Condensation Heat Transfer for Downward Flows of Steam-Air Mixture in a Circular Pipe*

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) and the cooling water in the annulus gap (8.5 mm) outside the pipe, and the temperature gradients in the pipe wall (thickness, 5.5 mm) between the steam-air mixture and cooling water. From the temperatures, we evaluated condensation heat fluxes \( q_c \), and derived an empirical correlation. In this study, we have calculated the heat transfer coefficient of the condensate film \( h_f \) with the Nusselt equation and evaluated the condensation heat transfer coefficient \( h_c \) from \( q_c \) and the measured temperature distributions. We also compared \( h_c \) with \( h_c \) correlations based on the heat and mass transfer analogy and the diffusion layer model. Results from both correlations agreed relatively well with the \( h_c \) data, but the correlations underestimated \( h_c \) for high \( h_c \) near the inlet with a thin thermal boundary layer and overestimated \( h_c \) for low \( h_c \).

Keywords: Condensation, Heat transfer coefficient, Non-condensable gas, Vertical circular pipe

1. Introduction

Our goal in this study was to validate condensation heat transfer models for a computational fluid dynamics (CFD) code and predict pressure and temperature change in the containment vessel (CV) of an actual light water reactor during the course of a loss-of-coolant accident (LOCA) using a CFD code. The structural integrity of CV under LOCA and severe accident (SA) conditions has been evaluated by various thermal-hydraulic codes (see the review by De la Rosa et al. [1]). Condensation in the presence of non-condensable gases is the dominant heat and mass transfer phenomenon in the CV during the course of a LOCA and SA. Green and Almenas [2] reviewed primary parameters determining condensation heat transfer to containment structures, and they showed that previously proposed correlations underestimated or overestimated the local condensation heat transfer coefficients measured in the Heiss-Dampf Reactor (HDR) containment.

There have been various experimental studies using pressure vessels simulating CV [3,4] and numerical studies [5,6] related to thermal hydraulics in CV under accident conditions. Good agreement between numerical results and measured values was obtained for macroscopic behavior in the test vessels, but local condensation heat transfer coefficients were not well validated due to lack of local heat transfer data. In the framework of the SARNET (Severe Accident Research Network of Excellence), benchmarking activities were done for the condensation rates on a vertical flat plate with a height of 2 m in a 0.34 x 0.34 m² square channel to improve CFD models for CV analysis [7]. In these benchmarking activities, good predictive capabilities were achieved for the condensation rates. However

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** 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
the temperature distributions of the steam-air mixture were not reported. In the CFD application to predict thermal-hydraulic phenomena in CV under LOCA and SA conditions, the condensation models should be validated against temperature distributions of the steam and non-condensable gas mixture in the thermal boundary layer. However, there are very few published databases for temperature distributions. In our previous study [8], we therefore measured radial and axial temperature distributions of the steam-air mixture in a simple vertical circular pipe (diameter, 49.5 mm) and the cooling water in the annulus gap outside the pipe, and the temperature gradients in the pipe wall between the mixture and cooling water to provide a database, which can be used to validate the prediction capability for temperature distributions of the steam and non-condensable gas mixture in the thermal boundary layer in CFD models. We evaluated condensation heat fluxes from the measured temperatures and derived an experimental correlation for heat fluxes, but we did not obtain condensation heat transfer coefficients.

In this study, we have evaluated the heat transfer coefficients of the condensate film and the condensation heat transfer coefficients using the previously obtained heat fluxes and measured temperatures [8]. We also compared the condensation heat transfer coefficients with existing correlations based on the heat and mass transfer analogy by Araki et al. [9] and the diffusion layer model by Liao and Vierow [10].

2. Outline of Experimental Method

Many experiments for condensation heat transfer of steam and non-condensable gas mixtures have been done in a circular pipe or on a flat plate. Huang et al. [11] have reviewed empirical correlations and theoretical models for condensation heat transfer. In most of the previous experiments, a one-dimensional temperature distribution along the mixture flow was measured and the proposed correlation or theoretical model was mainly for one-dimensional computation.

2.1 Experimental Apparatus

In our previous study [8], we measured the radial temperature distribution of the steam-air mixture in a simple vertical circular pipe in quasi-steady states to provide a database validating computation capability of a CFD code. The apparatus consisted of a steam supply system, an air supply system, a simple mixing tee for steam and air, an inlet section (the ratio of the length L to the diameter d, L/d = 10) to the test section, a test section, an exhaust pipe, and a cooling water system to feed low-temperature water into the cooling annulus in the test section. The test section consisted of a stainless steel pipe with the diameter d of 49.5 mm and the wall thickness of 5.5 mm, a polycarbonate outer pipe forming a cooling water annulus with the gap of 8.25 mm, and the inlet and outlet plenums for cooling water. The 49.5 mm diameter, which was the same as that by Araki et al. [9], was determined from the capacity of the steam boiler. The length of the cooling region was 0.61 m including the inlet and outlet plenums. The measurement error was ±2.5 % for the air flow rate and ±2 % for the steam flow rate. The pressure and temperature of the inlet steam-air mixture were measured in the inlet section. The measurement error was ±3 kPa for the pressure transducer in the pressure range of 0-0.3 MPa and ±0.5 °C for the T-type sheathed thermocouples used.

Locations of temperature measurement are shown in Fig. 1. A thermocouple array was used to measure radial temperature distributions of the stream-air mixture. Eleven and ten T-type sheathed thermocouples (Ø1.0) were fixed on the support plates at 50 mm and 20 mm upstream from the support plates, respectively. The 1.0 mm diameter was selected to prevent vibration of the thermocouple in the gas flow. Temperature distributions at z = 50, 90, 140, 240, 390 and 500 mm were measured by moving the thermocouple array (in Fig. 1, thermocouple locations for the 50 mm upstream from the support plates are shown). T-type sheathed thermocouples (Ø0.5, flat at the top end) were embedded in the stainless steel pipe at the radial positions of r = 26.25 mm (1.5 mm from the inner surface) and 28.25 mm (2.0 mm from the outer surface) and the axial locations of z = 50, 90, 140, 240, 390 and 500 mm from the top of the cooling region to measure
temperatures in the heat transfer wall. The same kind of T-type sheathed thermocouples were inserted from the outside of the test section at \( z = 50, 90, 140, 240, 390 \) and 500 mm to measure temperatures of the cooling water, and temperatures were measured at \( y = 0, 0.1, 1, 2, 4, \) and 8 mm from the outer surface of the heat transfer pipe by moving the thermocouples.

![Locations of temperature measurement](image)

**Fig. 1** Locations of temperature measurement [8].

| No. | \( W_{s,in} \) [g/s] | \( W_a \) [g/s] | \( x_{a,in} \) [-] | \( T_{g,in} \) [°C] | \( T_{cw,in} \) [°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   | 0.62              | 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         |

Table 1 Experimental conditions.

Pressure in the inlet section \( P_{in} = 0.124-0.127 \) MPa, Cooling water flow rate \( W_{cw} = 56 \) g/s.

2.2 Experimental Conditions

Table 1 lists the main experimental conditions for eleven cases [8]. Our interest was temperature distributions in the thermal boundary layer and the experimental conditions did not simulate thermal-hydraulic features such as velocities and temperatures in CV during LOCA conditions. The parameters were the steam flow rate at the inlet of the test section \( W_{s,in} \) and air flow rate \( W_a \) to change the air mass flow ratio \( x_{a,in} (=W_a/(W_{s,in}+W_a)) \). Nos. 3-5 were the base case and were for confirmation of repeatability. Nos. 1 and 2 were for the effect of small \( x_{a,in} \). Nos. 6-8 were for the effect of flow rates at the same \( x_{a,in} \) and Nos. 9-11 were for the effect of large \( x_{a,in} \) at a similar \( W_{s,in} \).

In the experiments, air at room temperature was mixed with steam to keep the saturated condition in the test section. We obtained \( W_{s,in} \) from the air flow rate \( W_a \), and the pressure \( P_{in} \) and mixture temperature \( T_{g,in} \) measured in the inlet section assuming a Gibbs-Dalton mixture of steam and an ideal non-condensable gas. The air mass flow ratio was in the range of \( x_{a,in} = 0.36-0.92 \) and the mixture temperature in the inlet section was in the range of \( T_{g,in} = 53.5-97.3 \) °C depending on \( x_{a,in} \). The Reynolds number of the mixture in the inlet section was in the range of \( Re_{g,in} = 3600-25000 \) (which was obtained using Eq. (6)). The Reynolds number of cooling water was about 500, which was selected to obtain a detectable temperature rise and avoid local natural circulations. The average heat flux \( q \) in the region of \( z = 0-0.5 \) m depended on the cooling water temperature, and the difference of \( q \) among Nos. 3-5 was relatively large.

2.3 Heat Fluxes

The uncertainty of the condensation heat flux strongly depends on its measurement method. The temperature gradient in the heat transfer wall is widely used to measure the condensation heat flux \( q_w \) (Araki et al. [9] for example). Nagae et al. [12] obtained the condensation heat flux from the enthalpy increasing rate of the cooling water or the enthalpy decreasing rate of the steam-air mixture based on the assumption of saturated conditions \( (q_{cw} \) or \( q_e) \). They used one temperature for the cooling water or the
steam-air mixture at the center in a cross section. We used a radial temperature distribution for the cooling water or the steam-air mixture in a cross section to obtain \( q_{cw} \) or \( q_{g} \). There were differences among the quantities \( q_{cw} \), \( q_{cw} \), and \( q_{g} \). Therefore, the average value of \( q_{cw} \), \( q_{cw} \), and \( q_{g} \) was used:

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

(1)

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 shown in Fig. 1.

3. Heat Transfer Coefficient

3.1 Data Processing

The effects of non-condensable gases on the heat transfer coefficient should be evaluated by considering different heat transfer mechanisms. The relationship between the heat flux \( q_{ave} \) and heat transfer coefficients \( h \) is expressed by:

\[
q_{ave} = h_{w}(T_{b} - T_{w}) = (h_{c} + h_{conv})(T_{b} - T_{f}) = h_{f}(T_{f} - T_{w})
\]

(2)

where \( h_{w} \) is the overall heat transfer coefficient, and \( h_{c} \), \( h_{conv} \), and \( h_{f} \) correspond to the heat transfer coefficients due to condensation of steam, convection of the gas phase and condensate liquid film, respectively. \( T_{b} \), \( T_{f} \) and \( T_{w} \) are temperatures in the bulk, at the condensate film surface and at the wall surface, respectively. The temperature at the center \( T_{c} \) was used for \( T_{b} \) (i.e. \( T_{b} = T_{c} \)). Eq. (2) is simplified to:

\[
1/h_{w} = 1/(h_{c} + h_{conv}) + 1/h_{f}
\]

(3)

which is widely used [9,10] to evaluate \( h_{c} \), \( h_{f} \) and \( T_{f} \) were obtained by using Nusselt’s model [13] expressed by:

\[
h_{f} = \lambda_{f} / \delta_{L} \quad \delta_{L} = (\gamma/\mu g)^{1/3} \quad T_{f} = T_{w} + q_{ave} \lambda_{f}
\]

(4)

where \( \Gamma \) is the liquid film flow rate per perimeter, \( \delta_{L} \) is the liquid film thickness, \( \lambda_{f} \) is the thermal conductivity of liquid, \( \mu_{g} \) is the liquid viscosity, and \( \gamma \) is the kinematic viscosity. The obtained liquid film thickness was less than 0.102 mm, and the Reynolds number (= \( \Gamma/\mu g \)) was in the range of 0.11-12.3. \( h_{conv} \) was estimated by using the Colburn formula [14] for fully-developed turbulent flow, which was modified for the effect of developing region of the thermal boundary layer \( Nu_{c}/Nu_{w} \) as:

\[
Nu = h_{conv} d/\lambda_{g} = 0.023 Re_{g}^{3/4} Pr_{g}^{1/3}(Nu_{c}/Nu_{w})^{3/4}
\]

(5)

for 2300 < \( Re_{g} < 10^{6} \)

where \( Nu \) is the Nusselt number, \( Pr \) is the Prandtl number, \( Re_{g} \) is the Reynolds number for the steam-air mixture, and \( \lambda_{g} \) is the thermal conductivity of gas. \( Re_{g} \) was evaluated using

\[
Re_{g} = ((W_{a} + W_{b})/(\rho \mu A))(\rho \mu d/\mu_{g})
\]

(6)

where \( A \) is the flow area, and \( \mu \) and \( \rho \) are the viscosity and density of the steam-air mixture, respectively. The correlation for \( Nu_{c}/Nu_{w} \) by Reynolds et al. [15] was used.

\[
Nu_{c}/Nu_{w} = 1+C(z/d), C = 0.8(1+70000/Re_{g}^{3/4})
\]

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

(7)

In Eqs. (2)-(7), the values of \( q_{ave} \), \( Re_{g} \), \( T_{b} \), \( T_{w} \) and \( \Gamma \) were taken from our previous study [8].

3.2 Temperature Distributions of Steam-Air Mixture

In our previous study [8], the dimensionless temperature was defined for the difference between the center and wall surface temperatures \((T_{c} - T_{w})\) because we did not obtain the liquid film surface temperature \( T_{f} \). This time, we obtained \( T_{f} \) from Eq. (4) and we redefined the dimensionless temperature for the difference between the center and liquid film surface temperatures \((T_{c} - T_{f})\). The thickness of the thermal boundary layer \( \delta_{T} \) was defined by the location at \( \delta_{T}/(T_{c} - T_{f}) = 0.99 \) as:

\[
\delta_{T} = \gamma/dT_{f}=0.99 \quad dT_{f} = (T_{c} - T_{f})/(T_{c} - T_{f})
\]

(8)

where \( T_{c} \) is the temperature of steam-air mixture. Fig. 2 shows the thermal boundary layer thicknesses \( \delta_{T}, \delta_{f} \) became thinner with decreasing air mass flow ratio \( x_{a,in} \) and increasing heat flux \( q \) (Table 1), due to the suction effect [6,16].

Fig. 3 shows dimensionless temperature distributions as a function of \((\gamma/\delta_{T})\). These distributions were expressed by a power function as:
where the constant $a$ and the exponent $n$ were obtained by the least square method. For Nos. 2, 3 and 6, dimensionless temperature distributions were similar, and the uncertainty was 0.015 for $a = 0.995$ and $n = 0.057$. For No. 1, $n$ was smaller than 0.057. For Nos. 9-11, dimensionless temperature distributions were similar, and $a = 1.000$ and $n = 0.083$. This indicated that $n$ is smaller as the suction effect increases. However, $n = 0.083$, 0.072, 0.070 and 0.125 for Nos. 4, 5, 7 and 8, respectively. For Nos. 4, 5 and 8, dispersion of temperature data was large. In No. 8 with $n = 0.125$, the heat flux was small and the total mass flow rate was the smallest (Table 1).

3.3 Condensation Heat Transfer Coefficient

Fig. 4 shows the condensation heat transfer coefficients $h_c$, which are the average values in the region between thermocouple locations. $h_c$ rapidly decreased with increasing $z$, but $h_c$ was almost constant in the region of $z > 0.1$ m in the case of high heat fluxes, Nos. 1-6. The average heat flux in No. 5 was the lowest among Nos. 1-6 due to the high cooling water temperature, but $h_c$ in Nos. 3-5 was large due to their large total mass flow rate. $h_c$ in Nos. 8-11 was small due to the low heat fluxes.

The liquid film heat transfer coefficient was in the range of $h_f = 6.3-22.2$ kW/m²K, and its effect on $h_i$ could not be neglected for $h_i > 1$ kW/m²K. The convection heat transfer coefficient was in the range of $h_{conv} = 0.008-0.097$ kW/m²K, and its effect on $h_i$ could not be neglected for the cases of low heat flux, Nos. 8-11.

3.4 Comparison of Condensation Heat Transfer Coefficients

Araki et al. [9] evaluated local condensation heat fluxes from the measured temperature gradient in the heat transfer pipe with a test section similar to that of our study. Their experimental conditions are listed in Table 2. The test section was longer, the steam flow rate was larger and the air flow rate was smaller in their experiments than they were in our experiments. They proposed an empirical correlation:

$$h_c \text{ [kW/m}^2\text{K]} = (2.11/10^4) \text{ Re}_g^{0.8} (P_t/P_a)^{0.99}$$

for $2300 < \text{ Re}_g < 21000$  (10)

where $P_t$ and $P_a$ are the total pressure and air partial pressure, respectively. In a preliminary comparison between Eq. (10) and measured $h_c$, we confirmed the followings. For Nos. 1-6 with large heat fluxes, Eq. (10) underestimated $h_c$ near the test section inlet. For Nos. 8-11 with small heat fluxes, on the other hand, Eq. (10) overestimated $h_c$ in the downstream region.

Uncertainty became large when an empirical correlation was used beyond the database for the correlation.

Araki et al. [9] also proposed a correlation based on the heat and mass transfer analogy:

$$h_c = m_c \text{ h}_{fg} I(T_b - T_i), \quad (11)$$

$$m_c = (D \rho M_c / R T_{ave} d) 2 \ln(P_{c,i}/P_{c,b})/ln(1 - 2/Sh), \quad T_{ave} = (T_b - T_i)/2,$$

where $D$ is the diffusion coefficient, $h_{fg}$ is the latent heat of condensation, $M_i$ is the molecular weight of steam, $m_i$ is the mass flux of steam, $R$ is the universal gas constant, and $Sh$ is the Sherwood number. The subscripts $b$ and $i$ denote the bulk and the gas-liquid interface, respectively. Eq. (11) is for fully-developed flow. Hence we modified the correlation for $Sh$ similar to Eqs. (5) and (7) as:

$$Sh = 0.023 \text{ Re}_g^{0.8} \text{ Sc}^{1/3} (Sh_{i}/Sh_{b}), \quad \text{for } 2300 < \text{ Re}_g < 10^7$$

$$Sh_i/\text{Sh}_{b} = \text{Nu}_{i}/\text{Nu}_{b} = 1+C/(z/d),$$

$$C = 0.8\{1+70000/\text{Re}_g^{3/2}\} \quad \text{for } 3000 < \text{ Re}_g < 500000,$$

where Sc is the Schmidt number.

Fig. 5 compares $h_{c,cal}$ calculated by using Eq. (11) of Araki et al. [9] and Eq. (12) with our data $h_{c,exp}$. Eq. (11) with Eq. (12) gave relatively good agreement with $h_{c,exp}$, Eqs. (11) and (12) underestimated $h_c$ near the test section inlet, and overestimated $h_c$ downstream for Nos. 8 and 9 with small total flow rates. Fig. 5 confirms that the heat and mass transfer analogy can be applied to a wide range of flow conditions including data by Araki et al. [9] and ours. The standard deviation of Eq. (11) from the experimental data was ± 30 %.
where the constant $a$ and the exponent $n$ were obtained by the least square method with the experimental data by Araki et al. [9] and ours. The standard deviation of Eq. (11) from the experimental data was ±30%.

Fig. 2 Thermal boundary layer thicknesses.

Fig. 3 Dimensionless temperature distributions.

Fig. 4 Condensation heat transfer coefficients $h_c$. 

$\frac{(T_g - T_f)}{(T_c - T_f)} = a \left( \frac{y}{\delta T} \right)^n$
In recent years, the diffusion layer model has been widely used for predicting condensation heat transfer coefficients in the presence of non-condensable gases. Therefore, we compared our data with the diffusion layer model by Liao and Vierow [10] given by:

\[
h_c = \frac{Sh (\lambda_c/d)}{1 + B_m},
\]

(13)

\[
\lambda_c = \left( \rho_n \beta_n \rho_f \beta_f \rho_g \beta_g \frac{h_{fg}}{h_{fg} + \rho_c \rho_f c_p (T_g - T_i)} \right) \ln \left( 1 + B_m \right)/B_m,
\]

\[
\phi_1 = (X_m / X_{ad}) \ln (1 + B_m)/B_m,
\]

\[
\phi_2 = M_s^2/(M_g b M_e)
\]

where \( B_m \) is the suction factor [17], \( h_{fg} = h_{fg} + \rho_c \rho_f c_p (T_g - T_i) \), \( X_m / X_{ad} \) is the log mean steam mass fraction, and \( \lambda_c \) is the condensation thermal conductivity. For \( Sh \) in Eq. (13), Eq. (12) was used.

Fig. 6 compares \( h_{c,cal} \) calculated by using Eq. (13) of Liao and Vierow [10] and Eq. (12) with our data \( h_{c,exp} \). The standard deviation of Eq. (13) from the experimental data was ± 34%.

Figs. 5 and 6 were similar and it was difficult to find the difference between the plotted lines for Eqs. (11) and (13) in the graphs. The average of \( h_{c,cal} \) values calculated using Eqs. (13) and (12) was 2.3% larger than the average of measured \( h_{c,exp} \) values and the average of \( h_{c,cal} \) values calculated using Eqs. (11) and (12) was 7.1% smaller than the average of measured \( h_{c,exp} \) values. On average, Eq. (13) gave good agreement with the data except for the underestimation near the test section inlet and the overestimation downstream for Nos. 8 and 9.

Lee et al. [18] proposed a modified diffusion layer model with an effective diffusion coefficient combining a molecular diffusion coefficient and the turbulent diffusion coefficient. They validated the modified diffusion layer model by using 157 condensation experiments from six different facilities and found good agreement with data. In this study, we did not use their model because it is difficult to obtain the turbulent diffusion coefficient.

4. Discussion

The aim of our experiments was to present a database for validation of condensation heat transfer models in a CFD code and confirm its capability to
predict temperature distributions in the thermal boundary layer like those shown in Fig. 3. In CFD codes, a condensation heat transfer correlation based on the heat and mass transfer analogy or the diffusion layer model is used. The heat and mass transfer analogy and the diffusion layer model are both based on a one-dimensional model and the condensation heat transfer coefficient is defined by heat transfer between the bulk and condensation surface. It is difficult for CFD users to define the bulk and the characteristic length in the Sherwood number for the CV analysis.

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

\[ q = m_x h_{i,s} = \frac{1}{\rho_d D} \left( \frac{\partial X_s}{\partial n} \right) \]  \hspace{1cm} (14)

where \( m_x \) is the mass flux of steam, \( n \) is the distance from the condensation surface, \( X_s \) is the steam mass fraction, and \( \rho_d \) is the density of the steam-air mixture. This model is suitable for a CFD application, because values in the cell contacting with the condensation surface can be used for \( X_s \) and \( \rho_d \). Fig. 7 shows heat fluxes calculated using Eq. (14) at \( z = 240 \) mm for Nos. 1, 8 and 11. The calculated heat fluxes depend on the distance from the condensation surface \( y \), and approach the measured heat flux \( q_{exp} \) with decreasing \( y \). The gradient of the steam mass fraction \( \left( \frac{\partial X_s}{\partial n} \right) \) close to the condensation surface should be used in Eq. (14). Hence Dehbi et al. [6] used a small cell with \( y^* = 0.47 \).

However, the small cell with \( y^* = 0.47 \) is not practical for the CV analysis because the small cell size requires many computational cells.

As described above, a reasonable model for the condensation heat transfer in the presence of non-condensable gases, which can be practically applied to the CV analysis, has not been proposed, and it is an important technical issue.

5. Conclusions

In this study, we evaluated the heat transfer coefficients of the condensate film \( h_f \) and the condensation heat transfer coefficients \( h_c \) for downward flows of steam-air mixture from the previously obtained heat fluxes and measured temperatures (Murase et al. [8]) in a vertical pipe of diameter \( d = 49.5 \) mm and 0.61 m long. We also compared \( h_c \) with correlations based on the heat and mass transfer analogy by Araki et al. [9] and the diffusion layer model by Liao and Vierow [10]. The results we obtained are as follows.

1. The thermal boundary layer thickness decreased with increasing condensation heat flux, and exponent fitting to the dimensionless temperature distributions of the steam-air mixture in the turbulent boundary layer decreased with increasing condensation heat flux due to the suction effect.

2. The measured \( h_c \) rapidly decreased with increasing \( z \) (distance from the test section inlet), but \( h_c \) was almost constant in the region of \( z > 0.1 \) m \((z/d > 2)\) in the case of the high heat fluxes of 39.9-47.6 kW/m².

3. Both the diffusion layer model and the heat and mass transfer analogy, which were modified for the effect of the developing region of the thermal boundary layer, gave good agreement with the data except for the underestimation near the test section inlet and the overestimation in the downstream for low \( q_e \) and small total mass flow rates. The average of \( h_{c,cal} \) values calculated using the diffusion layer model or the heat and mass transfer analogy was 2.3 % larger.
or 7.1 % smaller than the average of measured $h_{c, \text{exp}}$ values.

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**Nomenclature**

- **A**: flow area [m$^2$]
- **a**: constant [-]
- **$B_m$**: suction factor [-]
- **$c_p$**: specific heat [kJ/kg K]
- **d**: diameter of test section [m]
- **$D$**: diffusion coefficient [m$^2$/s]
- **$dT^*$**: dimensionless temperature difference [-]
- **g**: gravitational acceleration [m/s$^2$]
- **H**: height of test section [m]
- **$h_c$**: condensation heat transfer coefficient [kW/m$^2$K]
- **$h_{\text{conv}}$**: 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]
- **$h_{lg}'$**: $= h_{lg} + c_p(T_b - T_i)$ [kJ/kg]
- **$h_t$**: overall heat transfer coefficient [kW/m$^2$K]
- **M**: molecular weight [kg/mol]
- **m**: mass flux [kg/m$^2$s]
- **Nu**: Nusselt number [-]
- **n**: exponent number [-]
- **P**: pressure [Pa]
- **Pr**: Prandtl number [-]
- **q**: heat flux [kW/m$^2$]
- **R**: radius [m]
- **Re**: Reynolds number [-]
- **r**: radial coordinate [m]
- **Sc**: Schmidt number [-]
- **Sh**: Sherwood number [-]
- **T**: temperature [K]
- **$t_w$**: wall thickness [m]
- **v**: velocity [m/s]
- **W**: mass flow rate [kg/s]
- **X**: mass fraction [-]

**Greek letters**

- **$\Gamma$**: liquid film flow rate per perimeter [kg/s m]
- **$\delta_f$**: liquid film thickness [m]
- **$\delta_T$**: thermal boundary layer thickness [m]
- **$\lambda$**: thermal conductivity [kW/m K]
- **$\lambda_c$**: condensation thermal conductivity [kW/m K]
- **$\mu$**: viscosity [Pa s]
- **$\nu$**: kinematic viscosity [m$^2$/s]
- **$\rho$**: density [kg/m$^3$]

**Subscripts**

- **a**: air
- **ave**: average
- **b**: bulk
- **c**: center or condensation
- **cal**: calculation
- **conv**: convection
- **cw**: cooling water
- **exp**: experiment
- **f**: liquid film
- **g**: gas phase (steam-air mixture)
- **i**: gas-liquid interface
- **in**: inlet
- **j**: number of axial location
- **m**: measured or mean
- **s**: steam
- **t**: total
- **w**: wall
- **z**: z-direction

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