

# Performance Analysis of Electronics Cooling using Nanofluids in Microchannel Heat Sink

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**Abstract**—Performance analysis of thermal enhancement for cooled microchannel heat sink (MCHS) using nanofluids mathematical formulation was investigated and presented in this paper. Heat transfer capability in terms of thermal conductivity, heat transfer coefficient, thermal resistance, heat flux and required pumping power were evaluated on the effectiveness of copper oxide (CuO), silicon dioxide (SiO<sub>2</sub>) and titanium dioxide (TiO<sub>2</sub>) with water as a base fluid. The results showed that thermal performance augmented by 12.2% in thermal conductivity at particle volume fraction of 4% to CuO-water nanofluid, 11.8% for SiO<sub>2</sub>-water and 10.0% for TiO<sub>2</sub>-water. The maximum heat transfer coefficient enhances of 12.4% for CuO, SiO<sub>2</sub> is 8.22% and 7.4% for TiO<sub>2</sub> with the same inlet velocity of 3 m/s. The addition of nanoparticle concentration significantly enhances the heat transfer, but elevates the expenses of higher required pumping power to increase the pressure drop. The maximum enhancement of heat flux in CuO-water was found to be 2575 kW/m<sup>2</sup>, 2501 kW/m<sup>2</sup> for SiO<sub>2</sub>-water and 2485 kW/m<sup>2</sup> for TiO<sub>2</sub>-water nanofluid at 4% of volume fraction. The pressure drop is increased with the mass flow rate of 1021 kg/m<sup>3</sup> for CuO-water at 0.5% of volume fraction and 47925 Pa to 54314 Pa pressure drop at 4% of volume fraction. The CuO-water pumping power was found to be the highest at 4% of volume fraction with 102.3 W at 3 m/s inlet velocity compared to SiO<sub>2</sub> and TiO<sub>2</sub> also increased the pumping power of 75.0 W to 90.6 W with increasing volume fraction and pressure drop. The positive thermal results implied that CuO nanofluid is a potential candidate for future applications in MCHS. Further analysis is recommended to be done with various Reynolds number, pumping power and flow rate of nanoparticles to obtain better heat transfer performance of cooling fluids.

**Keywords:** Electronics cooling, nanofluids, microchannel heatsink

## I. INTRODUCTION

In the last three decades, the emergence of nanotechnology is rapidly approaching, were utilized to improve the heat transfer rate to apply on the electronic devices in order to reach a satisfactory level of thermal efficiency. The heat transfer rate can passively be improved by changing the geometry's flow, boundary conditions or by improving thermo physical properties such as increasing the thermal conductivity of fluid [1]. To meet the high dissipation rate requirements and maintain a low junction temperature in electronic devices, many cooling technologies have been pursued. Among of these, the microchannel heat sink (MCHS) was introduced because of its ability to produce high heat transfer coefficient, small size and volume per heat load and small coolant requirements [2].

Working fluids was applied to enhance the heat transfer by changing the fluid transport properties and flow features in MCHS. Recently, this concept has focused on heat transfer enhancement by using a nanofluid that has a nanoscale metallic or non-metallic particles in the base fluids. Besides, nanofluids has become a concern because they display higher potential as heat transfer fluid than normally utilized base fluids and micron sized particle-fluids. This is due to clogging in pumping and flow apparatus which is caused by rapid settling of the micron sized particle. Nanofluids do not indicate this behavior. This makes nanofluids a better choice as heat transfer fluid [3]. Nanofluids (1-100nm-size particles), often called as ultra-fine solid particles, engineered colloidal suspension, are stable and prepared by dispersing a certain percentage of nanoparticles in base fluids [4-6].

The factors that causes heat transfer enhancement are solid particles and host fluids chemical composition, size, shape and concentration of nanoscale particles, thermal condition and surfactants. Some of these factors also affect the stability of the nanofluids. There are three strategies to attain good stability, namely addition of surfactants, pH control and ultrasonication [7]. From the literature, heat transfer coefficient depends on Reynolds number, volume fraction of the nanofluids (concentration), temperature, base-fluid

thermal properties and nanofluids purity. Generally, nanofluids are highly potential to be used as coolant in electronic packaging since their heat transfer coefficient exceeded the predicted value in laminar flow region in many analyses and from the famous Dittus-Boelter correlation in turbulent correlations [8].

Zakaria et al.[9] has studied on numerical analysis of thermal enhancement for a single Proton Exchange Membrane Fuel Cell (PEMFC) cooling plate by using a low concentration of  $Al_2O_3$  in Water-Ethylene Glycol mixtures as a coolant. It shown that the higher volume percent concentration of  $Al_2O_3$  the better the heat transfer enhancement but at the higher expense of pumping power.

Moraveji et al. [10] used a model of MCHS with 20 x 20 mm bottom with five nanoparticle volume fractions in five inlet velocities for two types of nanoparticle containing  $TiO_2$  and  $SiC$ . By using different value of Reynolds Numbers, the effect of a nanoparticle volume fraction on the convective heat transfer coefficient was investigated. The modelling results was compared to analytical calculations and it showed that, it was accurate for the correlated equations that were obtained for Nusselt number and friction factor were acceptable.

The numerical simulations is studied on the laminar and turbulent forced convection heat transfer in a MCHS with a mixture of nanofluid consisting of CuO-water [11]. The method used to solve the continuity, momentum and energy equations was the finite volume method with the parameters of the particle volume fraction ( $\phi = 0.204\%, 0.256\%, 0.294\%$  and  $0.4\%$ ), and the volumetric flow rate ( $\dot{V} = 10 \text{ mL/min}, 15\text{mL/min}$  and  $20\text{mL/min}$ ). From the comparisons of thermal resistance predicted by the single-phase and two-phase models with the experimental results, it revealed that, two-phase model was more accurate than the single-phase model. Other than that, the thermal resistance of nanofluids is smaller than that of water, which decreases as the particle volume fraction and the volumetric flow rate increase in laminar flow. In addition, the pressure drop increases slightly for nanofluid-cooled MCHS in the laminar flow case.

This study deals with three types of nanoparticles which are copper oxide(CuO), silica( $SiO_2$ ) and titanium oxide ( $TiO_2$ ) suspended in a water as a base fluid. The microchannel heat sink (MCHS) operation was analyzed with the nanofluids serve as a working fluid. Performance of nanofluids as coolant is predicted in terms of thermal conductivity, heat transfer coefficient, thermal resistance, heat flux and required pumping power.

## II. METHODOLOGY

The performance of CuO-water,  $SiO_2$ -water and  $TiO_2$ -water nanofluids has been analyzed by using mathematical formulation to compare the performance of cooling of electronics. Thermophysical properties of the water and nanoparticles (CuO,  $SiO_2$ ,  $TiO_2$ ) at  $30^\circ\text{C}$  areas shown in Table I.

TABLE I. Thermophysical properties of the base fluid and nanoparticles [10, 12]

Fluid/Nanoparticles	Properties			
	$\rho(\text{kg/m}^3)$	$\mu(\text{N.s/m}^3)$	$C_p(\text{J/kg.K})$	$k(\text{W/m.K})$
Base Water, $H_2O$	994.2	$724.6 \times 10^{-6}$	4178	0.6248
Copper Oxide, CuO	6320	-	385	76.5
Silica Dioxide, $SiO_2$	3970	-	765	36
Titanium Dioxide, $TiO_2$	4157	-	710	8.4

### A. Nanofluid Properties

The thermophysical properties of the nanofluid with the volume fractions,  $\phi$ , of nanoparticles and base fluid were used; 0.5%, 1.0%, 1.5%, 2.0%, 2.5%, 3.0%, 3.5% and 4.0% are determined by utilizing the following equations:

Density of nanofluid is calculated by the use of Pak and Cho [13] correlation:

$$\rho_{nf} = \phi \rho_s + (1 - \phi) \rho \quad (1)$$

Effective thermal conductivity is given by Hamilton and Crosser [13] equation, can be expressed as follow:

$$k_{nf} = \frac{k_p + (n-1)k_f - (n-1)\phi(k_f - k_p)}{k_p + (n-1)k_f + \phi(k_f - k_p)} k_f \quad (2)$$

Nanofluid specific heat equation is evaluated from Xuan and Roetzel correlation [14] which shown below:

$$(\rho C_p)_{nf} = (1 - \phi)(\rho C_p)_f + \phi(\rho C_p)_p \quad (3)$$

Effective viscosity of nanofluid is given by Einstein equation as suggested for particle in volume fractions less than 5.0 vol.% and is defined [15]:

$$\mu_{nf} = \mu(1 + 2.5\phi) \quad (4)$$

Where subscript  $f$ ,  $p$ ,  $nf$  are correspond to fluid, particle and nanofluids respectively. Nanoparticle shape factor,  $n$ , was assumed to be 3 for spherical particles as tabulated in Table II.

TABLE II. Calculated thermophysical properties for nanofluids

<b>CuO/Water Properties</b>				
<b>Particle Volume Fraction(<math>\phi</math> %)</b>	<b><math>\rho_{CuO}</math></b>	<b><math>k_{CuO}</math></b>	<b><math>\rho_{CpCuO}</math></b>	<b><math>\mu_{CuO}</math></b>
0.5%	1020.83	0.63399	4145165	0.00073
1.0%	1047.46	0.64327	4136562	0.00074
1.5%	1074.09	0.65265	4127959	0.00075
2.0%	1100.72	0.66211	4119356	0.00076
2.5%	1127.35	0.67167	4110753	0.00077
3.0%	1153.97	0.68133	4102151	0.00078
3.5%	1180.60	0.69109	4093548	0.00079
4.0%	1207.23	0.70094	4084945	0.00080

<b>SiO<sub>2</sub>/Water Properties</b>				
<b>Particle Volume Fraction(<math>\phi</math> %)</b>	<b><math>\rho_{SiO_2}</math></b>	<b><math>k_{SiO_2}</math></b>	<b><math>\rho_{CpSiO_2}</math></b>	<b><math>\mu_{SiO_2}</math></b>
0.5%	1009.08	0.63374	4148184	0.00073
1.0%	1023.96	0.64277	4142600	0.00074
1.5%	1038.84	0.65189	4137017	0.00075
2.0%	1053.72	0.66109	4131433	0.00076
2.5%	1068.60	0.67038	4125850	0.00077
3.0%	1083.47	0.67977	4120266	0.00078
3.5%	1098.35	0.68924	4114682	0.00079
4.0%	1113.23	0.69881	4109099	0.00080

<b>TiO<sub>2</sub>/Water Properties</b>				
<b>Particle Volume Fraction(<math>\phi</math> %)</b>	<b><math>\rho_{TiO_2}</math></b>	<b><math>k_{TiO_2}</math></b>	<b><math>\rho_{CpTiO_2}</math></b>	<b><math>\mu_{TiO_2}</math></b>
0.5%	1010.01	0.63238	4147756	0.00073
1.0%	1025.83	0.64003	4141745	0.00074
1.5%	1041.64	0.64773	4135733	0.00075
2.0%	1057.46	0.65550	4129722	0.00076
2.5%	1073.27	0.66333	4123710	0.00077
3.0%	1089.08	0.67123	4117699	0.00078
3.5%	1104.90	0.67919	4111687	0.00079
4.0%	1120.71	0.68722	4105676	0.00080

### B. Microchannel Heat Sink (MCHS)

The MCHS which typically contains lot of parallel microchannel has the capability of producing high heat transfer coefficient, less dimension-volume per heat load and lesser requirement of coolant [2]. There were few assumptions have been made in order to simplify this analysis whereby[9]:

- The flow is incompressible, laminar and in steady state.
- The effect of body force is neglected.
- The fluid properties are constant and viscous dissipation is neglected.
- The fluid phase and nanoparticles are in thermal equilibrium with zero relative velocity and the resultant mixture can be considered as a conventional single phase.

The geometric configuration of this microchannel is considered based on research of Tsai and Chein[2] shown in Fig.1 whereby the nanofluids is forced to flow through the fin slot in x-direction with mean velocities of 3 m/s. Table III below shows the details of MCHS dimensions.

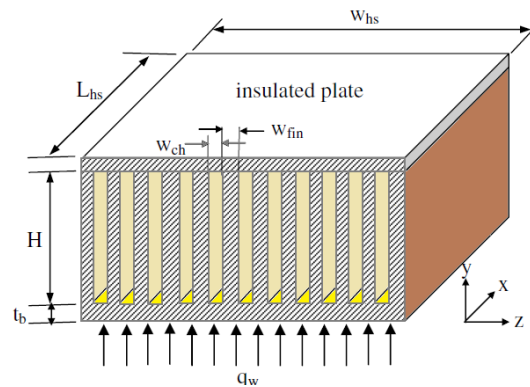


Fig. 1. MCHS schematic diagram [2]

TABLE III. Dimensions of MCHS [2]

Symbol	Size
Size $L_{hs} \times W_{hs}$	1 cm x 1cm
Channel height, H	365 $\mu\text{m}$
Channel width, $W_{ch}$	57 $\mu\text{m}$
Porosity, $\varepsilon$	0.5
Fin thickness, $W_{fin}$	57 $\mu\text{m}$
Number of channels, n	25
Aspect ratio, $\alpha_s$	6.4

Parameters like hydraulic diameter and other dimensionless correlations for Reynolds number, Prandtl number and Nusselt number are considered under the assumption of single phase, constant thermal properties applicable for laminar and turbulent flows in order to analyze the efficiency of MCHS by applying nanofluids to achieve the heat dissipation capability [16, 17].

The heat transfer coefficient is calculated by using:

$$h = \frac{Nu k_{nf}}{D_h} \quad (5)$$

Where;

$$\text{Hydraulic diameter} : D_h = \frac{4W_{ch}H}{2(W_{ch}+H)} \quad (6)$$

$$\text{Reynolds number} : Re = \frac{V_m D_h}{\nu} \quad (7)$$

$$\text{Prandtl number} : Pr = \left( \frac{C_p \mu}{k} \right)_{nf} \quad (8)$$

$$\text{Nusselt number} : Nu = 0.023 Re^{0.8} Pr^{0.3} \quad (9)$$

The calculation for Reynolds, Prandtl and Nusselt number are based on the assumption that the fluid flow with the inlet speed of 3m/s for various volume fraction of nanoparticle.

The efficiency of the microchannel is computed using:

$$\eta = \frac{\tanh(m \times H)}{(m \times H)} \quad (10)$$

Where;

$$m \times H = \sqrt{\frac{2h}{k_s W_{fin}}} \times H \quad (11)$$

$k_s$  is the MCHS thermal conductivity which is 400W/mK.

Furthermore, total thermal resistance is computed using the summation of all the three resistances:

$$R_t = \frac{1}{h A_s} + \frac{1}{\dot{m} C_p} + \frac{H}{k_s A_b} \quad (12)$$

where  $A_b$  is the area of MCHS bottom part and  $\dot{m}$  is the total nanofluid mass flowrate [12].

$$\text{The surface area is calculated by the formula of: } \dot{m} = n \rho_{nf} A_c \quad (13)$$

where  $A_c$  is the channel area.

$$A_s = n W_{ch} L_{hs} + 2n \eta H L_{hs} \quad (14)$$

Number of cooling channels,  $n = 25$  [18].

Thus, the total heat transfer ( $Q$ ) and the bottom heat flux  $\dot{q}$  is computed using:

$$Q = \frac{T_{max} - T_i}{R_t} \quad (15)$$

and

$$\dot{q} = \frac{Q}{A_b} \quad (16)$$

Where,  $T_i$  is the fluid temperature at the inlet and  $T_{max}$  is the largest of bottom temperature of microchannels. From Xie et.al [19] the maximum difference of ( $T_{max} - T_i$ ) is taken to be 50°C.

### C. Effectiveness Evaluation of Nanofluid for Electronics Cooling

The effectiveness of nanofluid as coolant is evaluated in term of performance of microchannel heat sink. This analysis is crucial in order to see the impact of the viscous pressure drop on the performance of MCHS.

Coefficient of performance, COP for MCHS is defined as the ratio of the dissipated heat to the invested pumping power [20].

$$COP = \frac{Q}{P} \quad (17)$$

Where;

$$P_{pow} = \dot{V} \Delta p \quad (18)$$

$$\dot{V} = n \rho_{nf} W_{ch} H V_m \quad (19)$$

( $P_{pow}$ : Idealized pumping power and: Volumetric flowrate of nanofluid,  $n$ : Number of channels).

The pressure drop is calculated using [21],

$$\Delta p = f \frac{L_{hs} \rho V_m^2}{D_h^2} \quad (20)$$

with Darcy friction factor :  $f = (1.82 \log Re - 1.64)^{-2}$  (21)

### III. RESULT AND DISCUSSION

Nanoparticle addition in fluid base will result higher thermal conductivity compared to a conventional liquid and conventional two-phase mixture coolant. Solid particle are added as they conduct heat much better than a liquid on its own. In this study, three types of nanofluids were used as coolant in microchannel heat sink. Performance of nanofluids as coolant were predicted in terms of thermal conductivity, heat transfer coefficient, thermal resistance, heat flux and required pumping power. The result was compared with conventional liquid which is water. Table IV shows the properties of water as coolant in MCHS with inlet velocity of 3 m/s.

TABLE IV. Properties of water in MCHS with inlet velocity of 3 m/s.

Properties	Values
Thermal conductivity, $k$ (W/m.K)	0.6248
Heat transfer coefficient, $h$ (W/m <sup>2</sup> .K)	30111.41
Reynolds No, Re	405.87
Prandtl No, Pr	4.85
Nusselt No, Nu	4.75

The prediction of performance of CuO-water, SiO<sub>2</sub>-water and TiO<sub>2</sub>-water nanofluids were calculated using Eq. (1)-Eq (21). The calculated data is shown in Table V, Table VI and Table VII.

TABLE V. Thermal conductivity, heat transfer coefficient and mass flow rate of CuO-water nanofluid.

CuO/Water			
Particle Volume Fraction( $\phi$ %)	$k$ (W/m.K)	$h$ (W/m <sup>2</sup> .K)	$\dot{m}$ (kg/s)
0.5%	0.63399	30584	0.01455
1.0%	0.64327	31055	0.01493
1.5%	0.65265	31525	0.01531
2.0%	0.66211	31993	0.01569
2.5%	0.67167	32460	0.01606
3.0%	0.68133	32926	0.01644
3.5%	0.69109	33392	0.01682
4.0%	0.70094	33856	0.01720

TABLE VI. Thermal conductivity, heat transfer coefficient and mass flow rate of SiO<sub>2</sub>-water nanofluid.

SiO <sub>2</sub> /Water			
Particle Volume Fraction( $\phi$ %)	$k$ (W/m.K)	$h$ (W/m <sup>2</sup> .K)	$\dot{m}$ (kg/s)
0.50%	0.63374	30419	0.01438
1.00%	0.64277	30727	0.01459
1.50%	0.65189	31037	0.01480
2.00%	0.66109	31347	0.01502
2.50%	0.67038	31658	0.01523
3.00%	0.67977	31970	0.01544
3.50%	0.68924	32283	0.01565
4.00%	0.69881	32597	0.01586

TABLE VII. Thermal conductivity, heat transfer coefficient and mass flow rate of TiO<sub>2</sub>-water nanofluid.

TiO <sub>2</sub> /Water			
Particle Volume Fraction(Ø %)	k(W/m.K)	h(W/m <sup>2</sup> .K)	m (kg/s)
0.50%	0.63238	30387	0.01439
1.00%	0.64003	30664	0.01462
1.50%	0.64773	30940	0.01484
2.00%	0.65550	31217	0.01507
2.50%	0.66333	31494	0.01529
3.00%	0.67123	31772	0.01552
3.50%	0.67919	32050	0.01574
4.00%	0.68722	32328	0.01597

#### A. Thermal Conductivity of nanofluids

Thermal conductivity enhancement as a function of particle volume fraction for the CuO-water, SiO<sub>2</sub>-water and TiO<sub>2</sub>-water nanofluids was obtained using Hamilton and Crosser static analysis method from Eq. (2). The plotted results show the thermal conductivity linearly increases as the particle volume fraction increases as seen in Fig.2. The difference in thermal conductivity enhancement increases with the increased particle volume fraction, and is as large as 12.2% at particle volume fraction of 4% for CuO-water, followed by 11.8% for SiO<sub>2</sub>-water and finally 10.0% for TiO<sub>2</sub>-water nanofluids. The highest thermal conductivity was about 0.70094 W/m.K for CuO-water, 0.69882 W/m.K for SiO<sub>2</sub>-water and 0.68722 W/m.K for TiO<sub>2</sub>-water nanofluids as shown in Table V, Table VI and Table VII for inlet velocity of 3 m/s. The increment of CuO-water nanofluid as compared to other nanofluids is due to large particle-liquid interfacial area thus exhibited higher heat transfer rate, which directly indicated the high thermal conductivity contained within the nanoparticles. The enhancement mechanism may be due to the addition of these nanoparticles to the base fluid (water) which has higher thermal conductivity compare to the base fluid led to improve thermal conductivity of the nanofluid. These results also comparable with other studies done by researchers using different nanofluids.

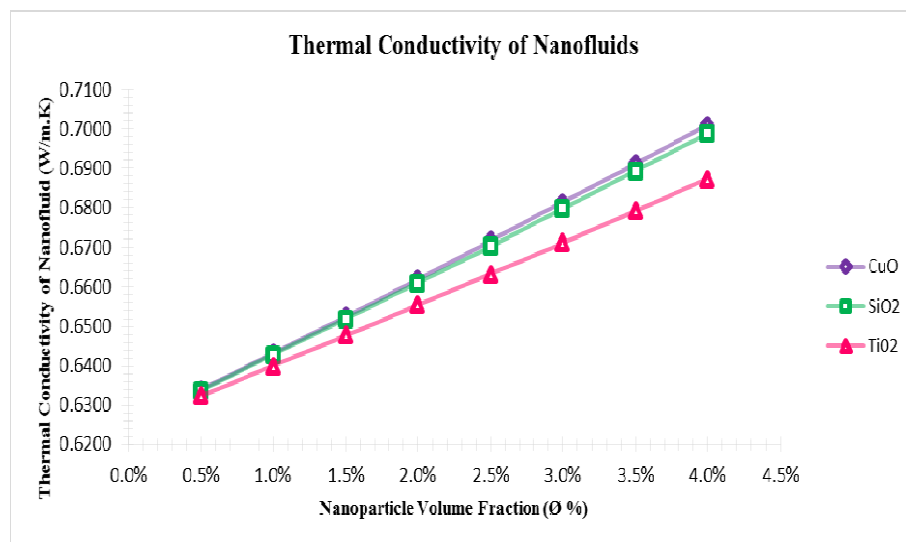


Fig 2: Increasing thermal conductivity nanofluids with particle volume fraction

#### B. Heat Transfer Coefficient of nanofluids to MCHS

The thermal performances of the nanofluids depend on the convective heat transfer coefficient. The increasing effects of the nanofluid concentrations promotes the heat transfer coefficient as shown in Fig. 3a). The heat transfer coefficient was calculated and expected to increase if thermal conductivity increases for all the coolants using Eq. (5). Fig. 3a) exhibits that there is an effective enhancement of the heat transfer coefficient in the range of Reynolds number from 400 and also with increasing of volume fraction from 0.5 to 4.0 vol.%. At the highest volume fraction, CuO-water nanofluid provided around 12.4% higher value of convective heat transfer coefficient with the comparison of the SiO<sub>2</sub>-water and TiO<sub>2</sub>-water nanofluid which found 8.22% and 7.4% respectively for the same inlet velocity of 3 m/s. Also heat transfer coefficient of CuO-water SiO<sub>2</sub>-water

and TiO<sub>2</sub>-water were 33847 W/m<sup>2</sup>.K, 32589 W/m<sup>2</sup>.K and 32328 W/m<sup>2</sup>.K, correspondingly at 4% volume percentage at the same 3.0 m/s inlet velocity. On the other hand, the heat transfer coefficient is also influenced by inlet velocity of the coolant and is a function of Reynolds number. With the increasing of nanofluid velocity, it will increase the nanoparticles movement or inter-collision which effectively enhances the heat transfer rate. So, the significant increase in Nusselt number and Reynolds numbers encourage the use of CuO-water nanofluid as a better coolant instead of a conventional coolant.

Addition of nanoparticles in liquid remarkably enhances energy transport process of the base liquid due to the chaotic movement of nanoparticles in the main flow. Increasing of volume fraction of nanoparticles will increase the density of nanofluids and lead to an increase in the mass flow rate. From Eq.(5) it shows that heat transfer coefficient increases with increasing thermal conductivity in conjunction of increasing volume fraction of nanoparticles. For example, CuO-water nanofluid at 0.5% volume fraction has mass flow rate of 0.015 kg/s and 30576 W/m<sup>2</sup>.K heat transfer coefficient with 3 m/s. At 4% volume fraction, the mass flow rate for CuO-water nanofluid is 0.017 kg/s and 33847.41 W/m<sup>2</sup>.K heat transfer coefficient with same inlet velocity as shown in Fig.3b).

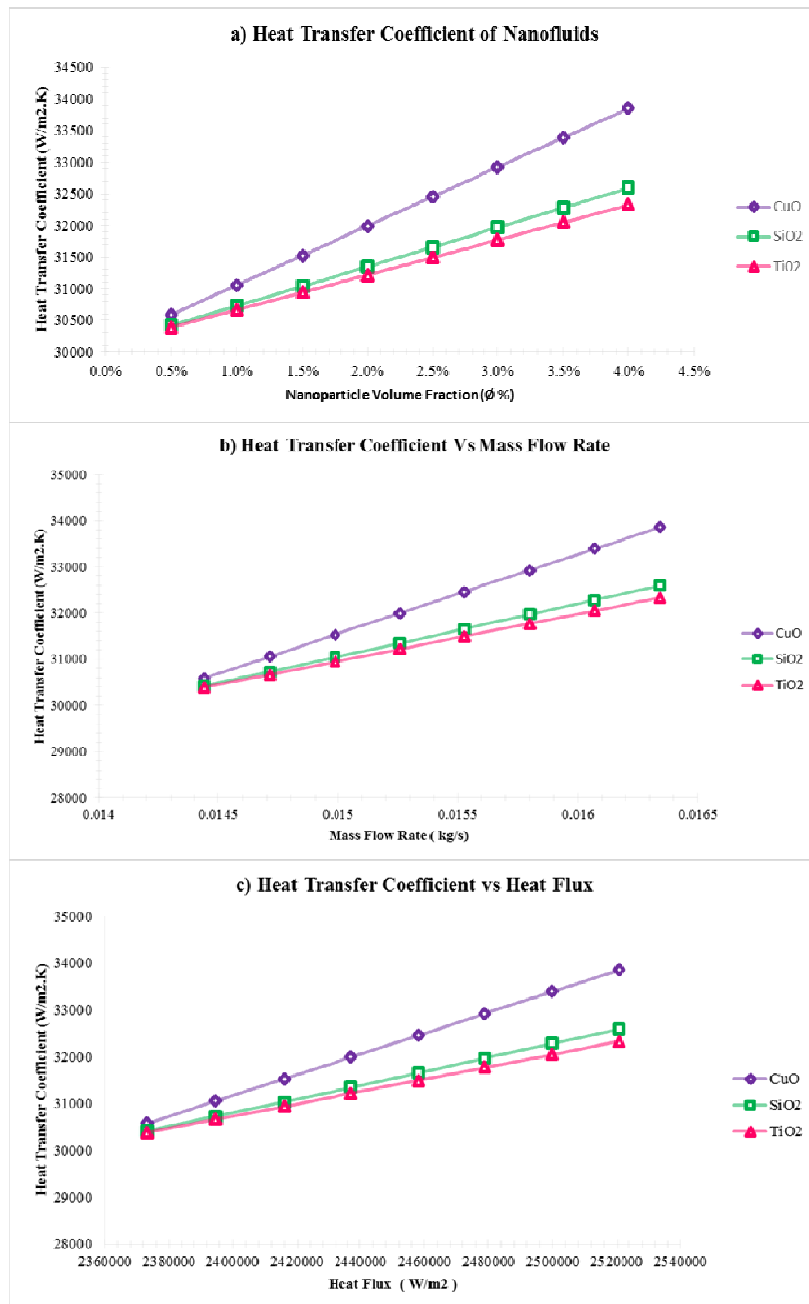


Fig 3a): Increasing heat transfer coefficient with particle volume fraction .Fig3b) Increasing heat transfer coefficient as function of mass flow rate Fig 3c) Increasing heat transfer coefficient with heat flux.



High heat flux removal is crucial for electronic devices. From this study, it was found that the heat flux increases for all types of nanofluids with increasing volume fraction of nanoparticles. The maximum enhancement of heat flux in CuO-water was found to be  $2575 \text{ kW/m}^2$ , followed by SiO<sub>2</sub>-water for  $2501 \text{ kW/m}^2$  and  $2485 \text{ kW/m}^2$  for TiO<sub>2</sub>-water nanofluid at 4% of volume fraction. Water has been used as a single-phase coolant however its thermal conductivity is lower than most metal and metal oxide. Since metal and metal oxide has higher thermal conductivity, the addition of metal or metal oxide nanoparticle in base-fluid or water can increase the thermal conductivity of the coolant which led to a decrease in thermal resistance and give high heat flux as shown in Fig.3c).

### C. Thermal Resistance of nanofluids

Thermal resistances are the functions of many parameters such as heat transfer coefficient,  $h$ , area of MCHS, mass flow rate of the nanofluid, thermal conductivity,  $k$ , and channel depth,  $H$ . By using Eq. (12) it is found that the increasing the volume concentration of the nanoparticle shows a declining trend to diminish the convective thermal resistance and thermal dispersion for all coolants and consistently can be seen at Fig. 4. Fig. 4 represents the diversity of the convective thermal resistance with the disparity of the volume concentration of the CuO-water, SiO<sub>2</sub>-water and TiO<sub>2</sub>-water nanofluid in the range of Reynolds number of 400. Thus, enhancing in convective heat transfer coefficient ultimately decreases the thermal resistance of the nanofluid.

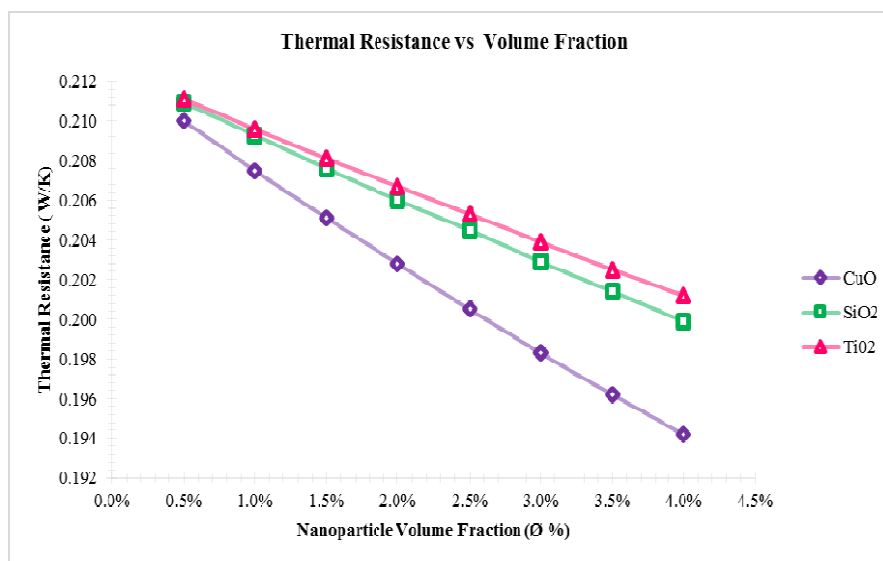


Fig.4: Decreasing thermal resistance with increasing particle volume fraction

### C. Pumping Power

Another important parameter in the MCHS operation is the pressure drop which relates to the required coolant pumping power. Parameters that influenced the pressure drop are channel width  $W_c$ , channel height  $H_c$ , bottom thickness  $W_b$ , and inlet velocity  $V_m$ . Using Eq. (20), the predicted results for pressure drops was obtained for all types of nanofluids are shown in Figure 7.

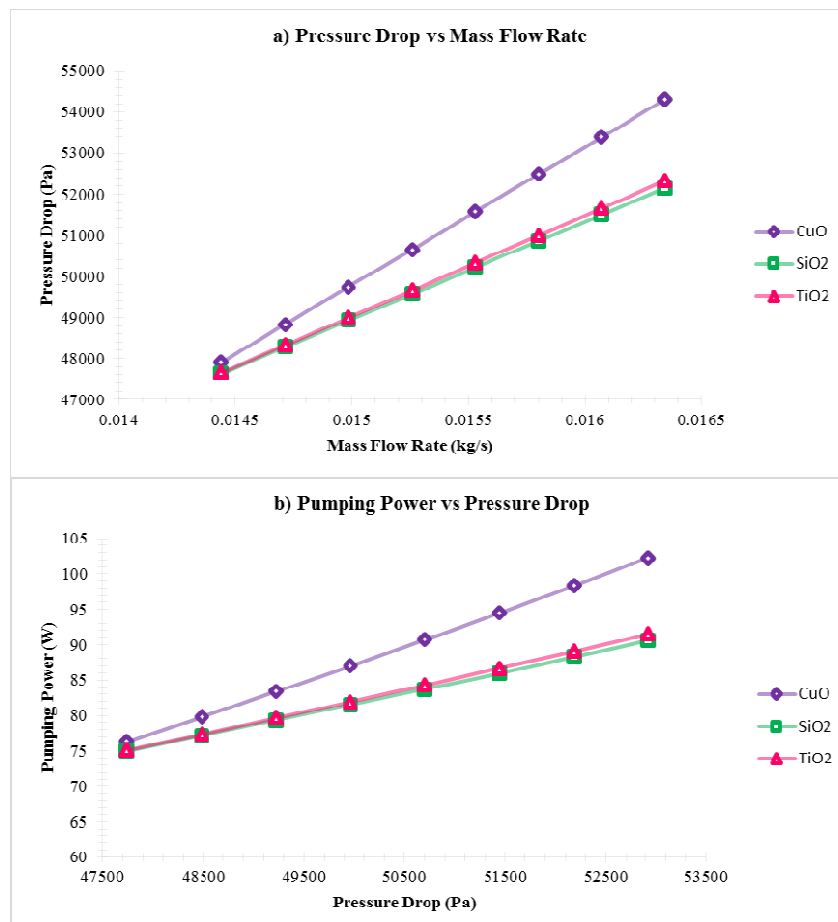


Fig.7: a): Increasing pressure drop with mass flow rate at 3m/s inlet velocity.b): Pressure drop with pumping power at 3m/s inlet velocity.

Fig.7a) demonstrates the increasing linearly in pressure drop as the mass flow rate augmented as well as the volume percentage for all types of nanofluids. From the result, it shows that CuO-water has a density of  $1021 \text{ kg/m}^3$  at 0.5% of volume fraction and 47925 Pa pressure drop. The pressure drop increases linearly with increasing volume fraction with maximum pressure drop of 54314 Pa at 4% of volume fraction at 3 m/s. Since the viscosity of nanofluid is larger than the pure fluid, a larger pressure drop is expected as well as the pumping power required in the nanofluids MCHS operation. From Eq. (18), it can be interpreted that pumping power directly proportional to pressure drop thus when pressure drop increases the pumping power increases as well. The pumping power also depends on the nanofluid concentration. Due to the higher density and viscosity of nanofluids, it has caused more power needed to drive the coolant passes through the narrow channel of the heat sink. The pumping power for CuO-water was found to be highest at 4% of volume fraction with 102.3 W at 3 m/s inlet velocity. The observed trending also applies to SiO<sub>2</sub> and TiO<sub>2</sub> where the pumping power increases from 75.0 W up to 90.6 W with increasing volume fraction and pressure drop. Fig.7b) shows the relationship between pumping power with pressure drop for all types of nanofluid. This result promotes that the pressure drop directly dependence on the higher density and viscosity of nanofluid to raise the pumping power of the nanofluid.

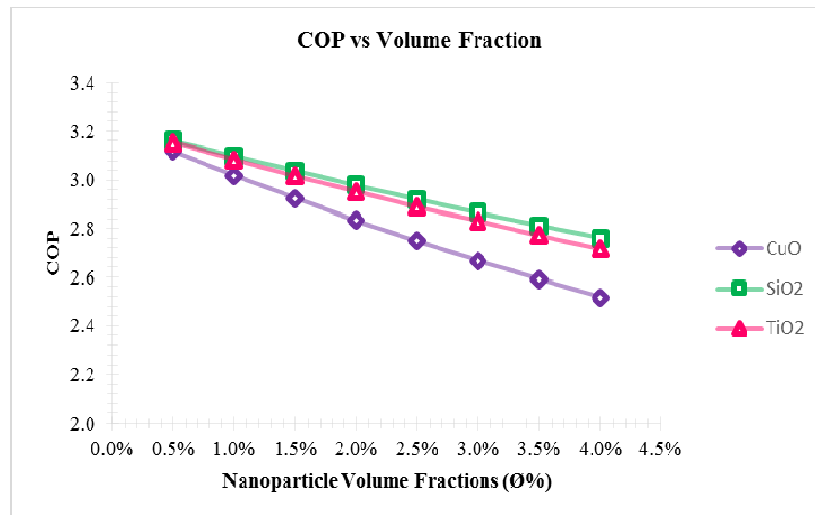


Fig.8: COP with nanoparticle volume fractions.

In order to measure efficiency of nanofluids as coolant for electronic devices, it is crucial to calculate the coefficient of performance of the MCHS. This analysis shows the impact of volume fraction to the performance of MCHS as thermophysical properties of nanofluid changed with addition of nanoparticle. Fig. 9 shows that COP decreases with the increasing volume fraction nanoparticles for all types of nanofluids.

#### IV. CONCLUSION

This study analysed the thermal performance of rectangular MCHS using copper oxide (CuO)/water, silica dioxide (SiO<sub>2</sub>)/water and titanium dioxide (TiO<sub>2</sub>)/water nanofluids at different volume fraction with constant inlet velocity. Mathematical formulation were used to compare their heat transfer capability in terms of thermal conductivity, heat transfer coefficient, thermal resistance, heat flux and required pumping power. Results confirmed that at higher particle volume fraction the CuO-water nanofluid has significant enhancement in thermal performances for electronics cooling compared to SiO<sub>2</sub>-water and TiO<sub>2</sub>-water nanofluid. Thus CuO could be proposed as potential substitutes for conventional coolants in laminar flow conditions especially in micro-scale equipments. The results revealed that there is considerable increase in conductivity and viscosity with the increase of particle volume fraction of nanofluids. For heat transfer coefficient, it was influenced by fluid velocity entrance effects and as function of Reynolds number. The pressure drop increases linearly with the increase of volume fraction and it is affected the amount of pumping power. High concentration of pumping power is required to move the coolant. The volume fraction of nanofluids, give impact to the performance of heat transfer by reducing the thermal resistance of MCHS and show a good agreement with the past experimental analysis.

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