Heat Transfer Analysis of Micro Channel Heat Sink

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Abstract: ***The paper presents the comparison of heat transfer characteristics of liquid coolants in forced convection cooling in a micro-heat sink with different pressure drops such as (35, 50 and 65kPa).The heat transfer characteristics of water and Propylene Glycol (PG) Water are obtained. The numerical results are validated against available results. Within the range of operating parameters, the heat transfer characteristics of PG water has been found better than water. Numerical results of a fluid flow micro-heat sink are obtained using commercial CFD software ANSYS-CFX.***

Keywords: micro-channel heat sink, CFD, heat transfer, fluid flow, forced convection, pressure drop, and liquid coolant.

1. Introduction

The progressive decrease of electronic device sizes and higher processing rates resulted in an increasing rate of heat generation per unit surface area, which means increased cooling requirements. The possibility of electronic equipment failure increases with the increase of temperature. Therefore cooling has become increasingly important in the design and operation of electronic equipment.

Tuckerman and Pease [1, 2] proposed the effective cooling techniques for electronic cooling. They built a water cooled integral heat sink with microscopic flow channels.

Weilin Qu and Issam Mudawar [3] analyzed the three-dimensional fluid flow and heat transfer processes in a rectangular silicon micro-channel heat sink numerically and a detailed description of the local and average heat transfer characteristics, i.e. temperature, heat flux and Nusselt number was obtained.

J.Li et al. [4] gave a detailed simulation of the heat transfer occurring in silicon based micro channel heat sink was conducted using a simplified three dimensional conjugate heat transfer model. The influence of the geometric parameters of the channel and the thermo physical properties of the fluid on the flow and the heat transfer are investigated using a temperature dependent thermo physical property method. The results indicate that the thermo physical properties of the liquid can significantly influence both the flow and heat transfer in the micro channel heat sink.

J. Li and G.P.Peterson [5] developed a full three dimensional numerical simulation for fluid flow and heat transfer in parallel micro-channel heat sink and evaluated the optimal geometric conditions. The single phase forced convective heat transfer and flow characteristics of water in micro channel structures/plates with small rectangular channel hydraulic diameters of 0.133-0.367 mm and distinct geometric configurations were investigated experimentally by Peng and Peterson [6] the laminar heat transfer was found to be dependent upon the aspect ratio and the ratio of hydraulic diameter to the center-center distance micro channel. The flow resistance of the liquid flow in the microstructures was also investigated experimentally and analytically, and correlations were proposed for the calculation of the flow resistance. Experimental investigation on liquid forced convection heat transfer through micro channels was studied by Peng and Wang [7] the experiments were conducted to investigate the single phase forced flow convection of water methanol flowing through micro channels with rectangular cross section. The transition and laminar heat transfer behavior on micro channels are very unusual and complex and are strongly affected by liquid temperature, velocity and micro channel size.

Lee P.S.et al. [8] investigated heat transfer in micro channels made of copper for Reynolds number 300 to 3500. The width of the studied channels range from 194 to 534μm, while the depths are five time the widths. In deducing the average Nusselt number, an average wall temperature based on a one-dimensional conduction model was used. For laminar flow the measured Nusselt number agreed with predictions for thermally developing flow over the entire length of the channel. Jiang Tao Liu et al. [9] investigated for fluid flow and heat transfer in micro channel cooling passages. Effect of viscosity and thermal conductivity variation on characteristics of fluid flow and heat transfer in micro channel heat sink are also calculated. They perform two dimensional simulations for low Reynolds number flow of liquid water in 100μm single channel subjected localized heat flux boundary conditions. The velocity field was highly coupled with temperature distributation and distorted through the variation of viscosity and thermal conductivity. Masud et al. [10] tried to validate the CFD package FLUENT with the experimental data obtained by then earlier. Here they have taken a heated chip with temperature 353 K and the air inlet velocity at temperature 293 K. The inlet velocities were varied from 1 m/s to 7 m/s. various turbulence models have been tested, and the effect of the channel inlet flow on the heat transfer rate has been determined by considering both a uniform and fully-developed condition. The substrate adiabatic heat transfer rate has been determined by considering both uniform and fully developed condition. The substrate adiabatic heat transfer coefficient is also numerically determined.

2. Mathematical Formulation

The micro heat sink model consists of a 10 mm long silicon substrate with silicon cover. The rectangular micro-channels have a width of 57μm and a depth of 180μm. A schematic of the structure of a rectangular micro-channel heat sink is shown in fig.1, where a unit of cell consisting of one channel was selected because of symmetry of the structure. The bottom surface of heat sink is uniformly heated

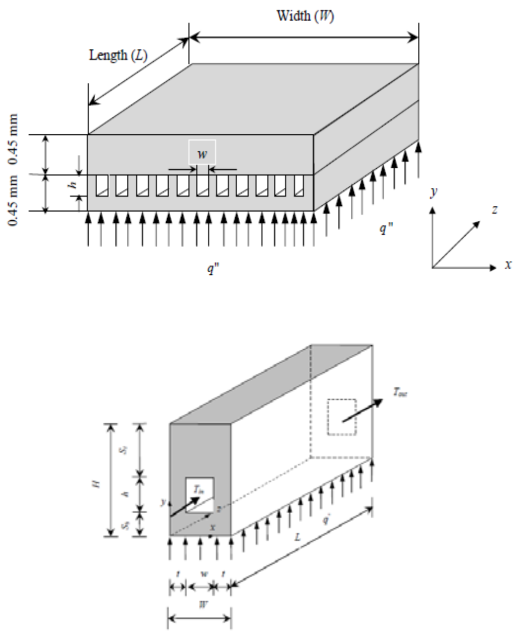

Figure 1: Structure of a rectangle micro-channel heat sink and the unit with a constant heat flux and at the top surface is well insulated.

The physical dimensions of the micro-channel are presented in Table 1, whereas the thermo-physical properties of the fluid (water or PG Water) and solid are presented in Tables 2 and 3, respectively.

Table 1: Governing dimensions of the single micro-channel

H (*μm*)	*h* (*μm*)	*W* (*μm*)	*w* (*μm*)	S_t (*μm*)	S_b (*μm*)	*t* (*μm*)	*L* (*mm*)
900	180	100	57	450	270	21.5	10

Table 2: Thermo physical Properties of fluid

Fluid	ρ *Kg/m*^{3}	*K* *W/m-K*	μ *Kg/m-s*	C_p *J/Kg-K*	*T* *K*
Water	998.2	0.6	0.001003	4182	293
PG Water	1062	0.36	0.0064	3400	293

Table 3: Thermo physical Properties of solid

Solid	ρ *Kg/m*^{3}	*K* *W/m-K*	C_p *J/Kg-K*
Silicon	2330	148	712

3. Governing Equations

For a fully developed laminar flow in a micro channel the entrance length=0.0575Re×Dh (1)

For a hydraulic diameter Dh = 86.58 and Reynolds number Re= 106, the entrance length = 527.705 μm which is less than 10 mm. So fully developed laminar flow is valid.

1. Continuity equation

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \qquad (2)$$

2. Momentum equation (Navier- stokes equations)

X-momentum equation

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=-\frac{\partial p}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right) \qquad (3)$$

Y-momentum equation

$$\rho\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}\right)=-\frac{\partial p}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right) \qquad (4)$$

Z- Momentum equation

$$\rho\left(u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right)=-\frac{\partial p}{\partial z}+\mu\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right) \qquad (5)$$

3. Energy equation

$$\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=\frac{1}{\alpha}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \qquad (6)$$

The hydrodynamic boundary condition can be stated as at the inner bottom wall surface of channel (no-slip condition)

u = 0, v = 0, w = 0, (7a)

at the inlet,

$z = 0, Pl = Pin, u = 0, v = 0$ (7b)

at the outlet,

$z = Lz, Pl = Pout, u = 0, v = 0$ (7c)

the following thermal boundary conditions at bottom wall

$$y = 0, -k\frac{\partial T}{\partial y} = q'' \quad (8a)$$

at inlet

$z = 0, T = Tin$ (8b)

4. Result and Discussion

The present numerical simulation has been done using ANSYS-CFX and meshing is done by ICEM-CFD with tetra mesh with 159468 nodes. The result has been partially validated with grid independence tests and there upon the lower mesh size is selected for rest of the simulations. The model of present study is validated against the numerical results of [4]. Fig. 2 shows the validated results of average heat transfer coefficient inside the channel for constant pressure drop of 50kPa and heat flux 90 W/cm2.

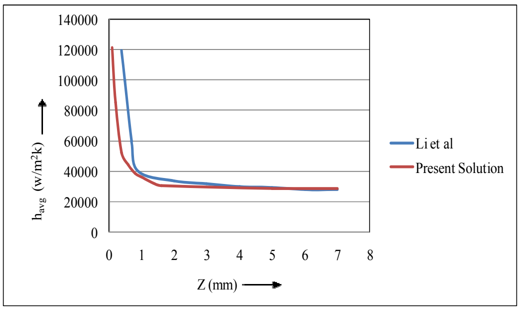

Figure 2: Model validation using average heat transfer coefficient inside the channel with the numerical result for Δp = 50kPa at q''= 90W/cm^2.

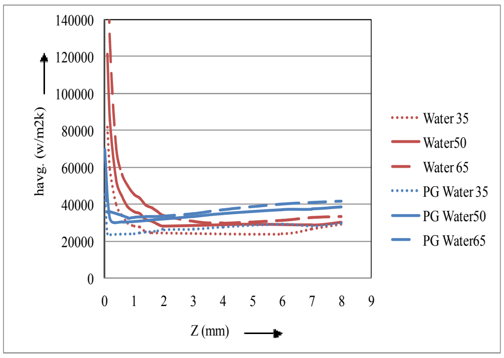

Figure 3: Comparison of average Wall heat transfer coefficient between water and PG Water of micro channel heat sink for different pressure drop such as (35, 50 and 65kPa) and q'' = 90 W/cm ^{2}.

The average wall heat transfer coefficient of PG Water and water are compared in fig.3 with different pressure drop. It is interesting to note that the average wall heat transfer coefficient sharply decreases due to growing boundary layer thickness. The heat transfer coefficient decreases along the flow direction and it is extremely high in the entrance region due to very thin boundary layer.

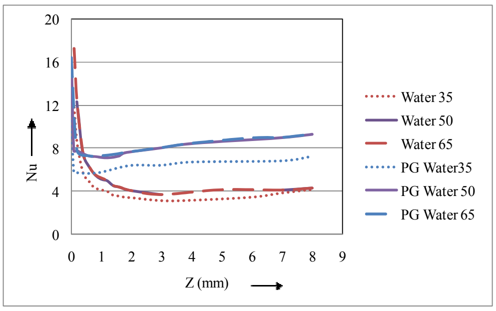

Figure 4: Comparison of Nusselt number between water and PG Water of micro channel heat sink for different pressure drop such as (35, 50 and 65kPa) and q'' = 90 W/cm ^{2}.

Fig.4 show the comparison of Nusselt number of water and PG water with different pressure drop .The average Nusselt number decreases along the channel due to formation of boundary layer.

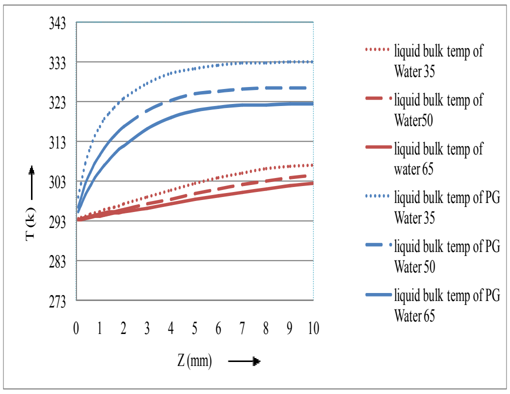

Figure 5: Comparison of temperature difference between water and PG Water of micro channel heat sink for different pressure drop such as (35, 50 and 65kPa) and q" = 90 W/cm ^{2}.

Fig.5 represents the comparison of the temperature variation of PG water and water with different pressure drop. It should be noted that the exit temperature of PG water is more than that of water which indicate that more heat can be transferred from the heat sink when using PG Water.

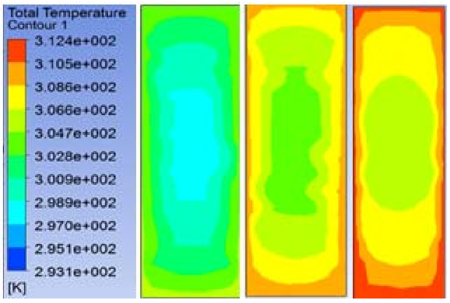

Figure 6: Temperature contour of Water in x-y plane of micro-channel heat sink at 1/3, 2/3, and outlet of the channel at Δp = 35kpa at q'' = 90 W/cm^2.

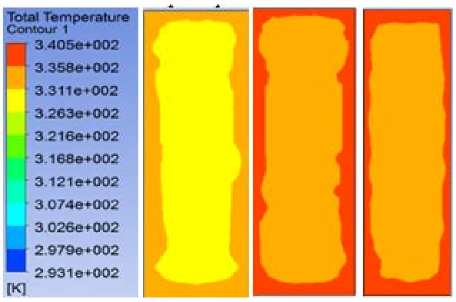

Figure 7: Temperature contour of PG Water in x-y plane of micro-channel heat sink at 1/3, 2/3, and outlet of the channel at Δp = 35kpa at q'' = 90 W/cm^2.

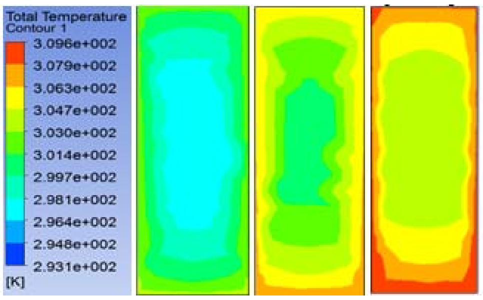

Figure 8: Temperature contour of Water in x-y plane of micro-channel heat sink at 1/3, 2/3, and outlet of the channel at Δp = 50kpa at q'' = 90 W/cm^2.

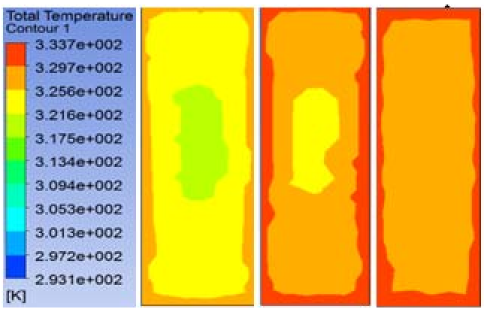

Figure 9: Temperature contour of PG Water in x-y plane of micro-channel heat sink at 1/3, 2/3, and outlet of the channel at Δp = 50kpa at q'' = 90 W/cm^2.

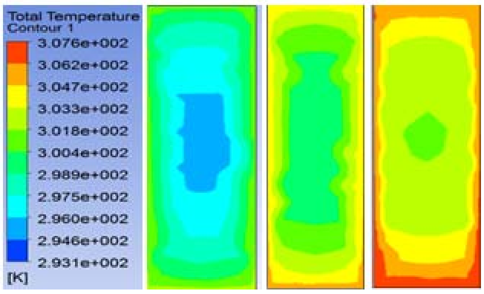

Figure 10: Temperature contour of Water in x-y plane of micro-channel heat sink at 1/3, 2/3, and outlet of the channel at Δp = 65kpa at q'' = 90 W/cm^2.

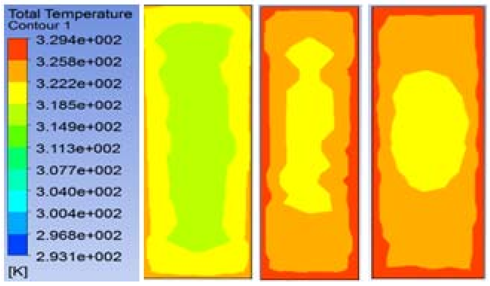

Figure 11: Temperature contour of PG Water in x-y plane of micro-channel heat sink at 1/3, 2/3, and outlet of the channel at Δp = 65kpa at q'' = 90 W/cm^2.

5. Conclusion

Heat transfer characteristics of micro channel heat sink using two liquid coolants, namely, water and PG Water has been compared. It has been found that the heat transfer characteristics of PG Water are better compared to Water for a different pressure drop such as (35, 50 and 65kPa).Which means that more heat can be transferred from the micro channel heat sink when using PG Water in place of Water.

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