CHAPTER 5: FINE-TUNING CHANNEL WIDTH, ASPECT RATIO AND FIN WIDTH OF THE NEW MICROCHANNEL HEAT SINK

5.1 INTRODUCTION

In chapter 3, design of a new microchannel heat sink (MCHS) containing four compartments with individual coolant inlet and outlet for each and microchannels with two ‘L’ sections (miter bends) in each channel was arrived at. However, the effect of microchannel width, aspect ratio and fin width on the performance of such a heat sink was not simulated. Hence, an attempt has been made to study the influence of microchannel width, aspect ratio and fin width on performance of such a new MCHS (Figure 5.1). The study assumes importance due to the fact that most of the reported works on influence of aspect ratio were performed on channels without such ‘L’ sections (miter bends) where intense fluid mixing occurs. With the understanding of the influence of width of the channel ($W_c$), aspect ratio and fin width on heat transfer performance of the new microchannel heat sink, they can be fine-tuned to achieve the best possible heat transfer performance in terms of total thermal resistance and substrate temperature gradient.

Three-dimensional numerical optimization, widely adopted [46,53,113] to identify the appropriate microchannel aspect ratio and width that lead to minimum total thermal resistance and substrate temperature gradient has been used here. Microchannel width (100-200 µm), aspect ratio (2 to 10) and coolant velocity (0.25 to 2 m/s) have been used as independent variables. In general, lower total thermal resistance is achieved with microchannels of lower width (<50 µm) and higher aspect ratio (>10), when the power consumption is greater than 0.01 W [114]. A reduction in channel width often leads to reduction in convection resistance and increase in bulk fluid resistance [115]. While
higher aspect ratio microchannels are considered to be generally advantageous [116], considerable difficulty exists in their fabrication and heat transfer at the fin tip is poor [13].

Figure 5.1. Schematic representation of the MCHS and the computational domain studied for identification of appropriate channel width, aspect ratio and fin width.
Microchannels of higher width containing ‘L’ sections (miter bends) can be fabricated with relative ease without exceeding the maximum permissible hydraulic diameter for microchannels [13], justifying the choice of range of microchannel width. Hence the smallest microchannel width has been fixed at 100 µm. A constant substrate thickness of 100 µm has been utilized to ensure that the changes in aspect ratio and channel width influence convective resistance while the resistance due to substrate thickness remains unchanged. Depending upon the microchannel width, the maximum values for aspect ratio and coolant velocity were fixed in such a way that the channel height did not exceed 1000-1200 µm and Reynolds number was within the limit for laminar flow regime.

A new parameter to characterize the substrate temperature gradient has been introduced that reflects both the design goals - lower maximum substrate temperature and closeness of average & minimum substrate temperatures. This would facilitate lower temperature window for fuel cell operation. The total thermal resistance based on maximum substrate temperature, instead of average substrate temperature has been used in this chapter for identification of appropriate channel width, aspect ratio and fin width enabling finetunement of design.

### 5.2 DATA REDUCTION

Hydraulic diameter ($D_h$), in terms of aspect ratio is given by

$$D_h = \frac{2W_e}{(1+\frac{1}{\alpha})}$$

(5.1)
In Eq. (5.1), the aspect ratio ($\alpha$) is the ratio of channel height ($H_c$) to the channel width ($W_c$).

Heat transfer coefficient ($h$) is related to heat flux ($q$), heat transfer area ($A_{wall}$), area of base plate ($A_b$) and temperatures using the following formula:

$$h = \frac{q \cdot A_b}{(T_{wall,avg} - T_{f,avg}) A_{wall}}$$  \hspace{1cm} (5.2)

Reynolds number is related to aspect ratio as follows:

$$Re = \frac{2 \cdot m \cdot W_c}{\mu \cdot N \cdot \alpha \cdot W_c^2 (1 + \frac{1}{\alpha})}$$  \hspace{1cm} (5.3)

The dimensionless pumping power ($P_{Dl}$) is related to pumping power, hydraulic diameter and properties of water as [79]:

$$P_{Dl} = \frac{P \cdot \rho^2 \cdot D_h}{\mu^3}$$  \hspace{1cm} (5.4)

The total thermal resistance ($R_{th,max}$) is related to heat flux ($q$), maximum temperature of bottom plate ($T_{b,max}$) and coolant inlet temperature ($T_{f,in}$) as follows:

$$R_{th,max} = \left( \frac{T_{b,max} - T_{f,in}}{q} \right)$$  \hspace{1cm} (5.5)

The dimensionless thermal resistance ($R_{th,Dl}$) is given by [117]

$$R_{th,Dl} = \frac{W_c \cdot k_f \cdot L_{hs} \cdot R_{th,max}}{H_c}$$  \hspace{1cm} (5.6)
The numerical simulations have been carried out using ANSYS FLUENT 13.0. The second order upwind method has been adopted to solve momentum and energy balance equations. The pressure and velocity terms were coupled using the SIMPLE algorithm. The computation domain was restricted to single quadrant of the domain as shown in Figure 5.1 to save the computational time. The grid sensitivity analysis was performed with 4 different grid numbers within the range of 1651680 to 7008572 mesh elements for the inlet velocity of 1 m/s. The sum of the scaled absolute residual for convergence of the momentum and continuity equations was $10^{-6}$. With respect to convergence criterion for energy equation, the sum of the scaled absolute residual was $10^{-8}$. Table 5.1 shows the details of microchannel dimensions (channel height and channel width) investigated to identify optimum microchannel dimensions.

Table 5.1. Details of microchannel dimensions investigated.

<table>
<thead>
<tr>
<th>$H_c$ (µm)</th>
<th>$W_c$ (µm)</th>
<th>$N$</th>
<th>$a$</th>
</tr>
</thead>
<tbody>
<tr>
<td>400</td>
<td>200</td>
<td>5</td>
<td>2</td>
</tr>
<tr>
<td>600</td>
<td>200</td>
<td>5</td>
<td>3</td>
</tr>
<tr>
<td>800</td>
<td>200</td>
<td>5</td>
<td>4</td>
</tr>
<tr>
<td>1000</td>
<td>200</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>1200</td>
<td>200</td>
<td>5</td>
<td>6</td>
</tr>
<tr>
<td>400</td>
<td>150</td>
<td>6</td>
<td>2.667</td>
</tr>
<tr>
<td>600</td>
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<td>6</td>
<td>4</td>
</tr>
<tr>
<td>800</td>
<td>150</td>
<td>6</td>
<td>5.334</td>
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<tr>
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<td>6</td>
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<tr>
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<td>100</td>
<td>9</td>
<td>8</td>
</tr>
<tr>
<td>1000</td>
<td>100</td>
<td>9</td>
<td>10</td>
</tr>
</tbody>
</table>
5.3 RESULTS AND DISCUSSION

5.3.1 Grid sensitivity

Table 5.2 shows the results of the grid sensitivity test. It may be observed from Table 5.2 that the temperature and coolant velocities predicted using intermediate grid varied from those predicted using fine grids by less than 1%. Hence the intermediate grid number was chosen for all the computations to reduce the computational time [53].

Table 5.2. Results of grid sensitivity tests carried out at simulation conditions of 1 m/s as coolant inlet velocity and heat flux of $1 \times 10^6$ W/m$^2$.

<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>Very coarse</td>
<td>1651680</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Coarse</td>
<td>2439276</td>
<td>0.094</td>
<td>-1.255</td>
</tr>
<tr>
<td>Intermediate</td>
<td>3208500</td>
<td>0.264</td>
<td>2.312</td>
</tr>
<tr>
<td>Fine</td>
<td>7008572</td>
<td>-0.044</td>
<td>-0.256</td>
</tr>
</tbody>
</table>

5.3.2 Pumping power

Figure 5.2 shows the effect of mass flow rate and aspect ratio on pumping power for 200 µm wide microchannels. The pumping power decreased with increase in aspect ratio at the same mass flow rate for 200 µm, 150 µm and 100 µm wide microchannels (Figure 5.2). This could be attributed to increase in cross-sectional area ($H_cW_c$) at higher aspect ratios that led to lower coolant velocity and hence lower pumping power. For the range of
coolant mass flow rates simulated in the present work, the lowest pumping power has been observed for 200 µm microchannels of aspect ratio 6.

The pumping power ($P$) is influenced by microchannel parameters such as hydraulic diameter ($D_h$), aspect ratio ($\alpha$), average channel length ($\bar{L}$) and the properties of fluid such as viscosity ($\mu$) and density ($\rho$). Accordingly, from the ‘pumping power – channel width – aspect ratio’ computational data, Eq. (5.7) has been developed, which encompasses Reynolds number range of 75 to 1160, hydraulic diameter range of 160 to 343 µm, aspect ratio range of 2 to 8 and the dimensionless pumping power ($P_{Dl}$) range of $4.06\times10^8$ to $3.00\times10^{11}$. The dimensionless pumping power ($P_{Dl} = P\rho^2D_h/\mu^3$) is related to Reynolds number ($Re$), aspect ratio ($\alpha$) and geometry ratio ($D_h/\bar{L}$) as

$$P_{Dl} = \frac{P\rho^2D_h}{\mu^3} = 23.241\text{Re}^{2.422}\alpha^{0.7695}\left(\frac{D_h}{\bar{L}}\right)^{-1.329}$$

(5.7)

Equation 5.7 predicts the dimensionless pumping power data with a maximum error of 14%, root mean square error of $4.14\times10^9$ and index of correlation of 0.9982.
Figure 5.2. Influence of mass flow rate and aspect ratio on pumping power for channels of different widths.
The parity plot for the dimensionless pumping power is shown in Figure 5.3. It may be observed from the parity plot that the maximum absolute percentage error was 14 %.

![Parity plot for dimensionless pumping power](image)

**Figure 5.3.** Parity plot for dimensionless pumping power.

### 5.3.3 Source and coolant temperature profile

Figure 5.4 shows the temperature profile of solid and liquid regions for different microchannel widths and aspect ratio at the pumping power of 0.2 W. It may be observed from Figure 5.4 that the lowest maximum solid temperature was obtained for 200 µm wide channels of aspect ratio 6. Figure 5.5 shows the temperature distribution in the bottom plate of the different microchannel widths and aspect ratio at the pumping power of 0.2 W. It may be observed from Figure 5.5 that the lowest substrate temperatures were recorded near the vicinity of ‘L’ section (miter bends) near the coolant inlet for all
microchannel widths and aspect ratios studied. Both maximum and minimum substrate temperatures were lowered with the use of high aspect ratio and wider microchannels. In all the cases, highest substrate temperatures were observed in uncooled regions of substrate between coolant inlet and outlet. Among the six channel dimensions compared in Figure 5.5 at the pumping power of 0.2 W, the lowest maximum substrate temperature (313 K) was observed for 200 µm wide channels of aspect ratio 6. The difference between maximum and minimum substrate temperatures was lowest for channel of these dimensions.
Figure 5.4. Effect of microchannel width and aspect ratio on temperature of solid and liquid regions at the pumping power of 0.2 W.
Figure 5.5. Effect of microchannel width and aspect ratio on substrate temperature contours.
5.3.4 Heat transfer coefficient

The influence of aspect ratio on heat transfer coefficient \( (h) \) – mass flow rate \( (m) \) relationship is shown in Figure 5.6. It is evident from Figure 5.6 that the heat transfer coefficient increased with increase in mass flow rate due to increase in coolant velocity. The heat transfer coefficient was also found to decrease with increase in aspect ratio. It may be understood from Eq. (5.2), that the heat transfer coefficient is inversely proportional to wall area. The wall area is related to aspect ratio as

\[
A_{wall} = NL_cW_c(1 + 2\alpha)
\]

(5.8)

where ‘\( N \)’ is number of channels, ‘\( L_c \)’ is length of the channel and ‘\( W_c \)’ is channel width.

The increase in wall area with increase in channel aspect ratio is appreciable. For instance, the wall area for aspect ratio of 6 is 2.5 times the wall area at the aspect ratio of 2 for 200 µm wide channels. Similarly, the wall area for aspect ratio of 10 in 100 µm wide channel is 2.27 times the wall area at aspect ratio of 4. The difference in temperature between the solid surface and the coolant too influences heat transfer coefficient (Eq. (5.2)). The presence of ‘L’ sections (miter bends) contributes to enhanced mixing, heat transfer, pressure drop and pumping power [111]. The difference in pumping power between channels of similar width and mass flow rate, but different aspect ratios may also affect heat transfer coefficient through mixing and influencing the temperature difference \( (T_{wall} - T_f) \).
Figure 5.6. Heat transfer coefficient vs mass flow rate data in microchannels of different widths and aspect ratio.
It may be recollected from the discussions in section 5.3.2 that the pumping power decreased with increase in aspect ratio. Hence to simultaneously take into account of influence of mass flow rate and pumping power on heat transfer coefficient, a plot of heat transfer coefficient per unit pumping power ($h/P$) as a function of mass flow rate and aspect ratio, as shown in Figure 5.7 can be made.

It may be observed from Figure 5.7 that the heat transfer coefficient per unit pumping power ($h/P$) increased with increase in aspect ratio for 200 µm wide microchannels. Similar trend was observed for the 150 µm and 100 µm wide microchannels also. This indicates that at the same pumping power and mass flow rate of coolant, use of high aspect ratio channels resulted in reduced wall temperatures. The volume of coolant available at any instant in high aspect ratio channels is higher, when compared to that in low aspect ratio channels of same widths. In addition, the volume of coolant in wider microchannels is higher than that in narrow microchannels of same aspect ratio. Hence, the higher coolant volume in wider, higher aspect ratio channels led to better substrate cooling resulting in lower wall temperature. This is reflected in trends observed in Figure 5.7.

Another factor that may have contributed to lower wall temperatures at higher aspect ratios (at a constant channel width) and higher width (at a constant height) is the higher area for heat transfer (wall area). At a fixed mass flow rate and channel width, coolant velocity decreases with increase in aspect ratio. Lower coolant velocity is expected to result in reduction in both heat transfer coefficient and pumping power [6], albeit to different extents.
Figure 5.7. Heat transfer coefficient per unit pumping power vs mass flow rate data in microchannels of different widths and aspect ratios.
At constant mass flow rate, pumping power is strongly dependent on coolant velocity when compared to heat transfer coefficient. Hence at constant mass flow rate, reduction in heat transfer coefficient is overcome by reduction in pumping power contributing to higher heat transfer coefficient per unit pumping power at higher aspect ratios. At constant coolant mass flow rate and fixed aspect ratio, increase in width too results in increase of cross-sectional area and reduction in velocity. Therefore, in line with the above discussions, heat transfer coefficient per unit pumping power decreased with increase in width at constant coolant mass flow rate and a fixed aspect ratio.

Though heat transfer coefficient per unit pumping power increased with both aspect ratio and channel width, the influence of aspect ratio was slightly more dominant than that of channel width, with exponents in \((h/P) = f(\alpha, W_c)\) being 1.6269 and 1.2724 respectively for ‘\(\alpha\)’ and ‘\(W_c\)’. While there are considerable difficulties in fabrication of high aspect ratio channels, channels of higher widths can be fabricated easily. Therefore, heat transfer coefficient per unit pumping power obtained using a higher aspect ratio channel of smaller width can be achieved or exceeded using a channel of lower aspect ratio and higher width also. For instance, the use of 200 μm wide channel of aspect ratio 6 resulted in 5 % increase in heat transfer coefficient per unit pumping power over that achieved using 100 μm wide channel of aspect ratio 10. Among the different aspect ratios and channel widths investigated in the present work, the highest heat transfer coefficient per unit pumping power was obtained for 200 μm channels of aspect ratio 6.

5.3.5 Nusselt number

Figure 5.8 shows the influence of Reynolds number on Nusselt number (on natural logarithmic scale) at different aspect ratios for microchannel widths of 200 μm, 150 μm
and 100 µm. It is evident from Figure 5.8 that the Nusselt number increased with Reynolds number inline with the several reported results [6,13,30,118].

For a channel with fixed width and aspect ratio, higher Reynolds number indicates higher coolant velocity. Heat capacity of coolant, which is the product of its specific heat and mass flow rate, is higher at higher Reynolds number leading to increased substrate cooling. It is also evident from Figure 5.8 that the Nusselt number – Reynolds number relationship is independent of aspect ratio, but dependent on channel width. At similar
Reynolds numbers, higher Nusselt numbers were obtained with microchannels of higher width. The dependence of Nusselt number on Reynolds number was higher in 200 µm wide microchannels, when compared to that in 150 µm and 100 µm wide microchannels.

Nusselt number is influenced by microchannel parameters such as hydraulic diameter \((D_h)\), aspect ratio \((\alpha)\), average channel length \((\bar{L})\) and the fluid properties of coolant such as viscosity \((\mu)\) and density \((\rho)\). Accordingly, from the Nusselt number – Reynolds number – channel width computational data, Nusselt number is related to Reynolds number \((Re)\), aspect ratio \((\alpha)\) and geometry ratio \((D_h/\bar{L})\) as

\[
Nu = 9.121 \, Re^{0.349} \alpha^{-0.065} \left(\frac{D_h}{\bar{L}}\right)^{0.461}
\]  

Equation 5.9 encompasses Reynolds number range of 75 to 1160, hydraulic diameter range of 160 to 343 µm, aspect ratio range of 2 to 8 and Nusselt number range of 7.2 to 22.3. Equation 5.9 predicts the Nusselt number data with a maximum error of 16 %, root mean square error of 0.4137 and index of correlation of 0.9931. The parity plot for the Nusselt number is shown in Figure 5.9. It may be observed from the parity plot that the maximum absolute percentage error was 16 %.
Total thermal resistance (based on maximum substrate temperature) is one of the important attributes of a microchannel heat sink. Several design modifications have been reported in the literature with an aim to reduce the total thermal resistance \cite{77,90,91,111,116} at a fixed pumping power. Microchannel heat sinks with lower total thermal resistance provide better substrate cooling and hence are preferred for thermal management. It is clear from discussions in section 5.3.4, that the aspect ratio influenced heat transfer coefficient, which in turn influenced the convective thermal resistance component of the total thermal resistance. For a microchannel of fixed dimension, pumping power consumption increases if pressure drop or coolant mass flow rates or
both are increased. Under these conditions, heat transfer coefficient too increases, which contributes to reduction in total thermal resistance. Hence total thermal resistance decreased with increase in pumping power for 100 µm, 150 µm and 200 µm microchannels (Figure 5.10), inline with several other reported data [60,115,119]. It may also be observed from Figure 5.10 that the total thermal resistance decreased with increase in aspect ratio at the same pumping power. At all aspect ratios, the decrease in thermal resistance at lower pumping powers was rapid. At higher pumping powers, a near saturation in total thermal resistance was observed for all aspect ratios at microchannel widths of 200 µm and 150 µm (Figure 5.10), with the minimum total thermal resistance decreasing with increase in aspect ratio. In the case of 200 µm wide microchannels, increase in aspect ratio from 2 to 6 influenced total thermal resistance – pumping power relationship ($R_{th} = xP^y$) through reduction in coefficient ($x$) and exponent ($y$). However, the reduction in coefficient and exponent were more pronounced when aspect ratio was increased from 2 to 4.

At a fixed pumping power, with increase in aspect ratio, coolant can be supplied at a higher mass flow rate as evident from discussions in section 5.3.2. In the case of 200 µm wide microchannel, the ratio of coolant mass flow rate in channel of aspect ratio 6 to coolant mass flow rate in channel of aspect ratio 2 was $\approx 7.2$. The cross-sectional area is increased by 3 times when aspect ratio is increased from 2 to 6. With increase in mass flow rate higher than the increase in cross-sectional area due to aspect ratio increase, the coolant velocity in channel of aspect ratio 6 was about 2.4 times higher than that in channel of aspect ratio 2 at the same pumping power. Therefore, the heat transfer coefficient and heat transfer rate were higher in channel of aspect ratio 6, when compared
to that in channel of aspect ratio 2, leading to better heat removal and lower substrate temperature in the former. Hence at the same pumping power, total thermal resistance decreased with increase in aspect ratio.

The effect of aspect ratio on total thermal resistance seems to depend on microchannel width also, as evident from difference in trends observed for 150 µm and 100 µm wide microchannels. For 100 µm wide microchannels, the variation of thermal resistance with aspect ratio is biphasic, with reduction in thermal resistance upon increasing aspect ratio from 2 to 8 followed by increase in thermal resistance upon increasing the aspect ratio to 10. This could be attributed to the fact that at very high aspect ratios & lower microchannel width, the coolant flowing near the tip of fin is least effective in removing heat from the substrate [13]. The increase in coolant velocity due to increase in aspect ratio from 8 to 10 in 100 µm wide microchannel was too low (~6.4% increase) to offset inefficient cooling at tip of the fin. Comparing the total thermal resistance - pumping power relationship for all channel widths and aspect ratios simulated in the present work, the lowest thermal resistance for a fixed pumping power was obtained for 200 µm wide microchannels with the aspect ratio of 6.
Figure 5.10. Influence of pumping power on total thermal resistance for microchannels of different widths and aspect ratios.
The dimensionless thermal resistance \( R_{th,Di} \) is influenced by microchannel parameters such as hydraulic diameter \( (D_h) \), aspect ratio \( (\alpha) \), average channel length \( (\bar{L}) \), properties of fluid such as viscosity \( (\mu) \) and density \( (\rho) \). Accordingly, from the total thermal resistance – pumping power – aspect ratio computational data, the dimensionless thermal resistance
\[
R_{th,Di} = \frac{W_{th} k_f L_h R_{th}}{H_c} \left( \frac{D_h}{L} \right)_{Dl} \tag{5.10}
\]
is related to Reynolds number \( (Re) \), aspect ratio \( (\alpha) \) and geometry ratio \( (D_h/L) \) as
\[
R_{th,Di} = 0.0658\ Re^{0.366} \alpha^{-1.28471} \left( \frac{D_h}{L} \right)^{-0.894}
\]
Equation 5.10 encompasses Reynolds number range of 75 to 1160, hydraulic diameter range of 160 to 343 µm, aspect ratio range of 2 to 8 and the dimensionless total thermal resistance range of 0.0092 to 0.1213. Equation 5.10 predicts the dimensionless total thermal resistance data with a maximum error of 16 %; root mean square error of 0.0040 and index of correlation of 0.9802. The parity plot for dimensionless total thermal resistance is shown in Figure 5.11. It may be observed from the parity plot that the maximum absolute percentage error was 16 %.
5.3.7 Substrate temperature gradient (θ)

The non-uniformity of substrate temperature is normally denoted by ‘θ’ and calculated using Eq. (3.15) given in chapter 3. However, there is an inherent drawback in utilization of ‘θ’ for comparison of different heat sink designs or operating conditions. For instance, the value of ‘θ’ will be low even for heat sinks with higher values of maximum and minimum substrate temperatures (high $T_{b,\text{max}}$ and high $T_{b,\text{min}}$ but still low $T_{b,\text{max}} - T_{b,\text{min}}$), despite the fact that higher maximum substrate temperature is detrimental and the heat sink is ineffective. Considering the fact that lower maximum substrate temperature and minimum substrate temperature close to average substrate temperature are attributes of a
near-ideal heat sink, the following parameter \( \omega \) may be used to denote non-uniformity in substrate temperature.

\[
\omega = \frac{\sqrt{(T_{b,\text{max}} - T_{b,\text{avg}})(T_{b,\text{avg}} - T_{b,\text{min}})}}{(q.A_i)}
\]  

(5.11)

The value of ‘\( \omega \)’ will be lower for heat sinks with lower maximum temperature and average substrate temperature closer to the minimum substrate temperature.

From the discussions shown in sections 5.3.4 and 5.3.6, it is evident that aspect ratio influenced heat transfer coefficient and total thermal resistance. Hence aspect ratio can be expected to influence the temperature distribution in substrate also. The influence of pumping power and aspect ratio on ‘\( \omega \)’ is shown in Figure 5.12 for microchannels of width 200 µm, 150 µm and 100 µm respectively. It is evident from Figure 5.12 that ‘\( \omega \)’ decreased with pumping power for all microchannel widths and aspect ratios simulated.

At a constant pumping power, ‘\( \omega \)’ decreased with increase in aspect ratio for 200 µm and 150 µm wide microchannels. This indicates that the uniformity of substrate temperature increased with the use of high aspect ratio channels. In other terms, either the maximum temperature in the substrate was lowered or average and minimum substrate temperatures were closer or both. This could be attributed to higher coolant velocity in high aspect ratio microchannels at the same pumping power. With respect to 100 µm wide microchannel, increase in aspect ratio beyond 8 did not lead to reduction in ‘\( \omega \)’, due to ineffectiveness of coolant to remove heat from the fin tip. Hence ‘\( \omega \)’ was found to be increased at the aspect ratio of 10, despite comparable substrate minimum temperature with that at the aspect ratio of 8.
Figure 5.12. Non-uniformity in substrate temperature ‘\( \omega \)’ vs pumping power.
Comparing the ‘ω’ – pumping power relationship for all channel widths and aspect ratios simulated, the lowest ‘ω’ for a fixed pumping power was obtained with microchannels of 200 µm wide at the aspect ratio of 6.

Hence it is evident from the above discussions that the use of microchannels of width 200 µm and aspect ratio 6 with the configuration shown in Figure 5.1 can be used as flow conduits to reduce the substrate temperature gradient.

5.3.8 Applicability at higher heat fluxes

In order to ascertain whether the optimum microchannel width and aspect ratio are dependent on the heat flux, simulations were carried out to obtain ‘ω’ vs ‘P’ relationship for a heat flux of $2 \times 10^6$ W/m$^2$ for MCHS containing 200 µm wide microchannels of aspect ratio 6 and 100 µm wide microchannels of aspect ratio 8.

Figure 5.13 shows the variation of ‘ω’ with pumping power for these two sinks at heat fluxes of $1 \times 10^6$ and $2 \times 10^6$ W/m$^2$. It is evident from Figure 5.13 that the sink with 200 µm wide channels of aspect ratio 6 performs better than that of 100 µm wide channels of aspect ratio 8. Therefore, the optimum values of ‘$W_c$’ and ‘$\alpha$’ are 200 µm and 6 respectively at higher heat flux also.
Figure 5.13. Non-uniformity in substrate temperature ‘\( \omega \)’ vs pumping power for two different heat fluxes and microchannel dimensions.

5.3.9 Effect of fin width \((W_{\text{fin}})\)

To evaluate the effect of the fin width on the heat transfer performance of the new microchannel heat sink, the fin width was varied from 50 \( \mu \text{m} \) to 200 \( \mu \text{m} \) with fixed channel width of 200 \( \mu \text{m} \) and channel depth of 1200 \( \mu \text{m} \).

Figure 5.14 shows the influence of fin width on the total thermal resistance – pumping power relationship of the new MCHS. A closer inspection of Figure 5.14 reveals that the total thermal resistance increased with decrease in fin width. This observation can be interpreted better if the contribution of conductive, convective and capacitive resistances to the total thermal resistance was considered.
The variation of convective thermal resistance with fin width, shown in Figure 5.15, indicates that the convective thermal resistance decreased with reduction in fin width. The convective thermal resistance is inversely proportional to the product of heat transfer area (wall area) and the heat transfer coefficient. It is evident from Table 5.3, that the reduction in fin width resulted in increase in number of channels as well as the wall area. The magnitude of increase in heat transfer area can be expected to reflect as the corresponding reduction in convective thermal resistance. For instance, the reduction in fin width from 150 μm to 100 μm resulted in ~5 % increase in wall area, while the total cross-sectional areas for coolant flow were same. The corresponding reduction in the convective thermal resistance due to reduction in fin width from 150 μm to 100 μm was
in the range of 1-4.5 % (calculated using the trend lines) with higher percentage reductions at lower pumping power. While 15 % increase in wall area was obtained due to reduction in fin width from 100 μm to 50 μm, the corresponding reduction in the convective thermal resistance was in the range of 7-8 % (calculated using the trend lines).

The total cross-sectional area for coolant flow increased by 16 % due to reduction in fin width (from 100 μm to 50 μm) resulting in reduction in coolant velocity, which could have partly offset the reduction in convective thermal resistance caused by increased wall area. This argument holds good with respect to reduction in convective resistance due to reduction in fin width from 200 μm to 150 μm also, for which the increase in both the wall area (15 %) and cross-sectional area (20 %) partly countered each other to provide only a feeble reduction in convective thermal resistance (<3 %). The reduction in fin width resulted in reduction in capacitive thermal resistance as well (Figure 5.16), with about 7-9 % reduction in capacitive thermal resistance in the pumping power range of 0.05-0.2 W, when the fin width was reduced from 200 μm to 150 μm. The reduction in fin width from 200 μm to 150 μm resulted in increase in channel cross-sectional area by 20 % and hence higher volume of coolant could be accommodated at any instant, contributing to reduction in capacitive thermal resistance. Similarly, reduction in capacitive thermal resistance due to reduction in fin width from 100 μm to 50 μm may also be attributed to the increase in coolant volume at any instant of time.

Among the conductive, convective and capacitive thermal resistances, the contribution of the conductive thermal resistance to the total thermal resistance was the highest. Across the entire range of fin widths and pumping powers studied, the conductive thermal resistance was about 40-67 % of the total thermal resistance. Hence the influence of fin
width on the conductive thermal resistance is likely to play an important role in the total thermal resistance – fin width relationship.

Figure 5.15. Influence of fin width on convective thermal resistance – pumping power relationship for the new MCHS.

Table 5.3. Variation of wall area and total cross-sectional area for coolant flow, with the fin width.

<table>
<thead>
<tr>
<th>Fin width (µm)</th>
<th>Channel width (µm)</th>
<th>Wall area (m²)</th>
<th>Source area (m²)</th>
<th>No of channels</th>
<th>Total flow cross-sectional area (m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>200</td>
<td>200</td>
<td>0.00011822</td>
<td>0.0000279</td>
<td>5</td>
<td>0.0000012</td>
</tr>
<tr>
<td>150</td>
<td>200</td>
<td>0.0001353</td>
<td>0.0000279</td>
<td>6</td>
<td>0.00000144</td>
</tr>
<tr>
<td>100</td>
<td>200</td>
<td>0.000142</td>
<td>0.0000279</td>
<td>6</td>
<td>0.00000144</td>
</tr>
<tr>
<td>50</td>
<td>200</td>
<td>0.00016362</td>
<td>0.0000279</td>
<td>7</td>
<td>0.00000168</td>
</tr>
</tbody>
</table>
Among the three resistances, conductive thermal resistance showed the weakest dependence on the pumping power, while theoretically, the same is independent of pumping power. The conductive thermal resistance is inversely proportional to the area for conduction, approximated as $N*Lc*W_{fin}$, where $N$, $Lc$ and $W_{fin}$ are the number of channels, length of a channel and the fin width respectively [57]. The reduction is fin width led to reduction in area for conduction. Hence, the conductive thermal resistance increased with reduction in fin width (Figure 5.17).
Figure 5.17. Influence of fin width on conductive thermal resistance – pumping power relationship for the new MCHS.

Summarizing the effect of fin width on the three resistances, conductive thermal resistance increased with decrease in fin width, while the convective and capacitive thermal resistances decreased with decrease in fin width. Being the significant contributor to the total thermal resistance, the conductive thermal resistance – fin width relationship influenced the total thermal resistance – fin width relationship. Hence, 200 μm was found to be the optimum fin width.

5.4 CONCLUSIONS

Numerical investigations were carried out to identify appropriate microchannel width (100-200 μm) and aspect ratio (2-10) of a four-compartment, microchannel heat sink
consisting of channels with miter bends. The pumping power was found to increase with aspect ratio. Heat transfer coefficient per unit pumping power increased with channel width and aspect ratio due to larger coolant volume, with the influence of aspect ratio being more predominant than that of channel width. The lowest thermal resistance for a fixed pumping power was obtained with 200 µm wide microchannels of aspect ratio 6. A new measure of substrate temperature non-uniformity (ω) has been introduced taking into account of substrate maximum, average and minimum temperatures. It is widely considered that high aspect ratio microchannels of lower width provide better heat transfer performance in non-compartmental, MCHS containing straight microchannels. However, our study revealed that the 200 µm wide microchannels with 200 µm fin width and aspect ratio 6 provided the best thermal performance in terms of heat transfer coefficient per unit pumping power, total thermal resistance and substrate temperature gradient at constant pumping power leading to ease of fabrication.