Development of a Prototypical Condensation Model for the Nearly Horizontal Heat Exchanger Tube of the APR+ PAFS(Passive Auxiliary Feed-water System)

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1. Introduction

Passive Auxiliary Feed-water System(PAFS) which is adopted in the Korean advanced nuclear power plant, APR+, removes decay heat by condensing steam from the secondary side in the nearly horizontal tubes under the accident condition. The comparison of prediction against PASCAL(2011) experimental data obtained by KAERI indicated that the best estimated safety analysis code such as MARS tends to underestimate the condensation heat transfer coefficient [1]. In this study, a new condensation heat transfer model was developed for the separated flow regime in the nearly horizontal tube.

2. Modeling

2.1 Prediction of void fraction

Prediction of void fraction was attempted by 1-D Separated flow model(SFM) assuming a fully developed stratified flow with steady state assumption. A momentum balance is schematically described in Fig. 1(a), and the final momentum equation is as follows.

$$\tau_{wg}\frac{S_g}{A_g} - \tau_{wl}\frac{S_l}{A_l} + \tau_i S_i\left(\frac{1}{A_l}+\frac{1}{A_g}\right) - \left(\rho_l - \rho_g\right) g \sin\theta = 0 \quad (1)$$

where $\tau_{wg,l}$ for wall shear stress of each phase, $A_{g,l}$ $S_{g,l}$ for the cross sectional area and the contact perimeter of each phase, respectively. τ_i and S_i for the shear stress and perimeter of the phase interface, respectively. Inclination angle θ is defined as in the Fig. 1(a),

To determine the void fraction(α) by the equation (1) in the nearly horizontal tube, three closure relations for the sheer stresses and an additional closure relation for flow cross-sectional geometry are required. The Blasius' power law form was used for wall sheer stresses and the assumption, $f_i = f_g$, was introduced for the interfacial sheer stress. The flow cross section of stratified flow is described in fig. 1(b). Assuming concaved interface shape with constant curvature, the geometrical relations are expressed as follows,

$$\gamma_2 = -4\tan^{-}\left(\frac{\frac{D}{2}\cos\frac{\gamma_1}{2} - \frac{D}{2} + \delta}{\frac{D}{2}\sin\frac{\gamma_1}{2}}\right) \quad (2)$$

$$R_2 = \left(\frac{D}{2}\sin\frac{\gamma_1}{2}\right) \Big/ \sin\frac{\gamma_2}{2} \quad (3)$$

Parameters appeared in the equations (2) and (3) are shown in Fig. 1(b). The following correlation suggested by Hart [2] was used for the wetted perimeter γ_1 .

$$\gamma_1 = 2\pi\left(0.52\left(1-\alpha\right)^{0.374} + 0.26 Fr^{0.58}\right) \quad (4)$$

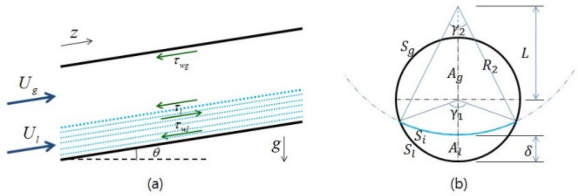

Fig. 1. Schematic description of the stratified flow configuration and coordinates: (a) flow direction, (b) cross section.

2.2 heat transfer coefficient correlation

The filmwise condensation takes place along the upper wetted perimeter of the tube. In the present work, vapor Reynolds number was added to the Nusselt's integral analysis to consider turbulent sheer stress effect on the interface. The effect of vapor Reynolds number to Nusselt number is shown in Fig. 2, comparing with experimental date including PASCAL_SS [1], ATLAS_PAFS [3]. Finally, a new average film condensation heat transfer coefficient was obtained by means of regression analysis with experimental data as follows,

$$h_{top} = 0.729\left(1 + 8.9\times10^{-4}\,\mathrm{Re}_g^{\;0.57}\right)\left(\frac{Gr_l\,\mathrm{Pr}_l}{Ja}\right)^{\frac{1}{4}}\left(\frac{k_l}{D_{hg}}\right) \quad (5)$$

The heat transfer in the tube bottom region is modeled by the single liquid convection. The following convective heat transfer coefficient from the Dittus-Boelter type correlation[4] was introduced for turbulent flow.

$$h_{bottom} = 0.023\,\mathrm{Re}_l^{\;0.8}\,\mathrm{Pr}_l^{\;0.3}\left(\frac{k_l}{D_{hl}}\right)\left(\frac{T_l - T_w}{T_{sat} - T_w}\right) \quad (6)$$

Here, subcooled temperature T_l of condensate is used to take into account of subcooling effect.

The hydraulic equilibrium diameter in the equations (5), (6) is based on cross sectional geometry and Reynolds numbers are defined by actual velocity of each phase. Overall condensation heat transfer coefficient in the entire tube is expressed with the equations (5) and (6) as follows.

$$h = \frac{h_{top}\left(2\pi - \gamma_1\right) + h_{bottom}\gamma_1}{2\pi} \quad (7)$$

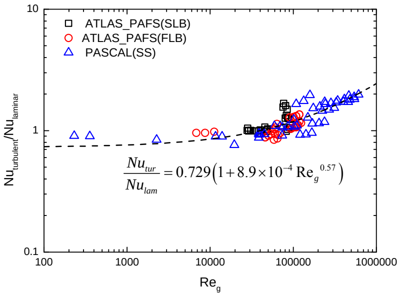

Fig. 2 Turbulent effect for the filmwise condensation phenomena

3. Evaluation of new model by comparison with experimental data

Available experimental data was summarized in the Tables 1 and 2 for the evaluation of developed heat transfer models. The evaluation results of a new model are shown in Figs. 3 and 4. As in the Figures, the new model can predict the data with a mean deviation about 10% and 30% for void fraction and heat transfer coefficient, respectively.

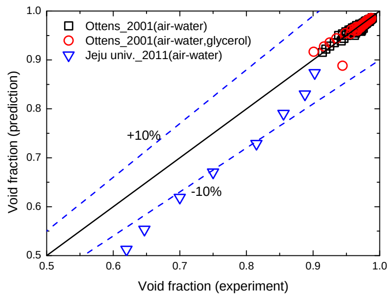

Fig. 3. Comparison of void fraction predicted with available experimental data.

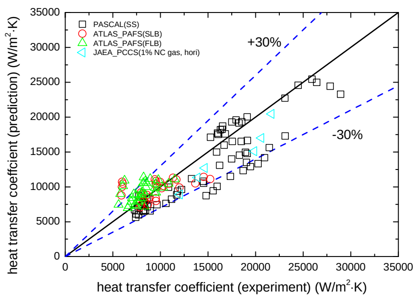

Fig. 4. Comparison of overall heat transfer coefficient predicted with available experimental data.

Table I : Void fraction data sources

Test	Fluid	θ (°)	I.D. (m)	No. of data points
Ottens (2001) [5]	Air-Water(Glycerol)	-2 ~ 0	0.052	107
Jeju univ. (2011) [6]	Air-Water	-3	0.045	10

Table Ⅱ : heat transfer coefficient data sources

Test	Fluid	θ (°)	I.D. (mm)	P_{sat} (bars)	G (kg/m^2s)	N. C. gas	No. of data points
PASCAL (SS) [1]	Water	-3	44.8	8.5~67	72~329	-	66
ATLAS_PAFS [3]	Water	-3	30.8	23~63	69~118	-	65
JAEA_PCCS [7]	Water	0	29	7	73	1%	6

4. Conclusion

A new heat transfer model which takes into account of different heat transfer mechanisms occurred in the upper and lower regions of perimeters was developed in the nearly horizontal tube. The present model shows good prediction capability against experimental data.

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