Numerical and Experimental Investigations on Heat Transfer of Aluminum Microchannel Heat Sinks with Different Channel Depths

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***Abstract*—In this study, heat transfer of aluminum microchannel heat sinks (MCHs) was investigated with both numerical and experimental methods. Five MCHs, each with twelve channels, were designed with the channel width of 500 µm, channel length of 33 mm, and channel depths varying from 200 µm to 900 µm. Water was used as the working fluid and Reynolds numbers, as independent variables, were in the range of 100 to 1000. For all cases done in this study, it is found that the heat transfer of microchannel heat sinks was significantly affected by the channel depth. At mass flow rate of 213 g/min, when the channel depths increased from 200 µm to 900 µm, the heat fluxes decreased from 31.8 W/cm^2 to 15.8 W/cm^2 and the heat transfer rate increased from 113.3 W to 143.8 W. Good agreement between numerical and experimental results was achieved, with maximum percentage errors less than 6%.**

***Index Terms*—microchannel heat sink, channel depth, heat transfer rate, heat flux**

I. Introduction

The needs of heat transfer devices for diverse applications and the fascinating characteristics observed in heat transfer field have attracted many scientists and engineers to investigate the heat transfer and flow in microchannel heat sinks. The inlet/outlet rectangular shape was designed by Lelea [1], and six different types of inlets/outlets were investigated numerically by Chein and Chen [2]. Lee *et al*. [3] studied the effects of channels' depth and width in rectangular copper microchannels with Reynolds numbers ranging from 300 to 3,500. Dang *et al.* [4] presented the investigations of heat transfer phenomena of an aluminum microchannel heat sink, and Tran et al. [5] presented studies on pressure drop and performance index of an aluminum microchannel heat sink. Wang *et al*. [6] presented an inverse geometric optimization for nano-cooled microchannel heat sink. In their research, water-based Al_2O_3 nanofluid with 1% particle volume fraction was used as the working fluid. Comparison of simulation data between CFD and ANSYS Fluent regarding a direct bond copper for power electronics packaging was discussed by Yin *et al*. [7]. Their research emphasized the importance of entrance effects on heat transfer and flow in the microchannel heat sink. With the foundation of the previous studies, the purpose of this study is to investigate experimentally and numerically the effects of channel depths on the heat transfer of aluminum microchannel heat sinks.

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II. Methodology

A. Mathematical Model

Given by the CFD–ACE$^+$ package, the governing equations for this system consist of the incompressible Navier-Stokes equations for the motion of fluid and the energy equation for the transfer of heat [8]. The incompressible Navier-Stokes equations can be expressed by

$$\rho\frac{\partial \mathbf{u}}{\partial t}+\rho(\mathbf{u}\cdot\nabla)\mathbf{u}=\nabla\cdot[-p\mathbf{I}+\mu(\nabla\mathbf{u}+(\nabla\mathbf{u})^T)]+\mathbf{F} \quad (1)$$

and

$$\nabla\cdot\mathbf{u}=0 \quad (2)$$

For steady-state conditions, $\partial\mathbf{u}/\partial t=0$; the boundary conditions of inlet flow are $\mathrm{u}=0$, $\mathrm{v}=0$, and $\mathrm{w}=w_0$; the boundary conditions of outlet flow are $\mu(\nabla\mathbf{u}+(\nabla\mathbf{u})^T)\boldsymbol{n}=0$, and $p=p_0$; where μ is dynamic viscosity, ρ is density, $\mathbf{u}$ is velocity field, u is velocity in the x-direction, v is velocity in the y-direction, w is velocity in the z-direction, p is pressure, **I** is the unit diagonal matrix, $\boldsymbol{n}$ is normal vector, and **F** is body force per unit volume ($\mathrm{F}_x=\mathrm{F}_y=\mathrm{F}_z=0$).

For the energy transport, no-slip conditions are assumed for velocity and temperature at the walls; these conditions are expressed by $\mathrm{u}_{wall}=0$ and $\mathrm{T}_{wall}=\mathrm{T}_{fluid\ at\ wall}$, respectively, where T_{wall} is wall temperature.

The heat transfer equation for the energy transport by the fluid is:

$$\rho Cp\frac{\partial T}{\partial t}+\nabla\cdot(-\lambda\nabla\mathrm{T})=Q_i-\rho Cp\boldsymbol{u}\cdot\nabla\mathrm{T}. \quad (3)$$

where Q_i is internal heat generation, T is temperature, Cp is specific heat at constant pressure, and λ is thermal conductivity. For steady-state conditions, $\partial T/\partial t=0$; the

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boundary condition of inlet flow is $T = T_0$; the boundary condition of outlet flow is convective flux, expressed by $\mathbf{n} \cdot (-\lambda\nabla T) = 0$; the thermal boundary conditions of the top wall and four side-walls of the microchannel heat sink are assumed to be constant heat flux, expressed by $-\mathbf{n} \cdot (-\lambda\nabla T) = q_0$; and the thermal boundary condition of the bottom wall is constant surface temperature $T = T_s$. The energy balance equation for the microchannel heat sinks is expressed by

$$Q = mCp(T_o - T_i). \tag{4}$$

where Q is heat transfer rate, m is mass flow rate, Cp is specific heat at constant pressure, T_i, and T_o are inlet and outlet temperatures, respectively.

Heat flux is calculated by

$$q = \frac{Q}{A}, \text{ or } q = k\Delta T_{lm} = \frac{\Delta T_{lm}}{\Sigma R}. \tag{5}$$

The log mean temperature difference is calculated by

$$\Delta T_{lm} = \frac{\Delta T_{max} - \Delta T_{min}}{ln\frac{\Delta T_{max}}{\Delta T_{min}}},\ \Delta T_{max} = T_s - T_i,$$

$$\text{and}\quad \Delta T_{min} = T_s - T_o. \tag{6}$$

where q is heat flux, A is heat transfer area, k is overall heat transfer coefficient, ΣR overall thermal resistance, T_s is surface temperature of the bottom wall.

Nusselt number is calculated by the following equation [2].

$$Nu_{Re} = \frac{h_{Re}D_h}{\lambda_f}, \tag{7}$$

where Nu_{Re} is the Nusselt number achieved by varying Reynolds number, h_{Re} is the heat transfer coefficient of a microchannel heat sink depending on the Reynolds number, and λ_f is the thermal conductivity of flow in the channels.

B. *Design and Set-up of the Test Facility*

The experimental system used in this study was designed with three major parts, including the test section, syringe system, and overall testing loop, as shown in Fig. 1. For verifying the numerical results, two aluminum substrates, with the same dimensions as shown in Fig. 2 except the differences in channel depths of 350 µm and 700 µm respectively for SD2 and SD4, were manufactured as the test specimens for the experiments. The substrates and its upper plates (top cover or PMMA plate) which are showed in Fig. 3 were manufactured by precision micromachining and bonded by UV (ultraviolet) light process.

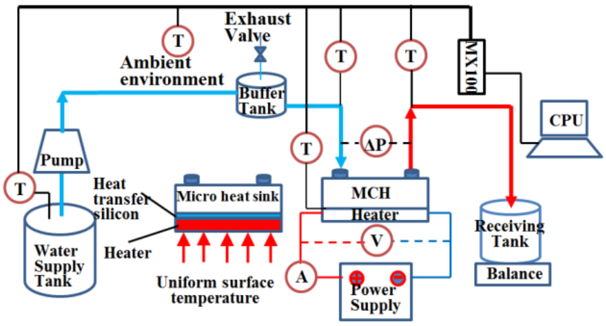

Figure. 1. Schematic diagram of the test loop for microchannel heat sinks

The substrates were designed with length of 48 mm, width of 27 mm, and thickness of 1.8 mm; 12 channels in each substrate were designed with a rectangular cross-section having width W_c of 500 µm, length L_c of 33 mm, distance between two adjacent microchannels of 500 µm, and channel depths D_c being 200 µm, 350 µm, 500 µm, 700 µm, and 900 µm for five MCHs from SD1 to SD5, respectively; two manifolds in each substrate were also designed with a rectangular cross-section having width of 3 mm, length of 18 mm, and depth of 900 µm. The substrates have the thermal conductivity of 237 W/(mK), the specific heat of 904 J/(kgK), and the density of 2,700 kg/m^3. The upper parts of the substrates were made of the transparent PMMA (polymethyl methacrylate) with the thickness of 10 mm, the thermal conductivity of 0.19 W/(mK), and the density of 1,420 kg/m^3.

To operate the experimental system, the temperature of the bottom wall of substrate was fixed at uniform temperature of 50 °C, the room temperature was controlled at around 25 to 26 °C, and the mass flow rate of water was varied from 39 g/min to 213 g/min.

C. *Numerical Simulation*

In order to investigate the heat transfer of the twelve microchannels inside the heat sinks, two manifolds, a combination of inlet-outlet as well as overall the microchannel heat sinks were simulated for numerical analyses.

Numerical simulations of three-dimensional single-phase heat transfer were performed by using CFD ACE$^+$ software. Algorithm of this software is based on the finite volume method. Five models have been built and 35 sets of data have been simulated. The dimensions of geometric configuration of the substrates are listed in Table I and shown in Fig. 2. Water was used as the working fluid. No internal heat generation was occurred, resulting in Q_i =0.

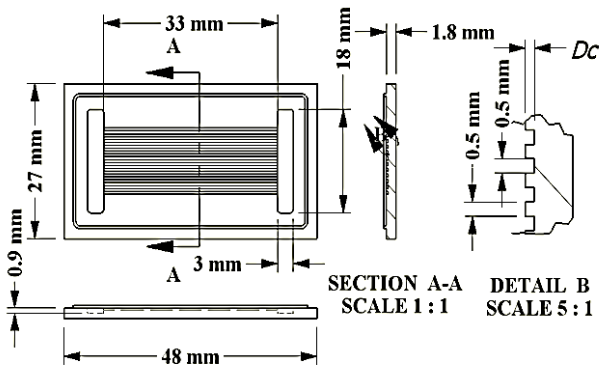

Figure. 2. Dimensions of the test section

III. Results and Discussions

A. *Numerical Results*

In this study, the boundary conditions for simulations were set as follows: the temperature of the inlet water was fixed at 25.5 ℃, the mass flow rates (MFRs) were varied from 39 g/min to 213 g/min, the bottom wall of the substrate was fixed at a uniform temperature of 50 ℃, and the channel depths were varied from 200 µm to 900 µm

TABLE I. DIMENSIONS OF THE TEST SECTIONS

No.	Dimensions of manifolds (mm)			Dimensions of channels		
				(mm)	(μm)	
	L_m	W_m	T_m	L_c	W_c	D_c
SD1	18	3	0.9	33	500	200
SD2	18	3	0.9	33	500	350
SD3	18	3	0.9	33	500	500
SD4	18	3	0.9	33	500	700
SD5	18	3	0.9	33	500	900

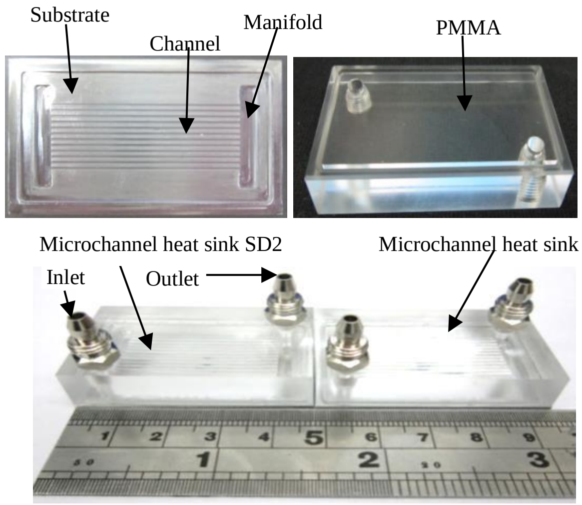

Figure. 3. Photos of substrates, PMMA plate and microchannel heat sinks SD2 and SD4

Fig. 4 shows the isothermal profiles of MCH SD2 at mass flow rate of 132 g/min. Results from Fig. 4 and such profiles of other models under this study indicate that for each microchannel heat sink, it always exists an isothermal surface throughout the channels, the manifolds, and the upper place.

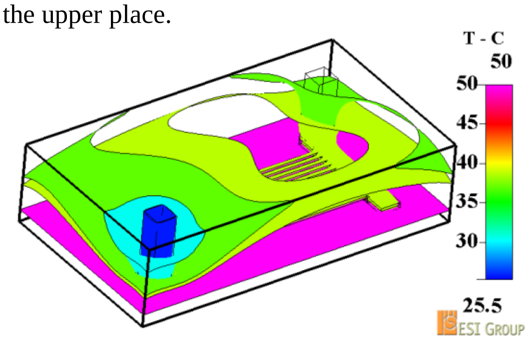

Figure. 4. Isothermal surface profiles

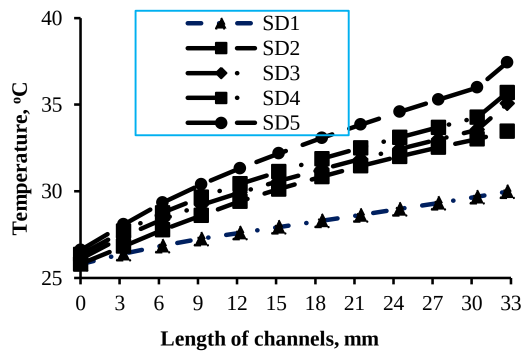

Figure. 5. The No. 1 channel temperature of the five MCHs

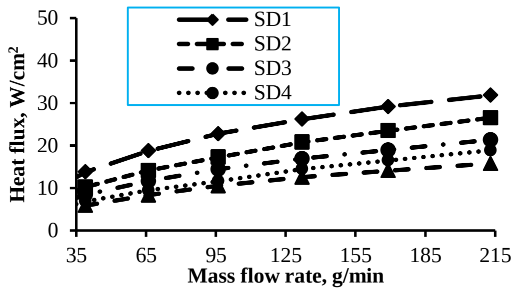

Figure. 6. Heat flux versus MFR of the five MCHs

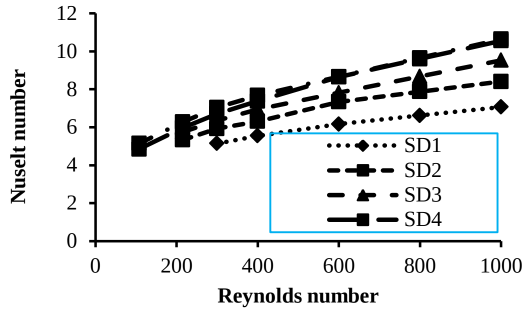

Figure. 7. Nusselt number versus Reynold number

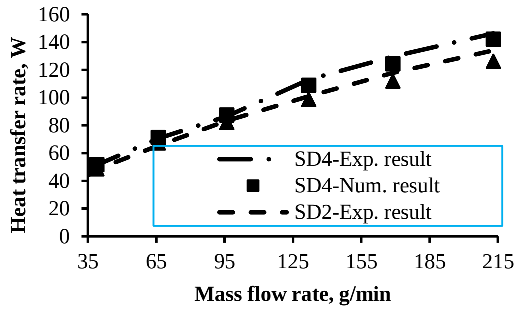

Figure. 8. Comparison Num. and Exp. Results

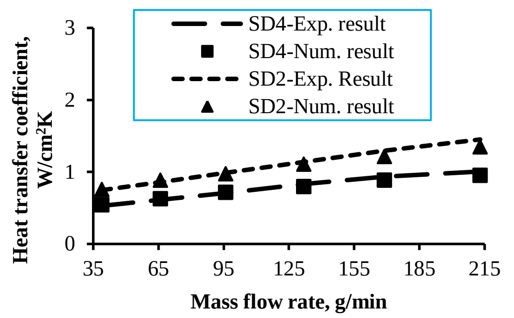

Figure. 9. Comparison num. and exp. results

The comparison of numerical results for temperature of the water moving in the No.1 channels of the five MCHs (MCHs SD1-SD5) is presented in the Fig. 5. It is found that the temperatures of water moving in channels with deeper channel depth are higher than those having shallower channel depth. It is noted that in this study, the No.1 channel shown in Fig. 5 is the channel nearest to the inlet of the MCH and the No. 12, the farthermost.

Fig. 6 shows a comparison of numerical results for heat fluxes of the five MCHs. The comparison shows that the heat fluxes increased from 15.8 W/cm^2 to 31.8 W/cm^2 when the channel depths decreased from 900 µm to 200 µm. In addition, the maximum heat flux of 31.8 W/cm^2 is obtained for the MCH SD1 at a MFR of 213 g/min.

Results obtained for the Nusselt number versus the Reynolds number for the MCHs from SD1 to SD5 in the laminar developing flow region are shown in Fig. 7. The results show that the Nusselt number increased when the Reynolds number increased or the channel depth decreased.

B. Experimental Results

The experiments were performed in a room maintained at the temperature around 25 to 26 ℃. Deionized water was used as the working fluid, the boundary conditions were discussed in the previous section, and the MFR was varied from 39 g/min to 213 g/min. Apparatuses used for the experiments are listed as follows:

- Thermocouple wires: model PT-100, made by Omega
- Pumps: model PU-2087, made by JASCO
- Differential pressure transducer: model PMP4110, made by GE Druck
- Thermoelectric heater: model TEC1-241.10, made by Championtech Technology

A comparison of numerical and experimental data for heat transfer rate of MCHs SD2 and SD4 is presented in Fig. 8. The results show that the heat transfer rate increased when the MFR increased, and good agreements between numerical and experimental results are achieved with maximum percentage error of 6 %. A comparison of numerical and experimental results for heat transfer coefficient of MCHs SD2 and SD4 is presented in Fig. 9. The results show that the heat transfer coefficient increased when the MFR increased or the channel depth decreased. Good agreement between numerical and experimental results is achieved, with maximum percentage error of 5.8 %.

IV. Conclusions

In this study, the effects of channel depth to heat transfer of five microchannel systems have been studied using both numerical and experimental methods. For the two cases being investigated experimentally in this study (with the channel depths of 350 µm and 700 µm), good agreement between numerical and experimental results was achieved, with maximum percentage errors to be less than 6%. A maximum heat flux of 31.8 W/cm^2 was achieved for the MCH SD1 (with the channel depth of 200 µm) at the MFR of 213 g/min. Besides, the maximum heat transfer rate of 143.8 W was achieved for the MCH SD5 (with the channel depth of 900 µm) at the MFR of 213 g/min. As far as the impact of the channel depth on the behaviors of heat transfer associated with the microchannel heat sink systems is concerned, when the channel depths increased from 200 µm to 900 µm, the heat fluxes decreased from 31.8 W/cm^2 to 15.8 W/cm^2 at the mass flow rate of 213 g/min.

Furthermore, the overall microchannel heat sink, including the channels, manifolds, substrate, inlet-outlet, and the top cover has been simulated by using the CFD – ACE$^+$ package.

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