Enhanced Microchannel Heat Transfer in Macro-Geometry using Conventional Fabrication Approach

K T Ooi¹, A L Goh²
¹School of Mechanical and Aerospace Engineering, Nanyang Technological University, Singapore 639798
²TUM-CREATE, Nanyang Technological University, Singapore 639798

E-mail: mktooi@ntu.edu.sg

Abstract. This paper presents studies on passive, single-phase, enhanced microchannel heat transfer in conventionally sized geometry. The intention is to allow economical, simple and readily available conventional fabrication techniques to be used for fabricating macro-scale heat exchangers with microchannel heat transfer capability. A concentric annular gap between a 20 mm diameter channel and an 19.4 mm diameter insert forms a microchannel where heat transfer occurs. Results show that the heat transfer coefficient of more than 50 kW/m²·K can be obtained for Re=4,000, at hydraulic diameter of 0.6 mm. The pressure drop values of the system are kept below 3.3 bars. The present study re-confirms the feasibility of fabricating macro-heat exchangers with microchannel heat transfer capability.

1. Introduction

Since Tuckerman and Pease [1] introduced the micro-channel heat transfer in solving high heat fluxes facing the electronics industries in 1981, today, microchannel heat transfer has been well studied and established. However the application of such a heat transfer technique is not widely employed but is only limited to few specific applications, where high heat fluxes occur.

It is believed that if micro-channel heat transfer phenomena can be applied to conventional heat exchangers, many benefits may result. Potential benefits are:

- Reduction in material cost. Since microchannel heat transfer is expected to provide higher heat transfer coefficients than the conventionally sized heat exchanger, thus a smaller heat transfer area is needed and hence a smaller heat exchanger will thus be required.
- Reduction in fabrication cost. Currently microchannels are fabricated using costly techniques involving MEMS and laser fabrication approaches. In this paper conventional machining

¹ To whom any correspondence should be addressed.
techniques such as turning, milling and cutting are used to produce microchannel in macro-geometry and this has significantly reduced the fabrication cost.

- Reduction in physical size. Compactness is an important consideration in many applications where space is a constraint. Having a smaller heat exchanger is an advantage in many applications where space is a constraint.

One common negative factor which microchannel flow imposed on actual application may be thought to be the high pressure drop which requires significantly higher pumping power. However, such an issue can be mitigated using design. For example, a shorter heat exchanger and hence lower pressure drop can be used if the heat transfer effect is significantly higher, when using microchannel.

In this paper, a purposed built test bed was fabricated to test enhanced microchannel heat transfer effects in macro-geometry using conventional fabrication techniques for single phase water flow.

Recent literature [2, 3] on microchannel heat transfer studies focused on implementation and understanding of the fluid flow and heat transfer phenomena. Kandlikar [3] in 2002, proposed that microchannel dimensions scale 10 to 200 μm. However, many others still classify microchannels broadly between 1 μm and 1 mm [4,5], or even simply ≤ 1 mm [6,7]. This paper adopts the generally accepted latter.

To increase single phase heat transfer in microchannel, the heat transfer enhancement techniques [8] were successfully introduced in microchannels in 1987. Passive heat transfer enhancement [9] can be achieved through: 1) thinning the thermal boundary layer, 2) increasing flow interruptions, and 3) increasing velocity gradient near the heated surface. Additionally, the redeveloping thermal boundary layer concept to enhance heat transfer has been experimentally verified [10].

Literature [4] shows that microchannel can be fabricated using: (i) micromechanical machining, (ii) X-ray micromachining, (ii) photolithographic-based processes, or (iv) surface and surface-proximity-micromachining. This paper proposed a readily available approach by combining two conventionally sized (hence forth called macro) geometries to form microchannel. This approach is relatively simple and more economical, since the macro geometries are fabricated using conventional machining processes.

Kong and Ooi [11] proved that microscale heat transfer in macro geometry using conventional fabrication techniques is indeed viable. In this paper, the focus is to enhance heat transfer by designing different geometrical profiles which disturb the flow, which is similar to those presented by Chang et al. [12] in 2005.

In this study single-phase liquid flow with Reynolds number range of 1,300-4600 is examined. Smooth microchannel with annular gap of 0.3 mm is used.

2. Experimental Setup

2.1. Test specimen

Figure 1 shows a 0.3 mm annular microchannel is created by placing a 19.4 mm diameter cylindrical insert within a 20 mm diameter circular channel. The flow region of interest has an axial length of 30 mm.
To disturb the flow, the enhanced heat transfer surface is achieved by creating grooves or steps as shown in figure 2. This geometrical profile is introduced on the outer surface of the cylindrical insert, suing conventional machining approach. Refer to figure 2, various parameters are defined and figure 3 shows the channels with different e/H ratio of 0.1, 0.3, 0.5 and 0.7, at constant P/e ratio of 10. Noted that H represents the channel height, which is fixed at 300 μm.

2.2. Experimental apparatus
Figure 4 illustrates the schematic of the experimental apparatus following works of researchers [6,13,14]. A chiller unit serves to cool the heated water to the desired inlet temperature before entering the test module. The air-vent loop serves to remove any air bubbles in the flow [2]. In the test loop, water is first channelled through a 40 micron filter to remove any solid particles. Coriolis flow meter was used to measure the flow. Temperature and pressure sensors are located at the entrance and exit of the test module to measure the corresponding key parameters.

2.3. Test module
Figure 5 shows the microchannel test module with the important components labelled. The heating coil supplied heat radially to the water via the inner copper surface. The length of the heat transfer section is 30 mm, the heat transfer surface area is always constant at 18.85 cm². Ten thermocouples are used to provide the copper wall temperature along the test section. The PEEK (k=0.92 W/m•K) and Mica (k=0.31 W/m•K) insulation are used to minimise heat loss.
2.4. Data reduction

Each steady-state measurement is computed as an average of 600 data points logged at 1 Hz. Steady state is deemed to be achieved when all temperature measurements fluctuate within 0.3 °C, all pressure measurements within 0.04 bar, and flow measurement within 0.05 L/min.

The steady-state sensible heat gain by the water is determined by the energy balance consideration:

\[ q = \rho c_p Q(T_o - T_i) \]  

(1)

The mean wall temperature, \( T_{w,m} \), is obtained by averaging ten local wall temperature measurements. The average heat transfer coefficient is then evaluated using eqn (2):
The heat transfer area, $A$, is fixed at 18.85 cm$^2$. The average Nusselt number is calculated from the average heat transfer coefficient:

$$ Nu = \frac{h D_h}{k_f} $$  \hspace{1cm} (3)

The pressure drop across the 30-mm microchannel is obtained by subtracting $\Delta P_{\text{entrance}}$ and $\Delta P_{\text{exit}}$ from $\Delta P_{\text{module}}$. The former two parameters are computed using the numerical model, whereas the latter is calculated from two pressure sensors positioned immediately before and after the test module.

$$ \Delta P_{\text{microchannel}} = \Delta P_{\text{module}} - (\Delta P_{\text{entrance}} + \Delta P_{\text{exit}}) $$  \hspace{1cm} (4)

The Darcy friction factor $f$ is determined by:

$$ f = \frac{\Delta P_{\text{microchannel}} D_h}{\frac{1}{2} \rho u_m^2 L} $$  \hspace{1cm} (5)

### 2.5. Operating conditions

The present study aims to investigate the effect of varying $e/H$ ratio of 0.1, 0.3, 0.5 and 0.7, at constant $P/e$ ratio of 10. The study is conducted at following conditions:

- i. $2.0 \leq \text{flow rates} \leq 7.0 \text{ L/min}$,
- ii. $1,300 \leq \text{Re} \leq 4,600$, covering both laminar and non-laminar flows, and
- iii. heat flux $\approx 53 \text{ W/cm}^2$.

### 2.6. Experimental Uncertainties

The experimental uncertainties [15-18] are listed in Table 1, with 95% confidence level. The low uncertainty in Reynolds number is due to the use of Coriolis flow meter, which provides an uncertainty of $\pm 0.1\%$ of flow reading. The uncertainties for pressure drop and friction factor falls within the general range of 10-15% reported by many researchers [7].

| Parameter       | Maximum uncertainty |
|-----------------|---------------------|
| $h$             | 7.6%                |
| Nu              | 7.7%                |
| Re              | 0.5%                |
| $\Delta P_{\text{module}}$ | 10.2%            |
| $\Delta P_{\text{microchannel}}$ | 10.2%           |
| $f$             | 10.2%               |

### 3. Results and Discussion

To ensure that the collected data from the measurement setup is valid, the results of the plain channel are used for validation. The pressure drop values across the plain microchannel, are first validated by
comparing with correlations from classical theory [19-21], the discrepancy ranging from 0.11% to 5.24% as shown in Table 4.

Table 2. Validation of friction factor

| Flow Rate (L/min) | Measured f | Predicted f | Discrepancy |
|-------------------|------------|-------------|-------------|
| 2.00              | 0.086      | 0.082       | 4.28%       |
| 3.00              | 0.063      | 0.060       | 3.78%       |
| 4.25              | 0.051      | 0.051       | 0.11%       |
| 7.00              | 0.042      | 0.045       | -5.24%      |

Table 3 shows that the measured Nusselt number varies within 3.3% when the Reynolds number and Prandtl number are maintained within 0.2%.

Table 3. Validation of Nusselt number

| Re   | Pr  | Nu  | Heat Input (W) |
|------|-----|-----|----------------|
| 3798 | 5.53| 46.21| 1000           |
| 3791 | 5.53| 47.72| 250            |

Discrepancy -0.2% 0.0% 3.3%

Furthermore, repeatability test has also been conducted and the maximum discrepancy in pressure drop and Nusselt number measurements are 0.57% and 0.39% respectively, indicating excellent repeatability.

Figure 6 shows that Nusselt number is directly proportional to e/H ratio, at constant P/e ratio. This is physically reasonable, considering that the minimum gap through which the fluid flows decreases as e/H ratio increases.

Figure shows that friction factor is directly proportional to e/H ratio. This is because a larger fluid velocity accompanies the decrease in minimum gap, as e/H ratio increases.

![Figure 6. Nusselt number against e/H ratio, at various Reynolds number](image)
4. Conclusion
An economical and readily available conventional fabrication technique coupled with a novel configuration for enhanced heat transfer in microscale passages has been tested. Major findings from the present study include:

- The highest convective heat transfer coefficient of 52.8 kW/m²·K at 6.5 L/min with pressure drops lower than 3.3 kPa can be achieved.
- The configuration proposed is potentially useful for many practical applications, especially in the design of heat exchangers.
- In the whole study, the pressure drop values are all less than 3.3 bars, which points towards feasible practical applications.

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