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The Thermal Performance of Three-Layered Microchannel Heat Sink with Tapered Channel Profile

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ABSTRACT

The present study numerically investigated the thermal performance for three-layered microchannel heat sink (MCHS) with horizontally tapered channel profile. The problem is solved using a three-dimensional conjugate heat transfer model. Parallel flow and counter flows configurations are investigated for their maximum thermal resistances. The effects of different tapered geometries of microchannel within the multi-layer MCHS are investigated by varying the channel inlet width and channel outlet width. The numerical results show that the three-layered MCHS with CF1 flow configuration give the lowest value of thermal resistance for low value of channel inlet size (converging channel profile) while the parallel flow configuration shows lowest value of thermal resistance among all the flow configurations at high value of channel inlet size (converging channel profile). The results also show that counter flow configurations alter the peak temperature location at the base of heat sink along the flow direction toward the center of the channel, rendering better temperature uniformity.

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.

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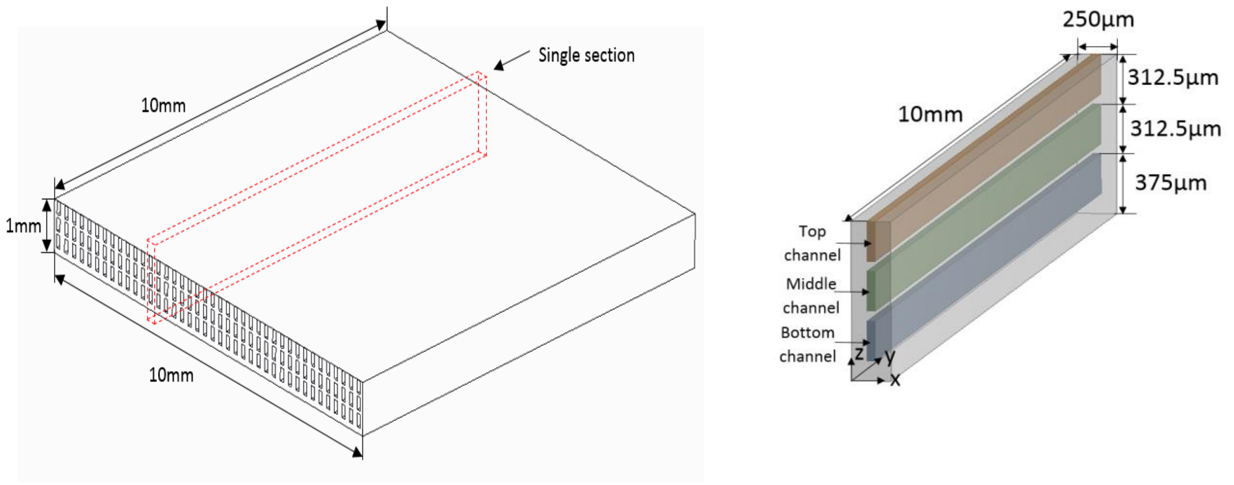
To reduce the temperature gradient along axial-direction, Vafai and Zhu [12] proposed the concept of DL-MCHS which is a substantial improvement over a conventional SL-MCHS. Following their work, other researches [12-17] have studied the performance for DL-MCHS. Wu *et al.,* [13] numerically investigated the parametric effects of channel number, aspect ratio and velocity ratio on 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. Wong *et at.,* [19] numerically investigated the thermal performance for DL-MCHS with tapered channel profile, they found that for most of the DL-MCHS designs, DL-MCHS with counter flow is found to have better thermal performance than those of parallel flow but the performance become reverse for those DL-MCHS designs having highly converging channel with small channel outlet size. These studies all confirmed that the DL-MCHS has its own advantages when compared to the SL-MCHS.

Although DL-MCHS has been investigated by many researchers [12-19], three-layered microchannel heat sink has seldom being investigated. Adewumi *et al.,* [20] presented a design method to optimize the geometry of double and three layered MCHS. Based on their results, they conclude that the DL-MCHS with counter flow under certain constrains.

Thus, the present study numerically investigated the thermal performance for three-layered MCHS with different channel inlet width, W_i and channel outlet width, W_o for different flow configurations.

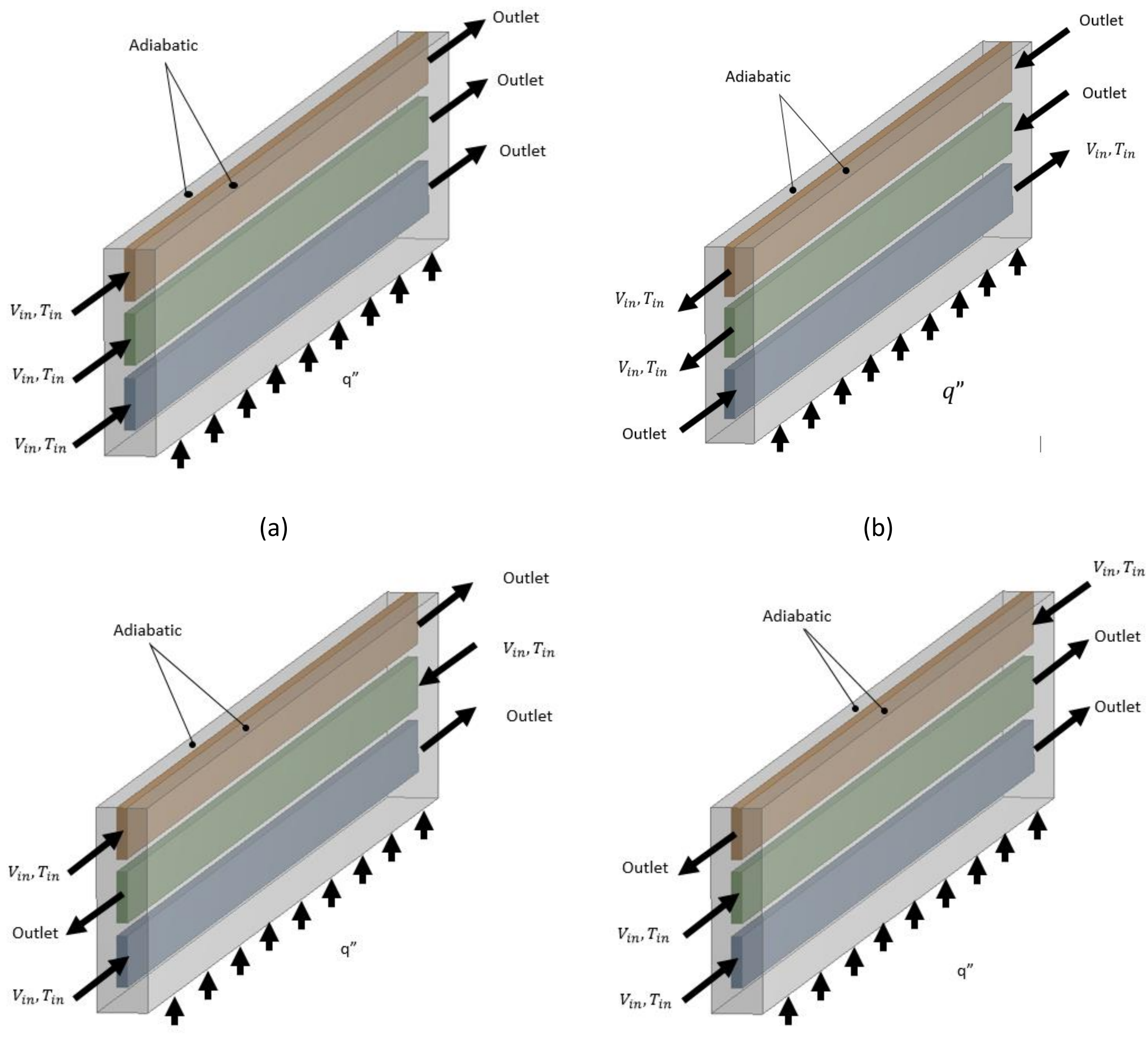
2. Methodology

The physical model of the three-layered MCHS is illustrated in Figure 1(a) with an overall width, length and height of 10 mm, 10 mm and 1 mm, respectively. The heat sink consists of 40 repeated sections with each section width *W* of 250 µm. The schematic of half of a single section is shown in Figure 1(b). The three-layered MCHS can be divided into top, middle and bottom layers with each layer consist of substrate materials and a channel. The present study considers 4 different flow configurations. One of them is the parallel flow configuration as illustrated in Figure 2(a). The rest are 3 different counter flow configurations, namely CF1, CF2 and CF3 as illustrated in Figure 2(b), Figure 2(c) and Figure 2(d), respectively. The substrate material used in the study is silicon and the coolant used is water. Note that the flow directions are illustrated with arrows in Figure 2 with fluid entering the channel inlets at different locations and with fluid exiting the channel outlets.



(a) (b)

Fig. 1. The schematic of (a) three-layered-MCHS (b) single section of three-layered-MCHS



(a) (b)

(c) (d)

Fig. 2. Schematics of boundary conditions for (a) parallel flow configuration PF, (b) counter flow configuration CF1, (c) counter flow configuration CF2 and (d) counter flow configuration CF3

Parametric studies were carried out to understand the effects of geometrical parameters of channel inlet width W_i and channel outlet width W_o. By varying the values of W_i and W_o, the tapered channel designs are obtained for investigation, eg. converging channels ($W_i > W_o$) and diverging channels ($W_i < W_o$). The study also includes straight channels ($W_i = W_o$). Based on different values of W_i and W_o , total 16 geometries of 3-layered MCHS were generated with the combination of parameters specified in Table 1. Each geometry will be set with 4 flow configurations as depicted in Figure 4. To solve the problem, 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.

Table 1
Different geometries of DL-MCHS of for parametric studies

Variables	Values
Channel inlet width, *Wi* (μm)	50, 100, 150, 200
Channel outlet width, *Wo* (μm)	50, 100, 150, 200

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:

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

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

- Energy equation for fluid:

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

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

- Energy equation for solid:

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

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

All the cases are subjected to the same boundary conditions of a uniform base heat flux *q”* of 100 W/cm^2 at the bottom surface of the heat sink, inlet temperature $T_{in}$of 300K, and an overall volumetric flow rate of 200 ml/min. All the outlet is set to be pressure outlet P_{out} with zero pressure. Top, front, and back solid surfaces of the domain are assumed to be adiabatic.

The boundary conditions are shown in Figure 2(a)-(d) for all the flow configurations. The inlet boundary conditions are stated as follows:

at inlets:

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

at outlets:

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

The boundary conditions at the fluid-solid interfaces are as follows:

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

The models were solved by using SIMPLE algorithm for pressure-velocity coupling. 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 continuity, velocity, and energy equation is set to below 10^{-5}.

3. Results

3.1 Validation

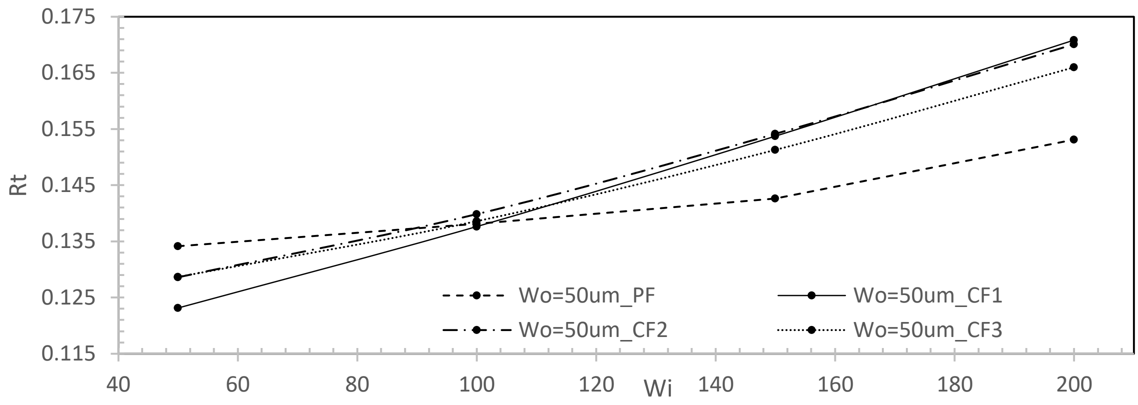
Since experimental data of 3 layered MCHS available, the numerical model of double layer MCHS is validated against the experimental and numerical results obtained by Wei *et al.,* [16], and the results has been presented in the previous paper [17] with good agreement. Then the model is further developed into 3 layer MCHS for the present study. Simulations have been conducted for three-layered MCHS with different W_i and W_o mentioned. The results of maximum bottom wall temperature were obtained. The results of thermal resistance R_T for different flow configurations were evaluated and presented hereafter, where R_T is defined as follows:

$$R_T = (T_{max} - T_{in})/q"A \tag{8}$$

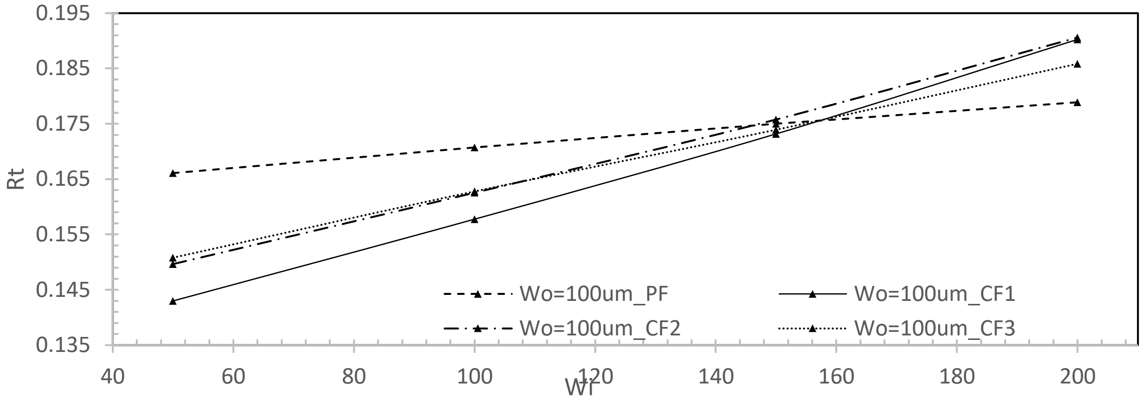
where T_{max} and A are the maximum bottom wall temperature and base area of the heat sink, respectively.

3.2 Thermal Resistance

To study the effect of W_i at fixed value of W_o for three-layered MCHS with different flow configurations, the results of R_T for W_o of $50\ \mu m$, $100\ \mu m$, $150\ \mu m$ and $200\ \mu m$ are plotted in Figure 3(a)-(d), respectively. It can be observed that, generally the increase of W_i increases the value of R_t for all the flow configurations investigated. This would mean the increase in channel inlet width leads to poorer heat transfer. The increase in channel inlet width will change the channel geometry from diverging channel to converging channel if the channel outlet width is unchanged. It is evident in Figure 3(a) that, the geometry with smallest value of Wo and Wi gives the lowest value of R_T,. The CF1 flow configuration provide the best thermal performance among the 4 flow configurations when Wi is small as observed in Figure 3(a). However, the PF flow configuration gives the lowest value of R_T when Wi increases to high value, which can be observed in Figure 3(a) and Figure 3(b).



(a)



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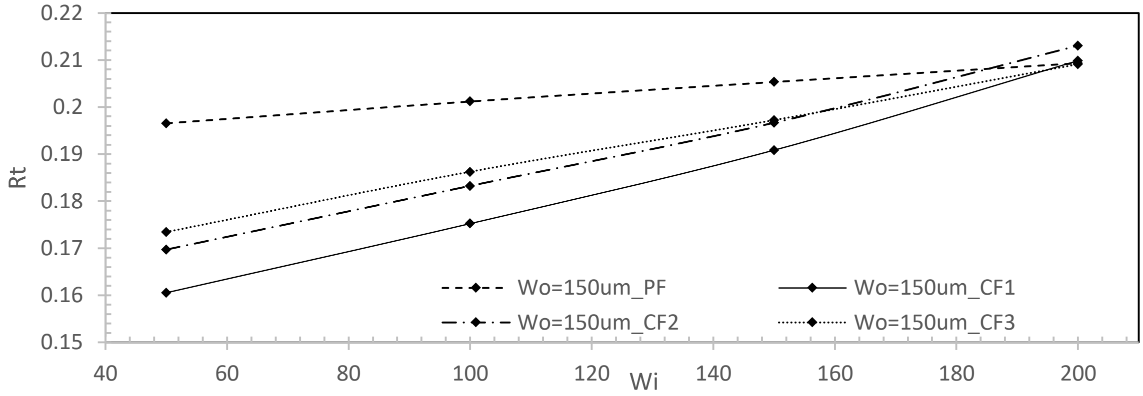

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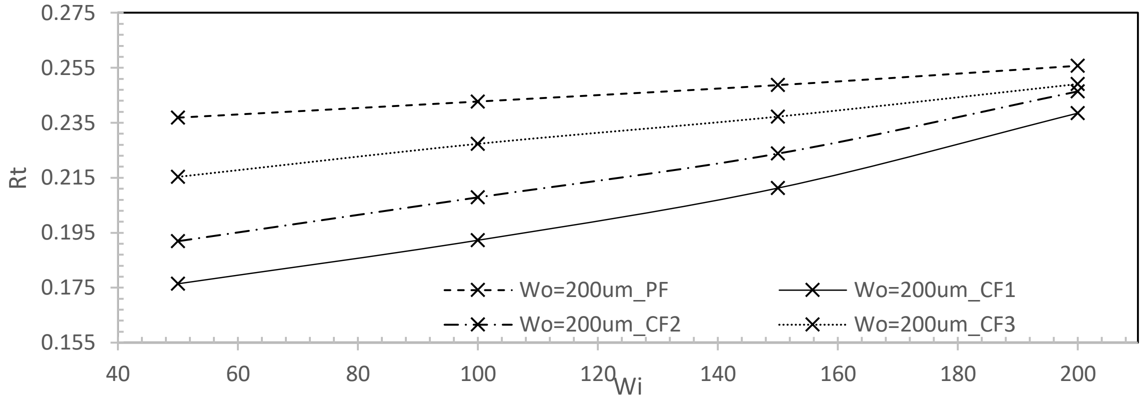


(d)

Fig. 3. Comparison of thermal resistance among PF, CF1, CF2 and CF3 flow configurations for (a)$W_o = 50\mu m$, (b)$W_o = 100\mu m$, (c)$W_o = 150\mu m$ and (d)$W_o = 200\mu m$

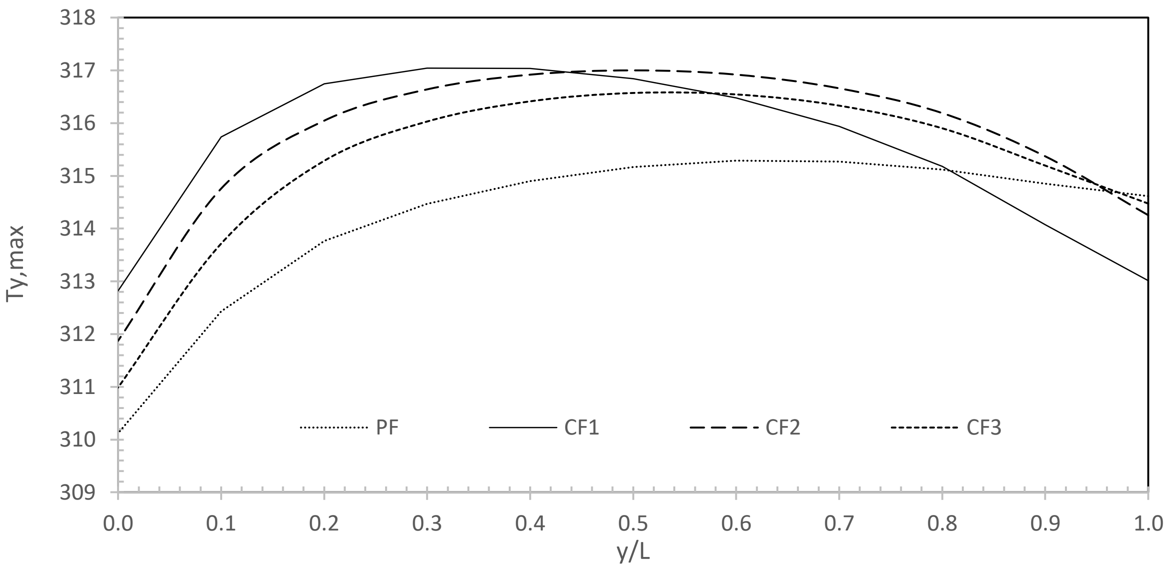
3.3 Temperature Distribution

To understand how tapered channel profile has affected the thermal performance, Figure 4(a), 4(b) and 4(c) are plotted to show the local variation of temperature , $T_{y,max}$ at the bottom surface of the heat sink along the flow direction for (i) converging channel geometry of $W_i = 200\mu m$ $W_o = 50\mu m$, (ii) straight channel geometries of $W_i = 50\mu m$ $W_o = 50\mu m$ and (iii) diverging channel geometry of $W_i = 50\mu m$ $W_o = 200\mu m$. $T_{y,max}$ here is the local maximum temperature at the bottom wall along the flow direction.

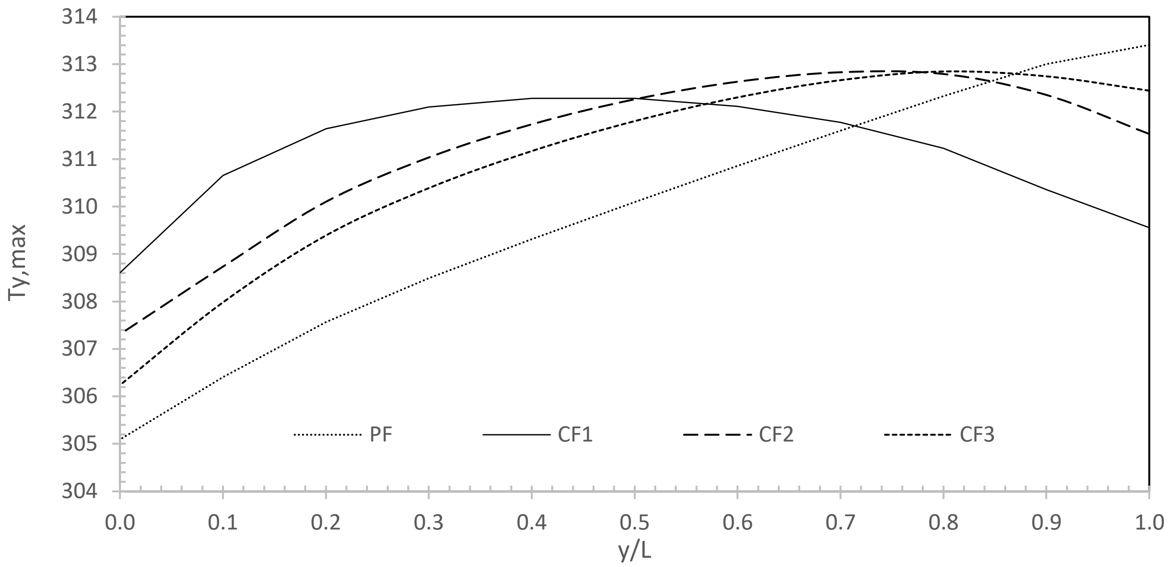
The trend of the temperature variation along the *y* direction for the 4 flow configurations are analyzed. It is found that, the temperature $T_{y,max}$ for parallel flow configuration increases from the inlet towards the outlet until it attains the peak temperature at the outlet as shown in Figure 4(b) and 4(c). The trend for PF shown in Figure 4(a) is similar as $T_{y,max}$ increases from inlet, with the peak close the channel exit, but not exactly at the exit. There is lightly difference for PF in Figure 4(a) as the curve becomes nearly horizontal from the middle towards the end of the channel.

When the counter flows (CF1, CF2 and CF3) are compared to the parallel flow (PF), it is found that, the counter flows have the location of peak temperature near the center of the channel rather than near the channel exit. This means the counter flows are able to moderate the heat transfer patterns to attain more balance heat transfer between both end of the channels. With such moderation, the counter flows are able to reduce the peak temperature as compared to the parallel flow as observed in Figure 4(b) (straight channel profile) and Figure 4(c) (converging channel profile), and improves the temperature non-uniformity.

When compared among the counter flows of CF1, CF2 and CF3, CF1 gives the lowest peak temperature for the case of straight channel (Figure 4(b)) and for the case of diverging channel (Figure 4(c)), which directly improves the thermal resistance. In Figure 4(c), CF3 gives the lowest peak temperature among counter flows.



(a)



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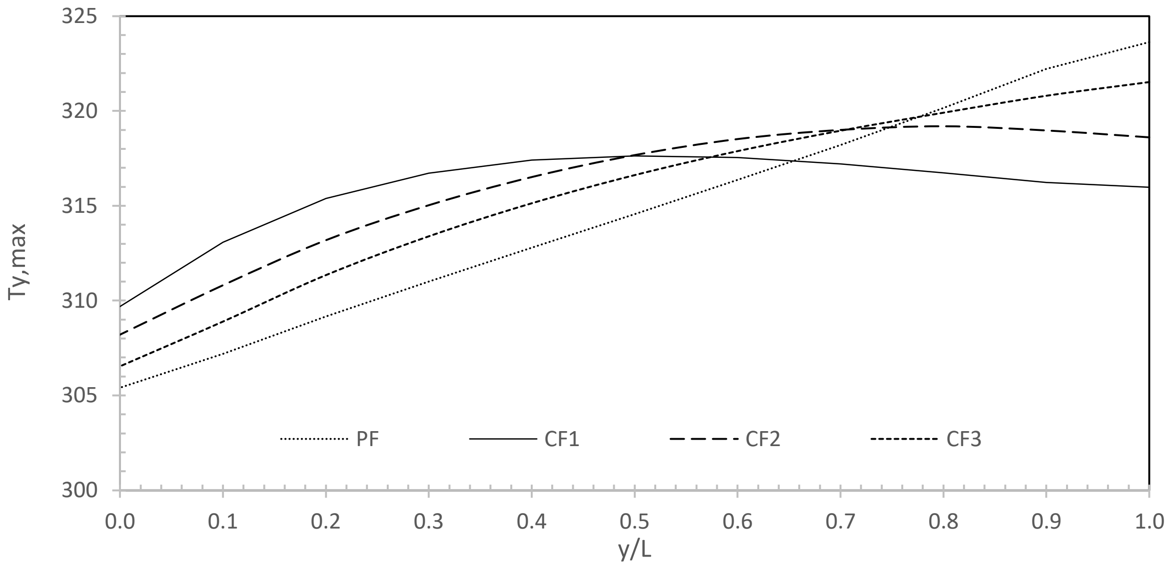

(c)

Fig. 4. Comparison of $T_{y,max}$ among 4 flow configurations for case of (a) converging channel with $W_i = 200\mu m$ $W_o = 50\mu m$, (b) straight channel with $W_i = 50\mu m$ $W_o = 50\mu m$ and (c) diverging channel with $W_i = 50\mu m$ $W_o = 200\mu m$

4. Conclusions

The thermal performance of three-layered MCHS with horizontally tapered channels have been investigated for PF, CF1, CF2 and CF3 flow configurations for different inlet width W_i at dfferent outlet width $W_o$through simulation. The thermal resistance and base temperature for counter and parallel flow has been analysed. The conclusion can be drawn as follows:

- For low value of channel inlet size (converging channel profile) in three-layered MCHS, CF1 flow configuration gives the lowest value of R_T
- For high value of channel inlet size (converging channel profile), parallel flow configuration gives the lowest value of R_T

- The counter flow configuration alters the temperature variation pattern of the parallel flow by moving the peak temperature location toward the center of the channel, giving better temperature uniformity.
- The three-layered MCHS with CF1 flow configuration demonstrate best temperature uniformity with lowest peak temperature for the case of straight and converging channel profile.

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