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AN EXPERIMENTAL STUDY ON CONVECTIVE BOILING HEAT TRANSFER IN MICRO HEAT EXCHANGERS

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ABSTRACT

Attributed to its high heat transfer coefficient, evaporating cooling involving the use of micro heat exchangers is considered a possible thermal management solution for cooling of high heat flux electronic devices. The desire to develop high-performance micro heat exchangers operating in the evaporation regime provides a major motivation for the present work. Methanol evaporated in two micro heat exchangers with chevron flow passages and straight flow passages respectively were tested in the present study. The test results show that the heat transfer coefficient increased with increasing flow rate in both chevron and straight flow passages micro heat exchangers. However, the effect of vapor quality on the heat transfer coefficient in the straight passages heat exchanger is in adverse to that in the chevron passages heat exchanger. The heat transfer coefficient increased with increasing vapor quality in the chevron passages heat exchanger but decreased in the straight passages heat exchanger. The flow visualization through transparent cover heat exchangers showed that the liquid film inside channel is partially dry out in the straight passages heat exchanger. The dryout portion area increased with increasing heating rate and exit vapor quality. This degraded the average heat transfer performance for evaporation in the straight passages heat exchanger. Because of the surface tension effect, the liquid film was dragged at the intersection corner of the upper and lower plate chevron passages. There is no significant dryout portion in the chevron passages heat exchanger. The relation of vapor quality with heat transfer performance in chevron passages heat exchanger is therefore similar to the boiling in a single channel prior to critical heat flux condition.

INTRODUCTION

Owing to the rapid development of semiconductor industry, the heat dissipated from electronic devices increases drastically with increasing device logic gate number and operation speed. Traditional air cooling methods would not be able to accommodate this high heat flux owing to their theoretical heat transfer limits. The cooling technologies have undergone evolutionary changes from direct air cooling to forced convective liquid cooling and two-phase evaporating cooling in recent years. Attributed to its high heat transfer coefficient, evaporating cooling involving the use of micro heat exchangers is considered a possible thermal management solution for cooling of high heat flux electronic devices. The desire to develop high-performance micro heat exchangers operating in the evaporation regime provides a major motivation for the present work.

Starting from Tuckerman and Pease [1], micro heat exchangers used for single phase liquid cooling have been developed for more than 20 years. Most of the researches in the field have been focused on flow paths having constant cross-sections straight micro channels (Wesberg et al. [2], Zhang et al. [3]). More geometrically complex micro channels may offer attractive performance advantages, but they have not been extensively investigated. Some conventional heat transfer enhancement techniques have been successfully applied in the micro heat exchangers. Colgan et al. [4] provided a practical implementation of a single-phase silicon microchannel cooler designed for cooling very high power chips. They concluded that the performance of 75 or 100 μm pitch silicon microchannel coolers with staggered fins was shown to be superior to continuous fin designs with equivalent geometries. Kandlikar and Grande [5] used three-dimensional micro channels that incorporated either microstructures in the channel or grooves in the channel surface to enhance its single-phase cooling performance. Fabrication methodologies for these devices have been successfully developed but their heat transfer performances have not been tested and proved. They also reviewed various enhancement approaches and proposed some of the conventional heat transfer enhancement techniques seem to hold promise in microchannels application.

As those described in Webb [6], conventional heat transfer enhancement techniques has been developed for several decades. Numerous enhancing methods have been proposed for internal or external laminar flow but those have not been verified that can be applied in microchannels. Several researches dealing with the single-phase forced convection heat transfer inside micro tubes have been published in the past years. Yang et al. [7] have successfully designed and fabricated a high performance micro heat exchanger with chevron flow paths based on the conventional plate heat exchanger heat transfer enhancement concepts. Their study still focused on single-phase liquid cooling but not two-phase evaporation cooling. It is expected that the two-phase evaporation will provide much higher heat transfer performance than the single-phase liquid heat transfer.

There are numerous researches have been dedicated on the evaporation heat transfer in micro heat exchangers. Most of them are for boiling in straight flow channels. (Qu and Mudawar [8], Lee and Garimella [9], Bertsch et al. [10]). Their test results did not agree with the conventional two-phase heat transfer correlations. The present study desired to provide a further understanding of flow boiling in micro heat exchangers with and without enhanced flow geometries. Both heat transfer performance test and boiling flow visualization were conducted.

EXPERIMENTAL SETUP

Heat Exchangers for Heat Transfer performance and Flow Visualization Test

The micro heat exchangers developed by Yang et al. [7] with straight and chevron flow passages were used in the present study. All heat exchangers were fabricated by chemical etching on copper plates. Both flow visualization and heat transfer performance were tested. The drawings of basic dimensions and flow passage patterns of these heat exchangers are shown in Figure 1 and 2 respectively. Details dimensions of the flow passages are listed in Table 1. For the flow visualization test, the copper upper plate of the heat exchanger was replaced by a transparent cover made of polycarbonate which has been machined to have the same flow passage pattern. The combination of the test section for flow visualization is shown in Figure 3. It includes a bottom frame, a lower plate of the heat exchanger and a transparent polycarbonate cover. These three parts were assembled together by 16 bolts and sealed by an O-ring between the lower plate and the transparent cover.

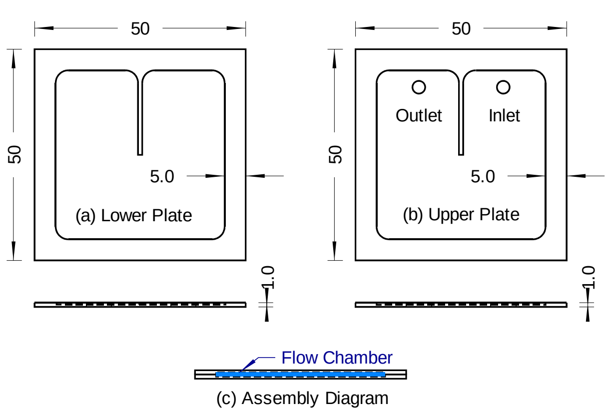
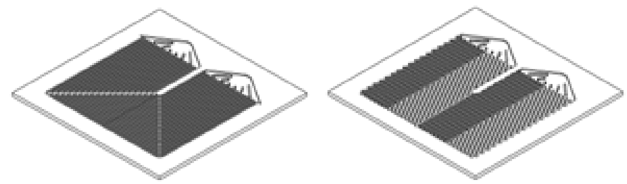


Figure 1. Basic dimensions of the micro heat exchanger [7] (unit: mm)



(a) straight (b) chevron

Figure 2. Flow passage patterns of heat exchangers in the present study.

Table 1. Dimensions of each heat exchanger

	straight (mm)	chevron (mm)
Outer dimension of plate	50×50×1	50×50×1
Height of channel	0.39	0.39
Width of channel	0.69	0.69
Space of channel	0.38	0.38
Chevron angle		30^o

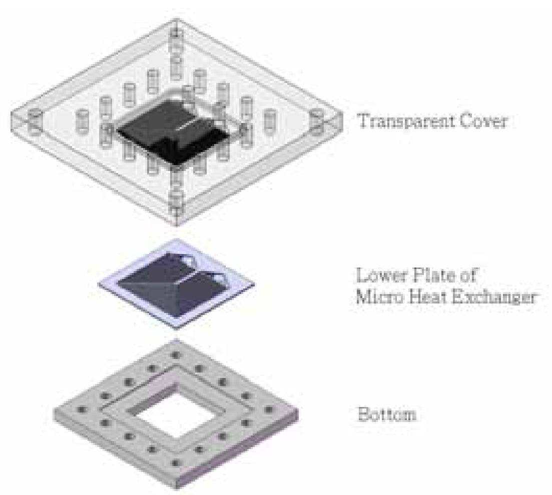


Figure 3. Test section assembly for flow visualization

Heat Transfer Performance Test System

A schematic diagram of the heat transfer performance test apparatus is shown in Figure 4. A gear pump was used to maintain a steady flow of the working fluid from the condenser to the preheater. A positive displacement flow meter was placed between the gear pump and the preheater to measure the fluid flow rate. The working fluid was heated in the preheater to reach the required inlet flow condition. A 25 μm filter was installed before the flow meter to prevent the micro-channels in the micro heat exchanger from possible clotting by foreign materials in the fluid. The test section consists of a micro heat exchanger, a heating block with square cross section area 10 mm x 10 mm and insulation material surrounding the heating block. The heating block was made of copper with imbedded cartridge heaters and three thermocouples. The cartridge heaters was connected to adjustable power suppliers and each power supply circuit was equipped with a multimeter that allows the voltage across and current through the adjustable power supplier to be measured. A thermocouple was attached on the lower surface of the micro heat exchanger to measure its heating center temperature (T_c). The pressure difference between the inlet and outlet of the test section was measured by a difference pressure transducer whereas the inlet and exit temperatures (T_{wi} and T_{wo} respectively) were measured by two thermocouple probes. The working fluid was evaporated in the test section and then condensed in the condenser to complete a closed loop. Methanol was used as the working fluid.

The exit vapor quality, x, can be evaluated from the equation listed below:

$$q = \dot{m}c_p(T_{in} - T_{sat}) + \dot{m}xi_{lv}$$

The heating rate q was derived by the one dimensional conduction law from the temperatures measured by the three thermocouples imbedded in the heating block. $\dot{m}$ is the flow rate, T_{in} and T_{sat} are the measured inlet and saturation temperature respectively, c_p and i_{lv} are the specific heat and latent heat of the working fluid. The heat transfer coefficient, h_{cs} was defined as

$$h_{cs} = \frac{q}{A(T_c - T_{sat})}$$

where A is the heating surface area. The experimental apparatus and derived parameters uncertainties are listed in Table 2.

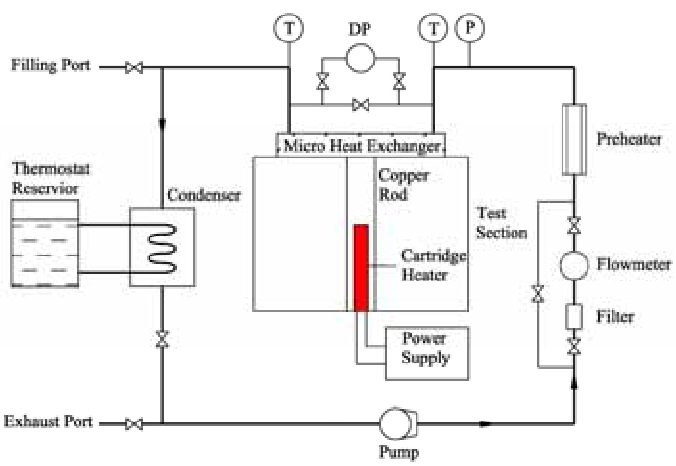


Figure 4. Schematic diagram of the heat transfer performance test apparatus

Table 2. Uncertainties of the experimental apparatus and derived parameters

Apparatus	Uncertainties
T type Thermocouple	±0.1 °C
Flow meter	±1.25 ml/min
Differential pressure transducer	±7.5 Pa
Derived parameters	
Heating power	0.78~1.4 %
Vapor quality	4.4~6.5 %
Thermal resistance	0.8~1.98 %

Experimental Apparatus for Flow Visualization

Figure 5 shows the schematic diagram of the visualization experimental apparatus. The heating and working fluid circulation system is basically the same as that for the heat transfer performance test. A high speed camera with speed of 1000 frames/sec was setup above the test heat exchanger to record the bubble generation of the working fluid in the heat exchangers. The inlet and saturation temperature during the flow visualization test are 58 and 60 °C respectively.

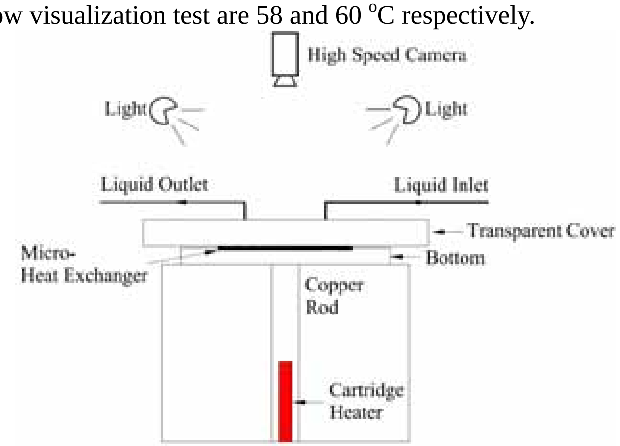


Figure 5. Schematic diagram of the visualization experimental apparatus

EXPERIMENTAL RESULTS AND DISCUSSIONS

Pressure Drop and Heat Transfer Performance

The pressure drop of methanol flow in straight and chevron passages heat exchangers at various flow rate and exit vapor quality conditions is shown in Figure 6. It shows that the pressure drop increases with increasing flow rate and vapor quality. The pressure drop in chevron flow passage heat exchanger is about four times higher than that in the straight flow passage heat exchanger. These results agree well with those tested by Yang et al. [2007].

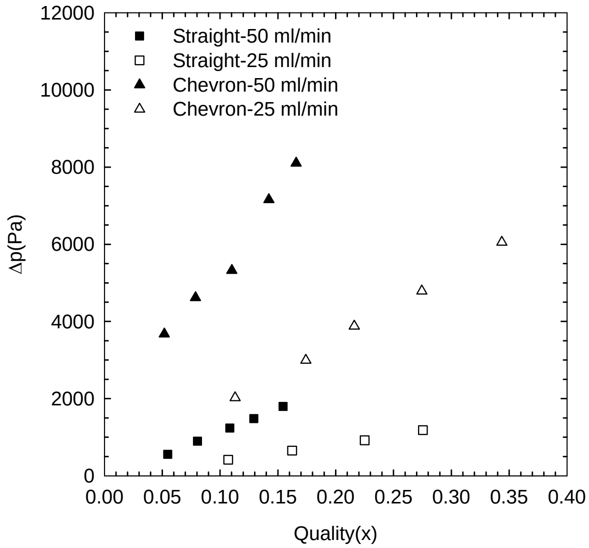


Figure 6. Pressure drop at various flow rate and exit vapor quality in heat exchangers

Figure 7 shows the heat transfer coefficients for flow in both heat exchangers at various flow rate and exit vapor quality. The heat transfer coefficient for flow rate of 50 ml/min is higher than that for flow rate of 25 ml/min in both heat exchangers. For flow in chevron flow passage heat exchanger, the heat transfer coefficients increases with increasing vapor quality. All of this satisfied the convective boiling theory. However, for flow in straight flow passage heat exchanger, the heat transfer coefficients decreases with increasing vapor quality. This result seems conflict to our experience.

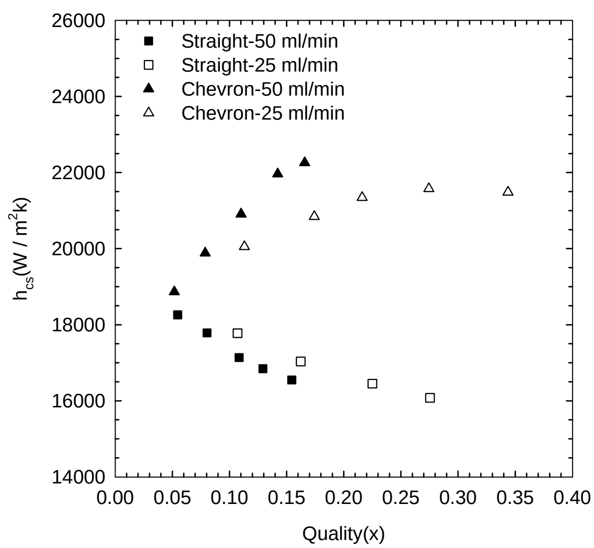
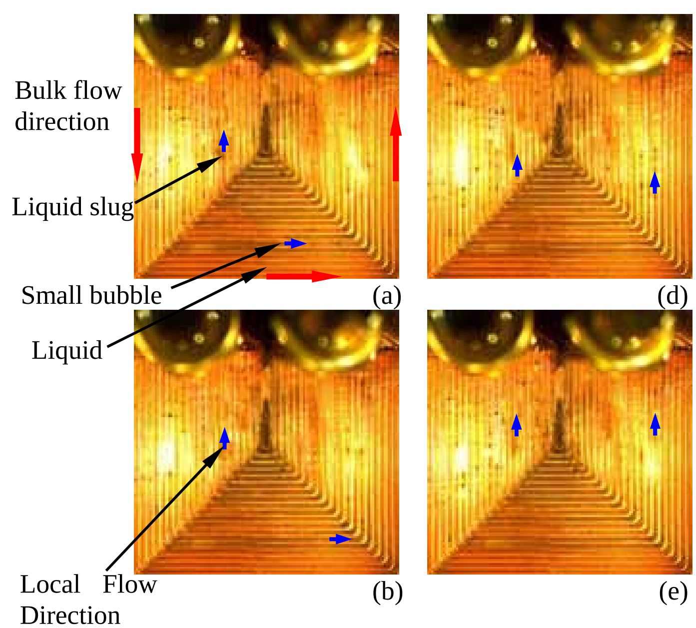


Figure 7. Heat transfer coefficient and vapor quality relation for different micro heat exchangers

Flow Visualization Results

The flow boiling visualization was conducted for flow in both straight and chevron passage heat exchangers at flow rate of 50 ml/min and vapor ranging from 0.05 to 0.35. Figure 8 shows the consecutive pictures of the flow boiling in the straight passages heat exchanger with 10 ms interval at heating rate of 40W and exit vapor quality 0.05. The pictures show that part of the entering subcooled liquid was evaporated at the center part of the heat exchanger but most of it still remained liquid phase with very few small bubbles at the rest part of the heat exchanger. Because of the sudden expansion of the fluid evaporated from liquid to vapor phase, part of the fluid flow reversed from the heating center to the inlet port direction. The two-phase flow oscillated at the center part of the heat exchanger. There is no sufficient liquid to feed the heating part and caused partial dryout at the center part. The area of the dryout portion increased with increasing heating power and therefore exit vapor quality. Comparing the heat transfer performance and flow visualization results, Figure 7 and Figure 8, we may conduct that the local dryout at the heating part degraded its heat transfer performance. The larger the dryout portion area, the lower the heat transfer coefficient.

For flow in chevron passage heat exchanger shown in Figure 9, the evaporated vapor at the center part expanded through the chevron channels to the outside of the heat exchanger but liquid feed back from other chevron passages. Because of the surface tension effect, the liquid film was dragged at the intersection corner of the upper and lower plate chevron passages. The flow oscillation still exists but there is no significant dryout at the heating center portion. The liquid film thickness decreased with increasing heating rate. And therefore, the heat transfer coefficient increased with increasing exit vapor quality as shown in Figure 7.



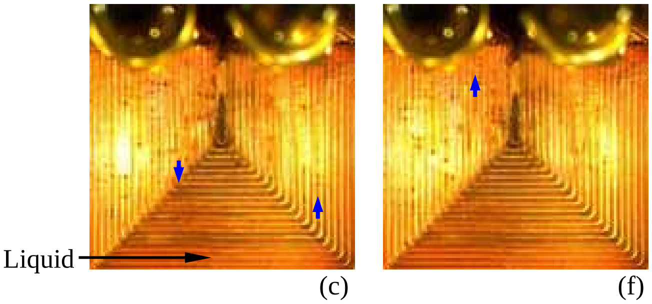
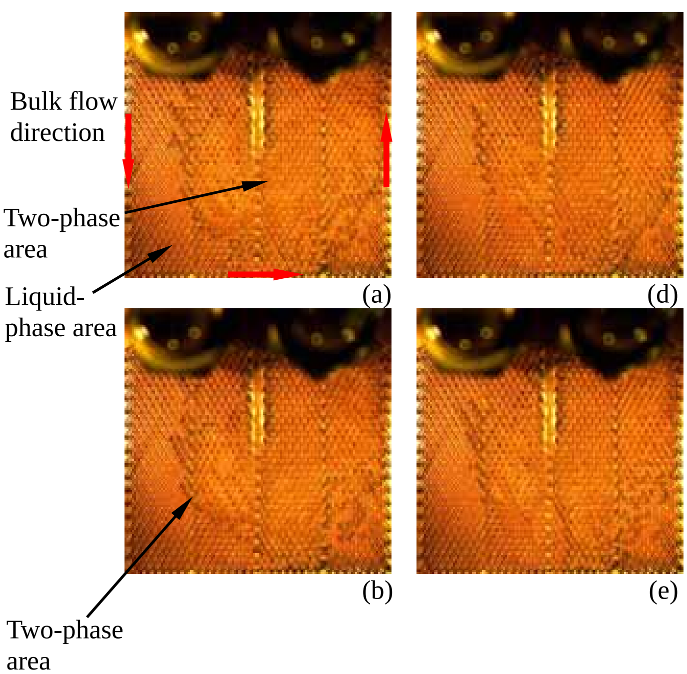


Figure 8. Consecutive pictures of flow boiling in straight passages heat exchanger with 10 ms interval from (a) to (f)



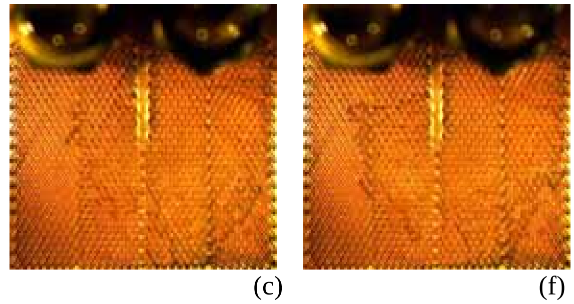


Figure 9. Consecutive pictures of flow boiling in chevron passages heat exchanger with 30 ms interval from (a) to (f)

CONCLUSIONS

Methanol evaporated in a micro heat exchanger with chevron flow passages which developed by Yang et al. [2007] were tested in the present study. Another micro heat exchanger with straight flow passages was also tested for comparison. The test results show that the heat transfer coefficient increased with increasing flow rate in both chevron and straight flow passages micro heat exchangers. This agrees with the conventional convective boiling theory. However, the effect of vapor quality on the heat transfer coefficient in the straight passages heat exchanger is in adverse to that in the chevron passages heat exchanger. The heat transfer coefficient increased with increasing vapor quality in the chevron passages heat exchanger but decreased in the straight passages heat exchanger. The flow visualization through transparent cover heat exchangers shows that the liquid film inside channel is partially dry out in the straight passages heat exchanger. The dryout portion area increased with increasing heating rate and exit vapor quality. This degraded the average heat transfer performance for evaporation in the straight passages heat exchanger. The flow visualization also showed that because of the surface tension effect, the liquid film was dragged at the intersection corner of the upper and lower plate chevron passages. There is no significant dryout portion in the chevron passages heat exchanger. The relation of vapor quality with heat transfer performance in chevron passages heat exchanger is therefore similar to the boiling in a single channel prior to critical heat flux condition.

NOMENCLATURE

A	Heating surface area, m^2
c_p	Specific heat of methanol, J/kg-^{o}C
h_{cs}	Two phase heat transfer coefficient, W/m^2-k
i_{lv}	Latent Heat of fluid, J/kg
$\dot{m}$	Mass flow rate, kg/s
q	Heating rate, W
T_c	Heating surface center point temperature, ^{o}C
T_{in}	Fluid inlet temperature, ^{o}C
T_{sat}	Saturated temperature, ^{o}C
x	Vapor quality

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