

Pool Boiling and Condensation Analysis for a Vertical Tube Bundle Condenser

W. Zhou¹, B. Wolf¹, and S. T. Revankar^{1,2}

¹ Purdue University, West Lafayette, Indiana, USA

² POSTECH, Pohang, Republic of South Korea

zhouw@purdue.edu, shripad@ecn.purdue.edu

Abstract

An experimental and theoretical study is performed for the steam condensation in a vertical tube bundle passive condenser simulating PCCS condenser in the ESBWR. Four condenser tubes are submerged in a water pool where the heat from the condenser tube is removed through boiling heat transfer. Condenser tubes with a full length/diameter scale are used to obtain the condensation data with various process parameters. The comparison of tube bundle experimental data with the single tube data by both the experiments and models shows that the single tube secondary heat transfer coefficient (HTC) is between 25% - 35% less than what was recorded for the tube bundle, and the tube bundle condensation rates are slightly higher than the data from the single tube test sections due to turbulent mixing effect which increases the condensation heat removal. The turbulent mixing on the secondary side decreases the ΔT between pool water and condenser tube outer wall, causing an increase in secondary HTC. This increase in secondary HTC thus results in higher condensate mass flow rates. Tube bundle boundary layer model and heat and mass analogy model were then developed for the prediction of the filmwise steam condensation with noncondensable (NC) gas in a vertical tube bundle. The predictions from the models are compared with the experimental data for various complete condensation and through flow conditions and the agreement is satisfactory. The local parameters predicted by the boundary layer model and heat and mass analogy model with tube bundle pool boiling can also be predicted with the axial distance from entrance for different NC gas fractions and system pressures.

Keywords: Tube bundle, boundary layer model, heat and mass analogy model, condensation, pool boiling.

1. Introduction

Condensation is an important mode of heat transfer that is widely applicable in the power industry due to its ability to achieve high heat transfer coefficients. General Electric's economic simplified boiling water reactor (ESBWR) includes a passive heat exchanger to depressurize the containment by condensing steam in vertical tubes through a pool of water [1]. This heat exchanger is called a passive containment cooling system (PCCS). The PCCS condenser must provide enough heat removal to keep the pressure in the containment less than the design pressure after a design basis accident, such as a loss of coolant accident (LOCA). A detailed knowledge of the PCCS heat transfer capabilities is necessary in predicting the containment

pressure following a design basis accident. The PCCS condenser has three modes of operation: through flow, cyclic venting, and complete condensation mode [2].

Experiments have been carried out by several researchers on condensation with and without the presence of NC gas in a vertical tube [3-11]. Most of these experiments use forced convection for the heat removal mechanism. However, the PCCS condenser uses pool boiling as the method for heat removal. Kim and No [7] did use pool boiling, but in a large rectangular tank and the experiments did not involve the presence of NC gas. Oh and Revankar [9,10] carried out tests with a single condensing tube using pool boiling in a cylindrical secondary pool and the presence of air as the NC gas. A new multi-tube test facility was designed and constructed to extend work of Oh and Revankar [9, 10] as well as to investigate the tube bundle effect on PCCS heat removal capabilities by Revankar et al [12]. The comparison of tube bundle experimental data with the single tube data by both the experiments and models shows that based on a linear fit for the trends, the single tube secondary HTC is between 25% - 35% less than what was recorded for the tube bundle, and the tube bundle condensation rates are slightly higher than the data from the single tube test sections [13]. In Fig. 1, data of the secondary HTC for tube bundle [13] and single tube tests [9, 10] are shown along with pool boiling HTC.

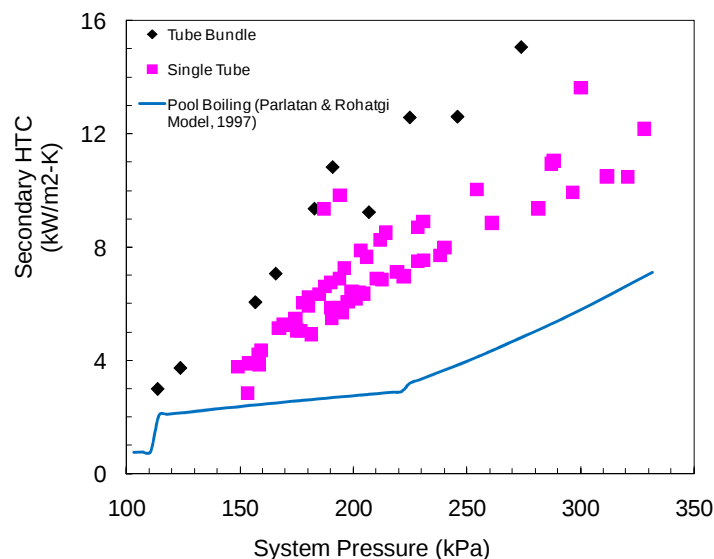


Figure 1. Comparison of Single Tube and Tube Bundle Secondary HTC

Similarly, in order to elucidate boiling heat transfer characteristics for each tube for tube bundles, Fujita and Hidaka [14] performed an experimental investigation of pool and flow boiling of Freon-113 at 0.1 MPa using two typical tube arrangements for horizontal tube bundles. It was found in this study that the boiling heat transfer coefficient (HTC) of each tube in a bundle was higher than that for an isolated single tube in pool boiling.

For vertical tube bundle in the PCCS, the bubbles are generated and rise up through the narrow clearance between tubes, induce turbulence and cause turbulence mixing. Such two-phase flow may impose certain influences on tubes in its passage. Its overall effect on heat transfer depends on various parameters such as bundle arrangement, tube pitch, tube location, thermal and hydrodynamic conditions, and system pressure. The larger secondary HTC is a result of

turbulent mixing of two phase flow in the secondary water pool. This increases the secondary pool temperature, which decreases the temperature difference between the outer wall of the condensing tubes and the secondary pool water. A decrease in the temperature difference will cause the secondary HTC to be larger. This increase in secondary HTC is enough to make the condensate mass flow rates higher than for a single tube. This effect could cause a better heat removal capability. The tube bundle effect thus affects the secondary pool boiling and then the in-tube condensation.

However, from the literature review conducted before, there is no such consideration in all the boundary layer models and the heat and mass analogy models. Here in this work we improved the existing boundary layer models on tube condensation by considering the tube bundle effects and the induced secondary heat transfer coefficient and condensate rate enhancement.

2. Tube Bundle Model

It is postulated that in pool boiling systems, two basic mechanisms take part in the heat transfer process for the pool boiling of saturated fluids: the ordinary convective mechanism of heat transfer, and the nucleate boiling mechanism associated with bubble nucleation and growth as proposed by Chen [15,16] for flow boiling. It is further postulated that these two mechanisms are additive in their contributions to total heat transfer.

Considering first the convective mechanism, it was recognized that at the two limits of 0 and 100% quality, the convective heat transfer should be described by the Dittus-Boelter type of correlation. It was then postulated that in the two-phase region where both liquid and vapor are present, the convective heat transfer should still be described by a modified form of the Dittus-Boelter equation.

$$h_{mac} = 0.023 Re^{0.8} Pr^{0.4} (k / D) \quad (1)$$

In this equation, the Prandtl and the Reynolds numbers and the thermal conductivity represent effective values associated with the two-phase fluid. We may define three parameters as ratios of these quantities divided by the liquid quantities.

$$\begin{aligned} \beta &= Pr / Pr_L \\ \gamma &= k / k_L \\ F &= (Re / Re_L)^{0.8} \end{aligned} \quad (2)$$

In the case of ordinary fluids, the Prandtl numbers of the liquid and of the vapor are normally of the same magnitude. The Prandtl number of the two-phase fluid should therefore also be of the same magnitude. Furthermore, since the heat is transferred through an annular film of liquid adhering to the wall, it is expected that the liquid properties would have the dominant effect. For these reasons it is reasonable to assume that β and γ may be taken to be unity as a first approximation. Equation 1 may then be rewritten as

$$h_{mac} = 0.023 (Re_L)^{0.8} (Pr_L)^{0.4} (k_L / D) F \quad (3)$$

The function Re_L in Equation 3, the liquid Reynolds number of two phase flow, can be described as below:

$$\text{Re}_L = \frac{(1-x)G \frac{A}{A_L} D}{\mu_L} \quad (4)$$

where

$x = G_g / G$, is the steam quality of the two phase flow;

$G_g = \dot{m} / A$, is the steam mass flux, which can be calculated as mass flow per unit area;

$q'' = \dot{m} h_{fg} \Rightarrow \dot{m} = q'' / h_{fg}$. $\dot{m}$, mass flow can be calculated using heat flux divided by latent heat;

$G = \rho_L A \Delta l / A = \rho_L \Delta l$, is the two phase mass flux, which can be calculated using the secondary side water level change with time (Δl) multiplied by liquid density;

A , is the effective two phase flow area, and A_L , is the effective liquid phase flow area given as [17],

$$\frac{A_L}{A} = \frac{\frac{(1-x)GA}{\rho_L}}{\frac{(1-x)GA}{\rho_L} + \frac{xGA}{\rho_v}} = \frac{(1-x)\rho_v}{(1-x)\rho_v + x\rho_L}, \quad (5)$$

and D , is the hydraulic diameter.

The function F in Equation 3, the ratio of the two-phase Reynolds number to the liquid Reynolds number, is based on the liquid fraction. This ratio is strictly a parameter by analogy to momentum transfer in two-phase flow. It may be assumed that F is a function of the Martinelli parameter, X_{tt} .

$$F = \begin{cases} 1.0 & \text{for } \frac{1}{X_{tt}} < 0.1 \\ 2.35 \left(0.213 + \frac{1}{X_{tt}} \right)^{0.736} & \text{for } \frac{1}{X_{tt}} > 0.1 \end{cases} \quad (6)$$

where

$$X_{tt} = \left(\frac{\mu_L}{\mu_v} \right)^{0.1} \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_v}{\rho_L} \right)^{0.5} \quad (7)$$

The analysis of Forster and Zuber [18] was taken as a basis for the formulation of a nucleate boiling mechanism of heat transfer with effective ΔT and Δp .

$$h_{mic} = 0.00122 \left(\frac{k_L^{0.79} c_{pL}^{0.45} \rho_L^{0.49} g_c^{0.25}}{\sigma^{0.5} \mu_L^{0.29} h_{fg}^{0.24} \rho_v^{0.24}} \right) (\Delta T_e)^{0.24} (\Delta p_e)^{0.75} \quad (8)$$

Then a suppression factor, S , was defined as proposed by Chen [15,16]:

$$S = (\Delta T_e / \Delta T)^{0.24} (\Delta p_e / \Delta p)^{0.75} \quad (9)$$

Combining with Equation 4, we then obtain an expression for nucleate boiling coefficient in terms of the suppression factor and the total superheat.

$$h_{mic} = 0.00122 \left(\frac{k_L^{0.79} c_{pL}^{0.45} \rho_L^{0.49} g_c^{0.25}}{\sigma^{0.5} \mu_L^{0.29} h_{fg}^{0.24} \rho_v^{0.24}} \right) (\Delta T)^{0.24} (\Delta p)^{0.75} S \quad (10)$$

The suppression factor, S , approaches unity at zero flow rate and zero at infinite flow rate. It was postulated, subject to experimental verification, that in all ranges of flow S can be represented as a function of the local two-phase Reynolds number.

The total heat transfer coefficient is then obtained as the sum

$$h = h_{mic} + h_{mac} \quad (11)$$

3. Tube Bundle Boundary Layer Model Results

Using our experimental conditions, test facility geometries and the above equations for pool boiling, we can calculate the nucleate boiling, convective and overall heat transfer coefficients for tube bundle pool boiling, and then calculate condensation heat flux and condensation HTC for the predictions using boundary layer model [9,10,19].

Complete Condensation

Using Equations 3, 10 and 11, we can calculate the convective, nucleate boiling, and the overall HTCs with different pressures. The results are plotted as shown in Figure 2, and compared with the experimental overall HTCs. From the figure, we can see the modeling results using Equations 3, 10 and 11 have a good agreement with the experimental results compared to the results using single tube model [9, 13,19].

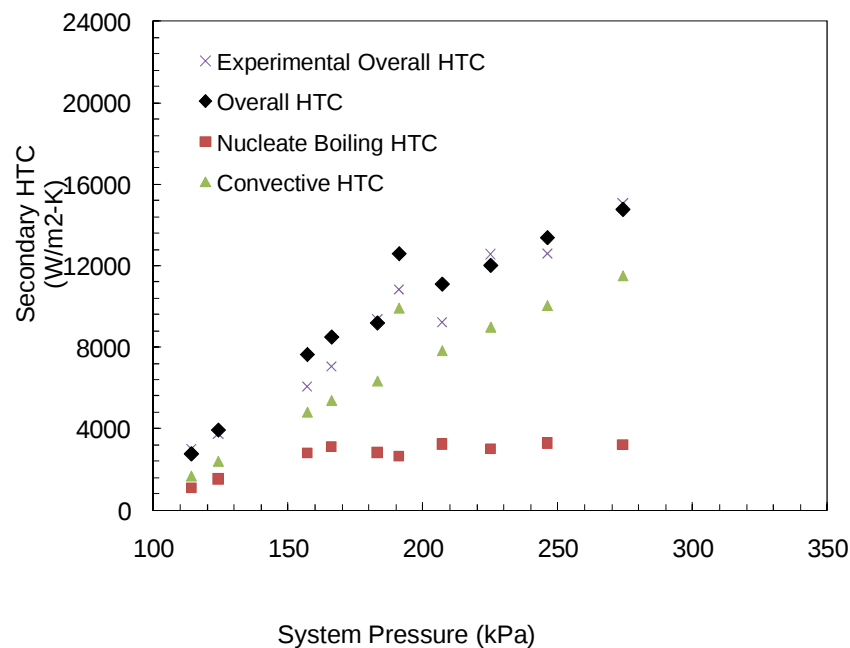


Figure 2 Overall, Nucleate Boiling, and Convective HTCs with Different System Pressures

Then the Equations 3, 10 and 11 were put into the boundary layer model to replace the existing secondary pool boiling correlation, and the results using tube bundle pool boiling correlation were obtained [19]. The model condensate mass flows and condensation HTC's were compared with the experimental data [13, 20] as shown in Figures 3 and 4. Good agreement between the model results with experimental results was obtained.

Through Flow

Using Equations 3, 10 and 11, we can calculate the convective, nucleate boiling, and the overall HTC's with different noncondensable gas fractions, inlet steam flow rates and system pressures. The results are plotted as shown in Figure 5, and compared with the experimental overall HTC's. From the figure, we can see the model results using Equations 3, 10 and 11 have a good agreement with the experimental results compared to the results using single tube model. For through flow, there are noncondensable gas fraction, system pressure and inlet steam flow rate which affect condensation. So, we compared the experimental and model secondary HTC's for through flow, which is different from what we plotted for complete condensation.

Then the Equations 3, 10 and 11 were put into the boundary layer model [9, 19] to replace the existing secondary pool boiling correlation, and the results using tube bundle pool boiling correlation were obtained. The modeling condensate mass flows and condensation HTC's were compared with the experimental ones as shown in Figures 6-7. Good agreement of modeling results with experimental results was obtained.

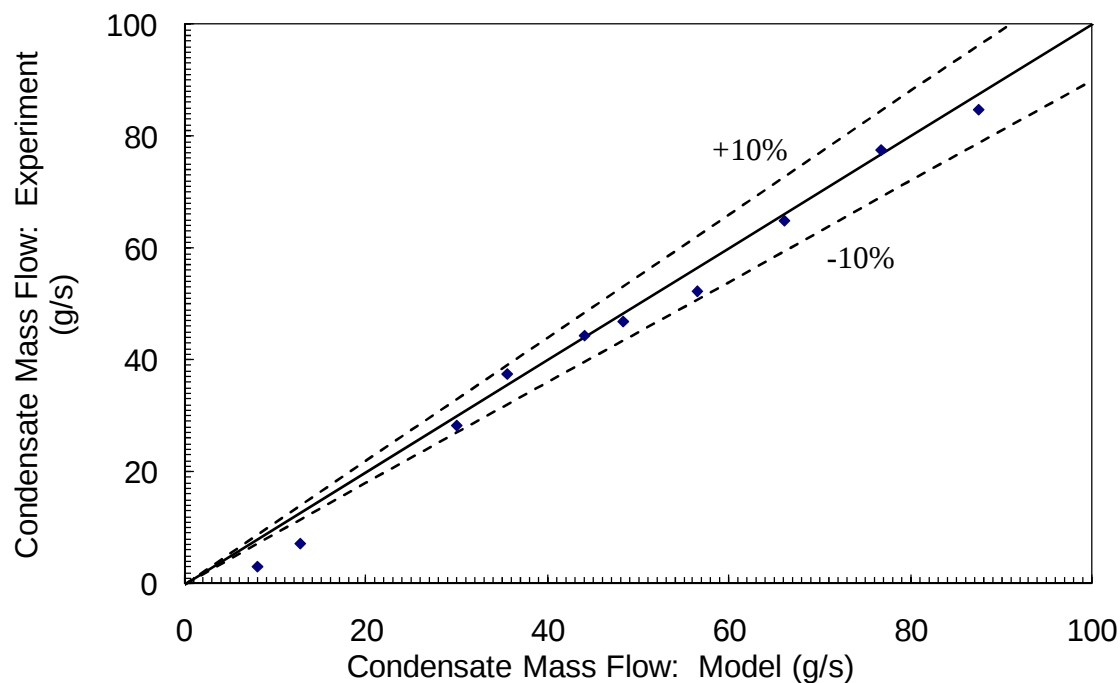


Figure 3 Comparison of Condensate Mass Flow for Tube Bundle Model Results with Experimental Results

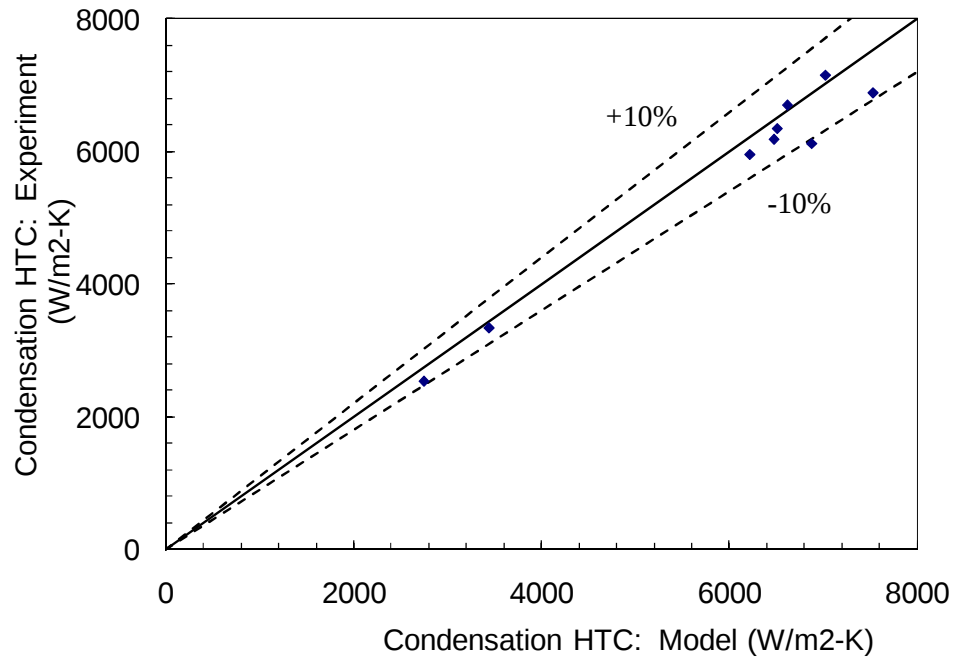


Figure 4 Comparison of Condensation HTC for Tube Bundle Model Results with Experimental Results

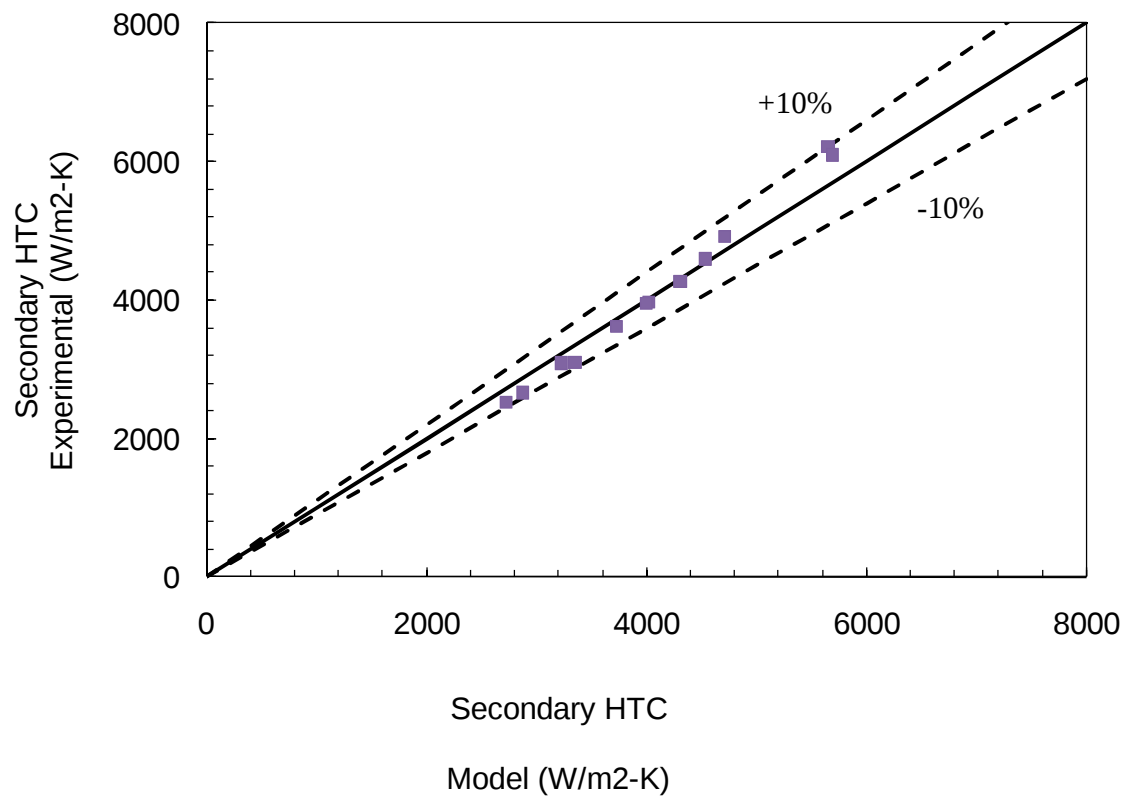


Figure 5 Comparison of Experimental and Modeling Results of Secondary HTCs for Through Flow

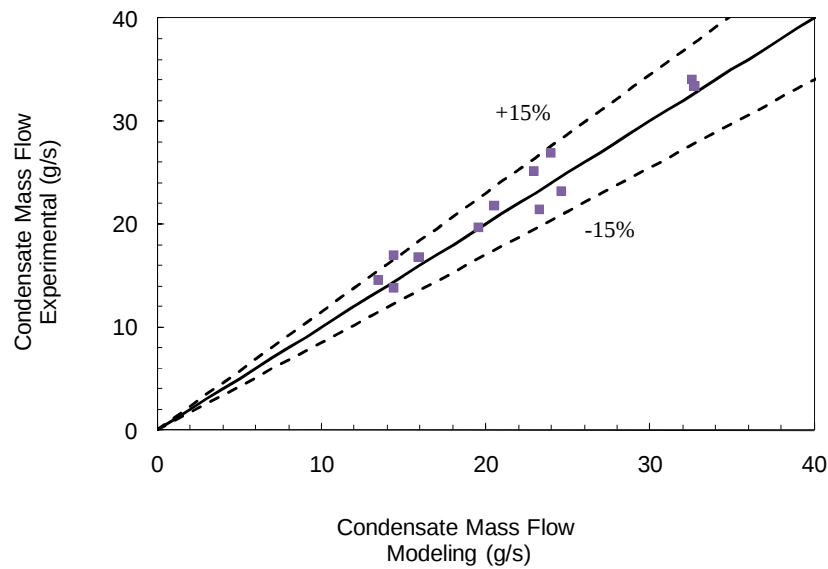


Figure 6 Comparison of Condensate Mass Flow for Tube Bundle Model Results with Experimental Results for Through Flow

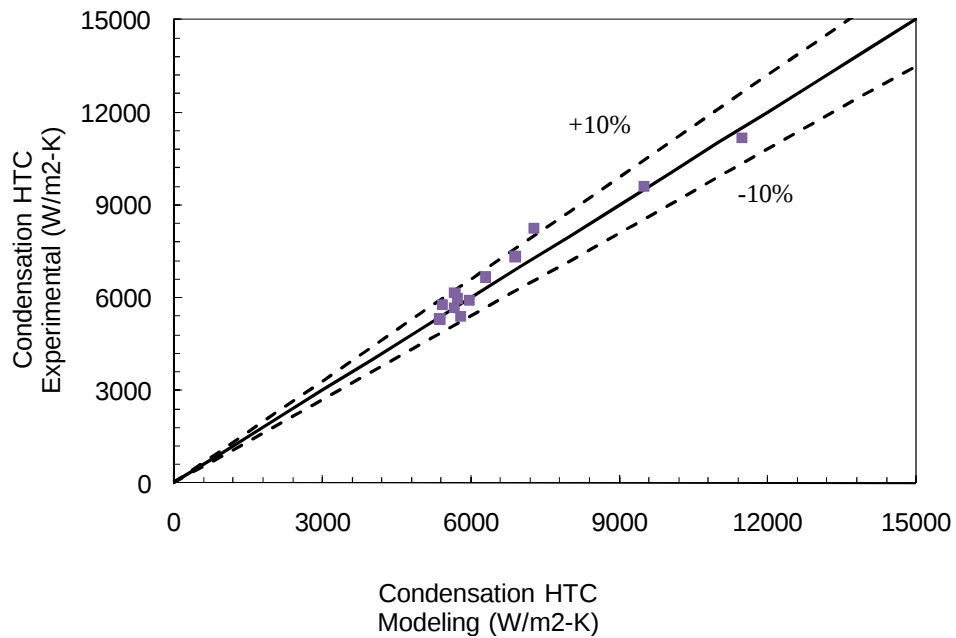


Figure 7 Comparison of Condensation HTCs for Tube Bundle Model Results with Experimental Results for Through Flow

Experimentally it is almost impossible to measure the condensate mass flow rate at different axial points, and the axial profile for condensation HTC could not be directly computed. To estimate the local condensation HTC, the boundary layer model [19] with tube bundle pool boiling was used to predict the axial condensate mass flow rates.

Figures 8 - 11 show the axial profiles of the condensate film thickness and HTC from the entrance under the conditions of system pressure 260 kPa and inlet steam mass flow rate 0.8 grams/sec for four tube bundle case. Figure 8 shows that the condensate film thickness increases with the axial distance from the tube entrance, and decreases with the increasing NC gas mass fraction at a fixed inlet steam flow rate and a fixed system pressure. The thickness increases rapidly initially, and then slowly as the axial distance from the entrance. Figure 9 shows that the average heat flux drops rapidly at first, and then decreases slowly as the axial distance from the entrance increases. The average heat flux decreases with increase in NC gas mass fraction. Figure 10 shows that the condensation mass flow rate has the same trends with the distance from the entrance and NC gas mass fraction as the average heat flux shown in Fig. 9. Figure 11 shows that the local condensation HTC drops rapidly at first, and then decreases slowly as the axial distance from the entrance increases. The condensation HTC decreased with increase in NC gas mass fraction and impact of NC gas mass fraction is significant at the entrance region where condensation is efficient.

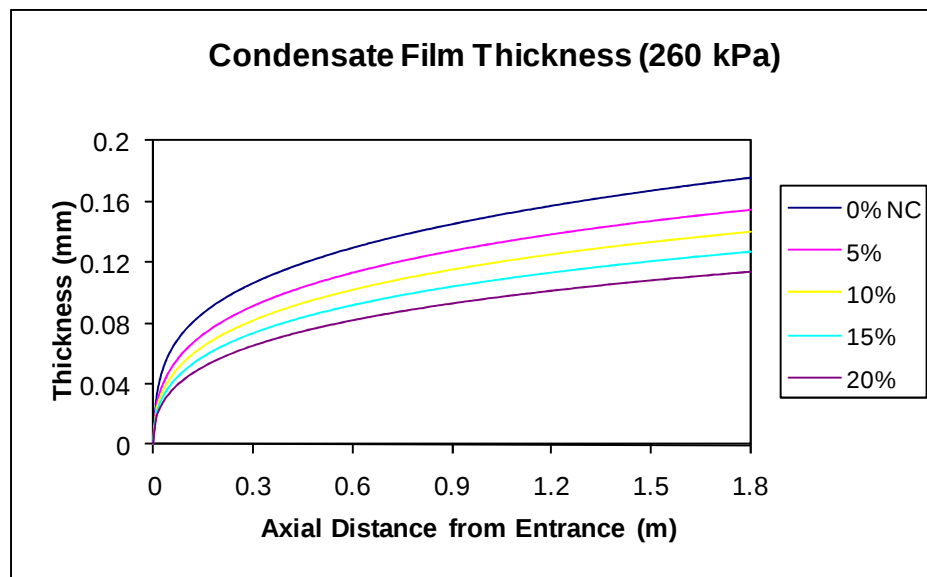


Figure 8 Condensate Film Thickness vs. Axial Distance from Entrance for Different NC Mass Fraction with Tube Bundle Model

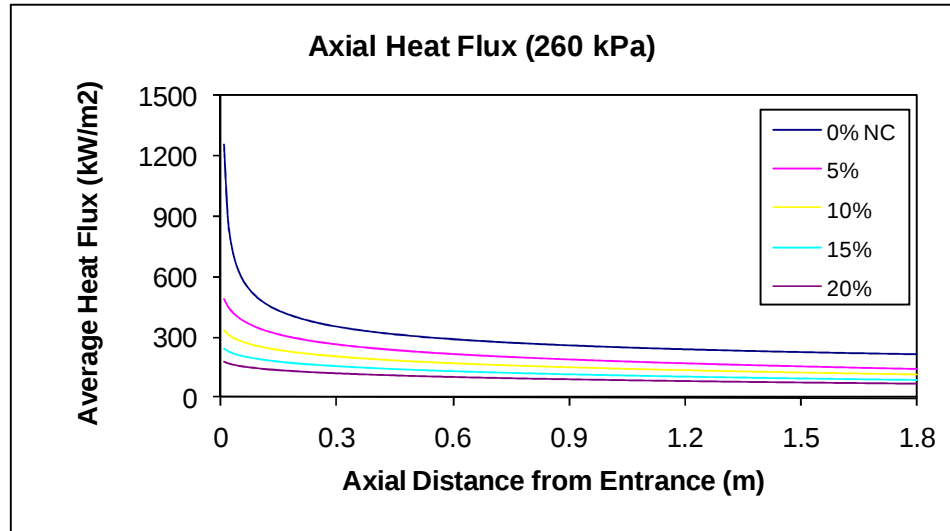


Figure 9 Average Heat Flux vs. Axial Distance from Entrance for Different NC Mass Fraction with Tube Bundle Model

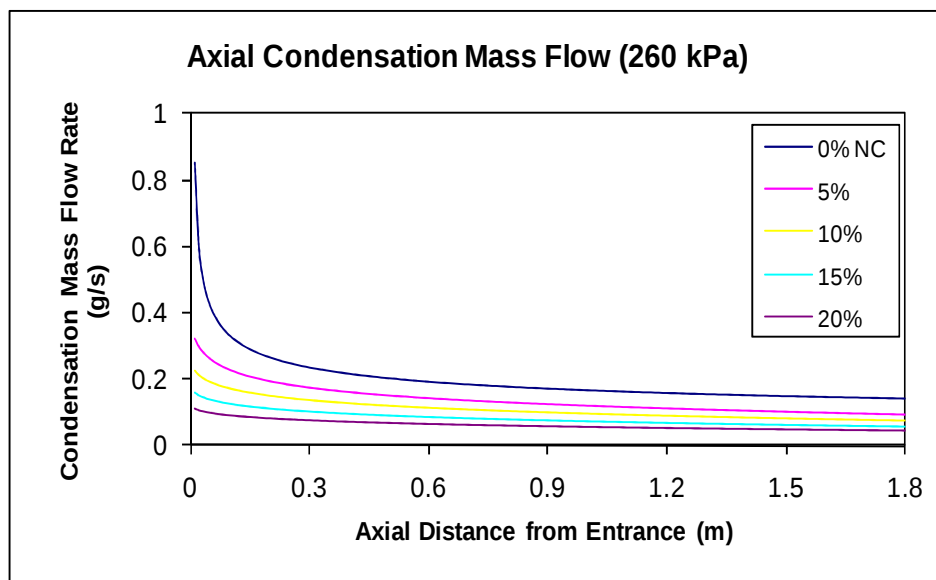


Figure 10 Condensation Mass Flow Rate vs. Axial Distance from Entrance for Different NC Mass Fraction with Tube Bundle Model

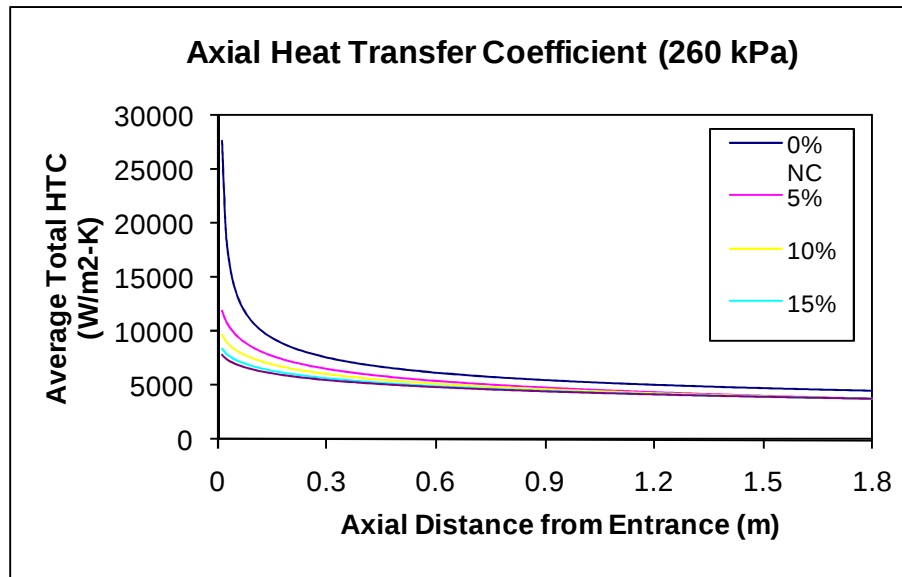


Figure 11 Average Heat Transfer Coefficient vs. Axial Distance from Entrance for Different NC Mass Fraction with Tube Bundle Model

4. Conclusion

The comparison of tube bundle experimental data with the single tube data by both the experiments and models shows that based on a linear fit for the trends, the single tube secondary HTC is between 25% - 35% less than what was recorded for the tube bundle, and the tube bundle condensation rates are slightly higher than the data from the single tube test sections due to turbulent mixing effect.

Tube bundle boundary layer model was then developed for the prediction of the filmwise steam condensation with noncondensable gas in a vertical tube bundle. The predictions from the model are compared with the experimental data. Secondary pool boiling heat transfer coefficients, condensation heat fluxes and heat transfer coefficients for various complete condensation and through flow tube bundle experimental data are compared and the agreement is satisfactory. The tube bundle model predictions of the condensation HTCs and condensate rates are within a range of ± 10 -15%, while the single tube model predictions are within a range of ± 25 %. So, the tube bundle model has a better prediction than the single tube model for the tube bundle experimental data. The local parameters predicted by the boundary layer model and heat and mass analogy model with tube bundle pool boiling can also be plotted with the axial distance from entrance for different noncondensable gas fractions and system pressures.

References

- [1] GE Nuclear Energy, 2006. ESBWR Design Control Document, Tier 2, Chapter 6 Engineered Safety Features,, Doc No. 26A6642AT.
- [2] Oh S., Gao H., Revankar S. T., 2007. Investigation Of A Passive Condenser System Of An Advanced Boiling Water Reactor, Nuclear Technology 158 (2): 208-218.

- [3] Vierow K. M., 1990. Behavior of steam-air systems condensing in concurrent vertical downflow, MS thesis, University of California at Berkeley, Berkeley, CA.
- [4] Kuhn S. Z., 1995. Investigation of heat transfer from condensing steam-gas mixtures and turbulent films flowing downward inside a vertical tube, PhD thesis, University of California at Berkeley, Berkeley, CA.
- [5] Siddique M. 1992. The effects of noncondensable gases on steam condensation under forced convection conditions, PhD thesis, Massachusetts Institute of Technology, Cambridge, MA.
- [6] Park H. S., No H. C., 1999. A condensation experiment in the presence of noncondensables in a vertical tube of a passive containment cooling system and its assessment with RELAP5/MOD3.2, Nuclear Technology 127 (2):160-169.
- [7] Kim S. J., No H. C., 2000. Turbulent film condensation of high pressure steam in a vertical tube, International Journal of Heat and Mass Transfer 43 (21):4031-4042.
- [8] Al-Shammari S. B., Webb D. R., Heggs P., 2004. Condensation of steam with and without the presence of non-condensable gases in a vertical tube, Desalination 169 (2):151-160.
- [9] Oh S., Revankar S. T., 2005. Effect of noncondensable gas in a vertical tube condenser". Nuclear Engineering and Design 235 (16):1699-1712.
- [10] Oh, S., Revankar, S. T., 2006. Experimental and theoretical investigation of film condensation with noncondensable gas, International Journal of Heat and Mass Transfer, 49:2523-2534.
- [11] Lee, K.Y., Kim, M.H., 2008. Experimental and empirical study of steam condensation heat transfer with a noncondensable gas in a small-diameter vertical tube, Nuclear Engineering and Design 238 (12):207-216.
- [12] Revankar, S.T., Zhou, W. and Henderson, G. 2008, Experimental and thermalhydraulic code assessment of the transient behavior of the passive condenser system in an advanced boiling water reactor. Technical Report, DE- FG07-04ID14605, DOE.
- [13] Henderson, G., Zhou, W., Revankar, S.T., 2010. Condensation in a vertical tube bundle passive condenser- Part 2: Complete condensation, International Journal of Heat and Mass Transfer, 53: 1156–1163.
- [14] Fujita, Y. and Hidaka, S. 1998. Effect of tube bundles on nucleate boiling and critical heat flux. Heat Transfer - Asian Research, 27:312-325.
- [15] Chen, J.C., 1963. A correlation for boiling heat transfer to saturated fluids in convective flow. ASME paper 63-HT-34.
- [16] Chen, J.C. 1966. A correlation for boiling heat transfer in convective flow. ISEC Process Design and Development, 5: 322-329.
- [17] Collier, J.G. 1981. Convective boiling and condensation. 2nd Ed. McGraw-Hill, New York.
- [18] Foster, K., Zuber, N. 1955. Dynamics of vapor bubbles and boiling heat transfer. AIChE Journal, 1: 531-535.
- [19] Zhou, W., 2010. Experimental and analytical study of the effects of noncondensable gas in a tube bundle passive condenser, Ph. D. Thesis, School of Nuclear Engineering, Purdue University.
- [20] Zhou, W., Henderson, G., Revankar, S.T., 2010. Condensation in a vertical tube bundle passive condenser - Part 1: Through flow condensation, International Journal of Heat and Mass Transfer, 53:1146–1155.