Measurements of Temperature Distributions and Condensation Heat Fluxes for Downward Flows of Steam-Air Mixture in a Circular Pipe*

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Abstract Three quantities were measured to provide a database for validation of computations with a CFD (computational fluid dynamics) code. The quantities were the radial temperature distributions of steam-air mixture in a vertical circular pipe (diameter, 49.5 mm), the temperature gradients in the pipe wall (thickness, 5.5 mm) and the radial temperature distributions of the cooling water in the annulus gap (8.25 mm) outside the pipe. From these temperature distributions, three kinds of condensation heat fluxes were obtained from the enthalpy decreasing rate of the steam-air mixture based on the assumption of saturated conditions, the temperature gradient in the pipe wall, and the enthalpy increasing rate of the cooling water. These three heat fluxes were different especially in the downstream region, where the heat flux was low, so that the average of the three heat fluxes was used to evaluate factors affecting the condensation heat transfer. This average was expressed by a function of the steam density difference between the main flow of the mixture and the mixture on the wall surface, the average steam velocity and the thermal boundary layer thickness. The suction effect due to condensation was also discussed.

Keywords: condensation, non-condensable gas, circular pipe, temperature distribution, heat flux

1. Introduction

Condensation in the presence of non-condensable gases is the dominant heat and mass transfer phenomenon in the containment vessel (CV) of a light water reactor during the course of a loss-of-coolant accident (LOCA). The structural integrity of the 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]); many of the codes employ a lumped parameter model, and an empirical correlation or the heat and mass transfer analogy is used for the condensation heat transfer in them. Green and Almenas [2] reviewed primary parameters for 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 with a volume of 11060 m$^3$ and a height of 60 m depending on fluid conditions such as local velocity and steam concentration. These discrepancies were mainly due to the difference between applicable conditions of correlations and local conditions in data.

Lumped parameter codes, which have been used to predict thermal-hydraulic phenomena in the CV, have the merit to provide a picture of the expected overall plant behavior with a reasonably limited computational effort. On the other hand, the developments of computational machines and techniques make it possible to address even complicated geometries by use of computational fluid dynamics (CFD) codes. Mimouni et al. [3] and Dehbi...
et al. [4] proposed a CFD model for the steam condensation in the presence of non-condensable gas, and both groups conducted a numerical simulation of TOSQAN experiments [5], where steam was injected into a closed cylindrical vessel (diameter, 1.5 m; height, 4.8 m) through a vertical pipe (diameter, 0.041 m) located centrally in the lower section of the vessel. Shibamoto et al. [6] carried out experiments similar to the TOSQAN experiments by using a test vessel called CIGMA to understand thermal hydraulic phenomena during severe accidents, and Ishigaki et al. [7] conducted a numerical simulation of the experiments taking condensation on the test vessel wall into account. All of them [3,4,7] obtained good agreement between computed results and measured values, but local condensation heat transfer coefficients were not well validated due to lack of local heat transfer data. In the framework of SARNET (Severe Accident Research Network of Excellence), benchmarking activities were done in four steps for the condensation rates on a vertical flat plate with a height of 2 m in a 0.34 x 0.34 m square channel to develop updated CFD models for reactor containment analysis [8]. In these benchmarking activities, CFD codes such as CFX and FLUENT were used, the bulk condensation in the steam-air mixture and the wall condensation on the vertical flat plate were addressed, and good predictive capabilities were achieved for the condensation rates. However, the temperature distributions of the steam-air mixture, which are important for validation of the bulk condensation model, were not reported. Kang and Kim [9] measured temperature and velocity profiles of a steam-air mixture flow over a nearly horizontal flat plate, and they did numerical simulations for the flow. They indicated that the gas mixtures became supersaturated near the interface, the gas mixture temperature profile was greatly concaved toward the interface, and the equilibrium model (where the phase changes at saturation conditions) underestimated the mixture temperature and the heat transfer coefficient, when the temperature difference between the saturated steam-air mixture and the cold wall was large. In the CFD application to predict thermal-hydraulic phenomena in the CV under LOCA and SA conditions, the bulk condensation in the steam and non-condensable gas mixture and the wall condensation on structural walls should be addressed and 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 of the steam and non-condensable gas mixture in the thermal boundary layer.

Utanohara et al. [10] did a numerical simulation of flow and temperature fields in the CV of a three-loop PWR (pressurized water reactor) during operation by using the CFD code FLUENT 13; this could be used as the initial conditions for a numerical simulation of thermal-hydraulic phenomena in the CV under LOCA conditions. Utanohara et al. [11] also did a numerical simulation for the experiment [12] of steam injection into a small-scale closed cylindrical vessel filled with air simulating a LOCA by using the CFD code FLUENT 15.0 to make a CFD model of the thermal flow field in the test vessel. This steam injection experiment, however, is not suitable for validation of condensation models because there are many influencing parameters such as temperature rise due to compression, condensation in the mixing boundary layer of the steam jet and ambient air, and a complex distribution of heat transfer coefficients on the vessel wall along the jet impingement and wall jet.

The objective of this study was 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. For this purpose, we measured radial temperature distributions of the steam-air mixture in a simple vertical circular pipe, the temperature gradients in the pipe wall and radial temperature distributions of the cooling water in the annulus gap outside the pipe (the geometry of the test section and test conditions did not simulate thermal-hydraulic features in the CV). From these temperature distributions, we obtained three kind of condensation heat fluxes from the enthalpy decreasing rate of the steam-air mixture based on
the assumption of saturated conditions, the temperature gradient in the pipe wall, and the enthalpy increasing rate of the cooling water. We also evaluated primary parameters affecting the condensation heat fluxes.

2. Experimental Method

Many experiments have been done to measure condensation heat transfer of steam and non-condensable gas mixtures in a circular pipe or a flat plate, and Huang et al. [13] reviewed empirical correlations and theoretical models proposed for condensation heat transfer in the presence of non-condensable gases. In most of the previous studies, one-dimensional temperature distributions along the mixture flow were measured and the proposed correlations or theoretical models were mainly for one-dimensional computations. In this study, therefore, we measured radial temperature distributions of a steam-air mixture in a simple vertical circular pipe to provide a database for validation of computations with a CFD code.

2.1 Experimental Apparatus

The schematic diagram of the experimental apparatus is shown in Fig. 1. The apparatus consisted of a test section, a steam boiler (Miura, ME-20), a steam regulator (TLV, S-COS-16), a steam heater (Nippon Heater, SGA, 2 kW), an air compressor (ANEST IWATA, SLP-1501EB), an air drier (ANEST IWATA, RDG-150C), an air heater (Nippon Heater, SGA-3425, 4 kW), a steam-air mixer (a simple mixing tee), an inlet section (L/D = 10) to the test section, an exhaust pipe, and a cooling water system to feed low-temperature water into the cooling annulus in the test section. The pipes for air after the air heater, steam and steam-air mixture were kept warm with low-power heaters. The steam-air mixture from the test section was exhausted to the atmosphere.

The measurement error was ±2.5 % for the air flow meter (Nippon Flowcell, FLT-N) and ±2 % for the steam flow meter (Nippon Flowcell, JBH-5-FD). 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 (Nagano Keiki, KH15-M33) in the pressure range of 0-0.3 MPa and ±0.5 ℃ for the T-type sheathed thermocouple (Okazaki MFG, T35, class 1, grounded).

The test section is shown in Fig. 2. The test section consisted of a stainless steel pipe with the diameter 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 length of the cooling region was 0.61 m including the inlet and outlet plenums.

Locations of temperature measurement are shown in Fig. 3. T-type sheathed thermocouples (φ 0.5, 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...
2.1 Experimental Apparatus

Fig. 1  Schematic diagram of experimental apparatus.

Photo 1  Thermocouple (TC) array.

Fig. 2  Test section.

Fig. 3  Locations of temperature measurement: Gas (U) and (L), thermocouples at 50 mm and 20 mm upstream of the support plates, respectively (see Photo 1).

(2.0 mm from the outer surface) and the axial locations of \( z = 10, 50, 70, 90, 140, 240, 390 \) 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 (\( \phi 0.5 \), flat at the top end) were inserted from the outside of the test section at \( z = 50, 70, 90, 140, 240, 390 \) and 500 mm to measure temperatures of the cooling water, and temperatures were measured at \( r = 30.25, 30.35, 31.25, 32.25, 34.25 \) and 38.25 mm \( ( y = 0, 0.1, 1, 2, 4, 8 \) mm from the outer surface of the heat transfer pipe) by moving the thermocouples.

2.2 Temperature Measurement of Mixture

The thermocouple array shown in Photo 1 was used to measure temperature distributions of the stream-air mixture. Eleven and ten T-type sheathed thermocouples (\( \phi 1.0 \)) were fixed on the support plates at 50 mm and 20 mm upstream from the support plates, respectively. The thermocouple array was inserted in the test section and was supported at the bottom flange with the O-ring seal. Temperatures at \( z = 50, 70, 90, 140, 240, 390 \) and 500 mm were measured by moving the thermocouple array. The radial locations of thermocouples were \( r = -23, -21.3, -18, -14, -10, 0, 11.5, 16, 18.5, 21.5 \) and 24 mm for the 50 mm upstream position and \( r = -22.5, -21, -18, -14, -10, 12, 16, 19, 21 \) and 24 mm for the 20 mm upstream position. These thermocouple locations were determined from photographs with a scale.

2.3 Experimental Conditions

Table 1 lists the main experimental conditions for eleven cases. The experimental conditions did not simulate thermal-hydraulic features such as velocities and temperatures during LOCA conditions in the CV. Our interest was temperature distributions in the thermal boundary layer. The parameters were the steam flow rate \( W_s \) and air flow rate \( W_o \) to change the air mass flow ratio \( x_{a,in} \). 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 \( x_{a,in} \). Nos. 9-11 were for the effect of large \( x_{a,in} \).

After mixing steam and air, the steam partial pressure decreases and the saturated temperature
Table 1: Experimental conditions.

| No. | $W_{s,s}$ [g/s] | $W_{s,in}$ [g/s] | $W_a$ [g/s] | $x_{a,in}$ [-] | $T_{g,in}$ [°C] | $T_{cw,in}$ [°C] |
|-----|-----------------|-----------------|------------|---------------|-----------------|-----------------|
| 1   | 3.98            | 3.58            | 2.0        | 0.36          | 97.3            | 18.5            |
| 2   | 4.4             | 4.03            | 4.5        | 0.53          | 91.5            | 12.5            |
| 3   | 5.97            | 5.77            | 9.0        | 0.61          | 87.3            | 12.7            |
| 4   | 5.94            | 5.91            | 9.0        | 0.60          | 87.7            | 22.5            |
| 5   | 5.86            | 5.86            | 9.0        | 0.61          | 87.5            | 28.5            |
| 6   | 4.14            | 3.94            | 6.0        | 0.60          | 87.7            | 12.7            |
| 7   | 2.0             | 1.91            | 3.0        | 0.61          | 87.4            | 12.7            |
| 8   | 1.07            | 0.84            | 1.5        | 0.64          | 85.9            | 11.2            |
| 9   | 1.07            | 0.82            | 3.0        | 0.79          | 74.6            | 11.0            |
| 10  | 1.09            | 0.79            | 6.0        | 0.88          | 61.8            | 11.2            |
| 11  | 0.94            | 0.74            | 9.0        | 0.92          | 53.5            | 11.2            |

Pressure in the inlet section $P_{in} = 0.124-0.127$ MPa, cooling water flow rate $W_{cw} = 56$ g/s, $W_{s,s}$, supplied steam flow rate; $W_{s,in}$, steam flow rate in the inlet section; $W_a$, air flow rate; $x_{a,in}$, air mass flow ratio ($=W_a/(W_{s,in}+W_a)$); $T_{g,in}$, mixture temperature in the inlet section; $T_{cw,in}$, cooling water inlet temperature.

Pressure in the test section $P_{out} = 0.114-0.127$ MPa, cooling water flow rate $W_{cw} = 56$ g/s. $W_{s,th}$, supplied steam flow rate; $W_{s,in}$, steam flow rate in the test section; $W_a$, air flow rate; $x_{a,in}$, air mass flow ratio ($=W_a/(W_{s,in}+W_a)$); $T_{g,th}$, mixture temperature in the test section; $T_{cw,th}$, cooling water inlet temperature.

Temperature $T_{cw}$ was different depending on the season during which the experiments were done. The Reynolds number of cooling water was about 500, which was selected to obtain a detectable temperature rise and avoid local natural circulations.

3. Temperature Distributions

3.1 Radial Temperature Distributions

Examples of radial temperature distributions are shown in Fig. 4. We assumed circumferentially symmetrical temperature distributions because clear differences were not found between radially plus and minus regions in Fig. 3 (this indicated good mixing of steam and air up to the test section). In No. 4, the condensation heat transfer was high and radial temperature differences were large in the wall and the cooling water. In No. 11, the condensation heat transfer was low, radial temperature differences were small in the wall and the cooling water, and the temperature increase of the cooling water along the flow direction was small.

3.2 Thermal Boundary Layer Thickness

The thermal boundary layer thickness is shown in Fig. 5. The thermal boundary thickness $\delta_T$ is defined by the radial location $y$ from the cold wall surface at which the dimensionless temperature difference $d\tau^*$ is 0.99:

$$\delta_T = y|_{d\tau^*=0.99}$$

$$d\tau^* = (T_c-T_w)/(T_c-T_{cw})$$

where $T_c$ and $T_w$ are temperatures at the center of the pipe and on the cold wall surface, respectively. For Nos. 3-5 in Table 1, the thermal boundary layer thickness was expressed by $\delta_T/R = 0.20 (z/D)^{0.40} \pm 0.03$, where the uncertainty of $\pm0.03$ was difference among Nos. 3-5. The $\delta_T/R$ for Nos. 2, 6 and 7 was within the uncertainty of Nos. 3-5, but the $\delta_T/R$ for No. 1 was thinner than that for Nos. 3-5. In No. 1, the air mass flow ratio $x_{a,in}$ was lowest in the experiments, and condensation heat fluxes were high. The $\delta_T/R$ for Nos. 8-11, where the steam flow rate $W_{s,in}$ was low and condensation heat fluxes were low, was thicker than that for Nos. 3-5.
Table 1  Experimental conditions.

| No. | $W_{s,s}$ [g/s] | $W_{s,in}$ [g/s] | $W_a$ [g/s] | $x_{a,in}$ [-] | $T_{g,in}$ [°C] |
|-----|-----------------|-----------------|-------------|----------------|----------------|
| 1   | 0.2             | 0.2             | 0.2         | 0.2            | 61.8           |
| 2   | 0.4             | 0.4             | 0.4         | 0.4            | 61.8           |
| 3   | 0.6             | 0.6             | 0.6         | 0.6            | 61.8           |

Fig. 4  Radial temperature distributions (Nos. 4 and 11: see Table 1).

Fig. 5  Thermal boundary layer thickness (Nos. 1-11: see Table 1).

Fig. 6  Dimensionless temperature distributions (Nos. 3-5).
Plotted values of Fig. 5 showed that the \( \delta T/R \) became thinner with increasing condensation heat fluxes. This is the well-known “suction effect” [4,15] in which the absorption of steam into the condensate liquid film causes shrinkage of the boundary layer.

### 3.3 Temperatures in Boundary Layer

Dimensionless temperature distributions in the thermal boundary layer for Nos. 3-5 are shown in Fig. 6. The maximum uncertainties of \( d(y/\delta_t) = \pm 0.1 \) and \( d(dT) = \pm 0.006 \) corresponded to uncertainties of \( dy = \pm 0.5 \) mm in the thermocouple locations and \( dT = \pm 0.5 \) °C in the temperature measurement, respectively.

Temperatures in Fig. 6 were clearly divided into two regions, where temperatures were expressed by a linear function or an exponent function. For the temperatures in the region expressed by a linear function, there were not enough data points and the uncertainties for the measurement locations were large, and we could not discuss this in detail. Most of the temperatures were within the maximum uncertainties. Two low temperature points in No. 5 (in the broken line) might be due to wetting of the thermocouples. In the turbulent boundary layer expressed by an exponent function, the temperature gradient was relatively low. It might be due to the suction effect.

Fig. 7 shows effects of experimental conditions on the dimensionless temperature distribution. Temperature distributions for Nos. 7 and 8 are not shown but they were similar to those for Nos. 6 and 9, respectively. Temperatures for Nos. 1, 2 and 6 were higher than those for Nos. 3-5. Four high temperature points in No. 1 (in the broken line) might be due to a local superheated state. Temperatures for Nos. 9-11 were lower than those for Nos. 3-5. Results in Fig. 7 showed that the suction effect affected not only the boundary layer thickness but also the temperature distribution in the boundary layer.

### 4. Condensation Heat Flux

#### 4.1 Evaluation Method

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 \( q_w \) [16]:

\[
q_w = \lambda_w \frac{(T_{r1} - T_{r2})}{\ln(r_2/r_1)}
\]

where \( q_w \) [kW/m²] is defined on the inner surface of the wall \((r = r_{cin})\), \( r_1 \) and \( r_2 \) are thermocouple locations, and \( \lambda_w \) [kW/mK] is the thermal conductivity of the wall.

Nagae et al. [14] 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. They used one temperature for the cooling water or the steam-air mixture at the center in a cross section. In this study, we used a radial temperature distribution for the cooling water or the steam-air.
mixture in a cross section. The condensation heat flux \( q_{cw} \) from the enthalpy increasing rate of the cooling water is expressed by:

\[
q_{cw} = -\frac{dQ_{cw}}{dz}/2\pi r_{w,in} \quad \text{and}
\]

\[
Q_{cw} = \sum c_{p,cw}(v_{cw,l}/v_{cw,ave}) W_{cw,Tcw,l} \times \pi (r_{i+1}^2 - r_i^2)/A_{cw},
\]

where \( A_{cw} [m^2] \) is the flow area, \( c_{p,cw} [kJ/kg \, K] \) is the specific heat, \( Q_{cw} [kW] \) is the thermal energy in a horizontal cross section transferred by the cooling water, \( v_{cw,l} [m/s] \) is the local velocity, and \( v_{cw,ave} [m/s] \) is the average velocity. The radial location \( r_i \) was defined by the center of the neighboring temperature measurement locations. The velocity distribution of a laminar flow was used for \( v_{cw,l} \).

Applying Eq. (1) to obtain the local steam mass flux, we express the condensation heat flux \( q_{g} \) from the enthalpy decreasing rate of the steam-air mixture by:

\[
q_{g} = \frac{dQ_{g}}{dz}/2\pi r_{w,in} \quad \text{and}
\]

\[
Q_{g} = \sum [M_w P_{g,i} h_a(T_{g,i})/(M_w (P_{w,i} - P_{g,i}) + h_a(T_{g,i}))]
\times (v_{g,l}/v_{g,ave}) W_{g} \{\pi (r_{i+1}^2 - r_i^2)/A_g\},
\]

where \( A_{g} [m^2] \) is the flow area, \( h_a(T_{g,i}) \) and \( h(T_{g,i}) [kJ/kg] \) are the enthalpy of air and the saturated steam at the temperature \( T_{g,i} \) respectively, \( Q_{g} [kW] \) is the thermal energy in a horizontal cross section transferred by the steam-air mixture, \( v_{g,l} [m/s] \) is the local velocity, and \( v_{g,ave} [m/s] \) is the average velocity. The velocity distribution of the mixture, which was obtained from a three-dimensional computation with the CFD code FLUENT 17 and standard k-\( \varepsilon \) turbulence model, was used for \( v_{g,l} \).

Fig. 8 compares heat fluxes obtained from the temperature gradient in the pipe wall \( q_{w} \), the enthalpy increasing rate of the cooling water \( q_{cw} \), and the enthalpy decreasing rate of the steam-air mixture \( q_{g} \). \( q_{cw} \) and \( q_{g} \) are shown by horizontal lines because they are average values in the region of two neighboring thermocouple locations. All \( q_{w} \) values were larger than the axially averaged heat flux obtained from the total enthalpy increase of the cooling water. This might be due to the heat loss through the thermocouples, because the ratio of the thermocouple inserted depth to the diameter was 4.0 (= 2 mm/0.5 mm). The \( q_{cw} \) value might be most reliable, but it sometimes showed an unreasonable value. This might be due to long measurement time, because the thermocouples to measure the cooling water temperature distribution were manually move in the radial direction for each axial location \( z \). The \( q_{g,a} \) and \( q_{g,l} \), which were respectively obtained from temperatures with the 50 mm and 20 mm upstream thermocouples (cf. Photo 1), sometimes showed an unreasonable value like the value at about \( z = 0.1 \) m. The reason for the small heat flux at about \( z = 0.1 \) m was not clear, but it might be due to wetting of the thermocouples as shown in Figs. 6 and 7 or the periodic change of the steam flow rate caused by the pressure change in the steam boiler because the
thermocouple array was axially moved during the measurement. The \( q_{g,u,i} \), which was obtained from the difference of enthalpies at locations of the 50 mm and 20 mm upstream thermocouples, showed relatively reasonable values in No. 4. However, the \( q_{g,u,i} \) sometimes showed unreasonable values. As mentioned above, each of the quantities \( q_w \), \( q_{cw} \), \( q_g \), and \( q_{g,u,i} \) had technical issues. Therefore, the average value of \( q_w \), \( q_{cw} \), and \( q_{g,u} \) was used in this study to mitigate effects of unreasonable values (this selection was made from an engineering point of view). \( q_{g,u} \) with the same measuring location as \( q_{cw} \) was used for \( q_g \).

\[
q_{ave} = \frac{q_{cw} + q_{g,u} + (q_{g,u,i} + q_{g,u})/2}{3} \tag{9}
\]

Fig. 9 shows the average heat fluxes \( q_{ave} \) in Nos. 3-5. The \( q_{ave} \) values in Nos. 3-5 were within ±20 %. The heat fluxes in this study were larger than heat fluxes obtained with the empirical correlation by Araki et al. [16] computed for No. 4 conditions.

\[
h_c [\text{KW/m}^2\text{K}] = 2.11 \times 10^{-4} \cdot \frac{0.8}{(P_g/P_l)^{0.99}}, \quad 2300 \leq Re_g \leq 21000 \tag{10}
\]

\( Re_g \) is the mixture Reynolds number. Heat fluxes with Eq. (10) increased due to the large temperature difference \( (T_{gc} - T_c) \) with increasing \( z \). In the experiments by Araki et al. [16], the ratio of the length \( L \) to the diameter \( D \) in the cooling section was 40 \((L/D = 2000 \text{ mm}/49.5 \text{ mm})\) and the measured heat transfer coefficients were well expressed by the heat and mass transfer analogy for fully developed flows (which is not shown here). The heat flux in this study agreed with that by Araki et al. [16] at \( z = 0.5 \text{ m} \).

### 4.2 Heat Fluxes

Fig. 10 shows the average heat fluxes \( q_{ave} \) in Nos. 1-2 and 6-11. The average heat fluxes in the region of \( z = 0-0.5 \text{ m} \) for Nos. 1, 2, 3-5, 6, 7, 8, 9, 10 and 11 were respectively, 47.5, 46.0, 44.0, 42.8, 27.9, 14.0, 11.0, 11.2 and 9.5 kW/m². Shrinkage of the thermal boundary layer due to the suction effect shown in Fig. 5 was in the order of Nos. 1, 2-6, 7, 8 and 9-11. This order was similar to the order of the average heat flux. This showed that the condensation heat flux (i.e. steam mass flux toward the cold wall) affected the shrinkage of the thermal boundary layer. The increase of the \( q_{ave} \) in the region of downstream was due to the increase of the temperature difference \( (T_r - T_n) \).

### 4.3 Empirical Correlation

Fujii et al. [17] used the ratio of the steam logarithmic mean concentration to the air logarithmic mean concentration \( C_{s,ave}/C_{a,ave} \) as a major factor for evaporation from the hot water pool surface and condensation on the cold vertical wall in the presence of air. The \( C_{s,ave}/C_{a,ave} \) gave relatively good results for Nos. 1-7 but it was not effective for Nos. 8-11. Due to the suction effect discussed in sections 3.2 and 4.2, we cannot apply the heat and mass transfer analogy without the suction effect. Therefore, the measured boundary thickness \( \delta_l/R \) was used as one of the effective factors to derive an empirical correlation of the condensation heat flux \( q_{ave} \). Based on the
examinations, we used the gradient of the steam density \((\rho_s - \rho_{s,v})/\delta t/D\) and the average steam velocity \(v_{s,\text{ave}}\) as major factors and we derived the following empirical correlation:

\[
q_{\text{ave}} = 0.0044 \left\{ \left( \rho_s - \rho_{s,v} \right)/(\delta t/D) \right\} v_{s,\text{ave}} h_{fg},
\]

\(3200 \leq Re_{g,in} \leq 25000, \ 0.36 \leq x_{a,in} \leq 0.92\) \(11\)

where \(\rho_s\) [kg/m\(^3\)] is the steam density and \(h_{fg}\) [kJ/kg] is the latent heat of evaporation or condensation. The coefficient of 0.0044 was obtained to fit heat flux data. In Eq. (11), \(\rho_s v_s\) shows the steam mass flux, and the effect of the air flow rate was small.

Fig. 11 compares the \(q_{\text{ave,cal}}\) calculated by Eq. (11) with the \(q_{\text{ave,m}}\) measured. The uncertainty of Eq. (11) was \(\pm 40\%\). Eq. (11) agreed well with data for Nos. 8-11 in the region of small \(q_{\text{ave}}\), but underestimated \(q_{\text{ave}}\) for No. 1 and overestimated \(q_{\text{ave}}\) for Nos. 3-5 in the region of large \(q_{\text{ave}}\) due to the small effect of the air flow rate in Eq. (11).

5. Discussion
5.1 Suction Effect

As discussed in section 4.2, the average heat flux affected shrinkage of the thermal boundary layer due to the suction effect. The suction effect with suction through the pipe wall from single phase turbulent flows was reported by Aggarwal and Hollingsworth [18] and Yeroshenko et al. [19], but we could find no reports of suction effect data under condensation in a vertical pipe (i.e. temperature distributions in a turbulent boundary layer).

The suction effect affected not only the thermal boundary layer thickness but also the temperature distribution in the turbulent boundary layer as shown in Fig. 12. The exponential value for the dimensionless temperatures decreased with increasing the condensation heat flux. The slope in the region of a linear function should increase with increasing the condensation heat flux, but this was not detected by the thermocouple array shown in Photo 1. This might be possible to measure by traversing a thermocouple in the thermal boundary layer. Fig. 12 will provide a database for validation of numerical simulations with a CFD code on the suction effect under condensation conditions.

5.2 Condensation Heat Flux

The main objective of this experiment was to provide a database for validation of numerical simulations. Therefore, we discussed important behavior to be validated like the suction effect and the heat flux as one of the boundary conditions in numerical simulations.

In numerical simulations, the boundary conditions on the wall surface are generally given by the distribution of the temperatures, heat fluxes, or heat transfer coefficients. In this study, the thermal
resistance of the condensate film was not separately evaluated and condensation heat transfer coefficients were not evaluated. In numerical simulations for the experiments, the heat flux distribution shown in Figs. 9 and 10 can be used.

6. Conclusions

We measured radial temperature distributions of steam-air mixture in a vertical circular pipe with the diameter of 49.5 mm, and the temperature distributions in the thermal boundary layer and condensation heat fluxes were evaluated. The main findings of the present study are as follows:

(1) The suction effect due to condensation, which shrinks the thermal boundary layer, was observed, and the thermal boundary thickness decreased with increasing the condensation heat fluxes.

(2) The gradient of the dimensionless temperature difference in the turbulent boundary layer decreased due to the suction effect with increasing the condensation heat fluxes.

(3) The condensation heat fluxes were in proportion to the steam density difference between the main flow of the mixture and the mixture on the wall surface, the average steam velocity and the inverse of the thermal boundary layer thickness.

Nomenclature

| Symbol | Description |
|--------|-------------|
| $T$   | temperature [°C] |
| $v$   | velocity [m/s] |
| $W$   | mass flow rate [kg/s] |
| $x_a$ | air mass flow ratio [-] |
| $y$   | distance from wall surface [m] |
| $z$   | axial coordinate [m] |

Greek letters

| Symbol | Description |
|--------|-------------|
| $\delta_l$ | thermal boundary layer thickness [m] |
| $\lambda_w$ | thermal conductivity of wall [kW/mK] |
| $\mu$   | viscosity [Pa s] |
| $\rho$  | density [kg/m$^3$] |

Subscripts

- $a$: air
- $ave$: average
- $c$: center
- $cal$: calculation
- $cw$: cooling water
- $g$: steam-air mixture
- $i$: number of radial location
- $in$: inlet or inner surface
- $j$: number of axial location
- $m$: measured
- $s$: steam
- $t$: total
- $w$: wall

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