TWO-PHASE FLOW PATTERNS, PRESSURE DROP AND HEAT TRANSFER DURING BOILING IN MINICHANNEL AND MICROCHANNEL FLOW PASSAGES OF COMPACT EVAPORATORS

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ABSTRACT

The small hydraulic diameters employed during flow boiling in compact evaporator passages are becoming more important as they are employed in diverse applications including electronics cooling and fuel cell evaporators. The high pressure drop characteristics of these passages are particularly important as they alter the flow and heat transfer, especially in parallel multi-channel configuration. The pressure drop oscillations often introduce dryout in some passages while their neighboring passages operate under single-phase mode. This paper presents a comprehensive review of literature on evaporation in small diameter passages along with some results obtained by the author for water evaporating in 1-mm hydraulic diameter multi-channel passages. Critical heat flux is not covered in this paper due to space constraint.

1. INTRODUCTION

Major progress in compact evaporator development has been made by the automotive, aerospace, and cryogenic industries over the last fifty years. The thermal duty and the energy efficiency increased during this period, while the space constraints became more vital. The trend was toward greater heat transfer rates per unit volume. The hot side of the evaporators in these applications was generally air, gas or a condensing vapor. The airside fin geometry also saw significant improvements resulting from increased heat transfer coefficients as well as greater surface area densities.

As the airside heat transfer resistance decreased, more aggressive fin designs were employed on the evaporating side, resulting in narrower flow passages. The narrow refrigerant channels with large aspect ratio were brazed in small cross-ribbed sections to provide a better refrigerant distribution along the width of the channel. The major changes in the recent evaporator designs involve individual, small hydraulic diameter flow passages, arranged in multi-channel configuration for the evaporating fluid.

Figure 1 shows a plate-fin evaporator geometry widely used in compact refrigerant evaporators. The refrigerant side passages are made from two plates brazed together, and airside fins are placed between two refrigerant flow passages. The plates have cross ribs that are oriented in opposite directions so that the top and bottom plates forming a refrigerant passage have only contact points at the intersection of these ridges. The two-phase flow of refrigerant gets distributed across the width of the flow passage. This feature is important in that the localized flow oscillations, caused by nucleate boiling and expanding vapor bubbles, is dissipated across the width and does not affect the upstream flow.

Figure 2 shows two geometries being used more widely in compact evaporators. The parallel minichannel geometry shown in Figure 2a is used extensively in condensation applications, whereas the geometry shown in Figure 2b has received quite a bit of attention for boiling applications. The channels are fabricated by a variety of processes depending on the dimensions and plate material. Conventional machining and electrical discharge machining are two typical options, while semiconductor fabrication processes are appropriate for microchannel fabrication in chip cooling application.

2. FLOW PATTERNS IN SMALL DIAMETER TUBES

2.1 Previous Studies

Visualization of boiling phenomena inside flow channels provides insight into the heat transfer mechanisms. Bubble formation, coalescence, formation of slugs or plugs, and local dryout conditions, are all important in understanding the heat transfer phenomena. Although some of the heat transfer and pressure drop equations employed in the design of commercial equipment are derived from flow pattern based models, the major benefit of flow pattern information lies in understanding the causes for premature dryout or CHF condition in an evaporator. Another major benefit is in the design of the inlet and the outlet manifolds in multi-channel evaporators.

The flow pattern maps available in literature were first developed for the petrochemical industry (Baker, 1954) for flow of oil and gas in large diameter pipes. Subsequently, the adiabatic flow pattern maps were developed as general flow pattern maps (for example, Hewitt and Roberts, 1969, and Taitel et al., 1980). In recent years, a number of flow pattern maps have been developed for specific conditions such as small diameter tubes, evaporation or condensation, and compact heat exchanger geometries.

Earlier investigators extensively studied flow patterns for gas-liquid flows in channels with small hydraulic diameters. A representative survey of the flow patterns was presented by Fukano et al. (1989). They identified bubbly, plug, slug, and annular flow patterns and compared the flow pattern transitions with the available flow pattern maps. Subsequently, a detailed study by Wambsganss et al. (1991) provides a more comprehensive summary and representation of gas-liquid flow patterns. The role of surface tension becomes more important in

smaller diameter channels. Triplett et al. (1999) explain that due to the dominance of surface tension, stratified flow is essentially absent, slug (plug) and churn flow patterns occur over extensive ranges of parameters, and the slip velocity under these patterns is small. Stratified flow can exist at very low flow rates, as observed by Kasza et al. (1997) for a mass flux of 21 kg/m^{2}s.

Hewitt (2000) gives a comprehensive summary of flow pattern studies available in literature. For large diameter tubes, the generalized flow pattern map for an air-water system, developed by Mandhane et al. (1974), was quite representative of other flow conditions as well. However, the theoretically based transition
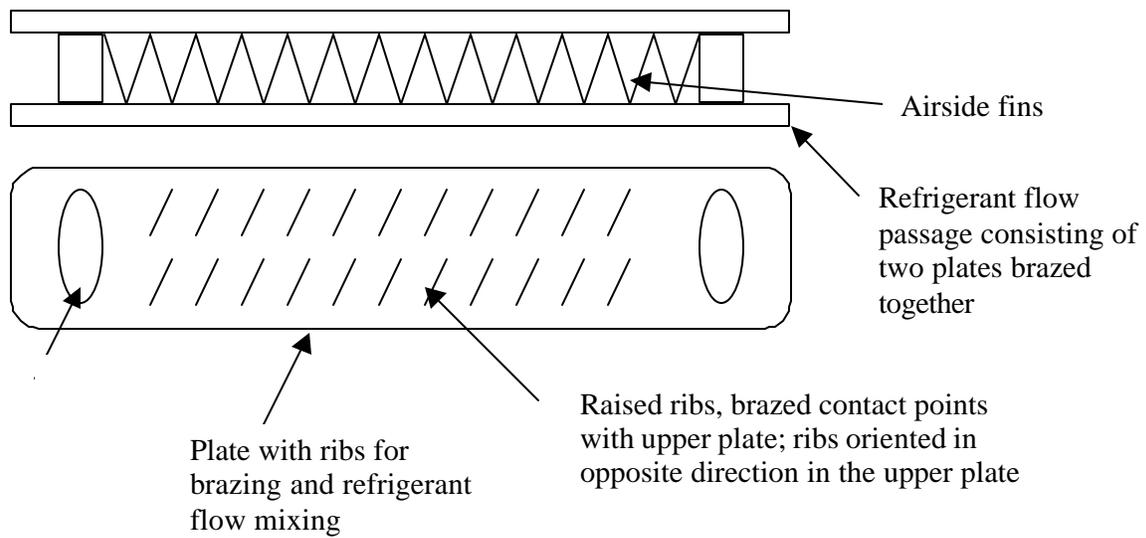
Airside fins

Refrigerant flow passage consisting of two plates brazed together

Raised ribs, brazed contact points with upper plate; ribs oriented in opposite direction in the upper plate

Plate with ribs for brazing and refrigerant flow mixing

Figure 1 Schematic of refrigerant and airside flow passages in a compact plate-fin type evaporator

Parallel mini-channels
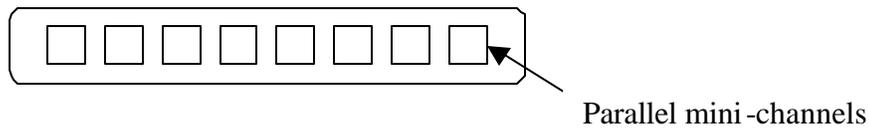
(a) Schematic cross sectional view of a multichannel evaporator with parallel mini-channels

Refrigerant manifold

Parallel mini- or micro-channels
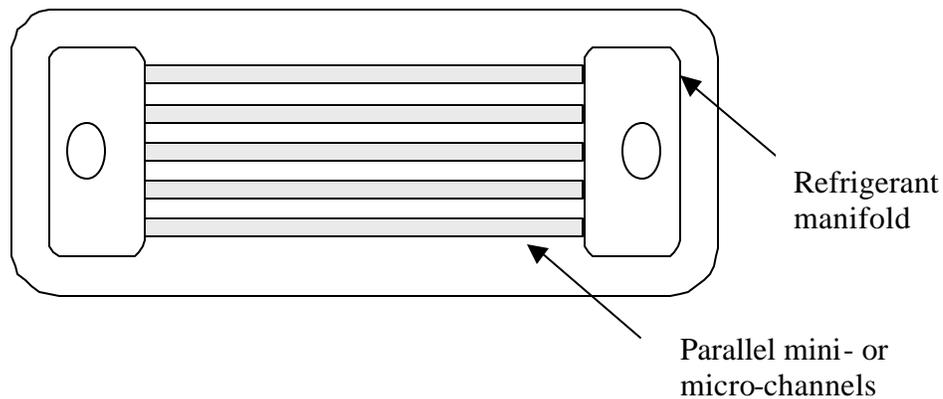
(b) Schematic of a multichannel evaporator with parallel mini- or micro-channels

Figure 2 Recent developments in multichannel evaporators

TABLE 1 SUMMARY OF INVESTIGATIONS ON EVAPORATION IN MINI- AND MICROCHANNELS

Author/ Year	Fluid and Ranges of parameters, G-kg/m^{2}s, q-kW/m^2	Channel Size/ D$_h$, mm, Horiz. (unless otherwise stated)	Heat Transfer	Pressure Drop	Flow Patterns	Remarks
Lazarek and Black, 1982	R-113, G=125-750. q=14-380	Circular, D=3.1, L=123 and 246	Heat transfer coeff. and CHF	pressure drop measured and correlated	Not observed	Subcooled and saturated data obtained, h almost constant in the two-phase region, dependent on heat flux. Behavior similar to large dia. tubes-combination of nucleate and convective boiling. Pressure drop correlations obtained.
Cornwell and Kew, 1992	R-113, G=124-627, q=3-33	Parallel rectangular, 75 channels-1.2x0.9 deep 36 channels-3.25x1.1 deep	Heat transfer coeff	Not reported	Isolated bubble, Confined bubble, annular-slug	The heat transfer coefficient was found to be dependent on the existing flow pattern. In the isolated bubble region, h~q$^{0.7}$, lower q effect in confined bubble region, convection dominant in annular-slug region
Moriyama and Inoue, 1992	R-113, G=200-1000, q=4-30	Narrow rectangular channel, 0.035-0.11 gap, width=30, L=265	Heat transfer coeff.	Pressure drop measured and components calculated	Flattened bubbles, w/coalescence, liquid strips, liquid film	Two-phase flow boiling data in narrow gaps obtained and correlated with an annular film flow model. Nucleate boiling ignored, although a dependence of h on q was observed.
Wambsganss et al. (1993)	R-113 G=50-100, q=8.8-90.7	Circular, D=2.92 mm	h as a function of x, G and q	Not reported	Not reported	Except at the lowest heat and mass fluxes, both nucleate boiling and convective boiling components were present.
Bowers and Mudawar, 1993, 1994	R-113, q=1000-2000, 0.28-1.1 ml/s	Mini and micro channels, D=2.54 and 0.51	Heat transfer rate	Pressure drop components	Not studied	Analytical (1993) and experimental (1994)studies comparing the performance of mini and micro channels. Mini-channels are preferable unless liquid inventory or weight constraints are severe.
Mertz, et al., 1996	Water and R-141b, G=50, 100, 200, 300 q=3-227	Rectangular channels, 1, 2 and 3 mm wide, aspect ratio up to 3	Heat transfer coeff and heat flux	Not measured	Nucleate boiling, confined bubble and annular	Single and multi-channel test sections. Flow boiling pulsation was observed in multi-channels, with reverse flow in some cases. Nucleate boiling dominant.
Ravigururajan et al., 1996	R-124, 0.6-5 ml/s, 20-400W	270 μm wide, 1 mm deep. 20.52 mm long,	Heat transfer coeff.	Not studied	Not studied	Experiments were conducted over 0-0.9 quality and 5 °C inlet subcooling. Wall superheat from 0-80 °C
Tran, et al., 1996	R-12, G=44-832, q=3.6-129	Circular, D=2.46, Rectangular, D$_h$=2.4	Heat transfer coeff.	Not studied	Not studied	Local heat transfer coeff. obtained up to x=0.94. Heat transfer in nucleate boiling dominant and convective dominant regions obtained. New correlation for nucleate boiling dominant region
Kasza et al., 1997	Water, G=21, q=110	Rectangular 2.5x6.0x500	Not reported	Not reported	Bubbly, Slug	Increased bubble activity on wall at nucleation sites in the thin liquid film responsible for high heat transfer in small channels

Author/Year	Fluid and Ranges of parameters, G-kg/m²s, q-kW/m²	Channel Size/ D_h, mm, Horiz. (unless otherwise stated)	Heat Transfer	Pressure Drop	Flow Patterns	Remarks
Tong, Bergles and Jensen, 1997	Water, G=25,000-45,000, CHF of 50-80 MW/m²	Circular, D=1.05-2.44	Subcooled flow boiling	Focus on pressure drop	Not studied	Pressure drop measured in highly subcooled flow boiling, correlations presented for both single phase and two-phase
Bonjour and Lallemand, 1998	R-113, q=0-20	Rectangular, 0.5 to 2 mm gap, 60 wide, 120 long, Vertical	Not studied	Not studied	3 flow patterns with nucleate boiling	Nucleate boiling with isolated bubbles, nucleate boiling with coalesced bubbles and partial dryout, criteria proposed for transitions
Table 1 continued						
Author/ Year	**Fluid and Ranges of parameters, G-kg/m²s, q-kW/m²**	**Channel Size/ D_h, mm, Horiz. (unless otherwise stated)**	**Heat Transfer**	**Pressure Drop**	**Flow Patterns**	**Remarks**
Peng and Wang, 1998	Water, ethanol and mixtures,	Rectangular, width=0.2-0.4, height=0.1-0.3, L=50; Triangular, D_h=0.2-0.6, L=120	Heat transfer coeff., heat flux	Not reported	Not observed	The heat transfer results indicate that both, nucleate and forced convection boiling are present. No bubbles were observed, and the authors propose a fictitious boiling. The authors did not use proper microscope and high-speed video techniques resulting in contradictory conclusions.
Kamidis and Ravigururajan, 1999	R-113, power 25-700 W, Re=190-1250	Circular, D=1.59, 2.78, 3.97, 4.62	Single and two-phase, max h~11 kW/m²C	Not reported	Not studied	Extremely high heat transfer coefficients, up to 11 kW/m²C were observed. Fully developed subcooled boiling and CHF were obtained.
Kuznetsov and Shamirzaev, 1999	R-318C, G=200-900, q=2-110	Annulus, 0.9 gap x 500	h~1-20 kW/m²C	Not studied	Confined bubble, Cell, Annular	Capillary forces important in flow patterns, thin film suppresses nucleation, leads to convective boiling
Lin et al., 1999	R-141b, G=300-2000 q=10-150	Circular, D=1	Heat transfer coeff.	Not studied	Not studied	Heat transfer coeff. obtained as a function of quality and heat flux. Trends are similar to large tube data
Downing et al., 2000	R-113, Ranges not clearly stated	Circular multi-channels in helical coils, D_h= 0.23-1.86, helix dia=2.8-7.9	Not studies	Single and Two-phase	Not studied	As the helical coil radius became smaller, pressure drop reduced – possibly due to rearrangement in flow pattern
Hestroni et al., 2000	Water, Re=20-70, q=80-360	Triangular parallel chann, θ=55° n=21,26 D_h=0.129-0.103, L=15	Measured, but data not reported	Not reported	Periodic annular	Periodic annular flow observed in microchannels. There is a significant enhancement of heat transfer during flow boiling in microchannels.
Lakshmi-narasimhan, et al., 2000	R-11, G=60-4586,	Rectangular, 1x20x357,	Subcooled and saturated flow boiling	Not measured	Boiling incipience observed through LCD	Boiling front observed in laminar flow, not visible in turbulent flow due to comparable h before and after, saturated flow boiling data correlated by Kandlikar (1990) correlation
Kandlikar, et al.,	Water,	Rectangular,	Subcooled and	Measured instants	High-speed photography	Flow oscillations and flow reversal linked to the severe pressure drop

et al., 2001	G=80-560 kg/m^{2}s	1x1x60 mm	and saturated flow boiling	instanta-neous values of pressure drop	photography on single and multiple channels	linked to the severe pressure drop fluctuations, often leading to flow reversal during boiling.

criteria presented by Taitel and Dukler (1976) is one of the most widely used flow pattern maps. Hewitt (2000) notes that both evaporation and condensation processes have a significant effect on the flow patterns.

The effect of evaporation on the flow pattern transitions was considered to be quite small in large diameter tubes. This is one of the reasons why the flow pattern studies from gas-liquid systems, such as air and water, were extended to the case of evaporation. In smaller diameter tubes, the effect of evaporation could be quite dramatic.

The evaporation of the liquid phase affects the flow in two ways. Firstly, it alters the pressure drop characteristics by introducing an acceleration pressure drop component that can be quite large at higher heat fluxes. Secondly, the tube dimension is quite small, and the effect of surface tension forces become more important in defining the two-phase structure.

Table 1 also includes the flow pattern studies for circular tubes and narrow rectangular channels with hydraulic diameters on the order of 3 mm or smaller. Cornwell and Kew (1992) conducted experiments with R-113 flowing in 1.2 mm x 0.9 mm rectangular channels. The parallel channels were machined in aluminum and a 6-mm thick glass plate was used to observe the flow pattern. In the flow ranges investigated, three flow patterns were observed as shown in Fig. 3. They also observed a strong link between the flow pattern and the heat transfer coefficient. In the isolated bubble region, $h \propto q''^{0.7}$, indicating the dominance of nucleate boiling. When the bubbles occupy the entire channel cross-section in the confined bubble region, h dependence on q decreased. The convection effects were dominant in the annular-slug region. In a subsequent study, Lin et al. (1998) compared the flow region transitions with those predicted by Barnea et al. (1983), developed for evaporating steam-water system in 4 mm diameter tubes, and Mishima and Hibiki (1996), developed for adiabatic air-water system in 2.05 mm diameter tube. The results indicate that the flow pattern maps developed for air-water flow are in general applicable, but the transition boundaries need to be refined for flow boiling in small diameter tubes and channels.
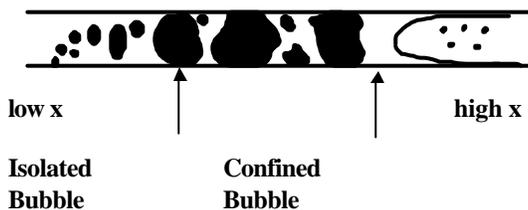

low x   **high x**

**Isolated   Confined
Bubble   Bubble**

Figure 3 Flow Patterns Observed by Cornwell and Kew (1992) in 1.2 mm x 0.9 mm parallel rectangular channels

Moriyama and Inoue (1992) conducted experiments in single narrow rectangular channels with R-113. They observed (i) flattened bubbles, confined by the flow channel (some of them coalescing), (ii) liquid strips flowing along the wall, and (iii) liquid film flow.

Mertz et al. (1996) investigated single and multi-channel test sections with water and R-141b flowing in rectangular multi-channels, 1, 2 and 3 mm wide. They observed the presence of nucleate boiling, confined bubble flow and annular flow and discovered that the bubble generation in the channels was not a continuous process. In addition, the vapor seemed to stay in the channels blocking the flow, and in some cases, causing a reverse flow to occur in the channels. In both single-channel and multi-channel cases, large pressure pulsations were noticed.

Kasza et al. (1997) present a detailed study on flow visualization of nucleation activity in a rectangular flow channel of 2.5 mm x 6 mm cross section. They viewed the flow using a high speed video camera with a maximum frame rate of 12,000 frames per second. The mass flux was 21 kg/m²s. They observed nucleate boiling on the wall, as the individual bubbles nucleated and grew on the wall. The bubble growth rates were similar to those in pool boiling case when the bubbles grew without interacting with the wall or any liquid-vapor interface.

Kasza et al. made interesting observations on individual bubbles and their interaction with other bubbles and vapor slugs. The vapor slugs were separated from the wall by a thin liquid film with a 0.67-mm average thickness. Nucleation was observed in this liquid film. The bubbles growing in this film did not easily coalesce with the vapor in the slug. Bubbles growing under the liquid slug were flattened and covered a larger wall area compared to those growing in the liquid flow. The heat transfer in the thin liquid microlayer underneath the bubbles improved the heat transfer; the bubble frequency and the vapor volume both increased for such bubbles. Their findings clearly indicate the occurence of nucleate boiling in thin liquid films that exist in both slug flow and annular flow conditions.

Bonjour and Lallemand (1998) report flow patterns of R-113 boiling in a narrow space between two vertical surfaces. The flow patterns observed are similar to those observed by other investigators: isolated bubbles, coalesced bubbles, and partial dryout. Comparing the bubble dimensions with the channel size is vital in determining whether the small channel size influences the bubble growth and leads to the confined bubble flow pattern. Following the work of Yao and Chang (1983), the Bond number,

$$= e\left(\frac{s}{g(r_L - r_G)}\right)^{-1/2}$$

, along with the Boiling number, $= q/(Gh_{fg})$, provided the basis for the transition from individual

to confined bubble flow. For smaller Bond number, the channel dimension is smaller than the departure bubble diameter resulting in confined bubble flow pattern. For large Bond numbers, the channel size does not interfere with the bubble flow. A more detailed treatment of the forces acting on the bubble is needed to clearly identify this boundary.

Kuznetsov and Shamirzaev (1999) studied the flow patterns during flow boiling of R318C in a 0.9-mm annular gap between two circular tubes. This work was an extension of previous research on air-water systems by Kuznetsov and Vitovsky (1995). The isolated bubble region was called as the small bubble region. It was followed by long Taylor's bubbles, similar to confined bubbles (defined by earlier investigators), which were elongated in the flow direction. These bubbles spread along the periphery of the annular gap and formed a cell structure referred to as the cell flow regime. As the vapor quality increased, the annular flow pattern was established with a ripple of waves.

Hestroni et al. (2000) studied the evaporation of water in multi-channel evaporators. The evaporators consisted of 21-26 parallel flow passages with channel hydraulic diameters of 0.103-0.129 mm. They observed periodic behavior of the flow patterns in these channels. The flow changed from single-phase flow to annular flow with some cases of dryout. The dryout, however, did not result in a sharp increase in the wall temperature. This clearly indicates that there is still some evaporating liquid film on the channel walls that could not be observed in the video images. They also reported the presence of vapor phase in the inlet plenum.

Lakshminarasimhan et al. (2000) studied the flow boiling in a narrow rectangular channel, 1 mm high × 20 mm wide × 357 mm long. They used LCD crystal display on the heated wall to observe the nucleation front and locate the onset of nucleate boiling. The ONB was clearly identified in the laminar flow region with subcooled R-11 entering the flow channel. As the flow rate increased to the turbulent region, the ONB could not be identified through this technique due to the high heat transfer coefficient in the subcooled single phase region. However, it is possibile that the ONB may have occurred at discrete locations rather than appearing as a clear identifiable front.

## 2.2 Discussion on Flow Patterns and Flow Pattern Transitions in Small Diameter Channels

The three flow patterns shown earlier in Fig. 1 represent the characteristic flow patterns associated with two-phase flow in minichannels. It is clear that discrete bubbles, resulting from nucleation activity on the wall, are present in the subcooled boiling and low quality regions. The presence of nucleation in the small diameter tubes is also evident through the available studies. The observations of nucleating bubbles in the thin liquid films by Kasza et al. (1997), shown in Fig. 4, are also very revealing.

The experimental conditions employed by Kasza et al. represent low Reynolds number conditions. With their hydraulic

diameter of 3.53 mm and a mass flux of 21 kg/m^{2}s, the single-phase Reynolds number is 262 at one atmosphere pressure. The shear stress effects under these conditions are expected to be quite low, and it should not come as a surprise that the growth rate exponent for nucleating bubbles reflects that under pool boiling conditions. In general, most of the visual studies are for low flow rates as the bubble activity cannot be easily traced at higher mass fluxes.

Kandlikar and Stumm (1995), and Kandlikar and Spiesman (1997) demonstrated presence of nucleating bubbles under highly sheared flow conditions in a rectangular channel. It was noted that bubble nucleation occurred when the nucleation criterion for available cavity sizes was satisfied. The effect of flow and wall temperature on bubble characteristics is illustrated by Kandlikar (2000) and is shown in Figs. 5 and 6.

Figure 5 shows the effect of flow Reynolds number on the bubble growth rate. As the flow velocity increased, flow changed from laminar to transition region with an associated increase in the single phase heat transfer coefficient. This caused the bubble to
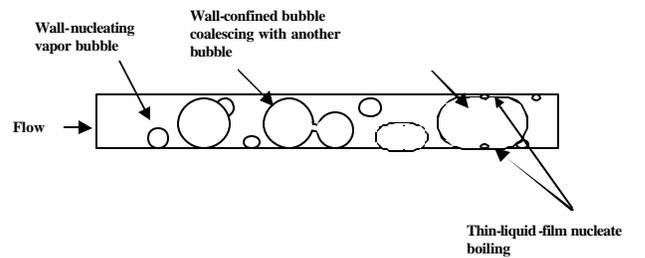
(a) Flow patterns
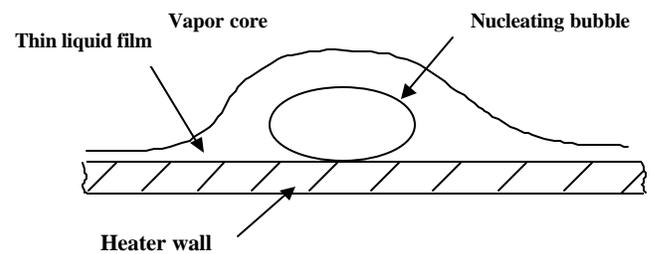
(b) Nucleation in the liquid film under a vapor slug

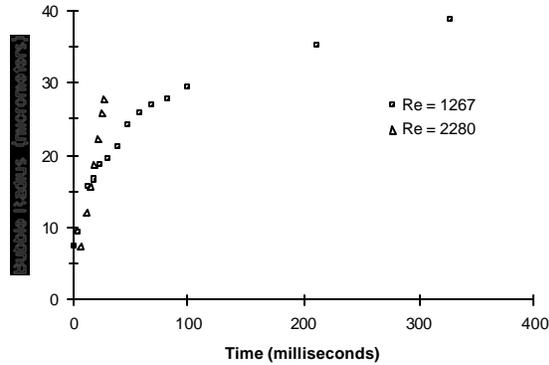**Figure 4 – Flow patterns and bubble nucleation in the liquid film observed by Kasza et al. (1997)**

Figure 5 – Effect of flow on bubble growth, subcooled flow of water at 1 atmospheric pressure in 3 mm x 40 mm rectangular channel, T$_{wall}$=108 °C, T$_{bulk}$=80 °C, cavity radius 3.2 mm, Kandlikar (2000).
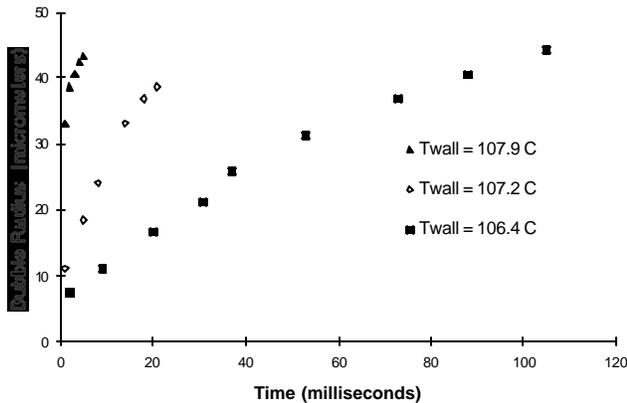

Figure 6 – Effect of wall temperature on bubble growth, subcooled flow of water at 1 atmospheric pressure in 3 mm x 40 mm rectangular channel, T$_{wall}$=108 °C, T$_{bulk}$=80 °C, cavity radius 3.2 mm, Kandlikar (2000).

grow much faster, reaching a departure condition in about 25 millisecond, at Re=2280, as opposed to 300+ millisecond at Re=1267. The departure bubble diameter is also influenced by the flow. As the flow velocity increases, the drag forces cause the bubbles to depart at smaller diameters.

Figure 6 further illustrates the sensitivity of the wall temperature conditions on the bubble growth rates. A higher wall temperature, with a greater associated heat flux, causes the bubbles

to grow rapidly and reach the departure conditions much sooner. The departure sizes appear to be more dependent on the flow velocity for the conditions depicted in these figures.

From the observations above, it may be concluded that the nucleating bubbles are present in flow boiling under high shear conditions. Kasza et al. (1997) observed such bubbles in thin films, shown in Fig. 4b, confirming that nucleation can occur in annular flow as well. Under these conditions, the nucleating bubble size decreases, and bubble departure frequency increases. This further confirms the conclusions made by Kandlikar and Stumm (1995) that a high speed camera with suitable magnification is needed to observe the nucleating bubbles (a) under high shear stress conditions, and (b) at high wall temperatures. In fact, the use of high-speed photography is essential in clearly observing the flow patterns in small diameter tubes.

The bubbles departing in the flow can exist as individual bubbles unless their size is smaller than the channel dimension normal to the nucleating surface. Further growth of these bubbles results in their confinement by the channel walls under the confined flow pattern. In reality, the confined flow pattern is similar to the early stages of the plug flow pattern seen in the conventional two-phase flow patterns for larger diameter tubes (>3 mm). Annular flow pattern then follows at higher qualities.

Flow Instabilities:

The flow instability is another concern in the design of evaporators that employ small channels. The flow instabilities can be reduced considerably by increasing the upstream pressure drop prior to the flow entering a channel. A large diameter flow section however generally precedes the test section in a number of experimental studies conducted on gas liquid flows in small diameter tubes. In the study conducted by Lin et al. (1998), air was introduced in a large mixing chamber upstream of the test section to reduce the disturbances resulting from gas injection. The presence of such low pressure drop regions immediately before the test section leads to significant flow instabilities that cause large pressure drop excursions, and occasionally result in a negative pressure drop with a corresponding flow reversal in the channel.

The instability occurs in the negative pressure drop region of the demand curve plotted as the pressure drop versus inlet flow velocity of the subcooled liquid. Kennedy et al. (2000) studied the onset of flow instability and noted that instability sets in at a slightly lower mass flux than the onset of significant void condition.

Cornwell and Kew (1992) conducted experiments on flow boiling of R-113 in parallel microchannels. They observed that the flow was unstable at lower flow rates. The pressure drop fluctuations were not reported.

Kandlikar et al. (2001) viewed the flow boiling of water in electrically heated multi-channel evaporators consisting of six parallel channels. The flow structure was visualized using high speed video camera up to a maximum speed of 1000 frames per second. The typical bubbly flow, slug flow and annular flow

patterns were observed. Nucleation was also observed in the bulk liquid as well as in the liquid film. However, the most interesting discovery made, in an attempt to understand the severe pressure fluctuations (described later under Pressure Drop section), was a visual confirmation of complete flow reversal in some of the channels.

Figure 7 shows the schematic of the multi-channel evaporator investigated by Kandlikar et al. (2001). The evaporator was heated electrically from the back wall of the test section. The front part was covered with a high temperature resistant glass for flow visualization.

Figure 8 shows the sequence of the events that lead to flow reversal in the flow channels. Two adjacent central channels are shown at 2 ms intervals in frames (a) through (e). Both channels (1) and (2) exhibit slug flow in the visible section.
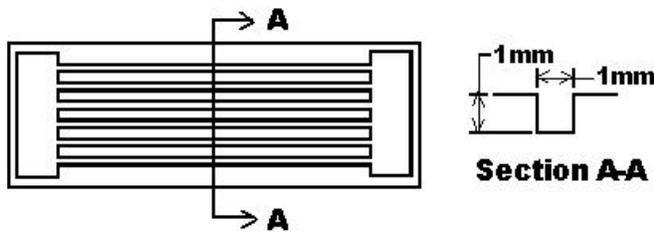

Figure 7 Multi-channel evaporator investigated by Kandlikar et al. (2001)

Figure 8 – Successive Frames (a) through (e) at 2ms Intervals of Two Channels Interacting- G=40kg/m^{2}s, Surface Temperature=110.6°C, Entering Water Temperature=24.7°C, Outlet Water Temperature=99.3°C, x>0

Vertical lines (y) and (z) indicate the initial boundaries of a vapor slug in channel (2) of Fig. 8(a). Vertical lines (x) and (w) are reference lines to aid visualization of slug motion throughout the frames.

In Fig. 8(b), channel (1) has flow in the direction of bulk flow (left to right). The vapor slug in channel (2) has expanded in the direction of bulk flow yet the inlet-side fluid/vapor interface has not moved. The fluid/vapor interface on the inlet side of the slug in channel (2) is still stationary in Fig. 8(c) although the outlet-side interface moves in the bulk flow direction. In Fig. 8(d) the flow in channel (1) moves along the direction of bulk flow, but the inlet-side fluid/vapor interface of the slug in channel (2) progresses in the direction counter to the bulk flow. In Fig. 8(e), the inlet fluid/vapor interfaces in both channels move in the direction counter to bulk flow. For this particular case, it appears that both of the channels experience a vapor-clogging condition where the differential pressure across the channels increases due to vapor generation, the mass flow rate is consequently reduced through
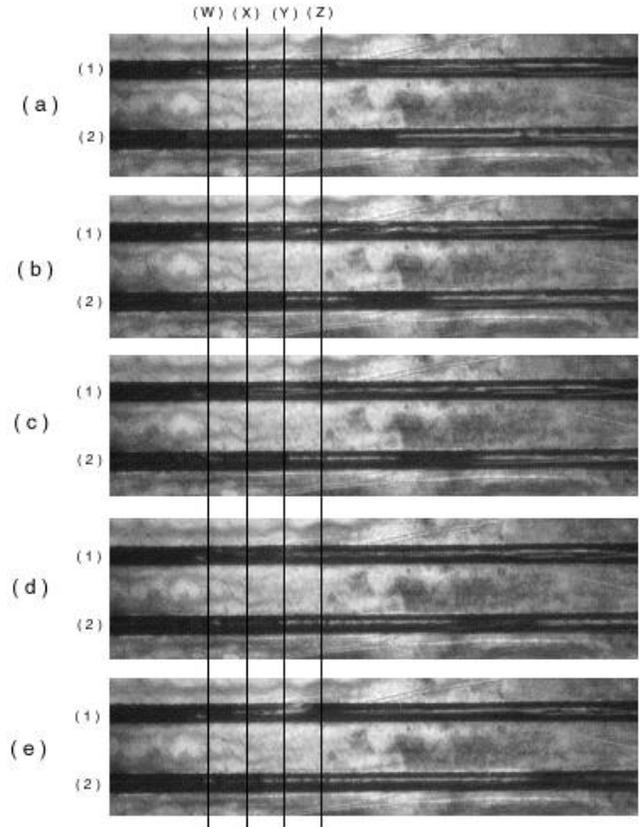
these two channels. The reaction to this condition in the other four channels would be an increased flow rate.

Concluding Remarks on Flow Patterns:

The flow patterns observed in the small channels indicate that the nucleating bubbles play an important role in small diameter channels. The three predominant flow patterns are- (a) Isolated bubbles, (b) Confined bubbles, and (c) Annular-Slug. Note that the flow patterns under high mass flux conditions (G>500 kg/m^{2}s) have not been studied in literature due to difficulty in capturing the high-speed movement under these conditions. Further work in this area is recommended.

3.   PRESSURE DROP IN SMALL DIAMETER FLOW BOILING CHANNELS

3.1 Single Channel

Pressure drop in small diameter tubes has been studied by a number of investigators. Lazarek and Black (1982) conducted systematic experiments to evaluate the three components of pressure drop. The desired quality was generated in the heated inlet section and the frictional pressure drop was measured under adiabatic conditions in the discharge section. The frictional pressure was correlated using a correlation recommended by Collier (1981) with the Martinelli parameter c_{tt}:

$$\frac{\Delta p_{TP}}{\Delta p_{LO}} = 1 + \frac{C}{c_{tt}} + \frac{1}{c_{tt}^2} \tag{1}$$

The subscript TP corresponds to the two-phase value, while LO corresponds to the value with the total flow in the liquid phase. Lazarek and Black found that using a value of C=30 produces results that are in good agreement with their experimental data. The value of C recommended by Collier is 20 for large tubes.

The acceleration pressure drop was accurately predicted using the Martinelli and Nelson's (1948) separated flow model with a multiplier K_{sa}.

$$\frac{\Delta p_{sa}}{G^2/(2r_L)} = K_{sa}\left[\frac{r_L}{r_G}\frac{x_{ex}^2}{a_{ex}} + \frac{(1-x)^2}{1-a_{ex}} - \frac{r_L}{r_G}\frac{x_{in}^2}{a_{in}} - \frac{(1-x_{in})^2}{1-a_{in}}\right] \tag{2}$$

Here x is the vapor quality and a is the void fraction. K_{sa} is an empirical constant. Lazarek and Black found that a value of K=2.5 correlated most of their data within ±20 percent.

Moriyama and Inoue (1992) measured pressure drop of R-113 boiling in narrow annular gaps of 35-110 µm. Their experimental values for frictional pressure drop were correlated by slightly modifying eq. (1). From their study, it is evident that the separated flow model is applicable for the narrow gaps typically encountered in microchannel applications.

On the other hand, Bowers and Mudawar (1994) employed a homogeneous flow model with f_{TP}=0.003 as recommended by Collier (1981). Their results for both minichannels and microchannels were correlated to within ±30% with this model.

Tong et al. (1997) present an exhaustive treatment of pressure drop during subcooled flow boiling in minichannels. In addition, they presented a correlation to predict the two-phase pressure drop. Since the void fraction was very small, a two-phase friction factor was applied. They observed a roughness effect on the single-phase laminar to turbulent transition in these tubes. The effect of tube diameter to length ratio was also noted to be quite significant.

From the studies available in literature, the effect of channel dimension on two-phase pressure drop is not clearly established.

Although several investigators provide different correlation schemes to correlate their data, they do not provide a clear indication of the effect of small passage sizes on pressure drop. The added effect of channel wall roughness on pressure drop, seen in the single phase data, is also not incorporated while analyzing the two-phase pressure drop parameters. These effects will become more important as the channel size becomes decreases from minichannel to microchannel geometries.

3.2 Multi-channels

As the tube diameter decreases, the vapor slugs fill the tubes. Under two-phase flow conditions, flow instabilities occur when the pressure drop in the upstream section is relatively small. Introducing a large pressure drop through a throttle valve in the liquid line immediately prior to the test section considerably reduces the instabilities. These instabilities have a significant effect on pressure drop and heat transfer under flow boiling conditions.
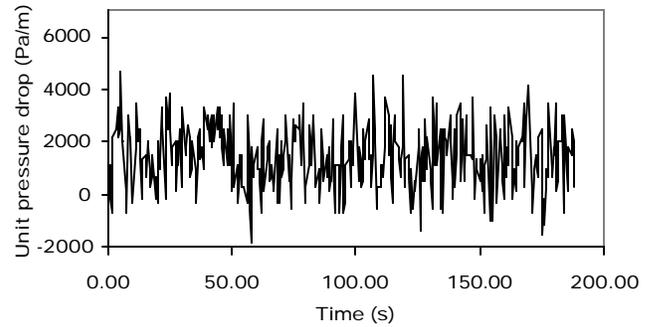

Figure 9 Differential Pressure History for a 6 Channel (1mm x 1mm) Parallel Configuration- G=48kg/m²s, $\Delta P_{max}/L$ = 4688Pa/m, $\Delta P_{min}/L$ = -1793Pa/m, Average Surface Temperature = 125°C, q''=74.3kW/m², Water Inlet and Outlet Temperatures 25.0 and 90.2°C, Kandlikar et al. (2001).

Figure 9 shows the pressure drop fluctuations measured in a multi-channel evaporator with six parallel 1mm × 1mm square microchannels. Similar observations were made by Kew and Cornwell (1996) during flow boiling of R-141b in 2-mm square channels and in 2.87-mm diameter circular tubes. The pressure drop fluctuations observed by Kandlikar et al. (2001) are quite large and result in flow reversals as discussed earlier under flow patterns in Section 2 of this paper. The compressibility of the two-phase mixture, in adjacent channels, acts in a manner similar to the negative slope in the upstream section of a single evaporator tube. The large pressure drop fluctuations lead to instantaneous localized flow reversal in some of the parallel channels. There are no models currently available that predict the pressure drop fluctuations and the flow reversals under flow boiling conditions. Knowledge of

these conditions is essential for safe operation of evaporators employing minichannels and microchannels.

4. HEAT TRANSFER IN SMALL DIAMETER FLOW BOILING CHANNELS

Flow boiling in small diameter tubes and compact heat exchanger passages has been a subject of interest in automotive, aerospace, air liquefaction, chemical, and petroleum industries, and in electronics cooling applications. Nakayama and Yabe (2000), and Kew and Cornwell (2000) present a good overview of the recent advances in this area. Table 1 includes some of the recent works on flow boiling heat transfer in minichannels.

Flow boiling heat transfer in 1-3 mm diameter channels has been a subject of investigation for a long time. In one of the earlier papers, Bergles (1964) studied the critical heat flux in 0.584, 1.194, and 2.388 mm diameter electrically heated tubes. They indicated that when the bubble diameter approaches the tube diameter, considerable non-equilibrium vapor volume exists in the evaporator section, and flow oscillations cause a premature burnout in the small diameter tubes.

Bowers and Mudawar (1993) studied flow boiling pressure drop and CHF in a minichannel of 2.54-mm diameter, and a microchannel of 510 μm diameter. Boiling curves for the two channels were obtained at nearly equal values of liquid Reynolds number. Their results are reproduced in Fig. 10. Despite the large variation in the tube diameter, the two curves overlap in the boiling region. It is believed that these experiments fall under the fully developed nucleate boiling regime. The differences between the two boiling curves are only evident at low heat flux (near single phase region) and high heat flux values (approaching CHF condition). This indicates that in spite of the differences in the flow characteristics of the channels, the flow boiling behavior is quite similar in the two geometries.
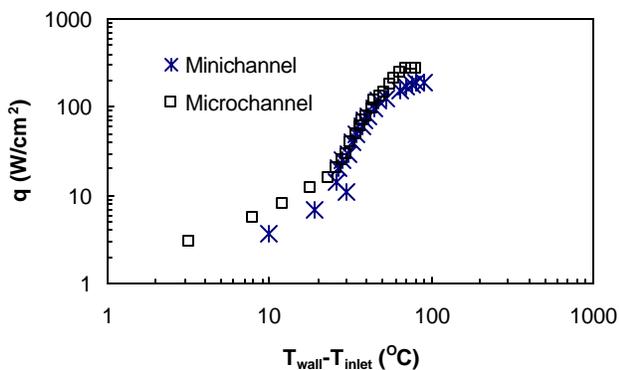

Figure 10 Flow Boiling Characteristics in Minichannel and Microchannel Evaporators, Bowers and Mudawar (1993)
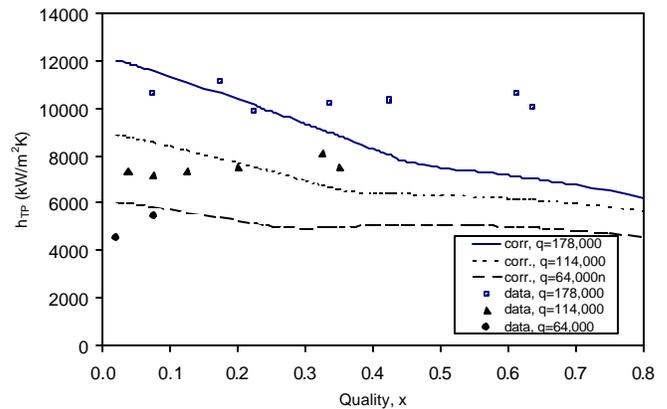

Figure 11 Comparison of Flow Boiling Heat Transfer Coefficient Data by Lazarek and Black (1982) with Kandlikar (1990) correlation

The detailed flow boiling data by Lazarek and Black (1982) provides a clear comparison between the flow boiling characteristics of minichannels and regular tubes (>3mm diameter). Figure 11 shows a comparison of Lazarek and Black's flow boiling data in a 3.1 mm diameter tube and Kandlikar's (1990) flow boiling correlation. The correlation represents the data quite well, although some differences exist in the high quality region. Although a detailed study is warranted, as a first order approximation, one may use the correlations developed for the large diameter tubes for predicting the heat transfer coefficients in minichannels.

Cornwell and Kew (1992) conducted experiments in two sets of parallel channel geometries, 1.2 mm x 0.9 mm deep, and 3.25 mm x 1.1 mm deep. Their results indicate that the flow boiling in such small channels exhibits fully developed nucleate boiling characteristics in the isolated bubble region at lower qualities. At higher qualities (when the bubbles fill the entire cross section), and in the annular flow region, convective effects dominate heat transfer. These characteristics are similar to those observed for the large diameter tubes. The isolated and confined bubble regions exhibit the characteristics similar to the nucleate boiling dominant region, while the annular-slug region exhibits the convective dominant trend seen in large diameter tubes (Kandlikar, 1991).

Another aspect of flow boiling heat transfer in small channels is the oscillatory nature of the flow. The time averaged value between two regions (i.e. between the confined bubble and the annular regions) would yield a combination of nucleate boiling dominant and convective boiling dominant regions. Purely flow pattern based correlations need to include this averaging effect. Since the large diameter correlations combine these effects, rather than using distinct boundaries, they are expected to provide the

basis for accurate correlation schemes for small diameter tubes and channels.

Continuing the work in this area, Lin et al. (1999) obtained flow boiling data with R-141b in 1-mm diameter tubes. Their results indicate that the heat transfer coefficient exhibits behavior similar to that reported by Cornwell and Kew (1992). The role played by bubbles is clearly an important one. Specifically they presented detailed results at G=500 kg/m^{2}s with q from 18 to 72 kW/m^2. At high heat fluxes, they observed considerable fluctuations in the wall temperature readings, indicative of flow oscillations that cause changes in the instantaneous values of local saturation temperature and heat transfer rate.

Wambsganss et al. (1993) conducted extensive experiments on flow boiling of R-113 in a 2.92-mm diameter tube. Their results indicate that the heat transfer coefficient was sensitive to both heat flux and mass flux changes, except for the lowest mass flux result. At the lowest value, G=50 kg/m^{2}s, changing the heat flux from 8 to 16 kW/m^2 did not have any influence on the heat transfer coefficient. One possible explanation is that for this case, the nucleate boiling is in the partial boiling region at low heat fluxes, and the effect of heat flux is therefore quite small. For their other test conditions, the mass flux was varied from 100 to 300 kg/m^{2}s, and the heat flux was varied from 16 to 63 kW/m^2. For these tests, the heat transfer coefficient exhibited a dependence on both heat flux and mass flux. This indicates the contributions from both nucleate boiling and convective boiling mechanisms. They also compared their data with the available correlations and found that the correlations by Liu and Winterton (1988), Shah (1976), and Kandlikar (1990) predicted their results with a mean deviation of less than 20 percent. In particular, Wambsganss et al. found that the specific correlation developed by Lazarek and Black (1982), who used their own small diameter tube data with R-113 in the correlation development, predicted the data with a mean error of 12.7 percent. The Chen (1966) correlation predicted the results with a mean deviation of 36 percent.

Mertz et al. (1996) conducted extensive experiments with water and R-141b boiling in six different minichannel configurations. The flow boiling was observed as oscillatory phenomena in multi-channels. It is interesting to note that although
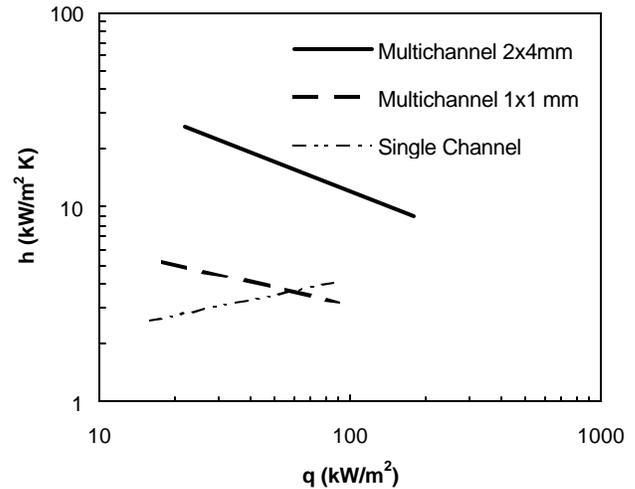

Figure 12 Comparisons of the average heat transfer coefficient trendlines for water in 2mm x 4mm single and multi-channel configurations, and 1mm x 1mm multi-channel configuration, G=200 kg/m^{2}s, T_{SAT}=120°C, Mertz et al. (1996)

the heat transfer coefficient increased with heat flux in almost all cases for single channels, the trend for water flowing in the multi-channel configuration at G=200 kg/m^2°C was different. For all channels in multi-channel configuration, the heat transfer coefficient decreased with increasing heat fluxes. It is suspected that the flow oscillations and reversals observed in multi-channel are responsible for the degradation in heat transfer. For R-141b boiling in multi-channel configuration, the heat flux effects were weak and somewhat mixed.

Mertz et al. observed that the heat transfer coefficient for both fluids in the multi-channel configuration was considerably higher than the corresponding single channel values under the same operating conditions. Figure 12 shows the comparison for water (at 200 kg/m^{2}s in 2 mm x 4 mm channels) in single and multi-channel configurations under the same operating conditions. The oscillatory behavior found in the multi-channel evaporator is therefore regarded to improve the heat transfer, although the improvement decreases at higher heat fluxes. Another fact to note is that the heat transfer coefficient is lower for the 1mm x 1mm multi-channel configuration than for its 2 mm x 4 mm counterpart as shown in Fig. 12.

Tran et al. (1996) conducted experiments with R-12 in small circular and rectangular channels both of 2.46 mm hydraulic diameter. Their results indicated two distinct regions, convective boiling dominant region at lower wall superheat values, and nucleate boiling dominant region at higher wall superheat values. They compared their data with the Kandlikar (1990) correlation, and found that it exhibited similar trends, but underpredicted the results. One reason for this difference is the fact that the single phase heat transfer coefficients in small diameter tubes is generally higher than those predicted by the Dittus-Boelter type correlations.

It is recommended that the measured single phase heat transfer coefficients be used in the correlation as recommended by Kandlikar (1991).

It is interesting to note that the heat transfer coefficients obtained by Tran et al. at higher qualities exhibited a dependence on heat flux alone. The mass flux had virtually no influence on the heat transfer coefficient. These observations are supported by the visual observations made by Kasza et al. (1997) who studied flow boiling of water in rectangular channels of 2.5 mm x 6 mm cross section. They concluded that the increased bubble activity at nucleation sites in the thin liquid film is responsible for high heat transfer coefficient in small hydraulic diameter tubes and channels. Kuznetsov and Shamirzaev (1999) conducted experiments with R-318C in an annular gap of 0.9 mm. They observed that at higher values of quality, nucleation was seen to be suppressed. However, their experimental results were in agreement with the correlation by Tran et al. (1999) which was developed for nucleate boiling dominant region.

Kamidis and Ravigururajan (1999) conducted flow boiling experiments with R-113 in circular tubes of 1.59, 2.78, 3.97 and 4.62 mm diameter tubes. Their results indicate that very high heat transfer coefficients of the order of 10 kW/m^{2}C are obtained in the flow boiling region. Figure 13 shows a comparison of their data for 1.59 mm diameter tube with the Kandlikar (1990) correlation. For the Reynolds number of 5720, the agreement is excellent. For Re=2370, the correlation underpredicts the data. It is suspected that the Reynolds number is in the transition region where the turbulent single phase heat transfer correlation is not applicable.
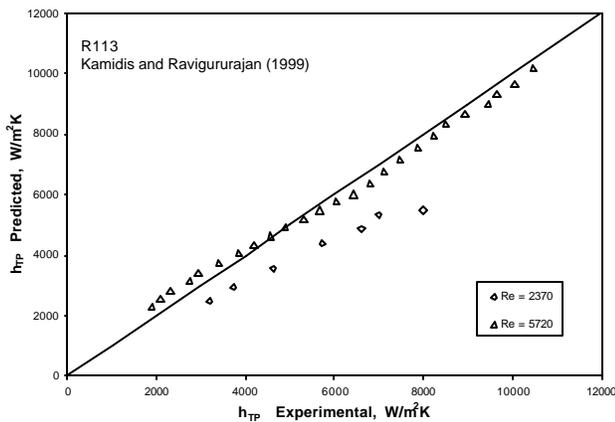

Figure 13 Comparison of Kamidis and Ravigururajan (1999) data with Kandlikar correlation (1990) for 1.59 mm diameter tube, R-113

Kennedy et al. (2000) studied the instability during flow boiling of water in circular tubes of 1.17 and 1.45 mm diameter. Based on their results, they proposed that the instability is initiated in the tube when the impressed heat flux is 90 percent of the value required to obtain saturated vapor condition at the exit section.

Lakshminarasimhan et al. (2000) conducted experiments with R-11 in 1 mm x 20 mm rectangular channels. They noted that the their saturated flow boiling data was accurately predicted using the large tube correlation by Kandlikar (1990). They also observed a clear boiling front with liquid flow in the laminar region. In the turbulent region, the boiling front could not be clearly viewed due to the high heat transfer coefficient in the single-phase region prior to nucleation.

The experimental studies available in literature provide some preliminary data on flow boiling heat transfer in small diameter tubes and channels. In the case of Lazarek and Black's (1982) and Lakshminarasimhan et al. (2000) data, the Kandlikar (1990) correlation for large diameter tubes predicted the results satisfactorily. However, Tran et al. (1996) indicated a significant enhancement over the large diameter correlations. Pressure fluctuations and multi-channel effects are not clearly understood for small diameter tubes. As noted by many investigators, including Lin et al (1999) and Mertz et al. (1996), pressure fluctuations have a significant effect on the flow characteristics and associated heat transfer performance. The periodic filling of the flow channel with large vapor plugs, followed by all liquid flow in the channel, make it very difficult to apply flow pattern based models to predict the heat transfer rates. Further research in this area is highly recommended.

Another class of flow channels that have received some attention in literature are those with hydraulic diameters below 500 µm. These channels are referred to as microchannels, although the precise boundary is not well defined. There are very few quantitative studies available for the microchannel geometry under flow boiling conditions. Further efforts are needed in this area to generate high-quality experimental data in microchannels under flow boiling conditions. This geometry has been investigated with its potential application in microelectronics cooling. Table 1 clearly indicates that there is very little quantitative data available for microchannels.

Ravigururajan et al. (1996) studied flow boiling in a microchannel 270 µm wide and 11 mm deep. The working fluid was R-124, and was tested over the entire quality range. They found that the heat transfer coefficient decreased from a value of 11 kW/m^{2}C at x=0.01 to about 8 kW/m^{2}C at x=0.65. Although no conclusions were drawn, this behavior may be the results of the two trends: a) the nucleate boiling heat transfer is dominant, leading to its suppression at higher qualities, or b) the higher vapor fraction leads to flow oscillations in multi-channels with a consequent change (increase?) in the heat transfer coefficient.

A number of investigators (for example, Peng and Wang, 1998, Peng et al., 1998) have indicated that the flow boiling heat transfer in microchannels may be quite different than that in larger diameter tubes. They also indicated that the regular nucleate boiling phenomenon does not exist in microchannels.

Peng and Wang (1998) conducted experiments with water, ethanol, and their mixtures in different shaped microchannel geometries (listed in Table 1). They noted the presence of both nucleate boiling and convective boiling in various regimes. They did not observe any bubble activity in the rectangular and triangular passages with hydraulic diameters between 0.1 to 0.6 mm. In turn, they called it a fictitious boiling phenomenon.

It is difficult to accept the notion of the fictitious boiling presented by Peng and Wang (1998). Similar studies reported by Kandlikar and Stumm (1995) and Kandlikar and Spiesman (1997) with a channel height of 3 mm indicates that bubbles as small as 10 μm are seen to depart from the nucleating sites. The key to observing bubble activity in small channels is to employ high speed photography along with a high resolution microscope.

The theoretical analysis presented by Peng and Wang (1998) considers a bubble nucleus that completely fills the tube. The microchannel dimension is on the order of 100 μm, while the cavity sizes for active nucleation are on the order of a few μm or smaller. It is expected that the nucleation criterion for flow boiling, established for large diameter tubes, will hold true unless the tube diameter approaches the cavity dimensions. Such a condition may exist only in submicron-sized tubes.

In conclusion, flow boiling in microchannels is an area where further research is needed. The difficulty in observing the bubbles and in the accurate measurement of heat fluxes at the wall make it very difficult to understand the mechanism of flow boiling heat transfer in this geometry. With the availability of more accurate data, we may be able to find some of the answers in the near future.

5. DESIGN CONSIDERATIONS IN MINI-CHANNEL EVAPORATORS

Mehendale et al. (1999,2000) present a good overview of the design consideration for heat exchangers employing mini- and microchannels.

Flow Instability in Multi-channel Evaporators

The small diameter multi-channel evaporators differ from the small hydraulic diameter compact heat exchangers in one vital aspect: there are no cross flow connections available for the fluid to flow across the width of the flow channel as it passes through the evaporator. This cross connection helps the nucleating bubbles grow in the cross-wise direction without blocking the entire flow passage, as in the case of small diameter channels. This flow structure is clearly illustrated by the flow pattern investigation conducted by Kuznetsov and Shamirzaev (1999) in an annular gap between two concentric tubes. They observed a cell pattern that effectively allows the bubbles and vapor to grow in the crossflow direction without blocking the flow. In the case of tubes, as shown by Kandlikar (2001), the vapor bubble growth leads to large pressure fluctuations that are not desirable for stable operation of the evaporator. The presence of fins or ribs in the gap is expected to further provide stability to the flow by increasing flow resistance.

In the case of multi-channel evaporators employing individual small-diameter tubes or channels, the channels running parallel to any given channel (which is experiencing vapor expansion in a direction opposite to the flow) act in a manner similar to reducing the upstream pressure drop characteristics in a two-phase system. The severe pressure drop fluctuations, coupled with the backflow of vapor into the inlet manifold, are not desirable. It could lead to premature CHF in some of the channels where vapor may flow, preferentially, without being accompanied by the liquid flow. Some of the systems that employ such evaporators may not tolerate such severe fluctuations in the flow rate.

With these considerations, it is necessary to design multi-channel evaporators that avoid severe pressure fluctuations found in the parallel channels. Further research in this area is warranted.

5.1 Design Guidelines for Sizing Small Diameter Multi-channel Evaporators

The small channels present a number of advantages making them attractive for specific systems. Their compact size, low weight, low liquid/vapor inventory, and fast response are just some of the desirable features. In this section, preliminary guidelines are presented for designing the multi-channel evaporators with small diameter channels. The design conditions considered in this analysis are as follows.

Design Conditions

It is assumed that an existing single or multi-channel evaporator employing large diameter (D) tubes is to be replaced with a multi-channel evaporator employing small diameter (d) tubes. Capital letter subscripts refer to the large diameter tube while lower case subscripts refer to smaller diameter tubes. For the qualitative analysis presented here, it is assumed that both geometries employ circular tubes, and that one larger diameter tube is being replaced by n number of smaller diameter tubes. The total mass flow rate and the total heat transfer rate are identical in both cases. The objective is to arrive at the number of small diameter tubes needed to replace each large diameter tube, and the new length the evaporator. Another consideration is the need for equal pressure drop in the two cases. The analysis is presented for a single-pass evaporator. It can be extended to a multipass evaporator configuration, but the treatment will introduce many additional parameters. The purpose of the following exercise is to provide some simple guidelines to help in designing the new evaporator. Extensive design efforts will be needed to arrive at the final design and all of the associated details.

5.2 Design Comparison

If n parallel channels are employed in the new evaporator (for each tube in the original design), the mass flux and heat flux for the new heat exchanger with small diameter (d) tubes would be different than those of the original heat exchanger with large diameter (D) tubes. Since the total mass flow rate remains constant between the two designs, we get:

$$n = \frac{G_D}{G_d}\frac{D^2}{d^2} \tag{3}$$

The total heat transfer rates in the two cases are also identical since the evaporators are being designed for the same heat duty. If we apply the heat transfer rate equations with the respective average heat transfer coefficients, and assume that the operating temperature difference in the two cases is identical, we can write:

$$\frac{L_d}{L_D} = \frac{\bar{h}_D}{\bar{h}_d}\frac{1}{n}\frac{D}{d} \tag{4}$$

Substituting n from eq. (3) into eq. (4), the ratio of the lengths in the flow direction is given by:

$$\frac{L_d}{L_D} = \frac{\bar{h}_D}{\bar{h}_d}\frac{G_d}{G_D}\frac{d}{D} \tag{5}$$

For the same exit quality conditions, the total heat transferred in the heat exchanger remains the same. The heat fluxes in the two cases are related by the following equation:

$$\frac{q_d}{q_D} = \frac{1}{n}\frac{D}{d}\frac{L_D}{L_d} \tag{6}$$

The flow boiling in the small diameter tubes exhibits a nucleate boiling dominant region at low qualities. Here the heat transfer coefficient varies as $q^{0.7}$. In the higher quality region, the flow becomes convective boiling dominant, with the heat transfer coefficient independent of q, and varying as $G^{0.7}$. In general, the dependence of h may be expressed as $h \propto q^m G^p$. The ratio of the two heat transfer coefficients is given by:

$$\frac{\bar{h}_d}{\bar{h}_D} = \left(\frac{q_d}{q_D}\right)^m \left(\frac{G_d}{G_D}\right)^p \tag{7}$$

The exponents m and p depend on the flow boiling heat transfer characteristics under the prescribed operating conditions.

Their values range from $m=0$ to 0.7, and $p=0$ to 0.8. Substituting the heat flux ratio from eq. (6) into eq. (7), combining it with eq. (3), and then substituting the heat transfer coefficient ratio into eq. (5), the ratio of the two lengths is obtained as:

$$\frac{L_d}{L_D} = \left(\frac{G_d}{G_D}\right)^{\frac{1-m-p}{1-m}}\frac{d}{D} \tag{8}$$

Equation (8) provides a preliminary estimate of the length of the heat exchanger needed to obtain the same vapor generation rate using a small diameter evaporator. The number of small diameter tubes replacing each large diameter tube is given by eq. (3).

A few observations can be made regarding the effect of boiling characteristics on the length ratio presented in eq. (8). If the nucleate boiling was the dominant mode in both cases, then $p=0$. This results in a mass flux ratio dependence given by $\left(L_d / L_D\right) = \left(G_d / G_D\right)\left(d / D\right)$. On the other hand, if the heat transfer was convective dominant, then $m=0$ and $p=0.8$. In the latter case, the mass flux ratio has a weak effect on the length ratio, $\left(L_d / L_D\right) = \left(G_d / G_D\right)^{0.2}\left(d / D\right)$. However, the actual variation lies between the two extreme cases discussed here.

Another important factor to be considered in the design of heat exchangers is the pressure drop. The design guideline provided by eq. (8) does not address this issue. A similar analysis is now presented for pressure drop comparison in the two cases.

The pressure drop analysis is a bit more complex. To simplify the analysis, the gravitational pressure drop is considered negligible when compared to the friction and acceleration pressure drop terms. Assuming the respective exit quality and exit void fractions to be equal in both cases, the pressure drop may be expressed in terms of the mass flux, tube length, tube diameter, exit quality, and fluid properties. The frictional pressure drop varies as $\Delta p_f \propto f_{TP}LG^2/D$, while the acceleration pressure drop varies as $\Delta p_f \propto G^2$. In addition, the void fraction plays a role in the pressure drop terms. The two-phase friction factor may be considered to vary as $f_{TP} \propto G^{-0.25}$. Assuming the same equations to apply for the small and large diameter tubes, the ratio of pressure drops may be expressed as

$$\frac{\Delta p_d}{\Delta p_D} = \frac{\left(L_d G_d^{1.75}/d\right)C_1 F_1(\mathbf{a}_d) + G_d^2 C_2 F_2(\mathbf{a}_d)}{\left(L_D G_D^{1.75}/D\right)C_1 F_1(\mathbf{a}_D) + G_D^2 C_2 F_2(\mathbf{a}_D)} \tag{9}$$

The constants C_1 and C_2 include additional variables in the respective pressure drop terms. F_1 and F_2 are functions of the void fractions in the two cases. Although the effects of mass flux,

diameter and length on pressure drop are not immediately obvious, assuming the frictional pressure drop to be dominant provides some degree of guidance. In addition, assuming the void fractions and their effects to be similar, eq. (9) may be simplified as follows.

$$\frac{\Delta p_d}{\Delta p_D} = \frac{\left(L_d G_d^{1.75} / d\right)}{\left(L_D G_D^{1.75} / D\right)} \tag{10}$$

For the case of equal pressure drop between the two configurations, the length ratio is obtained as,

$$\frac{L_d}{L_D} = \left(\frac{G_d}{G_D}\right)^{-1.75} \frac{d}{D} \tag{11}$$

Comparing eqs. (8) and (11), it is clear that the effect of mass flux is more severe on the pressure drop than on the heat transfer. The diameter effect is same in both cases. If the mass flux is held constant for the two configurations, then the length ratio is identical to the diameter ratio, $L_d / L_D = d / D$. However, in practical system designs, a higher pressure drop is generally accepted with evaporators employing small channels, and tube lengths larger than that given by eq. (11) are employed.

The negative exponent in the mass flux ratio of eq. (11) indicates that increasing the mass flux results in shorter tube lengths for the same pressure drop. In other words, increasing G_d causes the pressure drop to increase, and shorter tube lengths are needed to meet the pressure drop requirements. From the heat transfer perspective, a larger tube length may be needed to accommodate higher mass fluxes. Consequently, the design mass flux is a compromise between these considerations and other system requirements.

The preceding discussion provides a preliminary basis for the selection of a mass flux value for the smaller tube diameter heat exchanger being designed to replace an existing larger tube diameter evaporator. Needless to say, a number of additional parameters, including fluid properties, the local heat transfer coefficient and pressure drop relationships, differences in manifolding and number of passes, and the differences in allowable pressure drop will affect the design of the new evaporator with smaller diameter tubes. Another major consideration in the design of the evaporator is the performance on the hot fluid side. In the analysis presented here, the wall temperatures were considered to be identical in both evaporators. The comparisons should, therefore, be treated as qualitative in determining first-order effects.

5. CONCLUSIONS

On the basis of a critical literature review and the work consucted by the author, following conclusions can be drawn:

1. Three flow patterns are commonly encountered during flow boiling in minichannels: isolated bubble, confined bubble or plug/slug, and annular. The visual studies available in literature have been generally conducted for low mass flux values in tubes of 1-mm or larger hydraulic diameters.

2. The literature on flow patterns in microchannels is insufficient to draw any conclusions at this stage.

3. Large pressure drop fluctuations are noted in multi-channel evaporators. Flow pattern observations revealed a flow reversal in some channels with expanding bubbles pushing the liquid-vapor interface in both upstream and downstream directions.

4. Heat transfer studies in the microchannels indicate that, as a first order estimate, heat transfer may be predicted using the flow boiling correlations developed for large diameter tubes.

5. The heat transfer rate in multi-channel evaporators is different from that in single-channel evaporators under same set of operating conditions. The role flow rate fluctuations due to flow instabilities is not clearly understood at this stage.

6. The severe pressure drop fluctuations are not included in any pressure drop prediction schemes for minichannel and microchannel evaporators. Both separated and homogeneous flow models have been used with some degree of success by previous investigators.

7. In designing the evaporators with small diameter channels, the length to diameter ratio depends on the heat transfer and pressure drop characteristics. Larger pressure drops are generally accepted in evaporators with small diameter channels.

6. FUTURE RESEARCH NEEDS

The future research needs are summarized below:

a) Conduct high speed video studies to obtain flow pattern information under high mass flux conditions in small diameter tubes and channels.

b) Compare the performance of single tube evaporators and multi-channel evaporators under the same operating conditions and identify the reasons for the differences in their performance.

c) Study the effects of inlet flow conditions and manifold design on the performance of the multi-channel evaporator.

d) Conduct more experiments with microchannel evaporators to obtain accurate flow boiling heat transfer and pressure drop data as a function of quality, heat flux, mass flux, and tube/channel hydraulic diameter.

e) Critical heat flux is an important factor in the design of evaporators. Although not covered in this paper due to space constraint, there is a need for obtaining more experimental data for CHF in single and parallel minichannel and microchannels.

7. NOMENCLATURE

C constant in eq. (1)
C_1 and C_2 constants in eq. (9)
D   diameter of
e gap size, m
F_1 and F_2 constants in eq. (9)
G mass flux, kg/m^{2}s
g acceleration due to gravity, m/s^2
K_{sa} pressure drop multiplier in acceleration pressure drop eq. (2)
h_{ffg} latent heat of vaporization, J/kg
$\overline{h}$ average heat transfer coefficient in the evaporator, W/m^2 °C
L length of the evaporator tube
n   number of parallel small diameter tubes for each large diameter tube
Dp pressure drop, N/m^2
q heat flux, W/m^2
x quality

Greek Letters

α   void fraction
ρ   density, kg/m^3
σ   surface tension, N/m

χ_{tt} Martinelli parameter, $= \left(\dfrac{r_V}{r_L}\right)^{0.5} \left(\dfrac{m_L}{m_V}\right)^{0.1} \left(\dfrac{1-x}{x}\right)^{0.9}$

Subscripts

D   large diameter tube
d   small diameter tube
ex   exit
f   friction
in   inlet
L   liquid
LO   all flow as liquid
TP   two-phase
tt   turbulent-turbulent
V   vapor

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