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

K T Ooi1[1], A L Goh[2]
[1]School of Mechanical and Aerospace Engineering,
Nanyang Technological University, Singapore 639798
[2]TUM-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/m^2•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.

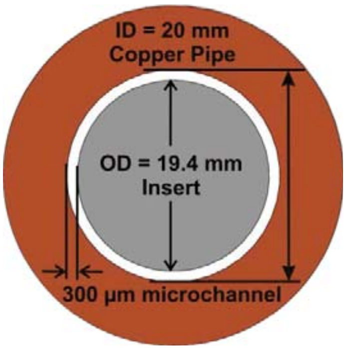

Figure 1. Front view of flow channel of interest

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.

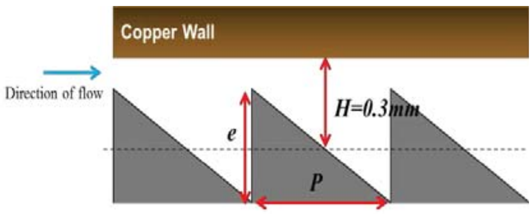

Figure 2. Inverted Fish Scale Profile parameters

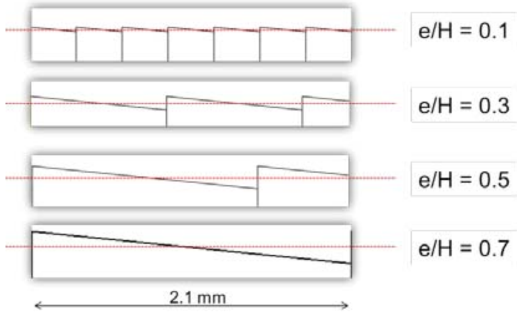

Figure 3. Flow Channels with different *e/H* ratios

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 cm^2 . 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.

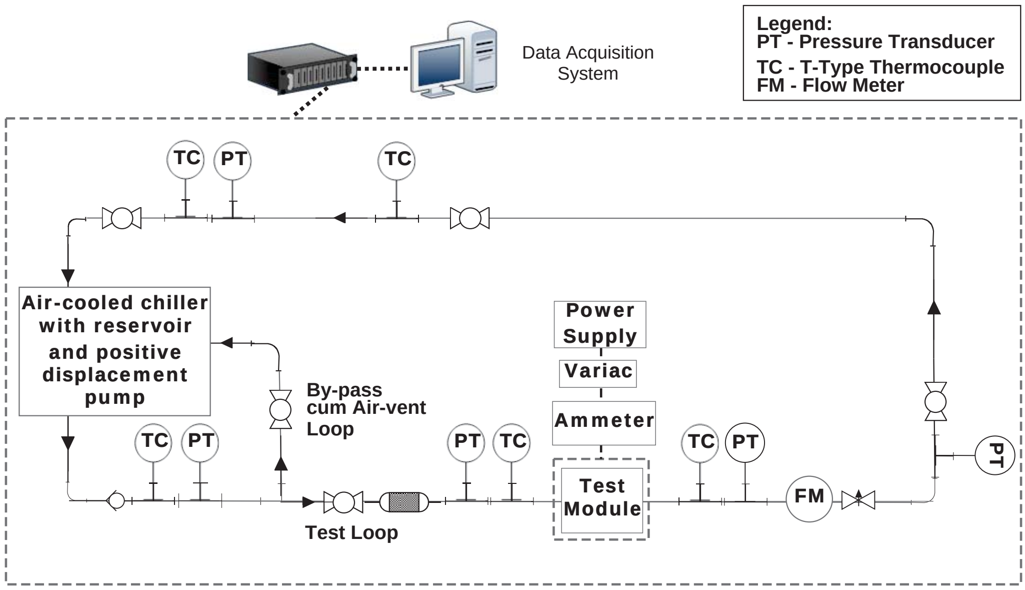

Figure 4. Experimental Apparatus

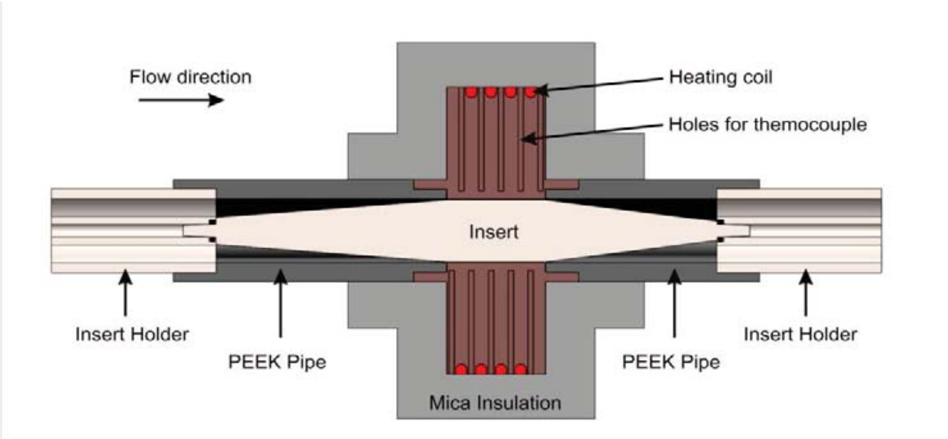

Figure 5. Sectional view of test module

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:

$$q = \rho c_p Q (T_o - T_i) \tag{1}$$

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):

$$h = \frac{q}{A(T_{w,m} - T_{f,m})} \tag{0}$$

The heat transfer area, A, is fixed at 18.85 cm^2. The average Nusselt number is calculated from the average heat transfer coefficient:

$$Nu = \frac{h\, D_h}{k_f} \tag{3}$$

The pressure drop across the 30-mm microchannel is obtained by subtracting $\Delta P_{\mathrm{entrance}}$ and ΔP_{exit} from $\Delta P_{\mathrm{module}}$. 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.

$$\Delta P_{microchannel} = \Delta P_{module} - (\Delta P_{entrance} + \Delta P_{exit}) \tag{4}$$

The Darcy friction factor f is determined by:

$$f = \frac{\Delta P_{microchannel}}{\frac{1}{2}\,\rho\, u_m^2} \frac{D_h}{L} \tag{5}$$

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. $2.0 \leq$ flow rates ≤ 7.0 L/min,
ii. $1{,}300 \leq \mathrm{Re} \leq 4{,}600$, covering both laminar and non-laminar flows, and
iii. heat flux ≈ 53 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 ± 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].

Table 1. Experimental Uncertainties

Parameter	**Maximum uncertainty**
h	7.6%
Nu	7.7%
Re	0.5%
ΔP_{module}	10.2%
$\Delta P_{microchannel}$	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.

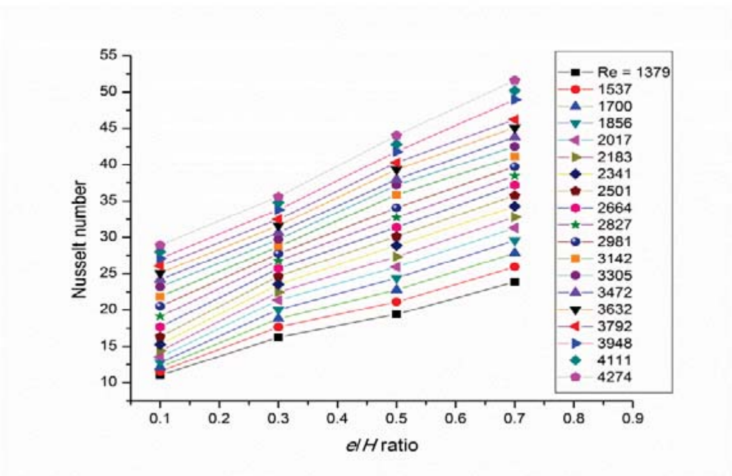

Figure 6. Nusselt number against *e/H* ratio, at various Reynolds number

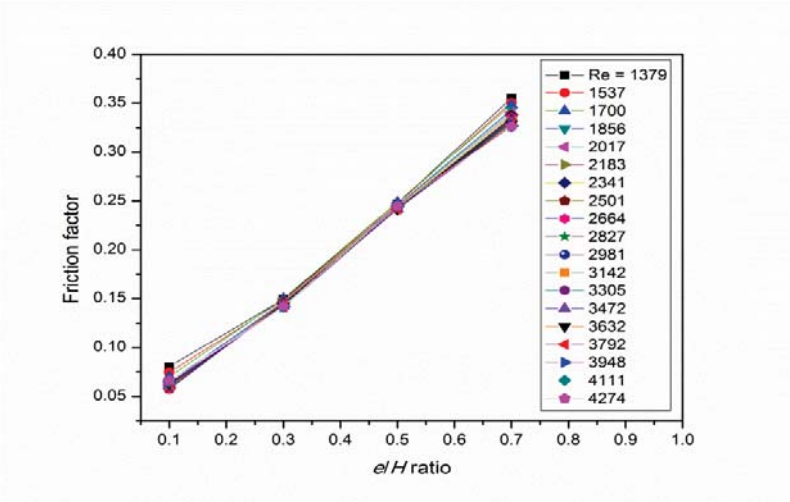

Figure 7. Friction factor against *e/H* ratio, at various Reynolds number

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/m^2•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.

References

[1] Tuckerman D B and Pease R FW 1981 High-performance heat sinking for VSLI, Electron Device Lett. 2, 126–129.
[2] Kandlikar S G 2012 History, Advances, and Challenges in Liquid Flow and Flow Boiling Heat Transfer in Microchannels: A Critical Review, ASME J. Heat Transf. 134 034001–1
[3] Kandlikar S G 2002 Fundamental issues related to flow boiling in minichannels and microchannels, Exp. Therm. Fluid Sci. 26 389–407
[4] Morin G L 2004 Single-phase Convective Heat Transfer in Microchannels: A Review of Experimental Results, Int. J. Therm. Sci. 43 631–651
[5] Fan Y and Luo L 2008 Recent Applications of Advances in Microchannel Heat Exchangers and Multi-Scale Design Optimization, Heat Transf. Eng. 29, 461–474
[6] Khan M G and Fartaj A 2011 A review on microchannel heat exchangers and potential applications, Int. J. Energy Res. 35, 553–582
[7] Obot N T 2002 Toward a better understanding of friction and heat/mass transfer in microchannels — a

literature review, Microscale Thermophys. Eng. 6, 155–173
[8] Kishimoto T and Sasaki S 1987 Cooling characteristics of diamond-shaped interrupted cooling fin for high-power LSI devices, Electron. Lett. 23, 456–457.
[9] Tao W Q, He Y L, Wang Q W, Qu Z G and Song F Q 2002 A unified analysis on enhancing single phase convective heat transfer with field synergy principle, Int. J. Heat Mass Transf. 45, 4871–4879
[10] Xu J, Gan Y, Zhang D and Li X, Microscale heat transfer enhancement using thermal boundary layer redeveloping concept, Int. J. Heat Mass Transf. 48, 1662–1674.
[11] Kong K S and Ooi K T 2013 A numerical and experimental investigation on microscale heat transfer effect in the combined entry region in macro geometries, Int. J. Therm. Sci. 68, 8–19
[12] Chang S W, Liou T M and L M H 2005 Heat transfer of rectangular narrow channel with two opposite scale-roughened walls, Int. J. Heat Mass Transf. 48, 3921–3931
[13] Lee P S, Garimella S V and Liu D 2005 Investigation of heat transfer in rectangular microchannels, Int. J. Heat Mass Transf. 48, 1688–1704
[14] Qu W and Mudawar I 2002 Experimental and numerical study of pressure drop and heat transfer in a single-phase micro-channel heat sink, Int. J. Heat Mass Transf. 45, 2549–2565
[15] Bell S 2001 A Beginner's Guide to Uncertainty of Measurement, National Physical Laboratory
[16] (UKAS), 2012 M3003 - The Expression of Uncertainty and Confidence in Measurement, United Kingdom Accreditation Services
[17] (JCGM), 2008 Evaluation of measurement data—Guide for the expression of uncertainty in measurement, Joint Committee for Guides in Metrology
[18] Taylor B N and Kuyatt C E1994 NIST Technical Note 1297 Edition - Guidelines for Evaluating and Expressing the Uncertainty of NIST Measurment Results
[19] Shah R K and London A L 1978 Laminar flow forced convection in ducts - Supplement 1 to Advances in Heat Transfer, Academic Press, New York
[20 Shah R K 1978 A Correlation for Laminar Hydrodynamic Entry Length Solutions for Circular and Noncircular Ducts, J. Fluids Eng. 100, 177–179.
[21] Phillips R J 1987 Forced-Convection, Liquid-Cooled, Microchannel Heat Sinks, M.S. Thesis, Massachusetts Institute of Technology