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Two-Phase Flows and Boiling Heat Transfer in Microchannels

Jong-Taek Oh†*, Ardiyansyah Ardiyansyah

*Department of Refrigeration and Air Conditioning Engineering, Chonnam National University, Yeosu, Chonnam 550-749, Korea
Graduate School, Chonnam National University, Yeosu, Chonnam 550-749, Korea

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Abstract

A study of literatures on flow boiling heat transfer and two-phase flows inside microchannels is summarized. The potential applications, fabrication method and efforts to determine certain dimensional threshold for microchannels classifications are discussed. For the last two decades, numerous two-phase flow and heat transfer models for micro-channels have been developed; many of them were derived from empirical models originally applied for conventional channels. Those models are discussed here along with a brief review on recent development of theoretical and phenomenological-based models for microchannels. This study is devoted to provide a review of important issues on flow boiling heat transfer and two-phase flows inside microchannels, including two-phase flow patterns, boiling heat transfer mechanism and correlations developments, pressure drop and prediction methods, and critical heat flux.

Key words: Microchannels, Boiling, Heat Transfer, Pressure drop, Two-phase flow

Nomenclature

C Chisholm's parameter
D diameter, mm
G mass flux, kg/m^{2}s
Co confinement number
g gravitational acceleration, 9.81 m/s^2
h heat transfer coefficient, kW/m^{2}K
i enthalpy, kJ/kg
K empirical constant for subcooled CHF
k thermal conductivity, kW/mK
L length of test section, m
M molecular weight, kg/kmol
n parameter used by Kandiklar-Steinke
P pressure, kPa
q heat flux, kW/m^2
R gas constant, Nm/kg.K
T temperature, K
We Weber number, $lG^2/\rho\sigma$
X Martinelli parameter
x mass quality

Greek Letters

μ dynamic viscosity, Ns/m^2
ρ density, kg/m^3
σ surface tension, N/m
ϕ^2 two-phase frictional multiplier
θ_{dry} dry angle of channel perimeter, rad
(dp/dz) pressure gradient, Pa/m

Subscripts

c critical
h hydraulic
l liquid
v vapor
co subcooled CHF
lo liquid only flow
max maximum
NcB nucleate boiling
nuc nucleation
sat saturation
sub subcooled
TP two-phase
wet wetted

*Corresponding author. Tel.: +82-61-659-3273, Fax.: +82-61-659-3003
E-mail address: ohjt@chonnam.ac.kr

1. Introduction

Research on heat transfer characteristic inside microchannels has become important topic as the advancement in the field of micro-scale technology such as micro electro-mechanical system. The impacts of miniaturization of the system involving high heat transfer processes create challenges for better system's thermal management that can only be solved by enhancement of the performance and efficiency of applied heat exchanger devices.

Microchannels have been considered as the potential solution for such devices due to their high thermal performance characteristic with low space requirements. As for current development, the fabrication methods of microchannels, which limit the microchannels applications in the early developments, have emerged beyond the ability to model the thermal system on the microchannels of such scale. Micromachining of silicon wafers, micro-extrusion of aluminum elements [1], anisotropic wet chemical etching and sawing, anisotropic dry etching, hybridization and system-on-chip integration [2] are examples of micro-fabrication methods capable of manufacturing microchannels.

Early research in microchannels heat exchanger can be traced back to works by Tuckerman and Pease [3] for a very-large-scale-integrated circuits cooling. Representative studies in two-phase flow inside minichannels and microchannels are summarized in Table 1. In fact, microchannels application can be found in cooling devices for computer's microprocessors and electronic components, laser diodes, radar and aviation components, and microchemical reactors. Some of microchannels geometries used in electronic cooling applications are depicted in Fig. 1 from Kandlikar [4]. For larger system, microchannels have been employed in compact heat exchanger for air conditioning system in automotive and domestic applications. Various microchannel configurations are used in these applications, including important feature of refrigerant headers which provide minimum effects on the upstream flow during two-phase flows inside the microchannel passages.

Despite the extensive applications of microchannels in heat exchanger devices, as for current applications, the devices are being implemented based on extensive testing which are expensive and time consuming. There has not been a fully-reliable thermal design methods for such applications due to the limited understanding in thermal-hydraulic phenomena occurred in micro-scale channels, importantly for two-phase flow boiling heat transfer mechanism.

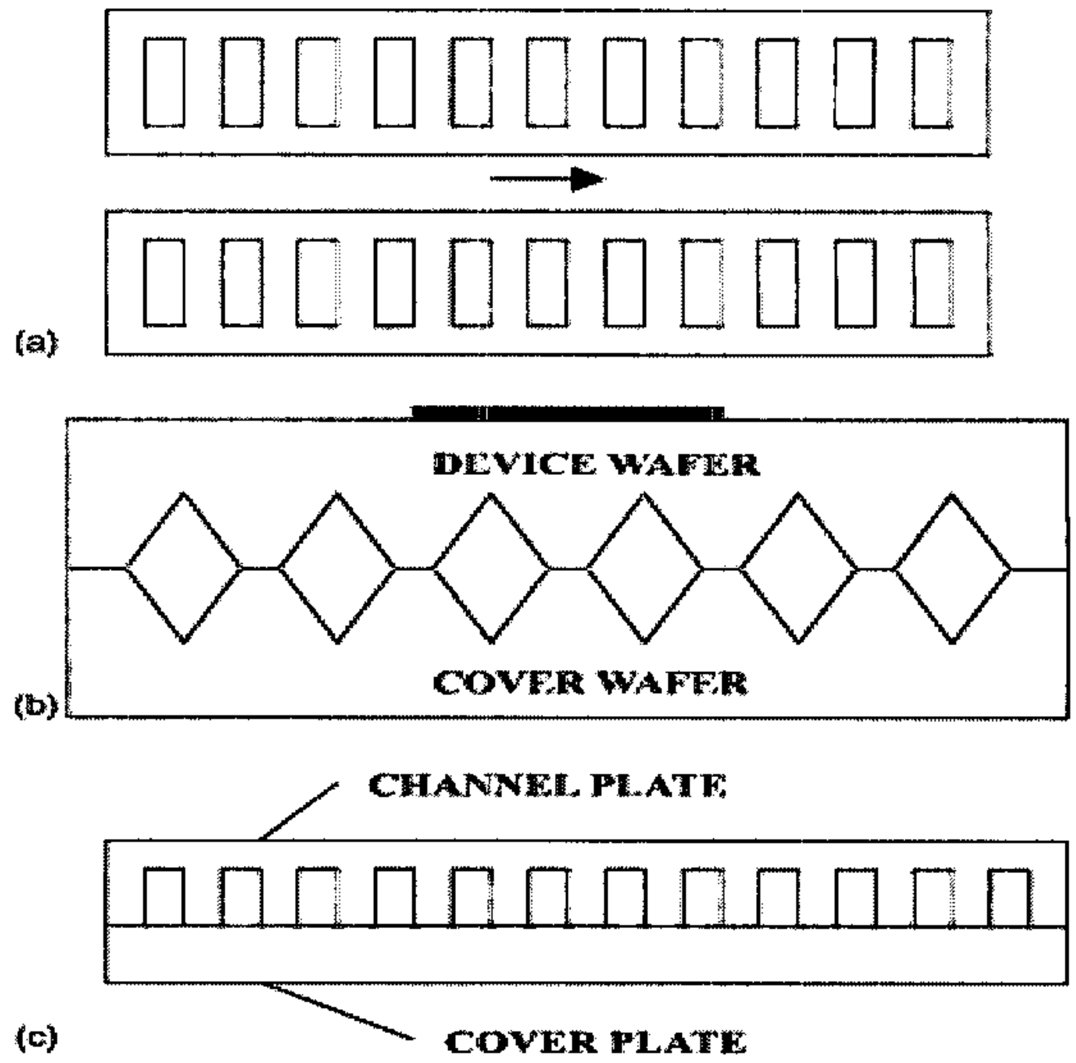

Fig. 1. Microchannels geometries in electronic cooling applications [4].

Literatures study of flow boiling heat transfer and two-phase flows inside microchannels are presented in the following sections. This study is devoted to provide a review of important issues on microchannels, including two-phase flow patterns, flow boiling heat transfer mechanism and model developments, two-phase pressure drop and prediction methods, and critical heat flux. Other issues which are also addressed here include microchannels classifications, flow boiling experimental data and trends and flow instabilities in microchannels.

2. Channel classifications

The heat transfer and two-phase flow phenomena inside microchannels show different characteristics compared to inside conventional channels. As numerous theoretical and experimental studies have been devoted to macro-scale two-phase flow and heat transfer models applied to conventional channels, it is necessary to determine such a threshold on which the model can be applied with acceptable reliability. Below the threshold, micro-scales models should be applied. However, a well-defined threshold for the transition of macro-to-micro-scale models has not been established yet despite several efforts made on microchannels classifications as discussed below.

The microchannels classification based simply on

Table 1. Summary of investigations on two-phase flow in minichannels and microchannels.

Author	Channel geometry	Fluid, test ranges G in kg/m^{2}s; q in kW/m^2	Remarks
Tuckerman and Pease (1981)	Compact-parallel heat sink w= 50-56μm; h= 300 μm; L= 10	Water Flow rate: 4.7-8.6 cm^3/s q= 1810-7900	Early research on heat transfer inside microchannels with single phase liquid flow. HTC were predicted using conventional relations
Lazarek and Black (1982)	Horizontal, single circular-channel D= 3.1; L= 123 and 246	R113 G=125-750 q=14-380	Heat transfer mechanisms during subcooled and saturated boiling were nucleate and convective boiling. HTC relatively constant values for range of quality; dependent on heat flux. Pressure drop and CHF model were developed.
Wambganss et al. (1993)	Horizontal, single circular-channel D= 2.92; L= 368	R113 G=50-300 q=8.8-90.75	High heat flux and low mass flux are inherent in small channels which results in high boiling numbers. Slug flow pattern is dominance over large range of parameters. Both lead to domination by a nucleation mechanism.
Bowers and Mudawar (1994)	Parallel heat sink D= 0.51 and 2.54	R113 Flow rate: 0.28-1.1 ml/s q=1000-2000	Experimental studies of heat transfer and pressure drop performance inside mini and microhannels
Mishima and Hibiki (1996)	Vertical, single circular-channel D= 1- 4 L=210-1000	Air-water 0.0896 $<j_G<$ 79.3 m/s 0.0116 $<j_L<$ 1.67 m/s	Flow pattern observation in minichannel with the transition predicted by Mishima-Ishii model. Frictional pressure drop using Chisholm model with a new C value as a function of diameter
Kew and Cornwell (1997)	Horizontal, single circular-channel D= 1.39-3.69 L=500	R141B G=188-1480 q=16.6-90	HTC increases with heat flux at low quality, while at higher qualities the heat transfer coefficient is a function of quality and independent of heat flux. Nucleate boiling was indicated. Introduce Co=0.5 as the threshold for narrow channel
Triplett et al. (1998)	Horizontal, single circular-channel D=1.1 and 1.45 semi-triangular Dh=1.09 and 1.49	Air and water 0.02 $<j_G<$ 80 m/s 0.02 $<j_L<$ 8 m/s	Flow pattern observation in small diameter channels. Void fraction was measured and predicted well with homogenous model, especially for bubbly and slug. Pressure drop correlation was developed.
Kattan et al. (1998)	Horizontal, single circular-channel D=12 and 10.92 L=3013	R134a, R123, R402A, R404A, R502 G=100-500 q=0.44-36	Flow pattern observation and map development. HTC increased with quality at low quality, peak point and drop at high quality; dependent on heat flux; proposed new flow pattern-based HTC model.
Serizawa and Feng (2001)	Horizontal circular-channel D=50 and 25μm	Air/steam-water 0.0012 $<j_G<$295.3 m/s 0.003 $<j_L<$ 17.52 m/s	Flow pattern observation in microchannel: observed liquid ring flow and liquid lump flow
Kawahara et al. (2002)	Horizontal circular-channel D=100 μm L=64.5	De-ionized water-nitrogen gas 0.1 $<j_G<$60 m/s 0.02 $<j_L<$ 4 m/s	Flow pattern observation in microchannel; gas core flows with a smooth/ring-shaped film or serpentine-like gas core surrounded by a deformed liquid film. Proposed probability-based flow pattern map
Qu and Mudawar (2003)	Horizontal parallel rectangular heat sink 231x713 μm	De-ionized water G=135-402	Saturated flow boiling HTC was strongly dependent on mass flux; weak dependent of heat flux; annular flow dominant in moderate and high quality; dominance of forced convective heat transfer; HTC decrease at higher quality; Introduced annular flow HTC correlation.
Zhang et al. (2004)	Data from 13 experiments; single horizontal, vertical; circular and rectangular D=0.78-6	Data from 13 experiments; water, R11, R12, R113 G=23.4-2939 q=2.95-2511	Two-phase flow in minichannels was liquid-laminar flow and gas-turbulent flow. Saturated flow boiling HTC was predicted well with modified Chen's correlation for the above flow conditions, i.e. Reynolds number factor and single-phase HTC h_{sp} in Chen's superposition model.

Qu and Mudawar (2004)	Horizontal parallel rectangular heat sink 215x821 μm	De-ionized water G=86-368	CHF study in microchannels. Flow instabilities and vapor back-flow were reported. CHF increase with mass flux, independent of inlet subcooling, CHF correlation were developed with additional data of R113 based on Katto-Ohno
Choi et al. (2007)	Horizontal, single circular channel D=1.5 and 3.0 L=2000	R22, R134a, CO_2 G=200-600 q=10-40	Saturated flow boiling HTC in minichannel were reported. Nucleate boiling was predominant in low quality; dependent on mass flux and heat flux; HTC correlation was developed based on Chen's superstition model.
Cheng et al. (2006) and Cheng et al. (2007)	Data from 6 experiments; horizontal, single and multi channel; circular and rectangular D=0.6-10	CO_2 G=50-1500 q=1.8-46	Flow pattern map for flow boiling CO_2 were developed. Phenomenology-based pressure drop and HTC models for CO_2 were proposed.

channel dimensions and its use in various applications were proposed by Mehendale et al. [5] as follows:

Conventional passage $D_h > 6$ mm
Compact passage 6 mm $\geq D_h > 1$ mm
Meso-channel 1 mm $\geq D_h > 100$ μm
Microchannel 100 μm $\geq D_h > 1$ μm
Where D_h is hydraulic diameter

Kandlikar and Grande [2] employed their classification based on the mean free path of molecules in the single-phase flow, surface tension effects and flow patterns in the two-phase applications. They considered the flow characteristics on which the rarefaction effects of the gas phase become important in microchannels and channels with smaller hydraulic diameter.

Their classification is as follows:
Conventional channel $D_h > 3$mm
Minichannels 3 mm $\geq D_h > 200$ μm
Microchannels 200 μm $\geq D_h > 10$ μm
Transitional channels 10 μm $\geq D_h > 0.1$ μm
  Transitional microchannels 10 μm $\geq D_h > 1$ μm
  Transitional nanochannels 1 μm $\geq D_h > 0.1$ μm
Molecular nanochannels 0.1 μm $\geq D_h$

Another method was introduced by Kew and Cornwell [6], which instead of using fixed dimensional channel as the threshold, they suggested physical criterion for the classification of microchannels. They used bubble confinement as the criterion of micro and macro-scale heat transfer. They suggested the use of confinement number *Co*, where the effects of confinement become significant at channels with hydraulic diameter such that *Co* is above 0.5.

$$Co = \frac{\left[\sigma/\left(g\left(\rho_l - \rho_v\right)\right)\right]^{1/2}}{D_h} > 0.5 \qquad (1)$$

Chinov and Kabov [7] proposed similar method as above to classify channel sizes but with smaller defined threshold for minichannels to microchannels transition. Their classification is as follows:

Large-scale channels D > $5l\sigma$
Gravity-capillary channels $0.5l\sigma <$ D $< 5l\sigma$
Capillary-gravity or minichannels $0.1\, l\sigma <$ D $< 0.5l\sigma$
Capillary or microchannels D $< 0.1l\sigma$
Where $l\sigma$ is a capillary constant defined as:

$$l_\sigma = \left[\sigma/\left(\left(\rho_l - \rho_v\right) g.\cos\xi\right)\right]^{1/2} \qquad (2)$$

and ξ is channel slope to the vertical.

Lin and Pisano [8] suggested the complete suppression of boiling nucleation, such that bubbles could not form, as the microchannel threshold. The nucleation radius, r_{nuc} used was defined as:

$$r_{nuc} = \frac{2\sigma}{\Delta T_{nuc}\left(dP/dT\right)_{sat}} \qquad (3)$$

where ΔT_{nuc} is the nucleation superheat. Based on this criterion, the threshold of micro-to-nanoscale was assigned to $D_h < 2r_{nuc}$.

An insight on the flow pattern consideration as the mean of channel's classification was given by Serizawa and Feng [9] from their works on two-phase flow inside 0.05 mm glass channel with steam-water flow. They observed a new defined flow pattern type, namely liquid ring flow. Similar type was also found in 0.002 mm channel but with another type of flow pattern, namely liquid lump flow. These results can be used as the criteria of microchannels classifications.

Their findings support the classification by Kandlikar and Grande [2] although transition zone between each classification is required and the actual thresholds need to be defined by further research.

Figure 2 shows a comparison of threshold diameter of carbon dioxide flows inside a microchannel. The diameter ranges of microchannels proposed by Chinov and Kabov [7] varies from 150 μm at low reduced pressure to about 10 μm at high reduced pressure, comparable to the ranges of 200-10 μm and 100-1 μm thresholds defined by Kandlikar and Grande [2] and Mehendale et al. [5], respectively. However, the classification from Kew and Cornwell [6] for macro-to-micro scale in CO_2 case gives much large values, i.e. from 3 mm to 200 μm at various reduced pressures.

These comparisons indicate there has been no distinct method on how to determine the threshold of microchannels diameter as the transition of micro-to-microscale two-phase flow and heat transfer phenomena. However, the Kandlikar and Grande [2] classification of microchannels will be used throughout the following sections for their simplicity and comparable channel's dimension ranges to other classifications as shown in figure 2.

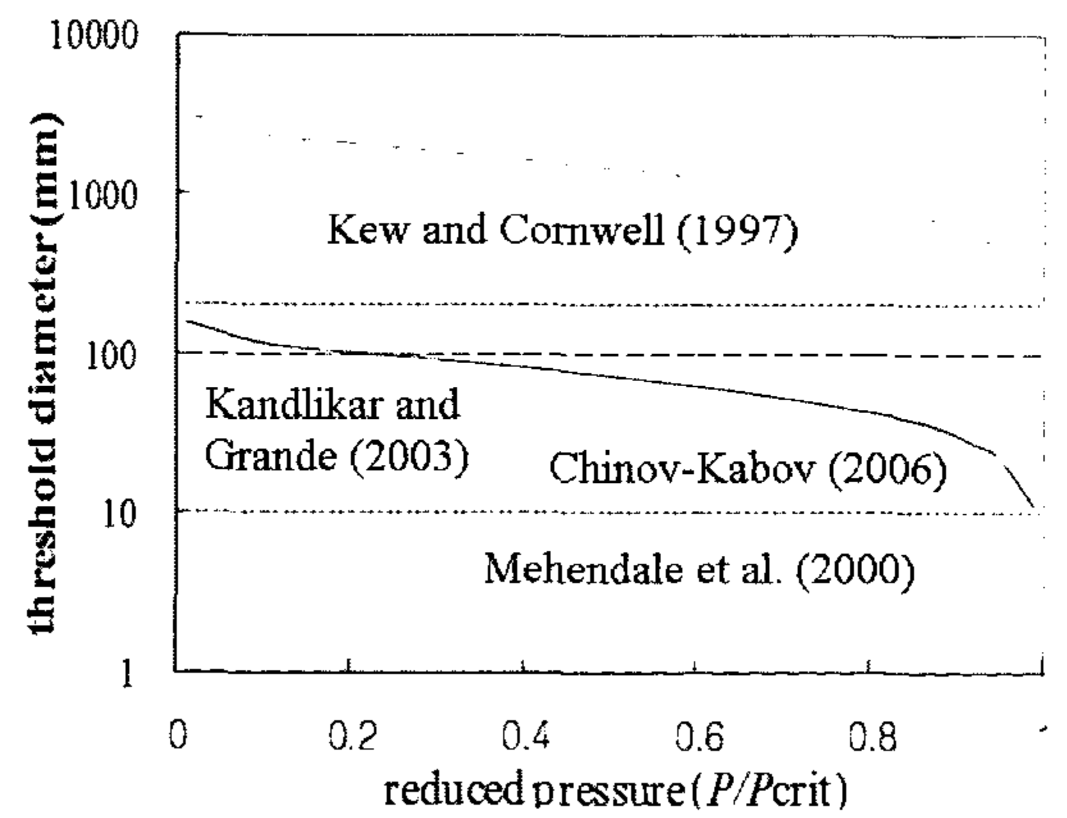

Fig. 2. Comparison of microchannels classification using CO_2:
.... Kandlikar-Grande [2], --- Mehendale et al. [5],
_._ Kew-Cornwell [6], ___ Chinov-Kabov [7]

3. Two-phase flow patterns

Identification of flow patterns provides important insight to form the basis of two-phase flow model in flow boiling inside microchannels. The flow pattern-based heat transfer and pressure drop prediction model through void fraction determination have shown considerable developments recently. The following discussions aim to give a brief summary of flow patterns studies and efforts to represent them in applicable flow-pattern maps. In addition, their uses as the basis of two-phase flow model developments will also be addressed.

Starting from the early efforts in mapping the flow regimes of two-phase flow inside conventional tubes of Baker (1954), the flow patterns of air-water flow inside horizontal tubes were represented as areas on graph. Taitel and Dukler [10], one of the most widely applied flow pattern map, introduced the map with Lockhart-Martinelli parameter, X as the horizontal axis and different parameters for each transition criteria for the vertical axis. They designated the flow regimes as bubbly, intermittent (plug/slug flow), stratified, wavy and annular flows, along with theoretically-based transition criteria.

Two-phase flow in evaporation affects the flow due to the introduction of acceleration pressure drop, especially at high heat fluxes, hence different characteristics in the flow pattern map and transitions for evaporation can be expected. Kattan et al. [11] developed a diabatic flow pattern map for evaporation of various refrigerants in horizontal tubes. Their comprehensive flow pattern database included both single and mixture refrigerants evaporation in 12.0 mm diameter and 3 m length cooper tube. Their two-phase flow designations include stratified, stratified wavy, intermittent, annular mist flow and transitions between them.

Their proposed flow pattern map was of Steiner's type and used vapor quality and mass velocity as horizontal and vertical axis of the map, respectively. Their map takes into account the effects of heat flux to the onset of dryout at the top of the tube during evaporation in horizontal tube. In addition, they argued that it can also be used for adiabatic flows. Further development of the map was discussed in Zurcher et al. [12] with additional flow boiling data of substitute refrigerants R134a and R407C and natural refrigerant ammonia. Their modifications include flow pattern transitions of stratified to stratified-wavy, intermittent to annular and stratified-wavy to annular.

Wojtan et al. [13] further improved the map by considering dynamic void fraction measurement from their experiment. Their map added annular-to-dryout and dryout-to-mist flow transition curve based on distinct trend of heat transfer coefficients. Figure 3 showed R-22 flow boiling heat transfer experimental data from Choi et al. [14] in their map.

The flow pattern maps described above were built from two-phase flow databases inside channels with larger diameter than any microchannel classifications explained earlier.

It is found that the designation of flow patterns in conventional and microchannels are almost similar, except for the stratified flow-type which does not exist in microchannels due to the dominance of surface tension effects and negligible gravitation effects as the channel's diameter decreases. However the flow- pattern maps for conventional channels have not been able to predict the flow pattern transitions in microchannels. Two-phase flow pattern designation as bubbly, slug, and annular flow which are evident in conventional channels were also observed in microchannels.

Kawahara et al. [15] suggested flow patterns designations from their work on adiabatic gas-liquid mixture flow inside a circular microchannel of 100 μm diameter. A liquid ring flow-type was observed which only specifically found in microchannels. The flow patterns designations are depicted in figure 4. Their results were plotted on flow pattern map with superficial gas and liquid velocity as horizontal and vertical coordinate respectively.

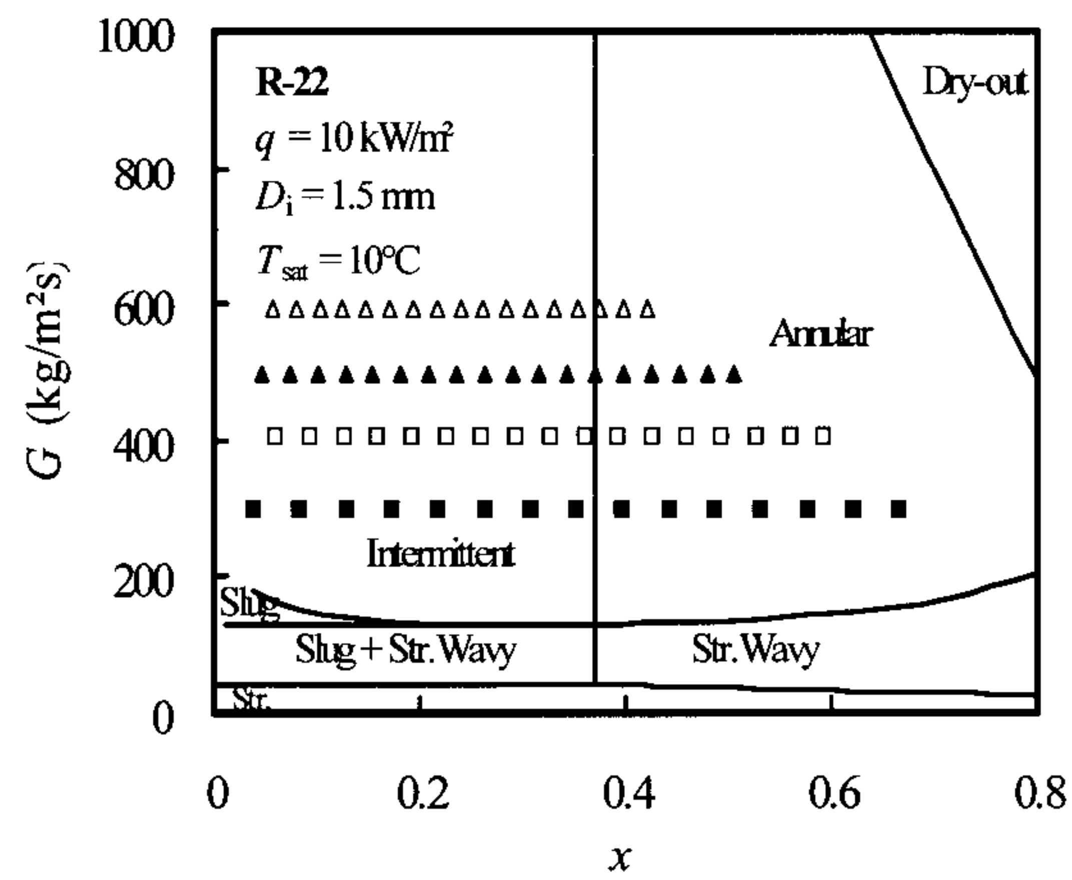

Fig. 3. Experimental data from Choi et al. [14] mapped in flow pattern map of Wojtan et al [13].

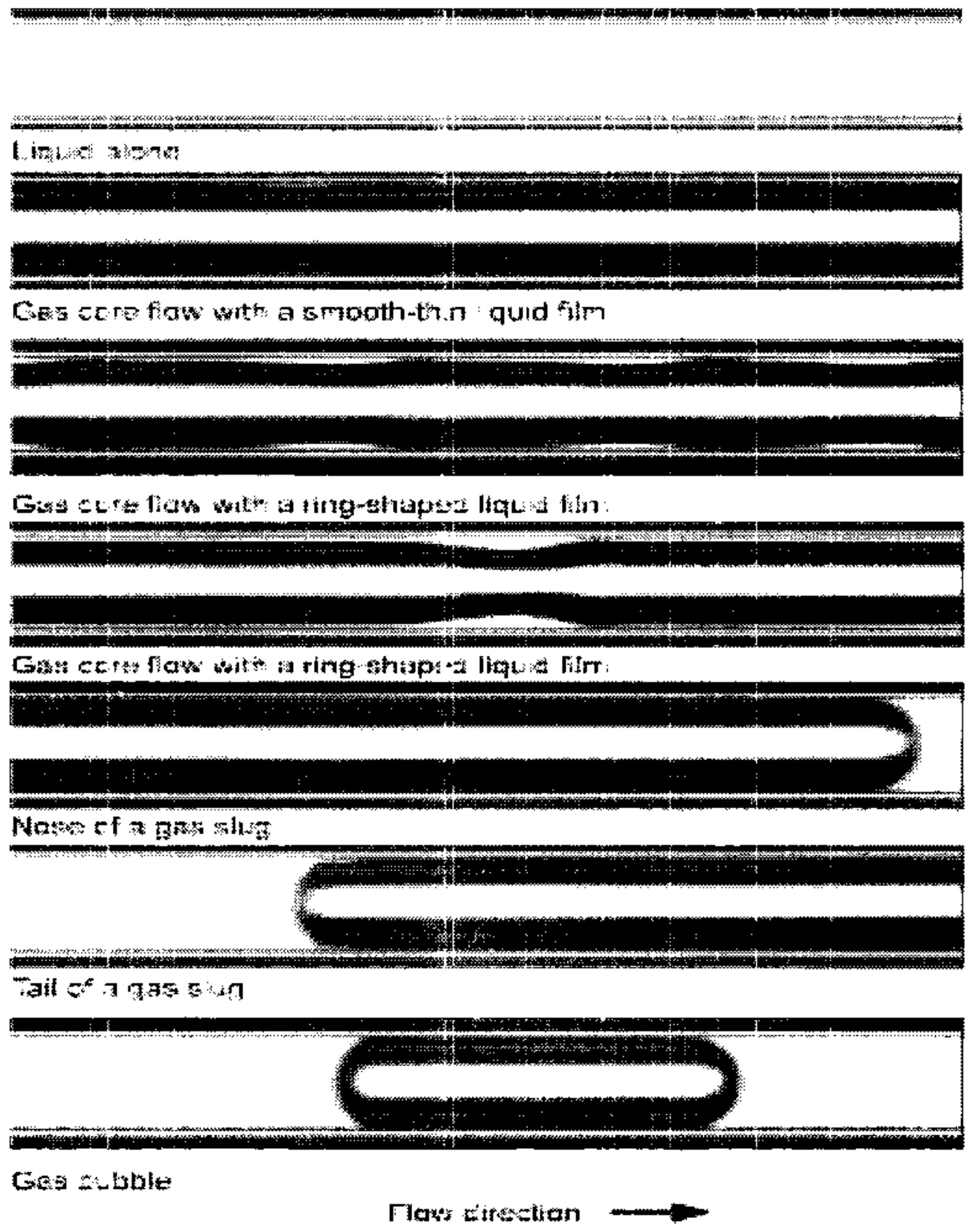

Fig. 4. Two-phase flow in a 100 μm diameter microchannel at liquid and gas flow rates are 0.15 m/s and 6.8 m/s respectively, from Kawahara et al. [15]. The designated flow patterns are: liquid alone, gas core with a smooth-thin liquid film, gas core with a smooth thick liquid film, gas core with a ring-shaped liquid film and gas core with a deformed interface.

Recently, Revellin and Thome [16] proposed a diabatic flow patterns classification based on Revellin [17] experimental data for R 134a and R245fa flow boiling inside 0.509 mm and 0.790 mm diameter channels and applied heat flux up to 597 kW/m^2. Observation of four principal flow patterns was reported i.e. bubbly, slug, semi-annular and annular flow along with transition regimes between each of these flow patterns. Their flow pattern map was an advancement of three-zone heat transfer model of Thome et al. [18] and Dupont et al. [19], which were limited to elongated bubble (slug) flow to include and generalize the model of the following flow patterns: (i) isolated bubble regime, includes both bubbly and/or slug flows, (ii) coalescing bubble regime, where the bubble coalescence is predominant, and (iii) annular regime, where the annular flow exists until the critical vapor quality corresponding to the onset of critical heat flux.

Further study on the phenomenology-based heat transfer and two-phase flow have been focused on the effort to integrate such flow pattern maps into the heat transfer and pressure drop models for flow boiling inside microchannels. One of the studies was reported recently by Cheng-Ribatski-Thome [20] for CO_2. Their flow boiling model was developed from boiling database of 0.6 mm to 10 mm diameter channels, wide ranges of mass velocity from 50 to 1500 kg/m^2s and heat flux range from 1.8 to 46 kW/m^2. However, more researches and experimental data for smaller diameters are needed to validate the implementation of the flow pattern map in microchannels dimensions. Moreover the flow instabilities observed in two-phase flow inside microchannels and their influences to flow transitions put challenges for flow pattern map developments.

4. Flow boiling heat transfer

As already widely recognized in microchannel, the surface tension effects are predominant in flow boiling heat transfer characteristics as opposite to gravitational effects in larger diameter channels. Even though such different mechanisms between microchannels and conventional channels are obvious, heat transfer correlations or models based on conventional channel have been widely used -with various modifications- to predict flow boiling heat transfer inside microchannels, sometimes with reasonable statistical fit. However it is becoming critical to develop heat transfer model which is based on specific characteristics of flow boiling inside microchannels.

The most prominent flow boiling heat transfer model which has been used widely is the superposition model of Chen. The model suggested that the flow boiling heat transfer mechanism was governed by nucleate boiling and convective evaporation contributions. The nucleate boiling contribution was obtained from Foster-Zuber pool boiling correlation and was considered to be suppressed with suppression factor S. The suppression factor was defined as the ratio of mean superheat to the wall superheat temperature. The convective evaporation was calculated by Dittus-Boelter equation for single-phase liquid turbulent convection inside tube. The convective contribution was corrected by factor F, a convection multiplier as a function of Martinelli parameter. Numerous researchers have proposed modification of this model, such as Zhang et al. [21] and Choi et al. [14] to account for laminar flow consideration for flow boiling inside mini and microchannels.

Another model which has been applied for conventional channels is asymptotic model of nucleate boiling heat transfer (h_{NcB}) and convective heat transfer (h_c) contribution in the form of:

$$h_{TP} = \left[(h_{NcB})^n + (h_c)^n \right]^{1/n} \tag{4}$$

The value of n was defined as 2 by Kutateladze using power law method, while Steiner-Taborek [22] used n=3. On one of the method presented in Collier and Thome [23], Gnielinski correlation was used to obtain h_c and Gorenflo type of relationship for h_{NcB}.

Kandlikar [24] proposed set of empirical correlations for heat transfer prediction inside horizontal and vertical tubes from extensive database of 24 experimental investigations. The model utilized non-dimensional convection and boiling number and introduced a fluid-dependent parameter F_{fl}. For the application in mini and microchannel, the correlations were modified in Kandlikar and Steinke [25] considering laminar flow occurred in small diameter channels due to the small hydraulic diameter and low mass flux commonly employed in microchannels applications. The all-liquid flow Nusselt number used in their correlation was given by:

$$Nu_{lo} = \frac{h_{lo}.D_h}{k_l} = n \tag{5}$$

where n is dependent on the channel geometry and the wall thermal boundary condition, for example, for circular channel under constant heat flux and constant temperature boundary condition, the value of n were defined as 4.36 and 3.66 respectively.

Kattan et al. [11] proposed a flow pattern-based heat transfer prediction model for horizontal tubes as follows:

$$h_{TP} = \frac{\theta_{dry}.h_v + (2\pi - \theta_{dry}).h_{wet}}{2\pi} \tag{6}$$

According to them, flow boiling (h_{wet}) occurs on the wetted tube wall perimeter and vapor-phase forced convection (h_v) occurs on the dry tube wall perimeter represented by dry angle θ_{dry}. The flow boiling heat transfer can take the form of asymptotic model explained earlier and the convective evaporation component is obtained from film flow model and local void fraction of Steiner model. The model was further developed by Wojtan et al. [13] by modifying the expression for the dry angle and extending the model for dryout and mist flow regimes.

The models explained above consider the nucleate boiling heat transfer mechanism takes important role in flow boiling heat transfer and suggest this mechanism become more and more important as the channel's diameter decreases as for the case of microchannels.

A different view on what mechanism governs the flow boiling heat transfer inside microchannels was proposed by Jacobi and Thome [26]. They suggested that the transient thin film evaporation in the elongated bubble mode governs the heat transfer in microchannels, instead of nucleate boiling mechanism noted earlier. Their model was further developed in Thome et al. [18] with three-zone flow boiling model, namely liquid slug, evaporating elongated bubble and vapor slug as shown in fig. 5.

The heat transfer mechanism includes liquid convection, vapor convection at the presence of dry zone and thin film evaporation as the main contributions. The model distinct approach considers the transient variation in the local heat transfer coefficient during sequential and cyclic passage, in opposite of the all-regime empirical models. Ribatski et al. [27] made comparison of extensive experimental data from 17 different studies of which most of them were performed in the test section with hydraulic diameter between 200 μm to 3 mm. Their results reported the three-zone model as the most accurate models, although with 50% of mean absolute error and only predicted 45% of the data within ± 30% error band. So far the model only covers heat transfer coefficient prediction in the elongated bubble regimes (slug flow) in circular channels; hence further researches are needed to extent for other flow regimes and channels geometries to develop a comprehensive phenomenology-based heat transfer coefficient prediction model.

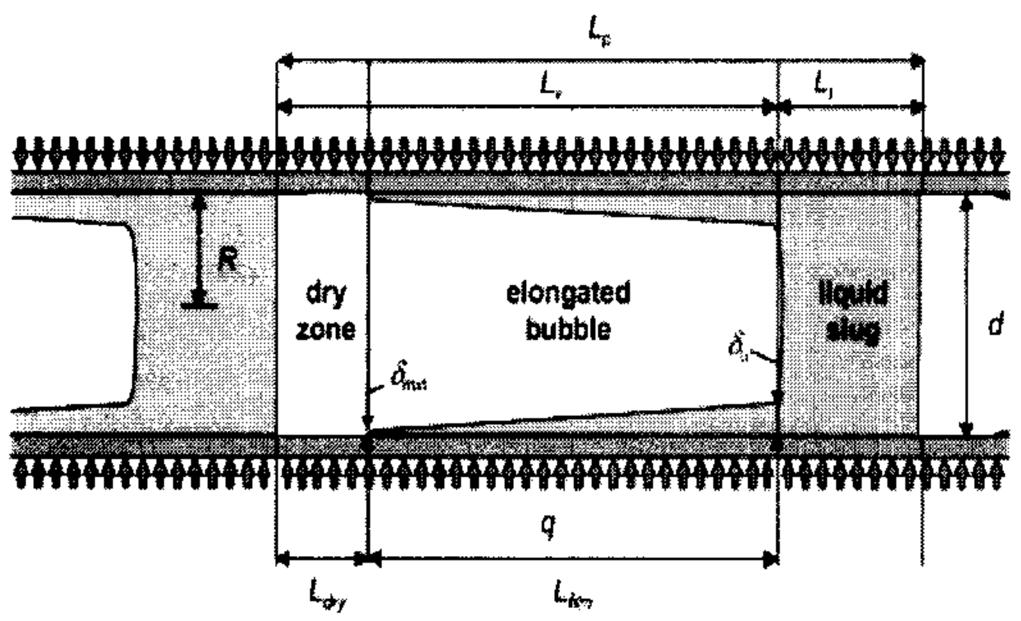

Fig. 5. Three-zone flow boiling model for elongated bubble flow regime in microchannels: liquid slug, elongated bubble and vapor slug, from Thome et al. [18].

Different trends of local heat transfer coefficient versus vapor quality and the effects of mass flux and heat flux in flow boiling inside microchannels have been reported until date. Analysis by Agostini and Thome [28] on 13 studies concluded that at low to medium vapor qualities, the heat transfer coefficient increase with the heat flux and decrease or relatively constant with respect to vapor quality. At the higher vapor quality, the heat transfer coefficient decreases sharply with vapor quality and independence on heat flux or mass flux. From the literatures studied, there are always positive effects of heat flux to the heat transfer coefficient, except at high vapor quality, while the mass flux effect varies from no effect, an increasing effect or a decreasing effect.

5. Pressure drop prediction

One of the drawbacks in flow boiling inside microchannels is its higher pressure drop due to friction increase compared to larger diameter channels. It is essential to develop two-phase pressure drop models since the pressure drop, together with void fraction, are the most important aspects of two-phase flow. In addition, the model is the basic element for the design of microchannels heat exchanger devices.

Collier and Thome [23] described two main approaches to model frictional pressure drop in two-phase flow inside tubes, namely homogenous model and separated-flow model. The homogenous model has been used in various forms in the steam generation, petroleum and refrigeration industries for a considerable time. The model considers the two-phase flow as a single phase possessing mean fluid properties. The two-phase mean viscosity used to determine two-phase frictional factor can be obtained from one of the correlations by McAdams et al. [29], Cicchitti et al. (1960) or Dukler et al. (1964).

The basic assumptions and derivations are explained in Collier and Thome [23]. The homogenous model was examined by Triplett et al. [30] using correlation from McAdams et al. [29] for the homogenous mixture's viscosity, while Friedel [31] correlation was used for the two-phase frictional multiplier. Their works had been developed from their flow pattern experimental results of air-water mixture flows within 1.1 and 1.45 mm diameter of circular channels and 1.09 and 1.49 mm hydraulic diameter of semi-triangular channels. They found that the homogenous model showed the best agreement for bubbly and slug flows but over-predicted the frictional pressure drop results for annular flow.

The separated flow model considers the phases to be artificially segregated into liquid and vapor stream. The simplest approach for this model suggests each stream flows at a mean velocity. Lockhart-Martinelli (1949) developed separated-flow model which proposed an empirical approach to determine the two-phase frictional multipliers ϕ^2 as a function of parameter X, where,

$$X^2 = \frac{\left(\frac{dp}{dz}F\right)_l}{\left(\frac{dp}{dz}F\right)_v} \tag{7}$$

$$\phi^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{8}$$

The value of Chisholm's parameter, C is a constant which depends on flow characteristic of each phase. For the two-phase flow in microchannels, the value of C would be 5 because liquid and gas flows are in laminar conditions. Mishima and Hibiki [32], using data of air-water two-phase flow inside circular and rectangular channels with hydraulic diameter range of 1-4mm, have proposed a correlation for C:

$$C = 21\left(1 - e^{-0.319D_h}\right) \tag{9}$$

Kawahara et al. [15] suggested the value of C=0.24 for their two-phase flow of de-ionized water and nitrogen inside a 100μm diameter microchannel.

Pressure drop in multi-microchannels showed fluctuations due to flow instabilities in the upstream section as addressed in Bergles and Kandlikar [33]. These instabilities may results in flow reversals (vapor back flow) which significantly affect the local pressure drop. The instabilities were classified into upstream compressible volume instability and excursive instability. The former instability is an oscillating flow which may lead to CHF, occurred when there is a significant compressible volume at the upstream of the heated test section. In two-phase flow inside microchannels the compressible volume may be caused by an entrained bubble, flexible hose or large volume of removed gases from liquid flow. The excursive instability, also called fundamental static instability, occurred in flow boiling inside microchannels due to unique pressure drop characteristics of boiling channels (see Fig. 5). Both instabilities can be reduced by utilizing throttle valve prior to the test sections even though this would require higher pressure drop and fabrication considerations.

Recently, Ribatski et al. [27] compared 12 well-known frictional pressure drop models of various channel diameters with experimental data from 9 different authors of which including data from mini and microchannels experiments. According to them, model proposed by Muller-Steinhagen and Heck [34], which were developed for conventional channels, showed the best results with 53.1% data were predicted and mean absolute deviation of 31.3%. The second best results were shown by homogenous model which utilizing two-phase viscosity suggested by Cicchitti et al (1960) together with the model proposed by Mishima-Hibiki [32]. They also found that for vapor qualities higher than 0.6, all models showed poor prediction results.

Their analyses indicated that the existing models can not predict the pressure drop characteristic in flow boiling microchannels with reliable results. Furthermore, the models failed to capture the effects of flow patterns transitions in microchannels on the pressure drop results.

6. Critical heat flux

Application of microchannels for heat exchanger devices with high heat dissipation and small space requirements need comprehensive understanding in critical heat flux of the two-phase flow in relation with maximum feasible heat removal ability and required coolant flow rate of such devices. At critical heat flux condition, the heat transfer coefficient drops abruptly on the surface on which boiling occurred, leads to sudden large increase in surface temperature, and may cause catastrophic failure to the system. Examples of these applications can be found in cooling devices for computer's microprocessor, advanced power and switching devices and laser diodes arrays.

Experimental method reported in CHF studies mainly conducted for stable flow inside single, thin and uniform wall thickness, circular tubes. Constant heat flux boundary condition is obtained by direct electrical heating to the tube's wall. The heat flux is increased slowly until CHF point on which vapor blanked and rapid rise in temperature and physical burnout of the surface occurred. According to the bulk fluid condition at channel outside, CHF can be classified as subcooled or saturated CHF. Another method using fluid heating (constant surface temperature) can also be applied with similar CHF values [33].

A widely used correlation for CHF was developed by Katto-Ohno [35] for single-conventional channel. They correlate the subcooled CHF as the function of liquid to vapor density ratio, liquid Weber number (We_l) and heated length (L_h) to diameter ratio as follows:

$$\frac{q_{co}}{Gi_{lv}} = f\left[\frac{\rho_l}{\rho_v}, We_l, \frac{L_h}{D}\right] \tag{10}$$

For saturated CHF, considering linear relation to liquid subcooling, the correlation become:

$$q_c = q_{co}\left(1 + K\frac{\Delta i_{inlet}}{i_{lv}}\right) \tag{11}$$

Where Δi_{inlet} is the inlet enthalpy of subcooling with respect to saturation and K is an empirical constant for subcooling. The correlation of CHF in conventional channel described above has been used as an alternative to develop correlation for microchannels. However, the low mass flux commonly found in microchannels together with unique pressure drop characteristics and flow instabilities (as will be discussed in latter section), lead to the need of further researches in these aspects.

Qu and Mudawar [36] found that Katto-Ohno correlation showed a fairly accurate prediction for saturated CHF data of single-circular-channel with diameter range of 1-3 mm for water and 3.15 mm for R113.

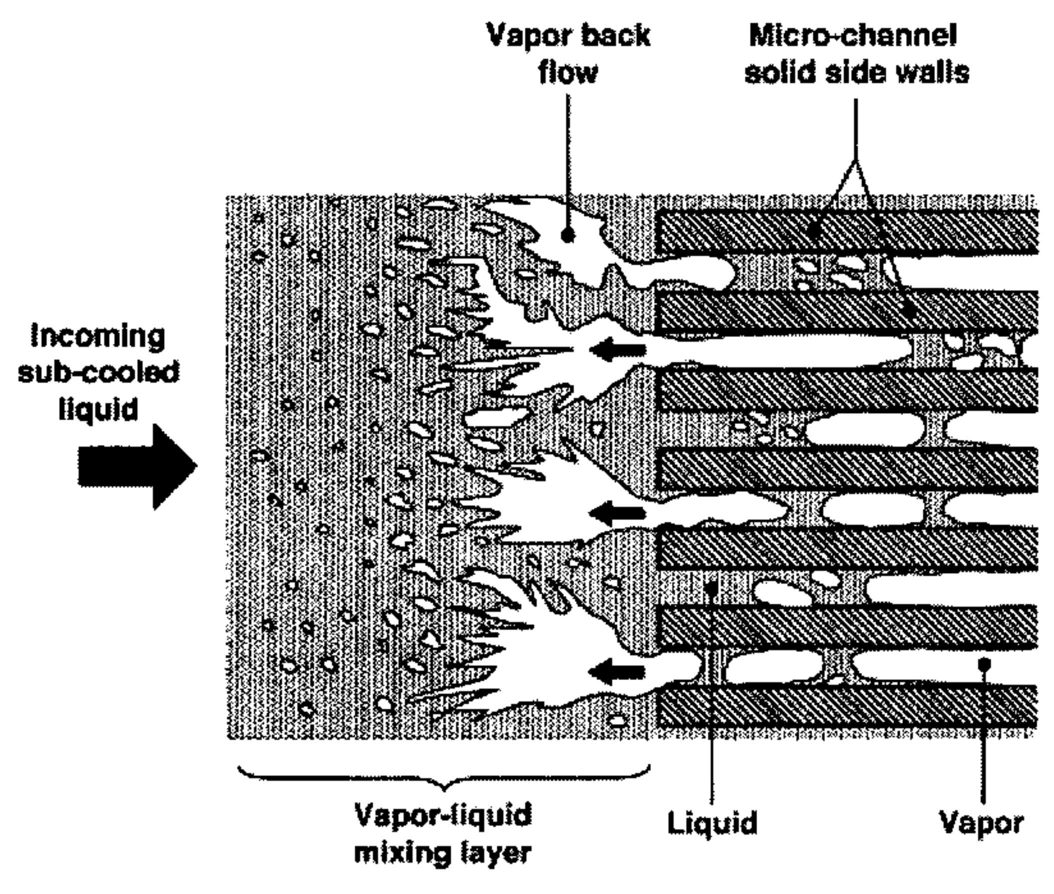

Fig. 6. Schematic representation of vapor backflow from Qu and Mudawar [36], considered to be onset of excursive instability by Bergles and Kandlikar [33].

However, the correlation largely over predicted their parallel microchannels heat sink CHF data. They argued that the deviation could be attributed to the flow instabilities occurred at the parallel channels (as previously discussed in pressure drop prediction section of this paper), as well as the rectangular shape of the channels.

Based on CHF data from their parallel microchannels experiments and Bowers and Mudawars [37], Qu and Mudawar [36] developed saturated CHF correlation as follows:

$$\frac{q_{co}}{Gi_{lv}} = 33.43\left(\frac{\rho_l}{\rho_v}\right)^{1.11} We_l^{-0.21}\left(\frac{L_h}{D}\right)^{-0.36} \tag{12}$$

They results indicated that the CHF was proportional to the mass velocity and for both mini- or microchannels heat sink databases, no effect was shown for the liquid inlet temperature. This was due to the loss of subcooling because of flow instabilities prior to the CHF in parallel channels, as described earlier, caused backflow of vapor into the upstream plenum which mixed with the inlet liquid and increased their temperature near to local saturation temperature.

Wojtan et al. [38] proposed a modification of Katto-Ohno correlation for CHF in single microchannel as:

$$\frac{q_c}{Gi_{lv}} = 0.437\left(\frac{\rho_l}{\rho_v}\right)^{0.073} We_l^{-0.24}\left(\frac{L_h}{D}\right)^{-0.72} \tag{13}$$

And the critical vapor quality could be obtained by:

$$x_c = \frac{q_c}{G(i_{lv} + i_{sub})}.4\frac{L_h}{D} \tag{14}$$

Their experimental data were saturated CHF of R-134a and R-245fa flow inside 0.5 and 0.8 mm diameter microchannels. Their results indicated strong dependence of CHF on mass velocity, heated length and microchannel diameter, but no influence of liquid subcooling. The authors noted that the correlation could be applied for $\rho v/\rho l < 0.15$ and for annular flow regime only.

Recently, a theoretical-based CHF model was proposed by Revellin and Thome [39]. Their assumptions were that the CHF occurs at two conditions: (i) dryout of the liquid film during evaporation in annular flow and (ii) vapor share overcoming surface tension forces to remove the liquid film from the wall at wavy flow transition. Their model were then compared to CHF experimental data of single circular microchannel from Wojtan et al. [38] and Lazarek and Black [40], and microchannels heat sink data from Bowers and Mudawar [37] and Qu and Mudawar [36]. Overall comparison showed good results with 89.6% of the all database were predicted within a 20% error band and mean absolute error of 9.3%. Their model simulation showed similar trend commonly seen in conventional channel experimental data, e.g. higher CHF at increased mass flux and channel diameter, and lower CHF at increased heated length and saturation temperature. The model also indicated limited influence of inlet subcooling, as shown also in the previous studies. Despite the ability to predict CHF for single and parallel microchannels and to be applied under non-uniform heat flux condition, the model involves non linear system of ordinary differential equations and requires numerical computation to solve.

The reported CHF from all the authors above are still far below the theoretical limit of the CHF suggested by Gambill and Lienhard [41] as follows:

$$q_{\max} = \rho_v i_{lv} \sqrt{RT/2\pi M} \tag{15}$$

Hence further improvements are still widely open to increase the operating CHF, such as increasing coolant mass flux and reduced pressure, reducing heated length or using multi-microchannels configurations.

7. Conclusion

The review presents literatures study of two-phase flow and boiling inside microchannels. There have been numerous classifications proposed to determine certain dimensional threshold for microchannels diameter, however no distinct method has been proven yet to set the transition of macro-to-micro scale two-phase flow and heat transfer phenomena occurred in microchannels.

While some flow patterns designations for conventional channels show similar patterns in microchannels, the flow pattern maps developed from conventional channels have not been able to predict flow patterns transitions observed in microchannels. Hence, it is important to develop flow pattern maps specifically for microchannels. Moreover, the maps are required for the development of flow patterns-based heat transfer and pressure drop prediction models which have attracted considerable attentions recently.

Numerous flow boiling heat transfer models have been suggested in the literatures. Many of them are based on conventional channel's correlations which have been modified for microchannels applications. The nucleate boiling heat transfer mechanism has been considered to take important role in flow boiling heat transfer inside microchannels and has been accommodated in their heat transfer models. However recent studies have also reported the transient thin film evaporation in the elongated bubble mode as the main mechanism that governs flow boiling heat transfer inside microchannels. Heat transfer models based on this mechanism are still being developed.

The homogenous and separated flow models to predict pressure drop in conventional and microchannels have been applied with different level of success. Analyses of existing models have indicated that they can not predict the pressure drop characteristic in flow boiling microchannels with reliable results. Furthermore, the models fail to capture the effects of flow patterns transitions in microchannels on the pressure drop results. Hence, together with heat transfer models, further researches on the flow-pattern based pressure drop prediction method are required.

Critical heat flux in flow boiling inside single and parallel microchannels have been discussed in this paper. The reported CHF values are still far below the suggested theoretical limit of the CHF. Hence, further improvements are still widely open to increase the operating CHF of microchannels, especially in parallel microchannels heat sink.

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