_l(\rho_l - \rho_v)g \cdot b \cdot \delta^2}{\mu} \right] ds \quad (7)$$

According to our assumption, heat transfer through the condensing layer is pure conduction, therefore,

$$dQ = \frac{k(bdx)}{\delta} (t_{sat} - t_s) \quad (8)$$

By unifying equations (7) and (8), we get,

$$\begin{aligned} \frac{h_{fg} \rho_l(\rho_l - \rho_v)g \cdot b \cdot \delta^2}{\mu} \cdot ds &= \frac{k(bdx)}{\delta} (t_{sat} - t_s) \\ \text{or } \delta^3 \cdot ds &= \frac{k \mu}{\rho_l(\rho_l - \rho_v)gh_{fg}} (t_{sat} - t_s) dx \end{aligned}$$

By integrating the above equation we get,

$$\frac{\delta^4}{4} = \frac{k \mu}{\rho_l(\rho_l - \rho_v)gh_{fg}} (t_{sat} - t_s)x + c_1$$

Substituting the boundary condition:  $\delta = 0$  at  $x = 0$  produces  $c_1 = 0$ , thus:

$$\delta = \left[ \frac{4k \mu (t_{sat} - t_s)x}{\rho_l(\rho_l - \rho_v)gh_{fg}} \right]^{\frac{1}{4}} \quad (9)$$

Equation (9) shows that the thickness of the temperature film increases with the fourth root of the dimension below the surface. The increase is somewhat rapid at the upper end of the vertical surface and slows down further.

### (d) Film Heat Transfer Coefficient

According to Nusselt's hypothesis, heat flow from vapor to the surface is by conduction through the liquid layer. Therefore,

$$dQ = \frac{k(bdx)}{\delta} (t_{sat} - t_s) \quad (i)$$

Heat flow can also be expressed as:

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

Where  $h_x$  is the local heat transfer coefficient.

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

$$\begin{aligned} \frac{k(bdx)}{\delta} (t_{sat} - t_s) &= h_x(b dx)(t_{sat} - t_s) \\ \text{or } h_x &= \frac{k}{\delta} \quad (10) \end{aligned}$$

Equation (10) shows that at a specific point on the heat transfer surface, the film modulus  $h_x$  is directly proportional to the thermal conductivity  $k$  and inversely proportional to the film thickness  $\delta$  at that point.

Substituting the value of  $s$  from equation (9), we get,

$$h_x = \left[ \frac{\rho_l(\rho_l - \rho_v)k^3gh_{fg}}{4\mu x(t_{sat} - t_s)} \right]^{\frac{1}{4}} \quad (11)$$

The local heat transfer coefficient at the lower end of the plate, *i. e.*  $x = l$

$$h_l = \left[ \frac{k^3\rho^2gh_{fg}}{\mu\mu h_l(t_{sat} - t_s)} \right]^{\frac{1}{4}} \quad (12)$$

It is noted that the condensation rate of heat transfer is greater at the upper end of the plate than that at the lower end. The average value can be obtained by integrating the position value of the coefficient (Equation (11)) as follows:

$$\begin{aligned} \bar{h} &= \frac{1}{l} \int_0^l h_x dx \\ &= \frac{1}{l} \int_0^l \left[ \frac{\rho_l(\rho_l - \rho_v)k^3gh_{fg}}{4\mu x(t_{sat} - t_s)} \right]^{\frac{1}{4}} dx = \frac{1}{l} \int_0^l \left[ \frac{\rho_l(\rho_l - \rho_v)k^3gh_{fg}}{4\mu(t_{sat} - t_s)} \right]^{\frac{1}{4}} \int_0^l x^{-\frac{1}{4}} dx \\ &= \frac{1}{l} \left[ \frac{\rho_l(\rho_l - \rho_v)k^3gh_{fg}}{4\mu(t_{sat} - t_s)} \right]^{\frac{1}{4}} \left[ \frac{x^{(-\frac{1}{4}+1)}}{-\frac{1}{4}+1} \right]_0^l \\ \text{or } \bar{h} &= \frac{4}{3} \left[ \frac{\rho_l(\rho_l - \rho_v)k^3gh_{fg}}{4\mu l(t_{sat} - t_s)} \right]^{\frac{1}{4}} \quad (13) \\ \bar{h} &= \frac{4}{3} h_l = \frac{4}{3} \times \frac{k}{\delta_l} \end{aligned}$$

Where  $h_l$  is the local heat transfer coefficient at the bottom edge of the plate.

This shows that the average heat transfer coefficient is 4/3 times the local heat transfer coefficient at the trailing edge of the panel.

Equation (13) is usually written in the form,

$$\bar{h} = 0.943 \left[ \frac{\rho_l(\rho_l - \rho_v)k^3gh_{fg}}{\mu l(t_{sat} - t_s)} \right]^{\frac{1}{4}} \quad (14)$$

The Nusselt solution derived above is an approximation since laboratory results have shown that it produces results that are approximately 20% lower than the measured values. McAdams suggested using a value of 1.13 in place of the factor 0.943, thus,

$$\bar{h} = 1.13 \left[ \frac{\rho_l(\rho_l - \rho_v)k^3gh_{fg}}{\mu l(t_{sat} - t_s)} \right]^{\frac{1}{4}} \quad (15)$$

While the above equation is used, it can be noted that all fluid properties are evaluated at temperature  $\left[ \frac{t_{sat} - t_s}{2} \right]$  and  $h_{fg}$  must be evaluated at  $t_{sat}$ .

Total heat transfer to the surface,

$$Q = h A_s(t_{sat} - t_s) \quad (16)$$

total condensation rate,

$$m = \frac{Q}{h_{fg}} = \frac{h A_s(t_{sat} - t_s)}{h_{fg}} \quad (17)$$

Figure 5 below shows the variation of film thickness and film modulus with plate height. Figure 6 below shows condensation on an inclined surface.



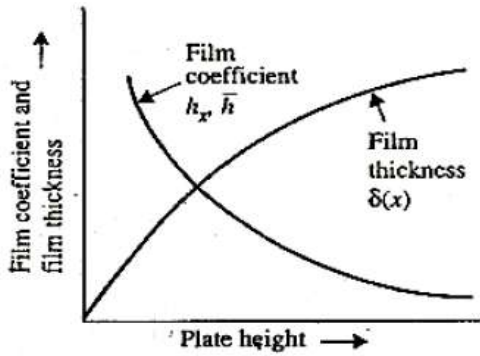


Figure 5: The Variation of Film Thickness and Film Modulus with Plate Height

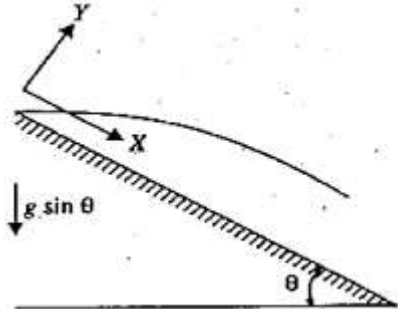


Figure 6: Condensation on An Inclined Surface

The thickness of the film increases as the plate height increases. The heat transfer rate decreases with increasing plate height since thermal resistance increases with increasing film thickness.

**(e) Inclined Flat Plate Surface**

For inclined flat surfaces, the gravitational acceleration  $g$  in equation (15) is replaced by  $g \sin \theta$ , where  $\theta$  is the angle between the surface and the horizontal (refer to Figure 6). Equation (15) is modified as follows:

$$h_{inclined} = 1.13 \left[ \frac{\rho_l (\rho_l - \rho_v) k^3 (g \sin \theta) h_f g}{\mu l (t_{sat} - t_s)} \right]^{\frac{1}{4}} \quad (18)$$

$$\text{or } h_{inclined} = h_{vertical} \times (\sin \theta)^{\frac{1}{4}} \quad (19)$$

Equation (19) is only applied for cases where  $\theta$  is small, and is not applicable at all for a horizontal plate.

**3.4 Turbulent Film Condensation**

When the plate where condensation takes place is elongated or the liquid film is robust enough, the flow of the condensate may become turbulent. Turbulence enhances heat transfer rates because heat is conveyed not just through condensation, but also via eddy diffusion. The criterion for this transition can be described using the Reynolds number, which is defined as follows:

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

Where:  $D_h$  = hydraulic diameter,  $u_m$  = Average flow velocity

$$D_h = 4 \times \frac{\text{cross sectional area of liquid flow}}{\text{wetted periphery}} = \frac{4A}{p}$$

Figure 7 below shows areas of film or laminar condensation on a vertical surface.

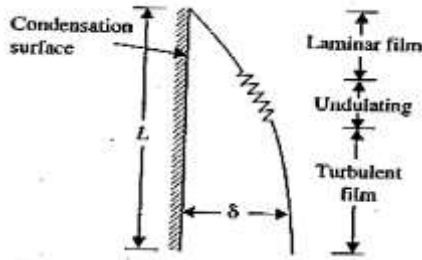


Figure 7: Areas of film or laminar condensation on a vertical surface

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  $\tau$ , such that:

$$Re = \frac{4\tau}{\mu_e} \quad (21)$$

With  $\tau = 0$  at the top of the plate and  $\tau$  increases with  $x$ .

The Reynolds number can also be linked to the heat transfer coefficient as follows:

$$\begin{aligned} Q &= \bar{h} A_s (t_{sat} - t_s) = \dot{m} h_{fg} \\ \dot{m} &= \frac{Q}{h_{fg}} = \frac{\bar{h} A_s (t_{sat} - t_s)}{h_{fg}} \\ Re &= \frac{4\bar{h} A_s (t_{sat} - t_s)}{h_{fg} \rho \mu_l} \quad (22) \end{aligned}$$

For a plate,  $A = L \times B$  and  $p = B$ , where  $L$  and  $B$  are the height and width of the panel respectively. Therefore,

$$Re = \frac{4\bar{h} L (t_{sat} - t_s)}{h_{fg} \mu_l} \quad (23)$$

When the value of  $Re$  exceeds 1800 (approximately), turbulence will appear in the fluid layer.

For  $Re > 1800$ , the following cross-correlation can be used:

$$\bar{h} = h_{turb} = 0.0077 \left[ \frac{\rho_l (\rho_l - \rho_v) k^3 g}{\mu_l^2} \right]^{\frac{1}{3}} (R_l)^{0.4} \quad (24)$$

### 3.5 Film Condensation on Horizontal Tubes

Nusselt analysis of film laminar condensation on horizontal tubes leads to the following relationships:

$$\bar{h} = 0.725 \left[ \frac{\rho_l (\rho_l - \rho_v) k^3 g h_{fg}}{\mu_l (t_{sat} - t_s) D} \right]^{\frac{1}{4}} \quad (25)$$

For a single horizontal tube,

$$\bar{h} = 0.725 \left[ \frac{\rho_l (\rho_l - \rho_v) k^3 g h_{fg}}{N \mu_l (t_{sat} - t_s) D} \right]^{\frac{1}{4}} \quad (26)$$

A horizontal tube with  $N$  tubes placed directly one above the other in the vertical direction.

Where,  $D$  = outer diameter of the pipe.

### 3.6 Slide Condensation in Horizontal Tubes (Film Condensation Inside Horizontal Tubes)

Condensing steam within pipes plays a crucial role in various engineering applications, including condensers found in refrigeration systems, air conditioning units, and numerous chemical and petrochemical industries. The processes occurring within these pipes are highly intricate, as the overall steam flow rate significantly influences both the heat transfer rate and the condensation rate on the pipe walls.

In 1962, Chato proposed the implementation of a specific mutual connection for low velocities in horizontal tubes, particularly for refrigerant condensation.

$$\bar{h} = 0.555 \left[ \frac{\rho_l(\rho_l - \rho_v)k^3 g h'_{fg}}{\mu_l D(t_{sat} - t_s)} \right]^{\frac{1}{4}} \quad (27)$$

$$\text{Where, } h'_{fg} = h_{fg} + \frac{3}{8} c_{pl}(t_{sat} - t_s) \quad (28)$$

Equation (28) is limited to a low steam Reynolds number such that:

$$Re_v = \left[ \frac{\rho_v u_{mv} D}{\mu_v} \right] < 3500$$

Where  $Re_v$  is evaluated at pipe entry states.

### 3.7 Impact of Non-Condensable Gases

The presence of a non-condensable gas, such as air, within a vapor that is subject to condensation can significantly affect the heat transfer coefficient. Research indicates that even a small volume percentage of air in the vapor can lead to a reduction of over 50% in the condensation heat transfer coefficient. This phenomenon occurs because, during the condensation of vapor mixed with a non-condensable gas, the gas remains at the surface. As a result, further condensation at the surface can only happen once the incoming vapor diffuses through the layer of non-condensable gas that accumulates in the vicinity. This neighboring gas acts as a thermal barrier, impeding the condensation process. Consequently, the rate of condensation declines markedly when the steam is mixed with even tiny amounts of non-condensable gases.

Given the detrimental effects that non-condensable gases have on condensing steam, it is essential for design practices to prioritize the effective venting of these gases whenever possible. Design practice should include venting the non-condensable gas as much as possible.

### 4. Conclusion

Dropwise condensation (DWC) is a phenomenon where vapor condenses into individual droplets, resulting in a heat transfer efficiency that is 5 to 7 times higher than that of filmwise condensation. To enhance and maintain DWC, researchers are exploring various surface treatments and chemical agents. Their efforts focus on boosting energy efficiency and promoting environmental sustainability across various industrial sectors, such as power generation, refrigeration, and cooling for electronics.

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