An Experimental Study of Gas-Liquid Heat Transfer in Micro-Channel Heat Sink

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Abstract

The study of two-phase heat transfer characteristics in 21 parallel micro-channels, 0.33 mm wide and 0.46 mm deep, is conducted in the present investigation. Prior to two-phase flow experiments, the apparatus and acquisition system are validated by the present single-phase flow data. Regarding gas-liquid flow experiments, air-water mixture from y-shape mixer is forced to flow in the micro-channel heat sink fabricated in copper, 40 mm long in the flow direction. Flow visualization is feasible by incorporating the stereozoom microscope into the camera system and different flow patterns are recorded. The dependence of Nusselt number on liquid and gas superficial Reynolds numbers are obtained. It is also shown from the results that the gas-liquid flow leads to the increment of Nusselt number up to 50% over single-phase flow.
Keywords*:* Two-phase flow; Heat transfer; Micro-channels; Heat sink

1. Introduction

Two-phase flow researches have been carried out extensively over the years. However, there have been a relatively small amount of publications dealing with micro-channels when compared with those for ordinarily sized channels. Capillary force is likely to play an important role in two-phase flow and heat transfer characteristics, resulting in significant differences in the flow phenomena between ordinarily sized channels and micro-channels. A micro-channel heat sink, first introduced by Tuckerman and Pease [1], is a cooling device incorporating small channels in which fluid flows to cool a heated substrate. Micro-channel heat sinks are employed in various applications such as cooling systems in turbine blades, fusion reactor blankets, rocket engines, hybrid vehicle power electronics, computer chips, etc. Discussions for key applications of micro-channel heat sinks are given in details by Mudawar [2]. Due to the need for continuously increased power dissipation of microelectronic devices, the single-phase micro-channel flow seems to be no longer highly effective cooling method because they are limited by low Nusselt number in laminar flow. Two-phase flow in small channels has become another effective means for dissipating heat. Noting that the researches on the two-phase micro-channel heat sinks have been mainly reported for flow boiling characteristics and such drawbacks, difficult to control, as backflow and instabilities are generally addressed [3]. In contrast, non-boiling two-phase heat transfer data is still lacking.

Bao et al. [4] carried out experiments to explore heat transfer performance of air-water flow in a channel having a diameter of 1.95 mm. They reported that at a fixed liquid flow rate, heat transfer coefficient increased with the increase in air flow rate, which caused by the flow pattern transition. Heat transfer of air-water flow in parallel micro-channels of 0.1 mm in hydraulic diameter was experimentally investigated by Hetsroni et al. [5]. Their results showed the decrease in Nusselt number with increasing gas flow rate, which is opposite to the results obtained by Bao et al. [4].

Betz and Attinger [6] showed segmented flow, an intermittent pattern of gas bubbles and liquid slugs, resulting in the heat transfer enhancement up to 140% in a micro-channel heat sink when compared with single-phase liquid flow.

Heat transfer characteristics of non-boiling two-phase flow in micro-channels with different diameters were studied by Choo and Kim [7]. Air and water were used as working fluids to examine the dependence of Nusselt number on the channel diameter. They found that with channel diameters of 0.506 and 0.334 mm, the Nusselt number increased with the increment of gas flow rate, but decreased with increasing gas flow rate when the channel diameters of 0.222 and 0.140 were employed.

According to previous works, the available results focusing on non-boiling two-phase heat transfer are not conclusive. Further research is needed to meet a general conclusion. The aim of the current research is to explore the heat transfer characteristics of air-water flow in parallel micro-channels with hydraulic diameter of 0.38 mm under low superficial velocities (j_G = 0.26 – 4.23 m/s, j_L = 0.07 – 0.34 m/s), which has never been seen before.

2. Experimental Apparatus and Procedure

A schematic diagram of the experimental apparatus is shown in Fig. 1. Peristaltic pump with adjustable flow rate is used to supply liquid flow through the micro-channel heat sink. The liquid mass flow rates are determined by using an electronic balance (320 ± 0.001 g) to measure weight of the liquid flowing from the test section outlet over a sufficient time whereas the flow rates of gas are measured by three sets of rotameters within the range of 5 – 50, 50 – 500, and 200 – 2500 sccm, respectively. An air-water y-shape mixer is served to introduce fluids smoothly along the test section. Instrumentation is installed at various positions to monitor the flow condition of the working fluids.

Fig. 2 illustrates an exploded view of the test section. The 21 parallel rectangular micro-channels of 0.33 × 0.46 mm (width × depth) with a 0.66 mm fin thickness are fabricated on the copper block in which a cartridge heater with adjustable input power is installed. The micro-channels are 40 mm long in the flow direction. On the top of the test section, the cover plate made of polycarbonate is placed to allow optical access for flow visualization. The copper block is well insulated by G10 epoxy. Metal plates and the insulator are bolted together with the copper block and the cover plate to firm the assembly.

To estimate the wall temperature needed for determining the heat transfer coefficient, type K thermocouples are inserted in the copper block. Their installations are assembled in a T-shape as shown in Fig. 2. Five thermocouples for each side of the test section are equally spaced along the axial direction, and located at a distance of 2 mm from the bottom of the micro-channels. Also, the temperature measurements at equal distance of 4.3 mm along the direction perpendicular to the flow are carried out with two additional thermocouples for each side. Based on the mentioned installations, the wall temperature can be calculated using extrapolation. In addition to the wall temperature, the thermocouples are installed at the inlet and outlet of the test section to measure the fluid temperature. All thermocouples, pressure transducer and relevant instruments installed in the experimental apparatus are well calibrated.

The cover plate, which is transparent, serves as a viewing window for flow visualization. The detailed formation of flow pattern is registered by precise stereozoom microscope mounted together with a camera having shutter speeds of 1/15 – 1/10,000s. An adjustable light source is placed, perpendicular to the viewing section.

In this work, the experiments are conducted in such a way that the air flow rate is increased by small increments while the water flow rate and the heat applied to the test section are kept constant at the desired value. The system is allowed to approach a steady state before the flow pattern and relevant data are recorded. During the experiment, the temperature and pressure drop are continuously recorded along the test section by the data logger.

3. Data Reduction

Due to the cover plate which is made of polycarbonate, adiabatic fin tip is assumed to obtain the wall heat flux, q_w, calculated based on the effective area, which is given by

$$q_w = \frac{q_b\left(W_{ch} + W_f\right)}{\left(W_{ch} + 2\eta H_f\right)} \qquad (1)$$

where q_b, W_{ch}, W_f, H_f and η represent, respectively, base heat flux, channel width, fin thickness, channel dept and fin efficiency.

The heat transfer coefficient is obtained from

$$h = \frac{q_w}{T_w - T_f} \qquad (2)$$

where T_f is fluid temperature and T_w denotes the average wall temperature estimated by an extrapolation.

For one dimensional heat conduction with adiabatic fin tip, the fin efficiency is expressed by

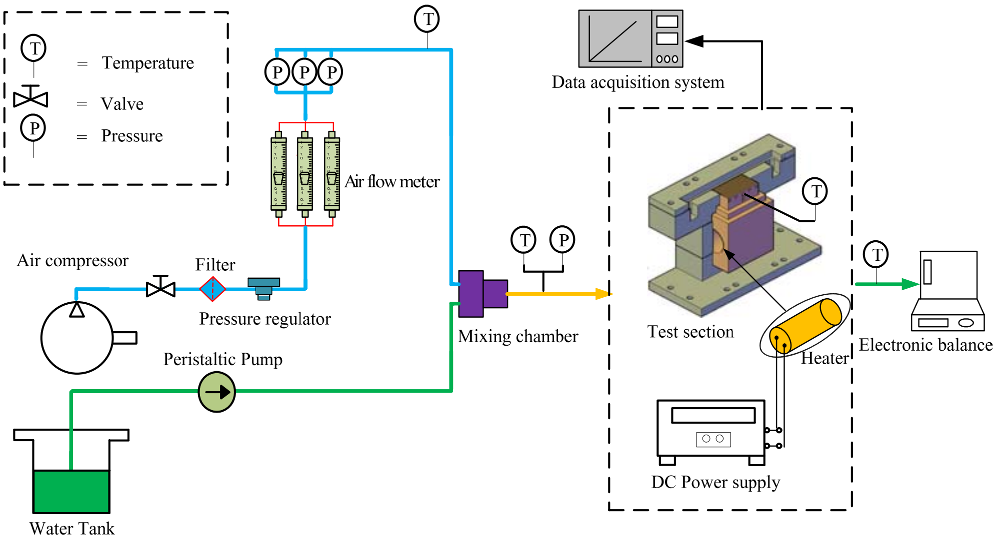

Fig. 1. Schematic diagram of experimental apparatus.

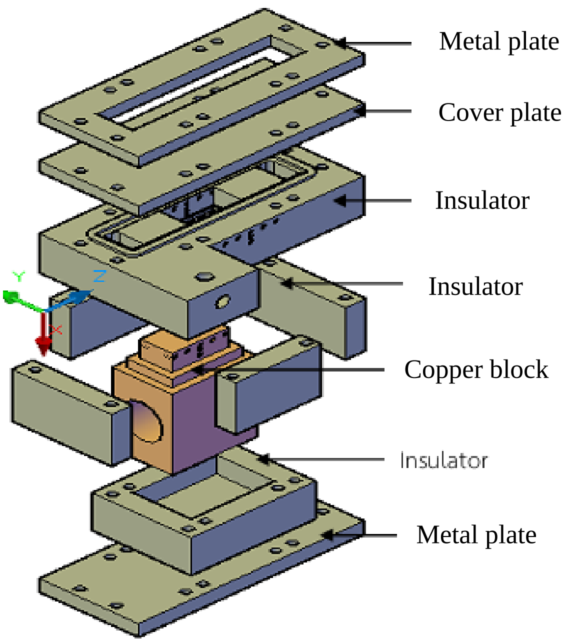

Fig. 2. Schematic diagram of test section.

$$\eta = \frac{\tanh(mH_f)}{mH_f} \quad (3)$$

where m is defined as

$$m^2 = \frac{hP}{kA_c} \quad (4)$$

In Eq. (4), P stands for fin perimeter, k for thermal conductivity and A_c for fin cross-sectional area.

Iteration process for Eqs. (1) – (4) is needed to determine the wall heat flux and heat transfer coefficient. The process is carried out until the fin efficiency converges to a fixed value.

4. Results and Discussion

4.1 Single-phase flow

Prior to perform two-phase flow experiments, the single-phase flow experiment is conducted to confirm the validity of the experimental system. Fig. 3 illustrates the energy balance over the test section. The figure indicates insignificant deviation between electrical input power and heat transfer rate absorbed by the fluid flow. In this experiment, heat balance is obtained by an average of 2.3% and, hence, heat losses are neglected in the experiment. The experimental results also show that Nusselt numbers match the predictions by Sieder and Tate [8] and Lee and Garimella [9], which are based on laminar thermally developing flow. In addition, single-phase friction factors are found to comply with the laminar developing flow predicted by Shah and London [10].

4.2 Two-phase flow

In this section, heat transfer and fluid flow characteristics of non-boiling air-water flow are presented. Fig. 4 shows the dependence of Nusselt number on superficial Reynolds number of gas (Re_{GS}) and liquid (Re_{LS}). As expected, heat transfer increases with increasing liquid superficial Reynolds number. As also shown in the figure, the two-phase heat transfer becomes better than the single-phase flow. Heat transfer enhancement up to 50% is obtained over the single-phase liquid flow. With the increment of

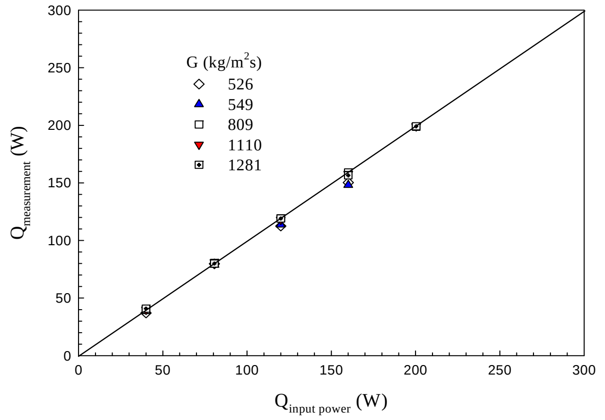

Fig. 3. Energy balance applied on the test section.

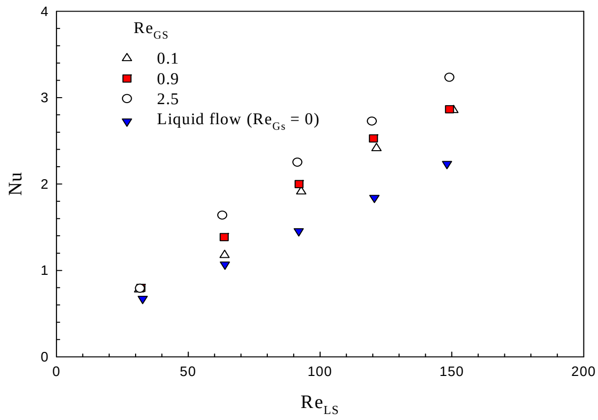

Fig. 4. The dependence of Nusselt number on superficial Reynolds number.

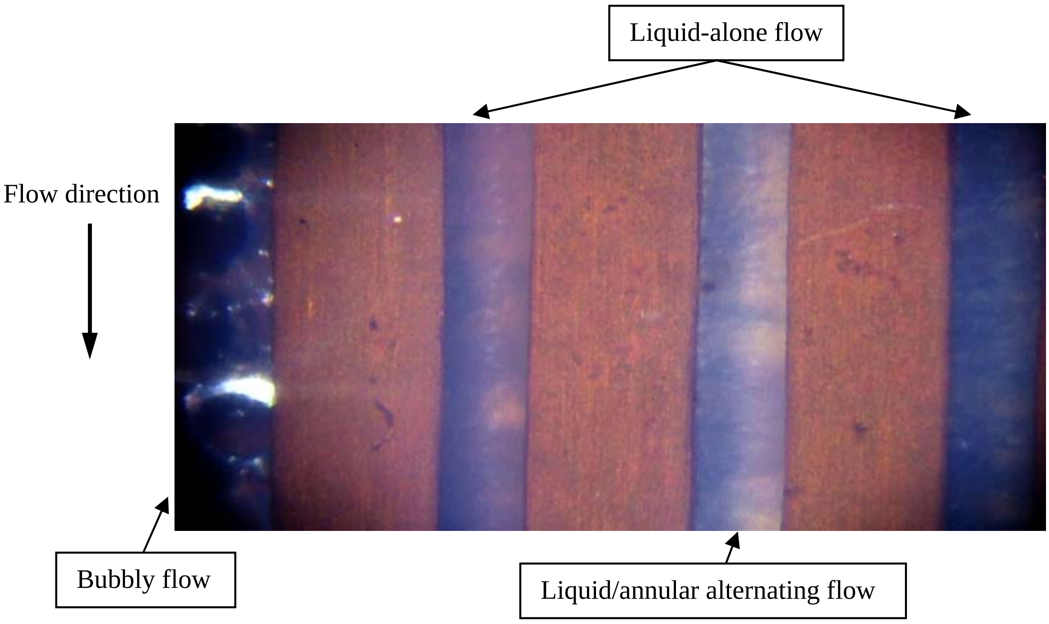

Fig. 5. Flow patterns in micro-channels.

gas flow rate, heat transfer enhancement is obtained, which agrees with the study done by Bao et al. [4]. The improved heat transfer is due to the gas injection in the channels resulting in turbulent mixing in the liquid film near the channel wall. Based on the flow visualization carried out by using stereozoom microscope and camera system, It was found that at a given condition, different flow patterns were observed in different channels as illustrated in Fig. 5. The observed flow patterns are classified differently according to the following definitions.

1) Bubbly flow: the bubbles with diameters less than the channel width.
2) Liquid-alone flow: the presence of only single-phase liquid flow in the channels.
3) Liquid/annular alternating flow: the alternating appearance of infrequent liquid slug and annular flow.

Liquid-alone flow and liquid/annular alternating flow were also observed in circular micro-channel as reported by Saisorn and Wongwises [11].

5. Conclusion

Heat transfer and fluid flow characteristics of non-boiling air-water flow in micro-channel heat sink are reported in this work. Both single-phase and two-phase flows are carried out in 21 parallel rectangular micro-channels. Each channel has a width and a depth of 0.33 and 0.46 mm, respectively. Two-phase heat transfer gives better results when compared with the single-phase flow. Heat transfer enhancement is obtained because air injected in the channels causes turbulent mixing in the liquid film near the channel wall. The flow visualization results show different flow patterns including bubbly flow, liquid-alone flow and liquid/annular alternating flow.

6. Acknowledgement

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7. References

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