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Heat Transfer Network and Correlation for A Double-Layered Microchannel Heat Sink

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ABSTRACT

This paper discusses the methods to obtain a correlation of local thermal resistance for heat transfer of a double-layered microchannel heat sink (DL-MCHS) with parallel flow configuration. Based on the energy balance in a thermal resistance network constructed for the DL-MCHS, the local thermal resistance which consists of conductive, convective and capacitive thermal resistance has been obtained as a function of governing parameters including the channel geometrical parameters. The results predicted for different channel width with the correlation are compared with the results obtained through numerical. Good agreement was found between simulation and correlation results, which indicates that this correlation could be used to predict the local temperature along the flow direction for DL-MCHS with different channel width.

1. Introduction

With the fast development of microelectronic equipment, heat removal has now become a serious problem to meet the demand for the thermal performance required. As an attempt to solve this issue, Tuckerman and Pease [1] proposed the design of silicon-based single-layered microchannel heat sink (SL-MCHS), that can achieve maximum thermal resistance, $\mathrm{R}_{\mathrm{t,max}}$ of 0.09˚C/cm^2 when subject to the heat flux q'' of 790 W/cm^2 by using water as coolant. From there onwards, scientists [2-7] have studied the performance for SL-MCHS with non-uniform channel design while some researchers [8-11] studied the performance for SL-MCHS by using Nano-fluid as coolant. Their findings concluded that the performance could be improved through different channel design and different coolant.

To reduce the temperature gradient along axial-direction, Vafai and Zhu [12] proposed the concept of DL-MCHS which is a substantial improvement over a conventional SL-MCHS. Following their work, other researches [12-16] have studied the performance for DL-MCHS. Wu *et al.,* [13] numerically investigated the parametric effects of channel number, aspect ratio and velocity ratio on

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the overall thermal performance. Lin *et al.,* [14] used a simplified conjugate-gradient method to optimize the flow and heat transfer for silicon-based DL-MCHS based on six design variables: channel number, bottom channel height, vertical rib width, thickness of two horizontal ribs and coolant velocity in the bottom. Wong and Muezzin [15] numerically studied the thermal performance for DL-MCHS with parallel and counter flow configuration. Wong and Ang [16] numerically studied the effects of vertically tapered and converging channel of a DL-MCHS on its thermal and hydraulic performance. Xie *et al.,* [17] numerically investigated the laminar heat transfer and pressure of DL-MCHS. Wei *et al.,* [18] experimental and numerical studied the thermal performance for stacked DL-MCHS by using silicon as substrate material and water as coolant. These studies all confirmed that the DL-MCHS has better performance when compared to the SL-MCHS.

Due to the difficulty of conducting experiment and long preparation and waiting time of running simulations for DL-MCHS, establishing correlation is the simplest way to predict the performance for MCHS. Tuckerman and Pease [1] proposed the correlation of maximum thermal resistance for SL-MCHS as the sum of three components: a) the conduction from the circuits through the substrate, package and heat sink interface, b) the convection from the heat sink to the coolant, c) the thermal resistance due to the heating of the fluid. Based on their results, the correlation results give good prediction of maximum thermal resistance when compared to the experimental results. After the work of Tuckerman and Pease [1], Mital [19] proposed a model that uses semi-empirical correlations to calculate the thermal resistance and pumping power in terms of channel width, wall width, flow velocity and particle volume fraction. Lee *et al.,* [20] found numerical predictions based on a classical, continuum approach. The predictions are in good agreement with the experimental data of copper-based single-phase flow through rectangular microchannels with different width and Reynolds number. Based on the results, they concluded that the conventional analysis approach could be employed in predicting heat transfer behaviour in MCHS when the entrance and boundary conditions are carefully matched in the simulation. Lee and Garimella [21] numerically investigated the laminar convective heat transfer performance in the entrance region of MCHS with rectangular cross-section. Based on the temperature and heat flux distribution, they proposed the local and average Nusselt number as a function of dimensionless axial distance and channel aspect ratio. Zhai *et al.,* [22] validated the related empirical correlation of laminar heat transfer in MCHS with both simulation and experiments in terms of pressure drop, friction factor, Nusselt number and different thermal resistances, etc. Although many researchers have investigated the correlation for SL-MCHS, the correlation for multi-layered MCHS remains blank.

Recently, Wei and Joshi [23] proposed the thermal resistance network for stacked multilayer MCHS and the solution was iterative in nature. Although the model was established, the solution still needs many assumptions and iterations, which is not efficient to predict the thermal performance. In the present work, a thermal resistance network is established to obtain a correlation to predict the temperature, and then the validity is checked by comparing with results obtained by simulation with a single-section 3D-conjugate model.

2. Methodology

The present study developed a correlation based on thermal resistance network and the predicted results will be compared with the simulation results. In this section, the physical model used for the study is discussed, followed by the simulation procedures. The thermal resistance network and the correlation is presented in next section.

2.1 Physical Model

Figure 1(a) illustrated the schematic of DL-MCHS with overall width, length and height of 10mm, 10mm and 1 mm, respectively. The DL-MCHS consists of $N = 40$ repeated sections with each section has width, W of $250\mu m$. For easy analysis, single section is used for analysis and the schematic of single section is illustrated in Figure 1(b). The DL-MCHS consists of two layers, which the top layer and bottom layer have same layer thickness of 500μm. In present study, the temperature distribution for DL-MCHS with different channel width, W_{ch} of $50\mu m$, $100\ \mu m$, $150\ \mu m$ and $200\ \mu m$ are investigated at fixed channel height, H_{ch} of 375μm. The present study considers parallel flow configuration in top and bottom channels, which the flow direction are illustrated in Figure 1(c). Thermal resistance network is constructed based on the model shown in Figure 1(b) and the procedures of the correlation is presented in the next section.

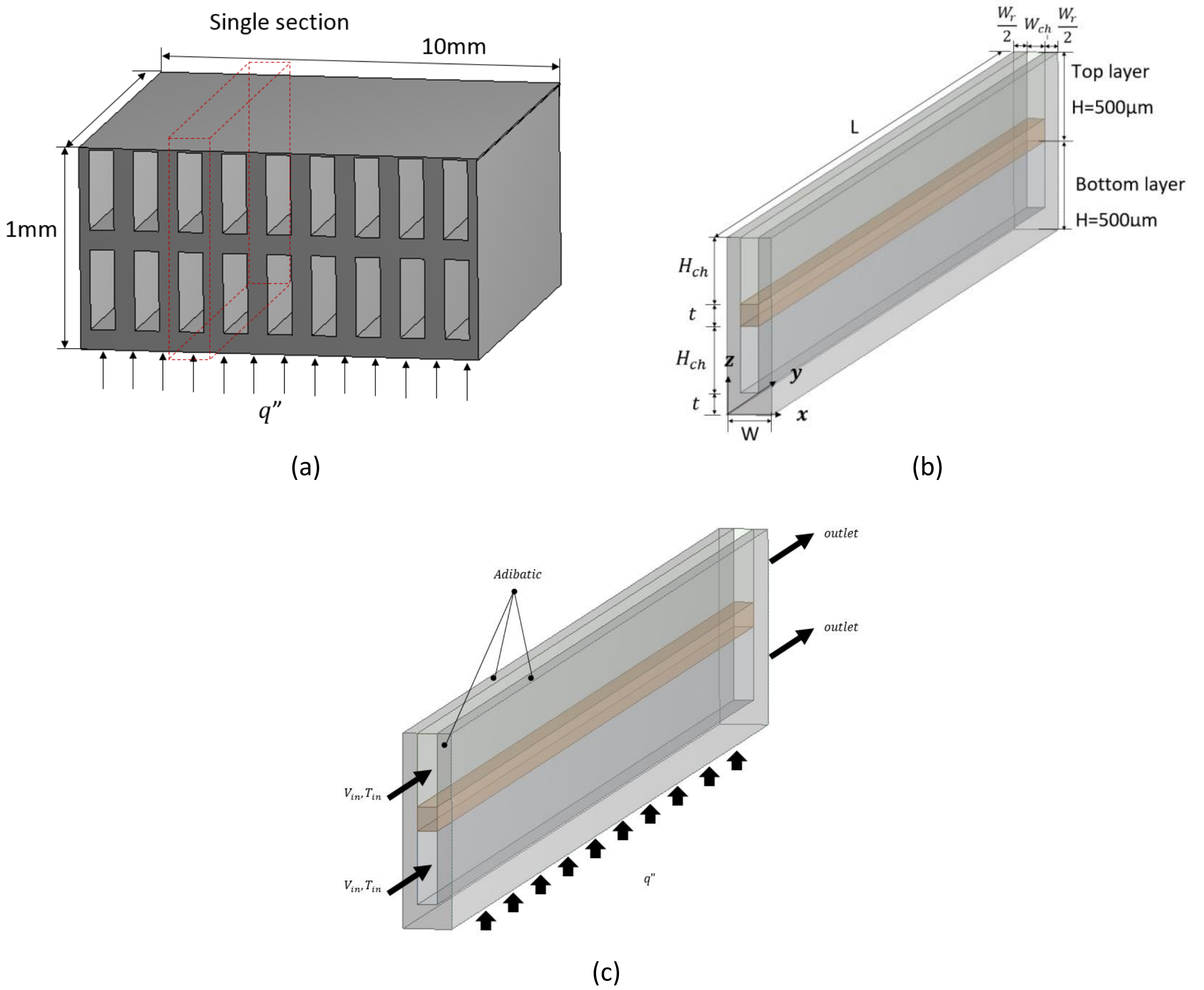


Fig. 1. The schematic of (a) DL-MCHS (b) single unit (c) DL-MCHS with parallel flow configuration

2.2 Simulation Procedures

Simulation is conducted with Ansys Fluent with the 3D conjugate model shown in Figure 1(b). To solve the conjugate model, the following assumptions are made:

I. The effects of gravity and other forms of body forces are negligible.
II. The fluid flow and heat transfer are in steady state.

III. The flow is incompressible and laminar
IV. The properties of fluid and solid are constant
V. Heat losses of the MCHS to the ambient are ignored.

Based on the assumptions above, the governing equations are as follows:

- Continuity equation:

$$\nabla \cdot \vec{V} = 0 \tag{1}$$

where, $\vec{V}$ is the velocity vector

- Momentum equation:

$$\rho_f\left(\vec{V} \cdot \nabla\right)\vec{V} = -\nabla p + \mu_f \nabla^2 \vec{V} \tag{2}$$

where, ρ_f and μ_f are the coolant density and viscosity, p is the coolant pressure.

- Energy equation for fluid:

$$\rho_f c_{p,f}\left(\vec{V} \cdot \nabla\right)T_f = k_f \nabla^2 T_f \tag{3}$$

where, T_f , $c_{p,f}$ and k_f are the temperature, specific heat and thermal conductivity of the coolant, respectively.

- Energy equation for solid part:

$$k_s \nabla^2 T_s = 0 \tag{4}$$

where, k_s and T_s are the thermal conductivity and temperature for solid, respectively.

For all cases investigated, inlet temperature, T_{in} of 300K is set at inlet and zero pressure outlet P_{out} is set at the outlets, adiabatic surface is assumed for top, front and back solid surfaces.
The boundary conditions are as follows:

at inlets:

$$v = V_{in},\ u = w = 0, T_f = T_{in} \tag{5}$$

at outlets:

$$P = P_{out},\ \frac{\partial T_f}{\partial y} = 0 \tag{6}$$

For fluid-solid interface:

$$\vec{V} = 0, \quad T_s = T_f, \quad k_s \nabla T_s = k_f \nabla T_f \tag{7}$$

The SIMPLE algorithm is used as the pressure–velocity coupling equation. The convection–diffusion formulation of the momentum and energy equation of the fluid was solved by using second-order upwind scheme. The convergence criteria of residuals for resolved continuity, velocity, and energy equation is set to below 10^{-5}.

2.3 Mathematical Formulation for Thermal Resistance Network

To establish the thermal resistance network and correlation for the DL-MCHS, it is assumed that each layer is consist of two section, i.e. the section of horizontal rib with thickness t and the section of vertical fin-fluid passage with height of H_{ch}.

Based on the energy equation for hydrodynamic fully developed region, the longitudinal (*y*-axis) conduction effect is negligible compared to the axial (*x*-axis) convection and axial (*x*-axis) conduction [24]. The local thermal resistance,R_y can be calculated based on axial conduction and convection in axial direction (*x*-axis). Assuming one-dimensional heat transfer at the horizontal rib and vertical fin of each layer, R_y is obtained with procedures presented hereafter.

3. Results

3.1 Thermal Resistance Correlation

Figure 2 shows the heat flow within a half of a repeated section of the DL-MCHS at location $y(0 \leq y \leq \mathrm{L})$. Uniform heat flux is applied at the bottom surface of the heat sink and consider total amount of heat $\mathrm{Q}_{1,y}$ is first conducted through the horizontal rib of the bottom layer, which can be written as:

$$Q_{1,y} = \frac{k_s A_y}{t}\left(T_{1,y} - T_{2,y}\right) = \frac{T_{1,y} - T_{2,y}}{\frac{t}{k_s A_y}} \tag{8}$$

where, $A_y = NWy$ is the based area of heat sink from $y = 0$ till the location y.

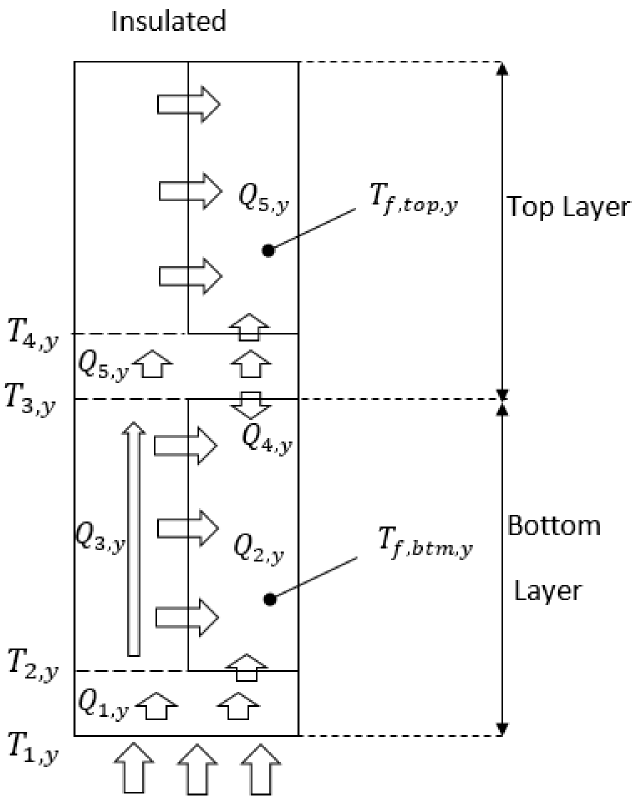


Fig. 2. The heat transfer flow within DL-MCHS

The heat is then transferred from the horizontal rib to the second section consist of vertical fin and fluid passage. Part of the heat of amount $Q_{2,y}$ is transferred into the fluid through the solid-fluid interface while the remaining heat $Q_{3,y}$ is transferred to top of the vertical fin of bottom layer.

Using the fin theory, convection can be presented in terms of $T_{2,y}$ and $T_{3,y}$. With that, $Q_{2,y}$ transferred through the fluid-solid interface can be obtained as follows:

$$Q_{2,y} = N\left(T_{2,y} - T_{f,btm,y}\right)\delta_{btm,y} + N\left(T_{3,y} - T_{f,btm,y}\right)\delta_{btm,y} + h_{btm,y}A_y\left(T_{2,y} - T_{f,btm,y}\right)$$
$$= \frac{T_{2,y}-T_{f,btm,y}}{\frac{1}{N\delta_{btm,y}}} + \frac{T_{3,y}-T_{f,btm,y}}{\frac{1}{N\delta_{btm,y}}} + \frac{T_{2,y}-T_{f,btm,y}}{\frac{1}{h_{btm,y}A_y}} \tag{9}$$

where, $T_{f,btm,y}$ is the averaged fluid temperature in the bottom channel at location y. $\delta_{btm,y} = \sqrt{h_{btm,y}P_{fin}k_sA_{c,y}}\frac{cosh_{btm,y}m_{btm,y}H_{ch}-1}{sinh_{btm,y}m_{btm,y}H_{ch}}$ is the heat transfer rate along the bottom fin area. $P_{fin,y} = 2(y + W_r)$ is the perimeter of a single fin, $A_{c,y} = yW_r$ is the cross-section area of a single fin and $W_r = W - W_{ch}$ is the rib width for a single channel. $Q_{3,y}$ can be expressed as follows:

$$Q_{3,y} = \frac{k_sA_{rib,y}}{H_{ch}}\left(T_{2,y} - T_{3,y}\right) = \frac{T_{2,y}-T_{3,y}}{\frac{H_{ch}}{k_sA_{rib,y}}} \tag{10}$$

where, $A_{rib,y} = W_r \cdot y \cdot N$ is total vertical rib area.

The heat amount of $Q_{3,y}$ that is transferred into the top layer horizontal rib is split into $Q_{4,y}$ and $Q_{5,y}$. $Q_{4,y}$ is transferred into the fluid from the bottom surface of top layer horizontal rib, and it is expressed as:

$$Q_{4,y} = Nh_{btm,y}A_y\left(T_{3,y} - T_{f,btm,y}\right) = \frac{T_{3,y}-T_{f,btm,y}}{h_{btm,y}A_y} \tag{11}$$

The remaining heat $Q_{5,y}$ is first conducted through the horizontal rib of the top layer and then transferred to the fluid in the top layer through convection. The convection thorough the top channel can be solved to obtain $T_{4,y}$ by fin theory with the following boundary conditions:

(i) insulated surface at the top of top layer
(ii) constant temperature at top surface of the horizontal rib at the top layer.

With that, $Q_{5,y}$ is expressed as:

$$Q_{5,y} = \frac{k_sA_y}{t}\left(T_{3,y} - T_{4,y}\right) = (N\delta_{top,y} + h_{top,y}A_y)(T_{4,y} - T_{f,top,y})$$
$$= \frac{T_{3,y}-T_{4,y}}{\frac{t}{k_sA_y}} = \frac{T_{4,y}-T_{f,top,y}}{\frac{1}{N\delta_{top,y}+h_{top,y}A_y}} \tag{12}$$

where, $\delta_{top,y} = \sqrt{h_{top,y}P_{fin,y}k_sA_{c,y}}\frac{sinhm_{top,y}H_{ch}}{coshm_{top,y}H_{ch}}$ is the heat transfer rate at the top fin area, $T_{f,top,y}$ is the averaged fluid temperature in the top channel at location y

Note that based on the energy balance of the fluid, the amount of heat transferred into the fluid will be stored in the fluid and result in fluid temperature increase. The amount of heat has been transferred to the fluid at top channel $Q_{top,y}$ and at bottom channel $Q_{btm,y}$ can be obtained as follows:

$$Q_{top,y} = \dot{m}c_{p,f}\left(T_{f,top,y} - T_{f,top,in}\right) = \frac{T_{f,top,y}-T_{f,top,in}}{\dot{m}c_{p,f}} \tag{13a}$$

$$Q_{btm,y} = \dot{m}c_{p,f}\left(T_{f,btm,y} - T_{f,btm,in}\right) = \frac{T_{f,btm,y}-T_{f,btm,in}}{\dot{m}c_{p,f}} \tag{13b}$$

The overall energy balance is presented by:

$$Q_y = Q_{1,y} = Q_{2,y} + Q_{3,y} = Q_{2,y} + Q_{4,y} + Q_{5,y} = Q_{top,y} + Q_{btm,y} \tag{14}$$

Substituting Eq. (8-14) into Eq. (15), the following can be obtained:

$$\begin{aligned}
Q_y &= \frac{T_{1,y} - T_{2,y}}{\frac{t}{k_s A_y}} \\
&= \frac{T_{2,y} - T_{f,btm,y}}{\frac{1}{N\delta_{btm,y}}} + \frac{T_{3,y} - T_{f,btm,y}}{\frac{1}{N\delta_{btm,y}}} + \frac{T_{2,y} - T_{f,btm,y}}{\frac{1}{h_{btm,y}A_y}} + \frac{T_{2,y} - T_{3,y}}{\frac{H_{ch}}{k_s A_{rib}}} \\
&= \frac{T_{2,y} - T_{f,btm,y}}{\frac{1}{N\delta_{btm,y}}} + \frac{T_{3,y} - T_{f,btm,y}}{\frac{1}{N\delta_{btm,y}}} + \frac{T_{2,y} - T_{f,btm,y}}{\frac{1}{h_{btm,y}A_y}} + \frac{T_{3,y} - T_{f,btm,y}}{\frac{1}{h_{btm,y}A_y}} + \frac{T_{3,y} - T_{4,y}}{\frac{t}{k_s A_y}} \\
&= \frac{T_{2,y} - T_{f,btm}}{\frac{1}{N\delta_{btm,y}}} + \frac{T_{3,y} - T_{f,btm}}{\frac{1}{N\delta_{btm,y}}} + \frac{T_{2,y} - T_{f,btm}}{\frac{1}{h_{btm,y}A_y}} + \frac{T_{3,y} - T_{f,btm}}{\frac{1}{h_{btm,y}A_y}} + \frac{T_{4,y} - T_{f,top,y}}{\frac{1}{N\delta_{top,y} + h_{top,y}A_y}} \\
&= \frac{T_{f,top,y}-T_{f,top,in}}{\dot{m}c_{p,f}} + \frac{T_{f,btm,y}-T_{f,btm,in}}{\dot{m}c_{p,f}}
\end{aligned} \tag{15}$$

The Eq. (15) can be converted into the thermal resistance network [16] presented in Figure 3(a). However, Wei *et al.,* [16] used an iterative method to solve the network to obtain thermal resistance, which is not efficient to predict the thermal performance. The present study simplified the network by imposing a modification of delta to wye to the network, which enable it to solve thermal resistance directly without iteration.

$R_{1,y}$ and $R_{3,y}$ represent the conductive thermal resistance of the horizontal ribs for bottom layer and top layer of the MCHS, respectively. $R_{2,y}$ represent the conductive thermal resistance along the vertical fin of bottom layer. $R_{4,y}$, $R_{5,y}$ and $R_{6,y}$ represent the convective thermal resistance between walls and coolant. $R_{7,y}$ and $R_{8,y}$ represent the capacitive thermal resistance due to the temperature increase of the coolant. The delta to wye conversion is applied and the converted thermal resistance is shown in Figure 3(b). With the modified thermal resistance network in Figure 3(b), correlation for local thermal resistance, R_y was obtained as:

$$\mathrm{R}_y = \mathrm{R}_{1,y} + \mathrm{R}_{a,y} + \frac{1}{\frac{1}{\mathrm{R}_{b,y}+\mathrm{R}_{7,y}} + \frac{1}{\mathrm{R}_{c,y}+\mathrm{R}_{3,y}+\mathrm{R}_{6,y}+\mathrm{R}_{8,y}}} \tag{16}$$

where,

$$R_{a,y} = \frac{R_{2,y}R_{4,y}}{R_{2,y}+R_{4,y}+R_{5,y}} \; ; \quad R_{b,y} = \frac{R_{4,y}R_{5,y}}{R_{2,y}+R_{4,y}+R_{5,y}} \; ; \; R_{c,y} = \frac{R_{2,y}R_{5,y}}{R_{2,y}+R_{4,y}+R_{5,y}} \tag{17a}$$

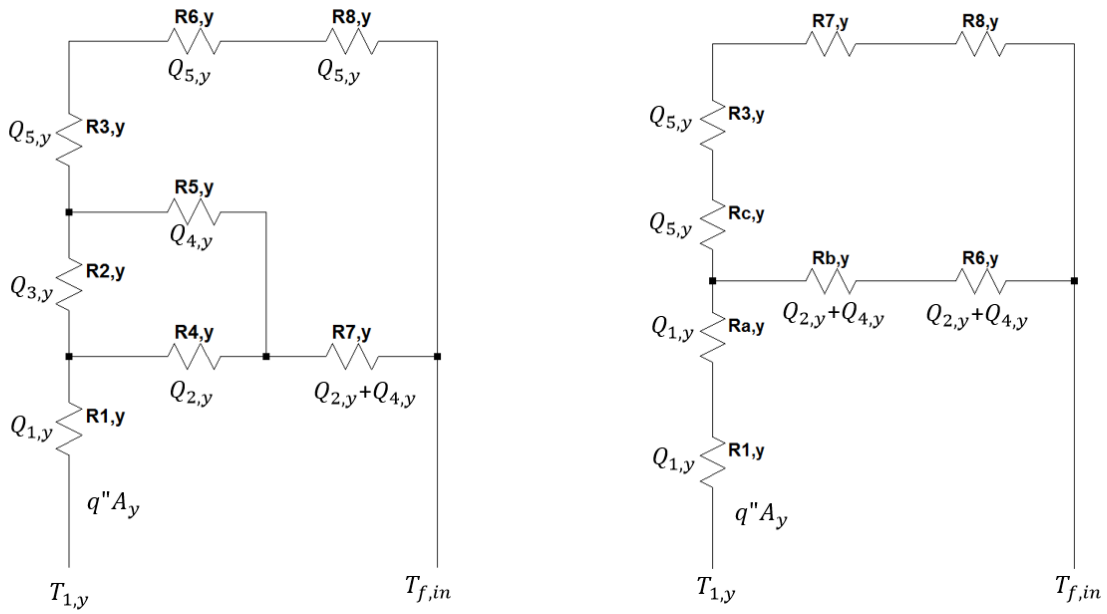
$$R_{1,y} = R_{3,y} = \frac{t}{k_s A_y} \; ; \; R_{2,y} = \frac{H_{ch}}{k_s A_{rib}} \tag{17b}$$

$$R_{4,y} = \frac{1}{N\delta_{btm,y}+h_{btm,y}A_y} \quad ; R_{6,y} = \frac{1}{N\delta_{top,y}+h_{top,y}A_y} \tag{17c}$$

$$R_{5,y} = \frac{1}{Nh_{btm,y}A_y} \qquad R_{7,y} = R_{8,y} = \frac{1}{\dot{m}c_{p,f}} \tag{17d}$$

$$h_{btm,y} = \frac{Nu_{btm,y}k_f}{D_h} \quad h_{top,y} = \frac{Nu_{top,y}k_f}{D_h} \tag{17e}$$

where, $D_h = 2(W_{ch}H_{ch})/(W_{ch} + H_{ch})$ is the hydraulic diameter of the channels.



(a) (b)

Fig. 3. (a) shows the thermal resistance network proposed by Wei *et al.,* [16] and (b) The thermal resistance after delta to wye conversion for DL-MCHS

It should be noticed that, in Eq. (17), variables t, W_{ch}, H_{ch}, W and y are the design parameters of DL-MCHS. The unknown variables $Nu_{btm,y}$ and $Nu_{top,y}$ are the Nusselt numbers for the convection at the fluid-solid interface at the bottom and top channel, respectively.

As seen in Figure 1(a), for the bottom fluid channel, the heat is transferred from four solid side walls into fluid. While for the top fluid channel, the heat is transferred from three solid side walls only into fluid as the top wall is considered as insulated wall. $Nu_{btm,y}$ can be solved using the correlation proposed by Lee *et al.,* [19] with four-sided heating condition as follows:

$$Nu_{4,y} = \frac{1}{C_1(y^*)^{C_2}+C_3} + C_4 \tag{18}$$

where, $y^* = y/(\text{RePrD}_h)$is the dimensionless axial distance, $\text{Re} = \frac{\rho_f v D_h}{\mu}$ is the Reynold's number , $\text{Pr} = \frac{C_{p,f}\mu}{k_f}$ is the Prandtl number and μ is the the dynamic viscosity of the fluid.
The coefficients in Eq. (18) are given as follows:

$$C_1 = -3.122 \times 10^{-3}\alpha^3 + 2.435 \times 10^{-2}\alpha^2 + 2.143 \times 10^{-1}\alpha + 7.325 \tag{19a}$$

$$C_2 = 6.412 \times 10^{-1} \tag{19b}$$

$$C_3 = 1.589 \times 10^{-4}\alpha^2 - 2.603 \times 10^{-3}\alpha + 2.444 \times 10^{-2} \tag{19c}$$

$$C_4 = 7.148 - 1.328 \times \frac{10^{-1}}{\alpha} - 5.936/\alpha^3 \tag{19d}$$

where, α is the channel aspect ratio. $\text{Nu}_{\text{top,y}}$ can be solved using correlation proposed by Lee [19] for three-sided heating condition as follows:

$$\text{Nu}_{top,y} = \text{Nu}_{4,y} \times (\text{Nu}_{\infty,3}/\text{Nu}_{\infty,4}) \tag{20}$$

where, $\text{Nu}_{\infty,3}$ and $\text{Nu}_{\infty,4}$ are the Nusselt numbers for fully developed flow for 3-sided heating and 4-sided heating conditions, respectively, which are expressed as follows:

$$\text{Nu}_{\infty,3} = 8.235(1 - \frac{1.883}{\alpha} + \frac{3.767}{\alpha^2} - \frac{5.814}{\alpha^3} + \frac{5.361}{\alpha^4} - \frac{2.0}{\alpha^5}) \tag{21a}$$

$$\text{Nu}_{\infty,4} = 8.235(1 - \frac{2.041}{\alpha} + \frac{3.0853}{\alpha^2} - \frac{2.4765}{\alpha^3} + \frac{1.0578}{\alpha^4} - \frac{0.1861}{\alpha^5}) \tag{21b}$$

3.2 Comparison of Correlation and Simulation Results For DL-MCHS

To check the validity of the correlation (16) obtained, the results of the correlation is compared with simulation results obtained by 3D modeling using Ansys Fluent. Before the comparisons, the simulation model is first validated with the experimental and numerical work by Wei *et al.,* [18] and the results are presented in Figure 4. As shown in Figure 4, the results of the simulation model is in good agreement with the results of Wei *et al.,* [18] with a maximum temperature difference of 1.7˚C.

Figure 5 presents the comparison of local max temperature difference ΔT_y along y direction between correlation and simulation results for DL-MCHS with different channel width,W_{ch} of $50\mu m$, $100\mu m$, $150\mu m$ and $200\mu m$. The results of ΔT_y for correlation and simulation were obtained respectively based on the following two equations:

$$\Delta T_{y,corr} = R_y \cdot q" \cdot NyW \tag{28a}$$

$$\Delta T_{y,simu} = T_{y,max} - T_{f,in} \tag{28b}$$

where, $T_{y,max}$ is the local maximum temperature long flow direction obtained by simulation.

It can also be seen in Figure 5 that value of $\Delta T_{y,,corr}$ is very close to the value of $\Delta T_{y,simu}$ for different values of W_{ch}. This indicates that the proposed correlated R_y may be employed to predict the temperature distributions along the flow direction for DL-MCHS.

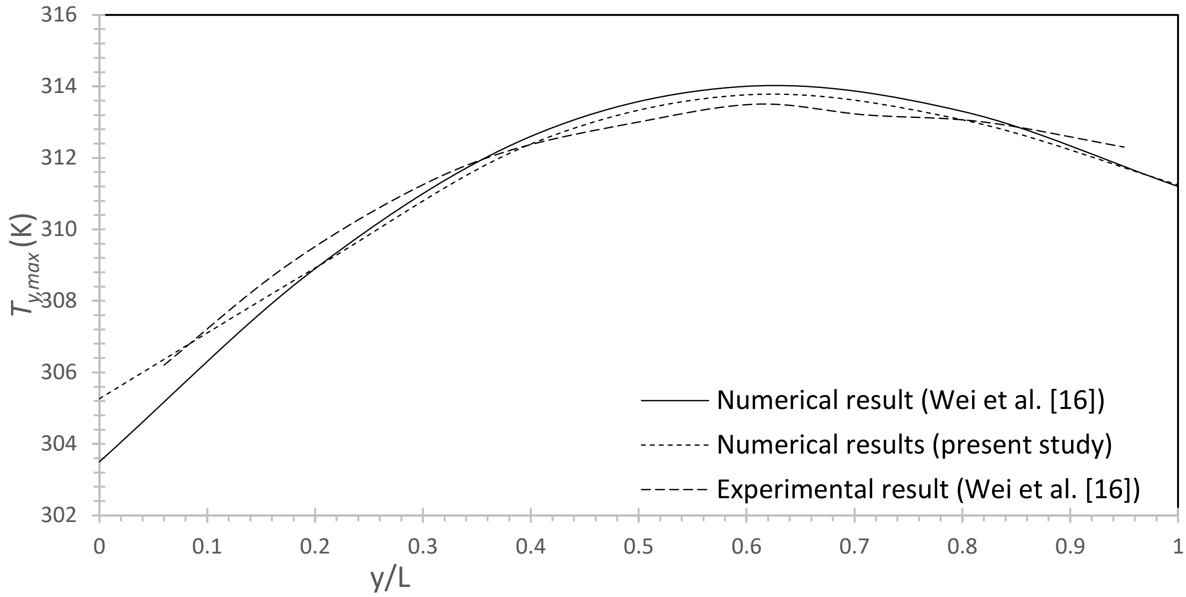


Fig. 4. The graph of max temperature along the bottom surface validated with Wei *et al.,* [16]

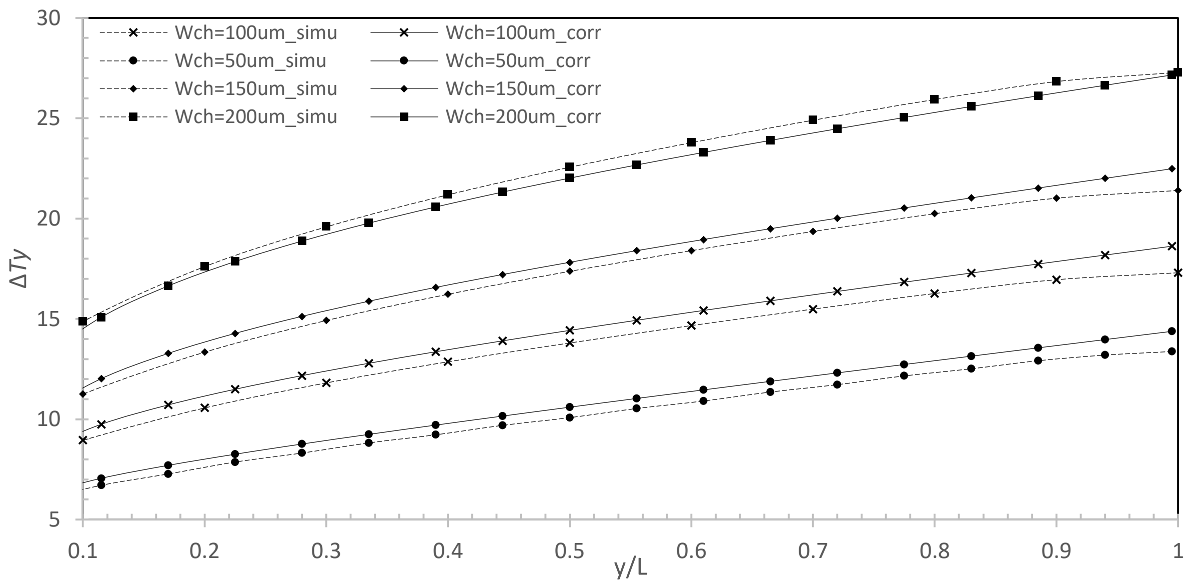


Fig. 5. The comparison of ΔT_y for DL-MCHS with different W_{ch}

4. Conclusions

In this paper, a thermal resistance network for DL-MCHS has been obtained. Modification has been carried on an existing thermal resistance network by imposing Delta to wye conversion. A correlation has been obtained from the modified thermal resistance network and solved to predict the local mean thermal resistance. Compared with ΔT_y obtained through numerical simulation results, the results of the proposed correlation results show good agreement, which indicates that this correlation could be used predict the local temperature along the flow direction for DL-MCHS with different channel width.

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