A condensation model for the PAFS heat exchanger tube under the pure steam conditions

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1. Introduction

APR+ PAFS adopts a horizontal-type condensation heat exchanger [1]. The MARS code under-predicts condensation heat transfer coefficients (HTCs) in a simulation of the PASCAL experiments conducted for the evaluation of PAFS heat exchanger [2]. From the results, it is necessary to improve the condensation model of the horizontal tube in MARS code. In the horizontal tube, the HTC is asymmetric because of the stratified flow formed by the condensate in the lower region of the tube. For the mechanistic modeling, therefore, the heat transfer needs to be considered separately for the upper and lower region.

In this study, experiments were carried out to investigate the characteristics of the local condensation heat transfer in the horizontal tube. Based on the experimental data, the prediction model for the condensation HTCs on the upper and lower region are improved.

2. Experiment

2.1 Experimental setup

Fig. 1 shows the schematic of the experimental setup. The in-tube condensation test section was a 40 mm inner diameter circular tube with a length of 3 m, and the downward inclination angle of -3°, corresponding to the PAFS heat exchanger tube orientation. Fig. 2 shows the local measurement points for the steam temperature and wall temperature on the condensation tube by rotating on its axis. The experiments were performed under 16 conditions of pure steam, corresponding to an inlet pressure of 1-5 bar and a steam mass flux of 10-50 kg/m^2s.

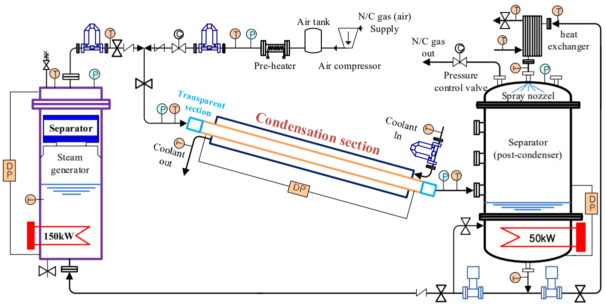

Fig. 1. Schematic diagram of inclined condensation test facility (PICON)

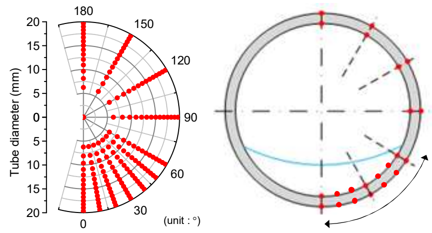

Fig. 2. Local measurement points for the steam temperature and wall temperature

2.2 Results of steam temperature distribution

Fig. 3 shows a typical distribution of the steam temperature. It can be seen that the stratified flow is formed by the subcooled condensate inside the condensation tube.

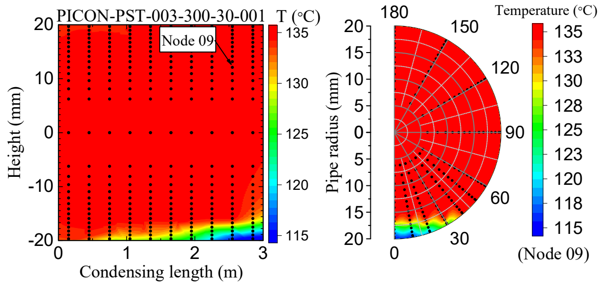

Fig. 3. Steam temperature distribution (3 bar, 30 kg/m^2s)

2.3 Results of circumferential distribution

Fig. 4 shows the local wall temperature and the heat flux measured along the tube circumference. The distributions of the heat flux and wall temperature were approximated as the following sigmoid function.

$$X = \frac{a-b}{1+(\theta/\theta_0)^p} + b \tag{1}$$

The heat transfer wall fraction angle (θ_{HTWF}), separating the upper and lower regions, is defined as the inflection point (where $d^2X/d\theta^2=0$) of heat flux distribution. From the fraction, the average HTC were defined separately for the upper and lower region of the tube as follows.

$$\bar{h}_{upper} = \frac{\overline{q''_{w\,upper}}}{T_{sat} - \overline{T_{w\,upper}}} \tag{2}$$

$$\bar{h}_{lower} = \frac{\overline{q''_{w\,lower}}}{T_{sat} - \overline{T_{w\,lower}}} \tag{3}$$

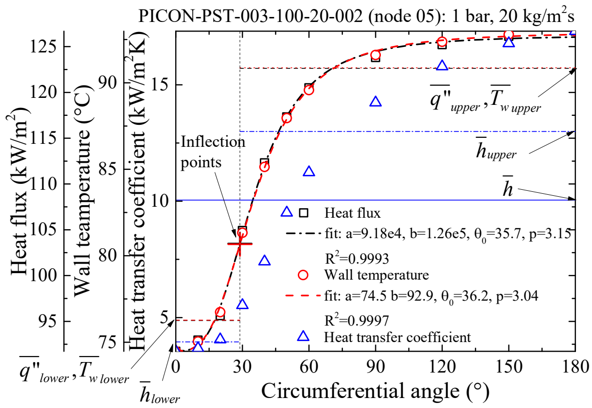

Fig. 4. Circumferential distribution of the local data

3. Modeling

In this study, the prediction correlation of HTCs and the wall fraction were improved based on the existing condensation model proposed by Ahn et al. [3].

3.1 Film condensation HTC on the upper region

The prediction model of the HTC of the upper region was developed based on the film condensation heat transfer. The average HTC of the laminar film on the upper region was defined as Eq. (4). In addition, to consider the heat transfer enhancement because of the axial shear stress by the steam flow, the influence of Reynolds number was considered in Eq. (5), as shown in Fig. 5.

$$\bar{h}_{f(\theta_{HTWF})} = \left[0.903-0.168\left(\frac{\pi-\theta_{HTWF}}{\pi}\right)^{2.22}\right]\left[\frac{g(\rho_L-\rho_G)\rho_L h_{fg} k_L^{3}}{D\mu_L\left(T_{sat}-\bar{T}_{w\,upper}\right)}\right]^{1/4} \quad (4)$$

For Re_G>4,000,

$$\bar{h}_{upper(\theta_{HTWF})} = \left(0.75+5.5\times10^{-4}\,Re_G^{\;0.59}\right)\bar{h}_{f(\theta_{HTWF})} \quad (5)$$

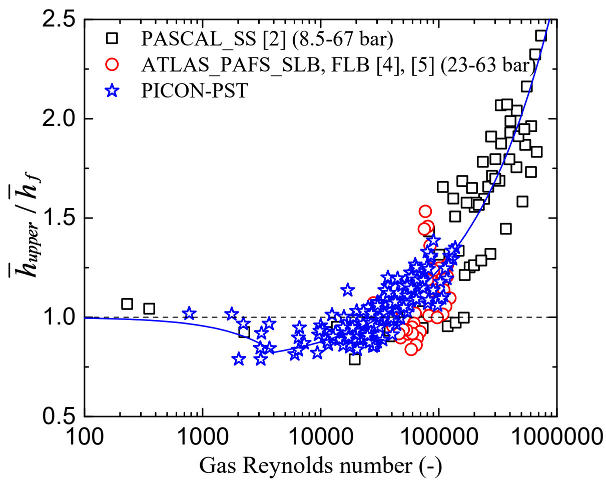

Fig. 5. Correlation of the film condensation HTC

3.2 Convection HTC on the lower region

The prediction correlation of the HTC on the lower region of the tube was based on the form of a two-phase flow multiplier model for convective heat transfer. The correlation was proposed as a function of the void fraction and wetted wall fraction as follow.

$$\bar{h}_{lower} = 0.023\,Re_{LS}^{\;0.8}\,Pr_L^{\;0.4}\left(\frac{k_L}{D}\right)\left[0.81\frac{(\theta_{WWF}/\pi)^{0.13}}{(1-\alpha)^{0.83}}\right] \quad (6)$$

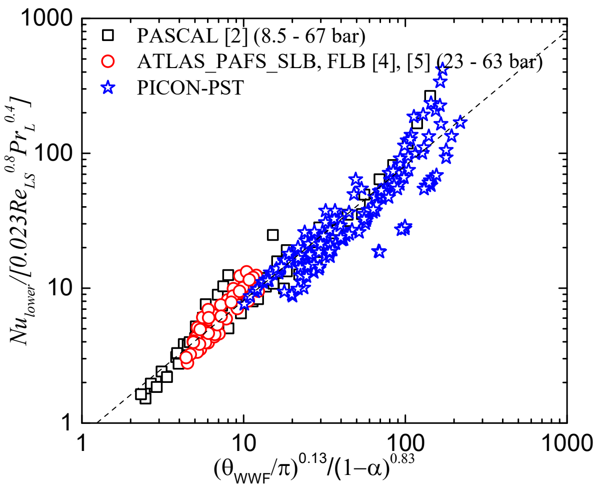

Fig. 6. Correlation of the convection HTC

3.3 Heat transfer wall fraction and average HTC

The heat transfer wall fraction (θ_{HTWF}/π) is calculated related to the wetted wall fraction (θ_{WWF}/π) predicted by Ahn et al. [6]. Fig. 7 shows the comparison between the predicted wetted wall fraction and the heat transfer wall fraction measured in the condensation experiment. From the result, heat transfer wall fraction angle was defined as follow.

$$\theta_{HTWF} = 0.6\theta_{WWF} \quad (7)$$

The wetted wall fraction is calculated simultaneously with the void fraction through the 1-D momentum balance equation [3]. In the calculation process, the modified interfacial friction factor was used [7].

From the heat transfer wall fraction, the average HTC for the whole circumference is calculated as follow.

$$\bar{h} = \frac{\bar{h}_{upper}\left(T_{sat}-\bar{T}_{w\,upper}\right)\left(\pi-\theta_{HTWF}\right)+\bar{h}_{lower}\left(T_{sat}-\bar{T}_{w\,lower}\right)\theta_{HTWF}}{\left(T_{sat}-\bar{T}_{w}\right)\pi} \quad (8)$$

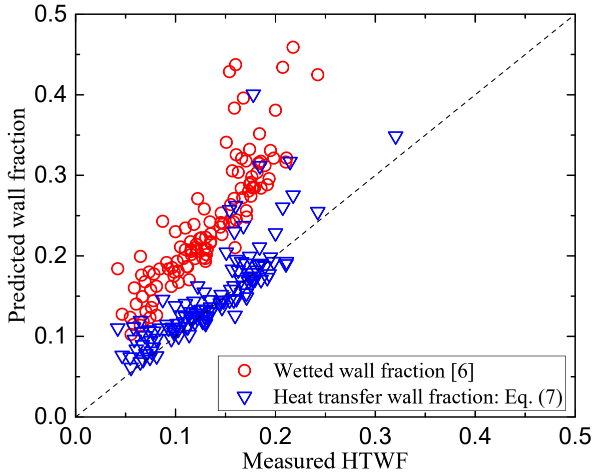

Fig. 7. Heat transfer wall fraction angle

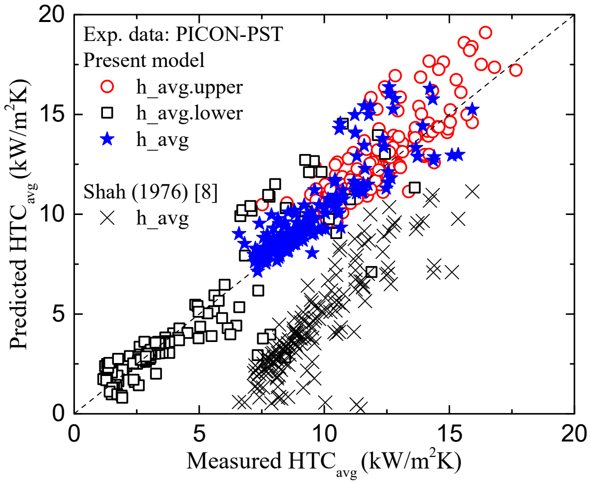

Fig. 9. Comparison result of the HTCs

4. Evaluation of the model

4.1 Effect of steam mass flux and pressure

Fig. 8 shows the average HTC measured at the different steam mass flux and pressure conditions in the steam quality of 0.8. As the flow rate increases and the pressure decreases, the HTC tends to increase. The present model shows good agreement with the experimental data.

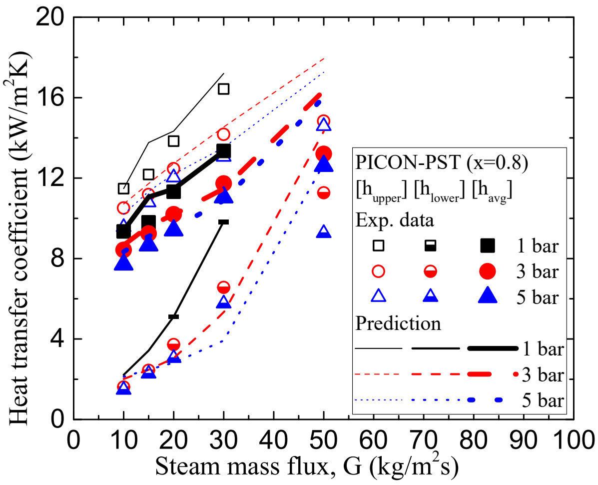

Fig. 8. Effects of steam mass flux and pressure on the HTC

4.2 Comparison of local HTCs

Fig. 9 shows the comparison of the predicted average HTC with the experimental data for the upper, lower, and overall region of the tube. The proposed model shows good agreement with the experimental data. In contrast, Shah's model [8], a default condensation model in MARS code, does not predict well the experimental data.

4.3 Benchmark evaluation with the MARS code

The PASCAL experiment was used for the benchmark analysis by applying the present condensation model into the MARS code to evaluate the new condensation model. Fig. 10 shows the MARS nodalization for the PASCAL experiment [9]. Fig. 11 shows that the original MARS code predicts the pressure of the steam generator about 50% higher than experimental data. On the other hand, when the present condensation model was applied into the code, the pressure prediction was similar to the experimental data. In addition, when the external wall temperature measured in the experiment was used as the boundary condition instead of the boiling heat transfer model, the predicted values were almost the same as the experimental data.

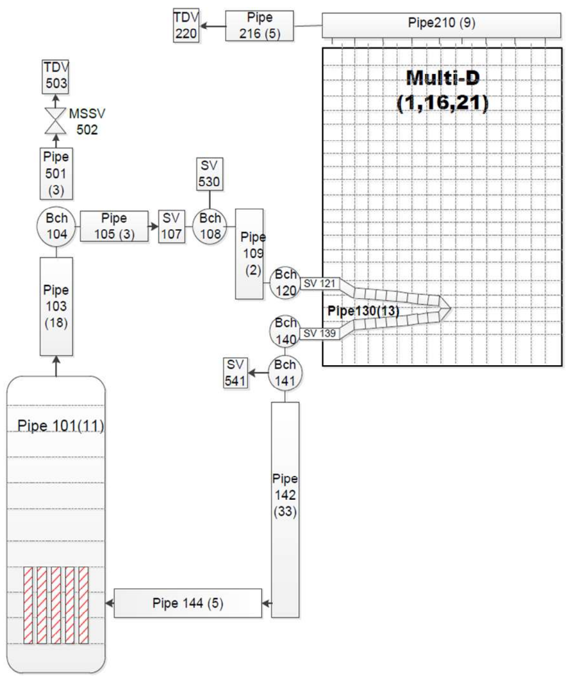

Fig. 10. Nodalization for the PASCAL test facility [9]

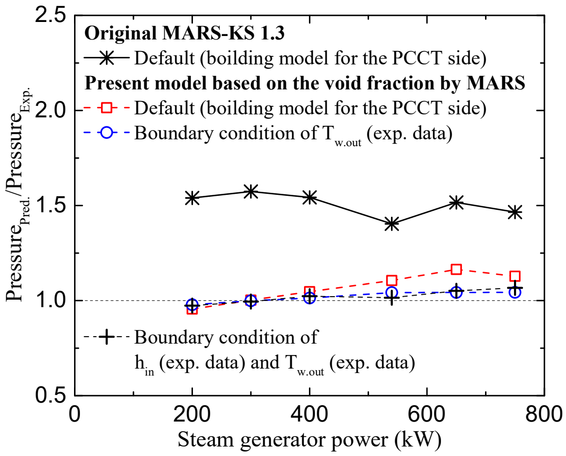

Fig. 11.Comparison of calculation result of the steady-state pressure

5. Conclusion

Condensation in a nearly horizontal tube was investigated to improve the condensation model for the better performance analysis of the heat exchanger of PAFS. The experiments were carried out under pure steam condition with a pressure of 1-5 bar a steam mass flux of 10-50 kg/m^{2}s using an inclined circular tube of a length of 3 m and an inner diameter of 40 mm. The local heat transfer parameters were measured in the circumferential direction and the axial direction. By analyzing the experimental data, a condensation model was developed and evaluated for the HTC on the upper and lower region of the condensation tube. The new condensation model was also evaluated by simulating PASCAL experiment using the MARS code.

Acknowledgement

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