Study of Critical Heat Flux Mechanism in Flow Boiling Using Bubble Crowding Model* (Application to CHF in Short Tube and in Tube with Twisted Tape under Non-Uniform Heating Conditions)

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Critical heat flux (CHF) of Subcooled Flow Boiling with water in a tube was investigated from the viewpoint of mechanistic models. The Weisman-Pei bubble crowding model was selected to predict CHF in a short tube and in a tube with an internal twisted tape under nonuniform heating conditions, Based on the results of bubble behavior observation and preliminary analysis. The original Weisman-Pei model was modified in order to explain the physical phenomena of CHF. The modified model predicted well CHF in a smooth tube including the very short tube and the tube with an internal twisted tape under uniform and nonuniform heating conditions.

Key Words: Critical Heat Flux, Subcooled Flow Boiling, Bubble Crowding Model, Nonuniform Heating, Twisted Tape

1. Introduction

In general, a cooling channel in high heat flux components, for example, that in a fusion reactor, will be under circumferentially nonuniform heating conditions, since the heat due to radiation approaches from one side of the wall. To enhance critical heat flux (CHF), an internal twisted tape is inserted into the cooling channel.

As for the model of CHF, much experimental research has been conducted under uniform heating conditions. Later, the mechanical energy criteria model and the boundary layer separation model were proposed. These models contributed to the formulation of the experimental equation to predict CHF.

Recently, as a mechanistic model, Weisman-Pei⁽¹⁾ proposed the bubble crowding model and Celata et al.⁽²⁾ proposed the sublayer dryout model.

Regarding experimental studies under nonuniform heating conditions, Nariyai et al.⁽³⁾ conducted experiments both in a straight tube and in a tube with an internal twisted tape, where the tube wall was thinned circumferentially, in order to simulate nonuniform heating conditions. As a result, in the straight tube, CHF was evaluated by correlation to uniform heating using actual averaged qualities at tube outlet under nonuniform heating conditions. In the tube with the internal twisted tape, CHF increased with increasing nonuniformity, which is defined by dividing the higher heat flux by the average heat flux in the whole tube. Kinoshita et al.⁽⁴⁾ investigated CHF under nonuniform heating conditions and measured the boiling curve of high heat flux under uniform heating conditions. It was confirmed that the net vapor generation (NVG) is almost equal to that by the Levy⁽⁵⁾ model. Then, they clarified that the main factor for CHF enhancement in the tube with the internal twisted tape under circumferentially nonuniform heating conditions was alternate development and disruption of the bubble boundary layer caused by the passage of fluid through the high and low heat flux regions alternately. The bubble boundary layer development was assumed by

* Received 5th October, 2000. Japanese original: Trans. Jpn. Soc. Mech. Eng., Vol. 65, No. 639, B (1999), pp. 3740-3747 (Received 4th December, 1998)

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consideration of NVG heat flux.

According to previous research, it is confirmed that CHF influenced by the condition of bubble generation in the upstream of the tube, and partially void fraction and qualities (equilibrium qualities) are important factors for CHF occurrence. Then Kinoshita et al.^{(6),(7)} conducted CHF experiments under axial nonuniform heating conditions and showed that CHF was enhanced under the bubble boundary layer alternate development condition and the short heated length condition.

In the present research, we aim to summarize vapor generation and bubble boundary layer development in the existing mechanistic model, and propose a mechanistic model to predict CHF at low pressure and high subcooling in a tube with an internal twisted tape under nonuniform heating condition.

Nomenclature

A	: flow area	m^2
A_h	: heat transfer area to bulk flow	m^2
a	: two-phase multiplier in turbulent intensity equation	Pa
D_b	: bubble diameter	m
D_i	: tube inside diameter	m
G	: mass velocity	$\text{kg}/\text{m}^2\text{s}$
H_l	: heat transfer coefficient	$\text{W}/\text{m}^2\text{K}$
h	: enthalpy	J/kg
h_{fg}	: evaporation enthalpy change	J/kg
i_b	: turbulent intensity at bubble layer core interface	
L	: heated length	m
L_b	: bubble length	m
m_3	: lateral mass flow rate from core to bubble layer	m^3/s
m_4	: lateral mass flow rate from bubble layer to core	m^3/s
NVG	: net vapor generation	W/m^2
P	: outlet pressure	Pa
p_H	: perimeter	m
q	: heat flux	W/m^2
q_c	: critical heat flux	W/m^2
q_{Levy}	: heat flux by levy correlation	W/m^2
q_b	: that portion of total heat flux effective for vapor generation	W/m^2
q_{cond}	: heat flux due to bubble condensation	W/m^2
r	: bubble radius	m
s	: thickness of bubble boundary layer	m
T_l	: temperature of liquid	K
T_{sat}	: saturation temperature	K
U_b	: bubble velocity	m/s
V	: volume	m^3
v_{11}	: radial velocity caused by bubble generation	m/s

x : quality

x_{ex} : equilibrium quality

y : twisted tape ratio

z : length from bubble generation point m

Greek symbols

α : cut thickness m

$\langle \alpha \rangle$: average void fraction

α_1 : void fraction at bulk

α_2 : void fraction at boundary layer

β : cut degree °

δ : sublayer thickness m

κ : heat conductivity W/mK

μ : viscosity $\text{Pa}\cdot\text{s}$

ρ : density kg/m^3

σ : surface tension N/m

σ_v : standard deviation of radial velocity

Ψ : factor of bubble movement at interface

Subscripts

1: bulk

2: bubble boundary layer

avg: average of whole tube

f : saturation condition

g : gas phase

l : liquid phase

ld : net vapor generation point

2. Characteristics of Previous Mechanistic Model

The Weisman-Pei model⁽¹⁾ is shown in Fig. 1. In this model, the CHF condition is that enthalpy transportation through the interface between boundary layer and the core reaches a critical value. Also

Fig. 1 Conceptual drawing of Weisman-Pei model

Fig. 2 Conceptual drawing of Celata model

enthalpy transportation is determined from turbulent radial velocity fluctuation. They assumed a critical condition for enthalpy transportation, that is, the elliptical bubbles, whose ratio of long to short elliptical axes is 3:1, filled the bubble boundary layer. When the bubbles are contacting each other, the void fraction under this condition is 0.82. They used the Levy model⁽⁶⁾ for bubble generation condition. The amount of bubbles generated is calculated by integration of the balance between the condensation heat flux and the bubble generation heat flux determined from the degree of all heat fluxes to NVG heat flux. The calculation is shown as Eq.(11). From the above, as the amount of bubbles are calculated by integration along the flow direction, it is said that the Weisman-Pei model considers the effect of heated length.

Next, the Celata model⁽²⁾ is shown in Fig. 2. This model assumed the existence of a vapor blanket or slugs adjacent to the tube wall. The critical heat flux condition is that departure nucleate boiling occurs when the liquid sublayer between the vapor blanket and the heated wall dry-out while the bubble blanket through the blanket length L_B . Then, CHF occurs. CHF is expressed by the following equation.

$$q_c = \frac{\rho_l \delta \lambda}{L_B} U_B, \quad (1)$$

where δ is the thickness of the liquid sublayer between the blanket and the heated wall, U_B is the velocity of bubbles determined from the balance of buoyancy and drag force. L_B is the blanket length determined from the critical wavelength of Helmholtz instability, which is expressed as follows using the surface ten-

sion and the density of vapor and liquid.

$$L_{Bcrit} = \frac{2\pi\sigma(\rho_l + \rho_g)}{\rho_l \rho_g U_B^2}, \quad (2)$$

In the calculation of U_B , the Karman velocity distribution, the friction factor by Colebrook-White equation, and the drag coefficient by Harmathy are used. δ is determined from the difference in superheated layer thickness in a three-layer model of temperature distribution for turbulent flow in a tube by Martinelli, and vapor blanket equivalent diameter, D_B , by Staub.

Next, the characteristics in the CHF prediction for qualities at the tube outlet of the Weisman-Pei model and the Celata model are considered. According to previous research^{(3),(4),(6),(7)}, subcooled flow boiling CHF increases with shortening of the heated length influenced by the development of a bubble boundary layer. This enhancement effect is thought to be a factor affecting CHF enhancement under circumferentially nonuniform heating conditions in the tube with the internal twisted tape. First, the effect of heated length on CHF prediction in each model is considered. Figure 3 shows the predicted values using the two models against tube exit qualities. To see the effect of heated length, calculated results for three heated lengths are also shown in the figure. L_2 , L_6 and L_{10} represent the three heated length, (20 mm, 60 mm, 100 mm) for the models, respectively. In this figure, the predicted values by the MD-Tong correlation proposed by Inasaka et al.⁽⁸⁾ are shown by the solid line. This correlation is thought to be the proper correlation of CHF under this condition.

The predicted values by the Weisman-Pei model are much higher than the MD-Tong correlation. Because the experimental CHF data used in the established Weisman-Pei model are obtained under relatively higher pressure conditions (3.0 - 14 MPa), the prediction gave higher CHF values under lower pressure condition near atmospheric pressure as this research. Considering the effect of heated length, when the heated length is shorter, CHF prediction is higher at the same exit quality. This result agrees with those of previous research. The Weisman-Pei model reflects well the effect of heated length on CHF.

The predicted values by the Celata model are almost the same as the MD-Tong correlation. Then, it is thought that Celata model makes good prediction of the absolute value. Considering the effect of heated length, the prediction gave the same value at the same exit quality even if the heated length is different. It is thought that the Celata model does not reflect the effect of heated length on CHF, because the Celata model is established by estimation of partial bubble conditions, such as bubble size. This estimation

Fig. 3 CHF prediction by mechanistic model

cannot reflect upstream conditions for the consideration of bubble generation and the development of a bubble boundary layer. That is to say, as in the previous report⁽⁶⁾, the Celata model predicts CHF under a long heated length condition that is more than several times the critical wavelength of Helmholtz instability.

3. Modification of Bubble Crowding Model

In the previous report⁽⁶⁾, the mechanism of CHF occurrence using the conceptual drawing of bubble growth is considered. Figure 4 shows the conceptual drawing of bubble motion during the CHF occurrence process.

(1) First, wall heat flux increases, and independent bubble generation starts. However, the bubbles attach to the heated wall. When heat flux reaches

Fig. 4 Conceptual drawing of bubble growth at CHF

NVG, bubbles start detaching from the heated wall.

(2) As the heat flux increases further, bubble generation increases and the bubble boundary layer is formed beneath the heated wall.

(3) The amount of bubbles in the boundary layer increases with increasing heat flux and void fraction of the bubble boundary layer at the tube outlet reaches the value at which bubbles start to influence each other. Then, the bubbles start to combine.

(4) The combined bubble (slug) continues to grow in size. When the heated length is sufficient, the bubble slug length reaches the critical wavelength of Helmholtz instability.

(5) Then, the slug obstructs the entry of bulk water into the thin liquid layer between the bubble and the heated wall. The thin liquid layer starts to dry out and burnout occurs.

The above is the assumption of CHF occurrence.

The bubble growth process at CHF occurrence could be recorded by video under atmospheric pressure, because the subcooling degree is lower than that under the high pressure condition. However, in the case of high heat flux, high pressure and high mass velocity, when the heated length is sufficient to increase the size of the bubble, it is thought that the burnout process proceed more rapidly in a shorter

Fig. 5 Void fraction concept

time. When the heated length is greater than the critical wavelength of Helmholtz instability, it is supposed that the void fraction in the bubble boundary layer reaches the value at which the combination starts. Then, the slug size increases further, and dryout occurs between the bubble and the heated wall. These process proceed at the same heat flux.

From the above, we selected that the bubble crowding model based on the Weisman-Pei model will be used to predict CHF at low pressure with consideration of the effect of heated length.

As CHF occurrence, which is between the start of bubble combination and burnout, requires a very short time, it is determined that the heat flux at the start of bubble combination is CHF in this present model. It is assumed that bubble combination starts when void fraction in the bubble boundary layer reaches the critical value at which the interaction of bubbles starts. When CHF occurs, the length of the slug reaches the critical wavelength of Helmholtz instability. When the heated length is less than the critical wavelength, CHF shows large enhancement.

The void fraction at CHF occurrence is shown in Fig. 5. It is assumed that independent bubbles are spherical under high velocity and high subcooling conditions. The bubbles are in cubes which are in contact each other. If the bubble fills in the cube,

which means that the bubble diameter is the same as the cube width, the bubbles will already be contacting each other. But according to Levy's⁽⁵⁾ assumption, mutual interference of bubbles occurred when the ratio of bubble diameter to bubble distance was 0.5. Then, it is thought that in this model, interference and combination start before the bubbles actually contact each other. Therefore, it is assumed that the void fraction to start interaction and combination of bubbles is half the void fraction value when the spherical bubble fills the cube. If the heat flux retain the same value, the slug size will increase to the critical wavelength of Helmholtz instability.

The void fraction is expressed by the following equation.

$$\alpha_2 = \frac{1}{2} \frac{V_{\text{Bubble}}}{V_{\text{Cube}}} = \frac{1}{2} \frac{4/3\pi r^3}{8r^3} = \frac{\pi}{12}, \quad (3)$$

We show next the calculation process of the Weisman-Pei model and the fundamental concept of the bubble crowding model. CHF q_c is the determined by the balance between enthalpy and quality. Then, it is expressed as follows.

$$\left(\frac{q_c}{h_{fg}G} \right) \left(\frac{h_l - h_{ld}}{h_f - h_{ld}} \right) = (x_2 - x_1) \psi i_b, \quad (4)$$

Here, h_l is liquid enthalpy, h_f is saturated liquid enthalpy and h_{ld} is the liquid enthalpy at NVG heat flux. x_1 and x_2 express the quality in the core region and the bubble boundary layer. ψ expresses the transfer rate of fluid at the interface between the core and the boundary layer, and i_b is the turbulent intensity at the interface.

NVG enthalpy h_{ld} is calculated using the Levy model. The dimensionless length from the wall is expressed as follows.

$$y_b^+ = 0.01 \frac{(\sigma D_i \rho_f)^{1/2}}{\mu_f}, \quad (5)$$

D_i is the tube inside diameter. Using this y_b^+ , h_{ld} is calculated from the three-layer model as follows.

$$\text{if } 0 \leq y_b^+ \leq 5.0; \quad h_f - h_{ld} = C_{pt} \frac{q}{H_l} - \frac{q}{G \left(\frac{f}{8} \right)^{1/2}} Pr y_b^+ \quad (6)$$

$$\text{if } 5.0 \leq y_b^+ \leq 30.0; \quad h_f - h_{ld} = C_{pt} \frac{q}{H_l} - 5 \frac{q}{G \left(\frac{f}{8} \right)^{1/2}} \left\{ Pr + \ln \left[1 + Pr \left(\frac{y_b^+}{5.0} - 1.0 \right) \right] \right\} \quad (7)$$

$$\text{if } y_b^+ \geq 30.0; \quad h_f - h_{ld} = C_{pt} \frac{q}{H_l} - 5 \frac{q}{G \left(\frac{f}{8} \right)^{1/2}} \left\{ Pr + \ln(1 + 5.0 Pr) + 0.5 \ln \left(\frac{y_b^+}{30} \right) \right\} \quad (8)$$

Here, dimensionless numbers are expressed as

follows. Nu is the Nusselt number, Re is the Reynolds number and Pr is the Prandtl number.

$$Nu = 0.023 Re^{0.8} Pr^{0.4} = \frac{H_i D_i}{\kappa_l} \quad (9)$$

$$Re = G D_i / \mu_l, \quad Pr = c_p \mu_l / \kappa_l$$

Next, we calculate the average quality and the liquid enthalpy. First we calculate the average quality, and then the calculate the void fraction.

$$x_{avg} = \frac{1}{G A h_{fg}} \left\{ \int_{z_d}^{z_{CHF}} \frac{p_H q_b}{1 + \varepsilon} dz - \int_{z_d}^{z_{CHF}} p_H q_{cond} dz \right\} \quad (10)$$

$$q_b = q \left[\frac{h_l - h_{ld}}{h_f - h_{ld}} \right] \quad (11)$$

$$q_{cond} = \begin{cases} H_0 \left(\frac{h_{fg}}{v_{fg}} \right) \left(\frac{A}{p_H} \right) \langle \alpha \rangle (T_{sat} - T_l), & \text{for } h_l \geq h_{ld} \\ 0, & \text{for } h_l < h_{ld} \end{cases} \quad (12)$$

$$\varepsilon = \frac{\rho_l (h_f - h_l)}{\rho_g h_{fg}} \quad (13)$$

$$T_l = F_1(h_l), \quad \rho_l = F_2(T_l)$$

we calculate the average void fraction in the tube using average quality and liquid enthalpy

$$h_l = \left\{ \frac{h_{avg} - h_g x_{avg}}{1 - x_{avg}} \right\}, \quad (14)$$

$$\rho_{avg} = \frac{\rho_g}{\left\{ \left(\frac{\rho_g}{\rho_l} (1 - x_{avg}) + x_{avg} \right) \right\}}, \quad (15)$$

$$\langle \alpha \rangle = \frac{x_{avg}}{\left\{ \left(\frac{\rho_g}{\rho_l} (1 - x_{avg}) + x_{avg} \right) \right\}} = \frac{x_{avg} \rho_{avg}}{\rho_g}, \quad (16)$$

Next, the turbulence intensity i_b is calculated using the average void fraction. i_b is estimated using the bubble diameter D_b and the tube inside diameter D_i in the Levy model⁽⁵⁾ and two-phase flow turbulent model by Serizawa⁽⁹⁾.

$$\mu_{avg} = \mu_l \exp \left[2.55 \alpha_{avg} / \left(1 - \frac{39 \langle \alpha \rangle}{64} \right) \right] \quad (17)$$

$$Re = \frac{G D_i}{\mu_{avg}} \quad (18)$$

$$D_B = 0.015 \left(\frac{8 \sigma D_i \rho_{avg}}{f G^2} \right)^{1/2} \quad (19)$$

$$i_b = 0.790 Re^{-0.1} \left(\frac{D_B}{D_i} \right)^{0.6} \left[1 + a \left(\frac{\rho_l - \rho_g}{\rho_g} \right) \right] \quad (20)$$

$$a = 0.135 \quad \text{if } G \leq 2695 \text{ kg/m}^2\text{s}$$

$$a = 0.135 \left(\frac{G}{2695} \right)^{-0.3} \quad \text{for } G > 2695 \text{ kg/m}^2\text{s} \quad (21)$$

Velocity distribution is expressed as follows.

$$\sigma_v = \frac{i_b G}{\bar{\rho}} = i_b \left[\frac{G}{\rho_l} + \left(\frac{G}{\rho_g} - \frac{G}{\rho_l} \right) x_{avg} \right] \quad (22)$$

$$v_{ll} = \frac{q_b}{\rho_g h_{fg}} \quad (23)$$

$$\Psi = \left\{ \frac{1}{\sqrt{2\pi}} \exp \left[-\frac{1}{2} \left(\frac{v_{ll}}{\sigma_v} \right)^2 \right] - \frac{1}{2} \left(\frac{v_{ll}}{\sigma_v} \right) \operatorname{erfc} \left(\frac{1}{\sqrt{2}} \frac{v_{ll}}{\sigma_v} \right) \right\} \quad (24)$$

Bubble boundary layer thickness s is determined in what follows using bubble diameter. Partial qualities

of core and boundary layer x_1 and x_2 , and void fractions α_1 and α_2 are expressed as follows.

$$s = 5.5 D_b \quad (25)$$

$$\rho_1 = \rho_{avg} \frac{r_0^2}{(r_0 - s)^2} - \rho_2 \frac{2 \left[r_0 - \frac{s}{2} \right] s}{(r_0 - s)^2} \quad (26)$$

$$\alpha_1 = \frac{\rho_{l1} - \rho_1}{\rho_{l1} - \rho_g}, \quad \alpha_2 = \frac{\rho_{l2} - \rho_1}{\rho_{l2} - \rho_g} \quad (27)$$

$$x_1 = \frac{\alpha_1 \rho_g}{\rho_l}, \quad x_2 = \frac{\alpha_2 \rho_g}{\rho_l} \quad (28)$$

Repeating the above process of calculation, we get the critical heat flux q_c using the Weisman-Pei model.

In the present model, NVG enthalpy h_{ld} is not used to calculate the average quality as in the Weisman-Pei model. NVG heat flux q_{Levy} , which is directly calculated using the Levy model, is used to calculate q_b . Condensation heat flux q_{cond} is expressed as follows using heat transfer coefficient H_l from the Dittus-Boelter correlation. Using an experimentally reasonable model, we attempt the simplification of the calculation method.

$$q_b = q - q_{Levy} \quad (29)$$

$$q_{cond} = H_l (T_{sat} - T_l) A_h / \pi D_i L \quad (30)$$

Here, A_h is the heat transfer area by bubbles on the inner surface of the tube, when the bubbles are in contact. It is expressed as follows.

$$A_h = \pi D_i L \times \frac{\pi}{4} \quad (31)$$

D_i is tube inside diameter, L is heated length, and the constant $\pi/4$ is the bubble ratio at the interface when the bubbles are in contact.

Using these modifications of critical void fraction and average qualities, CHF is calculated by the same process of the Weisman-Pei model. Then, CHF calculated by this modification model and the predicted value are compared with experimental data.

4. Consideration

4.1 Accuracy of Predicting CHF under Uniform Heating Conditions

Figure 6 shows the comparison of the predicted values and the experimental data by Kinoshita et al.⁽⁴⁾ at $G = 5000$ and $8700 \text{ kg/m}^2\text{s}$, $P = 0.1 - 1.5 \text{ MPa}$, against the experimental value. As shown in the figure, the ratio of the predicted values and experimental data is nearly equal to 1.0 for all data in two figures. Error was within 10%.

Figures 7(a) and (b) show the comparison with Celata's⁽¹⁰⁾ experimental data at comparatively low pressures (0.1 - 5.0 MPa) vs. exit quality and outlet pressure, respectively. As shown in Fig. 7, over 80% of data are within 25% of experimental data and prediction.

Fig. 6 Accuracy of the present model prediction for Kinoshita et al.'s data

Figures 8(a) and (b) show the comparison with Weisman-Pei⁽³⁾ experimental data at comparative high pressure (3.0 ~ 14 MPa) vs. experimental value and exit quality, respectively. As quality is nearly equal to zero, the predicted value is three times the experimental value. Since the conditions for obtaining this data are high pressure and low mass velocity, it is considered that the boiling phenomenon does not agree with the assumption in this presented model.

Moreover, in the case of the straight tube under non-uniform heating conditions, it is confirmed that the prediction by the present model is almost the same as the experimental data, when the average outlet quality is used for calculation.

4.2 Application to CHF in tube with internal twisted tape

For application to CHF in the tube with the internal twisted tape, the axial mass velocity G is replaced with the swirl flow mass velocity near the tube inner wall G_y . It is expressed as follows.

$$G_y = G \frac{\sqrt{4y^2 + \pi^2}}{2y} \quad (32)$$

Here, y is the twisted tape ratio that is determined as the 180-degree twisted pitch divided by tube diameter.

Figure 9 shows the comparison of the prediction and the experimental data by Kinoshita et al.⁽¹¹⁾ for the tube with the internal twisted tape under uniform heating conditions. As shown in Fig. 9, all errors are within 25%. The prediction is slightly larger at lower quality and smaller at higher quality. Then, it is thought that this prediction is good for CHF in the tube with the internal twisted tape. For application to CHF in the tube with the internal twisted tape, mass velocity is adjusted to swirl flow using tape twisted ratio of the internal twisted tape.

(a)

(b)

Fig. 7 Accuracy of the present model prediction for Celata et al.'s data

4.3 Application to CHF under non-uniform heating conditions

For application to CHF in the tube with the internal twisted tape under non-uniform heating conditions, the concept of effective heated length by Kinoshita et al.⁽⁷⁾ is used. It is considered in the case that the heat flux in the lower heat flux region is lower than NVG heat flux. In the tube with the internal twisted tape under non-uniform heating conditions as that of Kinoshita et al.⁽⁴⁾, the effective heated length decreases under large cut thickness α condition, because the heat flux in the lower heat flux region is lower than the NVG heat flux. The effective heated length is determined by the circumferentially cut degree β . Then, the effective heated length is the length of last higher heat flux region near tube outlet

(a)

(b)

Fig. 8 Accuracy of the present model prediction for Weisman-Pei's data

along the twisted tape, when the heat flux in the higher heat flux region is higher than the NVG heat flux. Then the effective heated length $L_{\text{effective}}$ is presented as follows.

$$L_{\text{effective}} = \frac{D_i}{2} \sqrt{4y^2 + \pi^2} \frac{360 - \beta}{180} \quad (33)$$

Here, D_i is tube inside diameter, y is twisted tape ratio, and β is circumference cut degree.

Figures 10(a), (b) and (c) show the comparison of prediction and Kinoshita's experimental data⁽⁴⁾ in the tube with the internal twisted tape under non-uniform heated conditions for $\beta=90^\circ$, 180° and 270° , respectively. In Fig. 10, the case of the cut thickness $\alpha=0.2$ mm is only shown, because that case indicates shortening of the effective heated length. For data obtained at 270° , the predictions are within 25%. This

Fig. 9 Accuracy of the present model prediction for uniform heating with twisted tape flow data

model shows that under non-uniform heating conditions, CHF is about twice that under uniform heating conditions. The predictions for small cut degrees $\beta=90^\circ$ are relatively high, because the low heat flux region is too short to disrupt the bubble boundary layer completely; thus, then the effective heated length estimated in the model is less than that obtained experimentally. However, it is thought that all data are well predicted under non-uniform heating conditions.

Then, the shortening of the effective heated length in the tubes with the internal twisted tape under circumferentially non-uniform heating conditions can be calculated. When heat flux circumferential distribution is known, the effective heated length is the length along the twisted tape in the region higher than NVG heat flux. Using this effective heated length, CHF in the tube with the twisted tape under circumferentially non-uniform heating conditions is predicted.

5. Conclusions

To determine the CHF occurrence mechanism, the Weisman-Pei bubble boundary layer crowding model is modified in terms of partially void fraction and average quality, and the concept of effective heated length is adopted. CHF's in a straight tube and in a tube with a twisted tape under uniform heating conditions and in a tube with a twisted tape under non-uniform heating conditions are predicted by this model. The predictions are compared with experimental data. The conclusions are summarized as follows:

(1) CHF in the straight tube under uniform heating conditions of relatively low pressure and high

Fig. 10 Comparison of the present model with experimental data under circumferentially nonuniform heating conditions in tube with twisted tape

subcooling degree is well predicted using this model.

(2) Under Kinoshita's experimental condition, for CHF in the tube with the twisted tape, the mass

velocity is modified to adjust to the increase in velocity by swirl flow near the wall.

(3) For CHF in tubes with the internal twisted tape under non-uniform heating conditions, the effective heated length is calculated as the length of the region where the heat flux is higher than NVG heat flux, by using the twisted tape ratio.

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