CONDENSATION HEAT TRANSFER
CONSIDERING ACTUAL FILM SURFACE

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ABSTRACT
This paper is concerned with the investigation of the effect of the nature of the surface of the condensate film on heat transfer during laminar film condensation. An idealized rippled nature of the film surface is proposed. Local heat transfer coefficients are calculated for film condensation on a vertical plate in laminar case. Calculations are performed numerically. For the proposed rippled nature of the film surface, the average heat transfer coefficients are up to 20% higher than that obtained for smooth condensate surface.

INTRODUCTION
Total or partial condensers for single vapor or mixture is one of the important problems in the chemical engineering. Has been described and analyzed by simplifying assumptions such as:
The liquid film are ignored;
Distribution through the liquid film is quid subcooling is not included.
Later improved by many investigators to
account for the above mentioned effects. Also boundary layer analysis has been applied to film condensation. Boundary layer analysis improves the accuracy of the representation of the film condensation process by including convection terms in the liquid layer.

In many circumstances, actual heat transfer rates during film condensation are substantially higher than predicted ones [2]. These discrepancies have been explained to arise mainly because the behavior of the actual film differs from that assumed. Many actual films flow in a rippled manner [4,5]. These ripples often arise because of such disturbances as uneven, though small, vapor velocities or as a result of condensate drainage from higher surfaces. The effect of this rippled structure is assumed to increase the heat transfer rates. Considering Fig. 1, the sensible heat flux \( q_g \) from the bulk of gas phase is given by:

\[
q_g = h_g (t_g - t_s)
\]

where \( h_g \) is the dry gas heat transfer coefficient, which is generally evaluated as if the gas phase were flowing alone [3]. The local overall heat transfer coefficient \( U \), defined by

\[
q_l = U (t_g - t_w) = h_f (t_s - t_w)
\]

is then given by:

\[
\frac{1}{U} = \frac{1}{h_f} + \frac{q_g}{q_l h_g}
\]

where \( q_l \) is the total heat flux and \( h_f \) is the condensate film heat transfer coefficient.

\[\text{Fig. (1) Temperature distribution in partial or total condensers}\]

To determine the dry gas heat transfer coefficient \( h_g \), there are many correlations which give acceptable values for laminar and turbulent flow of gas phase [1,2]. Such correlations apply to both smooth and rough surfaces.
Now, in the process of film condensation which is the major concern of this paper, the surface of the condensate film builds on the tube wall, whose structure affects both the dry gas heat transfer coefficient $h_g$ and the condensate film heat transfer coefficient $h_f$. The effect on the dry gas heat transfer may be compared with the effect of the surface roughness with two main differences (4):

1- The condensate has a velocity relative to the vapor phase; and 2- the ripple characteristics are not constant along the way of flow.

To account for this effect, Hampel (4) developed the following correlation to determine the friction factor to be used in the dry gas heat transfer correlations:

$$ f = f_0 \left[ 1 + 17.2 \left( \frac{\delta_f}{d_1} \right)^{0.9} \right] $$

where $\delta_f$ is the film thickness, $d_1$ is the tube inside diameter, and $f_0$ is the friction factor for smooth surface, which is function of Reynolds number $Re$.

On the other hand, the effect of the condensate film structure on the condensation heat transfer coefficient $h_f$ will be developed and described in the following section.

**FILM HEAT TRANSFER WITH PARTIALLY ACTUAL FILM SURFACE**

Consider a flat plate of length $L$ whose exposed face is at uniform temperature $T_w$ and which is inclined at an angle $\phi$ with the horizontal. Figure (2) illustrates a proposed idealized actual wavy film surface.

![Diagram](image)

**Fig. (2)** The proposed actual condensate surface nature for laminar film condensation on a plate.
The actual film thickness \( y_{ax} \) at any distance \( x \) along the plate in the direction of flow can be expressed by the following relation:

\[
y_{ax} = y_x \left[ 1.0 - \varepsilon \sin \left( \frac{2\pi x}{p} \right) \right]
\]  

In Eq. (5) \( \varepsilon \) is a factor less than unity, \( p \) is the period of the proposed wave, and \( y_x \) is the mean thickness of the condensate film evaluated on the basis of Nusselt theory for laminar flow film condensation and is given by:

\[
y_x = \left[ \frac{4 k_1 \mu_1 (l - t_w)}{\rho_1 (\rho_1 - \rho_g) g h_{fg} \sin \phi} \right] \left[ \frac{1}{x} \right]^{1/4}
\]  

It is recommended to use modified latent heat of the form \( h_{fg} = h_{fg} + 0.69 c_p (l_g - t_w) \) in lieu of \( h_{fg} \) in Eq. (6) [8].

According Nusselt theory, the local heat transfer coefficient \( h_x \) is given by:

\[
h_x = k_1 / y_x = \left[ \frac{\rho_1 (\rho_1 - \rho_g) g h_{fg} \sin \phi}{4 \mu_1 (l - t_w)} \right] \left( 1 / x \right)^{1/4}
\]  

Now, to calculate the local actual heat transfer coefficient \( h_{ax} \) which considers the surface structure one substitutes in Eq. (7) for \( y_x \) instead of \( y_x \). The average actual heat transfer coefficient \( h_a \) for the plate of length \( L \) is then given by:

\[
h_a = \frac{1}{L} \int_0^L \left( k_1 / y_x \right) \, dx
\]

\[
= \frac{1}{L} \int_0^L y_x \left( 1.0 - \varepsilon \sin \left( \frac{2\pi x}{p} \right) \right) \, dx
\]  

Since \( y_x \) is a function of \( x^{1/4} \), the above integration can be performed only numerically. In this work, numerical integration is performed using the trapezoidal rule. On the other hand, the integration can be performed analytically for \( \varepsilon = 0 \) (smooth film surface) to get the average heat transfer coefficient \( h \). The result is given by:

\[
h = 0.043 \left[ \frac{\rho_1 (\rho_1 - \rho_g) g h_{fg} \sin \phi}{L \mu_1 (l - t_w)} \right]^{1/4} = \frac{4}{3} h_L
\]

To investigate the effect of the proposed actual surface structure described by Eq. (5), condensation over a vertical plate of length \( L \) is considered, whose surface is maintained at 71°C. Vapor phase is steam at 0.5 bar and condenses in a filmwise manner. The following thermophysical properties are used:
RESULTS AND DISCUSSION

For comparison, the analytical solution of the problem according to Nusselt theory for normal condensate surface (ε = 0) is obtained and the results are plotted in Fig. (3). The figure illustrates the behavior of the local heat transfer coefficient $h_x$ and the local condensate film thickness $y_x$ along the plate.

According to the figure, the heat transfer decreases in the direction of the flow along the plate because the film thickness is increasing. The heat transfer coefficient at the end of the plate is 0.033 W/m$^2$.deg, while the average heat transfer coefficient $h$ for normal surface (Eq. (3)) is 0.072 W/m$^2$.deg.

On the other hand, the problem is solved numerically according to the proposed procedure which takes effect of the actual film surface nature into consideration. In the process of numerical integration, the distance step $Δx$ must be too small ($Δx > L/1000$) to account for the infinite heat transfer coefficient at the leading edge of the plate. The obtained results are plotted on figures 4-6 and some selected values are listed in the table below.

Figure 4 illustrates the local heat transfer coefficient and film thickness for specified surface parameters $ε = 0.2$ and $p = L/3$, that is the wave is repeated three times along the length L. Generally, it is clear that the local heat transfer coefficient $h_x$ decreases along the direction of flow because the local film thickness tends to increase in the same direction (Eq. (5)). In the same time, the behavior of local heat transfer coefficient gains the wavy characteristics as the film thickness. Most noticeable is the value of the actual average heat transfer coefficient $h_a$. The predicted value of $h_a$ for $ε = 0.2$ and $p = L/3$ is 0.072 W/m$^2$.deg., which is 2.0% higher than that obtained for smooth film surface $h_a = 0.072$ W/m$^2$.deg.). For the same $ε$ (0.2) and for $p = L$ (one wave over the length L of the plate), the predicted value of $h_a$ is 0.0342 W/m$^2$.deg., which is 3.3% higher than that obtained for normal surface. This result is physically acceptable because as $p$ increases the film surface suffers more local thinning or thickening.

According to Fig. 5, it is clear that the combined effect of both parameters $ε = 0.3$ and $p = L/3$ is great. The average heat transfer coefficient $h_a$ in this case rises to the value 0.088 W/m$^2$.deg.

Figure 6 illustrates the obtained results for $ε = 0.8$ and $p = L$, which means deeper thinning and thickening of the surface. The
predicted value of $h_\alpha$ for this case is 0.067 W/m$^2$.deg, which is about 20% higher than the value of $h$ for smooth surface ($0.072 \text{ W/m}^2$.deg). Comparison between the above obtained values leads to the fact that the factor $\varepsilon$ is more effective than the period $p$. In addition, a 21% increase in the average heat transfer rate is reported in [7] where the wave motion has been considered to be sinusoidal function in time at any distance $x$.

Examining the values in the table, it is found that in the extreme case—where $\varepsilon = 0.02$—the average heat transfer rate approaches the value of 0.094 W/m$^2$.deg. This value is in the order of the heat transfer coefficients in case of dropwise condensation. This result is satisfactory because in this case ($\varepsilon = 0.0$) the heat transfer mode changes from filmwise to dropwise condensation.

Average heat transfer coefficients for different actual surface parameters ($\varepsilon$ and $p$)

<table>
<thead>
<tr>
<th>$\varepsilon$</th>
<th>$\frac{L}{3}$</th>
<th>$\frac{L}{2}$</th>
<th>$L$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0</td>
<td>0.072</td>
<td>0.072</td>
<td>0.072</td>
</tr>
<tr>
<td>0.1</td>
<td>0.110</td>
<td>0.120</td>
<td>0.142</td>
</tr>
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<td>0.2</td>
<td>0.270</td>
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<td>0.342</td>
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<td>0.5</td>
<td>0.506</td>
<td>0.560</td>
<td>0.857</td>
</tr>
<tr>
<td>0.9</td>
<td>1.029</td>
<td>1.042</td>
<td>1.0594</td>
</tr>
</tbody>
</table>

CONCLUSION

In filmwise condensation, actual heat transfer rates are substantially higher than predicted ones. One reason for this is the structure of the actual surface of the condensate film. Actual films flow in a rippled (wavy) manner. According to the proposed physical surface nature of the condensate film the heat transfer rate rises up to 20% higher than the corresponding value for normal smooth film surface.

REFERENCES


NOMENCLATURE

c_p Specific heat (KJ/Kg.deg.)
f Friction factor
g Gravitational acceleration (m/s^2)
h Heat transfer coefficient (W/m^2.deg)
h_{lg} Latent heat (KJ/Kg)
k Thermal conductivity (W/m.deg)
L Plate length (m)
q Heat flux (W/m^2)
T Temperature (°C)
x Distance along the plate (m)

Subscripts
a actual
f film
l liquid
s film surface/saturation
g gas
w wall
Fig. (3) Development of condensate film thickness and heat transfer coefficient for smooth condensate surface.

Fig. (4) Development of condensate film thickness and heat transfer coefficient for rough condensate surface ($a = 0.2, p = L/3$).
Fig. (3) Development of condensate film thickness and heat transfer coefficient for rough condensate surface \( (e = 0.3, \rho = L/3) \)

Fig. (4) Development of condensate film thickness and heat transfer coefficient for rough condensate surface \( (e = 0.3, \rho = L) \)