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Condensation Analysis of Steam/Air Mixtures in Horizontal Tubes

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1. Introduction

Perhaps the most common flow configuration in which a convective condensation occurs is a flow in a horizontal circular tube. This configuration is encountered in air-conditioning and refrigeration condensers as well as condensers in Rankine power cycles. Although a convective condensation is also sometimes contrived to occur in a co-current vertical downward flow, a horizontal flow is often preferred because the flow can be repeatedly passed through the heat exchanger core in a serpentine fashion without trapping liquid or vapor in the return bends [1].

Many researchers have investigated a in-tube condensation for horizontal heat exchangers. However, almost all of them obtained tube section-averaged data without a noncondensable gas. Recently, Wu and Vierow [2,3] have experimentally studied the condensation of steam in a horizontal heat exchanger with air present. In order to measure the condenser tube inner surface temperatures and to calculate the local heat fluxes, they developed an innovative thermocouple design that allowed for nonintrusive measurements.

Here we developed a theoretical model using the heat and mass analogy to analyze a steam condensation with a noncondensable gas in horizontal tubes.

2. Modeling

The heat transfer through the vapor/noncondensable gas mixture boundary layer consists of the sensible heat transfer and the latent heat transfer given up by the condensing vapor, and it must equal that from the condensate film to the tube wall. Therefore, we get

$$h_f(T_i - T_w) = (h_c + h_s)(T_b - T_i) \qquad (1)$$

where h_f is the film heat transfer coefficient, h_c and h_s are the condensation and sensible heat transfer coefficients in the gas mixture respectively.

Then, the total heat transfer coefficient h_{tot} is given by

$$h_{tot} = \left[\frac{1}{h_f} + \frac{1}{h_c + h_s}\right]^{-1}. \qquad (2)$$

2.1 Condensate Film

In horizontal tubes at low vapor velocities, liquid that condenses on the upper portion of the inside tube wall tends to run down the wall toward the bottom, as indicated schematically in Fig. 1.

For the stratified annular flow condition, Chato [4] developed a detailed analytical model of the heat transfer for these circumstances as

$$h = 0.728Kc\left[\frac{g\rho_l(\rho_l - \rho_v)k_l^3 h_{lv}'}{\mu_l(T_{sat} - T_w)D}\right]^{1/4} \qquad (3)$$

where h'_{lv} is given by

$$h_{lv}' = h_{lv}\left[1 + 0.68\frac{C_{pl}(T_{sat} - T_w)}{h_{lv}}\right]^{1/4}. \qquad (4)$$

And then, Jaster and Kosky [5] suggested the following simple relations as a means of predicting the void fraction and the factor Kc in Eq. (3):

$$Kc = \alpha^{3/4} \qquad (5)$$

$$\alpha = \left[1 + \frac{1-x}{x}\left(\frac{\rho_v}{\rho_l}\right)\right]^{-1}. \qquad (6)$$

However, above model is for the tube section-averaged heat transfer coefficients. In real, the heat transfer across the thin film on the upper portion of the wall is very large compared to transport the liquid pool at the bottom of the tube (see Fig. 1). To consider the high heat transfer on upper portion, we suggest next model by modifying Eq. (3):

$$h = \frac{1}{0.728Kc}\left[\frac{g\rho_l(\rho_l - \rho_v)k_l^3 h_{lv}'}{\mu_l(T_{sat} - T_w)D}\right]^{1/4}. \qquad (7)$$

In other words, we use Eq. (3) for the bottom of the tube and Eq. (7) for the top of it.

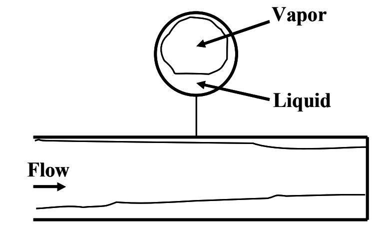

Fig. 1. Horizontal co-current annular flow with condensation

2.2 Vapor/Noncondensable Gas Mixture Flow

In this study, the heat and mass transfer analogy was used to analysis a steam condensation with air in horizontal tubes. Therefore, the sensible and latent heat transfer rates can be calculated simultaneously. The Nusselt and Sherwood numbers in the gas phase were modified to incorporate the effects of a condensate film roughness, suction, and a developing flow. Futher details are provided in [6]. Here, the averaged total heat flux in an axial node is calculated as

$$q_{tot}'' = (q_{top}'' + q_{bottom}'')/2\,. \qquad (8)$$

3. Results and discussion

We could obtain only two samples which contained all of the information to analyze them sufficiently from [2] and [3], as shown in Table I. The maximum uncertainties associated with the local heat fluxes and local heat transfer coefficients were reported to be 9.5 % and 10.0 %, respectively.

Figures 2 and 3 show that the theoretical model slightly underestimated the experimental data at the top of the tube. On the other hand, the model over-predicted the data at the bottom of it. The overestimation at bottom was dominant in the case of sample 2 because the flow was significantly stratified and the heat transfer through the liquid pool at the bottom was much smaller in the real situation. We need more data of condensation in horizontal tubes to improve the local models for film heat transfer around the tube. The comparisons of local heat fluxes are a little strange because there was a region that the result of top was lower than that of bottom. The reason is that the temperature difference between bulk and inner wall at the top was smaller than that at the bottom.

4. Conclusions

A theoretical model was developed to investigate a steam condensation with a noncondensable gas in horizontal tubes. The model predictions showed a good agreement with experimental data even though the film heat transfer model should be improved with affluent experimental data.

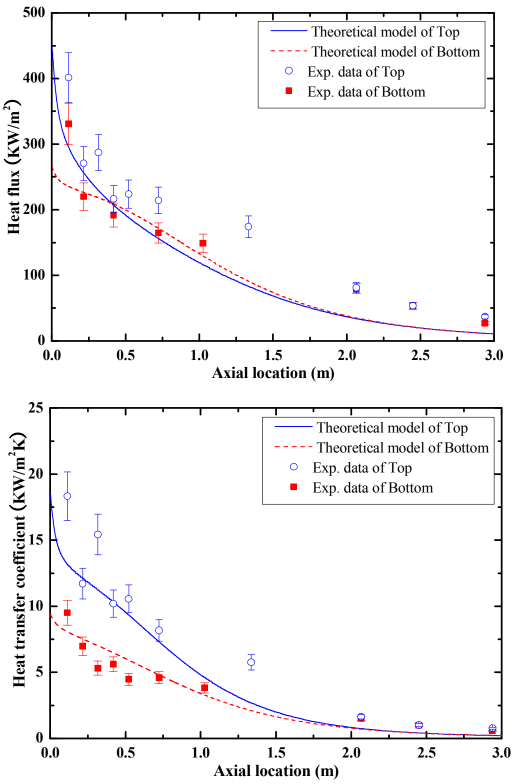

Fig. 2. Comparison of the local heat fluxes and local heat transfer coefficients for Sample 1.

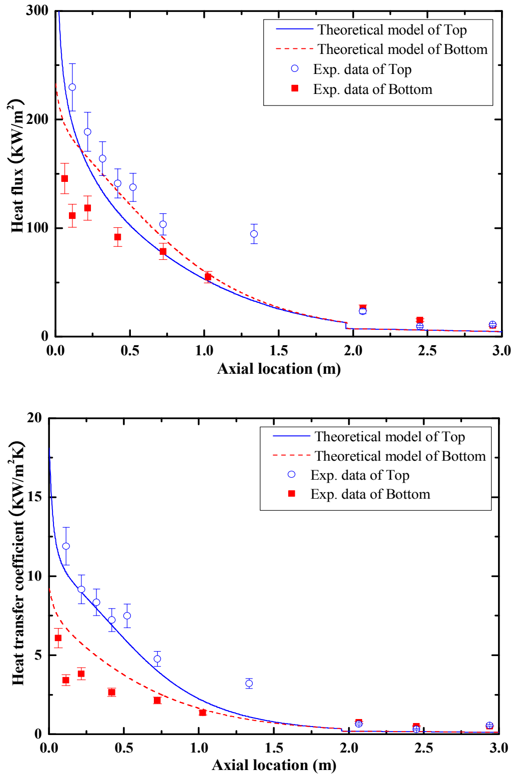

Fig. 3. Comparison of the local heat fluxes and local heat transfer coefficients for Sample 2.

Acknowledgements

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Table I: Samples of Experimental Data

	$\dot{m}_{steam}$ (g/s)	W_{air} (%)	Pressure(kPa)
Sample 1	11.5	5	200
Sample 2	6.0	5	100