Prediction Method of Condensation Heat Transfer from Steam-Air Mixture for CFD Application*

Michio MURASE*** Yoichi UTANOHARA** Shigeo HOSOKAWA***† Akio TOMIYAMA***

Abstract In our previous study, we measured the radial and axial temperature distributions of steam-air mixture in a vertical circular pipe (diameter, 49.5 mm; cooling height, 610 mm). The measured temperatures were used to evaluate the condensation heat flux $q_c$ and heat transfer coefficient $h_c$, where the bulk quantities were defined at the center of the circular pipe. It is sometimes difficult to define the bulk quantities in a three-dimensional computation for a complex geometry or complex flow field. In this study, therefore, we evaluated a prediction method for $h_c$ without the definition of the bulk quantities by using the measured temperature distributions. We applied existing $h_c$ correlations to an arbitrary radial location $y$ from the condensation surface, and obtained the distribution of local $h_c$. As a result, we found that $h_c$ was radially almost constant in the turbulent boundary layer (dimensionless distance of $y^+ > 20$). This means that existing $h_c$ correlations are applicable to computational cells of $y^+ > 40$ in a three-dimensional computation.

Keywords: CFD application, Condensation, Heat transfer coefficient, Non-condensible gas, Vertical circular pipe

1. Introduction

The goal of this study is to evaluate the pressure and temperature in a containment vessel (CV) of a nuclear power plant in detail during a loss-of-coolant accident (LOCA) by using a computational fluid dynamics (CFD) code. Prediction of condensation heat transfer coefficient $h_c$ from the mixture of steam and non-condensible gas (air or nitrogen) is important for this purpose. $h_c$ is generally for heat transfer between the bulk and condensation surface. It is however difficult to define the bulk quantities for the complex flow in the CV during the LOCA because many circulation flows occur in a large closed vessel. Therefore, in this paper we evaluated a prediction method for $h_c$ by using physical quantities in a region close to the condensation surface.

Structural integrity for the CV during the LOCA and a severe accident (SA) is evaluated by using several thermal-hydraulic computer codes [1]. Some thermal-hydraulic experiments with large-scale vessels simulating the CV [2–4] and CFD analyses for those experiments [4–6] have been carried out, and good agreements between data and analytical results were obtained for macroscopic behavior in the vessel. However, the local $h_c$ has not been well validated, and existing $h_c$ correlations gave overestimations for small $h_c$ and underestimations for large $h_c$ [7]. Benchmarking activities [8] were performed for the condensation rates on a vertical flat plate of 2 m height to improve the CFD models for CV analysis and good predictive capabilities were achieved for the condensation rates. The temperature distributions of the steam-air mixture however have not been reported. Legay-Desqueselles and Prunet-Foch [9] measured the temperature distribution in one cross section on a flat plate, and

---

* Received: 1 April 2021 / Accepted: 3 June 2021 / Published online: 2 August 2021
** Institute of Nuclear Safety System, Inc., 64 Sata, Mihama-cho, Mikata-gun, Fukui-ken, 919-1205 Japan
TEL:+81-(770)-37-9100 FAX: +81-(770)-37-2009 E-mail: murase@inss.co.jp
*** Kobe University †Present affiliation: Kansai University
showed that computations underestimated mixture temperatures. Kang and Kim [10] measured the temperature distribution for a nearly horizontal flat plate, and showed that computations underestimated mixture temperatures when a model for mist formation in the mixture was not employed. Many experiments have been done and many $h_c$ correlations have been proposed [1, 11], but $h_c$ has been defined using the bulk and condensation surface temperatures in all $h_c$ correlations.

In our previous study [12], we measured temperature distributions of steam-air mixture in a vertical pipe with the diameter of 49.5 mm. We compared the measured $h_c$ with the correlations based on the analogy between heat and mass transfer by Araki et al. [13] and based on the diffusion layer model by Liao and Vierow [14]. From these comparisons, good agreement was confirmed between the measured $h_c$ and correlations [15]. One technical issue however remained: the prediction method of $h_c$ from physical quantities in the computational cell next to the condensation surface. In this study, we therefore applied the $h_c$ correlation by Dehbi et al. [6] based on the gradient of steam concentration at the condensation surface to the measured temperature distributions [12], and evaluated local $h_c,\text{cal}$ for the dimensionless distance $y^+$ from the condensation surface. The correlation by Dehbi et al. [6] requires small $y^+$ value. Therefore, we also evaluated applicability of existing $h_c$ correlations [13, 14] to a local $y^+$ in the thermal boundary layer.

2. Outline of Experiment and Condensation Heat Transfer

2.1 Experimental Conditions

Locations of temperature measurement in the experiments [12] are shown in Fig. 1. The diameter $d$ of the test section was 49.5 mm, the wall thickness was 5.5 mm, the gap of the cooling water annulus was 8.25 mm, and the height of the cooling region was 610 mm. Experiments were carried out under steady-state conditions. Flow rates of steam and air, the pressure and temperature of the inlet steam-air mixture, and temperature distributions at locations shown in Fig. 1 were measured. The measurement error was ±0.5 °C for the T-type sheathed thermocouples used. An array with eleven T-type sheathed thermocouples ($\phi$0.10) were used to measure radial temperature distributions of the steam-air mixture. Temperature distributions at $z = 50, 90, 140, 240, 390$ and 500 mm were measured by manually moving the thermocouple array. T-type sheathed thermocouples ($\phi$0.5, flat at the top end) were embedded in the heat transfer pipe at 1.5 and 3.5 mm from the inner surface of the pipe to measure temperature gradients in the heat transfer wall. Temperature distributions of the cooling water were measured by moving T-type sheathed thermocouples ($\phi$0.5).

Table 1 lists the main experimental conditions [12]. The parameters were the steam flow rate at the inlet of the test section $W_{s,\text{in}}$ and air flow rate $W_a$. Nos. 3-5 were the base cases and were for confirmation of repeatability. Nos. 1 and 2 were cases for examining effects of small air mass flow ratio $x_{a,\text{in}}$. Nos. 6-8 were for checking effects of flow rates at the same $x_{a,\text{in}}$ and Nos. 9-11 were for effects of large $x_{a,\text{in}}$ at a similar $W_{s,\text{in}}$. The Reynolds number of the mixture in the inlet section was in the range of $Re_{p,\text{in}} = 3600-25000$ (where $Re_{p,\text{in}} = (W_{s,\text{in}} + W_a) \rho_c d \nu_c / \mu_c$ and $W_{s,\text{in}}$ is $W_s$ at the test section inlet). The Reynolds number of cooling water was about 500. In Table 1, $q$ is the heat flux averaged in the region of $z = 0-500$ mm.
Table 1  Experimental conditions [12].

| Case No. | Ws,in [g/s] | Wa [g/s] | xa,in [-] | Tg,in [°C] | Tcw,in [°C] | q [kW/m²] |
|----------|-------------|----------|-----------|------------|-------------|-----------|
| 1        | 3.58        | 2.0      | 0.36      | 97.3       | 18.5        | 47.5      |
| 2        | 4.03        | 4.5      | 0.53      | 91.5       | 12.5        | 46.0      |
| 3        | 5.77        | 9.0      | 0.61      | 87.3       | 12.7        | 47.6      |
| 4        | 5.91        | 9.0      | 0.60      | 87.7       | 22.5        | 45.8      |
| 5        | 5.86        | 9.0      | 0.61      | 87.5       | 28.5        | 39.9      |
| 6        | 3.94        | 6.0      | 0.60      | 87.7       | 12.7        | 42.8      |
| 7        | 1.91        | 3.0      | 0.61      | 87.4       | 12.7        | 27.9      |
| 8        | 0.84        | 1.5      | 0.64      | 85.9       | 11.2        | 14.0      |
| 9        | 0.82        | 3.0      | 0.79      | 74.6       | 11.0        | 11.0      |
| 10       | 0.79        | 6.0      | 0.88      | 61.8       | 11.2        | 11.2      |
| 11       | 0.74        | 9.0      | 0.92      | 53.5       | 11.2        | 9.5       |

Pressure in the inlet section was \( P_w = 0.124-0.127 \) MPa, and cooling water flow rate was \( W_{cw} = 56 \) g/s.  
\( W_{s,in} \), steam flow rate in the inlet section; \( W_a \), air flow rate; \( x_{a,in} \), air mass flow ratio \((=W_d(W_{s,in}+W_a))\); \( T_{s,in} \), mixture temperature in the inlet section; \( T_{cw,in} \), cooling water inlet temperature; \( q \), average heat flux.

2.2 Heat Flux

The temperature gradient in the heat transfer wall is widely used to measure the condensation heat flux \( q_w \). We also obtained \( q_{cw} \) and \( q_g \) from the enthalpy increasing rate of the cooling water and the enthalpy decreasing rate of the steam-air mixture, respectively. There were differences among the quantities \( q_w \), \( q_{cw} \), and \( q_g \), hence the average values of \( q_w \), \( q_{cw} \), and \( q_g \) were used [12]:

\[
q_{ave,j} = \frac{(q_{cw,j}+q_{g,j}+q_{w,j}+q_{q,j})/2)^{1/3}}{}
\]

where the subscript \( j \) shows the number of an axial location. \( q_{ave} \) defined by Eq. (1) is the average value between the neighboring thermocouple locations.

2.3 Heat Transfer Coefficient

The relationship between the heat flux \( q_{ave} \) and heat transfer coefficients \( h \) is expressed by:

\[
q_{ave} = h_t (T_b - T_w) = (h_c + h_{conv})(T_b - T_j)
\]

where \( h_t \) is the overall heat transfer coefficient, \( h_c \), \( h_{conv} \) and \( h_j \) correspond to the heat transfer coefficients due to condensation of steam, convection of the gas phase and condensate liquid film, respectively. \( T_b, T_j \) and \( T_w \) are temperatures in the bulk, at the condensate film surface and at the wall surface, respectively. Eq. (2) leads to:

\[
1/ht = 1/(hc + h_{conv}) + 1/ht.
\]

Eq. (3) is widely used [13, 14] to evaluate \( h_c, h_j \) and \( T_j \) were obtained by using Nusselt’s model [16] expressed by:

\[
h_j = \frac{\delta f / \delta y (\psi f)^{1/2}}{(3\nu f/\lambda f)^{1/3}},
\]

\[
T_j = T_w + q_{ave} lh_f
\]

where \( \Gamma \) is the liquid film flow rate per perimeter, \( \delta f \) is the liquid film thickness, \( \lambda_f \) is the thermal conductivity of liquid, \( \nu_f \) is the liquid viscosity, and \( \psi_f \) is the kinematic viscosity. The obtained liquid film thickness was less than about 0.1 mm, and the liquid film Reynolds number \((= \Gamma / \nu_f)\) was in the range of 0.11-12. \( h_{conv} \) was estimated by using the Colburn formula [17] for fully-developed turbulent flow and the effect of the developing region of the thermal boundary layer \( Nu_t/Nu_{∞} \):

\[
Nu = h_{conv} dl/\lambda_g = 0.023 Re_{g}^{0.8} Pr_{g}^{1/3} (Nu_t/Nu_{∞})
\]

for \( 2300 < Re_{g} < 10^7 \)

where \( d \) is the diameter, \( Nu \) is the Nusselt number, \( Pr \) is the Prandtl number, \( Re_{g} \) is the Reynolds number for the steam-air mixture \((Re_e = (W_s+W_a)d/\nu_A \nu g))\), and \( \lambda_g \) is the thermal conductivity of gas. For \( Nu_t/Nu_{∞} \), the correlation by Reynolds et al. [18] was used.

\[
Nu_t/Nu_{∞} = 1 + C/\sqrt{z/d},
\]

\[
C = 0.8(1+70000/Re_{g}^{3/2})
\]

for \( 3000 < Re_{g} < 500000 \)

In the previous study [15], \( h_f = 6.3-22.2 \text{ kW/m²K} \) and \( h_{conv} = 0.007-0.037 \text{ kW/m²K} \) for \( h_c = 0.060-4.5 \text{ kW/m²K} \), hence effects of \( h_f \) and \( h_{conv} \) on \( h_c \) could not be neglected.

2.4 Existing Correlations

Araki et al. [13] measured local \( h_c \) from the temperature gradient in the heat transfer pipe with \( d = 49.5 \text{ mm} \) and proposed the \( h_c \) correlation based on the analogy between heat and mass transfer:

\[
h_c = m_s h_g f (T_b - T_j),
\]

\[
m_s = (D P M_s/R T_{ave}) d \ln(P_{g,f}/P_{g,b})/ \ln(1 - 2/ Sh),
\]

\[
T_{ave} = (T_b + T_j)/2
\]
where $D$ is the diffusion coefficient, $h_{fg}$ is the latent heat of condensation, $M_s$ is the molecular weight of steam, $m_s$ is the mass flux of steam, $P$ is the pressure, $R$ is the universal gas constant, and $Sh$ is the Sherwood number. $Sh$ was computed by:

$$Sh = 0.023\, Re_g^{0.8}\, Sc^{1/3} \left( Sh_0/\text{Sh}_\infty \right)$$  \hspace{1cm} (10)

for $2300 < Re_g < 10^7$

$$Sh_0/\text{Sh}_\infty = Nu_c/\text{Nu}_\infty,$$

which is based on the analogy with Eq. (6).

Liao and Vierow [14] proposed the $h_c$ correlation based on the diffusion layer model expressed by:

$$h_c = Sh \left( \lambda_c \, \delta \right),$$ \hspace{1cm} (11)

$$\lambda_c = \left( \phi_2 \, h_{fg} \, h_{fg}/\rho_0 \, D \, M_s \, M_h/(\phi_1 \, R^2 \, T_g^3) \right),$$ \hspace{1cm} (12)

$$1/\phi_1 = (X_{cm}/X_{af}) \ln(1+B_m)/B_m, \quad \phi_2 = M_c^2/(M_{g,b} \, M_{s,f})$$

where $B_m$ is the mass transfer driving force and $\ln(1+B_m)/B_m$ is the suction factor [19], $h_{fg} = h_{fg} + c_p (T_b - T_f)$, $X_{af}$ is the air mass fraction at the condensation surface, $X_{cm}$ is the log mean steam mass fraction, and $\lambda_c$ is the condensation thermal conductivity. Eq. (10) was used for $Sh$ in Eq. (11). $R$ is the universal gas constant, and $R$ (kJ/kmol K) and $R$ (m$^3$ atm/kmol K) should be used for the term $R^2$ to keep dimensions in Eq. (12).

The $h_{c,cal}$ values computed with Eqs. (8) and (9) by Araki et al. [13] and with Eqs. (11) and (12) by Liao and Vierow [14] were similar, and they were smaller than the measured $h_{c,exp}$ for $z = 50$ mm near the test section inlet, and were larger than $h_{c,exp}$ for the downstream region in Nos. 8 and 9 with smaller $q_i$ [15]. The average of $h_{c,cal}$ by Liao and Vierow [14] was 9.5% greater than that by Araki et al. [13], and was 1.2% smaller than the average of $h_{c,exp}$. The standard deviation $\sigma$ of $h_{c,cal}$ from $h_{c,exp}$ was $\pm 30\%$ with Eqs. (8) and (9), and $\pm 34\%$ with Eqs. (11) and (12).

3. Correlation for CFD Application

3.1 Steam Mass Transfer Rate at Condensation Surface

Dehbi et al. [6] proposed the following condensation heat transfer model for a CFD application:

$$h_c = m_s \, h_{fg}/(T_g - T_f),$$ \hspace{1cm} (13)

$$m_s = \{1/(X_s - 1)\} \, \rho_s \, D \left( \partial X_s / \partial y \right),$$ \hspace{1cm} (14)

where $T_g$ is the temperature of steam-air mixture, $y$ is the distance from the condensation surface, $X_s$ is the steam mass fraction, and $\rho_s$ is the density of the steam-air mixture. They used a small computational cell (the dimensionless distance of $y^* = 0.47$) to evaluate $\partial X_s / \partial y$ at the condensation surface, and recognized that $y^* = 0.47$ was not practical for the CV analysis. Furthermore, Dehbi et al. did not confirm whether or not $\partial X_s / \partial y$ approached a constant value near the condensation surface. $y^*$ is defined by:

$$y^* = u_i y/v_i, \quad u_i = (\tau_w/\rho_s)^{1/2},$$ \hspace{1cm} (15)

where $u_i$ is the friction velocity, and $\tau_w$ is the wall shear stress. $\tau_w$ was calculated by using the Blasius formula.

$\partial X_s / \partial y$ in Eq. (14) was approximated by the gradient of $X_s$ between 0 and $y$ from the condensation surface ($\partial X_s / \partial y = \Delta X_s / \Delta y$) as:

$$m_s = \{1/(X_s - 1)\} \, \rho_s \, D \left( \Delta X_s / \Delta y \right).$$ \hspace{1cm} (16)

$D$, $X_s$, and $\rho_s$ depend on $T_g$. Fig. 2 shows $T_g$, $\rho_s$, $X_s$, and $\Delta X_s / \Delta y$ at $z = 0.14, 0.24, 0.39$ m in No. 1 as an example case. As $y$ became smaller, $\rho_s$ increased and $X_s$ decreased, because $T_g$ decreased. $D$ decreased, but its change was small and in the range of $2.7-3.2 \times 10^{-5}$. 
The ratio of \( h_{c,\text{cal}} \) computed with Eqs. (13) and (16) to \( h_{c,\text{exp}} \) is shown in Fig. 3 as a function of \( y^* \). \( h_{c,\text{exp}} \) was obtained by using Eq. (2) [15] and was defined between the bulk and condensation surface. \( h_{c,\text{cal}}/h_{c,\text{exp}} \) was small for small \( z \) like \( z = 0.09 \text{ m} \), and in the figure it is not shown for \( z = 0.05 \text{ m} \). In the turbulent boundary layer, changes of \( T_g \) and \( X_i \) were relatively small (see Fig. 2) and \( h_{c,\text{cal}} \) strongly depended on \( y \). In the region of the measured \( T_g \), most of \( h_{c,\text{cal}}/h_{c,\text{exp}} \) increased with decreasing \( y^* \).

Fig. 4 compares \( h_{c,\text{cal}} \) at the minimum \( y^* \) (= 3.3-18) with \( h_{c,\text{exp}} \). \( h_{c,\text{cal}} \) was smaller than \( h_{c,\text{exp}} \), but showed a similar trend to \( h_{c,\text{exp}} \). Figs. 3 and 4 show that Eq. (16) was qualitatively valid and that the temperature at small \( y^* \) was needed to confirm quantitative validity of Eq. (16).

It was also difficult to judge a suitable \( y^* \) predicting \( h_c \) from the temperature distributions measured by Legay-Desesquelles and Prunet-Foch [9] and Kang and Kim [10]. It is difficult to measure temperatures near the condensation surface, and we have to use temperatures computed by a CFD code for evaluation of Eq. (16).

The most effective factor for \( h_{c,\text{cal}} \) in Eq. (16) was \( y \) as shown in Fig. 2, and \( h_{c,\text{cal}} \) shown in Fig. 3 was approximately expressed by a power function of \( y^* \). From the relationship between \( h_{c,\text{cal}}/h_{c,\text{exp}} \) and \( y^* \) for Nos. 1-11, a function for the correction factor \( \eta \) was derived by using the least square method as:

\[
\eta = \min(10.5/\sqrt{y^*})^{0.92}, 1.
\]

\( h_{c,\text{cal}}/h_{c,\text{exp}} \) in the region of \( 3.3 \leq y^* \leq 250 \) was used to derive Eq. (18). In Eq. (18), \( \eta = 1 \) in the region of \( y^* < 12 \), and \( y^* = 12 \) was much larger than \( y^* = 0.47 \) used by Dehbi et al. [6]. This suggests that one-order larger \( y^* \) than that used by Dehbi et al. [6] might be applied to Eq. (16). Fig. 5 shows \( h_{c,\text{mod}} \) computed with Eqs. (13) and (16)-(18) comparing with \( h_{c,\text{exp}} \). \( h_{c,\text{mod}} \) depends on \( y^* \), hence the average of the maximum and minimum values in the region of \( 12 < y^* \leq 250 \) is shown in Fig. 5. The difference between the maximum and minimum values was large for Nos. 8, 9 and 1, and the differences of \( \pm 18 \% \), \( \pm 11 \% \) and \( \pm 10 \% \) for Nos. 8, 9 and 1 are shown by error bars. \( h_{c,\text{mod}} \) was much smaller than \( h_{c,\text{exp}} \) for \( z = 0.05 \text{ m} \) and was smaller than \( h_{c,\text{exp}} \) for \( z = 0.09 \text{ m} \). The standard deviation of \( h_{c,\text{mod}} \) was \( \sigma = \pm 33 \% \) for \( h_{c,\text{exp}} \) excluding \( h_{c,\text{mod}} \) for \( z = 0.05 \text{ m} \).

Eq. (14) is based on the assumption of constant \( \rho_g \), but \( \rho_g \) increases with decreasing \( y \) due to increased concentration of non-condensable gas. De la Rosa et al. [1] recommended \( "m_s = - D \frac{\partial \rho_s}{\partial y}" \) rather than \( "m_s = - D \frac{\partial \rho_s}{\partial y}" \) in their review, because \( "m_s = - D \frac{\partial \rho_s}{\partial y}" \) includes effects of \( \rho_g \). Addition of the convection term for \( "m_s = - D \frac{\partial \rho_s}{\partial y}" \) leads to:

\[
m_s = \left( \rho_{nc} \rho_{nc} \right) \left( \frac{\partial \rho_{nc}}{\partial y} \right) - \left( \frac{\partial \rho_{nc}}{\partial y} \right) D
\]

where \( \rho_{nc} \) is the density of non-condensable gas. There was no clear difference between the \( h_{c,\text{cal}} \) values computed with Eqs. (14) and (19).

### 3.2 Application of Existing Correlations

The \( y \) value in Eq. (16), which is based on Eq. (14), should be in the region of \( y^* < 12 \), where \( \eta = 1 \) in Eq. (18). A practical \( y^* \) value for CV analysis may be on the order of 100. In this case, Eqs. (13) and (16) should be modified by Eqs. (17) and (18), but applicability of Eqs. (17) and (18) to CV analysis is unknown.

Existing correlations, Eqs. (8) and (9) and Eqs. (11) and (12), were applied to evaluate \( h_{c,\text{cal}} \) for the local \( T_g \) at the temperature measuring location \( y \) like Fig. 3. \( S_h \) in Eqs. (9) and (11) was calculated with Eq. (10). For \( Re_g \) in Eq. (10), \( Re_g = u_{ave} d v_{ave} \) (where \( u_{ave} \) is the average velocity) is generally used. However, it is convenient to use values in the computational cell in contact with the condensation surface in the CFD analysis. Therefore, \( Re_g = u (2y)/v_g \) (where \( u \) is the local velocity) was used for \( Re_g \) in Eq. (10). The characteristic length of \( 2y \) was used to make \( 2y = d \) at the center of the pipe (\( y = d/2 \)). The turbulent velocity distribution in a circular pipe was used for the local velocity \( u \).

\( h_{c,\text{cal}} \) computed with Eqs. (8) and (9) by Araki et al. [13], which were applied to the \( T_g \) measuring location, is shown in Fig. 6. \( h_{c,\text{cal}}/h_{c,\text{exp}} \) (where \( h_{c,\text{exp}} \) is constant) for \( z = 0.05 \text{ m} \) was very small and is not shown in Fig. 6. \( h_{c,\text{cal}}/h_{c,\text{exp}} \) was small for \( z = 0.09 \text{ m} \). \( h_{c,\text{cal}}/h_{c,\text{exp}} \) was almost constant in the region of \( y^* > 20 \) and was within \( 1\pm 0.2 \) for \( z \geq 0.14 \text{ m} \). In the region
The ratio of $hc_{cal}$ computed with Eqs. (13) and (16) to $hc_{exp}$ is shown in Fig. 3 as a function of $y^+$. $hc_{exp}$ was almost constant in the region of $y^+ > 20$ and was within $1\pm0.2$ for $c > 0.14$ m. In the region of $0.01 < y^+ < 0.1$, $hc_{cal}/hc_{exp}$ was almost constant and was within $1\pm0.2$ for $z > 0.14$ m. In the region of $y^+ < 0.01$, $hc_{cal}$ was derived by using the least square method as:

$$hc_{cal} = \frac{1}{2} \left( \frac{1}{y^+} \right)^2 + 0.1$$

Fig. 3 Relationship between $hc_{cal}/hc_{exp}$ and $y^+$.

Fig. 4 $hc_{cal}$ for the smallest $y^+$ in Fig. 3.

Fig. 5 $hc_{mod}$ modified by Eqs. (17) and (18).

Fig. 6 Local $hc_{cal}$ computed by Eqs. (8) and (9).

Fig. 7 $hc_{cal}$ computed by Eqs. (8) and (9).

Fig. 8 $hc_{cal}$ computed by Eqs. (11) and (12).
of \( y^* < 20 \), \( \frac{h_c,cal}{h_c,exp} \) rapidly decreased with decreasing \( y^* \) due to low velocity and temperature at small \( y \). Fig. 6 showed that Eqs. (8) and (9) could be approximately applied to a wide region of \( y^* \).

Fig. 7 compares \( h_c,cal \) computed with Eqs. (8) and (9) by Araki et al. [13] with the measured \( h_c,exp \). \( h_c,cal \) depended on \( y^* \) as shown in Fig. 6, and the average of the maximum and minimum values in the region of \( y^* > 20 \) was used. The difference between the maximum and minimum values was large for Nos. 9 and 1, and it was \( \pm 10.7 \% \) and \( \pm 7.5 \% \) for Nos. 9 and 1, respectively. The standard deviation of \( h_c,cal \) was \( \sigma = \pm 35 \% \) for \( h_c,exp \) excluding \( h_c,cal \) for \( z = 0.05 \) m.

Fig. 8 compares \( h_c,cal \) computed with Eqs. (11) and (12) by Liao and Vierow [14] with the measured \( h_c,exp \). The difference between the maximum and minimum values was large for Nos. 9, 1 and 2, and it was \( \pm 13.5 \% \), \( \pm 9.9 \% \) and \( \pm 7.4 \% \) for Nos. 9, 1 and 2, respectively. The standard deviation of \( h_c,cal \) was \( \sigma = \pm 36 \% \) for \( h_c,exp \) excluding \( h_c,cal \) for \( z = 0.05 \) m.

\[ \sigma = \pm 35 \% \text{ in Fig. 7 and } \sigma = \pm 36 \% \text{ in Fig. 8} \] were close to \( \sigma = \pm 30 \% \) and \( \sigma = \pm 34 \% \) [15], respectively, where the correlations were applied between the bulk and the condensation surface. Figs. 7 and 8 showed that the existing \( h_c \) correlations [13, 14] could be applied to \( y^* > 20 \). This means that they can be applied to the computational cell with \( y^* > 40 \) \( (y^* > 20 \text{ at the center of the cell}) \) in contact with the condensation surface in CFD analysis, and this method may be applied to practical CV analysis. However, this method could not predict effects of the thin thermal boundary layer thickness expressed by Eq. (7), and gave small \( h_c \) for small \( z \).

4. Discussion

Figs. 6 to 8 showed that the existing \( h_c \) correlations, which are for the heat transfer between the bulk and condensation surface, could be applied to evaluate local \( h_c \) for the heat transfer between the location \( y \) and condensation surface. Therefore, we evaluated the relationship between the local Reynolds number \( Re_y(2y) \) and Sherwood number \( Sh(2y) \) by using the measured temperature distributions and Eqs. (11) and (12). The characteristic length of \( 2y \) was used, where \( 2y = d \) at the pipe center of \( y = d/2 \). The turbulent velocity distribution was used for \( u_k \) in \( Re_y(2y) = u_k(2y)/v_c \). \( \lambda_c \) was calculated from Eq. (12) and the measured temperatures, and \( Sh(2y) = h_c(2y)/\lambda_c \) based on Eq. (11) was obtained. \( Sh \propto Sc^{1/3} \) in Eq. (10) was used because \( Sc \) did not vary in the experiments.

Fig. 9 (a) shows the relationship between \( Sh(2y)/Sc^{1/3} \) and \( Re_y(2y) \) for \( z = 0.14-0.5 \) m, \( y^* > 14 \) and \( Re_y(2y) > 406 \). The standard deviation of the obtained \( Sh(2y) \) from Eq. (10) was \( \sigma = \pm 28.5 \% \) for 388 points. Dispersion was large for Nos. 8 and 9 with small mass flow rates, and \( \sigma = \pm 13.6 \% \) for 332 points except Nos. 8 and 9.
Fig. 9 (b) shows the relationship between Sh(2y)/Sc^{1/3} and Re_g(2y) for z = 0.14-0.5 m except Nos. 8 and 9. The correlation derived from the 332 point data by using the least-square method was Sh(2y)/Sc^{1/3} = 0.0232Re_g^{0.798}, which agreed very well with Eq. (10), Sh/Sc^{1/3} = 0.023Re_g^{0.80}.

Sh(2y) was large for z = 0.05 and 0.09 m due to the small boundary layer thickness for the temperature and steam concentration distributions. Sh(2y)/Sc^{1/3} = 0.084Re_g^{0.8} and σ = ±27.9 % for z = 0.05 m with 106 points, and Sh(2y)/Sc^{1/3} = 0.036Re_g^{0.8} and σ = ±21.5 % for z = 0.09 m with 104 points. For the CFD application, we have to use local physical values at y and y = 0, hence we have no methods to improve the Sh(2y) correlation for small z.

5. Conclusions

The methods to predict the condensation heat transfer coefficient \( h_c \) for CFD analysis were evaluated, where physical quantities in the computational cell in contact with the condensation surface are generally used. First, we applied the correlation based on the gradient of steam concentration at the condensation surface [6] to the measured temperature distributions [12]. This correlation requires a small \( y^* \) value, which is not practical for CV analysis. Therefore, we also applied existing \( h_c \) correlations [13, 14] to a local location \( y \) in the thermal boundary layer. The results we obtained are as follows (applicable conditions of the results are listed in Table 1).

1. \( h_{c,cal} \), which was computed from the gradient of steam concentration \( \Delta X/y \) for the measuring location nearest to the condensation surface \( (y^* = 3.3-18) \), showed a similar trend to the measured \( h_{c,exp} \), but was smaller than \( h_{c,exp} \).

2. \( h_{c,cal}/h_{c,exp} \) based on \( \Delta X/y \) decreased with increasing \( y^* \) in the region of 12 < \( y^* \leq 250 \), and a correlation of the correction factor \( \eta \) for \( h_{c,mod}/h_{c,exp} = 1 \) was derived as a power function of \( y^* \). The standard deviation of \( h_{c,mod} \) in the region of \( y^* > 12 \) for \( h_{c,exp} \) was \( \sigma = \pm 33 \% \) excluding \( h_{c,mod} \) for \( z = 0.05 \) m.

3. When existing \( h_c \) correlations such as the analogy between heat and mass transfer and the diffusion layer model were applied to an arbitrary location \( y \) in the thermal boundary layer, the change of the computed \( h_{c,cal} \) was small in the region of 20 < \( y^* \) < 400. This means that existing \( h_c \) correlations can be applied to the computational cell with \( y^* > 40 \) (\( y^* > 20 \) at the center of the cell) in contact with the condensation surface in CFD analysis.

4. The standard deviation of \( h_{c,cal} \) to \( h_{c,exp} \) was \( \sigma = \pm 35 \% \) for the correlation based on the analogy between heat and mass transfer and was \( \sigma = \pm 36 \% \) for the correlation based on the diffusion layer model. \( \sigma = \pm 35 \% \) and \( \sigma = \pm 36 \% \) (when existing correlations were applied to an arbitrary location \( y \) were close to \( \sigma = \pm 30 \% \) and \( \sigma = \pm 34 \% \) (when existing correlations were applied to the bulk), respectively.

5. The correlation for the local Sherwood number \( Sh(2y) \) with the characteristic length of 2y defined between an arbitrary location \( y \) and condensation surface \( (y = 0) \) was \( Sh(2y) = 0.0232Re_g^{0.798}Sc^{1/3} \) for 400 < \( Re_g < 28400 \), which agreed very well with the existing \( Sh \) correlation defined by the bulk and condensation surface, \( Sh = 0.023Re_g^{0.80}Sc^{1/3} \), except for \( Sh(2y) \) with small mass flow rates and in the region of small \( y \) or small \( z \).

Nomenclature

- \( A \) : flow area \([m^2]\)
- \( c_p \) : specific heat \([kJ/kg K]\)
- \( d \) : diameter of test section \([m]\)
- \( D \) : diffusion coefficient \([m^2/s]\)
- \( g \) : gravitational acceleration \([m/s^2]\)
- \( h_c \) : condensation heat transfer coefficient \([kW/m^2 K]\)
- \( h_{conv} \) : convection heat transfer coefficient \([kW/m^2 K]\)
- \( h_f \) : condensate film heat transfer coefficient \([kW/m^2 K]\)
- \( h_{lg} \) : latent heat of condensation \([kJ/kg]\)
- \( h_{lg}' \) : \( h_{lg} + c_p (T_s - T) \) \([kJ/kg]\)
- \( h_o \) : overall heat transfer coefficient \([kW/m^2 K]\)
- \( M \) : molecular weight \([kg/kmol]\)
- \( m \) : mass flux \([kg/m^2 s]\)
- \( Nu \) : Nusselt number \([-]\)
References

[1] De la Rosa, J. C., Escriva, A., Herranz, L. E., Cicero, T. and Munoz-Cobo, J. L., Review on Condensation on the Containment Structure, Progress in Nuclear Energy, Vol. 51, 32-66 (2009).

[2] Vendel, J., Malet, J. and Bentaib, A., Conclusions of the ISP-47 Containment Thermal-Hydraulics, Proc. the 12th Int. Topical Meeting on Nuclear Thermal-Hydraulics (NURETH-12), Paper No. 031 (2007).

[3] Sibamoto, Y., Ishigaki, M., Abe, S. and Yonomoto, T., Experimental Study on Outer Surface Cooling of Containment Vessel by Using CIGMA, Proc. the 17th Int. Topical Meeting on Nuclear Reactor Thermal Hydraulics (NURETH-17), Paper No. 21591 (2017).

[4] Studer, E., Abdo, D., Benteboula, S., Bernard-Michel, G., Cariteau, B., Coulon, N., Dabbene, F., Debesse, Ph., Koudriakov, S., Ledier, C., Magnaud, J.-P., Norvez, O., Widloecher, J.-L., Beccantini, A., Gounand S. and Brinster, J., Challenges in Containment Thermal Hydraulics, Nuclear Technology, Vol. 206(9), 1361-1373 (2020).

[5] Mimouni, S., Foissac, A. and Lavieville, J., CFD Modeling of Wall Steam Condensation by a Two-Phase Flow Approach, Nuclear Engineering and Design, Vol. 241, 4445-4455 (2011).

[6] Dehbi, A., Janasz, F. and Bell, B., Prediction of Steam Condensation in the Presence of Noncondensable Gases using a CFD-Based Approach, Nuclear Engineering and Design, Vol. 258, 199-210 (2013).

[7] Green, J. and Almenas, K., An Overview of the Primary Parameters and Methods for Determining Condensation Heat Transfer to Containment Structures, Nuclear Safety, Vol. 37(1), 26-48 (1996).

[8] Ambrosini, W., Forgione, N., Merli, F., Oriolo, F., Paci, S., Klijenak, I., Kostka, P., Vyskocil, L., Travis, J. R., Lehmkuhl, J., Kelm, S., Chin, Y.-S. and Bucci, M., Lesson Learned from the
SARNET Wall Condensation Benchmarks, Annals of Nuclear Energy, Vol. 74, 153-164 (2014).

[9] Legay-Desesquelles, F. and Prunet-Foch, B., Heat and Mass Transfer with Condensation in Laminar and Turbulent Boundary Layers along a Flat Plate, Int. J. Heat Mass Transfer, Vol. 29, 95-105 (1986).

[10] Kang, H. C. and Kim, M. H., Characteristics of Film Condensation of Supersaturated Steam-Air Mixture on a Flat Plate, Int. J. Multiphase Flow, Vol. 22, 1601-1618 (1999).

[11] Huang, J., Zhang, J. and Wang, L., Review of Vapor Condensation Heat and Mass Transfer in the Presence of Non-condensable Gas, Applied Thermal Engineering, Vol. 89, 469-484 (2015).

[12] Murase, M., Utanohara, Y., Goda, R., Shimamura, T., Hosokawa, S. and Tomiyama, A., Measurements of Temperature Distributions and Condensation Heat Fluxes for Downward Flows of Steam-Air Mixture in a Circular Pipe, Japanese J. Multiphase Flow, Vol. 33(4), 405-416 (2019).

[13] Araki, H., Kataoka, Y. and Murase, M., Measurement of Condensation Heat Transfer Coefficient inside a Vertical Tube in the Presence of Noncondensable Gas, J. Nuclear Science and Technology, Vol. 32(6), 517-526 (1995).

[14] Liao, Y. and Vierow, K., A Generalized Diffusion Layer Model for Condensation of Vapor with Noncondensible Gases, J. Heat Transfer (ASME), Vol. 129, 988-994 (2007).

[15] Murase, M., Utanohara, Y., Hosokawa, S. and Tomiyama, A., Condensation Heat Transfer for Downward Flows of Steam-Air Mixture in a Circular Pipe, Japanese J. Multiphase Flow, Vol. 34(4), 510-519 (2020).

[16] Nusselt, W., Die Oberflächenkondensation des Wasserdampfes, Z. Ver. Deut. Ing., Vol. 60(27), 541-546 (1916).

[17] Colburn, A. P., A Method of Correlating Forced Convection Heat Transfer Data and a Comparison with Fluid Friction, Trans. AIChE, Vol. 29, 174-210 (1933).

[18] Reynolds, H. C., Swearingen, T. B. and McEligot, D. M., Thermal Energy for Low Reynolds Number Turbulent Flow, J. Basic Engineering, Vol. 91(1), 87-94 (1969).

[19] Kays, M. K. and Crawford, M. E., Heat, Mass and Momentum Transfer, 3rd ed., McGraw-Hill, New York (1993).