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Numerical Investigation of Tapered DL-MCHS with Alternating Flow Configuration

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ABSTRACT

The present study numerically investigated the thermal and hydraulic performances of width-tapered Double-layered microchannel heat sink (DL-MCHS) with alternating flow configuration. Three-dimensional conjugate heat transfer model has been used to study the performances of different geometries of tapered microchannel within the DL-MCHS by varying the inlet width and outlet width of channels. The DL-MCHS with alternating flow configuration shows an increase in hydraulic performance and a decrease in thermal performance with the increase in inlet channel width or outlet channel width. The maximum temperature difference and temperature distribution on the bottom surface of the heat sink along the flow direction has been studied to find the effects of channel inlet width and channel outlet width on temperature uniformity. Better temperature uniformity was observed for DL-MCHS with small inlet channel width or outlet channel width under alternating flow. FOM (Figure of merit) is used to evaluate the thermo-hydraulic performance among MCHS with different geometries of tapered channel designs under alternating flow. As compared to the benchmark of DL-MCHS with straight channel design with inlet and outlet channel size of 50µm, the optimum DL-MCHS with tapered channel design shows 76% improvement in thermo-hydraulic performance. The alternating flow configuration has also been compared to both parallel and counter flow configurations. The DL-MCHS with alternating flow shows better thermal performance for the case of higher channel inlet size or smaller channel outlet size as compared with parallel flow configuration. However, the alternating flow configuration and counter flow configurations show large similar thermal performance.

Keywords:
Double-layer MCHS; Alternating flow; Thermos-hydraulic performance; Temperature uniformity

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1. Introduction

Nowadays, the problem of heat dissipation at high heat flux has received much attention due to the fast development of electronic components. As an attempt to solve this issue, Tuckerman and Pease [1] proposed the design of silicon-based single-layered microchannel heat sink (SL-MCHS), based on their experimental result, they found that by using water as coolant, the SL-MCHS can can achieve maximum thermal resistance, R_t of 0.09˚C/cm2 when subject to the heat flux q" of 790

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W/cm^2. From there onwards, scientists [2-7] have studied the performance for MCHSs by utilizing porous material as part of the substrate material 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.

Vafai and Zhu [12] proposed the concept of DL-MCHS which is a substantial improvement over a conventional single-layer MCHS by reducing the temperature gradient along axial-direction. 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 different design variables. 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 al.,* [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 with parallel and counter flow configuration has been investigated by many researchers [12-19], DL-MCHS with tapered channel profile and with alternating flow configuration has seldom being investigated. Thus, the present study numerically investigated the thermal, hydraulic and thermo-hydraulic performance for DL-MCHS with alternating flow configuration for different $\mathrm{W_i}$ and $\mathrm{W_o}$.

2. Methodology

The physical model of the DL-MCHS is illustrated in Figure 1 (a) with an overall width, $\mathrm{L_x}$ length, $\mathrm{L_y}$ and height, $\mathrm{L_z}$ of $10mm$, $10mm$ and $1mm$, respectively. The heat sink consists of 40 repeated sections with each section width W of 250μm. The geometrical parameters for a single unit is shown in Figure 1 (b). Each unit has identical top and bottom layer dimensions and each unit can be divided into left and right part. Each layer has channel height, $\mathrm{H_{ch}}$ of 375 $\mu\mathrm{m}$ and horizantal rib thickness, t of $125\mu\mathrm{m}$. For the left part and right part, the fluid flows along y-direction and negative y-direction, respectively. The computational domain of the single section is illustrated in Figure 1 (c).

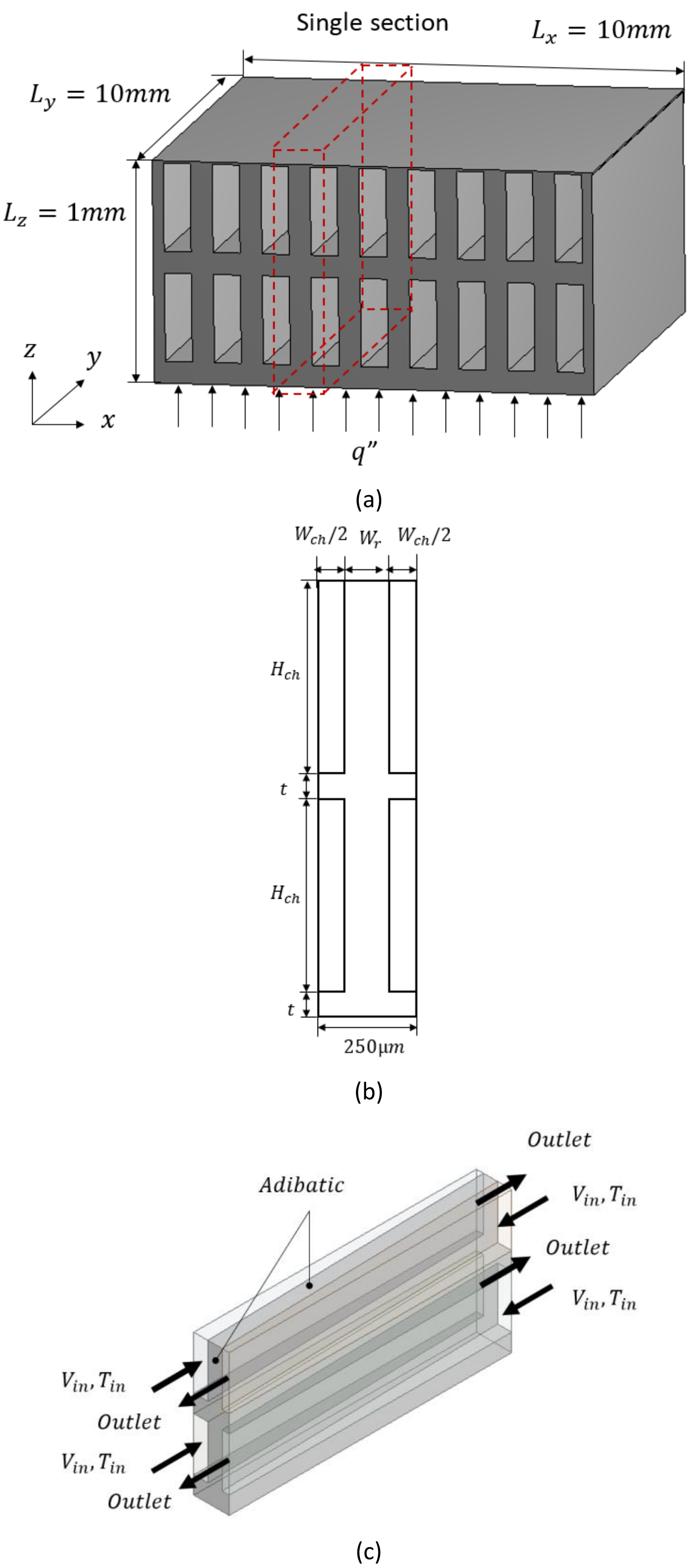


Fig. 1. (a) Schematics of double-layer MCHS (b) Repeating section of the DL-MCHS with alternating flow (c) Schematics of computational domain

Table 1
Geometrical Parameters used for DL-MCHS with alternating flow

Parameter	Minimum Value	Maximum Value
$\mathbf{W_i}(\mu\mathbf{m})$	50	200
$\mathbf{W_o}(\mu\mathbf{m})$	50	200
$\mathbf{W_r}(\mu\mathbf{m})$	50	200

Parametric studies were carried out to understand the effects of geometrical parameters channel outlet width, W_i and channel inlet width, W_o. In present study, W_o and W_i are both varied from $50\ \mu m$ to $200\ \mu m$ with step size of $10\ \mu m$. By varying W_i and W_o, the channel vertical rib thickness also varied from $50\mu m$ to $200\mu m$ with constant W of $500\mu m$. Based on different value of W_i and W_o , total 256 geometries were generated with the combination of parameters specified in Table 1. By varying the value of the inlet channel width W_i and outlet channel width, W_o, the MCHS designs with converging channel ($W_i > W_o$), straight channel ($W_i = W_o$) and then diverging channel ($W_i < W_o$) are created for investigation. 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.

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

Cotinuity 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 $100\ \mathrm{W/cm^2}$ at the bottom surface of the heat sink, inlet temperature T_{in} of 300K, and an overall volumetric flow rate,$\dot{Q}$ 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. Left and right surfaces of the domain are set to be symmetry.

The boundary conditions are shown in Figure 1 (c) for DL-MCHS with alternating flow configuration. The inlet boundary conditions are stated as follows:

At inlets:

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

The outlet boundary conditions are stated as follows:

At outlets:

$$P = P_{out} \tag{6}$$

The boundary conditions at the fluid-solid interface 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 model 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 resolved continuity, velocity, and energy equation is set to below 10^{-5}.

Other parameters used to evaluate the performance of MCHSs are described as follows:

Total pumping power:

$$\Omega = \dot{Q}_{bottom}\Delta P_{bottom} + \dot{Q}_{top}\Delta P_{top} = v_{in,bottom}W_{ch}H_{ch}\Delta P_{bottom} + v_{in,top}W_{ch}H_{ch}\Delta P_{top} \tag{8}$$

Average heat transfer coefficient, h_m:

$$h_m = \frac{q"}{(\bar{T}_b - T_{in})} \tag{9}$$

Thermal resistance, R_t, which assesses the heat dissipation capability of MCHSs:

$$R_t = \frac{T_{b,max} - T_{in}}{q"L_xL_y} \tag{10}$$

Maximum temperature difference on the bottom surface of the heat sink, ΔT_b, which represents the cooling uniformity of MCHSs:

$$\Delta T_b = T_{b,max} - T_{b,min} \tag{11}$$

Where $\overline{T}_b$ is the average wall temperature wall temperature along the centreline of the bottom surface of the MCHS. $T_{b,max}$ and $T_{b,min}$ are the maximum and minimum bottom wall temperature on the bottom surface of the heat sink.

To compare the thermal-hydraulic performance between different MCHS designs, Figure of merit (FOM) is applied to determine the improvement of one design in terms of heat transfer and pumping power at the same time:

$$FOM = \frac{(h_{m,new}/h_{m,base})}{(\Omega_{new}/\Omega_{base})^{1/3}} \tag{12}$$

3. Results

3.1 Thermal and Hydraulic Performance

The code has been validated in the previous work [19]. The thermal performance in terms of thermal resistance, R_t and hydraulic performance in terms of pressure drop, ΔP plotted in log_{10} scale for DL-MCHS with different value of W_i and W_o under alternating flow is shown in Figure 2 and Figure 3, respectively. As can be seen from the Figure 2 that, the design with small W_i or W_o gives lower R_t and R_t increases with either W_i or W_o increases, where the lowest value of R_t appears at design with $W_i = 50\mu m$ $W_o = 50\mu m$ and highest value of R_t appears at design with $W_i = 200\mu m$ $W_o = 200\mu m$. As for hydraulic performance as shown in Figure 3, opposite trend can be found from R_t that, the design with small W_i or W_o gives higher ΔP and ΔP decreases with the increase of either W_i or W_o increase. When the W_i or W_o is small, the decrement of ΔP is dramatic and the decrement becomes less significant with increase of either W_i or W_o. The lowest value of ΔP appears at design with $W_i = 200\mu m$ $W_o = 200\mu m$ and highest value of ΔP appears at design with $W_i = 50\mu m$ $W_o = 50\mu m$. This is as expected, since small channel width gives better thermal performance but in sacrifice of hydraulic performance [19].

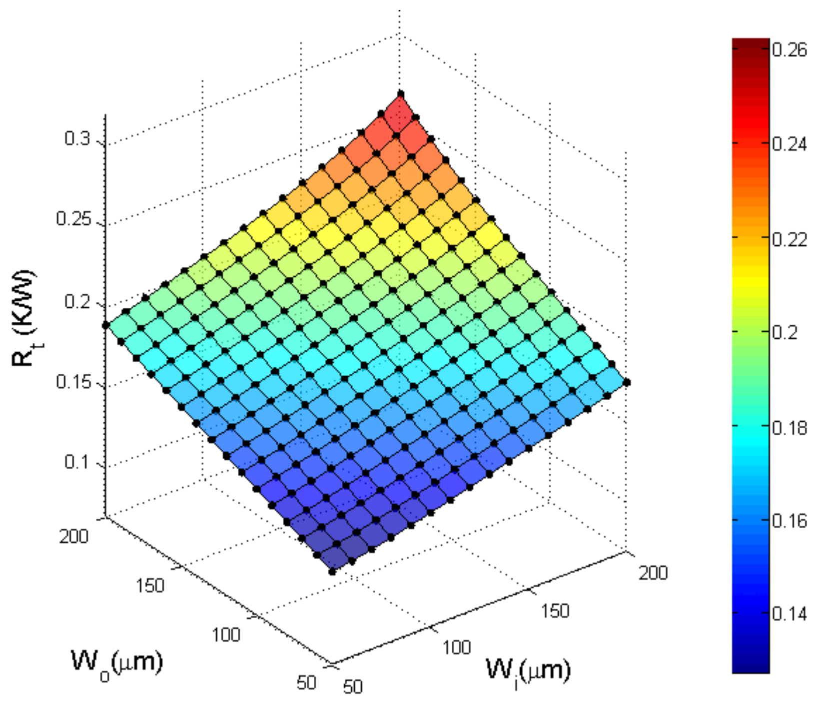


Fig. 2. The thermal resistance for DL-MCHS with alternating flow configuration

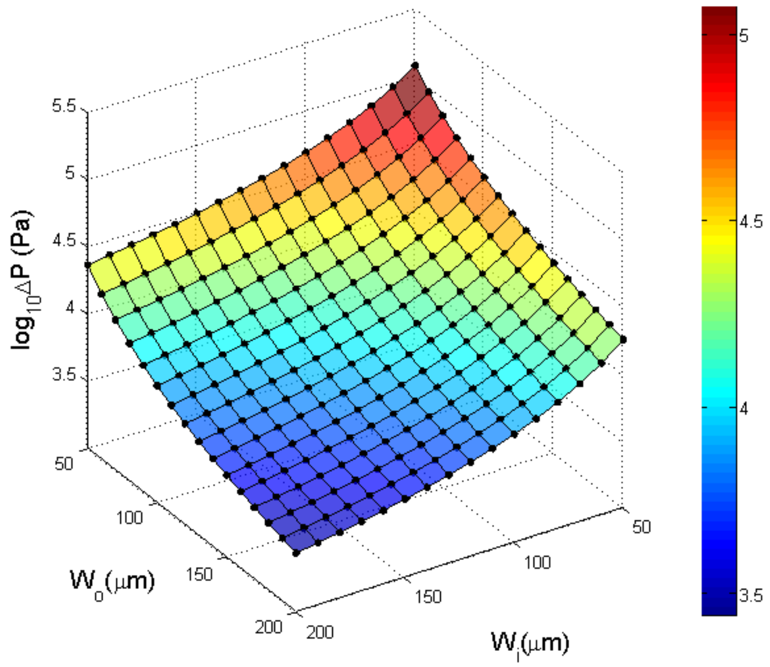


Fig. 3. The pressure drop for DL-MCHS with alternating flow configuration

To see the effect of different flow configurations on thermal performance, the results of R_t for DL-MCHS with alternating flow configuration is compared to our previous study's results [19]. Totally 32 designs with 4 different values of W_i of and 4 different values of W_o for DL-MCHS with parallel and counter flow configuration is used for comparison as shown in Figures 4 and 5, respectively. As can be seen from Figure 4 that, the DL-MCHS with alternating flow configuration shows lower value of R_t when W_i at high values or W_o at small values as compared to the DL-MCHS with parallel flow configuration. For all designs investigated, the alternating flow configuration with design of $W_i = 50\ \mu m\ W_o = 200\ \mu m$ shows 0.065K/W lower values of R_t as compared to the DL-MCHS with parallel flow configuration. As for DL-MCHS with counter flow configuration shown in Figure 5, the DL-MCHS with alternating flow configuration shows largely similar results as compared to the DL-MCHS with counter flow configuration.

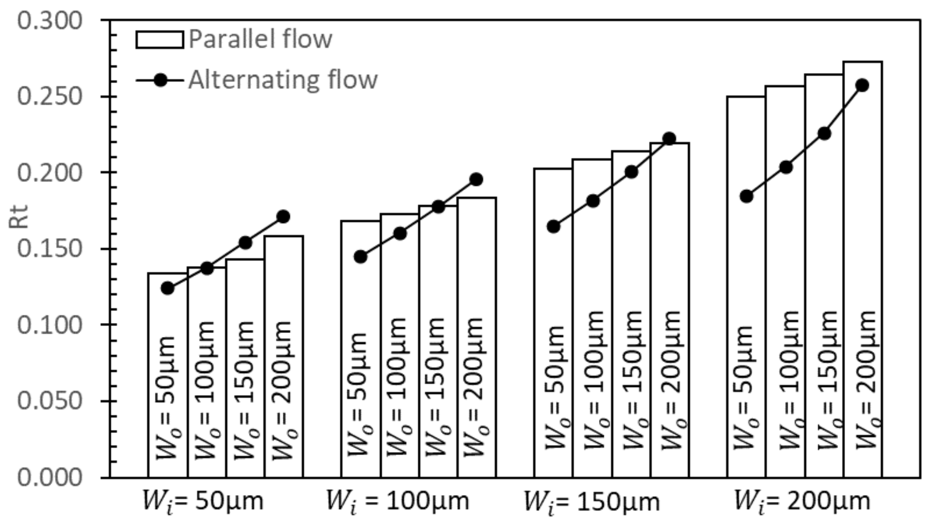


Fig. 4. The comparison of R_t between DL-MCHS with alternating and parallel flow configuration

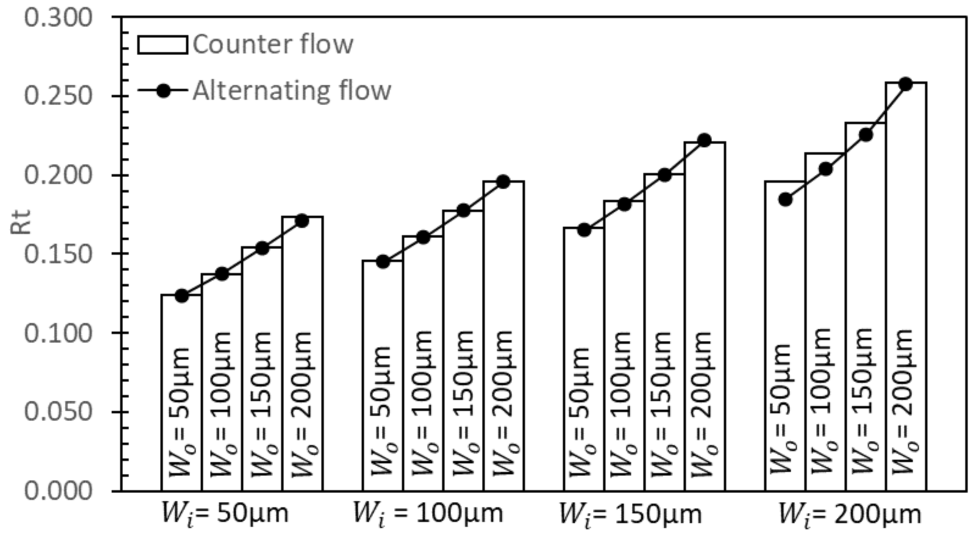


Fig. 5. The comparison of R_t between DL-MCHS with alternating and counter flow configuration

3.2 Temperature Uniformity

Besides of thermal resistance, ΔT_b is another parameter to evaluate the thermal performance in terms of temperature uniformity. Figure 6 shows the ΔT_b for DL-MCHS with alternating flow configuration and with different value of W_i and W_o. First looking into the effect of W_o, for any given value of W_i, ΔT_b increases with the increase of W_o, which $W_o = 50\mu m$ has the lowest value of ΔT_b and $W_o = 200\mu m$ has the highest value of ΔT_b for all values of W_i investigated. This indicate that the smaller W_o gives better temperature uniformity. Then looking into the effect of W_i, for any given value of W_o, ΔT_b decreases with the increase of W_i until it reaches the lowest value of ΔT_b and then increases. The lowest value of ΔT_b for different W_o varies from $W_i = 80\mu m$ to $W_i = 100\mu m$. This indicate that the design with small channel inlet size has better temperature uniformity. For all cases investigated, the DL-MCHS design with $W_i = 80\mu m$ $W_o = 50\mu m$ under alternating flow has the lowest ΔT_b of 3.297°C.

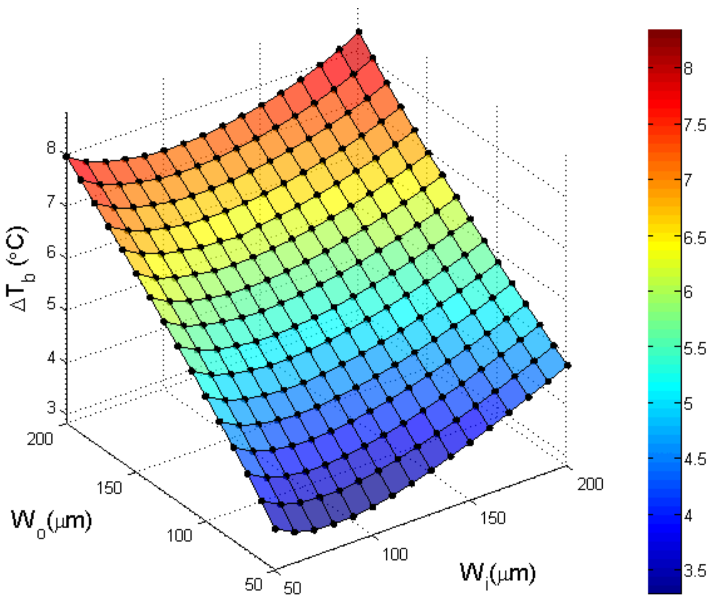
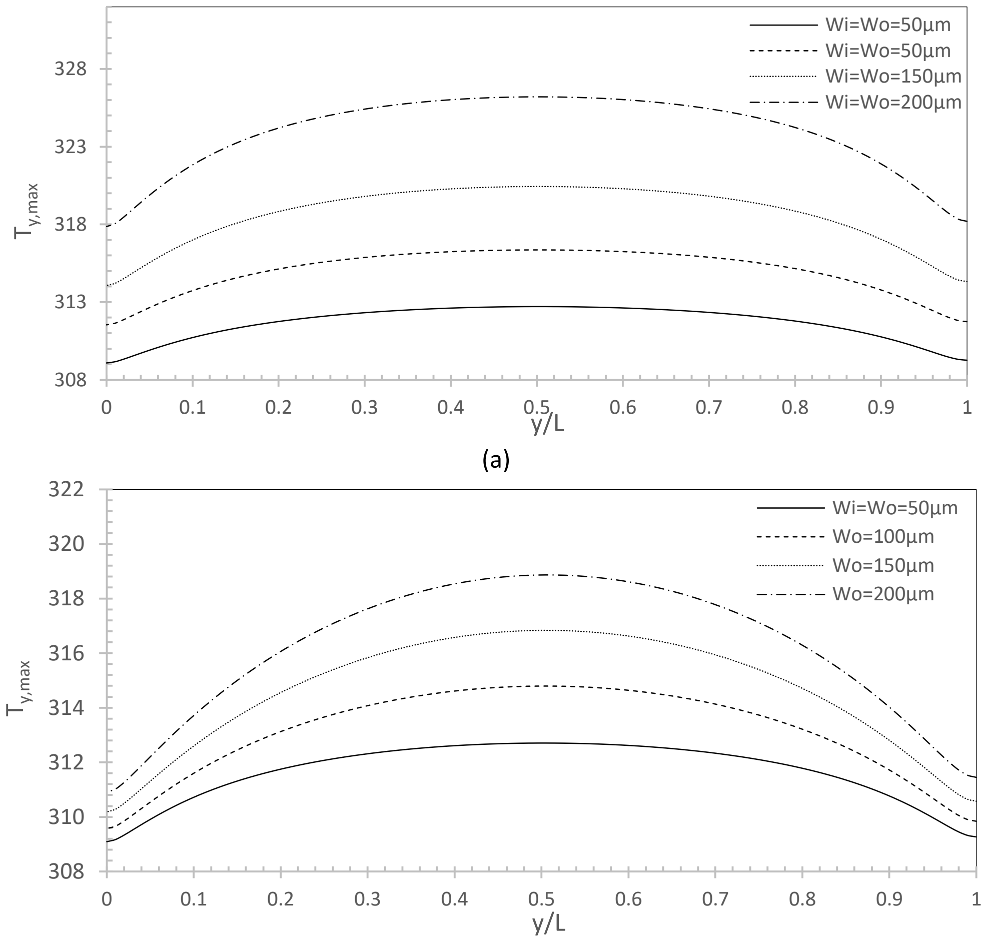
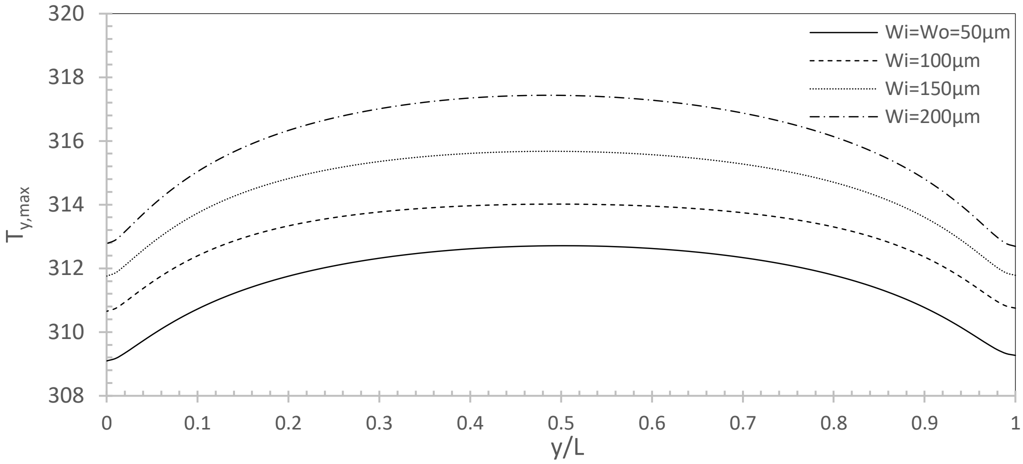


Fig. 6. The ΔT_b for DL-MCHS with alternating flow configuration

To explain the above results, the temperature distribution on the bottom surface of the heat sink along the flow direction, $T_{y,max}$ for DL-MCHS with alternating flow and different W_i and W_o is shown in Figure 7. As can be seen from Figure 7 that, for all designs investigated, the $T_{y,max}$ increases with the increase of y/L until $T_{y,max}$ reaches the symmetry position at y/L=0.5 and then decreases. This is as expected, since each DL-MCHS with alternating flow design has exact channel profile but different flow direction in each section's left and right part of the computational domain as mentioned in Figure 1. For DL-MCHS with alternating flow configuration and straight channel design($W_i = W_o$) as shown in Figure 7 (a), the ΔT_b increases with the increase of W_{ch}. The reason is that the increase of temperature at y/L=0.5($T_{b,max}$) is larger than the increase of temperature at y/L=0($T_{b,min}$) with the increase of W_{ch}, which indicated that $T_{b,max}$ is more sensitive to the changes of W_{ch} as compared to $T_{b,min}$. Comparing the DL-MCHS designs with alternating flow configuration and same W_i of $50\mu m$ for different W_o of $50\mu m, 100\mu m, 150\mu m, 200\mu m$ as shown in Figure 7(b), ΔT_b increases with the increase of W_o. This happens is because coolant's velocity decreases with the increase of W_o along the flow direction, which weakens the heat removal ability, greatly increases the $T_{b,max}$ and increases the ΔT_b. While comparing the DL-MCHS with alternating flow configuration and with same W_o for different W_i of $50\mu m, 100\mu m, 150\mu m, 200\mu m$ as shown in Figure 7 (c), ΔT_b first decreases and then increases with the increase of W_i. The reason is that, with the increase of W_i, the channel width change at y/L=0.5 is only half as compared to y/L=0, which gives lower temperature increase at y/L=0.5. However, $T_{b,max}$ is more sensitive to the channel width change as compared to $T_{b,min}$. As a combination of this two effects, ΔT_b shows a decrease with the increase of W_i when W_i is small and shows an increase with the increase of W_i after W_i reaches a certain value.



(b)



(c)

Fig. 7. Temperature distribution for DL-MCHS with alternating flow configuration with (a) Straight channel design (b) tapered channel design with fixed W_i of 50 µm (c) tapered channel design with fixed W_o of 50 µm

3.3 Thermo-Hydraulic Performance

To take into account of both thermal and hydraulic performance at the same time, Figure 8 shows the FOM for DL-MCHS with alternating flow configuration and different channel profile. For all designs investigated, the DL-MCHS with alternating flow and straight channel design of $\mathrm{W_i} = 50\ \mu\mathrm{m}\ \mathrm{W_o} = 50\mu\mathrm{m}$ is used for comparison. First looking into the effect of $\mathrm{W_i}$, for any given value of $\mathrm{W_o}$, increasing the $\mathrm{W_i}$ will increase the FOM until an optimum point is reached. After this point, increasing $\mathrm{W_i}$ results in a decrease of FOM. Then looking into the effect of $\mathrm{W_o}$, at low value of $\mathrm{W_i}(\mathrm{W_i} < 140\mu\mathrm{m})$, the FOM increases with the increase of $\mathrm{W_o}$. For higher value of $\mathrm{W_i}$ $(\mathrm{W_i} \geq 140\mu\mathrm{m})$, the FOM of the design with $\mathrm{W_o} = 180\ \mu\mathrm{m}, 190\ \mu\mathrm{m}, \text{and } 200\ \mu\mathrm{m}$ decrease rapidly with the increase of $\mathrm{W_i}$. This indicated that the DL-MCHS with alternating flow configuration with designs of larger $\mathrm{W_o}$ have better thermos-hydraulic performance when $\mathrm{W_i}$ is small. This happens is because when $\mathrm{W_i}$ and $\mathrm{W_o}$ is small, increase $\mathrm{W_i}$ or $\mathrm{W_o}$ only slightly increases the $\mathrm{R_t}$, but dramatically decreases the $\Delta\mathrm{P}$ as shown in Figures 2 and 3. While keep increasing $\mathrm{W_i}$ or $\mathrm{W_o}$, the decrements of $\Delta\mathrm{P}$ becomes less significant as shown in Figure 2. When $\mathrm{W_i}/\ \mathrm{W_o}$ increased till a certain value, the increment of hydraulic performance can't cover the decrement of the thermal performance, which results in a decrease in the thermo-hydraulic performance. For all cases investigated, The value of FOM is larger than 1 and the design of $\mathrm{W_i} = 160\mu\mathrm{m}\ \mathrm{W_o} = 190\mu\mathrm{m}$ gives the highest value of FOM of 1.7605 when compared to the design of $\mathrm{W_i} = 50\mu\mathrm{m}\ \mathrm{W_o} = 50\mu\mathrm{m}$, which means that the design with $\mathrm{W_i} = 160\ \mu\mathrm{m}\ \mathrm{W_o} = 190\ \mu\mathrm{m}$ has the best thermo-hydraulic performance among all cases investigated.

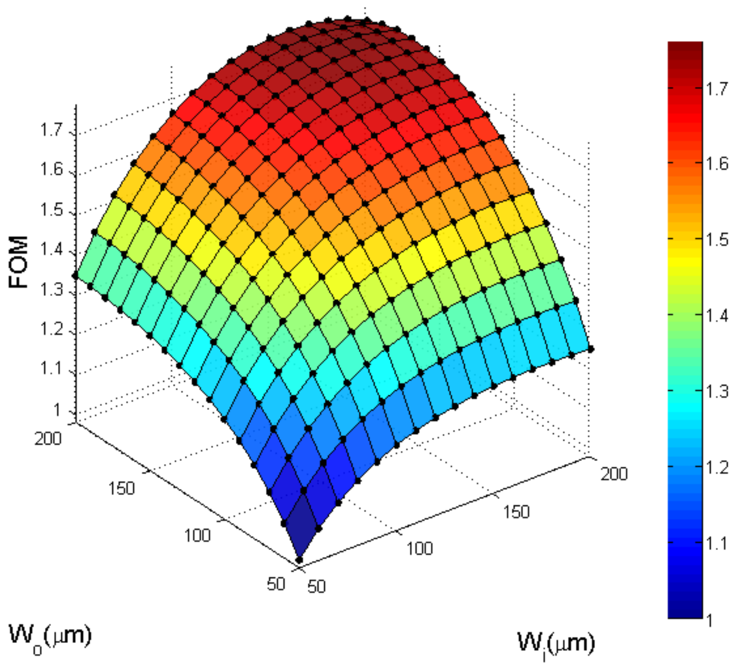


Fig. 8. FOM for DL-MCHS with alternating flow configuration

4. Conclusions

The thermal and hydraulic performance have been investigated for DL-MCHS with alternating flow and different inlet width W_i, outlet width W_o through simulation. The thermal resistance, pressure drop, temperature uniformity and FOM have been analyzed. The conclusions can be drawn as follows;

i. The design with small inlet channel width or outlet channel width gives lower thermal resistance but results in high pressure drop.
ii. The DL-MCHS with alternating flow configuration shows better thermal performance (lower thermal resistance) for the case of higher channel inlet size or smaller channel outlet size as compared to the DL-MCHS with parallel flow configuration.
iii. The DL-MCHS with alternating flow configuration shows largely similar results of thermal resistance as compared to the DL-MCHS with counter flow configuration
iv. The DL-MCHS with alternating flow design with small inlet channel width or small outlet channel width gives better temperature uniformity. The optimum DL-MCHS with alternating flow with channel design has the lowest temperature difference of 3.297°C.
v. The optimum DL-MCHS with alternating flow and tapered channel show about 76% better in thermo-hydraulic performance as compared with benchmark of DL-MCHS with straight channel design with inlet and outlet channel size of $50\mu m$.

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