Modified Modeling of Rivulet Flow during Dropwise Condensation Heat Transfer on a Long Vertical Tube under Atmospheric Condition

Taeseok Kim[1] and Sung Joong Kim[1,2]
[1] *Department of nuclear engineering, Hanyang University*
222 Wangsimni-ro, Seongdong-gu Seoul 04763 Republic of Korea
[2] *Institute of Nano Science and Technology, Hanyang University*
222 Wangsimni-ro, Seongdong-gu Seoul 04763 Republic of Korea
*tkim@hanyang.ac.kr; sungjkim@hanyang.ac.kr**

1. Introduction

Among various heat transfer mechanisms, condensation is deemed as an most effective as used for various heat exchange devices. For example, Passive Containment Cooling System (PCCS) suppresses overpressure inside a containment by using the condensation heat transfer and numerous studies have tried to evaluate the condensation heat transfer of PCCS. Some theoretical heat transfer models have been proposed for PCCS focusing only on the filmwise condensation. Recent studies reported the dropwise condensation on a commercial vertical tube [1, 2]. Of note is that conventional heat transfer models for dropwise condensation may be ineffective to evaluate the PCCS performance because these models were validated only on small size test specimens unlike the long vertical tube of the PCCS. Therefore, a new heat transfer model of dropwise condensation is proposed to incorporate the length effect of the vertical tube in this study.

2. Theoretical modeling of rivulet flow

Rivulet flows during dropwise condensation are formed by means of coalescence droplets on the upper side of the surface. So, the behavior of rivulet flows is directly related to the heat and mass transfer on the upper side surface. According to a previous study by Gajewski [1], the width of rivulets is constant as the flow rate increases, and the contact angle of rivulet flows is increased. In this study, the width of rivulet flows is assumed as two times of the droplet contact area ($W = 2r_{max}\sin\theta$). Through this assumption, the thickness and the cross-sectional area of the rivulet flow can be derived by geometric calculation as shown in Eqs. (1) and (2), and Fig. 1.

$$\delta_r = \frac{W}{2}\cdot\left(\frac{1-\cos\omega}{\sin\omega}\right) \tag{1}$$

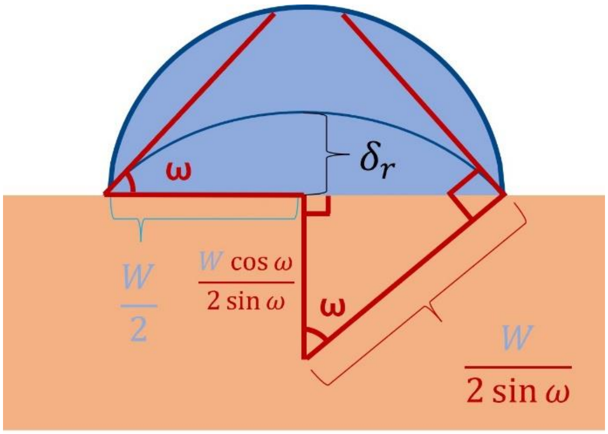
$$A_{rc} = \frac{W^2}{4}\,(\omega\csc^2\omega - \cot\omega) \tag{2}$$

Rivulet flows are formed by heat and mass transfer rate on the upper side surface. Thus, the volumetric flow rate of rivulet flows can be calculated by the heat transfer rate on the upper side and the number of rivulet flows. It is assumed that the number of rivulets and the width of a single rivulet can derive the circumference of the tube, and it can be expressed as follows:

$$n = \frac{\pi d_{out}}{2W} \tag{3}$$

The condensate volume should be formed by the rivulets, so the total flow rate of rivulet flows is given as,

$$n\dot{V} = \frac{Q_{up}}{h_{fg}\rho_l} \tag{4}$$



W: *Width of rivulet flow*
ω: *contact angle of rivulet*
θ: *contact angle of droplet*
$\dot{V}$: *Volume flow rate of liquid*

Fig. 1 Schematic of the cross-sectional view of the rivulet flow.

Hence, the velocity of rivulet flows can be calculated by assumption that it is a laminar flow on the free surface.

$$v_{rivulet} = \frac{\dot{V}}{A_{rc}} = \frac{\rho_l g \delta_r^2}{4\mu} \tag{5}$$

As a result, substituting Eq. (4) into Eq. (5) produces the contact angle of rivulet flows is given as,

$$(\omega\csc^2\omega - \cot\omega)\cdot\left(\frac{1-\cos\omega}{\sin\omega}\right)^2 = \frac{64\dot{V}\mu}{W^4\rho_l g} = \frac{16\mu}{\pi r_{max}^3\sin^3\theta\,\rho_l^2 g}\frac{Q_{up}}{d_{out}h_{fg}} \tag{6}$$

After the contact angle calculation, the previous single droplet heat transfer equations can be modified by applying the geometry of rivulet flows.

$$\Delta T_{curv,rivulet} = \frac{2T_{sat}\sigma}{h_{fg}\rho_l r} = \frac{2T_{sat}\sigma \sin\omega}{h_{fg}\rho_l r_{max}\sin\theta} \tag{7}$$

$$\Delta T_{lv,rivulet} = \frac{q_r}{W\frac{\omega}{\sin\omega}h_{lv}dy} = \frac{q_r \sin\omega}{2\omega r_{max}\sin\theta\, dy h_{lv}} \tag{8}$$

$$\Delta T_{ls,rivulet} = \frac{q_r}{2r_{max}\sin\theta\, dy h_{ls}} \tag{9}$$

$$\Delta T_{rivulet} = \frac{q_r}{4k_l dy}\int_0^\omega \frac{\tan\left(\frac{\varphi}{2}\right)}{\varphi}d\varphi \tag{10}$$

Therefore, temperature drop is derived as follows.

$$\Delta T = \Delta T_{curv,rivulet} + \Delta T_{lv,rivulet} + \Delta T_{rivulet} + \Delta T_{ls,rivulet} \tag{11}$$

Finally, substituting Eq. (7-10) into Eq. (11) produces the heat flux through the single rivulet flow:

$$\therefore q_r''(\omega(Q_{up})) = \frac{\left[\Delta T - \frac{2T_{sat}\sigma\sin\omega}{h_{fg}\rho_l r_{max}\sin\theta}\right]}{\frac{r_{max}\sin\theta}{2k_l}\int_0^\omega\frac{\tan\left(\frac{\varphi}{2}\right)}{\varphi}d\varphi + \frac{\sin\omega}{\omega h_{lv}} + \frac{1}{h_{ls}}} \tag{12}$$

3. Changing of droplet parameters by rivulet flow

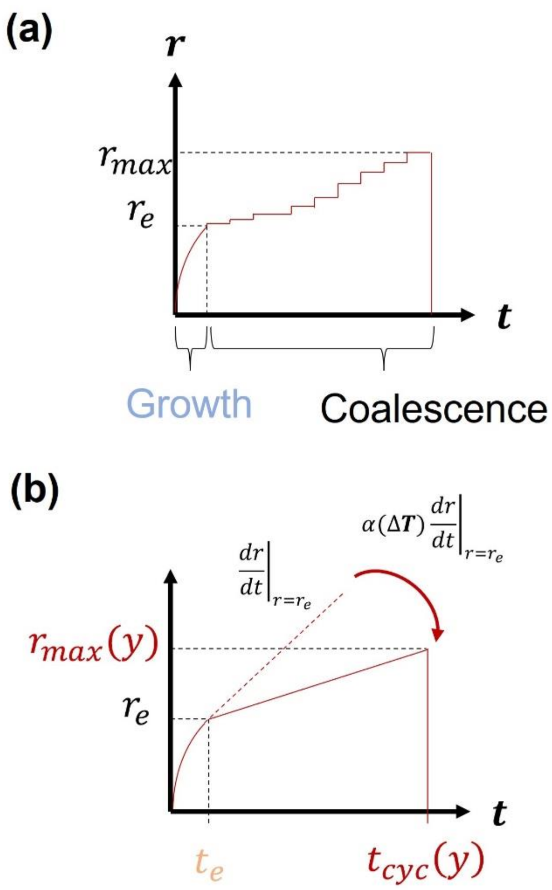


Fig. 2 Droplet growth modeling (a) in reality (b) modeling for a rivulet flow (not to scale).

In addition to the rivulet heat transfer, the rivulet flows can affect the droplet parameters such as the maximum droplet radius and droplet population density. In particular, the maximum droplet radius is an important factor to calculate the droplet population density. On a long vertical tube, a sweeping period of droplets determines the maximum droplet radius. To calculate the relationship between the sweeping period ($t_{cyc}(y)$) and the maximum droplet radius, the droplet growth is expressed as [2],

$$\frac{dr}{dt}\Big|_{r\le r_e} = G = A_1\frac{\left(1-\frac{\Delta T_{curv}}{\Delta T}\right)}{A_2 r + A_3} \tag{13}$$

The droplet grows by condensation before the droplet reaches to a sufficiently large size, and then the droplet grows further by coalescence with other droplets. Thus, the droplet radius can be described as shown in Fig. 1 (a). However, it is difficult to predict the droplet growth by the coalescence, so it was assumed as shown in Fig. 2 (b) and Eqs. (14) and (15).

$$r_e - r_{min} = \int_0^{t_e}\frac{\boldsymbol{dr}}{\boldsymbol{dt}}\Big|_{\boldsymbol{r\le r_e}} dt \tag{14}$$

$$\frac{r_{max}(y) - r_e}{(t_{cyc}(y) - t_e)} = \alpha\frac{dr}{dt}\Big|_{r=r_e} \tag{15}$$

The equivalent time can be solved using Runge-Kutta method, and the wall subcooling is an important factor determining the maximum droplet radius. The sweeping period is assumed to be a power function as proposed in Weisensee et al.'s study [3]. In addition, the sweeping period is related to the droplet growth, so the droplet growth is also the factor determining the sweeping period (Eq. (16)).

$$t_{cyc}(y, dT) = \frac{k}{\frac{dr}{dt}\Big|_{r=r_e}(dT)} \times y^{-0.3} \tag{16}$$

Eq. (16) includes an empirical coefficient, so it is determined by comparison with previous experimental data [4] as shown in Fig. 3.

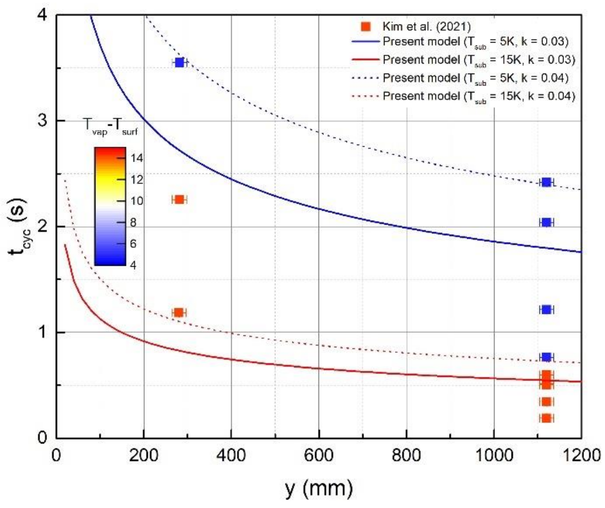


Fig. 3 Comparison between the sweeping period model and experimental data

Finally, the maximum droplet radius can be expressed as Eq. (17), and it is plotted as shown in Fig. 4.

$$r_{max}(y) = r_e + \alpha \left.\frac{dr}{dt}\right|_{r=r_e} (t_{cyc} - t_e) \tag{17}$$

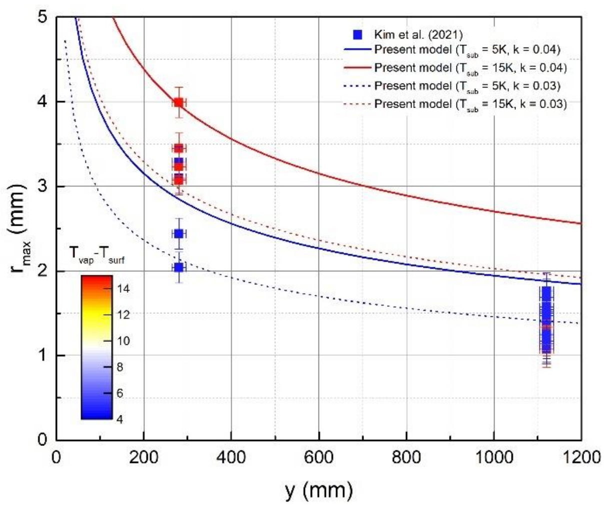


Fig. 4 Comparison between the maximum droplet radius model and experimental data

4. Calculation of overall heat transfer

Subsequently, the overall heat transfer in dropwise condensation was calculated based on modified rivulet heat transfer and droplet parameters. There are two regions on the condensation surface: the rivulet heat transfer region and dropwise condensation region. Thus, the rivulet heat transfer region is derived as,

$$A_r(y) = n \cdot W dy \frac{t_r}{t_{cyc}(y, dT) + t_r} \tag{18}$$

$$t_r(y) = \frac{dy}{v_{rivulet}(y)} \tag{19}$$

dy is a unit length for calculation of overall heat transfer, which is necessary because the rivulet heat transfer is related to the heat transfer rate on the upper side surface. Then the overall heat transfer can be derived as,

$$q''(y) = \frac{A_d(y)}{A}\left(q''_F(y) + q''_d(y)\right) + \frac{A_r(y)}{A} q''_r(y) \tag{20}$$

Finally, the condensation heat flux can be calculated by the overall heat transfer (Eq. (20)). The condensation heat flux is shown in Fig. 5, and it shows the comparison with the experimental data of the previous studies [5-8]. There is a significant discrepancy between the model and the experimental results, but the model clearly shows the dependence on the length and the wall subcooling successfully. Still the model needs to be improved to predict experiment data more accurately.

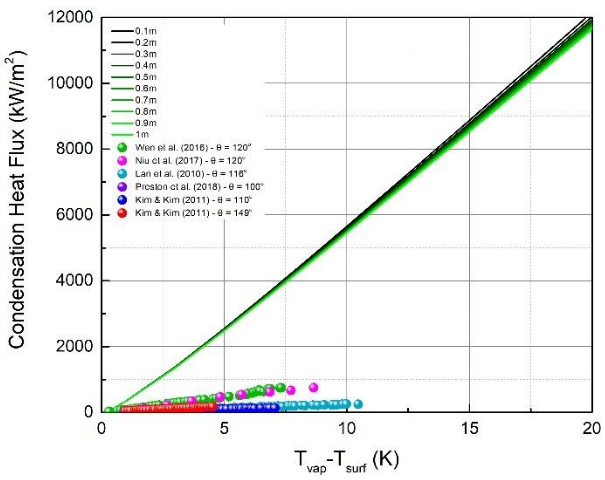


Fig. 5 Condensation heat flux of the modified model and experimental data

Even though the prediction model shows a poor agreement with the experimental data, the effect of wall subcooling and the length on the heat transfer rate could be captured in the model successfully. The normalized heat flux at the top of the surface is shown in Fig. 6, and it shows a good agreement with the previous study [3].

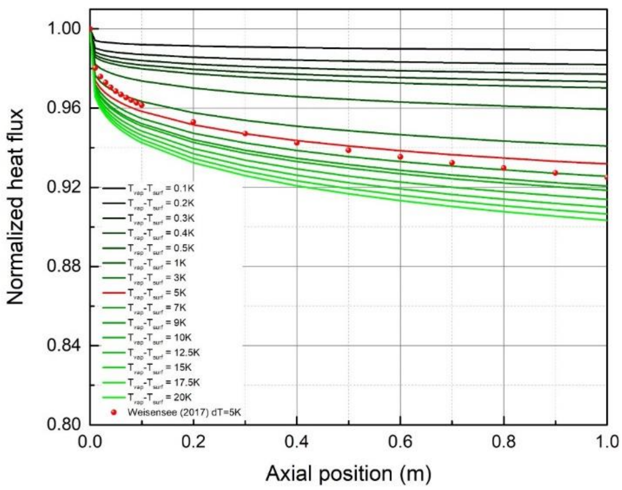


Fig. 6 Normalized heat flux of the modified model and the previous study

5. Conclusions

Improper models have been adopted in predicting the condensation heat transfer in PCCS; the filmwise condensation correlations have been used, but the dropwise condensation is shown predominantly during the abundant experimental database. However, the conventional dropwise condensation correlations are not applicable to the geometry of PCCS. Therefore, a new dropwise condensation correlation was proposed in this study, and it incorporates a length effect considering the length of the heat exchanger tubes in PCCS. This approach is comparable to the previous study, and it represents a new approach to predict the condensation heat transfer for PCCS application. However, the proposed model needs to be validated through the sufficient experimental data. So a future work will be focused on producing sufficient data to support the improved model.

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