Modification of the Horizontal Condensation Heat Transfer Model based on Slip Ratio Model

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1. Introduction

Over the past few decades, there have been many experimental and analytical researches to predict the horizontal condensation heat transfer and numerous models have been proposed. However, the proposed models which were based on the use of limited database, showed a considerable deviation against the experimental data, depending on different authors, different fluids and different flow conditions. The APR+ (Advanced Power Reactor) adopts several new safety features and the PAFS (Passive Auxiliary Feedwater System) is one of the advanced safety features which can cool down the nuclear reactor without any external power supply in case of accidents. For validation of the cooling and operational performance of the PAFS, PASCAL (PAFS Condensing Heat Removal Assessment Loop) facility was constructed and experimental investigation of the condensation heat transfer and natural convection phenomena in the PAFS was experimentally investigated at KAERI (Korea Atomic Energy Research Institute). From the experimental result, it was found that a thermal hydraulic system analysis code, MARS-KS (Multi-dimensional Analysis of Reactor Safety), underestimated the condensation heat transfer coefficient compared to the experimental data. Referring to the previous condensation heat transfer models in the literatures, a new condensation heat transfer model for the nearly horizontal tube, especially for the PCHX (Passive Condensate Heat Exchanger) in the PAFS, was proposed [1]. In this study, the modified correlation using the analytic slip ratio model was evaluated with the experimental data.

2. Condensation Heat Transfer

During condensation inside horizontal tubes, the two-phase flow pattern may be dominated by vapor or gravity forces. While annular flow pattern is associated with high vapor shear, stratified flows appear when gravity is the dominant force. In the fully developed annular flow pattern, there is a thin uniform condensate film on the entire tube wall, while the gas phase flows in the central core, and heat transfer is governed by vapor shear and turbulence. In the stratified flow regime, a certain thickness of the condensate layer forms at the bottom of the tube and a thin liquid film settles on the wall in the upper portion of the tube. In this condition, the heat transfer through the thin film is generally analyzed by the classical Nusselt theory.

Shah [2] proposed a simple dimensionless correlation for predicting heat transfer coefficients during film condensation inside plane tubes. It was verified by comparison with a wide variety of experimental data. It was shown to agree with data for water, refrigerants, and organics covering a wide range of conditions in horizontal, vertical, and inclined tubes. However, in a later paper, the author stated that his correlation will fail at very low flow rates and high reduced pressure condition and the deviations were found to be related to the viscosity ratio of phases and reduced pressure. He improved and extended his correlation to a wider range of parameters [3] suggesting a correction factor which was developed through data analysis. This result was found to give good agreement with data at higher flow rates for both horizontal and vertical tubes. Thome et al. [4] developed a flow pattern based on the condensation heat transfer model for the mist, annular, stratified wavy and fully stratified flow regimes. He proposed a new heat transfer model which considers both film and convective condensation effects in the stratified wavy and fully stratified flow regimes. The correlation includes the effect of liquid-vapor interfacial roughness on heat transfer. They tried to obtain a method with a minimum of empirical constants and exponents that not only gives a good statistical representation of the data but also correctly captures the trends in data. Their model predicts the heat transfer coefficient within 20% error bound. Cavallini et al. [5] also developed a condensation model for the annular flow and stratified flow regimes and validated it with over 2000 data points. Their stratified flow condensation model also includes both film and convective condensation heat transfer.

3. Modification of Shah Correlation

There are two kinds of method in modeling the condensation heat transfer phenomenon inside a nearly horizontal heat exchanger tube. One method is that the condensation phenomenon from the inlet to the outlet of the tube is modeled with one correlation. In the other method, the correlation is composed of two parts, convection and condensation heat transfer, separately. Shah's correlation [3] is also divided into two parts, convection and condensation heat transfer. The Shah's correlation uses the following two heat transfer equations;

$$h_I = h_{LT}\left(\frac{\mu_f}{14\mu_g}\right)^n \left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{P_r^{0.38}}\right] \quad (1)$$

where:

$$n=0.0058+0.557p_r \quad (2)$$

The second equation is

$$h_{NU} = 1.32 Re_{LS}^{-1/3}\left[\frac{\rho_l(\rho_l-\rho_g)gk_f^3}{\mu_f^2}\right]^{1/3} \quad (3)$$

Equation (3) is the Nusselt equation for laminar film condensation in vertical tubes. For horizontal tubes, following equation is recommended only if Re_{GT}>35000.

$$h_{TP} = h_I + h_{NU} \quad (4)$$

In the present study, the ratio of dynamic viscosity between each phase is replaced with a slip ratio in Equation (1). Precise modeling of heat transfer at the interface between each phase will contribute to reduce the difference between the correlation and experimental data. The modified correlation is used the slip ratio between each phase estimated from the thickness of the condensate water in place of the ratio representing the phase properties. To develop the slip ratio correlation using the condensate water level or the void fraction, it is assumed that a stratified steam-water mixture is flowing between two horizontal flat plates. Both phases is also assumed in fully steady state condition. The two-phase flow is shown in Figure 1, where δ represents the thickness of the liquid layer.

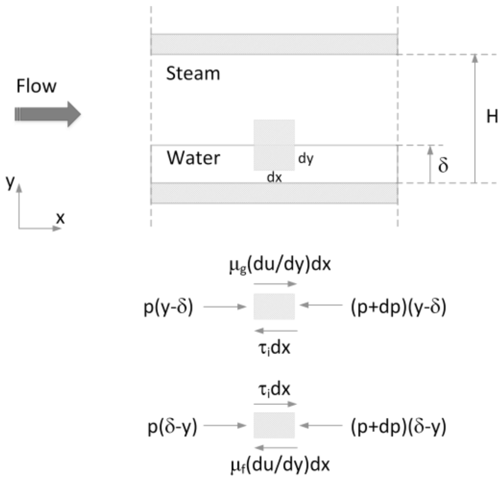

Fig. 1. Physical model for steam condensation in horizontal plate

A mass balance on the control volume shown in Figure 1 yields the following expressions for each phase:

$$v(x,y) - v(x,\delta^-) = \int_y^{\delta} \frac{\partial u}{\partial x} dy \quad \text{for water} \quad (5)$$

$$v(x,y) - v(x,\delta^+) = -\int_{\delta}^{y} \frac{\partial u}{\partial x} dy \quad \text{for steam} \quad (6)$$

For the fully develop flow, the terms on the right hand sides of the Equation (5) and (6) are set equal to zero. Neglecting any mass transfer provides the boundary condition. Consequently, the vertical component of the velocity field is equal to zero in the domain. Momentum transport in the flow direction yields the following equations for each phase, respectively:

$$0 = -\frac{dp}{dx}(\delta - y)dx + \tau_i dx - \mu_f \frac{du}{dy}dx \quad \text{for water} \quad (7)$$

$$0 = -\frac{dp}{dx}(y - \delta)dx - \tau_i dx + \mu_g \frac{du}{dy}dx \quad \text{for steam} \quad (8)$$

The interfacial shear τ_i is related to the velocity gradients in both phases by

$$\tau_i = \mu_f \left.\frac{du}{dy}\right|_{\delta^-} = \mu_g \left.\frac{du}{dy}\right|_{\delta^+} \quad (9)$$

From Equation (7) and (8), the velocity profiles for the two phases are given by integrating the two ordinary differential equations by separating of variables and applying the no-slip condition at the wall:

$$u(y) = \frac{1}{2\mu_f}\frac{dp}{dx}[(\delta-y)^2 - \delta^2] + \frac{\tau_i}{\mu_f}y \quad \text{for } 0 \le y \le \delta \quad (10)$$

$$u(y) = \frac{1}{2\mu_g}\frac{dp}{dx}[(y-\delta)^2 - (H-\delta)^2] + \frac{\tau_i}{\mu_g}(y-H)$$

for $\delta \le y \le H$ (11)

The interface the two values of u must be equal and the equation is rearranged with $\delta = H(1-\alpha)$:

$$\frac{dp}{dx} = \left[\frac{2}{H}\frac{\mu_g(1-\alpha)+\mu_f\alpha}{\mu_g(1-\alpha)^2 - \mu_f\alpha^2}\right]\tau_i \quad (12)$$

By integrating Equation (10) and (11), the average velocity of each phase is given as follows:

$$U_f = \frac{\delta}{2\mu_f}\left(\tau_i - \frac{2}{3}\frac{dp}{dx}\delta\right) \quad (13)$$

$$U_g = \frac{H-\delta}{2\mu_g}\left(-\tau_i - \frac{2}{3}\frac{dp}{dx}(H-\delta)\right) \quad (14)$$

From the definition of the slip ratio, S and the derived expression for the pressure gradient:

$$S = \left(\frac{\mu_f}{\mu_g}\right)^2 \frac{\alpha}{1-\alpha}\left[\frac{\frac{1}{3}\alpha^2 + \frac{4}{3}\frac{\mu_g}{\mu_f}\alpha(1-\alpha) + \frac{\mu_g}{\mu_f}(1-\alpha)^2}{\frac{\mu_f}{\mu_g}\alpha^2 + \frac{4}{3}\frac{\mu_f}{\mu_g}\alpha(1-\alpha) + \frac{1}{3}(1-\alpha)^2}\right] \quad (15)$$

Using the derived slip ratio, Equation (1) is modified as follows:

$$h_I = h_{LT}(aS)^b\left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{P_r^{0.38}}\right] \quad (16)$$

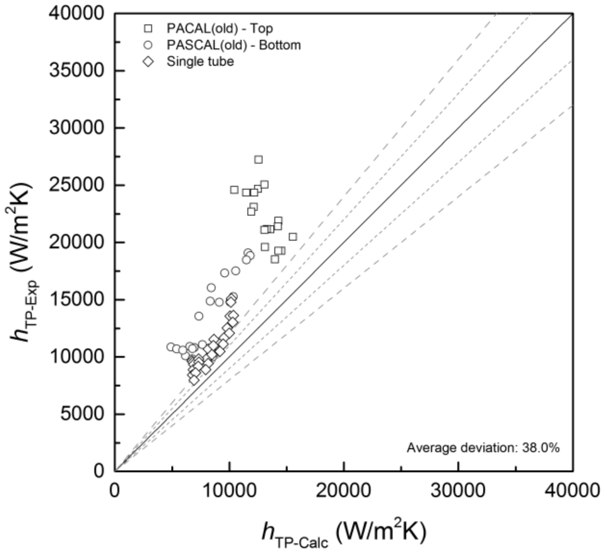

Fig. 2. Comparison for the total condensation heat transfer coefficient using Shah (2009) correlation

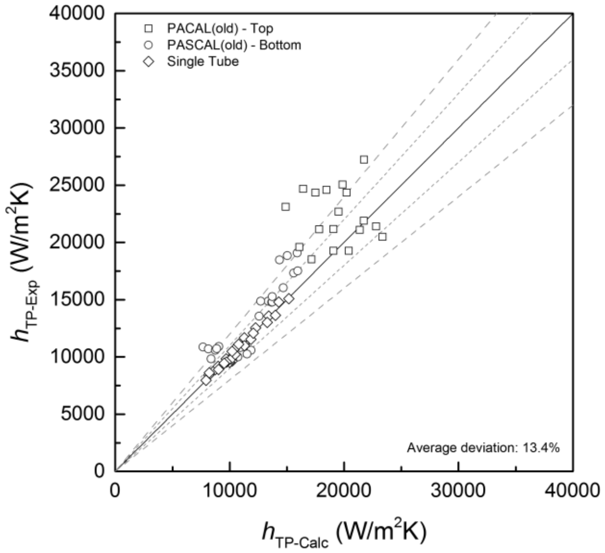

Fig. 3. Comparison for the total condensation heat transfer coefficient using the derived slip ratio model

4. Comparison of Results

The condensation heat transfer correlation modified as Equation (16) was compared with the experimental data acquired from the PASCAL experiment and the straight-tube experiment. The total number of data used for the evaluation was 69 except inlet, outlet and bending points of the PASCAL experiment data. The experimental data in the following range of conditions: reduced pressures, p_r, from 0.038 to 0.328, mass velocities from 46 to 273 (kg/m^{2}s), and vapor qualities from 0 to 1, and tube internal diameter is 0.0448 m. The vapor quality was assumed to vary linearly through the whole test section for PASCAL experiment data. In the straight-tube experimental data, the quality was derived from the heat balance calculation between the condensation heat exchanger tube and the water jacket. The total condensation heat transfer coefficient was compared with the experimental data and the result is shown in Figure 2. The heat transfer coefficients showed dispersed trend and underestimated compared to the experimental data. The percentage error from the comparison with the experimental data is about 38.0%.

Figure 3 showed the modified total condensation heat transfer coefficient compared with the experimental data. From equation (16), the constant *a* and *b* are selected to minimize the error from the comparison with the straight-tube experiment except PASCAL data. Then, the modified condensation heat transfer correlation using the developed slip ratio is as follows:

$$h_I = h_{LT} 0.4 S^{-0.05} \left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{P_r^{0.38}} \right] \quad (17)$$

The percentage error from the comparison with the experimental data is about 13.4%. Throughout this result, the derived slip ratio model for developing the heat transfer correlation in the condensation heat exchanger tube would appear quite reasonable method.

5. Conclusions

The condensation heat transfer correlation using the slip ratio model for the horizontal heat exchanger tube was developed to model the heat transfer phenomenon inside the PCHX tube. The proposed slip ratio model showed improved prediction performance around 14% of the error. To increase the prediction capability of the condensation heat transfer correlations developed in this study, more experimental data in the wide ranges should be compared as the further study.

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REFERENCES

[1] Park, Y.S., Report on the Model Package Development of the Condensation Heat Transfer for PAFS Best-estimate Analysis, APR+-PAFS-TR-14-01, 2014..

[2] Shah, M.M., A general correlation for heat transfer during film condensation inside pipes, Int. J. Heat Mass Transfer 22, 547–556, 1979.

[3] Shah, M.M., An Improved and Extended General Correlationfor Heat Transfer During Condensationin Plain Tubes, Int. J. HVAC&R Res. 15 (5), 889-913, 2009.

[4] Thome, J.R., El Hajal, J., Cavallini, A., Condensation in horizontal tubes. Part 2: New heat transfer model based on flow regimes, Int. J. Heat Mass Transfer 46, 3365–3387, 2003

[5] Cavallini, A., Censi, G., Del Col, D., Doretti, L., Longo, G.A., Rossetto, L., Condensation of halogenated refrigerants inside smooth tubes, Int. J. HVAC&R Res. 8 (4), 429–451, 2002.