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HEAT TRANSFER CHARACTERISTICS FOR LAMINAR FILMWISE CONDENSATION ALONG A FLAT VERTICAL PLATE WITH THREE DISTINCT COOLING ZONES

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ABSTRACT

For this paper, we studied laminar filmwise condensation along a flat plate with variable surface temperature by using Nusselt's theory. We calculated the local and averaged heat transfer coefficients on a condensing surface temperature on the heat transfer characteristic. Furthermore, heat transfer coefficients were higher for up-flow than for down-flow of water.

NOMENCLATURE

C_{pk}	:	Specific heat of liquid water
g	:	Gravitational acceleration
H_k	:	Local heat transfer coefficient
H_m	:	Average heat transfer coefficient
Nu_k	:	Local Nusselt number
Q_k	:	Local heat flux
Q_m	:	Average heat flux
S	:	Latent heat
T	:	Dimensionless temperature
t	:	Temperature
t_k	:	Surface temperature
t_s	:	Saturation temperature
u	:	Velocity of condensate on X axis
v	:	Velocity of condensate on Y axis
X_k	:	Location on X axis
X_p	:	Location of the thermal boundary region II on the condensing surface
ΔX	:	Length of region II
Y_k	:	Condensate film thickness
K_k	:	Thermal diffusivity
λ_k	:	Thermal conductivity
μ_k	:	Viscosity of water
ν_k	:	Kinematic viscosity of water
ρ_k	:	Density of water
ρ_v	:	Density of water vapour

INTRODUCTION

Filmwise condensation is the predominant form of the condensation phenomenon in heat exchangers. It has been investigated extensively both theoretically and experimentally.

Nusselt was the first person to analyze filmwise condensation theoretically. A number of papers have been reported since his pioneering paper. It is well known that the condensate film thickness greatly influences the heat transfer characteristics. Generally, the condensate film thickness is calculated assuming that the temperature is constant over the condensing surface. In practice, the local temperature on the condensing surface varies from location to location.

In this paper, an analysis is presented of the laminar filmwise condensation along a flat plate with variable surface temperatures using Nusselt's theory. The heat transfer characteristics of filmwise condensation was discussed. Furthermore, we also investigated the effects depending on whether the cooling water was flowing co- or counter-current to the direction of the condensate flow. There are very few papers in the literature which study this aspect of the filmwise condensation [1,2].

ANALYSIS

General Equations and Model

Classical analysis [3] of laminar filmwise flow on a flat plate (Figure 1) leads to the following equations for continuity, momentum and energy balances.

Vapour at saturation temperature t_s is condensed on a vertical flat plate a surface temperature t_k. Defining film thickness Y_k at location X_k from the upper edge, the temperature within the film, the velocity in the *X* direction and the velocity in the *Y* direction as *t*, *u* and *v* respectively we may write:

Continuity Equation,

$$\frac{\partial u}{\partial x}+\frac{\partial \upsilon}{\partial y}=0 \qquad (1)$$

Momentum Equation,

$$\rho_k\left(u\frac{\partial u}{\partial x}+\upsilon\frac{\partial u}{\partial y}\right)=\mu_k\frac{\partial^2 u}{\partial y^2}+g(\rho_k-\rho_\upsilon) \qquad (2)$$

Energy Equation,

$$C_{pk}\cdot\rho_k\left(u\frac{\partial t}{\partial x}+\upsilon\frac{\partial t}{\partial y}\right)=\lambda_k\frac{\partial^2 t}{\partial y^2} \qquad (3)$$

The heat balance on the condensate layer between X=0 and X_k may be written:

$$S\int_0^{Y_k} \rho_k u dy + \int_0^{Y_k} \rho_k u C_{pk}(t_s - t)dy = \int_0^{X_k} \lambda_k \left(\frac{\partial t}{\partial y}\right)_{y=0} dx \qquad (4)$$

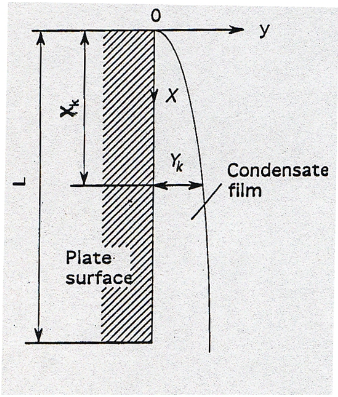

Figure 1: Filmwise condenstation

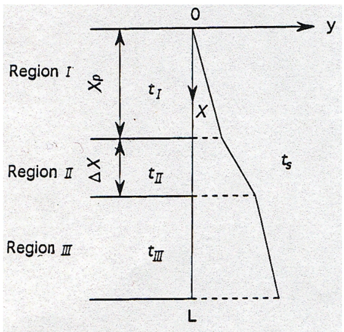

Figure 2: Calculation model.

Table 1 Calculated condition and surface temperatures

Condition	Region I	II(ΔX)	III	Length of ΔX
	T_I	T_{II}	T_{III}	
(A)	0.2	0.5	0.8	1/5•L
(B)	0.8	0.5	0.2	1/5•L

(A) ; The case of flowing water from the bottom.

(B) ; The case of flowing water from the top.

T_I, T_{II}, T_{III}; Dimensionless temperatures of the condensing surface.

$T_I, T_{II}, T_{III} = \frac{(t_s - t_I, t_{II}, t_{III})}{t_s}$

t_s; Saturation temperature (373.15K).

t_I, t_{II}, t_{III}; Surface temperatures of three regions on the condensing surface.

L; Length of the condensing plate (1m).

Table 1: Calculated condition and surface temperatures.

where,

S : latent heat,

From Equation (1) to (4) and some assumptions, the film thickness Y_k at X_k is finally given by Eq. (5).

$$Y_k = \left\{ Y_{k-1}^4 + \frac{4C_{pk}(t_s - t_k)\nu_k K_k}{Sg(\rho_k - \rho_v)/\rho_k}(X_k - X_{k-1}) \right\}^{\frac{1}{4}} \qquad (5)$$

where,

ν_k : Kinematic viscosity of liquid water (=μ_k / ρ_k),

K_k : Thermal diffusivity (=$\lambda_k / (C_{pk} \cdot \rho_k)$).

subscript,

k : a calculation region when dividing a condensing surface,

k-1 : one more upper calculation region of k region.

The local heat transfer coefficient H_k is obtained from the following equation.

$$H_k = \frac{\lambda_k \left(\frac{\partial t}{\partial y}\right)_{y=0}}{t_s - t_k} = \frac{\lambda_k}{Y_k} \qquad (6)$$

The average value of the local heat transfer coefficient over the whole surface, H_m is given by:

$$H_m = \frac{1}{n}\sum_{k=1}^{n} H_k \qquad (7)$$

and, the local Nusselt number ($N u_k$), the average Nusselt number ($N u_m$), the local het flux (Q_k) and the average heat flux (Q_m) are, respectively, given by:

$$Nu_k = \frac{H_k L}{\lambda_k} \qquad (8)$$

$$Nu_m = \frac{1}{n}\sum_{k=1}^{n} Nu_k \qquad (9)$$

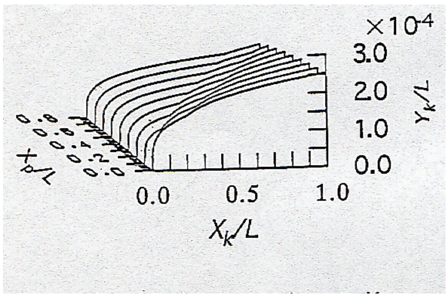

Figure 3: Boundary layer thickness Y_k
(T_I =0.2, T_{II} =0.5, T_{III} =0.8)

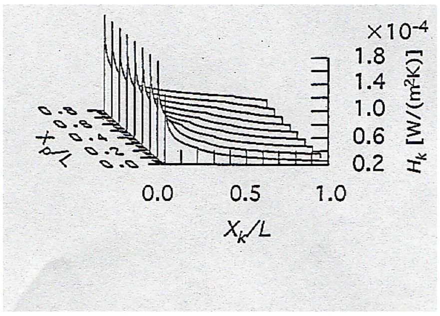

Figure 4: Local heat transfer coefficient H_k
(T_I =0.2, T_{II} =0.5, T_{III} =0.8)

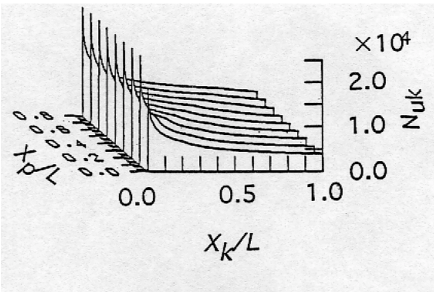

Figure 5: Nusselt number N_{uk}
(T_I =0.2, T_{II} =0.5, T_{III} =0.8)

$$Q_k = H_k(t_s - t_k) \tag{10}$$

$$Q_m = \frac{1}{n}\sum_{k=1}^{n} Q_k \tag{11}$$

Calculation Model and Non-Dimensionalization

The calculation model for the laminar filmwise condensation along a flat vertical plate with three distinct cooling zones is shown in Fig. 2. In Figure 2, X_p is the location of region II on the condensing surface and ΔX is the distance of X_p from the upper edge of the condensing surface. The heat transfer coefficient was calculated for $\Delta X = 1/5 \bullet L$.

We define dimensionless numbers X_k/L, X_p/L and Y_k/L, where X_k denotes the location under condensation relative to the top of the condensing surface at X_k and L is the height of the condensing surface. Dimensionless temperature T is defined by dividing the difference between the saturation stream temperature t_s and the local temperature t_k at the condensing surface by t_s.

In this paper, the condensing surface is vertically divided into three distinct cooling zones and the temperatures of the three regions are expressed at t_I, t_{II} and t_{III}. Table 1 shows the various calculation cases. Thermophysical properties are obtained at a temperature calculated as the mean of the saturation stream and the cooling surface temperatures.

CALCULATED RESULTS AND DISCUSSION

Cooling Water Flowing Up the Condensing Surface

The temperature on the condensing surface decreases as X_k increases. This condition corresponds to the calculation case A in Table 1.

The calculated results for ΔX=1/5•L are shown in Figures 3,4,5 and 6 (where L is the height in 1m of the condensing surface). Figure 3 shows the relation between X_k/L and Y_k/L with X_p/L as a parameter. As X_k/L increases, Y_k/L increases and the rate of increasing Y_k/L at the boundary regions increases further.

Figure 4 shows the distribution of H_k. Based on this model, the heat transfer coefficient is mainly influenced by the condensate film thickness. From this Figure, H_k decreases as X_k/L increases and H_k varies in inverse proportion to Y_k/L as indicated in Fig. 3. Furthermore, Figure 5 shows the distribution of Nusselt number Nu_k. The calculated results are similar to that of Fig. 4. The heat flux Ok in Fig. 6 decreases as X_k/L increases in the same thermal region. From this we obtain the increasing Q_k stepped at the thermal boundary regions. Note in particular, that Q_k suddenly decreases in the upper region I.

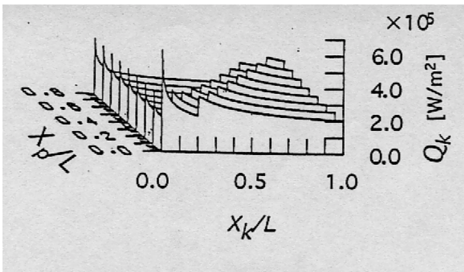

Figure 6: Heat flux Q_k
(T_I =0.2, T_{II} =0.5, T_{III} =0.8)

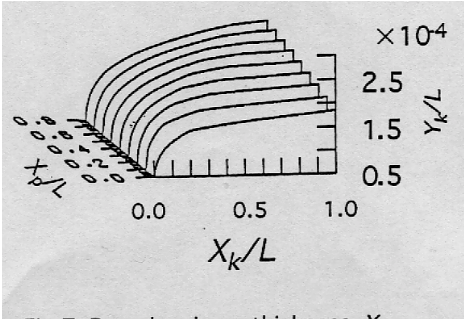

Figure 7: Boundary layer thickness Y_k
(T_I =0.2, T_{II} =0.5, T_{III} =0.8)

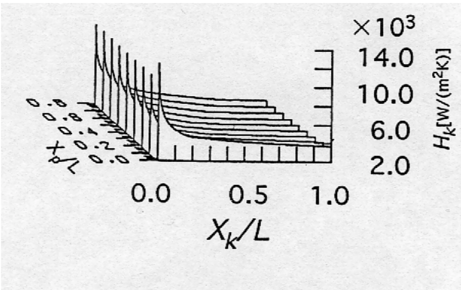

Figure 8: Local heat transfer coefficient H_k
(T_I =0.2, T_{II} =0.5, T_{III} =0.8)

Cooling Water Flowing Down the Condensing Surface

The calculated results for this case are tabulated as B in Table 1. The results (refer Figs. 7 to 10) were found using the same method as described under section 2.1. As may be seen in Fig. 7, Y_k/L increases with increasing X_k/L. However, the rate of increase in Y_k/L is smaller than that in the case of cooling water flowing up (see Fig. 3). As for the heat transfer coefficient H_k in Fig. 8, the value H_k is inversely proportional to the film thickness Y_k/L in Fig. 7 and H_k decreases with larger X_k/L. Note that Nusselt number Nu_k in Fig. 9 shows the same tendency to decrease as shown in Fig. 8.

The heat flux Q_k decreases with X_k/L as shown in Fig. 10, and the value of Q_k shows a tendency to decrease sharply at the thermal boundary surface. Comparing the heat flux with that in Fig. 6, it is evident that both heat fluxes show a similar tendency to decrease as X_k/L increases within the same thermal region. Note also that Q_k in Fig. 6 increases at the thermal boundary surface, while Q_k in Fig. 10 decreases at that surface. It is considered that the effect of the difference between the steam and surface temperatures at the thermal boundary surface on the heat flux is larger than that of the changing rate of the heat transfer coefficient causing a difference between the heat flux rate in the upward and downward flow of the cooling water.

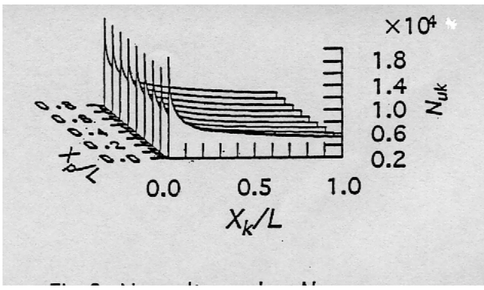

Figure 9: Nusselt number N_{uk}
(T_I =0.8, T_{II} =0.5, T_{III} =0.2)

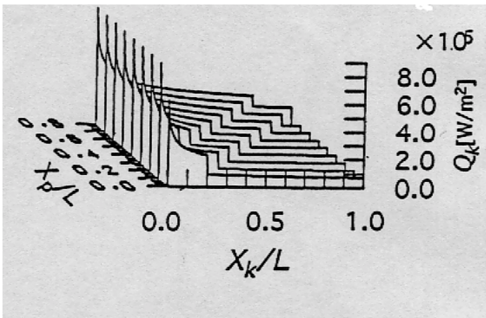

Figure 10: Heat flux Q_k
(T_I =0.8, T_{II} =0.5, T_{III} =0.2)

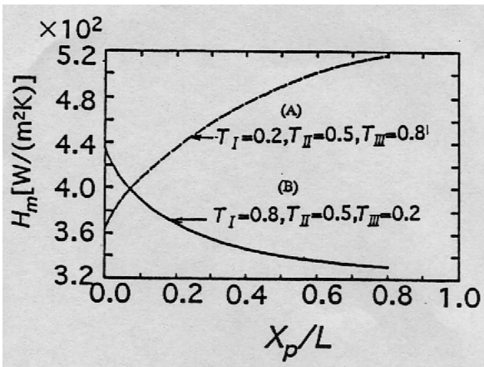

Fig.11 Averaged heat transfer coefficient H_m

Figure 11: Averaged heat transfer coefficient H_m.

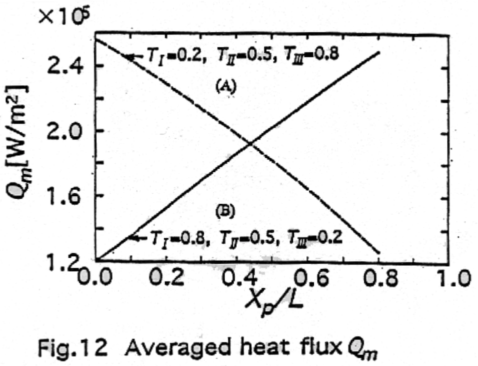

Figure 12: Averaged heat flux Q_m.

Average Heat Transfer Coefficient and Heat Flux on the Whole Condensing Surface

Figures 11 and 12 show an average heat transfer coefficient H_m and an average heat flux Q_m on the whole condensing surface respectively. The abscissa X_p indicates the location of the second thermal region II (length of the region ΔX=1/5•L) from the upper edge of Fig. 2

As X_p/L in Fig. 11 increases, H_m gradually increases under the calculation case A in Table 1 and H_m decreases in the case of the calculation case B. Except in the neighbourhood of the top region on the condensing surface, H_m for the case of the water flowing up (case A) shows a larger value than that of the water flowing down (case B). For example, H_m for case A increases about 30% more than that of the case B when T_{II}-0.5, ΔX=1/5•L and X_p/L=0.4. Also, according to this model, the Nusselt number shows a similar characteristic for the heat transfer coefficient.

As may be seen from Fig. 12 larger values of X_p/L results in a decrease of Q_m for the case A while it increases Q_m in the case of the case B. As may be seen from Fig. 12 with a larger X_p/L in the case of A, Q_m decreases and, in the case of the case B, Qm increases. At X_p/L=0.4 in Fig. 11, Q_m for the case A is about 10% larger than that of the case B. Under this case (X_p/L=0.4), for the present calculation model, H_m and Q_m are larger in the case of water flowing up the condensing surface.

CONCLUSION

The summary of the calculation results ins as follows:

1. Heat transfer characteristics on a condensing surface depend on the temperature distribution on the surface.

2. When Xp/L=0.4, the heat transfer with water flowing up is better than that for water flowing down.

ACKNOWLEDGMENT

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