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As the scale of devices becomes small, thermal control and heat dissipation from these devices can be effectively accomplished through the implementation of microchannel passages. The small passages provide a high surface area to volume ratio that enables higher heat transfer rates. High performance microchannel heat exchangers are also attractive in applications where space and/or weight constraints dictate the size of a heat exchanger or where performance enhancement is desired. This survey article provides a historical perspective of the progress made in understanding the underlying mechanisms in single-phase liquid flow and two-phase flow boiling processes and their use in high heat flux removal applications. Future research directions for (i) further enhancing the single-phase heat transfer performance and (ii) enabling practical implementation of flow boiling in microchannel heat exchangers are outlined. [DOI: 10.1115/1.4005126]

Keywords: microchannels, review, single-phase flow, flow boiling, enhancement, heat transfer modeling

Introduction

Smaller channel size in a heat exchanger provides higher surface area to volume ratio and results in higher heat and mass transfer rates and lower equipment size. Automotive and aerospace industries embraced the use of smaller sized passages in compact heat exchangers to meet the weight and size constraints, while high performance requirements in cryogenic industries, for example, necessitated the use of millimeter-sized passages in equipment with relatively large heat transfer rates and higher effectiveness requirements.

Tuckerman and Pease [1] in 1981 demonstrated for the first time the high heat flux removal capability of up to 800 W/cm² achieved with microchannels in single-phase and two-phase flows. After almost three decades, such high heat flux removal is now being accomplished in practical devices, although much of the work has been conducted in the last several years. A brief historical survey highlighting research in the areas of single-phase and two-phase cooling with microchannels, followed by recommendations for future work, is presented in this paper.

After a slow start, research interest in microchannels has increased significantly in the last decade. Figures 1 and 2 show histograms of technical publications in each year related to single-phase liquid flow and flow boiling, respectively. Application of microchannels with single-phase liquid flow for heat removal was investigated at only a few places in the world until 1996. Starting in 1997, the pace accelerated until 2008. It seems to have reached a plateau, with 2009 numbers falling below the 2007 level. The main thrust during this period was on understanding the fundamental mechanisms. Research is now focused on further refining the theoretical concepts, generating experimental data sets, enhancing heat transfer performance while reducing pressure drop, and extending the use of microchannels to new applications.

Figure 2 shows the publication histogram for research related to flow boiling in microchannels. Until 2004, research was conducted only at a few academic institutions as the basic questions regarding the viability of flow boiling were being examined. The interest since 2005 has somewhat leveled off, indicating some degree of saturation in our efforts in this field. The focus of research during this period was on the fundamental issues, mechanisms, and performance characterization. A breakthrough is awaited to yield higher heat transfer coefficients, higher critical heat flux (CHF) values, and stable operation before the flow boiling technology in microchannels can be implemented in practical devices.

This paper is organized to provide a historical perspective followed by the state-of-the-art developments in the two major topics in microchannels: single-phase flow and flow boiling. Condensation is not covered in this paper. A brief overview of research conducted in the last three decades is presented first, highlighting some of the important contributions in the respective fields. This is followed by a more in-depth review of the research contributions made in the last 5 years in each area. With over several thousand publications reported over this time, it is truly a formidable task to cover all the advancements published in this field. Although every effort has been made, the author is painfully aware that there may have been inadvertent omissions of some important publications. Excellent review articles covering the developments in microchannel research, giving the details of experiments, their uncertainty levels, and model/correlation descriptions have been published periodically and readers are encouraged to read these for an exhaustive coverage of the earlier work [2–28].

Single-Phase Liquid Flow

Historical Development Timeline—Single-Phase Liquid Flow. The developments in single-phase flow in microchannels may be divided into two aspects: fluid flow and heat transfer. Major milestones in our understanding in these fields are summarized in Fig. 3, covering a 25-year period. Starting with the Tuckerman and Pease [1] work in 1981, the next research emphasis was on the design and implementation aspect during 1986–1988 [29–32]. This was followed by research focused on fundamental understanding during the 11-year period, 1992–2002 [33–39]. Major emphasis during this period was on validation of continuum theory for incompressible fluid flow in microchannels. The period 2003–2004 reflects a renewed interest in this field, with a number of specialty conferences dealing with microscale phenomena emerging during this period. The focus shifted on channel classification and fundamental understanding of the fluid flow and heat transfer phenomena and on recognition of need for further heat transfer enhancement [40–47]. Finally, during the period 2005–2006, attention was focused on practical implementation, heat transfer enhancements using offset strip fins and nanofluids, developing an in-depth understanding of roughness effects and transition to turbulence, and influence of channel deformity [48–52]. The timeline is provided to illustrate general historical development in this field, with individual papers as representative developments during those periods. The period 2006 onward is left out as more time may be needed for the research community to recognize the significance of research reported in the last 4 years. However, a review of some of the important recent papers is presented later in this paper to indicate the state-of-the-art advancements in this field.

Historical Perspective—Single-Phase Liquid Flow. Significant advances have been made in the single-phase liquid flow research in microchannels in the last three decades. Microchannels were implemented in heat exchangers in some of the earlier studies. An excellent summary of early developments in microchannel heat exchangers is presented later in this paper to indicate the state-of-the-art advancements in this field.

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exchangers, providing a clear historical perspective, was presented in 1993 by Goodling [2]. It is a “must-read” article for all researchers in this field. It is interesting to note that even when microchannel research was in its infancy, 78 papers were cited by Goodling. During the period 1981–1993, the main centers engaged in research in this field were as follows: Stanford University and Lawrence Livermore National Labs—the pioneering thesis by Tuckerman [53] investigated for the first time microchannels in single-phase and two-phase modes for high heat flux removal, Mundinger et al. [54] applied the high performance silicon microchannel heat exchanger design to laser diode array cooling; Massachusetts Institute of Technology and Lincoln Laboratories—Phillips [55,56] and Phillips et al. [31] provided the entrance region Nusselt number and friction factor values in developing flows for three side heated channels under H1 boundary condition (constant heat flux with uniform circumferential temperature at any cross-section) along with a detailed design procedure, and Missaggia et al. [57] and Walpole et al. [32] demonstrated effective cooling of laser diodes with 500 W/cm² heat removal rate and a laser junction temperature rise of only 40°C. North Carolina State University—Turlik et al. [58] applied the microchannel design to multichip modules covering 25 cm² area; Auburn University—Knight et al. [33] provided optimization schemes for designing microchannel geometrical parameters for laminar and turbulent flows; Nippon Institute of Technology, Japan—Sasaki and Kishimoto [29] used microchannels for diode array cooling, and Kishimoto and Sasaki [30] employed diamond-shaped interrupted fins, perhaps the first application of enhanced microchannels. A number of individual contributions were also seen during this period—Bar-Cohen and
Jelinek [59] developed a geometrical optimization scheme for fin spacing in air cooled heat exchangers that could be extended to microchannels, and Bejan and Morega [60] applied pin fins to enhance heat transfer in a wide, single-flow passage (narrow gap) while keeping the pressure drop low. These developments show that practical implementation of microchannels with single-phase liquid cooling generated considerable interest by 1993.

In the later part of 1990s, the focus shifted to understanding the fluid flow and heat transfer phenomena in microchannel liquid flows. An excellent summary of the early developments is given by Mehemdale et al. [3]. A number of authors, as early as 1983, reported the single-phase studies for gas flows, which are not covered in this paper. In 1993–1994, Peng and Wang [61,62] reported the first study on single-phase flows in minichannel and Peng et al. [34,35] in microchannels. They suggested possible departure from the continuum theory. The Poiseuille and Nusselt numbers exhibited a dependence on Re, which could not be explained. The reasons for the discrepancies were subsequently explained by other investigators over the next 10 years.

The author feels compelled to write a word of gratitude to late Professor X. F. Peng, of Tsinghua University in China, whose untimely demise at the age of 48 years is difficult to accept. His contributions in exploring the single-phase and two-phase flows and heat transfer characteristics have made a significant impact on the research in this field. He is credited as one of the pioneering researchers to systematically study the single-phase and two-phase flows in microchannels.

In 2000, several papers focused on the application aspects. Harris et al. [36] developed a microchannel heat exchanger for liquid-to-air heat transfer, while Fedorov and Viskanta [37] studied the conjugate substrate conduction and microchannel convection problem and identified locations where severe thermal stresses may occur.

The fundamental understanding of the flow and heat transfer phenomena in microchannels was a major focus area for researchers in the first half of the first decade of this millennium. The experimental work presented by Peng and coworkers [34,35,61,62] in the mid-1990s and by Mala et al. [63], Mala and Li [64], and Xu et al. [65] in the late 1990s still left the question unresolved regarding the applicability of continuum models to liquid flows in microchannels. Xu et al. [65] reported the validity of the conventional theory for liquid flow in microchannels using fully developed flow; however, they neglected the entrance region effects. The uncertainties associated with entrance and exit loss coefficients and other factors such as nonuniformity in channel cross-sectional dimensions and experimental uncertainties, especially at low flow rates, seem to have fortuitously compensated for the entrance region effects. Palm [4] and Sobhan and Garimella [6] indicated that the current research at that time was inconclusive regarding the microscale effects on liquid flow and applicability of continuum equations. Judy et al. [38] in 2002 reported the first experimental data that followed the continuum theory for liquid flows in microchannels. At the same time, Qu and Madawar [39] confirmed the applicability of continuum theory for 231 µm × 731 µm minichannels. However, the discrepancy among various data sets could not be resolved. Wu and Cheng [41] indicated that surface roughness may affect the heat transfer in microchannels. Guo and Li [43] proposed the surface roughness, axial conduction, and measurement errors as possible sources for discrepancy in microchannel data. Lelea et al. [46] in 2004 presented data confirming the continuum theory; however, reviews by Obot [28] in 2001 and Morini [12] in 2004 left it as an unresolved open question. Obot [28] asserted that the transition to turbulence does not occur below Re = 1000 in smooth microchannels and that Nu varies as Re^{1/2}. In 2004, Sharp and Adrian [47] confirmed that the transition to turbulent flow occurs at approximately the same Reynolds numbers as in macroscale tubes. In 2005, Steinke and Kandlikar [66] presented experimental data confirming the validity of the continuum theory and also presented an analysis of earlier data showing the entrance region effects, entrance and exit losses, and experimental uncertainties as the major reasons for the discrepancy in the earlier data reported in the literature. They also presented a detailed analysis of the uncertainties associated with fluid flow and heat transfer data available in the literature. Lee et al. [49] verified the applicability of the continuum theory and suggested that careful attention should be given to the boundary conditions applied during the experiments.

The effects of property corrections and dissolved gases on single-phase diabatic flows were reported by Steinke and Kandlikar [67] in 2004. Their work indicates that, because of the steep temperature gradients in microchannels, it is important to apply property correction factors, generally of the form given below, in evaluating single-phase friction factors

\[ \frac{f}{f_{Re}} = \left( \frac{\mu_{w}}{\mu_{m}} \right)^{0.58} \]  

(1)

The presence of dissolved gases in water during diabatic single-phase experiments resulted in higher friction factors due to small bubbles forming on the walls and increasing resistance to the flow [67]. Figure 4 shows the effect of dissolved gas content on the friction factors after accounting for the variable property effects given by Eq. (1). It is seen that the friction factors are higher with a dissolved oxygen content of 8 ppm, but the results for 1 ppm concentration are in good agreement with the conventional theory for laminar flow. It is therefore recommended that degassed water be used during experiments for evaluating the friction factors in microchannels. Clearly, this poses a challenge in predicting the pressure drop in practical systems using liquids containing dissolved gases. It would be of practical advantage to avoid bubbles in microscale heat exchangers.

Defining microchannels has been a subject of considerable discussion in the literature. Kandlikar and Grande [40] in 2003 reported a channel classification scheme that identified microchannels as channels in the hydraulic diameter range of 10 µm to 200 µm. This scheme is followed in the present paper. The earlier scheme by Mehemdale et al. [3] had arbitrarily proposed 1 mm as the distinction between microchannel and mesochannel flows. It should be remembered that the channel classification scheme is intended to serve only as a guide. The associated flow and heat transfer phenomena in these channels are dictated by specific operating conditions, fluid properties and type of flow, such as single-phase or two-phase flow, gas, or liquid, etc. Further discussion on this topic is given by Kandlikar [68].

At the same time when the issue of the applicability of continuum theory to microchannel flows was resolved in 2005, the use...
of enhanced microchannels to meet the high heat flux cooling demands was recommended by Kandlikar and Grande [44]. Peles et al. [69] presented experimental data on micropin fins and obtained a very low thermal resistance value, comparable to the microchannel flows. Colgan et al. [50] reported a practical implementation of liquid cooling using enhanced microchannels with offset-fin geometry to dissipate very high heat fluxes. Steinke and Kandlikar [70] measured heat transfer coefficient values of over 500,000 W/m²·K using the same offset strip-fin microchannel geometry employed by Colgan et al. [50]. A few investigators have studied optimization of plain and enhanced microchannel geometrical parameters [71,72] and modeling of plain microchannels [73] to provide practical design guidelines. The effect of entrance region on heat transfer under H1 boundary condition (circumferentially uniform wall temperature and axially uniform wall heat flux thermal boundary conditions) has been studied by Lee and Garimella [74].

Nanofluids were introduced in microchannel heat sinks in the early part of the 2000 s. In 2005, Koo and Kleinstreuer [48] reported an important finding that lists the conditions under which nanofluids may provide enhancement for application in practical single-phase microchannel devices. They reported that (i) a nano-fluid concentration of at least 4%, (ii) elevated thermal conductivity of nanofluids, and (iii) treated channel walls, which prevent nanoparticle adhesion, may contribute in increasing the heat transfer rate.

The current literature clearly indicates that continuum modeling is applicable to liquid flow in microchannels. Rosa et al. [26] indicated in their review article that scaling effects need to be considered to identify if there are any special effects due to microscale channel dimensions in single-phase flow. Some of these issues were identified by the earlier investigators. The entrance and exit losses for macroscale channels are empirically correlated based on macroscale experiments. Their validity in microscale applications is open to question. The effects due to flow area changes need to be revisited as well. The property variation effects become important as the temperature gradients are quite steep near the wall. Parallel channels experience maldistribution effects, which need to be quantified for the specific header and channel geometry.

The areas where fundamental heat transfer related research with single-phase liquid flow has been focused in the last 5 years are design analysis for optimization, heat transfer augmentation techniques, and roughness effects. These topics are reviewed in the following sections.

Roughness Effect. The effect of surface roughness on single-phase liquid flow in microchannels has been studied in the literature both numerically and experimentally. These studies indicate that the conventional representation of constant friction factor in the laminar region is not accurate. The classical experiments by Nikuradse were found to have large uncertainties in the pressure drop measurements in the laminar region [75]. A number of different parameters are employed in the literature to model roughness features [76]. Kandlikar et al. [51] proposed replacing the root diameter with a constricted hydraulic diameter \(D_{h,cf}\)

\[
D_{h,cf} = D_h - 2\varepsilon
\]  

(2)

where subscript \(cf\) refers to the constricted flow parameters and \(\varepsilon\) is the roughness parameter defined as the distance between the mean floor line and the roughness peak height. The experimental results for the uniform roughness were correlated well. Using the constricted flow diameter, a modified Moody diagram was presented in which the laminar region was accurately represented by a straight line for all relative roughness values, and the friction factor reached a plateau beyond which the increased roughness had no effect on friction factor [51].

The results of regularly spaced ribs, however, showed a dependence on the pitch of the ribs [77]. The friction factor gradually shifted from the constricted flow diameter at close rib spacing to the smooth tube value at very high spacing. Rawool et al. [78] numerically simulated the flow in microchannels with trapezoidal, rectangular, and triangular shaped roughness structures. The trapezoidal shape yielded the lowest friction factors in the group. The circulation behind the roughness elements was seen to cause additional losses leading to an increase in the friction factor. The numerical work by Croce et al. [79] on conical-shaped three-dimensional elements of base diameter \(b\), spaced evenly at a pitch of \(s\) in both directions, indicates that the heat transfer and pressure drop results can be correlated by modifying \(\varepsilon\) in Eq. (2) to \(\varepsilon/s\). Their results show that the two-dimensional ridge roughness elements are more effective in providing relatively large heat transfer enhancement compared to the 3D elements as the large flow blockage in the valleys of the 3D elements more than offsets the enhancement in the heat transfer at the tips of the conical elements. A review of different roughness shapes and their effects and the instability mechanisms leading to turbulent transition in microchannels is presented in Ref. [22].

Several models were presented in the literature to account for roughness effects at the microscale level. Kleinstreuer and Koo [80] proposed a porous medium model for the roughness region, which is assumed to act in parallel to the main flow in the core of the channel. The flow resistance in the roughness region, modeled as a porous region, consists of a constant term and a velocity dependent power law term. The momentum equation for the open region (in the core of the channel) is solved by matching the slopes of the velocity profiles in the porous region and the open channel region. Bahrami et al. [81] proposed a model based on a Gaussian distribution of roughness elements for the prediction of pressure drop in fully developed laminar flow in microchannels. Earlier, Qu et al. [82] proposed a roughness–viscosity model in which the fluid viscosity was modified to include a viscosity term induced by the surface roughness. The roughness viscosity was considered similar to the eddy viscosity in the turbulent flow. In a subsequent paper on heat transfer effects, Qu et al. [83] modified their roughness viscosity model to include two components in the apparent viscosity. This included the sum of the roughness viscosity derived in their earlier paper and a friction viscosity. The experimental heat transfer results were correlated with this approach. However, this approach has not been tested with other data from the literature.

The effect of roughness on heat transfer is not as well studied in the literature. A recent publication by Morini et al. [84] indicates little effect of roughness on heat transfer in the laminar region for water and FC-72 in microchannels and minichannels. The earlier work by Kandlikar et al. [42] indicated an increase in the heat transfer coefficient with stainless steel minichannels with different relative roughness values obtained through acid etching.

Gamrat et al. [85] studied uniformly spaced roughness structures numerically and proposed a roughness-layer model for heat transfer in laminar fully developed tubes. They extended the porous region model of Kleinstreuer and Koo [80] by using the porosity of the roughness layer and roughness height as parameters. Their results indicate that the friction factor increases more than the heat transfer coefficient over the range of relative roughness up to 11%.

The effect of roughness is seen in inducing early transition to turbulence. This transition is believed to be one of the reasons for the higher friction factors for rough microchannels reported in the literature. The instabilities and relaminarization processes occurring in the flow around roughness elements have been a topic of intense research in the fluid mechanics community. The pioneering work of Narasimha and Sreenivasan [86] provides an insight into the underlying mechanisms for early turbulence transition. Based on the uniform roughness as well as ribbed geometries, the following correlation for transition Reynolds number is based on the constricted flow diameter defined in Eq. (2) [77].
For $0 < D_{h,	ext{cf}} < 0.08; \quad \text{Re}_{h,	ext{cf}} = \text{Re}_o - 18,750(\varepsilon/D_{h,	ext{cf}})$
For $0.08 < D_{h,	ext{cf}} < 0.15; \quad \text{Re}_{h,	ext{cf}} = 800 - 3270(\varepsilon/D_{h,	ext{cf}} - 0.08)$

(3)

where $\text{Re}_{h,	ext{cf}}$ is the transition Reynolds number for smooth tubes for the given channel cross-section. For the two-dimensional roughness elements, the experimental results of Brackbill and Kandlikar [77] indicate that the rib height and spacing both influence the transition.

Hu et al. [87] numerically studied the effect of three-dimensional roughness elements in channels of microfluidic devices for Reynolds number range of 0.1–10. The channel height was chosen to be 5 $\mu$m and the roughness elements were 0.1–2.1 $\mu$m in height and the spacing was varied from 0.1 to 1.5 $\mu$m. The effect of varying channel height from 5 to 50 $\mu$m was also investigated. They presented equations for the reduction in the channel height due to roughness elements to be applied in the conventional pressure drop equations.

Three-dimensional roughness has been studied so far in microfluidic applications at low Reynolds numbers, mainly to enhance mixing. It is a promising technique for liquid heat transfer applications. Further data is needed covering hydraulic diameters in the range of 10–200 $\mu$m in order to assess the effect of such structures on fluid flow and heat transfer in microchannels.

**Heat Transfer Augmentation of Single-Phase Liquid Flow.** In high heat flux removal applications such as chip cooling, further increases in heat transfer coefficients over plain microchannels are required to meet the cooling needs. A detailed review of various techniques applicable to heat transfer enhancement in microchannels and minichannels was presented by Kandlikar and Steinke [45]. The enhancement techniques are classified into two categories:

- Passive Techniques: surface roughness, flow disruptions, pin fins, offset-strip fins, channel curvature, re-entrant obstructions, secondary flows, out-of-plane mixing, and fluid additives (nanofluids);
- Active Techniques: vibration, electrostatic fields, flow pulsation, and variable roughness structures.

Some of these enhancement techniques have been investigated in the literature. A brief review of the available studies is presented here. Among passive techniques, surface roughness has been an area of growing interest as seen from the review presented in the earlier section. Xu et al. [88] utilized the concept of a redeveloping thermal boundary layer due to flow obstructions. These obstructions were created by a series of intersecting longitudinal (triangular, with $D_h = 155$ $\mu$m, pitch $= 45$ mm) and transverse microchannels (trapezoidal, sides of 1.015 mm and 0.712 mm, height $= 0.212$ mm, and pitch $= 3.694$ mm). In a silicon chip with 21.45 mm flow length, the heat transfer enhancement was found to be around 16% while friction factor was actually reduced by about 20%. Further performance improvements should be possible through optimization. Chang et al. [89] studied the effect of height on the heat transfer in rib-roughened side walls of microchannels. The heat transfer coefficient decreased immediately behind the ribs, but an overall performance improvement was noted. More systematic study is needed to clearly establish the efficacy of opposing ridges. Wei et al. [90] employed dimpled surfaces and obtained 10–35% improvement in heat transfer coefficient with very little increase in friction factor. Optimization of this geometry is needed to fully explore its potential.

Pin fins are extensively used in macroscale applications. Kosar and Peles [91] experimentally studied pin fins of circular cross-section extending over the entire height of microchannels. They found the performance of pin fins to be predictable by macroscale correlations. Significant enhancements, with heat transfer coefficients comparable to boiling, were reported. They emphasized the need to optimize the geometrical parameters for obtaining the desired benefits while keeping pumping penalties at a minimum.

A major milestone was achieved when Colgan et al. [50] employed offset strip fins in microchannels with water flow to dissipate heat fluxes of 400 W/cm$^2$. They employed 180 $\mu$m deep channels with fins at a constant pitch of 100 $\mu$m and fin width of 65 or 75 $\mu$m. The fin length was either 210 $\mu$m or 240 $\mu$m with a gap of 40 $\mu$m between the fin rows. Significantly, higher heat transfer performance was noted, with an improvement by a factor of approximately 1.5 to 3. By limiting the flow length to 3 mm, the pressure drop was limited to below 30 kPa. Steinke and Kandlikar [70] conducted experiments with fins of 250 $\mu$m length and 50 $\mu$m thickness, the same configuration as employed by Colgan et al. [50], and reported the average heat transfer coefficients to be in excess of 500,000 W/(m$^2$·K). This is perhaps the highest heat transfer coefficient value reported for single-phase internal flow.

Further enhancement is possible by reducing the channel width and fin length. A detailed geometrical optimization of the strip-fin geometry needs to be performed for the accompanying pressure drop penalty [72].

Other fin geometry studied in the literature is the longitudinal fins by Foong et al. [92]. The short fins seem to be promising. Another geometry that holds promise is microfins or helical grooves. This geometry has been very successful in refrigeration applications. Zhang et al. [93] evaluated aluminum, silicon, and metallic foam heat sinks and determined that the aluminum and foam heat exchangers could achieve a heat dissipation rate of 100 W/cm$^2$, while silicon microchannels could dissipate 200 W/cm$^2$. The thermal conduction resistance within the metal foam was the limiting factor. Tsuzuki et al. [94] introduced a wavy, S-shaped fin and optimized the geometry using a numerical solver to study parametric dependence of fin angle, guiding wing, thickness, length, and roundness. From these parametric studies, they identified geometrical parameters that are most suitable in enhancing heat transfer while keeping the pressure drop low.

Nanofluids are mixtures of nanoparticles and a base liquid used in a cooling system. Enhancement in thermal conductivity of nanofluids was reported by Lee et al. [95], followed by a study showing heat transfer enhancement with nanofluids by Xuan and Li [96] and Keblinski et al. [97]. Nanofluids have also been investigated extensively in microchannels. Introducing nanoparticles in a liquid has shown to enhance the energy transport in the boundary layer [48.98–101]. Bergman [102] showed that the combination of increased thermal conductivity and reduced specific heat in nanofluids affects their performance compared to the base liquid. Although the exact mechanism is not clear, Bergman introduced a dimensionless parameter to indicate whether the heat transfer performance will improve or deteriorate with nanofluids. Excellent recent reviews on the enhancement of thermal conductivity of nanofluids have been presented in the literature and the reader is referred to those publications, e.g., Refs. [103–110].

Among active enhancement techniques, synthetic jets have been studied by Chandratileke et al. [111]. They introduced synthetic jets from a neighboring volume through a pulsing device with a zero net mass flow rate. This technique introduces very little pressure drop penalty and is suitable for localized cooling of hot spots. Introducing a vapor compression refrigeration system to lower the coolant temperature or a Peltier cooler for cooling of hot spots on a chip are among some of the active systems being considered [112].

**Challenges and Future Research Needs in Single-Phase Liquid Flow.** A number of research needs have been identified in the discussion presented in the above section on single-phase liquid flow. Some of the major ones are summarized below.

**Roughness Effects.** The effects of two-dimensional and three-dimensional roughness elements on friction factor and heat transfer have been studied mostly using numerical techniques in the literature. Since the effect of recirculation and eddies generated...
behind the roughness structures are not accounted for accurately in the numerical schemes, experimental validation of the numerical results is essential. Developing theoretical models to account for these effects, especially on heat transfer, is an important area. Such models are expected to provide information regarding the underlying mechanisms, which can help in designing roughness surfaces with higher enhancement in heat transfer than the associated increase in pressure drop. There is also a need for developing optimized 2D and 3D geometries of the roughness structures that are specifically tailored for a given range of Reynolds numbers. In addition, new enhancement techniques need to be developed that utilize unique fabrication techniques that are available at microscale, such as etching, for mass production.

Heat Transfer Augmentation, Liquid Flow. Among the passive devices studied, fins hold the most promise. The offset strip fin geometry needs further investigation for geometrical optimization considering the suitable fabrication techniques. The header configuration is an important consideration in its design. Different cross-sectional geometries need to be explored for pin-fin configuration. The optimizing procedure outlined by Tsuzuki et al. [94] for designing S-shaped fins provides a very promising approach. It is expected that specific geometrical parameters will be optimized using numerical simulation in the future for a given liquid under a specific Reynolds number operation.

From its success at macroscale, microfin or microgroove geometry seems to be a strong candidate to provide heat transfer enhancement with relatively low pressure drop penalty. Micromixer is another type of device that has been extensively studied in the literature. It seems to be very promising in augmenting heat transfer as well, since it is able to effectively induce the mixing of fluids from the core of the flow channel to the heated surface. Among the active enhancement techniques, pulsed flow and synthetic jets are attractive for localized cooling. Concurrent research on the effect of pulsations/vibrations on the system integrity needs to be investigated before applying them in practical devices. It is expected that there will be significant research activity in this area in the coming years.

Numerical Results for Entrance Region. Among other topics, there is a need for generating numerical results for heat transfer and friction factors for microchannels of different aspect ratios under the more practical H2 boundary condition (constant heat flux axially and circumferentially).

Flow Boiling

Historical Development Timeline—Flow Boiling in Microchannels. Figure 5 shows major milestones in flow boiling research over the 25-year span since the first study on this topic by Tuckerman and Pease [1] in 1981. There was little research activity in the period 1981–1991. During 1992–2001, the major developments included a systematic study in minichannels and silicon microchannels, local heat transfer coefficient data, and identifying major obstacles [113–117]. In the period 2002–2004, research focus shifted to identifying flow patterns and differences between diabatic and adiabatic flows, modeling, and continuing work on local heat transfer coefficient data [8,67,118–124]. During 2005–2006, new work was presented on numerically modeling bubble growth, overcoming instabilities, and the use of enhanced structures for heat transfer [14,125–131]. The developments beyond 2006 onward are left out, as their relevance and...
importance are expected to become clear during a span of the next 5–10 years.

Historical Perspective—Flow Boiling in Microchannels. Flow boiling in microchannels was first studied by Tuckerman and Pease in 1991, but it took more than 10 years before a major effort was undertaken by other researchers to explore this topic further. The first major effort on understanding the thermohydraulic behavior of flow boiling systems in microscale passages was presented by Moriyama et al. [113] in 1992. They used R-113 refrigerant in 35–110 µm high and 30 mm wide rectangular passages. This geometry is sometimes referred to as microgap in the literature. The pressure drop results were correlated with the available two-phase correlations and the heat transfer was found to be 3–20 times larger than that for single-phase flow. One of the important findings was that the capillary number was identified as an important parameter in determining the liquid film thickness while modeling the slug flow. They also developed analytical models for the film flow with a vapor core and slug flow with vapor bubbles separated by the liquid slugs. The agreement between the model and experimental results was quite good and the effect of capillary number was shown to be an important group. Since this article addresses a number of issues that are being researched currently, a more in-depth review of their work is presented below.

The experimental data obtained by Moriyama et al. [113] are shown in Figs. 6 and 7. The heat transfer coefficient is plotted as a function of quality for the two mass fluxes in Fig. 6. The heat transfer coefficient increases dramatically with quality near a single quality, and then drops rapidly at higher qualities as dryout occurs. Figure 7 shows a comparison of the data with the pool boiling correlation by Armstrong [132]. Their results indicate a convergence trend is noted with an increase in mass flux. These trends are actually a combination of pool boiling and flow boiling characteristics, as the large width of the channels employed (30 mm) reduces the flow effect on nucleation but causes the flow of liquid around nucleating bubbles. Heat transfer is thus enhanced from the convective contribution due to the liquid flow.

In 1993, Peng and Wang [61] reported one of the first studies with subcooled flow boiling of water in microchannels. The fully developed boiling region had little influence from subcooling or flow velocity. These observations were helpful in determining the nature of flow boiling in microchannels, as will be discussed in later sections. Although the heat transfer data indicated the presence of nucleate boiling, they could not observe any bubbles, perhaps due to the lack of a high speed and high magnification imaging system.

A more systematic study covering a large range of minichannel diameters was conducted by Bowers and Mudawar [114] in 1994. They reported the average heat transfer, pressure drop, and CHF data in these channels. Jiang et al. [115] were the first researchers to incorporate the microchannel passages (D < 200 µm) with hydraulic diameters of 40 and 80 µm in a silicon heat sink. The thermal performance was unaffected by the flow and was dominated by the nucleate boiling, similar to the earlier studies on minichannels. Hetrsroni et al. [116] in 2000 used 103–129 µm hydraulic diameter microchannels for cooling silicon chips and studied the nonuniform temperature distribution on the chip. They observed single-phase flow followed by annular flow pattern during boiling. Kandlikar et al. [117] in 2001 observed the flow boiling instability associated with nucleation and rapid bubble growth in a minichannel. Instability will be reviewed in a later section.

In 2002, Serizawa et al. [119] conducted adiabatic flow experiments with steam–water and air–water flows in microchannels and studied flow patterns, which were affected by the surface roughness. They presented a flow pattern map based on their experimental observations. The differences between the adiabatic and diabatic flows were studied experimentally by Hetsroni et al. [120], who clearly demonstrated the role played by nucleation and bubble growth in diabatic flows. The main features being the absence of rapid flow oscillations and instabilities in adiabatic flows.

Moriyama et al. [113] in 1992 provided a plot showing the dependence of h on x as seen in Fig. 6. In 2003, Yen et al. [121] measured the local heat transfer coefficient in single microtubes as a function of x, q, and G and noted a strong decreasing trend in a single microchannel. They confirmed the contribution from the nucleate boiling component. In 2004, Steinke and Kandlikar [124] reported the local heat transfer coefficients as a function of heat flux, mass flux, and quality for water in parallel microchannels. The microchannels were etched in silicon and were heated with a copper heater deposited on the back surface. The heat transfer coefficient was found to decrease dramatically along the length as quality increased. Considering that the fluid is water at atmospheric pressure, the deterioration in heat transfer coefficient with increasing quality indicated a major departure from the flow boiling characteristics in conventional channels (Dh > 3 mm).

Moriyama et al. [113] compared their flow boiling data in microchannels with gaps of 65 µm and 110 µm with pool boiling correlation by Armstrong [132]. Their results indicate a convergence at high heat fluxes as shown in Fig. 7 (for 110 µm gap). The reduced dependence on the mass flux points to the dominance of nucleate boiling, which has been confirmed in microchannels by a
number of later studies. In these studies, the researchers reported experimental heat transfer data for water and several refrigerants (see review papers on this topic, e.g., Refs. [8–11]). The experimental data confirmed the earlier observations, showing a strong decreasing trend with quality at low qualities and mixed trends at moderate qualities, e.g., Refs. [114–116,120,121,124,129,133,134]. A number of correlations have been proposed in the literature, but none of them are able to predict the heat transfer coefficient consistently well with the available experimental data [24,135–137]. Although attempts are continuing to correlate the flow boiling data, it is becoming apparent that the underlying data sets suffer from some of the shortcomings listed below:

(a) large experimental uncertainty in measurements of channel dimensions, local wall temperatures, and estimation of wall heat flux;
(b) differences in wall boundary conditions (a subject not explored in flow boiling literature);
(c) differences in nucleation cavity sizes and distribution on the channel walls;
(d) uncertainty related to the presence of instability condition and its severity.

Until experimental data sets that are free from the above concerns become available, any correlation scheme cannot be expected to provide an accurate predictive methodology. The concerns regarding flow boiling instabilities are being addressed in some of the recent publications. There is an immediate need to generate experimental data in flow boiling systems using the same rigor that the researchers have applied in their quest for checking the applicability of the continuum models in single-phase liquid flow.

High speed visualization during flow boiling in microchannels is again receiving considerable attention with the focus now on flow patterns, e.g., by Martin-Callizo et al. [138], Schilder et al. [139], and Harirchian and Garimella [140]. These studies are aimed at linking the modeling effort to the flow pattern. Although it is difficult to predict the flow patterns with any degree of certainty, the insight gained through such visualization is extremely useful.

**Modeling of Flow Boiling Heat Transfer.** The first model of the flow boiling phenomenon was presented by Moriyama et al. [113] to predict the pressure drop and heat transfer coefficient during slug and film (annular) flow in a large aspect ratio microchannel passage with a gap of 35–110 μm as described earlier. Although this paper addresses a number of fundamental aspects using nondimensional parameters, it has not been considered by later researchers in their modeling effort. A more in-depth review of this paper is presented to bring out their important contributions in this area.

Moriyama et al. [113] modeled the slug flow as shown in Fig. 8 for the low quality region and film flow as shown in Fig. 9 for the high quality region. Since the liquid–vapor interface plays an important role in the thickness of the film deposited during the slug and film flows, capillary number \( Ca = \frac{\mu_L g L_{jv}}{\sigma} \) was introduced in the modeling; \( \mu_L \) is the liquid viscosity, \( j_{L} \) is the total volumetric flux, \( m/s \), and \( \sigma \) is the surface tension, \( N/m \). The volumetric flux may be replaced by the representative flow velocity. The capillary number represents the ratio of the viscous to surface tension forces.

Figure 10 shows the variation of the two-phase to single-phase heat transfer coefficient ratio for 65 μm and 110 μm gap sizes, two different mass fluxes in each plot, and three different \( q/G \) ratios (shown in the inset table). For a given fluid in the film flow region, a lower \( Ca \) value would indicate a thicker film, and hence a lower heat transfer coefficient, whereas a higher \( Ca \) value would indicate a thinner film, and correspondingly a higher heat transfer coefficient. Their modeling seems to capture this trend well for the smaller gap sizes (see the trend in the right hand region of the plots). It may be instructive to study and extend the model by Moriyama et al. [113] incorporating some of the recent findings on instability and liquid film thickness.

In 2002, Jacobi and Thome [118] presented an elongated bubble model, which was later refined into a three-zone model by Thome et al. [141]. The single-phase heat heat transfer in the slug was calculated using steady-state laminar flow equations, thin film evaporation is calculated using conduction equations in the liquid film, and heat transfer in the dry regions is considered by gas flow equations separately. Film thickness was back-calculated to match the data and did not provide accurate representation of the actual thicknesses. The authors used steady-state equations for representing flow boiling phenomena.

Heat conduction in the liquid, as the liquid–vapor interface sweeps over the heater, and associated microconvection are essentially transient phenomena. Fully developed flow equations for gas flow and liquid slugs do not provide accurate representation of the two-phase flow field.

Liquid film thickness plays an important role in modeling heat transfer in slug flow as well in film (or annular) flow. A recent work by Zhang et al. [142] provides useful information on liquid film thickness measurement using laser extinction method in a microgap as a function of interface velocity. An interesting observation made from these experiments was that the film thickness increased with the interface velocity. For interface velocities of 3–5 m/s, the film thickness was around 10 μm for water and 20 μm for ethanol and toluene. The film thickness in flow boiling in microchannels is expected to be higher during explosive boiling due to higher interface velocities.

Zhang et al. [143] in 2002 used the available correlations and developed a model to predict the local heat transfer coefficient variation along the length of 20–60 μm hydraulic diameter parallel microchannels. They compared their predictions with the local wall temperature measurements in the silicon chip. The variation and the actual heat transfer coefficient values along the length...
were predicted well. This model illustrates the practical implementation of the available information in the design of microchannel heat exchangers.

Many researchers noted that the flow boiling heat transfer in microchannels is dominated by the nucleate boiling contribution, which is mainly influenced by the heat flux. Correlations that are similar to pool boiling type correlations, e.g., Armstrong [132] and Cooper [144], with no mass flux term present in them have been found to predict the microchannel flow boiling data with some success [24,113,145]. Similarities and differences between pool boiling and microchannel flow boiling was explored by Kandlikar [146,147] and it was noted that the elongated bubbles provide advancing and receding interfaces on the heater surface that may be compared to the expanding and contracting interfaces at the base of a bubble during pool boiling. The heat transfer in the interface region consists of transient conduction and micro-convection. Further guidance could be derived from the recent research in the area of pool boiling, e.g., Refs. [148,149].

Models based on annular flow [150,151], annular flow in triangular microchannels considering the contact angle effects [152], and bubble coalescence [153] have also been proposed in the literature. Fogg and Goodson [154] considered the nucleation phenomenon and linked it to the instability leading to water hammer (which is similar to the backflow observed by other researchers). There are a number of recent papers published on the modeling aspects specific to different flow patterns. In general, these models perform well with the parent data sets used in their development. A broader testing using a large number of data sets has not been reported in the literature. The modeling effort also suffers from the nonavailability of reliable experimental data over a wide range of conditions.

Flow Boiling Heat Transfer Enhancement. Enhancement in flow boiling heat transfer was experimentally studied by Kosar and Peles [125] by incorporating fins. A recent study by Krishnamurthy and Peles [155] indicated significant heat transfer enhancement with microfins using HFE7000 in 222 μm hydraulic diameter channels. Nanofluids have been studied with modest improvements as reported in a review article by Cheng et al. [156]. Recently, Khanikar et al. [157] implemented nanostructures on the channel walls with modest enhancement. Repeated testing reduced the enhancement effect due to damage to the nanostructures. This field is still in its infancy and more research efforts are needed.

Use of nanofluids for flow boiling enhancement is a promising area that is currently being explored in pool boiling applications. A study by Lee and Mudawar [158] indicates that nanofluids caused surface deposition in microchannels. Recent findings by Ahn et al. [159] showed that using 0.01% concentration alumina nanoparticles, CHF was enhanced by about 55%. The enhancement was due to deposition of the nanoparticles on the surface causing the contact angle to decrease from 65 deg to about 12 deg. The work by Ahn et al. confirms that reducing contact angle of the surface by some other means may be an effective way to enhance the CHF in microchannels.

Flow Boiling Instabilities. Flow boiling instabilities were observed by a number of investigators [120,121,124,129,133,134,160,161]. They have been identified as a major concern in reliable operation of a microchannel heat exchanger. The high heat transfer coefficients during single-phase heat transfer prior to nucleation result in lower wall temperatures, thus delaying the onset of nucleation. It also leads to a high liquid superheat at the nucleation location. The bubble nucleation in this superheated liquid environment causes an explosive growth of the bubble, causing it to expand in both downstream and upstream locations [131,162]. The resulting backflow in the channels is a major source of instability in microchannels. Numerical simulation of the bubble growth phenomenon confirmed the role of local liquid superheat in causing the unstable condition [126]. The bubble expansion upstream results in a backflow into the header of the parallel microchannels. Providing artificial nucleation sites, introducing inlet restrictors, and incorporating heated regions to initiate nucleation were some of the remedies investigated by a number of researchers [122,128,130]. Although these remedies reduce the severity of the instabilities, flow oscillations are still present and may adversely affect the CHF condition. In the last 5 years, a number of studies have been reported on identifying the envelope of operating conditions for stable and unstable operations.

Brunin and Tadrist [163] modeled the instability phenomenon by considering the pressure forces across a slug formed during explosive boiling. The overpressure created by the slug was compared with the channel pressure drop. The ratio of the two pressure drops was then linked to the applied heat flux and the resulting ratio of channel pressure drop and heat flux (dimensional quantity) was used as a parameter indicative of the flow

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Fig. 10 Variation of heat transfer coefficient with capillary number, Ca, for two gaps: (a) 65 μm, and (b) 110 μm obtained by Moriyama et al. [113], redrawn from the original plot.
instability. The constants derived are for their specific setup geometry, and further generalization is needed to extend this concept to cover other system configurations and test fluids. Garrity et al. [164] studied the Ledinegg type instability in a 1 mm × 1 mm minichannel using 56 parallel channels in a stainless steel plate for a fuel cell cooling application using HFE-7100. The pressure drop versus flow rate was modeled and the instability region was predicted, with good agreement with the experimental data.

Zhang et al. [165] presented a detailed model to predict Ledinegg instability in microchannels using the available correlations for pressure drop in developing the pressure drop versus mass flow rate supply and demand curves. By comparing the slopes of the two curves, onset of instability was predicted. Using the model, they identified the ratio of kinematic viscosities of liquid to vapor phases as an important parameter. According to their findings, lower values of this ratio for HFE-7100 leads to a more stable operation compared to water. Effects of other parameters on instability were similar to those observed for larger systems. In another recent paper, Zhang et al. [166] proposed an active oscillatory mass flow controller to reduce the flow oscillations. Such dynamic control seems very promising to control oscillatory system behavior in systems where reliable operation is of critical importance.

CHF. CHF during flow boiling in microchannels is a topic of great interest as it limits the heat fluxes that can be safely and reliably dissipated. A number of studies have reported experimental data for CHF in microchannels. A detailed list of these data sources may be found in recent papers on CHF modeling using large databases [167,168]. The CHF in microchannels is clearly affected by the presence of instabilities, and large spread is observed in the data. This scatter is believed to be due to CHF values being reported under both stable and unstable operating conditions. As pointed out by Bergles and Kandlikar [14], the instabilities are responsible for lower CHF values.

A number of correlations have been proposed in the literature for CHF in microchannels. In general, they were developed with a small data set and provided useful insight into the CHF mechanism [169,170]. However, they had limited success in predicting other data sets that were not employed in the correlation development. Two correlations developed for macroscale applications by Katto and Ohno [171] and Shah [172], however, were found to predict the CHF generally within 20–40% mean absolute error. This is quite remarkable considering that the underlying data sets in their correlation development used tubes larger than 3 mm in diameter. For example, R-123 data by Kosar and Peles [173] was correlated within 24.7% mean absolute error by the correlation of Katto and Ohno. Park and Thome [174] report errors between 23% and 60% with this correlation. The success of these general correlations also shows the importance of utilizing large number of data sets in correlation development. However, Roday and Jensen [175] report that the correlation of Katto and Ohno resulted in large overprediction for \( D = 0.430 \) mm and 0.700 mm, and smaller overprediction with \( D = 0.286 \) mm tubes. In general, the correlation of Katto and Ohno underpredicted, while the correlation of Shah overpredicted the experimental CHF data. On the other hand, specific correlations have been developed in the literature based on the researchers’ own experimental data, e.g., Refs. [170,173,174,176,177]. They are able to correlate the parent data set well but generally do not fare as well with other data sets.

Recently, utilizing a scaling analysis [178], Kandlikar [179] proposed a CHF model that shows that the length-to-diameter (\( L/D \)) ratio acts as a step-function to yield high-CHF and low-CHF regions. The CHF was considered to be dependent on the local parameters and was a result of the vapor pushing under the liquid at the wall resulting in a vapor cut-back. The hydrodynamic instability resulting from this force was proposed by Palmer [180] in 1976 and used recently by Sefiane et al. [181] in modeling the interface instability during pool boiling. The new CHF model [179] utilizes Weber number, capillary number (earlier proposed by Moriyama [113] in film thickness modeling), and a new nondimensional number \( K_2 \) representing the ratio of the force due to momentum change resulting from evaporation at the interface and the surface tension force

\[
K_2 = \left( \frac{q}{h_{fg}} \right)^2 \left( \frac{D_k}{r_i \sigma} \right)
\]

The step-function form dependence on the \( L/D \)-ratio is believed to be linked to the instability during flow boiling. A higher \( L/D \)-ratio leads to a lower CHF, while short tubes with a lower \( L/D \)-ratio yields a higher CHF. The CHF correlation [179] covers a wide range of parameters: hydraulic diameter from 0.127 mm to 3.36 mm, quality from 0.003 to 0.98, \( \text{We} \) from 1.7 to 86,196, \( Ca \) from \( 10^{-5} \) to \( 10^{-3} \), and \( L/D \) from 10 to 352. A comparison of the model prediction with the experimental data is shown in Fig. 11. The parametric trends among Weber number, capillary number, and the new parameter \( K_2 \) reported in Ref. [179] provide a fundamental understanding of how inertia, viscous, and surface tension forces affect the CHF phenomenon at macroscale and microscale. In general, surface tension forces are dominant at very small channel sizes and lower mass fluxes. As the channel size or mass flux increases, viscous terms (\( Ca \)) become important, and finally for larger channel dimensions, or at higher mass fluxes, inertia forces (\( \text{We} \)) are dominant.

Challenges and Future Research Needs in Flow Boiling. The main challenges facing flow boiling in microchannels are in providing reliable system operation and a high value of heat transfer coefficient. Some of the research needs were presented in the discussion above. Some of the important areas are outlined below.

Heat Transfer. Careful experiments to generate reliable data under stable operating conditions and under controlled uncertainty conditions are needed. One of the major concerns is the validity of the flow boiling data. It is strongly recommended that all flow boiling experimental setups be first validated with the single-phase experiments before generating flow boiling data. Further, the experimental data is needed to represent the realistic operating conditions. For example, microchannels may be operated with subcooled entry or two-phase entry (such as in the evaporator of a
refrigeration system). The experimental data for the latter condition is severely lacking.

Instability. The work on instability modeling in microchannels is receiving increased attention in the last 4 years. The analysis presented by Zhang et al. [165] provides a useful direction. Future work is suggested to build on these concepts and integrate the actual system operation in the modeling that would include the upstream compressibility effects, explosive boiling, and other types of instabilities, including parallel channel and density wave instabilities. Active control to provide stable operation has been recently proposed [166]; however, research on developing passive techniques would be highly desirable from operational and cost standpoints. Use of infrared imaging provides a valuable tool in evaluating the transient processes inside channels without the need for optically transparent window [182,183].

Modeling. It is recommended that the transient nature of flow, heat transfer, and conjugate effects resulting from the wall thermal characteristics be included in developing more accurate models. The dynamic processes associated with the nucleation and rapidly moving interfaces will require some new techniques to replace the use of steady-state conduction and fully developed flow equations being employed in current models. Use of nondimensional parameters is strongly recommended to provide an understanding of the underlying mechanisms.

Enhancement. Enhancement in the flow boiling heat transfer coefficient is essential before this mode can be implemented in practical devices. Recognizing that the single-phase flow with water in an offset strip-fin geometry resulted in h values of around 500,000 W/(m² K) [50,70], it is reasonable to expect comparable or higher h values to justify the additional complexities introduced by the flow boiling systems. Effects of nanostructures on pool boiling are being investigated currently by many researchers, and their application in flow boiling needs to be examined.

CHF. Unstable operating conditions result in lower CHF values in microchannel flow boiling. This aspect needs to be further explored. Controlling unstable operation through active control for enhancing CHF offers a reliable technique [166] and should be pursued further.

Book Publications and Specialty Focused Conferences

A few books dedicated to microchannel fluid flow and heat transfer have been published in the literature. The first book by Kim et al. [184] focuses on the minichannel ranges and provides basic equations that are helpful in designing compact heat exchangers utilizing these channels. The second book by Zhang et al. [185] focuses on the silicon heat sinks, providing the reader with the fundamental aspects of the silicon cooling system design for applications as well as research. They confirm the validity of the Hu and Graham [186] nucleation model and provide performance characteristics of various coolers in two phase flow. The third book by Celata [187] provides an excellent survey of the research reported in the literature on single-phase and two-phase flows. It includes contributions from several experts in different fields. The single-phase liquid flow and two-phase adiabatic flow in minichannels and flow boiling in minichannels are covered in greater detail. The fourth book by Kandlikar et al. [188] is published in 2006 and covers gaseous flows, liquid flows, boiling, and condensation, along with some biological applications. The treatment in the single-phase liquid flow is on understanding the fundamentals with specific equations for entry region flows, roughness effects, and microchannel system design for chip cooling application. The coverage on boiling deals with the fundamentals of the boiling phenomena. The fifth book by Yarin et al. [189] provides a detailed coverage of literature related to single-phase and two-phase cooling systems. Special focus is on the single-phase and two-phase flows in heated capillaries. The sixth book by Kirby [190] focuses on fluid flow in microfabricated and nanostructured systems.

A number of specialty conferences are devoted to heat transfer and fluid flow in microchannels. The following list is not exhaustive but provides an overview of the scope of activities worldwide:

- Engineering Foundation Conference on Boiling Heat Transfer, since 1992, currently under ECI;
- ASME International Conference on Nanochannels, Microchannels and Minichannels, held annually since 2003;
- ECI International Conference on Transport Phenomena in Micro and Nanodevices, since 2004;
- ECI International Conference on Heat Transfer and Fluid Flow in Microscale, since 2005;
- ASME Micro/Nanoscale Heat and Mass Transfer International Conference, since 2007;
- Micro and Nano Flows Conference, since 2006 (Initially sponsored by IMechE/Royal Society of Edinburgh Colloquium);

In addition to the above specialty conferences, Mechanical Engineering societies in a number of countries (such as ASME, JSME, KSME, etc.) have a number of sessions devoted to the fluid flow and heat transfer at microscale.

Concluding Remarks

The advantages of using microscale passages in improving heat transfer have propelled research in this area. This subject will continue to receive attention as the focus shifts to further enhancing heat transfer, reducing pressure drop, and implementing these processes in practical devices. We are witnessing a paradigm shift that is expected to yield significant reduction in equipment sizes of some conventional heat transfer systems as well and introduce new products utilizing microscale technologies.

In single-phase liquid flow in microchannels, it is now well accepted that the continuum models are applicable. The reduction in scale is seen as an important consideration while considering the near wall region. Implementing nanofluids in the systems using microchannels is seen to provide significant changes in the performance. It is becoming clear that microchannels are gateways to flow modification through such nanoscale surface enhancement configurations. Enhancing heat transfer in single-phase flow through novel techniques will continue to be a high priority research endeavor in this area.

Flow boiling in microchannels is receiving renewed interest as the challenges for achieving stable operation are being overcome through a fundamental understanding of the underlying processes. Developing simple and inexpensive systems for operating flow boiling systems under stable conditions will be an important topic of research. Enhancing flow boiling heat transfer coefficient through novel techniques will continue to receive attention. Implementing some of the enhancement techniques that have been proven to be effective in compact heat exchanger passages (minichannels) will be a high priority area for research. Application of nanocoatings on the channel walls is seen to be another area for research in reducing the wall superheat at incipience and increasing CHF.

Another aspect of microchannel research that cannot be overlooked is the potential strides that can be made through cross-disciplinary implementation of ideas from the fields of micromixers, biological sciences, and microfluidic devices used in drug delivery and diagnostic applications.

Needless to say, we all should be thinking of the nanofluidic frontier that promises to provide even more exciting opportunities for heat transfer and fluid flow researchers.
Nomenclature

Roman Letters

\( C_a \) = capillary number

\( D_{in}, D_{out} \) = diameter, hydraulic diameter, m

\( f \) = friction factor

\( h \) = heat transfer coefficient, W/(m²°C)

\( h_g \) = latent heat of vaporization, J/kg

\( j \) = volumetric flux, m/s

\( q \) = heat flux, W/m²

\( Re \) = Reynolds number

\( V \) = mean flow velocity, m/s

\( We \) = Weber number

\( \alpha \) = quality

Greek Letters

\( \epsilon \) = roughness parameter

\( \mu \) = viscosity, Pa·s

\( \rho \) = density, kg/m³

\( \sigma \) = surface tension, N/m

Subscripts

\( q \) = corresponding to smooth wall

References


