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Analytical Modeling of Convective Condensation in Smooth Vertical Tubes

A. Nouri-Borujerdi[1]

In this paper, an analytical model is presented for forced convective film condensation inside smooth vertical tubes, in which Prandtl mixing length theory, Van Driest hypothesis and Reynolds analogy are used. Comparison of this model with the available experimental data and three other correlations reported in the literature demonstrates that the results are in satisfactory and close agreement.

INTRODUCTION

When a tube wall is cooled below the saturation temperature of a vapor flowing through the tube, condensation occurs. If the vapor velocity inside the tube is sufficiently high, the condensate flows as an annular film on the tube wall. Interfacial shear, due to vapor flow, tends to accelerate the liquid flow and makes the film thin. Consequently, the condensation heat transfer increases. Typically, much of the condensation processes occur in the annular flow regime, even for qualities as low as 25 percent [1-3].

Forced convective condensation inside tubes is widely adopted in process industry, e.g., air-conditioning and refrigeration condensers as well as condensers in Rankin power cycles. Although convective condensation is also contrived to occur sometimes in co-current vertical downward flows, horizontal flow is often preferred.

A variety of approaches for predicting heat transfer during annular flow condensation has been developed. These approaches can be divided into three categories: 1) Shear-based; 2) Boundary-layer and 3) Two-phase multiplier. Carpenter and Colburn [4] pioneered the development of shear-based correlations. They studied the influence of vapor velocity on film condensation in vertical tubes for co-current downward annular flow. On the basis of their experimental data, they reasoned that the transition from laminar to turbulent flow in the film occurs at much lower Reynolds numbers than in the absence of vapor shear. They also argued that the resistance to heat transfer in the turbulent liquid flow is entirely related to the laminar sublayer and that the wall shear stress is composed of additive components due to friction, acceleration and gravity. Altman et al. [5] also used the same method as Carpenter and Colburn [4] to correlate local heat transfer data obtained during the condensation of Freon-22 inside a horizontal tube. The shear stress at the wall was only based on the frictional pressure drop; i.e., the momentum and gravity contributions were neglected. They obtained the same expression as Carpenter and Colburn, with a higher value for the numerical coefficient.

Soliman et al. [6] accepted the basic validity of the form presented by Carpenter and Colburn; however, refined it in several ways. First, they corrected the equation for predicting the wall shear stress due to phase change, in which it was illustrated that this component is normally negligible in comparison to the frictional component except at low qualities. Secondly, they implemented an improved correlation for the frictional pressure drop. Finally, they used data from other researchers to determine new values of the constants.

Boundary layer approaches are similar to shear-based approaches, except that thermal resistance is considered throughout the entire liquid film thickness, not just in the laminar sublayer.

Azer et al. [7] studied this type of approach for annular flow condensation inside horizontal tubes theoretically. They obtained the temperature drop across the liquid film by integrating the energy equation and then utilized it in the definition of local heat transfer coefficient. Another example for boundary layer approaches is that of Traviss et al. [8], in which an analytical prediction of the heat transfer coefficient is provided under rather stringent assumptions. A simple force balance indicates the proportionality between the wall shear and the pressure drop, establishing the

1. *Department of Mechanical Engineering, Sharif University of Technology, Tehran, I.R. Iran.*

fact that the annular flow heat transfer coefficient is proportional to the square root of the pressure drop per unit length.

Two-phase multiplier approaches assume that the heat transfer process in annular two-phase flow is similar to that in single phase flow of the liquid and thus, their ratio may be characterized by a two-phase multiplier.

Akers et al. [9] utilized a two-phase multiplier approach and developed an in-tube condensation model, which defines all-liquid flow rate providing the same heat transfer coefficient as an annular condensing flow. This liquid flow rate is expressed by an equivalent Reynolds number and used in a single-phase turbulent flow equation to predict the condensation coefficient. Boyko and Kruzhilin [10] also evaluated this type of approach for condensation of steam inside tubes on the basis of the analogy between hydraulic resistance and heat transfer in accordance with Reynolds theory. They indicated that theoretical and experimental results demonstrate satisfactory agreement. The correlation also describes experimental results regarding the conditions of heat transfer during condensation flow of steam inside tube bundle. The other cited correlation of the two-phase multiplier type is that developed by Shah [11], based on his observation that the mechanisms of condensation and evaporation were very similar in the absence of nucleate boiling. Considering this idea, he modified the convective component of his flow boiling correlation for use during condensation.

The objective of the present study is to suggest an alternative calculation procedure for determining the local heat transfer coefficient for annular flow condensation inside tubes.

The present model differs from others such as Azer et al. [7], Traviss et al. [8], Chitti [12] and Carey [13] in evaluating the eddy diffusivity and the velocity distribution of the liquid film, for which the momentum equation in combination with the Prandtl mixing length is integrated; whereas, in other models the eddy viscosity is evaluated based on Von Karman universal velocity profile.

MATHEMATICAL FORMULATION

The system illustrated in Figure 1 is now considered. Co-current downward annular film condensation flows in a tube at moderate vapor flow rates ($G \geq 200$ kg/m^2.s). At this mass flux, the tube diameter and fluid properties influence the range of quality over which annular flow regime occurs [7]. For the purposes of this analysis, it is assumed that entrainment of liquid in the vapor core is negligible and the liquid flows in a thin film ($\delta << R$) in a smooth-walled tube. Convection terms in the energy and momentum equations are often neglected because the transport

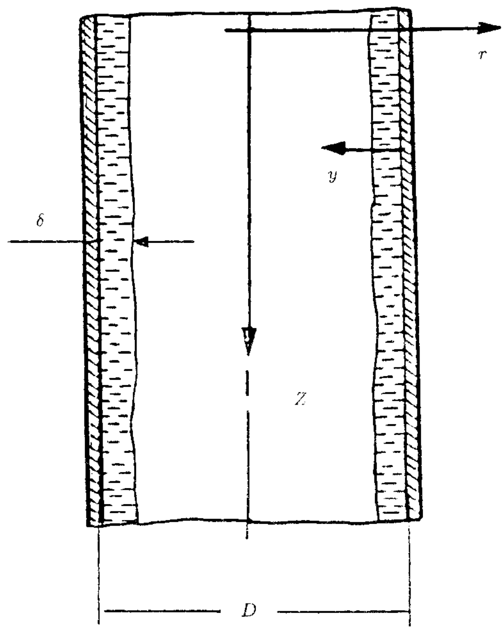

Figure 1. Physical situation for the flow condensation in a vertical tube.

rate across the film is much greater than downstream convection. Under these conditions, the momentum equation across the liquid film during turbulent flow can be expressed as follows:

$$\frac{d}{dy}[\rho_l(\nu_l + \epsilon_M)\frac{du}{dy}] + \rho_l g - \frac{dp}{dz} = 0, \tag{1}$$

where the static pressure gradient for the two-phase flow may be expressed as the sum of the pressure gradient due to friction, gravity and momentum changes. $y = \mathbf{R} - \mathbf{r}$ is the distance measured from the tube wall toward the center-line. ϵ_M is the eddy diffusivity for turbulent momentum in the liquid film. Integration of Equation 1 under the condition $\tau(y=0) = \tau_w$ yields the following distribution for shear stress:

$$\rho_l(\nu_l + \epsilon_M)\frac{du}{dy} = \tau_w - (\rho_l g - \frac{dp}{dz})y. \tag{2}$$

In treatment of internal vapor-liquid flow discussed here, the flow is considered to be steady and one-dimensional in the sense that the pressure term is idealized as being constant over any cross section of the tube, varying only in the axial direction.

Neglecting the pressure gradient due to the momentum change, the static pressure gradient along the tube is:

$$\frac{dp}{dz} = [\alpha\rho_v + (1-\alpha)\rho_l]\, g - \frac{4\tau_w}{D}. \tag{3}$$

α is area (or volume) average gas fraction and for a liquid film with thin thickness, δ, its value is approximated as:

$$\alpha = 1 - \frac{\delta}{D}. \tag{4}$$

From Equations 2 and 3, the velocity gradient can be obtained in the following dimensional form:

$$\frac{du^+}{dy^+} = \frac{1 - (4 + \frac{\alpha g D \Delta\rho}{\tau_w})\frac{y^+}{D^+}}{1 - \frac{\epsilon_M}{\nu_l}}, \tag{5}$$

where:

$$y^+ = \frac{yu^*}{\nu_l}, \quad D^+ = \frac{Du^*}{\nu_l}, \quad u^+ = \frac{u}{u^*},$$

$$\Delta\rho = \rho_l - \rho_v \quad \text{and} \quad u^* = \left(\frac{\tau_w}{p_l}\right)^{0.5}.$$

Using Van-Driest hypothesis [14] regarding the Prandtl mixing length, the turbulent eddy diffusivity for momentum can be expressed as:

$$\frac{\epsilon_M}{\nu_l} = (1^+)^2 |\frac{du^+}{dy^+}|, \tag{6}$$

and:

$$l^+ = Ky^+[1 - \exp(-\frac{y^+}{A^+})], \tag{7}$$

where K is empirical Von Karman constant. Fitting the law of the wall to experimental measurement, it is found that the constant is approximately $K = 0.41$. A^+ is also an empirically determined effective sublayer thickness. It is observed that both pressure gradient and transpiration have a pronounced effect upon A^+. A favorable pressure gradient induces a thicker sublayer and in the absence of the transpiration, Van Driest [14] proposed $A^+ = 26$.

Substituting Equation 5 into Equation 6 and solving for eddy diffusivity, results in:

$$\frac{\epsilon_M}{\nu_l} = -0.5 + 0.5[1 + 4(l^+)^2(1 - C\frac{y^+}{D^+})^{0.5}, \tag{8}$$

where,

$$C = 4 + \alpha g D \Delta\rho / \tau_w.$$

In a similar manner to the momentum equation, the energy equation across the liquid film during turbulent flow is:

$$\frac{d}{dy}\left[(K_l + \rho_l C_{pl}\epsilon_H)\frac{dT}{dy}\right] = 0, \tag{9}$$

where ϵ_H is the diffusivity for turbulent heat transfer in the liquid film. Using $T(y = 0) = T_w$, the integration of Equation 9 yields the temperature difference across the liquid film in dimensionless form as:

$$T_\delta^+ = \int_o^{\delta+} \frac{q''/q_w}{\frac{1}{Pr_l} + \frac{1}{Pr_t}\frac{\epsilon_M}{\nu_l}} dy^+, \tag{10}$$

where $T_\delta^+ = (T_\delta - T_w)\rho_l C_{pl} u^* / q''_w$ and T_δ is the liquid temperature at the outer edge of the film. Mass conservation of the liquid film requires that:

$$Re_l = 4\int_o^{\delta+} u^+(1 - 2\frac{y^+}{D^+})dy^+, \tag{11}$$

where $Re_l = G(1 - x)D/\mu_l$ and G is the total mass flux rate through the tube. Jischa and Rieke [15] have presented a theoretical relationship for the turbulent Prandtl number based on model transport equation for the turbulent kinetic energy. The final result is:

$$Pr_t = C + \frac{B}{Pr_l}. \tag{12}$$

In general, the values of B and C depend on the Reynolds and Prandtl numbers as well as the distance from the wall. For gas and liquid, the influence of Reynolds and Prandtl numbers seems to be weak. The following values have been suggested in [15] for the experimental constants, $C = 0.85$ and B as a function of the Reynolds number.

$$B = 0.012 - 0.05, \quad \text{for} \quad Re = 2 \times 10^4,$$

$$B = 0.005 - 0.015, \quad \text{for} \quad Re = 10^5.$$

Now, Equation 10 can be integrated, provided that the relation for $\dot{q}''/\dot{q}''_w$ is being known in terms of y^+. Assuming the analogy hypothesis between heat and momentum transfer across the liquid film, Equation 5 in the absence of gravity becomes:

$$\frac{\dot{q}''}{\dot{q}''_w} = \frac{\tau}{\tau_w} = 1 - 4\frac{y^+}{D^+}. \tag{13}$$

For an annular flow, the local heat transfer coefficient under the wall heat flux is:

$$h = \frac{\dot{q}''_w}{T_\delta - T_w}. \tag{14a}$$

Introducing $T_\delta^+ = (T_\delta - T_w)\rho_l C_{pl} u^* / \dot{q}''_w$ into Equation 14a, results in:

$$h = \frac{\rho_l C_{pl} u^*}{T_\delta^+}. \tag{14b}$$

The above equation in the form of Nusselt number is defined as:

$$Nu = \frac{hD}{k_l} = \frac{D^+ Pr_l}{T_\delta^+}. \tag{15}$$

To calculate D^+ in the above equation, u^* needs to be evaluated. From definition of the frictional pressure gradient, the wall shear stress caused by the two-phase flow is given by:

$$\tau_w = \frac{D}{4}(-\frac{dp}{dz})_f. \tag{16}$$

The two-phase frictional pressure gradient can be predicted using a variety of methods. The most popular method is to multiply the single-phase frictional pressure drop by a two-phase multiplier, ϕ_{lo}, as:

$$\left(-\frac{dp}{dz_f}\right) = \Phi_{lo}^2\left(-\frac{dp}{dz}\right)_{lo} = \Phi_{lo}^2\frac{2f_{lo}G^2}{D\rho_l}. \tag{17}$$

Introduction Equation 17 into Equation 16 yields:

$$D^+ = \Phi_{lo}Re_{lo}(\frac{f_{lo}}{2})^{0.5}. \tag{18}$$

Substituting Equation 18 into Equation 15 and using Blasius friction factor, the following is obtained:

$$Nu = 0.199\frac{\Phi_{lo}Pr_l Re_{lo}^{0.875}}{T_\delta^+}, \tag{19}$$

where f_{lo} and Re_{lo} are, respectively, friction factor and the Reynolds number that would result if the total mass flowed through the tube as liquid.

Several attempts have been made to develop improved correlations for the two-phase multiplier. Friedel [16] used a data base of 25000 points to develop the following correlation for predicting the two-phase multiplier for vertical upward and horizontal flow in round tubes when $\mu_1/\mu_v < 1000$:

$$\Phi_{lo}^2 = C_l + \frac{3.24C_2}{Fr^{0.045}We^{0.035}}, \tag{20a}$$

$$C_l = (1-x)^2 + x^2\frac{\rho_l}{\rho_v}\frac{f_{vo}}{f_{lo}}, \tag{20b}$$

$$C_2 = x^{0.78}(1-x)^{0.24}(\frac{\rho_l}{\rho_v})^{0.91}(\frac{\mu_v}{\mu_l})^{0.19}(1-\frac{\mu_v}{\mu_l})^{0.7}, \tag{20c}$$

$$Fr = \frac{G^2}{gD\rho_{TP}^2}, \tag{20d}$$

$$We = \frac{DG^2}{\sigma\rho_{TP}}, \tag{20e}$$

$$\rho_{TP} = \left(\frac{x}{\rho_v} + \frac{1-x}{\rho_l}\right)^{-1}. \tag{20f}$$

f_{vo} is the friction factor for the total mass flowing as vapor.

SOLUTION METHOD

As can be seen from the above equations, once the quality x is specified at a given location z, along with the other parameters such as the total mass flux rate, G, and saturation properties at the condensing temperature, the friction velocity, u^*, can be calculated. Once the friction velocity is calculated, then the velocity profile, film thickness, drop temperature across the film, T_δ^+, and the local heat transfer coefficient, h, are calculated by a FORTRAN computer program as follows.

The velocity profile, u^+, is determined by integration of Equation 5 using Simpson's rule. For particular cases, a step size is chosen for the integral prior to the evaluation. The step size later reduced until the percentage error in the integral is less than 0.1%. Then, a value is assumed for δ^+ and Equation 11 is numerically integrated across the liquid film. If this condition is not satisfied, a new value of δ^+ must be chosen and the procedure is repeated until Equation 11 is satisfied. In the next step, Equation 10 is numerically integrated to compute the liquid temperature at the outer edge of the condensate film. Substituting the value of T_δ^+ and the two-phase multiplier from Equation 20a into Equation 19, the value of the local Nusselt number for a given vapor quality and total mass flux rate is obtained.

RESULTS AND DISCUSSION

The results of the predictions of the present model for the local Nusselt number in comparison with the experimental data of Dobson and Chato [17] for R-134a are depicted in Figure 2. This figure shows the local Nusselt number at the mass flux of 650 kg/s.m^2 and saturation temperature of 35°C in a 7.04 mm inside diameter tube. In the figure, the predictions of the present model have been calculated using Equation 19 for different values of vapor quality and a smooth curve is drawn through the points. As can be seen from the figure, the values predicted by this model are in good agreement with the experimental results except for the low vapor quality. This discrepancy is attributed to the flow regime of the condensation film that may not be an annular flow at low qualities. Since the flow regimes clearly affect the heat transfer processes, it is necessary to evaluate the flow regime for condensation conditions on the Mandhane map [18]. The condensation path for mass flux at 650 kg/s.m^2 of R-134a on the Mandhane map (Figure 3) shows that the data are all predominantly in the annular or annular-mist flow regime. Moreover, the figure illustrates that the flow regime is a slug flow for all qualities below 25 percent. The points on the map cover a range of quality from 5 percent to 95 percent in 5 percent increments. Decreasing the

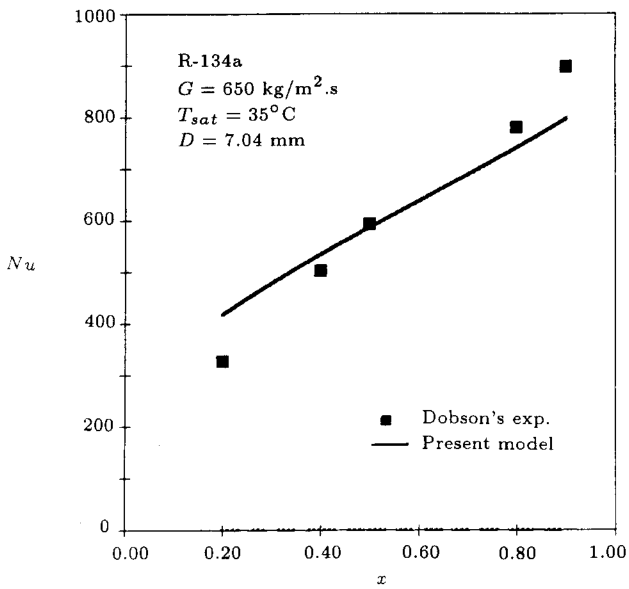

Figure 2. Comparison of the local Nusselt number measured by Dobson [17] with the predictions of the present model.

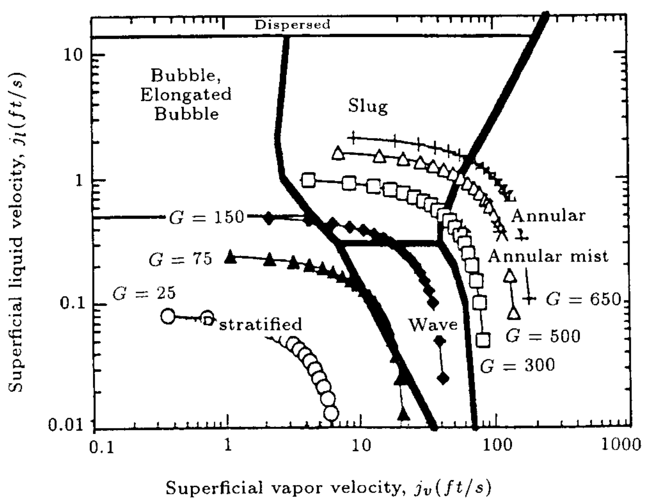

Figure 3. Predicted flow patterns for R-134a on the Mandhane et al. [18] map using corrected superficial velocities.

quality corresponds to moving from right to left on the map.

Figure 4 compares experimental and predicted Nusselt numbers versus quality for R-22 in the 7.04 mm inside diameter tube at the mass flux of 225 $Kg/s.m^2$ and saturation temperature of 35°C.

Several annular flow correlations have been selected for comparison with the current model. According to the Mandhane map, this mass flux should be annular at all qualities above 12 percent. The figure shows that the predictions of the current model agrees very well with the experimental data. Furthermore, the current model is in close agreement with Traviss et al.

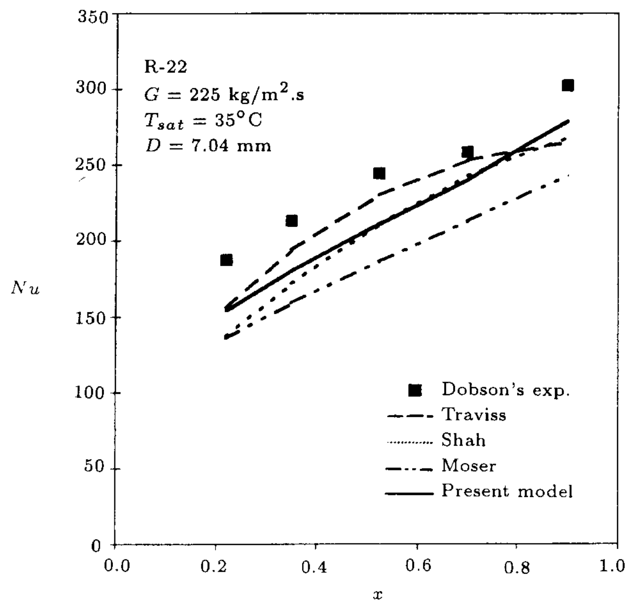

Figure 4. Comparison of the local Nusselt numbers measured by Dobson [17] with the predictions of different correlations and the present model.

[8] and shah [11] correlations in comparison with those of Moser et al. [19].

Table 1 summarizes the parametric range at which each data point was predicted using the local Nusselt number expression defined in Equation 19. In addition, each point was also predicted with shah [11], Traviss et al. [8] and Moser et al. [19] correlations. The overall ability of each correlation to predict the experimental data was determined by the mean absolute deviation, defined as:

$$DE_{ave} = \frac{1}{N} \sum_{j=1}^{N} \frac{|Nu - Nu_{expj}|}{Nu_{expj}}, \tag{21}$$

where N is the total number of data points. The mean absolute deviation gives the average difference between predicted and experiment values for each data set, without considering whether the difference was positive or negative. Figure 5 compares the Nusselt numbers calculated from the present predictions, Equation 19, with the experimental data of Chitti and Anand [12], Dobson and Chato [17] and Goodykoontz [20,21]. From this figure, it can be observed that the experimental data points scatter around the present predictions with a maximum discrepancy of 17 percent, i.e., it indicates that most of the experimental values are within a range of ±17 percent of the predicted values with a mean deviation of 11 percent. This is larger than the mean deviation values of Traviss et al. [8] correlation (10.6%), although it is smaller than mean deviation values of Shah [11] correlation (16.6%) and significantly less than those of Moser et al. [19] correlation (20.3%). This indicates that the present

Table 1. Local heat transfer data analyzed for the verification of the present model.

Source	Fluid	Tube I.D (mm)	T_{sat} (C)	Mass Flux ($kg/m^2.s$)	Quality
[12]	R-22	8	36-46	204.8-439	0.23-0.76
[17]	R-134a	7.04	35	650	0.29 – 0.91
	R-22	7.04	35	225 – 650	0.2 – 0.9
	R-32/R-125	3.14	35	800	0.18 – 0.87
[20,21]	Steam	15.88	129	30 – 130	0.16 – 0.93
		7.44	109		

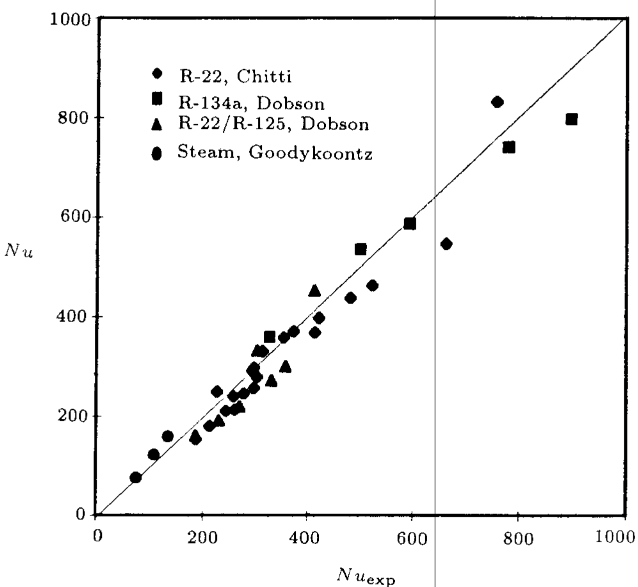

Figure 5. Comparison of the present model results with the local Nusselt numbers measured by Chitti [12], Dobson [17] and Goodykoontz [20,21].

model predicts the local heat transfer data as well as Shah [11] and Traviss et al. [8] and better than Moser et al. [19].

CONCLUSIONS

In this paper, a theoretical study of local heat transfer coefficients during condensation of a saturated vapor in turbulent annular forced flow through a vertical tube is proposed. The present work, although involves an iteration procedure to calculate the film thickness, eliminates the calculations of void fraction, pressure drop and shear stress distribution.

For the local heat transfer data, the present model predicted the data as accurately as the Traviss et al. [8] correlation and more accurately than the Shah [11] and Moser et al. [19] correlations. The mean deviation value is 11 percent for the present model, 10.6 percent for the Traviss et al. [8] correlation, 16.5 percent for the Shah [11] correlation and 20.3 percent for the Moser et al. [19] correlation.

NOMENCLATURE

B, C	constant
C_p	constant pressure specific heat
D	tube inner diameter
f	fanning friction factor
Fr	Frude number, $G^2/Dg\rho^2_{Tp}$
g	gravity acceleration
G	mass flux, ρu
h	heat transfer coefficient
k	thermal conductivity, Van Karman const.
l	Prandtl mixing length
P	pressure
Pr	Prandtl number, ν/α
q	heat flux
Nu	local Nusselt number
Re	Reynolds number
T	temperature
u	velocity
we	Weber number, $DG^2/\sigma\rho_{TP}$
x	vapor quality
y	transversal coordinate
z	longitudinal coordinate

Greek Letters

α	void fraction
δ	film thickness
Δ	difference
ϵ	eddy diffusivity
μ	dynamic viscosity
ν	Kinetic viscosity
ρ	density
σ	surface tension
τ	shear stress
ϕ^2	two-phase multiplier

Subscripts

f	frictional

H	thermal
l	liquid
lo	entire flow as a liquid
M	momentum
t	turbulent
TP	two-phase
v	vapor
vo	entire flow as a vapor
w	wall

Superscript

$()+$	$\frac{()u^*}{\nu}$

REFERENCES

1. Soliman, M. and Azer, N.Z. "Flow patterns during condensation inside horizontal tubes", *ASHRAE Trans.*, **77**(1), pp 210-224 (1971).
2. Stoecker, W.F. and DeGrush, D. "Measurements of heat transfer coefficients of nonazeotropic refrigerant mixtures condensing inside horizontal tubes", Oak Ridge National Laboratory/Sub/ 81-7762/6 and 01 (1987).
3. Taitel, Y. and Dukler, A.E. "A model for predicting flow regime transitions in horizontal and near horizontal gas liquid flow", *AIChE J.*, **22**, pp 47-55 (1976).
4. Carpenter, E.F. and Colburn, A.P. "The effect of vapor velocity on condensation inside tubes", *Proceedings of General Discussion on Heat Transfer*, published by the Institute of Mechanical Engineers and ASME, pp 20-26 (1951).
5. Altman, M., Staub, F.W. and Norris, R.H. "Local heat transfer and pressure drop for refrigerant 22 condensing in horizontal tubes", *Chemical Engineering Progress Symposium Series*, **56**(30), pp 151-159 (1960).
6. Soliman, H.M., Schuster, J.R. and Berenson, P.J. "A general heat transfer correlation for annular flow condensation", *ASME J. of Heat Transfer*, **90**, pp 267-276 (1968).
7. Azer, N.Z., Abis, L.V. and Soliman, H.M. "Local heat transfer coefficients during annular flow condensation", *ASHRAE Trans.*, **78**(1), pp 135-143 (1972).
8. Traviss, D.P., Rohsenow, W.M. and Baron, A.B. "Forced convection condensation inside tubes: A heat transfer equation for condenser design", *ASHRAE Trans*, **79**(2), pp 31-39 (1973).
9. Akers, W.W., Dean, H.A. and Crosser, O.K. "Condensing heat transfer within horizontal tubes", *Chemical Engineering Progress Symposium Series*, **55**(29), pp 171-176 (1959).
10. Boyko, L.D. and Kruzhilin, G.N. "Heat transfer and hydraulic resistance during condensation of steam in a horizontal tube and in a bundle of tubes", *Int. J. of Heat and Mass Transfer*, **10**, pp 361-373 (1967).
11. Shah, M.M. "A general correlation for heat transfer during film condensation inside pipes", *Int. J. of Heat and Mass Transfer*, **22**, pp 547-556 (1979).
12. Chitti, M.S. and Anand, N.K. "An analytical model for local heat transfer coefficients for forced convective condensation inside smooth horizontal tubes", *Int. J. of Heat and Mass Transfer*, **38**(4), pp 615-627 (1995).
13. Carey, V.P., *Liquid-Vapor Phase-Change Phenomena*, Hemisphere Publishing Corporation, New York (1992).
14. Van Driest, E.R. "On turbulent flow near a wall", *J. of Aerospace Sci.*, **23**, pp 1007-1011 (1956).
15. Jischa, M. and Rieke, H.B. "About the prediction of turbulent Prandtl and Schmidt number from modeled transport equations", *Int. J. of Heat and Mass Transfer*, **22**, pp 1547-1555 (1979).
16. Friedel, L. "Improved friction pressure drop correlation for horizontal and vertical two phase pipe flow", Paper E_2, European Two-Phase Flow Group Meeting, Ispra, Italy (1979).
17. Dobson, M.K and Chato, J.C. "Condensation in smooth horizontal tubes", *J. of Heat Transfer, Trans. of the ASME*, **120**, pp 193-213 (1998).
18. Mandhane, J.M., Gregory, G.A. and Aziz, K. "A flow pattern map for gas-liquid flow in horizontal pipe", *Int. J. of Multiphase Flow*, **1**, pp 537-553 (1974).
19. Moser, K.W., Webb, R.L. and Na, B. "A New equivalent Reynolds number model for condensation in smooth tubes", *J. of Heat Transfer, Trans. of the ASME*, **120**, pp 410-417 (1998).
20. Goodykootz, J.H. and Dorsch, R.G. "Local heat transfer coefficients for condensation of steam in vertical down flow with 5/8 in. diameter tube", Report NASA TN D3326 (1966).
21. Goodykoontz, J.H. and Dorsch, R.G. "Local heat transfer coefficients and static pressures for condensation of high velocity steam within a tube", Report NASA TN D3953 (1967).