On the Nature of Critical Heat Flux in Microchannels

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The critical heat flux (CHF) limit is an important consideration in the design of most flow boiling systems. Before the use of microchannels under saturated flow boiling conditions becomes widely accepted in cooling of high-heat-flux devices, such as electronics and laser diodes, it is essential to have a clear understanding of the CHF mechanism. This must be coupled with an extensive database covering a wide range of fluids, channel configurations, and operating conditions. The experiments required to obtain this information pose unique challenges. Among other issues, flow distribution among parallel channels, conjugate effects, and instrumentation need to be considered. An examination of the limited CHF data indicates that CHF in parallel microchannels seems to be the result of either an upstream compressible volume instability or an excessive instability rather than the conventional dryout mechanism. It is expected that the CHF in parallel microchannels would be higher if the flow is stabilized by an orifice at the entrance of each channel. The nature of CHF in microchannels is thus different than anticipated, but recent advances in microelectronic fabrication may make it possible to realize the higher power levels. [DOI: 10.1115/1.1839587]

Introduction: General Description of Microchannel Heat Exchangers

This paper will be directed toward microchannels, which invariably involve cooling channels in blocks, as opposed to mini and larger diameter channels that have individual confining walls and are usually thermally well controlled. Using commonly accepted definitions of microchannels [1,2], the hydraulic diameter \( D_h \) will be in the range 10–200 \( \mu \text{m} \). The length of the flow passages, \( L \), will be on the order of 10,000 \( \mu \text{m} \). (The length will be less in the case of interrupted or “cross-linked” channels [3].) The channels will usually be cut in a block; for silicon, microelectronic fabrication techniques will be used [2], whereas for copper or other metals, an end mill, or a lamination-and-bonding process will be used. Typically, there will be of the order of 100 parallel channels. Heat is usually supplied to one side of the block. Assuming that the channels are cut from one side [4], a cover plate is provided on that side (Fig. 1), and inlet and exit headers couple the microchannel heat exchanger to the flow system.

The result of this arrangement is that two problems experienced in conventional heat exchangers are aggravated. The first of these is flow distribution among many parallel channels—a particularly serious concern with boiling and evaporating flows. The second is conjugate effects, circumferential and axial heat conduction in the material forming the channel. These complications make it extremely difficult to ascertain the flow in each channel, and it is virtually impossible to measure the heat flux and temperature distributions around the periphery of each channel.

It should be noted that boiling is desirable with heat sinks for cooling of electronic components, because the wall temperature is more likely to be uniform. This is due to the fact that the wall temperature is constrained to the fluid saturation temperature. In single-phase flow, the wall temperature tracks the fluid temperature, which may rise greatly at high heat input. The pressure drop is to be kept at a reasonable level, for structural reasons and to avoid wall temperature variation as the saturation temperature falls along the channel length. To minimize the pressure drop, the flow rate is so low that large heat inputs result in large enthalpy changes; thus, it is saturated boiling that is of interest. The occurrence of CHF must be regarded as an undesirable condition, as it will cause overheating of an individual channel or even the entire substrate containing the microchannels.

The main purpose of this paper is to highlight the existing CHF studies, deduce the cause of CHF, and give guidelines for new studies.

CHF in Small Diameter Tubes

Several hundred thousand CHF points have been reported in the boiling literature of the past 50 years. These data were overwhelmingly obtained with stable flow (see below) in single, thin-walled, circular tubes. The tubes were usually of uniform wall thickness, and direct electrical heating was utilized to simulate the constant heat flux boundary condition. The electric power (heat flux) was increased slowly until a vapor blanketing occurred, as evidenced by physical burnout of the tube or activation of a burn-out protection device by the increased temperature. In the latter case, the power to the tube was rapidly disconnected before burn-out occurred. Numerous correlations have been proposed for subcooled exit conditions (e.g., [5]) and for boiling with net vapor generation (e.g., [6]). Some studies of CHF have considered axial and/or circumferential flux tilts, usually by machining the tube. The critical heat flux condition is as well defined there as with uniform heating. Fluid heating of uniform-wall tubes has also been used (essentially simulating the uniform wall temperature boundary condition), and the CHF values are similar to those for uniform heat flux.

Single-tube CHF data are not available for microchannels, because of fabrication and instrumentation considerations. At the present time, the only alternative seems to be to derive guidance from the available CHF data in mini channels. Extensive experiments were conducted with subcooled boiling in tubes as small as \( D = 0.3 \text{ mm (300 } \mu\text{m)} \) [7]; however, the mass fluxes—and pressure drop—were very high. As mentioned above, the interest for microchannel heat exchangers is in even smaller channels, with bulk boiling at low mass flux and low pressure.

The modeling of saturated flow boiling CHF in microchannels under stable conditions has not received much attention in the literature. The pool boiling model by Kandlikar [8] employs an additional momentum force caused by the evaporating interface near the heater wall. The resulting nondimensional groups utilizing the evaporating momentum, inertia, and surface tension forces show promise [9]. However, accurate experimental data under stable operating conditions are essential for validating any model.

Flow Distribution in Microchannels

Two-phase heat sinks for cooling of microelectronic components will typically operate with single-phase inlet conditions. The inlet header pressure drop will, therefore, be small compared to the pressure drop in the channels (vaporization and large \( L/D \)). In spite of the fact that the outlet header has two-phase flow, the pressure drop of that header is also relatively small. The result of all this is a uniform flow distribution, provided the heat input is uniform. Even for nonuniform placement of electronic devices, the heat input at the top of the channels will tend to be uniform,
Incipient Boiling

It is important to determine whether there are any barriers to vapor formation in microchannels. There seems to be no reason why conventional theory cannot be used to predict incipient boiling in microchannels. There should be numerous imperfections on the heated surface and sidewalls that act as nucleation sites. These cavities are likely to contain a preexisting gas phase; that is, the wall temperature increased at incipient boiling. The available evidence seems to corroborate the preceding description of nucleation behavior. Bowers and Mudawar [11] reported some hysteresis in their microchannel with R-113. Jiang et al. [12] observed temperature profiles that indicated smooth vaporization of water in silicon channels of 40 or 80 μm hydraulic diameter. In a more recent study, however, Zhang et al. [13] reported unusual, reverse-temperature-overshoot behavior with water; that is, the wall temperature increased at incipient boiling. The magnitude of the temperature overshoot was reduced by etching artificial cavities into the surface. It is noted that there is precedent for such reverse-temperature overshoots with boiling of water [14].

Qu and Mudawar [15] reported a sophisticated analysis of incipient boiling in a microchannel. Their model examines the hydrodynamic and thermal conditions for bubble departure rather than simply the nucleation condition at a cavity. This is because the first bubbles in a low flow condition in a microchannel were observed to depart into the liquid flow, rather than sliding along the wall as they would in larger diameter channels employed in conventional evaporator applications. The shape of the bubbles varied depending whether the bubbles formed at the flat or corner regions of a rectangular channel. Since the bubble has already grown to a rather large size, the heat flux and wall superheat are larger than those for conventional incipient boiling. In any case, microscopic observations of bubble formation in subcooled water are in good agreement with the predictions, confirming the various assumptions in the numerical calculations. The conjugate nature of the heat flow in the copper block was acknowledged in defining the conditions for a large, stable bubble.

Conjugate Effects

There will usually be nonuniform heat flow around the circumference of a rectangular channel. This leads to preferential nucleate boiling from the channel base (heat input side), variations in heat transfer coefficient within the channel, and ultimately initiation of critical heat flux at the hottest surface. Numerous numerical studies of conjugate single-phase flow in microchannels have been carried out, but none have been reported for two-phase heat transfer. A numerical approach seems to be the only option to handle conjugate effects, as it is impossible to place temperature sensors around the circumference of a microchannel, so as to get the local wall temperature and heat flux.

As an example of the experimental difficulty, consider what it takes to get thermal information in simulated cooling channels for the plasma-facing components (PFCs) for the International Thermonuclear Experimental Reactor. Rather than use varying tube wall thickness to create the flux tilts, circular tubes are surrounded by a large block, heated over part of the outside perimeter. Boyd et al. [16,17] described a massive cylindrical test section heated over 180°C by five resistance heaters. Thermocouples placed at 48 stations give the three-dimensional distribution of the wall temperatures and heat fluxes. See Fig. 2. For reference, the tube inside diameter is 10.0 mm. The system is very complex, but it is considered capable of obtaining boiling curves around the channel circumference, including the local CHF. Earlier in the program, it was postulated that three different flow regimes could coexist within the channel: single-phase turbulent flow, subcooled flow boiling, and subcooled film boiling. Recently, Boyd et al. [17] demonstrated this coexistence.

The same phenomenon is expected in blocks with microchannels, the difference being that the small size makes it impossible to install a large number of temperature sensors around the flow channel. Implanted temperature sensors can get representative block temperatures along a channel length, but they are insufficient to calculate local heat transfer coefficients.

Character of Bulk Boiling

In conventional heated channels with bulk boiling, incipient boiling is normally followed by a progression of well-known flow patterns: bubbly flow (first at the wall, then in the core), slug flow, and annular flow (usually annular/dispersed flow). Mist flow follows CHF—that is dryout of the annular film. In microchannels, on the other hand, the bubbles depart—and the departure size is considerably smaller than that in the channel width, according to the visual observations of Qu and Mudawar [15]. They then observed that there was an abrupt transition to annular flow at zero quality [18]. From a variety of published observations, Koo et al. [19] earlier concluded that bubbly flow and slug flow do not exist in microchannels; the flow goes suddenly to annular/dispersed flow. The same group concludes that a mist flow forms immediately after incipient boiling (Jiang et al., [20]; Zhang et al., [13]). It would be more reasonable, however, to envision that the wall remains wetted, perhaps with rivulets or heavy droplet deposition—until some form of dryout (CHF). Predictions of the heat transfer coefficient in annular flow are presented by Koo et al. [19] and Qu and Mudawar [18]. It should be noted that the recommendations of Tabatabai and Faghri [21] appear to be contrary to the above observations; they suggest that only flow patterns dominated by...
surface tension (bubble and slug) occur in tubes of 100 μm diameter. On the other hand, they do acknowledge that when the bubble size approaches the channel size (which is the case in microchannels), the result is annular flow.

The general conclusion from these studies is that annular flow is the dominant flow regime in a two-phase microchannel. It is then reasonable to suggest that the critical heat flux condition is caused by dryout of the annular liquid film. However, this may not cause failure of the cooling system, because of conduction in the microchannel block. Vapor blanketing of the portions of the channels near the heat input will lead to a major redistribution of the heat flow toward the opposite portions of the channels. If those portions are not vapor blanketed, they can efficiently accommodate the heat transfer, and no CHF will be recorded in the channel block.

An important aspect of bulk boiling is the fluctuations in flow and pressure, because these fluctuations can initiate instabilities. Two-phase-flow noise is always present in flow bulk boiling, due to bubble formation or the passage of liquid and vapor. A comprehensive discussion of fluctuations in minichannels is given by Kandlikar [22]. Pressure fluctuations, and associated temperature oscillations, seem to be associated with boiling in microchannels to a greater extent than in conventional channels. This is because the flow velocities are very low, and bubble formation can cause a significant disruption of low-quality flow. Observations of boiling in microchannels are reported by Hetsroni et al. [23]. They reported pressure drop fluctuations of about 1 kPa max and outlet temperature fluctuations of about 1°C max with the dielectric fluid Vertrel XF. Similar small-scale fluctuations were reported for water by Wu and Cheng [24].

CHF Experiments for Microchannels

The pioneering work on CHF in small channel arrays was carried out by Bowers and Mudawar [11]. They had an array of 17 circular channels, 510 μm diameter, 28.6 mm long, in a 1.59-mm-thick nickel block, heated over the central 10 mm. Although these were actually minichannels, the tests were carried out with R-113 at low mass fluxes typical of microchannels (7.0–28.2 kg/m²·s.). Inlet pressure was 1.38 bar, and inlet subcooling ranged from 10 to 32°C. A single thermocouple monitored the average temperature of a 10 mm×10 mm copper block coupling the electric heater to the block with the channels. Good conduction around the circumference of each channel was assumed, so that the heat flux was taken as the heat input divided by the channel wall area underneath the heater. Boiling curves (q'' versus T_w–T_μ) were generated that terminated in well-defined critical heat fluxes. Unusually, the CHF was found to be independent of inlet subcooling, and almost directly proportional to mass flux. Although a dimensionless correlation was proposed, most of the quantities in the correlation were not varied. The bulk fluid condition at the end of the heated section was high quality or even superheated.

Jiang et al. [12] developed a microchannel heat sink integrated with a heater and an array of implanted temperature sensors. There were up to 58 or 34 channels of rhombic shape, having a hydraulic diameter of 40 or 80 μm, respectively, in the 10-mm-wide×20-mm-long test section. Due to the fabrication method, there was no transparent cover plate to view the two-phase flow. CHF data were taken for once-through water entering at 20°C. It appears that the CHF condition was characterized by a rapid rise in the average of all the temperature sensors. The critical power was found to be a linear function of the total volume flow rate, which ranged from 0.25 to 5.5 ml/min, as shown in Fig. 3. Good conduction in the silicon wafers can be assumed, but it is not possible to determine the critical heat flux or the mass flux for the eight data points, because the channel cross-sectional areas are not reported by the authors.

Mukherjee and Mudawar [25] reported CHF data for ultralow...
flow rates of water and FC-72, driven by a natural circulation thermosyphon. Both flat boiling surfaces and finned surfaces (vertical, 0.2 mm width, 0.2 mm spacing, and 0.66 mm height) were used in the 21.3 × 21.3 mm vertical heated surface, i.e., 53 fins. The gap width was varied from 0.13 to 21.5 mm. Thermocouples were installed to measure the fluid inlet, fluid outlet, and heater midpoint temperatures. The CHF data for the smallest gap are indicative of flow in a microchannel array; however, the data are of limited usefulness because there was no direct measurement of either the flow rate or the pressure. It is emphasized that the tests were designed primarily to demonstrate the advantages of the thermosyphon, pumpless system.

In a recent paper, Qu and Mudawar [26] report the first comprehensive study of CHF in rectangular microchannels. The heater block contained 21–215 × 821 μm channels. The heat flux was based on the heated three sides of the channel; in other words, no conjugate effects were considered. De-ionized, deaerated water was supplied over the mass flux range of 86–368 kg/m² s, with an inlet temperature of 30 or 60°C and an outlet pressure of 1.13 bar. In order to eliminate upstream compressible volume instabilities, it was necessary to install a throttle valve upstream of the test section. For these typical microchannel conditions, an unusual phenomenon was observed as CHF was approached; there was vapor backflow from all of the channels into the inlet plenum. This is shown in the author’s sketch, reproduced here as Fig. 4.

The following correlation was developed for the 18 points of this study, as well as R-113 data from the previous study [11]

\[
\frac{q^*}{Gh_{fg}} = 33.43 \left( \frac{\rho_v}{\rho_f} \right)^{1.11} \left( \frac{G^2 \Delta L}{\sigma \rho_f} \right)^{-0.21} \left( \frac{L}{D_h} \right)^{-0.36}
\]

(1)

It is noted that conventional correlations for water grossly overpredict the data, yet it is commonly observed that CHF increases with decreasing diameter. This trend is opposite to that expressed by the correlation. This fact, as well as the vapor backflow from the channel inlets (which contributed to the lack of a subcooling effect), suggests that the data do not actually characterize the real CHF of the channels.

We suggest that all of the CHF tests discussed above were affected by instabilities. The two major instabilities that affect microchannel heat exchangers are upstream compressible volume instability and excursive instability.

**Upstream Compressible Volume Instability**

When there is a significant compressible volume upstream of the heated section, an oscillating flow may lead to CHF. The compressibility could be caused by an entrained air bubble or a flexible hose, but for small channels, a large volume of degassed liquid is sufficient to cause the instability. The only investigators of microchannel heat exchangers reporting problems with upstream compressible volumes were Qu and Mudawar [26,27]. Note, though, that other studies of forced CHF used enclosed channels so that the flow behavior near CHF could not be observed. So, in those studies without an upstream throttle valve, a compressible volume instability was likely responsible for CHF.

The cause of instability with a large upstream compressible volume is a minimum in the pressure drop versus flow rate curve, as demonstrated analytically [28] and experimentally [29]. In other words, the criterion for the constant volume instability (CVI) is

\[
\frac{\partial \Delta P}{\partial \dot{m}_{CVI}} = 0
\]

(2)

Operation at the minimum is unstable, because boiling fluctuations, discussed above, cause a transition to a condition where the heat flux can no longer be accommodated, and CHF is the result. As classified in [30], this is a pressure drop oscillation initiated by a flow excursion (see below), causing a dynamic interaction between the boiling channel and the compressible volume. This instability can be eliminated by isolating the boiling channel from the compressible volume by a throttle valve. Of course, this means an increased system pressure drop.

**Excursive Instability**

Microchannel heat exchangers invariably consist of an array of parallel channels conveying the coolant. The temptation is to design these channels using heat transfer and pressure drop data for single channels that have flow imposed. When boiling is involved, however, this procedure is not valid, because it fails to capture the excursive or Ledineg instability that results from the unique pressure drop characteristics of a boiling channel.

The pressure drop characteristics of a single channel (including entrance and exit losses) are necessary to predict this “hydrodynamic” instability. Typical data for these “demand” curves are shown in Fig. 5; the pressure drop is given as a function of mass flow rate for a fixed geometry, but varying inlet temperature, heat flux, and exit pressure. Although this figure is for subcooled boiling, the general concept holds equally well for bulk boiling. The pressure drop starts out lower than the isothermal value for pure liquid, because of the reduced viscosity at the wall with heating.

As the flow rate is reduced, boiling is initiated, whereupon the pressure drop starts out lower than the isothermal value for pure liquid is sufficient to cause the instability. The only investigators of microchannel heat exchangers reporting problems with upstream compressible volumes were Qu and Mudawar [26,27]. Note, though, that other studies of forced CHF used enclosed channels so that the flow behavior near CHF could not be observed. So, in those studies without an upstream throttle valve, a compressible volume instability was likely responsible for CHF. The cause of instability with a large upstream compressible volume is a minimum in the pressure drop versus flow rate curve, as demonstrated analytically [28] and experimentally [29]. In other words, the criterion for the constant volume instability (CVI) is

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As the flow rate is reduced, boiling is initiated, whereupon the curve starts to turn around. A well-defined minimum is observed, and the curve rises sharply until CHF is observed at “d.” If the experiment were not terminated by CHF, the curve would rise further, eventually reaching a maximum and joining the all-vapor curve.

The key to the parallel-channel behavior is the minimum, that is for the excursive instability (EI)

\[
\frac{\partial \Delta P}{\partial \dot{m}_{EI}} = 0
\]

(3)

If the “supply” curve is that of a centrifugal pump, “A,” there will be two intersections on the “demand” curve: “b” and “c.” Actually, the intersection b is not possible, since operation beyond the minimum cannot occur; at the minimum, a first-order instability occurs, since a perturbation in flow rate (assumed negative) causes an excursion to point “a.” The classification of [30] considers this a fundamental static instability.

As demonstrated analytically and experimentally by Maultsby and Griffith [28], the heat flux for this hydrodynamic instability is considerably lower than the heat flux that would be
obtained with a stabilizing pressure drop at the tube inlet such that the supply curve (pump inlet throttle) has a steeper negative slope than the demand curve chosen, “B.” Prediction of the minimum in the channel pressure drop versus flow rate curve is essential, but sufficient data to do this have been obtained only for minichannels, not microchannels. Once again, the stabilizing pressure drop will require a higher head pump.

Improved microfabrication techniques may make it possible to build flow restrictions at the inlet of each channel to accomplish the inlet pressure drop. Each orifice must be identical. Since the orifice will necessarily be smaller than the channels, it will require a tighter manufacturing tolerance than the channel itself. The papers by Thorsen et al. [31] and Kandlikar and Grande [32] discuss advanced fabrication methods that might be used to realize this.

It is noted that the criteria for instability for both the upstream compressible volume instability and the excursive instability are identical, Eqs. (2) and (3). Furthermore, the cures for the instability are identical: upstream throttling. This means that it would only be necessary to install the inlet orifices to cure both types of instability. On the other hand, it is insufficient to install a valve in the inlet line before the header; that will cure one type of instability but not the other. We suggest this is what happened in the experiments of Qu and Mudawar [26]. They installed the throttle valve to avoid the compressible volume instability, but were still subject to the excursive instability.

The first step in confirming that this phenomenon is prevalent in microchannel CHF, especially the observations of [26] as shown in Fig. 4, is to show that there is a minimum in the pressure drop curve. The model and earlier data of Koo and co-workers [19] shown in Fig. 6 demonstrate a well-defined minimum. It is emphasized that they obtained these results for a single channel; this must be the case, as parallel channels would exhibit CHF at the minimum. Although these results are for pressure drop versus power input, they can be transformed to pressure drop versus flow rate [33], also with a conspicuous minimum. It is concluded that microchannels with vaporization are prone to excursive instability. The excursive instability results in a lower CHF than would be obtained with stable flow in the individual channels.

Postdryout Behavior

The heat transfer coefficient beyond the CHF condition, whatever the cause, is needed to assess the consequences of exceeding the critical heat flux. Jiang et al. [12] report the CHF temperature excursions for the test section described earlier, and for data at flow rates of water less than 2 ml/min, as shown in Fig. 3. The postdryout wall temperatures were 185–250°C. While these temperatures are rather high, they may not result in destruction of the heat sink, unless gaskets, bonds, and the electronic device itself fail. Note that this is the situation for compressible volume or excursive instability; postdryout temperatures would be higher for true CHF.

Instrumentation

The experience of previous investigators provides meaningful guidance for the CHF instrumentation of microchannel heat exchangers. Visual observations of the boiling in rectangular channels through a transparent cover plate are useful to infer flow patterns, flow stagnation, and flow reversal. However, given the small dimensions of the channels and two-phase phenomena, such
as bubbles, microscopic observation must be used. Even with this, it is difficult to distinguish important features, such as the evaporation film in annular flow [23].

In terms of quantitative measurements, the heat input to the channel should be accurate. This usually means correcting for the heat losses to the ambient from actual electronic devices or electrical heaters. There are several ways to express the heat flux. The overall performance can be expressed as a heat flux based on the area of the heat input to the microchannel array. However, the channel behavior requires the effective channel surface area. Because of good conduction in the microchannel block, this will usually be the channel area exposed to the fluid (excluding the cover plate).

A knowledge of the channel surface temperature and heat flux would be desirable; however, thermocouples or other sensors cannot be readily installed around a channel to get the local behavior. The smallest practical thermocouple (36 gauge) has a bead size exceeding the upper limit of channel size (100 µm gap for a rectangular channel or hydraulic diameter equal to 200 µm). Multiple thermocouples [12] are preferable to a single thermocouple recording the entire block temperature [11]. They can be placed in the side near the heater, along the flow direction. Due to expected uniform flow distribution, it is necessary to do this only for a single channel.

Experimental uncertainties are another area where particular attention needs to be given. The small passage dimensions, and associated errors in measuring temperature differences, heat fluxes, and pressure gradients make it extremely difficult to achieve a high degree of accuracy with conventional measurement techniques.

Conclusions

This paper has considered major issues associated with the use of microchannels. The emphasis is on bulk boiling heat transfer, which occurs when high heat loads are to be accommodated in microelectronic heat sinks. The upper limit of heat flux is the critical heat flux (CHF).

Existing small-tube CHF data are inapplicable to microchannels, mainly because of being taken at high mass fluxes.

Flow distribution in parallel microchannels is not considered to be a problem with the usual subcooled inlet conditions.

There appear to be no barriers to incipient boiling in microchannels, so vapor will appear without difficulty. Water is clearly not a problem, but highly wetting fluids may experience temperature overshoots.

Conjugate effects may become important in microchannels such that CHF occurs in the region near the heater, whereas the opposite side of the channel may have conventional boiling. Many microchannel blocks have a high thermal conductivity and the heat transfer coefficient is rather uniform circumferentially; thus, the heat flux is uniform around the circumference (except for the adiabatic cover plate in the case of rectangular channels).

The few available CHF experiments for parallel microchannels are discussed. The database is only of order 10, as opposed to conventional channels having a database of order 100,000. Clearly, more data are needed. But, it is important to recognize the character of CHF in microchannels that have been studied to date. The case is made that all of the available CHF data were taken under unstable conditions. The critical condition is the result of an upstream compressible volume instability or the parallel channel, Ledinegg instability. As a result, the CHF values are lower than they would be if the channel flow were kept stable by an inlet restriction at the inlet of each channel. A proper inlet restriction can eliminate both kinds of instability. It is suggested that the next step in microchannel research be the fabrication and study of the effects of such orifices.

Acknowledgment

It is indeed a pleasure to acknowledge the contribution of Richard C. Chu to this paper. Nearly 40 years ago, he caused the career of A.E.B. to include cooling of computers, particularly “high end” thermal management. Microchannels with vaporization are a natural outgrowth of the many studies promoted by Dick Chu. The second author, Satish Kandlikar, would like to gratefully acknowledge the continued IBM faculty award for conducting research in this area.

Nomenclature

\[ D = \text{tube diameter, m} \]
\[ D_h = \text{hydraulic diameter, m} \]
\[ G = \text{mass flux, kg/m}^2\text{s} \]
\[ h_{fg} = \text{latent heat of vaporization, J/kg} \]
\[ L = \text{length of channel, m} \]
\[ P_h = \text{pressure, bar} \]
\[ \Delta P = \text{heated section pressure drop, kPa} \]
\[ Q = \text{volumetric flow rate, ml/min} \]
\[ q = \text{power, W} \]
\[ q_* = \text{power at CHF, W} \]
\[ q_*/A = \text{heat flux, W cm}^{-2} \]
\[ T_{in} = \text{inlet temperature, °C} \]
\[ T_w = \text{wall temperature, °C} \]
\[ w = \text{mass flow rate, kg/s} \]

Greek Symbols

\[ \rho_1 = \text{density of liquid, kg/m}^3 \]
\[ \rho_v = \text{density of vapor, kg/m}^3 \]
\[ \sigma = \text{surface tension, N/m} \]

References


