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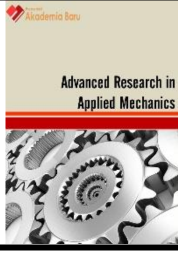

Hybrid Microchannel Heat Sink with Sustainable Cooling Solutions: Experimental Analysis

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ABSTRACT

Miniaturization and utilization of low-dimensional structures of recent electronic devices have witnessed some new micro cooling methods which can fulfil the cooling demand for the electronic devices. Microchannel heat sink (MCHS) is one of the micro cooling method which appears as a promising method that can provide high heat transfer rate due to small hydraulic diameter. Furthermore, microchannel heat sink is easy to be fabricated compare to other micro cooling device. Due to fast development in electronic industry, hybrid microchannel heat sink with optimal design has received a great deal of attention in order to provide sustainable cooling solutions. However, most of the studies of hybrid microchannel heat sink only provided the numerical analysis without any validation of the proposed design experimentally. This is very important since it also will determine whether the proposed hybrid microchannel heat sink can be fabricated or not. Therefore, the aim of this article is to validate the numerical model of hybrid microchannel heat sink (TC-RR-SC MCHS) experimentally. The validation result showed that the maximum discrepancy between both simulation and experimental analyses for Nusselt number and friction factor were 15.8% and 17.4%, respectively, which is less than 20%.

Keywords:
Micro-cooling; microchannel heat sink; rib, cavity; sustainable cooling solutions

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1. Introduction

Cooling system application in thermal engineering have been studied both theoretically and practically in building energy system, electronic devices, chemical vapour deposition instruments, solar energy collectors, furnace engineering and many more [1]. Many methods have been proposed to improve the cooling performance, such as air cooling method [2], heat pipe method [3], use of liquid material as a coolant [4], and micro-cooling method [5]. Micro-cooling is a good technique due to its high cooling efficiency. Tuckerman *et al.,* [6] reported that rectangular microchannel heat sink

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(MCHS) removed heat-flux up to 790 W/cm^2. However, it generated high-pressure drop. This is because of high wall shear stress in the developing region of laminar flow [7]. The discovery motivated many scholars to develop hybrid MCHS. MCHS with rib-cavity structure is an innovation that can improve the heat transfer performance with less pressure drop. Li *et al.,* [8, 9] investigated the effects of ribs and cavities on the hydrothermal performance. They revealed that combined effect of interruption, re-development of the thermal boundary layer, intensified mainstream disturbance, and chaotic mixing enhanced the heat transfer performance.

The most critical issue in the design development of hybrid MCHS is its fabrication process as there are many limitations such as the size of cutting tools and micro-machining process. It will determine whether the proposed design can be commercialised and transferred to the mass production in electronic industry or not. Besides that, most of researchers still use straight-channel MCHS for their numerical model validation [8, 10-12] because experimental data of the fabricated hybrid MCHS are very limited in the open literature. Therefore, in this study, hybrid MCHS (TC-RR-SC MCHS) was fabricated and its hydrothermal performance was compared with the simulation result. Regression equations were formulated from the regression analysis, which it can be used as reference for validation purpose in the future study for an innovated MCHS that have similar design with the proposed MCHS.

2. Methodology

In this section, the relevant expressions related to Nusselt number, Nu_{ave}, and friction factor, f_{app} are discussed in detail. Before the Nu_{ave} was calculated, the average convective heat transfer coefficient, h_{ave}, had been determined first based on the sensible heat, Q_{sen}, the average wall and fluid temperatures, and the total area of convective heat transfer as follows:

$$Nu_{ave} = \frac{h_{ave} D_h}{k_f} \tag{1}$$

$$h_{ave} = \frac{Q_{sen}}{A_t \left(T_w - T_{f,ave}\right)} \tag{2}$$

$$Q_{sen} = \rho_f c_{pf} \dot{V} \left(T_4 - T_1\right) \tag{3}$$

The volume flow rate, $\dot{V}$, of the distilled water was measured experimentally via a flow meter. The inlet and outlet temperatures of the distilled water in the fabricated MCHS were measured by the T_1 and T_4 thermocouples, respectively. The total area of convective heat transfer, A_t, was calculated by the following equation:

$$A_t = A_{bc} + \eta A_{fin} \tag{4}$$

Where A_{bc} is the unfinned area at the bottom surface of microchannels, while A_{fin} is the fin area. The tip of the sidewalls and rib geometries were assumed in an adiabatic tip condition as the top cover of the TC-RR-SC MCHS device is made of a low-conductivity material (0.2 W/m-K). Thus, the coefficient of area, η, can be written as [13]:

$$\eta = \frac{\tanh(mHc)}{mHc} \tag{5}$$

$$m = \sqrt{\frac{h_{ave} P_w}{k_{al} A_c}} \tag{6}$$

Where P_w, k_{al}, and A_c are the wall perimeter, the thermal conductivity of Aluminium 6061 T6 (167 W/m-k), and the cross-section area of the channel wall, respectively. They are calculated as follows:

$$P_w = 2Hc + W_w \tag{7}$$

$$A_c = HcW_w \tag{8}$$

The W_w represents the channel wall thickness. The final value of the convective heat transfer coefficient was determined after the total area of convective heat transfer had been being calculated by the iterative approach. It had been being repeated until the difference between η_{old} and η_{new} approached to 0.1. The average fluid temperature, $T_{f,ave}$ was calculated based on T_1 and T_4, while the wall temperature, T_w measured by Eq. (9). The direct measurement of T_w is not available. Thus, extrapolation of the measured temperatures (T_2, T_3 and T_5) were performed by assuming 1D heat conduction [14].

$$T_w = T_{al} - \frac{Sq_{w,s}}{k_{al}} \tag{9}$$

Where T_{al} is the average temperature (T_2, T_3 and T_5) of aluminium block. The vertical distance between those thermocouples and the bottom surface of the microchannel, S, is 5.8 mm. The supplied heat flux in the aluminium block was calculated as the sensible heat, Q_{sen}, gained by the distilled water over the base area of the aluminium block upper body, A_{ubb}, as shown in Eq. (10).

$$q_{w,s} = \frac{Q_{sen}}{A_{ubb}} \tag{10}$$

The velocity of the distilled water, U, in the TC-RR-SC MCHS device was calculated by Eq. (11). The pressure transducer was not directly connected to the inlet and outlet of the microchannels. Thus, the pressure transducer measured the total pressure loss, ΔP_{tot}, as shown in Eq. (12):

$$\mathrm{Re} = \frac{\rho_f U D_h}{\mu_f} \tag{11}$$

$$\Delta P_{tot} = \Delta P_{mchs} + \Delta P_{se} + \Delta P_{sc} \tag{12}$$

Where ΔP_{mchs} is the actual pressure drop in the fabricated MCHS. Pressure loss due to sudden expansion and contraction is denoted by ΔP_{se}, and ΔP_{sc}, respectively.

Pressure loss due to sudden expansion, ΔP_{se}:

$$\Delta P_{1-2} = \frac{1}{2}\rho_f\left(u_2{}^2 - u_1{}^2\right) + \frac{K_{e1}}{2}\rho_f u_1{}^2 \tag{13}$$

$$\Delta P_{4-5} = \frac{1}{2}\rho_f\left(u_5{}^2 - u_4{}^2\right) + \frac{K_{e2}}{2}\rho_f u_4{}^2 \tag{14}$$

$$\Delta P_{5-6} = \frac{1}{2}\rho_f\left(u_6{}^2 - u_5{}^2\right) + \frac{K_{e3}}{2}\rho_f u_5{}^2 \tag{15}$$

Velocity in the cross-section of manifold and MCHS areas are denoted by u_1, u_2, u_4, u_5, and u_6, respectively. The loss coefficient of the sudden expansion values (K_{e1}, K_{e2}, and K_{e3}) were determined from the Handbook of Heat Transfer [15].

Pressure loss due to sudden contraction, ΔP_{sc}:

$$\Delta P_{2-3} = \frac{1}{2}\rho_f\left(u_3{}^2 - u_2{}^2\right) + \frac{K_{c1}}{2}\rho_f u_3{}^2 \tag{16}$$

$$\Delta P_{3-4} = \frac{1}{2}\rho_f\left(u_4{}^2 - u_3{}^2\right) + \frac{K_{c2}}{2}\rho_f u_4{}^2 \tag{17}$$

$$\Delta P_{6-7} = \frac{1}{2}\rho_f\left(u_7{}^2 - u_6{}^2\right) + \frac{K_{c3}}{2}\rho_f u_7{}^2 \tag{18}$$

Where u_3 and u_7 are velocity in the cross-section of manifold and MCHS areas. The loss coefficient of sudden contraction values (K_{c1}, K_{c2}, and K_{c3}) were also could be found in the Handbook of Heat Transfer [15]. The friction factor, f_{app}, in the fabricated MCHS was measured by substituting the calculated ΔP_{mchs} in the Eq. (19).

$$f_{app} = \frac{2\Delta P_{mchs} D_h}{\rho L t U^2} \tag{19}$$

3. Result and Discussion

Figure 1 shows the experimental result of Nu_{ave}. Based on the correlation analysis, the correlation coefficient, r for the experimental result of Nu_{ave} was 0.98. It indicated that there was a significant positive relationship between Re and Nu_{ave} ($r(7) = .98,\ p < .05$). Besides that, regression analysis was also conducted in order to formulate regression equations. The analysis showed that the R^2 value for this analysis was 0.9514. Figure 2 shows the validation of Nu_{ave} result. The maximum discrepancy between both simulation and experimental results was 15.8%. There are factors that contributed to the discrepancy of the data. The fabricated TC-RR-SC MCHS have burrs and debris at the top and bottom surface of microchannels. It affected the Nu_{ave} result as the heat transfer coefficient, h increased attributed to the increasing of the heat transfer area provided by the surface area of burrs and debris. Besides that, the fabricated TC-RR-SC MCHS and simulated TC-RR-SC MCHS were different in their number of microchannels. The fabricated TC-RR-SC MCHS have 49 microchannels, while the simulated TC-RR-SC MCHS only have a single microchannel.

The result of f_{app} also showed a significant negative relationship to the Re (see Figure 3). The r of f_{app} was -0.95 ($r(7) = -.95$, $p < .05$). The regression analysis also demonstrated that 90% of the variability in f_{app} was explained by the variability in Re. Next, f_{app} the was also validated experimentally. Figure 4 shows that the maximum discrepancy between both simulation and experimental data was 17.4%. The reason is that the pressure transducer was not connected directly to the inlet and outlet of the fabricated TC-RR-SC MCHS. The width of the pressure transducer is wider than the block length of the fabricated TC-RR-SC MCHS. Therefore, the pressure transducer need to be connected separately from the inlet and outlet of the fabricated TC-RR-SC MCHS. The burrs and debris at the top and bottom surface of microchannels also had affected the f_{app} as the pressure drop increased attributed to the decreasing of microchannels flow area. Furthermore, the different number of microchannels between the simulated TC-RR-SC MCHS and fabricated TC-RR-SC MCHS also had contributed to the increasing of data deviation.

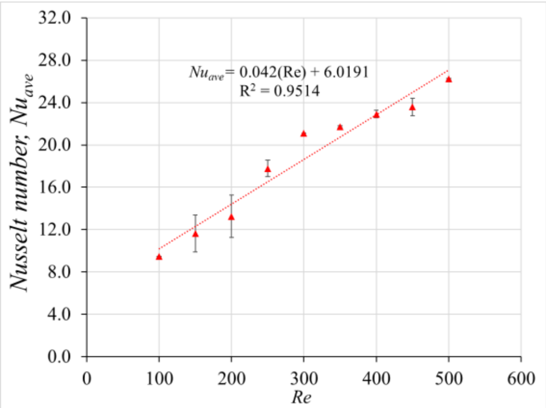

Fig. 1. Experimental result of the Nu_{ave} of the optimised TC-RR-SC MCHS

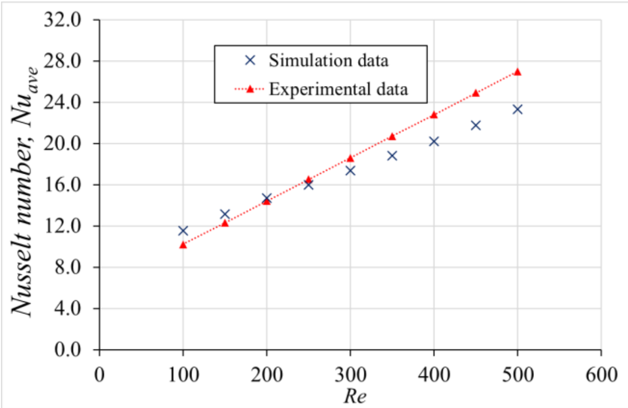

Fig. 2. Validation of the numerical result of Nu_{ave} with experimental result

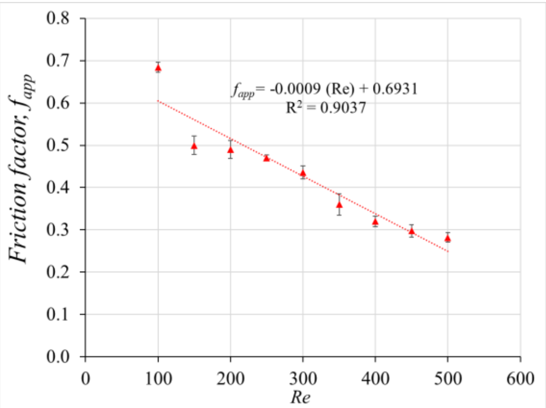

Fig. 3. Experimental result of the f_{app} of the optimised TC-RR-SC MCHS

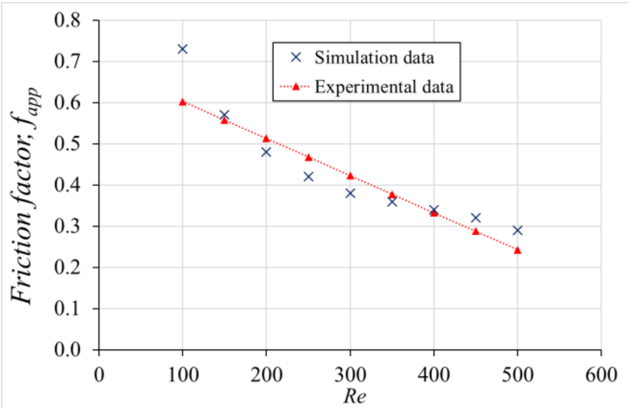

Fig. 4. Validation of the numerical result of f_{app} with experimental result

4. Conclusion

As overall, this analysis showed that the burrs and debris at the top and bottom surface of microchannels, and the number of microchannels affected both Nu_{ave} and f_{app}. To overcome these issues, a harder material than aluminium should be used in the fabrication of TC-RR-SC MCHS so that the inspection, deburring and finishing process can be continued until the end of the fabrication process. Besides that, a computer with higher RAM and uses parallel computing system is required to simulate TC-RR-SC MCHS with 49 microchannles. This is because the generation of meshing process for the design requires multiple computer cores to run the process simultaneously.

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