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Investigations on Single-Phase Liquid Flow through Semi-Circular Microchannels

Ravindra Kumar[1], Mohd. Islam[2] and M.M.Hasan[2]

[1]*Senior Lecturer, Department of Mechanical Engineering, G.B.Pant Institute of Technology, New Delhi, India.*
[2]*Professor, Department of Mechanical Engineering, Jamia MilliaIslamia, New Delhi, India.*

Abstract

An experimental investigation has been conducted to explore the validity of classical correlations based on conventional size channels for predicting the flow and heat transfer behaviour in single-phase liquid flow through semi-circular microchannels. The test piece used in the experiment was made of copper and consists of ten microchannels in parallel having hydraulic diameter of 214 µm and 60 mm length of semi-circular shape. The experiments were performed with deionised water as coolant and Reynolds number ranging from 238 to 1250. The experimental results obtained for friction factors showed that conventional theory for fully developed laminar flow is applicable with the range of our experiments. Classical correlations for macrochannels are evaluated with average Nusselt number obtained for microchannel with the assumption of constant wall heat flux. Slight deviations were observed in the experimental results obtained for Nusselt number when compared with earlier correlations, due to conjugate effect, variation in measurement of interface temperature and neglecting heat transfer through other surfaces of MCHS. 3-D conjugate heat transfer analysis is performed numerically for steady state condition. To predict heat transfer rates accurately, the entrance effects, conjugate effects, axial heat conduction and boundary conditions must have to be carefully considered.

Keywords: Semi-Circular Microchannels, friction factor, Reynolds number, Nusselt Number

INTRODUCTION

With the development of new micro-machining technologies, the size of Micro-Electro-Mechanical Systems (MEMS) and other electronic gadgets have drastically reduced in recent years. As the size of a device reduces the heat flux density increases, which may cause overheating due to which proper functioning and overall well being of these devices is a big concern for researchers. So highly efficient cooling technologies and heat dissipation methods are required to develop which can cater the safety and stable operation. In order to overcome the problem of high heat flux removal from small area, microchannel heat sink is right choice, as it provides high surface area to volume ratio, high convective heat transfer coefficient with small amount of coolant compared to conventional heat exchanger. In a microchannel heat sink (MCHS) the heat from a hot surface is taken away by forcing a fluid through small passages of hydraulic diameter varying from 10 µm to 200 µm[1].The first landmark study in the field of heat transfer using microchannel heatsink for electronic cooling was demonstrated by Tuckerman and Pease [2] in 1981, this pioneer work have paved the door for further research in the area of microchannel heat transfer.

Mala & Li[3] conducted experiments on microtubes (50µm $\leq Dh \leq$ 254µm) made of fused silica and stainless steel using water as test fluid. At higher values of Reynolds number, they obtained larger friction factor than the predicted values based on conventional theory and reported that deviation from the macroscale theory increases as the Reynolds number increases and diameter decreases.. Jiang et al.[4] in their experimental study on water flow through trapezoidal microchannels with(35 µm $\leq Dh \leq$120 µm) for Reynolds number less than 30 found experimental friction factor was lower than the conventional theory. Further Xu et al.[5] also found similar results while performing experimental investigations on water flows through microchannels (35 µm $\leq Dh \leq$ 120 µm) and revealed that experimental friction factor was smaller than that predicted by Hagen-Poiseulle flow theory for channel dimensions below 100µm. Judy et al.[6] conducted experimental study on frictional characteristics of flows of distilled water, methanol and isopropanol through circular and square microchannel of fused silica and stainless steel (47 µm $\leq Dh \leq$ 101 µm).Their results did not found any significant deviation in experimental pressure drop data from classical theory data. Park and Punch[7] conducted experimental investigations on deionised water through rectangular silicon microchannels (106 µm $\leq Dh \leq$307µm) and observed excellent coherence between experimental and theoretical values for friction factor when Reynolds number kept in the range($69<Re<800$).

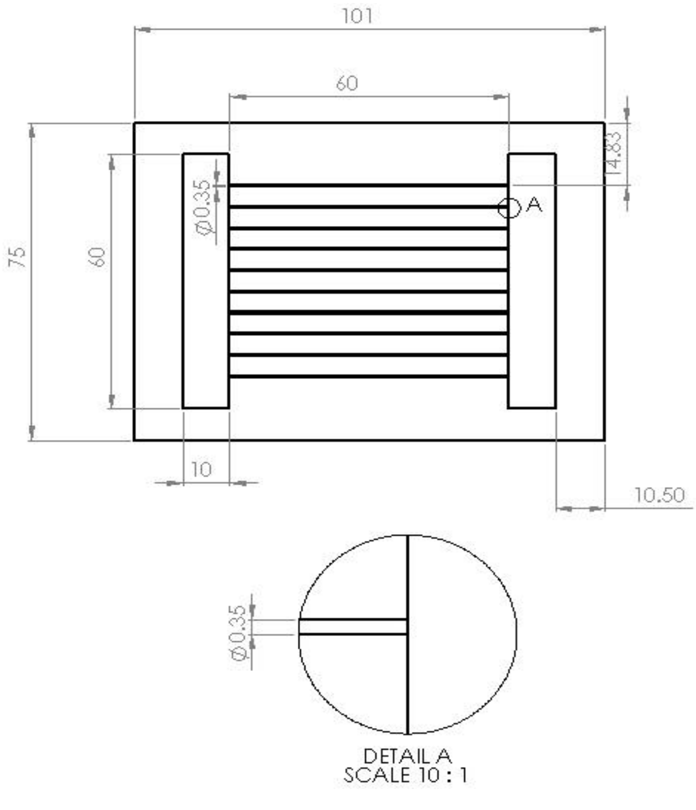
Experimental study conducted by Peng and Peterson[8] in their experimental study on water flows through rectangular microchannels(123µm$\leq Dh \leq$ 367µm) reported that Nusselt number was mainly depended on microchannel aspect ratio in both laminar and turbulent flow regions. They also found lower experimental values of Nusselt number than the values based on conventional theory for laminar regions. Warrier et al.[9] found good agreement between experimental and theoretical values of local Nusselt number for fully developed laminar flow, which also supported by the work of Qu and Mudawar[10], Owhaib and Palm[11],Wang et al.[12]. As reported by Mokrani et al.[13] macroscale theories can be applied to microchannels having hydraulic diameter greater than 100 µm. Harms et al.[14] have suggested that classical correlations can be used for single and multichannel microchannels and their experimental results have illustrated slight deviations in Nusselt number at low Reynolds number, and deviations have been attributed to measurement uncertainty. Rahman & Gui [15] performed experimental study for trapezoidal microchannels in laminar regime for water and reported friction factor values agrees well with conventional theory and Nusselt number obtained were higher than predicted by conventional correlations. Bucci et al.[16] experimentally measured the friction factor and Nusselt number for water in circular tubes. They found experimental friction factor values agreed well with conventional values , while Nusselt number values were found higher than the values predicted by conventional correlations. Peng et al.[8],[17] obtained higher experimental values of friction factor and lower values of Nusselt number for laminar water flow in rectangular microchannels when compared with conventional values. Gao et al.[18] obtained lower values of experimental nusselt number than predicted by conventional theory, while experimental friction factor values were found in agreement with conventional theory during experiments performed in rectangular microchannels for laminar water flow.

Based on the literature review [19], [20], [21] it is found that almost no experimental work has been performed on semi-circular cross section of microchannel heat sink. Also, in terms of manufacturing of semi-circular channels casting and extrusion are the most economical methods available for mass production of MCHS [22], compared to other MCHS design, semi-circular channel is the simplest design available. No such study on semi-circular channel is found to exist. Hence the authors take this opportunity to perform experimental analysis on semi-circular MCHS. Therefore, an attempt has been made to understand the heat transfer characteristics in semi-circular cross-sectional shape of microchannels.

METHODOLOGY

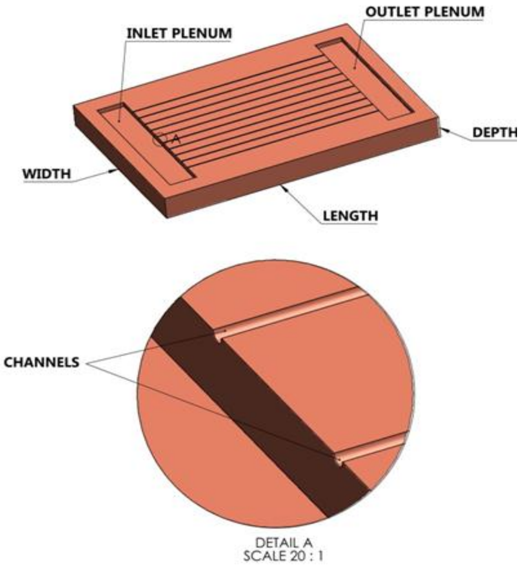
Experimental Test Section

The test sections is copper MCHS containing 10 number of parallel semi-circular shape microchannels having hydraulic diameter 214 µm and length 60 mm as shown in fig.1. It is machined from square block of copper (105mmx105mmx10mm), and semi-circular shape of microchannels were cut in the top surface of the specimen using Electronica Spring CNC wire cut machine. The measurements were accurate to ± 20 µm giving a mean uncertainty of 9.34 % for the hydraulic diameter which is 214 µm. The substrate thickness of test section is kept 8 mm. The test section is provided with inlet and outlet plenums (10x70x1.2mm) which act as buffer zone and help in getting uniform flow distribution to the channels. The cover plate made up of copper is bolted to the heat sink. One heater is attached to the bottom of the MCHS to provide heat flux. Water inlet and outlet temperature was measured by using four (PT-100) thermocouples while axial channel wall surface temperature was measured by two (PT-100) type thermocouples.



1 (a) *Detailed drawing of Microchannel*

1(b) *Experimental test specimen.*



1(c) *Full Domain geometrical model*

Figure 1. Details of test section

Experimental Facilities

The experimental facility utilized in the present investigation is shown in fig. 2. A flow pump is used to drive the deionised water from a water holding tank; the flow pump provides smooth and steady flow over a wide range of flow rates that corresponds to a Reynolds number ranging from 238 to 1250. In order to avoid blockage of the microchannels, a submicron filter of 0.1μm was installed between outlet of pump and inlet of test section. The test section is provided with a cartridge heater 150 watt through the blind temperature controller (BTC) which cut the electric supply of the heater when the temperature reaches at the desired point, for heat supply to microchannel. A wattmeter is used to measure the heat supply to the MCHS. A digital manometer model (RS-232), with an accuracy of ±0.3% of its full scale at 25°C) is used to measure the differential pressure between the inlet and outlet of the test section.

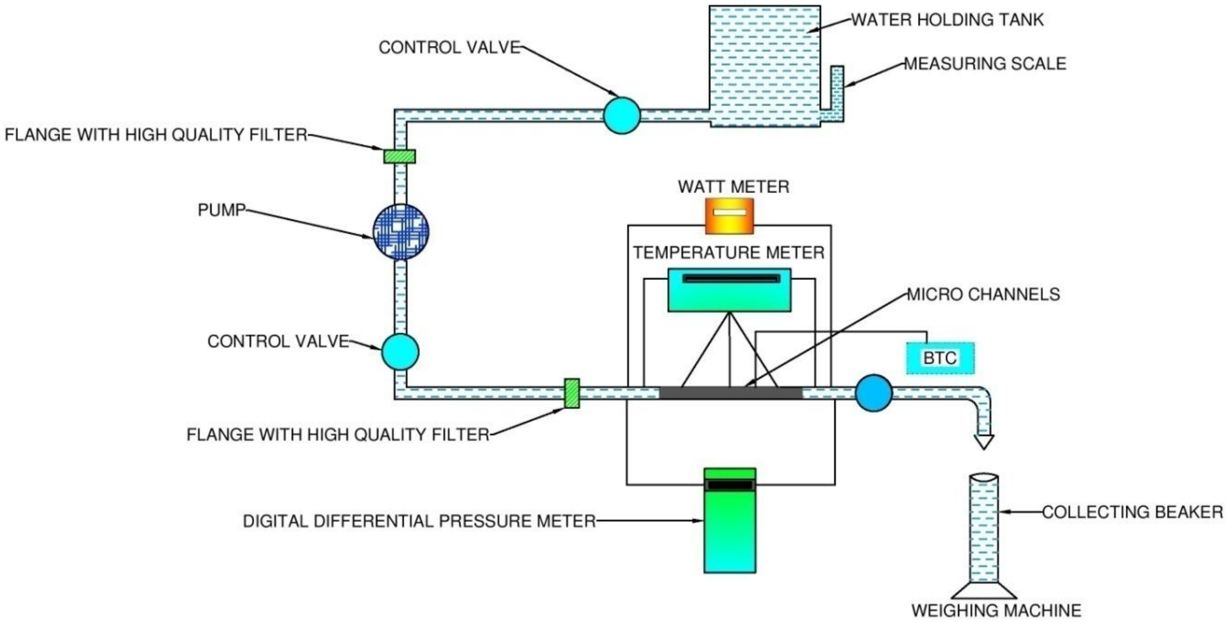


Figure 2. Schematic diagram of experimental setup

DATA REDUCTION

Flow Characteristics

In this Experimental study, Reynolds number is defined by

$$Re = \frac{\rho V_m D_h}{\mu} \tag{1}$$

Where ρ is density of the fluid flowing through microchannel and V_m is the mean velocity of the flowing fluid, μ is the viscosity of fluid and D_h is the hydraulic diameter defined by

$$D_h = \frac{4\, A_c}{P} \tag{2}$$

The Experimental friction factor based on the pressure drop measurements in microchannel flow are represented by Darcy friction factor which is given as

$$f_{exp} = \frac{2 \Delta p D_h}{\rho V_m^2\, L} \tag{3}$$

$$f_{exp-actual} = \frac{2 \Delta p_{actual}\, D_h}{\rho V_m^2\, L} \tag{4}$$

$$\Delta p_{actual} = \Delta p - K_{in}\, \rho \frac{V_m^2}{2} - K_{out}\, \rho \frac{V_m^2}{2} \tag{5}$$

Where K_{in} and K_{out} are the loss coefficients for the abrupt entrance and abrupt exit respectively [23],which are determined on the basis of the experimental values widely available for circular tubes .The values of K_{in} and K_{out} are recommended for circular tubes with abrupt entrance and abrupt exit as 0.5 and 1.0 respectively.

Where Δp, is the experimental pressure drop across the microchannel test section from inlet to the outlet which was measured during the experiment.

The experimental friction factor is compared with theoretical friction factor for fully developed laminar flow in circular pipes based on classical macroscale theory given as

$$f_{th} = C/Re \tag{6}$$

Where C = 64 constant depend on the cross-sectional shape of microchannel.

Heat Transfer

The overall performance of a Microchannel heatsink is calculated in terms of convective heat transfer along with the pressure drop. For the steady flow in the microchannels and in the absence of viscous dissipation effects, the actual heat transfer through convection from the inside surface of microchannels to the working fluid can be equated to the sensible heat carried away by the fluid

$$q = \dot{m} c_p (T_o - T_i) \tag{7}$$

The inlet and outlet temperature were taken from the thermocouples inserted into the inlet and outlet position of the test section. The heat transfer from heater is transferred to the MCHS walls, which is convected to the fluid flowing through micrchannels. With the assumption of constant wall heat flux boundary conditions for microchannel, the average heat transfer coefficient (h) for single microchannel can be written in terms of

$$h = q / n\, A_{sur} (T_{w,m} - T_{f,m}) \tag{8}$$

Measured heat transfer coefficients were converted to dimensionless Nusselt numbers based on the average hydraulic diameter of the microchannels.

$$Nu = \frac{h\, D_h}{K_f} \tag{9}$$

NUERICAL ANALYSIS

A numerical model with unit cell is developed as shown in Fig.3-5 So as to reduce computation effort. The unit cell consists of height, width and length of 8 mm, 5 mm and 60 mm respectively. Fluid flows through single semi-circular cross-sectional microchannel. Following assumptions are considered:

1. 3-D steady state laminar fluid flow and heat transfer.
2. Constant solid and variable thermo-physical properties of fluid.
3. The effect of gravity and other body forces are negligible.
4. Viscous dissipation is excluded.

The numerical model is based on Navier–Stokes and energy equations used to solve conjugate heat transfer process. Based on the above assumptions, following equations are used,

Continuity equation

$$\frac{\partial \rho_f}{\partial t} + \frac{\partial (\rho_f u_i)}{\partial x_i} = 0 \tag{10}$$

Momentum Equation

$$\rho_f \frac{\partial u_i}{\partial t} + \rho_f u_j \frac{\partial u_i}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \rho_f g_i + \frac{\partial}{\partial x_j}\left(\mu_f \frac{\partial u_i}{\partial x_j}\right) + \frac{\partial}{\partial x_j}\left(\mu_f \frac{\partial u_j}{\partial x_i}\right) - \frac{2}{3}\frac{\partial}{\partial x_i}\left(\mu_f \frac{\partial u_k}{\partial x_k}\right) \tag{11}$$

Energy Equation

For Fluid

$$\rho_f \frac{\partial h}{\partial t} + \rho_f u_j \frac{\partial h}{\partial x_i} = \frac{\partial P}{\partial t} + u_i \frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_i}\left(k_f \frac{\partial T_f}{\partial x_i}\right) + \left(\tau_{ij} \frac{\partial u_i}{\partial x_j}\right) \tag{12}$$

For Substrate Conduction

$$\frac{\partial}{\partial x_i}\left(k_s \frac{\partial T_s}{\partial x_i}\right) = 0 \tag{13}$$

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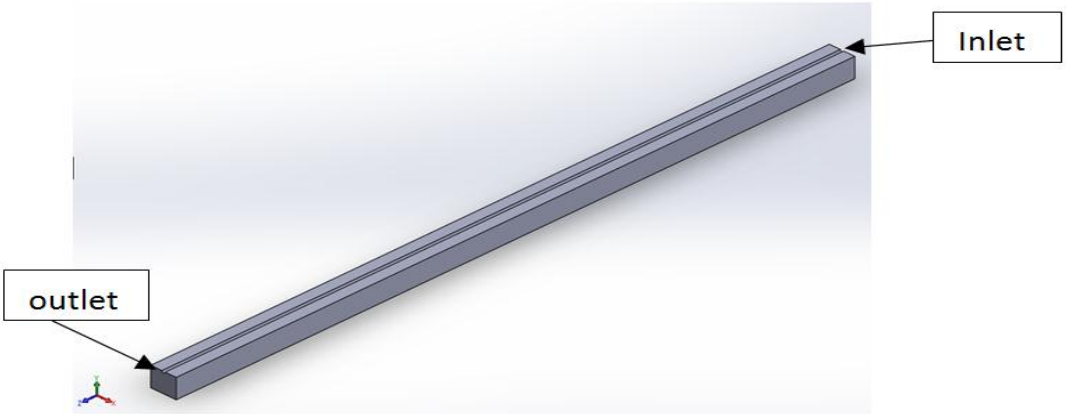

Figure 3. Single microchannel substrate for numerical simulation

The boundary conditions are

The left and right surfaces of the computational domain are assigned Symmetrical Boundary conditions.

At z = 0

$w = U_{in}$, u=v=0

$$k_s \frac{\partial T_s}{\partial z} = q$$

At z = L

$\frac{\partial w}{\partial x}$=u=v=0

$$k_s \frac{\partial T_s}{\partial z} = 0$$

At x = 0

u=v=w=0

At x = w

u=v=w=0

At y = 0

u=v=w=0

At y = h

u=v=w=0

The equations are solved using commercial CFD code Ansys CFX. The meshing is done using ICEM CFD software. Hexahedral mesh is obtained as shown in fig. 4 The code is validated from the experimental work of Tuckerman & Pease[2]. Grid independence test is conducted for different mesh sizes as shown in table.1 case is simulated with mesh 01, mesh 02, mesh 03 number of elements, the pressure drop, outlet average temperature and maximum outlet velocity found not to deviate within 2% and hence we use mesh 02 (280238).The solution is converged when maximum residual value were less than 10^{-6} for all the variables.

$$E\% = \left|\frac{F_2 - F_1}{F_1}\right| X100 \tag{14}$$

Where F is any parameter.

Solution Methods

Table 1

Heat 12500 W/m2 & Re = 200					
	Mesh 01	*E%*	Mesh 02	*E%*	Mesh 03
No. of	221284		280238		510462
ΔP (bar)	0.0694	*-1.9774*	0.0698	*-1.4124*	0.0708
Tout (K)	336.39	*0.4839*	336.04	*0.3794*	334.77
Vmax (m/s)	0.595711	*0.2730*	0.59361	*-0.0806*	0.594089

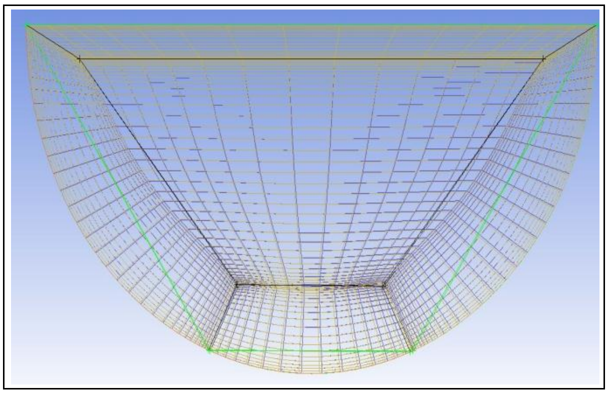

Figure 4. Fluid Mesh Generation in Semi-circle Cross section

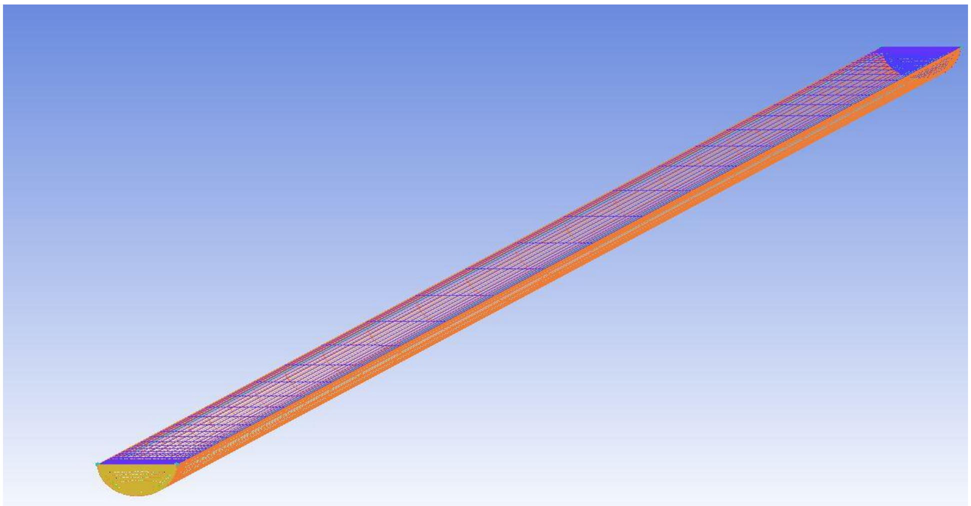

Figure 5. Complete Fluid Mesh generation in semi-circular channel

RESULTS AND DISCUSSIONS

Validation

The numerical model with single channel substrate is validated by the experimental work of Tuckerman & Pease [2]. The numerical model dimensions of solid substrate are in accordance with the actual dimensions of solid substrate used, in which length, width and height is 7.4 mm, 0.4 mm, 0.1 mm. The microchannel depth and width is 302 µm and 50 µm. In case of flow rate of 8.6 cm3/sec, the thermal resistance obtained is 0.09 K/W. In case of numerical simulation thermal resistance obtained is 0.07 K/W. The temperature rise in the numerical simulation is 52.5 °C, which is comparable to the experimental results obtained. The pumping power in both the cases is 0.019 W and 0.02 W. Summary of results shown in Table 2.

Table 2

	Experimental	Numerical Results
Temperature rise	50 °C	52.5 °C
Pumping Power	0.019	0.02
Thermal Resistance	0.09	0.07

Flow Characteristics

Fig.5 shows the total pressure drop and which is indicating the total measured pressure drop including the pressure losses at inlet and exit to the microchannel ,while pressure drop actual is indicating the actual pressure drop in microchannels by subtracting pressure losses at entry and exit. Pressure losses has been found ranging from (1-10)% of the measured pressure drop. It has been observed that for lower values of Reynolds number the entry and exit pressure losses are minimum and can be neglected, but for higher Reynolds number pressure losses are increasing at fast rate and should be taken into consideration.

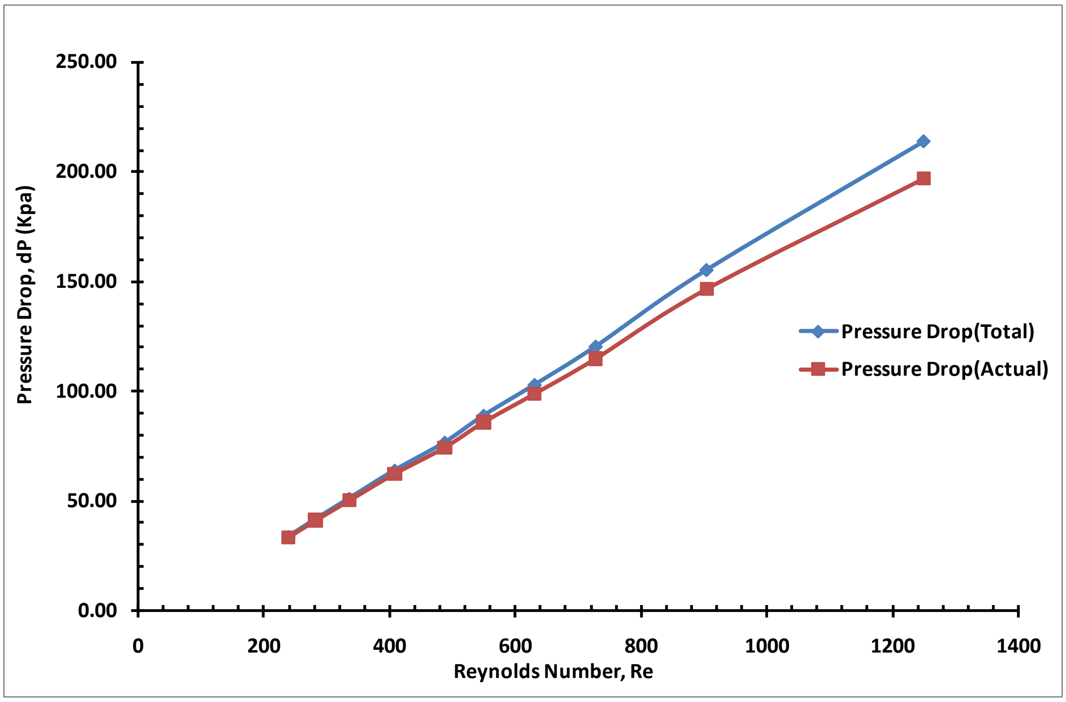

Figure 5. Pressure drop variation with Reynolds number

As shown in Fig.6 the values of experimental friction factors are compared with numerical simulated values, theoretical predicted values for fully developed laminar flow in circular microchannel, and Shah & London correlation [24]. It is observed that numerically predicted value of friction factor is slightly lower than experimental friction factor. This can be attributed to single channel configuration without inlet/outlet manifold is used in numerical simulation, while in experiment a multichannel configuration with entrance and exit bends is used, which is resulting higher experimental friction factor. Other reason of getting higher $f_{(Exp)}$ than $f_{(\mathrm{Num})}$may be non-achievement of uniform flow distribution among channels for multichannel flow [25]. Since a critical value of Re is not reached therefore it can be safely assumed that flow is in laminar regime. In our experimental investigation we have obtained x+ values (0.224-1.18)and x^* values in the range of(0.04-0.22), this indicates that the flow inside the microchannel is fully developed laminar flow and thermally developing [26] in the range of experiments conducted (238<Re<1250) inside the semi- circular microchannels.

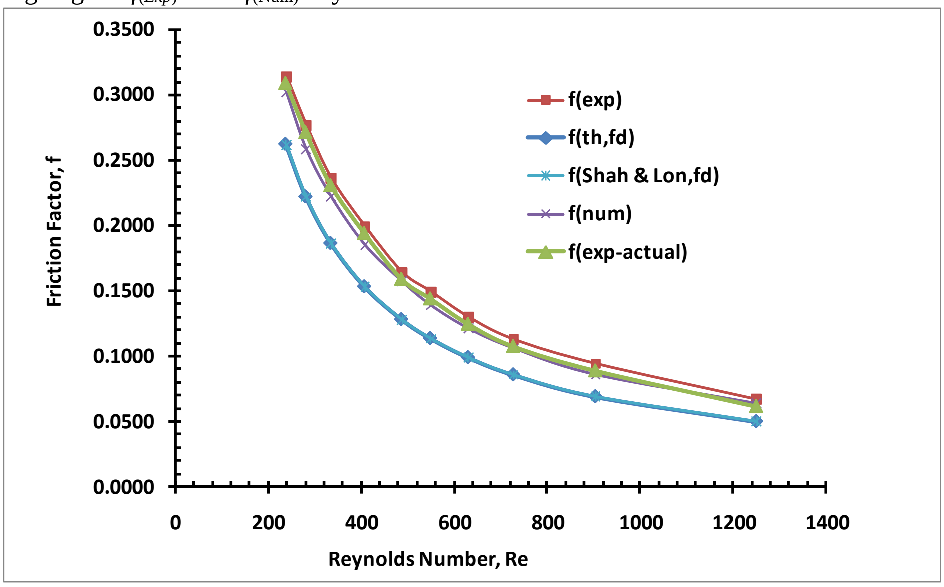

Figure 6. Friction factor variation with Reynolds number

Heat Transfer Characteristics

As shown in Fig.7 for lower Reynolds number the experimental values of Nusselt number under predict the values obtained by numerical simulation and theoretical correlations. This is because for Nu_{exp} it is assumed that the average temperature of interface is that of solid substrate as it is very difficult to measure experimental interface temperature and in our calculations, we have taken value of solid substrate, while it does not hold true for 3-D conjugate heat transfer model taken in Numerical analysis in which predicted value of average interface temperature is taken to calculate Nu_{num}. Also, in reality it is difficult to achieve an adiabatic boundary at inlet and outlet of the MCHS as assumed in the numerical model and a significant amount of heat is transferred to the inlet and outlet manifolds especially at low Reynolds number[27], [28]. Therefore, the numerically obtained Nusselt number is higher than the experimental Nusselt number at low value of Reynolds number. It is observed that when calculating heat transfer at low Re, a particular attention should be given to the effects of heat conduction through the wall as reported by Hetsroni et al.[29]

Slight discrepancy is found in Nusselt number at low Re ,this may be attributed to dominance of conjugate effects[30].When conjugate effects exists then conduction mechanism and convection mechanism produces counter effects and hence temperature distribution will not remain linear along the microchannel. Due to this non-linearity, the Nusselt number calculated by equation (9) is underestimated, because the mean value of the bulk temperature along the microchannel is underestimated. As reported by Li et al.[31] that during the heat transfer process in microtubes, due to high wall thickness -to-diameter ratio of microtubes axial conduction in the tube wall may take place and can be a possible cause for deviation between experimental and numerical heat transfer values. Also the general assumption of temperature distribution being one dimensional could be erroneous and hence deviations in observations is obtained as the temperature of actual microchannel is varying along longitudinal as well as transverse directions [32].

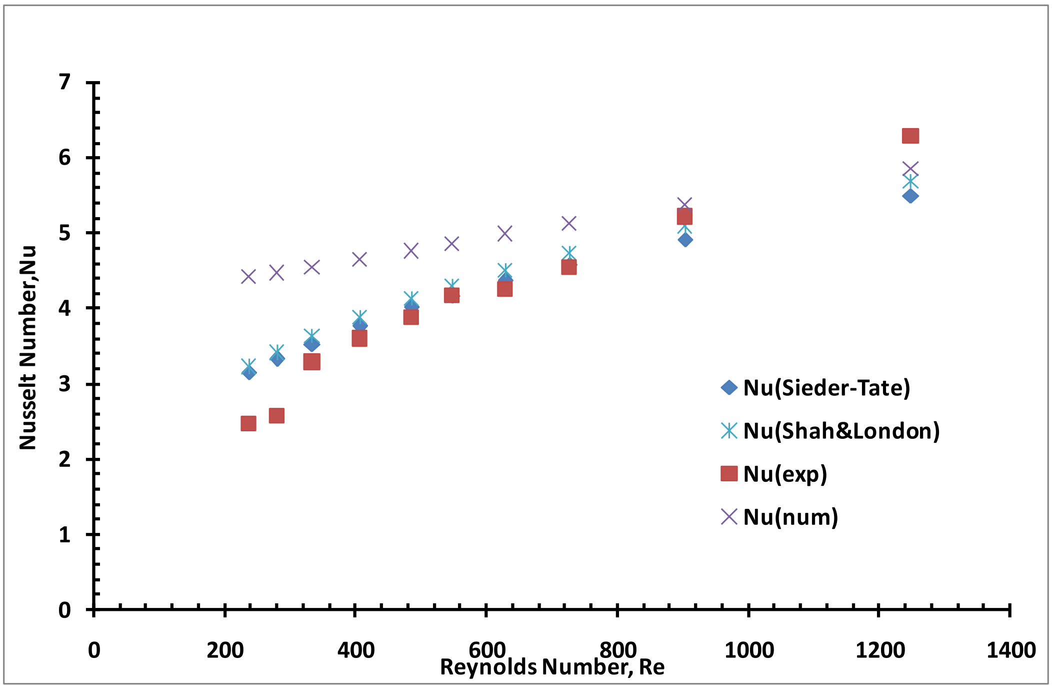

Figure 7. Nusselt number variation with Reynolds number

In order to have a clear understanding of the thermal performance in the microchannel region of the MCHS the graph is plotted between thermal resistance, pumping power with respect to Reynolds number as shown in Fig.8.

Eq. 15 and Eq. 16 shows thermal resistance and pumping power.

$$R_{th} = \frac{T_{max} - T_{min}}{q} \qquad (15)$$

$$\Omega = \Delta p.Q \qquad (16)$$

We have got a trade-off between thermal resistance and pumping power at Re=750. Since with increase in Reynolds number the thermal resistance decreases as the heat transfer coefficient increase, but with increase in frictional pressure drop the pumping power also increases. It is suggested that a low Reynolds number flow is preferable over high Reynolds number to optimize the tradeoff between thermal resistance and pumping power. It can be seen that there is a sharp increase in pumping power compared to lower decrease in thermal resistance after trade off point.

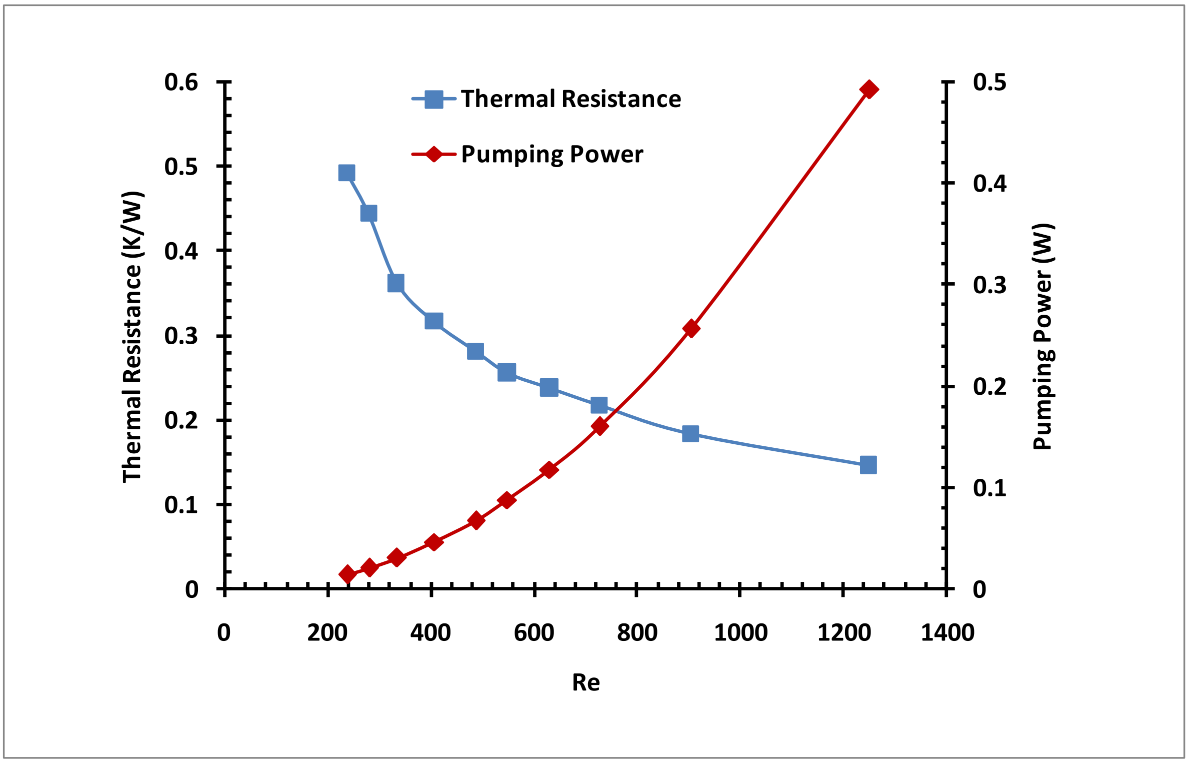

Figure 8. Thermal Résistance (K/W),Pumping Power(W) variation with Reynolds Number

CONCLUSIONS

Experimental investigations have been carried out for friction factors in single phase liquid flow inside semi- circular microchannels. The experimental results obtained showed good agreement with conventional hydraulic theory and it is also observed that the flow inside the microchannels was fully developed laminar flow in the range of experiments conducted (238<Re<1250).The flow in microchannels can be well predicted by applying conventional friction factor theory and the flow inside the microchannels is fully-developed.

However at lower Reynolds number ,the experimental values of Nusselt number differs slightly from the correlations developed by Shah & London[24] and Sieder-Tate[33]. It is found closer to the results predicted by correlation developed by Sieder-Tate, as they have chosen simultaneously developing flow.It confirms that in our case the flow remains simultaneously developing. A calibrated numerical model may be used to predict the experimental results in case of MCHS with thick substrate. In literature, the results predicted by numerical simulation for MCHS with very thin substrate agrees well with experimental results. It is suggested that further investigations need to carry out to develop new correlations suitable for semi-circular cross-sectional MCHS. It is suggested that a low Reynolds number flow is preferable over high Reynolds number to optimize the trade-off between thermal resistance and pumping power.

Nomenclature

Δp	Experimental pressure drop
$\dot{m}$	Mass flow rate ,Kg/Sec
V	Flow velocity
A	Area
c	Specific heat, J/Kg-K
D_h	Microchannel hydraulic diameter
f	Friction factor
h	Convective heat transfer coefficent

Subscripts

act	Actual values
c	Cross section
exp	Experimental value
f	fluid
i	Inlet
m	mean value
max	maximum
num	Numerical value

K	Thermal Conductivty
L	Length of microchannel
Nu	Nusselt Number
P	Perimeter
Q	Volumetric flow rate, m^3/sec
q	Heat transfer rate, W
Re	Reynolds Number
T	Temperature, K
x^*	Dimensionless Thermal entrance length
x+	Dimensionless Hydraulic entrance length
o	Outlet
p	at constant pressure
s	solid
sur	Surface
th	Theoretical value
w	wall

Greek Symbols

μ	Viscosity
ρ	Density of fluid
Ω	Pumping Power, W

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