NUMERICAL APPROACH FOR COCURRENT STRATIFIED STEAM WATER FLOW IN A HORIZONTAL CONFIGURATION

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The present study deals with the one-dimensional numerical approach in predicting the local flow properties for cocurrent stratified steam-water flow in a horizontal configuration. The turbulence-centered model, developed principally for gas absorption, has been modified and introduced for the condensation heat transfer coefficient using interfacial parameters, such as turbulent velocity and length scales. The calculated condensation rates, pressures and mean water layer thicknesses are in good agreement with Lim's experimental data obtained from cocurrent stratified steam flow on a fairly thick layer of water. In addition, the approach was applied to the case of relatively thin liquid films, and the results were compared with Linehan's experimental data. The comparison indicates that the one-dimensional numerical approach with the present condensation heat transfer correlation developed from the thick liquid data can be applicable to the prediction of the flow properties for thin liquid films.

Key Words: Turbulence-Centered Model, Turbulent Gas Absorption Model, Transport Process, Single Driving Force, Interfacial Structure

NOMENCLATURE

A : Cross sectional area
B : Channel width
C_p : Specific heat
D_h : Hydraulic diameter
f : Friction factor
g : Acceleration of gravity
H : Channel height
h : Local heat transfer coefficient
i : Enthalpy
i_{fg} : Latent heat of evaporation
k : Thermal conductivity
Nu_t : Turbulent Nusselt number
P : Static pressure
Pr : Prandtl number
q : Heat flux
Re : Reynolds number
Re_t : Turbulent Reynolds number
St : Stanton number
T : Temperature
t : Height of water layer
t^+ : Dimensionless length scale of water layer
u : Streamwise velocity
u^+ : Dimensionless velocity scale, u/u^*
u^* : Shear velocity
u' : Turbulent velocity scale
W : Mass flow rate per unit width
x : Coordinate of flow direction
y : Coordinate normal to flow
y^+ : Dimensionless length scale, u^*y/ν
β : Interfacial velocity factor
μ : Viscosity
ν : Kinematic viscosity
ρ : Density
τ : Shear stress
ϕ : Reynolds momentum flux

SUBSCRIPTS

a : Adiabatic
b : Upper wall surface
c : Condensation
E : Eddy
f : Liquid phase
g : Gas phase
i : Interface
in : Inlet
o : Liquid wall surface
sat : Saturation
t : Turbulent

SUPERSCRIPTS

$-$: Average

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1. INTRODUCTION

Direct contact condensers have widespread industrial applications but usually involve dispersed flows, such as bubbles in cold liquid or drops surrounded by vapor. Stratified, two-phase flow occurs in indirect contact condensers, but the liquid films are relatively thin, and the controlling factor is the temperature difference between the solid wall and the vapor. There are, however, stratified direct contact situations in which the vapor flows either cocurrently or countercurrently to fairly thick layers of cold liquid, which can be of practical importance. For example, in a small break loss of coolant accident in a pressurized water reactor(PWR), subcooled emergency core cooling(ECC) water is injected into the pressure vessel through the hot and/or the cold legs in order to prevent overheating of the reactor core. The subcooled ECC water is then brought into contact in several locations with escaping steam, which is generating within the core. The resulting condensation rate can determine whether or not the reactor core will remain covered with liquid. This is obviously a desirable situation, as illustrated by the Three Mile Island experience. A knowledge of the thermal hydraulic interactions in such situations is important for the quantitative prediction of the system characteristics and then the course of the accident.

A number of specific examples of condensation induced fluid motions and common features of rapid condensation situation are discussed in Block(1980). These examples are qualitatively dealt with situations in which the condensation process influences, and in turn is influenced by fluid motions. Condensation heat transfer in indirect condensers, involving thin liquid film at nearly saturation temperatures, has been studied in a variety of configurations both theoretically and experimentally. The principal resistance to heat transfer in this case was assumed to be in the region of the film in close proximity to the wall, which is viscous and largely unaffected by the presense of the interfacial waves. However, the newer applications of present interest involve the contact of steam with fairly thick layers of cold water, where the heat transport to the solid boundary is either absent or negligible, and then the previously developed models which emphasize the wall viscous layer may not be applicable here since the resistance to heat transfer is related to the overall characteristics of the film. Several experimental studies have been carried out on local transport properties in direct contact stratified two phase flow over the years(Linehan, 1968 ; Lim, 1981 ; Jensen, 1982 ; Kim, 1983 ; Cook, 1981 ; Segev, 1981). As a result, a number of empirical correlations on the condensation heat transfer coefficient have been developed ; part of them will be described in Section 2. In an alternate approach, there have been some theoretical approaches using the transport equations for the turbulent kinetic energy and dissipation rate. Daly and Harlow(1980) employed a one-equation model of turbulence to describe the mass, momentum, and energy transfer in a condensing two phase flow. They obtained results in good agreement with the data of Lee(1979) for horizontal, cocurrent, stratified steam water flow, using an empirical correlation for the mean water layer thickness. In a different stratified cocurrent flow system without condensation, Akai, et al.(1981) employed a two-equation model of turbulence to predict the turbulent transport mechanism for both phases. Their analytical model, when solved numerically, gave good agreement with their own data. The work of Daly and Harlow(1980) seems to be especially attractive, since it deals with stratified condensing flows, which is similar to the present case.

The present study deals with the numerical approach in predicting the local flow properties for the cocurrent, stratified, steam-water flow in a horizontal configuration. Some correlations on condensation heat transfer coefficient are reviewed and each of them is introduced to put the conservation equations into a closed form.

2. ONE-DIMENSIONAL NUMERICAL APPROACH

For the stratified, cocurrent, steam-water flow in a rectangular horizontal channel, it is assumed that the flow is one-dimensional, incompressible, steady, and that the steam is at saturation condition in the entire region. In addition, the heat transfre through the solid wall is assumed to be negligible. The one-dimensional numerical approach is adopted because it provides enough information for the prediction of general system characteristics, even with its inherent disadvantage of losing detailed system characteristics. It is a practical approach and is usually pursued in the simulation of giant systems such as nuclear power plants.

2.1 Conservation Equation

Considering the control volume as shown in Fig. 1, one can obtain the continuity equation,

$$dW_g = -dW_f \tag{1}$$

where
$$W_g = \rho_g (H-t)\,\bar{u}_g \tag{2}$$

$$W_f = \rho_f\, t\, \bar{u}_f \tag{3}$$

The momentum balance for each phase gives, neglecting the terms with the product of differential,

vapor phase :
$$-(H-t)\,dP_g/dx - (\tau_b + \tau_i) = W_g d\bar{u}_g/dx + (\bar{u}_g - u_{g,i})\,dW_g/dx \tag{4}$$

liquid phase :
$$-t dP_f/dx + (P_g - P_f)\,dt/dx + (\tau_i - \tau_o) = W_f d\bar{u}_f/dx + (\bar{u}_f - u_{f,i})\,dW_f/dx \tag{5}$$

The steam pressure is assumed to be constant across steam space due to the lower steam density. Then the pressure at the interface can be expressed as

$$P_f :_{y=t} = P_g \tag{6}$$

Pressure variation across the liquid layer is assumed to be proportional to gravity only, where

$$P_f = P_g + (t-y)\,\rho_f g \tag{7}$$

Differentiating Eq. (7) with respect to x, one obtains

$$\partial P_f/\partial x = dP_g/dx + \rho_f g\, dt/dx \tag{8}$$

With no slip condition at interface, one can obtain :

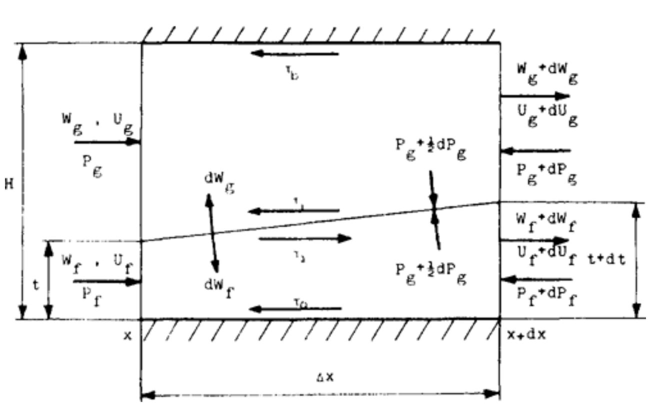

Fig. 1 One-dimensional model for two-phase flow over element of horizontal channel

$$u_{g,i}=u_{f,i}=u_i \tag{9}$$

The interfacial velocity, u_i, may be anywhere between the average vapor and liquid velocity. In the analysis of turbulent film condensation, Carpenter & Colburn(1951) chose the vapor velocity as the interfacial velocity. However, due to the order-of-magnitude difference in the viscosities of the vapor and the liquid, the interfacial velocity is probably closer to the average liquid velocity than to the average vapor velocity. The relationship between u_i and u_f is frequently expressed as;

$$u_i=\beta\bar{u}_f \tag{10}$$

Several investigators have chosen constant, β, in the range of 1 to 2. Soliman, Schuster and Berenson(1968) used $\beta=1.25$, assuming the liquid layer to be similar to the boundary layer of single phase, in the study of condensation in annular flow. Linehan(1968) proposed $\beta=1.14$, assuming the velocity profile in the liquid layer to follow the 1/7 power law. The contribution of the momentum terms containing the interfacial velocity is found to be of minor importance(Rohsenow, Webber & Ling, 1956). This may be attributed to the fact that the steam velocity is much higher than the water velocities. Then, the value of 1.14 is used in this study.

Substituting Eqs. (1), (8), (9), (10) into Eqs. (4), (5), and rearranging the terms, one can easily obtain the one-dimensional momentum equations for each phase:

$$\text{vapor phase}: \frac{W_g^2}{\rho_g(H-t)^2}\frac{dt}{dx}+(H-t)\frac{dP_g}{dx}=\{\frac{2W_g}{\rho_g(H-t)}-\frac{1.14\,W_f}{\rho_f t}\}\frac{dW_f}{dx}-(\tau_b+\tau_i) \tag{11}$$

$$\text{liquid phase}: \{-\frac{W_f^2}{\rho_f t}+\rho_f g t\}\frac{dt}{dx}+t\frac{dP_g}{dx}=-\frac{0.86\,W_f}{\rho_f t}\frac{dW_f}{dx}+(\tau_i-\tau_o) \tag{12}$$

The energy conservation equation due to condensation can be expressed as follows:

$$dq_c=d(W_f i_f)=-d(W_g i_g)=-i_g dW_g \tag{13}$$

Integrating Eq. (13) from the entrance to the distance, x.

$$C_{pt}(W_f T_f-W_{f,in}T_{f,in})=-i_g(W_g-W_{g,in}) \tag{14}$$

Eliminating W_g with Eq. (1) and rearranging the terms, one can obtain

$$T_f=T_{f,in}+\frac{(W_f-W_{f,in})[i_{fg}+C_{pf}(T_g-T_{f,in})]}{C_{pf}W_f} \tag{15}$$

or

$$W_f=W_{f,in}\frac{i_{fg}+C_{pf}(T_g-T_{f,in})}{i_{fg}+C_{pf}(T_g-T_f)} \tag{16}$$

2.2 Constitutive Equations

(1) Wall Shear Stress

The evalution of the wall shear stress in the liquid phase requires an assumption concerning the turbulent nature of the film, particularly in the region near the wall. Carpenter & Colburn(1951) suggested what is termed the "wall-layer model" of film flow, which assumes that a liquid film flowing under the influence of high gas flow acts as the wall layer of a single-phase liquid flow with the same thickness and wall shear stress. The wall-layer model allows the use of a single-phase velocity profile to establish an expression for the wall shear stress. The von Karman universal velocity distribution(1939) is expressed as

$$u^+=y^+ \qquad 0<y^+<5 \tag{17}$$

$$u^+=-3.05+5.0\ln y^+ \qquad 5<y^+<30 \tag{18}$$

$$u^+=5.5+2.5\ln y^+ \qquad 30<y^+<t^+ \tag{19}$$

Integrating the universal velocity distribution from 0 to t^+, one can obtain

$$Re_f=3.0t^++2.5t^+\ln t^+-64 \tag{20}$$

Hershman(1960) obtained the values of t^+ versus Re_f from experiment in which three-dimensional waves are present on the film surface, and found that Eq. (20) fits the experimental data $\pm4\%$ over the range of t^+. The deviation might be within the experimental scatter of the data. Wall shear stress in the liquid phase can be obtained from the definition of t^+,

$$\tau_0=\rho_f(\nu_f t^+/t)^2 \tag{21}$$

The wall shear stress in the vapor phase may be determined from the Blasius equation(Schlicting, 1968) for the friction factor in turbulent flow, which is defined as

$$f_b=0.079Re_{g,Dh}^{-0.25} \tag{22}$$

In applying this equation, derived for a single phase, turbulent flow in a circular pipe, to two-phase flow in a rectangular channel, there arises some question as to the appropriate definition of the wetted perimeter, since it requires a knowledge of the wetted perimeter to evaluate the hydraulic diameter. Gazley(1949) and Russell & Etchells(1969) suggested that for gas liquid systems the interface can be considered to act as a free surface with respect to the liquid phase, and as a stationary surface or solid boundary with respect to the gas:

$$D_{h,g}\overset{\Delta}{=}\frac{4A_g}{P_G}=\frac{4B(H-\delta)}{2\{B+(H-\delta)\}} \tag{23}$$

$$D_{h,f}\overset{\Delta}{=}\frac{4A_f}{P_L}=\frac{4B\delta}{B+2\delta} \tag{24}$$

where P_f and P_g are the perimeter of the portion of the solid wall in contact with the liquid and gas phase, respectively. For a rectangular channel with a fairly large aspect ratio, where $B\gg(H-\delta)$ and δ, this gives

$$D_{h,g}\approx2(H-\delta) \quad\text{and}\quad D_{h,f}\approx4\delta$$

Theofanous, et al.(1976) estimated the smooth wall (upper wall) and interfacial friction factors for cocurrent stratified air-water flow in a rectangular channel by measuring the pressure drop, the position of the maximum in the mean velocity profiles, and the mean liquid film thickness. Upon comparison of the Blasius Eq. (22) with the wall shear stress data, using the same definition for the hydraulic diameter of

Eq. (23), good agreement was found at low liquid and gas Reynolds numbers. However, at high liquid and gas Reynolds numbers, Eq. (22) underestimates the friction coefficient. This is to be expected, since the interface becomes characterized by more irregular waves as the gas and liquid Reynolds numbers increase. One can note that at zero water flow the data agree well with the Blasius equation; this implies that the definition for the hydraulic diameter, Eq. (23), is strictly correct only if the interfacial shear stress is equivalent to the wall shear stress.

Cohen(1964) measured the wall shear stress for the flow of air in a rectangular channel of aspect ratio 12 : 1. He found that the Blasius friction factor increases by about 9%. Thus, replacing the coefficient in Eq. (22) by Cohen's coefficient and rearranging, it gives

$$\tau_b = 0.0865 Re_{g,D_h}^{0.25} W_g / \{2\rho_g (H-t)^2\} \tag{25}$$

(2) Interfacial Shear Stress

The effect of condensation on interfacial shear stress may be visualized by analogy with the familiar problem of suction for the gas phase, in the view of thinning the momentum boundary layer, which results in increasing the shear stress.

Silver & Wallis(1965), took into account the effect of condensation on the interfacial shear stress by means of the Reynolds flux concept, arriving at;

$$f_i / f_{i,a} = exp(\phi / 2f_{i,a}) - \phi / f_{i,a} \tag{26}$$

where f_i and $f_{i,a}$ are the interfacial friction factors with and without condensation, respectively.

An alternative approach, developed by Mickley, et al. (1954), employs a stagnant film theory to relate the friction factor with suction to the friction factor without suction for the same free-stream flow conditions, resulting in

$$\frac{f_i}{f_{i,a}} = \frac{\phi}{f_{i,a}} \frac{1}{exp(\phi / f_{i,a}) - 1} \tag{27}$$

Based on these ideas, Linehan(1968) proposed a simple approximate equation

$$f_i / f_{i,a} = 1 - \phi / f_{i,a} \tag{28}$$

which is equivalent to

$$\tau_i = 1/2 f_{i,a} \rho_g \bar{u}_g^2 - \bar{u}_g dW_g / dx \tag{29}$$

This relation indicates that the interfacial shear stress in the presence of condensation is simply augmented by an amount equal to the condensation rate times the average vapor velocity, where the interfacial velocity has been neglected in comparison to the vapor velocity. A more exact expression would thus be

$$\tau_i = 1/2 f_{i,a} \rho_g (\bar{u}_g - u_i)^2 - (\bar{u}_g - u_i) dW_g / dx \tag{30}$$

Linehan found, using the air-water data of Cohen(1964), that the interfacial friction factor without mass transfer could be correlated with the liquid Reynolds number only:

$$f_{i,a} = 1.875 \times 10^{-5} Re_f + 0.0068 \qquad Re_f < 340 \tag{31}$$

$$f_{i,a} = 0.23 \times 10^{-5} Re_f + 0.0131 \qquad Re_f > 340 \tag{32}$$

(3) Heat Transfer Coefficient

Assuming the interface to be always at T_{sat}, the heat transferred to the liquid is related to the temperature gradient at the interface times an effective turbulent thermal conductivity, where

$$q_c = -k_E|_t \cdot \partial T_f / \partial y|_t = i_{fg} dW_f / dx \tag{33}$$

The condensation heat transfer coefficient is defined as

$$q_c = h(T_g - T_f) = i_{fg} dW_f / dx; \tag{34}$$

hence

$$h = i_{fg} / (T_g - T_f) dW_f / dx \tag{35}$$

This equation relates the local condensation heat transfer coefficient to the local condensation rate. In order to identify the condensation rate, which is needed to put the conservation equation into a closed form, the heat transfer coefficient must be correlated with the flow parameters and/or easily accessible parameters.

Linehan(1968) proposed a simple constant Stanton number relationship based upon the hypothesis of constant eddy diffusivity. Based upon his data for cocurrent stratified steam-water flow, he found

$$St = h / (\rho C_p u_f) = 0.0073 \tag{36}$$

Lim, et al.(1981) measured condensation rates for cocurrent, stratified, steam-water flow in a horizontal channel and found the Stanton number is a function of the steam and water Reynolds number and the liquid Prandtl number, which contrasts with the constant Stanton number result of Linehan;

$$\text{rough surface : } \overline{St}(x) = 0.0344(\overline{Re}_g(x))^{0.58} (\overline{Re}_f(x))^{-0.58} (Pr_f)^{-0.7} \tag{37}$$

$$\text{smooth surface: } \overline{St}(x) = 0.63(\overline{Re}_g(x))^{0.58} (\overline{Re}_f(x))^{-0.91} (Pr_f)^{-0.7} \tag{38}$$

Jensen(1982) suggested that the appropriate constant grouping for the condensation heat transfer coefficient is $st_t Pr^a$, based upon an examination of some gas absorption data together with his own horizontal cocurrent steam water data. The correlations, based on the critical Reynolds number and the interface condition, are

rough surface : $a = 1/2$
smooth surface: $= 2/3$

and the constants are

$C = 0.14 \qquad u_f x / \nu > 1.8 \times 10^5$
$C = 0.1 \qquad u_f x / \nu < 1.8 \times 10^5$

The principal drawback of the Lim and the Jensen approaches is that they require detailed information about the interfacial wave structure, since the correlations are classified by the interface condition. For practical applications, it is necessary to express the heat transfer coefficient in terms of easily accessible flow properties rather than estimates of the interfacial structure. This point has been emphasized in the present author's consideration on the heat transfer coefficient(Kim, et al.,1986). The present authors correlated the heat transfer coefficient in condensing cocurrent two phase flow by means of a power-law relationship involving the turbulent liquid Reynolds number and the liquid Prandtl number. This draws upon the analogy between heat and mass transfer at the interface, starting with the trubulent gas absorption models that relate the mass transfer coefficient to the local turbulent properties. The analogy is reasonable because a single driving force in the liquid, due either to a concentration gradient or to a temperature difference, is dominant in the transport process. Based upon the Lim, et al. experimental data for cocurrent, stratified, steam-water flows, it is found that

$$Nu_t = 0.417 Re_t^{0.863} Pr^{0.194} \tag{39}$$

3. EXPERIMENTAL DATA

There are only two experiments of cocurrent, stratified, steam-water flow known to the authors. The data, obtained from the experiments, are used in comparisons with the results of the numerical approach mentioned above.

Lim, et al.(1981) made measurements of local condensation rates, mean water layer thicknesses and pressure gradients for the cocurrent, stratified, steam-water flow in a horizontal channel. The test section is a rectangular horizontal channel with an aspect ratio of 4.8; a length of 1.6m, a width of 0.3048m and a height of 0.0635m, as shown in Fig. 2. Steam and water enter uniformly through the inlet plenum and exit through the outlet plenum. At the entrance, the mass flowrates of steam and water, water thicknesses and temperatures were measured. The steam velocity distributions, steam pressures, water thicknesses and temperatures were measured at 5 stations along the channel centerline; the distances of each station from entrance are 0.157m, 0.306m, 0.586m, 0.868m, 1.233m. The local mass flowrates of steam are obtained by integrating the measured steam velocity distributions through pitot probes. The present analysis starts with the conditions of the first measurement location since the characteristic of the flow very near the channel inlet is, as of yet, not well understood. The range of experimental conditions is given in Table 1.

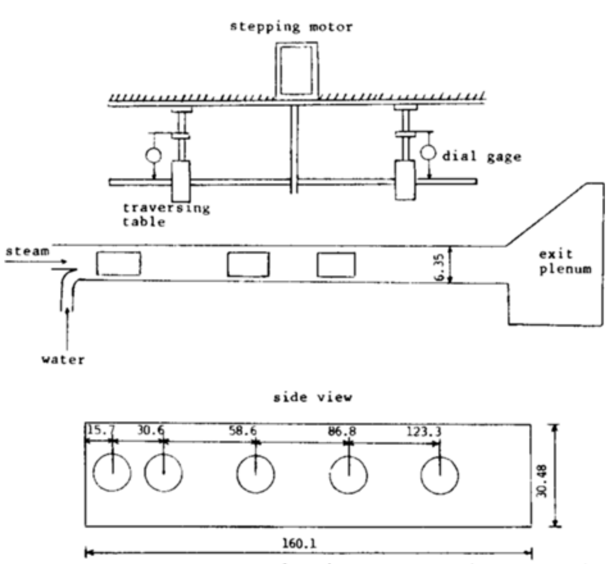

Fig. 2 Channel view of lim's experimental facility(Lim, 1981)

Table 1 Lim's experimental range

Re_f	800 ~12700
Re_g	2100~18600
Pr_f	2 ~ 5
h(kW/m²s)	2 ~ 21

Linehan(1968) made measurements of local water temperatures at the wall and mean film thicknesses for a cocurrent, stratified flow of saturated steam over a subcooled water film. The test section is a rectangular horizontal channel with an aspect ratio of 9.1; a height of 0.0167m, a length of 0.4572m and a width of 0.1524m. The water layer thicknesses were measured at 8 stations of 0.0508m intervals from the entrance, and the wall surface temperatures were measured at 13 stations of 0.0254m intervals. Liquid bulk temperatures were obtained from the measured wall surface temperatures based on the assumption of a uniform temperature profile across the liquid film. Linehan's experiment for a thin liquid film of about 1mm thickness is in contrast to the Lim, et al. experiment for the relatively thick liquid layer of about 10mm thickness.

4. RESULTS AND DISCUSSION

The local flow properties of interest, such as flow rate, pressure and liquid layer thickness, etc., can be calculated from the conservation equations described in Sec. 2. The constitutive equations for the shear stress and the condensation heat transfer coefficient have been introduced to put the conservation equations into a closed form; with respect to the shear stress, Eq. (21) for the liquid-phase, Eq. (25) for the gas-phase and Eq. (30) for the interface. With respect to the condensation heat transfer correlation, each of the correlations of Linehan, Lim, et al., Jensen and the present authors is introduced to calculate the local condensation rates. The calculated local condensation rates, pressure and liquid layer thickness are compared with the experimental data of Lim, et al.(1981) for the cocurrent, stratified flow of saturated steam

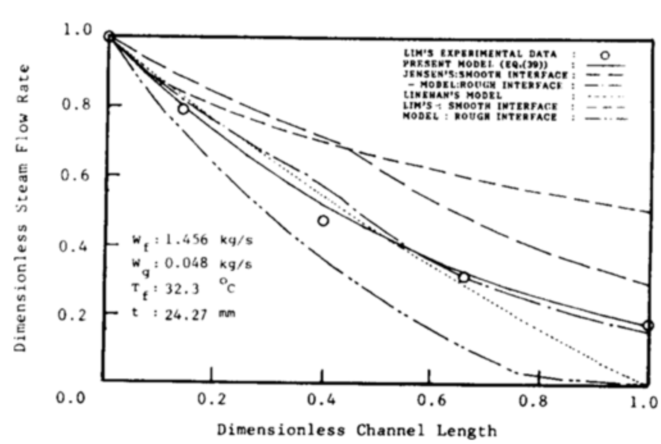

Fig. 3 Comparison of models with Lim's data for steam flow rate

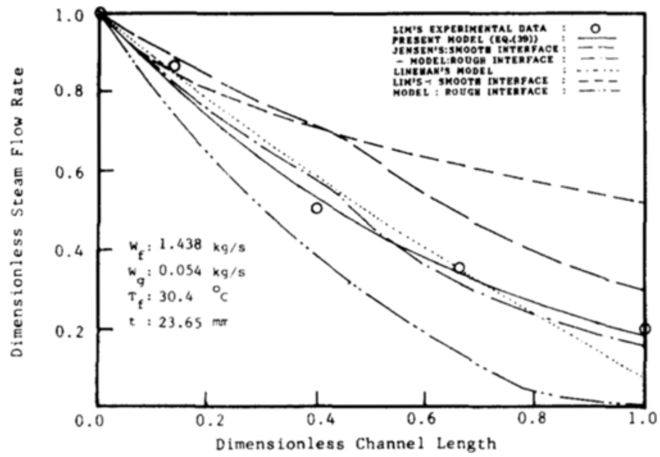

Fig. 4 Comparison of models with Lim's data for steam flow rate

over a fairly thick layer of cold liquid in a horizontal channel. In addition, the numerical approach is further extended to compare the result with the data of Linehan(1968) for the thin liquid film case.

The numerical method uses the marching procedure where the results of one location give information for the next location. By substituting the constitutive equations for shear stress into the momentum conservation equations, two equations with respect to 3 unknown variables, i.e. dt/dx, dP_g/dx, and dW_f/dx, are obtained. The condensation rate is obtained from the energy conservation equation using the correlations on the condensation heat transfer coeffieient. The condensation rate is then used to obtain the rate of change of liquid thickness, dt/dx, and pressure gradient, dP_g/dx. Through the step calculations of this procedure, the localized information through the entire length of the channel can be obtained with the given inlet conditions.

4.1 Application to Thick Liquid Layer

The local condensation rates, obtained from the several heat transfer coefficient models, are compared with the experimental data of Lim, et al., as shown in Fig. 3, 4, 5, 6. As expected above from Lim's and Jensen's models, each gives

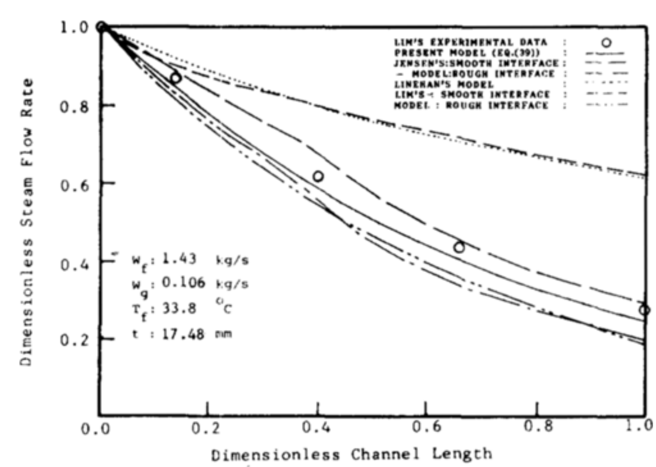

Fig. 5 Comparison of models with Lim's data for steam flow rate

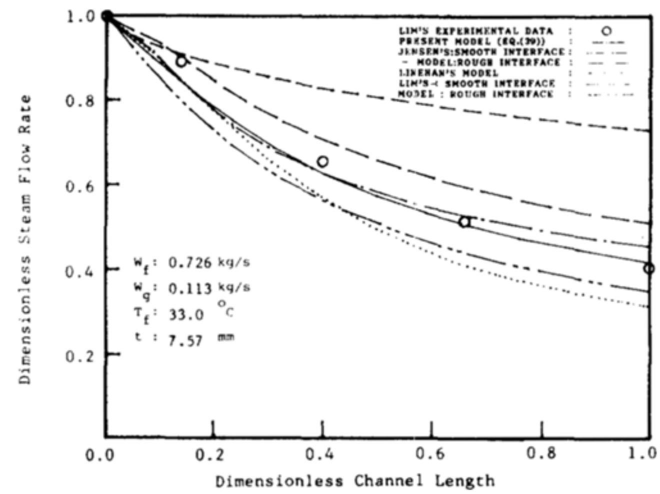

Fig. 6 Comparison of models with Lim's data for steam flow rate

different results for rough and smooth interfaces, and Linehan's model, which suggests constant Stanton number, results in considerable discrepancy with Lim's experimental data. In general, Lim's correlation for the rough or the smooth interface either overestimates or underestimates in comparison with the experimental data, whereas Jensen's correlation for the rough or the smooth interface may be applicable individually according to the entry conditions. However, there are difficulties in practical applications, since the boundary criteria on the interfacial structure are not suggested. For a constant water flowrate, as the steam flowrate increases, the interface becomes rougher, bringing about the increase of the interfacial shear stress, and consequently, an increase in the heat transfer coefficient. As shown in Fig. 4 and Fig. 5, it may be expected that the interface in Fig. 5 is rougher than that in Fig. 4, since the initial steam flowrate in Fig. 4 is smaller than that in Fig. 5. But one can observe that Jensen's correlation for rough interface in Fig. 4 and that for smooth interface in Fig. 5 fits the experimental data well. Therefore, the application of this correlation for a single interface may cause large deviation, since the interface roughness varies frequently and is affected strongly by the localized flow conditions. As

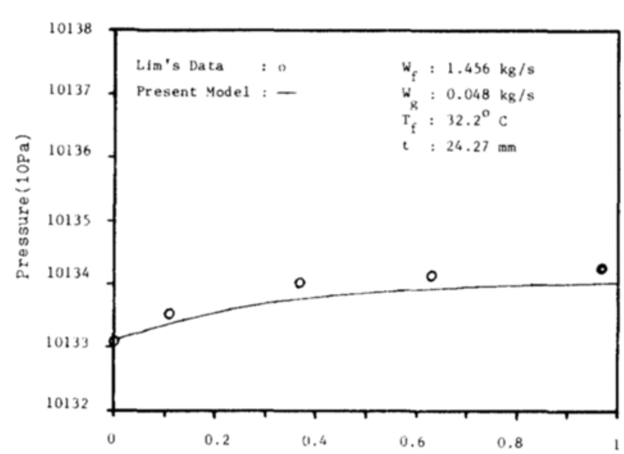

Fig. 7 Comparison of the calculated pressure with Lim's data

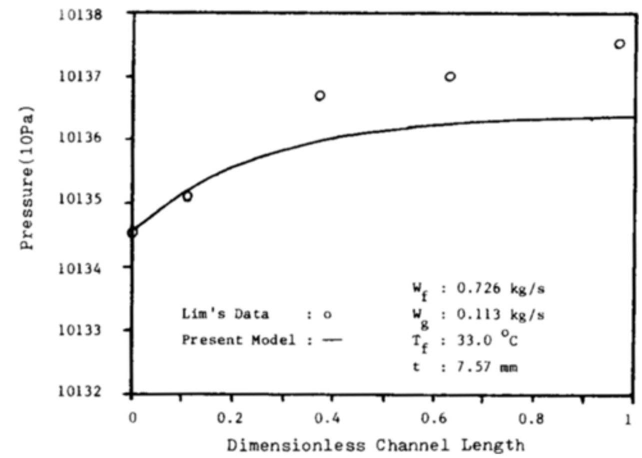

Fig. 8 Comparison of the calculated pressure with Lim's data

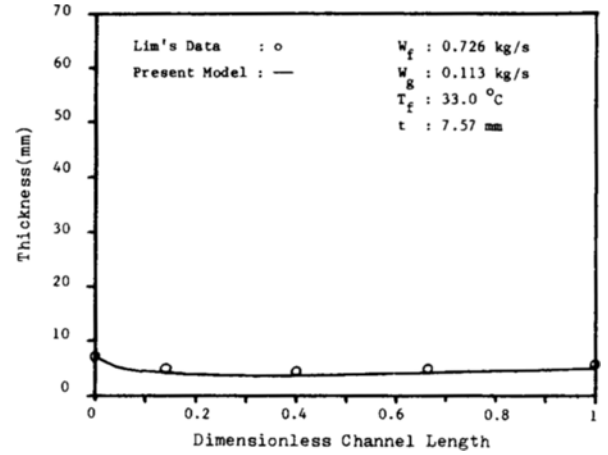

Fig. 10 Comparison of the calculated water layer thickness with Lim's data

shown in Fig. 6, Jensen's correlation for the smooth interface agrees with the experimental data near the inlet whereas the correlation for the rough interface agrees with the experimental data near the outlet. However, the correlation suggested by the present authors agrees well with the experimental data regardless of various entry conditions. This agreement may be ascribed to the fact that the present correlation is composed of local variables without taking into account the entry condition.

Pressure, calculated from one-dimensional numerical approach using the condensation rate obtained from the present correlation, is compared with Lim's experimental data and is in good agreement as illustrated in Figs. 7, 8. The pressure gradient with the flow direction in the horizontal channel is composed of the friction term and the momentum change term due to condensation. The pressure is reduced by the friction, but increased by the reduction of momentum. Since the effect on pressure due to steam momentum change is much greater than that due to friction in the case of large condensation, the numerical result using the present correlation, which predicts the condensation rate relatively well, describes the momentum change appropriately and also serves as a good predictor of the pressure gradient.

In Figs. 9, 10, the numerical results using the present

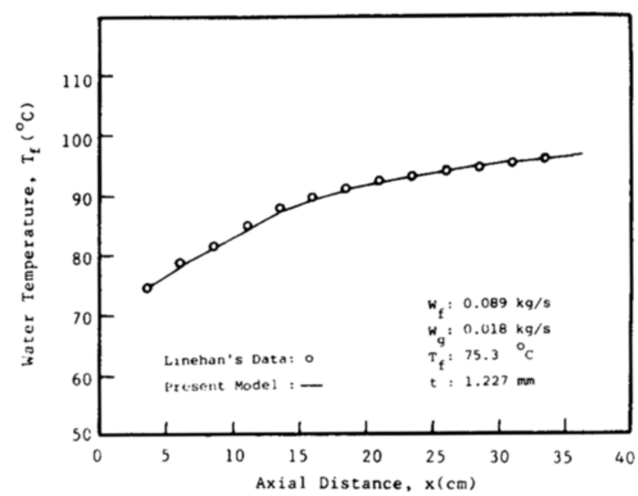

Fig. 11 Comparison of the calculated temperature with Linehan's data

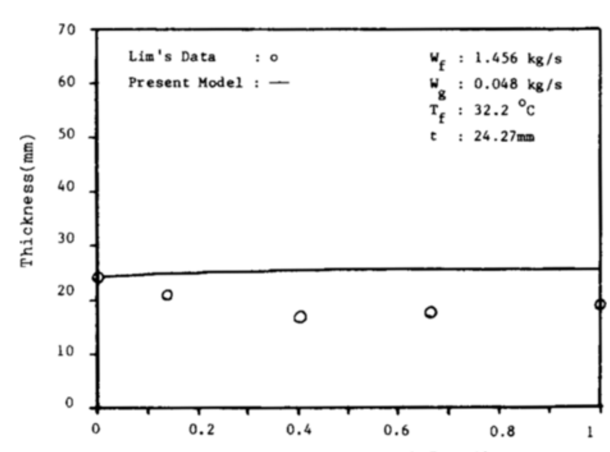

Fig. 9 Comparison of the calculated water layer thickness with Lim's data

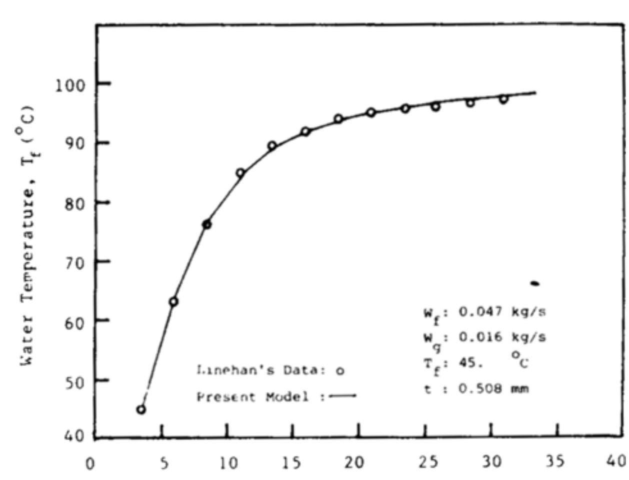

Fig. 12 Comparison of the calculated temperature with Linehan's data

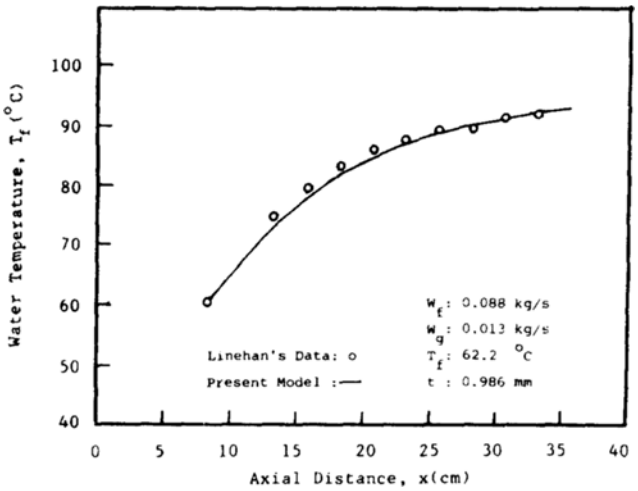

Fig. 13 Comparison of the calculated temperature with Linehan's data

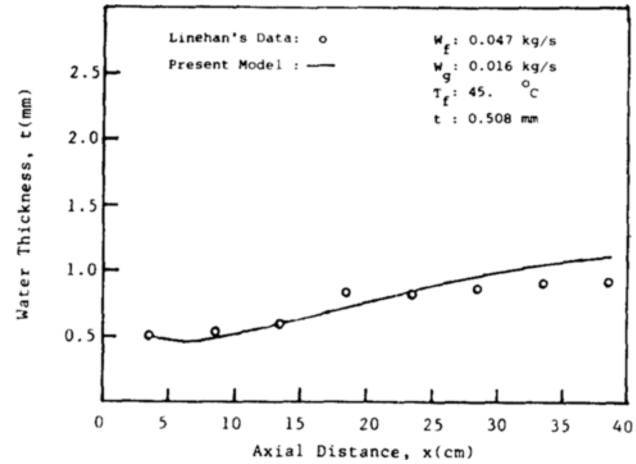

Fig. 15 Comparison of the calculated water layer thickness with Linehan's data

correlation are compared with experimental data with respect to the liquid layer thickness. When the liquid flowrate is constant for cocurrent, stratified flow, as steam velocity increases, the liquid layer thickness decreases because of accelerating liquid surface due to increasing interfacial drag force. The drag force changes the interfacial wave structure, thereby changing the horizontal and vertical velocity distributions near the interface. These changes in wave structure and velocity distributions may be important factors in determining liquid layer thickness since they strongly govern the overall velocity distributions of the liquid layer and the vertical distribution of turbulence intensity. The wave structure and the velocity distribution can not be described with one-dimensional approach assumed here. Especially for a thick liquid layer, a fairly accurate prediction of the liquid layer thickness may be difficult without enough consideration of these effects. It points out that the boundary layer type approximation based on the assumption of a thin boundary layer may not be applicable to a fairly thick layer of liquid. This is well evidenced in Daly & Harlow's numerical analysis with complementary correlation of liquid layer thickness taken directly from Lim's experimental data for cocurrent, stratified flow. This discrepancy is also noted in the present study, as shown in Fig. 9. Since low liquid flowrate results in low condensation rate and thin liquid layer thickness, one-dimensional numerical approach similar to the boundary layer type approximation may be applicable to thin liquid layer thinkness, as shown in Fig. 10.

4.2 Application to Thin Liquid Layer

It may be interesting to apply the numerical approach with the present heat transfer correlation obtained from the thick liquid layer data to the thin liquid layer of Linehan's experiment. Linehan replaced the liquid bulk temperature with the temperature measured at the wall by assuming that the liquid temperature is nearly constant throughout the cross section as the liquid temperature gradient throughout the liquid layer is minimized with high turbulence intensity due to interfacial wave structure of the trubulent flow of the thin liquid layer. Though this assumption may be reasonable in the case of the existence of the higher turbulence intensity, the liquid temperature gradient through the liquid layer exists or, at least, the liquid bulk temperature is higher than the wall temperature. Therefore, it is reasonable that the liquid temperature

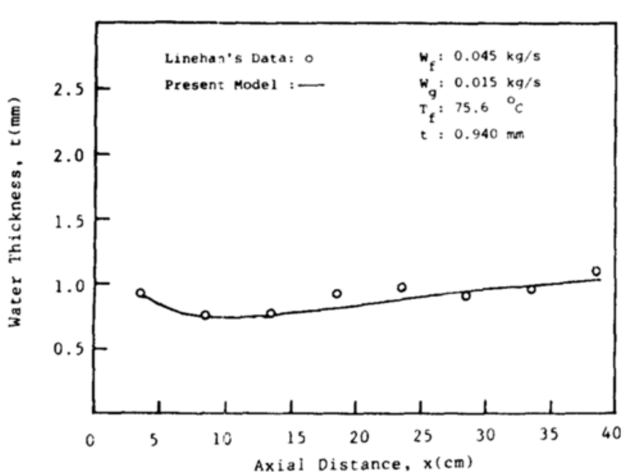

Fig. 14 Comparison of the calculated water layer thickness with Linehan's data

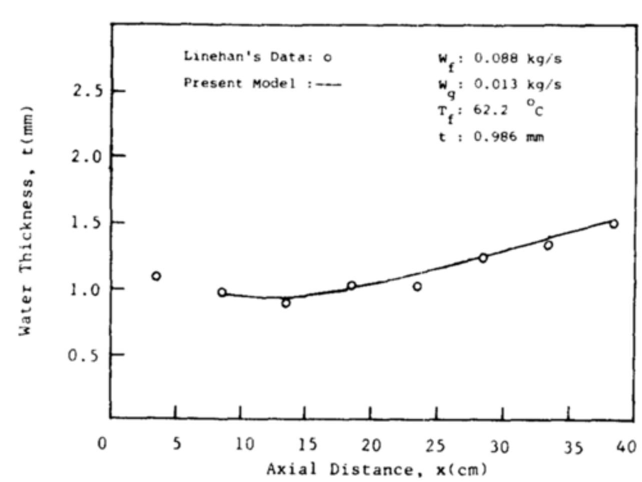

Fig. 16 Comparison of the calculated water layer thickness with Linehan's data

calculated from the present method is somewhat higher than Linehan's data, as shown in Figs. 11, 12 & 13. Figs. 14, 15 & 16 show the comparison of the calculated liquid layer thickness with the measured data. The present numerical approach agrees well with the experimental data for the thin liquid layer. It may be because the present one-dimensional approach is similar to the boundary layer type approximation, and the approximation seems to be especially applicable to the thin liquid layer. The comparisons of the calculated liquid temperatures and liquid layer thicknesses with the measured data for the thin liquid layer indicate that the one-dimensional numerical approach with the present correlation developed from the thick liquid tata is applicable to the prediction of the variables for thin liquid layer flow.

5. CONCLUSION

The one-dimensional numerical approach with the present heat transfer correlation, Eq. (39), results in a good agreement with the experimental data for the condensation rate, pressure and mean water layer thickness for the cocurrent, stratified steam flow on a fairly thick liquid layer in a horizontal configuration. It indicates that the present heat transfer correlation based on the gas absorption model, which relates the mass transfer due to concentration gradient, may be applicable to the mass transfer due to temperature gradient through the consideration of appropriate turbulent properties.

In addition, the comparison of the calculated liquid temperatures and liquid layer thicknesses with the measured data for the thin liquid layer indicates that the one-dimensional numerical approach with the present heat transfer correlation developed from the thick liquid data is applicable to the prediction of the flow properties for the thin liquid layer flow.

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