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Experimental Analysis of Laminar Flow and Heat Transfer in a Multi-Port Finned Minichannel

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ABSTRACT

Due to their high heat transfer rate, small scale channels have been a popular area of study for the past three decades, especially for heat exchangers. In this study, fluid flow and heat transfer in a multi-port finned minichannel with a rectangular cross section was investigated experimentally under the constant heat flux boundary condition. The minichannel, which has a length of 638 mm, consists of 16 ports, 14 of which are identical finned rectangular channels with a width of 2.10 mm and a height of 5.85 mm while the remaining two ports at the outer edges of the channel were considered to be identical with the other ports. Deionized water was used as the working fluid with Reynolds number ranging between 75 and 190 in a single port. In order to correctly evaluate local heat transfer and friction coefficient values, thermal entrance effects and varying thermo-physical properties of the working fluid were taken into consideration throughout the study. Local Nusselt number varying with dimensionless axial thermal length, friction factor and average Nusselt number values varying with Reynolds number, and temperature distribution along the wall were evaluated to study fluid flow and heat transfer of the finned minichannel. The results were compared to theoretical values and presented graphically. Local Nusselt number values indicated fairly good agreement with theory while friction factor was overestimated. This was considered to be due to the effect of fins. In addition, three correlations were suggested in order to evaluate friction factor, local Nusselt number and average Nusselt number for the minichannel in given ranges of parameters.

Key Words:
Fin; Convective Heat Transfer; Laminar; Multi-Port; Nusselt Number

NOMECLATURE

A_c	channel port cross-section area, m^2	m	mass flow rate, kg/s
A_{ht}	channel heated area, m^2	Nu	Nusselt number
a	channel port width, m	P_w	wetter perimeter, m
b	channel port height, m	Pr	Prandtl number
C	constant	ΔP	total pressure difference, Pa
c	fin height, m	PEC	performance evaluation criteria
c_p	specific heat capacity, J/kg K	Q	total rate of heat gained by fluid, W
D	diameter, m	q	heat flux, W/m^2
d	fin width, m	R	indirectly measured parameter
e	port wall thickness	Re	Reynolds number
f	friction factor	T	Temperature, °C
G	volumetric flow rate, m^3/s	u	velocity, m/s
Gz	Graetz number	x	axial distance, m
L	length, m	x*	dimensionless thermal axial distance

y	directly measured parameter

Greek symbols

α	aspect ratio (b/a)
α_c	aspect ratio (a/b)
ε	relative roughness
μ	dynamic viscosity, Pa s
ρ	density, kg/m^3
ω	uncertainty

Subscripts

exp	experimental
i	inlet
f	fluid
h	hydraulic
m	mean
o	outlet
t	thermal
theo	theoretical
w	wall
x	local

INTRODUCTION

With the rapid development of MEMS (micro electro mechanical systems), high amounts of heat have to be dissipated from these systems. The initial studies began with Tuckerman and Pease [1], who tested their compact silicon substrate with an integral part and indicated that up to 790 W/cm^2 heat dissipation could be achieved. This encouraged researchers to investigate fluid flow and heat transfer characteristics in small scale channels. Consequently, those researches contributed to the area of heat exchangers. Even though there are numerous papers published about heat transfer in small scales, a few of them cover the gap of minichannels, which are neither micro- nor conventional channels, but in between [2,3].

Wang et al. [4] both experimentally and numerically investigated friction and heat transfer characteristics in a trapezoidal microchannel using deionized water with the constant heat flux boundary condition, neglecting axial heat conduction. They found good agreement between their experimental data and numerical predictions, showing that classical Navier-Stokes approach was valid in hydraulic diameters as low as 155 μm. Their numerical results also indicated that thermal entry length and fully developed Nusselt number for a trapezoidal cross sectional channel with bottom heating are $z = 0.15 \cdot Re \cdot Pr \cdot D_h$ and $Nu = 4.00$, respectively.

Gunnasegaran et al. [5] investigated the effects of geometric parameters on thermohydraulic performance of microchannel heat sinks with Reynolds number ranging between 100-1000. Their study mainly focused on triangular, trapezoidal, and rectangular cross sectional geometries. Results showed that between these three geometries, the rectangular channel showed the highest friction factor and heat transfer rate, while the triangular channel had the lowest, with the trapezoidal channel in between. They specifically pointed out that width to height ratio of rectangular channels has a major influence on Poiseuille number. Moreover, for the rectangular channel, the smaller the hydraulic diameter gets, the greater the value of the heat transfer coefficient is.

Wu and Cheng [6] experimentally investigated laminar convective heat transfer and pressure drop of water flow in trapezoidal microchannels with various surface conditions. They found out that the Nusselt number values and friction coefficient greatly depends on geometrical parameters of the channel which is considered more important than surface roughness and surface hydrophilic property. Experimental results indicated that at low Reynolds numbers ($Re < 100$), Nusselt number increases sharply and is practically linear with Reynolds number, while the rate of Nu to Re gradually decreases as Reynolds number passes the value of 100. Afterwards, a numerical study was conducted by Zhuo et al. [7] to show the applicability of the former study by using field synergy principle. It was found that Navier-Stokes and energy equations with no-slip boundary condition can be used to predict thermohydraulic performances of fluid flow for hydraulic diameter as low as tens of micrometers with reasonable accuracy. They also indicated for the two microchannels they analyzed, fully developed Nusselt number increases with the increase of Reynolds number differing from conventional theory where fully developed Nusselt number is constant.

Hetsroni et al. [8,9] compared experimental data available in literature with theory and numerical results to investigate fluid flow and heat transfer characteristics in microchannel. In their fluid flow study, it was concluded that Poiseuille number is independent of Reynolds number for single phase fluid flow in smooth microchannels with hydraulic diameter varying from 15 to 4010 μm in laminar region. Additionally, flow behavior in microchannels with hydraulic diameters as small as 50 μm showed no difference from macroflow. Heat transfer studies indicated that axial conduction in both fluid and the wall has a significant effect on heat transfer in microchannels. Thus, it was suggested to use the exact model in numerical applications in order to evaluate correct variation of Nusselt number.

Lee et al. [10] conducted an experimental investigation to find out the validity of classical correlations used in conventional sized channels to evaluate thermal behavior of rectangular microchannels. Deionized water was used as working fluid. Reynolds number and hydraulic diameter varied between 300-3500 and 318-903 μm. Numerical analysis was also carried out to verify their work. Numerical results showed good agreement with experimental data (average deviation of 5%), suggesting that classical continuum approach can be used to predict heat transfer behavior of microchannels with dimension in given range. The results also confirmed that instead of full 3D conjugate analysis, a simplified this wall analysis can be employed as an alternative to decrease computational time.

Also, several reviews were conducted by various researchers [11,12] about flow and heat transfer characteristics in small scale channels. Their research consist of vast amount of experimental and numerical data. They conclude that, although most of liquid flow investigations agree well with conventional theory, there are still numerous amount of studies that deviate significantly, either over or under predict, especially on friction coefficient. They predict that, due to improvement on manufacturing and measuring technology, therefore reduction on surface roughness and having control over hydraulic diameter, older studies might not provide accurate comparison as thermohydraulic properties of small scale channels greatly depend on geometry. Another review was done by Rosa et al. [13] which is focused on scaling effects in small scale channels. They recommended that, although classical theories are reliable to predict friction factor and heat transfer characteristics, scaling effects should be accounted for accurate results.

Considering the abovementioned literature review, more research should be done in small scale channels to further prove their applicability to the theory. Additionally, while most of the authors focused on simple geometries to clarify the approaches, a few of them considered the effects of fins. Therefore, an experimental investigation of heat transfer and fluid flow characteristics in a multi-port finned minichannel with rectangular cross section was conducted under various constant heat flux rates. Reynolds number ranged from 75 to 190 in a single port. In order to correctly evaluate local heat transfer and friction coefficient values, the effect of thermal entrance and varying thermo-physical properties of the working fluid were taken into consideration. Specifically, a finned multiport rectangular channel was chosen due to their applicability in industry, therefore, experimental analysis of the channel would provide assistance for heat exchanger design.

EXPERIMENTAL SETUP

Test Facility

A schematic illustration of the test facility, which is composed of a cooling bath (Cole Parmer®, KH-13500-30), a by-pass line, a control valve, a gear pump (Cole Parmer®, KH-74014-55), a flowmeter (Flowtech®, DK800S-6), thermocouples (Cole Parmer®, KH-08542-04, T type), resistance temperature detectors (Cole Parmer®, KH-08117-80), a differential pressure transmitter (Validyne®, 1-N-1-28-S-4-A) and the test section, is given in Fig. 1a.

The working fluid is stored and cooled in constant temperature cooling bath. Cooling bath consists of its own pump, which extends the maximum flowmeter limit. Thus, a by-pass line was implemented to the circuit from the outlet to the inlet of the bath in order to reduce the flow rate. Joint use of control valve and gear pump is used to stabilize fluid pressure in addition to adjusting the flow rate to the desired value by manually increasing or decreasing the flow area and rpm, respectively. Once the fluid passes the flowmeter, where adjusted flow rate is visually measured, it reaches test section.

Test Section

The test section consist of heating layer and test channel.

Heating Layer

The heating layer can be seen in Fig. 1c. The test section is designed to apply constant heat flux on the minichannel. In order to achieve this condition, a thin bronze bar was wrapped with dielectric tape and resistance wire was stationed on it. Thus, electric current was ensured not to flow through bronze bar. Figure 1b. shows the thermocouple locations along the channel. They are 70 mm apart from one another except the first one at the inlet, which is 25 mm apart from the channel edge. As they were likely to get affected by the temperature of bronze bar and give inaccurate results, a thin mica layer was placed between the bronze bar and the minichannel. This whole process posed small gaps between layers, hence increased thermal resistance. In order to prevent this problem, a wooden stock was put on top of the heating section, then compressed with clamps and the whole system was isolated. Table 1 lists the material properties in the heating layer.

Table 1. Material properties in heating layer

Materials	Thickness[mm]	k[W/m K]
Glass Wool	30	0.031
Wooden Stock	15	0.13
Resistance Wire	0.5	[-]
Bronze Bar	1	401
Mica	0.5	0.3
Minichannel	0.44	237

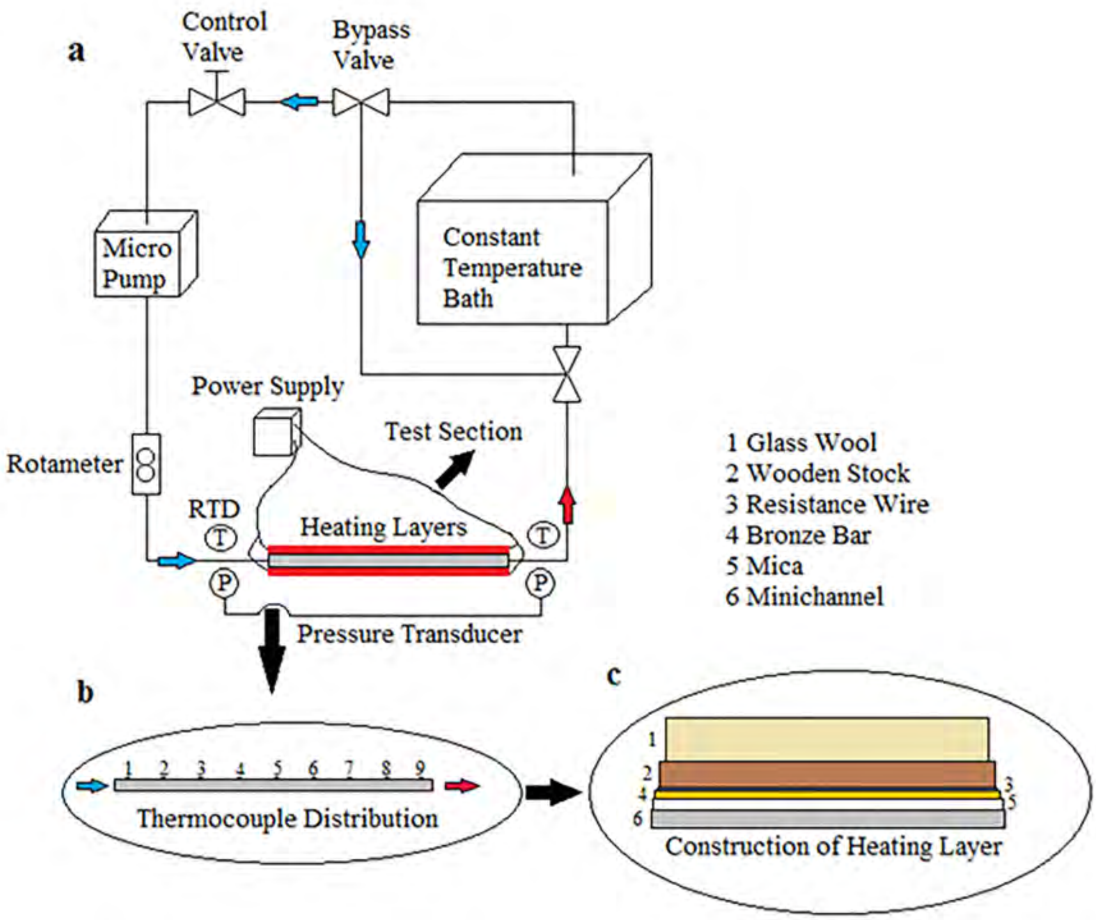

Figure 1. (a) Schematic representation of test facility (b) Thermocouple distribution along test channel wall (c) Heating layer materials and construction

Test Channel

A visual representation of the minichannel cross section is given in Fig. 2. The minichannel consists of 16 ports, 14 of which are identical to each other. The remaining 2 ports at the outer edges are considered to be similar to the others for convenience. Geometrical specifications of the channel are provided in Table 2.

Table 2. Geometrical specifications of the channel.

Parameters	*Definitions*	*Values*
L [mm]	*Length of the channel*	*638*
W [mm]	*Width of the channel*	*38.05*
H [mm]	*Height of the channel*	*6.73*
a [mm]	*Width of the port*	*2.10*
b [mm]	*Height of the port*	*5.85*
c [mm]	*Height of the fin*	*0.87*
d [mm]	*Width of the fin*	*0.25*
e [mm]	*Channel port thickness*	*0.25*

Manifold design is stated as an important problem considering the working fluid is desired to flow in each port with equal velocity [14]. Therefore, manifolds are designed by means of FLUENT©. Average axial velocity at the ports on the edge of microchannel inlet have relative error of 13%.

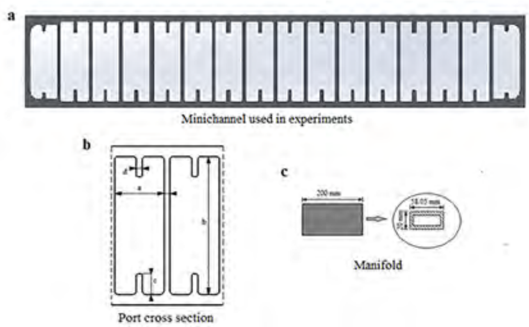

Figure 2. (a) Minichannel (b) Cross section of the port (c) Manifold

The reason behind this is that, those ports are not identical with inner ports, although they are assumed identical throughout the study. Remaining ports have average relative error of 3%, which is practical. In each manifold, 2 holes are drilled to implement differential pressure transducer and resistance temperature detectors. By doing so, pressure difference between inlet and outlet manifolds, inlet and outlet temperature of the fluid was measured.

Experimental Method

Before starting the experiments, data logger and power supply are turned on. The control valve is opened fully to allow the working fluid to flow through the test section. After preliminary preparations, constant temperature bath and micro pump are turned on. The fluid maintained in the constant temperature bath is cooled to the desired temperature by manually adjusting the bath setting. After fluid temperature stabilizes, control valve is gradually closed and flowmeter is constantly checked in order to ensure that desired flow rate value is approximately achieved. Afterwards, the micro pump is used to adjust the flow rate to the exact desired value by increasing or decreasing the flow rate by small amounts. Then, current and voltage values are adjusted from the power supply to initiate heating at the test section. Temperature and pressure readings are started visualized in the computer and recorded. In approximately 20 minutes, steady state is achieved. The temperature and pressure data are digitally recorded through the data acquisition unit every 10 seconds for 15 minutes. For each repeated experiment, above mentioned procedure is followed.

DATA ANALYSIS

Data Reduction

Calculations were performed using conventional equations [15,16]. Mass flow rate is calculated through the measured volume flow rate and water density.

$$m = G \cdot \rho \tag{1}$$

Average Reynolds number in one port is defined as below.

$$\mathrm{Re} = \frac{1}{16}\frac{m \cdot D_h}{A_c \cdot \mu} \tag{2}$$

Where A_c is the cross sectional area of a channel port. Hydraulic diameter and aspect ratio is defined, respectively, as follows.

$$D_h = \frac{4 \cdot A_c}{P_w} \tag{3}$$

$$\alpha = \frac{b}{a},\ \alpha_c = \frac{1}{\alpha},\ 0 < \alpha_c < 1 \tag{4}$$

Hydrodynamic development length, thermal development length and dimensionless axial thermal development length for rectangular cross sectional channels in laminar region were calculated, respectively, as:

$$L_h = 0.05 \cdot D_h \cdot \mathrm{Re} \tag{5}$$

$$L_t = 0.1 \cdot D_h \cdot \mathrm{Re} \cdot \mathrm{Pr} \tag{6}$$

$$x^* = \frac{x}{\mathrm{Re} \cdot \mathrm{Pr} \cdot D_h} \tag{7}$$

Friction factor was calculated including entrance and exit loses.

$$f = \frac{1}{2}\frac{D_h}{L}\frac{\Delta P}{\rho \cdot u_m^2} \tag{8}$$

ΔP is the pressure difference measured by the differential pressure transmitter. Heat transferred to the fluid, which is total heat transferred throughout the test section, is given as:

$$Q = m \cdot c_p \cdot (T_{f,o} - T_{f,i}) \tag{9}$$

$T_{f,i}$ and $T_{f,o}$ are the inlet and outlet temperature of the working fluid, which are measured by resistance temperature detectors. The heat flux is calculated by dividing by the channel heating area, A_{ht}:

$$q = \frac{Q}{A_{ht}} \tag{10}$$

Longitudinal temperature of the fluid in the test section is assumed to be varying linearly and calculated as follows:

$$T_f(x) = T_{f,i} + (T_{f,o} - T_{f,i}) \cdot \frac{x}{L} \tag{11}$$

The local heat transfer coefficient is defined as:

$$h(x) = \frac{q}{\left[T_w(x) - T_f(x)\right]} \tag{12}$$

Where, $T_w(x)$ are the temperature measurements along the wall via thermocouples. The local Nusselt number is defined as:

$$\mathrm{Nu}(x)=\frac{h(x)\cdot D_h}{k_f(x)} \tag{13}$$

Average Nusselt number is calculated by taking the mean value of nine local Nusselt number values for each experiment.

$$Nu_m=\frac{1}{9}\sum Nu(x) \tag{14}$$

An add-in, which is generated using industrial standard IAPWS-IF97, is used to calculate thermophysical properties of the fluid at required temperatures. In order to validate the accuracy of the program, $k_f(x)$ values were compared with tabular values [17] and maximum relative difference was found to be %.0.87 [18].

Uncertainty

In this study, uncertainty analysis is done according to reference [19].

$$R=R(y_1,y_2,y_3,\dots,y_n)$$

$$\omega_R=\left[\left(\frac{\partial R}{\partial y_1}\omega_1\right)^2+\left(\frac{\partial R}{\partial y_2}\omega_2\right)^2+\dots+\left(\frac{\partial R}{\partial y_n}\omega_n\right)^2\right]^{1/2} \tag{15}$$

Where, ω_R is the uncertainty in the result, $y_1, y_2, \dots, y_n$ are independent variables, and $\omega_1, \omega_2, \dots, \omega_n$ are uncertainties in the independent variable. Operating conditions of the system and uncertainty of major parameters are provided in Table 3.

As it can be seen in Table 3, uncertainties related to hydraulic performance of the flow are higher than usual. Geometrical parameters are the major cause of this phenomenon. Since the channel is miniscale, conventioanal measurements techniques are used to evaluate the parameters. However, increasing accuracy of measurement, for instance by using electron microscopy, will provide lower uncertainties, thus more accurate results.

RESULTS AND DISCUSSIONS

In this study, the flow is known to be in laminar region, since Reynolds number varied between 75 and 190 in a single port. In the graphically presented data, there were no significant slope changes in experimental data for friction factor and average Nusselt number before Re = 190. Therefore, only laminar flow evaluations are done for both friction and heat transfer characteristics.

In order to experimentally determine viscous heating, at first, inlet temperature of the test section was maintained at 20°C without heating. It was observed that the temperature difference between the inlet and the outlet of the test section was less than 0.1°C, which indicated that viscous heating could be neglected throughout the study. Pressure drop experiments were conducted without heating as well since heat flux has no significant effect on friction factor in laminar region [20]. Experiments were repeated four times and the maximum relative error of repeated data and heat loss to exterior were less than 8% and %10, respectively. Overall heat loss reduced as Reynolds number increased.

Friction Characteristics

For fully developed laminar friction factor, Shah and London [21], Leon and Roman [22], Spiga and Morini [23] provided analytically solved equations for smooth rectangular ducts. Shah equation is given as:

$$f\,\mathrm{Re}=26(1-1.3553\alpha_c+1.9467\alpha_c^2-1.7012\alpha_c^3+0.9564\alpha_c^4-0.2537\alpha_c^5) \tag{16}$$

Hydrodynamic development length is calculated using Eq. (5) and maximum development was found out as 25 mm, which corresponds to 3.9% of the channel. Therefore, hydrodynamic development effect inside the channel was ignored and results were presented with respect to fully developed friction factor throughout the study.

Figure 3 shows the plot of friction factor as a function of Reynolds number along with the results of Shah and London[21] for comparison. It can be seen that experimental data are found to be following the same trend with work of Shah and London [21], however higher. It is a well-known fact that roughness affects friction factor significantly and it has been researched widely. A recent numerical study of Pelevic and Meer [24] shows that an increase in relative roughness by 2.93 % increases friction factor by 7 %. Correspondingly, the higher friction factor values are considered to be an accumulation of the presence of the

Table 3. Operating parameters and uncertainties

Parameters	*Values*	*Uncertainties*
G [lt/h]	20 - 48	2.50 %
Q [W]	170 - 321	0.2-0.3 %
T_w[K]	295 - 311	± 0.5
T_f[K]	294 - 306	± 0.1
D_h[mm]	2.52	10 %
h_x[W/m²K]	992-3242	8.4 - 13.1 %
Re	75 - 190	14 - 15.5 %
Nu	5.25 - 7.87	11.2 - 14.6 %
f	0.171 - 0.453	16.3 - 22.3 %

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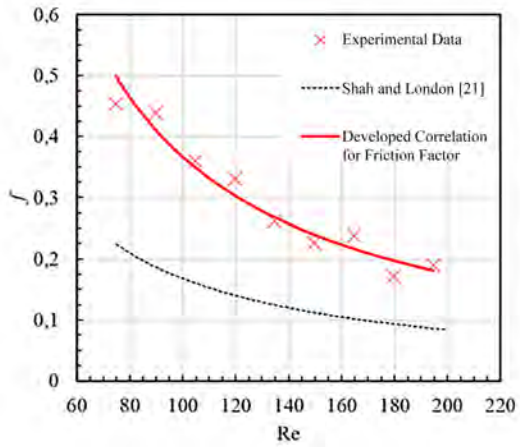

Figure 3. Friction factor versus Reynolds number

fins and possible roughness effect in this study. The aspect ratio was calculated ignoring the fin geometry and relative roughness is unknown for the channel.

While developing a correlation, as friction coefficient is known to be a function of Reynolds number, Eq. (17). is used. It is generalized formulation of laminar flow in ducts, given as:

$$f = \frac{C}{\mathrm{Re}} \tag{17}$$

Where C is constant for different geometrical shapes. For example, C is 16 for smooth circular tubes while its value depends on aspect ratio for rectangular channels [25]. In this study, C was found as 37 and Eq. (18) is suggested to be utilized in the design of heat exchangers in the experimental range studied here.

$$f = \frac{37}{\mathrm{Re}} \tag{18}$$

Heat Transfer Characteristics

In order to properly comprehend the effects of heat flux, two different heat flux rates, which are 3685 W/m^2K and 5560 W/m^2K, are used while conducting experiments. Figure 4 shows the effect of different heat flux rates on temperature along the wall. It can be seen from the figure that increasing heat flux increases the wall temperature and wall temperature difference between outlet and inlet of the channel. As Reynolds number increase, temperature difference between outlet and inlet of the channel reduces. This indicates that, in cooling applications, higher Reynolds numbers should be considered in laminar region for cooling applications. Theoretical Nusselt number for thermally developing flow is given for smooth rectangular cross sectional ducts with aspect ratios of 0.33 and 0.5, respectively, as [15]:

$$\mathrm{Nu}_x = \frac{31.297 + 14867x^* + 622440x^{*2}}{1 + 2131.3x^* + 144550x^{*2} - 13297x^{*3}} \tag{19}$$

$$\mathrm{Nu}_x = \frac{28.315 + 27038x^* + 1783300x^{*2}}{1 + 3049x^* + 472520x^{*2} - 35714x^{*3}} \tag{20}$$

As the current test channel has an aspect ratio of 0.359, a linear interpolation between Eq. (19) and (20) is done and the resulting equation is also provided in Fig. 5 along with the local Nusselt values of the experimental data varying with dimensionless thermal development length.

It can be seen in Fig. 5 that local Nusselt values show fairly good agreement and same trend with theory. However, before the flow converges into fully developed Nusselt value ($x^* < 0.1$), a distinct alteration occurs. While laminar developing theory shows a sharp change, experimental results are smoother. This is thought to be originating from the presence of the fins. Flow disturbances caused by fins further delays the formation of thermal boundary layer and increases heat transfer coefficient, correspondingly Nusselt

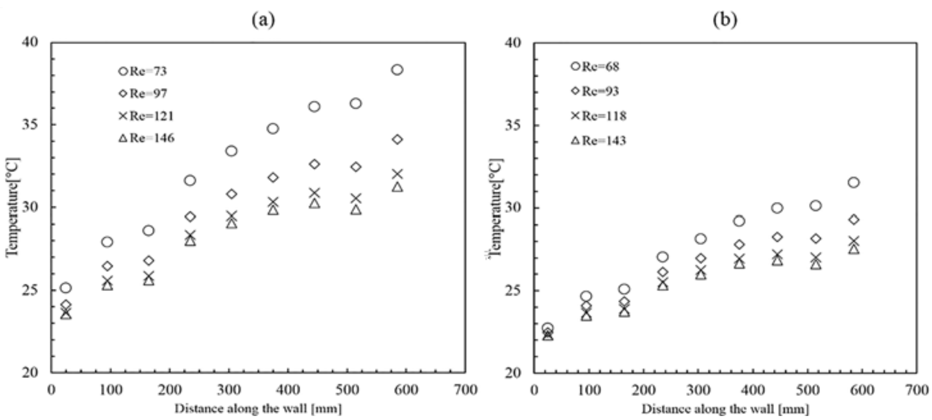

Figure 4. Temperature difference along the channel wall for heat fluxes (a) 5560 W/m^2K (b) 3685 W/m^2K

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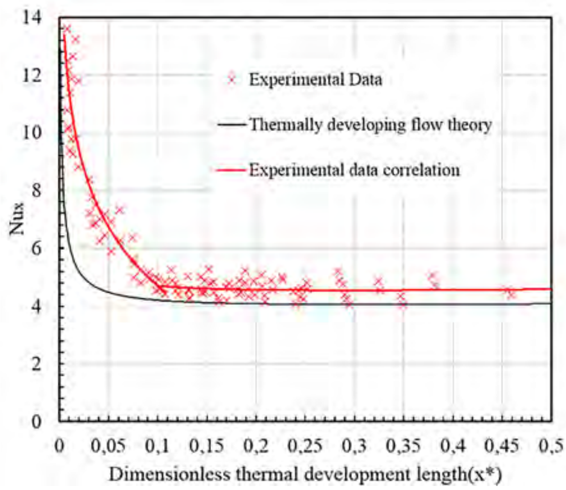


Figure 5. Local Nusselt number vs dimensionless thermal development length.

number. Therefore, in short channels where thermal development plays an important role, fin usage might be considered while designing small scale heat exchangers.

In order to determine local Nusselt values for the current channel, a new correlation is proposed considering the threshold is x* = 0.1, as:

$$\mathrm{Nu}_x = \begin{cases} \mathrm{Nu}_x = -2.898\ln(x^*) - 1.9606 & x^* < 0.1 \\ \mathrm{Nu}_x = 4.71 & x^* > 0.1 \end{cases} \quad (21)$$

Figure 6 shows the average Nusselt number variation with Reynolds number with two Prandtl numbers (Pr). Thermal development length theory is plotted alongside with experimental results. It can be seen that both Reynolds number and Prandtl number enhance average Nusselt number as they increase. Study of Zhang et al [26] indicates the thermohydraulic performance of different multiport microchannels with sawtooth fin structures. According to their conclusion, finned structures increase heat transfer rate as well as friction factor and fin geometry has a great impact on both characteristics. Experimental data of this study slightly overestimated thermal development theory. Bearing in mind the work of Zhang et al. and the results of the current study, the differences between experimental data and the theory can be due to the fins. In addition, as fluid temperature is assumed to be varying linearly along the channel, while it practically varies differently, this may be another reason of overestimation. It can also be concluded according to heat transfer results that, conventional theory is better used to predict heat transfer phenomenon in minichannels with hydraulic diameter as small as 2.519 mm.

General correlation form for thermally developing flow to predict Nusselt number in laminar flow is given as a function of Graetz number, which is defined as:

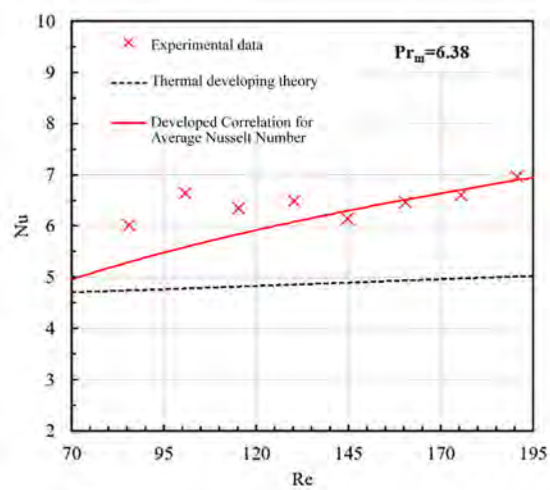
$$Gz = \frac{D_h}{L}\,\mathrm{Re}\,\mathrm{Pr} \quad (22)$$

Therefore, a new correlation is proposed to predict average Nusselt number in the given range of parameters, as:

$$\mathrm{Nu}_m = 4.1\left(\mathrm{Re}\,\mathrm{Pr}\frac{D_h}{L}\right)^{0.33}\left(\frac{\mu_f}{\mu_w}\right)^{0.14} \quad (23)$$

Performance Analysis

Experimental results of the fluid flow and heat transfer indicated the enhancement of heat transfer together with pressure drop. Therefore, in order to truly evaluate the usability of the channel, a performance evaluation criteria is defined as [27]:



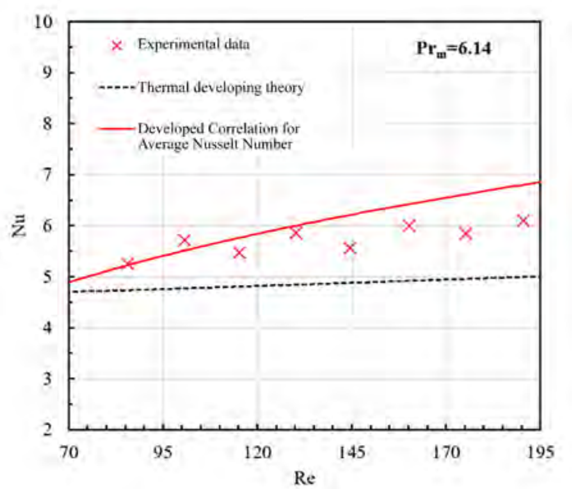


Figure 6. Average Nusselt number results in comparison with thermal development theory for (a) Pr = 6.38 (b) Pr = 6.14

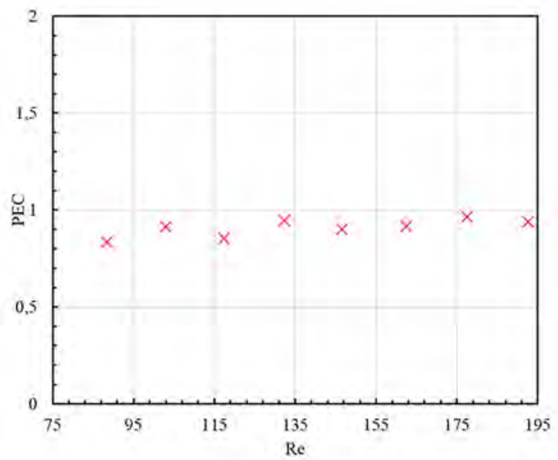


Figure 7. Performance evaluation criteria for the minichannel

$$PEC=\frac{Nu_{exp} / Nu_{theo}}{(f_{exp} / f_{theo})^{1/3}} \quad (24)$$

Since proposed correlations show the predictable values of thermohydraulic performance of the flow, in order to better visualize PEC values, those correlations were divided by theoretical values and used instead. Figure 7 shows the PEC values of the channel. As can be seen from the figure, PEC values are smaller than 1, which indicates that roughness and fin presence affect pressure drop more than they affect heat transfer rate.

CONCLUSION

Experimental investigation of laminar heat transfer and fluid flow analysis in a rectangular cross sectional multiport finned minichannel was conducted with constant heat flux boundary condition using deionized water. Reynolds number ranged between 70 and 190. Varying thermophysical properties of the working fluid and entrance effects were taken into consideration. Friction factor and convective heat transfer Nusselt numbers were calculated and compared with conventional and small scale correlations in literature and theory. Both friction factor and convective heat transfer Nusselt numbers showed higher values than theory. Three new correlations were developed to estimate friction factor, local Nusselt number and average Nusselt number in the given range of parameters. Performance evaluation criteria (PEC) analysis was conducted to assess the usability of the channel. Further studies should be done with similar channels without fins and with different fin geometries in order to fully comprehend the fin effect on minichannel.

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