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Experimental Investigation of Fluid Flow and Heat Transfer in Microchannels

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***Abstract* - An experimental investigation was conducted to explore the characteristics of fluid flow and heat transfer in rectangular microchannels using water as a working fluid in single-phase flow. The micro channels considered having width of 350 microns; with the channel depth 2.5 mm. Test piece was made of copper and contained twenty three microchannels in parallel. The experiments were conducted with water, with the Reynolds number ranging from 375 to 2000. Numerical predictions obtained based on a classical, continuum approach were found to be in good agreement with the experimental data, suggesting that a conventional analysis approach can be employed in predicting heat transfer behavior in microchannels of the dimensions considered in this study.**

Keywords: **Microchannel; Electronics cooling; liquid cooling; Heat flux; Heat sink**

Nomenclature
A ,heat transfer area m^2
cp , specific heat, kJ/kg
f ,friction factor
Dh hydraulic diameter
P, perimeter of microchannel
k ,thermal conductivity, W/m°C
h, convective heat transfer coefficient, W/ m^2
L ,channel length
N, number of microchannels
Nu ,Nusselt number
Pr ,Prandtl number
q ,heat transfer rate, W
Re, Reynolds Number
b, channel height
Q, volumetric flow rate, m^3/s
w,channel width
ρ density, kg/m^3
Subscripts
i inlet
o outlet
t, thermal
f, fluid
w, wall
m, mean

I. Introduction

Heat transfer in microchannels has been studied in a number of investigations, and has been compared with the behavior at ‘‘conventional’’ (i.e., larger- sized) length scales. However, there have been wide discrepancies between different sets of published results. Measured heat transfer coefficients have either well exceeded [1], or fallen far below [2, 3], those predicted for conventional channels. Possible reasons advanced to account for the deviation from classical theory have included surface roughness [4] electrical double layer [5] and aspect ratio [6] effects. The capability of Navier–Stokes equations to adequately represent the flow and heat transfer behavior in microchannels has been called into question in some of these studies. Tuckerman and Pease [7] suggested the use of microchannels for high heat flux removal; this heat sink is simply a substrate with numerous small channels arranged in parallel, such that heat is efficiently carried from the substrate into the coolant. Their study was conducted for water flowing under laminar conditions through microchannels machined in a silicon wafer. Heat fluxes as high as 790 W/cm^2 were achieved with the chip temperature maintained below 110ºC. Peng *et al.* [2, 8] experimentally investigated the flow and heat transfer characteristics of water flowing through rectangular stainless steel microchannels with hydraulic diameters of 133–367 µm at channel aspect ratios of 0.33–1. Their results showed that characteristics of flow in microchannels agree with conventional behavior predicted by Navier–Stokes equations. They suggested that deviations from classical behavior reported in earlier studies may have resulted from errors in the measurement of microchannel dimensions, rather than any micro scale effects. More recent studies have confirmed that the behavior of microchannels is quite similar to that of conventional channels. The present work complements the detailed flow field and pressure drop measurements of Liu and Garimella [10]. A systematic investigation is conducted of single phase heat transfer in microchannels of hydraulic diameter 0.61 mm, at flow Reynolds numbers of 380–2000. An important focus of this work is to examine the validity of conventional correlations and numerical analysis approaches in predicting the heat transfer behavior in microchannels, for correctly matched inlet and boundary conditions.

II. Experimental Setup and Procedures

A schematic of the experimental facility used in this investigation is shown in fig: 2.1.

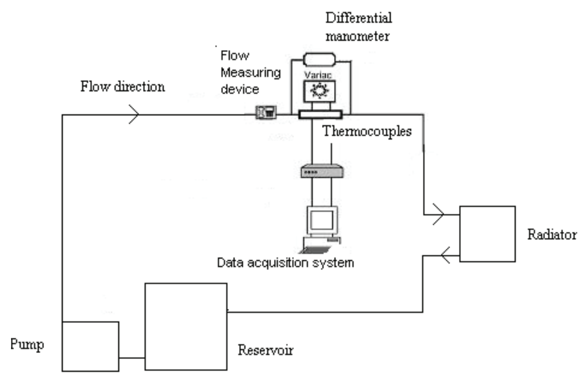

Fig. 2.1 Schematic of experimental setup

Water from a holding tank (reservoir) is driven through the flow loop using pump and provided smooth and steady flow over wide range of flow rates. A ball valve downstream of the flow meter allows fine adjustments of the flow rate from 0.5 L/min. to 2.5 L/min., which corresponds to a Reynolds number, range 385-1851. The fluid then enters into microchannel test section through inlet plenum. Heated water exits the test section and is passed through a radiator and goes to the holding tank again. Here we have used closed loop cooling system in our experimental set-up.

The microchannels have following dimensions:
Width = 350 microns and depth = 2.5 mm, using these dimensions the hydraulic diameter was calculated using the following expression.

$$D_h = 4A/P \tag{2.1}$$

The test section consists of a copper test block, a clear acrylic lid. The heat sink was machined from square block of dimensions 50 mm X 50 mm X 5 mm.The microchannels were cut into the top surface using EDM machine. Holes were drilled into four corners of the test block to house acrylic lid, which is mounted with help of four screws and a packing is also provided to make it leak proof. A pocket of 1.5cm X 1.5 cm is provided into the bottom of the copper block to house the cartridge heaters that can provide a combined maximum heat flux of 200 W/cm^2. Five thermocouples of 'J' type were embedded in the copper test section at 5 mm axial intervals. The temperature readings from these thermocouples are extrapolated to provide average microchannel wall temperature. Two thermocouples out of five were also located at the inlet and outlet of the test section to measure the fluid temperatures at these locations all thermocouples were read into a data acquisition system. The voltage input to the cartridge heaters was controlled by a watt hour meter. The steady state sensible heat gain by the coolant can be determined from an energy balance.

$$q = \rho c_p Q(T_{m,o} - T_{m,i}) \tag{2.2}$$

The volumetric flow rate Q is measured with a flow meter. The inlet and outlet fluid temperatures $(T_{m,o} and T_{m,i})$ are obtained using the two thermocouples positioned immediately upstream and downstream of the microchannels,respectively. Density and specific heat are calculated based on the mean fluid temperature T_m (average of the fluid inlet and outlet temperatures).The average heat transfer coefficient is determined from:

$$h = q/[NA(T_w - T_m)] \tag{2.3}$$

In which A is the area available for convection per channel (w+2b), N is the total number of channels, T_w is the average

Table I Convection Correlations For Flow in A Micro Channel Heat Sink

Conditions	Correlation
Laminar fully developed	$\frac{64}{\mathrm{Re}}$
Laminar fully developed, uniform heat flux $\Pr \geq 0.6$	$Nu = 4.36$
Laminar fully developed, uniform wall temperature $\Pr \geq 0.6$	$Nu = 3.66$
Laminar, thermal entry length, uniform wall temperature	$\overline{Nu} = 3.66 + \frac{0.0668(D/L)\mathrm{Re}\Pr}{1+0.04[(D/L)\mathrm{Re}\Pr]^{2/3}}$

temperature of the channel wall and T_m is the mean fluid temperature. Due to high thermal conductivity of copper, the uncertainty involved with such an estimation of the wall temperature is less than 1%.The corresponding average Nusselt number is calculated as :

$$Nu = hD_h / k_f \, .$$

A standard error analysis revealed uncertainties in the reported Nusselt numbers in the range of 6-17%.The uncertainties were greatest for a given micro channel test piece at the highest flow rates due to smaller increase in mean fluid temperatures from inlet to outlet. The following procedure was followed for the conduct of test. The test piece to be investigated was covered with the acrylic cover; water tight seal was effected using a silicone sealant between the mating surfaces. Once the test section was assembled, the pump was switched on, to provide the necessary pressure head to drive the coolant through the flow loop.

The desired flow rate for each test was set using the valve. After the flow rate stabilized, the heater power supply was switched on and maintained at the required level, and a steady state was reached in 30-45 minutes. Readings from all the thermocouples were stored using the data acquisition system throughout the duration of the experiment.

III. Results and Discussion

Commonly used heat transfer correlations for laminar flows in channels are enumerated in Table 3.1; these correlations have been widely employed in the literature for comparison against experimental results for microchannels. The correlations in the table are categorized according to the state of development of the flow and thermal fields and boundary conditions. It may be noted that although correlating equations given in Table I were originally developed for circular tubes, they have often been used for non-circular tubes with substitution of the hydraulic diameter D_h based on comparison with these conventional correlations, various conclusions have been drawn regarding their applicability to microchannel heat transfer which are following.

A. Variation of Pressure Drop with Reynolds Number

The variation of experimental pressure drop for Reynolds number varying from 375 to 1860 for the laminar range is shown in Fig. 3.1; it was observed that as the Reynolds number increases for a particular value of heat flux, pressure drop increases. As we know pressure drop is directly proportional to the square of velocity. Reynolds number is directly proportional to velocity of fluid so increase in pressure drop with increase in Reynolds number was justified. It was also observed that the increase in pressure drop with increase in heat flux was very small.

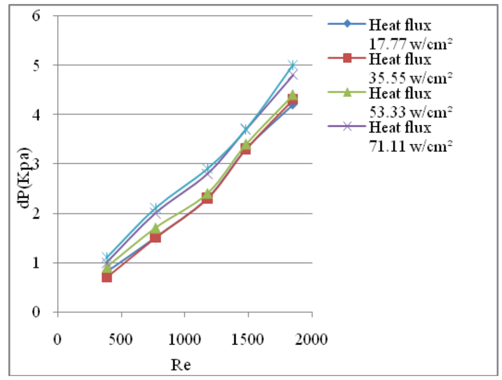

Fig. 3.1 Reynolds number versus pressure drop at different heat fluxes

B. Variation of Fanning Friction with Reynolds Number

The variation of experimental fanning friction factor with Reynolds number varying from 375 to 1860 for a microchannels having hydraulic diameter 0.61 mm and depth 2.5 mm is shown in figure 3.2. It was observed that as the flow rate increases friction factor decreases. The decrease in friction factor implies decrease in pressure drop. It was observed that fanning friction decreases with increase in Reynolds number up to a certain value, i.e. from 384 to 767,after Reynolds number 767, fanning friction increases at faster rates as observed from the figure 3.2. This increase is due to decrease in dynamic viscosity of the water with increase in temperature, which leads to more fanning friction factor. It was also observed that after certain value the fanning friction again starts decreasing.

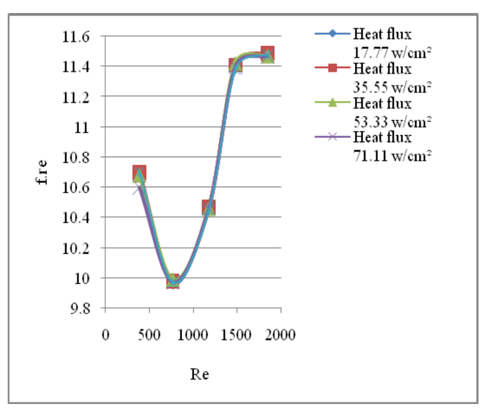

Fig. 3.2 Reynolds number versus experimental fanning friction factor at different heat fluxes

C. Variation of Fanning Friction with Heat flux

The variation of experimental fanning friction factor with heat flux varying from 17.77 W/cm^2 to 88.88 W/ cm^2 is shown in figure. It was observed that the variation of the fanning friction with heat flux was negligible. It was also observed that fanning friction increases with increase in Reynolds number, which was justified as we have also seen in figure 3.2 the variation of fanning friction with Reynolds number that fanning friction increases with increase in Reynolds number at various heat flux rates.

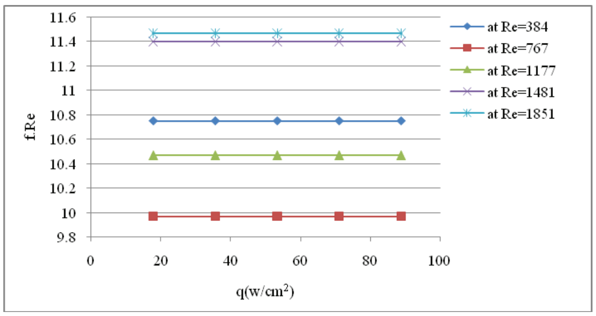

Fig. 3.3 heat flux verses fanning friction at different Reynolds numbers

D. Variation of Temperature Distribution Along Length of the Microchannel at Different Heat Fluxes

Figure 3.4 shows the variation of temperature along the length of microchannel heat sink (MCHS) at heat flux 17.77 W/cm^2 and at different flow rates. It was observed as liquid flow rate increases the heat removal rate also increases, that was due to the fact that heat removed is directly proportional to the mass flow rate or volume flow rate as given by the following relation: $q'' = mc_p\Delta T$

Similar trends were observed for other heat fluxes, i.e. for 35.55 W/cm^2, 53.33W/cm^2,71.11W/cm^2 and 88.88W/cm^2 as shown in figures 3.5, 3.6, 3.7 and 3.8, as flow rate was increased from 0.5 LPM to 2.5 LPM, and it was observed that the heat removal rate also increases rapidly.

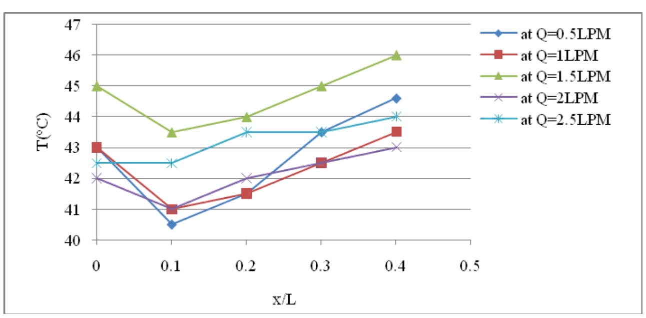

Fig. 3.4 temperature distribution along the length of micro channel heat sink (MCHS) in non-dimensional form at different flow rates and at heat flux 17.77 W/cm^2

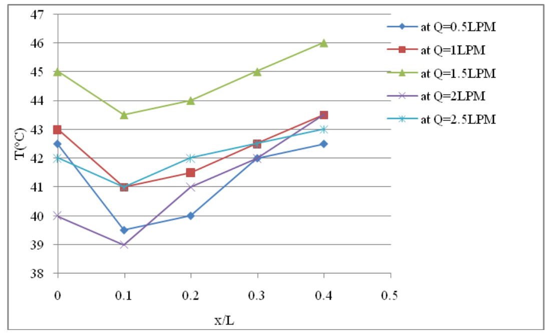

Fig. 3.5 Temperature distribution along the length of micro channel heat sink (MCHS) in non-dimensional form at different flow rates and at heat flux 35.55 W/cm^2

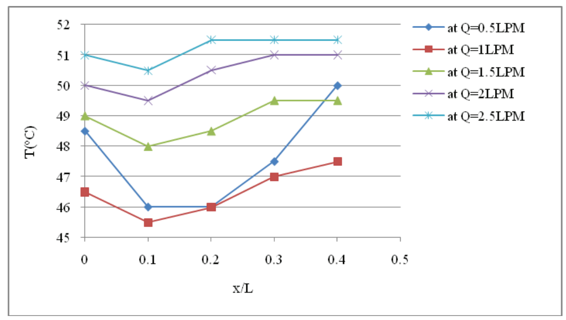

Fig. 3.6 Temperature distribution along the length of micro channel heat sink (MCHS) in non-dimensional form at different flow rates and at heat flux 53.33 W/cm^2

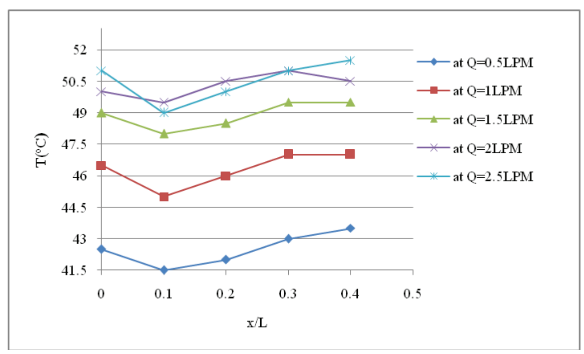

Fig. 3.7 Temperature distribution along the length of micro channel heat sink (MCHS) in non-dimensional form at different flow rates and at heat flux 71.11 W/cm^2

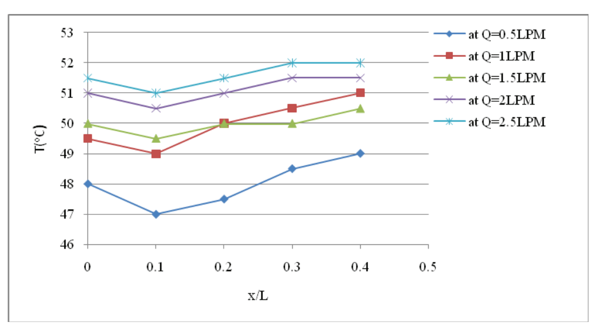

Fig. 3.8 Temperature distribution along the length of micro channel heat sink (MCHS) in non-dimensional form at different flow rates and at heat flux 88.88 W/cm^2

IV. Conclusion

Experimental analysis was made for the fluid flow and heat transfer with microchannels of hydraulic diameter 0.61 mm and length of channels 20 mm. The following conclusions were made through the experimental investigation:

- When mass flow rate decreases, friction factor also decreases due to decrease in velocity of flowing fluid.
- The pressure drop increases with increase in velocity of fluid, thereby increases heat flux removal rate.
- Initially fanning friction factor decreases as there is increase in mass flow rate. But after certain value of mass flow rate it starts increasing.
- The heat removal rate from microchannel heat sink increases with increase in mass flow rate, as with increase in mass flow rate pressure drop also increases, so that pressure loss is converted into heat which is carried by the water which was used as coolant in the flow loop.

Similar trends were also observed in conventional channels, which lead to the conclusion, even below 1 mm hydraulic diameter, conventional heat transfer equations/ methodologies hold well. Various literatures [2, 3, 5, 7, and 12] also validate the same.

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