Heat Transfer and Fluid Flow in Microchannels—2012 Status and Research Needs

Heat transfer and fluid flow in microchannels have been topics of intense research in the past decade. A critical review of the current state of research is presented with a focus on the future research needs. After providing a brief introduction, the paper addresses six topics related to transport phenomena in microchannels: single-phase gas flow, enhancement in single-phase liquid flow and flow boiling, flow boiling instability, condensation, electronics cooling, and microscale heat exchangers. After reviewing the current status, future research directions are suggested. Concerning gas phase convective heat transfer in microchannels, the antagonist role played by the slip velocity and the temperature jump that appear at the wall are now clearly understood and quantified. It has also been demonstrated that the shear work due to the slipping fluid increases the effect of viscous heating on heat transfer. On the other hand, very few experiments support the theoretical models and a significant effort should be made in this direction, especially for measurement of temperature fields within the gas in microchannels, implementing promising recent techniques such as molecular tagging thermometry (MTT). The single-phase liquid flow in microchannels has been established to behave similar to the macroscale flows. The current need is in the area of further enhancing the performance. Progress on implementation of flow boiling in microchannels is facing challenges due to its lower heat transfer coefficients and critical heat flux (CHF) limits. An immediate need for breakthrough research related to these two areas is identified. Discussion about passive and active methods to suppress flow boiling instabilities is presented. Future research focus on instability research is suggested on developing active closed loop feedback control methods, extending current models to better predict and enable superior control of flow instabilities. Innovative high-speed visualization and measurement techniques have led to microchannel condensation now being studied as a unique process with its own governing influences. Further work is required to develop widely applicable flow regime maps that can address many fluid types and geometries. With this, condensation heat transfer models can progress from primarily annular flow based models with some adjustments using dimensionless parameters to those that can directly account for transport in intermittent and other flows, and the varying influences of tube shape, surface tension and fluid property differences over much larger ranges than currently possible. Electronics cooling continues to be the main driver for improving thermal transport processes in microchannels, while efforts are warranted to develop high performance heat exchangers with microscale passages. Specific areas related to enhancement, novel configurations, nanostructures and practical implementation are expected to be the research focus in the coming years. [DOI: 10.1115/1.4024354]

Keywords: microscale thermal transport, heat transfer, single-phase flow, rarefaction effects, slip flow, flow boiling, enhancement, instabilities, condensation, review, research needs

1 Introduction

As has been observed in natural systems such as lungs and kidneys in humans and other living animals, transport processes become more efficient at microscale passage dimensions. The increase in the area/volume ratio of the passages and the change in the relative importance of different forces create a new class of transport processes that can lead to significant size reductions in practical devices utilizing these processes. Although these benefits have been well-known, as demonstrated by the pioneering work of Tuckerman and Pease [1], significant research in this area started only after their successful application in high heat flux electronics cooling and microfluidic devices. The definition of microscale is somewhat flexible, depending on the dimensions at which the transport characteristics are affected. In general, channel dimensions of 10–200 μm constitute the bulk of the microscale focus; larger dimensions of up to 1–3 mm are also of interest in some applications such as single-phase liquid flow, boiling and condensation.

The resurgence in microscale transport research started in its earnest in the first decade of the new century, marking the period 2000–2012 for a large number of publications dealing with thermal transport in microchannels. This was aided by a number of dedicated conferences, such as ASME International Conference on Nanochannels, Microchannels and Minichannels started in 2003, and the European MicroFlu conference on microfluidics started in 2008 highlighting the increased research activity worldwide in this area.
This paper is aimed at capturing the recent advances in the field of microscale convective transport, focusing on four main topics: single-phase gas flows, single-phase liquid flows, flow boiling, and flow condensation. A brief survey of the current status is presented, followed by a critical evaluation of research opportunities in the respective areas. Two specific applications, electronics cooling and microscale heat exchangers, are considered in more detail as they seem to be the most promising candidates benefitting from microscale transport process advancements. The need to shift the research focus to a more fundamental approach is highlighted in recognizing some of the unique characteristics of the microscale phenomena, and utilizing them in further enhancing the thermal transport processes.

### 2 Single-Phase Gas Flow in Microchannels

A large number of recent investigations have been undertaken to analyze convective heat transfer in gas microflows addressing various new applications connected to the increasing development of MEMS. Gas microflows with heat transfer occur, for example, in microscale heat exchangers, pressure gauges, mass flow and temperature microscale sensors, micropumps and microsystems devoted to mixing or separation for local gas analysis, mass spectrometers and Knudsen micropumps. The main elements of these microsystems which are subject to heat transfer are long or short, possibly bent, microchannels with various possible cross-sections.

In these microchannels, due to rarefaction effects, a local thermodynamic nonequilibrium is observed within the Knudsen layer close to the wall. It results in a velocity slip and a temperature jump at the wall, with thermal antagonist effects: compared with the classic behavior in microchannels, the first phenomenon increases, whereas the second one decreases, convective heat transfer. In addition, a third mechanism has to be taken into account in microchannels: shear work due to gas slip at the wall increases the contribution of viscous heating to the convective heat transfer.

#### 2.1 Specificities of Heat Transfer in Gas Microflow

2.1.1 Rarefaction and Slip Flow Regime. In addition to compressibility, rarefaction of the flow in these microchannels has a significant impact on heat transfer. Shrinking down the dimensions of fluidic microsystems under an internal gas flow leads to an increase of the Knudsen number

\[
\text{Kn} = \frac{\lambda}{D_h}
\]

defined as the ratio of the mean free path, \(\lambda\), of the molecules over the hydraulic diameter \(D_h\) of the microchannel. The Knudsen number encountered in typical microsystems is frequently between \(10^{-3}\) and \(10^{-1}\), which is the typical range of the well-known slip flow regime [2]. In this moderately rarefied regime, the Navier-Stokes equations are still valid in the bulk flow, but a local nonequilibrium is observed within the Knudsen layer. This thin wall layer, the thickness of which is on the order of the molecular mean free path, experiences complex nonlinear phenomena that result in velocity slip, temperature jump and thermal creep at the wall [3], which strongly influence convective heat transfer.

2.1.2 Boundary Conditions and Heat Transfer. Several kinds of boundary conditions are proposed in the literature to take into account this nonequilibrium near the wall surface [4]. The main ones are based on the Maxwell-Smoluchowski conditions

\[
u - u_w = \sigma_T \frac{\partial u}{\partial n} \bigg|_w + \frac{\sigma_T \mu R}{P} \frac{\partial T}{\partial t} \bigg|_w
\]

In Eq. (2), the difference between the fluid velocity at the wall, \(u\), and the wall velocity, \(u_w\), depends on both the velocity and temperature gradients. The first term of the RHS is the velocity slip which results from the normal gradient, \(\partial u / \partial n\), of the stream-wise velocity and also from the tangential gradient, \(\partial v / \partial t\), of the stream-wise velocity, which should be taken into account in case of curved surfaces. The viscous slip coefficient, \(\sigma_T\), is in practical cases of the order of unity and it depends on the wall nature and roughness, as well as on the gas species.

The thermal creep effects due to the tangential temperature gradient, \(\partial T / \partial t\), are taken into account by the second term of the RHS of Eq. (2). The thermal slip coefficient, \(\sigma_T\), is close to \(3/4\) for specular reflection and of the order of unity in case of diffuse reflection. A detailed discussion on the values of \(\sigma_T\) and \(\sigma_T\) can be found in Ref. [5]. The temperature jump is proportional to the normal gradient, \(\partial T / \partial n\), of the temperature and can be calculated by Eq. (3).

#### 2.2 Main Theoretical and Experimental Results.

A large number of recent papers have been devoted to the analysis of pressure driven flows of gases with heat transfer in straight microchannels with constant cross-section, focusing on the determination of the Nusselt number. Most of them concern circular or parallel-plate microchannels, and to a lesser extent, microchannels with rectangular, trapezoidal and triangular sections, frequently encountered in fluidic microsystems. In these studies, the flow is assumed laminar and the regime is the slip flow regime. Investigations relate to thermally fully developed and thermally developing flows, with uniform heat flux (UHF) or constant wall temperature (CWT) conditions. The most significant contributions and results are summarized in Secs. 2.2.1 to 2.2.3.

2.2.1 Analytical Solutions for Simplest Configurations. Analytical solutions can be derived in the simplified case of a fully laminar developed flow in a straight microchannel, assuming incompressible fluid with constant properties and without taking into account thermal creep and viscous heating. For a flow in a circular (C) microtube with UHF, this analytical solution reads [6]

\[
\text{Nu}_{(C\text{-UHF})} = \left( \text{Kn} \sigma_T \frac{11 + 128 \text{Kn} \sigma_T + 384 (\text{Kn} \sigma_T)^2}{48 (1 + 8 \text{Kn} \sigma_T)^2} \right)^{-1}
\]

For the same problem with CWT, the analytic solution has no explicit formulation, but its resolution leads to tabulated results [7]. For a flow in a parallel-plate (PP) microchannel with one insulated wall and one heated wall with uniform heat flux (UHF-I1W), the analytical solution with the same simplifying hypothesis reads:
Nusselt number was deduced assuming that the heat flux at the wall \( \varphi_w \) is fully transmitted by conduction in the fluid. However, in case of slip flow, the heat transfer at the wall should be modified to include the shear work done by the slipping fluid and should read [14]

\[
\varphi_w = k \frac{\partial T}{\partial n} + \mu \frac{\partial u}{\partial n}|_w
\]

This condition has rarely been implemented in slip flow heat transfer analysis. Miyamoto et al. [15] analyzed the effect of viscous heating in UHF heat transfer between parallel plates in the slip flow regime using Eq. (10) and obtained

\[
\text{Nu}_{\text{PP, UHF, Br} \neq 0} = \left( \frac{9Br_\varphi}{C_3} \left( \frac{3 + 84\sigma_rKn + 560(\sigma_rKn)^2}{35C_3} + 8\sigma_r\xi_TKn^2 \right) \right)^{-1} - 1
\]

This equation gives the same results as Eq. (9) in the case of no rarefaction (\( Kn = 0 \)) and in the case of no viscous dissipation (\( Br_\varphi = 0 \)), which was expected as shear work in the Knudsen layer scales with the Brinkman number [16]. On the other hand,
when rarefaction and viscous heating are both taken into account, the influence of rarefaction is stronger when shear work is taken into account (Fig. 1(b)). Comparing with macrochannels, the decrease of hydraulic diameter leads to a decrease of convective heat transfer efficiency (\( \text{Nu} \) decreases as \( \text{Kn} \) increases) but the work of shear stress due to slip at the wall—which is observed only in the case of microchannels—increases the role of viscous dissipation and results in a higher heat transfer in the case of cooled surface.

Developing flows have also been analyzed by a number of authors who have treated the so-called extended Graetz problem [6]. In some papers, such as Refs. [12,13], the axial conduction (finite Peclet number) is taken into account in addition to viscous dissipation. The influence of the variation of gas properties (\( \mu, k \)) within the cross-section due to temperature variations has been considered by Hooman et al. [17] who observed negligible effect on \( \text{Nu} \) in the UHF case in a microtube.

#### 2.2.3 Experimental Studies

There are very few published experimental data on convective slip flow heat transfer in microchannels. Some experiments have revealed very low Nusselt numbers, compared to the expected ones, but the huge discrepancy between experiment and theory can be explained by experimental errors discussed in Ref. [18]. The most accurate available data concern flows between parallel plates. The group of Miyamoto measured the surface temperature distribution of a slip choked flow of low-density air through a millimeter-size parallel-plate channel with adiabatic walls [19] and with uniformly heated walls [15]. In the configuration with uniform heat flux at the wall, the relative deviation between the measured and calculated wall temperature distribution was very small, of the order of 1% or 2% for the reported data, with an increased discrepancy for increasing Knudsen number. The experimental Knudsen number, however, was limited to values lower than \( 5 \times 10^{-7} \) at the inlet and \( 5 \times 10^{-7} \) at the outlet for the reported data and in this range slip flow and temperature jump effects are not very significant. In the configuration with adiabatic walls, a few data were reported for higher Knudsen numbers, up to \( 6 \times 10^{-3} \) at the inlet and \( 6 \times 10^{-2} \) at the outlet, well in the slip flow regime. There was a very good agreement with the simulated wall temperature distribution, taking into account shear work at the wall. On the contrary, when this shear work was not taken into account, the wall temperature was underestimated with a deviation of a few percent. This analysis confirms that Eq. (10) should be used and that most of the analytical and numerical models proposed in the literature for convective heat transfer in gas microflows should be corrected to take into account the shear work at the wall.

### 2.3 Single-Phase Gas Flow—Research Needs

The literature on slip flow heat transfer is now well stocked, with a series of available analytical solutions or numerical analysis for various heat conditions and microchannel geometries. The limits of the analytical solutions, based on simplifying assumptions, are not, however, always well documented. Concerning the influence of gas properties temperature dependence on heat transfer, there are still few studies in the slip flow regime [20]. When viscous dissipation has to be taken into account, the effect of shear work at the wall is generally not considered, and most of the analytical solutions should be extended to include its non-negligible effect. In addition, the thermal creep is not analyzed, except in a few numerical papers [21]. On the other hand, although the theory of gas hydrodynamics in the slip flow regime is now supported by smart experiments, there is a crucial lack of experimental data concerning heat transfer in this regime. Providing accurate experimental data on slip flow heat transfer is a challenge for the next years, which would allow a real discussion on the validity of velocity slip and temperature jump boundary conditions, on the appropriate values of accommodation coefficients, as well as on the limits of applicability of slip flow theory, in terms of degree of rarefaction. Some promising emerging techniques such as MTT and interferometry should be developed for this analysis of heat transfer in gas microflows. Up to now, these techniques are successfully used for thermography in liquids microflows [22,23] and their application to gas microflows is a challenging goal. Preliminary results detailed in Ref. [22] give high hopes that MTT could be soon implemented for successful analyze of temperature fields in gas microflows. Even if the technique is limited to millimeter-size channel sections, due to the minimal achievable size of the laser beam—typically a few tens of micrometers—, the same flow and heat transfer configurations than those observed in microchannels can be obtained by reducing the pressure. Current research, for example, at the Stokes Research Institute, also shows that interferometry could be an operative tool for gas thermometry in a couple of years.

### 3 Enhancement in Single-Phase Flow and Flow Boiling in Microchannels

The basic liquid flow behavior at microscale remains unchanged for as small as 10 \( \mu \)m passage dimensions. The liquid molecules are closely packed together compared to the channel dimensions and the equations developed for macroscale applications, such as Navier-Stokes equation and convective heat transport equations are still applicable. During flow boiling, however, the bubble sizes are quite comparable to the channel dimensions and are responsible for changes in the two-phase flow behavior and thermal characteristics. The main issue has been instability, which is being addressed in Sec. 4. The overall progress made in the area of liquid flow and flow boiling in microchannels in the last three decades has been reviewed recently by Kandlikar [24]. Recent publications indicate that there is a need for further enhancing heat transfer in microchannels. This section is therefore devoted to reviewing the literature in this field and identifying specific research directions.

#### 3.1 Single-Phase Liquid Flow

Among the excellent reviews available on this topic, the work by Goodling [25], Sobhan and Garamella [26], Palm [27], and Guo and Li [28] are noteworthy. By 2005, it was well accepted that the liquid flow in microchannels follows the continuum theory, and the discrepancies were mainly attributed to the experimental errors, axial conduction effects, roughness effects. A recent exhaustive review of the early work on heat transfer during single-phase flow is given by Kandlikar [24].

In addition to the work on fundamental aspects of friction factor and heat transfer at microscale, considerable progress has been made in applying the high heat removal capability of microchannels with liquid flow. Microchannels are sought after in high heat flux dissipation applications. A heat flux in excess of 1 kW/cm\(^2\) is anticipated in advanced electronic cooling applications. To meet this cooling demand, employing single-phase liquid with an average temperature difference of 20 °C would require a heat transfer coefficient of 500,000 W/m\(^2\)°C. Thus, heat transfer enhancement has been identified as a major research thrust for future research.

Microchannels used in heat transfer applications with single-phase liquid flow are generally operated in the laminar region. Heat transfer coefficient in the fully developed laminar region is inversely related to the channel hydraulic diameter. Under a constant heat flux boundary condition, in order to obtain the desired heat transfer coefficient of 500,000 W/m\(^2\)°C, a hydraulic diameter of around 5 \( \mu \)m is needed with a plain microchannel. However, the pressure drop increases as \( D^2 \) and an extremely large pressure drop would result with this configuration. The need for enhancement within microchannels was identified earlier in 2004 by Kandlikar and Grande [29], who proposed different enhancement features in silicon substrates. Employing enhancement features in microchannels offers a practical solution to meet this high heat flux dissipation demand. Tuckerman [30] in 1984 had generated tall pin fins of aspect ratios 8:1 by using precision
sawing to enhance the performance of microchannels. As the heat transfer coefficient increases over the fin surface, fin efficiency decreases with fin height, and longer fins may not be desirable. A good review of various enhancement studies is given by Tullius et al. [31]. Steinke and Kandlikar [32] presented a list of enhancement techniques applicable in microchannels. Figure 2 shows one of their proposed designs with cross mixing of adjacent liquid streams. Some of the other promising designs include surface roughness, flow disruptions, and secondary flows. The two enhancement techniques that have received attention in the recent literature are fins and roughness elements. These are reviewed in greater detail.

Fabrication techniques for the enhancement features have been developed for low cost production. A review of fabrication techniques for silicon substrates is provided by Kandlikar and Grande [29], and Jasperson et al. [33] provide a detailed review for metal substrates.

3.1.1 Fins in Single-Phase Liquid Flow. Fins increase the available heat transfer surface area and are used extensively in macroscale heat transfer applications. Tuckerman [30] first employed high aspect ratio pin fins to enhance microchannel performance. A number of researchers have recently applied this geometry in microchannels. Kosar and Peles [34] studied circular pin fins, 99.5 \( \mu \text{m} \) in diameter and placed at a pitch of 150 \( \mu \text{m} \) in a staggered configuration on a silicon chip inside an 1800 \( \mu \text{m} \) wide and 10 mm long channel. A heater was placed on the backside of the silicon chip. Using water flow, they obtained a maximum heat transfer coefficient of around 55,000 W/m²°C. John et al. [35] compared the performance of square and circular pin fins and noted that the figure of merit, defined as the heat transfer performance divided by pressure drop penalty, was higher for circular pin fins at Re < 300, while reverse was true at higher Re. Vanapalli et al. [36] studied additional fin shapes of ellipse and diamond in gas flow, but these have not been studied in liquid flows. Lee et al. [37] studied their patented parallelogram cross-sectional fins with the side walls parallel to the flow and the oblique front and rear sides facing at an angle. Their fin geometry is depicted in Fig. 3. This design is similar to that shown in Fig. 2, except the secondary flow is induced in only one direction. This geometry was implemented on a copper cooler base of 25 mm × 25 mm with a fin height of 1.2 mm and a channel width of 300 \( \mu \text{m} \). The secondary flow induced in the oblique interconnecting passages resulted in an eighty percent enhancement in heat transfer with little pressure drop penalty. However, the heat transfer coefficient was relatively low due to the large hydraulic diameter employed. Implementing this geometry at microscale seems to be attractive.

Prasher et al. [38] studied various square and circular pin fin geometries of diameters (or side walls for square fins) ranging between 55 and 153 \( \mu \text{m} \). Their results indicate that the area enhancement combined with the improvement in heat transfer coefficient provided a maximum enhancement factor of 25 as compared to the channel without any fins. They presented correlations for heat transfer and friction factor as a function of geometrical parameters and flow rate.

Among other fin shapes, offset strip fins have been widely utilized for heat transfer enhancement. Colgan et al. [39] used this geometry in their silicon chip cooler to dissipate over 500 W/cm² with water and over 250 W/cm² with a fluorinated liquid. The fin length was 500 \( \mu \text{m} \) with a pitch of 50–100 \( \mu \text{m} \). The heat transfer coefficients for these fins were measured by Steinke and Kandlikar [40] and were seen to be one of the highest reported (in excess of 500,000 W/m²°C). An SEM image of one of the fin geometries used in these two studies is shown in Fig. 4.

From the literature discussed above, it is seen that the offset strip-fin geometry provides significant heat transfer enhancement in microchannels. The increased pressure drop due to the tortuous path followed by the liquid has been countered by reducing the flow length with multiple inlet and outlet headers [39]. This multiple header concept was originally suggested by Tuckerman [30] with alternate input and output headers that were placed along the flow length in the cover plate over the microchannels. It is also noted that the similarity between microscale and macroscale flows provides an opportunity to explore the fin geometries that have
been successful in macroscale applications be investigated for microscale liquid flow systems. The added constraints due to manufacturability and more stringent pressure drop limitations may be applied as additional selection criteria.

Another important aspect in the cooling of electronic devices is the uniformity of temperature along the coolant flow path. As the coolant flows, its temperature rises, which in turn causes the substrate temperature to rise almost in the same proportion since the heat transfer coefficient is uniform along the flow length from inlet to outlet (except in the entrance region). To reduce the temperature variation, Rubio-Jimenez et al. [41] numerically investigated a microscale pin-fin structure with increasing fin density along the flow direction. The resulting increase in the surface area and heat transfer coefficient compensate for the decrease in the available temperature difference between the fluid and the substrate along the flow direction, and a more uniform substrate temperature is obtained as shown in Fig. 5. Here the fin density is changed twice along the flow length in the MF-50 \( \times 100 \times 200-66 \) fin structure with variable fin density. For comparison, the temperature variation in a conventional microchannel and an open gap of the same height without the fins are also shown. The variable fin density design also offers a lower pressure drop as higher fin density is employed only toward the exit where the temperature difference is smaller.

### 3.1.2 Roughness in Single-Phase Flow

Surface roughness to improve heat transfer at microscale is a relatively simple technique. Preliminary work on the effect of uniform roughness was conducted by Kandlikar et al. [42]. Stainless steel tubes were chemically etched to produce different surface roughness. The results showed that for a 0.62 mm diameter circular tube, a surface roughness of 0.355% enhanced heat transfer by about 50% over a smooth tube with little pressure drop penalty in the Re range from 900 to 2800. The effect of roughness on friction factor in laminar and turbulent flows was investigated by Kandlikar et al. [43], who proposed a modified Moody diagram based on the concept of constricted flow diameter. Weaver et al. [44] conducted an experimental study with rectangular channels 500 \( \mu \text{m} \) high with a roughness height of 6.1 \( \mu \text{m} \) resulting from their manufacturing process. They noted that the heat transfer was enhanced by only a factor of 1.1 to 1.2, while friction factor increased by a factor of 2.2 in the Re range 1000–3000. Lin and Kandlikar [45] conducted a systematic study to investigate the effect of structured roughness on heat transfer and pressure drop in a 10 mm wide channel with a gap of 1 mm. Figure 6 shows a surface profile that was generated on the stainless steel sidewalls. For a roughness element height of 96.3 \( \mu \text{m} \) shown in Fig. 6, an enhancement of about 1.8 was obtained in the laminar region over a plain surface. However, transition to turbulent flow occurred earlier and a significantly higher heat transfer coefficient of 150,000 W/m\(^2\)C was obtained. This geometry represents one of the best performing enhancement techniques available in the literature (regular twisted tapes, coiled wire insert, alternate clockwise and anticlockwise twisted tapes, and porous media insert). Figure 7 shows the heat transfer performance comparison of three roughness profiles studied by Lin and Kandlikar [45]. Structured roughness provides a very effective way for enhancing heat transfer at microscale. A number of numerical studies are available that provide further insight, e.g., Refs. [46,47], but are not reviewed here due to space constraint. Further research on designing specific roughness features to match the fluid properties and operating flow rate is warranted.

### 3.2 Research Needs in Single-Phase Liquid Flow in Microchannels

Although efforts are needed for developing a fundamental perspective of microscale heat transfer, enhancing heat transfer in microchannels is of critical importance in successful development of microscale thermal devices. The heat transfer coefficient needs to be further increased, while dramatic reductions in pressure drop are warranted. This leads to another concern regarding the small area available for manifolds. Another area recommended for research is the development of high performance/large heat duty heat exchangers at competitive costs using microscale passages to replace those using macroscale passages.
3.3 Flow Boiling in Microchannels. Since boiling phenomenon is in general considered to be more efficient than the single-phase flow due to the large latent heat transfer at a constant temperature in a liquid, the advantages have not yet been realized in microchannels. Some of the fundamental work on flow boiling was presented by Tuckerman and Pease [1] in 1981, followed by Motyjama et al. in 1992 [48]. Bowers and Mudawar [50] in 1994 for micromachans, and Hetronini [51] in 2000. The heat transfer coefficients were reported to be low during flow boiling in microchannels by Steinke and Kandlikar [52] in 2004. In 2001, Kandlikar [53,54] visualized the explosive bubble growth that is responsible for instability and low critical heat flux. High-speed visualization was employed by a number of researchers to understand the link between flow patterns and heat transfer performance, e.g., Schilder et al. [55] and Harirchian and Garimella [56] and Cheng et al. [57]. Although these efforts have shed considerable light on the two-phase flow behavior, a clear path toward enhancing the heat transfer performance still remains elusive. Recent work by Kandlikar [58] shows the similarities between pool flow boiling and flow boiling in microchannels, while it is seen that the enhanced pool boiling with an open microchannel surface by Cooke and Kandlikar [59,60] has significantly outperformed flow boiling in microchannels with water. This indicates an immediate need to focus future research on flow boiling enhancement in microchannels. The complexity of a flow boiling system cannot be justified when the performance of a simpler pool boiling system or a single-phase system is superior. In Secs. 3.3.1 and 3.3.2, two enhancement techniques will be reviewed to evaluate their potential in enhancing heat transfer coefficient and CHF.

3.3.1 Fins With Flow Boiling. Fins have been used extensively in flow boiling application at macroscale. Krishnamurthy and Peles [61] employed 100 μm circular pin fins in 250 μm tall microchannels. They obtained a maximum heat transfer coefficient of around 75,000 W/m²°C. Their video images indicated the presence of vapor patches which led to premature CHF. Lu and Su-Ho [62] studied 200 μm square pin fins in 670 μm high microchannels with water. A maximum heat flux of 248.5 W/cm² was attained with a mass flux of 417 kg/m²s. The highest heat transfer coefficient obtained was 125,000 W/m²°C at lower qualities (vapor fractions). Xue et al. [63] employed 30 μm square pin fins in 60 μm high silicon microchannels. They obtained a CHF of around 65 W/cm² with FC-72. The flow boiling heat transfer coefficients improved over pool boiling with increasing flow velocities. Although fins are attractive in macroscale flow boiling applications, the existing studies show that they are not able to compete with the high heat transfer coefficients obtainable with single-phase liquid flow. Further research is warranted on developing new fin geometries that will provide stable high performance. High heat flux and high heat transfer coefficient) under flow boiling conditions. Such studies are needed for both silicon and copper substrates because of their application in electronics cooling and high performance heat exchangers, respectively.

3.3.2 Nanostructures With Flow Boiling. Nanowires have been shown to enhance pool boiling due to their improved surface wettability characteristics. Khanikar et al. [64] employed carbon nanotubes (CNT) on copper microchannel surfaces. They found a modest improvement in heat transfer at low heat fluxes, but only a marginal improvement in heat transfer characteristics at high heat fluxes. Since CNTs yielded a hydrophobic surface, the nucleation characteristics improved but the large contact angles adversely affected the performance at higher heat fluxes. Similar results were obtained by Singh et al. [65] with CNTs on silicon substrates. Recently, Kousalya et al. [66] employed ultraviolet light on the CNT structures and reported an improvement in heat transfer performance with a 4.6°C reduction in the wall superheat at a heat flux of around 50 W/cm². The ultraviolet light was utilized to make a CNT surface hydrophilic as originally reported by Takata et al. [67]. Morshed et al. [68] employed carbon nanowires on silicon substrates and tested for flow boiling enhancement with water. Their reported results for heat transfer coefficients that have been erroneously reported as around 5000 W/m²°C, while the correct units are believed to be 5000 W/m²°C. The surface became hydrophilic with the introduction of nanowires and the heat transfer performance improved. The wall superheat was reduced by about 8°C at a heat flux of 55 W/cm² under a mass flux of 251 kg/m²s. The maximum heat fluxes employed in their tests were below 60 W/cm².

The nanostructures are seen to be effective in increasing boiling performance through an increase in the surface wettability. This is an area where further research is warranted.

3.4 Research Needs in Flow Boiling Heat Transfer Enhancement in Microchannels. Heat transfer enhancement during flow boiling in microchannels has not been extensively investigated in literature as the instability issues were of prime concern. With their current low heat transfer coefficient and CHF limits, flow boiling in microchannels does not compete favorably with single-phase and pool boiling techniques. The fins are promising as they enhance the CHF limit, but the high wall superheat noted in tall fins make them unsuitable. Shorter fins are desirable from the wall superheat standpoint, but the associated area increase becomes limited. Porous surfaces are attractive, but care has to be taken to limit the additional thermal resistance introduced by the porous matrix. The nanostructures seem to be an attractive option, as they are able to alter and enhance the heat transfer mechanism. The benefits of microscale have not yet been fully realized for flow boiling in microchannels, and progress in that direction can be made only through a fundamental understanding of the boiling phenomena at microscale. Further benefits can be realized by combining such high performing microscale techniques with nanoscale surface features.

4 Flow Boiling Instabilities in Microchannels

Flow boiling instabilities are well recognized as a hindrance to the realization of evaporative microscale heat exchangers. They introduce temperature oscillations, thermal stress cycling, vibration, and they can compromise the structural integrity of the system. More importantly, they can lead to premature transition to the limiting CHF condition. To overcome this difficulty, an effort to better understand the mechanisms controlling flow instabilities at the microscale and to develop methods to suppress their deleterious effects have been undertaken by several independent research groups. As detailed in a recent book dedicated to this topic [71], much is qualitatively known about the nature of flow instabilities in microscale domains because of these studies. However, quantitative measures and models to better predict the occurrence and magnitude of these flow instabilities are far from being satisfactory. In addition, better methods need to be developed to overcome the challenges presented by flow instabilities.

4.1 Instability Modes. Studies performed over the last decade suggest that five instability modes control the small scale [49–51], namely, rapid bubble growth, Ledinegg (excursive), parallel channel, upstream compressible volume, and the CHF condition. Only the rapid bubble growth instability is unique to the microscale. The other four were initially identified for conventional scale channels; some dating back to the late 1930s, such as the Ledinegg instability [72]. However, the manifestations of these instabilities in microchannels are somewhat different than in their large scale counterparts.

4.1.1 Rapid Bubble Growth Instability. When a nucleation site inside a microchannel is activated, a bubble will form and very often will grow very rapidly before occupying the entire
channel’s cross-sectional area. Once this occurs, the bubble will continue to grow explosively downstream and upstream as well, if preventive means are not properly taken, causing flow reversal. As discussed by Kuo and Peles [73] and detailed by Peles [71], two mechanisms control this instability: the high liquid superheat sustained in microchannels made of silicon and the pressure waves generated during the growth and collapse of a bubble.

4.1.2 Ledinegg Instability. The first successful attempt to systematically study flow boiling instabilities was published by the German scientist M. Ledinegg who formulated the basis for the widely known static instability that bears his name [72]. From the various flow boiling instability modes this is perhaps the most important and most extensively studied. It is therefore, not surprising that it is also one of the first instability modes that were revealed in the course of studying flow boiling in microchannels. Many papers and books have detailed this instability that arises from the pressure drop-mass flux characteristics in a channel with flow boiling. This instability can only occur when the slope of the pressure drop-mass flux curve is negative.

4.1.3 Parallel Channel Instability. Connecting two or more channels in parallel to the same inlet and exit manifolds can modify the stability of the individual channel in two distinct ways. It can change the mass flow rate supply curve of the channel and, if the channels are connected through a conductive solid along their length, it induces thermal interaction that can modify the stability condition.

If the slope of the channel’s supply curve moderates, the system stability—the Ledinegg instability—will deteriorate. Because microchannel heat sinks are formed by etching many deep trenches into a silicon chip or a metal slab, the inlet manifold to exit manifold pressure drop will not be significantly affected by a temporary change in an individual channel, such as a temporary change in the mass flow. It follows that the system cannot react to changes occurring in a single channel, and the nominal supply curve of a single channel is flat (i.e., the inlet to exit pressure drop is constant irrespective of mass flow). This makes microchannels very susceptible to the parallel channel instability, which in this case, is directly related to the Ledinegg instability. On the other hand, thermal interaction between adjacent channels tends to stabilize the flow at the microscale [74].

4.1.4 Upstream Compressible Volume. This instability is common in systems with a significant volume of noncondensable gas trapped upstream or in a heated section. Since gas is compressible, changes to the inlet pressure will cause the section upstream a heated channel to become compressible—the extent of the compressibility depends on the volume of trapped gas upstream the channel relative to the channel’s volume. For very small channels, Bergles and Kandlikar [70] noted that a large volume of degassed liquid is sufficient to introduce considerable compressibility. It is often associated with pressure drop oscillations, having a characteristic frequency of about 0.1 Hz, combined with density wave oscillations, having a characteristic frequency of about 1 Hz.

4.1.5 Critical Heat Flux Condition. The critical heat flux condition is perhaps one of the most deleterious instability that can occur in flow boiling system. It marks the transition from a very effective heat transfer process to a very ineffective process. Since the heat transfer coefficient drops very rapidly following the CHF condition, the surface can reach unacceptable temperatures causing complete burnout of the channel under uncontrolled heating conditions.

Because of the importance of the CHF conditions—primarily avoiding it—literature concerning CHF conditions at the micro-scale are plentiful (e.g., Ref. [75]). After examining available literature and superimposing microchannel CHF data on Gambill and Lienhard [76] CHF limits, Carey [77] speculated that CHF at the microscale is hindered in comparison to the conventional scale. However, Kuo and Peles [78] suggested that the low CHF values in microchannels might be the results of suboptimal experimental conditions used in early studies, such as low mass fluxes and long channels. They used some of their own data to suggest that higher CHF values can be achieved in microchannels if appropriate conditions prevail.

4.2 Mitigating Instabilities. Because instabilities are detrimental in the implementation of flow boiling systems, several methods have been developed and proposed to suppress, or at least delay, their appearance. Most of these techniques are aimed at suppressing the Ledinegg instability, parallel channel instability, upstream vapor compressibility, and the critical heat flux conditions, but, several can also be used to suppress the rapid bubble growth.

Inlet restrictors have been used in conventional scale channels to overcome the Ledinegg instability and parallel channel instability. They add pressure drop that increases linearly with mass flow rate and, therefore, reduces the negative slope of the pressure drop-mass flux demand curve. This technique was found to be equally effective in microchannels [69,79]. Diverging channels is a variation of inlet restrictors used to suppress the Ledinegg instability and was successfully demonstrated to be effective at the microscale (e.g., Ref. [80]).

Fractal tree-like microchannels [81] were studied partly as a method to mitigate flow boiling instabilities. Although not immediately apparent, this flow configuration is yet another variation of the inlet restrictors as far as flow instabilities are concerned. This is because the cross-sectional flow passage increases downstream and, therefore, shifts the demand for the channel’s pressure drop toward the inlet where the flow is either single-phase flow or at low quality.

Extraction of vapor from a microchannel (Fig. 8) in an effort to reduce the void fraction and, by proxy, reduce the apparent effect of the boiling process on the overall channel’s pressure drop demand curve was proposed by David et al. [82].
and novel idea is in its infancy and will require much effort, innovation, design, and careful microfabrication processes to become a viable and superior solution for flow instabilities in microscale domains.

Several methods to address the rapid bubble growth have been studied. Forming reentrant cavities along a microchannel, such as the one shown in Fig. 9, in an effort to promote nucleation and reduce the wall superheat at the onset of nucleate boiling (ONB), was proposed by several groups [73,78,80,83]. Since one of the mechanisms responsible for the rapid bubble growth is the liquid superheat, reduction in the liquid temperature at ONB helps suppress this instability. However, since other mechanisms can also trigger the rapid growth of a bubble, i.e., the bubble dynamics and elevated pressures, this remedy might not be sufficient.

Since reentrant cavities reduce the liquid superheat, they also tend to suppress the oscillation generated by the bubble ebullition process. Macroscale studies [84] have shown that flow oscillations tend to prematurely trigger CHF conditions. Microscale studies [73] confirmed that the reduction in flow oscillation attributed to surface reentrant cavities can also be used to augment the CHF conditions.

Analytical efforts to study the stability of flow boiling in microchannels and develop techniques to suppress flow instabilities were recently reported by Zhang et al. [85,86]. Active feedback control models were used to simulate modulation of the supply pump and the inlet valve in an effort to suppress the Ledinegg and the upstream compressible volume instabilities in a single channel. Based on these models a controller design was presented. In a subsequent paper [86], the stability and active flow and temperature control of parallel channel system were analyzed. With identical channels, the system was shown to be uncontrollable. For nonidentical channels, instabilities can be controlled through modulating the pump. A controller design approach for such cases was presented.

4.3 Research Needs—Flow Boiling Instabilities. Despite the several efforts to suppress flow boiling instabilities detailed above, methods developed so far are far from being perfected either because they introduce drawbacks, such as increased pressure drop (e.g., inlet restrictors to suppress the Ledinegg instability), or they lack sufficient capabilities to suppress them. Some of the most advanced techniques suggested may be superior, but, they have not been successfully proven and are not mature enough.

New active and passive methods need to be developed to overcome penalties associated with current proven techniques and at the same time provide robust inexpensive suppression techniques. Efforts pertinent to active flow control should include both open loop and closed loop control.

Regardless of the effort to develop new techniques, fundamental studies need to be continued to map flow boiling instabilities and identify their modes. This is desirable because instabilities are multidimensional problems with many variables controlling their occurrence. Experimental and numerical studies should be conducted to expand available data under large operating conditions, different geometries, and different types of fluids. In addition, multiscale numerical models need to be further developed to predict instabilities, such as the rapid bubble growth. These models should span length scales starting from the nanoscale to several millimeters, and will require a range molecular dynamic simulations coupled with continuum models. Such studies should have an eye toward current and future applications.

5 Condensation in Microchannels

5.1 Recent Advances. Condensation, evaporation and boiling share several underlying aspects; however, the drivers for research in these fields have been different. While boiling and evaporation have been critical for the advancement of high-heat-flux heat removal from compact devices in applications such as electronics cooling, condensation is usually associated with heat rejection, typically coupled with larger thermal resistances from single-phase coupling fluids or ambient air. With the increasing emphasis on system-level metrics such as overall energy efficiency, reductions in heat transfer equipment size, and the use of environmentally friendly working fluids, obtaining a fundamental understanding of condensation at the microscale assumes increasing importance. As the flow passage size decreases, the relative importance of surface tension compared to gravity increases. Therefore, condensation models developed for larger tubes (where gravity and inertial forces dominate) are not expected to reliably predict the behavior at small scales. Thus, there is a critical need for robust condensation heat transfer and pressure drop models applicable to a wide variety of working fluids, microscale geometries and operating conditions. Measurements and modeling of condensation processes pose challenges different from those encountered in boiling and evaporation, even in the most basic aspect of measuring heat transfer rates. Over the past decade, advances in experimental and modeling techniques have gradually begun to establish the groundwork for an in depth understanding of condensation at the microscale.

5.1.1 Condensation Flow Regimes and Void fractions in Microchannels. Heat transfer and pressure drop in microchannel condensation depend strongly on the mechanisms for two-phase flow (e.g., stratified, intermittent, annular, and their variants) as well as corresponding phase distributions characterized by void fraction, slip ratio, etc. Thus, heat and momentum transfer models require knowledge of the flow regimes for the applicable operating conditions and geometries. Over the past decade, flow regime mapping has progressed from visualization of air/water mixtures under adiabatic, ambient pressure conditions, to phase change of representative fluids at operating conditions of interest. These advances are slowly reducing the need for unreliable extrapolations of findings on air–water mixtures with their large phase property differences (e.g., liquid/vapor density ratio, surface tension) to typical condensing fluids with completely different properties and the accompanied phase change. One of the first visualizations of the actual condensation process in microchannels was the work of Coleman and Garimella [87–89] on R134a using an innovative pressurized transparent enclosure around glass microchannels to enable visualization at high pressures, representing a departure from adiabatic air–water mixtures. They classified flow regimes into annular, wavy, intermittent, and dispersed flows, with further subdivisions into patterns, and developed empirical flow regime transition criteria as functions of $G$, $x$, and $D_h$ for circular ($D = 4.91$ mm) and rectangular ($1 < D_h < 4.8$ mm;
0.5 < AR < 2) microchannels at 150 < G < 750 kg m\(^{-2}\) s\(^{-1}\). In addition to the documentation of intermittent and other flows during condensation, their most significant observation was that as \(D_h\) decreased, the intermittent and annular flow regimes increased in extent at the expense of stratified/wavy flow regimes, due to the increased importance of surface tension over gravity. Following up on that initial work from the early 2000s, Keinath and Garimella [90] extended the visual documentation of condensation (Fig. 10) in two important aspects: operation at high reduced pressures (0.38 < \(P_r\) < 0.77) approaching the critical pressure, and b) the use of azeotropic refrigerant mixture R404A. In addition, they extended the range of geometries (0.5 < \(D < 3\) mm), and used advanced image processing techniques to obtain quantitative information about the dynamic phase distributions in such flows.

Moreover, such visualizations of condensation at the microscale with high spatial and temporal resolution have also enabled the measurement and modeling of local void fractions, with working fluids of interest operating at the relevant conditions. For example, Winkler et al. [91] developed an edge-detecting and spline-fitting technique to identify the vapor–liquid interface in wavy and intermittent flow during condensation of R134a for 1 < \(D_h\) < 4.8 mm, which was used to obtain void fractions based on video frames of the respective regimes. Keinath and Garimella [90,92] extended the work of Winkler et al. further through the use of better optical resolution and image processing to obtain data for diameters as small as \(D_h = 0.5\) mm, and for a wider range of reduced pressures 0.38 < \(P_r\) < 0.77 for condensing R404A. This wide range of \(P_r\) enabled the assessment of the influence of fluid property variation on void fraction. In their experiments with 200 < \(G\) < 800 kg m\(^{-2}\) s\(^{-1}\) in round tubes with \(D = 0.5, 1.0,\) and 3.0 mm, the vapor core is assumed to be circular or circular on top with a flat bottom at the appropriate height (Fig. 11(a)). Like Winkler et al. [91] and others before, Keinath and Garimella [92] did not find a significant influence of \(G\) on void fraction, particularly at low \(x\). The bulk average void fraction also did not depend appreciably on \(D_h\); however, \(D_h\) did affect the distribution of the liquid and vapor within the flow. This is evident in Fig. 11, which shows axial and bulk average void fractions for \(G = 200\) kg m\(^{-2}\) s\(^{-1}\) flow at \(T_{\text{sat}} = 30^\circ\) C in tubes with \(D = 1\) and 3 mm. For both tubes, the bulk average void fraction is the same, even though one image shows intermittent flow, and the other wavy flow. Such emerging insights show that basing transport models only on bulk average void fractions is inadequate; different interfacial areas, even at the same void fraction, would lead to different \(\Delta P\)s and heat and mass transfer coefficients.

### 5.1.2 Microchannel Condensation Pressure Drop

Due to the strong coupling of heat and momentum transfer, accurate determination and modeling of microchannel condensation pressure drop is essential for the development of heat transfer models. Cavallini and co-workers [93–97] have, during the past decade, achieved progress on condensation \(\Delta P\) at small \(D_h\). They investigated \(\Delta P\) in a rectangular multiport minichannel (\(D_h = 1.4\) mm) for R-134a and R-404A [93] and R-134a, R-236ea, and R-410A [95] for 200 < \(G\) < 1400 kg m\(^{-2}\) s\(^{-1}\) at 40°C. The refrigerants spanned a range of reduced pressures, high \((P_R = 0.49, R410A)\), medium \((P_R = 0.25, R134a)\) and low \((P_R = 0.14, R236ea)\). They found that most correlations could not predict the frictional gradient over the range of fluids and operating conditions, and developed a two-phase multiplier correlation [97] based on the Friedel [98] model for annular flow using data from several studies of condensation in microchannels. They concluded that gravitational effects were not important in annular minichannel flow and eliminated the Froude number dependence in the model, and also attempted to account for liquid entrainment. More recently, Cavallini et al. obtained \(\Delta P\) data for R134a [94] and R1234yf [96] in a single circular microchannel with \(D = 0.96\) mm for 200 < \(G\) < 1000 kg m\(^{-2}\) s\(^{-1}\). With this, they extended the above model [97] to account for surface roughness and to nonshear...
dominated flow regimes through modifications in the liquid film friction factor.

Condensation $\Delta P$ models with special attention to the applicable flow regime include the initial multiregime model of Garimella et al. [99] for condensing R134a in circular microchannels, followed by Agarwal and Garimella [100] for circular and noncircular microchannels ($0.42 < D_h < 0.8$ mm). This model was in turn based on models of $\Delta P$ in intermittent flow by Garimella et al. for circular [101] and noncircular [102] microchannels, and for annular flow [103]. The intermittent flow $\Delta P$ model was based on the unit cell shown schematically in Fig. 12. The insights about intermittent flow structure gained from their earlier flow visualization studies enabled them to idealize the total $\Delta P$ in each unit cell as the sum of the $\Delta P$ in the liquid slug, vapor bubble and transitions between the two (Eq. 12). A correlation for slug frequency was then developed as a function of slug Reynolds number for circular and noncircular tubes.

$$
\frac{dP}{dz} = \left( \frac{dP}{dz} \right)_{\text{film-bubble}} \left( \frac{L_{\text{bubble}}}{L_{\text{UC}}} \right) + \left( \frac{dP}{dz} \right)_{\text{slug}} \left( \frac{L_{\text{slug}}}{L_{\text{UC}}} \right) + \Delta P_{\text{transition}} \left( \frac{N_{\text{UC}}}{L} \right)
$$

(12)

For the annular flow model [103], the frictional $\Delta P$ is related to the interfacial friction factor as shown in Eq. (13), where the void fraction, $\xi$, is calculated from the Baroczy [104] model

$$
\frac{\Delta P}{L} = 2 \cdot \frac{G^2}{\rho_w \cdot g \cdot \xi^2 \cdot D} \left( \frac{1}{h} \cdot \frac{2}{\rho_v} \cdot \frac{G^2}{\rho_w} \cdot \xi^2 \cdot \frac{1}{D} \right)
$$

(13)

The interfacial friction factor ($f_i$) is then correlated to the liquid-phase Darcy friction factor, by the expression in Eq. (14), where $X$ is the Martinelli parameter and $\psi$ is the ratio of viscous-to-surface tension forces

$$
\frac{f_i}{f_L} = A \cdot X^a \cdot Re_L^{-b} \cdot \psi
$$

(14)

5.1.3 Microchannel Condensation Heat Transfer: Accurate measurement of condensation heat transfer in microchannels poses unique challenges. Unlike boiling experiments, heat transfer rates cannot be measured through the electrical input, and must therefore be inferred from secondary measurements. This is in turn made difficult by the small mass flow rates in microchannels that lead to small condensation heat duties, especially when experiments must be conducted with small changes in $x$ across the vapor–liquid dome to achieve enough resolution to track flow regime variations. Further compounding the challenges is the fact that the low heat duties coupled with the high $h$ in microchannels leads to a very small $\Delta T$ upon which $h$ measurements must be based.

Models for microchannel condensation have focused predominantly on annular flow, with adjustments and interpolations with single-phase heat transfer to address for intermittent flow. Cavallini and co-workers conducted condensation measured heat transfer in concert with their $\Delta P$ studies [93,95–97] in circular and rectangular channels with $D_h < 3$ mm. From these data and their $\Delta P$ model, they developed a heat transfer model based on the heat and momentum analogy [97]. Cavallini et al. [105] then developed a model from a database for tubes with $D > 3$ mm. Despite being comprised of data for large tubes, Del Col et al. [96] found that the model predicted condensation for R1234yf for $D = 0.96$ mm well.

An innovative solution to the issue of accurate measurement of condensation $h$, the “thermal amplification” technique, was developed by Garimella and Bandhauer [106]. In this technique, heat of condensation is removed by a high flow rate primary coolant, which provides a low thermal resistance, followed by rejection of this heat of condensation to a secondary coolant at a low thermal capacitance rate. This approach decouples, and separately addresses, the conflicting challenges of measurement of accurate heat duties of a few Watts, while also accurately deducing condensation $h$ from the measured overall conductance values without intrusive temperature measurements. The resulting heat transfer coefficients were then used to develop models for condensation of R134a in circular ($0.506 < D < 1.524$ mm) [107] and noncircular ($0.424 < D_h < 0.839$ mm) [108] channels for $0.25 < G < 750$ kg m$^{-2}$ s$^{-1}$. To predict the shear-driven annular condensation, they used an approach similar to Traviss et al. [109]. To account for microchannel effects, the $\Delta P$ and interfacial shear for this model were determined from the corresponding annular $\Delta P$ models (Eqs. (13) and (14)).

Advances in the measurement of $h$ at even smaller scales ($0.1 < D_h < 0.4$ mm) were made by Agarwal and Garimella [110,111] for condensing R134a in rectangular geometries using photolithography and diffusion bonding for the fabrication of the test sections. Heat transfer coefficients were obtained from the data through spatially and temporally resolved analyses of the conjugate heat transfer between the copper channels and the refrigerant and coolant, including developing a closed-form solution to address the effect of intermittent vapor and liquid flow over the channel surface. These heat transfer coefficients yielded a flow regime based model, in which the heat transferred in the vapor bubble and liquid slug regions were combined to obtain a composite time-averaged $h$, as a function of the correlated slug length ratio (Eq. (15)). For highly annular flows (high $x, G$), the liquid slug length is very small compared to the vapor bubble, and the heat transfer is dominated by that occurring across the thin annular film.

$$
h = h_1 \cdot \left( \frac{L_{\text{slug}}}{L_{\text{slug}} + L_{\text{bubble}}} \right) + h_2 \cdot \left( 1 - \frac{L_{\text{slug}}}{L_{\text{slug}} + L_{\text{bubble}}} \right)
$$

(15)

Such a technique allows for the modeling of a wide range of conditions with inherently smooth transitions between the two dominant regimes in microchannels, intermittent and annular. Numerical models for predicting microchannel condensation heat transfer for laminar annular flow have been developed by Wang and Rose [112,113], Nebuloni and Thome [114] and Da Riva and Del Col [115]. Wang and Rose’s model accounts for the streamwise shear stress on the condensate film and the transverse pressure gradient due to surface tension. The transverse pressure gradient is particularly important in noncircular microchannels, where fluid is preferentially drawn to sharp corners resulting in local thinning of the condensate film and high $h$. This phenomenon is generally not accounted for in correlations for large channels. Closure to the model is provided through an empirical prediction of $\Delta P$. Their technique enables the investigation of the
individual contributions of gravitational, surface tension and viscous effects by “switching on or off” the corresponding terms. They use this model to identify surface tension and viscosity dominated regions where $h$ can be predicted via a Nusselt-type correlation for a variety of fluids, with good agreement with the data of several investigators including Cavallini et al. [95] and Bandhauer et al. [107]. Nebuloni and Thome [114] investigate laminar annular condensation in microchannels with varying internal shape using a finite volume formulation of the energy equations in the liquid phase. They showed that as $D_h$ decreases, the effect of axial conduction on local tube wall temperature and $h$ can be significant. Da Riva and Del Col [115] simulate condensation of R134a in a 1-mm diameter circular channel using a volume of fluid method to assess the influence of surface tension. For the circular channel, they found condensation to be gravity dominated. This is in contrast to the other numerical studies [112–114], which show a surface tension effect when condensation occurs in geometries with sharp corners.

5.2 Research Needs—Condensation in Microchannels.

The above discussion shows that the past decade has seen a substantial increase in the understanding of condensation at the microscales, where investigators are not simply extrapolating from air–water two-phase flows, or proposing minor modifications to models originally developed for large $D_h$. Microchannel condensation is now being studied as a unique entity with its own governing influences. Innovative high-speed visualization techniques have yielded detailed qualitative and quantitative information about the prevalent flow regimes, the criteria for transitions between them, and the liquid–vapor phase distribution information needed for void fraction models. Flow mechanism information, together with shear and momentum balance analyses in different regions of intermittent and annular flow, has yielded models that specifically account for microscale phenomena. Measurement techniques developed to address challenges specific to high flux heat transfer at low heat duties have yielded heat transfer coefficients that appear to be in agreement across different groups. Probably the best known example was introduced publicly in 1980 when IBM announced the “thermal conduction module” [116]. By coincidence, Tuckerman and Pease [1,30] recognized that the thermal resistivity of silicon was not the problem. The problem was heat removal from the back surface.

Somewhat surprisingly (at the time) this advance was not exploited by the computer manufacturers. This was because at the same time the scaling down of dimensions in ICs led to faster, less power-hungry and cheaper computing circuitry. Complementary metal oxide semiconductor (CMOS) circuits became attractively fast and, more importantly, eliminated static power dissipation, and so displaced emitter-coupled logic (ECL) circuits in computers. So this eliminated the need for water cooling in most computers. An additional factor was that the use of computers had evolved from a few expensive, fast, central computers to many cheaper ones but not quite so fast. Thus, computing electronics did not need the Tuckerman heat sink. The only market for many years was for cooling laser diode arrays.

As a result microchannel cooling was revived. As an example Colgan et al. [39] at IBM have described a large variety of the microchannel structures to allow rational choices for different conditions and a demonstrated specific thermal conductance of about 5 W/K/cm$^2$ for single chip modules. One contribution to the thermal resistance is that of the interfaces. This was avoided in the original configuration of Tuckerman and Pease by fashioning the heat sink into the chip itself. Such an approach was not well received by those manufacturing ICs, so a novel interface structure based on microcapillaries was introduced. This was the first use of microstructures to improve a thermal interface and was very effective, with an interfacial conductance of more than 30 W/K/cm$^2$ [30], but has been little used since then.
As a second example, the company Cooligy was formed specifically to develop aggressive technologies for cooling high-speed chips. Cooligy technology not only employed microchannels but also employed phase change (originally used in heat pipes) and novel pumping technology [117]. The cooling liquid was directly pumped into the “microstructure heat collector” attached on the back of a silicon chip. The rest of the cooling loop consisted of a pump and an air-to-coolant heat exchanger with a fan. Cooligy technology was used initially in the Apple G4 table top computer.

3D integrated circuitry is now being developed in various forms. In its simplest and most widely used form, completed chips are stacked on top of each other and are interconnected at the periphery. The next level being considered is to arrange interchip connections by using an array of through silicon vias; this is under very active development. These two techniques can be regarded as advanced packaging. A further development is “wafer stacking” in which prefabricated layers of active circuitry are laminated together [118]. This allows larger number of interconnects between the layers of circuitry. Finally there is monolithic 3D-IC in which the layers of circuitry are fabricated in situ [119]. This approach allows the highest density of interconnects between the two layers of active circuitry. All forms of 3D-ICs have the challenge of removing heat from the upper layers. Anomalous, usually lower, thermal conductivity of materials in thin film form is one complication. This is yet another example of the importance of developing an understanding of thermal transport at the atomic level. Another related issue is the tortuous route of removing heat from the upper layers, i.e., through many interfaces. Numerical simulation is starting to play a big role in modeling these interfaces. Improved simulation using more powerful computers and imaginative new algorithms should be a really fruitful area of research. Microchannels are uniquely suited to provide cooling between two IC layers. However, the issues related to thermal vias and interconnects require close collaboration among the circuit designers, thermal engineers and fabricators [120].

In cooling of high power electronics, many microchannel devices now use either evaporative cooling of water or refrigerants, or impingement jet cooling [121]. The heat fluxes reached with these technologies are in the range of some hundred W cm⁻² and, thus, not significantly higher than those reached a decade ago. The same values have been successfully obtained with single-phase laminar cooling to obtain low pressure losses [122–124]. Figure 13 shows an example of such a microstructure fabricated in copper.

However, high power electronics will be further developed. While in former times the key application for cooling was computing processors, it is now changed to full computers (i.e., blade server systems) and high power electronics for automotive industry. Especially in the latter, a combination of very high heat flux and low pressure losses, obtained at minimum volume, is targeted. For any type of electronic cooling application, the quality and reliability of the thermal interface between the electronic device (e.g., a semiconductor chip) and the microchannel heat sink or “cold plate” is of critical importance, and there is ongoing research employing nanotechnology to improve the heat transfer coefficients of such interfaces while maintaining minimal mechanical stress on the chips [125].

6.2 Research Needs in Application of Microchannels in Electronics Cooling. Advances in thermal management are needed for the development of further generations of electronics to continue. Such advances can be brought about by exploiting the opportunities afforded by new materials and structures and by greatly improved modeling techniques that themselves rely on advanced computing electronics and by a resurgence in new algorithms and related techniques emanating from computer science departments. Inventions of new configurations are hard to predict but funding agencies should also be ready to review, and fund, seed projects to test out new ideas that offer rational promise of meeting some of the above challenges. Cooling solutions in the future will require more collaboration among the electronics circuit designer, microfabrication engineers, reliability experts, material scientists and thermal engineers. Novel solutions that dramatically extend the cooling capabilities may be explored independently in laboratory setting, but their ultimate success depends on our ability to manufacture functional devices in a cost-competitive manner.

7 Applications of Microchannels in Heat Exchangers

7.1 Microscale Heat Exchangers: Theory, Design and Applications. Microscale heat exchangers have been the subject of research and application for more than 30 yr now. The developments of the last 10 yr are briefly described in the following text.

7.1.1 Theoretical Overview. The potential advantages of miniaturization for improving heat transfer have been appreciated for many decades; publication of the book “Compact Heat Exchangers” by W. M. Kays and A. L. London in 1955 was particularly influential in educating the engineering community of that era. By miniaturization, the dimensions of geometrical elements of heat transfer devices are changed. The influence of the change of dimensions can be quantified by using a scaling factor S. Based on the physical base units mass, length and time, the volume is scaled by S³, the surface area by S¹ and the characteristic length (i.e., the hydraulic diameter) by S¹. Flow velocity is scaled by S⁰. A detailed description of the influences of scaling is provided in Ref. [126].

Many theoretical and experimental results have been published within the last 10 yr dealing with the validity of classical heat transfer correlations, as applied to the prediction of heat transfer behavior in microchannel heat exchangers [43,126–128]. Looking at this literature, it is obvious that there is no consensus on this topic yet, neither for laminar nor turbulent flow. However, Morini et al.
showed, by analyzing systematically many published results and combining them with experiments, that in many cases no deviation from predictions can be shown, after taking into account measurement uncertainties or influences of heat conduction and heat dissipation, geometry of the device or other confounding factors [129,130]. They showed that in most cases the Gnielinski correlation given in Ref. [131] gives satisfactory predictions. The effects of roughness and other factors influencing the single-phase heat transfer have been discussed in Sec. 3.1.2.

7.1.2 Microscale Heat Exchanger Devices and Applications. Inspired by the pioneering work of Tuckerman and Pease [1], microscale heat exchangers have been designed, manufactured and tested worldwide in different applications. Today a main advantage over earlier work is that there are now several commercial providers for microscale heat exchangers in metal, ceramics, glass and/or plastics, whereas in former times there was only the possibility to obtain very specific prototype devices for lab experiments. Thus, microscale heat exchangers have been taken out of the laboratory status into commercial products, to a certain extent. Nevertheless, in many cases they are still prototype pieces, or tailor-made for specific applications. Within the last 10–12 yr, the major applications have been high power electronics cooling, heat exchangers in offshore facilities, and heat transfer applications in the automotive and aerospace industries. Aside from these, microscale heat exchangers have been identified as valuable tools in process engineering, bio engineering and other topics of research. The objectives in the applications addressed so far are fairly different. In electronics cooling, the goal is usually to minimize thermal resistance in a compact volume while maintaining a very low fluid pressure drop. In contrast, the heat exchangers used in offshore applications and process engineering are typically designed to provide very high power density at high pressure levels and high temperatures combined with minimized susceptibility to fouling and corrosion. Here, pressure losses are important but not a key driver for the development, which is to save volume on the site of an offshore platform or a chemical plant, as well as to gain efficiency. Especially in process engineering, process intensification by recuperation of energy (namely, thermal energy) and integration of process steps into modular devices have attracted great interest [132]. Coatings and specific materials have been implemented for corrosion resistance and antifouling behavior. Moreover, integration of catalytic materials to enable using the heat exchangers as chemical reactors have been developed to obtain improvements in chemical process engineering [132]. Recently, the integration of microscale heat exchangers for recuperation of thermal power in mobile systems (automotive, railroad, ships, air and space traffic) as well as the improvement of decentralized generation of electrical power and heat has gained major interest. Here, the superior heat transfer capacities combined with the small volume are the main advantages taken into account.

Materials used for microscale heat exchangers depend heavily on the application. Whereas for electronic cooling in most cases silicon, copper or aluminum are used (in rare cases ceramic coolers have been employed), heat exchangers for industrial applications are mainly made of stainless steel or highly corrosion resistant alloys. For extremely high temperatures, devices made of SiC have been developed. Glass devices are used for very corrosive fluids in chemical reaction engineering, and in some very low temperature heat recovery applications using liquid helium. For biomedical applications, polymers are often used due to the need to have disposable devices. Devices which need not be disposable are most often made of stainless steel. In terms of biomedical or nutrition applications, the principles of hygienic design and manufacturing have generally not yet been met with microscale heat exchangers.

The basic microchannel heat exchanger designs are more or less derived from those existing in macroscale, namely, crossflow, countercurrent and cocurrent flow designs. Figure 14 shows examples of those designs, manufactured in stainless steel.

Additionally, special microstructure designs have been implemented to increase the heat transfer and decrease the pressure drop, in most cases linked to specific applications. Fluids used have not changed significantly from those used in conventional heat exchangers; water or mixtures of hydrocarbons with water, alcohols, refrigerants and gases are most commonly used. A
relatively new trend is to use so-called nanofluids to increase the heat transfer [133]. Nanofluids are suspensions of nanoparticles (i.e., Al₂O₃ or similar) in liquids. They can provide an improvement in heat transfer from several percent up to roughly an order of magnitude, but may be prone to fouling inside the microstructures. As might be expected, it has been observed that fouling and blocking is an even more serious problem for microscale heat exchangers than for macroscale ones. For example, accelerated fouling led to abandonment of an otherwise promising attempt to develop an energy-efficient microscale milk pasteurizer [134]. A reliable solution for this problem has not yet been found. In another study for the same topic, a simple microscale heat exchanger design was used for homogenization and pasteurization of dairy products, but did not succeed very well. However, a solution was suggested and tested, and proofed to work, at least for the fouling problems. Recently, the thermal treatment of hydrocarbon monomers to create polymerized hydrocarbons has been performed successfully. This is an especially interesting accomplishment, considering the fouling problems that can occur during polymerization processes. By careful device and process design, fouling effects were completely avoided. Some countermeasures used successfully in macroscale (such as antifouling coatings) do not work at microscale due to the fact that the shear forces for the drag-off of fouling layers or particles are simply too low in laminar flow. However, stainless steel microscale heat exchangers have successfully been applied to biotech processes with living cells as well as in the nutrition industry, showing no increased tendency to fouling. Here, precise design and process control seems to be more important than antifouling precautions.

7.2 Research Needs in Microchannel Heat Exchangers. Within the last 10 yr, the theoretical description of heat transfer in microscale has advanced only gradually. The same is true for development of new devices, with the exception of some very specific devices. Several basic tasks, such as the validity of conventional models at the microscale, are still not completely clarified, as well as the influence of the surface quality or how to overcome fouling. However, the technical applicability of microstructure heat exchangers has been widely enhanced owing to the emergence of commercial providers of such devices, which makes it easier to apply them to a wide variety of applications and technologies. In the future, a homogenization of the model description as well as many more applications are expected.

Some of the most interesting future applications may result from the very short thermal time constants of microchannel structures (resulting from the combination of very high heat transfer coefficient and very low heat capacity), coupled with their very short diffusion times (high mass transfer coefficients) and the short residence times inside the devices which can allow acceleration of various physical and chemical process flows. One example of exploiting the short thermal time constant (about 10 ms) of the microchannel heat sink coupled with a high thermal conductance compliant interface is currently being developed to improve the accuracy of “nanoimprint lithography,” a technology which may be key to future progress in semiconductor manufacturing. The idea is to use locally controlled expansion to assure accurate overlay in large area (e.g., 300 mm diameter) nanoimprinting. In this technique, a quartz template with a relief image of the required pattern is pressed into a liquid polymer on top of the wafer, which sits on a microchannel heat sink chuck. The pattern on the template is accurately positioned over the existing structure on the wafer, then a laser pulse crosslinks the polymer into a solid resist film. With appropriate chemistry, the quartz template can be removed without damaging the polymeric resist pattern. Features finer than 10 nm can be patterned in this manner. Achieving accurate overlay over the entire pattern requires that local distortion of template and wafer match. By using an array of light microscopes to monitor local overlay errors and feeding back to local heating elements within the microchannel heat sink chuck, one can rapidly correct these errors [135].

As mentioned before, the influences of surface roughness are not described in a homogeneous way yet. Due to the fact that technical heat transfer systems (at least those not made from silicon) always show a certain level of surface roughness, there is need for a consistent treatment of the influences of surface roughness on the heat transfer. More generally, a consistent model description of the heat transfer in microscale is needed, which is still under discussion. In many cases, effects like entrance lengths or developing flows as well as flow mal-distributions by parallelization of microstructures with inadequate distribution systems at the entrance and outlet manifolds (plenums) are not given sufficient attention with when designing and manufacturing microscale heat exchangers. This is of special importance when dealing with modular systems like those already mentioned for chemical process engineering or energy-efficient processes. Moreover, there is no standardized method for the practical design of microscale heat exchangers yet. Together with a homogenized and accepted theoretical description (which could be implemented into numerical models) this could improve the precision of experimental work and the comparability between numerical simulation results and experimental results worldwide. Most likely this would also improve the business possibilities for manufacturers of microstructure heat exchanger devices, which can be fairly large sized units employing microscale passages.

8 Conclusions and Outlook

Fundamental issues related to transport processes—single-phase gas flow, single-phase liquid flow, flow boiling and condensation—have been extensively explored and a sound understanding of these processes is available through the published literature. New opportunities are being presented with the simultaneous exploration of nanoscale features and transport processes. Advances in microfluidics, MEMS technology and nanomaterial fabrication and characterization are opening up new possibilities in making the microchannel transport processes more efficient and applicable to practical devices. As seen from the surge in research in the last decade, fundamental research and practical implementation are expected to synergistically lead the advancement in this area. After presenting a brief overview of the current status, research needs in these areas are listed for the respective microchannel application. Research needs and product development opportunities for electronics cooling as well as heat exchangers employing microscale passages are also highlighted.

Acknowledgment

The authors would like to thank U.S. federal agencies for the encouragement and support they have provided through research grants to a number of researchers working in the area of microscale transport processes. In particular, the support provided by National Science Foundation in supporting conferences, specifically the ASME International Conference on Nanochannels, Microchannels, and Minichannels, is gratefully acknowledged. Special thanks to the NSF CBET Program Director Professor Sumanta Acharya for his encouragement in writing this status report. The progress has been truly made possible by the large number of researchers worldwide through their creative endeavors. The support of the European Community under Grant No. PITN-GA-2008-215504 is gratefully acknowledged by the authors Stéphane Colin and Juergen Brandner.

Nomenclature

- \( A \) = surface, m\(^2\)
- \( AR \) = aspect ratio, W/H
- \( C \) = free variable
- \( D, d \) = diameter, m

Journal of Heat Transfer
SEPTEMBER 2013, Vol. 135 / 091001-15
\( D_h = \) hydraulic diameter, m  
\( f = \) friction factor  
\( G = \) mass flux, kg \( m^{-2} s^{-1} \)  
\( h = \) heat transfer coefficient, W m\(^{-2}\)K\(^{-1}\)  
\( K = \) free variable  
\( k = \) thermal conductivity, W m\(^{-1}\)K\(^{-1}\)  
\( L = \) length, m  
\( LUC = \) length of unit cell, m  
\( n = \) normal coordinate, m  
\( P = \) pressure, Pa  
\( Pr = \) reduced pressure  
\( R = \) specific gas constant, J kg\(^{-1}\)K\(^{-1}\)  
\( \delta = \) scaling factor  
\( s = \) length, m  
\( t = \) streamwise tangential coordinate, m  
\( T = \) temperature, K  
\( u = \) streamwise velocity, m s\(^{-1}\)  
\( v = \) velocity component normal to the wall, m s\(^{-1}\)  
\( w = \) quality or length (in Sec. 7)  
\( X = \) Martinelli parameter  

### Dimensionless Numbers

\( Br_p = \) modified Brinkman number  
\( Ca = \) capillary number  
\( Ec = \) Eckert number  
\( Gr = \) Graetz number  
\( Kn = \) Knudsen number  
\( Ma = \) Mach number  
\( Nu = \) Nusselt number  
\( Pe = \) Peclet number  
\( Pr = \) Prandtl number  
\( Re = \) Reynolds number  

### Greek Symbols

\( \alpha = \) void fraction  
\( \rho = \) density, kg m\(^{-3}\)  
\( \psi = \) dimensionless surface tension term  
\( \chi = \) mean free path, m  
\( \Lambda = \) dimensionless length  
\( \mu = \) dynamic viscosity, Pa s  
\( \sigma = \) surface tension, N m\(^{-1}\)  
\( \sigma_v = \) viscous slip coefficient, dimensionless  
\( \sigma_t = \) thermal slip coefficient, dimensionless  
\( \phi = \) heat flux, W m\(^{-2}\)  
\( \beta = \) temperature jump coefficient, dimensionless  

### Subscripts

\( D \) = dimensionless  
\( I, j = \) index  
\( L = \) liquid  
\( n = \) normal  
\( SAT = \) saturated  
\( t = \) tangential  
\( V = \) vapor  
\( w = \) at the wall  
\( z = \) in z-direction  
\( \infty = \) fully developed conditions  

### Acronyms

C = circular microtube  
CHF = critical heat flux  
CWT = constant wall temperature  

PP = parallel-plate microchannel  
UHF = uniform heat flux  
1IW = one insulated wall  

### References


