KSME Journal, Vol. 1, No. 2, pp. 101~107, 1987. 

# FORCED CONVECTION LAMINAR FILM CONDENSATION AT A CHANNEL ENTRANCE

**Sung Hong Lee* and M.C.Yuen****

*(Received March 2, 1987)*

The influence of film condensation of a pure vapor on the laminar flow at the entrance of a symmetrical two-dimensional channel is studied on the basis of the integrated equations of change by the approximate integral method. As the condensation rate increases the vapor speed is drastically reduced which causes the inlet region to decrease in size. The reduction in vapor speed tends to support the growth of the vapor boundary layer in spite of the opposite effect of suction. The dependence of pressure, liquid film thickness, vapor bound-ary layer thickness and vapor quality is studied as a function of the streamwise coordinate and the condensation parameter, $C_{pl}(T_s-T_w)/(P_r \cdot H_{fg})$. Particular attention is paid to the influences of condensation on the interfacial shear stress, the temperature dependence of liquid viscosity and the orientation of the channel with respect to gravity.

**Key Words:** Momentum Integral, Channel Entrance, Temperature-Dependent Property, Interfacial Velocity, Condensation Parameter, Interfacial Shear Stress

## NOMENCLATURE

$a, b, c, d, e, f, g$ : Constant
$A_0, A_1, B_0, B_1, C_0$ : Coefficient
$C_p$ : Specific heat
$E=\dfrac{C_{Pl}(T_S-T_W)}{P_r \cdot H_{fg}}$ : Nondimensional condensation parameter
$g_x$ : Gravitational acceleration of the liquid film in $x$-direction
$H$ : Height of channel
$H_g$ : Enthalpy of steam
$H_{fg}$ : Heat of vaporization
$m''$ : Condensation rate
$P$ : Static pressure
$Pr$ : Prandtl number of liquid
$Re_o$ : Reynolds number of vapor based on $U_o$
$T_s$ : Temperature of saturation
$T_w$ : Temperature of wall
$U, V$ : Streamwise and normal velocity
$W$ : Mass flowrate
$W_t$ : Total mass flowrate of vapor and liquid
$x, y$ : Streamwise and vertical distance
$\alpha$ : Coefficient
$\delta$ : Film thickness of condensate
$\mu$ : Dynamic viscosity
$\rho$ : Density
$\tau$ : Shear stress

### SUBSCRIPTS

$f$ : Film
$g, l$ : Vapor, liquid phase quantity
$i$ : Liquid-vapor interfacial quantity
$o$ : Initial value quantity
$\infty$ : Free stream or core flow quantity
$t$ : Total

## 1. INTRODUCTION

Recently, there has been considerable interest in the study of rapid condensation of vapor. This problem is of particular relevance to nuclear reactor safety analysis, where in the hypothetical loss of coolant in a pressurized water reactor, steam comes into contact with cold water at several locations(Lim, et al. 1984). Quantitative prediction of these events requires knowledge of the local condensation phenomena at the liquid vapor interface(Lee 1983a, Kim and Bankoff, 1983; Lee and Bankoff, 1983).

Analytical models are developed to describe the forced convection laminar film condensation inside a horizontal and vertical two-dimensional channel entrance with constant wall temperature(Fig. 1). Similarity solutions of the two-phase boundary layer equations concerning laminar film condensation(Lee, 1987) on a horizontal flat plate were obtained by Cess(1960), Koh(1962) and Kim(1987) for the case of forced convection only i.e., no body force. Laminar film condensation of a pure vapor in a tube with constant fluid properties was studied by the approximate integral method by Lucas(1979). His approach will be extended by considering the temperature-dependent property of liquid viscosity(Lee, 1983 b).

Integrated forms of the momentum equations in the vapor and condensate flow with assumed velocity profiles are used. The analysis will, therefore, be only an approximation which is considered appropriate for the somewhat qualitative purpose of the investigations.

*Department of Machine Design Engineering, Pusan National University, Pusan 607, Korea
**Department of Mechanical Engineering, Northwestern University, Evanston, Illinois 60201, U.S.A.

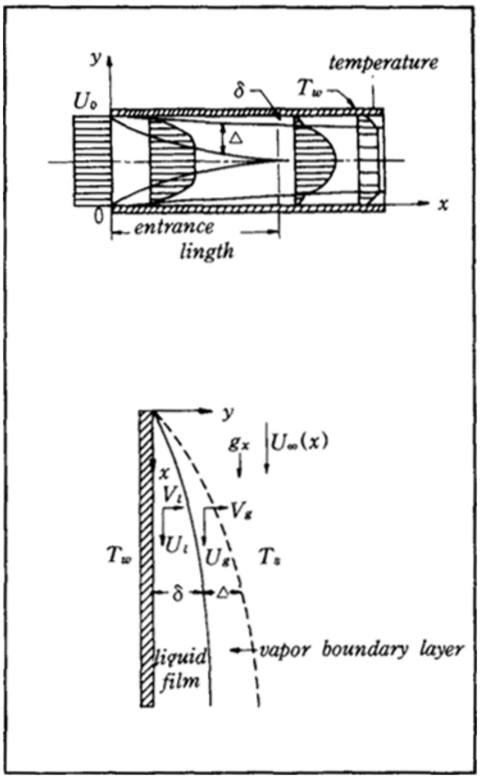


**Fig. 1** Physical model and coordinate system

## 2. PHYSICAL AND ANALYTICAL MODEL

### 2.1 Flow Model

The system of coordinates is shown in Fig. 1. The centerline is an axis of symmetry. For the horizontal channel, gravity is neglected. Flows are steady, laminar and wave-free. Mainstream flow is parallel to the channel direction($x$) and velocity distribution is uniform at the inlet. Boundary layer flows are assumed in the immediate neighborhood of the interphase of liquid-vapor and solid-liquid, while potential flow is assumed in the core region of the vapor flow. Pressure may be written as a function of axial distance, $x$

$$P = P_g(x) = P_l(x) \tag{1}$$

The inlet vapor is assumed to be saturated and is cooled on the wall at the channel entrance. The normal direction($y$) momentum jump due to mass conversion at the vapor-liquid interface, $\rho_g V_g^2 - \rho_l V_l^2$, is assumed to be negligible in comparison to other terms. For laminar, two-dimensional, steady flow buoyancy and energy dissipation effects are neglected. A body force by gravity($g_x$) is applied to the liquid film only. Momentum equation:

$$\rho\left(U\frac{\partial U}{\partial x} + V\frac{\partial U}{\partial y}\right) = -\frac{\partial P}{\partial x} + \frac{\partial \tau}{\partial y} + \rho g_x \tag{2}$$

Energy(liquid) equation:

$$\frac{\partial^2 T_l}{\partial y^2} = 0 \tag{3}$$

### 2.2 Properties

The variation of vapor density and other vapor properties can be assumed constant at the saturation temperature. Vapor density relative to liquid density may be neglected. All properties of liquid are assumed constant except the liquid viscosity. Viscosity of the liquid is assumed to depend significantly upon the temperature of the liquid. Drew(1938) suggested the following equation for most liquids over a moderate temperature range.

$$1/\mu_l = a + b\ T_l \tag{4}$$

### 2.3 Energy Equation and Temperature Distribution

Temperatures of the vapor core and channel wall are assumed to be uniform and fixed. Interfacial mass transfer resistance is negligibly small.

Therefore, the temperature at the liquid-vapor interphase and in the vapor core region is assumed to be the saturation temperature. Heat conduction in the axial flow direction, thermal convection, and viscous dissipation terms in the energy equation are neglected in both phases. This implies that the energy released by condensation is transferred by conduction through the liquid film. Even though vapor temperature is above the saturated temperature, heat transfer in the vapor does not have a strong influence on the result because the energy transferred to the interface by convection and conduction in the vapor phase is small relative to the heat of condensation. Thus the contribution of conduction from the vapor phase can be neglected. For the liquid phase, Bromley(1952) and Rohsenow(1956) estimated the effect of thermal convection within the film, and for the forced convection condensation, Cess and Koh indicated that the effect of thermal convection can be neglected if subcooling parameter(Jacob number), $C_{pl}(T_s - T_w)/H_{fg}$, is equal to or less than 0.2. For liquid Prandtl numbers from 1 to 10 and high values of property ratio, $\rho_l\mu_l/\rho_g\mu_g$, the convection effect is negligible for the whole range of the subcooling parameter. For most applications in refrigeration and air conditioning, the above conditions are satisfied and the effect of thermal convection may be neglected. In the steam-water condensation case, latent heat, $H_{fg}$, is very large, so the maximum subcooling parameter is approximately 0.2 at atmospheric pressure. Therefore, thermal convection may be neglected. The energy Eq. (3) for the condensate can be simplified, and the linear temperature distribution of the liquid is obtained as follows:

$$T_l = c + dy \tag{5}$$

### 2.4 Velocity Profiles in the Vapor and Liquid Phases

The velocity profile in the vapor phase is assumed to be parabolic at the entrance in the developing boundary layer as shown by Schiller(1922):

$$U_g = e + fy + gy^2 \tag{6}$$

For the forced-convection condensation, the incoming vapor may have a high velocity, which produces large shear stress at the liquid-vapor interface particularly at the entrance region.

Since the pressure gradient in a very thin, liquid film is small compared to the shear force, the pressure term can be neglected in the momentum equation for the film :

$$(dp/dx)\cdot\delta \ll \mu_l \partial U_l/\partial y \tag{7}$$

Cess and Koh indicated that the effect of the acceleration within the liquid film is neglible when the value of subcooling parameter, $C_{pl}(T_s-T_w)/P_r\cdot H_{fg}$, is much less than 30. By assuming a creeping motion in a thin liquid film (Nusselt, 1916) and neglible momentum change in the film, the following simplified equation for the condensate film can be suggested :

$$\frac{\partial}{\partial y}(\mu_l \partial U_l/\partial y) = \text{constant} = -\rho_l g_x \tag{8}$$

From the simplified liquid momentum Eq. (8) and the temperature distribution(Eq. 5) of the film, with the coupled liquid-vapor interfacial boundary conditions, the following velocity distribution in the liquid film can be obtained :

$$\begin{aligned} U_l(y) &= \int_0^y \frac{\tau_l}{\mu_l(T_l)}dy \\ &= \frac{\tau_i\delta}{\mu_S}\left[\eta-\left(1-\frac{\mu_S}{\mu_W}\right)\left(\eta-\frac{\eta^2}{2}\right)\right] \\ &+ g_x\frac{\rho_l\delta^2}{\mu_S}\left[\left(\eta-\frac{\eta^2}{2}\right)-\left(1-\frac{\mu_S}{\mu_W}\right)\left(\eta-\eta^2+\frac{\eta^3}{3}\right)\right] \end{aligned} \tag{9}$$

where $\eta=\dfrac{y}{\delta}$

and $U_i = U_l(y=\delta)$

## 3. METHOD OF SOLUTION

### 3.1 Integral Equations

The integral equations of the two-phase boundary layer flow provide the basis for a number of approximate methods of solution. The velocity profile for vapor boundary layer flow, $U_g$, is given above(Eq. 6), and the velocity distribution for liquid film may be deduced from the simplified momentum equation of the thin condensate film(Eq. 9). Temperature filed of the liquid phase, $T_l$, is also obtained from the simple energy equation of the thin film and is given as a linear profile(Eq. 5).

Upon integrating the governing conservation equations of continuity and vapor-momentum for steady, two dimensional, incompressible flow with respect to y from the wall ($y=0$) to a centerline ($y=H/2$) of the symmetric channel geometry, we obtain :

continuity of two phase flow is given by

$$\frac{d}{dx}\int_0^\delta \rho_l U_l dy + \frac{d}{dx}\int_\delta^{H/2}\rho_g U_g dy = 0 \tag{10}$$

Momentum(vapor) integral from Eq. (2) is

$$\begin{aligned}\int_\delta^{H/2} U_g\frac{\partial U_g}{\partial x}dy + \int_\delta^{H/2} V_g\frac{\partial U_g}{\partial y}dy \\ = -\int_\delta^{H/2}\frac{1}{\rho_g}\frac{dP}{dx} + \int_\delta^{H/2}\frac{1}{\rho_g}\frac{\partial \tau_g}{\partial y}dy\end{aligned} \tag{11}$$

Additionally, two more equations, the continous interfacial shear stress condition and the thermal boundary condition at the interface, will be utilized to solve the film condensation at the entrance by taking derivatives of the continuous interfacial shear stress equation with respect to $x$.

Energy balance

$$H_{fg}m'' = k\frac{\partial T_l}{\partial y}\bigg|_{y=\delta} \tag{12}$$

Shear stress

$$\frac{d}{dx}\left(\mu\frac{\partial U_b}{\partial y}\bigg|_{y=\delta} - \mu_l\frac{\partial U_l}{\partial y}\bigg|_{y=\delta}\right) = 0 \tag{13}$$

There are four simultaneous, first-order, ordinary differential equations. There are four unknown variables: vapor core velocity, $U_\infty(x)$, the liquid-vapor interfacial Velocity, $U_i(x)$, the condensate film thickness, $\delta(x)$, and the vapor boundary layer thickness, $\Delta(x)$, which are functions of an axial distance, $x$, only(see Fig. 1).

### 3.2 Estimation of Initial Values

Estimation of the initial values is possible by postulating that condensation phenomena over a flat plate is similar to the condensation at a horizontal channel entrance.

For the case of $U_\infty$ = constant, the overall energy balance equation,

$$\delta\frac{d}{dx}\int_0^\delta u_l dy = \frac{k(T_S-T_W)}{\rho_l H_{fg}} = \nu_l E \tag{14}$$

and the momentum Eq.(11) of vapor are reduced, respectively, to

$$\delta\frac{d}{dx}(U_i\delta) = \text{constant} \tag{15}$$

$$\int_\delta^{H/2} U_g\frac{\partial U_g}{\partial y}dy + \int_\delta^{H/2} V_g\frac{\partial U_g}{\partial y}dy = \int_\delta^{H/2}\frac{1}{\rho_g}\frac{\partial \tau_g}{\partial y}dy \tag{16}$$

The particular solutions of these simultaneous Eqs.(15, 16) result from the power series expansion method(Fujii et al. 1972, Lucas 1979) in the range of small $z(=x/H)$, that is, the formulas,

$$U_i = A_0 U_\infty + A_1 z + \cdots\cdots \tag{17}$$

$$(\delta/H)^2 = B_0 z + B_1 z^2 + \cdots\cdots \tag{18}$$

can be simplified by choosing first terms on the right side of the Eqs. (17, 18) and neglecting other higher order terms. By substituting the above two equations into those simplified equations, the dimensionless coefficients $A_0$ and $B_0$ are solved and the initial values can be determined ; thus, this larminar film condensation problem. can be solved as an initial value problem(Lee, 1983 b ; Lucas et al., 1979).

## 4. NUMERICAL RESULTS AND DISCUSSION

Solutions are calculated for the saturated water vapor and for the various kinds of subcooling parameters(modified Jacob number, $E=\dfrac{C_{pl}(T_s-T_w)}{H_{fg}\cdot P_r}$) with a variable viscosity of liquid at atmospheric pressure(Figs. 2, 3, 4, 5, 6, 7). For a particular fluid(i.e., when density, viscosity and Prandtl num-

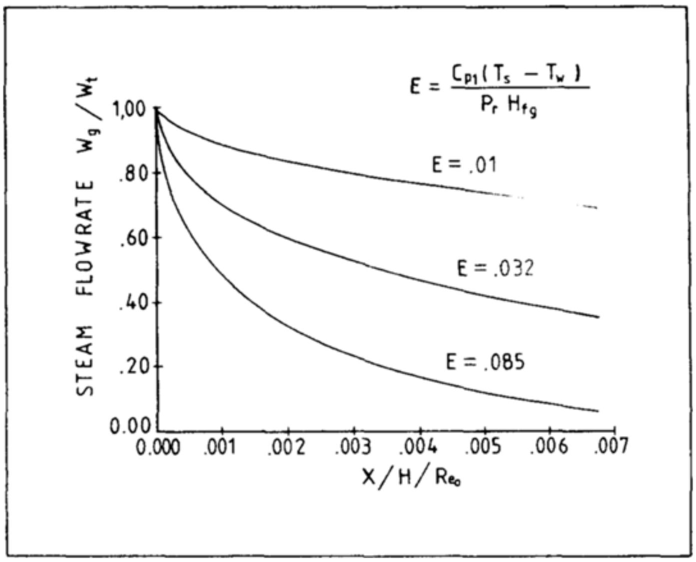


Fig. 2 Steam flowrate along a horizontal channel

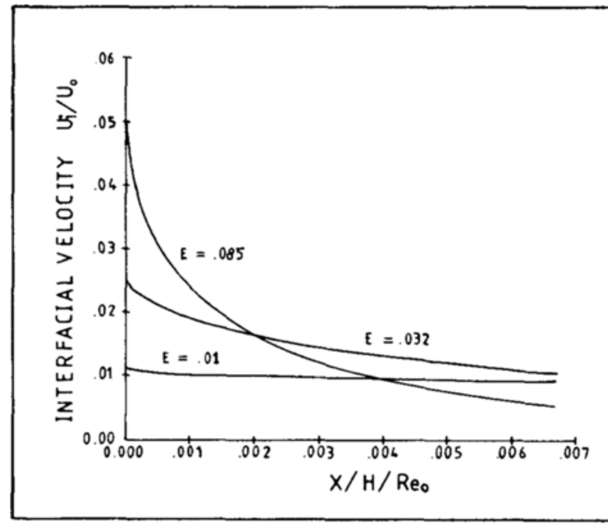


Fig. 4 Interfacial velocity along a horizontal channel

ber of the fluid are fixed), the subcooling parameter, $E$, and nondimensional location of the channel $x/H/Re_o$ are involved as important factors. Nondimensional solutions of the steam flowrate, $W_g$, pressure drop, and interfacial velocity for a horizontal channel($g_x = 0$ case) are shown in Figs. 2, 3 and 4.

When subcooling parameter, $E$, is large, steam flowrate is reduced very fast($E = .085$ on Fig. 2), and fluid of the core region moves slowly(pressure goes up in Fig. 3), and the interfacial velocity at the liquid-vapor boundary is also reduced very rapidly(Fig. 4).

Solutions for the vertical channel are compared with horizontal film condensations in Figs. 5, 6 and 7. The end of the entrance region for the vertical case is shown in these figures. The orientation of the channel has a significant effect on film condensation. Figure 5 shows that steam inside the vertical channel is condensed more rapidly than that in the horizontal case. Figure 7 shows that the interfacial velocity inside the vertical channel becomes faster due to the gravitational effect on liquid and slows down inside the horizontal channel due to the reduction of the core velocity and the interfacial shear stress. The effect of the interfacial velocity is shown in the film thickness: it is thick in the horizontal film and thin in the verticla film(Fig. 6). The local heat transfer coefficient is inversly proportional to the thickness of the condensate film.

Solutions for constant viscosity at a reference temperature are compared with the solutions for variable viscosity of a liquid. Reference film temperatures are selected for three values of $\alpha$, 1/2, 1/3, and 1/4 in the equation, $T_{film} = T_w + \alpha(T_s - T_w)$. Solutions for $\alpha = 1/3$ agrees well with the variable viscosity solution for forced convection laminar film condensation at a channel entrance(Figs. 8, 9 and 10) excepting the interfacial velocity on Fig. 11. When the viscosity of the condensate is constant, that is, when the reference film temperature is selected for the value of $\alpha$ in the above

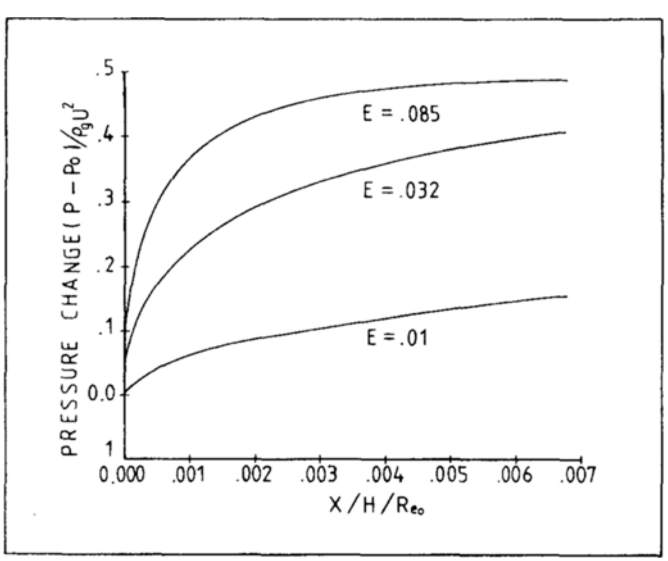


Fig. 3 Pressure chang along a horizontal channel

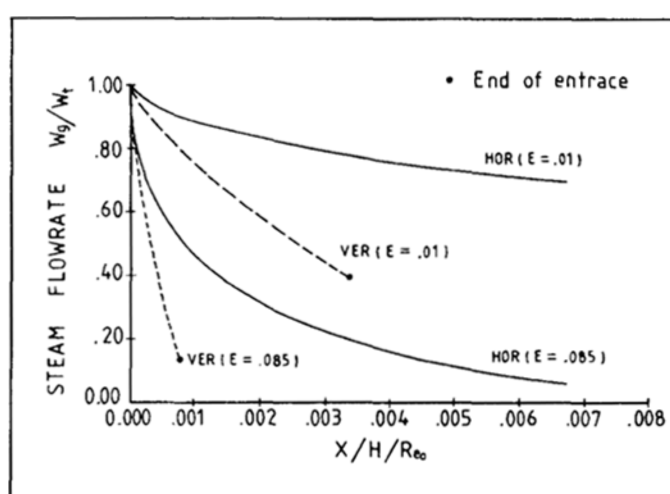


Fig. 5 $W_g/W_t$ comparision of horizontal with vertical film condensation

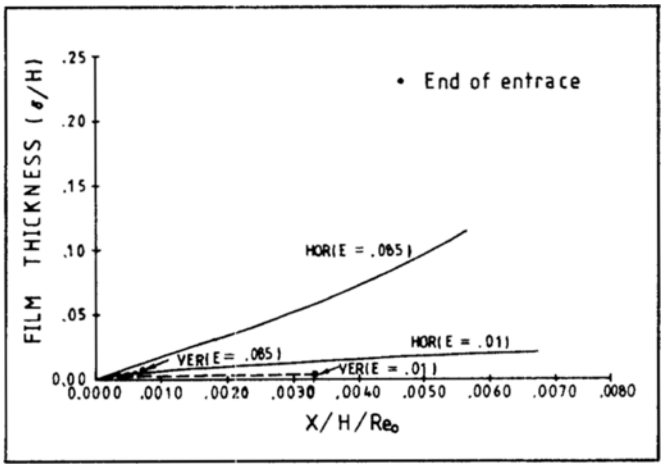


**Fig. 6** $\delta/H$ comparison of horizontal with vertical film condensation

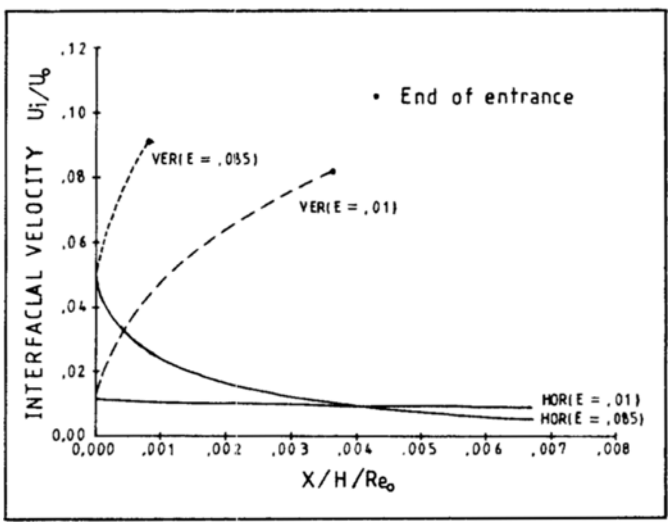


**Fig. 7** $U_i/U_o$ ccomparison of horizontal with vertical film condensation

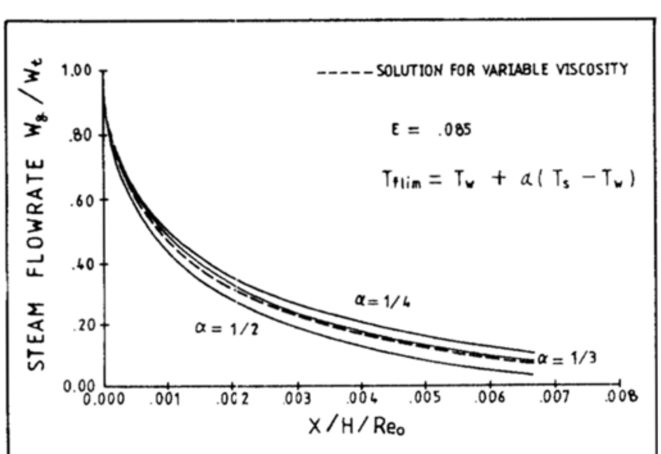


**Fig. 8** Solutions of steam flowrate depending on the reference film temperature

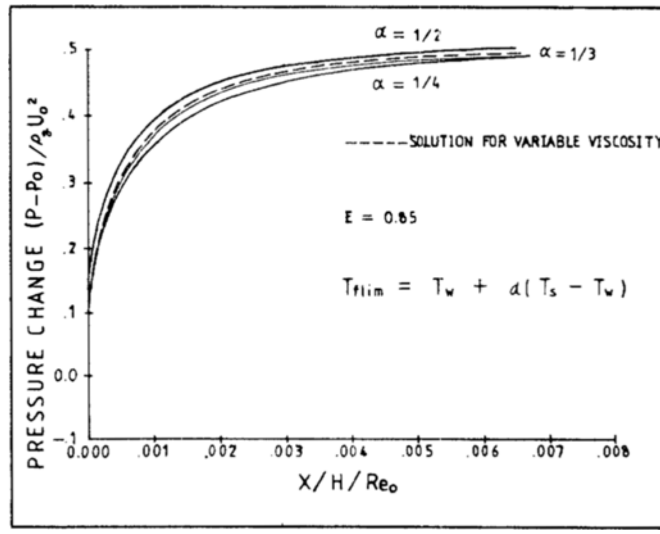


**Fig. 9** Solutions of pressure change depending on the reference film temperature

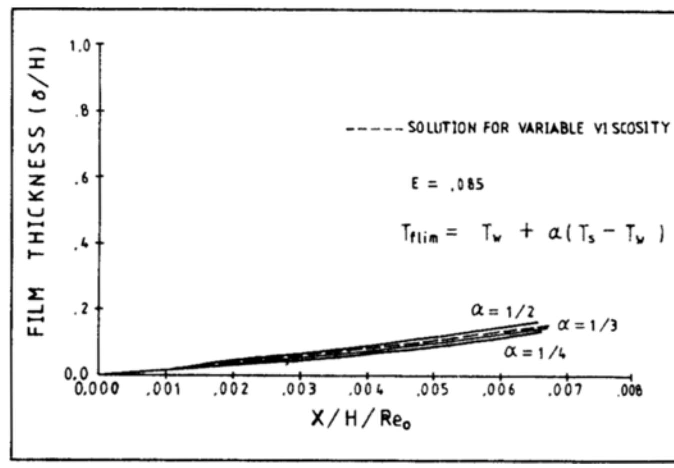


**Fig. 10** Solutions of film thickness depending on the reference film temperature

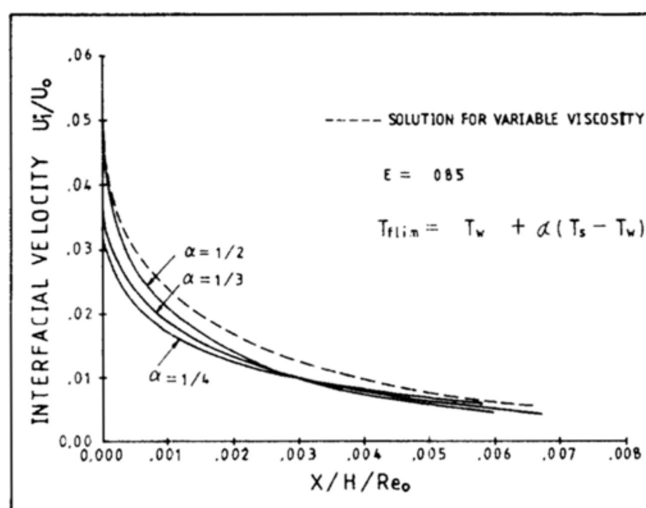


**Fig. 11** Soulutions of interfacial velocity depending on the reference film temperature

**Table 1** Solution of the horizontal laminar film condensation for
$U_o=10m/s$, $T_g=T_s=100℃$, $T_w=20℃$, $H=0.0625$m $\mu_g/\mu_l=0.0445$

| $x$ | $\frac{U_\infty}{U_o}$ | $\frac{U_i}{U_o}$ | $\frac{\delta}{H}$ | $m''$ | $h_x$ | $\tau_i$ | $\frac{\tau_i}{m''(U_\infty-U_i)}$ |
|---|---|---|---|---|---|---|---|
| m | | | | kg/ $m^2\cdot s$ | kw/ $m^2$℃ | Pa | |
| .0001 | 1.000 | .0492 | .00006 | 6.554 | 184.9 | 62.61 | 1.005 |
| .0004 | .994 | .0489 | .00016 | 2.254 | 63.6 | 21.40 | 1.005 |
| .0013 | .986 | .0485 | .00028 | 1.282 | 36.2 | 12.08 | 1.005 |
| .0039 | .974 | .0479 | .00050 | 0.730 | 20.6 | .679 | 1.004 |
| .0114 | .953 | .0469 | .00086 | 0.422 | 11.9 | 3.841 | 1.004 |
| .0295 | .923 | .0454 | .00141 | 0.257 | 7.25 | 2.265 | 1.004 |
| .0919 | .865 | .0425 | .00261 | 0.139 | 3.93 | 1.149 | 1.002 |
| .2180 | .796 | .0390 | .00425 | 0.085 | 2.41 | .647 | 1.001 |
| .7168 | .651 | .0318 | .00887 | 0.041 | 1.15 | .253 | .996 |
| 1.7155 | .501 | .0243 | .01656 | 0.022 | .62 | .103 | .988 |
| 2.9593 | .391 | .0188 | .02614 | 0.014 | .39 | .051 | .979 |
| 5.4530 | .262 | .0123 | .04829 | 0.008 | .21 | .018 | .957 |
| 9.1968 | .161 | .0072 | .09370 | 0.004 | .11 | .005 | .914 |
| 12.509 | .112 | .0048 | .15022 | 0.002 | .07 | .002 | .865 |

equation for $T_{film}$, the velocity profile is a straight line for horizontal case, However, the real velocity distribution is not straight when the viscosity of the liquid is dependent upon the temperature of the liquid as in Eq. (4). From the Fig. 11, the interfacial velocity for the variable viscosity is larger than that for the three solutions at the constant reference temperature.

The differences between the constant property solutions for the vertical film condensation depending on the selection of $\alpha$ for the reference film temperatures are smaller than the horizontal cases. These kinds of differences are also smaller for the small value of $E$ than for the large value.

The interfacial shear stress can be estimated from the numerical solution. Using the momentum integral equation of the vapor phase, the effect of condensation on the interfacial shear stress may be estimated by the following equation:

$$\tau_i=m''(U_\infty-U_i)+C_o \tag{19}$$

$$C_o=U_\infty\int_\delta^{\delta+\Delta}\rho_g U_\infty dy-\frac{d}{dx}\int_\delta^{\delta+\Delta}\rho_g U_g^2 dy-\frac{dp}{dx}\Delta \tag{20}$$

It is found that for the relatively high condensation case the local interfacial shear stress($\tau_i$) is alomost equal to the momentum transferred by the condensing vapor across the liquid-vapor interface which results from the phase change (Table 1). This trend is shown more clearly in the vertical channel (Table 2).

$$C_o\approx 0. \tag{21}$$

**Table 2** Interfacial shear stress on the vertical film condensation for
$U_o=10m/s$, $T_g=T_s=100℃$, $T_w=20℃$, $\mu_g/\mu_l=0.0445$

| $x$ (m) | $\frac{U_\infty}{U_o}$ | $\frac{U_i}{U_o}$ | $m''$ | $\tau_i$ | $h_x$ | $\frac{\tau_w}{\tau_i}$ | $\frac{\tau_i}{m''(U_\infty-U_i)}$ |
|---|---|---|---|---|---|---|---|
| .0001 | 1.0000 | .0518 | 4.903 | 46.68 | 138. | 1.00 | 1.00 |
| .0002 | 9.975 | .0517 | 3.279 | 31.14 | 92.5 | 1.00 | 1.00 |
| .0005 | .9941 | .0516 | 2.253 | 21.32 | 63.5 | 1.00 | 1.00 |
| .0008 | .9903 | .0515 | 1.682 | 15.86 | 47.5 | 1.01 | 1.00 |
| .0013 | .9865 | .0514 | 1.334 | 12.52 | 37.6 | 1.01 | 1.00 |
| .0021 | .9819 | .0513 | 1.066 | 9.959 | 30.1 | 1.02 | 1.00 |
| .0040 | .9732 | .0512 | .778 | 7.197 | 22.0 | 1.04 | 1.00 |
| .0071 | .9625 | .0512 | .585 | 5.352 | 16.5 | 1.07 | 1.00 |
| .0115 | .9508 | .0514 | .465 | 4.191 | 13.1 | 1.12 | 1.00 |
| .171 | .9385 | .0517 | .385 | 3.425 | 10.9 | 1.17 | 1.01 |
| .0295 | .9167 | .0526 | .301 | 2.604 | 8.5 | 1.29 | 1.00 |
| .0545 | .8826 | .0543 | .232 | 1.920 | 6.5 | 1.51 | 1.00 |
| .0919 | .8418 | .0568 | .188 | 1.475 | 5.3 | 1.81 | 1.00 |
| .1418 | .7963 | .0595 | .160 | 1.175 | 4.5 | 2.20 | 1.00 |
| .2181 | .7366 | .0631 | .137 | .920 | 3.9 | 2.79 | 1.00 |
| .4040 | .6145 | .0698 | .111 | .603 | 3.1 | 4.36 | 1.00 |
| .7168 | .4458 | .0781 | .092 | .338 | 2.6 | 8.19 | 1.00 |

## 5. CONCLUSION

The above results demonstrate that the interfacial shear stress($\tau_i$) at the liquid-vapor boundary at a channel entrance is coupled with the mass transfer process and signigicantly affected by phase change and approximately determined by the momentum change rate, $m''(U_\infty-U_i)$, when the condensation parameter, E,i.e., condensation rate, is large. These kinds of trends are identical to those in film condensation in forced convection on a flat plate (Lee et al. 1986), and the findings of this paper are expected to hold for other condensation conditions as well. The effects of the orientation of the channel can be quite considerable. Putting the channel in a vertical position improves the performance of the coindensation in the channel.

For the liquid film, it is generally accepted that Nusselt's assumption may by used for liquid water at the approximate solutions, when the thermophysical properties(viscosity) are evalusted at a suitable reference conition, more specifically, at

$$T_{film}=T_w+\frac{1}{3}(T_s-T_w).$$

## ACKNOWLEDGMENT

This reaearch was partially supported by Korea Science and Engineering Foundation. Much of the Graphical work was done by T. H. Cho.

## 6. REFERENCES

Bromley, L.A., 1952, "Effect of Heat Capacity of Condensate", Ind. and Eng. Chem., Vol. 44, p. 2966.

Cess, R.D., 1960, "Laminar Film Condensation on a Flat Plate in the Absence of a Body Force", Z. Angew. Math. Phys. Vol. 11, pp. 426~433.

Drew, T.B., 1938, Personal Communication guoted by W.H. McAdams.

Fujii, T. and Uehara, H., 1972, "Laminar Filmwise Condensation on a Vertical Surface", Int. J. Heat Mass Transfer, Vol. 15, pp. 217~233.

Kim, H.J. and Bankoff, S.G., 1983, "Local Heat Transfer Coefficients for Condensation in Stratified Countercurrent Steam Water Flows", ASME Journal of Heat Transfer, Vol. 105, pp. 706~712.

Kim, J. H., 1987, M. S. Thesis, Pusan National University.

Koh, J.C.Y., 1962, Int. J.: Heat Mass Transfer, Vol. 5, pp. 941~954.

Lee, S.C. and Bankoff, S. G., 1983, "Stability of Steam -Water Countercurrent Flow in an Inclined Channel: Flooding", ASME Journal of Het Transfer, Voil. 105. pp. 713~718.

Lee, S. H., 1983a, "Effect of Condersation on Interfacial Shear Stress", Ph. D. Dissertation, Northwestern University, Evanston, Il. U.S.A.

Lee, S.H., 1983b, "Laminar Film Condensation at a Horizontal Channel Entrance", KSME Fall Conference Proceeding, pp. 131~136.

Lee, S.H. and Yuen, M.C., 1986, Korea-USA Seminar on Thermal Enginnering and High Technology, pp. 233~239, Edited by J.H. Kim, S.T. Ro. and T.S.Lee, Hemisphere Publishing Corpooration and Springer-Verlag.

Lee, S.H., 1987, "Effects of Condensation on the Interfacial Shear Stress in Laminar Film Condensation of Freon -11 on a Flat Plate", Pusan National University, Research Review, Vol. 33, pp. 25~29.

Lee S.H. and Yuen, M.C., 1987, "Effects of Condensation on the Interfacial Shear Stress in Laminar Film Condensation on a Flat Plate", KSME Journal, Vol. 1. No. 1, pp. 36~29.

Lim, I.S., Tankin, R.S. and Yuen, M.C., 1984, "Condensation Measurement of Horizontal Cocurrent Stema/Water Flow", ASME Journal of Heat Transfer, Vol. 106, pp. 425~432.

Lucas, K., 1979, "Influence of Film Condensation on the Laminar Flow in the Entrance" (In German), Wärme-und Stoffüber-tragung, Vol. 12, pp. 131~136.

Lucas, K. and Moser, B., 1979, "Laminar Film Condensation of Pure Vapours", Int. J. Heat Mass Transfer, Vol. 22, pp. 431~435.

Nusselt, W., 1916, VDI Zeitschrift, 60: 541, 569.

Rohsenow, W.M., 1956, Trans. ASME, 78, 1945-1648.

Schiller, L., 1922, Gorschungarbeiten V.D.I. 248.

**Errata (Vol. 1, No. 1, June 1987)**

| Page | Column | Line | For | Read |
|---|---|---|---|---|
| 37 | right | Eq. (8) | $\tau_i = \dot{m}''(U_\infty - U_i) + U_\infty^2 \frac{d}{dx} \int_\delta^{\delta+\Delta} \rho_g \frac{U_\infty}{U_\infty}\left(1 - \frac{U_g}{U_\infty}\right) dy - \frac{dP_1}{dx}\Delta$ | $\tau_i = \dot{m}''(U_\infty - U_i) + U_\infty \frac{d}{dx} \int_\delta^{\delta+\Delta} \rho_g U_g dy - \frac{d}{dx} \int_\delta^{\delta+\Delta} \rho_g U_g^2 \, dy - \frac{dP}{dx}\Delta$ |