CHAPTER 3: COMPUTATIONAL ANALYSIS OF NEW MICROCHANNEL AND MINICHANNEL HEAT SINK CONFIGURATIONS

3.1 INTRODUCTION

The importance of heat sinks in ensuring the optimum performance of devices is well realized as the minichannel heat sinks and microchannel heat sinks are used in the thermal management of fuel cells, battery packs, laser devices, electronic devices, etc. For instance, the space between cathode and anode in membrane fuel cells comprise cooling channels, often referred as bipolar plate. Bipolar plates are made from glass, silicon, graphite and composite materials. Bipolar plates consist of microchannels that serve as conduits for flow of coolant, enabling removal of heat [103].

The survey of literature presented in the Chapter 2 revealed the importance of channel dimensions, shape and the heat sink configuration. Hence heat sink designers attempt to arrive at different heat sink designs by tuning these factors such that the heat removal performance of the sink is enhanced. Numerical investigations are normally performed to optimize channel dimensions and design, before verifying the same through heat transfer experiments.

As a part of this work, several new configurations of microchannel heat sinks were developed conceptually. These included (i) two heat sink designs containing serpentine microchannels of two different channel widths, (ii) a heat sink with spiral microchannels with one coolant inlet and coolant outlet located at diagonally opposite corners, (iii) a heat sink with spiral microchannels with coolant inlet from the top at the center and two coolant outlets at the diagonally opposite corners, (iv) two heat sink designs dividing the total heat sink into four compartments comprising parallel microchannels and dedicated coolant inlet and outlet for each compartment and (v) four heat sink designs dividing the
total heat sink into four compartments comprising microchannels with miter bends (‘L’ sections) and dedicated coolant inlet and outlet for each compartment. The profile of bottom plate temperature was obtained for each design from the computational study and was used as the basis for preliminary screening. Designs that resulted in higher total hot spot areas were eliminated. Accordingly, the four heat sink designs in which the heat sink was divided into four compartments with each compartment containing microchannels with miter bends (‘L’ sections) were selected for detailed study. The performance of these four heat sink designs were compared with a conventional microchannel heat sink design reported in the literature. The new MCHS designs were extended to minichannel heat sinks (mCHS) also, maintaining a scale ratio of 10 between mCHS and MCHS.

3.2 NEW MICROCHANNEL HEAT SINK AND MINICHANNEL HEAT SINK DESIGNS

A new perspective of dividing the heat sink into four quadrants with separate coolant inlet and outlet as shown in Figure 3.1, is attempted here. This is likely to provide better thermal management in terms of substrate temperature gradient. The I-type MCHS design reported by Chein and Chen [104] has been modified with respect to microchannel length and used as conventional MCHS here and is also shown in Figure 3.1. In the work of Chein and Chen [104], the microchannel length was 10 mm, while the inlet and outlet plenums were 3 mm long. However, in this thesis, to make effective use of heat sink area, the length of inlet and outlet plenums was fixed at lower value (1 mm), which in turn led to increase of channel length by 4 mm (2 mm each near inlet and outlet plenum).
Figure 3.1. (a) Schematic diagram of different microchannel/minichannel heat sink configurations and (b) CAD models of the computational domain.

The overall length and width of the microchannel heat sink were 18 mm and 6.2 mm respectively. Channel width, channel height and fin width were fixed at 200 µm, 400 µm and 200 µm for all the four new heat sink configurations (configurations A to D) and the conventional heat sink. Each quadrant of the new MCHS consisted of

(i) an inlet plenum for supply of coolant to microchannels located in that region

(ii) an entrance region comprising of a coolant entry from the inlet plenum to parallel microchannels
(iii) an exit region comprising of coolant exit from microchannels to the outlet plenum

(iv) an outlet plenum to collect coolant leaving the microchannels for recycling

The overall length and width of the minichannel heat sink were 18 cm and 6.2 cm respectively, maintaining a scale up ratio of 10 over that of MCHS. The channel width, channel height and fin width for mCHS were fixed at 2 mm, 4 mm and 2 mm maintaining this ratio.

The arrangement of channels in configuration A and configuration C are similar. However, the designs of inlet and outlet plenums in configuration A and configuration C are different. Similarly, while the arrangement of microchannels in configuration B and configuration D are similar, the designs of inlet and outlet plenums of these two configurations (B & D) are different. The performance of each of the four designs has been evaluated numerically to compare the designs on the basis of substrate temperature gradient, total thermal resistance and pumping power.

### 3.3 MODEL ASSUMPTIONS & GOVERNING EQUATIONS

The following assumptions have been made

1. Steady, three dimensional, incompressible, single phase and laminar flow of coolant has been considered.
2. Steady and three-dimensional heat transfer has been considered.
3. The effect of temperature on properties of coolant and heat sink has been neglected.
4. A constant heat flux condition to simulate the heat generation has been imposed at the bottom. All other surfaces have been considered to be insulated.

Accordingly, the continuity, momentum and energy balance equations for the current problem can be written as follows [105]:

For fluid flow:

\[ \nabla \cdot \vec{V} = 0 \]  
\[ \rho \left( \nabla V \cdot \vec{V} \right) = -\nabla p + \mu \nabla^2 \vec{V} \]  
\[ \rho c_p \left( \nabla \cdot \vec{V} T \right) = k \nabla^2 T \]

The viscous dissipation term has been excluded in Eq. (3.3), as the maximum Brinkman number \[ Br = \mu \nu^2 / k \left( T_{wall,avg} - T_{f,avg} \right) \] was estimated to be 0.003, far lower than 1 for viscous dissipation term to be considered [106].

For energy in solid part of heat sink

\[ k_s \nabla^2 T_s = 0 \]

The boundary conditions are as follows:

At the inlet:

\[ \vec{V} = \vec{V}_{in} \]

At the outlet:
\[ p = p_{\text{out}}, \quad \frac{\partial T}{\partial n} = 0 \quad (3.6) \]

At the fluid-solid interface:

\[ \bar{V} = 0, \quad T = T_s, \quad -k_s \frac{\partial T_s}{\partial n} = -k \frac{\partial T}{\partial n} \quad (3.7) \]

At the bottom plate:

\[ q = -k_s \frac{\partial T_s}{\partial n} \quad (3.8) \]

The hydraulic diameter \((D_h)\) is related to height \((H_c)\) and width \((W_c)\) of the channels as follows:

\[ D_h = \frac{4 \times \text{cross-sectional area}}{\text{wetted perimeter}} = \frac{2H_cW_c}{H_c + W_c} \quad (3.9) \]

The length of thermally developing region for laminar flow in a channel is given by

\[ x = 0.05 Re.D_h.Pr \quad (3.10) \]

where

\[ Re = \frac{\rho V D_h}{\mu} \quad (3.11) \]

\[ Pr = \frac{c_p \mu}{k_f} \quad (3.12) \]

Silicon was chosen as the material for MCHS, while deionized water was used as the coolant. The inlet temperature of the water was set to 293 K. A constant heat flux of \(1 \times 10^6 \) W/m\(^2\) was applied at the bottom plate of the heat sink. The properties of fluid and solid used in the computation are \(\rho = 1000 \) kg/m\(^3\), \(c_p = 4182 \) J/kg K, \(\mu = 0.001 \) kg/m s, \(k_f = 0.6 \) W/m K and \(k_s = 148 \) W/m K.
Copper with thermal conductivity of 400 W/m K was chosen as the material for mCHS due to its wide use in minichannel heat sinks. Deionized water was used as the coolant.

The pumping power is related to the pressure drop ($\Delta p$) and the volumetric flow rate ($\dot{V}$) as

$$Pumping \ power = \Delta p \dot{V} \quad (3.13)$$

The average Nusselt number ($Nu_{avg}$) can be related to heat flux ($q$) and hydraulic diameter as

$$Nu_{avg} = \frac{qD_h}{(T_{wall,avg} - T_{f,avg})k_f} \quad (3.14)$$

The non-uniformity of temperature in the heat sink is given by

$$\theta = \frac{(T_{b,max} - T_{b,min})}{q} \quad (3.15)$$

The total thermal resistance ($R_{th}$) can be related to substrate and coolant temperature as follows:

$$R_{th} = \frac{T_{b,avg} - T_{f,in}}{q} \quad (3.16)$$

Knudsen number is related to mean free path ($\lambda$) and hydraulic diameter as

$$Kn = \frac{\lambda}{D_h} \quad (3.17)$$

### 3.4 NUMERICAL PARAMETERS AND PROCEDURES

The mean free path of water is 0.3 nm [107]. The Knudsen number in microchannels and minichannels were calculated to be $1.125 \times 10^{-6}$ and $1.125 \times 10^{-7}$ respectively. Since the
Knudsen number in microchannels and minichannels were much lower than 0.001, the continuum assumption is valid [108].

A commercial software ANSYS FLUENT (version 13.0) has been used for the numerical simulations. The governing equations were discretized using implicit method. The convective terms in the momentum and energy conservation equations were solved by the second order upwind scheme. The SIMPLE method was chosen for the coupling of pressure and velocities.

In the computational analysis, only one quadrant of the heat sink was investigated adopting symmetry conditions. The grid sensitivity was carried out for MCHS using three different grid sizes. The computational cells with coarse, intermediate and fine grids (934836, 1805440 and 4339755 elements) were tested for grid sensitivity at the inlet velocity of 1 m/s. With respect to the grid sensitivity study for simulations involving mCHS, three grids namely coarse, intermediate and fine (comprising 934836, 1805440 and 2734080 elements respectively) were tested at inlet velocity of 0.14 m/s. The computational domain for each of the four new MCHS and mCHS designs is also shown in Figure 3.1. The new MCHS and mCHS designs are very much different from conventional MCHS and mCHS reported in the literature. Hence the comparison of new MCHS and mCHS with others on the basis of same inlet velocity or pressure drop or mass flow rate may be misleading. Power consumption accounts for both the coolant flow rate and pressure drop. Hence to enable comparison of performance of different MCHS and mCHS designs, pumping power has been chosen as the independent variable.
3.5 RESULTS AND DISCUSSION

Figure 3.2 shows the temperature values at the centreline of the bottom plate of MCHS for the three grid sizes tested, at the coolant velocity of 1 m/s and heat flux of $1 \times 10^6$ W/m$^2$.

![Figure 3.2. Grid sensitivity study for MCHS: Centre line temperature of the bottom plate of configuration ‘A’ at x = 1.55 mm.](image)

Tables 3.1 and 3.2 revealed that the simulation results were identical for computations with intermediate and fine grid for both MCHS and mCHS, as the percentage deviation in the simulated values between intermediate and fine grids for both velocity and temperature were < 1 % [109]. Hence all the computations were carried out using intermediate grid to reduce the computational load. The percentage deviations were calculated using the formulae
\[ \text{Percentage deviation} = \left( \frac{X_{\text{lower}} - X_{\text{higher}}}{X_{\text{lower}}} \right) \times 100\% \quad (3.18) \]

\(X_{\text{lower}}\) – value of parameter calculated using the lesser number of mesh elements (Example: intermediate)

\(X_{\text{higher}}\) – value of parameter calculated using the higher number of mesh elements (Example: fine)

Table 3.1. Results of grid sensitivity analysis for MCHS \((q = 1 \times 10^6 \text{ W/m}^2\) and inlet velocity = 1 m/s).

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Number of mesh elements</th>
<th>% deviation in (T_{b,\text{max}})</th>
<th>% deviation in (V_{\text{avg}})</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>934836 (Coarse)</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>1805440 (Intermediate)</td>
<td>0.471</td>
<td>0.509</td>
</tr>
<tr>
<td></td>
<td>4339755 (Fine)</td>
<td>0.117</td>
<td>0.128</td>
</tr>
</tbody>
</table>

Table 3.2. Results of grid sensitivity analysis for mCHS \((q = 5\times10^4 \text{ W/m}^2\) and velocity inlet = 0.14 m/s).

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Number of mesh elements</th>
<th>% deviation in (T_{b,\text{max}})</th>
<th>% deviation in (V_{\text{avg}})</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>934836 (Coarse)</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>1805440 (Intermediate)</td>
<td>0.073</td>
<td>0.182</td>
</tr>
<tr>
<td></td>
<td>2734080 (Fine)</td>
<td>0.053</td>
<td>-0.27</td>
</tr>
</tbody>
</table>
Figure 3.3 shows the computational grids corresponding to intermediate number of mesh elements for the configuration ‘A’.

Figure 3.3. Computational grids employed for simulation of configuration ‘A’.

The comparison of numerical results with those of experimental data for microchannel heat sink design of Tuckerman and Pease [11] shown in Table 3.3 confirms the ability of numerical procedure to simulate experimental data. It is pertinent to note that experimental data on the performance of the four compartment microchannel heat sink is unavailable in literature and hence the numerical results are compared with the data of Tuckerman and Pease [11].
Table 3.3. Comparison of simulation result and experimental data of Tuckerman and Pease [11] for validation of numerical method.

<table>
<thead>
<tr>
<th>S. No.</th>
<th>$W_{fin}$ (µm)</th>
<th>$W_c$ (µm)</th>
<th>$H_c$ (µm)</th>
<th>$\Delta p$ (kPa)</th>
<th>$q$ (W/m²)</th>
<th>$R_{th}$ (K/W)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Experimental data [11]</td>
</tr>
<tr>
<td>1</td>
<td>55</td>
<td>45</td>
<td>287</td>
<td>117.21</td>
<td>2.77×10⁶</td>
<td>0.113</td>
</tr>
<tr>
<td>2</td>
<td>50</td>
<td>50</td>
<td>302</td>
<td>213.73</td>
<td>7.90×10⁶</td>
<td>0.090</td>
</tr>
</tbody>
</table>

Also, the computational method was validated by comparing the simulation results with those of Chein and Chen [104] for the same geometry and simulation conditions and are presented in Figure 3.4. The variation of average Nusselt number with pressure drop obtained as a result of present simulation work is in good agreement with those of Chein and Chen [104].

Figure 3.4. Variation of average Nusselt number with pressure drop.
3.5.1 Coolant velocity contours

The velocity contours of coolant in the computational domain of the four different MCHS and mCHS are shown in Figure 3.5a and 3.5b at mid plane of the channel in Z direction (at the depth of 0.2 mm for microchannel heat sink and at the depth of 2 mm for the minichannel heat sink).

The centrelines at which the velocity values were taken are shown in Figure 3.6. The centreline velocity profile in the third channel of each of the four minichannel and microchannel heat sinks is shown in Figures 3.7a and 3.7b. It may be observed from Figure 3.7a that the coolant velocity increased gradually in the axial direction for the straight microchannel regions (marked as Y’ in Figures 3.6 and 3.7) of configurations ‘A’ to ‘D’. The velocity profile for configuration ‘C’ was qualitatively similar to that of configuration ‘A’. However, the coolant velocity in configuration ‘C’ was lower than that of configuration ‘A’ due to energy losses in the expansion section of inlet plenum. The configurations ‘B’ and ‘D’ had similar velocity profiles qualitatively, with lower velocities in configuration ‘D’ attributed to energy loss in the expansion section of inlet plenum. The coolant velocity profiles in configurations ‘B’ and ‘D’ showed several regions of developing flow due to (i) short microchannels along ‘X’ direction near inlet plenum (shown as X’ in Figures 3.6 and 3.7a) (ii) long microchannels after ‘L’ section (miter bends) and (iii) short microchannels along ‘X’ direction near outlet plenum. Apart from these developing regions, coolant recirculation was observed near two ‘L’ sections (miter bends) connecting the long microchannels with short microchannels of configurations ‘B’ and ‘D’.
Figure 3.5. Velocity contour at the mid plane in depth of the channel for different configurations of (a) microchannel heat sinks for coolant inlet velocity of 2 m/s and heat flux of $1 \times 10^6$ W/m$^2$ and (b) minichannel heat sinks for coolant inlet velocity of 0.3 m/s and heat flux of $5 \times 10^4$ W/m$^2$. 
It is pertinent to recollect that larger disturbances in the flow, through change in flow direction and mixing contribute to enhanced heat transfer [90]. Also, coolant velocity in the axial direction plays an important role in enhancement of heat transfer [110]. Hence, the increasing coolant velocity in the flow direction for all the configurations A to D (as seen from Figure 3.7a) is expected to promote heat transfer. Figure 3.7b shows the centreline velocity profile of the third channel of the four minichannel heat sinks. The trends in Figure 3.7b are qualitatively similar to those in Figure 3.7a indicating the presence of developing regions in the four minichannel heat sinks also.
Figure 3.7. Coolant velocity profile in the third channel of different configurations of (a) microchannel heat sinks for coolant inlet velocity of 2 m/s and heat flux of $1 \times 10^6$ W/m$^2$ and (b) minichannel heat sinks for coolant inlet velocity of 0.3 m/s and heat flux of $5 \times 10^4$ W/m$^2$.

### 3.5.2 Substrate temperature profiles

The temperature profile of bottom plate of four different microchannel and minichannel heat sink configurations at comparable conditions is shown in Figures 3.8a and 3.8b. It may be observed from Figures 3.8a and 3.8b that substrate temperature was the lowest for configuration ‘B’ among the four configurations simulated. Hot spots were observed at the edges of the heat sink due to the absence of the channels. Those regions were indirectly cooled by conduction across the substrate. The substrate temperature gradients were relatively small for all the four configurations, except near the regions of inlet and
outlet plenum. Haller et al. [111] have reported increased heat transfer rates due to the presence of bends and T-junctions. The presence of bends caused the formation of coolant vortices and was responsible for change in the thermal gradient near the bends [111].

In line with the observations of Haller et al. [111] lower substrate temperature was observed near the two L-sections (miter bends), as shown in Figures 3.8a and 3.8b for the configurations ‘B’ and ‘D’, in which the fluid mixing near the bends reduced coolant temperature gradient, but increased the driving force for heat transfer, leading to better cooling of substrate.

The temperature contours shown in Figure 3.8a pertain to the coolant inlet velocity of 2 m/s. As the pressure drops for coolant flow in these configurations were different, the temperature contours in the four new configurations and in the conventional heat sink were studied at the same pumping power. Pumping power was calculated as the product of total volumetric flow rate of coolant and total pressure drop for the entire region of heat sink as shown in Eq. (3.13). Figure 3.9a shows the substrate temperature contours for the four new microchannel heat sinks (entire sink) and the conventional heat sink for the pumping power of 0.15 W. One may observe from Figure 3.9a that the maximum substrate temperature was the lowest for configuration ‘B’.
Figure 3.8. Temperature contour of the bottom plate of different new (a) microchannel heat sink at the inlet velocity of 2 m/s and $q = 1 \times 10^6$ W/m$^2$ and (b) minichannel heat sink at the velocity of 0.3 m/s and $q = 5 \times 10^4$ W/m$^2$. 
Figure 3.9. Temperature contour of the bottom plate of conventional and different new (a) MCHS at pumping power of 0.15 W and (b) mCHS at pumping power of 0.0486 W.

Among the four new microchannel heat sinks, maximum substrate temperature was highest in configuration ‘C’. However, the maximum substrate temperature in the
conventional microchannel heat sink was 25 K greater than that in the configuration ‘C’ of new microchannel heat sink. Similarly, the maximum substrate temperature in the conventional minichannel heat sink was 8 K greater than that in the configuration ‘C’ of the new minichannel heat sink.

Also, it is pertinent to note that the lowest substrate temperature in the conventional heat sink was closer to the maximum substrate temperature in configuration ‘B’ for both the microchannel and minichannel heat sinks. All these indicate that at the same pumping power, the four new configurations of microchannel and minichannel heat sinks were more effective in maintaining a lower substrate temperature, compared to that of conventional heat sink. While the dedicated coolant distribution to each quadrant of the new microchannel heat sinks have resulted in reduction of minimum substrate temperature, the coolant distribution through multiple jets simulated by Lelea [110], have shown higher minimum substrate temperature, in comparison with the sink comprising single coolant inlet. The observations of Lelea [110] was predominantly attributed to lower axial velocities in certain regions with the use of multiple jets, in comparison with that of single coolant inlet.

3.5.3 Average Nusselt number

The average Nusselt number in heat sink is an indicator of heat transfer rates achievable. It is widely known that the local heat transfer coefficients and the local Nusselt numbers are high in the regions of developing flow. In heat sinks with several regions of developing flow and coolant recirculation, higher local Nusselt numbers and hence higher average Nusselt number can be expected. Hence, while comparing heat sinks of different designs under similar heat flux and pumping power, heat sinks with higher average
Nusselt number provide higher rates of heat removal. In laminar flow regime, the length of developing flow is a function of fluid velocity [105]. Hence simulation experiments were performed to determine average Nusselt numbers for different coolant velocities leading to different pumping power. Such an exercise permits evaluation of different heat sink designs through comparison of average Nusselt number-pumping power data as shown in Figures 3.10a and 3.10b. The simulation results corresponding to the conventional microchannel and minichannel heat sink have also been included in Figures 3.10a and 3.10b.

It may be observed from Figures 3.10a and 3.10b that higher average Nusselt numbers were observed for configurations ‘B’ and ‘D’, when compared to that of configurations ‘A’ and ‘C’. The higher average Nusselt numbers for configurations ‘B’ and ‘D’ may be attributed to the presence of larger number of developing regions in configurations ‘B’ and ‘D’.

While comparing the configurations ‘B’ and ‘D’ on the basis of average Nusselt number, there is very little difference in configuration ‘B’ and ‘D’ at lower pumping power for MCHS and mCHS. However, for MCHS, at pumping powers greater than 0.1 W, higher average Nusselt numbers were observed for configuration ‘D’. The average Nusselt numbers in configurations ‘A’ and ‘C’ were similar over the range of pumping powers simulated. It is also pertinent to note that the average Nusselt numbers for all the four new configurations of MCHS and mCHS (A–D) were greater than that of the conventional heat sinks, over the entire range of pumping power simulated. Hence, it is clear that higher heat transfer rates can be achieved in all the four new MCHS and mCHS
configurations, when compared to heat transfer rates achievable in conventional heat sinks.

It is reasonably well-established that higher Nusselt numbers are achieved in the entrance/acceleration regions in comparison with that in fully-developed regions [112]. Hence, entrance regions in microchannels contribute significantly towards higher Nusselt numbers [112]. It is evident from the velocity profiles in Figures 3.7a and 3.7b, that the lengths of thermally developing regions were longer due to higher velocity and Reynolds number in accordance with Eq. (3.10). Hence higher average Nusselt numbers of the new configurations ‘A’ to ‘D’ may be attributed to longer regions of thermally developing flow. Among the new configurations, the average Nusselt number in configuration ‘D’ was the highest at higher pumping powers due to presence of larger number and hence longer regions of developing flow.
Figure 3.10. Variation of average Nusselt number with pumping power for (a) microchannel heat sink \( (q = 1 \times 10^6 \text{ W/m}^2) \) and (b) minichannel heat sink \( (q = 5 \times 10^4 \text{ W/m}^2) \).
3.5.4 Substrate temperature gradient ($\theta$)

One of the main objectives of devising new configurations for microchannel heat sinks is to reduce the temperature gradient of the substrate. In conventional microchannel heat sinks and minichannel heat sinks of parallel microchannels and minichannels with inlet plenum at one end and the outlet at the other, the substrate temperature gradient is high. A substrate with uniform cooling will have smaller difference between the maximum and minimum temperature. The ratio of difference between maximum and minimum substrate temperatures ($T_{b,\text{max}} - T_{b,\text{min}}$) to heat flux ($q$) is ‘$\theta$’, which is used as measure of non-uniformity of substrate temperature. Lower values of ‘$\theta$’ indicate better uniformity of substrate temperature. The variation of ‘$\theta$’ with pumping power for the four new MCHS and mCHS configurations and the conventional MCHS and mCHS are shown in Figures 3.11a and 3.11b.

It is evident from Figures 3.11a and 3.11b that ‘$\theta$’ decreased with pumping power, with higher rate of decrease at lower pumping powers. For a fixed heat sink geometry, pumping power increases with coolant flow rate. With increase in coolant flow rate, coolant velocity increases, which in turn increases heat transfer rate though increase in heat transfer coefficient.
Figure 3.11. Variation of ‘θ’ with pumping power for (a) microchannel heat sink ($q = 1 \times 10^6 \text{ W/m}^2$) and (b) minichannel heat sink ($q = 5 \times 10^4 \text{ W/m}^2$).
At higher pumping powers, the lower values of maximum substrate temperatures and hence lower ‘\( \theta \)’, as per Eq. (3.15), can be achieved, when compared to that at lower pumping powers. This explains the pumping power dependence of ‘\( \theta \)’ for the heat sinks. Comparing ‘\( \theta \)’ value of different configurations at lower pumping powers (<0.1 W), it decreased in the following order: \( \theta_{\text{Conventional}} > \theta_C > \theta_A > \theta_B \). It seems that several regions of undercooling in conventional microchannel heat sink existed at lower pumping power, compared to those in new configurations (A-D).

The MCHS and mCHS of configuration ‘B’ possessed the lowest ‘\( \theta \)’. The ‘\( \theta \)’ vs pumping power relationship for configuration ‘B’ seemed to saturate at lower pumping power for MCHS, when compared to other three new MCHS configurations. This probably indicates that configuration ‘B’ has been optimized to minimize substrate temperature gradient even at lower pumping powers. The saturation ‘\( \theta \)’ (minimum value of ‘\( \theta \)’ in the pumping power range investigated) of new MCHS configurations was less than 50% of ‘\( \theta \)’ of conventional MCHS. Similarly, the saturation ‘\( \theta \)’ (minimum value of ‘\( \theta \)’ in the pumping power range investigated) of new mCHS configurations was about 30% of ‘\( \theta \)’ of conventional mCHS. One of the methods to reduce substrate temperature gradient is to ensure mixing of hotter coolant with the relatively lower temperature coolant, leading to averaging their temperature and help level the substrate temperature [91]. In other words, coolant mixing is desirable to ensure uniformity of substrate temperature. A combination of better distribution of coolant, presence of regions of developing flow and coolant recirculation in the four new MCHS and mCHS have contributed to reducing the substrate temperature gradient. The ‘\( \theta \)’ values for the four new MCHS configurations were lower than those reported by Ramos Alvarado et al.
The present results are in qualitative agreement with those of Lelea [110], who observed lower substrate temperature gradient in the microchannel heat sink with multiple inlets attached tangentially to the microtube, in comparison with the sink comprising single coolant inlet.

3.5.5 Total thermal resistance

Total thermal resistance, the ratio of maximum driving force to the heat flux is a measure of total resistance for heat transfer. Hence a lower value of total thermal resistance is preferred, a criteria normally used to compare MCHS configurations [13]. The total thermal resistances for the four new MCHS and mCHS configurations and that of conventional MCHS and mCHS, as a function of pumping power are shown in Figures 3.12a and 3.12b. The simulation results for the conventional MCHS and mCHS indicate higher total thermal resistance compared with that of new configurations, over the entire range of pumping power simulated. The total thermal resistance of microchannel heat sink is comprised of conductive, convective and capacitive resistances [51]. The conductive resistance is independent of pumping power, while the convective resistance decreases with increasing pumping power due to higher heat transfer coefficient achieved at higher pumping power [51].
Figure 3.12. Variation of total thermal resistance with pumping power for (a) microchannel heat sink ($q = 1 \times 10^6$ W/m$^2$) and (b) minichannel heat sink ($q = 5 \times 10^4$ W/m$^2$).
Increased velocity or coolant flow or coolant mixing contribute to higher pumping power. The capacitive resistance, which is a measure of temperature difference between average coolant temperature and the coolant inlet temperature [51], decreases with increase in pumping power, either due to increased fluid mixing or due to higher heat capacity at higher coolant flow rates achieved at higher pumping powers. The total thermal resistances of the four new MCHS and mCHS configurations ‘A’ to ‘D’ were lower than that of conventional MCHS and mCHS at the same pumping power due to reduction in convective and capacitive resistances achieved by efficient coolant distribution and the presence of acceleration regions.

Among the new sinks, the lowest total thermal resistances were achieved with configuration ‘B’ over the range of pumping powers simulated. The thermal resistance values of the new MCHS were compared with the best configuration evaluated by Chai et al. [51] (Figure 3.12a) and it is evident that the total thermal resistances of the proposed MCHS configurations were lower at least by 50 % in comparison to design of Chai et al. [51]. Taking both ‘θ’ and the thermal resistance into account, the configuration ‘B’ seems to be the most suitable of the four new MCHS and mCHS configurations simulated.

In this chapter, a constant aspect ratio of 0.5 has been used for all the heat sink designs simulated. It may be possible to reduce the total thermal resistance and '0' further through the choice of higher aspect ratio. Hence the performance of MCHS and mCHS of configuration 'B' may be improved further by increasing the aspect ratio of the channels.

3.6 CONCLUSIONS

Numerical analyses of flow and heat transfer in four new MCHS and mCHS configurations were performed and compared with conventional MCHS and mCHS. At
the same pumping power and heat flux, the new MCHS and mCHS designs are superior to conventional MCHS in the following aspects:

1. lower maximum and minimum substrate temperature
2. lower temperature gradient of the substrate
3. lower total thermal resistance
4. higher Nusselt number and heat transfer rate

Among the new MCHS and mCHS designs, the configuration ‘B’ provided the lowest total thermal resistance and ‘0’. The total thermal resistance and substrate temperature gradient in configuration ‘B’ of MCHS were respectively at least 50% and 30% lower than that in the conventional MCHS.

Similarly, the total thermal resistance and substrate temperature gradient in configuration ‘B’ of mCHS were respectively at least 30% and 68% lower than that in the conventional mCHS.