Pressure Drop and Heat Transfer for Flow-Boiling of Water in Small-Diameter Tubes*

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Pressure drop and heat transfer for flow-boiling of water in small-diameter tubes under atmospheric pressure were discussed. The results of experiments were compared with those of several existing correlations and models. The experiments were carried out using atmospheric-pressure water in tubes with inner diameter D ranging from 2.0 to 6.0 mm, heated length L from 4.0 to 680.0 mm, inlet water subcooling ΔT_{in} from 70 to 90 K, and mass velocity G from 100 to 10 170 kg/(m²s). The highest heat flux q attained was 33 MW/m². Interesting phenomena were observed in small-diameter tubes with a very short heated length. The most suitable correlations for predicting the pressure drop and heat transfer in small-diameter tubes were also discussed.

Key Words: Pressure Drop, Heat Transfer, Forced Convection, Boiling, Multiphase Flow, Pipe Flow

1. Introduction

The pressure drop and heat transfer in forced convection boiling in small-diameter tubes are important in relation to the thermal-hydraulic design of high heat load equipment such as plasma-facing components of a nuclear fusion reactor, solid targets of a high-power accelerator and a large-scale radiation light device. Much research has been performed on normal-diameter tubes (namely, those with diameters larger than 8 mm) in relation to heat removal from various engineering equipment such as heat exchangers and nuclear reactors. However, very little work has been done on small-diameter tubes. As Mishima and Hibiki⁽¹⁾ reported for an adiabatic two-phase flow, Chisholm's parameter $C^{(2)}$ for small-diameter tubes is different from that for normal-diameter tubes, and the characteristics of gas-liquid two-phase

flow in small-diameter tubes are usually different from those in normal-diameter tubes. Therefore, it is anticipated that empirical correlations and models proposed for normal-diameter tubes lose their accuracy when used in the prediction of pressure drop and heat transfer in small-diameter tubes. Moreover, critical heat flux increases when tube diameter and heated length decrease^{(3),(4)}. For example, critical heat flux reaches an extremely high value of 158 MW/m² when mass velocity $G=19\,740$ kg/(m²s), tube diameter $D=1$ mm, heated length $L=5$ mm, and inlet water temperature $T_{in}=10^{\circ}\text{C}$. Very little has been reported concerning pressure drop and heat transfer in liquid single-phase and boiling two-phase flows under such a high heat flux. Regarding pressure drop in boiling two-phase flow in small-diameter tubes, we found only the work of Inasaka et al.⁽⁵⁾, who addressed the pressure drop characteristics under low void fraction and high flow rate conditions.

In this study, the effects of tube diameter, distance from the entrance of the heated section, mass velocity and heat flux on pressure drop and heat transfer are investigated for liquid single-phase and boiling two-phase flows in small-diameter tubes. In addition, the experimental results are compared with existing correlations, and those that are applicable to small-diameter tubes are selected.

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2. Experiment

2.1 Experimental apparatus

Figure 1 shows a schematic of the experimental loop. Purified water was used as the working fluid. The system pressure was adjusted to be 1.3 MPa in the upstream of the flow control valve located in front of the test section, and the relief valve from which boiling water flows out of the test section was kept open to the atmosphere. Measured data were logged in a personal computer, and were recorded and displayed on a screen.

Figure 2 shows a schematic of two kinds of test sections for measuring the pressure drop. Figure 3 shows the test section for measuring the heat transfer. The pressure drop across the test section was measured with a strain-gauge type differential-pressure transducer by setting up the test section vertically in the case of Fig. 2(a) and horizontally in the case of Fig. 2(b). Sheathed chromel-alumel thermocouples of 0.5 mm diameter were silver-soldered onto the outer surface of the heated wall and the entire heated section was thermally insulated. Since heat loss from the heated section was estimated to be 1% or less relative to the amount of heat in the working fluid, the

Fig. 1 Schematic of experimental apparatus

Fig. 2 Schematic of test sections for measuring pressure drop

heat loss was considered to be negligible. The heater tube was made of nickel or stainless steel. In this experiment, no effect of heater material on the heat transfer coefficient was observed. The test section was joule-heated by a stabilized direct-current (D.C.) electric power supply. The experiment was conducted by increasing the power to the test section stepwise, and by setting the inlet water temperature and the mass velocity at constant values. Data were then taken at each step. The maximum temperature of the exit water was 101°C, which indicates that the exit pressure was kept at atmospheric pressure throughout the experiment. Measurement accuracy was estimated as follows: electric power to the test section within $\pm 2\%$, mass velocity within $\pm 3\%$, pressure within $\pm 2\%$, wall temperature and inlet water temperature within $\pm 2\%$ and errors in the differential pressure and heat flux were estimated to be within $\pm 2\%$ to $\pm 10\%$ depending on the length of the heated section.

2.2 Experimental condition

2.2.1 Pressure drop First, pressure drop was measured under conditions (TEST (A): tube diameter $D=6$ mm, heated length $L=680$ mm, and vertical upward flow) that are comparatively close to those for normal-diameter tubes, and the results were compared with the homogeneous flow and annular flow models. Next, pressure drop was measured under conditions of small diameter and short heated length (TEST (B): $D=2$ mm, $L=100$ mm, and horizontal flow), and again, the results were compared with existing correlations and models. Inlet water temperature T_{in} was set at 30°C, and the range of mass velocity G was 50 - 4 000 kg/(m²s). In TEST (A), to determine the relationship among pressure drop ΔP , mass velocity G and heat flux q , heat flux was set at a constant value and mass velocity was decreased gradually. At each step, pressure drop across the test

Fig. 3 Schematic of test sections for measuring heat transfer

section was measured. In TEST (B), to eliminate head loss, the test section was mounted horizontally. Then, mass velocity was set at a constant value and heat flux was increased gradually to determine the relationship between pressure drop and heat flux. At each step, pressure drop across the test section was measured. The measurement location for differential pressure is shown in Fig. 2. In this case, the measured pressure drop is the sum of the pressure drops across the heated section and the unheated sections. When boiling occurs in the heated section, pressure drop in the upstream unheated section can be disregarded since it is 2% or less of the total pressure drop, because of the short length and the small pressure drop due to single-phase flow in the unheated section. On the other hand, pressure drop in the downstream unheated section may contribute to the total pressure drop across the test section by 5% or less if boiling occurs in the heated section.

2.2.2 Heat transfer The heat transfer coefficient was measured using two kinds of test sections with tube diameter $D=2.0, 6.0$ mm and heated length $L=100.0, 360.0$ mm. The inlet water temperature T_{in} was set at 20°C , and the mass velocity G at 100, 500, 1 000 and 5 000 $\text{kg}/(\text{m}^2\text{s})$. The inside wall temperature $T_{w,inside}$ was calculated by solving the heat conduction equation, assuming that the outside of the wall was thermally insulated as follows:

$$T_{w,inside} = T_{w,outside} - \frac{q \cdot r_i}{2 \cdot k} \frac{\left(\frac{r_i}{r_o}\right)^2 - 2 \cdot \ln \frac{r_i}{r_o} - 1}{1 - \left(\frac{r_i}{r_o}\right)^2}, \quad (1)$$

where k is the thermal conductivity of nickel or stainless steel, and r_i and r_o are the inside and outside radii of the tube, respectively. The heat transfer coefficient h and the Nusselt number Nu at the position of interest are calculated using the following equations:

$$h = \frac{q}{T_{w,inside} - T_f}, \quad (2)$$

$$Nu = \frac{h \cdot D}{k_f}. \quad (3)$$

The bulk water temperature T_f at the position of interest is obtained from the thermal balance equation. It is assumed that $T_f = T_{sat}$ when $T_f \geq T_{sat}$.

3. Experimental Results and Discussion

3.1 Pressure drop

Figure 4 shows pressure drop ΔP as a function of mass velocity G . In this case, mass velocity was decreased gradually while keeping the heat flux constant. From Fig. 4, it can be seen that the ΔP - G curve has a negative slope in the two-phase flow region and has a minimum when the heat flux is $0.8 \text{ MW}/\text{m}^2$ or

Fig. 4 Dependence of pressure drop on mass velocity and heat flux in vertical tubes

Fig. 5 Dependence of pressure drop on heat flux at a constant mass velocity in horizontal tubes

more. The location of the minimum moves toward higher mass velocity and higher pressure drop loss as the heat flux increases. The numerical results obtained using the MINI-TRAC code⁽⁶⁾ as a driver code are also shown in Fig. 4. In using the code, correlations based on the homogeneous flow model were selected to calculate wall friction and wall heat transfer coefficient. The figure shows that the agreement between the numerical result and the experimental result is good. Therefore, in this case, pressure drop can be predicted well by the homogeneous flow model. For small-diameter tubes or short heated length tubes, however, the conditions exceed the applicable range of constitutive equations built in the code. Therefore, to predict pressure drop in small-diameter and/or short heated length tubes accurately, it is necessary to accumulate pressure drop data for tubes with a small diameter and/or a short heated length and to obtain appropriate correlations that reproduce the experimental results well. Figure 5 shows pressure drop for the horizontal test section with $D=2$ mm, $L=100$ mm and $T_{in}=25^\circ\text{C}$. In the figure, predictions by the model based on existing pressure drop correlations as well as by Akagawa

Table 1 Applicable ranges of existing correlations

Martinelli-Nelson ⁽⁸⁾	0.1 - 20 MPa
Thom ⁽⁹⁾	15 - 3000 psia, outlet steam quality = 3 - 100 %
Dittus-Boelter ⁽¹⁰⁾	$10^4 \leq Re \leq 10^5$, $1 \leq Pr \leq 10$
Petukov ⁽¹⁶⁾	$10^4 < Re < 5 \times 10^6$, $0.5 < Pr < 2000$
Gnielinski ⁽¹⁷⁾	$3000 < Re < 10^6$, $0.5 < Pr < 2000$
Sleicher-Rohsenow ⁽¹⁸⁾	$10^4 < Re < 10^6$, $0.1 < Pr < 10^5$
Bergles-Rohsenow ⁽¹⁹⁾	15 - 2000 lb _f /in. ² abs
Dengler-Addoms ⁽²⁰⁾	0.06 - 0.28 MPa, Water, Up-flow in tubes
Shah ⁽²¹⁾	2.4 - 27.1 mm tube, 0.1×10^6 - 13.8×10^6 N/m ² , $\Delta T_{\text{sub}} = 0 - 153^\circ\text{C}$, 0.2×10^6 - 87×10^6 kg/hm ² , $0.01 - 22.9$ MW/m ²
Sani ⁽²³⁾	0.1 - 0.27 MPa, Water, Down-flow in tubes

et al.'s method⁽⁷⁾ are also shown. The applicable ranges of the correlations used in this paper are shown in Table 1. In general, the total pressure loss ΔP across the heated section is expressed as follows.

$$\Delta P = \Delta P_f + \Delta P_a + \Delta P_h, \quad (4)$$

where ΔP_f is friction loss, ΔP_a is acceleration loss and ΔP_h is head loss. In this experiment, head loss is zero because the test section was set up horizontally. The thermodynamical equilibrium quality at the exit of the heated section (hereafter, called exit quality X_{ex}) is calculated as follows:

$$X_{\text{ex}} = 4 \cdot \left(\frac{L}{D} \right) \cdot \left(\frac{q}{G \cdot h_{fg}} \right) - \left(\frac{h_{\text{sat}} - h_{\text{in}}}{h_{fg}} \right). \quad (5)$$

In most of the pressure drop studies performed so far, the contribution of friction loss was predominant and those of acceleration loss and position loss were small. It is thought, however, that acceleration loss reaches a value comparable to friction loss under certain conditions in small-diameter tubes. Then, we attempted to test various combinations of existing correlations and models against experimental results. The following are the equations used for the comparison.

The length of the subcooled region L_{sub} is expressed as follows, based on heat balance:

$$L_{\text{sub}} = \frac{D}{4} \cdot \frac{G \cdot (h_{\text{sat}} - h_{\text{in}})}{q}. \quad (6)$$

Friction loss in the subcooled region $\Delta P_{f, \text{sub}}$ can be calculated by:

$$\Delta P_{f, \text{sub}} = 2 \cdot f \cdot \frac{L_{\text{sub}} \cdot G^2}{D \cdot \rho_f}. \quad (7)$$

Friction factor f is obtained as follows:

$$\begin{aligned} \text{Laminar flow: } f &= 16/Re \\ \text{Turbulent flow: } f &= 0.079/Re^{0.25}. \end{aligned} \quad (8)$$

The length of the boiling two-phase flow region L_{TP} is obtained by $L_{TP} = L - L_{\text{sub}}$. The friction loss of boiling two-phase flow can be calculated by the Martinelli-Nelson method⁽⁸⁾. The acceleration loss of boiling two-phase flow was calculated in two ways, i.e., using both the homogeneous flow and the annular flow models. Martinelli and Nelson gave the friction

loss of boiling two-phase flow in graphical form, which can be expressed by the following correlation given by Ueda⁽¹¹⁾:

$$\frac{\Delta P_f}{\Delta P_{L0}} = 1 + 1.20 \cdot X_{\text{ex}}^{3/4} \cdot (1 + 0.01 \sqrt{\frac{\rho_f}{\rho_g}}) \cdot \left[\left(\frac{\rho_f}{\rho_g} \right)^{0.8} - 1 \right], \quad (9)$$

where ΔP_{L0} is friction loss for saturated liquid single-phase flow, and is given by the following equation. This correlation was used in the present study to calculate friction loss of a boiling two-phase flow region.

$$\Delta P_{L0} = 2 \cdot f \cdot \frac{L_{TP} \cdot G^2}{D \cdot \rho_f}, \quad (10)$$

where friction factor f can be calculated using Eq. (8).

Acceleration loss was calculated as follows.

[1] Acceleration loss calculated by the homogeneous flow model

Acceleration loss is expressed as a function of X_{ex} in the following equation.

$$\Delta P_a = \frac{G^2}{\rho_f} \left[X_{\text{ex}} \cdot \left(\frac{\rho_f}{\rho_g} - 1 \right) \right] \quad (11)$$

[2] Acceleration loss calculated by the annular flow model

Acceleration loss is given by the following equation as a function of X_{ex} and the void fraction at the exit of the heated section a_{ex} .

$$\Delta P_a = \frac{G^2}{\rho_f} \left[\frac{X_{\text{ex}}^2}{a_{\text{ex}}} \cdot \left(\frac{\rho_f}{\rho_g} \right) + \frac{(1 - X_{\text{ex}})^2}{1 - a_{\text{ex}}} - 1 \right] \quad (12)$$

In the annular flow model, it is necessary to predict the exit void fraction a_{ex} accurately. Thus, in this study, the Thom correlation⁽⁹⁾, the Smith correlation⁽¹²⁾ and the Bankoff correlation⁽¹³⁾ were examined as correlations for the exit void fraction in a heated tube. It was found from the comparison that a_{ex} predicted by the Thom correlation is similar to that predicted by the homogeneous flow model, the Bankoff correlation gives a considerably low value ($a_{\text{ex}} < 0.7$), and the Smith correlation predicts values of a_{ex} that lie between the Thom and the Bankoff correlations. We decided to use the Smith correlation to calculate the exit void fraction, because prediction by this correlation appeared to be consistent with the previous observation⁽⁴⁾. The results obtained using the method for calculating pressure drop in a vertical boiling tube proposed by Akagawa et al.⁽⁷⁾ were also compared with the present experimental result. In Fig. 5, the thin solid line denotes the calculated values of ΔP_f using the Martinelli-Nelson method (MN) for ΔP_f and the homogeneous flow model (HM) for ΔP_a (Method 1). The bold, solid line denotes the calculation results obtained using the annular flow model for ΔP_a in which the void fraction was calculated using the Smith correlation (SM) (Method 2). The dotted line denotes the prediction by Akagawa et al.'s

method. The maximum and minimum measured values are denoted by $\bullet$ and $\blacktriangle$ in Fig. 5, respectively, because the differential pressure in the heated section oscillated markedly during the measurement.

Figure 5 shows that Method 2 gives a prediction closest to the experimental result. It should be noted that the measured pressure drop takes a local maximum value at point (1) and a local minimum at point (2). The reason for the local maximum may be attributed to the growth of a slug flow region, which was indicated by the increase in the amplitude of pressure oscillations near point (1). The reason for the local minimum at point (2) may be due to the shrinkage of a slug bubble region and the growth of churn-turbulent and annular flow regions. Beyond point (3), it is thought that the stable annular flow region became dominant because the amplitude of pressure oscillations became small.

3.2 Wall heat transfer coefficient

3.2.1 Distribution of the heat transfer coefficient Figure 6 shows the relationship between wall heat flux q and inside wall temperature $T_{w, \text{inside}}$ under the conditions of $D=2$ mm, $L=360$ mm, $G=1000$ kg/(m²s) and $T_{\text{in}}=20^\circ\text{C}$ in a vertical tube. Results similar to Fig. 6 were obtained for other conditions. Figure 7 shows the profile of the Nusselt number Nu vs. elevation z , with wall heat flux q as parameter. The local quality X is shown by the dotted lines in Fig. 7 for reference. It is thought that the flow in the region where the wall temperature is lower than that indicated by the dotted line in Fig. 6 is characterized by the forced convection of liquid single-phase flow, and in Fig. 7, all the data points for wall heat fluxes lower than 0.15 MW/m² are included in this region. On the other hand, the increase in Nusselt number in Fig. 7 indicates the occurrence of nucleate boiling for heat fluxes higher than 0.41 MW/m², and in Fig. 6, the region where the inside wall temperature is higher than that corresponding to the dotted

line is considered. Figure 6 and 7 indicate that the heat transfer coefficient in the liquid single-phase flow is high in the entrance region and decreases as the flow develops along the distance from the inlet. Although the distance from the inlet to the well-developed flow depends on the experimental conditions, we define here the thermal entrance region as the region where $z/D < 50$ because the heat transfer coefficient changes significantly in that region. Figure 7 indicates that boiling started from the downstream edge when the heat flux was increased to 0.405 MW/m² and the heat transfer coefficient started to increase rapidly. The figure also shows that the nucleate boiling region expanded toward the upstream with increasing heat flux and the heat transfer coefficient increased with increasing elevation z . The Nusselt number is shown in Fig. 8 as a function of local quality X . It was found from Fig. 8 that the Nusselt number increases rapidly with increasing local quality in the subcooled region ($X < 0$) due to the onset of subcooled boiling. At the same local quality in the subcooled region, the heat transfer coefficient is higher at a shorter elevation z from the inlet. It was also found that the trend of the Nu - X curve changes at $X=0$. In

Fig. 6 Dependence of heat flux on inside wall temperature and elevation in vertical tubes

Fig. 7 Dependence of Nusselt number on elevation and heat flux in vertical tubes

Fig. 8 Dependence of Nusselt number on local quality and elevation in vertical tubes

the region where $X > 0$, the dependence of Nu on X diminishes. At $z = 345$ mm, a local peak in the Nusselt number is observed at a low local quality near $X = 0$. This may be attributed to the heat transfer enhancement due to the stirring effect in bubbly and slug flows. Beyond this local peak, the Nusselt number starts to increase again monotonically with a small slope. The reason for this monotonic increase in the Nusselt number is attributed to the higher heat transfer rate caused by forced convection evaporation of a thinner annular liquid film and the higher steam velocity at higher local qualities. This trend in the Nusselt number near $X = 0$ was observed in all the experiments regardless of the mass velocity.

3.2.2 The effect of tube diameter on heat transfer characteristics Figure 9 shows the effect of tube diameter on the Nusselt number as a function of local quality. It is seen that the Nu - X curve changes its slope at $X = 0$. At a constant tube diameter, the Nusselt number increases when mass velocity is high, and this trend is particularly remarkable for $D = 6$ mm. At a constant mass velocity, the Nusselt number becomes larger for a larger tube diameter. The reason for this behavior of the Nusselt number for small-diameter tubes is attributed to the flow laminarization in the single-phase region and to the suppression of two-phase turbulent mixing in the nucleate boiling region. It is noted here that the sharp decrease in the Nusselt number at a high local quality for $D = 6$ mm as shown in Fig. 9 is attributed to the oscillation of the wall temperature due to boiling transition.

3.3 Discussion on the wall heat transfer coefficient

3.3.1 Heat transfer in liquid single-phase forced convection region Figure 10 shows the experimental results obtained under the conditions of $D = 2$ mm, $L = 360$ mm and $G = 1000$ kg/(m²s), and the predictions obtained using existing correlations for heat transfer including the modified Dittus-Boelter

Fig. 9 Dependence of Nusselt number at the exit on local quality and diameter of vertical tubes

correlation⁽¹⁰⁾. In Fig. 10, the predictions by the correlations for laminar flow heat transfer, namely $Nu = 4.0$ and $Nu = 4.36$, and the Collier equation⁽¹⁴⁾ are shown for comparison since the Reynolds number at the tube entrance is as small as 1994. The figure also shows the predictions by the Dittus-Boelter correlation, the Deissler correlation⁽¹⁵⁾, the Petukov correlation⁽¹⁶⁾, the Gnielinski correlation⁽¹⁷⁾ and the Sleicher-Rouse correlation⁽¹⁸⁾, all of which were developed for calculating turbulent flow heat transfer. From these results, it was found that the experimental results fall between the predictions by $Nu = 4.36$ and the Collier correlation. The same result was obtained for $D = 6$ mm. From these results, the discrepancy between the experimental results and the existing correlations for laminar flow is estimated to be about $\pm 10^\circ\text{C}$ in wall temperature. It should be noted here that the wall temperatures predicted by the turbulent flow correlations are lower than the experimental results, which means a larger heat transfer coefficient for turbulent flow and the same tendency as normal-diameter tubes. Figure 11 shows the experimental results obtained under the conditions of $D = 6$ mm, $L = 360$ mm and

Fig. 10 Comparison between the experimental results and several existing heat transfer correlations and a new correlation

Fig. 11 Comparison between the experimental results and several existing heat transfer correlations and a new correlation

Fig. 12 Comparison between the experimental results and the new heat transfer correlation

$G=500 \text{ kg}/(\text{m}^2\text{s})$ and the predictions obtained using the heat transfer correlations for turbulent flow. From Fig. 11, it can be seen that the existing correlations agree well with the experimental results. This means that the existing correlations are applicable to small-diameter tubes as well.

In this study, the Dittus-Boelter correlation was modified as follows for correlation with the experimental result for liquid single-phase flow in small-diameter tubes in both laminar and turbulent flow regimes.

$$Nu_{\infty} = 0.023 \cdot Re^k \cdot Pr^{0.4}, \quad (13)$$

where

$$k = 0.65; D = 2 \text{ mm}, 2.2 \text{ mm}, Re_{in} < 3000$$

$$k = 0.78; D = 2 \text{ mm}, 2.2 \text{ mm}, Re_{in} \geq 3000$$

$$D = 6 \text{ mm}, Re_{in} < 3000$$

$$k = 0.80; D = 6 \text{ mm}, Re_{in} \geq 3000,$$

where Re_{in} is the single-phase Reynolds number at the inlet of the heated section. In Eq. (13), that index k of the Reynolds number for laminar flow is smaller than that for turbulent flow by 0.8 reflects the experimental results that the flow is laminarized in small-diameter tubes and that the heat transfer coefficient of laminar flow is lower than that of turbulent flow. In general, index k should be a continuous function of tube diameter. In Eq. (13), k is given separately for different tube diameters, because of the limited experimental data. The solid bold lines in Figs. 10 and 11 denote the prediction by Eq. (13), it is seen that Eq. (13) reproduces the experimental results in the thermally developed region very well.

It is shown in Figs. 6 and 7 that the heat transfer coefficient is large in the thermal entrance region in liquid single-phase flow. The heat transfer coefficient in the thermal entrance region can be expressed as follows, based on Eq. (13) for a thermally developed region:

$$Nu = Nu_{\infty} \cdot \coth [0.014 \cdot \{(z/D) + 17\}]. \quad (14)$$

Figure 12 shows the experimental results

Fig. 13 Comparison between the experimental results and several existing heat transfer correlations

obtained under the conditions of $D=2 \text{ mm}$, $L=360 \text{ mm}$ and $G=500 \text{ kg}/(\text{m}^2\text{s})$, and the predictions by Eq. (14). The figure shows good agreement between the experimental results and Eq. (14) except in the region very close to the inlet of the heated section where the accuracy of prediction is poor. It was expected that the heat transfer coefficient in a short tube could be calculated using Eq. (14) for the entrance region in a long heated tube. Then, Eq. (14) was compared with the experimental results for a short tube under the conditions of $D=2 \text{ mm}$, $L=100 \text{ mm}$ and $G=500 \text{ kg}/(\text{m}^2\text{s})$, and the results are shown in Fig. 13. From this figure, it was confirmed that the heat transfer coefficient along the test section with a short heated length is reproduced well by Eq. (14) for the entrance region in the test section with a long heated length.

3.3.2 Onset of nucleate boiling The Bergles-Rohsenow correlation⁽¹⁹⁾ is used in many cases to calculate the onset of nucleate boiling. The location of the onset of nucleate boiling as predicted by the Bergles-Rohsenow correlation is also shown in Figs. 10 and 11. Here, it is assumed that nucleate boiling is initiated at the intersection of the modified Dittus-Boelter correlation and the Bergles-Rohsenow correlation. From the present experiments, it was found that the boiling started at a wall temperature slightly lower than that predicted by the Bergles-Rohsenow correlation. This indicates that the flow is more laminar in a smaller diameter tube and bubble growth occurs at a lower wall temperature. The present experimental results for onset of nucleate boiling can be correlated by the following equation:

$$q = 3.0 \times 10^{-4} \cdot \Delta T_{\text{sat}}^{2.6} \quad (15)$$

$$q [\text{MW}/\text{m}^2], \Delta T_{\text{sat}} [\text{K}].$$

Predictions by Eq. (15) are shown in Figs. 10 through 12.

3.3.3 Heat transfer in the nucleate boiling region Figure 14 shows the experimental results obtained under the conditions of $D=2 \text{ mm}$, $L=360$

Fig. 14 Comparison between the experimental results and the new heat transfer correlations

mm and $G=1000 \text{ kg/(m}^2\text{s)}$, and the predictions obtained using existing heat transfer correlations for a saturated nucleate boiling region. The heat transfer correlations and models used for comparison are the Dengler-Addams correlation⁽²⁰⁾, the Shah correlation⁽²¹⁾, the Chen correlation⁽²²⁾, the Sani correlation⁽²³⁾, the homogeneous flow model⁽⁶⁾, the Jens-Lottes correlation and the Thom correlation⁽²⁵⁾. It was found that the measured wall temperature is higher than all of the predicted values. The same results were obtained over the entire range of the present experiments. The reason for this may be explained as follows. Since vapor bubbles flow one-dimensionally in a small-diameter tube⁽⁴⁾, the heat transfer enhancement due to turbulent mixing agitated by vapor bubbles is suppressed, and as a consequence, the heat transfer coefficient decreases relative to that of normal-diameter tubes. It is shown in Fig. 14 that the existing correlations can reproduce only a limited part of the experimental results. Thus, the following correlation was developed to reproduce the whole range of the present experimental results:

$$\begin{aligned} Nu &= a \cdot X + b \\ a &= 9.7 \times 10^{-6} \times D^{1.93} \\ b &= 1342 \cdot D \cdot Re_{in}^{0.44}, \end{aligned} \quad (16)$$

where D [m], Re_{in} [-]: Reynolds number at the inlet of the heated section.

The predictions by Eq.(16) are shown in Figs. 10 through 13 as solid, bold lines.

3.4 Interesting wall temperature behaviors in small-diameter tubes

Interesting wall temperature behaviors were observed in boiling two-phase flow in small-diameter tubes when the heated length was long or extremely short. (A) When the tube diameter was small and the heated length was long, it was observed that the boiling curve was almost flat in the transition-boiling region and a comparatively steady post DNB condition was generated. (B) When the heated length was

Fig. 15 Special phenomenon in small-diameter tube with a long heated section

Fig. 16 Special phenomenon in small-diameter tube with a very short heated section

extremely short, the wall temperature oscillated with a period of 50 sec or longer immediately before burnout. In general, DNB progresses rapidly in normal-diameter tubes. However, in case (A), the heat-up started at a slightly negative exit quality and continued quasi-steadily, as shown in Fig. 15. It is deduced from this observation that the flow is one-dimensional in a long and narrow channel, than it becomes inverted-annular in the slightly negative exit quality region, and finally it assumes the post-DNB heat transfer mode. On the other hand, in case (B), slowly changing, large wall temperature peaks were observed occasionally. This phenomenon was remarkable when heat flux was high in a test section with short heated length, in which the effects of heat capacity of the heater material and heat loss in the axial direction on wall temperature fluctuation cannot be neglected.

4. Summary

Experiments were performed to investigate forced convective boiling in small-diameter tubes, and pressure drop and heat transfer coefficient were measured. Moreover, the applicability of existing correlations and models to small-diameter tubes was

examined by comparing them with the experimental results. The following conclusions are drawn from this study.

(1) For a tube with a long heated section of $D = 6$ mm, pressure drop was reproduced by the homogeneous flow model for wall friction loss of a tube with a small diameter and a short heated length. The experimental results agreed well with the model in which acceleration loss was calculated based on the annular flow model together with the Smith correlation for the void fraction.

(2) The axial profiles of the heat transfer coefficients for forced-convection single-phase and boiling two-phase flows in small-diameter tubes and the effect of tube diameter on the heat transfer coefficients were clarified. The experimental results for forced-convection single-phase heat transfer coefficient were reproduced well by the existing correlations. However, it turned out that the temperature at the onset of nucleate boiling was overestimated by the existing correlations, and the accuracy of prediction decreased when the existing correlations for nucleate boiling heat transfer were applied to small-diameter tubes. New empirical correlations for the onset of nucleate boiling and nucleate boiling heat transfer were proposed to reproduce the present experimental results.

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