

Thermal and Hydraulic Performances of Nanofluids Flow in Microchannel Heat Sink with Multiple Zigzag Flow Channels

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Abstract. This article presents an experimental investigation on the heat transfer performance and pressure drop characteristic of two types of nanofluids flowing through microchannel heat sink with multiple zigzag flow channel structures (MZMCHS). SiO₂ nanoparticles dispersed in DI water with concentrations of 0.3 and 0.6 vol.% were used as working fluid. MZMCHS made from copper material with dimension of 28 x 33 mm. Hydraulic diameter of MZMCHS is designed at 1 mm, 7 number of flow channels and heat transfer area is about 1,238 mm². Effects of particle concentration and flow rate on the thermal and hydraulic performances are determined and then compare with the common base fluid. The results indicated that the heat transfer coefficient of nanofluids was higher than that of the water and increased with increasing particle concentration as well as Reynolds number. For pressure drop, the particle concentrations have no significant effect on the pressure drop across the test section.

1 Introduction

A decade ago, thermal performance of the air-cooled was quite small due to limitation of the thermal performance. Later, liquid cooled system such as water or other liquids were used to instead of the air-cooled system. Because of the thermal performance of liquid-cooled system are very high compared to the air-cooled system. Now a day, the microscale and nanoscale heat transfer devices became the crucial tool in many common applications such as integrated circuits (IC), VLSI devices, laser-diode arrays, micro heat exchanger and computer chips. Thus, the heat transfer area was limited and unprecedented high load was also generated. Improving the thermal performance of the coolants and innovative cooling equipment with special geometric structures were required. Tuckerman and Pease [1] who first introduced the concept of the microchannel heat sink (MCHS) to increase the thermal performance of the cooling system. Later, Masuda and colleges [2] was first to improve the thermal performance of the coolants by using very small solid particles dispersed in the common fluids. In 1995, Choi [3] was first named the common heat transfer fluids with nanoparticle suspension that "Nanofluids". Many researchers reported that the thermal performance of nanofluids was higher than the common heat transfer fluids and little penalty drop in pressure [4]-[11]. As a result, nanofluid-cooled heat sink was estimated to be a very effective cooling system for removing high heat densities generated by the modern electronic devices.

The benefits of nanofluid-cooled heat sinks have been summarized by Daungthongsuk and Wongwises [9]-[11].

Thus, the target of the present work is to evaluate the heat transfer performance of nanofluids flowing through MCHS with multiple zigzag flow channel structures (MZMCHS). Effect of particle concentration and flow rate on the heat transfer coefficient and pressure drop are reported.

2 Experimental apparatus

The experimental system is constructed in the present study to evaluate the thermal performance and flow characteristic of nanofluids flowing through MZMCHS. Schematic diagram of the experimental apparatus used in the present work is demonstrated as shown in Fig. 1. It mainly consists of the test section, a pump with speed controller, two storage tanks, and a receiver tank. The configuration of the MZMCHS used in this work is shown in Fig. 2. Copper material is used to make the MZMCHS with dimension of 28 x 33 mm. It also consist of 7 number of flow channels, heat transfer area of 1,238 mm² and hydraulic diameter of 1 mm. The heat sink cover plate is made from acrylic material and attached to the top of the heat sink to completely seal the flow passages. To simulate heat load, a 95 W electric heater is used and is attached at the below side of the test section. Two storage tanks with capacity of 15 liters are made from stainless steel. For the storage tank No.1, it consist of a cooling coil with cooling capacity of 3.5 kW and a 2 kW electric heater. In order to adjust the fluid temperature, a thermostat is used to control the temperature at the desired value. To meet the steady state condition, the

storage tank No.2 is used to decrease the fluid temperature leaving from the MZMCHS to the temperature of the tank No. 1. It consist of a 3.5 kW cooling coil and a thermostat. To adjust the flow rate of the working fluid, a pump with speed controller is used. T-type thermocouples are installed at the both ends on the test section to measure the bulk fluid temperature. To evaluate the surface temperature of the test section, two thermocouples are placed at 5 mm and 15 mm from the

top surface of the test section for measuring the temperature gradient and then used to estimate the surface temperature. For pressure drop measurement, two digital pressure gauges are located at the inlet and exit of the test section. The receiver tank is used to measure the mass flow rate by the time taken for a given volume of the working fluid to be discharged. All instruments are calibrated to ensure the accuracy and reliability of the measured data.

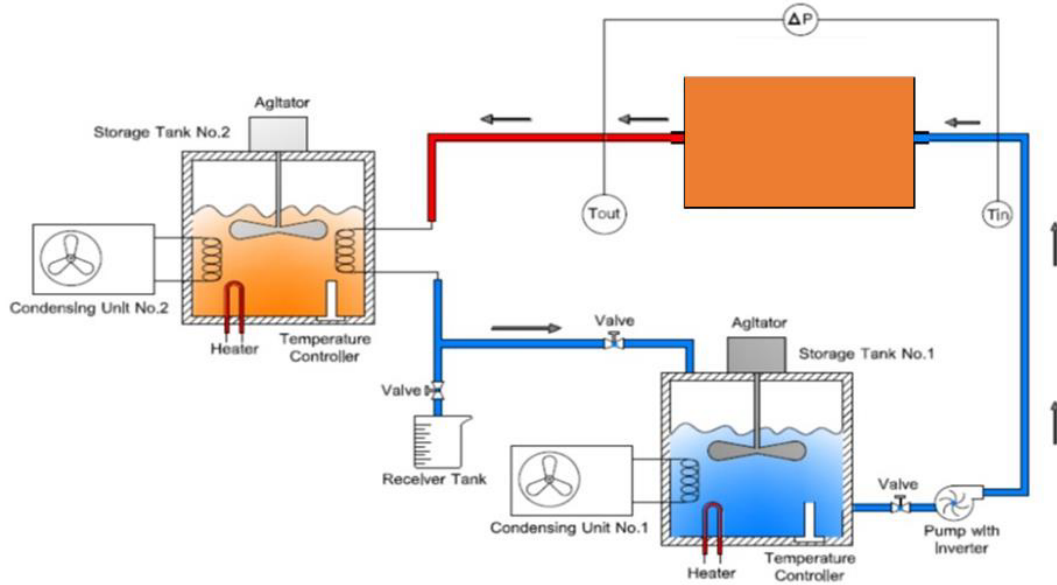


Figure 1. Schematic diagram of the experimental system

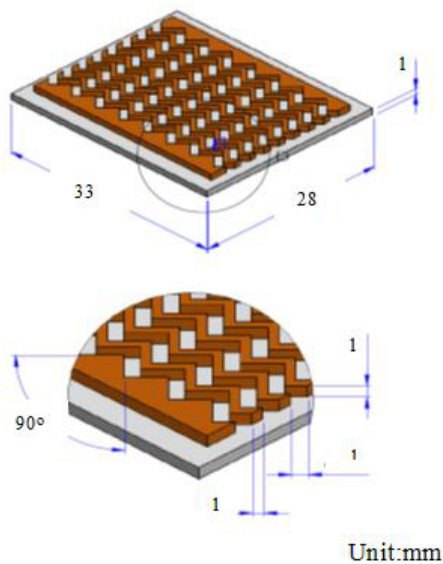


Figure 2. MZMCHS used in this work

To evaluate the thermal performance and pressure drop across the test section, the inlet and exit temperatures of the working fluid, surface temperatures, mass flow rates, and pressure drop across the test section. However, all data are measured after steady state condition reached.

3 Sample preparation

The 2-step method was used to prepare the nanofluids with the desired particle volume fractions. SiO₂ nanoparticles with an average diameter of 15 nm were used and dispersed in the DI water at particle volume fractions of 0.3 and 0.6 vol.% . Firstly, SiO₂ nanoparticles were dispersed and stirred in the DI water at the required particle concentration. Secondly, a 500 W ultrasonic vibrator was continuously sonicated the nanofluid for 2 hours in order to break down the agglomeration of the ultrafine nanoparticle. The thermophysical properties of each nanoparticle are expressed in Table 1.

Table 1. Thermophysical properties of SiO₂ nanoparticle.

| Properties | Value |
|---|-------|
| Thermal conductivity (W/m ² K) | 1.37 |
| Specific heat (kJ/kgK) | 0.742 |
| Density (kg/m ³) | 2,648 |

4 Data reduction

In the present study, SiO₂-water nanofluids with particle fractions of 0.3 and 0.6 vol.% are tested and compared with the data for water-cooled heat sink. The thermal performance can be calculated by the following equations.

The heat transfer rate of the nanofluids is computed from:

$$Q_{nf} = \dot{m}_{nf} C_{p,nf} (T_{out} - T_{in})_{nf} \quad (1)$$

in which, Q is the heat transfer rate, \dot{m} is the mass flow rate, T is the temperature and subscript nf , in and out are nanofluid, inlet and exit, respectively.

The Nusselt numbers of the nanofluids are defined as follows:

$$Nu_{nf} = \frac{h_{nf} D_H}{k_{nf}} = \frac{Q_{nf} D_H}{A_S (T_S - T_{nf}) k_{nf}} \quad (2)$$

where h is the heat transfer coefficient, k is the thermal conductivity. T_S and A_S are the surface temperature and the heat transfer area, Nu is the Nusselt number and D_H is the hydraulic diameter based on the each flow channel.

The Reynolds number based on hydraulic diameter are calculated from:

$$Re_{D_H} = \frac{\rho_{nf} u_{ch} D_H}{\mu_{nf}} = \frac{\rho_{nf} u_{ch}}{\mu_{nf}} \frac{4A_{ch}}{p_{ch}} \quad (3)$$

where ρ is the density, μ is the viscosity, u_{ch} and p_{ch} are the mean velocity and the perimeter of the each flow channel and A_{ch} is the cross-sectional area of the each flow channel.

The Pak and Cho [12] correlations are used to calculate the specific heat and density of the nanofluids which are expressed as follows:

$$Cp_{nf} = \phi Cp_p + (1 - \phi) Cp_w \quad (4)$$

$$\rho_{nf} = \phi \rho_p + (1 - \phi) \rho_w \quad (5)$$

where Cp is the specific heat, ϕ is the particle volume fraction, and subscript nf , p and w are nanofluid, nanoparticles and water, respectively.

The Hamilton and Crosser model [13] and Einstein model [14], are used to calculate the thermal conductivity (k) and viscosity (μ) of nanofluids which are expressed as follows:

For thermal conductivity;

$$k_{nf} = \left[\frac{k_p + (n-1)k_w - (n-1)\phi(k_w - k_p)}{k_p + (n-1)k_w + \phi(k_w - k_p)} \right] k_w \quad (6)$$

$$n = 3/\psi \quad (7)$$

For viscosity of the nanofluids;

$$\mu_{nf} = (1 + 2.5\phi)\mu_w \quad (8)$$

where ψ is the sphericity which is defined as the ratio of the surface area of a sphere to the surface area of the particle. The sphericity is 1 and 0.5 for the spherical and cylindrical shapes, respectively. n is the empirical shape factor.

5 Results and discussion

The variation of the surface temperature with heat load as a function of particle concentration was shown in Fig. 3. The measured data indicated that the surface temperatures

of MZMCHS increased with increasing heat load and decrease as particle concentration increased. This mean that nanofluid-cooled MCHS gave better thermal performance than that of the water-cooled MCHS which led to increase in the heat removal rate from the system. This is due to the fact the nanofluid have a large thermal property compared to the conventional pure water. As a result, the surface temperature of the test section was lower than usual.

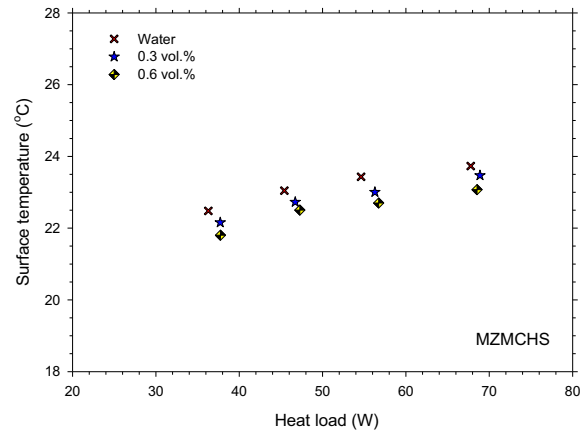


Figure 3. Variation of the surface temperature of heat sink as a function of heat load

Comparison of the measured Nusselt number (Nu) between nanofluids-cooled and water-cooled MCHS was shown in Fig. 4. The measured data illustrated that Nu increases with increasing Reynolds number as well as particle concentrations. Moreover, it evidently seen that the nanofluid-cooled heat sink gave higher heat transfer performance than that of the water-cooled heat sink about 8%. This means that the use of nanofluids as coolant gave better transfer performance than that of the common pure water and can be suitable for used in the case of high heat load dissipation

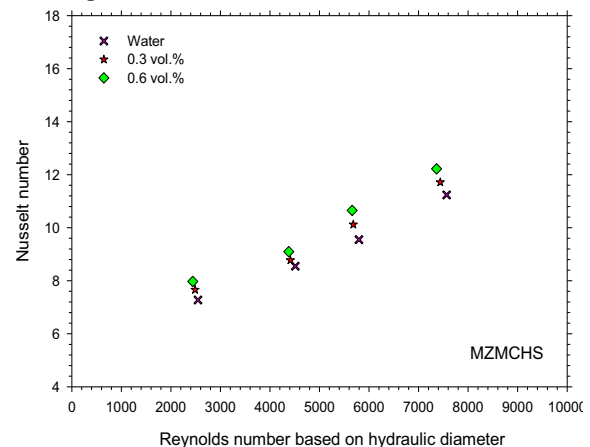


Figure 4 Comparison of the Nusselt number between nanofluids-cooled and water-cooled heat sink

Fig. 5 shows the comparisons between the pressure drops across the MZMCHS obtained from the nanofluids and common pure water. The results indicated that the pressure drop across the test section increases with Reynolds number increased. Furthermore, it clearly seen that the use of nanofluids had no significant effect on the pressure drop. Thus, it may concluded that nanofluids

have high potential to enhance the thermal performance of the cooling system without penalty drop in pressure.

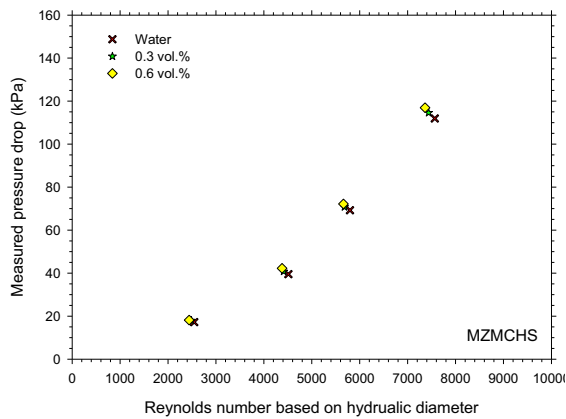


Figure 5. Measured pressure drop as a function of particle concentration and Reynolds number

6 Conclusions

The heat transfer performance and pressure drop characteristic of nanofluids-cooled heat sink were experimentally studied. Microchannel heat sink with multiple zigzag flow channel (MZMCHS) was used. SiO₂-water nanofluids with particle volume concentration of 0.3 and 0.6 vol.% were used as coolant and then compared to the common water. The effect of particle concentrations and the Reynolds number on the Nusselt number and pressure drop were examined. Important conclusions have been described as follows:

- The Nusselt number increased with increasing particle concentration as well as Reynolds number. And, nanofluids-cooled heat sink gave larger thermal performance than that of the water-cooled heat sink by approximately 8% (average at particle concentration of 0.6 vol.%).

- The use of nanofluids as coolant had no significant effect on the pressure drop.

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