PERGAMON

International Journal of Heat and Mass Transfer 45 (2002) 845–854

International Journal of HEAT and MASS TRANSFER

www.elsevier.com/locate/ijhmt

Non-iterative condensation modeling for steam condensation with non-condensable gas in a vertical tube

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Received 5 November 1998; received in revised form 11 May 2001

Abstract

Based on a heat and mass transfer analogy, an iterative condensation model for steam condensation in the presence of a non-condensable gas in a vertical tube is proposed including the high mass transfer effect, entrance effect, and interfacial waviness effect on condensation. A non-iterative condensation model is proposed for easy engineering application using the iterative condensation model and the assumption of the same profile of the steam mass fraction as that of the gas temperature in the gas film boundary layer. It turns out that the Nusselt number for condensation heat transfer is expressed in terms of air mass fraction, Jakob number, Stanton number for mass transfer, gas mixture Reynolds number, gas Prandtl number and condensate film Nusselt number. The comparison shows that the non-iterative condensation model reasonably well predicts the experimental data of Park, Siddique, and Kuhn.

1. Introduction

Steam condensation in the presence of non-condensable gas in vertical tubes is an important thermal-hydraulic phenomenon which occurs in the isolation condenser of passive reactors, such as SBWR and CP-1300 [1]. Several experiments have been performed on the condensation of steam in the presence of non-condensable gas in a vertical tube and several empirical correlations and mechanistic models for a condensate layer and a gas mixture layer have been developed based on experimental data.

Several methods have been developed to calculate the film thickness, and the film side heat transfer coefficient was also calculated for both the laminar and turbulent condition. Kim [2] used a reliable model for the condensate film thickness and the condensate film side heat transfer coefficients for a flat plate. The default model of RELAP5/MOD3.2 [3] uses the Nusselt correlation [4] to calculate the heat transfer coefficient in laminar conditions and the Shah correlation [5] in turbulent conditions. The maximum of predictions from the laminar and turbulent correlations is used to calculate the condensate film heat transfer coefficient. Kuhn [6] calculates the condensate film thickness by solving the momentum balance equation considering the thinning effect of the interfacial shear and the condensate film heat transfer coefficient is calculated using Blangetti's film model [7]. Munoz-Cobo [8] developed a model to predict the accurate film thickness and the local condensation heat transfer coefficient inside a vertical tube and an approximate method to calculate the condensate film thickness was also developed, which does not need any iteration to solve the transcendental equation for the film thickness. After evaluating the condensate film thickness, it is shown that the film thickness calculated with tube geometry is slightly thinner than that calculated with plate geometry, which is negligible with this 2 in. tube and that the calculated condensate film thickness very much changes with consideration of the shear of the mixture flow. The film side heat transfer

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PII: S0017-9310(01)00176-4

Nomenclature

b_h	blowing parameter
B	mass transfer driving force
c_f	friction factor
C_p	specific heat (J/kg K)
D	diameter (m)
g	mass transfer conductance
G	mass flux (kg/m^2 s)
h	heat transfer coefficient (J/kg K)
i	enthalpy (J/kg)
Ja	Jakob number
k	thermal conductivity (W/m K)
m''_v	mass transfer rate
M	molecular weight
N_A	a defined non-dimensional parameter
Nu	Nusselt number
P	pressure (Pa)
P_A	a defined non-dimensional parameter
Pr	Prandtl number
q''	heat flux (W/m^2)
Q	volumetric flow rate (m^3/s)
Re	Reynolds number
Sc	Schmidt number
Sh	Sherwood number
St	Stanton number
T	temperature (°C)
u	velocity (m/s)
v	specific volume (m^3/kg)
W	mass fraction
	mass flow rate (kg/s)
x	local axial distance (m)
X	molar fraction

Greek symbols

β	McAdams modifier
δ	film thickness (m)
ϵ_s	sand roughness (m)
ρ	density (kg/m^3)
μ	viscosity (kg/m s)

Subscripts

0	without suction
∞	free stream
AB	mass transfer
b	bulk
cd	condensation
cv	convection
e	entry
f	liquid
fg	evaporation
g	steam-non-condensable gas mixture
h	hydraulic
i	liquid–gas–vapor interface
s	saturated, smooth
t	total
v	vapor
w	wall

coefficient can be obtained by various calculation methods of film thickness or by using a McAdams modifier. The McAdams modifier is used by Araki [9] and Siddique [10] to consider the shear effect roughly. The McAdams modifier, β, which is multiplied to account for increase in heat transfer as a result of interfacial waviness and rippling, is known to be 1.28 for the laminar condensate film flow, whose film Reynolds number, Re_f, is below 1800.

There are three types of gas mixture layer modeling. The first one is the model in which the original correlations of the Nusselt number and Sherwood number are modified with several multipliers to consider the effects of high mass transfer, developing flow, film roughness and property variations. This model is first introduced by Siddique [10] and is extended by Hasanein [11] to the in-tube steam condensation in the presence of air/helium mixtures.

The second one is diffusion layer modeling using the effective condensation thermal conductivity. Kageyama [12] developed diffusion layer modeling for condensation in vertical tubes with non-condensable gases and Peterson [13] also developed diffusion layer modeling for turbulent vapor condensation both in vertical tubes and on vertical surfaces with non-condensable gases. The local condensation rate is predicted using the analogy between heat and mass transfer, coupled with a reasonable condensate film model. An effective condensation thermal conductivity is derived by expressing the driving potential for mass transfer as a difference in saturation temperatures and using appropriate thermodynamic relationships. Munoz-Cobo [8] developed a condensation model similar to Kageyama's [12] effective condensation thermal conductivity concept. The model considers the effects of high mass flux, mist and film roughness, and it was performed for the air mass fraction below 10%.

The last one uses mass transfer conductance modeling. Condensation in a vertical tube with non-condensable gases can also be represented in terms of mass transfer relations in dealing with the mass transfer problem, and the concept of mass transfer conductance and mass transfer driving potential is used to calculate the mass transfer rate [14]. This mass transfer condensation modeling does not need any assumption inherent in the condensation thermal conductivity, k_c. The mass transfer conductance, the mass transfer driving potential and blowing parameters are derived by considering Couette flow with transpiration at the interface of a flat plate. Araki [9] developed a condensation model similar to Mills [14] mass transfer conductance modeling. The mass fluxes of steam in a flat plate and in a cylinder are compared, and it is shown that the steam mass flux in a cylinder is always smaller than that in a flat plate.

Kuhn [6] developed three kinds of correlations for the condensation heat transfer problem with non-condensable gases for vertical down flow. The simple degradation factor method is developed as an empirical correlation, and two more mechanistic models, the diffusion layer theory and mass transfer conductance modeling, are also developed. Regardless of these differences they all assume the temperature at the liquid–gas interface, which is necessary to calculate the heat transfer coefficients of both the condensate film and the steam–gas mixture, respectively.

Following the above literature survey, a reference mechanistic model of vertical in-tube condensation, which is an iterative method, is developed for steam condensation in the presence of a non-condensable gas in a vertical tube. A non-iterative model is developed based on the reference mechanistic model to enhance applicability to the code, which does not need iteration to find the temperature and pressure at the liquid–gas interface. Without using any interfacial data, the condensation heat transfer coefficient can be expressed in terms of non-dimensional bulk parameters.

2. Reference modeling of vertical in-tube condensation

The total heat flux is expressed as

$$q''_t = h_t(T_b - T_w), \quad (1)$$

where the total heat transfer coefficient, h_t, is divided into the condensate film side heat transfer coefficient, h_f, and the mixture side heat transfer coefficient, h_g, which is composed of convective and condensation terms, h_{cv} and h_{cd}, respectively.

$$\frac{1}{h_t} = \frac{1}{h_f} + \frac{1}{h_g} = \frac{1}{h_f} + \frac{1}{h_{cd} + h_{cv}}. \quad (2)$$

Eq. (2) is based on the assumption that the mixture and the condensate film are in a saturated state, the radiation heat transfer is negligible, and the condensation and sensible heat transfer rate can be calculated simultaneously using the heat and mass transfer analogy. The condensate film thickness is calculated using Munoz-Cobo's approximate method [8] with its accuracy and simplicity, and the condensate film heat transfer coefficient is calculated with Blangetti's film model [7]. The steam–gas mixture side heat transfer coefficients, h_{cd} and h_{cv}, are calculated using the momentum, heat, and mass transfer analogy. The heat flux through the condensate is balanced with the mass transfer through the vapor–gas mixture boundary layer. The condensation heat transfer coefficient can be expressed with the mass transfer rate, m''_v, as follows:

$$h_{cd}(T_b - T_i) = m''_v(i_{g,b} - i_{f,i}), \quad (3)$$

where $i_{g,b}$ is the mixture bulk enthalpy and $i_{f,i}$ the liquid enthalpy at the interface. The mass transfer rate is expressed as follows:

$$m''_v = -g\frac{W_{v,i} - W_{v,b}}{1 - W_{v,i}} = -gB, \quad (4)$$

where g is the mass transfer conductance, B is the mass transfer driving force, and $W_{v,i}$ and $W_{v,b}$ are the mass fractions of the steam at the interface and at the tube centerline, respectively.

From Eqs. (3) and (4), the condensation heat transfer coefficient, h_{cd}, can be derived.

$$h_{cd} = g\frac{i_{g,b} - i_{f,i}}{1 - W_{v,i}}\frac{W_{v,i} - W_{v,b}}{T_i - T_b}. \quad (5)$$

The convective heat transfer, h_{cv}, in Eq. (2) and mass transfer conductance, g, in Eq. (5) can be calculated together using the heat and mass transfer analogy

$$St = \frac{h_{cv}}{\rho_g u_g} = \frac{Nu}{Re_g Pr_g}, \quad (6)$$

and

$$St_{AB} = \frac{g}{\rho_g u_g} = \frac{Sh}{Re_g Sc_g}. \quad (7)$$

There are several methods to calculate the Stanton number, St. Gnielinski's calculation method [15] is used for smooth tubes and Dipprey's calculation method [16] is used for rough tubes, which is applied to this modeling.

$$St = (c_f/2)\Big/\left(1.0 + \sqrt{c_f/2}(5.19[Re_g\sqrt{c_f/2}\epsilon_s/D]^{0.2}Pr^{0.44} - 8.48)\right), \quad (8)$$

where

$$\epsilon_s/D = e^{(3.0-0.4/\sqrt{c_f/2})}. \quad (9)$$

Using the heat and mass transfer analogy, the Stanton number for mass transfer, St_{AB}, is calculated similarly.

$$St_{AB} = (c_f/2)\Big/\left(1.0 + \sqrt{c_f/2}(5.19[Re_g\sqrt{c_f/2}\epsilon_s/D]^{0.2} \times Sc^{0.44} - 8.48)\right). \quad (10)$$

As the heat transfer coefficient strongly depends on the interfacial shear stress, it is very important to adopt the appropriate interfacial friction factor. The friction factor, c_f, is calculated using Wallis's [17] correlation for the interfacial friction factor in the vertical annular flow.

$$c_f/2 = c_{f,s}/2\left(1 + 300\frac{\delta}{D}\right), \tag{11}$$

where $c_{f,s}$ is the friction factor for the smooth tube.

The high mass transfer effect is considered. The Stanton number with blowing, St_b, can be expressed with the Stanton number for no transpiration, St_0, and the blowing parameter, b_h:

$$St_b = St_0\frac{b_h}{e^{b_h} - 1}, \tag{12}$$

where b_h is the alternative heat transfer blowing parameter, which has explicit relation for St_0 rather than the implicit equation, and it can be expressed with several non-dimensional parameters.

$$\begin{aligned} b_h &= \frac{m''_v/G_\infty}{St_0} \\ &= -\frac{JaNu_{cd}}{St_0 Pr_g Re_g}\frac{Nu_f}{Nu_f + [Nu_{cv} + Nu_{cd}]k_g/k_f}, \end{aligned} \tag{13}$$

where Ja is the Jakob number which is defined as $C_{p,g}(T_b - T_w)/i_{fg}$, m''_v the mass transfer rate of the vapor and G_∞ the mass flux of the free stream.

The entrance effect is also considered. For short tubes, where the region of fully developed flow is a small percentage of the total length, the local value of the Nusselt number for uniform velocity and temperature profile in the entrance region is given based on the experimental data for gas [18].

$$Nu = 1.5\left(\frac{x}{D}\right)^{-0.16} Nu_0 \quad \text{for } 1 < \frac{x}{D} < 12 \tag{14}$$

and

$$Nu = Nu_0 \quad \text{for } \frac{x}{D} > 12. \tag{15}$$

3. Non-iterative modeling of vertical in-tube condensation

A non-iterative model for the condensation heat transfer coefficient is developed without any liquid–gas interface information such as interface temperature. The condensate film heat transfer coefficient, h_f, can be calculated by empirical correlation, and both h_{cv} and h_{cd} can be calculated by analogy between heat and mass transfer. The convective and condensation heat transfer coefficients can be calculated separately without using the interface temperature, T_i.

From the energy balance, the amount of heat transferred by the condensing vapor to the liquid–vapor interface by diffusing through the steam-non-condensable gas mixture boundary layer is equal to that transferred through the condensate film. The heat flux through the condensate film layer is calculated by

$$q''_f = h_f(T_i - T_w), \tag{16}$$

where h_f is the heat transfer coefficient in the condensate boundary layer and the heat flux through the mixture boundary layer is

$$q''_v = (h_{cd} + h_{cv})(T_b - T_i), \tag{17}$$

where h_{cd} and h_{cv} are the heat transfer coefficients in the mixture boundary layer by condensation and convection, respectively. The heat fluxes are balanced at the interface.

$$h_f(T_i - T_w) = (h_{cd} + h_{cv})(T_b - T_i) \tag{18}$$

and

$$(h_f + h_{cd} + h_{cv})(T_b - T_i) = h_f(T_b - T_w). \tag{19}$$

Using Eqs. (18) and (19), the temperature difference between the bulk and the interface is expressed with the temperature difference between the bulk and the condensing wall.

$$\begin{aligned} (T_b - T_i) &= \frac{h_f}{h_{cd} + h_{cv}}(T_i - T_w) \\ &= \frac{h_f}{h_f + h_{cd} + h_{cv}}(T_b - T_w). \end{aligned} \tag{20}$$

The mass fraction of steam at the interface, $W_{v,i}$, can be expressed in terms of the bulk mass fraction of steam, $W_{v,b}$, by Taylor expansion.

$$W_{v,i} = W_{v,b} + \left.\frac{\partial W_v}{\partial T}\right]_b (T_i - T_b) + \left.\frac{\partial W_v}{\partial T}\right]_b^2 (T_i - T_b)^2 + \cdots \tag{21}$$

Here $W_{v,i}$ in Eq. (21) can be approximated by taking the first-order differential term only. The properties of temperature and concentration are assumed to change proportionally in the gas mixture boundary layer. This is another expression of heat and mass transfer analogy. The terms of $W_{v,i} - W_{v,b}$ and $1 - W_{v,i}$ can be calculated and inserted into Eq. (5) as follows:

$$W_{v,i} - W_{v,b} \approx \left.\frac{\partial W_v}{\partial T}\right]_b (T_i - T_b), \tag{22}$$

$$\begin{aligned} 1 - W_{v,i} &\approx 1 - W_{v,b} - \left.\frac{\partial W_v}{\partial T}\right]_b (T_i - T_b) \\ &= 1 - W_{v,b} + \frac{h_f}{h_f + h_{cd} + h_{cv}}(T_b - T_w)\left.\frac{\partial W_v}{\partial T}\right]_b \end{aligned} \tag{23}$$

and

$$\begin{aligned} h_{cd} = g i_{fg}[(\partial W_v/\partial T]_b)/(1 - W_{v,b} + h_f/(h_f + h_{cd} + h_{cv}) \\ (T_b - T_w)(\partial W_v/\partial T)]_b)]. \end{aligned} \tag{24}$$

When Eq. (24) is rearranged, a simple quadratic equation for the condensation heat transfer coefficient, h_{cd}, is derived as follows:

$$Ah_{cd}^2 + Bh_{cd} + C = 0, \tag{25}$$

where

$$A = 1 - W_{v,b}, \tag{26}$$

$$B = (h_f + h_{cv})(1 - W_{v,b}) + [h_f(T_b - T_w) - gi_{fg}]\left.\frac{\partial W_v}{\partial T}\right]_b, \tag{27}$$

and

$$C = -gi_{fg}(h_f + h_{cv})\left.\frac{\partial W_v}{\partial T}\right]_b. \tag{28}$$

If the unknown variable, $\partial W_v/\partial T]_b$, is constant, calculated solutions should be exact. The vapor molar fraction and the vapor mass fraction is expressed in terms of the pressure ratio as follows:

$$X_v = P_v/P_t \tag{29}$$

and

$$W_v = \frac{M_v P_v/P_t}{M_g(1 - P_v/P_t) + M_v P_v/P_t}. \tag{30}$$

The partial differentiation of vapor mass fraction about temperature are derived and approximated to be expressed with the bulk properties using the Clausius–Clapeyron equation as follows:

$$\frac{\partial W_v}{\partial T} = \frac{\partial W_v}{\partial P_v}\frac{\partial P_v}{\partial T} = \frac{1}{P_t}N_A\frac{\partial P_v}{\partial T} \approx \frac{i_{fg}\rho_v}{P_t T}N_A, \tag{31}$$

where

$$N_A = \frac{M_v M_g}{\left[M_g(1 - X_v) + M_v X_v\right]^2}. \tag{32}$$

Using the above relation from Eq. (31), B and C in Eqs. (27) and (28) can be rewritten as follows:

$$B = H_1 A + H_2 B_{2T} - B_{3T} \tag{33}$$

and

$$C = -H_1 B_{3T}, \tag{34}$$

where $H_1 = h_f + h_{cv}$, $H_2 = h_f$, $B_{2T} = (i_{fg}\rho_v)/(P_t T)\,(T_b - T_w)N_A$ and $B_{3T} = (gi_{fg}^2\rho_v)/(P_t T)N_A$.

As the coefficients A and C are always positive and negative, respectively, Eq. (25) has the following unique positive solution:

$$h_{cd} = \frac{-B + |B|\sqrt{1 - 4AC/B^2}}{2A}. \tag{35}$$

Eq. (35) can be non-dimensionalized using Eqs. (26), (33) and (34) as follows:

$$Nu_{cd} = \frac{1}{2}\frac{k_f}{k_g}\frac{Nu_f}{\text{div1}} \times \left[-\text{div} + |\text{div}|\sqrt{1 + 4(1 + h_{cv}/h_f)\frac{\text{div1}\cdot\text{div3}}{\text{div2}}}\right], \tag{36}$$

where div = div1 + div2 − div3, div1 = $N_B P_A$, div2 = Ja, and div3 = $Pr_g St_{AB} Re_g/Nu_f k_g/k_f$.

St_{AB} and h_{cv} in Eq. (36) are corrected to consider the effects of high mass transfer and entrance. Eq. (12) is used for the former effect and Eqs. (14) and (15) for the latter effect. The non-dimensional parameters in Eq. (36) are expressed as follows: $W_{g,b} = 1 - W_{v,b}$; $X_{g,b} = 1 - X_{v,b}$; $Nu_{cd} = h_{cd}D_h/k_g$; $Nu_f = h_f D_h/k_f$; $St_{AB} = g/\rho_g u_g$; $Re_g = \rho_g u_g D_h/\mu_g$; $Pr_g = C_{p,g}\mu_g/k_g$; $Ja = C_{p,g}(T_b - T_w)/i_{fg}$; $P_A = P_t^2/(\rho_v^2 i_{fg}^2)C_{p,g}/R_v$; $N_B = X_{g,b}(1 - X_{g,b})[1 + X_{g,b}(M_g/M_v - 1)]$.

As div is always positive and y is a very small value compared with 1, the square root term of Eq. (36) can be expanded and approximated from the expansion of the Taylor series:

$$\sqrt{1 + y} \approx 1 + \frac{1}{2}y, \tag{37}$$

where

$$y = 4(1 + h_{cv}/h_f)\frac{\text{div1}\cdot\text{div3}}{\text{div2}}. \tag{38}$$

Using the approximation of Eq. (37), Eq. (36) can be simplified as follows:

$$Nu_{cd} = (1 + h_{cv}/h_f)\frac{Pr_g St_{AB} Re_g}{N_B P_A + Ja - Pr_g St_{AB}(Re_g/Nu_f)(k_g/k_f)} \tag{39}$$

As the convective heat transfer coefficient, h_{cv}, is negligibly small compared with the film side heat transfer coefficient, h_f, Eq. (39) can be further simplified as follows:

$$Nu_{cd} = \frac{Pr_g St_{AB} Re_g}{N_B P_A + Ja - Pr_g St_{AB}(Re_g/Nu_f)(k_g/k_f)} \tag{40}$$

The definition of Nusselt number for condensation includes the parameters of St_{AB}, Re_g, Pr_g, Nu_f, k_g/k_f, Ja, N_B, and P_A. The developed correlation for the condensation Nusselt number is composed of several non-dimensional parameters used for empirical correlations by several investigators [6,9,11,19–21]. The condensation Nusselt number explicitly shows its dependencies on several parameters such as the gas Reynolds number, gas mole fraction, liquid Nusselt number, Jakob number, interfacial friction factor, roughness and other physical properties. It increases as both the liquid Nusselt number and the gas Reynolds number increase, while it decreases as both the Jakob number and the gas mole fraction increase.

4. Calculation procedures

Two kinds of modeling were performed to be compared with available experimental data. Calculation procedures are quite different between the reference model and the non-iterative model. The reference modeling separately calculates the heat flux through the liquid film and through the mixture boundary layer with an assumed interface temperature. Iteration is needed to get reasonable heat transfer coefficients of h_f, h_{cv} and h_{cd} by modifying the interface temperature, T_i, until the heat fluxes converge within a specified accuracy. The non-iterative modeling separately calculates the heat flux through the liquid film and through the air–vapor boundary layer without an assumed interface temperature.

The condensing tube is divided into axial control volumes of a specific size. The calculations are performed at the center position of each control volume for all parameters and physical properties used. The calculation procedures at each axial location of the tube are explained in Figs. 1 and 2 for the reference modeling and the non-iterative modeling, respectively.

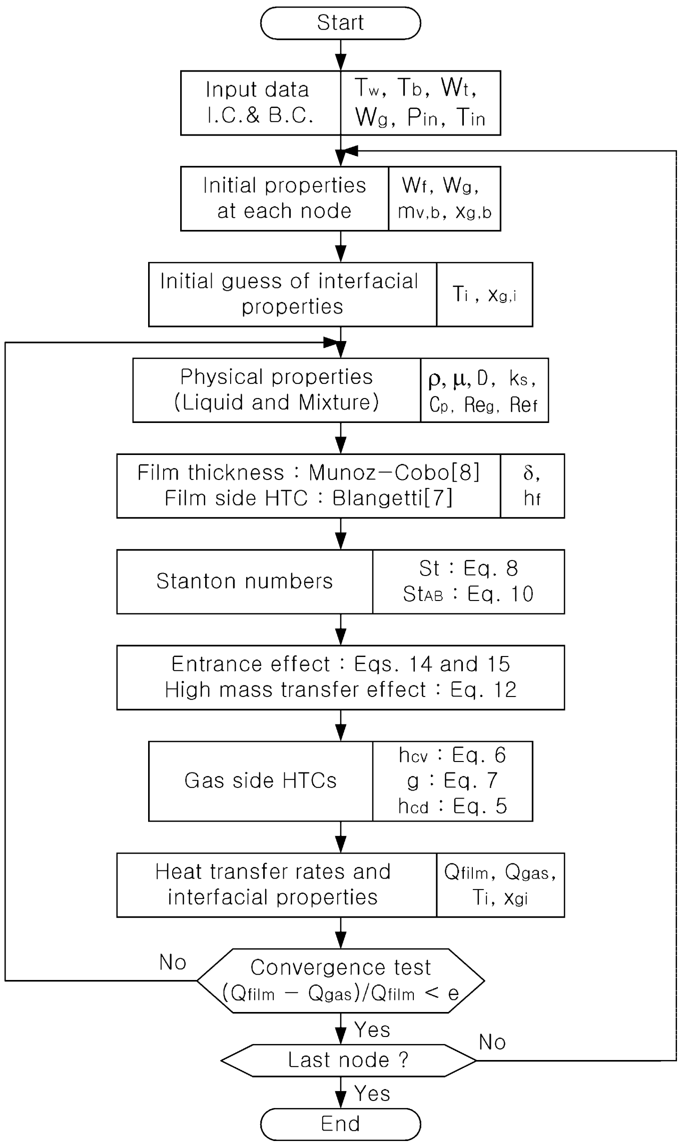

Fig. 1. Calculation procedure of reference simulation of vertical in-tube condensation of steam with non-condensable gas.

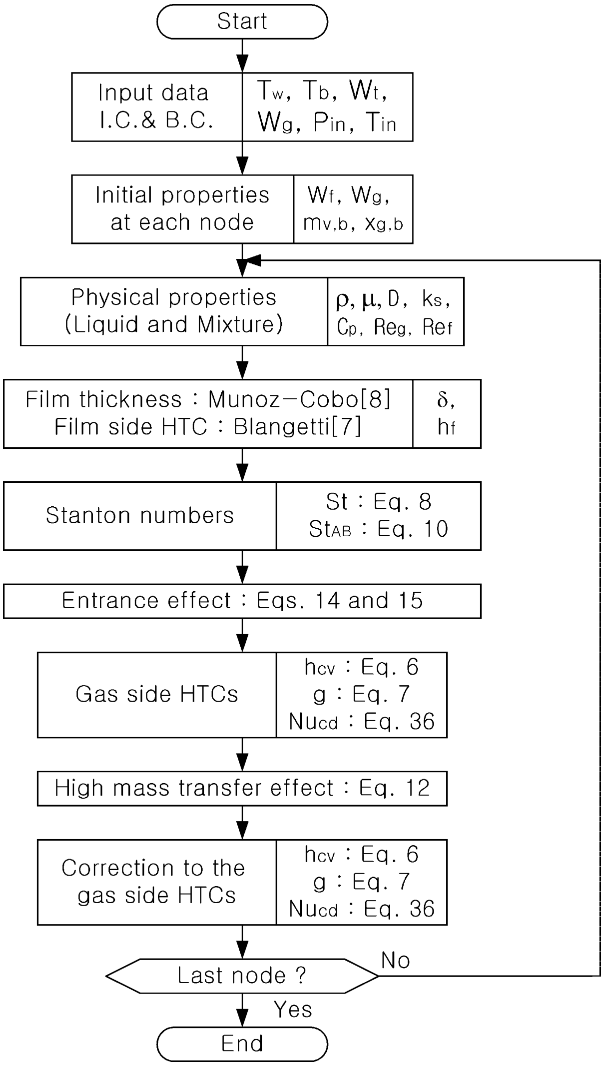

Fig. 2. Calculation procedure of non-iterative simulation of vertical in-tube condensation of steam with non-condensable gas.

5. Results and discussion

For assessment of both the reference and non-iterative models developed here, all 19 sub-tests of Park's experiment [22] for vertical in-tube condensation are used. They span the range of conditions expected for the design of CP-1300 PCCS; the inlet saturated steam temperature ranges from 100°C to 140°C, the inlet air mass fraction from 10% to 40%, and the inlet mixture flow rate from 10.8 to 44.6 kg/h.

The predicted condensation heat transfer coefficients with the non-iterative model are compared with those with reference model, which is shown in Fig. 3.

There is excellent agreement between predictions from the non-iterative model based on Eq. (40) and the reference model with the root mean square error of 7.9%.

Fig. 4 shows a comparison of the predictions of the total heat transfer coefficient from the non-iterative model with Park's experimental data of E11d and E11f.

The experiment E11d was performed with the inlet saturated steam temperatures of 121.4°C, the inlet air mass fraction of 20% and the inlet mixture flow rate of 26.5 kg/h, and the experiment E11f was performed with the inlet saturated steam temperatures of 120.5°C, the inlet air mass fraction of 10.3% and the inlet mixture flow rate of 28.6 kg/h.

Both the reference modeling and the non-iterative modeling well predict the experimental data of both E11d and E11f except for the lower part of the test section. The heat transfer coefficients decrease greatly near the tube

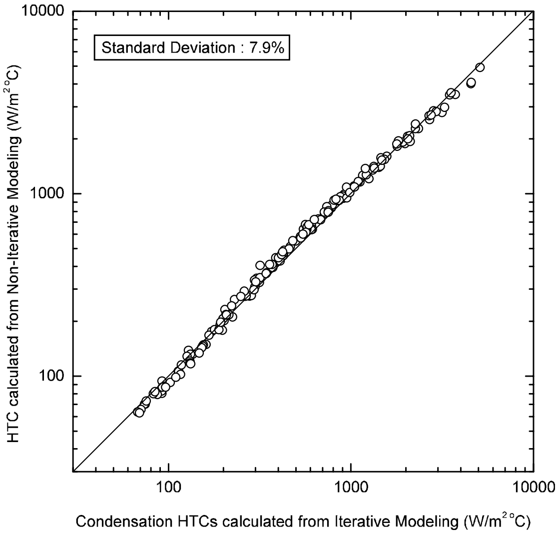

Fig. 3. Comparison of condensation heat transfer coefficients calculated from non-iterative model with those from reference model.

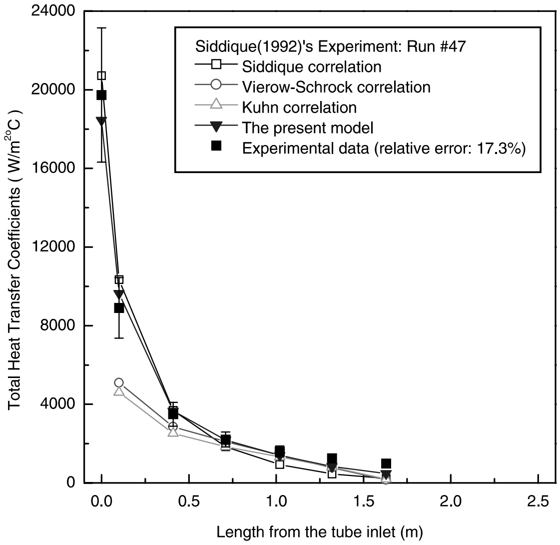

Fig. 5. Comparison of modeling result of the total heat transfer coefficient with Siddique's steam–air experimental data of no. 47.

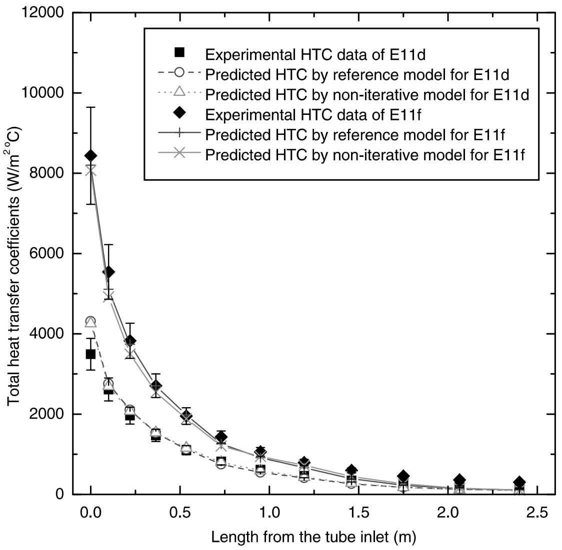

Fig. 4. Comparison of modeling result of the total heat transfer coefficient with Park's experimental data of E11d and E11f.

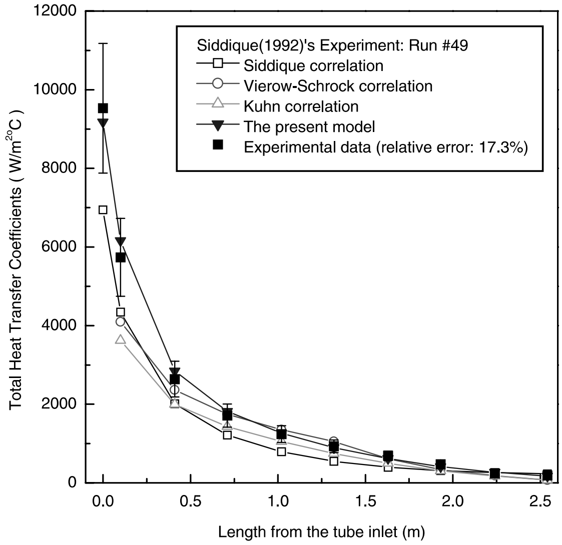

Fig. 6. Comparison of modeling result of the total heat transfer coefficient with Siddique's steam–air experimental data of no. 49.

inlet, where the mixture Reynolds number is highest and the local air mass fraction is lowest. With non-iterative modeling the calculation results show a little lower values than those from the reference modeling.

The non-iterative modeling is also compared with the experimental data of Siddique [21] and Kuhn [6]. Figs. 5 and 6 show the predictions from the non-iterative model for run numbers 47 and 49 of Siddique's steam–air experiment [21], with the similar pressure and mixture Reynolds number, but with different non-condensable gas mass fractions of 0.1 and 0.2, respectively.

The predictions show good agreement with Siddique's data throughout the tube. The agreement is excellent near the tube inlet which has high heat transfer coefficients. The experimental data are also compared with predictions from three existing empirical correlations, Siddique [21], Vierow et al. [20], and Kuhn [6].

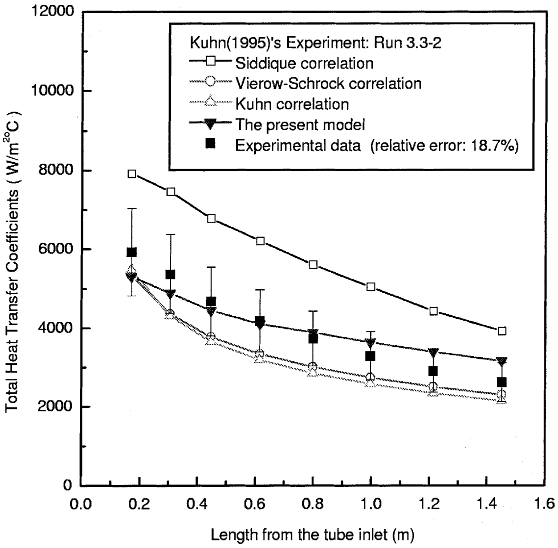

Fig. 7. Comparison of modeling result of the total heat transfer coefficient with Kuhn's steam–air experimental data of no. 3.3-2.

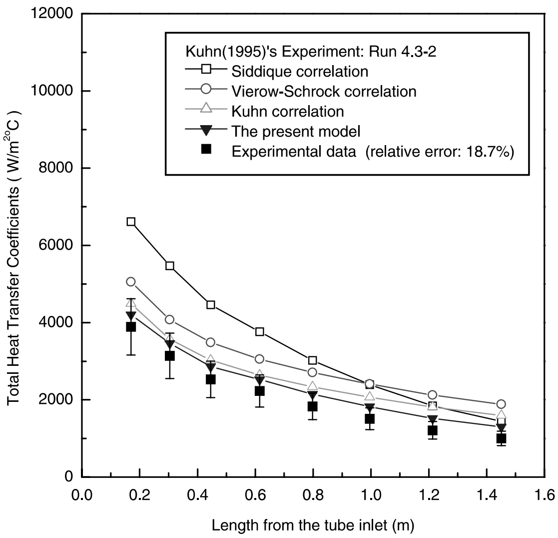

Fig. 8. Comparison of modeling result of the total heat transfer coefficient with Kuhn's steam–air experimental data of no. 4.3-2.

Siddique's empirical correlation well predicts his own experimental data, but the other correlations underpredict the experimental data with high deviations, especially in the range of high heat transfer coefficients.

Figs. 7 and 8 show the predictions from the non-iterative model for run numbers 3.3-2 and 4.3-2 of Kuhn's steam–air experiment [6], with the same air mass fraction and pressure, but with different mixture Reynolds numbers.

The experiments 3.3-2 and 4.3-2 are performed with the similar mass fraction and pressure, but with different inlet mixture flow rates of 59.5 and 31.5 kg/h, respectively. The predictions shows good agreement with Kuhn's data of both steam–air experiments throughout the tube. The maximum errors for run numbers 3.3-2 and 3.4-2 are 26.8% and 23.4% near the tube outlet, respectively.

6. Conclusions and recommendation

Both reference and non-iterative models are developed for steam condensation with non-condensable gas in a vertical tube. Basically the analogy between momentum, heat and mass transfer is used and two boundary layers are simulated separately in this modeling. For the condensate boundary layer Blangetti's film model is used to calculate the local film heat transfer coefficients using the film thickness calculated with Munoz-cobo's approximate modeling method, and for the mixture boundary layer a new iterative model is developed to predict the mixture layer side heat transfer coefficients. The reference model is based on the energy balance at the liquid–gas interface. To eliminate the complexities caused by the iteration, the non-iterative model is developed to provide a correlation which has a physical background and is expressed with several non-dimensional parameters.

The predictions of the non-iterative model show excellent agreement with those of the reference model over the entire region of the test section. The predictions from the non-iterative model are compared with the experimental data of Park [22] and agreement is reasonable in all cases compared. Also, the non-iterative model is assessed against the experimental data of Siddique [21] and Kuhn [6] and shows good agreement.

With its simplicity and meaningful derivation, the non-iterative model can be used to improve the condensation models in the presence of non-condensable gases in thermal-hydraulic codes such as RELAP5 and RETRAN-3D.

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