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REVIEW ARTICLE

Review of the calculation of the boiling heat transfer coefficient in mini-channels and micro-channels

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ABSTRACT

The need to dissipate high heat flux densities has led researchers and designers to employ phase change as a mechanism to achieve this goal and thereby achieve more compact heat exchanger equipment. In the present work, a study of the literature on boiling in mini-channels and microchannels was carried out. For this purpose, bibliographies dating from the 1990s to the present were consulted, which revealed the main parameters or topics that characterize this process in mini-channels and microchannels. Thus, the terms mini-channels and microchannels, forced flow boiling and flow regimes (map) are addressed. In addition, a summary of the equations for the determination of the heat transfer coefficient in two-phase regime (h_{df}) is presented.

Keywords: Boiling; Microchannels; Mini-channels; Heat Transfer Coefficient

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1. Introduction

The need to dissipate high heat flux densities by air flow has forced designers to consider liquid cooling without phase change as an option. The other option has been to employ phase change for this purpose, i.e., the use of boiling in small diameter channels. Heat pipes, fuel cells, compact evaporators of advanced designs, among others, are equipment or devices that use channels of hydraulic diameter of the order of 1 mm. It was Tuckerman and Pease[1] who demonstrated experimentally that a heat flux density (qp) of 1300 W/cm^2 can be dissipated while maintaining a temperature difference of less than 70 °C.

That is why boiling in mini-channels and microchannels has great expectations to obtain an effective heat dissipation, mainly in small equipment[2], hence this line of research has taken interest mainly when it is necessary to dissipate high heat flux densities in electronic equipment among other uses[3,4].

Shah[5] defined the compact exchanger as having an area to volume ratio equal to or greater than 700 m^2/m^3. Many of the electronic circuit fabrication techniques are used in the fabrication of compact exchangers. Mini-channels and microchannels constitute a new technology in the dissipation of large energy densities through small areas. They are an alternative for the replacement of conventional finned exchangers used mainly in the automotive, air conditioning and refrigeration industries, among others. A surface of mini-channels and microchannels is usually formed by several of these elements in parallel[6]. The cooling medium flows through these channels in order to extract the heat from the energy source, having as a characteristic that the flow is laminar. In addition, in such a surface, high values of heat transfer coefficient, high area/volume ratio, small mass and volume are obtained and small amount of the cooling medium or cooling agent is needed.

In a heat exchanger consisting of mini-channels and microchannels, heat transfer is improved in two ways: first, the small dimensions of the ducts increase the heat transfer coefficient and second, the flat orientation of the ducts reduces the resistance on the air flow, which leads to a higher flow or a decrease in fan power. These attributes make minichannel and microchannel surfaces suitable media for cooling equipment[7].

Compared to a conventional exchanger, the main advantage of a mini-exchanger or a micro-exchanger is its high area/volume ratio, which results in a high overall heat transfer coefficient per unit volume that can be greater than 100 MW/m^3K; higher by 1 or 2 orders of magnitude than that of a conventional exchanger[8].

From the literature reviewed, it can be said that the theory for single-phase flow is applicable for conventional channels as well as for mini-channels and microchannels[9]. However, the theory for two-phase flow in conventional channels is not appropriate for mini-channels and microchannels[10].

The objective of this work is to review the literature on the characteristics of heat transfer during boiling in mini-channels and microchannels. For this purpose, the terms mini-channels and microchannels, forced flow boiling and flow regimes (map), among others, are discussed. In addition, a summary of the equations for the determination of the heat transfer coefficient in two-phase regime (h_{df}) that have been obtained and published by different authors is presented.

2. Methods and materials

2.1 Minichannel and microchannel

When are we in the presence of a mini-channel or microchannel? The terms mini-channel and micro-channel are used in the literature without any universal criteria, although many works have been carried out in an attempt to find a general criterion for these terms. Some researchers define the same transition criterion, between macrochannel and minichannel/microchannel, for both single-phase and two-phase flow in channels, while others define the criterion independently for single-phase and two-phase flow.

Mehendale[11] used the following classification to define channels:

$1\ \mu m \leq D_h \leq 100\ \mu m$: Microchannel
$100\ \mu m \leq D_h \leq 1$ mm: Minichannel
$1\ mm \leq D_h \leq 6$ mm: compact channel
$6\ mm < D_h$: conventional channel

Kandlikar and Balasubramanian[12] use a classification based on the mean path of molecules in single-phase flow, surface tension effects and two-phase flow structure.

Conventional channel: $D_h \geq 3$ mm
Mini-trunking: $200\ \mu m \leq D_h < 3$ mm
Microchannel: $10\ \mu m \leq D_h < 200\ \mu m$
Nanochannel or molecular: $D_h \leq 0.1\ \mu m$

Based on the observation that as the channel dimensions become smaller, the surface tension becomes important and on the other hand that the effect of gravity loses its effect, Kew and Cornwell[13] proposed as a criterion to define macrochannels, minichannels and microchannels, the confinement number (Co), given by:

$$Co = \frac{\left[\frac{\sigma}{g(\rho_l - \rho_g)}\right]^{0.5}}{D} \tag{1}$$

Where σ, ρ_l, ρ_v, g and D are liquid surface tension, liquid density, vapor density, gravity and channel diameter respectively.

This number is the ratio between the size of the bubble at the time of its detachment from the surface and the diameter of the duct. This same criterion has been used by Thome *et al.*[14] and Barber[15]. Under this criterion, a channel whose confinement number is greater than 0.5 can be classified as a mini/microchannel and the opposite would be a conventional channel.

Harirchian and Garimella[16] suggested the confining convective number as a criterion to define the macrochannel and the mini-channel or macrochannel, which is given by the Bond number (Bo) and the Reynolds number (Re) whose expression is:

$$Bo^{0.5} \cdot \mathrm{Re} = 160 \quad (2)$$

The Bond number expresses the ratio between the buoyancy force and the force due to tension.

$$Bo = \frac{D_h}{\sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}}} \quad (3)$$

The Reynolds number expresses the ratio between the dynamic force and the viscous force.

$$\mathrm{Re} = \frac{uD\rho}{\mu} \quad (4)$$

For a value less than 160 the channel is catalogued as a mini-channel or micro-channel and above this value it is a macro-channel. With this criterion the authors tried to take into account the effects of mass flux density and viscosity on the confinement of flow in mini-channels and microchannels together with surface tension, gravity and density.

Brauner [17] in his analysis proposed the Eötvös number (Eo) as a criterion to consider or not the influence of surface tension and gravity. The discriminant value is Eo < $(2\pi)^2$.

$$E\ddot{o} = \frac{D^2 g(\rho_l - \rho_v)}{\sigma} \quad (5)$$

On the other hand, Lee[18], considering the relationship between the drag force on the bubble and the force due to surface tension, proposes the following transition criterion:

$$D_{trans} = 17.8(\frac{\sigma\rho_l - 3\mu_l G}{G^2}) \quad (6)$$

From the above it can be said that there is no unity of criteria for the definition of mini-channel and micro-channel.

2.2 Ebullition in forced flow

In forced flow boiling it is common to consider that heat is transferred by two mechanisms: forced convection and nucleated boiling. In forced convection heat is transferred in the same way as in convection without phase change, where the heat transferred increases with increasing mass flux density (G). This mechanism is modeled with equations similar to those of convection without phase change including a fluid flow enhancement factor[13,19]. In nucleated boiling, heat is transferred by bubbles arising on the heating surface. These bubbles grow and eventually separate from the heating surface. This mechanism is similar to that of large volume boiling and is modeled with the equations of the latter. Here the heat transfer coefficient increases with increasing heat flux density and is independent of vapor quality and mass flow rate[20,21].

Heat transfer in the nucleated boiling regime is characterized by the formation of bubbles, which is influenced by the density of nucleation centers, the diameter of bubble detachment and the frequency of bubble formation.

The nucleation center density (Na) is the number of cavities or sites in which bubbles are generated per unit area of the channel and gives a measure of the energy transferred with the bubble. This nucleation center density depends on the heat flux density $(q_w^{"})$[22,23], vapor velocity[24], cavity size[25] and fluid type.

The expression obtained by Kuo[26] is based primarily on the heat flux density

$$Na = 0.29 q_W^{"1.4} \quad (7)$$

The boiling cycle is the process of liquid heating, bubble formation, bubble growth and bubble release. The number of bubbles formed per unit time or bubble formation frequency (f) and bubble detachment diameter (Ddb) are factors that have great influence on heat transfer[9]. During forced flow boiling the bubble formation frequency depends on the heat flux density, mass flux density, fluid type, size and nature of the nucleation center (cavity) and the bubble release diameter. In general, the bubble detachment frequency is evaluated by an expression of the form[26-28]:

$$f^n D_{db} = f(g, \rho_l, \rho_v, Ja) \quad (8)$$

$$f^n D_{db} = \mathrm{cte} \quad (9)$$

Where *Ja* is the Jakov criterion which expresses the ratio between the sensible heat required to heat a mass of liquid to its saturation temperature (T_{sat}) and the latent heat h_{lv} to evaporate the same mass of liquid.

$$Ja = \frac{c_p(T - T_{sat})}{h_{lv}} \tag{10}$$

The value obtained by Kuo[26], for an average absolute error of 17%, was:

$$fD_{db} = 5.65 \times 10^{-3}\frac{m}{s} \tag{11}$$

In the literature[29], the authors proposed the dimensionless nucleation frequency as a function of the dimensionless temperature difference through the expression:

$$f^* = 0.0013\exp\,(3 \cdot 10^{-5}\varphi) \tag{12}$$

Where frequency and dimensionless temperature were defined as:

$$f^* = \frac{fW^2\rho_l}{\mu_l} \tag{13}$$

and

$$\varphi = \frac{(T_s - T_\infty)Wk_l\rho_l}{\mu_l\sigma} \tag{14}$$

Where W, μ_l, k_l, T_s, T_∞ are the width of the channel, the dynamic viscosity and thermal conductivity of the liquid, the temperature of the surface where the liquid boils and the temperature of the liquid.

Applying a dimensional analysis, in the literature[29], they obtained a relationship between the size at which the bubble detaches and the Reynolds criterion, which expresses the exponential decrease in the size of the bubble at the time of detachment (V_{db}) as the Reynolds criterion (Re) increases, that is:

$$\frac{V_{db}}{H^3} = 2 \cdot 10^4\exp\,(-5 \cdot \mathrm{Re}^{0.25}) \tag{15}$$

Where H is the characteristic dimension of the channel.

2.3 Map of flow regimes

The determination of the different flow regimes has been studied by several researchers[16,30,31], since, once the regime is known, the development of models for the calculation of both the heat transfer coefficient and the pressure drop is facilitated.

In mini-channels and microchannels the flow regime depends on the interaction between the forces due to surface tension and inertia. Surface tension is dominant in the bubble and bullet flow regimes; and inertia is dominant in the annular and eddy flow regimes. **Figure 1** shows a representation of each of these flow types.

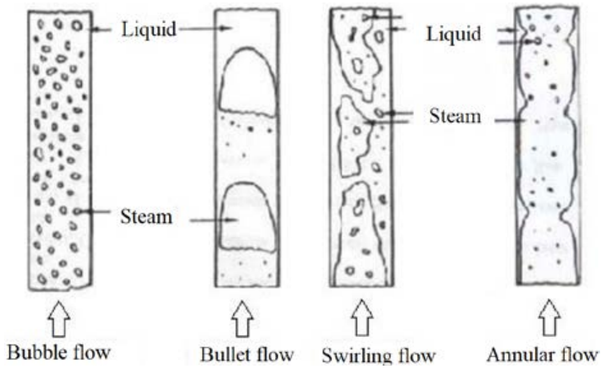

Figure 1. Flow regimes.

The flow regime map proposed by Harirchian and Garimella[16,31] proposes that for values of $Bo0.5\mathrm{Re} < 160$ vapor confinement is observed in both the bullet and swirl/annular flow regimes, while for $Bo0.5\mathrm{Re} > 160$ no confinement is observed. For low heat flux density with $Bl < 0.017$ ($Bo0.4\mathrm{Re}$-0.3) and $Bo0.5\mathrm{Re} < 160$ bullet flow is observed and with $Bo0.5\mathrm{Re} > 160$ bubble flow is observed. For high heat flux density i.e., for $Bl > 0.017$ ($Bo0.4\mathrm{Re}$-0.3) the bubbles coalesce and give rise to swirling/annular flow. Bl and Bo are the boiling and bond numbers respectively, defined as:

$$Bl = \frac{q_p}{G \cdot h_{lv}} \tag{16}$$

$$Bo = \frac{g(\rho_l - \rho_v)D^2}{\sigma} \tag{17}$$

In the literature[16], the authors propose the following as the length at which the transition from bubble to annular flow occurs:

$$L_t = 96.65(Bo^{0.5}\mathrm{Re})^{-0.258}Bl^{-1}\frac{\rho_g}{\rho_l - \rho_g}\frac{A_{cs}}{P_H} \tag{18}$$

Where A_{cs} and P_H are the cross-sectional area of the channel and the wetted perimeter of the channel, respectively.

Revellin and Thome[32] proposed a map of flow regimes from data obtained for boiling refrigerant R134a and R245fa in circular tubes. In their work they distinguished three types of regimes: isolated bubble, coalesced bubble and annular. The geometrical and flow conditions used were: 0.509 and 0.790 mm ducts, heating length from 20 to 70 mm, mass flux density from 210 to 2094 kg/m^{2}s, heat flux density from 3.1 to 597 kW/m^2, saturation temperature from 26, 30 and 35 °C and subcooling from 2 to 15 °C. The transition between the regime of isolated bubbles to coalesced bubbles is fulfilled when:

$$x_{ba/bc} = 0.763(\frac{\mathrm{Re}, Bo}{We_v})^{0.41} \tag{19}$$

For the transition from the coalesced to the annular bubble regime, the criterion is taken:

$$x_{bc/a} = 14 \cdot 10^{-5}\left(\mathrm{Re}_l^{1.47} We_l^{-1.23}\right) \tag{20}$$

Where *We* is the Weber number:

$$We = \frac{G^2 D}{\sigma\rho} \tag{21}$$

From the flow regimes that appear in minichannels and microchannels, it can be concluded that they are similar to those that appear in conventional channels: bubble, bullet, swirl and annular.

2.4 Equations used for the determination of the heat transfer coefficient in mini and microchannels

For the calculation of heat transfer during forced flow boiling in ducts, the correlations used can be divided into two groups:

Correlations that give an average heat transfer coefficient for the entire boiling process.

Correlations giving a local heat transfer coefficient as a function of steam quality. The group of correlations for the local heat transfer coefficient can be divided into:

(i) Improved model
(ii) Superposition model
(iii) Asymptotic model

In the improved model the heat transfer coefficient for two-phase flow (h_{df}) is calculated in the same way as for single-phase flow, considering the whole fluid as a liquid (h_L) and is affected by a factor that takes into account the influence of the presence of the two phases (E).

$$h_{df} = E \cdot h_L \tag{22}$$

The heat transfer coefficient for a single phase h_L is calculated by the equations of Gnielinski, cited by Kandlikar[21].

The superposition model assumes that the two-phase heat transfer coefficient is the sum of the single-phase convective component (h_{sf}) and the nucleated boiling component (h_{en}), that is:

$$h_{df} = h_{sf} + h_{en} \tag{23}$$

The asymptotic model is similar to the superposition model but in potential form, that is:

$$h_{df}^n = h_{sf}^n + h_{en}^n \tag{24}$$

In general, these models can be summarized as[33,34]:

$$h_{df} = \left[(E \cdot h_{sf})^n + (S \cdot h_{en})^n\right]^{1/n} \tag{25}$$

Here, the contribution of both parts is intensified and/or inhibited by the intensifying factor E or by the inhibition factor S. The exponent n takes into account the transition from one mechanism to the other. Depending on the value of n the superposition model can be subdivided into: Linear (n = 1)[35,36] and nonlinear ($n \neq 1$). Example of model with n = 2 is that of Liu[37] and with n = 3 is that of Steiner[38].

Models based on the flow structure are based on the superposition method and take into account the characteristics of the flow structure:

$$h_{df} = \frac{\theta_{\mathrm{seco}} h_v + (2\pi - \theta_{\mathrm{seco}}) h_{\mathrm{hum}}}{2\pi} \tag{26}$$

Where θ_{seco} is the angle of the dry perimeter, h_v is the heat transfer coefficient of the vapor phase, h_{hum} is the heat transfer coefficient of the wet perimeter corresponding to a non-linear superposition of effects taking into account the equivalent thickness of the liquid.

2.5 Empirical correlations

The correlations for the determination of the heat transfer coefficient in two-phase flow, in general, are based on heat transfer coefficients for liquid phase h_L[33,39] and on dimensionless criteria, that is:

$$h_{df} = h_L \cdot f(Bo, Fr, Bl, We, Co, X_{tt}, \ldots)$$

(27)

The mean absolute error (MAE) of the correlations is determined according to the expression[40,41]:

$$EAM = \frac{1}{Np}\sum_{1}^{Np}\frac{|Y_{cal} - Y_{\exp}|}{Y_{\exp}} \times 100\%$$

(28)

Where Np is the number of points (data) analyzed, Y_{cal} is the calculated value and $Y_{\exp}$ is the experimental value.

Of the expressions used, the one with the lowest mean absolute error is the equation proposed by Basu[42]:

$$h_{df} = 1.44 \times 10^5 (Bl^2 We_D)^{0.32} (\frac{\rho_l}{\rho_v})^{0.31}$$

(29)

3. Conclusions

(1) It is necessary to develop a general criterion based on the thermophysical properties of the fluids and on the operating conditions to know the boundaries between conventional channel, mini-channels and microchannels.

(2) The product of the frequency and diameter of bubble detachment is an order of magnitude smaller in mini-channels and microchannels than in conventional channels.

(3) The structure of boiling flow in mini-channels and micronals is similar to that of conventional channels: bubble flow, bullet flow and annular flow.

(4) For the determination of the two-phase heat transfer coefficient the Basu expression, equation (29), is the one with the lowest mean absolute error.

Conflict of interest

The author declared no conflict of interest.

References

1. Tuckerman DB, Pease RFW. High performance heat sinking for VLSI. IEEE Electron Device Letters 1982; 2(5): 126–129.
2. Kandlikar SG. Nucleation characteristics and stability considerations during flow boiling in microchannels. Experimental Thermal and Fluid Science 2006; 30(5): 441–447. DOI: 10.1016/j.expthermflusci.2005.10.001.
3. Mudawar I. Assessment of high-heat-flux thermal management schemes. IEEE Transactions on Components and Packaging Technologies 2001; 24(2): 122–141.
4. Thome JR. Boiling in microchannels: A review of experiment and theory. International Journal of Heat and Fluid Flow 2004; 25(2): 128–139. DOI: 10.1016/j.ijheatfluidflow.2003.11.005.
5. Shah RK, Sekulic DP. Fundamental of heat exchanger design. New Jersey, USA: John Wiley & Sons, INC.; 2003. p. 9.
6. Kaew-On J, Sakamatapan K, Wongwises S, *et al.* Flow boiling heat transfer of R134a in the multiport minichannel heat exchangers. Experimental Thermal and Fluid Science 2011; 35(2): 364–374.
7. Kandlikar SG. A roadmap for implementing minichannels in refrigeration and air-conditioning systems-current status and future directions. Heat Transfer Engineering 2007; 28(12): 973–985. DOI: 10.1080/01457630701483497.
8. Jiang P, Fan M, Si G, *et al.* Thermal-hydraulic performance of small-scale micro-channel and porous-media heat-exchangers. International Journal of Heat and Mass Transfer 2001; 44(5): 1039–1051.
9. Okawa T. Onset of nucleate boiling in mini and micochannels: A brief review. Frontiers in Heat and Mass Transfer 2012; 3: 013001. DOI: 10.5098/hmt.v3.1.3001.
10. Kandlikar SG. Similarities and differences between flow boiling in microchannels and pool boiling. Heat Transfer Engineering 2010; 31(3): 159–167. DOI: 10.1080/01457630903304335.
11. Mehendale SS, Jacobi AM, Shah RK, *et al.* Fluid flow and heat transfer at micro-and meso-scales with application to heat exchanger design. Applied Mechanics Reviews 2000; 53(7): 175–193.
12. Kandlikar SG, Balasubramanian P. An extension of the flow boiling correlation to transition, laminar, and deep laminar flows in minichannels and microchannels. Heat Transfer Engineering 2004; 25(3): 89–93. DOI: 10.1080/01457630490280425.
13. Kew PA, Cornwell K. Correlations for the prediction of boiling heat transfer in small-diameter channels. Applied Thermal Engineering 1997; 17(8–10): 705–715.
14. Thome JR, Dupont V, Jacobi AM, *et al.* Heat transfer model for evaporation in microchannels. Part I: Presentation of the model. International Journal of Heat and Mass Transfer 2004; 47(14–16): 3375–3385.
15. Barber J, Brutin D, Sefiane K, *et al.* Bubble confinement in flow boiling of FC-72 in a “rectangular” microchannel of high aspect ratio. Experimental Thermal and Fluid Science 2010; 34(8): 1375–1388.
16. Harirchian T, Garimella SV. Flow regime-based modeling of heat transfer and pressure drop in microchannel flow boiling. International Journal of Heat and Mass Transfer 2012; 55(4): 1246–1260. DOI: 10.1016/j.ijheatmasstransfer.2011.09.024.
17. Brauner N, Maron DM. Identification of the range

of small diameters, conduits, regarding two phase flow pattern transitions. International Communications in Heat and Mass Transfer 1992; 19(1): 29–39.
18. Lee J, Mudawar I. Critical heat flux for subcooled flow boiling in microchannel heat sinks. International Journal of Heat and Mass Transfer 2009; 52(13–14): 3341–3352.
19. Zhang W, Hibiki T, Mishima K, *et al.* Correlation for flow boiling heat transfer at low liquid reynolds number in small diameter channels. Journal Heat Transfer 2005; 127(11): 1214–1221.
20. Tibiriçá CB, Ribatski G. Flow boiling heat transfer of R134a and R245fa in a 2.3 mm tube. International Journal of Heat and Mass Transfer 2010; 53(11–12): 2459–2468.
21. Kandlikar SG. A scale analysis based theoretical force balance model for critical heat flux (CHF) during saturated flow boiling in microchannels and minichannels. Journal of Heat Transfer 2010; 132(8): 081501. DOI: 10.1115/1.4001124.
22. Cooke D, Kandlikar SG. Pool boiling heat transfer and bubble dynamics over plain and enhanced microchannels. Journal of Heat Transfer 2011; 133: 052902. DOI: 10.1115/1.4003046.
23. Liu G, Xu J, Yang Y, *et al.* Seed bubbles trigger boiling heat transfer in silicon microchannels. Microfluidics and Nanofluidics 2010; 8(3): 341–359.
24. Thome JR, Consolini L. Mechanisms of boiling in micro-channels: Critical assessment. Heat Transfer Engineering 2010; 31(4): 288–297.
25. Zhuan R, Wang W. Simulation on nucleate boiling in micro-channel. International Journal of Heat and Mass Transfer 2010; 53(1–3): 502–512.
26. Kuo CJ, Kosar A, Peles Y, *et al.* Bubble dynamics during boiling in enhanced surface microchannels. Journal of Microelectromechanical Systems 2006; 15(6): 1514–1527. DOI: 10.1109/JMEMS.2006.885975.
27. Fu X, Zhang P, Huang CJ, *et al.* Bubble growth, departure and the following flow pattern evolution during flow boiling in a mini-tube. International Journal of Heat and Mass Transfer 2010; 53(21–22): 4819–4831.
28. Karayiannis TG, Shiferaw D, Kenning DBR, *et al.* Flow patterns and heat transfer for flow boiling in small to micro diameter tubes. Heat Transfer Engineering 2010; 31(4): 257–275.
29. Lee M, Cheung LSL, Lee YK, *et al.* Height effect on nucleation-site activity and size-dependent bubble dynamics in microchannel convective boiling. Journal of Micromechanics and Microengineering 2005; 15(11): 2121–2129. DOI: 10.1088/0960-1317/15/11/018.
30. Cheng L, Ribatski G, Quibén JM, *et al.* New prediction methods for CO_2 evaporation inside tubes Part I—A two-phase flow pattern map and a flow pattern based phenomenological model for two-phase flow frictional pressure drops. International Journal of Heat and Mass Transfer 2008; 51(1–2): 111–124.
31. Harirchian T, Garimella SV. A comprehensive flow regime map for microchannel flow boiling with quantitative transition criteria. International Journal of Heat and Mass Transfer 2010; 53(13–14): 2694–2702. DOI: 10.1016/j.ijheatmasstransfer.2010.02.039.
32. Revellin R, Thome JR. A new type of diabatic flow pattern map for boiling heat transfer in microchannels. Journal of Micromechanics and Microengineering 2007; 17(4): 788–796.
33. Gnielinski V. New equations for heat and mass transfer in turbulent pipe and channel flow. International Chemical Engineering 1976; 16(2): 359–368.
34. Bertsch SS, Groll EA, Garimella SV, *et al.* A composite heat transfer correlation for saturated flow boiling in small channels. International Journal of Heat and Mass Transfer 2009; 52(7–8): 2110–2118.
35. Gungor KE, Winterton RHS. A general correlation for flow boiling in tubes and annuli. International Journal of Heat and Mass Transfer 1986; 29(3): 351–358.
36. Jung D, Radermacher R. Prediction of evaporation heat transfer coefficient and pressure drop of refrigerant mixtures. International Journal of Refrigeration 1993; 16(5): 330–338.
37. Liu Z, Winterton RHS. A general correlation for saturated and subcooled flow boiling in tubes and annuli, based on a nucleate pool boiling equation. International Journal of Heat and Mass Transfer 1991; 34(11): 2759–2766.
38. Steiner D, Taborek J. Flow boiling heat transfer in vertical tubes correlated by an asymptotic model. Heat Transfer Engineering 1992; 13(2): 43–69.
39. Shiferaw D, Karayiannis TG, Kenning DBR, *et al.* Flow boiling in a 1.1 mm tube with R134a experimental results and comparison with model. International Journal of Thermal Sciences 2009; 48(2): 331–341.
40. Basu S, Ndao S, Gregory J, *et al.* Flow boiling of R134a in circular microtubes-Part II study of critical heat flux condition. Journal of Heat Transfer 2011; 133(5): 051503. DOI: 10.1115/1.4003160.
41. Fang X, Shi R, Zhou Z, *et al.* Correlations of flow boiling heat transfer of R-134a in minichannels: Comparative study. Energy Science and Technology 2011; 1(1): 1–15.
42. Basu S, Ndao S, Michna GJ, *et al.* Flow boiling of R134a in circular microtubes-Part I study of heat transfer characteristics. Journal of Heat Transfer 2011; 133(5): 051502. DOI: 10.1115/1.4003159.