Prediction of Void Fraction in Subcooled and Low Quality Boiling Regions*

By Kotohiko SEKOGUCHI**, Osamu TANAKA***
Shuji ESAKI****, Takuo IMASAKA****

A new theoretical model, based on the radial temperature distribution of water that has been measured at the subcooled boiling region, is presented for the prediction of an actual quality in the subcooled flow boiling at elevated pressure. The predicted actual quality is converted into the void fraction using Smith's void correlation. Applying a part of the theoretical results, a simple calculation method of void fraction is also proposed.

These prediction methods are compared with the experimental results which cover the following conditions: pressure: 1.0~140.6 ata, heat flux: 1.6 × 10³~326 × 10³ kcal/m²h, mass velocity: 130.5~5097 kg/m²s.

Agreement between void fraction calculations and experiments is good.

1. Introduction

The need for predicting void fraction in boiling two-phase systems has led to numerous experimental investigations and some analytical studies(1)~(9). This field, however, is still in the process of exploration and clarification. A void fraction for subcooled flow boiling results from a heat balance among three parts of the fluids such as vapor, subcooled water, and superheated water close to the heated surface, which is complicatedly affected by the bubble-growth, -departure from the wall, and-dissipation due to condensation. The existence of two boiling regions is recognized; the first region is one of little vapor generation and of a very low void fraction. The second region is one of significant net vapor generation and of a void fraction rapidly increasing along the tube where some of the bubbles are detached from the heated wall and the others slide on it. Thus, most models presented hitherto aimed at the determination of void fraction in the second region.

Bowring proposed a method for calculation of actual vapor quality (or true vapor quality) which takes into account the heat transports consisting of the latent heat carried by departure bubbles, the forced convective heat, and the additional convective heat due to departure bubbles while neglecting the condensation at the top of bubbles staying or sliding on the wall(2). Rouhani and Axelsson considered the effect of vapor condensation which was regarded as one of the principal parameters in the heat exchange between the heated wall and the subcooled liquid(9).

Larsen and Tong developed such a model that the thickness of the bubble layer, assumed to be a constant void fraction, increases with a decrease of subcooling(7). There were also more simplified methods representing the true vapor quality by a simple function of the thermal equilibrium quality(5),(6),(8). The comparison between the results predicted by the above methods and our experimental data revealed that much more trials are necessary for the improvement of modeling(10).

In this paper a new analytical model is proposed for determining the true vapor quality in a subcooled flow boiling, which is based on evaluating the deformation in radial temperature profile(11) caused by the vapor generation. Employing the result of the present model, a simple correlation is also developed, and a reasonable agreement with experimental data is demonstrated.

2. Nomenclature

\[ A = \text{constant defined by Eq. (22)} \]
\[ B = \text{constant defined by Eq. (23)} \]
\[ Bo = \text{Boiling number ( = q/Gr)} \]
\[ Bo^* = \text{constant defined by Eq. (25)} \]
\[ C = \text{constant defined by Eq. (26)} \]
\[ c_p = \text{specific heat of liquid at constant pressure, kcal/kg°C} \]
\[ D = \text{pipe diameter, mm} \]
\[ De = \text{equivalent diameter, mm} \]
\[ G = \text{mass velocity, kg/m²s} \]
\[ Gg = \text{mass velocity of vapor, kg/m²s} \]
\[ h = \text{specific enthalpy, kcal/kg} \]
\[ h_r = \text{specific enthalpy of liquid at saturation temperature, kcal/kg} \]
\[ h_{fg} = \text{latent heat of vaporization, kcal/kg} \]
\[ h_{fg} = \text{heat transfer coefficient defined by Eq. (5), kcal/m²h°C} \]
\[ K_m = \text{constant defined by Eq. (4)} \]
\[ K_n = \text{constant defined by Eq. (3)} \]

* Received 2nd June, 1977.
** Professor, Faculty of Engineering, Kyushu University, Fukuoka.
*** Research Assistant, Faculty of Engineering, Kyushu University.
**** Post Graduate Student, Faculty of Engineering, Kyushu University.
k = thermal conductivity of liquid, kcal/m2h°C
m = exponent in Eq.(2)
n = exponent in Eq.(1)
Pr = Prandtl number
P = pressure, at
q = heat flux, kcal/m2h
r = pipe radius, mm
Re = Reynolds number ( =Dq/υ)
s = slip ratio
T = temperature, °C
T_b = bulk temperature, °C
T_c = temperature of centerline, °C
T_s = saturation temperature, °C
T_w = temperature of heated wall, °C
T_wexp = experimental temperature of heated wall, °C
T_{wp} = temperature of heated wall for single-phase temperature distribution, °C
u = liquid velocity, m/s
υ_c = liquid velocity of centerline,m/s
υ_{in} = inlet liquid velocity, m/s
x = thermal equilibrium quality
x_v = true vapor quality
x_{v0} = true vapor quality at x=0
x_s = thermal equilibrium quality at onset of saturated boiling
x_d = thermal equilibrium quality at onset of bubble detachment defined by Eq.(24)
y = distance from wall, mm
y* = dimensionless distance from wall
( = y/R)
y_{d} = dimensionless distance from wall corresponding to saturation temperature of single-phase temperature distribution
α = local void fraction
δ = average void fraction
δ_{0} = average void fraction at x=0
ε = constant defined by Eq.(30)
ν = kinematic viscosity, m²/s
ξ = function defined by Eq.(19)
ρ_g = mass density of vapor, kg/m³
ρ_l = mass density of liquid, kg/m³
ψ = ratio of the mass of liquid flowing in the homogeneous mixture to the total mass of liquid flowing (defined by Eq.(17))

3. Analysis

3.1 Physical model

Water temperature and void fraction distributions in both the radial and axial directions were obtained at elevated pressures using the electric resistivity probe technique; from these local values it may be possible to imagine the development of a bubble flow. The axial development of the radial water temperature and void fraction distributions is shown in Fig.1 for three qualities, respectively (11).

Temperature distributions for the single-phase turbulent flows with the same averaged velocities of boiling flows are also drawn by dotted lines. Naturally the water temperature and the void fraction increase as thermal equilibrium quality increases in the flow direction. The subcooled water temperature distributions in the radial direction are classified into two regions: one is the core region where vapor bubbles are not observed and the temperature distributions are consistent with those for the single-phase flows. The other is the wall region where subcooled boiling occurs and the water temperatures are obviously different from those for the single-phase flows; i.e., the water temperature between the wall and the radial position of maximum void fraction is lower than the value for the single-phase, whereas the water temperature is higher between the maximum void fraction point and the core region.

This suggests that an enhanced heat transfer is caused by surface boiling in the superheated water layer, condensation in the subcooled water core and intensified turbulence due to the relative velocity between bubble and water. Wall temperatures in this boiling region (T_{wexp}) remain essentially constant and are lower by 13°C to 23°C than those for the single-phase flows (T_{wp}). The heat supplied from the heated surface to the boiling flow is spent for heating the subcooled water and for generating vapor bubbles. The temperature distributions are too complex to calculate the sensible heat quantities of water exactly. However, considering that the measured temperature distribution in the water core is consistent with that of the single-phase flow and the superheated layer on the wall is extremely thinner than for the single-phase flow, the sensible heat quantity of water is estimated on the basis of the temperature distribution shown in the Fig.2. Here it is first postulated that the temperature in subcooled water
core has the same value for the corresponding single-phase flow and the temperature in the superheated part of the single-phase temperature distribution is constant at saturation temperature in flow boiling. Strictly speaking, actual temperature distribution near the heated wall on which surface boiling occurs is somewhat different from that defined above.

It would be reasonably expected, however, that the deviations of calculated sensible heat quantity from the actual one compensate each other, because the deviations are partly plus and partly minus as seen in Fig. 2. Next, all the heat of superheated water evaluated as a single-phase flow at given thermal and flow conditions is assumed to be converted into generating bubbles.

3.2 Analytical procedure

3.2.1 Evaluation of the superheat in the temperature distribution for imaginary single-phase flow

The amount of bubbles or steam flowing through a cross section of the tube is assumed to correspond to the heat quantity for superheat which is determined by extending the temperature curve for the imaginary single-phase flow to the wall surface. For evaluating the superheat, it is necessary to decide how to express the temperature and velocity distributions for the imaginary single-phase flow. The present analysis is first performed by applying the power law to these distributions. Further, the results for the velocity profile of von Kármán and the temperature profile of Martinelli will be discussed.

According to the power law, the distributions of velocities and temperatures are expressed as

\[
\frac{u}{u_c} = \left(\frac{y}{R}\right)^{1/n} = y^{1/n} \quad \cdots \quad (1)
\]

\[
\frac{T_w - T}{T_w - T_c} = \left(\frac{y}{R}\right)^{1/n} = y^{1/n} \quad \cdots \quad (2)
\]

where, \(u\) and \(T\) are the local liquid velocity and temperature, \(u_c\) and \(T_c\) are the values of center line, \(T_w\) is the wall temperature, and \(y^* = (y/R)\) is a dimensionless distance from the wall.

Integrating Eqs. (1) and (2) the liquid mean velocity at the inlet \(u_{in}\) and the bulk temperature \(T_B\) can be expressed by

\[
\frac{u_{in}}{u_c} = \frac{2a^2}{(n+1)(2n+1)} = K_a \quad \cdots \quad (3)
\]

\[
\frac{T_w - T_c}{T_w - T} = \left(\frac{y}{y^*}\right)^{1/n} = y^{1/n} \quad \cdots \quad (4)
\]

The wall temperature for single-phase \(T_w\) is obtained by means of the following heat transfer coefficient \(\chi_{hp}\) which is evaluated by taking into account the increase of liquid velocity with an increased void fraction

\[
\chi_{hp} = \frac{q}{T_w - T_s} = 0.023 \frac{T_w - T_s}{R_k \left(\frac{1 - z_R}{1 - z^*}\right)^{1/3}} \quad \cdots \quad (5)
\]

where \(q\) is the heat flux, \(k\) is the thermal conductivity of liquid, \(D\) is the tube diameter, \(R_k\) and \(Pr\) are the Reynolds and Prandtl numbers, \(\hat{h}\) is the average void fraction, and \(T_R\) is the true vapor quality in subcooled flow boiling. Thermal equilibrium quality is described as

\[
x = \frac{h - \hat{h}}{h_{hp} - \hat{h}} = G_T (T_w - T_s) \quad \cdots \quad (6)
\]

where \(h\) and \(\hat{h}\) are the enthalpies of two-phase mixture and liquid at saturation temperature, \(h_{hp}\) is the latent heat of vaporization, \(c_p\) is the specific heat of liquid at constant pressure, and \(T_R\) is the saturation temperature. Eliminating \(T_w\) and \(T_c\) from Eqs. (2), (4) and (5) yields

\[
T = \left(1 - \frac{1}{K_a y^{1/n}}\right) \frac{q}{h_{hp}} + T_s \quad \cdots \quad (7)
\]

From Eqs. (6) and (7) the single-phase temperature distribution at a thermal equilibrium quality \(x\) is given by

\[
T = \left(1 - \frac{1}{K_a y^{1/n}}\right) \frac{q}{h_{hp}} + \frac{T_w - T_s}{x} \quad \cdots \quad (8)
\]

According to the assumption the true vapor quality \(x_R\) can be expressed as the ratio of the vapor mass flow rate generated by dissipating the superheat of fluid to the total mass flow rate,

\[
x_R = \frac{G_T \int_{T_s}^{T} (T - T_s) u(1-y) dy^*}{h_{hp} \int_{T_s}^{T} u(1-y) dy^*} \quad \cdots \quad (9)
\]

where \(y^*\) is a dimensionless distance from the wall to the position with saturation temperature of the single-phase temperature profile. From Eqs. (1), (3) and (8), Eq. (9) becomes

\[
x_R = \frac{B_R \left(\frac{2m + m + n}{n(n+1)} \cdot y^{(n+1)/m}\right) - \frac{m}{(n+1)} \cdot y^{(n+1)/m}}{h_{hp} \left(\frac{2m + m + n}{n(n+1)} \cdot y^{(n+1)/m}\right) - \frac{m}{(n+1)} \cdot y^{(n+1)/m}} \quad \cdots \quad (10)
\]

Fig. 2 Physical model for temperature distribution in subcooled flow boiling.
\[
\beta_x = \left( \frac{K_x}{1 + \frac{1}{2\beta_0}} \right)^{\frac{1}{2}} \tag{11}
\]

and
\[
\beta_x = \frac{\rho_v}{\rho_l} = \frac{R_x^{1+2\beta_x}}{1 - \frac{2\beta_x}{m+1}} \beta_0 \tag{12}
\]

The relation between \( x \) and \( x_B \) obtained from Eqs. (10), (11) and (12) is shown in Fig.3. The functions to give \( x \) and \( x_B \) at three locations such as the onset of surface boiling, which is assumed to occur at \( T_x = T_B \), the point of \( x = 0 \), and the onset of bulk boiling are as follows:

(1) For the onset of surface boiling (\( T_x = T_B \)):
\[
x = 0, \quad x_B = \beta_0 \left( \frac{2m+1}{m+1} \right) \left( 2 + \frac{m+1}{K_x^{1+2\beta_x}} \right) \tag{13}
\]

(2) For the point where thermal equilibrium quality \( x \) is equal to zero:
\[
x = 0, \quad x_B = \beta_0 \left( \frac{m(3m+2m+n)}{m+1} \right) \left( 2 + \frac{m+n+1}{K_x^{1+2\beta_x}} \right) \tag{14}
\]

(3) For the onset of bulk boiling (\( T_x = T_B \)):
\[
x = 1, \quad x_B = \rho_v / \rho_l \left( 1 - \frac{1}{\beta_0} \right) \tag{15}
\]

For \( x_B < x < 1 \), \( x_B \) can be taken equal to \( x \).

3.2.2 Axial profile of void fraction

In general, the average void fraction \( \bar{\alpha} \) is interrelated with the slip ratio \( \alpha \) and the true vapor quality \( x_B \) as follows:
\[
\bar{\alpha} = \rho_v / \rho_l \left( 1 - \frac{1}{\beta_0} \right) \tag{16}
\]

where \( \rho_v \) and \( \rho_l \) are the mass densities of vapor and liquid. Proper choice of a correlation for slip ratio is important for successful void prediction. There are a number of correlations of void fraction available in the literature. As a result of comparison between our data and those calculated by the correlations, Smith's correlation \(^{10} \) is employed, which is given by
\[
\bar{\alpha} = \frac{1 + \rho_v / \rho_l \left( \frac{1}{2\beta_0} \right) \left( \frac{1 + \beta_0}{1 + \beta_0} \right)}{1 + \beta_0} \tag{17}
\]

where \( \psi \) is the ratio of the liquid flow rate as a homogeneous mixture to the total liquid flow rate, and Smith recommended 0.4 for \( \psi \).

The procedure for calculating the average void fraction in subcooled flow boiling can be summarized in the following steps:

(1) From Eq. (5), assume the average void fraction \( \bar{\alpha} \) and calculate the value of \( \bar{\beta}_B \).
(2) From Eqs. (10), (11) and (12), determine the vapor quality \( x_B \). Then put \( m \) and \( n \) as 7.0, respectively.
(3) Calculate the value of \( \bar{\alpha} \) using Eq. (17).
(4) Calculate the value of \( \bar{\beta}_B \) again using the estimated \( \bar{\alpha} \) from Eq. (5). Return to step (2) and iterate between steps (2) and (4) to obtain consistent values for the average void fraction \( \bar{\alpha} \).

4. Simplified method for determining void fraction

The prediction method of void fraction mentioned above needs iteration to solve. In order to avoid such an iteration, a more simplified method is proposed using a part of the analysis described in the previous section. The existence of two regions has been fully recognized for subcooled flow boiling. The first region is defined as one of little or no net vapor generation, and thus the void fraction in this region is extremely low. The second region presents a significant net vapor generation where the void fraction increases rapidly with the heated length. As for the purpose of avoiding only one value of void fraction, \( x_B \) can sufficiently be regarded as zero up to the starting point of the second region \( (x = x_B) \), and the range of \( x = x_B = 1 \) may be enough to evaluate the void fraction. Except for the case of post dryout in which steam is superheated, the true vapor quality is equal to thermal equilibrium quality after the start of bulk boiling \( (x = x_B) \). Consequently, the true vapor quality necessary to determine the void fraction in subcooled boiling ranges at most from \( x_B \) to \( x_B \) as thermal equilibrium quality. For convenience, a simple functional relationship between \( x \) and \( x_B \) is here introduced into this quality range. The relationship is considered to have four constraints that it should satisfy three points such as \( (x = x_B, x_B = 0) \), \( (x = 0, x_B = x_B) \), \( (x = x_B, x_B = 0) \), and \( \delta x_B / dx \) should also be unity at \( x_B \), in which \( x_B \) and \( x_B \) are determined by Eqs. (14) and (15).

4.1 Simplified functional relation between the true vapor and thermal equilibrium qualities

It is assumed that the true vapor quality \( x_B \) is related to the thermal equilibrium quality \( x \) by (the dotted line in Fig.3).
4.3 Expression for the true vapor quality $x_0$ at the thermal equilibrium quality equal to zero

From Eqs. (12) and (14), $x_0$ can be written as

$$x_0 = f(m, n) \frac{R e^{m} P r^{n/3}}{(1 - d_a) B_i}$$

From a standpoint of accuracy, it is the most desirable to use Eq. (17) for calculating $\delta_0$ from $x_0$, but iteration is necessary to solve. As the purpose of this section is to derive a simple calculation method of void fraction, Eq. (29) described later is used instead of Eq. (17). According to the examination of Eq. (29), the predicted values of $\delta_0$ have a tendency to be lower than the experimental. To obtain better results it is necessary to modify Eq. (25). Generally, $m$ is a function of Reynolds and Prandtl numbers, and $n$ is a function of Reynolds number. For simplification, however, $m$ and $n$ are regarded as constant. Further, $x_0$ is assumed to be negligibly small as compared to unity. Taking account of the modification and the approximation, Eq. (25) is written as:

$$x_0 = \frac{C R e^{m} P r^{n/3} B_i}{(1 - d_a) B_i}$$

where $C$ is the modification coefficient to be determined experimentally. Figure 4 shows the values of $x_0$ obtained by substituting experimental data (12) into Eq. (29), against $Re^{m} P r^{n/3} B_i$. If $C = 4.4$, Eq. (26) correlates 80% of the data with an accuracy of about 10% for void fraction. The accuracy of the simple calculation method is practically acceptable. To confirm the validity of the parameters included in Eq. (26), the equation of void fraction given by $x_0$ is compared with the other correlations of $\delta_0$.

Equations (26) and (29), by taking $x_0 < 1$,

$$\frac{d_0}{1 - d_a} = C R e^{m} P r^{n/3} B_i$$

where $C$ is a constant defined by Eq. (30). Martin (19) proposed a purely empirical correlation as follows:

$$\frac{d_0}{1 - d_a} = 0.69 \left( \frac{0.31}{P r} \right) B_i$$

**Table 1** Range of experimental parameters used for comparison with the predictions

<table>
<thead>
<tr>
<th>Run No.</th>
<th>Authors</th>
<th>Pressure data</th>
<th>Heat flux kcal/m²h</th>
<th>Mass velocity kg/m²h</th>
<th>Channel geometry and dimension, mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 ~ 27</td>
<td>Sekoguchi et al.</td>
<td>1.3 ~ 16.0</td>
<td>4.03x10⁴</td>
<td>1.40x10⁴</td>
<td>circular</td>
</tr>
<tr>
<td>28 ~ 38</td>
<td>Bartolomei et al.</td>
<td>14.8 ~ 44.4</td>
<td>32.7</td>
<td>68.8</td>
<td>circular</td>
</tr>
<tr>
<td>39 ~ 53</td>
<td>Mauer</td>
<td>84.4 ~ 140.6</td>
<td>26.3 ~ 326</td>
<td>555 ~ 5097</td>
<td>circular</td>
</tr>
<tr>
<td>54 ~ 61</td>
<td>Martin</td>
<td>34.4 ~ 140.6</td>
<td>34.4 ~ 146</td>
<td>750 ~ 2200</td>
<td>rectangular</td>
</tr>
<tr>
<td>62 ~ 63</td>
<td>Firstenberg et al.</td>
<td>1.0</td>
<td>1.6 ~ 2.3</td>
<td>451</td>
<td>rectangular</td>
</tr>
<tr>
<td>64 ~ 65</td>
<td>Cook</td>
<td>40.7</td>
<td>12.6</td>
<td>339 ~ 391</td>
<td>rectangular</td>
</tr>
<tr>
<td>66 ~ 69</td>
<td>Christenssen</td>
<td>28.1 ~ 70.3</td>
<td>18.2 ~ 42.6</td>
<td>637 ~ 915</td>
<td>rectangular</td>
</tr>
<tr>
<td>70 ~ 75</td>
<td>Dyen et al.</td>
<td>140.3</td>
<td>40.7 ~ 135.6</td>
<td>868 ~ 1153</td>
<td>rectangular</td>
</tr>
<tr>
<td>76 ~ 109</td>
<td>Rouhani</td>
<td>9.5 ~ 49.9</td>
<td>52.2 ~ 106.2</td>
<td>130 ~ 1485</td>
<td>rectangular</td>
</tr>
<tr>
<td>110 ~ 162</td>
<td>Foglia et al.</td>
<td>47.9 ~ 91.0</td>
<td>29.0 ~ 152.7</td>
<td>1718 ~ 2842</td>
<td>rectangular</td>
</tr>
</tbody>
</table>
Equation (28) indicates that $\xi_0$ is a function of Bo and is quite similar to Eq. (27). It should be noted that $\xi_0$ correlated by Boving(2) and Foglia et al.(18) are also a function of Bo.

4.4 Calculation procedure for the simplified method

The relation between true vapor quality and average void fraction can be written as(10):

$$\frac{\xi}{\xi_0} = \frac{x}{1 - x}$$

(29)

where

$$\xi = 1.5 \left( \frac{\rho_1}{\rho_g} \right)^{0.5} - 0.3$$

(30)

Equation (29) is so simple that it is not needed. Equations (29) and (30) were formulated by one of the authors based on Thom's experimental data (10).

The simple procedure for calculating average void fraction in subcooled flow boiling can be summarized in the following steps:

1. From Eq. (24) calculate $x_0$.
2. Determine the value of $x_0$ using Eq. (26).
3. From Eqs. (18), (19), (22) and (23), obtain the relation between $x$ and $x_0$.
4. Evaluate $\xi$ from Eqs. (29) and (30).

5. Results and discussion

For the calculation of average void fraction, physical properties at saturation temperature are used and the exponents m and n in Eqs. (1) and (2) are taken as 7.0. Void fraction data for the rectangular and annular channels are treated on the basis of equivalent hydraulic diameter.

Both the analytical and simplified methods proposed are compared in Figs. 5(a) with experimental results available in literature (12)(20) and our data. The solid lines are the results for the analytical method and the dotted lines for the simplified calculation method. The experimental conditions indicated in the figures are pressure P (ata), heat flux $q$ ($\times 10^4$ kcal/m²h), mass velocity $G$ (kg/m²s) and diameter $D$ (mm) or hydraulic diameter $D_e$ (mm). The ranges of the experimental conditions are listed in Table 1 together with test channel geometry. The experimental conditions covered by the present

Fig. 5 Average void fraction

(Comparison of calculated values with experimental data)

- - - - - : Analytical model
- - - - - : Simplified calculation method

P: pressure, ata  q: heat flux, $\times 10^4$ kcal/m²h
G: mass velocity, kg/m²s  D: pipe diameter, mm
D_e: equivalent diameter of channel, mm
Fig. 5 Average void fraction
(Comparison of calculated values with experimental data)
comparison are pressure; 1.04×10^4 atm, heat flux; 1.6×10^4×326×10^4 kcal/m^2 h, mass velocity; 130.5×5097 kg/m^2 s.

As seen from Figs.3-4(a) the calculation results show that these void fractions are in satisfactory agreement with the experimental data except for a low mass velocity of about 130 kg/m^2 s. However, the experimental data with a relatively lower mass velocity of 310 kg/m^2 s (run numbers 1 to 3) indicate fairly good consistency with the predictions. Clearly, much more measurements will be necessary for the low mass velocity condition, because experiments are extremely meager for such a low velocity.

As mentioned earlier, the present analysis was performed by applying the power law to express the velocity and temperature distributions for simplicity. However, it was made clear that when the velocity profile of von Kármán and the temperature profile of Martinelli are used, the difference of void fractions between these two calculations is 2% at the most.

6. Conclusions

A new theoretical model based on the measured radial temperature profile of water is presented for the prediction of void fraction in the subcooled and low quality boiling regions. Applying a part of the theoretical result, a simplified calculation method of void fraction is also proposed. From the comparison of the predictions with experimental results available in literature and our data, the following may be concluded:

(1) The proposed analytical method for void prediction is in satisfactory agreement with experimental data which cover the experimental conditions: pressure; 1.04×10^4 atm, heat flux; 1.6×10^4×326×10^4 kcal/m^2 h, mass velocity; 130.5×5097 kg/m^2 s.

(2) When using the equivalent hydraulic diameter, void predictions for a rectangular channel and an annular channel are also consistent with experimental data.

(3) The results of the simplified calculation method without iteration are also in good agreement with the experimental results as those of analytical method.

Acknowledgements

The authors wish to express their thanks to Dr. M. Nakatsuyama of the Technical College for his advice on the experiments, and Miss N. Mori of Kyushu University for her assistance in this study. Thanks are also due to Kyushu University Computation Center for calculation of experimental data.

References