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Mohamed Mahgoub
Associate Professor., Mechanical Power Engineering Department., Faculty of Engineering., El-Mansoura University., Mansoura., Egypt., magnerm@mans.edu.eg

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CONDENSATION HEAT TRANSFER CONSIDERING ACTUAL FILM SURFACE

BY

M. Mahgoub
Mansoura University
Mansoura, EGYPT

انتقال الحراره بالتكثيف مع اخذ السطح الحقيقي لطبقة التكثيف في الاعتبار

خلاصه - يهدف هذا البحث النظري الى دراسة تأثير الطبيعه العمليه لسطح طبقة التكثيف على معامل الانتقال الحراري اثناء التكثيف الرقائقي للابخره ، وقد تم في هذا البحث فرض طبيعه موجيه لسطح طبقة التكثيف مع تطبيق نظرية نسلت للتكثيف الرقائقي . ولقد دلت النتائج على ان معامل الانتقال الحراري يزداد كلما كان تردد الموجه التي تمثل السطح كبيرا ، وهذه الزياده تصل الى حوالي ٢٠ ٪ من معامل الانتقال الحراري لسطح مستوي بدون مثل هذه الاضرابات الموجيه .

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.

...RODUCTION

...f 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;
... tribution 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 :

$$q_g = h_g (t_g - t_s) \quad (1)$$

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

$$q_t = U (t_g - t_w) = h_f(t_s - t_w) \quad (2)$$

is then given by :

$$\frac{1}{U} = \frac{1}{h_f} + \frac{q_g}{q_t h_g} \quad (3)$$

where q_t is the total heat flux and h_f is the condensate film heat transfer coefficient.

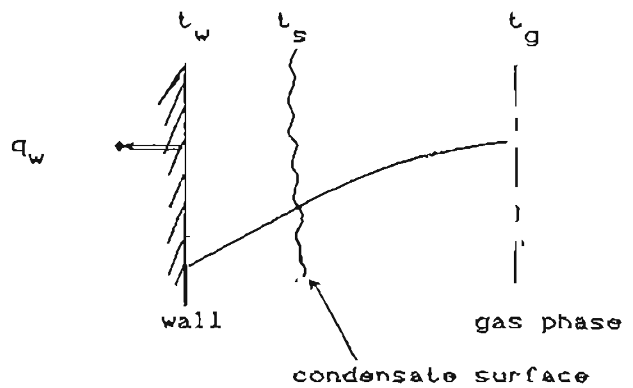

Fig.(1) Temperature distribution in partial or total condensers

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:

$$f = f_o\left[1 + 17.2\,(\delta_f/d_i)^{0.9}\right] \qquad (4)$$

where δ_f is the film thickness, d_i is the tube inside diameter, and f_o is the friction factor for smooth surface, which is function of Reynold 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 ϕ with the horizontal. Figure (2) illustrates a proposed idealized actual wavy film surface.

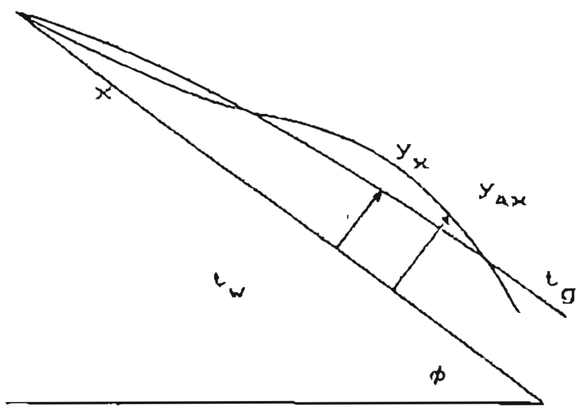

Fig.(2) The proposed actual condensate surface nature for laminar film condensation on a plate

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 :

$$y_{ax} = y_x \left[1.0 - \varepsilon \sin (2\pi x/p) \right] \qquad (5)$$

In Eq.(5) ε 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 :

$$y_x = \left[\frac{4\, k_1 \mu_1 (t_g - t_w)}{\rho_1 (\rho_1 - \rho_g)\, g\, h_{fg} \sin\phi} \right]^{1/4} (\, x\,)^{1/4} \qquad (6)$$

It is recommended to use a modified latent heat of the form h'_{fg} = h_{fg}+ 0.68 $c_{p,1}(t_g - t_w)$ in lieu of h_{fg} in Eq.(6) [6].

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

$$h_x = k_1/y_x = \left[\frac{\rho_1 (\rho_1 - \rho_g)\, k_1^3 g\, h_{fg} \sin\phi}{4\, \mu_1 (t_g - t_w)} \right]^{1/4} (1/x)^{1/4} \qquad (7)$$

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

$$h_a = (1/L) \int_0^L (k_1/y_{ax})\, dx$$

$$= (k_1/L) \int_0^L \frac{dx}{y_x (1.0 - \varepsilon \sin (2\pi x/p))} \qquad (8)$$

Since y_x is 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 ε = 0 (smooth film surface) to get the average heat transfer coefficient h . The result is given by :

$$h = 0.943 \left[\frac{\rho_1 (\rho_1 - \rho_g)\, k_1^3 g\, h_{fg} \sin\phi}{L\, \mu_1 (t_g - t_w)} \right]^{1/4} = \frac{4}{3} h_L \qquad (9)$$

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 ^oC. Vapor phase is steam at 0.5 bar and condenses in a filmwise manner. The following thermophysical properties are used :

$t_g = 82\ ^\circ C$, $t_w = 71\ ^\circ C$, $k_l = 670\times10^{-3}$ W/m.deg.

$h_{fg} = 2304$ KJ/Kg $\mu_l = 3510\times10^{-7}$ Kg/m.s

$\rho_l = 970.5$ Kg/m^3 $\rho_g = 0.3085$ Kg/m^3

$g = 9.81$ m/s^2

In this analysis liquid properties are assumed to be constant through the film thickness, i.e temperature independent.

RESULTS AND DISCUSSION

For comparison, the analytical solution of the problem according to Nusselt theory for normal condensate surface ($\epsilon = 0$) 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 6053 W/m^2.deg. while the average heat transfer coefficient h for normal surface (Eq.(9)) is 8072 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 $\epsilon = 0.2$ 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_{ax} 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 $\epsilon = 0.2$ and p =L/3 is 8287 W/m^2.deg., which is 2.8% higher than that obtained for smooth film surface (h = 8072 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 8342 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 =0.5 and p = L/3 is great. The average heat transfer coefficient h_a in this case rises to the value 9508 W/m^2.deg.

Figure 6 illustrates the obtained results for $\epsilon = 0.8$ and p = L, which means deeper thinning and thickening of the surface. The

predicted value of h_a for this case is 9667 W/m^2.deg. which is about 20% higher than the value of h for smooth surface (8072 W/m^2.deg). Comparison between the above obtained values leads to the fact that the factor ε 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 funtion in time at any distance x.
Examining the values in the table, it is found that in the extreme case- where ε = 0.9- the average heat transfer rate approaches the value of 19694 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 (ε = 0.9) the heat transfer mode changes from filmwise to dropwise condensation.

Average heat transfer coefficients for different actual surface parameters (ε and p)

	p		
ε	L/3	L/2	L
0.0	8072	8072	8072
0.1	8110	8120	8142
0.2	8278	8229	8342
0.5	9506	9560	9667
0.9	19258	19420	19694

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.

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NOMENCLATURE

c_p	Specific heat (KJ/Kg.deg.)		
f	Friction factor	y, δ	Film thickness (m)
g	Gravitational acceleration (m/s^2)	μ	Viscosity (Kg/m.s)
h	Heat transfer coefficient (W/m^2.deg)	ε, p	Surface paramaters
h_{fg}	Latent heat (KJ/Kg)	ρ	Density (Kg/m^3)
K	Thermal conductivity (W/m.deg.)	ϕ	Angle
L	Plate length (m)	Re	Reynolds number
q	Heat flux (W/m^2)		
t	Temperature (^{o}C)		
x	Distance along the plate (m)		

Subscripts

a	actual	f	film	g	gas
l	liquid	s	film surface/saturation	w	wall

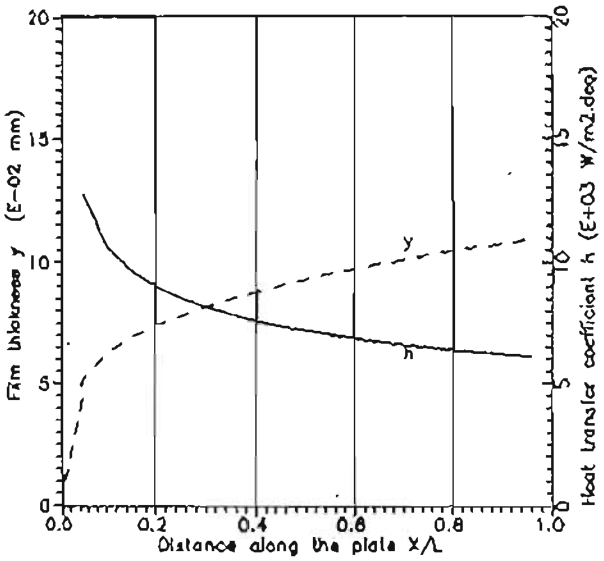

Fig. (3) Development of condensate film thickness and heat transfer coefficient for smooth condensate surface

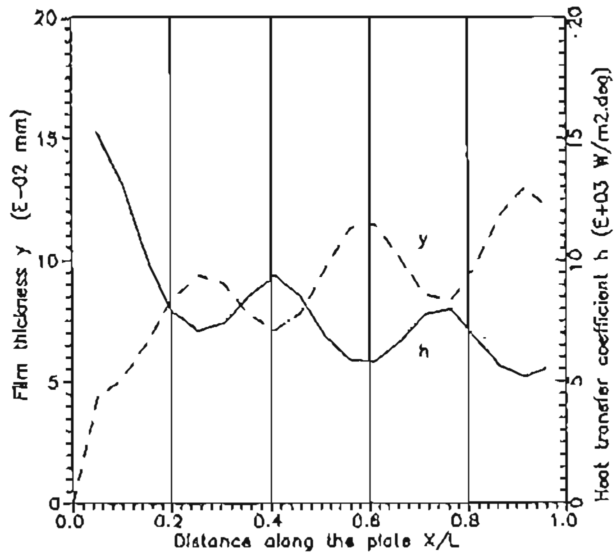

Fig. (4) Development of condensate film thickness and heat transfer coefficient for rough condensate surface (e = 0.2, p =L/3)

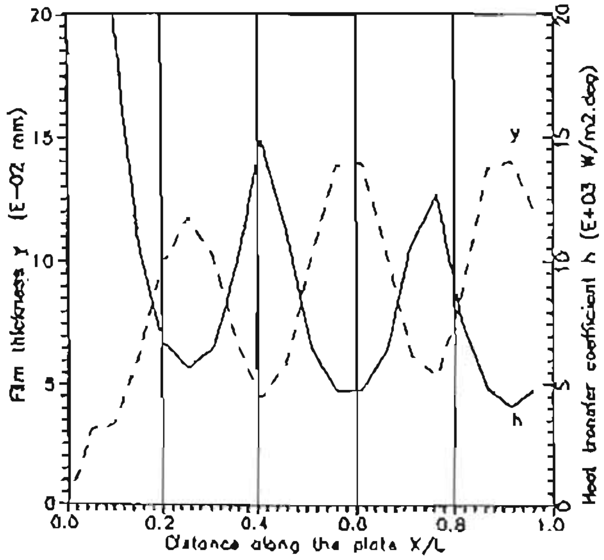

Fig. (5) Development of condensate film thickness and heat transfer coefficient for rough condensate surface (a = 0.5, p =L/3)

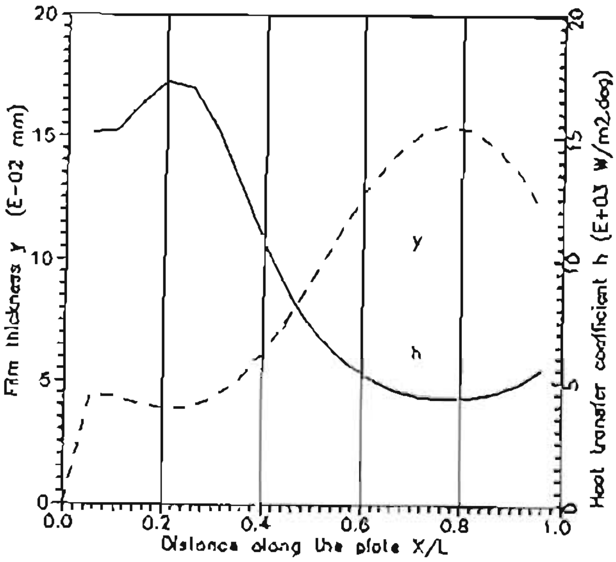

Fig. (6) Development of condensate film thickness and heat transfer coefficient for rough condensate surface (a = 0.5, p =L)