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Design and Experimental Issues with Heat Exchangers for Microfluidics

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Abstract

In this paper we explore issues in designing a heat exchanger for a microfluidic channel. Among important issues specific to heat exchangers in microfluidics are axial conduction and shear rate dependant phenomena. A heat exchanger has been designed for use in a further study to examine the effect of liquid contact angle on the convective heat transfer coefficient. We show that by applying a constant heat flux to the outer channel wall surface, axial conduction causes the boundary conditions to change from uniform temperature at lower Reynolds numbers, to approach uniform heat flux at higher Reynolds numbers.

Introduction

Microfluidics, the generic name given to the study of fluid flow in confined channels with dimensions generally less than approximately 1mm, has exploded onto the fluid mechanics scene in the last 5 years or so [1]. In general, microfluidics research has been driven by two main application areas: high surface area heat sinks for high-flux computer chip cooling using multiple microchannels arrays [2]; and functional devices on a chip for biotechnology (lab-on-a-chip or micro total analytical systems) that increase the speed and sensitivity of processes that often involve heating and temperature control of reagents and products [3]. As both these applications involve convective heat transfer from a solid to a fluid, a fundamental understanding of microscale processes is required for device design. The question then arises: Can we use standard macro scale heat transfer correlations for microscale flows?

There has been considerable debate in the literature over the applicability of macro scale Nusselt number (non dimensional heat transfer coefficient) correlations to microfluidic flow. This is due, primarily, to the particularly large variation in experimental data (see for example [4-9]). There are various reasons for the lack of consistency in the experimental data but most are due to the following:

- Approximations and assumptions made due to the difficulty in measuring bulk fluid temperatures where it is almost impossible to place a temperature measurement device in a microchannel without effecting the flow [5, 8]
- Heat loss due to the increased surface area to volume ratio in microchannels relative to macrochannels [10]
- Relative dimensional tolerances in fabrication (e.g one micron tolerance in a 30 micron channel is significant compared to a 0.1mm tolerance in a 100mm channel)
- Axial conduction due to the thickness of the wall relative to the channel destroying ideal conditions [10, 11]
- Ill-defined surface variability such as roughness and hydrophobicity [9]

Slip flow, where there is exists a none zero fluid velocity at a solid wall, has been well documented in the literature both experimentally [12-15] and by using molecular dynamics simulations [16-18]. Slip flow is important as it may allow a significant reduction in the friction pressure drop and thus the pumping power required for micro heat-exchangers and other microfluidic devices. While the underlying physical cause of slip is not fully understood, what is clearly known is that an apparent slip occurs more readily on non-wetting surfaces (hydrophobic), on rough surfaces, and at high shear rates. The last two phenomena are why slip may becomes important in microchannels, for it is in microchannel flow where surface roughness becomes significant relative to the channel size and where it is possible to obtain large shear rates (see shear rate section below).

Given that there must be a weaker interaction between the fluid and wall molecules for slip to occur, the corollary to the reduced pressure drop is a reduced heat transfer coefficient. To date in the literature there is only one study that investigates the effect of hydrophobicity on the heat transfer rate [9]. This study is limited to flow inside silicon and hydrophilic glass microchannels, and it shows an increase in the Nusselt number for the glass microchannels, relative to the silicon channels equal to approximately 10% (depending on the Reynolds number).

In this paper we consider several issues in the design of a heat exchanger for a microchannel that will be used for testing the effect of different fluid/wall contact angles on heat transfer coefficient. The main design considerations were surface access prior to assembly to allow surface modification via plasma polymerisation, small enough hydraulic diameters to obtain high enough shear rates to expect slip, the ability to measure the heat transfer surface temperature accurately and the minimisation of expensive microfabrication techniques.

Experiments

Figure 1 shows the schematic design of the heat exchanger. It has been designed to mimic flow between parallel plates (width >> channel depth) with one wall at a constant heat flux (or temperature depending on the flow rate) and the other wall adiabatic. This corresponds to a well studied geometry for which the Nusselt number is a constant 5.35 (or 4.86) [19].

The heat transfer surface consists of 2mm thick brass that has been imbedded into a Poly(methyl methacrylate), (PMMA or its trade name Perspex) base and polished flat using a Logitech PM5 precision lapping and polishing machine (the RMS roughness as measured by an atomic force microscope was found to be approximately 50nm). PMMA is a good thermal insulator (see table 1) meaning that the flow should be hydrodynamically fully developed before flowing over the brass heat transfer surface.

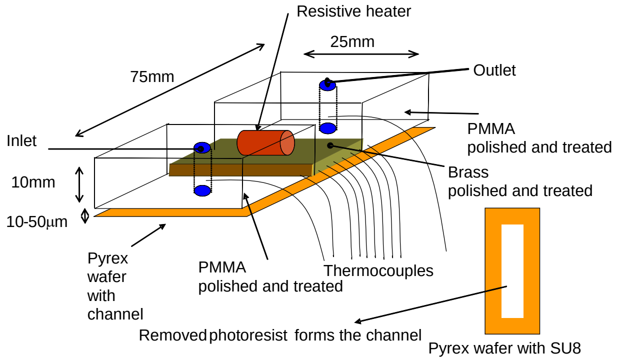

Figure 1: Schematic of the microfluidic heat exchanger.

The channel is formed by spinning a photoresist (in this case we used SU8, due to its excellent mechanical strength and solvent resistance) on to a 100mm diameter Pyrex wafer to the desired thickness (10-50µm) and removing the 8mm by 50mm channel (by UV exposure through a mask and developing). The wafer was then cut into the desired size (75mm by 25mm). The glass wafer with channel was sealed to the heat exchanger with a clamping system. The channel depth was measured with a microscope after sealing. Heating was provided by a 10W 47Ω resistor bonded to the back of the brass using high thermal conductivity epoxy (Tra-Con 2902) and heated with a calibrated HP power supply. Six 70µm diameter type T thermocouples were imbedded in the brass, and one thermocouple was imbedded on each side of the brass in the PMMA, with their tips at the centre of the channel. Small diameter wires were used to minimise conduction losses. A thermocouple was also placed at the centre of the inlet and outlet channels.

	PMMA	Brass
Thermal Conductivity (W/m.K)	0.18	112
Thermal Expansion coeff (m/m//K)	73e-6	18.5e-6
Specific heat (J/kg.K)	1460	380
Density	1160	8500

Table 1: Physical properties of heat exchanger material.

The temperature measurement system was calibrated against a traceable platinum resistance thermometer in a uniform temperature water bath. Data was captured to computer via a high precision National Instruments 4350 data logger and a constant temperature terminal block.

Theory

Shear rate

The velocity profile for flow between parallel plates is given by [19]

$$\frac{u}{u_m} = \frac{3}{2}\left(1-\left(\frac{y}{w}\right)^2\right) \quad , \qquad (1)$$

where u is the velocity, u_m is the mean velocity, y is the distance between the channel walls with the origin at the centre of the channel, and w half the channel height. The maximum shear rate is at the wall and is

$$\left.\frac{du}{dy}\right|_{y=w} = \dot{\gamma}_{\max} = \frac{3u_m}{w} = \frac{3\mathrm{Re}_{Dh}\,\nu}{2w^2} \quad , \qquad (2)$$

where Re_{Dh} is the Reynolds number based on the hydraulic diameter, $D_h=2w$, and ν is the kinematic viscosity. Equation (2) illustrates the fact that for the same Reynolds number the maximum shear rate scales as the reciprocal of the channel height squared. That is why it is much more likely to observe slip in microchannels. In fact there tends to be a critical value where slip has been observed as summarised in Table 2.

	Critical shear rate (s^{-1})	Comments
Wu and Cheng, 2003 [9]	~50,000	Roughness ~10nm
Zhu and Granick 2002 [13]	~10,000	Roughness ~10nm
Choi et al 2002 [12]	~10,000	

Table 2: Example of some critical shear rates extracted from the literature from which either a critical slip length was specified or for which it has been inferred from the departure of hydrophobic and hydrophilic data.

The maximum channel height used for this heat exchanger is approximately 50µm, which, with the available flow rates from our pump corresponds to a maximum shear rate of approximately $10^6 s^{-1}$.

Axial conduction

The ratio of conduction heat transfer through a channel wall, to the convective heat transfer to the fluid from the wall, assuming the same temperature difference for the two processes, has been presented previously as a non-dimensional number to define compact heat exchanger efficiency [20] (see equation 3). This number, re-arranged, has been recently applied to microfluidics where walls are generally thicker than the fluid channels [11, 21] and is given by:

$$\lambda = \frac{A_s D_h \kappa_s}{A_f L \kappa_f}\frac{1}{Pe} \quad , \qquad (3)$$

where A_s and A_f are the cross sectional areas of the solid wall and fluid respectively, L is the channel length, κ is the thermal conductivity, and Pe is the Peclet number given by $\mathrm{Re}_{Dh}\mathrm{Pr}$. Given a constant heat flux on a channel outer wall (as in Figure 1) for low values of λ (<0.01) axial conduction can be considered negligible, and the fluid/solid surface boundary condition can be considered as having constant heat flux. For higher values of λ axial conduction becomes more important, and the boundary conditions approach constant temperature for infinite λ. Microfluidic channels are generally characterised by large As/Af and low Pe, hence the tendency for axial conduction to become significant.

Numerical Simulations

In order to aid in the understanding of the heat transfer processes in the heat exchanger design the commercial package CFD-ACE (ESI-Group) was used to solve the coupled flow, convection and conduction heat transfer. The mass conservation, the Navier-Stokes and the energy -in the form of total enthalpy- equations were solved by numerically integrating over each of computational cells or control volumes defined by the grid.

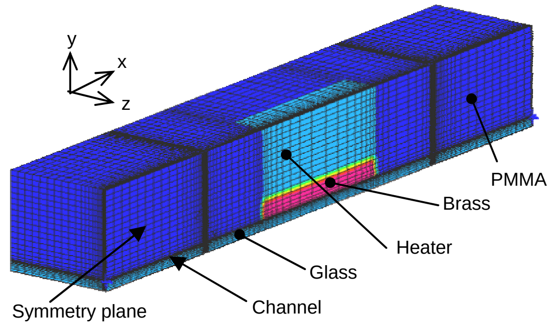

Figure 3: Three dimensional grid used for the numerical modelling.

Results and Discussion

Inlet configuration

In order to assume flow between parallel plates, the flow from a single inlet should spread evenly along the width of the channel. The 3D CFD model was used to determine the uniformity of the flow under various conditions. Numerical simulations and flow visualisation experiments indicate that a spreader at the inlet is not required. It has been found that with the flow entering the microchannel from the top through an inlet with diameter much greater than the channel height, the impinging jet tends to spread the liquid evenly across the channel width. This was verified experimentally with streak line visualisation using fluorescent 2μm particles. Figure 4 shows a superposition of streak lines from experiments with those from CFD results for Re = 2 (note that the inlet was not perfectly centred in the channel for the experimental streaklines).

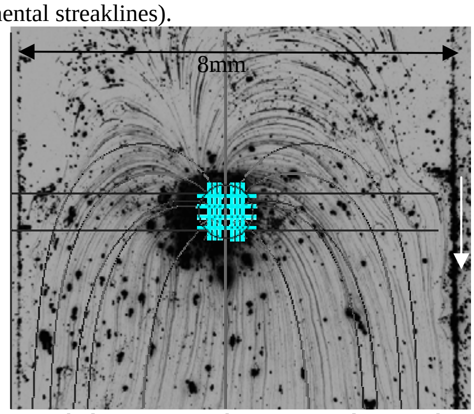

Figure 4: Streak line images taken using a long working distance fluorescent microscope and by using CFD simulations for Re =2. Flow from the inlet in the direction as indicated by the arrow.

Figure 5 shows the predicted velocity profiles at the centre plane in the y direction (middle of the channel depth) for a range of Reynolds numbers. The excellent uniformity along the width of the channel further than 200μm from the wall indicates that the assumption of flow between parallel plates is reasonable. Figure 5b shows that the flow becomes fully developed within 5mm from the inlet, before which little heat is transferred to the fluid (see next section).

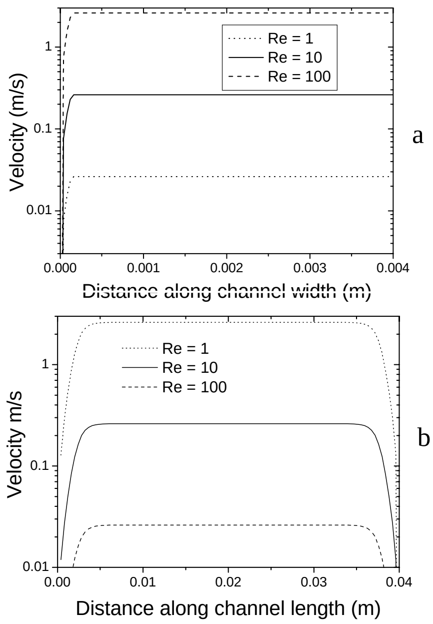

Figure 5: Plots of velocity profile from CFD simulations across the width (a) and the length (b) of the channel. In both plots the data is taken in the centre line of the channel.

Heat transfer results

The effect of different Reynolds numbers on the wall temperature distribution in the heat exchanger is shown in Figure 6 from both experiments and from the simulations. For the low Reynolds number where the effect of wall conduction is more significant the simulations tend to under-predict the amount of heat being transferred axially to the PMMA. This discrepancy requires further investigation but it is most likely at least partly due to the fact that the heat transfer coefficient is not uniform over all the external surface area. Future experiments with more insulation should help reduce the axial conduction. The agreement between experiments and the simulations is better for the higher Reynolds number case where heat loss and axial conduction effects are reduced. In general however the results show that the brass wall is almost at a constant temperature at Re =2 and slightly increasing at Re =20.

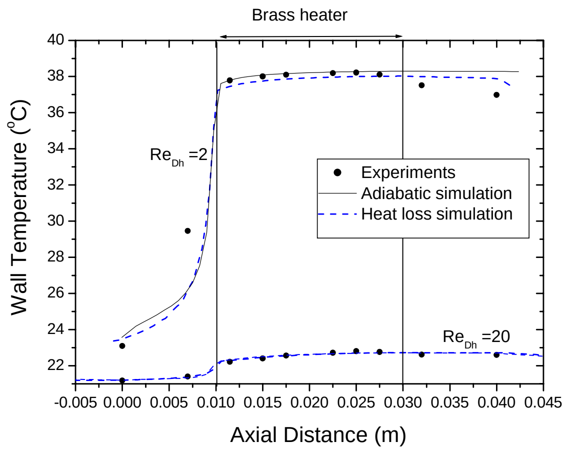

Figure 6: Measured and simulated wall temperatures along the wall of the heat exchanger with a heat input of 0.53W, for two different Reynolds numbers. The simulation includes a perfectly insulated case (adiabatic) and a simulated heat loss case that assumed a convective heat transfer coefficient of approximately 1W/m^2.K (calculated from cooling experiments).

The Nusselt number is the standard non-dimensional heat transfer coefficient and is defined as:

$$Nu_x = \frac{hD_h}{\kappa} = \frac{q''D_h}{\kappa\left(T_w(x) - T_m(x)\right)}, \qquad (4)$$

where h is the convective heat transfer coefficient, $T_m = \frac{1}{UA_c}\int_{A_c} uT dA_c$ is the mean fluid temperature, T_w is the wall temperature, q'' is the wall heat flux and κ the thermal conductivity of the fluid. One of the major issues with experimental evaluations of the Nusselt number in microfluidics is that it is almost impossible to measure the mean fluid temperature, especially because the temperature gradients, like the velocity gradients, are so large across the channel. Thus it is probably not the most useful number to use for experimental micro heat exchanger data.

Figure 7 shows the normal heat flux distribution from the channel wall into the fluid along with the Nusselt number calculated using equation (4) from numerical simulations. Due to axial conduction the brass plate is closer to a uniform temperature boundary condition than to the uniform heat flux boundary condition that was applied to the outer wall. As the Reynolds number is increased, however, the heat flux becomes more uniform along the channel length due to the decrease in the axial conduction effect (see equation (3)).

An interesting feature of the Nu results is the drop to a negative value just before the brass heater. As the glass lid has a thermal conductivity five times higher than the PMMA, heat is conducted axially towards the inlet. In the region adjacent to the heater the glass wall is hotter than PMMA surface and the fluid is heated primarily from the glass. Thus the fluid temperature is slightly warmer than the PMMA surface adjacent to the brass. This gives rise to the negative Nusselt number shown in figure 7.

The important feature from figure 7 is the fact that although a uniform heat flux was applied to the external wall the internal wall is essentially at a uniform temperature with a Nusselt number very close to the theoretical value of 4.86. In the PMMA which heated via conduction from the brass, the Nusselt number is very close to the uniform heat flux theoretical value of 5.35, although an insignificant amount of heat is transferred in that region.

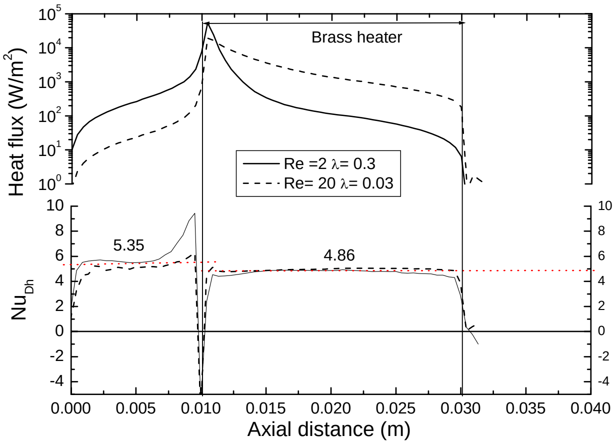

Figure 7: Local heat flux and Nusselt number distributions for the same cases given in Figure 6.

Conclusions

We have demonstrated a heat exchanger design for microfluidics that will allow the testing of the effect of contact angle on the heat transfer coefficient at high wall shear rates. We have shown using CFD that due to the large channel wall cross-sectional area relative to the fluid channel dimensions, the affect of axial heat conduction becomes important. In fact, when the Reynolds number, or more importantly the axial conduction ratio, λ, is small, it is not possible to achieve the preferred constant heat flux boundary condition on the channel wall. However at higher Reynolds numbers and thus shear rate, where apparent wall slip effects will become visible, λ increases and the boundary conditions tend to constant heat flux, which is more desirable to measure small changes in temperature due to slip. Finally the complex nature of the Nusselt number variation along the flow direction indicates how experimental studies of heat exchangers for microfluidics can become rather confused if detailed coupled simulations are not carried out simultaneously.

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