

Evaluation of Steam-Air Mixture Condensation Heat Transfer Models on Tube-Bundle

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1. Introduction

The Passive Containment Cooling System (PCCS), developed in Republic of Korea, located in the containment building of APR+ or iPOWER reactor and located in the contain vessel of i-SMR. When the accident that reactor coolant ejected to the containment such as Loss of Coolant Accident (LOCA), the PCCS cool down the containment with condensation of the steam-Non Condensable Gas (NCG) mixture. Unlike the ESBWR PCCS, where the condensation heat transfer occurs inside the tubes, the PCCS heat exchangers (HXs) condense the steam outside the tubes under natural convection conditions.

To accurately predict this outer wall condensation heat transfer, numerous researchers have developed heat transfer models. In this study, the model that combines the previously developed condensation heat transfer models for steam-air mixture was evaluated with various experimental data.

2. Outer wall condensation model

Previous researchers have performed many condensation heat transfer experiments, and models have been presented in various forms. Usually, empirical correlations developed with experimental data show a simple formation; however, the applicability is limited to the range of experiments. On the other hand, the Heat and Mass Transfer Analogy (HMTA) method is a theoretical model, and it can be applied to a wide range of flow conditions. However, the iterative calculation is needed for this HMTA method.

2.1. Single tube condensation heat transfer model

Kang et al. [1] conducted a single tube condensation experiment and developed an empirical model with previous researchers' experimental data. The non-dimensional numbers that configure the correlation are W_a , Ja , Re^* , and A_r . These numbers are used for air mass fraction, wall subcooling, gas diffusion, and heat exchanger tube geometry effect, respectively.

$$h_{single} = 14.8 \left(\frac{1-W_{a,\infty}}{W_{a,\infty}} \right)^{0.37} Ja^{-0.62} Re^{*0.39} A_r^{0.2} \left(\frac{k_{m,Avg.}}{d_{tube}} \right) \quad (1)$$

Also, Kang et al. [2] developed a single tube condensation model using mass transfer analogy at the film layer and diffusion layer interface to improve predictability and apply to various geometries of heat exchanger surface. In the film layer, Kang et al. [2] considered the film Reynolds number (Re_{film}) to reflect the film interface shape. The used film heat transfer models are as follows in Table I.

Table I: Film heat transfer models according to Re_{film}

Models
Laminar flow ($Re_{film} \leq 30$)
$h_{film} = 1.47 k_{film} Re_{film}^{-1/3} \left(\frac{\mu_{film}^2}{g \rho_f (\rho_f - \rho_g)} \right)^{-1/3}$
Laminar-wavy flow ($30 < Re_{film} \leq 1800$)
$h_{film} = \frac{k_{film} Re_{film}}{1.08 Re_{film}^{1.22} - 5.2} \left(\frac{\mu_{film}^2}{g \rho_f (\rho_f - \rho_g)} \right)^{-1/3}$
Turbulent flow ($1800 < Re_{film}$)
$h_{film} = \frac{k_{film} Re_{film}}{8750 + 58 Pr_{film}^{-0.5} (Re_{film}^{0.75} - 253)} \left(\frac{\mu_{film}^2}{g \rho_f (\rho_f - \rho_g)} \right)^{-1/3}$

For the calculation of the Heat Transfer Coefficient (HTC) with mass transfer analogy in the diffusion layer interface, Kang et al. [2] developed the Sherwood number (Sh). Developed Sh considers the intensity of natural convection (Ra_L), the suction effect (Θ), vessel geometry (AR), and the curvature effect ($\eta_{curvature}$) of the HX tube.

$$Sh = 0.19 Ra_L^{0.3} \Theta (1 + 0.2 AR^{0.8}) \eta_{curvature} \quad (2)$$

2.2. Tube-bundle condensation heat transfer model

Kang et al. [3] performed a tube-bundle condensation experiment and proposed the local and average bundle factors using a previously developed single tube condensation model. The average bundle factor is defined as follows.

$$\psi_{bundle} = \frac{h_{bundle}}{h_{single}} \quad (3)$$

This factor contains pressure loss ($\overline{\Delta P^*}$) by tube arrangement, suction effect (Θ) by gas diffusion, and blockage effect (α_{block}) caused by the diffusion layer. Especially, in the blockage effect term, the diffusion

layer thickness (δ) is calculated from the Sh number as follows in Eq. (6).

$$\psi_{bundle} = 1 + 210(\overline{\Delta P^*} - \Theta^{-0.005})\alpha_{block} \quad (4)$$

$$\alpha_{block} = \frac{2\delta}{p-d_{tube}} \quad (5)$$

$$Sh = \frac{L_{tube}}{\delta} = 0.13 Ra_L^{1/3} \Theta \quad (6)$$

3. Evaluation of condensation models

The single tube condensation heat transfer model and bundle factor model are needed to calculate the tube-bundle condensation heat transfer coefficient as in Eq. (3). In this study, tube-bundle experimental data performed by Kang et al. [3] were used to evaluate the single tube condensation heat transfer model and select a reference model. After this selection, the bundle factor models have been evaluated.

3.1. Combinations of condensation models

Fig. 1 and 2 show the evaluation results of each single tube condensation model. As shown in the figures, Kang et al. [2] model predicts well the experimental data with in Mean Absolute Percentage Error (MAPE) 6.97% and Root Mean Square Error (RMSE) 9.10%. So, this single heat transfer model was selected for the reference model.

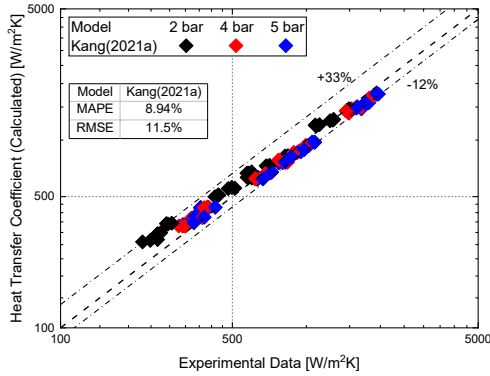


Fig. 1. Evaluation result of Kang et al. [1] model

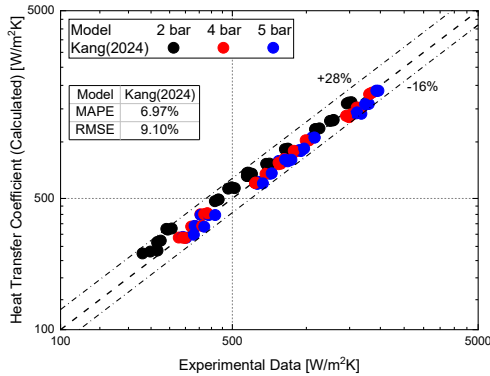


Fig. 2. Evaluation result of Kang et al. [2] model

Table II shows the combinations of the tube-bundle condensation heat transfer model. In this study, the same average bundle factor was employed; however, different correlations for the Sh number within the model were used, specifically those proposed by McAdams [4, 5] and Kang [2].

Table II: Combinations of tube-bundle condensation model

Combination	Bundle factor	Sh correlation
Case 1	Kang et al. [3]	McAdams [4, 5]
Case 2		Kang et al. [2]

3.2 Evaluation results

The evaluation result in Fig. 4 shows a higher prediction error than in Fig. 3, in the entire region of vessel conditions. This is because the diffusion layer thickness calculated from the Sh number correlation by Kang et al. [2] is smaller than that calculated from the McAdams [4, 5] correlation. This diffusion layer thickness may be affected by pressure and the air mass fraction of the vessel. So, under the high-pressure condition, the error of the heat transfer coefficient of the Kang et al. [2] correlation is lower than that of the McAdams [4, 5] correlation. And under the high air mass fraction condition, McAdams [4, 5] correlation shows higher predictability than Kang et al. [2] correlation, as shown in Table III and Table IV.

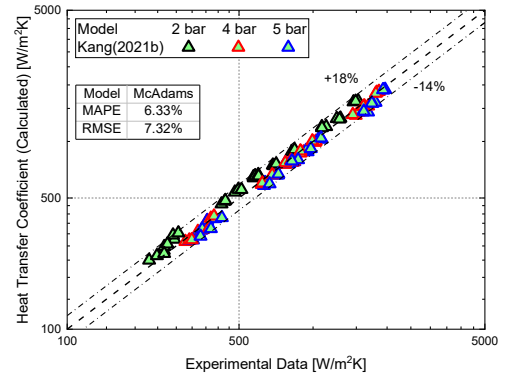


Fig. 3. Evaluation result of HTC with Case 1

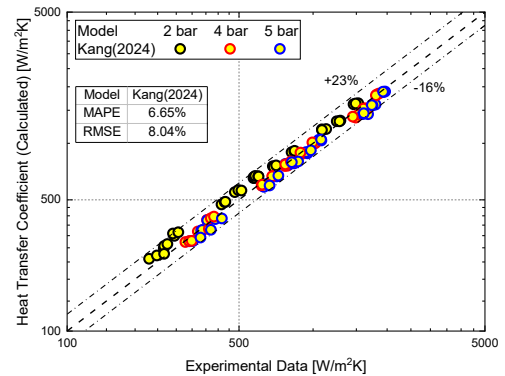


Fig. 4. Evaluation result of HTC with Case 2

Table III: MAPE and RMSE of tube-bundle average HTC according to the vessel pressure

Pressure	Sh correlation			
	McAdams [4, 5]		Kang et al. [2]	
	MAPE	RMSE	MAPE	RMSE
2 bar	9.10%	9.66%	10.64%	11.48%
4 bar	3.54%	4.19%	3.30%	3.97%
5 bar	6.36%	7.07%	6.01%	6.80%

Table IV: MAPE and RMSE of tube-bundle average HTC according to the vessel air mass fraction

Air mass fraction	Sh correlation			
	McAdams [4, 5]		Kang et al. [2]	
	MAPE	RMSE	MAPE	RMSE
0.2	6.34%	6.25%	7.31%	7.24%
0.4	5.46%	5.55%	6.46%	6.74%
0.5	6.93%	7.11%	7.66%	8.32%
0.7	6.60%	7.68%	7.79%	9.55%

4. Conclusion

This study evaluated the steam-air mixture outer wall condensation heat transfer models using Kang et al. [3] tube-bundle experimental data.

For the single tube condensation heat transfer model, the predictability of the Kang et al. [2] model, which is based on the HMTA method, is higher than the empirical model proposed by Kang et al. [1].

Also, this study evaluated the tube-bundle models according to the correlations of the Sh number. The latest Sh number correlation presented by Kang et al. [2] predicts with higher accuracy under the high-pressure condition and the low air mass fraction condition. However, this correlation is considered to require improvement to enhance the accuracy of diffusion layer thickness calculations under low-pressure and high air mass fraction conditions.

NOMENCLATURE

A_{CS}	Cross sectional area of vessel-tube (m^2)
AR	Area ratio ($= A_{HT}/A_{CS}$, -)
Ar	Aspect ratio ($= L_{tube}/r_{vessel} - r_{tube}$, -)
A_{HT}	Heat transfer area of HX tube (m^2)
d_{tube}	Tube diameter (m)
g	Gravitational acceleration (m/s^2)
h_{bundle}	Tube-bundle averaged HTC ($W/m^2 \cdot K$)
h_{single}	Single tube averaged HTC ($W/m^2 \cdot K$)
Ja	Jakob number (-)
k_{film}	Condensate film thermal conductivity ($W/m \cdot K$)
$k_{m,avg}$	Steam-air mixture thermal conductivity at average temperature ($W/m \cdot K$)
L_{tube}	Tube heat transfer length (m)
p	Pitch (m)
Pr_{film}	Condensate film Prandtl number (-)
$\overline{\Delta P^*}$	Averaged dimensionless pressure loss (-)
r_{tube}	Tube radius (m)

r_{vessel}	Vessel radius (m)
Ra_L	Rayleigh number (-)
Re^*	Diffusion Reynolds number (-)
Re_{film}	Film Reynolds number (-)
Sh	Sherwood number (-)
α_{block}	Blockage fraction (-)
δ	Diffusion layer thickness (m)
η	Curvature factor (-)
θ	Bird's suction factor (-)
μ_{film}	Condensate film dynamic viscosity (-)
ρ_f	Liquid density (kg/m^3)
ρ_g	Gas density (kg/m^3)
ψ_{bundle}	Bundle factor (-)

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