EXPERIMENTAL STUDY OF FLOW BOILING OF FC-72 IN PARALLEL RECTANGULAR MICROCHANNELS

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Abstract. *Using FC-72 as the coolant, a closed-loop two-phase microchannel heat sink driven by a micro-gear pump was developed in this work. The cooling system was made of oxygen-free copper containing 14 parallel channels of the dimension of 0.8mm (W)×2mm(D)×20mm(L). The cooling performance of the heat sink was evaluated by the heat transfer coefficient and the pressure drop under different conditions. Experiments were performed under the conditions of initial pressure of Pin = 73 and 101 kPa, mass velocity of G = 71–250 kg/m^{2}s, outlet quality of xe,out = 0–90% and heat flux of q"= 25–140 W/cm^2. The preliminary result is reported here which shows that, a maximum heat dissipation rate of 96 W/cm2 can be achieved while keeping the heating surface temperature below 85^{o}C within the experimental conditions investigated. However under low flow rate conditions, critical heat flux could be occurred easily that deteriorates the performance of the cooling system.*

***Keywords**: Microchannel, flow boiling, heat transfer coefficient, pressure drop, FC-72, critical heat flux*

1. INTRODUCTION

Research on high heat flux compact cooling technologies is motivated by the increasing heat generation rates in VLSI circuits, high power semiconductors and laser thermal weapons. Microchannel heat transfer system is one of the promising techniques for these applications due to its compactness, large surface-to-volume ratio, high aspect ratio and minimal coolant usage especially under phase-change conditions. Compared with single-phase counterpart, flow boiling in microchannels possesses higher heat transfer coefficient and provides an opportunity for a nearly constant surface temperature decided by the saturation temperature of the fluids. Tremendous efforts have been undertaken to reveal two-phase heat transfer characteristics in microchannels. Most of these previous studies are focused on flow boiling of water in open-loop systems because of the relative ease of machining (Yu et al. 2002, Wen et al. 2004, Wen and Kenning, 2004). For close-loops, Jiang et al. (2002) developed a closed-loop two-phase microchannel cooling system using an electroosmotic pump for de-ionized water. It removed 38 W from a 1cm×1cm chip while keeping the chip temperature rise below 100°C. Lee and Mudawar (2005a, b) studied two-phase heat transfer characteristics and pressure drop of R134a in a microchannel heat sink, and revealed different heat transfer mechanisms at low, medium and high vapor qualities. A heat transfer coefficient correlation was developed accordingly, which agreed well with experimental data for both R134a and water. There were a few studies on split flow arranged microchannels (Myung et al. 2006, Liu, 2006), which could reduce the flow length resulting in a lower pressure drop and a more uniform surface temperature distribution. Cross-linked microchannel heat sink was also investigated (Cho and Koo, 2003,Colgan et al, 2005) and showed better cooling performance than that of a regular microchannel heat sink.

However most of these previous investigations are focused on boiling characteristics of water under either atmospheric condition due to the simplicity or high pressure conditions for power plant applications. As the boiling point of water at atmospheric conditions is fixed at 100°C, it is difficult to apply directly on many electronic devices that require low peak temperatures. Possible ways around would be either employing alternative coolants that have lower boiling temperatures or still operating microchannel boiling systems but operating at reduced pressures. FC-72 is such an attractive coolant that could be used to achieve low operation temperature. FC-72 with boiling point of 56°C is thermally and chemically stable, nonflammable and practically nontoxic, which makes it ideal for many applications. Although the latent heat of FC-72 of ~ 93 kJ/kg at a saturation temperature of 56°C is far less than that of water, it still compares favorably with the cooling capability of many other refrigerants. Quite a lot of studies have been conducted on FC-72, mainly on single channels for refrigeration applications. Relatively little is known for FC 72 in a multi-microchannel system under different system pressures.

For a particular space application where it requires to control the device temperature below 85°C but to dissipate heat up to 100 W/cm^2, flow boiling of FC-72 in microchannels is proposed as a possible solution. A two-phase microchannel system is constructed for this purpose. The preliminary results of the cooling system using FC-72 as the coolant is reported in this paper, which includes the influence of mass flux, heat flux and exit quality on boiling heat transfer coefficient and two-phase pressure drop, as well as the performance assessment of the cooling system up to the point of Critical Heat Flux (CHF).

2. EXPERIMENTAL SYSTEM AND VALIDATION

2.1 Experiment setup

A schematic representation of the microchannels system is shown in Figure 1. It is composed of three primary subsystems: a main flow loop, a secondary heat removal loop, and a compute-controlled data acquisition system. The main flow loop includes a reservoir, a micro-gear pump (Micro Pump Inc., GB-P23.JVS.A+DB380A), the microchannel heat sink, and the inner tube of the serpentine cooler. The micro-gear pump is selected to maintain a relatively constant flow rate. The working fluid is pumped from the reservoir through a filter, which is located upstream of the pump to keep the working fluid clean and avoid blockage in the microchannels. The working fluid is delivered to the heat sink at the desired flow rate by adjusting power supply voltage of the pump or through the by-pass valve. After the heating section, the working fluid is cooled by the serpentine cooler downstream of the heat sink to subcooled state, which is pumped into the reservoir to form a closed-loop. The secondary heat removal loop includes the outer tube of the serpentine cooler and a water cooling unit.

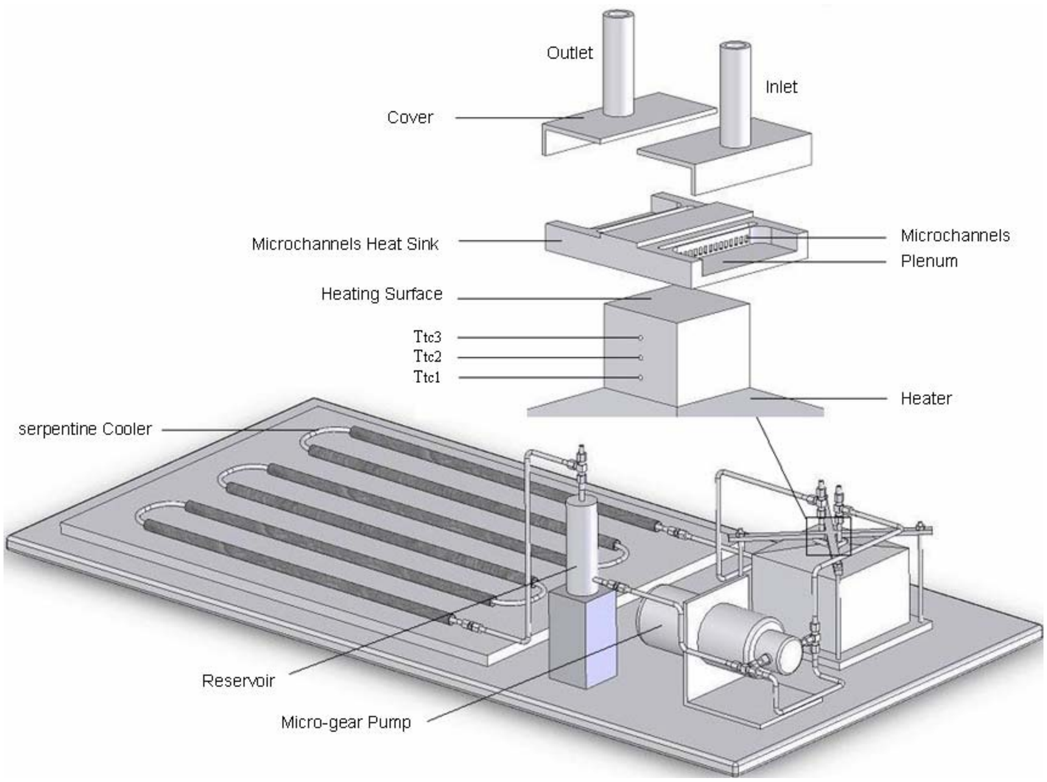


Figure 1. Schematic of the microchannel heat sink experimental system

The microchannel heat sink is designed to be completely hermetic with an appropriate shape for easy usage, and to achieve the lowest possible thermal resistance with a reasonable pressure drop for given conditions. Two microchannel heat sinks, made from oxygen-free copper, are developed in the project. Each consists of two cover plates and one substrate with grooved parallel rectangular micro-channels, which are spaced 0.6mm apart. These two heat sinks have the same overall dimensions, but with different number and dimension of microchannels grooved on the substrate plate. One heat sink has 14 channels with the dimension of 0.8mm(W)×2mm(D)×20mm(L), and the other has 17 channels with the dimension of 0.6mm(W)×2mm(D)×20mm(L). The microchannels are fabricated by the electric spark drilling technique, which is followed by computer controlled linear-cutting to ensure that the channels are separated from each other and the whole heat sink is hermetic. The superior thermal conductivity of the copper block ensures that the working fluid flowing through the microchannels can be heated from all four surfaces. The two cover plates are fitted over the two sides of the heat sink to seal the channels and form the inlet and outlet manifolds. Plenums are formed in the inlet and outlet of the test section to ensure a quasi-steady flow before entering the microchannels and avoid flow mal-distribution in parallel channels. The nominal area of the microchannel region in a test heat sink is 20 mm by 20 mm.

(a) (b)

Figure 2. Image of the microchannel heat sink: (a) the substrate (b) the with two cover plates

Heat loads are simulated by a high heat flux electrical resistance heater made of copper, as shown in Figure 3. The shape of the heating block is tapered from the middle of the heater to the root of the heater neck. Apart from the heating surface, all other surfaces of the heater block are carefully insulated to minimize heat loss to the ambient and obtain a high heat flux at the heating surface. Three thermocouples are embedded in the heater neck along the axial direction, which are spaced 4mm apart. Based on the assumption of one-dimensional heat diffusion in the heater neck, the heat flux can be calculated using the Fourier's law, which will be validated later through experiments. The heating area is 20mm×20mm, which is designed to match the dimension of the microchannel heat sink. Rather than using conventional thermal grease, the heating block was soldered onto the back of the microchannel heat sink to minimize the contact resistance.

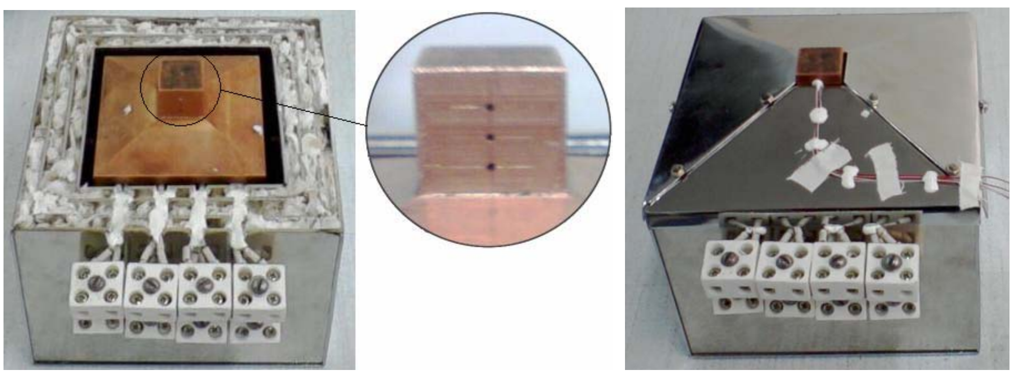


Figure 3. Images of the high heat flux electrical resistance heater and TC positions

The performance of the microchannel heat sink is evaluated by measuring the fluid temperature, the heater temperature, and the pressure drop across the micro-channels for given volumetric flow rates. Two type-K thermocouples are installed in the inlet and outlet plenums to measure the temperature of the working fluid, and another four type-K thermocouples are installed in the copper block along the microchannels. Furthermore, eleven type-T thermocouples are installed in other key locations in the flow loop. An absolute pressure transducer (JYB-KO-PVA, ±0.5% kPa) is located at the inlet plenum of the test heat sink, and a differential pressure transducer (JYB-3151, ±0.25% kPa) is connected to the inlet and outlet to obtain the pressure drop. Two flowmeters are used ub the range of 0-2.5ml/s (Alicat, 40-0-18-0-200-150-0-15-KM101, ±0.5% ml/min) and 2.5-6ml/s (MBLD, ±1% ml/min) respectively. All thermocouple and pressure transducer measurement signals are conditioned by a signal conditioner (Agilent34970A) prior to the data acquisition card (DAQ).

2.2 Experimental procedure, data reduction and system validation

The microchannel heat sink is evaluated under different heat flux and mass flux conditions. For a given flow rate, the voltage applied to the heater is increased at controlled intervals after a steady state is reached, until the occurrence of the burnout, which is always accompanied by a sudden rapid increase of surface temperature. The input power that induces this sudden surface temperature is defined as the critical heat flux in this work. The correspondent exit quality in the experiment covers a range of 0-0.9.

Based on the assumption of one-dimensional heat diffusion in the heater neck, the base heating flux q_b'' is calculated by the Fourier's law:

$$q_b'' = -\lambda \frac{\partial T}{\partial x} = \frac{\lambda(T_{tc1} - T_{tc3})}{Z_1 - Z_3} \quad (1)$$

where λ is the thermal conductivity of copper. In a like manner, the temperature of the heating surface is calculated as follow:

$$T_h = \frac{Z_3(T_{tc3} - T_{tc2})}{Z_2 - Z_3} + T_{c3} \quad (2)$$

The mass flow rate $\dot{m}$ is determined from the volumetric flow rate and the density of subcooled liquid based on the temperature and pressure measured just downstream of the flow-meter.

The working fluid is supplied to the microchannels in a subcooled state, and maintains its liquid state along the microchannels up to a location where the thermodynamic equilibrium quality x_e, Eq. (3), reaches zero.

$$x_e = \frac{h - h_l}{h_{\lg}} \quad (3)$$

The length of the microchannel is divided into two parts: single-phase length L_{sp} and two-phase length L_{tp}. The single-phase length L_{sp} can be calculated from the thermodynamic equilibrium:

$$L_{sp} = \frac{\dot{m} c_{p,l}(T_{sat} - T_{in})}{q_b'' W_h} \quad (4)$$

where W_h is the width of heating area, $c_{p,l}$ is the specific heat of liquid, T_{in} and T_{sat} are the inlet working fluid temperatures and the saturation temperature, respectively.

The two-phase length is then calculated by:

$$L_{tp} = L_{ch} - L_{sp} \quad (5)$$

where L_{ch} is the total length of the microchannel.

The two-phase heat transfer coefficient is calculated by:

$$h = \frac{q_w''}{T_w - T_{sat}} \quad (6)$$

where T_w is the channel wall temperature, which is calculated from the measured surface temperature assuming one-dimensional heat conduction. q_w'' is calculated using the extended area of the channels($N \cdot A_{ch}$), whereas q_b'' is related to the heater footprint area(A_h). N is the number of channels.

$$q_w'' = \frac{q_b'' \times W_h}{2N(W_{ch} + H_{ch})} \quad (7)$$

The outlet quality of the microchannels, xe,out, is determined from the energy balance as below:

$$x_{e,out} = \frac{\left(q_b'' A_h - c_{p,l}\dot{m}(T_{sat} - T_{in})\right)/r}{\dot{m}} \quad (8)$$

where r is the latent heat of evaporation.

The heat loss of the whole system is assessed prior to FC-72 experiments, which typically smaller than 10%. Among different ways of heat losses, a major part is from the heater, though all surfaces of the heater are carefully insulated apart from the top surface, i.e. the heating surface. The heat flux calculation by Eq. (1) is also validated under single-phase convection conditions, where the total heat input is compared with the sensible heat increase of the working fluid. Very good agreement is reached as shown in Figure 4.

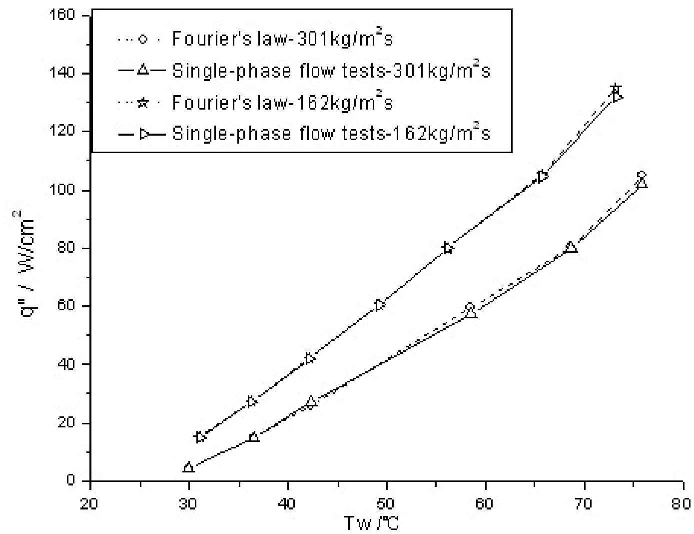
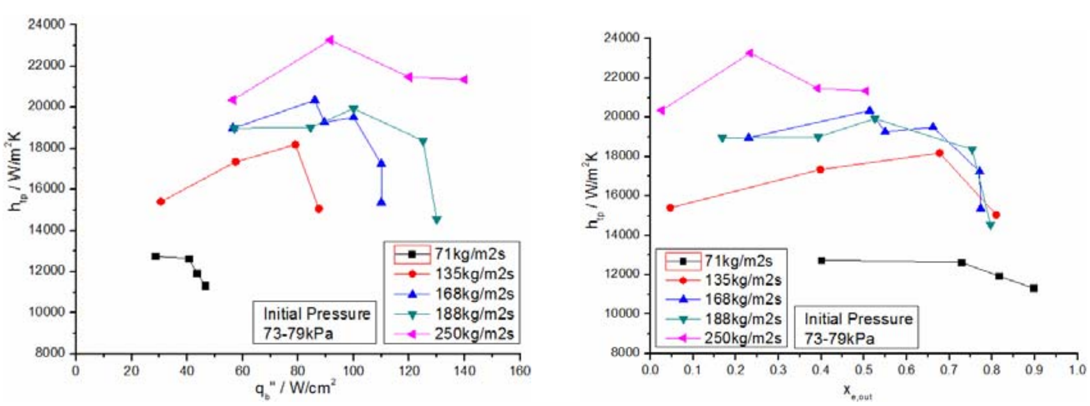


Figure 4. Validation of the heat flux calculation through single phase convection tests.

3. RESULTS AND DISCUSSIONS

After building up the confidence on the test system, a number of experiments are conducted and some preliminary results are shown here. Figure 5 shows the average two-phase heat transfer coefficient of the heat sink as a function of heat flux and exit quality under five different mass fluxes. It shows a strong dependence of the heat transfer coefficients (HTC) on both mass flux and heat flux. Apart from the lowest mass flow rate, G=71 kg/m^{2}s, similar dependence of the HTC on the heat flux is found for all other mass flux conditions. There is a general trend of HTC increase at low heat fluxes, reaching a peak value and then starting decrease, which is believed to be associated with the local dryout. The peak heat transfer coefficient is found to be shifted to a higher heat flux with the increase of mass flow rate, suggesting an increase in the CHF value. At low heat fluxes, the heat transfer is dominated by the mass flow effect. The higher the mass flux, the higher the heat transfer coefficient. At higher qualities, all HTC data appears converging into one decline line.

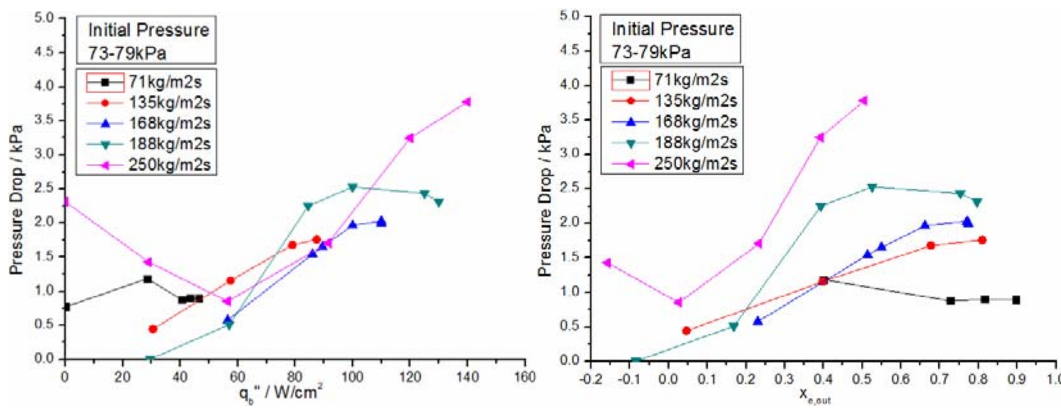


(a) heat flux influence (b) exit quality influence

Figure 5. Boiling heat transfer coefficient as a function of heat flux and exit quality

The mechanisms of flow boiling in conventional channels have been conventionally classified either as nucleation-dominant or convection dominant, depending on different sensitivities of the heat transfer coefficient to mass flux, exit quality, and heat flux (Chen, 1963). Although it is still not fully clear about the heat transfer mechanisms in microchannels, the conventional concept is used widely to distinguish boiling mechanisms, either as nucleation dominant (Mehendale and Jacobi, 2000, Bao et al. 2000, Lazarek and Black, 1982, Yu et al. 2002) or convection dominant (Lee and Lee, 2001, Huo et al. 2004, Ravigurrajan, 1998, Warrier et al. 2002, Yan and Lin, 1998, Klimenko, 1990.), or a combination of both contributions (Lin et al. 2001, Qu and Mudawar, 2003, Wen et al. 2004), depending on different dependences of the heat transfer coefficient upon mass flux, heat flux and vapor qualities. Our current study shows that for FC-72 boiling in parallel microchannels, there is a strong dependence of on both the flow rate and heat flux. However it should be remembered that the data obtained in this study are the average value of the heat sink (time and space averaged), calculated from Eq. (6), which does not give details of heat transfer distribution along the channels. Transient experiments have revealed that microchannels heat transfer is dominant by the formation and development of confined bubbles (Wen et al. 2004, Thome et al.2004). The confined bubble could grow in both downstream and upstream, which accelerates a slug of liquid along the channel, causing a pressure and temperature fluctuations. In a region of established confined bubble flow, the averaged heat transfer coefficient is contribution from both forced convection by liquid slugs and evaporation from confined bubbles. We are performing a further visualization study to understand better the mechanism of flow boiling of FC-72 in the microchannel heat sink, which shall be reported later.

For cooling systems based on phase change mechanisms, the safe operation of the system is limited by the critical heat flux, which is the upper limit of heat removal without incurring serious surface temperature jump. Beyond that, a vapor blanket or local dryout will occur on the surface that prevents further direct contact of liquid. As a result, sudden temperature rise will occur due to the poor thermal conductivity of vapor. It is of vital importance to predict the critical heat flux in small channels. In the experiments, the critical heat flux is observed by a rapid increase in the surface temperature, typically over 150^{o}C. The CHF value is dependent upon the flow rate and can exceed 1MW/m^2 under relative high mass flow rate.

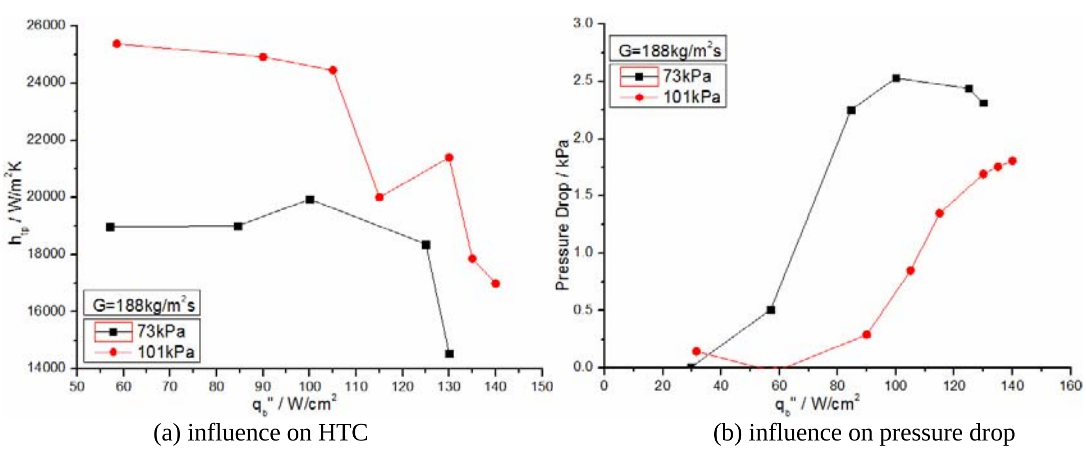


(a) heat flux influence (b) exit quality influence

Figure 6. Pressure drop through microchannels heat sink

The corresponding pressure drop through microchannels heat sink on input power and exit quality is illustrated in Figure 6. In general, the pressure drop increases with the increase of heat flux until a peak value is reached. One exception is for the highest mass flux investigated, G=250 kg/m^{2}s, the pressure drop decreases initially, which is due to a decrease in viscosity at single phase flow regime. Subsequent boiling and generation of vapor increases the pressure drop. Similar to the heat transfer coefficient case, the peak value of the pressure drop is associated with the occurrence of the CHF, which increases with the increase of mass flux. Similar to conventional-sized pipes, the pressure drop increases saliently with the increase of mass flux. Some fluctuations in pressure drop is also observed in the experiments, which is associated with the confinement of bubbles and subsequent rapid growth, accompanied by occasional reversed flow and liquid re-distribution among different channels.

Figure 7 shows an example of the influence of system pressure on the heat transfer coefficient and pressure drop. It is apparent that the pressure drop decrease with system pressure, while a reverse trend is observed for the heat transfer coefficient, which exhibits the same trend of other studies for single channels.



(a) influence on HTC (b) influence on pressure drop

Figure 7. The influence of system pressure on HTC and pressure drop

Checking all the temperature data shows that for a specified surface temperature of 85°C, the maximum heat dissipation rate achieved is 96 W/cm^2 under the highest flow rate tested, i.e. G=250 kg/m^{2}s. Such a performance is slightly less than the required value of 100 W/cm^2, however it is reasonable to believe that a higher heat dissipation value can be achieved by increasing further the flow rate of the coolant, due to the strong dependence of heat transfer coefficient on the mass flux, as shown in Figure 5. Under such a condition, the pressure drop of the microchannel heat sink is only a couple of *kPa* and is acceptable for typical applications. It appears feasible that FC-72 can be used as an effective coolant for high heat flux applications. Further detailed studies are still ongoing.

4. SUMMARY

This paper shows the feasibility of using FC-72 cooled two-phase microchannel system for high heat dissipation applications. Within the parameters investigated, the microchannel heat sink can achieve a heat dissipation rate of ~96 W/cm^2 while keeping the device temperature below 85°C. In addition, the study showed that the average heat transfer coefficient of the heat sink is a strong function of mass flux and heat flux, the CHF can occur earlier under low mass flow rate conditions, which is associated with a rapid increase in surface temperature, and the system pressure affect both the heat transfer coefficient and pressure drop of the system. Further optimization of the microchannel system to achieve better performance is currently undergoing.

5. ACKNOWLEDGEMENTS

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