

EXPERIMENTAL RESEARCH AND CFD SIMULATION OF CROSS FLOW MICROCHANNEL HEAT EXCHANGER

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ABSTRACT

In this study, a cross flow microchannel heat exchanger has been manufactured out of standard sizes using aluminum material. The plate dimensions of heat exchangers have been 50x50x3 (mm³) that composed of two plates in cross flow arrangement. All evaluated geometries have been consisted of square microchannels with 490 µm width and 490 µm depth. An appropriate experimental facility has been established to perform the fluid flow and heat experiments. Moreover, heat transfer and fluid flow characteristics in microchannels have been simulated by ANSYS Fluent V15 Computer Program and experimental results have been compared with Computational Fluid Dynamics (CFD) results. Results showed that experimental heat transfer data was a very good agreement between data obtained by CFD simulation. However, the numeric pressure drop values have not been compatible with experimental ones.

Keywords: *Microchannel, Cross Flow Microchannel Heat Exchanger, CFD*

INTRODUCTION

Nowadays, the use of mini and micro channel heat exchangers has got a wide application area from electronic devices to cooled reactors. In our country, many researches have conducted on simulations of microchannels with different dimensions and boundary conditions could be seen in [1]. Mini scale heat exchangers have been widely used for many applications including cooling of electronic equipment, fuel cell, air conditioning, heat pumps, gas turbines and etc [2-4]. While many industrial organization have capacity to invest on mini-and micro-scale systems, they have imported this technological products and this mini-micro systems aren't became popular in the consumer market.

The experimental and numerical studies in microchannel flow show the importance of surface roughness, entrance effects, shear flow, viscous dissipation and axial heat conduction parameters. Therefore, the micro device geometry to determine the micro channels in the fluid flow to ensure that the mass and heat transfer analysis, which enables development of new methods. Also, verification existing models by experimental work is very important. Generally hydraulic diameter of up to 500 µm in the literature, the heat exchangers are called the micro heat exchanger [5]. Kang et al. [6] have examined a microchannel cross flow heat exchanger made of silicon. They have analyzed interaction between the effectiveness and the pressure drop in the microchannels of heat exchanger. Alm et al. [7] have fabricated and tested counter and cross flow micro heat exchangers made of alumina. Water used as a working fluid and the maximum flow rate has been 120 kg/h. They have stated that the efficiency values are in the range of 0.10-0.22. The heat transfer coefficients are reached to 22 kW / m².K. Experimental heat transfer and pressure drop results have been reported to be higher than theoretical results. Hasan et al. [8] have analyzed the effect of shape and hydraulic diameter of channels on the performance of microchannel heat exchanger numerically. It has been found that the increase in the number of heat exchanger channels leads to an increase in both efficiency and pressure drop. In addition, circular geometry has provided better overall thermal and hydraulic performance. Dang et al. [9] have performed experimental and numerical analysis of single-phase heat transfer and fluid flow behavior in a microchannel heat exchanger. Effects of mass flow rate, heat flux and gravity on heat and fluid flow have been examined. Results have shown that numerical predictions are in good agreement with experimental values. Mathew and Hegab [10] have presented a thermal model for a counter-flow microchannel heat exchanger exposed to thermal interaction with its surroundings. The thermal model in numerical analysis have consisted of two fluid equations including the temperature change in the axial direction. It has been reported that experimental and theoretical data are compatible with each other

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for both conditions of external heat transfer. Hasan et al. [11] have fabricated counter and cross flow micro heat exchangers made of alumina and tested with water as working fluid. Also, numerical calculations have been made with FORTRAN code to investigate the axial heat conduction effects. They have reported that wall thickness, thermal conductivity, channel volume plays an important role on heat transfer. Dixit and Ghosh [12] have investigated the effect of heat in-leak in a two-stream crossflow heat exchanger. Crossflow mini-channel heat exchanger made from stainless steel have been tested in order to validate the numerical model. Results have shown that numerical results for the thermal interaction between the heat exchanger and the environment coincide with the experimental results. Zhou et al. [13] have performed numerical and experimental analysis to investigate effects of microchannel height, width and space on performance of heat exchanger. Parallel flow microchannel heat exchanger has been fabricated with milling process on copper plate and tested with water. They have found that average Nusselt number increased with increased width and decreased height of channel. In addition, the friction factor has increased with decreased height and width. Ling et al. [14] designed three kinds of microchannel configurations to investigate the effects of channel configurations and operation parameters. The experiments have been carried out with the counter flow arrangements. The flow rate of hot and cold water are set to 100–400 ml/min and 100–600 ml/min respectively. They have reported that the heat transfer is significantly improved, there is a significant increase in the pressure drop. Considering energy savings, a higher inlet temperature and moderate cold water flow are recommended during operation. Xu et al. [15] carried out experimental study on microchannel (hydraulic diameter of 0.39 mm) heat exchanger made of aluminum and silicon. They reported that the friction factor results in silicon channels are compatible with the literature, but high values in aluminum channels. Peng and Peterson [16] examined the fluid flow and heat transfer characteristics of water in microchannel with hydraulic diameters of 130-340 μm . All the channels were machined on a stainless steel surface. Deviations between predicted data were observed for experimental results of both flow friction and heat transfer. There have been several studies of the thermal and hydraulic performance capabilities of heat exchangers with micro channels. However, the thermal and hydraulic performance of a fluid flow and heat transfer under numerically and experimentally constant temperature conditions have not yet been studied.

This paper mainly focuses on fabricating a miniaturized cross flow heat exchanger, testing and also numerical investigating the heat and fluid flow inside aluminum heat exchanger plate which has square channels under constant wall temperature boundary condition. Primarily, the main objectives of this study are (1) to fabricate a cross flow microchannel heat exchanger, (2) to investigate heat transfer and fluid flow behavior, (3) to simulate with a commercial CFD program and (4) to compare CFD results to experimental data.

METHODOLOGY

The numerical simulation has been performed using ANSYS FLUENT 15. The steady state continuity, conservation of momentum and energy equations for fluid are defined by eq [1], eq [2], and eq [3], respectively.

$$\frac{D\rho}{Dt} + \rho \operatorname{div} V = 0 \quad (1)$$

$$\rho \frac{DV}{Dt} = \rho g - \nabla p + \mu \nabla^2 V \quad (2)$$

$$\rho c_p \frac{DT}{Dt} = \beta T \frac{Dp}{Dt} + \operatorname{div}(k \nabla T) + \phi \quad (3)$$

where ϕ is the viscous energy distribution function, and it becomes zero when the viscous dissipation effect is neglected. A commercial CFD code FLUENT is employed for the numerical simulation. The standard SIMPLE algorithm is selected and viscous laminar model was used for laminar ($\text{Re} < 2000$). The following assumptions were adopted: 1. Steady state fluid flow and heat transfer. 2. Incompressible fluid 3. Negligible radiative heat transfer 4. Constant fluid properties. When the viscous dispersion function is omitted and the continuous conditions are taken into

account, the energy equation can be written as follows. Then, two dimensional energy equation (Equation (3)) simplifies to [17];

$$\frac{1}{\alpha} \left(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} \right) = \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \quad (4)$$

where α is thermal diffusivity, $\alpha = k/(\rho c_p)$. For the case of no flow (that is, $u=v=0$) the energy equation simplifies to the two-dimensional, steady-state heat conduction equation with no heat generation. For the experimental investigation carried out in this study, the effects on the heat transfer and fluid flow of the heat exchanger will be discussed as follows.

Directly derived from 3D cylindrical coordinated Navier-Stokes momentum equations, the Hagen-Poiseuille equation gives the relationship between volumetric flow and pressure loss in a fully developed laminar flow [18]. Theoretical pressure loss across the flow in the cross flow heat exchanger microchannel can be calculated as follows;

$$\Delta P = 128 \frac{\mu L}{\pi D^4} \dot{V} \quad (5)$$

The frictional pressure drop can be defined following expression with inlet and outlet losses conventionally;

$$\Delta P = \Delta P_{i-o} - \left[\sum K_c \frac{\rho U_m^2}{2} + \sum K_e \frac{\rho U_m^2}{2} \right] \quad (6)$$

where ΔP_{i-o} , is pressure difference measured between inlet and outlet. K_c , and K_e , represents minor loss coefficients due to contraction, expansion and their values were taken as 0.8-1.0 for laminar flow [19], respectively.

Heat transfer during the phase change of liquid fluid in heat exchangers is written as follows [20];

$$\dot{Q} = \dot{m} h_{fg} \quad (7)$$

where \dot{m} mass flow rate associated with condensation or evaporation and h_{fg} is the enthalpy of evaporation at the specified temperature and pressure. In heat transfer calculation, the heat in hot side is equal to $\dot{Q}_{hot} = \dot{Q}$. This heat transfer value is equal to the heat transfer value on the hot fluid side of the heat exchanger. The heat transfer on the cold fluid side is:

$$\dot{Q}_{cold} = \dot{m} c_p (T_{c,i} - T_{c,o}) \quad (8)$$

In the microchannel heat exchanger, the energy balance equation can be calculated by the following equation.

$$\dot{Q}_{hot} - \dot{Q}_{loss} = \dot{Q}_{cold} \quad (9)$$

The rate of heat transfer across the surface are of cross flow heat exchangers can also be expressed as flowing equation,

$$\dot{Q} = FUA\Delta T_{lm} \quad (10)$$

F is the correction factor based on the logarithmic mean temperature difference ΔT_{lm} for a cross flow arrangement and if the temperature change of one fluid is negligible F equals to 1.

$$\dot{m}c_p(T_{c,i} - T_{c,o}) = UA\Delta T_{lm} \quad (11)$$

where U represents overall heat transfer coefficient and A is the heat transfer area;

$$\frac{1}{UA} = \frac{\Delta T_{lm}}{\dot{m}c_p(T_{c,i} - T_{c,o})} \quad (12)$$

T_{in} , T_{out} are respectively the inlet and outlet temperatures of the fluid. ΔT_m , the logarithmic temperature difference between the two fluids is determined by the following relation;

$$\Delta T_{lm} \equiv \frac{(T_{h,i} - T_{c,o}) - (T_{h,o} - T_{c,i})}{\ln \frac{(T_{h,i} - T_{c,o})}{(T_{h,o} - T_{c,i})}} \quad (13)$$

The logarithmic average temperature difference accurately reflects the average temperature difference between cold and hot fluids. Finally, the effectiveness ε , is determined by [21];

$$\varepsilon = \frac{Q_{cold}}{Q_{max}} \quad (14)$$

DESIGN AND FABRICATION

Micro heat exchanger geometry details such as plate's channel number, length, width and height were projected in the CAD (Figure 1).

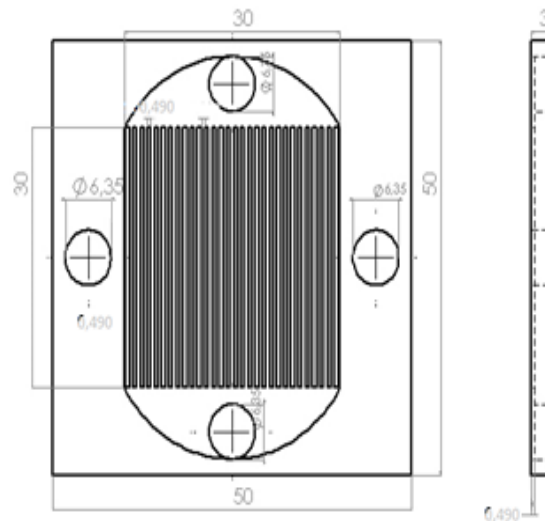


Figure 1. Microchannel plate's technical drawing

The heat exchanger was composed of two thermal exchanger plates in cross flow arrangement. The micro heat exchangers studied in this work has an architecture based on a large number of parallel microchannels showing a square cross section connected to two common manifolds. The microchannel plates that make up the heat exchanger were made of aluminum and these plates are 3mm thick. It has 31 microchannels on both plates where hot and cold

water passes. All microchannels were 30 mm long. 31 each microchannel is designed and manufactured with width and height equal to 490 μm . The plates were manufactured with precision CNC machining. Two stainless steel plates were connected to connect the top and bottom sides of the microchannel plates. The outside of heat exchanger was well insulated with glass wool. Geometrical dimensions of the tested micro heat exchanger are shown in Table 1.

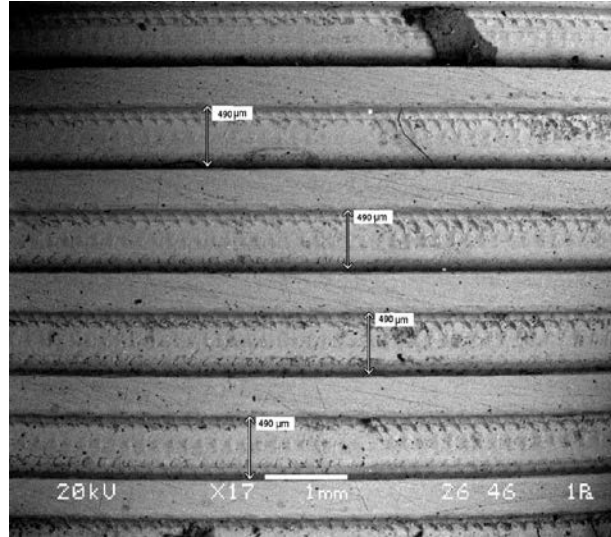


Figure 2. SEM view of a microchannel plate

Table 1. Geometrical dimensions of the tested micro heat exchanger

Geometrical details	Value
Number of channels	31
Channels width	490 μm
Channels height	490 μm
Channel length	30 mm
Dimension of the plate	50*50*3 mm
Number of the plate	2
Thickness of the plate	3 mm

THE EXPERIMENTAL FACILITY

The schematic representation of the experimental setup is shown in Figure 3. The test setup consists of pump, micro filter, water reservoir, heat exchanger, constant temperature water bath, computer and data collection system. The microchannel heat exchanger tested consists of parallel microchannel aluminum plates in a cross flow pattern. One of the fluids in the heat exchanger is water vapor in atmospheric conditions and the other is water. Vapor is generated by an electric steam boiler. The boiler operates under atmospheric conditions. Water vapor was sent to the heat exchanger through a copper tube. The mass of the water and duration in the boiler were measured so that water flow to the heat exchanger was calculated. In addition, a large collector was placed at the heat exchanger outlet and the water vapor was condensed on the wall of the water container under room conditions.

Water flow was provided by the micro pump operates in the range of 0-200 ml/min. The mass flow rate of water was determined by measuring on a digital balance (AND GX-600, ± 0.001 gr). The mass flow rate was measured after condensing the water vapor used in the experiments. The fluid temperature at the inlet to the heat exchanger was adjusted and controlled by a programmable constant temperature bath (Cole-Palmer 12108-25). The pressure difference in the heat exchanger was measured with pressure transducers (Keller 0-4 bar, $\pm 3.4\%$). Fluid temperatures at the inlet and outlet of the heat exchanger were measured using insulated K-type thermocouples (± 0.4 °C). The measured data were collected by a data acquisition system (Iotech PersonalDaq-3000) and transferred to a computer. Experimental tests were repeated at least three times under the same conditions to ensure reproducibility of the measurements. Uncertainty analysis was performed and the maximum uncertainties of heat transfer rate was estimated to be $\pm 9.2\%$.

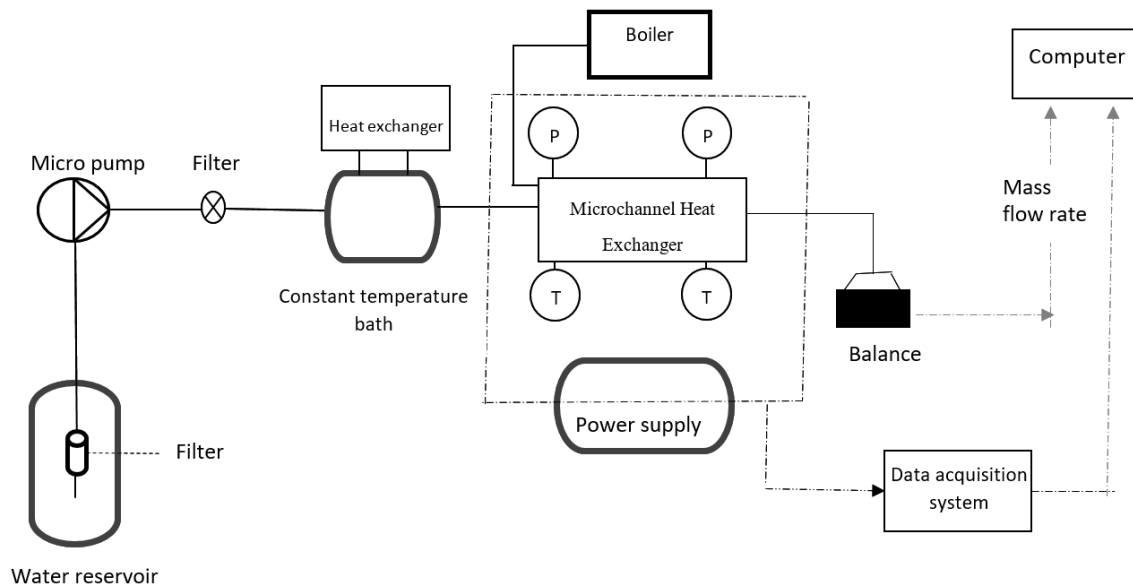


Figure 3. Experimental apparatus

NUMERICAL SIMULATIONS

The geometry of plates consisting of micro-channel heat exchanger with the dimensions of $50 \times 50 \times 3$ mm³ experimentally tested has been based on in numerical simulation. Micro-channel plate given in Figure 4 as three dimensions in the shape has been designed in SOLIDWORKS drawing program and defined in order to be analyzed in FLUENT. The meshing has been created by ANSYS ICEM CFD and grid independence sensitivity has been tested by three different grids as given in Table II. Percentage deviation has been calculated by $\left[\frac{f_2 - f_1}{f_1} \times 100 \right]$. The structure of grid include 1367720 nodes and 1187955 hexahedral volume elements. In addition, the solution was considered to converge and the relative residuals of continuity, momentum and energy equations were less than 0.0001. A zoom of the border zones of the model plate has been given in Figure 5.

Table 2. Grid independence

Parameters	Number of mesh elements: 7.86.E+5	Deviation [%]	Number of mesh elements: 1.01.E+6	Deviation [%]	Number of mesh elements: 1.18.E+6
Pressure drop ΔP [Pa]	2558.0	2.9	2638.1	1.3	2665.5
Outlet Temperature, T_{out} [K]	336.6	1.8	340.2	0.9	343.4

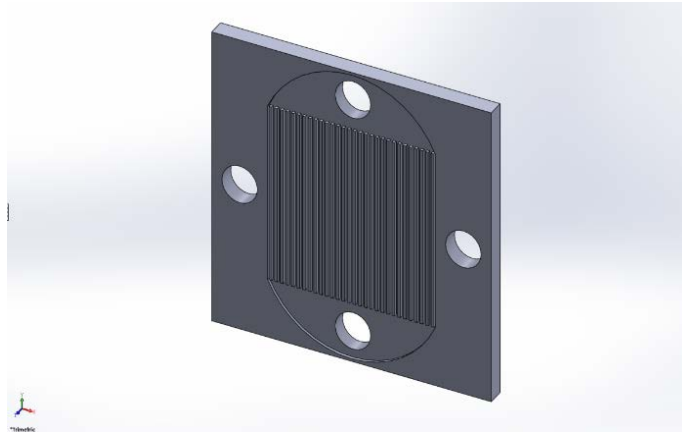


Figure 4. Microchannel plate's 3D technical drawing

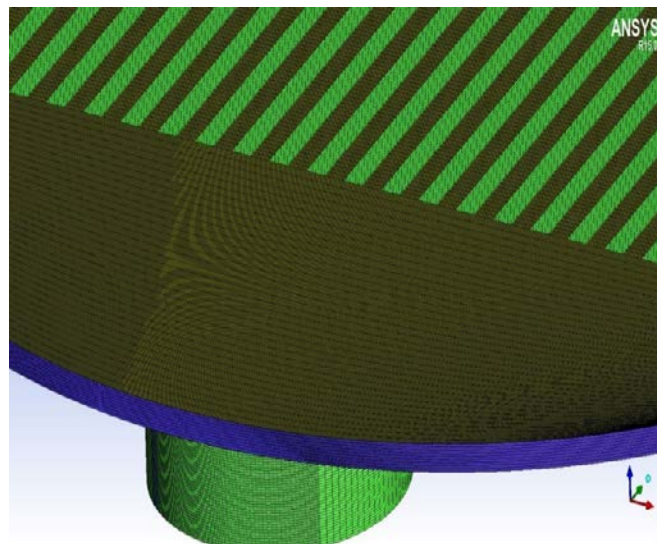


Figure 5. Zoom of grid mesh diagram profile

The fluid flow is assumed incompressible, laminar and viscous dissipation is negligible. At the inlet, water temperature is 19 °C. For different case, the mass flow rates are ranged from $1.64 \cdot 10^{-4}$ kg/s to $33.04 \cdot 10^{-4}$ kg/s. Due to the phase change, the upper wall of the plate was assumed to be at constant temperature. At the water outlet uniform atmospheric pressure is applied. The experiments and analysis have been conducted for different 20 mass flows rate

mentioned under the same conditions. The image of flow volume of working flow analyzed in ANSYS and all boundaries of the computational domain are marked schematically in Figure 6.

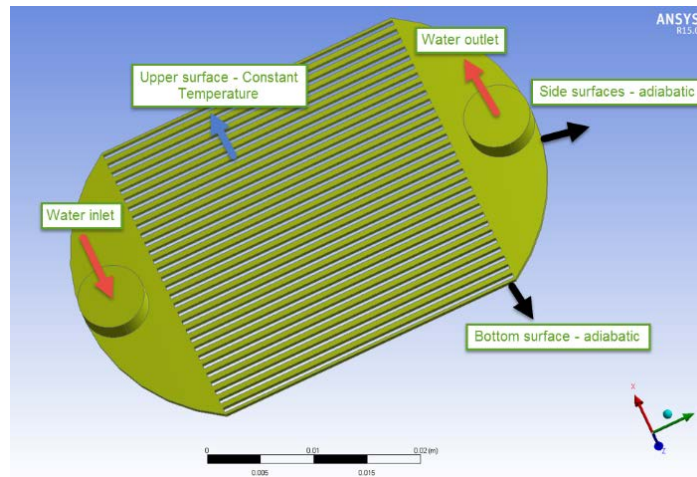


Figure 6. ANSYS computational domain and boundaries conditions

RESULTS AND DISCUSSION

CFD simulation was performed to determine the pressure drop and temperature distribution in the cross flow microchannel heat exchanger. The inlet temperatures of water and mass flow rates are given in the CFD simulation. The pressure distribution along microchannel plate for inlet mass flow rate of 0.001813 kg/s has been given in Figure 7. Maximum pressure value of 1388 Pa and minimum pressure value of 2354 Pa have been observed in inlet and outlet of heat exchanger plate, respectively. Also calculated pressure values continues to decrease towards to the heat exchanger plate outlet, it is indicating color change from red to blue in Figure 7. In the experimental study, experiments have been repeated three times under the same conditions, meanwhile the pressure drops have been measured.

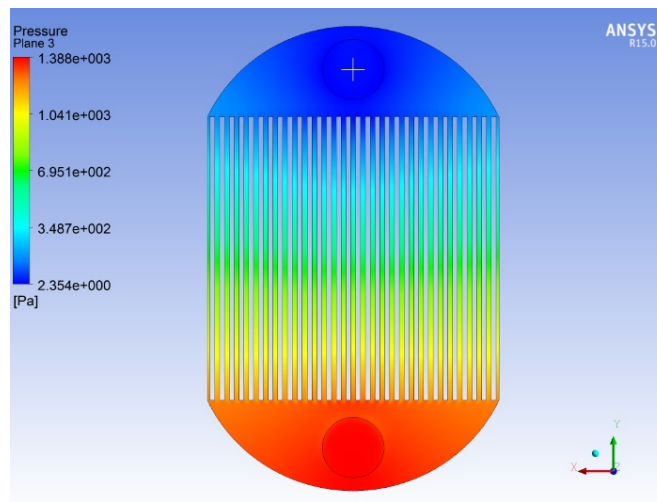


Figure 7. Pressure distribution along the channel

The pressure drop obtained by the CFD simulation according to the mass flow increase has been given in Figure 8. In addition, theoretical pressure drop values have been calculated by Equation 5. Comparison the pressure drop values obtained by the CFD modeling with the experimental values and also obtained by Equation 5 has been given in Figure 8. When experimental data is compared to the literature, deviations are observed, as seen in Figure 8.

The deviations observed are thought to be measurement errors. There are 31 parallel microchannels in the aluminum plate in this study. Although microchannel measurements are made with an advanced microscope, even the hydraulic diameter of one of the 31 microchannels is smaller than the others, resulting in increased pressure drop. For example, if the hydraulic diameter of microchannel is 10% smaller, the frictional pressure loss increases by one-half times. This causes errors in the frictional pressure drop values to fall outside the acceptable $\pm 10\%$ engineering band. There are studies in the literature that support our findings. Peng and Peterson [16] conducted heat and fluid flow experiments on microchannels made of stainless steel material and stated that their experimental results are different from the predicted results. Also, Xu et al. [15] have studied on aluminum heat exchangers and stated that their results are not compatible with the literature. As a result, they have reported that the pressure drop errors have been due to measurement errors due to production. Also, another reason other than measurement uncertainties is the structure of the inlet and outlet manifolds. Alm et. al. [7] has stated that the values of pressure drop obtained experimentally have been higher than the pressure drop values obtained from modelling. Another possibility of high experimental pressure drop is the roughness effect of the surfaces. But, the friction factor in laminar flow is independent of the surface roughness and depends only on the Re number. With the help of the equation $\frac{\epsilon}{R} = 2 \frac{\epsilon}{D} = \frac{5}{1.41\sqrt{Re}}$ relative roughness is $\epsilon / D \sim 4\%$ for critical Reynolds number (≈ 2300). Below 4% surfaces can be accepted as smooth [22]. Hydraulic diameter of microchannels is 0.49 mm and ϵ / D ratio equals to 0.2% ($\epsilon = 0.001$ for aluminum material). Consequently, the effect of surface roughness on the friction pressure drop is ignored. The effect of pressure drop on manifolds on the results will be examined in future studies.

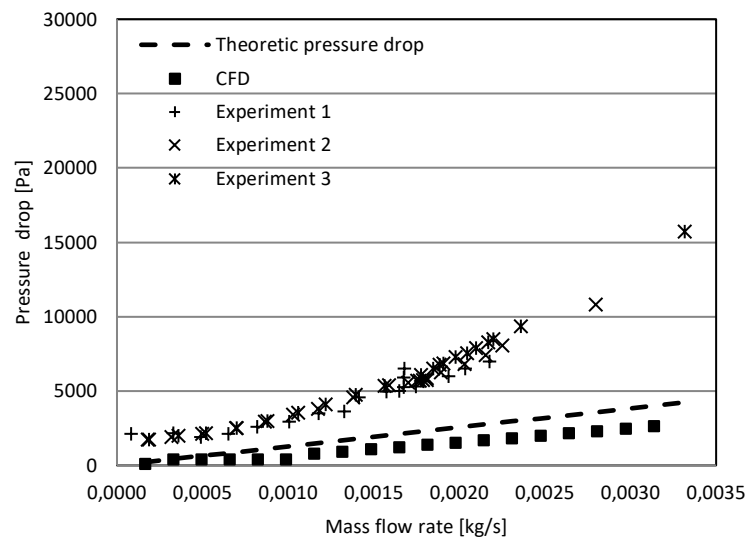


Figure 8. Comparison of the experimental and theoretical pressure drop values with CFD model

The temperature distribution along the microchannel heat exchanger plate by CFD model for mass flow rate of 0.001813 kg/s has been given in Figure 9. Temperature values are varying from center to border of the microchannel heat exchanger and more temperature rise occurs in border channels while water is flowing to outlet. The temperature of the fluid in channel inlet is at lowest level 19.7 °C and it is increasing to temperature of 95.6 °C towards the outlet. Figure 10 shows measured temperatures of cold and hot side of cross flow microchannel heat exchanger.

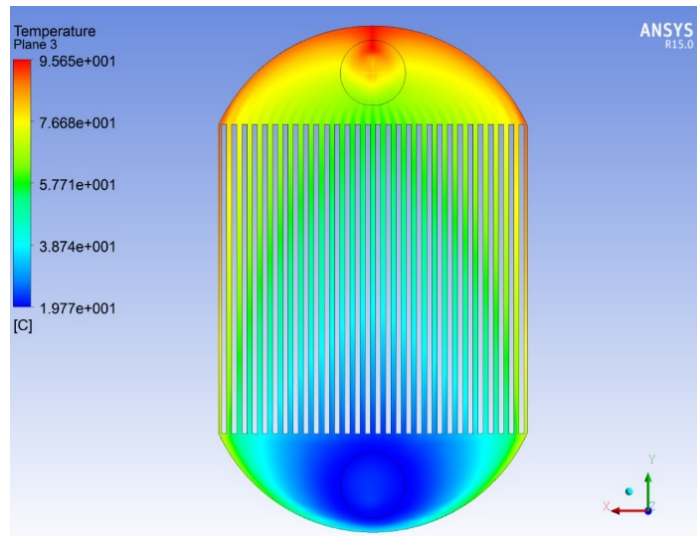


Figure 9. Temperature distribution along the channel

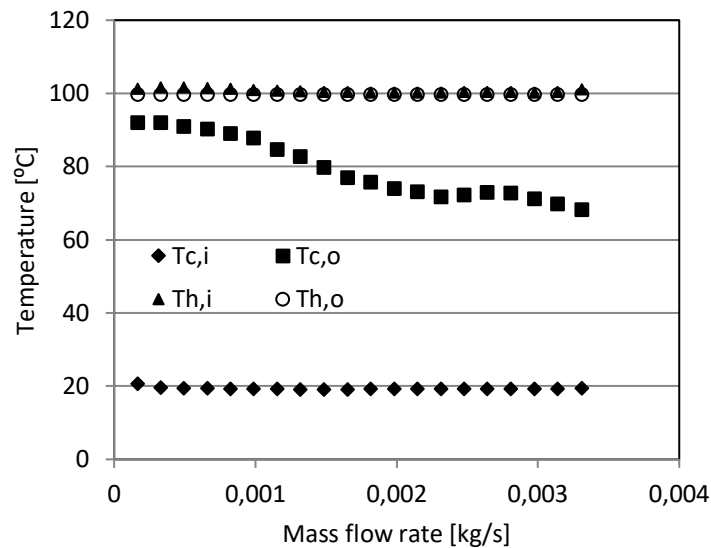


Figure 10. Measured temperatures of cold and hot side of cross flow microchannel heat exchanger

Boiler have generated vapor at atmospheric pressure. Vapor entries at temperature of 99.8 °C in the inlet of hot side of cross flow microchannel heat exchanger. Phase change occurs and hot side temperature maintain constant. The water goes into the cold side at 19 °C and it is getting hot. Outlet temperatures of water flow are varying with mass flow rate. Theoretically, the more the fluid remains in the channel, the more it heats. The outlet temperatures values acquired by CFD modelling for different mass flow rates are compared with experimental values in Figure 11. There is good consistency between experimental temperature values and CFD data.

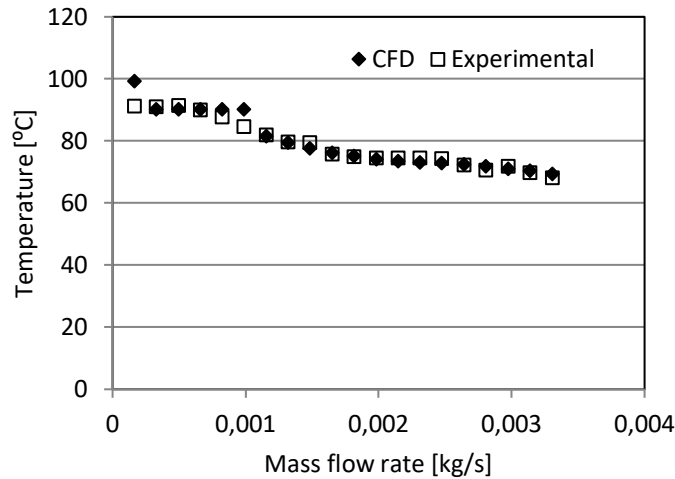


Figure 11. Comparison of experimental outlet temperatures with CFD modelling results

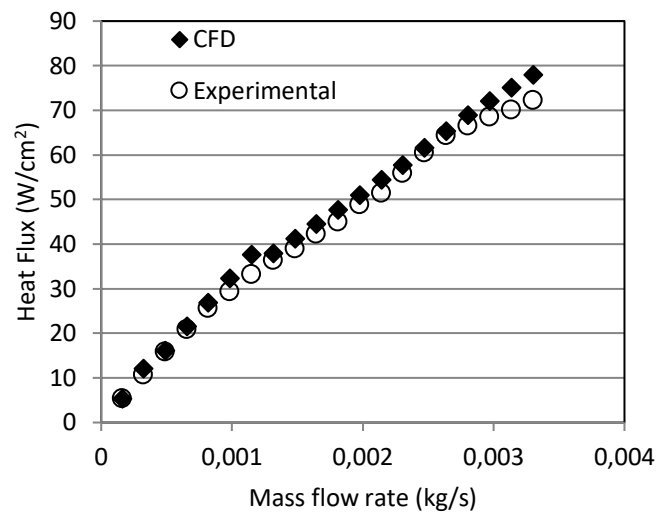


Figure 12. Comparison experimental heat flux values with CFD results

Figure 12 shows the change of experimental and CFD results of heat flux with mass flow rates. The mass flow rate increases from 0.11641 to 3.304 g/s and the heat flux increases from 5.3 to 77.9 W / cm². The heat flux values obtained from the experimental and CFD analyzes are generally compatible with each other. However, at high mass flow rates, the heat flux obtained from the CFD results is higher than the experimental data. Maximum heat flux difference is obtained at the highest mass flow rate and it is 4 W/cm². This difference may be occurred by reason of the fact that heat losses in the experiments. In addition, Dang et al. [9] reported an experimental heat transfer of 6.5 W/cm² for 0.2 g/s mass flow rate and 7.5 W/cm² for 0.3 g/s. As seen in Figure 12, experimental heat transfer was measured for 5.32 W/cm² for 0.16 g/s mass flow rate and 10.5 W/cm² for 0.3 g/s. Although the measured data do not correspond exactly with the data reported by Dang et al. [9], values support each other.

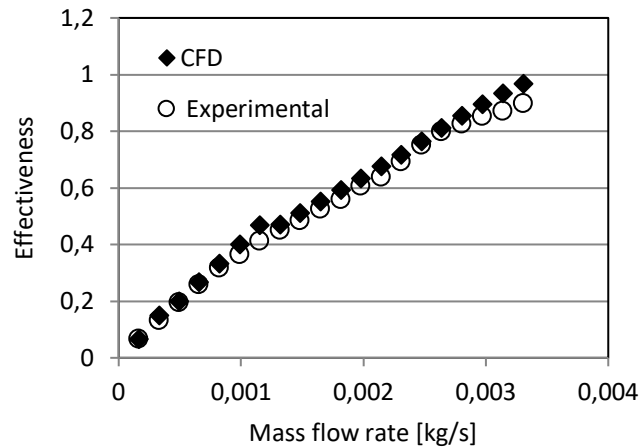


Figure 13. A Comparison between numerical and experimental effectiveness

The effectiveness values obtained from CFD simulation and the effectiveness calculated from experimental data are compared in Figure 13. As shown in Figure 13, the effectiveness of the cross-flow microchannel heat exchanger increased with increasing mass flow rate. When the experimental effectiveness compared with CFD results, experimental results are found to be compatible with those. Dixit and Ghosh [12] reported the efficiency value as approximately 0.37 versus 1.6 g/s mass flow rate of cold fluid. Efficiency was found to be 0.5 in similar mass flow rate, as seen in Figure 13.

CONCLUSION AND DISCUSSION

In this study, a cross flow microchannel heat exchangers made of aluminum material was manufactured and tested. Fluid flow and heat transfer for only one plate was modelled by ANSYS V15 FLUENT at constant surface temperature. Experimental results was verified by experimental data published in open literature. Numerical results were compared with experimental results; it was observed that the temperature data was compatible with each other. However, the experimental pressure drops and CFD results were not matched. It was concluded that there could be measurement uncertainty in determining hydraulic diameter of microchannels as the cause of deviations in pressure drop. There are studies in the literature that reached similar conclusions. Experimental and numerical results supporting the existing studies were obtained. As a result, exemplary results on topics such as pump selection and thermal efficiency for researchers working on microchannel heat exchangers were obtained.

NOMENCLATURE

A	Surface area, m^2
c_p	Specific heat capacity at constant pressure, $J/kg\ K$
D	Hydraulic diameter, m
f	Darcy friction factor
g	Gravitational acceleration, $m\ s^{-2}$
h	Heat transfer coefficient, $W/m^2\ K$
k	Thermal conductivity, $W/m\ K$
L	Micro channel length, m
\dot{m}	Mass flow rate, kg/s
P	Pressure, Pa
Pr	Prandtl number
Re	Reynolds number
T	Temperature, K
U	Overall heat transfer coefficient, $W/m^2\ K$
u, v	Velocity components in x, y directions, ms^{-1}

x, y Cartesian coordinates, m

Greek symbols

μ Dynamic viscosity, Pa s
 ρ Density, kg/m³
 Q Heat energy, kJ/s
 ε Average roughness height, m
 β Thermal expansion coefficient of fluid, K⁻¹
 ϕ Solid volume fraction
 α Thermal diffusivity of the fluid, m² s⁻¹

Subscripts

c Cold
 h Hot
 in Inlet
 out Outlet

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