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Non-Iterative Condensation Model for Steam Condensation with Noncondensable Gas in a Vertical Tube

Hyun Sik PARK and Hee Cheon NO

Dept. of Nuclear Eng., Korea Advanced Institute of Science and Technology,
371-1 Ku-song Dong, Yu-song Gu, Taejon 305-701, Korea
e-mail:hcno@nesun1.kaist.ac.kr

Abstract

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 the Stanton number for mass transfer, gas mixture Reynolds number, air mass fraction, Jakob number, gas Prandtl number and liquid film Nusselt number. The comparison shows that the non-iterative model reasonably well predict the experimental data.

1 Introduction

The steam condensation in the presence of noncondensable gas in vertical tubes is an important thermal-hydraulic phenomenon which occurs in an 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 noncondensable 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 their experimental data.

Several methods have been developed to calculate the film thickness, δ, and the film side heat transfer coefficient, h_f, was also calculated both for the laminar and turbulent condition.

There are three types of gas mixture layer modeling. The first one is the model in which the original correlations of 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[2, 3]. The second one is the diffusion layer modeling using the effective condensation thermal conductivity[4, 5, 6]. 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. The last one uses the mass transfer conductance modeling. Condensation in a vertical tube with noncondensable gases can also be represented in terms of mass transfer relations in dealing with the mass transfer problem, and the concept of the mass transfer conductance and mass transfer driving potential is used to calculate the mass transfer rate[7, 8]. 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 noncondensable gas in a vertical tube[9]. 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 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 the interface temperature. The condensate film heat transfer coefficient, h_f, can be calculated by the empirical correlation, and both h_{cv} and h_{cd} can be calculated by the analogy between heat and mass transfer. The convective and condensation heat transfer coefficients can be calculated separately without using the interface temperature, T_i.

The total heat flux is expressed as

$$q_t'' = h_t \cdot (T_b - T_w), \tag{1}$$

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.

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

Equation 2 is based on the assumption that the mixture and the condensate film are at saturated state, the radiation heat transfer is negligible, and the condensation and the sensible heat transfer rate are calculated simultaneously using the heat and mass transfer analogy. The condensate film thickness is calculated using Munoz-Cobo's approximate method[6] with its accuracy and simplicity, and the condensate film heat transfer coefficient is calculated with Blangetti's film model[10]. 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.

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

$$q_f'' = h_f \cdot (T_i - T_w), \tag{3}$$

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

$$q_v'' = (h_{cd} + h_{cv}) \cdot (T_b - T_i), \tag{4}$$

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.

$$h_f \cdot (T_i - T_w) = (h_{cd} + h_{cv}) \cdot (T_b - T_i) \tag{5}$$

and

$$(h_f + h_{cd} + h_{cv}) \cdot (T_b - T_i) = h_f \cdot (T_b - T_w). \tag{6}$$

Using Equations 5 and 6, the temperature difference between the bulk and the interface is expressed with the temperature difference between the bulk and the condensing wall.

$$(T_b - T_i) = \frac{h_f}{h_{cd} + h_{cv}} \cdot (T_i - T_w) = \frac{h_f}{h_f + h_{cd} + h_{cv}} \cdot (T_b - T_w). \tag{7}$$

Also the condensation heat transfer coefficient, h_{cd}, can be derived as follows:

$$h_{cd} = g \cdot \frac{i_{g,b} - i_{f,i}}{1 - W_{v,i}} \cdot \frac{W_{v,i} - W_{v,b}}{T_i - T_b}. \tag{8}$$

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.

$$W_{v,i} = W_{v,b} + \frac{\partial W_v}{\partial T}]_b \cdot (T_i - T_b) + \frac{\partial W_v}{\partial T}]_b^2 \cdot (T_i - T_b)^2 + ... \tag{9}$$

Here $W_{v,i}$ in Equation 9 can be approximated by taking the first order differential term only and the terms of $W_{v,i} - W_{v,b}$ and $1 - W_{v,i}$ can be calculated and inserted into Equation 8 as follows:

$$W_{v,i} - W_{v,b} \approx \frac{\partial W_v}{\partial T}]_b \cdot (T_i - T_b), \tag{10}$$

$$1 - W_{v,i} \approx 1 - W_{v,b} - \frac{\partial W_v}{\partial T}]_b \cdot (T_i - T_b) = 1 - W_{v,b} + \frac{h_f}{h_f + h_{cd} + h_{cv}} \cdot (T_b - T_w) \cdot \frac{\partial W_v}{\partial T}]_b \tag{11}$$

and

$$h_{cd} = g \cdot i_{fg} \cdot \frac{\frac{\partial W_v}{\partial T}]_b}{1 - W_{v,b} + h_f/(h_f + h_{cd} + h_{cv}) \cdot (T_b - T_w) \cdot \frac{\partial W_v}{\partial T}]_b}. \tag{12}$$

When Equation 12 is rearranged, a simple quadratic equation for the condensation heat transfer coefficient, h_{cd}, is derived as follows:

$$A \cdot h_{cd}^2 + B \cdot h_{cd} + C = 0, \tag{13}$$

where

$$A = 1 - W_{v,b}, \tag{14}$$

$$B = (h_f + h_{cv}) \cdot (1 - W_{v,b}) + [h_f \cdot (T_b - T_w) - g \cdot i_{fg}] \cdot \frac{\partial W_v}{\partial T}]_b, \tag{15}$$

and

$$C = -g \cdot i_{fg} \cdot (h_f + h_{cv}) \cdot \frac{\partial W_v}{\partial T}]_b. \tag{16}$$

If the unknown variable, $\frac{\partial W_v}{\partial T}]_b$, 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:

$$X_v = P_v/P_t, \tag{17}$$

and

$$W_v = \frac{M_v \cdot P_v/P_t}{M_g \cdot (1 - P_v/P_t) + M_v \cdot P_v/P_t}, \tag{18}$$

and its partial differentiation about temperature are expressed as follows:

$$\frac{\partial W_v}{\partial T} = \frac{\partial W_v}{\partial P_v} \cdot \frac{\partial P_v}{\partial T} = \frac{1}{P_t} \cdot N_A \cdot \frac{\partial P_v}{\partial T}, \tag{19}$$

where

$$N_A = \frac{M_v \cdot M_g}{[M_g \cdot (1 - X_v) + M_v \cdot X_v]^2}. \tag{20}$$

Equation 19 can be expressed with the bulk properties using the Clausius-Clapeyron equation.

$$\frac{\partial W_v}{\partial T} = \frac{1}{P_t} \cdot N_A \cdot \frac{\partial P_v}{\partial T} = \frac{1}{P_t} \cdot N_A \cdot \frac{i_{fg}}{T \cdot v_{fg}} \approx \frac{i_{fg}\rho_v}{P_t T} \cdot N_A. \tag{21}$$

Using the above relation from Equation 21, B and C in Equations 15 and 16 can be rewritten as follows:

$$B = H_1 \cdot A + H_2 \cdot B_{2T} - B_{3T}, \tag{22}$$

and

$$C = -H_1 \cdot B_{3T}, \tag{23}$$

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

As the coefficients A and C are always positive and negative, respectively, Equation 13 has the following unique positive solution:

$$h_{cd} = \frac{-B + \sqrt{B^2 - 4AC}}{2A}. \tag{24}$$

Equation 24 is rearranged as follows:

$$h_{cd} = \frac{(H_1 \cdot A + H_2 \cdot B_{2T} - B_{3T})}{2A} \cdot \left(\sqrt{1 + \frac{4A \cdot H_1 \cdot B_{3T}}{(H_1 \cdot A + H_2 \cdot B_{2T} - B_{3T})^2}} - 1 \right) \tag{25}$$

To simplify Equation 25, the term in the square root is defined as

$$y = \frac{4 \cdot A \cdot H_1 \cdot B_{3T}}{(H_1 \cdot A + H_2 \cdot B_{2T} - B_{3T})^2}. \tag{26}$$

As y is a very small value compared with 1, the square root term of Equation 25 can be expanded and approximated from the expansion of the Taylor series:

$$\sqrt{1 + y} = 1 + \frac{1}{2}y - \frac{1}{4}y^2 + ... \approx 1 + \frac{1}{2}y. \tag{27}$$

Several calculation results show that Equation 27 produces higher values than the original one does within 8%.

Now, Equation 25 can be nondimensionalized using Equations 26 and 27 as follows:

$$Nu_{cd} = \frac{(1 + h_{cv}/h_f) \cdot Pr_g}{(1 + h_{cv}/h_f) \cdot N_B \cdot P_A / (St_{AB} \cdot Re_g) + Ja / (St_{AB} \cdot Re_g) - Pr_g / Nu_f \cdot k_g / k_f}, \tag{28}$$

where $W_{g,b} = 1 - W_{v,b}$, $X_{g,b} = 1 - X_{v,b}$, $Nu_{cd} = h_{cd} D_h / k_g$, $Nu_f = h_f D_h / k_f$, $St_{AB} = g/\rho_g u_g$, $Re_g = \rho_g u_g D_h / \mu_g$, $Pr_g = Cp_g \mu_g / k_g$, $Ja = Cp_g \cdot (T_b - T_w)/i_{fg}$, $P_A = P_t^2 / (\rho_v^2 \cdot i_{fg}^2) \cdot Cp_g / R_v$, and $N_B = X_{g,b} \cdot (1 - X_{g,b}) \cdot [1 + X_{g,b} \cdot (M_g/M_v - 1)]$. St_{AB} and h_{cv} in Equation 28 are corrected to consider the effects of high mass transfer and entry.

As the convective heat transfer coefficient, h_{cv}, is negligibly small compared with the film side heat transfer coefficient, h_f, Equation 28 can be further simplified as follows:

$$Nu_{cd} = \frac{Pr_g \cdot St_{AB} \cdot Re_g}{N_B \cdot P_A + Ja - Pr_g / Nu_f \cdot k_g / k_f \cdot St_{AB} \cdot Re_g} \tag{29}$$

The definition of Nusselt number for condensation includes the parameters of St_{AB}, Re_g, Nu_f, Pr_g, k_g/k_f, Ja, N_B, and P_A. The developed correlation for the condensation Nusselt number is composed of several nondimensional parameters used for empirical correlations by several investigators[3, 8, 11, 12, 13, 14].

3 Calculation procedures

Three kinds of modeling were performed to be compared with available experimental data. Calculation procedures are quite different amomg the reference model, the non-iterative model and the iterative model without any interface information. The first reference modeling[9] separately calculates the heat flux through the liquid film and through the mixture boundary layer with an assumed interface temperature. It needs iteration 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 reference modeling separately calculates the heat flux through the liquid film and through the air-vapor boundary layer. In the second non-iterative modeling the condensation Nusselt number is calculated considering the entrance effect to calculate the blowing parameter, and it is corrected considering the high mass transfer effect with the calculated blowing parameter. In the third iterative modeling without any interface information the condensation Nusselt number is calculated iteratively.

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 figures 1 and 2 for the non-iterative modeling and the iterative modeling without any interface information, respectively.

4 Results and discussion

For assessment of the reference model, the non-iterative model and the iterative model without any interface information developed here, all 19 sub-tests of Park's experiment[15] for vertical in-tube condensation are used. They span the ranges of conditions expected for the design of CP-1300 PCCS; the inlet saturated steam temperature ranges from 100^oC to 140^oC, the inlet air mass fraction from 10% to 40%, and the inlet mixture flow rate

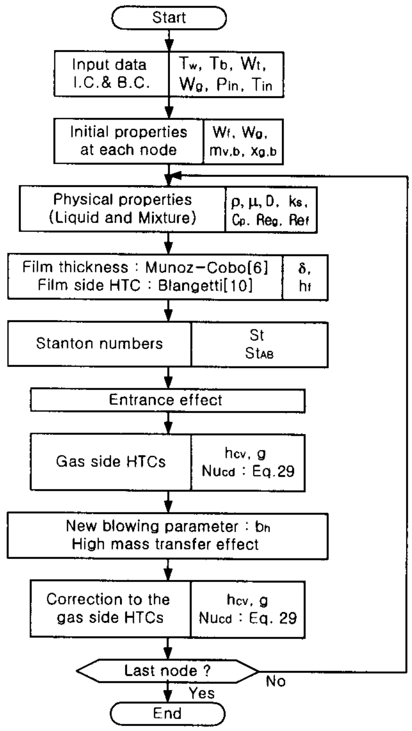

Fig. 1: Calculation procedure of non-iterative simulation of vertical in-tube condensation of steam with noncondensable gas

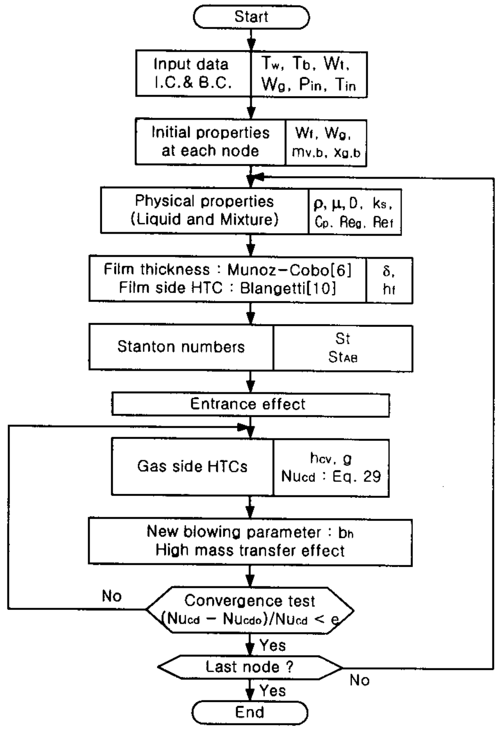

Fig. 2: Calculation procedure of iterative simulation of vertical in-tube condensation of steam with noncondensable gas : no interface information

from $10.8kg/hr$ to $44.6kg/hr$.

The predicted condensation heat transfer coefficients with the non-iterative model are compared with those with reference model, which is shown in Figure 3. There are excellent agreement between predictions from the non-iterative model and the reference model with the root mean square error of 7.9%.

Figure 4 shows comparisons of the predictions of the total heat transfer coefficients from the above three condensation models with Park's experimental data of *E11d* and *E11f*. The experiment *E11d* was performed with the inlet saturated steam temperatures of 121.4^oC, the inlet air mass fraction of 20% and the inlet mixture flow rate of $26.5kg/hr$, and the experiment *E11f* was performed with the inlet saturated steam temperatures of 120.5^oC, the inlet air mass fraction of 10.3% and the inlet mixture flow rate of $28.6kg/hr$. Both the reference modeling and the non-iterative modeling predict well 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 inlet, where the mixture Reynolds number is highest and the local air mass fraction is lowest. With the non-iterative modeling the calculation results show little lower values than those from the reference modeling. Also the predictions from the non-iterative modeling are in good agreement with those from the iterative modeling without any interface information.

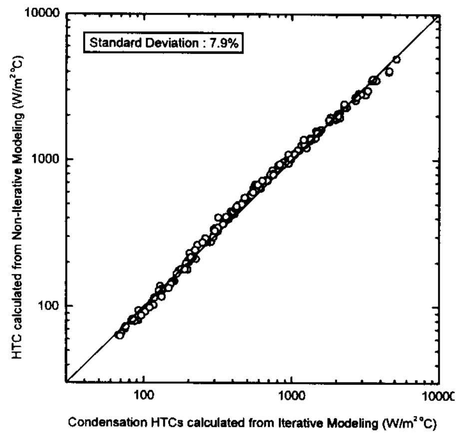

Fig. 3: Comparison of condensation heat transfer coefficients calculated from non-iterative model with those from reference model

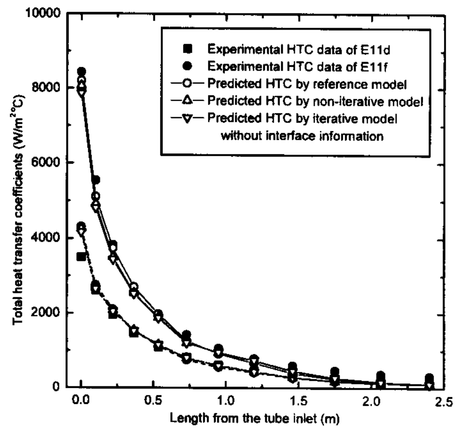

Fig. 4: Comparison of modeling result of the total heat transfer coefficient with Park's experimental data of *E11d* and *E11f*

5 Conclusion and Recommendations

A non-iterative models are developed for the steam condensation with noncondensable 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 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 the correlation which has physical background and is expressed with several nondimensional parameters. The predictions by the non-iterative model also show excellent agreement with those by 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 and agreement is reasonable in all cases compared except for the region of the tube outlet.

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

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