Development of Model for Steam Condensation in the Presence of Non-Condensable Gas on the Outer Wall of the Plate and Tube

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1. Introduction

The integrity of the nuclear power plants (NPPs) in the station blackout accident is an important design requirement after the Fukushima accident. To enhance the integrity of the NPPs, researches and development related to passive safety systems operating without electricity have been carried out in the nuclear industry. Passive containment cooling system (PCCS), which is one of the passive safety systems, improves the integrity of the containment building. In design basis accidents such as loss of coolant accident (LOCA) and main steam line break (MSLB), a large amount of steam with high energy is discharged into the containment building, and the steam increases pressure inside the containment building. In these accidents, the PCCS condenses the steam on the heat exchanger and reduces the pressure.

For the development of PCCS, experimental studies on steam condensation in the presence of air on flat plates or tubes were performed. Table I presents the geometric conditions of the heat exchangers in experiments. Uchida [1] and Matthew et al. [2] performed experiments using the flat plate. The tube data were produced by Dehbi [3], Lee et al. [4], and Kim et al. [5]. Furthermore, many correlations to predict the heat transfer coefficient were proposed. However, most of the correlations do not consider the effect of condensate film.

In the present study, the condensation model, which can consider the effect of condensate film, was developed.

Table I: Geometry of heat exchangers

Experiment	Plate/ Tube	Geometry
Uchida [1]	Plate	Width (W): 140 mm Height (L): 300 mm
Matthew al. [2]	Plate	Width: 457, 914 mm Height :2,130 mm
Dehbi [3]	Tube	Diameter: 38.0 mm Length: 3,500 mm
Lee et al. [4]	Tube	Diameter: 40.0 mm Length: 1,000 mm
Kim et al. [5]	Tube	Diameter: 21.5 mm Length: 1,000 m
Present	Tube	Diameter: 21.5, 33.6, 42.4 mm Length: 1,328, 1,280 mm

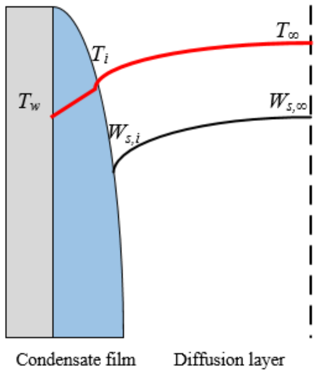

Fig. 1. Schematic of condensation phenomenon.

2. Development of Condensation Model

2.1 Condensation phenomenon

The condensation is subdivided into the diffusion layer and condensate film regions, as shown in Fig. 1. In the diffusion layer region, the steam is diffused from the bulk to condensate film, and then the steam is condensed on the condensate film. The diffusion is caused by the difference in the steam mass fraction (W_s) between bulk and interface. In the condensate film region, the heat is transferred from the interface to the outer wall of a heat exchanger. The condensate heat transfer was affected by the wave and film thickness.

The heat transfer coefficient (h) between bulk and outer wall can be expressed as follows:

$$h = \frac{h_{film} h_{cond}}{h_{film} + h_{cond}} \tag{1}$$

where h_{film} and h_{cond} are the heat transfer coefficients for condensate film and diffusion layer regions.

2.2 Condensate Film

The heat transfer of the condensate film was affected by the wave at the gas-condensate interface. Accordingly, the correlations considering the wave effect based on the Reynolds number of condensate were proposed from the previous studies and are as follows:

$$\bar{h}_{film} = 1.47 k_{film} Re_{film}^{-1/3} \left[\frac{\mu_l^2}{g \rho_l \left(\rho_l - \rho_g \right)} \right]^{-1/3},$$
$$\mathrm{Re}_{film} \le 30 \text{ [6]} \tag{2}$$

$$\bar{h}_{film} = \frac{k_{film} Re_{film}}{1.08 Re_{film}^{1.22} - 5.2} \left[\frac{\mu_l^2}{g \rho_l \left(\rho_l - \rho_g \right)} \right]^{-1/3},$$
$$30 < \mathrm{Re}_{film} \le 1800 \ \ [7] \tag{3}$$

$$\bar{h}_{film} = \frac{k_{film} Re_{film}}{8750 + 58 Pr_{film}^{-0.5} \left(Re_{film}^{0.75} - 253 \right)} \left[\frac{\mu_l^2}{g \rho_l \left(\rho_l - \rho_g \right)} \right]^{-1/3}$$
$$1800 < \mathrm{Re}_{film} \ \ [8] \tag{3}$$

The correlations were validated in previous studies and in good agreement with the data of condensate film under pure steam conditions. Therefore, the heat transfer coefficient of condensate film was obtained from the correlations.

2.3 Diffusion Layer

The steam diffused in the diffusion layer region condenses on the condensate film. Accordingly, the heat transfer coefficient (h_{cond}) was determined by the mass flux of the diffused steam, and it is derived based on the diffusion:

$$h_{cond} = \frac{\rho_{s-a} D_{s-a}}{1 - W_{s,i}} \frac{W_{s,\infty} - W_{s,i}}{\delta} \frac{h_{fg}}{T_\infty - T_i}, \tag{5}$$

where δ is the diffusion layer thickness and is obtained from Sherwood number (Sh_L):

$$Sh_L = \frac{L}{\delta}. \tag{6}$$

The Sherwood number is related to the diffusion mass transfer. The diffusion was affected by the free convection, suction effect, curvature of condensing surface, and vessel size. Therefore, the correlation for Sherwood number was proposed considering these effects.

The effect of the free convection was taken into account by the Rayleigh number (Ra_L). The Rayleigh number was defined as follows:

$$Ra_L = Gr_L Sc \tag{7}$$

where Gr_L and Sc are the Grashhof number and Schmidt number, and they are related to the free convection and the property of steam-air mixture, respectively. They are expressed:

$$Gr_L = \frac{\rho_{s-a} \left(\rho_{s-a,i} - \rho_{s-a,\infty} \right) g L^3}{\mu_{s-a}^2} \tag{8}$$

$$Sc = \frac{\nu_{s-a}}{D_{s-a}} \tag{9}$$

The suction phenomenon occurs when the steam condenses on the heat exchanger. Bird [9] proposed the suction factor ($\Theta_{suction}$) as follows:

$$\Theta_{suction} = \frac{\ln\left(1 + B\right)}{B} \tag{10}$$

$$B = \frac{W_{s,i} - W_{s,\infty}}{1 - W_{s,i}} \tag{11}$$

It was known that the mass diffusion on the tube was affected by the tube radius (R) and length (L). This effect is called the curvature effect. To consider the curvature effect, the curvature factor ($\eta_{curvature}$) is proposed based on the curvature parameter (γ_R) as follows:

$$\eta_{curvature} = \max\left[1, 4.9 \gamma_R^{0.6}\right] \tag{12}$$

$$\gamma_R = \frac{L}{R \cdot Ra_L^{1/4}} \tag{13}$$

The curvature effect is negligible when the curvature parameter is less than 0.07. Thus, the curvature factor is 1 in this region, as shown in Fig. 2.

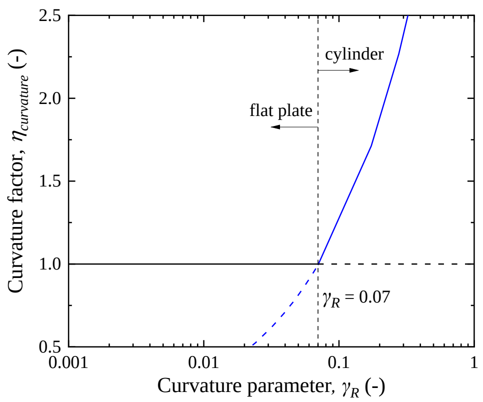

Fig. 2. Curvature factor according to curvature parameter.

The steam-air mixture flow in the vessel influences the diffusion layer thickness, and the flow is affected by the vessel size. The velocity of the steam-air mixture supplied is soon dissipated when the vessel size is much larger than the heat transfer area. Accordingly, the aspect ratio (AR) of heat transfer area (A_{HT}) and cross-sectional area ($A_{CS,vessel}$) of the vessel was defined to consider the vessel size effect. The aspect ratio was defined:

$$AR = \frac{A_{HT}}{A_{CS,vessel}} \tag{14}$$

The AR range of the data set is from 0.007 to 2.627, and the AR of the single train of the PCCS with respect to APR 1400 is about 0.61 [10].

The correlation for the Sherwood number was proposed based on Ra_L, $\Theta_{suction}$, $\eta_{curvature}$, and AR as follows:

$$Sh_L = 0.19 Ra_L^{0.3} \Theta_{suction} \eta_{curvature} \left(1 + 0.2 AR^{0.8}\right) \tag{15}$$

The coefficients in the correlation were determined by the regression analysis using the experimental dataset.

2.4 Evaluation of Model

The proposed condensation model was evaluated using the experimental dataset. Fig. 3 shows the predictive results of the proposed model. The model predicted 95% of data within the 24% errors.

The predictive performance of the proposed model was compared with those of the existing correlations proposed by Dehbi [11], Kim et al. [5], and Lee et al. [12]. The correlations are tabulated in Table II. As shown in Fig. 4, Dehbi correlation tended to overestimate the experimental data. Besides, the error bands of the correlations proposed by Kim et al. and Lee et al. were greater than that of the proposed model. Therefore, the predictive performance of the model is better than those of the existing correlations.

Table II: Existing correlations

Dehbi correlation [11]: $h=0.185D^{2/3}\left(\rho_{s-a,w}+\rho_{s-a,\infty}\right)\left(\frac{\rho_{s-a,w}-\rho_{s-a,\infty}}{\mu_{s-a}}\right)^{1/3}\frac{h_{fg}}{T_\infty-T_w}\ln\left(\frac{1-W_{s,w}}{1-W_{s,\infty}}\right)\eta_{curvature}$ $\eta_{curvature}=1+15.83\left(Gr_L^{-1/3}\frac{L}{d}\right)^{0.81}\left(1-0.35W_{a,\infty}^{0.25}\right)\left(\frac{P}{P_{atm}}\right)^{-0.09}\left(\frac{T_\infty-T_w}{T_w}\right)^{0.14}$
Kim et al. correlation [5]: $h=890Gr_L^{0.125}\left(1-W_{a,\infty}^{0.01}\right)^{0.966}Ja^{-0.327}\frac{k_{s-a}}{d}\eta_{curvature}$ $\eta_{curvature}=\frac{d}{0.04}\left(1-1.25\ln\frac{d}{0.04}\right)$
Lee et al. correlation [12]: $h=\frac{L^{0.347}\left[\left(-0.542+0.00085P_t\right)-\left(18.8+0.0354P_t\right)\log W_{a,\infty}\right]}{d^{1.356}\left(\frac{\Delta T_{w,sub}}{T_{crit}}\right)^{\left(0.351+0.0488W_{a,\infty}\right)\left(1-\frac{T_{crit}}{\Delta T_{w,sub}}\right)-\frac{29.97}{P_t}}}$

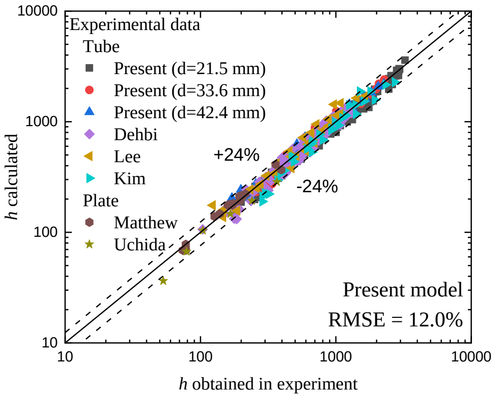

Fig. 3. Evaluation of proposed condensation model.

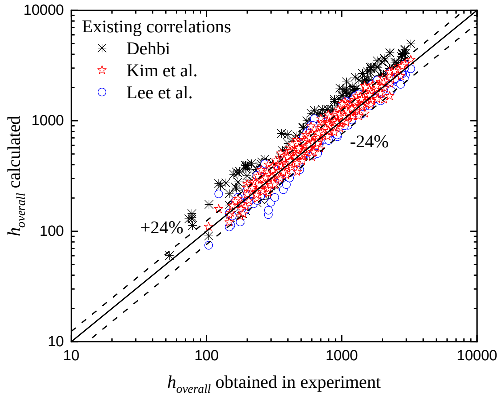

Fig. 4. Evaluation of existing correlations.

3. Conclusions

The condensation model was developed considering the characteristics of heat transfer in the condensate film and diffusion layer regions. For this, the correlation of Sherwood number was proposed based on the effects on steam diffusion, such as free convection, suction, curvature, and vessel size. The proposed condensation model can predict the heat transfer coefficients of condensation on the flat plates and tubes. Furthermore, the predictive performance was significantly improved.

REFERENCES

[1] H. Uchida, A. Oyama, Y. Togo, Evaluation of post-incident cooling systems of light water power reactors, in: Proc. Third Int. Conf. Peac. Uses At. Energy, 1964.

[2] M.M. Swartz, S.C. Yao, Experimental study of turbulent natural-convective condensation on a vertical wall with smooth and wavy film interface, Int. J. Heat Mass Transf. 113 (2017) 943–960.

[3] A.A. Dehbi, The effects of noncondensable gases on steam condensation under turbulent natural convection conditions, Diss. Massachusetts Inst. Technol. (1991).

[4] Y.G. Lee, Y.J. Jang, D.J. Choi, An experimental study of air–steam condensation on the exterior surface of a vertical tube under natural convection conditions, Int. J. Heat Mass Transf. 104 (2017) 1034–1047.

[5] U.K. Kim, J.W. Yoo, Y.J. Jang, Y.G. Lee, Measurement of heat transfer coefficients for steam condensation on a vertical 21.5-mm-O.D. tube in the presence of air, J. Nucl. Sci. Technol. 57 (2020) 905–916.

[6] W. Nusselt, Die oberflachenkondensation des wasserdamphes, VDI-Zs. 60 (1916) 541.

[7] S.S. Kutateladze, Fundamentals of Heat Transfer, Acad, Press. New York. (1963).

[8] D.A. Labuntsov, Heat transfer in film condensation of pure steam on vertical surfaces and horizontal tubes, Teploenergetika. 4 (1957) 72–79.

[9] R.B. Bird, Transport phenomena, Applied Mechanics Reviews, 2002.

[10] H. Ha, S. Lee, H. Kim, Optimal design of passive containment cooling system for innovative PWR, Nucl. Eng. Technol. 49 (2017) 941–952.
[11] A. Dehbi, Correcting for tube curvature effects on condensation in the presence of a noncondensable gas in laminar regimes, Int. J. Heat Mass Transf. 151 (2020) 119384.
[12] D.H. Lee, J.M. Yoo, H.Y. Kim, D.J. Hong, B.J. Yun, J.J. Jeong, Application of the machine learning technique for the development of a condensation heat transfer model for a passive containment cooling system, Nucl. Eng. Technol. (2021).