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Article *in* Sadhana · December 2013

DOI: 10.1007/s12046-013-0192-7

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Sādhanā Vol. 38, Part 6, December 2013, pp. 1083–1126.

Two-phase flow boiling in small channels: A brief review

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MS received 11 July 2012; revised 17 January 2013; accepted 31 January 2013

Abstract. Boiling flows are encountered in a wide range of industrial applications such as boilers, core and steam generators in nuclear reactors, petroleum transportation, electronic cooling and various types of chemical reactors. Many of these applications involve boiling flows in conventional channels (channel size $\geq$ 3 mm). The key design issues in two phase flow boiling are variation in flow regimes, occurrence of dry out condition, flow instabilities, and understanding of heat transfer coefficient and vapor quality. This paper briefly reviews published experimental and modeling work in these areas. An attempt is made to provide a perspective and to present available information on boiling in small channels in terms of channel size, flow regimes, heat transfer correlations, pressure drop, critical heat flux and film thickness. An attempt is also made to identify strengths and weaknesses of published approaches and computational models of boiling in small channels. The presented discussion and results will provide an update on the state-of-the-art and will be useful to identify and plan further research in this important area.

Keywords. Boiling; two-phase flow; small channel; dry-out; heat transfer; instabilities.

1. Introduction

Boiling in small channels is a topic of current interest with its application to steam tubes in boilers, compact evaporators, compact heat exchangers, tubes in refrigerant industries, and microelectronics devices with heat pipes etc. Practical development of these devices is creating a need of better understanding of transport phenomenon in small channels. In each application it is observed that a phase-change heat transfer is playing a vital role. Phase change process particularly flow boiling are important in the micro and macro scale devices but relatively little work has been done in this area. Systems employing small channels offer many advantages such as compact physical size, small thermal resistances between the device and coolant, high heat transfer coefficients, reduced inventory requirements, low capital cost etc. Boiling flows inside small channels provide a very effective way of heat transfer; generally more than single phase flow.

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The rapid development of small, mini and micro fluidic devices is motivating the demand for comprehensive understanding of transport mechanisms involved in boiling through such small channels.

Flow boiling has been studied extensively in the last few decades. However, flow boiling heat transfer and two-phase flow transport phenomena in small, mini and micro hydraulic diameter channels are quite different from conventional large channels. Some of the industrial applications falling over a channel size ranging from 0.33 mm to 20 mm are shown in figure 1a–b.

Basically, two-phase flow boiling heat transfer in tubes or channels can be characterized by either nucleate or convective boiling component or both. All three possibilities have been

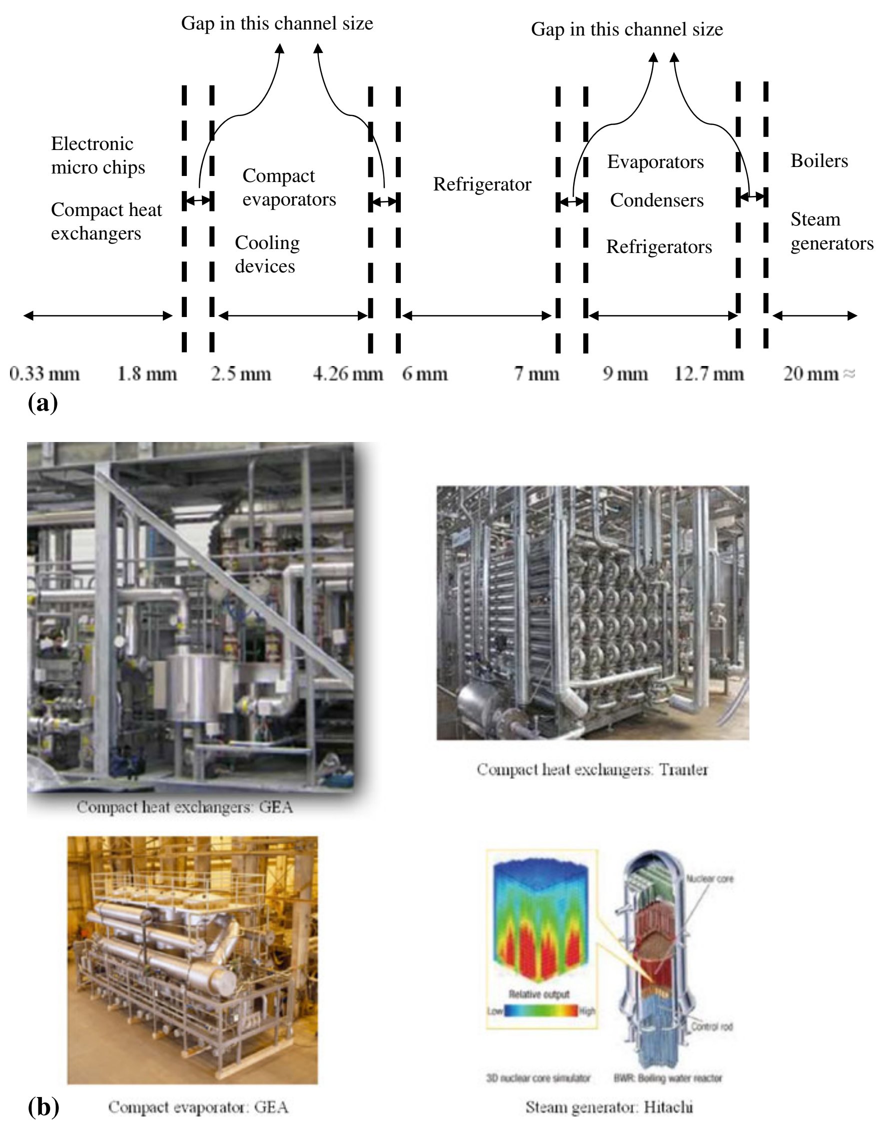


Figure 1. (**a**) Schematic representation of gaps identified in literature on channel size. (**b**) Pictorial view of industrial equipments where two-phase flow is shown.

reported based on experiments under different operating parameters. Over the past decades, a lot of work has been done to understand the fundamental aspects of two-phase flow and flow boiling of mixtures in various channel sizes. These include mainly experiments on flow boiling and development of heat transfer correlations for mixtures. A widely used approach is to assume that heat is transferred by a combination of nucleate boiling and convective boiling. In nucleate boiling, thin liquid films are created cyclically under small bubbles growing in contact with the wall at individual nucleation sites and the wall superheat. The heat transfer coefficient increases nonlinearly with increasing heat flux but it is insensitive to the flow rate through the channel. In convective boiling, heat is transferred from the wall through the laminar sub-layer to the turbulent flow of the mixture of liquid and bubbles that have detached from the wall and coalesced and results into different flow regimes as shown in figure 2.

As may be seen from figure 2, a sub-cooled liquid enters the tube (heated uniformly along its length at a relatively low heat flux) from the bottom and then completely evaporates over the length of the tube. While the liquid is being heated up to its saturation temperature, the wall temperature initially is below that necessary for nucleation (zone A). Thus, the heat transfer process in zone A is sub-cooled, single-phase heat transfer to the liquid, which may be laminar or turbulent. Then the wall temperature rises above the saturation temperature and boiling

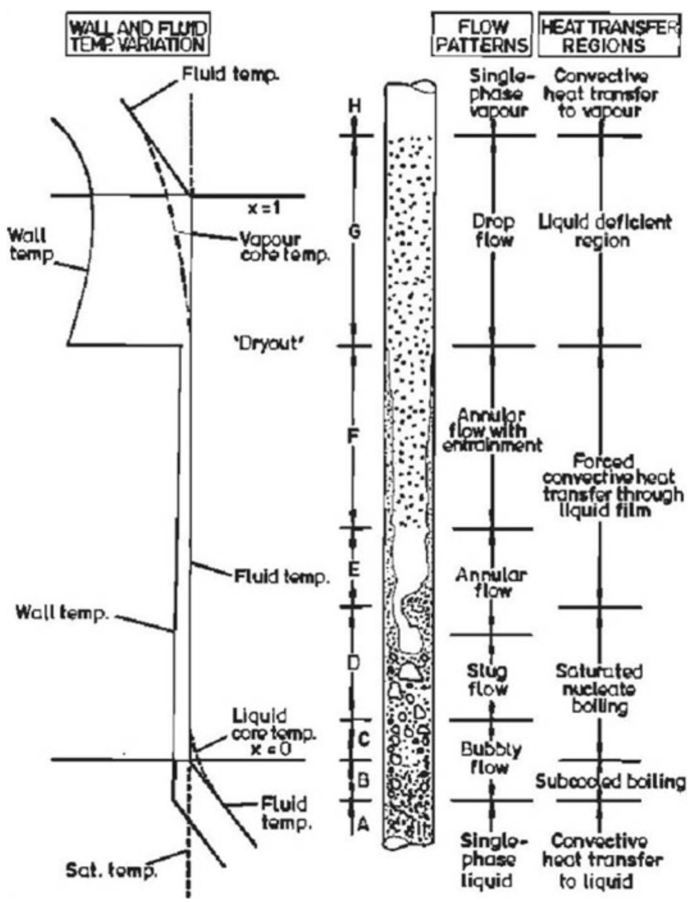


Figure 2. Heat transfer regions in convective boiling in a vertical tube from Collier & Thome (1994).

nucleation takes place in the superheated thermal boundary layer on the tube wall, such that sub-cooled flow boiling occurs in zone B with the vapour bubbles condensing as they drift into the sub-cooled core. The liquid then reaches its saturation temperature and saturated boiling in the form of bubbly flow begins in zone C. Saturated boiling continues through the slug flow regime (zone D), the annular flow regime (zone E) and then the annular flow regime with liquid entrainment in the vapour core (zone F). At the end of zone F, the annular film is either dried out or sheared from the wall by the vapour, a point that is referred to as the onset of 'dry-out' or simply dry-out. Above this point, mist flow in the form of entrained droplets occurs with a large increase in wall temperature of an imposed wall heat flux (zone G). The temperature of the continuous vapour phase in zone G tends to rise above the saturation temperature where the vapour quality is 1.

As can be seen from figure 2, two-phase flow boiling occurring inside a tube leads to be unstable because of random movements of liquid and vapour. Two-phase flow instabilities induced during flow boiling in thermal systems are undesirable as they can result in mechanical vibrations and system control problems, affect normal operation, restrict operating parameters and influence system safety. The proper characterization of these instabilities and the condition for their occurrence would facilitate optimal and safe operation of the involved systems. The safe operating regime of a two-phase flow can be determined by the instability values of system parameters such as flow rate, pressure, wall temperatures, and exit mixture quality.

As mentioned above, design of small channel depends on heat transfer coefficient, flow regimes, heat and mass fluxes and vapour quality. Despite significant research efforts, it is observed that traditional heat transfer coefficient correlations for saturated boiling with liquid-turbulent and gas-turbulent flows may not be suitable to predict heat transfer coefficients in small channels due to possibility of laminar flow. Although many analytical and experimental studies related to flow boiling heat transfer in small channels have been performed during the past decades, adequate systematic and accurate data bases are still unavailable and reliable heat transfer coefficient correlation applicable to a wide range of conditions in small-channels is often difficult to identify.

In the present review, we focus specifically on boiling in small channels. Existing experimental and modelling studies on flow boiling in small channels are critically reviewed. Summary of published results are presented in the form of easy to read and easy to assimilate tables. We try to identify key conclusions as well as the key gaps from already published studies and suggest some general issues to keep in mind in any further research on boiling in small channels.

2. Two-phase flow boiling

We focus on "classification" of channels based on their sizes due to a general lack of agreement in the definition of such classification as can be seen in existing literature. The need for such a classification is important for comparative studies of two-phase flow boiling in a given category of "channel" under various experimental conditions.

In the following sub-sections, we discuss briefly the role of channel size and several other parameters necessary in the understanding of two-phase flow.

2.1 *Channel size*

As mentioned above, boiling inside small passages provides a very effective way of fluid movement, as well as relatively large heat dissipation capabilities for specialized applications.

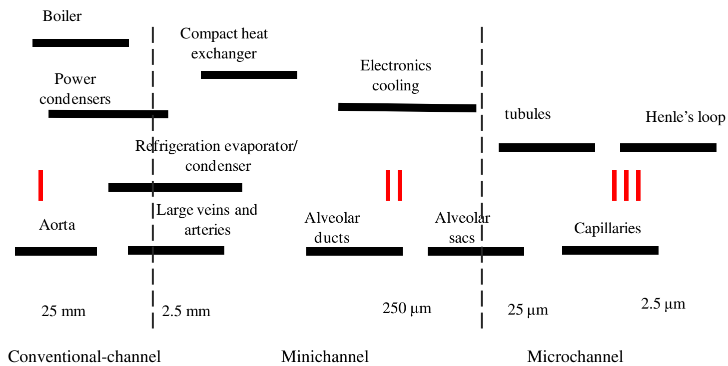


Figure 3. Ranges of channel diameters employed in various applications, Kandlikar & Steinke (2003).

Therefore it is prudent to distinguish between small, mini, and micro channels based on engineering practice and application areas employing these channels. Kandlikar & Steinke (2003) showed ranges of channel dimensions employed in various systems (figure 3) and also few applications as per channel dimensions (table 1).

Table 1. Applications of boiling in small channels.

Author and year	Channel size	Fluid	Applications
Tran *et al* (1996)	Circular – 2.46 mm Rectangular – 1.76 × 4.06 mm	R12	Compact evaporators
Peles & Haber (2000)	Circular – 300 μm	Water	Micro heat pipes
Lee & Lee (2001)	Rectangular – 0.4 × 2 mm	R113	Compact heat exchangers
Lin *et al* (2001)	Circular – 1.1, 1.8, 2.8, 3.6 Square – 2 × 2 mm	R141b	Compact evaporators and condensers
Zhang *et al* (2004)	Circular – 0.78 to 6 mm	Water and R11, R12 and R113	Compact heat exchangers, cooling of electronic chips
Huo *et al* (2004)	Circular – 4.26 and 2 mm	R134a	Micro devices, compact heat exchangers and devices for cooling electronic equipment
Kandlikar & Balasubramanian (2004)	Circular – 2.92, 1.1, 1.8, 0.51 mm	R113, R141b	Electronics cooling and fuel cells
Bruno & Bontemps (2005)	11 parallel rectangular channels – 3.28 × 1.47 mm	R134a	Compact heat exchangers in air conditioning systems
Balasubramanian & Kandlikar (2005)	Six parallel channels – 1054 × 197 $\mu\mathrm{m}^2$	Water	Effect of microgravity used in space application
Zhang *et al* (2006)	Circular – 0.33 to 6.22 mm	Water	Compact heat exchangers, electronic chips, nuclear fusion reactors
Shiferaw *et al* (2007)	Circular – 2.01 and 4.26 mm	R134	Compact evaporators
Shiferaw *et al* (2009)	Circular – 1.1 mm	R134a	Compact evaporators
Koichi *et al* (2004)	3–6 mm vertical tube	Sub cooled water	Nuclear reactors

However there is no universal agreement on the threshold channel size that distinguishes small and conventional channels. Shah (1986) defined heat exchanger of a hydraulic diameter <6 mm as 'compact'. Mehendale *et al* (2000) considered the hydraulic diameter range of 1–6 mm for describing compact heat exchangers.

Thus according to both Shah (1986) and Mehendale *et al* (2000), the threshold hydraulic diameter distinguishing between small and conventional sizes is 6 mm.

Based on engineering practice and application areas such as refrigeration industry, compact evaporators employed in automotive, aerospace, air separation and cryogenic industries, Kandlikar (2001, 2002) defined the tube diameter ranges from 3 to 25 mm. It was observed that for all the data was given by Kandlikar (2001, 2002) was in the turbulent region considering all flow in the liquid phase. As the tube diameter becomes smaller, liquid flow Reynolds number falls in the laminar region.

According to Kandlikar (2001, 2002) and Kandlikar & Grande (2003), the channel size defining the boundary between small and conventional channel is 3 mm.

Similarly, various other studies quote threshold diameters for distinguishing small channels from conventional ones. However their values depend on the specific experimental conditions (see table 2).

2.2 *Dimensionless numbers*

Many researchers think the criterion ought to be based on the combination of channel size and fluid thermo-hydraulic properties rather than only on channel dimension. Apart from channels size there are several important dimensionless numbers which are used to represent the feature of fluid flow in small channels (see table 3). According to these dimensionless numbers, the distinction between small diameter and normal size channels may be classified as well. In the study of small channels, it is observed that the various hydrodynamic forces, e.g., inertia, viscosity, buoyancy and surface tension, all play an important role concerning the motion of the liquid and vapor flowing in the channel. These forces are taken into account for various non-dimensional groups. Non-dimensional groups are useful in arriving at key basic relations among system variables that are valid for different fluids even under different operating conditions. Some of these groups have been derived empirically, often on the basis of experimental data. Fundamental considerations of the governing forces and their mutual interactions lead to non-dimensional groups that provide important insight into the physical phenomenon.

Triplett *et al* (1999) defined the scales in flow channels in terms of the Laplace constant L defined as

$$L = \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}}.$$

Here σ is surface tension, g is gravitational acceleration, and ρ_l and ρ_v are liquid and vapour densities respectively.

Kew & Cornwell (1997) proposed Confinement number, Co for small diameter channels:

$$\mathrm{Co} = \frac{1}{D_h}\sqrt{\frac{4\sigma}{g(\rho_l - \rho_v)}}.$$

They reported that two-phase flow exhibits different flow and heat transfer characteristics when the confinement number, Co is greater than 0.5. For instance, isolated bubbles prevail when $\mathrm{Co} > 0.5$ and form a typical flow regime in small tubes, i.e., confined bubble flow.

Table 2. Threshold hydraulic channel diameters serving as distinction between small and conventional channels.

Author and year	Threshold size, mm	Remarks
Shah (1986)	6	Hydraulic diameter based on definition of compact heat exchanger as the one offering surface area to volume ratio higher than 700 m^2/m^3
Mehendale *et al* (2000)	6	Based on experimental work on R134a in a heat exchanger consisting of an inlet manifold, 52 parallel channels, and an exit manifold
Kandlikar (2001, 2002)	3	Based on the mean free path of molecules in the single-phase flow, surface tension effects, and flow patterns in the two-phase flow applications. 3–25 mm range studied for turbulent flow
Wongwises *et al* (2000)	7.5	Based on studies in evaporative heat transfer involving HFC 132a in horizontal tubes
Cheng & Mewes (2006)	3 to 15.8 for water 1 to 6.8 for R134a 6 for mixture fluids	Studied single component fluid as well as mixtures over a range of reduced pressures. Threshold diameters found decreasing as reduced pressure approached unity

Similarly, Brauner & Moalem-Maron (1992) used the Eotvos number Eo for identifying the criterion to distinguish channel sizes. They stated that when

$$\mathrm{Eo} = \frac{(2\pi)^2\sigma}{(\rho_l - \rho_v)D_h^2 g} > 1$$

Table 3. Dimensionless numbers in two-phase boiling.

Dimensionless parameter	Formula	Physical significance
Bo (boiling number)	$q/(m \times h_{fg})$	Ratio of heat exchanged with the surroundings to the heat that would be liberated by the complete vapourization of the input liquid
Co (confinement number)	$\mathrm{Co} = \frac{1}{D_h}\sqrt{\frac{4\sigma}{g\,(\rho_l - \rho_v)}}$	The criterion that enables the determination of the critical size of the channel defining the shift from isolated bubble regime to confined bubble regime for a given fluid pressure
Eotvos number	$\mathrm{Eo} = \frac{(2\pi)^2\sigma}{(\rho_l - \rho_v)D_h^2 g}$	Ratio of capillary pressure to inter-phase differential hydrostatic head
Nu (Nusselt number)	$\alpha d_e/k$	Ratio of the convective heat transfer coefficient to the conductive heat transfer coefficient
We (Weber number)	$(G^2L/\rho\sigma)$	It is a measure of the relative importance of the fluid's inertia compared to its surface tension
Nu_f (Film Nusselt number)	$(\alpha\delta/k)$	Ratio of the convective heat transfer coefficient to the conductive heat transfer coefficient for film

surface tension dominates, and this marks the boundary for small passages. Triplett *et al* (1999) found that stratified flow became impossible when Eo > 100 in their experiments indicating size effects. Akbar *et al* (2003) summarized the previous studies and concluded that the buoyancy effect can be negligible when the Bond number is less than 0.3 where flow regimes are insensitive to the channel orientation.

In fact, all the above dimensionless numbers, Co, Eo and Bo (reported in table 3), consider a combined effect of fluid density, surface tension and channel size on two-phase flow in associating a scale for the system under various experimental conditions.

2.3 *Flow regimes*

The flow pattern maps available in literature were first developed for the petrochemical industry (Baker 1954) for flow of oil and gas in large diameter pipes. Subsequently, the adiabatic flow pattern maps were developed as general flow pattern maps (e.g. Taitel *et al* 1980). In recent years, a number of flow pattern maps have been developed for specific conditions such as small diameter tubes, evaporation or condensation, compact heat exchanger geometries, etc.

For large diameter tubes, although the generalized flow pattern map for an air–water system, developed by Mandhane *et al* (1974) specifically for horizontal tubes, was to a large extent, representative of other flow conditions as well, the theoretically based transition criteria presented by Taitel & Dukler (1976) is one of the most widely used flow pattern maps today. Whereas Hewitt & Roberts (1969) developed flow regime map for vertical flow as shown in figure 4, for wide range of flow rates and the fact that they correlated both air/water data at atmospheric pressure and steam/water flow at high pressure. Hewitt (2000) gives a comprehensive summary

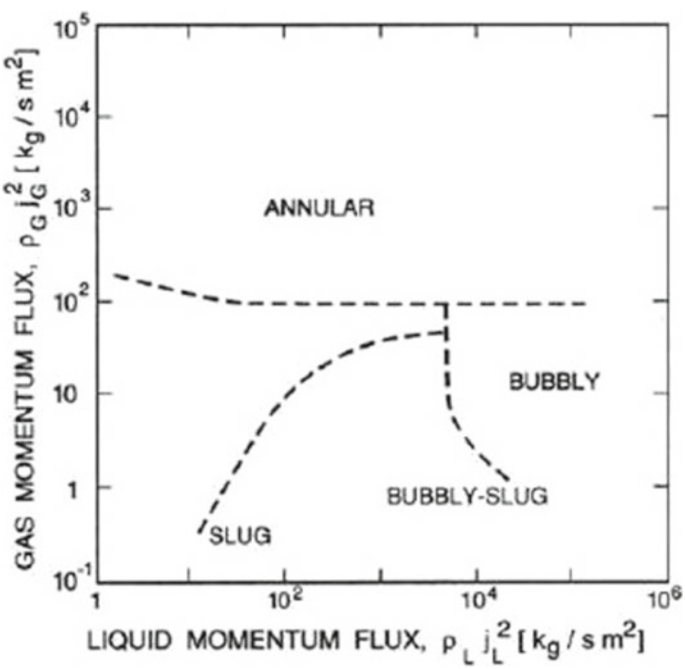


Figure 4. Flow regime map for Hewitt & Roberts (1969) for vertical channel validated for air–water at atmospheric pressure and steam–water at high pressure.

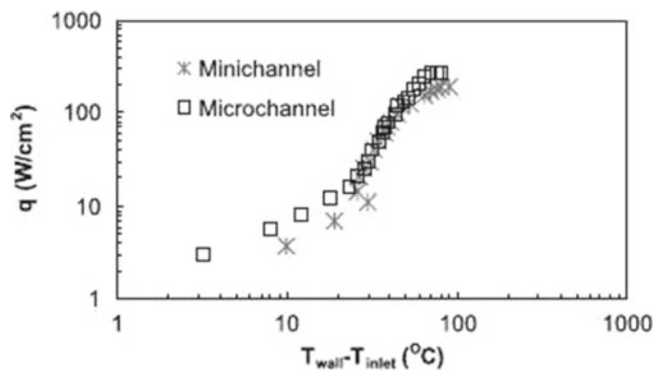


Figure 5. Flow boiling characteristics in mini and microchannel evaporator (Bowers & Mudawar 1994).

of flow pattern studies available in literature. He noted that both evaporation and condensation processes have a significant effect on the flow patterns. During evaporation, the annular film eventually dries out. For condensations no dry out occurs; the condensate at the wall coats the tube perimeter with the liquid film. The effect of evaporation on the flow pattern transitions was considered to be quite small in large diameter tubes. This is one of the reasons why the flow pattern studies from gas–liquid systems, such as air and water, were extended to the case of evaporation. In smaller diameter tubes, the effect of evaporation could be quite significant. The evaporation of the liquid phase affects the flow in two ways. First, it alters the pressure drop characteristics by introducing an acceleration pressure drop component that can be quite large at higher heat fluxes. Secondly, the tube dimension is quite small, and the effects of surface tension forces become more important in defining the two-phase structure. Bowers & Mudawar (1994) studied the flow boiling in a minichannel 2.54 mm diameter and microchannel 510 μm diameter. Boiling curves for the two channels were obtained at nearly equal values of liquid Reynolds number. Their results are shown in figure 5. Despite the large variation in the tube diameter, the two curves overlap in the boiling region. It was believed that these experiments fall under the fully developed nucleate boiling regime. The differences between the two boiling curves are only evident at low heat flux (near single-phase region) and high heat flux values (approaching CHF condition). This indicated that in spite of the differences in the flow characteristics of the channels, the flow boiling behaviour is quite similar in the two geometries.

Experimental observations by various researchers have shown that three flow regimes are sufficient to describe the patterns observed. These have been defined as follows:

2.3a *Isolated bubble flow*: Similar to bubble flow in large channels, bubbles detach from the nucleation sites and flow as discrete units in the liquid. As the flow proceeds further heat addition results in an increase in the number and size of the bubbles.

2.3b *Confined bubble flow*: Bubbles span the gap (in spaces confined in one dimension) or fill the channel (in spaces confined in two dimensions). They are separated from the wall by a layer of liquid which evaporates and causes the bubble to grow exponentially. Either through coalescence or through individual growth, single bubbles may reach a sufficient size to be regarded as confined before becoming detached from their nucleation sites. This latter phenomenon has been observed in relatively large tubes having polished surfaces and has been referred to as slug flow.

A similar phenomenon also occurs in the boiling of liquid metals. The rapid growth of confined bubbles can lead to significant fluctuations in the pressure at entry to the channel and apparent instability in multi-channel arrangements.

2.3c *Annular-slug flow*: As the confined bubbles expand, liquid in the slugs between them is deposited on the channel wall and the flow becomes annular with random, irregular slugs of liquid inter-spread with the vapour.

Though there are still arguments on the classification schemes of the flow patterns, each class could be subdivided into subclasses. For example, Taitel (1990) and Barnea (1987) defined six typical flow patterns in their vertical upward flow pattern map (figures 6a–b):

- *Dispersed bubble*: numerous small bubbles float in a continuous liquid phase.
- *Bubbly*: bubble size is comparable to channel size but not as large as the tube diameter.
- *Slug*: bubbles develop into bullet shape due to the tube wall restriction. Sometimes the bullet bubbles are followed by a stream of small bubbles creating a trail.

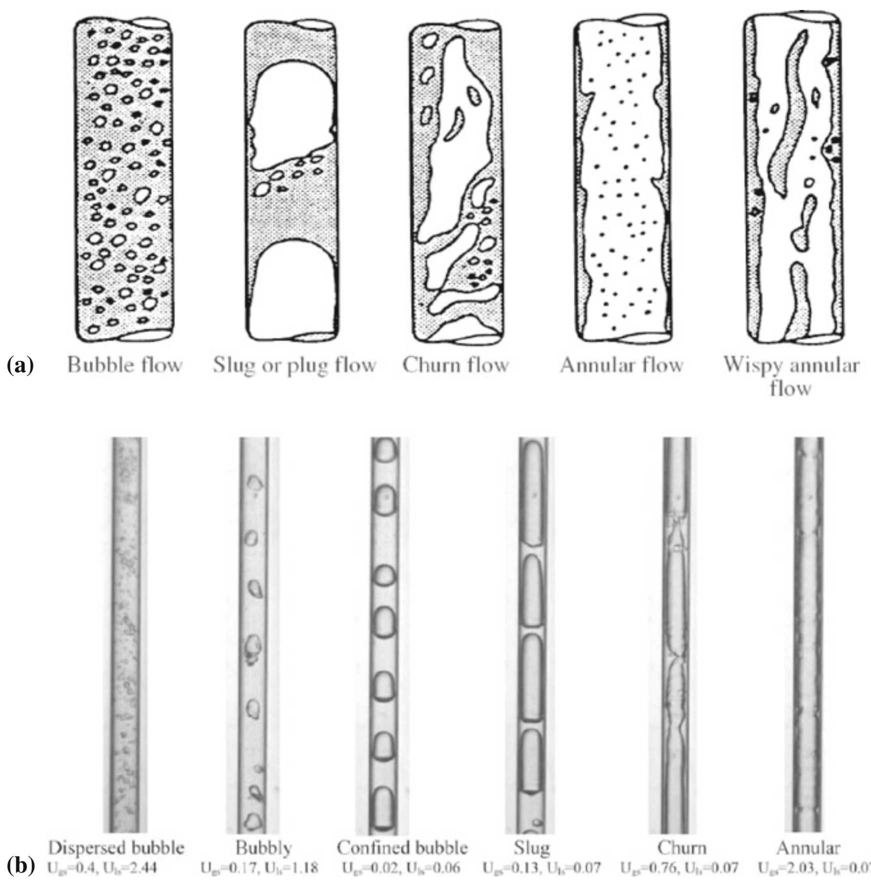


Figure 6. (**a**) Flow patterns in vertical flow (Hewitt 2007). (**b**) Experimental observation of 2.01 mm vertical tube observed by Shiferaw *et al* (2009).

Table 4. Flow behaviour reported in literature for tubes and channels of various sizes.

References	Passage size and fluid	Flow behaviour reported
Barnea (1987)	4–12 mm tubes Air–water	Dispersed bubble, elongated bubble, slug, churn and annular
Damianides & Westwater (1988)	1–5 mm tubes	Dispersed bubble, bubbly, plug, slug, pseudo slug, wavy and annular
Mishima & Hibiki (1996)	1–4 mm tubes Air – water	Bubbly, slug, churn, annular and annular-mist flow
Kew & Cornwell (2000)	1.39 and 3.69 mm tubes R141b flow boiling	Isolated bubble, confined bubble and slug/ annular flow
Lin *et al* (1999)	0.5–4 mm tubes Air–water	Bubbly, confined bubble, slug, churn and annular
Triplett *et al* (1999)	1.1 and 1.45 mm tube 1.097, 1.49 mm semi-triangular microchannel	Bubbly, slug, aerated slug-churn, slug-annular, annular
Zhao & Bi (2001)	Triangular channels 2.886, 1.443, 0.866 mm Air–water	Dispersed bubble, slug, churn, annular Dispersed flow was not present in the 0.866 mm
Serizawa *et al* (2002)	0.02, 0.025, 0.05 and 0.1 mm tubes Air–water	Bubbly, slug, liquid ring flow, liquid lump flow. Lump flow was not observed when using steam–water
Kawahara *et al* (2002)	0.1 mm tube Nitrogen – deionised water	Liquid alone and gas core with (i) smooth-thin liquid film (ii) smooth-thick liquid film (iii) ring shaped liquid film (iv) deformed interface
Chung & Kawaji (2004)	0.05 to 0.53 mm tubes Nitrogen–water	Bubbly, churn, gas core flow with smooth thin liquid film. Slug-annular, annular. Churn observed only in 0.25 and 0.53 mm
Chen & Garimella (2006)	1.1, 2.01, 2.88 and 4.26 mm tubes R134a	Dispersed bubble, bubbly, slug, churn, annular and mist flow. Confined flow was observed in the 2.01 and 1.1 mm tubes
Revellin & Thome (2007)	0.51 and 0.8 mm tubes	Isolated bubble, coalescing bubble and annular flow
Shiferaw *et al* (2007)	Circular 2.01 and 4.26 mm R134	Slug, churn and annular
Shiferaw *et al* (2009)	Circular – 1.1 mm, R134a	Confined, slug, churn, annular
Julia *et al* (2011)	Vertical annular – 19 mm Air – Water	Bubbly, cap-slug, churn-turbulent and annular flows, axial and radial flow regime transitions studied and surveyed
Tadrist (2007)	Circular 10 mm and higher	Flow instabilities studied and analysed for mini and macrochannels
Zhang *et al* (2010)	Circular 1–5 mm, refrigerents-vapour, ammonia-vapour, water–air	Reported that void fraction and pressure drop correlations based on separated flow model work better in comparison with homogeneous flow model for flow boiling in minichannels
Venkatesan *et al* (2010)	Circular 0.6–3 mm, air–water	Bubbly, dispersed bubbly, slug, slug-annular, and wavy-annular flow. Unique flow patterns were identified for different tube diameters that confirm the diameter effect on flow patterns in two-phase flows

Table 4. (continued).

References	Passage size and fluid	Flow behaviour reported
Zhihu *et al* (2006)	Circular 50 mm with horizontal rod, air – water	Rod affects flow transitions and bubble shape and size. All standard regimes observed
Ould Didi *et al* (2002)	Circular 10.92–12 mm, horizontal tube, R134a, R123, R402A, R404A and R502	Best method for pressure drop prediction reported for each type of flow pattern. A single best correlation applicable for all flow regimes attempted but with high deviations
Karayiannis *et al* (2008)	0.52–4.26 mm Vertical tubes, R134a	Above 0.52 mm dia. pressure drops were predicted well by Chisholm and the Lockhart-Martinelli method. No existing correlation predicted well for 0.52 mm dia.

- *Churn*: bullet bubbles start to distort and small bubbles in liquid slug coalesce into gas clump with increase of gas velocity. It is a highly oscillatory flow with chaotic interface.
- *Annular*: gas phase becomes a continuous flow in the core of the tube.
- *Mist*: liquid film is blown away from tube wall and numerous liquid droplets float in high-speed vapour flow.

Flow behaviour reported in literature for tubes and channels of various sizes are reported in table 4.

Harirchian & Garimella (2010) reviewed the literature on flow regime maps for boiling in microchannels. They developed a new type of comprehensive flow regime map for wide range of experimental parameters and channel dimensions. They also developed quantitative transition criteria of flow regime based on non-dimensional parameters (figure 7). This comprehensive flow regime map is based on experimental results and flow visualization performed with Fluorinert FC-77.

The factors affecting flow patterns are numerous and complex. Within the transition zone the flow patterns possess characteristics of more than one of the flow patterns. Despite significant research in development of flow maps with similar tubes as well as similar experimental conditions, it is observed that studies are not in agreement with each other (for e.g., Oya 1971, Barnea *et al* 1983 and Mishima & Hibiki 1996). This is because the identification of flow patterns may be greatly affected by the observer's subjectivity and the experimental technique employed.

2.4 *Two-phase pressure drop*

The prediction of two-phase pressure gradient is an essential step in the design of a variety of plant in the power and process industries. The total pressure drop of fluid is due to the variation of kinetic and potential energy of the fluid and also due to friction at the wall of flow channel (see Equation 8). Calculation of pressure drop across the full length of the heated channel is more complex due to the existence of several regimes with varied physical conditions along the channels.

$$\Delta P_{\text{Total}} = \Delta P_{\text{static}} + \Delta P_{\text{momentum}} + \Delta P_{\text{frictional}}. \qquad (1)$$

The most significant and difficult term is the frictional pressure drop. In order to get frictional pressure drop one should have a knowledge of two-phase friction factor, mixture density, vapour

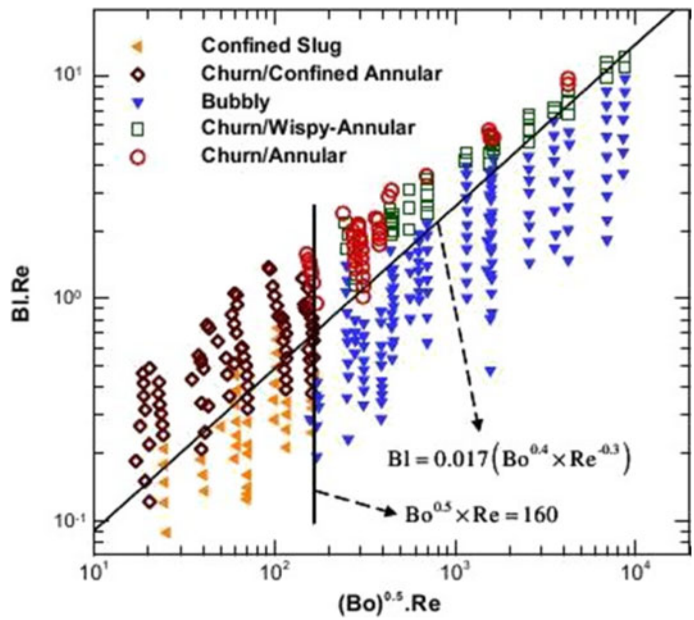


Figure 7. Comprehensive flow regime map for FC-77.

quality and void fraction, velocity of vapour and liquid (i.e., slip ratio) as shown in figure 8. It has been shown that total pressure drop is the summation of acceleration, static and frictional pressure drop. If acceleration and static pressure drop are present (figure 8), judicious selection of void fraction correlation becomes necessary for reliable quantitative measurements of these two pressure drops. Similarly, from figure 9, it can be seen that frictional pressure drop is estimated by various concepts such as in adiabatic and diabatic two-phase flows. Two-phase pressure drop can be measured for gas–liquid adiabatic flow or it can also be measured for vapour–liquid non-adiabatic, boiling or condensing flow. Because of ease of operation, laboratory measurements tend to be made with adiabatic gas–liquid flow e.g., air–water, rather than non-adiabatic flow. Two-phase pressure drop correlations are largely based on data for such adiabatic gas–liquid two-phase flow. On the other hand, two-phase heat transfer equipment, by default, will have boiling or condensing flows. It is generally accepted that the two-phase pressure drop and flow pattern characteristics for non-adiabatic flow could be similar to adiabatic gas–liquid flow. Adiabatic two-phase frictional pressure drop is developed from empirical correlations, friction multiplier concepts and flow pattern specific models. Few correlations considering two-phase friction multiplier concept have been reported in table 5.

Whalley (1980) made an extensive comparison based on frictional multiplier concept between various published correlations. Following are his recommendations:

- $\left(\frac{\mu_L}{\mu_G}\right) < 1000$ and mass velocities less than 2000 $\mathrm{kg/m^2\,s}$, the Friedel correlation (Friedel 1979) should be used.
- $\left(\frac{\mu_L}{\mu_G}\right) > 1000$ and mass velocities greater than 100 $\mathrm{kg/m^2\,s}$, the Chisholm correlation (Chisholm 1973) should be used.

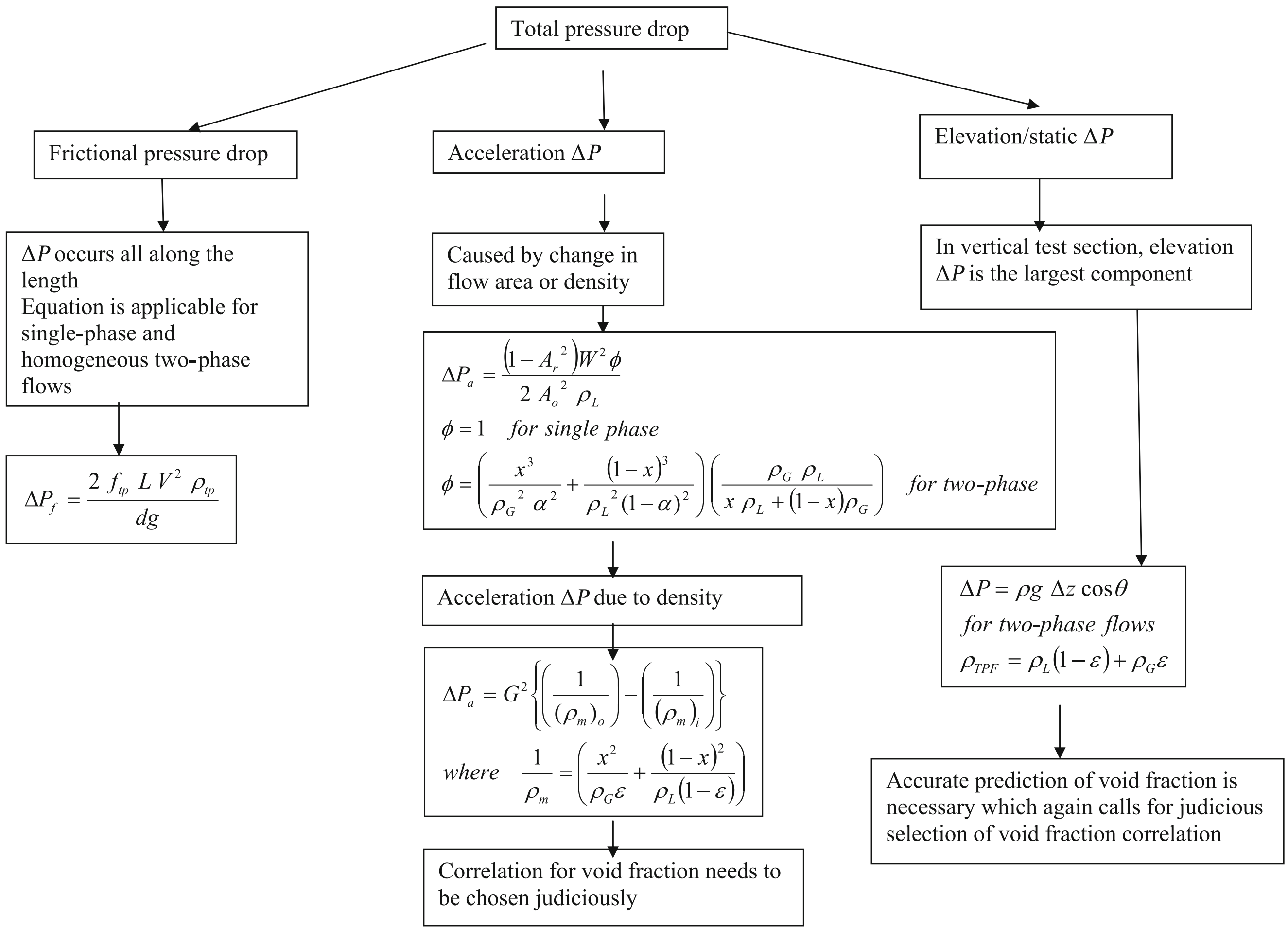


Figure 8. Total pressure drop in two-phase flow boiling.

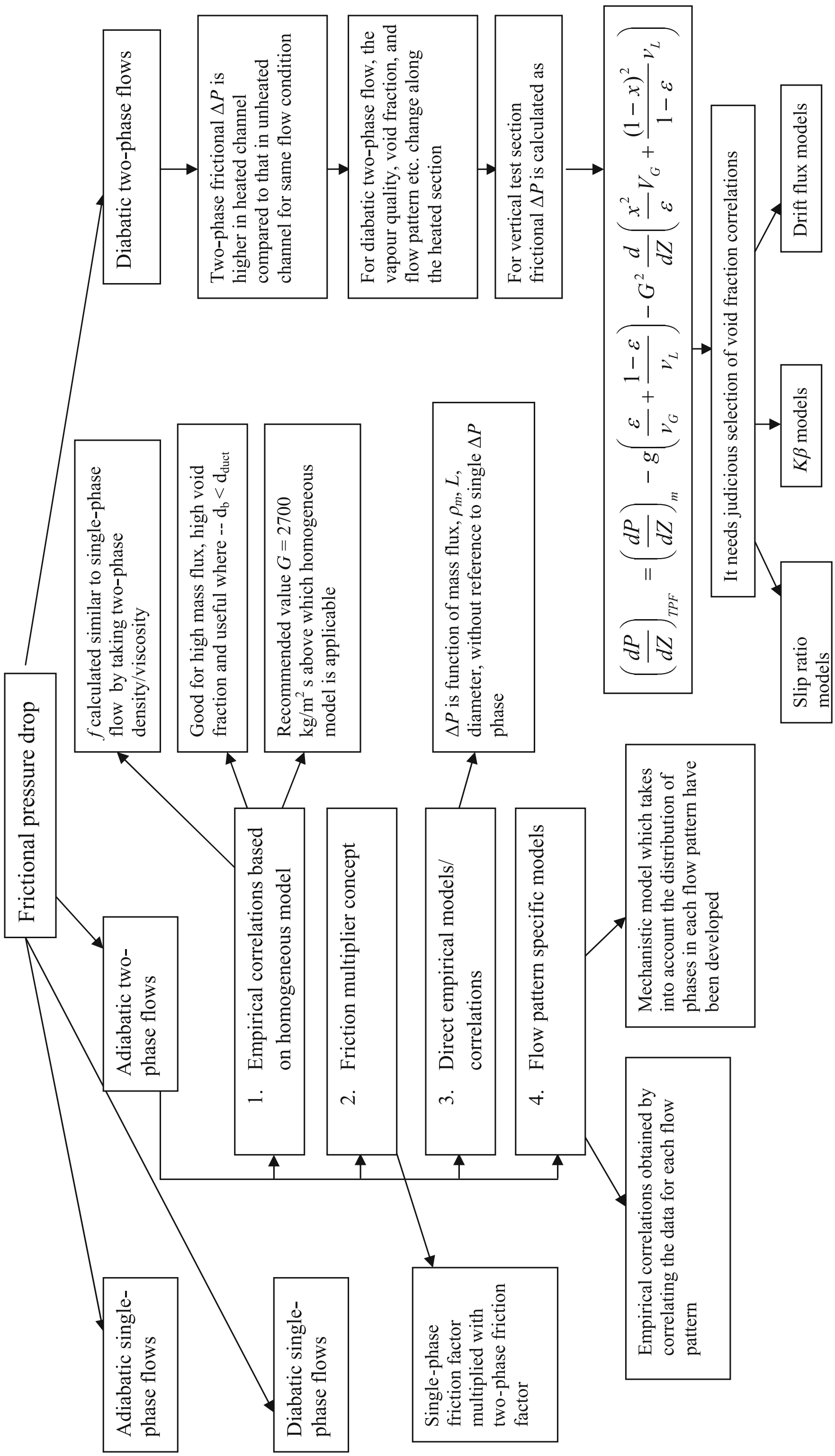


Figure 9. Frictional pressure drop in two-phase flow boiling.

Table 5. Correlations for two-phase frictional pressure drop.

Authors	Correlations	Remark
Friedel (1979)	$\Delta P_{\text{fric}} = \Delta P_l \Phi_{fr}^2$ $\Phi_{\text{fr}}^2 = E + \dfrac{3.24FH}{Fr_H^{0.045} We_L^{0.035}}$	Applicable when $\dfrac{\mu_L}{\mu_G} < 1000$
Lockhart & Martinelli (1949)	$\Delta P_{\text{fric}} = \Delta P_l \Phi_{Ltt}^2$ $\Delta P_{\text{fric}} = \Delta P_l \Phi_{Gtt}^2$	Applicable for laminar and turbulent flow
Gronnerud correlation (1979)	$\Delta P_{\text{fric}} = \Delta P_l \Phi_{gd}$ $\Phi_{gd} = 1 + \left(\dfrac{dp}{dz}\right)_{Fr} \left[\dfrac{\left(\frac{\rho_L}{\rho_G}\right)}{\left(\frac{\mu_L}{\mu_G}\right)^{0.25}} - 1\right]$	Applicable specifically for refrigerants
Chisholm (1973)	$\left(\dfrac{dp}{dz}\right)_{\text{fric}} = \left(\dfrac{dp}{dz}\right)_L \Phi_{ch}^2$ $\Phi_{ch}^2 = 1 + \left(Y^2 - 1\right)\left[Bx^{\frac{(2-n)}{2}} (1-x)^{\frac{(2-n)}{2}} + x^{2-n}\right]$	Used for wide range of operating condition
Bankoff (1960)	$\left(\dfrac{dp}{dz}\right)_{\text{fric}} = \left(\dfrac{dp}{dz}\right)_L \Phi_{Bf}^{\frac{7}{4}}$ $\Phi_{Bf} = \dfrac{1}{1-x}\left[1 - \gamma\left(1 - \dfrac{\rho_G}{\rho_L}\right)\right]^{\frac{3}{7}} \left[1 + x\left(\dfrac{\rho_L}{\rho_G} - 1\right)\right]$	Extension of homogenous model
Chawla (1967)	$\left(\dfrac{dp}{dz}\right)_{\text{fric}} = \left(\dfrac{dp}{dz}\right)_G \Phi_{Chawla}$ $\Phi_{Chawla} = x^{1.75}\left[1 + S\left[\dfrac{1-x}{x}\dfrac{\rho_G}{\rho_L}\right]\right]$	Based on vapour pressure gradient

- $\left(\dfrac{\mu_L}{\mu_G}\right) > 1000$ and mass velocities less than 100 kg/m^2 s, the Lockhart and Martinelli correlation (Lockhart & Martinelli 1949) should be used.
- For most fluids, $\left(\dfrac{\mu_L}{\mu_G}\right) < 1000$ and hence the Friedel correlation will be the preferred method for in tube flow according to Whalley (1980).

In case of diabatic two-phase flows, frictional pressure drop is a function of flow properties and significantly on void fraction as shown in figure 9. It is observed that vapour quality and void fraction parameters are continuously changing. This affects overall flow pattern (refer to section 2.3) and thus affects pressure drop of the system.

Therefore, it is important to distinguish the difference between void fraction of vapour phase and thermodynamic vapour quality. The vapour quality is the ratio of mass of vapour to the total mass of liquid plus vapour. The most widely utilized void fraction definition is the cross sectional average void fraction, which is based on the relative cross sectional areas occupied by the respective phases.

Thermodynamic vapour quality and void fraction are closely related to each other. The classic definition of x as a mass flow rate ratio clearly shows that it includes slip ratio S using equation 4. This phase velocity ratio or slip ratio is defined as the cross sectional areas of mean vapour velocity to cross sectional area mean liquid velocity.

$$x = \frac{m_v}{m_v + m_l}. \tag{2}$$

$$x = \frac{u_v \rho_v A_c \alpha}{[u_v \rho_v A_c \alpha + u_l \rho_l A_c (1 - \alpha)]}. \tag{3}$$

$$x = \frac{\rho_v \alpha S}{[\rho_v \alpha S + \rho_l (1 - \alpha)]}. \tag{4}$$

$$\alpha = \frac{1}{\{1 + S(1 - x)\frac{\rho_v}{\rho_l}}. \tag{5}$$

$$x = \frac{\rho_v \alpha}{\rho_m}. \tag{6}$$

$$\rho_m = (1 - \alpha)\rho_l + \alpha \rho_v. \tag{7}$$

Thus, there is a direct connection between void fraction and thermodynamic quality through mixture densities. It is evident that actual vapour mass fraction, which corresponds to the actual void fraction is equivalent to thermodynamic vapour quality.

This cross sectional void fraction is usually predicted by one of the following methods:

- Homogeneous model (which assumes the two-phases travel at the same velocity).
- 1-D models (which account for differing velocities of the two-phases).
- Drift flux models including radial variations in local void fraction and flow velocity.
- Models based on the physics of specific flow regimes.
- Empirical and semi empirical methods.

Thus, relation between void fraction and vapor quality represented by several correlations; few of them are reported in Table 6. Where it observed that void fraction is function of vapor quality.

Industrial applications, where two-phase flow exists, requires the formidable task of predicting the phase distribution in the system, given their operating conditions. Once this phase distribution is known, the problem may be simplified. One of the critical unknown parameters involved in predicting the pressure loss and heat transfer in any gas–liquid system is the void fraction (ε) or the liquid holdup ($1 - \varepsilon$). ε is the volume occupied by the gas and ($1 - \varepsilon$) is the volume occupied by the liquid. The accurate prediction of void fraction in vertical channels is of immense importance in determining the two-phase pressure drop. This variation in void fraction resulting from the change in flow direction requires a reliable, accurate, and flexible correlation to predict void fraction. Many researchers have contributed to the development of the void fraction correlations in vertical orientation. The theoretical considerations used for the development of these correlations are based on the separated flow model or the drift flux model. Most of these correlations are flow pattern-specific and are limited in their application in terms of the range of flow variables. The void fraction correlation proposed by Woldesemayat (2006), is independent of the flow direction, system pressure, and fluid thermo-physical properties. However, it is subject to some inaccuracies in predicting the void fraction (ε) typically in the range of $0 < \varepsilon < 0.4$ in vertical upward flow directions. To choose the right correlation for the desired application from a pool of available correlations in the literature (Godbole *et al* 2011), it is of the utmost importance to sort out and identify the top-performing correlations applicable to a range of flow

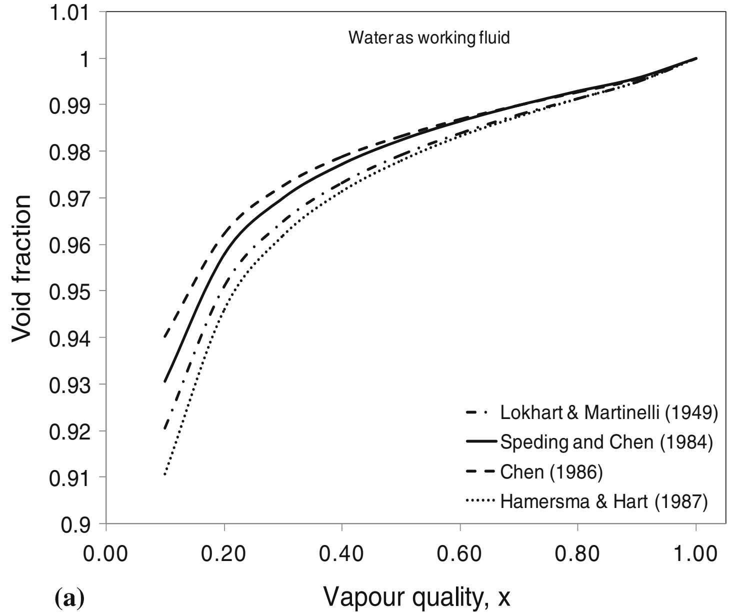
Table 6. Void fraction correlations.

Author	Correlations
Slip ratio models	
Homogeneous	$\varepsilon = \left[1 + \left(\frac{1-x}{x}\right)\left(\frac{\rho_G}{\rho_L}\right)\left(\frac{\mu_L}{\mu_G}\right)\right]^{-1}$
Lockhart & Martinelli (1949)	$\varepsilon = \left[1 + 0.28\left(\frac{1-x}{x}\right)^{0.64}\left(\frac{\rho_G}{\rho_L}\right)^{0.36}\left(\frac{\mu_L}{\mu_G}\right)^{0.07}\right]^{-1}$
Chisholm (1973)	$\varepsilon = \left[1 + \sqrt{1 - x\left(1 - \frac{\rho_L}{\rho_G}\right)}\left(\frac{1-x}{x}\right)\left(\frac{\rho_G}{\rho_L}\right)\right]^{-1}$
Spedding & Chen (1984)	$\varepsilon = \left[1 + 2.22\left(\frac{1-x}{x}\right)^{0.65}\left(\frac{\rho_G}{\rho_L}\right)^{0.65}\right]^{-1}$
Chen (1986)	$\varepsilon = \left[1 + 0.18\left(\frac{1-x}{x}\right)^{0.6}\left(\frac{\rho_G}{\rho_L}\right)^{0.33}\left(\frac{\mu_L}{\mu_G}\right)^{0.07}\right]^{-1}$
Hamersma & Hart (1987)	$\varepsilon = \left[1 + 0.26\left(\frac{1-x}{x}\right)^{0.67}\left(\frac{\rho_G}{\rho_L}\right)^{0.33}\right]^{-1}$
Zhao *et al* (2000)	$\varepsilon = \left[1 + \varepsilon^{-0.125}\left(\frac{1-x}{x}\right)^{0.875}\left(\frac{\rho_G}{\rho_L}\right)^{0.875}\left(\frac{\mu_L}{\mu_G}\right)^{0.875}\right]^{-1}$
$K\varepsilon_H$ correlations	
Chisholm (1983), Armand (1946)	$\varepsilon = \frac{1}{\varepsilon_H + \frac{1}{(1-\varepsilon_H)^{0.5}}}\varepsilon_H$
Bankoff (1960)	$\varepsilon = \left[0.71 + \left(1.45 \times 10^{-2}\right)P\right]\varepsilon_H$
Drift flux correlations	
Hughmark (1965)	$\varepsilon = \frac{U_{SG}}{1.2U_M}$
Rouhani & Axelsson (1970)	$\varepsilon = \frac{x}{\rho_G}\left[\mathrm{CO}\left(\frac{x}{\rho_G} + \frac{1-x}{\rho_L}\right) + \frac{U_{GM}}{G}\right]^{-1}$

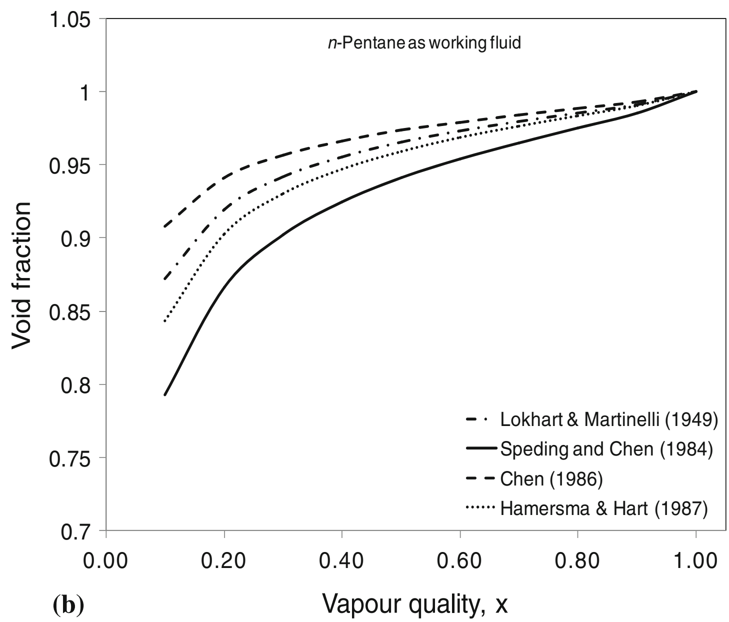
variables and channel orientations. Some of the most cited correlations are reported in table 6. In the present study, few correlations compared (with water and *n*-pentane as working fluid) and concluded that as vapour quality increases void fraction is also increased (see figures 10a–b).

2.5 *Heat transfer coefficient*

In vertical channel two phase flow system, boiling will not occur until the wall temperature reaches a certain value (wall superheat) above the saturation temperature. As soon as evaporation of fluid starts, there is qualitatively progressive variation in local heat transfer coefficient along the heated tube because of variation in flow regimes (as discussed in section 1). Based on analyses of various experimental investigations of flow boiling, it was found that liquid-laminar



(a)



(b)

Figure 10. (**a**) Comparison of homogeneous model correlations using water as working fluid. (**b**) Comparison of homogeneous correlations using n-pentane as working fluid.

and gas-turbulent flow is a common feature in many applications of small channels. Therefore, many researchers formulated various heat transfer coefficient correlations and are as follows:

$$h_{tp} = Sh_{nb} + Fh_{sp} \tag{8}$$

where

$$\text{for} \quad \frac{1}{X_{tt}} > 0.1 \qquad F = 2.35\left(\frac{1}{X_{tt}} + 0.213\right)^{0.736} \tag{9}$$

$$\text{for} \quad \frac{1}{X_{tt}} < 0.1 \qquad F = 1 \tag{10}$$

$$h_{sp} = 0.023\mathrm{Re}_f^{0.8}\,\mathrm{Pr}_f^{0.4}\left(\frac{k_f}{D_h}\right) \tag{11}$$

$$S = \frac{1}{\left(1 + 2.53 \times 10^{-6}\mathrm{Re}_f^{1.17}\right)} \tag{12}$$

$$h_{nb} = 0.00122\left(\frac{k_f^{0.79}C_{pf}^{0.45}\rho_f^{0.49}}{\sigma^{0.5}\mu_f^{0.29}h_{fg}^{0.24}\rho_g^{0.24}}\right)\Delta T_{sat}^{0.24}\Delta P_{sat}^{0.75} \tag{13}$$

$$X_{tt} = \left(\frac{1 - x_{eq}}{x_{eq}}\right)\left(\frac{\rho_g}{\rho_f}\right)^{0.5}\left(\frac{\mu_f}{\mu_g}\right)^{0.1}. \tag{14}$$

Here Chen introduced two dimensionless factors, the suppression factor S and the enhancement factor F, to account for the smaller effective superheat due to forced convection, and for the increase in convective turbulence due to the presence of vapour phase, respectively. This correlation is widely used and taken as a benchmark in the literature.

To correlate flow boiling heat transfer coefficients Shah (1987) proposed a different approach, where the nucleate boiling component was represented by the boiling number Bo, while the convective number Co was used for the forced convection component. He first used a graphical chart, later curve-fitted equations, to select the larger of the two components:

$$\frac{h_{tp}}{h_{sp}} = \mathrm{MAX}(\psi_{nb}, \psi_{fc}) \tag{15}$$

where the nucleate boiling component $\psi_{nb} = fn(\mathrm{Bo})$ and the convective component $\psi_{fc} = fn(\mathrm{Co})$.

After Chen (1963) and Shah (1976) various researchers developed heat transfer coefficient correlations depending upon specific design and operating conditions. We list them below:

- *Lazarek and Black (1982)*:

$$h_{tp} = \left(30\mathrm{Re}^{0.857}\mathrm{Bo}^{0.714}\right) \tag{16}$$

- *Modified Lazarek and Black (1982)*:

$$h_{tp} = \left\{30\mathrm{Re}^{0.857}\mathrm{Bo}^{0.714}(1 - x)^{-0.143}\right\}k/L \tag{17}$$

- *Cooper correlation*: Cooper (1984) proposed the following correlation for nucleate boiling heat transfer.

$$h_{nb} = 55P_R^{0.12-0.087\ln\varepsilon}(-0.4343\ln P_R)^{-0.55}M^{-0.5}q^{0.67} \tag{18}$$

- *Gungor and Winterton correlation*: Gungor & Winterton (1986) proposed a correlation that is a modification of the Chen correlation:

$$h_{tp} = Sh_{nb} + Fh_{sp} \tag{19}$$

where

$$S = fn(F, \mathrm{Re}_f) \qquad \text{and} \qquad F = fn(\mathrm{Bo}, X_{tt}) \tag{20}$$

- *Liu & Winterton (1991)* pointed out the deficiencies of Gungor and Winterton (1986) correlation and developed a general correlation for flow boiling in tubes

$$h_{tp} = \left[\left(F h_{sp,fo}\right)^2 + (S h_{nb})^2\right]^{1/2} \tag{21}$$

- *Kandlikar (1990)* proposed a correlation of the form

$$h_{tp} = h_{sp}\left(C_1 \mathrm{Co}^{C_2}\left(25\mathrm{Fr}_{fo}\right)^{C_5} + C_3 \mathrm{Bo}^{C_4} F_f\right). \tag{22}$$

Corresponding to the forced convection and nucleate boiling regions, two sets of constants C_1 through C_5 were established by careful regression analysis, using a large bank of water data. The heat transfer coefficient at any given condition is evaluated using the two sets of constants for the two regions, and the higher of the two heat transfer coefficient values is the choice. In addition, when tested on a refrigerant, a fluid-dependent correction factor F_f is introduced. Therefore a new factor F_f is needed when this correlation is used for any new fluid beyond the tested ones.

Steiner & Taborek (1992) followed the power-type addition model for two boiling components and built a heat transfer correlation for flow boiling based on an extensive databank that is similar to Dittus-Boelter correlation (Dittus & Boelter 1930).

- *Tran et al correlation*: Tran *et al* (1996) conducted flow boiling heat transfer experiments for R12 in small channels and proposed

$$h_{tp} = 840{,}000 \mathrm{Bo}^{0.6} \mathrm{We}_l^{0.3} \left(\rho_l/\rho_g\right)^{-0.4} \tag{23}$$

- *Kew-Cornwell correlation*: Kew & Cornwell (1997) modified the Lazarek-Black equation to allow for an observed increase in the heat transfer coefficient with the vapour quality in larger tubes.

$$h_{tp} = 30 \mathrm{Re}_{lo}^{0.857} \mathrm{Bo}^{0.714} \frac{\lambda_l}{D_h}\left(\frac{1}{1-x}\right)^{0.143} \tag{24}$$

Lin *et al* (2001) presented a work on four circular and one square channels to measure local heat transfer coefficients at various heat flux and mass flux. It was observed that in circular and rectangular channels local heat transfer coefficients are not only function of heat flux but also a function of vapour quality and weaker function of mass flux.

- *Yu et al (2002)* modified the correlation suggested by Tran *et al* and proposed the following instead:

$$h_{tp} = 640{,}000 \mathrm{Bo}^{0.54} \mathrm{We}_l^{0.27} \left(\rho_l/\rho_g\right)^{-0.2} \tag{25}$$

- Flow boiling heat transfer correlations, such as those in Kandlikar (2001, 2002), utilize the single-phase, all-liquid heat transfer coefficient in predicting the nucleate boiling and convective boiling components as given by the following equation:

$$\left(h_{tp}/h_{sp}\right) = \left\{C_1 \mathrm{Co}^{C_2}\left(25 Fr_{fo}\right)^{C_5} + C_3 \mathrm{Bo}^{C_4} \mathrm{Fr}_f\right\} \tag{26}$$

- *Kandlikar-Balasubramanian (2004)* extended the above Kandlikar correlation to laminar flow and mini- and microchannels. The flow regions are classified as turbulent region

($\mathrm{Re}_{LO} \geq 3000$), transition region ($1600 \leq \mathrm{Re}_{LO} < 3000$) and laminar region ($\mathrm{Re}_{LO} < 1600$). They considered the effect of tube orientation for flow boiling in small diameter tubes negligible, and thus deleted the Froude number effect in the above correlation by setting $f(\mathrm{Fr}_{lo}) = 1$.

$$h_{tp,nb} = \left[0.6683\mathrm{Co}^{-0.2} + 1058.0\mathrm{Bo}^{0.7} F_f\right] (1 - x)^{0.8} h_{lo} \tag{27}$$

$$h_{tp,nb} = \left[1.136\mathrm{Co}^{-0.9} + 667.2\mathrm{Bo}^{0.7} F_f\right] (1 - x)^{0.8} h_{lo} \tag{28}$$

where h_{lo} is constant for the laminar region, h_{lo} is calculated using Eq. 29 for the turbulent region and determined by linear interpolation in the transition region.

$$h_{lo} = \frac{(f/8)\,\mathrm{Re}_{lo}\mathrm{Pr}_l\,(\lambda_l/D)}{1 + 12.7\,(f/8)^{1/2}\left(\mathrm{Pr}_l^{2/3} - 1\right)} \quad for \quad 10^4 \leq \mathrm{Re}_{lo} \leq 5 \times 10^6 \tag{29}$$

$$h_{lo} = \frac{(f/8)\,(\mathrm{Re}_{lo} - 1000)\,\mathrm{Pr}_l\,(\lambda_l/D)}{1 + 12.7\,(f/8)^{1/2}\left(\mathrm{Pr}_l^{2/3} - 1\right)} \quad for \quad 3000 \leq \mathrm{Re}_{lo} \leq 10^4 \tag{30}$$

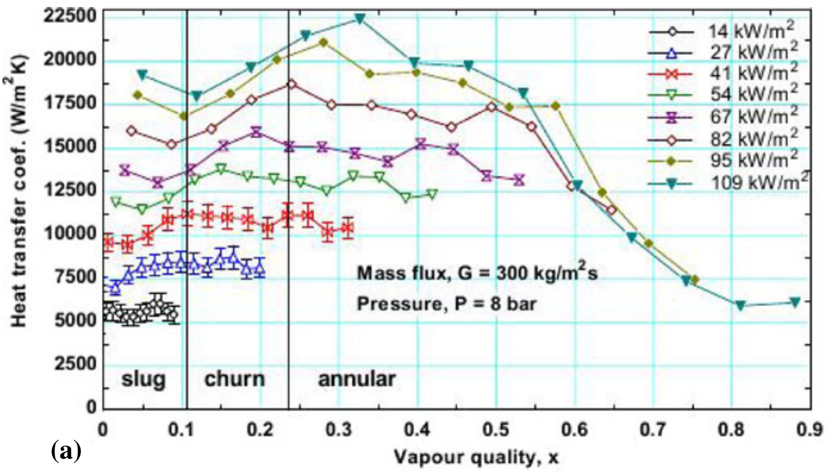
$$f = (0.79 \ln \mathrm{Re}_{lo} - 1.64)^{-2} \tag{31}$$

F_f values are listed in table 7. The applicable vapour quality range is $x < 0.7 \sim 0.8$.

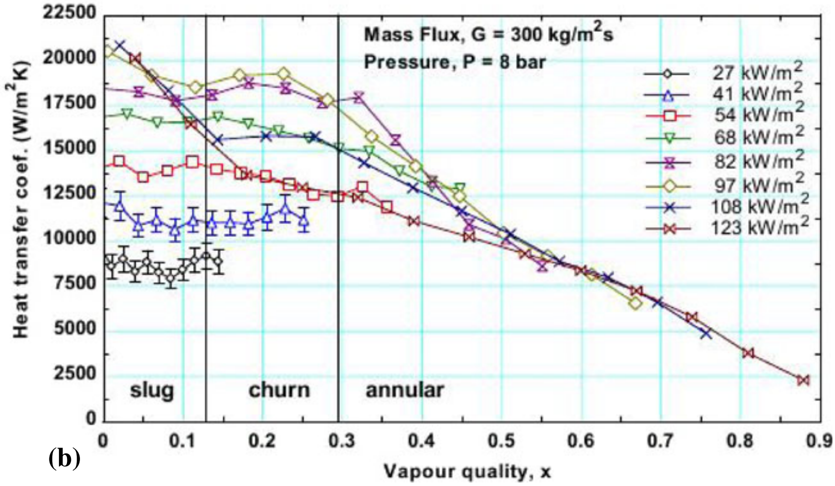
Shiferaw *et al* (2007) carried out experiments using channel diameter 4.26 and 2.01 mm and gathered experimental data for local heat transfer coefficient as a function of vapour quality. Basically, heat transfer coefficient is based on prevailing flow regimes. Therefore, Shiferaw *et al* (2007) showed flow pattern transition boundaries predicted by Chen & Garimella (2006) with heat transfer coefficient vs vapour quality as shown in figure 11. The experimental results for the 4.26 mm tube demonstrate that heat transfer coefficient increases with heat flux but does not change with vapour quality when the quality was less than 40–50% for low heat flux. For 2.01 mm tube this boundary moves to 20–30% vapour quality. This behaviour of low quality is conventionally interpreted as evidence that nucleate boiling is the dominant heat transfer mechanism.

Table 7. Recommended F_f values in Kandlikar flow boiling correlation.

Fluid	F_f	Fluid	F_f
Water	1.00	R134a	1.63
R11	1.30	R152a	1.10
R12	1.50	R31/R-132	3.30
R13B1	1.31	R141b	1.80
R22	2.20	R124	1.00
R113	1.30	Kerosene	0.488
R114	1.24		



(a)



(b)

Figure 11. (**a**) Local heat transfer coefficient as a function of vapour quality at different heat fluxes (a) d − 4.26 mm. (**b**) Local heat transfer coefficient as a function of vapour quality at different heat fluxes (b) d − 2.01 mm.

2.6 *Critical heat flux (CHF)*

Critical heat flux (CHF) in flow boiling is the heat flux that results in a sudden rise in the wall temperature. There are basically two classes of CHF situations, namely departure from nucleate boiling (DNB) and dry-out (DO). DNB occurs at very low qualities or even under sub-cooled liquid conditions; here the vapour generation rate is higher than the rate of vapour evaporation by the flowing liquid. This leads to an accumulation of vapour at the generation site and prevents liquid from contacting the surface. The vapour therefore gets superheated and the wall temperature rises suddenly due to the relatively poor heat transfer characteristics of vapour. In contrast to DNB, dry-out occurs at high qualities and is a result of the gradual disappearance of the liquid film adjacent to the wall as a result of both entrainment and evaporation. Dry-out occurs in annular flow and a significant amount of liquid may still be flowing in the form of droplets entrained by the vapour.

The terms critical heat flux (CHF) and dry-out are both used to describe a situation when the heated surfaces are no longer wetted by the liquid. Strictly speaking, the way in which this situation is reached determines whether it should be termed critical heat flux or dry-out. Critical heat flux could appear at any vapour fraction and represents the upper limit of the heat flux where liquid is still in contact with the surface. Dry-out appears at high vapour fraction, independent of heat flux, when there is not sufficient liquid left to fully wet the walls of the channel. At dry-out condition, the wall temperature starts increasing due to the heat transfer deterioration.

CHF refers to the heat transfer limit causing a sudden decrease in the heat transfer coefficient and possible catastrophic failure of a device in which evaporation or boiling is occurring. The ability to predict the CHF is therefore of vital importance to the safety of flow boiling system.

Critical heat flux has been studied extensively over the past several decades. Authoritative reviews on the subject may be found in Tong *et al* (1997), Hewitt & Hall-Taylor (1970) and Katto & Ohno (1984).

Flow boiling CHF can be classified as either sub-cooled CHF or saturated CHF. For the sub-cooled CHF, several theories have been proposed to explain its mechanism: intense boiling causes the boundary layer separation from the heated wall and the resulting stagnant liquid depletion (Kutateladze & Leont 1966; Tong 1968), bubble crowding within the boundary layer inhibits liquid replenishment near the wall surface (Weisman & Pei 1983), and dry-out of a liquid sub layer beneath vapour blankets causes the appreciable rise of local wall temperature (Lee & Mudawar 1988; Vijayarangan *et al* 2006). For the saturated CHF, the liquid film dry-out near the channel outlet is widely regarded as the trigger mechanism (Hewitt & Hall-Taylor 1970). At low flow rates in small-diameter tubes this type of CHF may be prone to occur due to the thinner liquid film thickness.

As the transport process behind the flow boiling CHF is extremely complex, CHF prediction relies heavily on empirical correlations derived from experimental CHF databases. There are many correlations (Maqbool *et al* 2012) in the literature that apply exclusively to water. Some of them are reported in table 8. Few of them are compared using water and *n*-pentane as working fluid respectively (see figures 12a–b). It is observed that there exists a strong dependence of CHF on mass flux, heated length and inner diameter.

Therefore, the applicability of existing CHF correlations to small-diameter channels should be carefully examined in detail. Although many analytical and experimental studies related to flow boiling CHF in small-diameter channels have been performed in the past decades, a reliable CHF correlation applicable to a wide range of parameters for small-diameter channels has not been developed yet.

Summary of investigations on boiling in mini and microchannels regarding heat transfer coefficient, pressure drop and flow regimes, CHF was done and reported in table 9.

2.7 *Film thickness*

The development of techniques to measure liquid film thickness has been an important topic in the area of two-phase flow and many methods have been developed for it. The methods can be classified into four groups according to their measurement principles: 1) optical, 2) acoustic, 3) radiological and 4) electrical methods. In optical methods various optical devices such as high speed cameras and lasers are used for detecting an interface (Coney *et al* 1989; Ursenbacher *et al* 2004), total internal reflection of light (Yu *et al* 1996), fluorescence intensity (Schubring *et al*

Table 8. Critical heat flux correlations.

Author	Correlation
Bowring (1972)	$q_c = \dfrac{A' + 0.25 D_h G \Delta h_{in}}{C' + L}$
Katto & Ohno (1984)	$\dfrac{q''_{CHF}}{G i_{lv}} = f\left\{\dfrac{\rho_l}{\rho_v}, \dfrac{\sigma \rho_l}{G^2 l}, \dfrac{L}{D_h}\right\}$
Shah (1987)	$\dfrac{q''_{CHF}}{G i_{lv}} = 0.124 \left(\dfrac{L}{D}\right)^{0.89} \left(\dfrac{10^{4^n}}{Y}\right)(1 - x)$ $Y = G^{1.8} D^{0.6} \left(\dfrac{C_p}{k_l \rho_l^{0.8} g^{0.4}}\right)\left(\dfrac{\mu_l}{\mu_v}\right)$
Callizo *et al* (2008)	$\dfrac{q''_{CHF}}{G i_{lv}} = 0.3216 \left(\dfrac{\rho_v}{\rho_l}\right)^{0.084} \left(\dfrac{L}{D}\right)^{-0.942} we_l^{-0.034}$
Wojtan *et al* (2006)	$\dfrac{q''_{CHF}}{G i_{lv}} = 0.437 \left(\dfrac{\rho_v}{\rho_l}\right)^{0.073} \left(\dfrac{L}{D}\right)^{-0.72} we_l^{-0.24}$
Zhang *et al* (2006)	$\dfrac{q''_{CHF}}{G i_{lv}} = 0.0352 \left(we_D + 0.0119 \left(\dfrac{\rho_v}{\rho_l}\right)^{0.361} \left(\dfrac{L}{D}\right)^{2.31}\right)^{-0.295}$ $\left(\dfrac{L}{D}\right)^{-0.311} \left[2.05 \left(\dfrac{\rho_v}{\rho_l}\right)^{0.170} - x_{in}\right]$
Bowers & Mudawar (1994)	$\dfrac{q''_{CHF}}{G i_{lv}} = 0.16 \left(\dfrac{L}{D}\right)^{-0.54} we_l^{-0.19}$
Qi *et al* (2007)	$\dfrac{q''_{CHF}}{G i_{lv}} = (0.214 + 0.14\mathrm{Co}) \left(\dfrac{\rho_v}{\rho_l}\right)^{0.133} we_l^{-0.133} \dfrac{1}{1 + 0.03\frac{L}{D}}$

2010) and so on. Acoustic methods are based on the fact that ultrasonic waves are reflected at the gas–liquid interface and the time interval between the emitted and the reflected signal is converted into the film thickness (Lu *et al* 1993; Pederson *et al* 2000). Radiological approaches use radiation, such as X-ray, gamma-ray, and neutrons. In radiological methods, the different attenuation characteristics in gas and liquid are used to measure the liquid film thickness (Stahl & Rohr 2004; Saito *et al* 2005). Electrical methods are classified into two approaches: conductance and capacitance. The conductance (or the capacitance) between electrodes placed on the inner surface of the pipe is dependent on the conductivity (or the dielectric constant) of each phase and its morphology. The conductance method is cheaper than the other methods: it is non-intrusive and does not disturb flows. In addition, the response to the change of the liquid film thickness is very fast. To measure liquid film thickness, Coney (1973) proposed a conductance probe composed of two parallel electrodes with unequal widths and separated by an insulator and developed a theoretical model to analyze the relationship between the conductance and the liquid film thickness. Fukano (1998) suggested in-line and concentric ring-type probes and successfully applied them to the liquid film thickness measurement in a rectangular test section. Conte (2003) and Geraci *et al* (2007) proposed flush-mounted pin probes. The method is used for very thin films, typically up to 2 mm (pipe diameter = 127 mm). However, their work dealt with measuring thicker films up to 11 mm (pipe diameter = 80 mm). More recently, Liu *et al* (2007) used Electrical Capacitance Tomography (ECT) based image reconstruction to extract the liquid film thickness information.

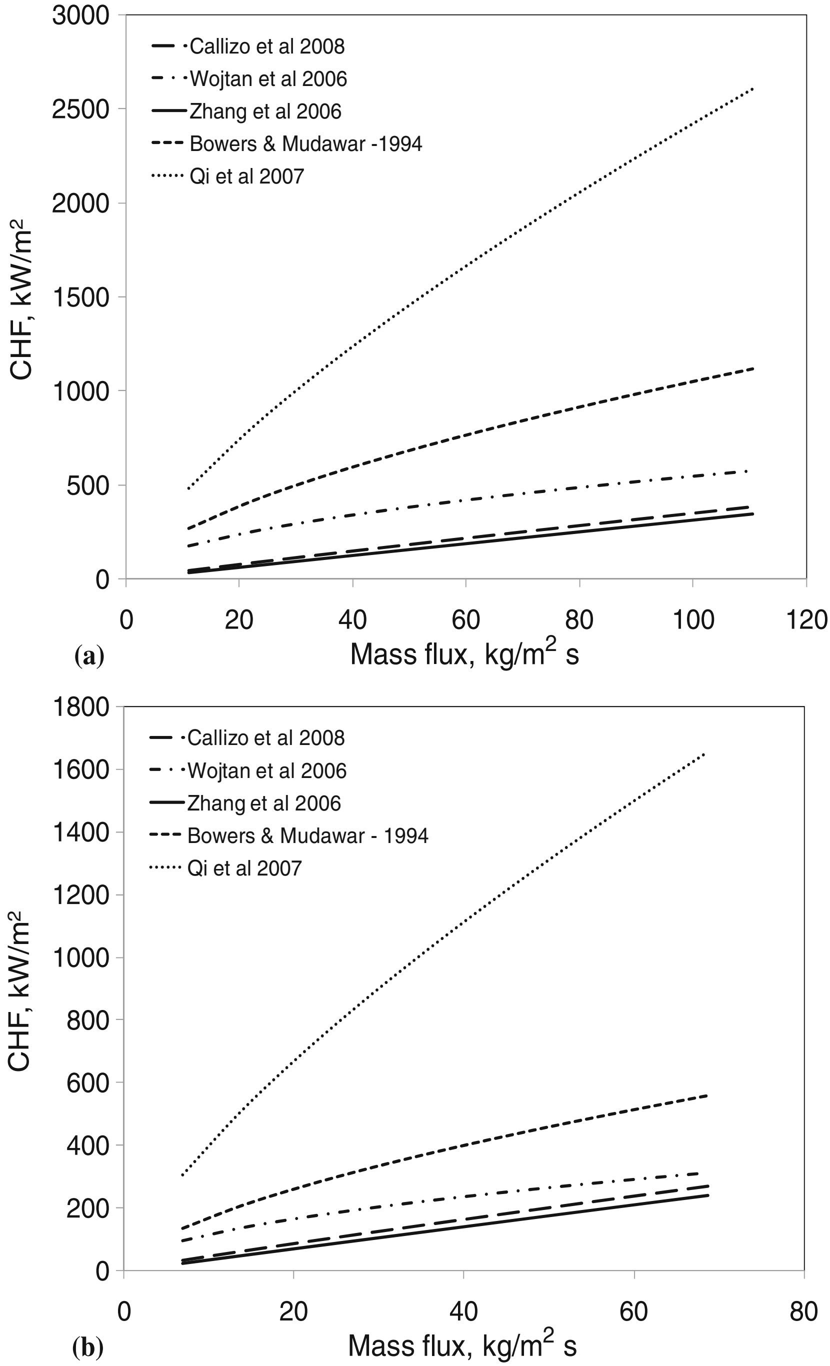


Figure 12. (**a**) Comparison of CHF correlation using water as working fluid. (**b**) Comparison of CHF correlation using *n*-Pentane as working fluid.

Table 9. Summary of investigations on boiling in mini and microchannels.

Author/year	Fluid ranges of parameters	Channel size/D_h	Heat transfer	Pressure drop	Flow patterns	Remarks
	G (kg/m^2), q (kW/m^2)	(mm), Horiz. (unless otherwise stated)				
Downing *et al* (2000)	R113, ranges not clearly stated	Circular multichannels in coils, $D_h = 0.23$–1.86, helix dia. $= 2.8$–7.9	Not studied	Single and two-phase	Not studied	As the helical coil radius became smaller, pressure drop reduced possibly due to different flow pattern
Hetsroni *et al* (2003)	Water, Re = 20–70, $q = 80$–360	Triangular parallel channels, $h = 55$, $n = 21, 26$, $D_h = 0.129$–0.103, $L = 15$	Measured, but data not reported	Not reported	Periodic annular	Periodic annular flow observed in microchannel. There is a significant enhancement of heat transfer during flow
Kennedy *et al* (2000)	Water, $G = 800$–4500, $q = 0$–4 MW/m^2	Circular, $D = 1.17$ and 1.45, $L = 160$	Onset of nucleate boiling starts instability	Not reported	Not studied	Heat flux at the onset of flow instability was 0.9 of the heat flux required for sat. Vapour at exit, similarly, G at OFI was 1.1 times G for saturated exit vapour condition
Lakshminarasimhan *et al* (2000)	R11, $G = 60$–4586,	Rectangular, $1 \times 20 \times 357$	Subcooled and saturated flow boiling	Not measured	Boiling incipience observed through LCD	Boiling front observed in laminar flow, not visible in turbulent flow due to comparable **h** before and after, saturated flow boiling data correlated by Kandlikar correlation

Table 9. (continued).

Author/year	Fluid ranges of parameters	Channel size/D_h	Heat transfer	Pressure drop	Flow patterns	Remarks
Kandlikar *et al* (2001)	Water, $G =$ 80–560 kg/m^2 s	Rectangular, $1 \times 1 \times 60$	Subcooled and saturated flow boiling	Measured instantaneous values of pressure drop	High-speed photography on single and multiple channels	Flow oscillations and flow reversal linked to the severe pressure drop fluctuations, often leading to flow reversal during boiling
Khodabandeh & Palm (2001)	R134a and R600a, mass flow rate not measured, $q = 28$–424 kW/m^2	Circular tube, 1.5 dia.	Htc determined from experiments	Not studied	Not studied	Htcs compared with 11 correlations. Mass flow rate is not measured, it is assumed to be constant in all experiments perhaps causing large discrepancies with correlations at higher **h**
Kim & Bang (2001)	R22, $G = 384$–570, $q = 2$–10	Square tube, 1.66 dia.	Htc obtained as a function of quality 0–0.8	Not studied	Flow pattern observed in rectangular channels of 1:32, 1:78 mm c/s	Heat transfer coefficient somewhat higher than Shah (1960) and Gungor & Winterton (1986) correlation predictions. Slug flow pattern seen as the dominant flow pattern
Koo *et al* (2001)	Water, 200 W heat sink	Parallel rectangular microchannel $50 \times 25\ \mu m^2$	Thermal simulation of the VLSI chip with the heat sink	Two-phase pressure drop using homogeneous flow model	Thermal profile predicted on the chip and compared with experiments	The pressure drop predictions using homogeneous flow model were in good agreement with data. The Kandlikar correlation was used for predicting the heat transfer rate. Predictions in good agreement with data

Table 9. (continued).

Author/year	Fluid ranges of parameters	Channel size/D_h	Heat transfer	Pressure drop	Flow patterns	Remarks
Lee *et al* (2003)	R113 $G = 50$–200, $q = 3$–15	Rectangular 0.4–2 high, 20 wide	Saturated, $x = 0.15$–0.75	Measured over 260 mm long section	Not reported	Pressure drop correlated using Martinelli-Nelson parameter, H.T. predicted will by Kandlikar correlation for film Re > 200, new correlation developed using film flow model for film Re < 200
Serizawa & Feng (2001)	Air–water, $j_L = 0.003$–17.52 m/s, $j_G = 0.0012$–295.3 m/s, Steam–water, ranges not given	Circular tubes, diameters of 50 μm for air–water, and 25 μm for steam–water	Air–water two-phase flow, steam–water details not given	Not studied	The ranges of flow rates studied	Two new flow patterns identified: liquid ring flow and liquid lump flow
Warrier *et al* (2001)	FC84, $G = 557$–1600, $q =$ 59.9	Rectangular, dimensions not available, hydraulic dia. = 0.75	Single-phase and two-phase	Pressure drop measured over 325.1 mm long test section	Not studied	Overall pressure, drop and local htc determined. Htc correlated as a function of boiling number alone
Huo *et al* (2007)	R134a, $G = 100$–500 system pressure 9–12 bar exit quality up to 0.9; $q = 13$–150 W/m^2	Two stainless steel tubes of internal diameter 4.26 and 2.01	HTC versus quality, for different pressure drops, heat and mass fluxes	Not studied	Not studied	Good agreement between experimental data and modern flow models such as 3 zone evaporation model without predicting the effect of channels diameter and partial dry-out
Shiferaw *et al* (2009)	R134a $G = 100$–600 kg/m^2 s; $q = 16$–150 kW/m^2 and pressure 6–12 bar	Stainless steel tube of 1.1 internal diameter	HTC plotted vs quality, for different pressure drops, heat fluxes and mass fluxes	Not studied	Dispersed bubble, bubbly, confined bubble, slug, churn, annular	3-zone evaporation model predicts the annular flow especially at relatively low pressure. The dry-out region is highly over-predicted by the model

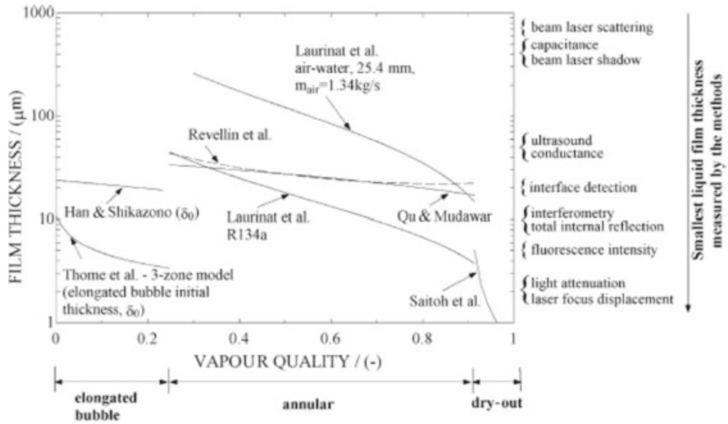


Figure 13. Comparison between the smallest film thickness measured by different methods and some estimated film thicknesses according to models from the literature. R134a, diameter of 770 lm, mass velocity of 467 kg/m^2 s, heat flux of 10 kW/m^2 and saturation temperature of 100°C.

Recently, it is reported that liquid film thickness provides information to define the small to micro-scale transition in two-phase flow using high speed imaging (Ong & Thome 2010). Attempt was also made by Revellin & Thome (2007) for collecting experimental data for 0.5 to 0.8 mm channel size with two refrigerants (R134a and R245 fa). They developed a model and compared their results with the experimental data of Tibiriçá & Ribatski (2009). It was observed that the model failed in predicting when its estimated film thickness was used to calculate the local heat transfer coefficient. Tibiriçá *et al* (2010) also carried out experiments in order to measure film thickness in microchannels. It is also mentioned that once the local liquid film thickness is known, it becomes easier to estimate the superficial void fraction.

Despite its importance, only few authors have addressed the issue of measuring the liquid film thickness in micro-scale two-phase flow system. This is mainly because the main challenge in measurement of film thickness is related to the small dimensions involved, and the inherent difficulties in developing non-intrusive transducers at these scales. Moreover, the small film thickness requires a non-invasive measuring method to avoid disturbance to the flow. As the experimental data reported is limited, measurement of film thickness in a small channel diameter looks to be a promising area for further research. The comparison between different film thickness measurement methods and model based estimation of film thicknesses from the literature is presented in figure 13.

2.8 *Modelling of boiling in small channels*

There are a considerable number of heat transfer correlations that can be used for modelling of flow boiling in different channels. Comprehensive review on this topic has been carried out by Thome (1996). A paramount difficulty in development of correlations for flow boiling at small mass flow rates and moderate heat fluxes (refrigeration, air-conditioning) or alternatively at large mass flow rates and high heat fluxes (power engineering). Capturing all these trends with only single correlation is a very difficult task.

Kattan *et al* (1998a) argues that the models based on distinguishing between flow regimes should be seriously considered for general use. Wojtan *et al* (2005) postulated models for stratified wavy and mist flow regimes. These models are very promising; however they fall under the class of the regime-dependent models. They require knowledge of the particular flow regime, as well as the proportion of liquid and gaseous phase in the flow. For that reason such model is difficult to be applied in engineering practice, as it requires prior knowledge of the flow regimes and conditions. Heat transfer issues in small diameter channels received attention in numerous papers, e.g., Bergles *et al* (2003), Kandlikar (2001) etc.

One of the major difficulties in modelling two-phase flow is the determination of the geometry of the flow field which can adopt several configurations commonly referred to as flow pattern or flow regimes. Significant engineering design values such as heat transfer and pressure drop are closely related to this distribution, the calculation of two-phase flow pattern by means of CFD modelling can be very useful (Schepper *et al* 2008). These flow calculations are computationally very demanding. They made use of existing CFD code for modelling of two-phase flow in horizontal tubes for air–water and liquid–vapour flow. For simulations the Eulerian–Eulerian as well as VOF model approach is chosen. In the VOF model, the Eulerian framework is taken for both the phases combined with reformulation of interface forces on volumetric basis. Simulation work involves 3D simulations of horizontal flow regimes observed in two-phase liquid–vapour system. The simulation results for the different horizontal flow regimes can be found in figure 14. It can be concluded that the VOF model in combination with PLIC interface reconstruction method implemented in existing code predicts the expected flow regime accurately.

Similarly, Anglart & Caraghiaur (2011) developed two-phase annular mist flows with evaporating liquid films. This type of flow exists in boiling channels prior to onset of dry-out. Annular-mist flows have a very complex structure since they contain liquid and vapour, both as the continuous and the dispersed phase. Due to this feature, both the Eulerian–Eulerian as well as the Eulerian–Lagrangian approaches were often used to model such flows. The two approaches, with pertinent conservation equations and closure relationships were used by Anglart & Caraghiaur (2011) for modelling annular-mist flows.

In the CFD code, for example in Fluent software, the two-phase flow phenomenon can be modeled in several ways. The two-phase flow models used in Fluent are Euler–Euler models.

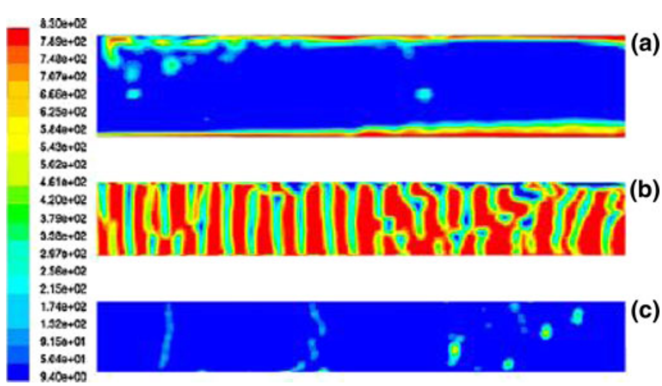


Figure 14. Contours of mixture density for liquid–vapour gasoil flow. (**a**) Annular flow, (**b**) bubbly flow, (**c**) dispersed flow (Schepper *et al* 2008).

These models treat the different phases as interpenetrating continua. Since the volume of a phase cannot be occupied by the other phases, the concept of phase's volume fraction is introduced. The conservation equations for each phase are derived to obtain a set of equations, which have a similar structure for all phases. For the higher void fraction cases, such as slug flow it is recommended to use the VOF model (Dostal *et al* 2008). The slug flow is the most important flow regime in two-phase flow system primarily because of numerous industrial and practical applications. Taha & Cui (2006) studied CFD modelling of slug flow in vertical tubes. They have used Fluent software to simulate the motion of single Taylor bubble rising in motionless or flowing liquid through a vertical tube using VOF approach. A fully developed velocity profile is imposed at the inlet and relative movement between the liquid and wall generates a velocity profile as shown in figure 15a. The geometric reconstruction scheme that is based on piece linear interface calculation method (PLIC) is applied to reconstruct the bubble free interface. 3D simulation of bubble was conducted to capture the 3D nature of the flow. The unstable and transient nature of the flow surrounding the bubble from laminar to turbulent, produces significant modification to the bubble profile and stability as shown in figure 15b. There bubble tail oscillates and small bubble can be seen shed at the bubble rear.

Similarly, numerical simulation by CFD using the volume of fluid (VOF) method (Hirt & Nicholos 1981) has been applied in modelling two-phase bubbly flow since 1980s. The advantages of the VOF method may lie in its simple concept, reasonable accuracy and is relatively easy to implement. Thus it could be used to simulate complex phenomena such as interface

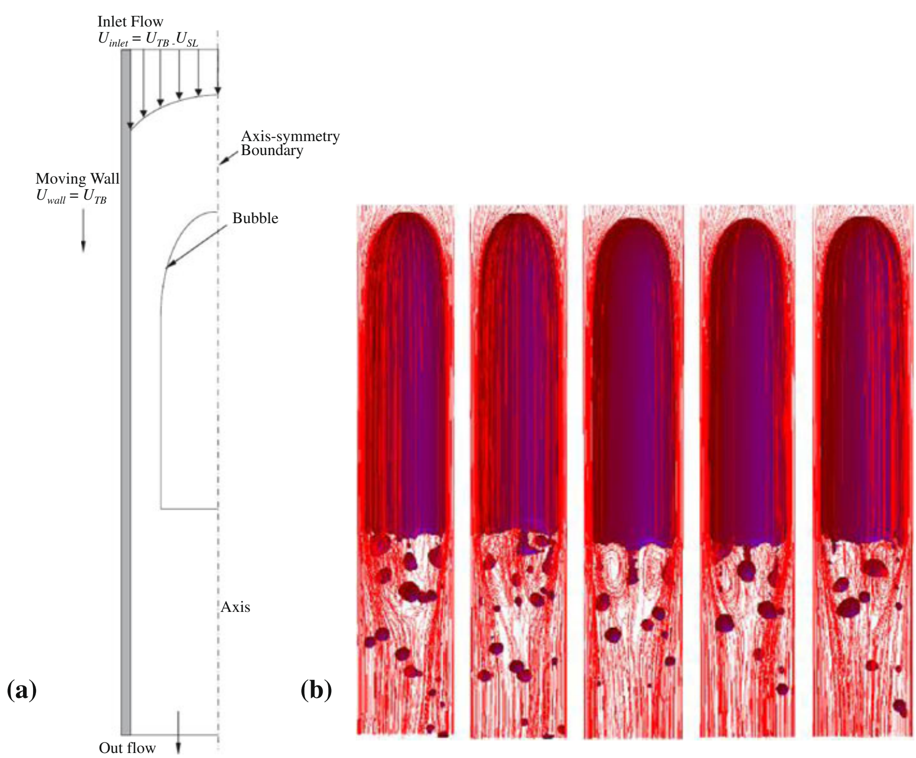


Figure 15. (**a**) Initial BC for Taylor bubble rising in vertical pipe (Taha & Cui 2006). (**b**) Progression of wake flow pattern behind Taylor bubble rising through a stagnant liquid.

Table 10. Review of computational fluid dynamic study in two-phase flow modelling.

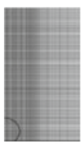
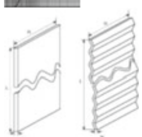
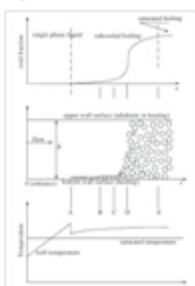
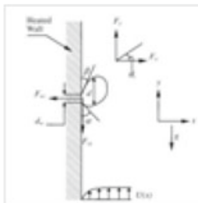
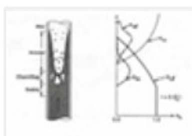
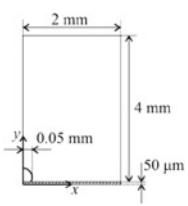
Author	Geometry	Software	Approach	Studied parameters	Remarks
Kocar & Sokmen (2009)		FLUENT – 6.3	Two fluid E–E	Void distribution data	Validated a nucleate boiling flow model to provide sensitivity and uncertainty analysis for flow boiling model parameters
Zu & Yan (2009)		FLUENT – 6.3	VOF for gas–liquid interface	Bubble deformation, evolution of bubble shape and temperature field	EHD effects on bubble deformation under pseudo-nucleate boiling conditions; VOF is employed to track the interface between two-phases
Guo *et al* (2011)		FLUENT + mathematical model	VOF for gas–liquid interface	Impact of wall contact angle on bubble shape, nucleation, ΔP, HTC	Wall contact angle has a significant effect on the bubble shape
Xu *et al* (2006)		Mathematical modelling	1-D, non-equillibrium, two-fluid model	Void fraction vs channel length, two-phase velocity distribution	Predicted results highlight that buoyancy force plays an important role on the void fraction evolvement, especially at low velocity for vertical down-flow

Table 10. Continued.

Author	Geometry	Software	Approach	Studied parameters	Remarks
Yeoh *et al* (2008)		Mathematical modelling	Wall heat partition model, mechanistic approach for bubble frequency evaluation, two fluid model and population balance model	Bubble diameter vs height, void fraction vs height, local void fraction profiles	Bubble departure and lift-off diameters that were determined mechanistically based on the force balance model showed good agreement against experimental observations
Lahey (2007)		CMFD solver i.e., NPHASE	Multidimensional single and multiphase flows using DNS and CMFD models	Air/water bubbly flow, void fractions vs radial positions, RMS velocity fluctuations, interacting 3-D bubble, bubble formation	Remarkable progress has been made to reliably predict multiphase flow phenomena
Kunkelmanna & Stephana (2011)		OpenFOAM	VOF	Bubble growth, temperature profile with time	Study deals with 1-D sucking interface problem and growth of spherical bubble. Results showed good agreement with analytical methods

breakup and coalescence etc. (Gopala & van Wachem 2008). However, the 3-D interface reconstruction in VOF is numerically prohibitive and therefore may result in some uncertainties on the interface curvature and thus on the surface tension forces. Son & Dhir (1999) developed a two-dimensional numerical model of growth and departure of single vapour bubbles during nucleate pool boiling. In the model, the level set technique was used for tracking the liquid vapour interface. The model was then extended by Mukherjee & Kandlikar (2005) to 3-D cases to simulate the growth of a bubble from a nucleation site on the side wall of a rectangular channel to confinement. The channel was restricted in length and the interaction between pressure changes and evaporation was not taken into account. The main drawback of the level set methods is that its numerical computation can generate mass loss in under-resolved regions (Sussman *et al* 1994). Yan & Zu (2007, 2008) developed their lattice Boltzmann method (LBM) with the capability of modelling of two-phase flow of fluids with large liquid–gas density ratio, and fluid wall interactions to simulate bubbles behaviour in mini-/microchannels. Despite its numerical advantages, the method is time consuming and generally difficult. Various studies on this are summarized in table 10 which covers different geometries, modelling approaches and key parameters for two-phase flow boiling.

3. Summary, conclusions and path forward

A review on flow boiling in small channels is presented. It was observed that flow boiling has been studied extensively for last few decades. However issues concerning flow boiling heat transfer and two-phase flow transport phenomena in small diameter channels are quite different from these encountered in conventional large channels. Published literature on this is summarized in this paper. It was highlighted that the channel size classification is merely a guide to help us identify the channel size ranges in various applications. The operating parameters, such as system pressure, and fluid properties, such as density, viscosity, and surface tension, have a major influence on prevailing flow characteristics.

Identification of flow regimes in a heated section is very difficult and complicated task. This is because spatially varying flow regime ranging from bubbly flow to mist flow occur in such a heated section. Relatively limited research have been carried out so far on flow boiling patterns and heat transfer in small diameter tubes for different fluids. It was also observed that different experimental conditions were used (such as different temperature, mass flux, heat flux and pressure) by different investigators, which makes it difficult to compare results directly. In addition, the identification of flow patterns may be greatly affected by the observer's subjectivity and the experimental technique employed.

In two-phase flow boiling, continuously varying flow regimes also make development of generalized correlations for frictional pressure drop and effective heat transfer coefficient rather complex. The prediction of critical heat flux (CHF) also relies heavily on empirical correlations derived from experimental data. Available experimental and modelling studies on flow boiling in small channels are critically reviewed in this work. Efforts have also been made to evolve some broad guideline for the future research in this important area. Some of the key conclusions drawn from this review can be outlined as:

- Classification of small diameter channels into mini and micro (where surface phenomena dominates) channels is not clearly established in the literature. Further work is necessary to clarify changes in regimes of heat transfer and boiling over intermediate (between micro- and mini) range of diameters of small channels.

- Flow visualization and transition criteria of two-phase flow in small channels should relate to flow boiling heat transfer mechanisms which intrinsically relate to flow patterns. The continuous variation in flow regimes along the boiler tubes complicates development of generalized correlations for effective pressure drop and heat transfer. Further work on relating the observed overall pressure drop and heat transfer data to underlying spatio-temporal scales and fluid dynamics is needed to enhance appropriate interpretation of reported experimental data and therefore subsequent applications.
- The accurate prediction of void fraction in vertical channels is of immense importance in determining the two-phase pressure drop. Therefore, a few void fraction correlations are reported in table 6 and compared in figures 7a–b. It was observed that Lockhart & Martinelli (1949) correlation is most useful for the design calculations.
- It is observed that a strong dependence of CHF on mass flux, heated length and inner diameter. Reported CHF correlations tried to compare using water and *n*-pentane as working fluid. It was observed that CHF predicted by Qi *et al* (2007) correlation was maximum at particular mass flux as compared to other reported correlations.
- A systematic and consistent experimental data base on flow boiling characteristics of a small-channel is needed. The lack of such data is also hindering further development in computational modelling.

Suggestions for further work

- It is important that flow regime maps be first developed based on simultaneous high-speed visualization and local heat transfer and pressure drop measurements in carefully characterized experimental set-ups. Such a broad database would then allow the development and validation of models which are based on the prevailing flow regimes.
- A generalized effective pressure drop and heat transfer model should be targeted by incorporating various flow regimes, heat transfer mechanisms, channel sizes, component compositions, instabilities and so on.
- The film thickness measurement requires a non-invasive measuring method to avoid disturbance to the flow. Hence, it is observed that this is a promising area for further research where very limited information is available on the measurement of film thickness. This may also shed light on dry-out phenomena and related instabilities.
- Better computational models and modelling approaches are needed to develop reliable, accurate and yet tractable, numerical simulations of boiling flows. Accurate prediction of bubble generation, growth and departure from wall is still difficult and remains a challenge. Detailed experimental data for validation and possibly multi-scale modelling may provide a way forward.

It is envisaged that a systematic research on two-phase flow and flow boiling in small channels will bring advancement of knowledge and meet the practical requirements in various applications. This review may provide a starting point for this long and arduous journey.

Nomenclature

A_S heat transfer surface area, m^2
Bo boiling number
Co convection number $(\rho_G/\rho_L)^{0.5}\ ((1-x)/x)^{0.8}$
D_h tube diameter, mm

f	fanning friction factor
F_{Fl}	fluid-surface parameter
Fr	Froude number
G	mass flux, $\mathrm{kg/m^2\ s}$
h_{LO}	all-liquid flow single-phase heat transfer coefficient, $\mathrm{W/m^2\ {}^\circ C}$
h_{TP}	two-phase heat transfer coefficient, $\mathrm{W/m^2\ {}^\circ C}$
i_{LG}	specific enthalpy, J/kg
L	length of tube, m
Nu	Nusselt number
Pavg	test section average pressure, kPa
q	heat flux, $\mathrm{W/m^2}$
Re_{Lo}	Reynolds number
T	temperature, °C
X	vapour quality
d	hydraulic diameter = 4 (cross-sectional area/perimeter of tube), m
F	Enhancement factor, or Chen's Reynolds number factor
g	acceleration due to gravity, $\mathrm{m/s^2}$
k	liquid thermal conductivity, W/mK
M	molecular mass
N	number of data points
δ	thickness of film
$\mathrm{p}r$	reduced pressure
S	suppression factor
$(1-x)$	dryness fraction
$\Delta P_{\mathrm{static}}$	static pressure drop
$\Delta P_{\mathrm{momentum}}$	momentum pressure drop
$\Delta P_{\mathrm{friction}}$	frictional pressure drop
ρ_H	homogenous density
H	vertical height
θ	angle with respect to horizontal
ρ_L	liquid density
ρ_G	gas density
ε_H	void fraction
V_V/V_L	velocity ratio or slip ratio
x	vapour quality
f_{tp}	two-phase friction factor
$\dot{m}$	mass flow rate, kg/s
D_i	inner diameter, m
ρ_{tp}	two-phase density
μ_{tp}	quality averaged viscosity
ΔP_L	liquid phase flow pressure drop
Φ_{fr}	Friedel two-phase multiplier
Φ_{Ltt}	two-phase multiplier for liquid phase
Φ_{Gtt}	two-phase multiplier for gas phase
Φ_{gd}	Gronneurd correlation two-phase multiplier
Φ_{ch}	Chishlom two-phase multiplier
Φ_{Bf}	bankoff two-phase multiplier

Φ_{chawla} Chawla two-phase multiplier
μ_L liquid dynamic viscosity
μ_G gas dynamic viscosity
h heat transfer coefficient, W/m^2 K
q_{in} heat flux, W/m^2
$\mathrm{T_o}$ Temp. at outlet
$\mathrm{T_{in}}$ Temp. at inlet
z axial location, length
C_P specific heat at constant pressure
A_c cross sectional area
i_{lv} latent heat of vaporization
T_{sat} saturation temperature
D_i internal diameter
q_c critical heat flux, W/m^2,
D_h hydraulic tube diameter

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