Numerical investigation on onset of significant void during water subcooled flow boiling

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ABSTRACT

Accurate evaluation of the location of onset of significant void (OSV) is particularly important in predicting the void fraction profile in the subcooled flow boiling. In this study, a Computational Fluid Dynamics (CFD) model based on a wall heat flux partition algorithm is presented. Good consistency between simulation results and available experimental data for void fraction, liquid temperature, and wall temperature is demonstrated for a given case. To further evaluate the feasibility of the CFD model in predicting the OSV condition, more simulations are performed under various operational parameters. The simulation results show that an acceptable prediction is observed in the inlet and outlet sections for the void fraction in all the cases compared with experimental data. In addition, the relative enthalpy of the flow at OSV decreases

1 with a rise in heat flux and with decrease in mass flux and with increase in pressure
2 obviously, which coincide well with the experiments. Meanwhile, the change trends for
3 the liquid subcooling at OSV of the simulation results and the results calculated by
4 Saha and Zuber correlation are similar and the error is within -7.15 ~ 8.09 K. Therefore,
5 the CFD model can obtain a comparatively accurate liquid subcooling at OSV.

6 **KEY WORDS:** subcooled flow boiling; void fraction; onset of significant void;
7 subcooling

Nomenclature

A_b	area covered by nucleating bubbles (m^2)
A_{if}	interfacial area concentration (m^{-1})
A_r	surface area to volume ratio (m^{-1})
C	coefficient
C_p	specific heat capacity at constant pressure ($\text{J}/(\text{kg K})$)
d_{bw}	bubble departure diameter (m)
D	tube diameter (m)
f	frequency of bubble departure (s^{-1})
$\bar{F}$	interfacial force (N)
g	gravitational acceleration (m/s^2)
G	mass flux ($\text{kg}/(\text{m}^2 \text{ s})$)
h	heat transfer coefficient ($\text{W}/(\text{m}^2 \text{ K})$)
h_{fg}	latent heat of evaporation (J/kg)
i	enthalpy (J/kg)
Ja	Jacob number
k	thermal conductivity ($\text{W}/(\text{m K})$)
K	empirical constant
$\vec{n}_w$	unit normal pointing away from the wall
N_a	active nucleate site density (m^{-2})
P	pressure (Pa)
Pe	Peclet number
Pr_T	turbulent Prandtl number

q	heat flux (W/m ²)
R_a	surface roughness (μm)
T	temperature (K) or quenching period (s)
$\vec{u}$	velocity (m/s)
$\vec{u}_{\tan}$	phase relative velocity component tangential to the wall surface (m/s)
V_d	volume of the bubble based on the bubble departure diameter (m ³)
x_{rel}	relative enthalpy of the flow
y_w	distance to the nearest wall (m)
z	axial height (m)
<i>Abbreviations</i>	
CFD	Computational Fluid Dynamics
DNB	Departed Nucleate Boiling
ONB	Onset of Nucleate Boiling
OSV	Onset of Significant Void
SST	Shear Stress Transport
VOF	Volume of Fluid
<i>Greek</i>	
α	void fraction
Γ	interfacial mass transfer rate (kg/s)
ΔT_w	wall superheat (K)
ΔT_{sub}	liquid subcooling (K)
λ	thermal diffusivity (m ² /s)
μ^{eff}	effective viscosity (kg/(m s))
μ_T	turbulent viscosity (kg/(m s))
ρ	density (kg/m ³)
<i>Subscript</i>	
b	bubble
c	convective
$cond$	condensation
D	drag
e	evaporative
if	interfacial

<i>in</i>	inlet
<i>l</i>	liquid
<i>lv</i>	vapor to liquid
<i>L</i>	lift
<i>q</i>	quenching
<i>s</i>	single phase
<i>sat</i>	saturation
<i>sub</i>	subcooled
<i>tot</i>	total
<i>vl</i>	liquid to vapor
<i>w</i>	wall
<i>WL</i>	wall lubrication

1. Introduction

Subcooled flow boiling occurs in many technical applications such as steam generators, nuclear reactors and refrigeration systems [1]. It is characterized by high heat transfer coefficient [2]. Similar to any boiling process, subcooled flow boiling is a phase change process where exists thermodynamic non-equilibrium between the liquid and vapor phases. Thus, the complexity of the process is increased and the analysis is very challenging [3-5].

The schematic drawing of a subcooled flow boiling accompanied by a typical boiling curve describing the void fraction distribution is shown in Fig. 1. When the subcooled liquid is injected into a heated channel, its temperature rises gradually in the axial direction in the single-phase region. The two-phase flow region is started at the location of onset of nucleate boiling (ONB) where the wall temperature sufficiently exceeds the saturation temperature at the local pressure to permit the first bubble to appear on the heated surface [6]. However, the temperature of the bulk liquid away

1 from the wall remains below saturation value and the flow region is thus known as
2 being subcooled. The void fraction remains low within a certain region for the reason
3 that the bubbles still attach to the heated surface. Further downstream, as the bulk
4 liquid temperature increases, the bubbles grow larger and begin to detach from the
5 heated surface [7]. Then, a rapid increase of the void fraction commences at the
6 location called onset of significant void (OSV). It is known that accurate evaluation of
7 the location of OSV is particularly important in predicting the void fraction profile in
8 the subcooled flow boiling region [8, 9].

9 With the rapid development of Computational Fluid Dynamics (CFD), the
10 theoretical description and analysis of subcooled flow boiling have been performed.
11 Nemitallah et al. [10] presented a detailed analysis of two-phase flow boiling
12 characteristics inside high pressure systems focusing on non-uniform axial heating
13 profiles. The influences of heat flux profile in the axial direction were investigated
14 while maintaining the same total power. Chen et al. [11] performed a CFD study on
15 vertical subcooled flow boiling of refrigerant-113 using the two-fluid model. Their
16 results showed that the bubble boundary layer became thicker when the wall heat flux
17 was increased. Hua et al. [12] adopted a three-dimensional two-fluid model coupled
18 with wall boiling model to simulate the local two-phase characteristics of boiling at
19 different passages under low-pressure condition. The simulation results showed that
20 present two-fluid model could get an accurate temperature field for cylinder head.
21 Končar et al. [13] modeled the local two-phase flow parameters in upward subcooled
22 flow boiling at low pressure. A good overall agreement between the calculated and
23 experimental data was obtained. Krepper et al. [14-16] performed several CFD studies
24 for the subcooled flow boiling. They applied CFD to solve the described task and

1 confirm its capability to contribute to the fuel assembly design. Cheung et al. [7]
2 developed a model for the subcooled flow boiling in vertical channels at low pressure.
3 In their study, the empirical correlations for the active nucleation site density, bubble
4 departure diameter and bubble departure frequency were assessed. Li et al. [17]
5 performed a numerical study of nitrogen flow boiling in a vertical tube using the
6 two-fluid model. They reported some parameters which are very important for accurate
7 predictions of flow boiling characteristics. These parameters included the distribution
8 of bubble diameter, the lift force, and the active site density. Subcooled flow boiling of
9 a nanofluid consisting of water and Al_2O_3 has been investigated numerically by
10 Abedini et al. [18]. It was shown that the simultaneous effect of different heat transfer
11 mechanisms (forced convection, bubble generation and quenching heat flux) involves
12 on the overall heat transfer coefficient. Zhuan and Wang [19] simulated the bubble
13 behavior to analyze the mechanism of subcooled boiling in a micro-channel. And the
14 results showed that both onset of nucleate boiling and boiling heat transfer
15 are influenced by micro-channel size. Yang et al. [20] proposed and discussed a new
16 wall boiling model for a 2-mm narrow rectangular channel with a copper heating
17 surface at atmospheric pressure. Compared to the default model that had been
18 developed in CFX code, the new model provides better predictions of the distribution
19 of wall temperature in the 2-mm narrow channel. Talebi et al. [21] applied a two-fluid
20 mathematical model to numerical simulation of subcooled flow boiling at low-pressure
21 condition. Good agreement was achieved with the predicted results against the
22 experimental data available in the literatures. Pan et al. [22] performed a numerical
23 investigation for the behavior of condensing single vapor bubble in subcooled boiling
24 flow by using the Volume of Fluid (VOF) multiphase flow model. The simulation

1 results were well consistent with previous experimental data in the bubble size
2 variation and lifetime.

3 From the above, there are several researches which are studying the CFD modeling
4 in predicting the subcooled flow boiling characteristics. However, among these
5 researches, the OSV condition has seldom been discussed in detail, which is of great
6 importance in predicting the void fraction profile in the subcooled flow boiling region.
7 In the present work, the CFD code ANSYS Fluent is used as a framework for solving
8 the two-fluid model of subcooled flow boiling. The model is validated against
9 available experimental data in predicting the profiles of the void fraction, liquid
10 temperature, and wall temperature. To evaluate the feasibility of the CFD model in
11 predicting the OSV condition, the Saha and Zuber correlation [9] is introduced. Then,
12 the validity of the CFD model is also discussed under different conditions using more
13 experiment data.

14 **2. Modeling of subcooled flow boiling**

15 In the subcooled flow boiling process, the energy is transferred directly from the
16 wall to the liquid. Part of this energy will cause the temperature of the liquid to
17 increase and part will generate vapor. Interphase heat transfer will also cause the
18 average liquid temperature to increase, however the saturated vapor will condense.
19 Modeling subcooled boiling flow in the context of CFD can be broadly classified
20 according to two categories: (i) heat flux partitioning during subcooled boiling flow at
21 the heated wall and (ii) two-phase flow in the bulk subcooled flow away from the
22 heated wall.

2.1. Heat flux partitioning model

The wall boiling phenomenon is modeled by the nucleate boiling model of Kurual and Podowski [23] and an extended formulation for the departed nucleate boiling regime (DNB) by Laviéville et al. [24]. As shown in Fig. 2, the total heat flux from the wall q_w is partitioned into three components, namely the convective heat flux q_c , the quenching heat flux q_q , and the evaporative heat flux q_e

$$q_w = q_c + q_q + q_e \quad (1)$$

The heated wall surface is subdivided into area A_b , which is covered by nucleating bubbles and a portion $(1 - A_b)$, which is covered by the fluid.

The convective heat flux q_c is expressed as

$$q_c = h_s (1 - A_b) (T_w - T_l) \quad (2)$$

where h_s is the single phase heat transfer coefficient, T_w is the wall temperature and T_l is the liquid temperature.

The quenching heat flux q_q models the cyclic averaged transient energy transfer related to liquid filling the wall vicinity after bubble detachment, and is expressed as

$$q_q = \frac{2k_l}{\sqrt{\pi\lambda_l T}} A_b (T_w - T_l) \quad (3)$$

where k_l is the liquid thermal conductivity, T is the quenching period between the departure of a bubble and the beginning of the growth of a subsequent one, and λ_l is the liquid thermal diffusivity.

The evaporative heat flux q_e is given by

$$q_e = V_d N_a \rho_v h_{fg} \quad (4)$$

where N_a is the active nucleate site density, ρ_v is the vapor density, h_{fg} is the latent heat of evaporation, and V_d is the volume of the bubble based on the bubble departure

1 diameter. It can be expressed as

$$V_d = \frac{\pi d_{bw}^3}{6} \quad (5)$$

2 where d_{bw} is the bubble departure diameter.

3 The abovementioned equations need closure for the following parameters:

4 1. Active nucleate site density

5 The active nucleate site density depends on the wall surface roughness, wettability
6 of the solid-liquid pair and wall superheat. For water and steel, the active nucleate site
7 density is correlated according to Lemmert and Chawla [25] depending solely on the
8 wall superheat as

$$N_a = C^n \Delta T_w^n \quad (6)$$

9 where $n=1.805$, $C=210$, ΔT_w is the wall superheat, and $\Delta T_w = T_w - T_{sat}$.

10 2. Bubble departure diameter

11 For the bubble departure diameter, the Tolubinsky and Kostanchuk correlation [26]
12 is applied

$$d_{bw} = \min \left(0.0014, 0.0006 \exp \left(-\frac{\Delta T_{sub}}{45} \right) \right) \quad (7)$$

13 where ΔT_{sub} is the liquid subcooling, and $\Delta T_{sub} = T_{sat} - T_l$. The correlation is derived for
14 the high-pressure water boiling experimental data with the upper limit for the bubble
15 departure diameter ($d_{bw}=1.4\text{mm}$).

16 3. Frequency of bubble departure

17 For the frequency of bubble departure, the model of Cole [27] based on inertia
18 controlled growth is used

$$f = \frac{1}{T} = \sqrt{\frac{4g(\rho_l - \rho_v)}{3d_{bw}\rho_l}} \quad (8)$$

1 where g is the gravitational acceleration.

2 4. Area of influence

3 Its definition is based on the bubble departure diameter and the nucleate site density

$$A_b = K \frac{N_a \pi d_{bw}^2}{4} \quad (9)$$

4 Note that in order to avoid numerical instabilities due to unbound empirical
5 correlations for the nucleate site density, the area of influence has to be restricted. The
6 area of influence is limited as follows

$$A_b = \min \left(1, K \frac{N_a \pi d_{bw}^2}{4} \right) \quad (10)$$

7 where the value of the empirical constant K has been implemented based on Del valle
8 and Kenning's findings [28]

$$K = 4.8 \exp \left(- \frac{Ja_{sub}}{80} \right) \quad (11)$$

9 where Ja_{sub} is the subcooled Jacob number. It is defined as

$$Ja_{sub} = \frac{\rho_l C_{pl} \Delta T_{sub}}{\rho_v h_{fg}} \quad (12)$$

10 **2.2. Two-phase flow model**

11 In the two-phase flow model, the field equations are expressed by two sets of
12 conservation equations of mass, momentum and energy. One set is dedicated to resolve
13 the vapor bubble phase while the other is solved for the liquid water phase. Through an
14 appropriate averaging of the local instantaneous balance equations, phase interaction
15 terms appear in each of the averaged balance equations. These interfacial terms
16 represent the mass, momentum and energy transfers through the interface between the
17 phases. In the mass conservation equations, the interfacial mass transfer is accounted

1 between phases due to vapor bubbles being condensed in the bulk subcooled liquid. In
 2 the momentum conservation equations, the interfacial momentum transfer between the
 3 vapor and liquid phases due to the drag force and non-drag force such as lift force,
 4 turbulent dispersion force and wall lubrication force are incorporated. In the energy
 5 conservation equations, the interfacial heat transfer accounts for the phase change due
 6 to condensation.

7 2.2.1. Conservation equations

8 The continuity equations for the liquid phase and vapor phase can be defined as:

$$\frac{\partial}{\partial t}(\alpha_l \rho_l) + \nabla \cdot (\alpha_l \rho_l \vec{u}_l) = \Gamma_{lv} \quad (13)$$

$$\frac{\partial}{\partial t}(\alpha_v \rho_v) + \nabla \cdot (\alpha_v \rho_v \vec{u}_v) = \Gamma_{vl} \quad (14)$$

9 where α_l and α_v are the void fractions of the liquid phase and vapor phase, ρ_l and ρ_v are
 10 the densities of the liquid phase and vapor phase, $\vec{u}_l$ and $\vec{u}_v$ are the velocities of the
 11 liquid phase and vapor phase, and Γ_{lv} and Γ_{vl} are the interfacial mass transfer rates
 12 across the phase boundary.

13 The momentum equations for the liquid phase and vapor phase can be defined as:

$$\begin{aligned} \frac{\partial}{\partial t}(\alpha_l \rho_l \vec{u}_l) + \nabla \cdot (\alpha_l \rho_l \vec{u}_l \vec{u}_l) = & -\alpha_l \nabla P + \alpha_l \rho_l \vec{g} + \nabla \cdot \left[\alpha_l \mu_l^{eff} \left(\nabla \vec{u}_l + (\nabla \vec{u}_l)^T \right) \right] \\ & + (\Gamma_{lv} \vec{u}_v - \Gamma_{vl} \vec{u}_l) + \vec{F}_{lv} \end{aligned} \quad (15)$$

$$\begin{aligned} \frac{\partial}{\partial t}(\alpha_v \rho_v \vec{u}_v) + \nabla \cdot (\alpha_v \rho_v \vec{u}_v \vec{u}_v) = & -\alpha_v \nabla P + \alpha_v \rho_v \vec{g} + \nabla \cdot \left[\alpha_v \mu_v^{eff} \left(\nabla \vec{u}_v + (\nabla \vec{u}_v)^T \right) \right] \\ & + (\Gamma_{vl} \vec{u}_l - \Gamma_{lv} \vec{u}_v) + \vec{F}_{vl} \end{aligned} \quad (16)$$

14 where P is the pressure, $\vec{g}$ is the gravitational vector, μ_l^{eff} and μ_v^{eff} are the
 15 effective viscosities of the liquid phase and vapor phase, $\vec{F}_{lv}$ and $\vec{F}_{vl}$ are the total
 16 interfacial forces.

1 The energy equations for the liquid phase and vapor phase can be defined as:

$$\frac{\partial}{\partial t}(\alpha_l \rho_l i_l) + \nabla \cdot (\alpha_l \rho_l \vec{u}_l i_l) = \nabla \cdot \left[\alpha_l \lambda_l \nabla T_l + \frac{\mu_{Tl}}{Pr_{Tl}} \nabla i_l \right] + (\Gamma_{lv} i_v - \Gamma_{vl} i_l) \quad (17)$$

$$\frac{\partial}{\partial t}(\alpha_v \rho_v i_v) + \nabla \cdot (\alpha_v \rho_v \vec{u}_v i_v) = \nabla \cdot \left[\alpha_v \lambda_v \nabla T_v + \frac{\mu_{Tv}}{Pr_{Tv}} \nabla i_v \right] + (\Gamma_{vl} i_l - \Gamma_{lv} i_v) \quad (18)$$

2 where H_l and H_v are the enthalpies of the liquid phase and vapor phase, μ_{Tl} and μ_{Tv} are
 3 the turbulent viscosities of the liquid phase and vapor phase, Pr_{Tl} and Pr_{Tv} are the
 4 turbulent Prandtl numbers of the liquid phase and vapor phase.

5 2.2.2. Interfacial area concentration

6 Interfacial area concentration is defined as the volumetric interfacial area between
 7 two phases. This is an important parameter for predicting mass, momentum and energy
 8 transfer through the interface between phases. When using the Eulerian multiphase
 9 model with non-granular secondary phases, the interfacial area is computed using an
 10 algebraic relationship between a specified bubble diameter and the interfacial area
 11 concentration. The algebraic interfacial area models are derived from the surface area
 12 to volume ratio, A_r , for a spherical bubble

$$A_r = \frac{\pi d_b^2}{\frac{1}{6} \pi d_b^3} = \frac{6}{d_b}. \quad (19)$$

13 Then, the interfacial area concentration is expressed as

$$A_{if} = \alpha_v A_r = \frac{6\alpha_v}{d_b}. \quad (20)$$

14 2.2.3. Interfacial heat and mass transfer

15 In subcooled boiling flow, the interfacial heat and mass transfer between the two
 16 phases are described by the bubble evaporation on the heated wall and the bubble

1 condensation in bulk flow. As the bubbles depart from the wall and move towards the
 2 subcooled region, heat is transferred from the bubble to the liquid. The heat transfer
 3 rate due to condensation is expressed as

$$q_{cond} = h_{if} A_{if} (T_{sat} - T_l), \quad (21)$$

4 where h_{if} is the interfacial heat transfer coefficient which is modeled by Ranz-Marshall
 5 correlation [29].

6 Correspondingly, the interfacial mass transfer or condensation rate across the phase
 7 boundary is deduced as

$$\Gamma_{lv} = -\Gamma_{vl} = \frac{h_{if} A_{if} (T_{sat} - T_l)}{h_{fg}}. \quad (22)$$

8 2.2.4. Interfacial momentum transfer

9 The total interfacial force in the momentum equations includes the following forces:

$$\overrightarrow{F_{lv}} = -\overrightarrow{F_{vl}} = \overrightarrow{F_D} + \overrightarrow{F_L} + \overrightarrow{F_{WL}} + \overrightarrow{F_{TD}}, \quad (23)$$

10 The drag force $\overrightarrow{F_D}$ is aroused by interfacial friction force and pressure difference
 11 acted on the vapor bubbles, which is computed by Ishii-Zuber correlation [30]

$$\overrightarrow{F_D} = \frac{1}{8} C_D A_{if} \rho_l |\overrightarrow{u_v} - \overrightarrow{u_l}| (\overrightarrow{u_v} - \overrightarrow{u_l}), \quad (24)$$

12 where C_D is the drag coefficient.

13 From Drew [31], the lift force $\overrightarrow{F_L}$ in terms of the slip velocity and the curl of the
 14 liquid phase velocity is described by

$$\overrightarrow{F_L} = C_L \alpha_v \rho_l (\overrightarrow{u_v} - \overrightarrow{u_l}) \times (\nabla \times \overrightarrow{u_l}), \quad (25)$$

15 where C_L is the lift coefficient, the model developed by Moraga et al. [32] is used in
 16 this study.

17 The wall lubrication force acts in lateral direction away from the wall and tends to

1 push the vapor bubble away from the heated wall. Herein the model of Antal et al. [33]
 2 is used for computing the wall lubrication force

$$\overrightarrow{F_{WL}} = -\frac{\alpha_v \rho_l |\overrightarrow{u_{\tan}}|^2}{d_b} \max\left(0, C_{w1} + C_{w2} \frac{d_b}{y_w}\right) \overrightarrow{n_w}, \quad (26)$$

3 where y_w is the distance to the nearest wall, $\overrightarrow{n_w}$ is the unit normal pointing away
 4 from the wall, $C_{w1} = -0.01$ and $C_{w2} = 0.05$ are non-dimensional coefficients, and
 5 $\overrightarrow{u_{\tan}}$ is the phase relative velocity component tangential to the wall surface. It can be
 6 expressed as

$$\overrightarrow{u_{\tan}} = (\overrightarrow{u_v} - \overrightarrow{u_l}) - [(\overrightarrow{u_v} - \overrightarrow{u_l}) \cdot \overrightarrow{n_w}] \overrightarrow{n_w}. \quad (27)$$

7 The effect of the turbulent dispersion of the vapor bubbles in the liquid phase due to
 8 the turbulent eddies is considered in the present model through the application of the
 9 turbulent dispersion force term. The considered definition of the turbulent dispersion
 10 force in the present model is based on the Favre averaging of the interfacial drag force
 11 by Burns et al. [34].

12 3. Validation of the model

13 3.1. Literature review on subcooled flow boiling experiments

14 There has been copious research in the experimental investigation of subcooled flow
 15 boiling. The working mediums include water and other refrigerants and the study
 16 parameters include void fraction, wall temperature, liquid temperature, liquid and
 17 vapor velocities, and heat transfer coefficient. Some of the typically researches are
 18 summarized in Table 1. Among these researches, the one of Bartolomei and Chanturiya
 19 [35] provides the most detailed data of the void fraction, wall temperature, and liquid
 20 temperature for water subcooled flow boiling, which are the most important and

effective for the validation. Therefore, the experiment data of Bartolomei and Chanturiya [35] are chosen for the validation of the model.

3.2. Physical model and numerical method

As shown in Fig. 3, the physical model is a 2 m long stainless steel pipe with an inside diameter of 15.4 mm and an outside diameter of 25.4 mm, which is same to that studied by Bartolomei and Chanturiya [35]. Upward subcooled flow boiling in a vertical tube subjected to uniform heat flux is considered in this study. The heat flux imposed on the inside wall is 0.57 MW/m^2 , thus the corresponding heat flux on the outside wall is 0.3456 MW/m^2 . The mass flux of the water is $900 \text{ kg/(m}^2\cdot\text{s)}$. The saturation temperature corresponding to an operating pressure of 4.5 MPa is 530.6 K and the inlet subcooling is set to 58.2 K. The temperature dependent liquid and vapor properties are taken from the REFPROP software [43].

The subcooled flow boiling is simulated with a 2D axisymmetric model, using the commercial software of FLUENT. The Menter's $k-\omega$ Shear Stress Transport (SST) [44] turbulence model has been used, as it can provide a more realistic prediction of the turbulent dissipation close to the wall. For the bubble induced turbulence, the Sato's model [45] is applied. The QUICK spatial discretization method [46] is imposed for the volume fraction term. The second order upwind scheme is used for the discretization of momentum, turbulent kinetic energy, specific dissipation rate, and energy terms. The calculations of heat transfer are performed through both the solid domain and the fluid domain.

3.3. Grid independency

It is well known that the Euler-Euler method with RPI Model does not show grid

convergence when the grid is refined. It can only be done for the coarse grids, with the first near-wall grid cell covering the whole boundary layer thickness. In the present study, a constant value of 250 is used for y^+ . To obtain a grid independent solution, a sensitivity analysis is carried out to determine the number of grids for a good compromise between the numerical accuracy and the computational effort. Five uniform grid numbers, i.e., 7,800, 28,600, 62,400, 109,200, and 169,000 nodes, are tested, and all of the grids are quadrilateral. The comparisons of the distributions of the void fraction and liquid temperature at the axial height of 1.5 m for the five grid numbers are shown in Fig. 4(a) and (b). It can be seen that the grid number of 109,200 is sufficient to obtain a grid independent result.

In this study, the effect of surface roughness, R_a , on the void fraction and liquid temperature is considered in the numerical simulations, and which was not illustrated in the experimental research studied by Bartolomei and Chanturiya [35]. The range of variation in the surface roughness, R_a , was 0 ~ 10 μm . The simulation results of the void fraction, α , and liquid temperature, T_l , for various conditions of surface roughness, R_a , are shown in Fig. 5(a) and (b). It can be seen that the void fraction, α , and liquid temperature, T_l , have little difference for various conditions of surface roughness, indicating that the surface roughness has little impact on the simulation results under the present operating parameters. Therefore, the surface roughness, R_a , is set to 0 μm in the later simulations.

3.4. Validation of void fraction and liquid and wall temperature

Fig. 6 shows the comparisons of the experimental data recorded by Bartolomei and Chanturiya [36] and the simulation results of the void fraction, liquid temperature, and

1 wall temperature. The void fraction and liquid temperature are cross-sectional average
 2 values. It can be seen that the simulated void fraction is very close to the experimental
 3 data. Meanwhile, the simulated wall temperature and liquid temperature are also in a
 4 good agreement with the experimental data, and the errors are within -3.85 ~ 1.26 K
 5 and -2.16 ~ 2.72 K, indicating an acceptable agreement between our simulation and the
 6 experiment.

7 3.5. Validation of OSV condition

8 As shown in Fig. 6, the OSV location is easy to be recognized according to the
 9 evident change of the void fraction curve. The location of OSV is indicated with the
 10 arrow in Fig. 6. Since the Saha and Zuber correlation [9] is recognized to be very
 11 reliable [47], it is used for the validation of the OSV condition.

12 Saha and Zuber [9] suggested that OSV is controlled by local thermal conditions at
 13 low mass fluxes, whereas at high mass fluxes, it is hydrodynamically controlled

$$\Delta T_{sub,OSV} = 0.0022 \frac{qD}{k_l} \quad Pe \leq 70000, \quad (28)$$

$$\Delta T_{sub,OSV} = 153.8 \frac{q}{GC_{pl}} \quad Pe > 70000, \quad (29)$$

14 where Pe is the Peclet number. It is described as

$$Pe = \frac{GDC_{pl}}{k_l}. \quad (30)$$

15 Table 2 shows the comparison of the calculated and simulated $\Delta T_{sub,OSV}$ values. The
 16 calculated $\Delta T_{sub,OSV}$ value by the Saha and Zuber correlation [9] matches the simulated
 17 $\Delta T_{sub,OSV}$ value with the error of 3.86 K. Thus, the CFD model can provide a
 18 comparatively accurate prediction for the liquid subcooling at the location of OSV.

4. Validation under various operational parameters

To further validate the accuracy of the CFD model in predicting the OSV condition, more simulations are compared with experiments performed by Bartolomei et al. [35] under different heat flux, mass flux, and pressure conditions. The channels simulated are made up of tubes with the internal diameter of 12 mm and wall thickness of 2 mm. Simulations with heat flux uniform over the length are carried out on tubes with heated lengths from 800 to 1,500 mm. It is in accordance with the experimental conditions. The range of variation in operational parameters was: with respect to heat flux 0.4 ~ 1.1 MW/m², with respect to mass flux 1000 ~ 2000 kg/(m²·s), and with respect to pressure 3 ~ 15 MPa, as shown in Table 3. In their work, the void fractions in all the cases were plotted against the relative enthalpy of the flow, x_{rel} , which is calculated as a function of the axial height, z , by

$$x_{rel} = \frac{i_{tot}(z) - i_{l,sat}}{i_{v,sat} - i_{l,sat}}. \quad (31)$$

where $i_{l,sat}$ and $i_{v,sat}$ are the enthalpy of the liquid and vapor at the saturated state under the operating pressure. $i_{tot}(z)$ is the total enthalpy at a constant axial height, z , which can be calculated by

$$i_{tot}(z) = i_{in} + \frac{\pi D \cdot q_w}{G \cdot \frac{\pi D^2}{4}} z = i_{in} + \frac{4q_w}{GD} z. \quad (32)$$

where i_{in} is the enthalpy of the liquid at the inlet.

The comparisons between the simulation results and experiment data for the dependence of the void fraction, α , on relative enthalpy of the flow, x_{rel} , for various heat flux, mass flux and pressure are shown in Fig. 7(a) ~ (e). An acceptable prediction is observed in the inlet section and the outlet section for all the cases, while the

simulated void fraction is somewhat underpredicted in the middle section compared with experiment data for some cases. In addition, the relative enthalpy of the flow at the location of OSV decreases with a rise in heat flux and with decrease in mass flux and with increase in pressure obviously, which coincide well with the experiments.

The simulated $\Delta T_{sub, OSV}$ under various operational parameters are also compared with $\Delta T_{sub, OSV}$ calculated by the Saha and Zuber correlation [9], and the results are shown in Table 4. z_{OSV} and $T_{l, OSV}$ are the axial height and liquid temperature at the location of OSV. According to Eq. (30), the values of Pe are larger than 70000 in all the cases. Thus, $\Delta T_{sub, OSV}$ should be calculated by Eq. (29). The simulated $\Delta T_{sub, OSV}$ increases with a rise in heat flux and with decrease in mass flux or pressure, except P3. With the combined effect of the heat flux and pressure in case P3 are larger than those in case P1 and P2, the simulated $\Delta T_{sub, OSV}$ of case P3 is larger than that of case P1 and P2. The results show that the change trends for $\Delta T_{sub, OSV}$ of the simulation result and the calculation result are similar and the error is within $-7.15 \sim 8.09$ K. Therefore, the CFD model can obtain a comparatively accurate liquid subcooling at the location of OSV.

5. Conclusions

A detailed CFD model based on a wall heat flux partition algorithm is presented for water subcooled flow boiling. The model is validated against available experimental data in predicting the profiles of the void fraction, liquid temperature, and wall temperature. In addition, the subcooling at the location of OSV is calculated by the Saha and Zuber correlation [9] and compared with the simulation result to evaluate the feasibility of the CFD model in predicting the OSV condition. Then, the simulations

1 are also performed under different conditions. The range of variation in operational
2 parameters was: with respect to heat flux $0.4 \sim 1.1 \text{ MW/m}^2$, mass flux $1000 \sim 2000$
3 $\text{kg}/(\text{m}^2\cdot\text{s})$, and pressure $3 \sim 15 \text{ MPa}$. The main conclusions of the present work are
4 summarized as follows:

5 (1) Good consistency between the void fractions of the simulation results and
6 experimental data is demonstrated. Meanwhile, the simulated wall temperature and
7 liquid temperature are also in a good agreement with the experimental data, and the
8 errors are within $-3.85 \sim 1.26 \text{ K}$ and $-2.16 \sim 2.72 \text{ K}$, indicating an acceptable
9 agreement between our simulation and the experiment.

10 (2) An acceptable prediction is observed in the inlet section and the outlet section for
11 the void fraction in all the cases, while it is somewhat underpredicted in the middle
12 section compared with experiment data for some cases. In addition, the relative
13 enthalpy of the flow at the location of OSV decreases with a rise in heat flux and with
14 decrease in mass flux and with increase in pressure obviously, which coincide well
15 with the experiments.

16 (3) The change trends for $\Delta T_{sub, OSV}$ of the simulation result and the calculation result
17 are similar and the error is within $-7.15 \sim 8.09 \text{ K}$. Therefore, the CFD model can obtain
18 a comparatively accurate liquid subcooling at the location of OSV.

19 This work is focused on water subcooled flow boiling (15.4 mm inside diameter) to
20 evaluate the feasibility of the CFD model in predicting the OSV condition. On this
21 basis, the CFD model for other refrigerants subcooled flow boiling and also
22 micro-scale subcooled flow boiling will be developed in the future work.

23

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Table Captions

Table 1. Experimental researches on subcooled flow boiling

Table 2. Comparison of calculated and simulated $\Delta T_{sub,OSV}$ values

Table 3. Simulation conditions for various heat flux, mass flux and pressure

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Figure Captions

Fig. 1. A schematic illustration of subcooled flow boiling

Fig. 2. Heat flux partition of the wall boiling model

Fig. 3. Physical model and boundary conditions

Fig. 4. Distributions of the void fraction (a) and liquid temperature (b) at the axial height of 1.5 m with different grid number

Fig. 5. Simulation results of the void fraction (a) and liquid temperature (b) for various conditions of surface roughness

Fig. 6. Measured vs. simulated averaged void fraction, averaged liquid temperature, and wall temperature

Fig. 7. Comparisons between the simulation results and experiment data for $\alpha(x_{rel})$ under different conditions with $G \approx 1000 \text{ kg}/(\text{m}^2 \cdot \text{s})$, $P \approx 7 \text{ MPa}$ (a), $q_w \approx 0.8 \text{ MW}/\text{m}^2$, $P \approx 7 \text{ MPa}$ (b), $q_w \approx 1.1 \text{ MW}/\text{m}^2$, $P \approx 11 \text{ MPa}$ (c), $q_w \approx 1.0 \text{ MW}/\text{m}^2$, $G \approx 1000 \text{ kg}/(\text{m}^2 \cdot \text{s})$ (d), and $q_w \approx 1.1 \text{ MW}/\text{m}^2$, $G \approx 2000 \text{ kg}/(\text{m}^2 \cdot \text{s})$ (e)

Table 1. Experimental researches on subcooled flow boiling

Authors	Application ranges	Study parameters
Bartolomei and Chanturiya [35,36]	Water, $q = 0.4 \sim 1.1 \text{ MW/m}^2$, $G = 1,000 \sim 2,000 \text{ kg/(m}^2\cdot\text{s)}$, $P = 3 \sim 15 \text{ MPa}$	Void fraction, wall temperature, liquid temperature
Zeitoun and Shoukri [37]	Water, $q = 0.286 \sim 0.706 \text{ MW/m}^2$, $G = 151.4 \sim 411.7 \text{ kg/(m}^2\cdot\text{s)}$, $P = 0.117 \sim 0.168 \text{ MPa}$	Void fraction
Garnier et al. [38]	R12, $q = 0.074 \sim 0.076 \text{ MW/m}^2$, $G = 1,985 \sim 2,030 \text{ kg/(m}^2\cdot\text{s)}$, $P = 1.46 \sim 2.62 \text{ MPa}$	Void fraction, wall temperature, liquid temperature, liquid and vapor velocities
Lee et al. [39]	Water, $q = 0.115 \sim 0.320 \text{ MW/m}^2$, $G = 476 \sim 1,061 \text{ kg/(m}^2\cdot\text{s)}$, $P = 0.1 \sim 0.2 \text{ MPa}$	Void fraction, liquid and vapor velocities
Roy et al. [40]	R113, $q = 0.095 \sim 0.116 \text{ MW/m}^2$, $G = 568 \sim 784 \text{ kg/(m}^2\cdot\text{s)}$, $P = 0.269 \text{ MPa}$	Void fraction, liquid and vapor velocity
Hata and Masuzaki [41]	Water, $u = 4.0 \sim 13.3 \text{ m/s}$, $P = 0.739 \sim 1.064 \text{ MPa}$	Wall temperature, heat transfer coefficient
Yan et al. [42]	Water, $q = 5 \sim 12.5 \text{ MW/m}^2$, $G = 6,000 \sim 10,000 \text{ kg/(m}^2\cdot\text{s)}$, $P = 3 \sim 5 \text{ MPa}$	Heat transfer coefficient

Table 2. Comparison of calculated and simulated $\Delta T_{sub,OSV}$ values

z_{OSV} (m)	α_{OSV}	Simulated $\Delta T_{sub,OSV}$ (K)	Calculated $\Delta T_{sub,OSV}$ (K)	Error (K)
0.92	0.00389	24.61	20.75	3.86

Table 3. Simulation conditions for various heat flux, mass flux and pressure

Case	q_w (MW/m ²)	G (kg/(m ² ·s))	P (MPa)	T_{in} (K)
H1	0.44	998	6.81	521
H2	0.78	965	6.89	493
H3	1.13	961	6.84	466
M1	0.78	965	6.89	493
M2	0.77	1467	6.89	519
M3	0.78	2024	6.79	520
M4	1.13	966	10.81	502
M5	1.16	1554	10.81	563
M6	1.13	1959	10.84	563
P1	0.98	990	3.01	445
P2	0.90	994	4.41	463
P3	1.13	961	6.84	466
P4	1.13	2037	6.81	504
P5	1.13	1959	10.84	563
P6	1.13	2123	14.75	583

Table 4. Comparison of calculated and simulated $\Delta T_{sub,OSV}$ values for various heat flux,
mass flux and pressure

	z_{OSV} (mm)	$T_{l, OSV}$ (K)	T_{sat} (K)	Simulated $\Delta T_{sub, OSV}$ (K)	Pe	Calculated $\Delta T_{sub, OSV}$ (K)	Error (K)
H1	540	537.52	557.12	19.60	127021	13.53	6.07
H2	690	532.28	557.91	25.63	119620	25.17	0.46
H3	730	527.52	557.42	29.90	116592	37.05	-7.15
M1	690	532.28	557.91	25.63	119620	25.17	0.46
M2	470	536.10	557.91	21.81	185278	16.18	5.63
M3	640	536.99	556.93	19.94	256931	11.85	8.09
M4	770	560.33	589.92	29.59	137130	33.84	-4.25
M5	100	567.84	589.92	22.08	233496	20.90	1.18
M6	150	568.51	590.13	21.62	295880	16.10	5.52
P1	460	480.35	507.19	26.84	104588	33.63	-6.79
P2	590	502.95	529.36	26.41	111162	29.81	-3.4
P3	730	527.52	557.42	29.90	116592	37.05	-7.15
P4	780	533.84	557.12	23.28	254510	17.20	6.08
P5	150	568.51	590.13	21.62	295880	16.10	5.52
P6	350	593.02	613.96	20.94	389656	13.22	7.72

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(a)

(b)

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(a)

(b)

(c)

(d)

(e)

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