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Review of two-phase flow and flow boiling of mixtures in small and mini channels

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Abstract

Two-phase flow and flow boiling phenomena of fluidic mixtures in small and mini channels are becoming important in the miniaturization of thermal systems. This paper aims to present a state-of-the-art review in this important area and to identify what have been done so far and what still need to be done in the future. Firstly, various definitions of small and mini channels are described and the criteria based on these definitions are compared with each other. Comments on different viewpoints of the channel size classifications are acknowledged. Secondly, the background of two-phase flow and flow boiling of mixtures is described. Then, the current research status of two-phase flow and flow boiling of mixtures in normal size channels is presented as it is a basis for the study of two-phase flow and flow boiling of mixtures in small and mini channels. Finally, an overall review of two-phase flow and flow boiling of mixtures in small and mini channels is presented. It is concluded that the available study of two-phase flow and flow boiling of mixtures in small and mini channels is rather scarce and a systematic knowledge of two-phase flow and flow boiling of mixtures in small and mini channels is required. Based on this review, the future research directions including both fundamental and applied research in this area have been indicated.

Keywords: Two-phase flow; Flow boiling; Mixtures; Small and mini channels; Micro-channels; Micro heat transfer

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1. Introduction

Miniaturization has recently become the key word in many advanced technologies as well as traditional industries. The miniaturized systems are being progressively applied in commercial sectors such as the electronics, pharmaceutical, and medical industries, as well as in the military sector, for biological and chemical warfare defense, as an example. With the emergence of micro- and meso-scale thermal, fluidic, and chemical systems, the development of ultra-compact heat exchangers, miniature and micro pumps, miniature compressors, micro-turbines, micro thermal systems for distributed power production has become an important

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agenda of many researchers, research institutions, and funding agencies (Shekarriz, 2000). Heat exchanger miniaturization is essential not simply from the standpoint of producing compact systems, but for the challenge of exchanging heat with yet smaller and smaller surface areas, as is the case for cooling of electronic devices and sensors. In addition, it has been shown that proper miniaturization by use of micro and mesochannels can result in higher system efficiency (Shekarriz and Call, 1999). As one of the most important heat transfer modes, flow boiling in small hydraulic diameter channels is becoming increasingly important in various applications. These include the applications of compact and micro heat exchangers in traditional industries and the wide practical applications in highly specialized fields such as bioengineering, micro-fabricated fluidic systems, microelectronics, aerospace, micro heat pipes and others (Kandlikar, 2001, 2002; Ghiaasiaan and Abdel-Khalik, 2001). The advantage of small hydraulic diameter channels lies in their high heat transfer coefficient and significant potential in decreasing the heat exchanger surface area. However, flow boiling heat transfer and two-phase flow transport phenomena in small hydraulic diameter channels are quite different from that in normal size channels. Therefore, it is of great significance to understand the fundamental phenomena including flow patterns, flow boiling heat transfer, critical heat flux (CHF) and two-phase flow pressure drop so as to develop related new high technologies.

Due to the significant differences of transport phenomena in small diameter channels as compared to normal size channels, one very important issue should be clarified about the distinction between small diameter channels and normal size channels at first. However, a universal agreement is not clearly established in the literature. Instead, there are various definitions on this issue. Shah (1986) defined a compact heat exchanger as an exchanger with a surface area density ratio >700 m^2/m^3. This limit translates into a hydraulic diameter of <6 mm. According to this definition, the distinction between small diameter channels and normal size channels is 6 mm. Mehendale et al. (2000) defined various small and mini heat exchangers in terms of hydraulic diameter D_h, as:

- Micro heat exchanger: $D_h = 1$–100 μm.
- Meso heat exchanger: $D_h = 100$ μm–1 mm.
- Compact heat exchanger: $D_h = 1$–6 mm.
- Conventional heat exchanger: $D_h > 6$ mm.

According to this definition, the distinction between small diameter channels and normal size channels is 6 mm. Based on engineering practice and application areas such as refrigeration industry in the small tonnage units, compact evaporators employed in automotive, aerospace, air separation and cryogenic industries, the application in the field of microelectronics and micro-electro-mechanical-systems (MEMS), Kandlikar (2001, 2002) defined the following ranges of hydraulic diameters which are attributed to different channels:

- Conventional channels: $D_h > 3$ mm.
- Mini channels: $D_h = 200$ μm–3 mm.
- Microchannels: $D_h = 10$–200 μm.

According to this definition, the distinction between small diameter and normal size channels is 3 mm. In addition, there are several important dimensionless numbers which are used to represent the feature of fluid flow in small and mini channels. According to these dimensionless numbers, the distinction between small diameter and normal size channels may be classified as well. Triplett et al. (1999) defined flow channels with hydraulic diameters D_h of the order, or smaller than, Laplace constant L as follows:

$$L = \sqrt{\frac{\sigma}{g(\rho_l - \rho_v)}} \tag{1}$$

as small diameter channels which are extensively applied in compact heat exchangers, microelectronic cooling systems, nuclear reactors, chemical processing and small-sized refrigeration systems. Where σ is surface tension, g is gravitational acceleration, ρ_l and ρ_v are respectively liquid and vapor densities. Kew and Cornwell (1997) proposed Confinement number Co for small diameter channels, as

$$Co = \frac{1}{D_h}\sqrt{\frac{4\sigma}{g(\rho_l - \rho_v)}} \tag{2}$$

which is based on the definition of Laplace constant. Based on a linear stability analysis of stratified flow and the argument that neutral stability should consider a disturbance wavelength of the order of channel diameter, Brauner and Moalem-Maron (1992) derived the following Eotvös number *Eö* criterion for the dominance of surface tension for small diameter channels:

$$E\ddot{o} = \frac{(2\pi)^2\sigma}{(\rho_l - \rho_v)D_h^2 g} > 1 \tag{3}$$

It seems that the definitions of small diameter channels are quite confused because there are different criteria of small diameter channels as described above. In the literature, some researchers took a hydraulic diameter of 6 mm as the criterion of small diameter channel in their study (e.g. Wolk et al., 2000). Some researchers took a hydraulic diameter of 7.5 mm as the criterion of small diameter channels (e.g. Wongwises et al., 2000a). Apparently, there is no unified definition of small diameter channels until now. The main purpose of the present paper is to review two-phase flow and flow boiling of mixtures in small and mini channels. Taking into account of the effect of physical properties of fluids, the threshold diameters (the threshold diameter is defined as the transition diameter from normal size channels to small diameter channels in this paper.) based on the available criteria with respect to small diameter channels are compared respectively for pure fluids and mixture fluids so as to have an overview about the dimension scales of small diameter channels. The criteria used here are respectively the fixed threshold diameters defined by Kandlikar (2001, 2002) and the threshold diameters based on Laplace constant (Eq. (1)), Confinement number (Eq. (2)) and Eotvös number (Eq. (3)). The selected fluids are pure fluids including water, R134a, R22 and carbon dioxide (CO_2), and mixture fluids including R404a, R407c, R410a and R507a. The physical properties of these fluids are obtained from software REFPROP Version 6.01 (1998). Table 1 shows the critical pressures and temperatures of these fluids. Fig. 1(a)–(d) show the variation of the threshold diameters with reduced pressure for pure fluids which are water, R134a, R22 and CO_2, respectively. Fig. 2(a)–(d) show the variation of the threshold diameters with reduced pressure for mixture fluids which are R404a, R407c, R410a and R507a, respectively. Unlike the fixed values of threshold diameters defined by Kandlikar, the threshold diameters defined by the dimensionless numbers decrease when reduced pressures increase. It can been seen that the threshold diameters are greatly affected by the physical properties of the fluids. For example, according to the criteria based on Confinement number, water has a maximum threshold diameter of about 5 mm while R134a has a maximum threshold diameter of about 2.3 mm. It can also be seen that there are big differences among the threshold diameters

Table 1
Critical pressures and temperatures of the selected fluids

Fluids	Composition by weight	Critical pressure (MPa)	Critical temperature (°C)
Water	100% Water	22.1	374
R134a	100% R134a	4.06	101.1
R22	100% R22	4.99	96.2
CO_2	100% CO_2	7.38	31.1
R404a	44% R125 52% R143a 4% R134a	3.74	72.2
R407c	23% R32 25% R125 52% R134a	4.63	86.1
R410a	50% R32 50% R125	4.77	70.1
R507a	50% R125 50% R143a	3.72	70.7

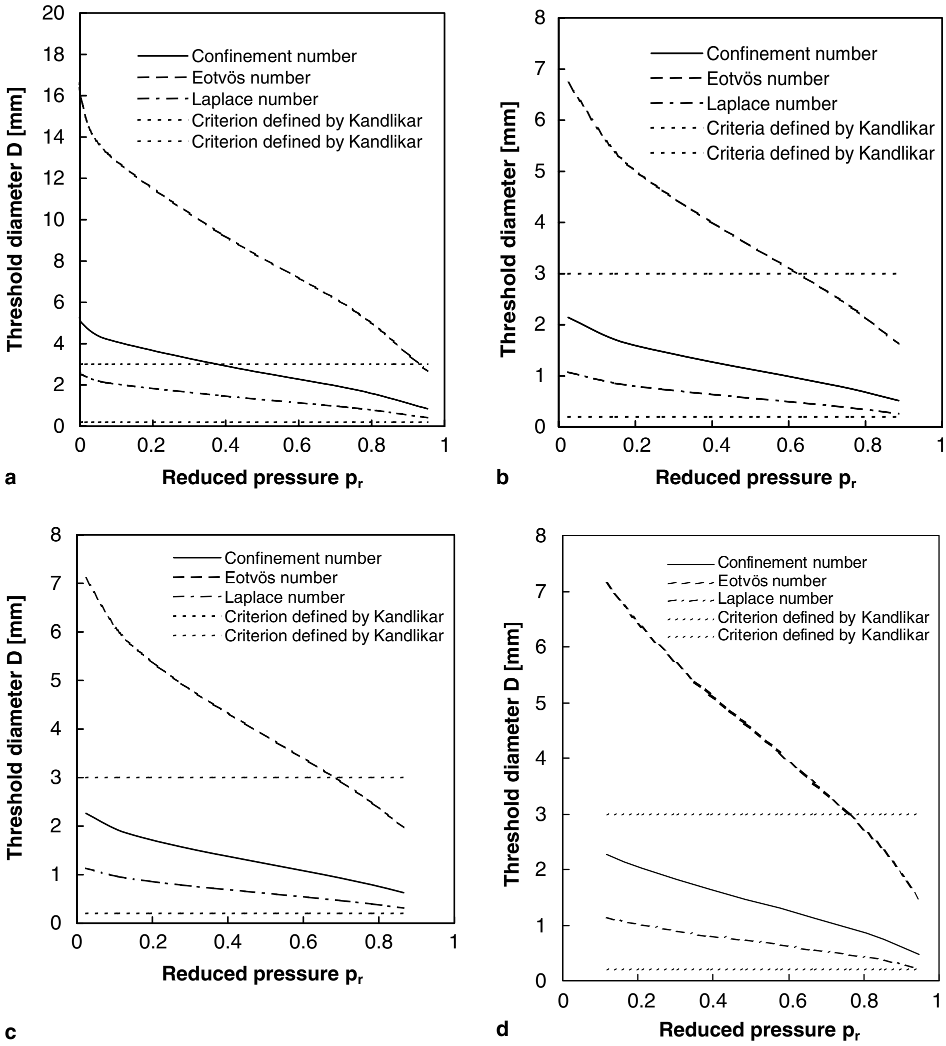

Fig. 1. Comparison of various definitions of threshold diameters for different pure fluids: (a) water; (b) R134a; (c) R22; (d) CO_2.

defined by various criteria. According to all the criteria, the threshold diameter ranges from around 3 mm to 15.8 mm for water while the threshold diameter ranges from around 1 mm to 6.8 mm for R134a. For the mixture fluids, the maximum threshold diameter is mostly around 6 mm. By comparison, it can be obtained that the threshold diameters based on the available definitions of small diameter channels are quite different. It must be pointed out that the channel size classifications made in the literature cover all processes including single-phase gas and liquid flows, adiabatic two-phase flow, flow boiling as well as condensation. The use of dimensional numbers such as Confinement number etc for channel size classifications is rather inaccurate as the effect of gravity is rather small. It can be seen that there are big differences among these classifications from Figs. 1 and 2. These dimensional numbers are not recommended for small channels. Therefore, fixed values are used for the channel size classification by Kandlikar (2001, 2002). However, as the channel size classification is an open topic, it is quite normal that there are different viewpoints of channels size classifications in the literature. Considering the maximum threshold diameter of mixtures are around 6 mm (some are around 7 mm) according to the definition of Confinement number, a hydraulic diameter of about 6 mm is used as the distinctive line in the present review. Thus, it may cover all other definitions of small diameter channels and can provide more information on two-phase flow and flow boiling of mixtures in small and mini channels although there is no unified definition on the channel size classification.

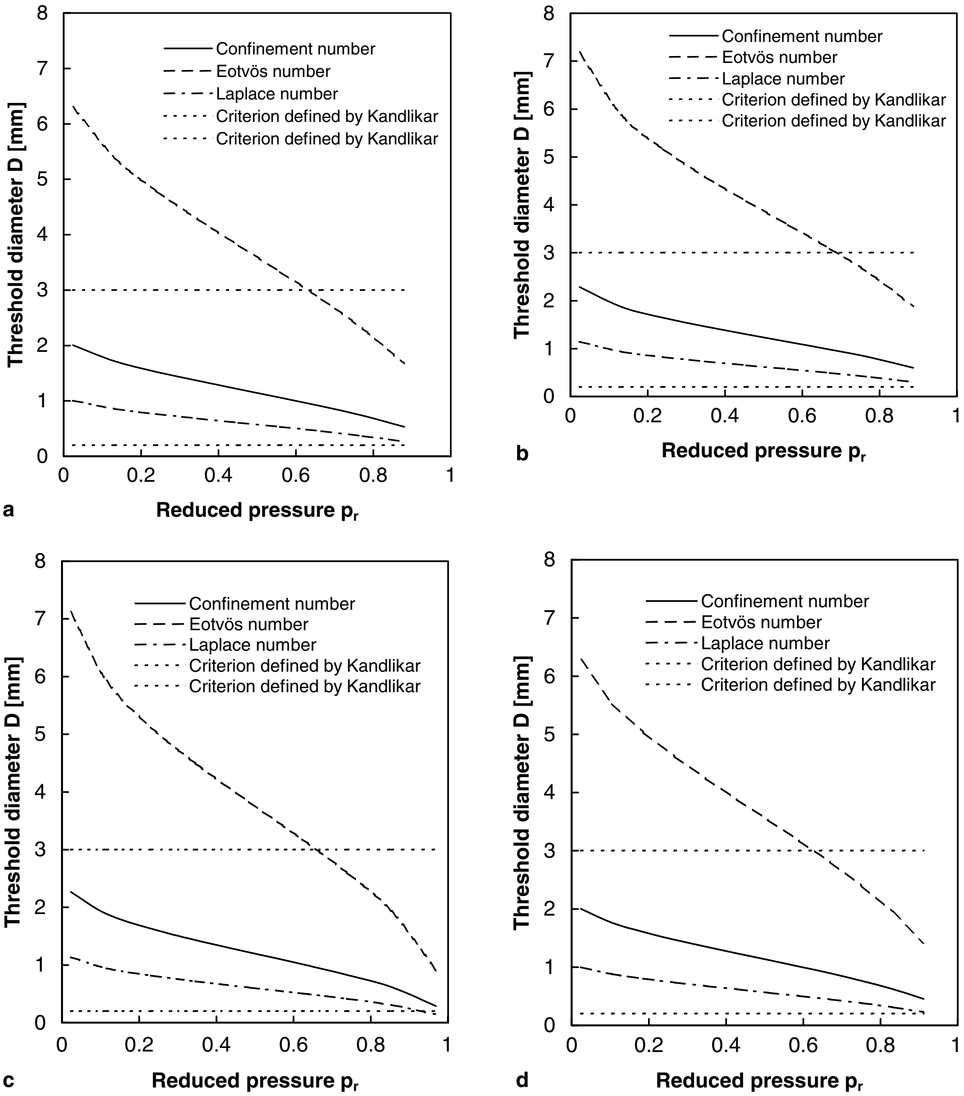

Fig. 2. Comparison of various definitions of threshold diameters for different mixtures: (a) R404a; (b) R407c; (c) R410a; (d) R507a.

Over the past decade, a number of investigations on two-phase flow and flow boiling characteristics in small diameter channels have been reported to explore the fundamental aspects of two-phase flow and flow boiling in small diameter channels. Ghiaasiaan and Abdel-Khalik (2001) performed an overall review of two-phase flow in micro-channels with hydraulic diameters ranging from 0.1 mm to 1 mm. They discussed the criteria to distinct small and normal size channels. They reviewed the research of two-phase flow patterns, flow boiling heat transfer, critical heat flux (CHF) and two-phase flow pressure drop in small channels. However, their review only covers the hydraulic diameters ranging from 0.1 mm to 1 mm. As discussed in the forgoing, the threshold diameters based on dimensionless numbers to distinction between small diameter and normal size channels obviously depend on the physical properties of fluids and they may be beyond a value of 1 mm. Therefore, it is important to include larger diameters when discussing flow boiling in small diameter channels because the definitions of small diameter channels are quite different. Kandlikar (2001, 2002) performed a review of flow boiling in small diameter channels as well. In his review paper, he defined the threshold diameters as fixed values as indicated in the forgoing. He tabulated the previous research papers according to author/year, the fluid and ranges of test parameters, the study contents such as heat transfer, pressure drop and flow patterns and some remarks. He reviewed an overall status of the research in this important area. Total 27 research papers are included in his review papers. Most of the previous papers on flow boiling in small diameter channels are among 1990s except that one earlier paper is in 1982. The latest research papers which he cited are up to 2001. These are typical papers on the study of flow boiling in small diameter channels although some related papers in the literature are not included in his review, for example, the research of Wolk et al. (2000), Xu et al. (1999), Zhao et al. (2000) etc., just to name a few here. Previous studies generally confirm significant differences between small diameter and normal size channels, with respect to two-phase pressure drop, flow boiling heat transfer, CHF and two-phase flow patterns (by flow visualization). The test channels include circular tubes, rectangular square tubes, parallel rectangular triangular tubes, annulus, and circular multi-channels in coils. The working fluids include water, air/water, refrigerants such as R113, R11, R141b, R124, R12, R318C, R134a, R22, FC84, CO_2 etc., and ethanol and water binary mixture. Among these papers reviewed by Kandlikar, there is only one paper which deals with flow boiling of mixture by Peng and Wang (1998). Although it is reported that they experimentally studied flow boiling of ethanol and water binary mixture in their paper, they used the method for pure fluid to reduce their experimental data and did not consider the effect of component composition on the flow boiling heat transfer characteristics (more review about this paper is presented in Section 4). Recently, Hetsroni et al. (2005) summarized the study of fluid flow in micro-channels with hydrodynamic diameter ranging from 1.01 μm to 4010 μm in the literature. They compared the predictions of the conventional theory and experimental data obtained during the last decade and discussed the possible sources of unexpected effects which were revealed by a number of previous investigations. Nearly all the researchers focused on the experimental investigations of two-phase flow and flow boiling with pure fluids in single small channels in the first stage of this new promising research area. There is little research on two-phase flow and flow boiling of mixtures in small and mini channels in the literature. The reason is possibly because two-phase flow and flow boiling phenomena of mixtures are much more complex than that of pure fluids. In recent years, the research of two-phase flow and flow boiling in small and mini channels is increasingly growing as indicated by a large amount of publications in the literature. Extensive studies of two-phase flow and flow boiling in small and mini channels are increasing fast due to the very important applications of this area in various aspects as indicated in numerous research papers (e.g. Zhao and Bi, 2001; Yang, 2001; Chen et al., 2002; Kawahara et al., 2002; Yarin et al., 2002; Yu et al., 2002a,b; Smith et al., 2003; Bergles et al., 2003; Rupani et al., 2003; Yun and Kim, 2003; Watel, 2003; Zhang et al., 2004; Steinke and Kandlikar, 2004; Warrier and Dhir, 2004; Molki et al., 2004; Kandlikar, 2004; Brutin and Tadrist, 2004; Jacob and Thome, 2002; Cheng et al., submitted for publication; Hetsroni et al., 2004, 2005), just to list a few here. The research on two-phase flow and flow boiling in small diameter channels is not only limited to experimental work but it is also extended to extensive exploration of the mechanisms of flow boiling in small diameter channels, flow boiling heat transfer modeling, heat transfer and pressure drop correlations, compact and micro heat exchangers including multi-channels, two-phase flow instability in small channels and so on. For example, Zhang et al. (2004) studied the correlations for flow boiling heat transfer for mini channels. They found that liquid-laminar and gas turbulent flow are common feature in many applications of mini channels. They modified traditional Chen correlation for flow boiling heat transfer in mini channels. Kandlikar (2004) studied the flow boiling heat transfer mechanisms in micro-channels. He analyzed the forces acting on the liquid–vapor interface during flow boiling in micro-channels and concluded that heat transfer during flow boiling in micro-channels seemed to be nucleate boiling dominant, which has been confirmed by his experimental observation. Brutin and Tadrist (2004) studied two-phase flow instabilities in mini channels. Jacob and Thome (2002) proposed a heat transfer model for evaporation of elongated bubble flows in micro-channels. Cheng et al. (submitted for publication) recently proposed a new flow pattern based flow boiling heat transfer model for natural refrigerant CO_2, which includes both small diameter and normal size channels. Watel (2003) extensively reviewed flow boiling in small passages of compact heat exchangers which are multichannel passages with hydraulic diameters less than about 10 mm. The review includes small circular and rectangular channels including studies of individual channels and multichannel arrangements of parallel channels and multichannel arrangements of straight perforated and serrated fin passages. His paper concerns two-phase flow regimes, local heat transfer coefficient in small passages of compact heat exchangers and heat transfer correlations. In all, the research of two-phase flow and flow boiling in small and mini channels is going into extensive aspects. However, as one of the important aspects, there is still very little information of two-phase flow and flow boiling of mixtures in small and mini channels in the literature although there are relatively a large number of papers related to two-phase flow and flow boiling of mixtures in normal size channels (as described in the following section of the review on this aspect). Previously, the studies of two-phase flow and flow boiling of mixtures in small and mini channels were mainly related to the performance of capillary tubes which are used as throttle in air conditioning, refrigeration and heat pump systems. With the rapid development of micro-electro-mechanical systems (MEMS), the development of compact and micro evaporators for mixture fluids is becoming essential. For example, potential development of micro heat pumps (Munkejord et al., 2002) using newly friendly refrigerant mixtures requires to understand two-phase flow and flow boiling phenomena of mixtures in small and mini channels. Other new technologies such as micro energy systems, micro chemical and bio-medical systems etc also require to understand the fundamental aspects of two-phase flow and flow boiling of mixtures in small and mini channels. Therefore, it is of significance to study two-phase flow and flow boiling phenomena of mixtures in small and mini channels. It should be mentioned here that there are a lot of studies of fluid flow and boiling heat transfer of fluids with the addition of surfactants (e.g. Hetsroni et al., 2004; Klein et al., 2005) in recent years. The addition of surfactants in fluids can enhance nucleate boiling heat transfer and reduce flow drag. In this review paper, the study of two-phase flow and flow boiling of aqueous surfactant solutions is not included because it is a kind of heat transfer enhancement and drag reduction technology with very small amount addition (only at ppm grade).

The objectives of the present paper are to review the state-of-the-art study of two-phase flow and flow boiling of mixtures in small and mini channels and to identify the future research directions in this important area. In the following sections, several related aspects are addressed. The importance of the study of two-phase flow and flow boiling of mixtures is described at first. As a basis of the study of flow boiling of mixtures in small and mini channels, the studies of flow boiling of mixtures in normal size channels are then summarized. Finally, the current research status of flow boiling of mixtures in small and mini channels is presented. Based on the review of this important area, the future research directions of two-phase and flow boiling of mixtures in small and mini channels are indicated.

2. Background of study on two-phase flow and flow boiling of mixtures

Many researchers used refrigerants as working fluids in their studies of flow boiling in small and mini channels as summarized by Kandlikar (2001, 2002). They are mostly chlorofluorocarbons (CFCs) and halogenated carbon-based compounds (HCFCs). However, the impact that these refrigerants have on environment has been ignored in most of these studies. As CFCs and HCFCs are known to provide a principal cause to ozone depletion and global warming, production and use of these refrigerants have been restricted. Since the advent of the Montreal Protocol, much work has been done in an attempt to find replacements for CFCs and HCFCs (Wang, 2001). This includes studies on pure refrigerants as well as refrigerant mixtures. Several potential substitutes to CFCs and HCFCs as refrigerants are mixtures of fluids because single-component fluids have not been found for many CFCs and HCFCs. For example, no pure substance may replace R22 and refrigerant mixtures have been developed to replace R22. Refrigerant mixtures having zero ozone depletion potential (ODP) and low global warming potential (GWP) have been suggested as drop-in or mid-term replacement. Although it is helpful to explore the mechanisms of two-phase flow and flow boiling in small and mini channels using working fluids such as R113, R11, R12, R22 etc., it is of no practical use because of the phaseout of these refrigerants. Therefore, it is of great significance to select accepted refrigerants as working fluids in studying two-phase flow and flow boiling phenomena. The accepted refrigerants include chlorine-free substances such as pure hydrofluorocarbons (HFCs), refrigerant mixtures and natural refrigerants. As this paper focuses on the review of two-phase flow and flow boiling of mixtures in small and mini channels, refrigerant mixtures and other mixture fluids in normal size channels are discussed in the next section to provide a basis for the study of two-phase flow and flow boiling in small and mini channels. According to the international agreement about reducing and banning the use of CFCs and HCFCs, the refrigerant mixtures are classified as three different categories (Vorster and Meyer, 2000) as shown in Table 2. Most of these accepted refrigerant mixtures show more or less special behavior when they evaporate. Therefore, it is important to study the two-phase flow and flow boiling characteristics of these refrigerant mixtures so as to design or retrofit heat pumps and refrigeration systems. In addition, it is also an important aspect to study two-phase flow and flow boiling

Table 2
Classification of refrigerant mixtures (Vorster and Meyer, 2000)

Phase-out refrigerant mixtures	Temporarily acceptable refrigerant mixtures	Acceptable refrigerant mixtures
R11/R114	R22/R124	R32/R134a
R12/R22	R22/R134a	R32/R143
R12/R152a	R22/R142b	R32/R152a
R13B1/R152a	R22/R152a	R125/R143
R22/R114	R32/R124	R125/RC318
	R123/R134a	R134a/R143
	R124/R125	R134a/143a
	R124/R143a	R143/R143a
	R125/R142b	R218/RC318
	R142b/R143a	R290/R600

of refrigerant and lubrication oil mixtures which appear in the practical air-conditioning, heat pumps and refrigeration systems.

Refrigerant mixtures can be divided into two main groups, i.e. azeotropic and non-azeotropic mixtures (including near azeotropic mixtures) (Wang, 2001). Azeotropic mixtures behave as pure substances at phase change. When they boil or condense, the temperature will be constant during the whole change of phase if the pressure is kept constant. Fig. 3 shows a schematic phase equilibrium diagram for azeotropic mixture. Azeotropic point is indicated by the arrow. T_b and T_d are respectively bubble point and dew point temperatures. In non-azeotropic mixtures, boiling or condensation occurs over a temperature range. Fig. 4 shows the variation of bubble and dew point temperatures of a typical binary non-azeotropic mixture as a function of mixture composition. The difference between the dew point and the bubble point temperatures is known as the temperature glide. In refrigeration systems, the temperature glide can vary between 2 and 20 K depending on the type of mixtures. Temperature glide in mixtures can be used beneficially in reducing the entropy generation in heat exchangers by matching the glides of the refrigerant with that of the heat transfer fluid. Thus, the cycle efficiency may be improved (Vorster and Meyer, 2000; Kruse, 1981; Didion and Bivens, 1990). In addition to the application of flow boiling of mixtures in air-conditioning, heat pumps and refrigeration systems, there are a number of applications of two-phase flow and flow boiling of mixtures in various industries such as chemical engineering, petroleum industry, biomedical engineering, micro-electro-mechanical systems and so on. Therefore, it is of significance to study two-phase flow and flow boiling phenomena of mixtures.

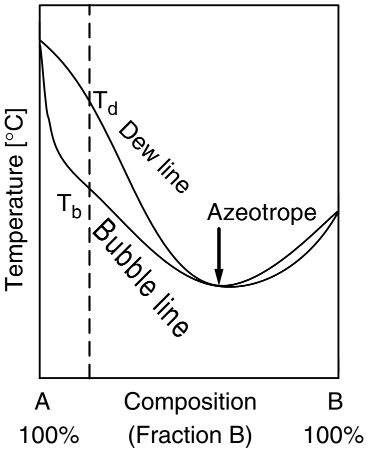

Fig. 3. Phase equilibrium diagram for azeotropic mixtures (A and B components).

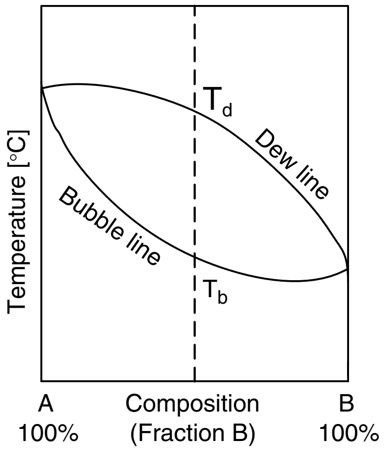

Fig. 4. Phase equilibrium diagram for non-azeotropic mixtures (A and B components).

3. Two-phase flow and flow boiling of mixtures in normal size channels

As a basis for the research of two-phase flow and flow boiling of mixtures in small and mini channels, it is necessary to review the research of two-phase flow and flow boiling of mixtures in normal size channels at first. 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 normal size channels. These mainly include experiments on flow boiling of refrigerant mixtures and development of heat transfer correlations for mixtures. Stephan (1995) gave a general analysis on boiling of mixtures in forced flow as one section in his review paper. He analyzed the heat transfer mechanisms of flow boiling of mixtures and explained the reason why the heat transfer coefficients of mixtures are lower than that of pure fluids. He also pointed out that a flow boiling heat transfer model based on an asymptotic addition of the two boiling components presented by Steiner and Taborek (1992) can be applied to mixtures and has good accuracy. Thome (1996) reviewed the research of boiling of new refrigerants including both pool boiling and flow boiling. He addressed several key points on flow boiling of mixtures such as the experimental data reduction procedures for mixtures, the calculation of isobaric change in enthalpy of mixture during evaporation along a tube, heat transfer mechanism of flow boiling of mixture by analyzing the available experimental results and the prediction of the non-azeotropic flow boiling heat transfer coefficient. He suggested that the recommended approach for non-azeotropic mixtures is to utilize an accurate, general type of pure refrigerant correlation that is modified to include the mixture mass diffusion effect. The effect of non-linear variation in mixture physical properties is easily included by using local mixture properties in evaluating the pure refrigerant as he showed several correlations such as Gungor and Winterton (1986, 1987) and compared the model with the experimental data. In addition, he addressed the issue of refrigerant-lubricating oil mixtures and reviewed the previous study of flow boiling of refrigerant-lubricating oil mixtures. Kandlikar (1998b) performed a historical review on modeling flow boiling of binary mixtures in his research paper. He presented a summary of several important available correlations and his previous correlation as a basis for developing a new correlation for binary mixtures.

In this section, the research status of two-phase flow and flow boiling of mixtures in normal size channels is described. Several typical and new studies in the literature are presented.

3.1. An overview of experimental study of two-phase flow and flow boiling of mixtures in normal size channels

Flow boiling heat transfer of mixtures is degraded as compared with pure components because the effect of mass diffusion (Stephan, 1995; Steiner and Taborek, 1992). This has been confirmed by an extensive of experimental results in the literature. Radermacher et al. (1983) conducted experiments of flow boiling with a R13b1/R152a mixture in a horizontal stainless steel tube (2.7 m long, 9 mm inside diameter). It was observed that the local and average heat transfer coefficients for the mixture were significantly lower than those for either pure component. As a follow onto this work, Ross et al. (1987) conducted a series of experiments with an R13b1/R152a mixture having composition of 0.07, 0.22, 0.36 and 0.64 mole fraction R13b1. Their results also showed a substantial degradation of heat transfer coefficients with R13b1/R152a mixture. In addition, Ross et al. reported, for the first time, a circumferential variation of wall temperatures with mixtures in an annular flow regime. In annular flow, heat is conducted across the liquid layer and especially in a horizontal geometry, the liquid film at the bottom is thicker than that at the top due to gravity. Consequently, for pure components the wall temperature at the bottom of the tube was higher than that at the top because of the increased resistance to heat transfer conduction. This results in a lower heat transfer coefficient at the bottom rather than at the top of the heated tube. For mixtures, however, a new behavior was observed: the wall temperature at the bottom was lower than the one at the top. Ross et al. conjectured that the cause for this phenomenon might be a composition variation around the circumference of the heated tube. They also indicated that heat transfer coefficients for mixtures were more or less constant over the composition range of 0.1–0.64 mole fraction of R13b1. Of the numerous studies of flow boiling with mixtures, one very good study of flow boiling of mixture is the work by Jung et al. (1989a), which has been cited as a typical example by many researchers. They studied flow boiling heat transfer characteristics of pure refrigerants and a non-azeotropic refrigerant mixture in horizontal stainless steel tube (4 m long, 9 mm inside diameter). A series of tests were carried out for pure and mixed refrigerants of R22 and R114 at several compositions. Their results indicated a full suppression of nucleate boiling for pure and mixed refrigerants beyond transition qualities and the majority of the data belonged to the convective evaporation region. The heat transfer coefficients of mixtures in this region are as much as 36% lower than the ideal values under the same flow condition as shown in Fig. 5. A composition variation of up to 0.07 mole fraction in the annular liquid film were measured between the top and bottom of the tube, which caused a corresponding circumferential variation of wall temperature with mixtures. The other part of the variation in heat transfer coefficients is caused by nonlinear variation in physical properties with composition and temperature. In addition, it is also possible for the type of flow pattern to change with composition, although that is not apparent unless flow regimes are observed experimentally. Shin et al. (1997) studied flow boiling of pure refrigerants and refrigerant mixtures in a horizontal stainless steel tube (5.9 m long, 7.7 mm inside diameter). The refrigerant mixtures include R32/R134a, R290/R600a (non-azeotropic refrigerant mixtures) and R32/R125 (azeotropic refrigerant mixture). They are mixtures with zero ozone depletion and are used to replace CHCs and HCFCs. They concluded that heat transfer coefficients depend strongly on heat flux at low quality region and become independent as quality increases. Boissieux et al. (2000) studied experimentally flow boiling of three refrigerant mixtures including R404a, R407c and

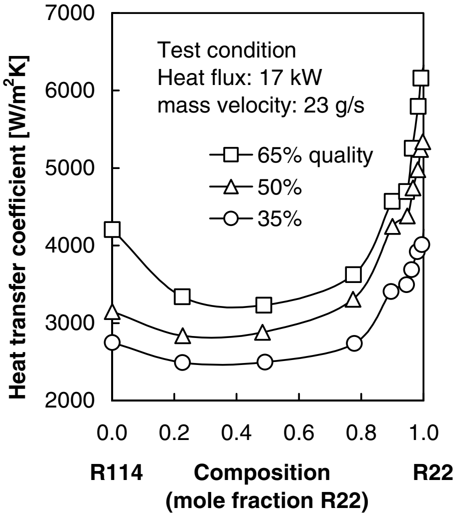

Fig. 5. Flow boiling heat transfer coefficient of binary mixture R22/R114 mixture as a function of composition at various qualities (Jung et al., 1989a).

Isceon 59 in horizontal normal size tube. The local heat transfer coefficients were measured and compared with the available correlations. They found that big discrepancies have been produced. They proposed a modified heat transfer correlation based on their own data. Actually, there are a lot of studies on flow boiling of refrigerant mixtures including both binary and ternary mixtures in the literature (Singal et al., 1983; Murata and Hashizume, 1988, 1990, 1993; Sami et al., 1993; McGillis and Carey, 1993; Celata et al., 1993, 1994; Sivagnanam et al., 1994; Wenzel et al., 1994; Niederkrüger and Steiner, 1994; Kedrierski and Kaul, 1998; Wettermann and Steiner, 2000; Meyer et al., 2000; Sami and Comeau, 2002, 2004; Sami and Aucoin, 2003; Zhang et al., 1997; Sami and Song, 1997; Barbosa et al., 2002). These studies generally include experimental work on flow boiling heat transfer, two-phase flow pressure drop, subcooled flow boiling in tubes and in heat exchangers. In addition, the study of flow boiling of other mixtures such as water/ammonia, ammonia/lithium nitrate, water/lithium bromide, mixtures of ethylene glycol and water, and binary mixed magnetic mixtures were also performed by some researchers (Rivera and Best, 1999; Ruvera et al., 2003; Kandlikar and Bulut, 2003; Shuchi et al., 2002; Shuchi et al., 2004). As the present paper aims to review the study of two-phase flow and flow boiling of mixtures in small and mini channels, here just to list a few papers in this section to show the generalized research status of flow boiling with mixtures in normal size channels.

3.2. Models of two-phase flow and flow boiling of mixtures in normal size channels

Boiling of mixtures differs substantially from that of pure fluids due to a number of factors such as the effect of composition on nucleation (Shock, 1977), significant change in physical properties of mixtures with composition (Stephan and Preusser, 1979) and the retardation of vapor-liquid exchange and evaporative mechanisms (Thome, 1982). The heat transfer coefficient of mixtures is generally lower than that of the equivalent pure fluid with the same physical properties as shown in the foregoing. Some limited work has been reported on the mechanisms and attended to model the processes. The empirical evaluation of the heat transfer coefficient in forced convective boiling has been performed by different authors using modified heat transfer correlations, derived from the original ones developed for pure components. Bennett and Chen (1980) proposed a correlation to binary mixtures, which is based on the Chen (1966) correlation, as

$$h_{\mathrm{mix}} = h_{\mathrm{mic}} S_{\mathrm{mix}} + h_{\mathrm{mac}} F_{\mathrm{mix}} \tag{4}$$

where h_{mix} is the flow boiling heat transfer coefficient for mixtures, and microscopic heat transfer coefficient h_{mic}, i.e. the nucleate boiling contribution to the heat transfer is as follows:

$$h_{\mathrm{mic}} = 0.00122\left[\frac{k_{\mathrm{l}}^{0.79} c_{\mathrm{pl}}^{0.45} \rho_{\mathrm{l}}^{0.49}}{\sigma^{0.5} \mu_{\mathrm{l}}^{0.29} h_{\mathrm{fg}}^{0.24} \rho_{\mathrm{v}}^{0.24}}\right] [T_{\mathrm{w}} - T_{\mathrm{b}}]^{0.24} [p_{\mathrm{sat}}(T_{\mathrm{w}}) - p_{\mathrm{sat}}(T_{\mathrm{b}})]^{0.75} \tag{5}$$

where k_{l} is liquid thermal conductivity, c_{pl} is liquid specific heat at constant pressure, μ_{l} is liquid dynamic viscosity, h_{fg} is latent heat of vaporization, T_{w} is wall temperature, T_{b} is bubble point temperature, $p_{\mathrm{sat}}(T_{\mathrm{w}})$ is saturation temperature at T_{w} and $p_{\mathrm{sat}}(T_{\mathrm{b}})$ is saturation temperature at T_{b}. The nucleate suppression factor is as

$$S_{\mathrm{mix}} = S\left[1 - \frac{c_{\mathrm{pl}}(Y_{\mathrm{m}} - X_{\mathrm{m}})}{h_{\mathrm{fg}}}\left(\frac{\mathrm{d}T_{\mathrm{b}}}{\mathrm{d}X_{\mathrm{m}}}\right)_{\mathrm{pbulk}}\left(\frac{\alpha}{D}\right)^{1/2}\right]^{-1} \tag{6}$$

where Y_{m} is local vapor mass fraction of the less volatile component, X_{m} is local liquid mass fraction of the less volatile component, α is thermal diffusivity, D is mass diffusivity, the nucleate boiling suppression factor S is as

$$S = \frac{1}{1 + 2.53 \times 10^{-6} Re_{\mathrm{tp}}^{1.17}} \tag{7}$$

where two-phase Reynolds number is as

$$Re_{\mathrm{tp}} = \frac{G D_{\mathrm{in}}(1 - x)}{\mu_{\mathrm{l}}}\left[F\left(\frac{Pr_{\mathrm{l}} + 1}{2}\right)^{0.444}\right]^{1.25} \tag{8}$$

where G is mass flux, D_{in} is channel inner diameter, x is quality, Pr_{l} is liquid Prandtl number, F is nucleate boiling factor.

$$F = 1 \quad \text{if } \frac{1}{X_{tt}} \leqslant 0.1 \tag{9}$$

$$F = 2.35\left[\frac{1}{X_{tt}} + 0.213\right]^{0.736} \quad \text{if } \frac{1}{X_{tt}} > 0.1 \tag{10}$$

where Martinelli number X_{tt} is as

$$X_{tt} = \left(\frac{\rho_{\mathrm{v}}}{\rho_{\mathrm{l}}}\right)^{0.5}\left(\frac{\mu_{\mathrm{l}}}{\mu_{\mathrm{v}}}\right)^{0.1}\left(\frac{1-x}{x}\right)^{0.9} \tag{11}$$

where μ_{v} is vapor dynamic viscosity.

The macroscopic heat transfer coefficient h_{mac}, i.e. the convective boiling contribution to the heat transfer is as follows:

$$h_{\mathrm{mac}} = h_{\mathrm{l}} = 0.023 Re_{\mathrm{l}}^{0.8} Pr_{\mathrm{l}}^{0.4} \frac{k_{\mathrm{l}}}{D_{\mathrm{in}}} \tag{12}$$

where liquid phase Reynolds number Re_{l} and liquid phase Prandtl number Pr_{l} are as

$$Re_{\mathrm{l}} = \frac{GD_{\mathrm{in}}(1-x)}{\mu_{\mathrm{l}}} \tag{13}$$

$$Pr_{\mathrm{l}} = \frac{\mu_{\mathrm{l}} c_{\mathrm{pl}}}{k_{\mathrm{l}}} \tag{14}$$

The nucleate boiling factor for mixture is as

$$F_{\mathrm{mix}} = F\left(\frac{Pr_{\mathrm{l}}+1}{2}\right)^{0.444}\left[\frac{\Delta T}{\Delta T_{\mathrm{s}}}\right]_{\mathrm{mac}} \tag{15}$$

$$\left[\frac{\Delta T}{\Delta T_{\mathrm{s}}}\right]_{\mathrm{mac}} = 1 - \frac{(1-Y_{\mathrm{m}})q}{\rho_{\mathrm{l}} h_{\mathrm{fg}} h_{\mathrm{m}} \Delta T_{\mathrm{s}}}\left(\frac{\mathrm{d}T_{\mathrm{b}}}{\mathrm{d}X_{\mathrm{m}}}\right)_{\mathrm{pbulk}} \tag{16}$$

where q is heat flux, ΔT_{s} is superheat temperature, h_{m} is mass transfer coefficient.

$$h_{\mathrm{m}} = 0.023\left(\frac{D}{D_{\mathrm{in}}}\right) Re_{\mathrm{tp}}^{0.8} Sc^{0.4} \tag{17}$$

where Sc is Schmidt number as

$$Sc = \frac{\mu}{\rho D} \tag{18}$$

In theoretical models developed by Sardesai et al. (1982), not only does the mixture affect nucleate boiling (this is recognized as an effect of diffusive resistance on nucleate boiling), but the thermal resistance in the vapor phase also causes a degradation in heat transfer coefficient. However, the effect of diffusive resistance on two-phase forced convection is neglected. According to the study using mixtures of refrigerants R11 and R114 by Murata and Hashizume (1988, 1990), the heat transfer coefficient of mixtures is significantly lower than that of pure fluid in the boiling-dominant region, while it is almost equal in the convection-dominant region. Jung et al. (1989a,b) reached the same conclusion through their study with mixtures of R22/R114 and R12/R125a. This is because the effect of diffusive resistance is small in the convection-dominant region.

Mishra et al. (1981) correlated their experimental data with the following equation for mixture heat transfer coefficient h_{mix}:

$$h_{\mathrm{mix}} = C h_{\mathrm{l}}\left(\frac{1}{X_{tt}}\right)^{m} Bo^{n} \tag{19}$$

where boiling number Bo is as

$$Bo = \frac{q}{Gh_{\mathrm{fg}}} \tag{20}$$

and h_{l} is calculated with Eq. (12). The coefficient values for two different mixture composition are as: (a) R12 (23–27%) and R22 (77–73%): $C = 5.62$, $m = 0.23$, $n = 0.5$; (b) R12 (41–48%) and R22 (59–52%): $C = 21.75$, $m = 0.29$, $n = 0.23$. This correlation shows a mean deviation of 30% on the data of composition type (a) while a better performance is obtained if used on the data of composition type (b).

Jung et al. (1989b) found that Chen correlation was unable to correlate the heat transfer data for refrigerant mixtures. Therefore, they developed the following correlation for flow boiling inside horizontal tubes for pure and mixed refrigerants:

$$h_{\mathrm{mix}} = \frac{S}{C_{\mathrm{UN}}} h_{\mathrm{UN}} + C_{me} F h_{\mathrm{l}} \tag{21}$$

which represents a modification of the Chen correlation. The heat transfer enhancement factor F and the suppression factor S are defined as follows:

$$F = 2.37 \left[0.29 + \frac{1}{X_{tt}} \right]^{0.85} \tag{22}$$

$$S = 4048 X_{tt}^{1.22} Bo^{1.13} \quad \text{if } X_{tt} < 1 \tag{23}$$

h_{UN} is the pool boiling heat transfer coefficient, developed by Unal (1986) as

$$h_{\mathrm{UN}} = \frac{1}{C_{\mathrm{UN}}} \frac{h_1 h_2}{h_1 X_1 + h_2 X_2} \tag{24}$$

where h_1 and h_2 are nucleate pool boiling heat transfer coefficients of component 1 and 2 calculated by Stephan and Abdelsalam's (1980) pool boiling correlation, as

$$h_{\mathrm{SA}} = 207 \frac{k_{\mathrm{l}}}{d} \left(\frac{q d_e}{k_{\mathrm{l}} T_{\mathrm{sat}}} \right)^{0.745} \left(\frac{\rho_{\mathrm{v}}}{\rho_{\mathrm{l}}} \right)^{0.581} Pr_{\mathrm{l}}^{0.533} \tag{25}$$

where d_{e} is equilibrium break-off-diameter.

$$d_e = 0.0146 \beta [2\sigma / (g(\rho_{\mathrm{l}} - \rho_{\mathrm{v}}))]^{0.5} \tag{26}$$

where the contact angle $\beta = 35°$.

C_{UN} and C_{me} are correction factors for mixtures in pool boiling and evaporation respectively. C_{UN} and C_{me} are complicated functions of liquid–vapor-phase concentrations. For pure fluids, the mixture correction factors become 1. C_{UN} is defined as

$$C_{\mathrm{UN}} = [1 + (b_2 + b_3)(1 + b_4)](1 + b_5) \tag{27}$$

where

$$b_2 = (1 - X) \ln \left(\frac{1.01 - X}{1.01 - Y} \right) + X \ln \left(\frac{X}{Y} \right) + |Y - X|^{1.5} \tag{28}$$

where X and Y are respectively is liquid and vapor phase compositions based on mole.

$$b_3 = 0 \quad \text{if } X \geqslant 0.01 \tag{29}$$

$$b_3 = (Y/X)^{0.1} - 1 \quad \text{if } X < 0.01 \tag{30}$$

$$b_4 = 152 (p/p_{\mathrm{cmvc}})^{3.9} \tag{31}$$

where p is pressure, p_{cmvc} is critical pressure of the more volatile component.

$$b_5 = 0.92 |Y - X|^{0.001} (p/p_{\mathrm{cmvc}})^{0.66} \tag{32}$$

$$X/Y = 1 \quad \text{if } X = Y = 0 \tag{33}$$

C_{me} is a correction factor which considers mass transfer resistance in the convective evaporation region, as

$$C_{\mathrm{me}} = 1 - 0.35|Y - X|^{1.56} \tag{34}$$

For the construction of this correlation, more than 3000 data points were used for R22, R12, R152a, R114 and their mixtures. The correlation was compared with experimental data for R12/R152a and R22/R114 mixtures at various concentrations and for a wide range of operating conditions. The mean deviation was 9.6% for the mixture data examined.

Kandlikar (1998b) developed a flow boiling model for mixtures by looking at the fundamental bubble growth process and estimating the interface concentration. He used his previous correlation for pure fluids (Kandlikar, 1990) as the starting point and incorporated his pool boiling model (Kandlikar, 1998a) in his flow boiling model for binary mixtures. He proposed the following correlation for flow boiling of binary mixtures (The properties of the mixtures at saturation are employed in the following equations.):

A volatility parameter V_1 is used to distinguish between the regions where the diffusion effects are significant and where they are small as in azeotropic systems.

$$V_1 = \frac{c_{\mathrm{pl}}}{h_{\mathrm{fg}}}\left(\frac{\alpha}{D}\right)^{0.5}\frac{\mathrm{d}T}{\mathrm{d}X_1}(Y_1 - X_1), \tag{35}$$

where X_1 and Y_1 are respectively mass fraction of component 1 in liquid phase and in vapor phase. Subscript 1 represents more volatile component. The other symbols in Eq. (35) have the same meanings as that defined above.

Region I. Near-azeotropic region; $V_1 < 0.03$, flow boiling heat transfer coefficient for mixtures h_{mix} is calculated as

$$h_{\mathrm{mix}} = \text{lager of } h_{\mathrm{NBD}} \text{ and } h_{\mathrm{CBD}} \tag{36}$$

where h_{NBD} and h_{CBD} are the nucleate boiling dominant (NBD) and the convective boiling dominant (CBD) heat transfer coefficients which are given respectively by

$$h_{\mathrm{NBD}} = 0.6683Co^{-0.2}(1 - x)^{0.8}h_{\mathrm{lo}} + 1058Bo^{0.7}(1 - x)^{0.8}F_{\mathrm{Fl}}h_{\mathrm{lo}} \tag{37}$$

$$h_{\mathrm{CBD}} = 1.136Co^{-0.9}(1 - x)^{0.8}h_{\mathrm{lo}} + 667.2Bo^{0.7}(1 - x)^{0.8}F_{\mathrm{Fl}}h_{\mathrm{lo}} \tag{38}$$

Additionally, for horizontal tubes with Froude number, Fr_{lo}, less than 0.04, a multiplier $(25Fr_{\mathrm{lo}})^{0.324}$ is applied to the first terms in Eqs. (37) and (38). For $Fr_{\mathrm{lo}} > 0.04$, and for vertical tubes, no correction is needed. This correction is usually not needed for the range of mass fluxes employed in the refrigerant evaporators. Boiling number Bo is defined by Eq. (20). Convection number Co is defined as

$$Co = \left(\frac{\rho_{\mathrm{v}}}{\rho_{\mathrm{l}}}\right)^{0.5}\left(\frac{1 - x}{x}\right)^{0.8} \tag{39}$$

Fr_{lo} is Froude number with all flow as liquid and is defined as

$$Fr_{\mathrm{lo}} = \frac{G^2}{\rho_{\mathrm{l}}^2 g D_{\mathrm{in}}} \tag{40}$$

F_{Fl} in Eqs. (37) and (38) is a fluid-surface parameter related to the nucleation characteristics. where X_1 and X_2 are mass fraction of components in liquid phase. Subscribes 1 and 2 are components of binary system and 1 is more volatile component and 2 is less volatile component. The fluid-surface parameter is obtained as the mass fraction-averaged value given by the following equation:

$$F_{\mathrm{Fl}} = X_1F_{\mathrm{Fl,1}} + X_2F_{\mathrm{Fl,2}} \tag{41}$$

Table 3 lists the values of F_{Fl} for different refrigerants flowing in copper or brass tubes. F_{Fl} for all liquids in stainless steel tubes is 1.0. The single-phase heat transfer coefficient, h_{lo}, is obtained from the Petukhov and Popov (1963), and Gnielinski (1976) correlations.

Table 3
Fluid-surface parameter F_{Fl} for refrigerants in copper or brass tubes (Kandlikar, 1998a,b)

Fluid	F_{Fl}
Water	1.0
R11	1.3
R12	1.5
R13B1	1.31
R22	2.2
R113	1.3
R114	1.24
R124	1.9
R134a	1.63
R152a	1.1

For all fluids in stainless steel tubes, $F_{\mathrm{Fl}} = 1.0$.

Petukhov and Popov (1963) for $0.5 \leqslant Pr_{\mathrm{l}} \leqslant 2000$ and $10^4 \leqslant Re_{\mathrm{lo}} \leqslant 5 \times 10^6$

$$Nu_{\mathrm{lo}} = \frac{h_{\mathrm{lo}} D_{\mathrm{in}}}{k_{\mathrm{l}}} = Re_{\mathrm{lo}} Pr_{\mathrm{l}} (f/2) \Big/ \left[1.07 + 12.7(Pr_{\mathrm{l}}^{2/3} - 1)(f/2)^{0.5}\right] \tag{42}$$

Gnielinski (1976) for $0.5 \leqslant Pr_{\mathrm{L}} \leqslant 2000$ and $2300 \leqslant Re_{\mathrm{lo}} \leqslant 10^4$

$$Nu_{\mathrm{lo}} = \frac{h_{\mathrm{lo}} D_{\mathrm{in}}}{k_{\mathrm{l}}} = (Re_{\mathrm{lo}} - 1000) Pr_{\mathrm{l}} (f/2)/[1.0 + 12.7(Pr_{\mathrm{l}}^{2/3} - 1)(f/2)^{0.5}] \tag{43}$$

where Pr_{l} is liquid Prandtl number which is defined by Eq. (14), Re_{lo} is Reynolds number with all flow as liquid which is defined as

$$Re_{\mathrm{lo}} = \frac{G D_{\mathrm{in}}}{\mu_{\mathrm{l}}} \tag{44}$$

Nu_{lo} is Nusselt number with all flow as liquid. The friction factor in Eqs. (42) and (43) is given by

$$f = [1.58 \ln(Re_{\mathrm{lo}}) - 3.28]^{-2} \tag{45}$$

The symbols not defined here have the same meanings of that in the forgoing. The first terms in Eqs. (37) and (38) for the NBD (nucleate boiling dominant region) and CBD (convection boiling dominant region) regions, respectively, represent the convective components, while the second terms, which include the heat flux, represent the nucleate boiling component. The demarcation between the NBD and CBD regions is made automatically by comparing heat transfer coefficients of mixtures h_{mix} predicted by Eqs. (37) and (38), respectively, and taking the larger of the two as indicated by Eq. (36).The near-azeotrope region covers azeotropes and low-volatility difference mixtures.

Region II. Moderate diffusion-induced suppression region, $0.03 < V_1 < 0.2$, and $Bo > 10^{-4}$,

$$h_{\mathrm{mix}} = h_{\mathrm{CBD}} = 1.136 Co^{-0.9} (1-x)^{0.8} h_{\mathrm{lo}} + 667.2 Bo^{0.7} (1-x)^{0.8} F_{\mathrm{Fl}} h_{\mathrm{lo}} \tag{46}$$

F_{Fl} for mixtures is obtained from Eq. (41). In the moderate diffusion-induced suppression region, the nucleation effects are suppressed due to the mass diffusion resistance, and the convective heat transfer becomes dominant. In the CBD region, the bubble growth is primarily limited to the early stages in the growth cycle. The correlation for the CBD region without any suppression factor is therefore able to predict this region well.

Region III. Severe diffusion-induced suppression region, (a) for $0.03 < V_1 < 0.2$ and $Bo > 10^{-4}$ and (b) $V_1 > 0.2$,

$$h_{\mathrm{mix}} = 1.136 Co^{-0.9} (1-x)^{0.8} h_{\mathrm{lo}} + 667.2 Bo^{0.7} (1-x)^{0.8} F_{\mathrm{Fl}} h_{\mathrm{l\ddot{o}}} F_{\mathrm{D}} \tag{47}$$

This region covers the two ranges as indicated in (a) and (b) above. F_{D} is obtained from

$$F_{\mathrm{D}} = 0.678[1 + (c_{\mathrm{pl}}/h_{\mathrm{fg}})(\alpha/D)^{1/2} |(Y_1 - X_1)(\mathrm{d}T/\mathrm{d}X_1)|]^{-1} \tag{48}$$

where dT/dX_1 is the slope of the bubble point temperature versus X_1 curve. The severe diffusion-induced suppressed region is dominated by the convective effects. The nucleate boiling contribution in this region is further reduced due to the large difference in composition between the two phases, and the resulting mass diffusion resistance at the liquid–vapor interface of the growing bubble.

The Kandlikar correlation is able to correlate over 2500 data points within 8.3–13.3% mean deviation for each data set. The $h - x$ trend is represented well for R12/R22, R22/R114, R22/R152a and R132a/R123 systems. Electrically heated stainless steel test sections as well as fluid heated copper test sections are both covered under this correlation.

Shin et al. (1997) compared their results with the Thome–Shakil correlation (Thome, 1994) for refrigerant mixtures and found that the correlation overpredicted the heat transfer coefficients measured in their study.

One often limitation of past research is the fact that most of the flow boiling correlations for horizontal flow were obtained using vertical tube with some correction to account for flow stratification at low flow rates. In vertical upflow, dryout in annular flow tends to occur at vapor qualities in the range from 50% to 75% and hence, few test data are taken above this threshold. Consequently, these vertical tube results are not particularly suitable for predicting local coefficients in horizontal tubes where complete evaporation of the fluid has to be modeled. Kattan et al. (1998a,b,c) proposed a new heat transfer model for flow boiling in horizontal plain tubes that incorporates the effects of local two-phase flow patterns, flow stratification, and partial dryout in annular flow. It is reported that the new method accurately predicts a large, new database of flow boiling data in large size passages, and is particularly better than existing methods at high vapor qualities ($x > 85\%$) and for stratified types of flows.

Compared with that of flow boiling of pure fluids, the models of flow boiling of mixtures are relatively less. Most researchers proposed the heat transfer correlations based on their own experimental data and modified the available correlations for pure fluids. There is no general heat transfer model for flow boiling heat transfer of mixtures.

In addition to the models of flow boiling heat transfer, other aspects such as two-phase flow pressure drop of mixtures and critical heat flux (CHF) have been studied as well. Cavallini et al. (1997) studied pressure drop during vaporization of refrigerants including zeotropic and nearly-azeotropic refrigerant mixtures such as R407a, R407c, R404a and R410a inside enhanced tubes. Two traditional models for adiabatic pressure losses inside smooth tubes were adapted for condensation and boiling inside enhanced tubes. Ould Didi et al. (2002) compared an extensive new two-phase pressure drop of refrigerants including mixtures (R402a and R404a) with the most quoted pressure drop prediction methods in the literature. They found that the method of Müller-Steinhagen and Heck (1986) and the method of Grönnerud (1979) consistently gave the best predictions. Celata et al. (1994) studied experimentally CHF in forced convective upflow at different compositions of binary mixture R12 and R114. The experimental results show an almost linear dependence of critical heat flux on the composition of the mixture. A comparison of their CHF data with available correlations was also performed and good agreement was obtained.

The previous experimental work and modeling of two-phase flow and flow boiling of mixtures in normal size channels have laid a solid foundation for the study of two-phase flow and flow boiling of mixtures in small and mini channels.

4. Two-phase flow and flow boiling of mixtures in small and mini channels

Two-phase flow and flow boiling of mixtures in small and mini channels have been found a wide of applications in petrochemical engineering, process engineering, air-conditioning, refrigeration and heat pump systems, cryogenic engineering, biological engineering, MEMS and so on. For example, applications of compact evaporators in hydrocarbon separation, liquefaction of gas (nitrogen, helium, natural gas) and the separation of oxygen and nitrogen, applications of micro-channels and micro heat exchangers in reactors for modification and separation of biological cells, selective membranes and liquid/gas chromatographies, just to name a few in here. One typical example of two-phase flow and flow boiling (evaporation) of mixtures is the application of capillary tubes in refrigeration, heat pump and air-conditioning systems. As shown in Fig. 6, a capillary tube is a constant area expansion device used in small refrigeration, heat pump and air-conditioning systems. It is a simple tube with an inner diameter ranging from 0.5 to 2 mm and a length ranging from 400 to 2500 mm.

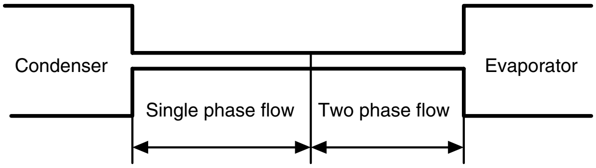

Fig. 6. Schematic diagram of a capillary tube used in air-conditioning, refrigeration and heat pump systems.

During the past decades, extensive data for adiabatic capillary tubes have been reported. However, the early investigations focused exclusively on the performance of CFC and HCFC such as R12, R22 etc. Due to their negative effect on environment, a search for alternative refrigerants became an important task for the air-conditioning, heat pump and refrigeration industry. The alternatives include a lot of refrigerant mixtures as already described in the forgoing. Consequently, the performance of capillary tubes for alternative refrigerant mixtures has become an important subject for designers. The main concern is to determine the appropriate length and diameter of the capillary tube at given refrigeration capacity and operation conditions. Therefore, it is necessary to understand the characteristics of gas–liquid two-phase flow and flow boiling (evaporation) of refrigerant mixtures in capillary tubes. In addition, due to the development of MEMS such as micro heat pumps with dimensions in the order of centimeters, a number issues have yet to be solved. These including the determination of micro heat exchanger heat transfer capacities (Munkejord et al., 2002). In this aspect, it is necessary to understand the fundamental phenomena of two-phase flow and flow boiling in small and mini channels. These generally include flow boiling heat transfer, two-phase flow patterns and two-phase pressure drop.

However, only a limited number of studies have been reported on the fundamental issues of two-phase flow and flow boiling of mixtures in small and mini channels although there are a lot of studies of two-phase flow and flow boiling of pure fluids in small and mini channels in the literature. The reason is possibly due to the complexity and difficulty of the two-phase flow and flow boiling phenomena of mixtures in small and mini channels. As one very important topic, it is necessary to know the progress of the research on two-phase flow and flow boiling of mixtures in small and mini channels and to identify the future research directions in this area. As the major objective of this paper, a state-of-the-art review of study on two-phase flow and flow boiling of mixtures in small and mini channels in the literature is presented to provide an overview of the study in this important area. Table 4 shows a summary of investigations on two-phase flow and flow boiling of mixtures in small and mini channels in the literature so far. The available papers in the literature are listed according to author/year, fluids used, channel size and position, modeling study, experimental study and some remarks. Compared with the study of two-phase flow and flow boiling of pure fluids in small and mini channels, the study of two-phase flow and flow boiling of mixtures in small and mini channels is rather rare. Most of the studies are related to capillary tubes used in refrigeration, heat pump and air conditioning systems so as to use new refrigerant mixtures as working fluids as in publications (Chang and Ro, 1996; Chung, 1998; Bittle et al., 1998; Sami and Tribes, 1998; Jung et al., 1999; Wei et al., 1999; Chang et al., 2000; Wongwises et al., 2000a,b; Wongwises and Pirompak, 2001; Chen et al., 2001; Kim et al., 2002; Motta et al., 2002; Fiorelli et al., 2002). These include two-phase flow frictional pressure drops, mass flow rates, qualities and temperatures. Both experimental and modeling studies have been performed to provide practical and theoretical basis for the design and operation of capillary tubes in refrigeration, heat pump and air-conditioning systems. However, as for flow boiling of mixtures in small and mini channels, there are very limit studies in the literature. Peng et al. (1996) and Peng and Wang (1998) studied experimentally flow boiling of water methanol binary mixture in horizontal micro channels with hydraulic diameter ranging from 0.133 to 0.343 mm. They studied the influence of liquid compositions of more volatile liquid on the heat transfer performance. They also observed the flow boiling phenomena in the micro channels and did not observe bubble phenomenon. They proposed the “boiling space” and “ficitious boiling” concepts and tried to explain flow boiling phenomena in micro-channels. It should be pointed out that the authors did not use proper microscope and high-speed video techniques to observe the boiling phenomena in their study, which resulted in contradictory conclusions as others. In addition, they applied the data reduction method for pure fluids to their experimental data for flow

Table 4
Summary of investigations on two-phase flow and flow boiling of mixtures in small and mini channels

Author/Year	Fluids	Channel size and position	Modeling study	Experimental study	Remarks
Peng et al. (1996)	Water/methanol binary mixture	Rectangular channel ($D_h = 0.133$–0.343 mm, $L = 45$ mm). Horizontal	N/A	Subcooled flow boiling heat transfer	Liquid composition has great effect on heat transfer
Peng and Wang (1998)	Water/methanol binary mixture	Rectangular channel $D_h = 0.133$–0.343 mm, $L = 50$ mm; Triangular channel $D_h = 0.2 \sim 0.6$ mm, $L = 120$ mm. Horizontal	N/A	Subcooled and nucleate flow boiling heat transfer	The channel size, geometrical configuration and mole fraction have significant impact on flow boiling, nucleate bubble formation and growth
Chang and Ro (1996)	R32/R134a, R32/R125, R125/R134a, R32/R125/R134a	Capillary tube $D_h = 1.2$ and 1.6 mm, $L = 1.5$ m	Two-phase frictional pressure drop	Two-phase pressure drop during evaporation	The homogeneous flow model is appropriate to calculate the pressure drop of two-phase flow
Chung (1998)	R407c	Capillary tube (General tube size)	Modeling of mass flow rate	N/A	A numerical procedure for the solution of two-phase Fanno flows of refrigerants in capillary tubes was proposed
Bittle et al. (1998)	R410a	Capillary tube $D_h = 0.66$–2.29 mm, $L = 508$–5080 mm	Modeling of mass flow rate	N/A	Generalized dimensionless correlations were developed for predicting refrigerant mass flow rate in an adiabatic capillary tube
Sami and Tribes (1998)	R410a, R410b, R507, R32/R134a	Capillary tube $D_h = 1.07$, 1.4, 1.5 1.9 mm	Modeling of friction pressure drop, quality and temperature	N/A	The proposed numerical model predicted well the refrigerant behaviour inside capillary tubes
Jung et al. (1999)	R407c, R410a	$D_h = 1.2$–2.4 mm	Modeling of the performance of capillary tubes	N/A	A simple practical equation for mass flow is regressed
Wei et al. (1999)	R 407c	$D_h = 0.8$–2 mm, $L = 600$–2000 mm. Straight and Coiled ($D_c = 52$ mm and 130 mm)	A correlation for predicting mass flow was proposed	Mass flow rate	The effect of coiling on the performance of capillary tubes was quantitatively investigated. A correlation was proposed to describe the relation between straight and coiled capillary tubes
Chang et al. (2000)	R-410a	Circular tube, $D_h = 5$ mm and $= 995$ mm. Horizontal	A modified Friedel correlation was proposed	Two-phase frictional pressure drop	Slight modification to the Friedel correlation was proposed that can extend its capability in the small diameter range while retaining its capability in larger diameter tube
Wongwises et al. (2000a,b)	R401a, R401b R401c, R407c R410a, R404a R507a	Capillary tubes $D_h = 1.4$, 1.5 and 1.9 mm	Modeling of two-phase pressure drop	N/A	A two-phase homogeneous flow model has been applied to determine the refrigerant flow characteristics

Wongwises and Pirompak (2001)	R404a, R507a R407b, R410a R410b	Capillary tubes $D_h = 0.77$, 1.07, 1.41 and 1.5 mm	Modeling of two-phase pressure	N/A	drop was conducted and compared with conventional refrigerants
Chen et al. (2001)	R410a	$D_h = 3.17$ mm and 5.05 mm, $L = 700$ mm	N/A	Two-phase flow pressure drop	Various correlations for two-phase flow pressure drop were compared with experimental data
Kim et al. (2002)	R407c, R410a	$D_h = 1.2$, 1.3, 1.5, 1.7 and 2 mm, $L = 500$–1500 mm. Straight and coiled ($D_c = 40$, 120 and 200 mm),	A dimensionless correlation was developed	The performances of adiabatic capillary tubes were studied	The proposed dimensionless correlation predicted mass flow rates well
Motta et al. (2002)	R404a, R404a/oil	$D_h = 0.8$ mm, $L = 1000$ mm	N/A	Flow visualization	There is a shift of the vaporization point in the capillary tube
Fiorelli et al. (2002)	R410a, R407c	$D_h = 1.067$, 1.372 and 1.626 mm, $L = 1$, 1.25 and 1.5 m	N/A	Performance of capillary tube	The main operational parameters affect in a similar way the performance of capillary tubes for both refrigerants
Hsieh and Lin (2002)	R410a	Plate heat exchanger with vertical flow channels, $D_h = 6.6$ mm	N/A	Flow boiling heat transfer, pressure drop	Both the boiling heat transfer coefficient and frictional pressure drop increase almost linearly with heat flux
Wellsandt and Vamling (2003)	R134a/R32, R124a/R125/R143a	Plate-type evaporator with vertical flow channels, $D_h = 4.2$, 5.24 and 6 mm	A simple heat transfer model was proposed	Heat transfer, pressure drop	The proposed model predicts heat transfer well
Chen et al. (2004)	R410a	Four U-type return bends with tube diameter $D_h = 3.3$ and 5.07 mm curvature ratio ranged from 3.91 to 8.15 mm	A modified two phase friction factor correlation was proposed	Two-phase pressure drops	The proposed correlation gives a good agreement to the experimental data
Greco and Vanoli (2005a,b)	R410a and R404a	Horizontal circular tube $D_h = 6$ mm, $L = 6000$ mm	A modified heat transfer coefficient correlation was proposed	Flow boiling heat transfer coefficient	Heat transfer coefficients increase with increasing saturation pressures for a constant refrigerant mass flux. With increasing pressure the nucleate boiling contribution to the heat transfer coefficient increases

boiling of binary mixture although they tried to study the effect of liquid compositions of more violate component on flow boiling heat transfer in their research. It seems that the method used in their study was impropriate. For binary mixture, the following basic equation should be used to determine the heat transfer coefficient as

$$h_{\text{mix}} = \frac{q}{T_{\text{w}} - T_{\text{b}}} \tag{49}$$

where the bubble point temperature T_{b} should be used for binary mixture. Only for a pure fluid, the bubble point temperature equals the saturation temperature. However, as pointed out by Thome (1996), many publications do not mention which definition was used. Some misused the definition of pure fluids for mixtures. In the process of evaporation of binary mixtures, as the component concentrations in the liquid and vapor phase change along the test section, the local bubble-point temperature rises as the heavy component builds up the liquid phase. Thus not only must latent heat be added to the fluid to evaporate the fluid, but also sensible heat must be added to both phases to heat them up to the new local bubble-point temperature. The isobaric change in enthalpy of mixture during evaporation along a tube is comprised of three terms expressed in the following thermodynamic relationship:

$$\mathrm{d}H = h_{\text{fg}}x + (1-x)(\mathrm{d}T_{\text{b}})c_{\text{pl}} + x(\mathrm{d}T_{\text{b}})c_{\text{pv}} \tag{50}$$

where H is enthalpy and c_{pv} is vapor specific heat at constant pressure.

Apparently, there is no such information in their studies (Peng et al., 1996; Peng and Wang, 1998). It is unclear why they did not consider this issue in their study.

For the sake of practical application, Studies of flow boiling of refrigerant mixtures in plate heat exchangers with small hydraulic diameter channels have been performed by several researchers. Hsieh and Lin (2002) studied experimentally saturated flow boiling heat transfer and frictional pressure drop of the ozone friendly refrigerant R410a in a vertical plate heat exchanger with a hydraulic diameter of 6.6 mm. They found that both boiling heat transfer coefficient and frictional pressure drop increased almost linearly with the imposed heat flux. Furthermore, the refrigerant mass flux exhibits significant effect on the saturated flow boiling heat transfer coefficient only at higher heat flux. For a rise of refrigerant pressure, the frictional pressure drops were found to be lower to a noticeable degree. The refrigerant pressure has very slight influence on the saturated flow boiling heat transfer coefficient. They proposed empirical correlations for both boiling heat transfer coefficients and friction factor based on their data. Wellsandt and Vamling (2003) studied flow boiling of mixtures including R134a/R32 and R124a/R125/R143a in a plate-type evaporator with vertical small channels. The hydraulic diameters included 4.2 mm, 5.24 and 6 mm. Both flow boiling heat transfer and two-phase flow pressure drop were studied. They proposed a simple heat transfer model based on their experimental data.

Chen et al. (2004) studied experimentally two-phase pressure drops of R410a in a four U-type return bends with tube diameters of 3.3 and 5.07 mm and curvature rations of 3.91 and 8.51 mm, respectively. They proposed a modified two-phase friction factor correlation based on their experimental data.

Greco and Vanoli (2005a) studied experimentally flow boiling with HFC mixtures in a smooth horizontal tube (6 mm ID and 6 m length stainless tube). The local heat transfer coefficients of HFC mixtures R410a and R404a were measured. Both are near-azeotropic mixtures. Greco and Vanoli (2005b) compared their experimental data of the two HFC mixtures with theoretical results predicted with the correlations of Chen (1966), Kandlikar (1990), Shah (1976), Yoshida et al. (1994), Gungor and Winterton (1986, 1987), Yu et al. (2002a), Steiner and Taborek (1992), Jung and Radermacher (1991) and Kattan et al. (1998a,b,c). Apparently, they used some correlations developed for pure fluids to evaluate the heat transfer coefficients for mixtures. Although they concluded that the Kandlikar correlation was the best one among the correlations selected according to their experimental data. However, it can be seen that the standard deviation is still high. Based on their experimental data, a modification of the Kandlikar correlation was proposed. The modified correlation predicted the experimental results much better than all other correlations.

It is obvious that the research on two-phase flow and flow boiling of mixture in small and mini channels is much less than that of pure fluids in small and mini channels. In fact, as one important aspect of two-phase flow and flow boiling, there is no study of flow patterns and flow visualization of mixtures in the literature. Furthermore, all the heat transfer correlations available in literature are for normal size tubes and passages.

As for small diameter channels, no correlation and heat transfer mechanisms are available. Even for the experimental studies of flow boiling heat transfer available in the literature, there are two main limits. One is that some data reduction method is not appropriate. The other is that there are limited test channel diameter range for both small and micro dimensions. In all, there lacks a systematic knowledge of two-phase flow and flow boiling of mixtures in small and mini channels. As the emergence of some new technologies related to this area, it is of significance to develop research on two-phase flow and flow boiling of mixtures in small and mini channels. For both fundamental (the theory of two-phase flow and flow boiling of mixtures) and applied studies, the future research should focus on the following aspects to fulfill this void in this important area:

(i) Systematic experiments on two-phase flow and flow boiling of mixtures in a wide of channel diameter ranges covering both macro and micro scales. These should include the effect of component composition on both heat transfer coefficient and two-phase pressure drop.
(ii) Flow pattern visualization and transition criteria of two-phase flow and flow boiling of mixtures in small and mini channels. These should relate to flow boiling heat transfer mechanisms which intrinsically relate to flow patterns.
(iii) Modeling of flow boiling heat transfer and two-phase pressure drop of mixtures in small and mini channels. A generalized heat transfer model should be targeted by incorporating the heat transfer mechanisms, flow patterns, channel sizes, component compositions, diffusion of components etc.
(iv) Development of the theory of two-phase flow and flow boiling of mixtures in small and mini channels. This should include a detailed study of the mechanisms of mass, momentum and heat transfer under conditions of interaction of hydrodynamic and thermal effects and phase changes in small diameter channels including micro-channels.
(v) Study of two-phase flow and flow boiling of mixtures in compact and micro heat exchangers. For the sake of practical application, effort should be focused on the development of generalized heat transfer and pressure drop correlations for both compact and micro heat exchangers.

It is envisaged that a systematic research on two-phase flow and flow boiling in small and mini channels will bring advancement of knowledge and new theory of two-phase flow and flow boiling of mixtures in small diameter channels and meet the practical requirements in various applications. However, it is still a long-term task to achieve a systematic knowledge in this important area because of the complexity and difficulty of two-phase flow and flow boiling phenomena of mixtures in small and mini channels. Efforts should be made to contribute to both experimental and theoretical studies in this important area in the future.

5. Conclusions

A state-of-the-art review of two-phase flow and flow boiling of mixtures in small and mini channels is presented this paper. The various definitions of small and mini channels are presented at first. The various definitions of small diameter channels were evaluated with several typical fluids including both pure and mixture fluids and were compared with each other. Comments on different viewpoints of the channel size classifications are acknowledged. Then, the background of two-phase flow and flow boiling of mixtures is described. As the fundamental for the study of two-phase flow and flow boiling phenomena of mixtures in small and mini channels, previous experimental work and models of two-phase and flow boiling of mixtures in normal size channels are presented. As the main objective of the present paper, the current research status on two-phase flow and flow boiling of mixtures in small and mini channels is presented. It is obvious that the research in this important area is still rare so far. For both fundamental and applied studies in this area, the future research directions have been indicated so as to fulfill this void in the field of two-phase flow and flow boiling of mixtures in small and mini channels.

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