Numerical and Experimental Study on the Heat Transfer Characteristics in the Minichannel Heat Sinks

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ABSTRACT

In the present study, experimental and numerical results on the heat transfer characteristics and thermal performance of the minichannel heat sinks are presented. An experimental apparatus is set-up to analyze the problem. The minichannel heat sinks are fabricated from the aluminum by the wire electrical discharge machine with the length, the width and the base thickness of 110, 60, 2 mm, respectively. The in-line configuration heat sinks with two different fin heights are 1.00, 1.50 mm. Experiments are done at various coolant Reynolds numbers in the ranging of 80-200. The k-ε two equations low Reynolds number turbulence model is employed to simulate the turbulent heat transfer characteristics. A finite volume method with a structured uniform grid system is employed to solve the model. The predicted results are verified by comparing with the measured data. The predicted results are good agreement with the present experiments. The results of this study are of technological importance for the efficient design of cooling systems of electronic devices to enhance cooling performance.

Keywords: *Heat transfer characteristics; minichannel heat sink; electronic cooling*

1. INTRODUCTION

Due to the small physical size of the electronic devices, the conventional air or liquid cooling systems limitation, the development of the miniaturized technology, mini and micro-components has been increased especially in the electronic devices. The development of the miniaturized technology, mini and micro-components has been introduced as one of the heat transfer enhancement techniques. The heat transfer and pressure drop in the mini and micro-channel has been widely studied by researchers. Peng and Peterson [1] experimentally investigated the forced convective heat transfer and flow characteristics of water in microchannel. Adams et al. [2] investigated the single-phase turbulent flow in circular microchannels. A generalized correlation for the Nusselt number for single-phase turbulent flow in circular microchannels was proposed. Gillot et al. [3] applied the single-phase and two-phase micro heat sinks for cooling of power components. Qu and Mudawar [4] experimentally and numerically investigated the pressure drop and heat transfer characteristics of a single-phase micro-channel heat sink. Hetsroni et al. [5] applied the heat sink for cooling the electronic devices. Agostini et al. [6], [7] experimentally studied on the friction factor and heat transfer coefficient of R134a in a vertical liquid up-flow. Owhai and Palm [8] studied the heat transfer characteristics of R134a single-phase forced convection through single circular micro-channels. Yue et al. [9] considered the pressure drop characteristics of single and two-phase flows through two T-type rectangular microchannel mixers. Abdelall et al. [10] experimentally investigated the single-phase and two-phase pressure drops caused by abrupt flow area expansion and contraction in small circular channels. Lee et al. [11] verified the thermal behavior obtained from the classical correlations with the experimental data in the rectangular microchannels. Zhang et al. [12] reported a single-phase liquid cooling in the microchannel heat sink for electronic packages. Shen et al. [13]

investigated the single phase convective heat transfer of de-ionized water in a compact heat sink consisting of 26 rectangular microchannels. Timothy and Brackbill [14] studied on the roughness elements affect internal flows in different ways. Steinke and Kandlikar [15] considered the validity of friction factor theory based upon conventional sized passages for microchannel flows is still an active area of research. Hrnjak and Tu [16] focused on investigating fully developed liquid and vapor flow through rectangular microchannels. R134a liquid and vapor were used as the testing fluids. Xie et al. [17] studied the development of the information technology industry, the heat flux in integrated circuit chips cooled by air flowing in the microchannel. Mohammed et al. [18] numerically investigated the water heat transfer and flow characteristics in the rectangular cross-section microchannel heat sink. It can be seen that the friction factor and wall shear stress tend to increase with the wavy microchannels amplitude increases. Hashemi et al. [19] considered effect of channel aspect ratio and porosity on the nanofluid thermally developing temperature profile and fully developed velocity profile in a miniature plate fin heat sink. Shafeie et al. [20] numerically studied the laminar forced convective heat transfer of water in micro pin-fin heat sinks. Shkarah et al. [21] conducted the micro-channel heat sinks with three different shapes. Seyf et al. [22] described the thermal and hydrodynamic characteristics of a microtube heat sink with tangential impingement. Kuppusamy et al. [23] numerically studied the thermal and nanofluid flow fields in a trapezoidal grooved microchannel heat sink. Fan et al. [24] experimentally and numerically studied the fluid flow and heat transfer on cylindrical oblique-finned heat sink. Wong et al. [25] predicted the thermal performance of a parallel flow two-layered microchannel heat sink. Lelea and Laza [26] studied the nanofluid heat transfer and flow characteristics in tangential microtube heat sink.

As mentioned above, the numerous papers presented the study on heat transfer and pressure drop in the channel or heat sink with various configurations. However, only few work reported on the application of minichannel heat sink for the cooling electronic components. In addition, there still remains room to discuss on the heat transfer characteristics in the heat sinks both experiment and simulation. Therefore, the paper focuses on the experimental and numerical study on the cooling characteristics in the minichannel heat sinks using water as coolant. Effects of fin height, heat flux, and coolant flow rate on the cooling performance are considered.

2. EXPERIMENTAL APPARATUS AND METHOD

A schematic diagram of the experimental apparatus is shown in Fig. 1. The test loop consists of a set of cooling loop, test section and data acquisition system. The close-loop of fluids consists of a 10^{-3} m^3 storage tank, and the flow rate measurement system. After the temperatures of the fluids are cooled to achieve the desired level, the fluids is pumped out of the storage tank, passed through the minichannel heat sink, and returned to the storage tank. The flow rates are varied during a set of experiments with the help of a dimmerstat connected to the pump. The flow rates of the coolant are controlled by adjusting the valve and measured by collecting the fluid with the precise cylinder for a period of time during 15 min and the fluid mass is measured by an electronic weight scale. The pressure drops across the test section are measured by the differential pressure transducer with an accuracy of 0.02% of full scale. There are two pressure taps on the cover plate upstream and downstream of the test section. The type-T copper–constantan thermocouples with an accuracy of 0.1% of full scale are employed to measure fluids temperatures. The inlet and outlet temperatures of fluids are measured by thermocouples with 1 mm diameter probes extending into the cover plate in which the coolant flow. All the thermocouples probes are precalibrated by dry-box temperature calibrator with 0.01 °C precision.

The schematic diagram of the test section is shown in Fig. 2. The test sections are fabricated from the copper by a wire electrical discharge machine (WEDM) with the widths*length of 60*110 mm. Two heat sinks with fin height of 1.00, 1.50 mm. are tested. Experiments are conducted with various coolant flow rates, channel height of fin, inlet temperature of the coolant, and heat flux. The supplied voltage and current to the heaters are measured by the digital clamp meter with an accuracy of 0.2% of full scale. Data collection is carried out using a data acquisition system (DataTaker, DT800).

The energy balance between the supplied heat by the heater plate and the absorbed heat by the fluids is ±10%. The average value of heat transfer rate is obtained by the supplied heat by heater plate and the absorbed heat by the fluids. The uncertainties of measurements data and the relevant parameters obtained from the data reduction process are calculated. The maximum uncertainties of the relevant parameters in the data calculation are based on Coleman and Steel method [27]. The maximum uncertainties of relevant parameters are ±4% for Reynolds number, ±7% for thermal resistance and ±5% for the Nusselt number.

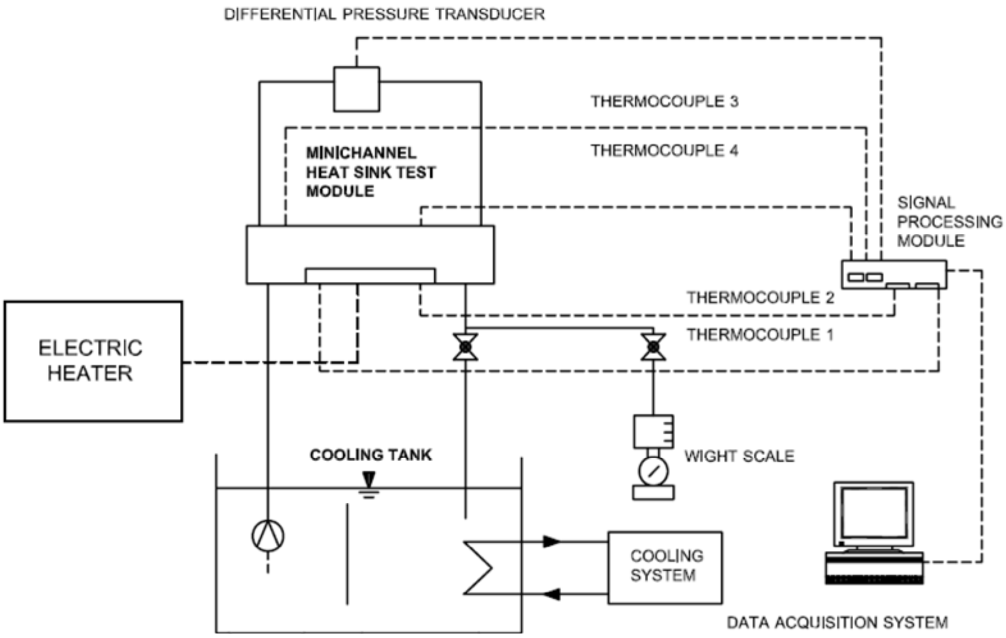

Fig. 1 *Schematic diagram of experimental apparatus.*

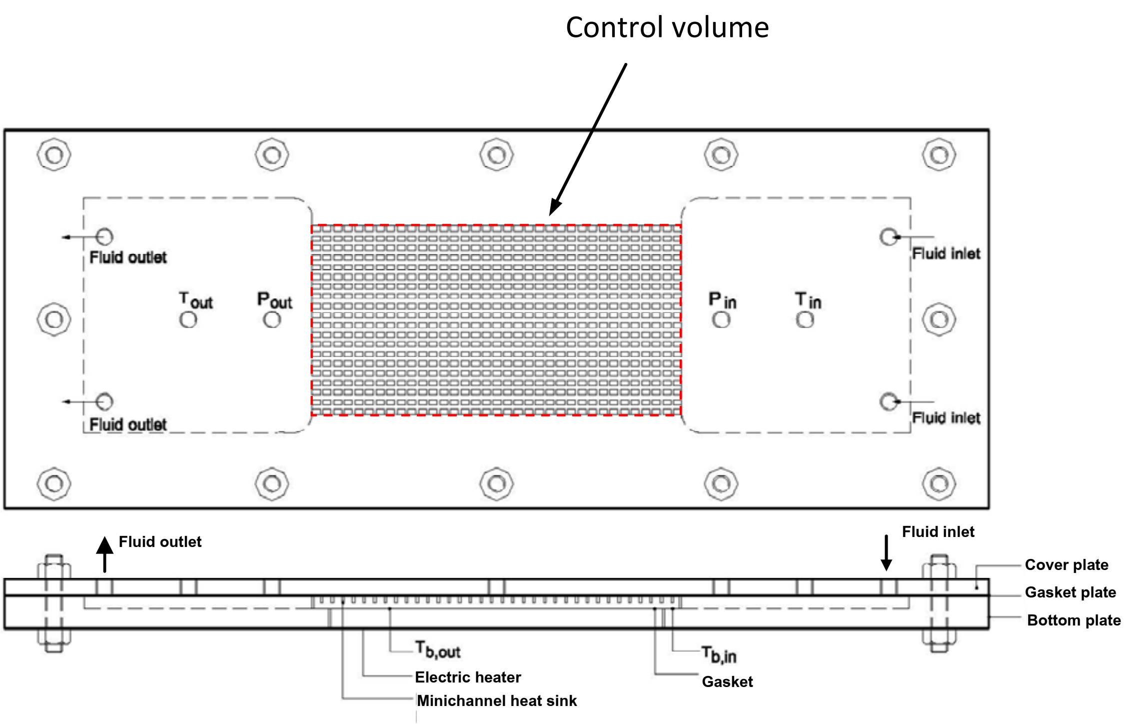

Fig. 2 *Schematic diagram of the test section.*

3. MATHEMATICAL MODELING

To focus on the effect of fin height, coolant flow rate and heat flux on the heat sink performance, the following assumptions are made:

- Fluid flow and heat transfer are in steady-state and three dimensional.
- Single phase fluid flow is turbulent.
- Properties of both fluid and heat sink material are temperature independent.
- All the heat sink surfaces exposed to the surroundings are assumed to be insulated except the heat sink base plate where constant heat flux simulating the heat generation from electronic chip is specified.

By considering the geometry and physical problem as shown in Fig. 2, the k-ε low Reynolds number standard turbulence model [28], [30] is employed to simulate the turbulent heat transfer characteristics. The main governing equations can be written in the following form:

Continuity equation:

$$\frac{\partial \rho}{\partial t} + div\left(\rho \mathbf{U}\right) = 0 \tag{1}$$

Momentum equation:

x-momentum: $$\rho \frac{Du}{Dt} = -\frac{\partial p}{\partial x} + div\left(\mu\, grad\, u\right) + S_{M_x} \tag{2}$$

y- momentum: $$\rho \frac{Dv}{Dt} = -\frac{\partial p}{\partial y} + div\left(\mu\, grad\, v\right) + S_{M_y} \tag{3}$$

z-momentum: $$\rho \frac{Dw}{Dt} = -\frac{\partial p}{\partial z} + div\left(\mu\, grad\, w\right) + S_{M_z} \tag{4}$$

Energy equation:

$$\rho \frac{Di}{Dt} = -p\, div\, \mathbf{U} + div\left(\Gamma\, grad\, T\right) + \Phi + S_i \tag{5}$$

Turbulent kinetic energy (k) equation:

$$\frac{\partial\left(\rho k\right)}{\partial t} + div\left(\rho k \mathbf{U}\right) = div\left[\left(\frac{\mu_t}{\sigma_k} grad\, k\right)\right] + 2\mu_t E_{ij} \cdot E_{ij} - \rho\varepsilon \tag{6}$$

Turbulent kinetic energy dissipation (ε) equation:

$$\frac{\partial\left(\rho \varepsilon\right)}{\partial t} + div\left(\rho \varepsilon \mathbf{U}\right) = div\left(\frac{\mu_t}{\sigma_\varepsilon} grad\, \varepsilon\right) + C_{1\varepsilon}\frac{\varepsilon}{k} 2\mu_t E_{ij} \cdot E_{ij} - C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \tag{7}$$

The empirical constants for the turbulence model are arrived by comprehensive data fitting for a wide range of turbulent flow of Launder and Spalding [28], [29]:

$$C_\mu = 0.09,\ C_{\varepsilon 1} = 1.47,\ C_{\varepsilon 2} = 1.92,\ \sigma_k = 1.0,\ \sigma_\varepsilon = 1.3 \tag{8}$$

Based on the computational domain as shown in Fig.4, the boundary conditions of the simulation processes are as follows:

- Inlet: uniform velocity and temperature.
- Outlet: zero pressure condition.
- Outside surface: no heat loss for all outside surface.
- Wall: no slip condition for all directions
- For the heat source, constant heat flux condition

4. NUMERICAL SIMULATION

By considering the geometry and physical problem as shown in Fig. 3, second-order upwind scheme and structured uniform grid system are used to discretize the main governing equations. Based on the control volume method, SIMPLEC algorithm of Van Doormal and Raithby [31] is employed to deal with the problem of velocity and pressure coupling. Inlet coolant temperature with uniform velocity entering the test section is applied. Velocity boundary condition is applied at the inlet section while the pressure boundary condition is used at the outlet section. In the present study, the commercial program ANSYS/FLUENT is employed as the numerical solver to solve the problem. The results are considered converged when the normalized residual falls below 10^{-5} for all variables.

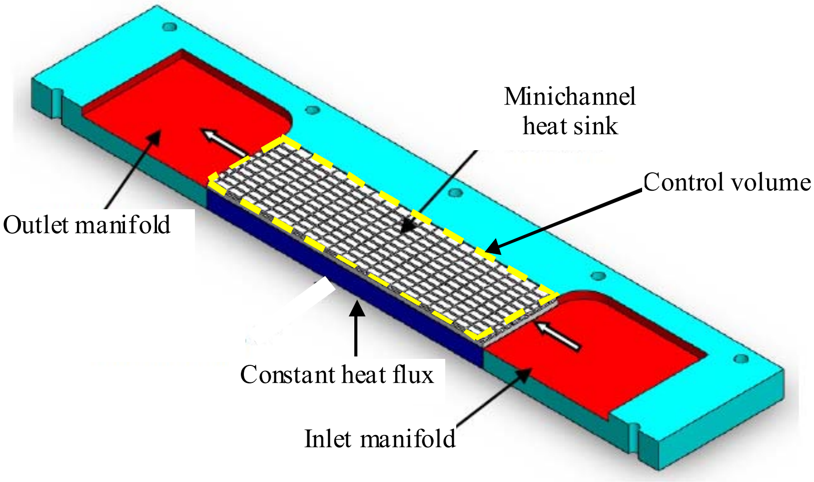

Fig. 3 *Computational domain of the minichannel heat sink.*

5. GRID OPTIMIZATION

In order to obtain the accuracy of computations processes, the grid independence is performed in the analysis by adopting different grid sizes of 730,000 1,130,000 and 2,090,000. The grid independence check indicated that the grid systems of 1,130,000 ensure a satisfactory solution. This is verified by the fact that the difference of the outlet coolant temperature with grid finer than the 1,130,000 (e.g. 2,090,000) within 1%. After the above two comparisons and confirming that the computational model is generating correct results, coolant with various Reynolds numbers, fin heights and heat flux are analyzed.

6. RESULTS AND DISCUSSION

The Reynolds numbers based on the hydraulic diameter of the channel are tested in the range between 80 and 200. Fig. 4 shows the variation of the outlet coolant temperature with Reynolds number for different inlet coolant temperatures. It can be seen that the heat transfer rate tends to increase as coolant mass flow rate increase. However, increase of heat transfer rate is less than that of coolant mass flow rate. Therefore, outlet coolant temperature tends to decrease as coolant mass flow rate increases. In addition, the outlet coolant temperature depends directly on the inlet coolant temperature. Figure 4 also compares the results obtained from the present experiment and those from the numerical study. It can be clearly seen from figure that the results obtained from the model slightly underpredicts the measured data.

Fig. 5 shows the variation of the average Nusselt number with Reynolds number for different inlet coolant temperatures. It can be seen that the heat transfer rate depends on the cooling capacity rate of coolant. Therefore, average Nusselt number increases with increasing coolant Reynolds number. In addition, for a given coolant flow rate, the Nusselt number at lower inlet coolant temperature are higher than those from higher ones. This is because the cooling capacity depends directly on the temperature difference between the coolant and the heat sink. Considering the results obtained from the numerical study and those from the measured data, it can be clearly seen that the predicted Nusselt number are slightly lower than the measured ones. Higher surface area and surface roughness result in increase heat transfer rate, therefore, the Nusselt number of the heat sink with fin height of 1.50 mm are higher than those of 1.00 mm as shown in Figure 6. Across the whole range of coolant, average deviation of computed values from experiment is 2.96%. Reasonable agreement is obtained from the comparison between the measured data and the predicted results.

In this study, the overall heat sink performance can be shown in the thermal resistance form. The thermal resistance across the test section is produced from: thermal contact term between the heater unit and heat sink unit, conduction term through the heat sink unit, and convection term of the fin surface and the coolant.

For the thermal contact resistance between the heater unit and the heat sink unit, it can be minimized by using a thin film of high thermal conductivity grease at the junction interface. Using this definition, thermal resistance of heat sink is function of heat transfer coefficient. It is found that the heat sink thermal resistance decreases with increasing coolant Reynolds number and its slope decreases gradually. Increasing in coolant Reynolds can reduce the convective thermal resistance which is only a part of the total thermal resistance. However, at relative high coolant Reynolds number, the gain in the total thermal resistance reduction is obtained as shown in Fig. 7. Based on results shown in Fig. 7, it is seen that the heat sink with higher fin height give the thermal resistance lower than those for lower one. It can be clearly seen from figure that the results obtained from the model are excellent agreement with average deviation of computed values from experiment being 4.17% over the range of Reynolds numbers studied.

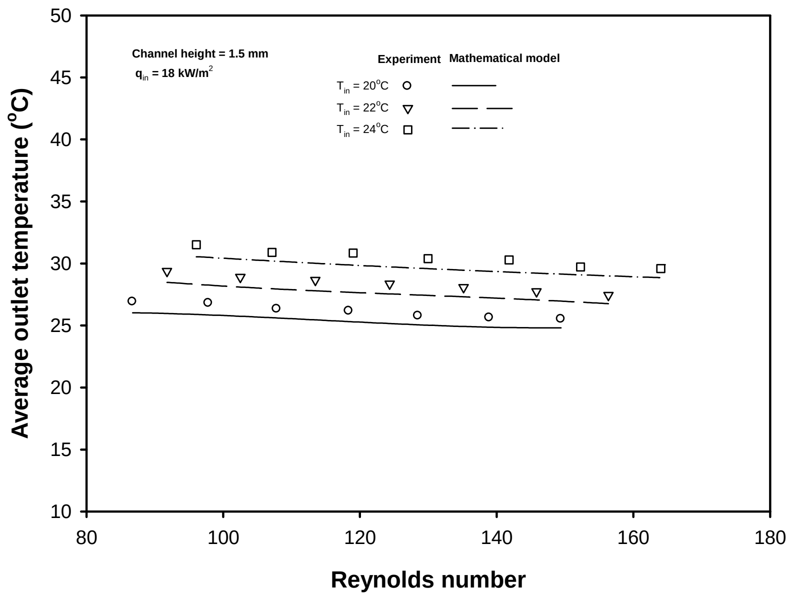

Fig. 4 *Variation of average outlet temperature of coolant with Reynolds number.*

7. CONCLUSIONS

New experimental data and numerical results on the heat transfer characteristics of the minichannel heat sinks are presented. The configurations and dimensions of the minichannel heat sink have significant effect on the thermal performance of the minichannel heat sink. The reasonable agreement is obtained from the comparison between the predicted results and the measured ones. The results of this study are of technological importance for the efficient design of cooling systems of electronic devices to enhance cooling performance.

Nomenclature

$C_{\varepsilon 1}$	turbulent model constant
$C_{\varepsilon 2}$	turbulent model constant
C_{μ}	turbulent model constant
Re	Reynolds number
T	temperature, °C
u_m	water velocity, m/s
$\mathbf{U}$	velocity vector

Greek symbols

ρ	density
μ	viscosity
ε	dissipation kinetic energy, m^2/s^3
σ_k	diffusion Prandtl number for k
σ_{ε}	diffusion Prandtl number for ε
ε	dissipation kinetic energy, m^2/s^3
μ	viscosity, kg/ms
Φ	viscosity energy dissipation function
σ_k	diffusion Prandtl number for k
σ_{ε}	diffusion Prandtl number for ε

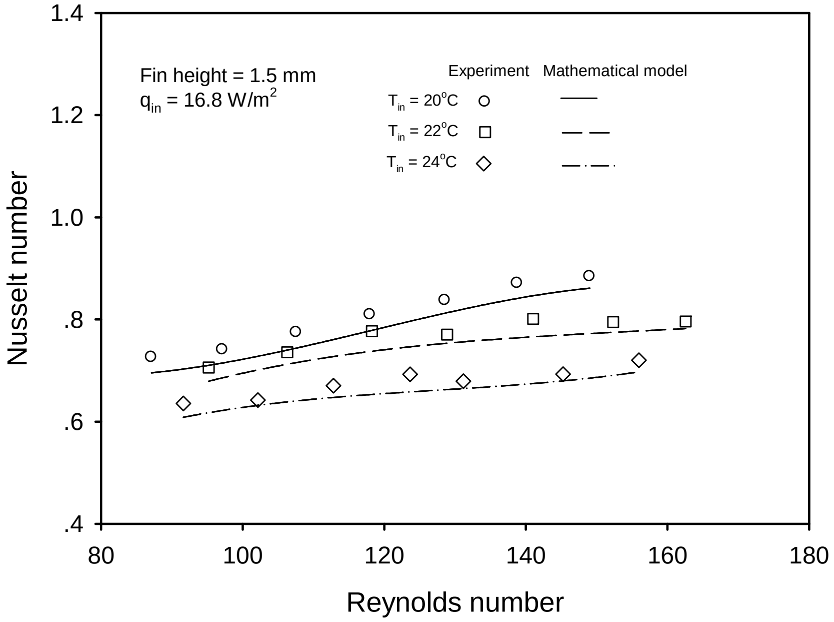

Fig. 5 *Comparison between the predicted Nusselt number and the measured Nussselt number.*

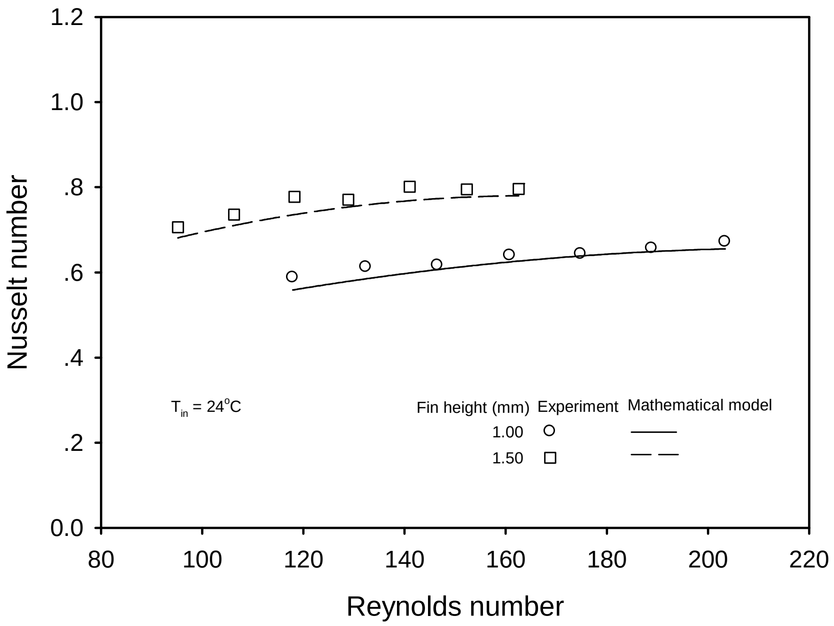

Fig. 6 *Effect of fin height on the Nusselt number.*

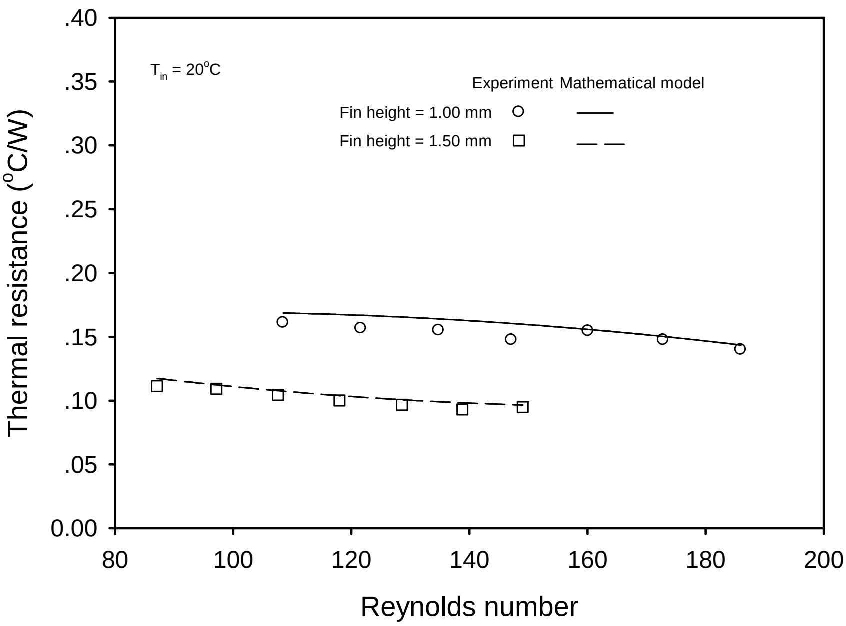

Fig. 7 *Comparison thermal resistance between the experiment and the mathematical model.*

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