Heat transfer enhancement and flow characteristic of Al_2O_3-water nanofluids flowing through a microchannel heat sink

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Abstract

The research presents an experimental study on the heat transfer and pressure drop characteristics of Al_2O_3-water nanofluids flowing through a microchannel heat sink (MCHS). The effects of Reynolds number and particle concentrations on the heat transfer and flow behavior are investigated. Comparison of the heat transfer coefficient obtained from water-cooled MCHS and nanofluids-cooled MCHS are also presented. MCHS with rectangular flow channel made from aluminum with dimension of 5x5 mm are used as the test section. Al_2O_3-water nanofluids with particle concentrations of 1.0, 2.0 and 3.0 wt.% are tested. Two electric heaters with each capacity of 50 W are used to supply heat to the test section. The experimental conditions are described as follows: 1) fluid temperature is setted at 15 ^{o}C and 2) Reynolds number ranging between 1500 and 3000. The results indicated that the heat transfer performance of MCHS increased with increasing Reynolds number as well as particle concentrations. Compared with pure water, the result indicated that heat transfer performances of nanofluid-cooled MCHS are higher than those of water by about 7 – 15%. Finally, the pressure drop of nanofluids is close to the water.

Keywords: heat transfer coefficient, Pressure drop, nanofluid, microchannel heat sink

1. Introduction

The concept of nanofluid is to disperse some solid particles with nanometer size in conventional heat transfer fluids such as water, oil and ethylene glycol. This concept was carried out by Masuda et al. [1] who studied the heat transfer performance of liquids with solid nanoparticles suspension in 1993. However, the term of "Nanofluid" was first introduced by S.U.S Choi [2] in 1995, and successively gained popularity. A number of researchers reported that nanofluids gave higher heat transfer performance than that of common heat transfer fluids [3-18]. Similarly, based on advanced electronic devices, these devices become smaller and are employing high speed and high power density such as computers, power electronics, car engines, and high-powered lasers or x-rays. Normally, these devices generate the unprecedented of high load whereas their surface area for heat removal is limited. Thus, proper cooling technologies for advanced electronic devices are necessary required and it is challenging task for a number of researchers. As mentioned above, it is the concept of microchannel heat sink (MCHS) for dissipating large amounts of heat from these devices in order to maintain the required performance and reliability.

Two decades ago, pioneer researchers Tuckerman and Pease [19] reported the heat transfer performance of microchannel heat sink (MCHS) for cooling the very-large-scale integrated (VLSI) circuit. However, single-phase liquid is used as working fluid for cooling the microchannel heat sinks and shows higher potential for heat removal than air cooling system. In general, there are three approaches for increasing the cooling performance of advanced electronic devices with high level of heat generation. The first is to find an optimum geometry of cooling devices in which the cooling performance is maximized. The second is to reduce the channel diameter for enhancing the heat transfer coefficient which is reported by Tuckerman and Pease [19]. The last is to improve the heat transfer performance of coolants. Thus, the use of MCHS combined with nanofluid as coolant is an innovative idea for cooling small electronic devices. So far, some existing published articles which associate with this idea are described in the following sections.

Lee and Choi [20] investigated the thermal performance of MCHS with NF_2 and NF_3 nanofluid compared with pure water and liquid nitrogen, theoretically. Their results showed that thermal resistances of nanofluid were lower than that of a pure water and liquid nitrogen, respectively. On the contrary, cooling rate and power density of nanofluid were much larger than those of pure water and liquid nitrogen.

Chein and Huang [21] proposed a mathematical model for predicting the heat transfer performance and pumping power of $MCHS_S$ using nanofluids as working fluid. Cu nanoparticle dispersed in water with various particle concentrations were used as testing fluids. Similarly, two specific geometries of MCHS were tested. Their results indicated that addition of nanoparticles in the base liquid remarkably enhanced the heat transfer performance of the base liquid and no penalty in pressure drop.

Koo and Kleinstreuer [22] presented a numerical study on the heat transfer and flow characteristics of MCHS with CuO nanoparticles dispersed in water and ethylene glycol, respectively. Their simulation data showed that heat transfer performance of ethylene glycol was higher than that of water. Moreover, they also suggested that nanoparticles with high thermal conductivity and

MCHS with high aspect ratio of flow channel should be used.

Jang and Choi [23] presented the cooling performance of a MCHS by using Cu-water and diamond-water nanofluids as coolant, numerically. The results showed that the diamond–water nanofluid with particle concentration of 1.0 vol.% gave a higher heat transfer performance than the base fluid by about 10%. Moreover, their results indicated that use of nanofluids can reduce both the thermal resistance and temperature difference between the heated surface of MCHS and the working fluid. They also demonstrated that nanofluids-cooled MCHS is the next generation cooling system for cooling the ultra-high heat flux devices.

Abbassi and Aghanajafi [24] investigated the heat transfer performance of MCHS using Cu-water nanofluid as coolant, numerically. The thermal dispersion model was adopted for heat transfer analysis and thermal dispersion coefficient was considered. Their data showed that nanofluid gave larger heat transfer enhancement than those of base fluid and this enhancement increased with increasing particle concentration as well as Reynolds number.

Chein and Chuang [25] presented the thermal performance of MCHS using CuO-water nanofluids as coolants. Particle concentrations of 0.2 and 0.4 vol.% were tested in their study. The results showed that the presence of nanoparticles have higher energy absorption than that of pure water at a low flow rate. In contrast, there is no contribution from heat absorption when the flow rate is high.

Tsai and Chein [26] analytically investigated on the MCHS performance with nanofluids as coolant. Cu-water and CNT-water nanofluids were used as the working fluid. The porous medium model was used to simulate the MCHS performance. Their simulation results demonstrated that the use of nanofluids can reduce the temperature difference between MCHS surface and fluid temperature compared with that of pure water. Moreover, at the porosity and aspect ratio less than the optimum porosity and aspect ratio, it was clearly seen that heat transfer performance of nanofluid-cooled MCHS was larger than that of base fluid. In contrast, for the case of the porosity and channel aspect ratio greater than optimum porosity and aspect ratio, use of nanofluid did not create a significant change in thermal resistance of the MCHS.

Jung and colleague [27] reported an experimental investigation on the heat transfer coefficient and friction factor of Al_2O_3-water nanofluids flowing through MCHS with rectangular channel under laminar flow condition. Compared with pure water, the results indicated that the heat transfer coefficient of nanofluid was about 32% higher than that of water without penalty in pressure drop. This enhancement increased with increasing Reynolds number and particle concentration.

Ghazvini and Shokouhmand [28] determined the heat transfer performance of a MCHS using CuO-water nanofluid as coolant, both analytically and numerically. The fin model and the porous media approach were used to analyze the thermal behavior of MCHS. The results showed that the overall heat transfer coefficient of nanofluid was larger than that of pure water and increased with increasing Reynolds number. The porous media approach gave higher heat transfer coefficient ratio than the fin model. Moreover, The dimensionless temperature increased with an increasing of porosity of the MCHS.

Ho et al. [29] investigated the forced convective cooling performance of Al_2O_3-water nanofluid flowing through MCHS under laminar flow regime, experimentally. MCHS made from copper consists of 25 parallel rectangular microchannels with length of 50 mm was used as the test section. The results showed that the nanofluids have much higher average heat transfer coefficient than those of pure water. Moreover, their results also indicated that nanofluids gave lower thermal resistant and wall temperature than those of base fluid. Finally, use of nanofluid has little penalty in pressure drop.

Ebrahimi et al. [30] numerically investigated on the cooling performance of a nanofluids-cooled MCHS. Carbon nanotubes (CNTs) dispersed in water was used as coolant. The results showed that thermal conductivity of nanofluid increase with increasing nano-layer thickness which leads to decrease in temperature gradient in MCHS.

As aforementioned, the above literatures have focused on the heat transfer performance of nanofluid-cooled MCHS both experimentally and numerically. Their results showed that nanofluid-cooled MCHS gave higher heat transfer potential than that of the common base liquid. Moreover, many researchers recommended that MCHS using nanofluids as coolant are expected to be the next generation of the novel electronic cooling technology. Thus, the authors would like to address the heat transfer performance of MCHS with use of nanofluids as coolant for dissipating large amounts of heat from a MCHS. Al_2O_3-water nanofluids with particle concentrations of 1 and 3 wt.% are used as working fluids and flowing through MCHS with rectangular flow channel.

2. Sample preparation

In the present study, nanofluids provided by a commercial source (DEGUSSA, Aerodisp. W630) were used as working fluid. This mixture was composed of Al_2O_3 nanoparticles with an average diameter of 120 nm dispersed in water. The original particle concentration was 30 wt%. In order to produce other required particle volume fractions, dilution with water followed by a stirring action was effected. Moreover, an ultrasonic vibrator was used to sonicate the solution continuously for about two hours in order to break down agglomeration of the nanoparticles. The desired concentrations used in this study were 1.0%, 2.0%, and 3.0 wt.%.

3. Experimental apparatus and procedure

Fig. 1 shows schematic diagram of the experimental system used in the present study. It mainly consists of four main parts as follows: a test section (MCHS with rectangular channel configuration), a micro pump with an inverter, two storage tanks, and a receiver tank. A 50x50 mm MCHS with 25 rectangular flow channels is made from aluminum. Focused on the MCHS, the width and height of the channel are 1 mm and 1 mm, respectively. Similarly, hydraulic diameter and heat transfer area of MCHS are 1 mm and 0.003 m^2, respectively. Plastic tubes were placed at both ends of the MCHS for reducing the heat loss along the respectively. Similarly, 3 positions of wall temperatures measurement were carried out using T-type thermocouples. Two storage tanks with 20 L capacity were made from acrylic plate and used for adjusting the nanofluid temperature and reduce the nanofluid temperature leaving from the test section, respectively. The storage tank No.1 with a 9,000 Btu/hr cooling capacity, 2 kW heaters and a thermostat was used to control the temperature of

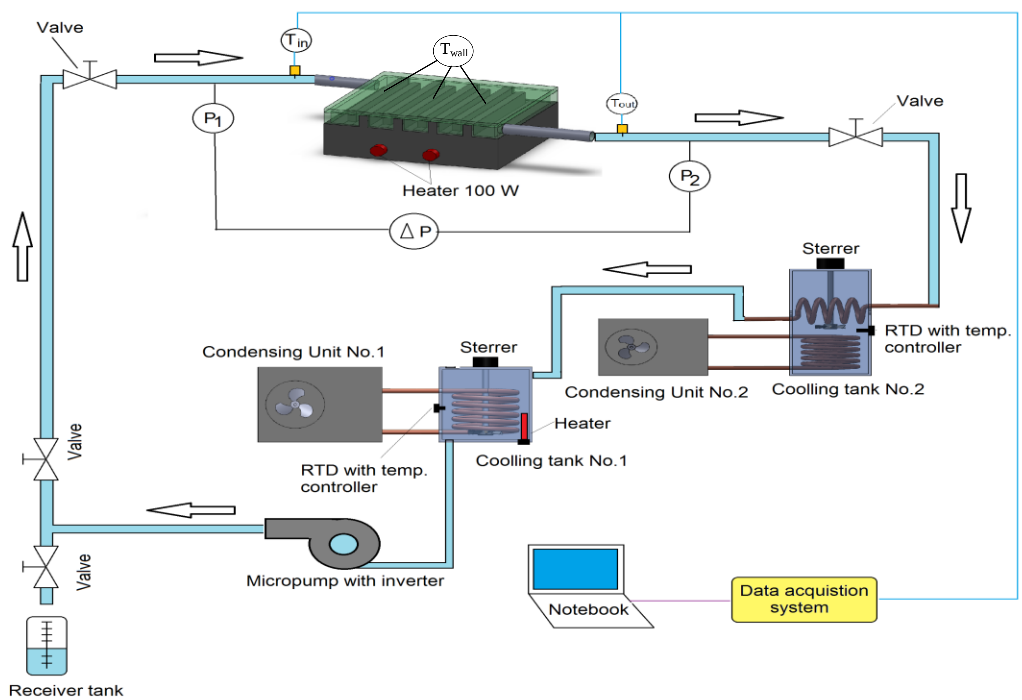

Fig. 1 Schematic diagram of the experimental apparatus

axial direction. Similarly, acrylic plate was placed on the top of MCHS to prevent heat loss. Two 50 W heaters were used to supply heat load to the MCHS. The pressure drop and the bulk temperature of the nanofluid at inlet and exit of the test section were measured by using the differential pressure transmitter and T-type thermocouples, nanofluid constant. Similarly, the storage tank No. 2 with a 9,000 Btu/hr cooling capacity and a thermostat was used to cool down the nanofluid temperature leaving from the test section to the setting temperature of tank No.1. The speed of the micro pump was controlled by using an inverter for adjusting the flow rate of nanofluids. The receiver tank was used to measure the nanofluid flow rate by the time taken for a given volume of nanofluid to be discharged.

The differential pressure transmitter was calibrated using an air operated dead weight tester. The uncertainty of the pressure measurement is $\pm$ 0.030 kPa. The nanofluid flow rates were determined by electronic balance. The uncertainty of the electronic balance was $\pm$ 0.001 kg. A standard thermometer was used to calibrate all of the T-type thermocouples with a maximum precision of 0.1 °C. Therefore, the uncertainty of the measured Nusselt number was around 5%.

During the test run, the pressure difference in the nanofluid, wall temperatures of the MCHS, mass flow rates of the nanofluids, and the inlet and exit temperatures of the nanofluids were recorded.

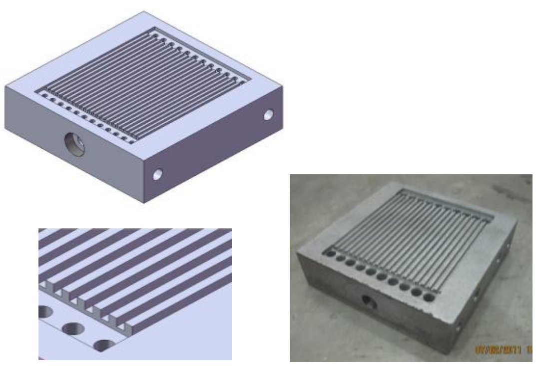

Fig. 2 MCHS configuration used in the this study

4. Data reduction

In the present study, Al_2O_3-water nanofluids with particle fractions of 1.0%, 2.0%, and 3.0 wt.% are used to evaluate the heat transfer performance of MCHS using nanofluids as coolant. Thus, the heat transfer performance of nanofluids can be computed from the following equation.

The heat transfer rate into the nanofluid is computed from:

$$Q_{nf} = \dot{m}_{nf} Cp_{nf} (T_{out} - T_{in})_{nf} \qquad (1)$$

where Q_{nf} is the heat transfer rate of the nanofluid, $\dot{m}_{nf}$ is the mass flow rate of the nanofluid and T_{in} and T_{out} are the temperature of the nanofluid at inlet and exit of the test section.

The heat transfer coefficient and the Nusselt numbers of the nanofluid are evaluated from the following equations.

$$h_{nf} = \frac{Q_{nf}}{A_S (T_S - T_{nf})} \qquad (2)$$

$$Nu_{nf} = \frac{h_{nf} D_H}{k_{nf}} \qquad (3)$$

where h_{nf} is the heat transfer coefficient of the nanofluid, T_S is the average temperature of the heated surface of MCHS, T_{nf} is the bulk temperature of the nanofluid which is the average fluid temperature across the test section, Nu_{nf} is the Nusselt number of the nanofluid, D_H is the Hydraulic diameter of the channel and k_{nf} is the thermal conductivity of the nanofluid.

Moreover, the Reynolds number can be calculated by using the following equation:

$$\mathrm{Re} = \frac{\rho_{nf} u_m D_H}{\mu_{nf}} \qquad (4)$$

where ρ_{nf} is the density of the nanofluid, u_m is the mean velocity of the nanofluid and μ_{nf} is the viscosity of the nanofluid.

The density and specific heat of the nanofluids presented in the above equation are computed by use of the Pak and Cho [6] correlations, which are expressed as follows:

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

and

$$Cp_{nf} = \phi Cp_p + (1-\phi)Cp_w \qquad (6)$$

where Cp_{nf} is the specific heat of the nanofluid, Cp_p is the specific heat of the nanoparticles, Cp_w is the specific heat of the base fluid and ϕ is the particle volume fraction of the nanoparticles.

Similarly, the thermal conductivity and viscosity of nanofluids are calculated from Hamilton and Crosser model [31] and Einstein equation suggested by Drew and Passman [32] which are expressed as follows:

For thermal conductivity of the nanofluid;

$$k_{nf} = \left[\frac{k_p + (n-1)k_w - (n-1)\phi(k_w - k_p)}{k_p + (n-1)k_w + \phi(k_w - k_p)}\right]k_w \qquad (7)$$

$$n = 3/\psi \qquad (8)$$

where n is the empirical shape factor and ψ is the sphericity, defined as the ratio of the surface area of a sphere (with the same volume as the given particle) to the surface area of the particle. The sphericity is 1 and 0.5 for the spherical and cylindrical shapes, respectively. Moreover, k_p is the thermal conductivity of the nanoparticles and k_w is the thermal conductivity of the base fluid.

At temperature of 15 ^{o}C, the thermal conductivity of water is 0.589 W/mK. Similarly, the thermal conductivity of the nanofluids with particle concentrations of 1.0, 2.0 and 3.0 wt.% are 0.594, 0.599 and 0.604 W/mK, respectively.

For viscosity of the nanofluid;

$$\mu_{nf} = (1+2.5\phi)\mu_w \qquad (9)$$

where μ_w is the viscosity of the base fluid.

The thermophysical properties of the nanofluid shown in the above equations are calculated from water and nanoparticles at average bulk temperature.

For the case of pressure drop, the measured pressure drop of nanofluid-cooled MCHS are compared with the water-cooled MCHS.

5. Results and discussion

In the present study, the experimental data are divided into two groups: 1) Heat transfer performance and 2) Pressure drop. The detail results are shown in the following subsections.

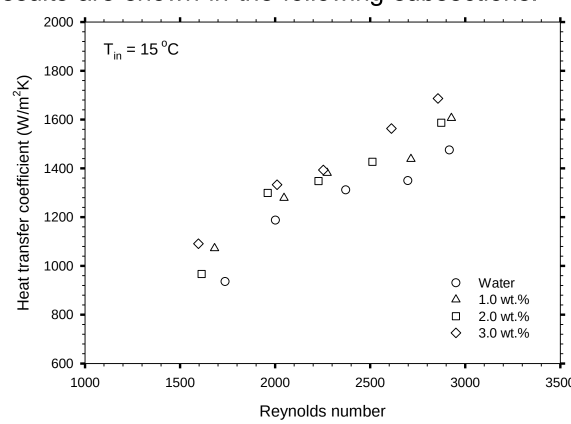

Fig. 3 Experimental heat transfer coefficient for water and Al_2O_3-water nanofluids

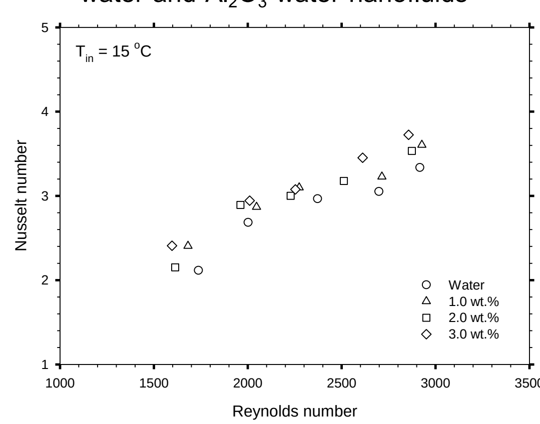

Fig. 4 Measured Nusselt number for water and Al_2O_3-water nanofluids versus Reynolds number

As shown in Figs. 2 and 3, the heat transfer coefficient and the Nusselt number of the nanofluids are higher than those of the base liquid, and increase with increasing the Reynolds number as well as the particle concentrations. The enhancement range between 7 to 15% is obtained. This behaviour due to the nanoparticles presented in the base liquid increase the thermal conductivity, which leads to an increase in the heat transfer performance.

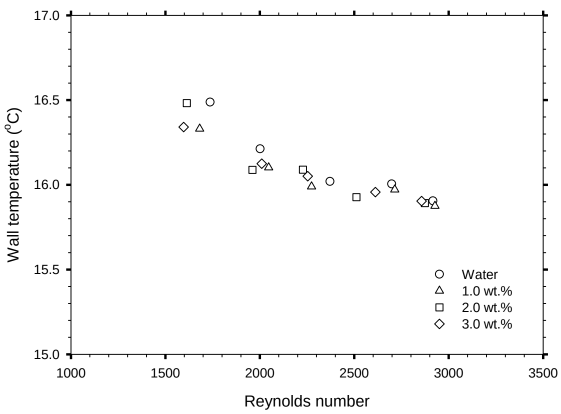

Fig. 5 Effect of particle concentrations on the wall temperature of MCHS

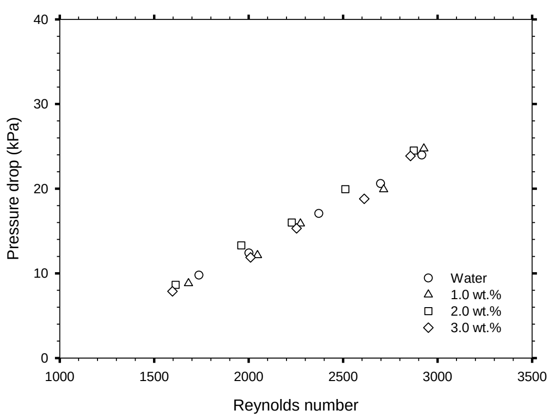

Fig. 6 Comparison of pressure drop obtained from water and that from the Al_2O_3-water nanofluids at different particle concentrations

Fig. 5 shows the wall temperature of MCHS as a function of Reynolds number and the particle concentrations. The figure shows that the wall temperature of MCHS decreases with increasing Reynolds number as well as particle concentrations especially at lower Reynolds number. This is due to the fact that the addition of nanoparticles in the base fluid increases the energy exchange process which leads to increase in the heat transfer performance. At lower Reynolds number, higher wall temperature difference between water-cooled MCHS and nanofluid-cooled MCHS are obtained. This may be caused by heat exchange period for lower Reynolds number is longer than that for higher Reynolds number.

As shown in Fig. 6, the results indicate that the pressure drop of the nanofluids increases with increasing Reynolds number and that there is a small increase with increasing particle concentrations. This means that the use of nanofluids will not cause a penalty in pressure drop. This is due to small particle size and very low particle concentrations. This is one of the benefits of nanofluids to use them as heat transfer fluid in practical applications.

6. Conclusions

The convective heat transfer performance and pressure drop characteristics of MCHS using Al_2O_3-water nanofluids as coolant were experimentally investigated. The particle concentrations of 1 and 3 wt.% were tested. The effect of particle concentrations and the Reynolds number on the heat transfer performance and pressure drop of nanofluids-cooled MCHS were examined and then compared with the data for water-cooled MCHS. Important conclusions have been obtained and summarised as follows:

- The use of nanofluid-cooled MCHS gives significantly higher heat transfer coefficients than those of the water-cooled MCHS by about 7 - 15%.
- The wall temperature of MCHS decreases with increasing Reynolds number as well as particle concentrations. For the case of nanofluid-cooled MCHS, the wall temperatures are lower than those of water-cooled MCHS, especially at lower Reynolds number.
- The pressure drop of water-cooled and nanofluids-cooled MCHS increases with increasing Reynolds number and there is a small increase with increasing particle concentrations.

7. Acknowledgment

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8. Nomenclature

C_p	=	specific heat, J/kgK
D_H	=	Hydraulic diameter, m
f	=	friction factor
h	=	heat transfer coefficient, W/m^2K
k	=	thermal conductivity, W/mK
L	=	length of the test tube, m
$\dot{m}$	=	mass flow rate, kg/s
Nu	=	Nusselt number
ΔP	=	Pressure drop, Pa
Q	=	heat transfer rate, W
Re	=	Reynolds number
T	=	temperature, ^{o}C
u	=	velocity, m/s

Greek symbols

ϕ	=	volume fraction
ρ	=	density, kg/m^3
μ	=	viscosity, kg/ms

Subscript

f	=	fluid
in	=	inlet
m	=	mean
out	=	outlet
p	=	particles
nf	=	nanofluid
s	=	surface

9. References

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