IMPROVING THERMAL PERFORMANCE OF MICROCHANNEL ELECTRONIC HEAT SINK USING SUPERCRITICAL CO$_2$ AS COOLANT

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Abstract

In view of increasing tendency of power density of electronic systems, cooling performance improvement of microchannel heat sink is an emerging issue. In the present article, supercritical CO$_2$ is proposed as a heat transfer fluid in micro-channel heat sink for power electronics cooling. Energetic and exergetic performance analyses of microchannel heat sink using supercritical CO$_2$ have been done and compared with conventional coolant, water. To take care of sharp change in properties in near critical region, the discretization technique has been used for simulation. Effects of both operating and geometric parameters (heat flux, flow rate, fluid inlet temperature, channel width ratio and channel numbers) on thermal resistance, heat source (chip) temperature, pressure drop, pumping power and entropy generation are presented. Study shows that the thermal resistance, heat source temperature and pumping power are highly dependent on CO$_2$ inlet pressure and temperature. Supercritical CO$_2$ yields better performance than water for certain range of fluid inlet temperature. For the studied ranges, maximum reduction of thermal resistance by using CO$_2$ is evaluated as 30%. Present study reveals that there is an opportunity to use supercritical CO$_2$ as coolant for power electronic cooling at lower ambient temperature.

Keywords: Supercritical CO$_2$, microchannel heat sink, electronics cooling

1. Introduction

The conventional cooling technologies, including natural air convection and forced air convection, have become hard to cool the high heat flux (≥100 W/cm$^2$) in high power electronic applications; such as, integrated power electronics module (IPEM). The two important objectives in electronics cooling—minimization of the maximum substrate temperature and reduction of substrate temperature gradients—can be achieved by the use of liquid-cooled microchannels. The microchannel heat sink (MCHS) exhibits great potential in power electronics cooling since the first fabrication by Tuckerman and Pease [1], which cooled a heat flux of 790 W/cm$^2$ with a temperature rise of 71°C. This landmark has activated great interests in application of MCHS for electronics cooling and many subsequent R&D works [2] have been done to make it as one of the most efficient technologies for high power electronics applications.

Within last three decades, large numbers of research and development activities have been done on MCHS, including heat transfer and pressure drop characteristics [3-4], effects of fluid property variation [5], the importance of scaling effects of fluid flow [6] and dimensional optimization [7]. In general, there are two options for performance improvement of MCHS: (i) material and design optimization of MCHS and (ii) use of suitable heat transfer fluid. Use of
suitable heat transfer liquid should fulfill several criteria such as, performance, operating range, thermal and chemical stability, etc. Water was initially proposed as heat transfer fluid and it is still best choice for electronics applications. Apart from this, ethylene glycol-water brine, methanol-water mixture, FC-72, etc. were also proposed. Recently, nanofluids also have been proposed as heat transfer fluid and lots of numerical and experimental works for nanofluid-cooled MCHS have been conducted [8-14]. In the present study, supercritical CO$_2$ is being proposed as a heat transfer fluid in MCHS for electronics cooling.

Recently, the CO$_2$ has been emerged as working substance for various engineering applications. Apart from its easy availability, environmental friendliness and personal safety natures, supercritical carbon dioxide yields excellent thermophysical properties. Hence, the CO$_2$ has been revived as a suitable refrigerant for various refrigeration and air-conditioning devices [15-16]. Apart from this, it has been emerged as working substance in supercritical Brayton cycle for next generation nuclear reactor and solar applications [17]. It has also proposed working fluid for supercritical Rankine power cycle for the utilization of low grade heat [18]. Recently, the supercritical CO$_2$ has been proposed as heat transmission fluid for geothermal heat extraction [19], solar thermal collector [20] and many other applications [21-25]. However, to the best of the authors’ knowledge, no study on MCHS has been conducted by using supercritical CO$_2$ as a coolant for electronic cooling.

In this study, the energetic as well as exergetic performance analyses of microchannel heat sink using supercritical CO$_2$ have been conducted for electronic applications and compared with conventional coolant, water. To take care of the sharp change in thermophysical and transport properties in near critical region, the discretisation technique has been used for simulation of MCHS. Effects of heat flux, fluid inlet temperature, fluid mass flow rate and channel geometry (channel with ratio, number of channels and aspect ratio) on chip temperature, thermal resistance, pumping power and entropy generation are presented.

Fig. 1 Cooling circuit and a computational segment
2. Mathematical Modeling and Simulation

The cooling circuit of a high power integrated circuit (IC) chip is shown in Fig.1 [8]. A microchannel heat sink (Fig. 2) is used to extract heat generated in chip and a heat exchanger (gas cooler) is used to reject heat to the ambient. The coolant flows through parallel rectangular channels of heat sink (Fig. 2) and is acted as a heat transmission fluid between heat sink and heat exchanger. A pump is used to maintain required pressure rise for flow through MCHS – heat exchanger system. Single layered MCHS structure [14] is considered in this study and supercritical CO$_2$ is proposed as a coolant. As the main aim of this study to promote supercritical CO$_2$ as coolant in MCHS, the performance of MCHS using supercritical CO$_2$ has been only studied. The following assumptions have been made for the analysis:

1) Single-phase, both heat transfer and fluid flow are in steady state
2) The effects of natural convection and radiation are neglected
3) Heat generation through chip surface is uniform
4) Coolant is equally distributed through channels and both heat transfer and fluid flow characteristics in each channel are identical (only varying with channel length)

It may be noted that properties of CO$_2$ changes abruptly near the critical point. To consider the lengthwise property variation, the MCHS has been discretised, and momentum, energy and entropy balance equations have been applied to each segment. For given channel width ratio and number of flow channels, channel pitch, channel width and fin thickness can be calculated by,

\[
P_{ch} = \frac{W_{hs} - 2t_w}{N_c + \beta - 1} \quad W_{ch} = \beta P_{ch} \quad W_{fin} = (1 - \beta) P_{ch}
\]

(1)

MCHS has been divided lengthwise into number of small segments of length $\Delta L$ as shown in Fig. 2. Applying energy balance at $i^{th}$ computational segment:

\[
\dot{m}_f \left( h_{f_i}^{i+1} - h_{f_j}^i \right) = \dot{Q}^i = W_{ch} \Delta L q''
\]

(2)

\[
\frac{T_{s_i}^i - T_{f_i}^i}{R_{tot}} = \dot{Q}^i = W_{ch} \Delta L q''
\]

(3)
Assuming that the heat generated in chip is absorbed by coolant only and neglecting thermal resistance of solder, the total thermal resistance of a segment is consist of conductive thermal resistances of copper and AlN layers, convective thermal resistance and capacitive thermal resistance. Hence,

\[ R_{\text{tot}} = R_{\text{Cu}} + R_{\text{AlN}} + R_{\text{conv}} + R_{\text{cap}} = \frac{t_{\text{Cu}}}{k_{\text{Cu}} W_{\text{hs}} \Delta L} + \frac{t_{\text{AlN}}}{k_{\text{AlN}} W_{\text{hs}} \Delta L} + \frac{1}{\eta_0 \alpha_j A_{\text{eff}}} + \frac{1}{2 m_f c_{pf}} \]  

(4)

Here, product of effective area and overall fin efficiency is given by [2,13],

\[ \eta_0 A_{\text{eff}} = N_c \left[ W_{\text{ch}} + 2 H_{\text{ch}} \eta_{\text{fin}} \right] \Delta L \]  

(5)

Where, fin efficiency, \( \eta_{\text{fin}} \) is given by [2],

\[ \eta_{\text{fin}} = \frac{\tanh(m H_{\text{ch}})}{m H_{\text{ch}}}, \quad m = \sqrt{\frac{2 \alpha_f}{k_{\text{Cu}} W_{\text{fin}}}} \]  

(6)

The coolant heat transfer coefficient in a segment is given by,

\[ \alpha_f = Nu \frac{k_f}{D_h} \]  

(7)

No correlation is available particularly for supercritical CO\(_2\) heating in rectangular microchannel. Many studies (e.g. [23]) showed that single-phase correlation can be applied for supercritical CO\(_2\) also. Ducoulombier et al. [26] used Hausen correlation (widely used for laminar flow with entrance effect) for supercritical CO\(_2\) cooling in circular microchannel. Hence, Hausen correlation applicable for rectangular channel with four sides heated [13] has been used. Therefore, the average Nusselt number of a segment is given by [13]:

\[ Nu = Nu_\infty + \frac{0.14 Gz}{1 + 0.05 Gz^{2/3}} \]  

(8)

Where, the Nusselt number for the fully developed laminar flow is given by (Re<2200) [13],

\[ Nu_\infty = 8.235 \left( 1 - \frac{2.0421}{Ar} + \frac{3.0853}{Ar^2} - \frac{2.4765}{Ar^3} + \frac{1.0578}{Ar^4} - \frac{0.1861}{Ar^5} \right) \]  

(9)

And, the dimensionless Graetz number determines the importance of entrance effect which should be taken into account when Gz > 10

\[ Gz = \frac{Re Pr D_h}{L} \quad L = (i - 1 + i / 2) \Delta L \]  

(10)

Now, the pressure in a segment is given by [4],

\[ p_j^i - p_j^{i+1} = \Delta p_j = \frac{\left( m_f / N_c \right)^2}{2 \rho_f A_c} 4 f_{\text{app}} \frac{\Delta L}{D_h} \]  

(11)

Where, apparent Fanning friction factor is expressed as \((\text{Re}<2300, \text{Ar}<10)\) [4]:

\[ f_{\text{app}} = \frac{1}{\text{Re}} \left[ 3.2 \left( \text{Re} D_h / \Delta L \right)^{0.57} \right]^{0.5} + \left[ 4.7 + 19.64 \left( \text{Ar}^2 + 1 \right) / (\text{Ar} + 1)^2 \right]^{0.5} \]  

(12)

The pressure drop across the heat sink consists of three parts: the pressure drop across the flow channels, the pressure drop at the inlet due to the flow constriction and at the exit due to the flow expansion. Hence,
\[ \Delta p_{\text{tot}} = \sum \Delta p_{\text{segment}} + \left( \frac{m_f}{N} \right)^2 \frac{K}{2 \rho_f A_c^2} \]  

Where, the loss coefficient \( K \) at inlet and outlet of heat sink is given as [2]:

\[ K = 0.6 \left[ \left( \frac{N_f A_c}{W_{hc} H_{ch}} \right)^2 - 2.4 \left( \frac{N_c A_c}{W_{hc} H_{ch}} \right) \right] + 1.8 \]  

Hence, the pumping power is calculated by,

\[ PP = \left( \frac{m_f}{\rho_{f,inlet}} \right) \Delta p_{\text{tot}} \eta_{\text{pump}} \]  

Now, the overall thermal resistance is expressed as:

\[ R_{\text{th,hs}} = \frac{(T_s - T_{f,inlet})}{\sum \dot{Q}_i} \]  

The entropy generation in the heat sink can be calculated by,

\[ \dot{S}_{\text{gen}} = m_f \left( s_{f,\text{exit}} - s_{f,\text{inlet}} \right) - \frac{(h_{f,\text{exit}} - h_{f,\text{inlet}})}{T_s} \]  

A computer (FORTRAN) code has been developed to simulate the MCHS using water and supercritical CO\(_2\) as coolants for various geometry and operating parameters. The temperature dependent correlations available in open literatures [13] have been used for thermophysical properties of water and the previously developed property code CO2PROP [27] is used for thermodynamic and transport properties of supercritical carbon dioxide. Property variation is very abrupt near the critical region and the MCHS encompasses this region. To consider this variation, the entire length of the MCHS has been divided equally into several discrete segments and in each segment, heat transfer coefficient for CO\(_2\) is calculated based on mean values. This way, the MCHS is made equivalent to a number of MCHSs arranged in series and the combined heat transfer of all the segments is the total heat transfer of the MCHS. Therefore, fast changing properties of supercritical CO\(_2\) have been modelled accurately in the MCHS. For given heat flux, dimensions and fluid inlet temperature, the followings are calculation steps for each segment: (i) total resistance (Eqs. 1,4-10), (ii) source temperature and fluid exit temperature (Eqs. 2-3) and pressure drop (Eqs. 11-12). Then the heat sink performance parameters (Eqs. 13-17) and average chip temperature has been calculated. The grid dependent test has showed that the result becomes nearly independent on number of segments (e.g. error in fluid exit temperature is less than 0.1% per segment) for number of segments above 15. Hence the number of segment has been taken as 20 for simulation to confirm accuracy.

### 3. Results and Discussion

Suitability of using supercritical CO\(_2\) as heat transmission fluid in single-layered single-sided MCHS is examined in this study for power electronics cooling. Following dimensions are considered [13]: length and width of heat sink are 10mm both, thickness of Cu layer is 300\(\mu\)m, thickness of AlN is 635\(\mu\)m and channel height is 300\(\mu\)m. Thermal conductivities of Cu and AlN are 387.6W/mK and 180W/mK, respectively [13]. CO\(_2\) pressure is taken as 75bar and to sustain this high pressure, thicker side wall of 0.3mm has taken (no restriction has been used for internal fin thickness due to similar pressure in both sides. Unless otherwise stated, channel number and fluid inlet temperature have been taken as 100 and 30\(^\circ\)C, respectively.

To the best of authors’ knowledge, no previous result is available in open literature for MCHS with supercritical CO\(_2\). Hence, the code has been quantitatively validated with the experimental data given in Ref. [13] for water. Comparison has been done for channel number of
100, channel height of 287µm, channel width of 55µm, fin width of 45µm, which will yield channel width ratio of 0.55 and aspect ratio of 5.22. For inlet water temperature of 297K, inlet water velocity of 2m/s and heat flux of 277W/cm², the experimental and simulation values of thermal resistance reported in the Ref. [13] are 0.113K/W and 0.105K/W, respectively, whereas, the present simulation code yield the value of 0.109K/W. Hence, the present code is under-predicted about 4% from experimental data and this better matching is due to discretization. The various performance parameters of electronic microchannel heat sink are exhibited graphically as elucidated below.

![Graph](image)

**Fig. 3** Effect of channel width ratio for constant mass flow rate

Figs. 3 and 4 show the variations of thermal resistance and pumping power with channel width ratio for constant fluid mass flow rate of 0.001kg/s and number of channels of 100, respectively (Re varies from 575 to 1745 and increases lengthwise). For given constant mass flow rate, with the increase in width ratio, R_{cap} remains constant and R_{conv} increases due to dual effects of decrease in heat transfer coefficient (as Re decreases due to decrease in mass velocity) and insignificant increase in surface area, and hence thermal resistance increases monotonically. Whereas, the pressure drop as well as pumping power decreases due to increase in hydraulic diameter for nearly constant volume flow rate. However, for given constant inlet velocity, the mass flow rate increases and hence R_{cap} monotonically decreases with increase in channel width ratio, whereas, the heat transfer coefficient decreases due to increase in hydraulic diameter (as the heat transfer coefficient is inversely proportional to hydraulic diameter) and effective area increases insignificantly, and hence as a result, R_{conv} increases. Due to this dual effect of R_{cap} and R_{conv}, the thermal resistance decreases initially and after reaching some minimum value, again increases with increase in channel width ratio and yield some optimum value corresponding to minimum resistance. Thus, the optimal resistance is a trade-off between the heat transfer area and the channel flow resistance. As shown in Fig. 4, the optimum channel width ratio is 0.7 for water (similar to previous studies), whereas, 0.75 for CO₂. As shown, the pumping power insignificantly decreases because the increase of channel cross-section area and decrease of
Poiseuille number. Practically, maintaining constant mass flow rate is easier than constant inlet velocity of channel.

Fig. 4 Effect of channel width ratio for constant inlet velocity

Figures 5-9 show the variations of average heat source (chip) temperature, thermal resistance, pressure drop, pumping power and entropy generation with number of channels by using supercritical CO₂ for channel width ratio of 0.75 and mass flow rate of 0.001kg/s for various heat flux conditions (Re varies from 427 to 2219). With $N_c$ increasing, $R_{\text{cap}}$ keeps independent due to constant mass flow rate, whereas, $R_{\text{conv}}$ decreases due to continuous increase in both heat transfer coefficient (decrease in hydraulic diameter) and effective heat transfer area, and hence thermal resistance decreases monotonically (Figure 6). Hence, the overall temperature deference decreases due to constant heat transfer rate and hence source temperature decreases with increase in $N_c$ (Figure 4). On the other hand, the mass flow rate per channel decreases as well as the cross section area of a channel also decreases. Hence, as the second effect is more predominant than first one, the pressure drop as well as pumping power increases due to decrease in mass velocity with increasing $N_c$. Entropy generation occurs due to thermal resistance (temperature difference) and flow resistance (pressure drop). Here, entropy generation due to thermal resistance decreases and entropy generation due to flow resistance increases with increasing $N_c$. As the first part is more predominant, total entropy generation decrease with increase in $N_c$. As the overall dimensions and mass flow rate are same, it is obvious that both thermal resistance and the heat source temperature and hence entropy generation will be more for higher heat flux. Whereas, the differences in pressure drop and pumping power with variation of heat flux are mainly due to property variation associated with change in operating condition. Figure 10 shows the performance characteristic curve (thermal resistance versus pumping power). The thermal resistance decreases rapidly with the pumping power and then tended to reach a saturated. As discussed, it seems to be always desirable to increase $N_c$ for decrease in thermal resistance; however, the extremely large pressure drop for high $N_c$ makes it impractical to further increase the channel number after a critical value, hence, some trade-off between these two
parameters are needed to optimize the channel numbers. As shown in Figure 8, optimum channel number should be in the range of 90 to 100 as the decrease in entropy generation is negligible thereafter.

Fig. 5 Effect of channel number on heat source temperature

Fig. 6 Effect of channel number on thermal resistance
Fig. 7 Effect of channel number on channel pressure drop

Fig. 8 Effect of channel number on pumping power
Fig. 9 Effect of channel number on entropy generation

Fig. 10 Variation of pumping power with thermal resistance

Table 1: Performance comparison (\(m_f=0.001\)kg/s, \(N_c=100\), \(\beta=0.75\))

<table>
<thead>
<tr>
<th>Coolant</th>
<th>(q'') W/cm(^2)</th>
<th>(T_{f,\text{inlet}}) °C</th>
<th>(T_s) °C</th>
<th>(R_{h,Ja}) K/W</th>
<th>(PP) W</th>
<th>(\sum Q_i) PP W</th>
<th>(\dot{S}_{\text{gen}}) W/K</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>200</td>
<td>20</td>
<td>44.6</td>
<td>0.1226</td>
<td>0.163</td>
<td>1225</td>
<td>0.0352</td>
<td></td>
</tr>
<tr>
<td>200</td>
<td>30</td>
<td>54.5</td>
<td>0.1221</td>
<td>0.143</td>
<td>1401</td>
<td>0.0328</td>
<td></td>
</tr>
<tr>
<td>300</td>
<td>20</td>
<td>56.8</td>
<td>0.1223</td>
<td>0.154</td>
<td>1947</td>
<td>0.0747</td>
<td></td>
</tr>
<tr>
<td>300</td>
<td>30</td>
<td>66.7</td>
<td>0.1219</td>
<td>0.136</td>
<td>2206</td>
<td>0.0698</td>
<td></td>
</tr>
<tr>
<td>Supercritical CO(_2)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>200</td>
<td>20</td>
<td>55.2</td>
<td>0.1755</td>
<td>0.068</td>
<td>2955</td>
<td>0.0566</td>
<td></td>
</tr>
<tr>
<td>200</td>
<td>30</td>
<td>45.1</td>
<td>0.0751</td>
<td>0.095</td>
<td>2101</td>
<td>0.0446</td>
<td></td>
</tr>
<tr>
<td>300</td>
<td>20</td>
<td>58.9</td>
<td>0.1353</td>
<td>0.069</td>
<td>4347</td>
<td>0.0971</td>
<td></td>
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<tr>
<td>300</td>
<td>30</td>
<td>60.7</td>
<td>0.0961</td>
<td>0.113</td>
<td>2648</td>
<td>0.0937</td>
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</tr>
</tbody>
</table>
Performance comparison of supercritical CO\textsubscript{2} with water is shown in Table 1. As the property variation of water in negligible, the thermal resistance is nearly constant and hence the heat source temperature is only dependent on heat flux and inlet temperature. Whereas, the thermal resistance for supercritical CO\textsubscript{2} is highly dependent on CO\textsubscript{2} inlet pressure as well as temperature due to abrupt change in thermophysical and transport properties. Re for supercritical CO\textsubscript{2} is about 10-12 times more than water and increases significantly along the channel length due to decrease in viscosity. Hence, the heat source temperature is a strong function of fluid pressure, temperature and heat flux. Similar explanation is applicable for pumping power also and pumping power of supercritical CO\textsubscript{2} is lower due to lower viscosity. Hence, performance index (ratio of heat transfer rate to pumping power) is higher for supercritical CO\textsubscript{2}. Due to lower temperature rise, the overall temperature difference between heat source and fluid is more for supercritical CO\textsubscript{2}, which leads to higher entropy generation rate compared to water. Interestingly, the thermal resistance for CO\textsubscript{2} is higher at 20°C and lower at 30°C, which can be detailed illustrated in Figure 11. As shown, the supercritical CO\textsubscript{2} can give better performance than water for inlet temperature higher than 22°C and maximum reduction of thermal resistance is evaluated as 22% for inlet temperature of 29°C. Hence, the superiority of supercritical CO\textsubscript{2} as coolant in MCHS for electronic cooling can be observed for certain range of fluid inlet temperatures, which is dependent on heat flux, flow rate and geometry. At near critical operation, CO\textsubscript{2} is better as it exhibits superior thermophysical and transport properties (such as very high specific heat capacity and thermal conductivity, and lower viscosity) than water (however, water is better in most of the operating conditions). Supercritical CO\textsubscript{2} can be used for the inlet temperature up to critical temperature (≈31°C) only.

4. Conclusions
Energetic as well as exergetic performance analyses of microchannel heat sink using supercritical CO\textsubscript{2} have been done to study effects of various operating and design parameters for electronic applications and compared with water. For given fluid inlet temperature and mass flow rate, the thermal resistance increases with increase in channel width ratio. However, for given
fluid inlet temperature and inlet velocity, the thermal resistance yields some minimum value at optimum channel width ratio. The optimum channel width ratio is 0.7 for water whereas, 0.75 for supercritical CO$_2$. Result shows that the pumping power requirement decreases with increase in channel width ratio. Variation with number of channels shows that with channel number increasing, thermal resistance decreases and hence source temperature decreases, pressure drop as well as pumping power increases and entropy generation decreases. Both thermal resistance and the heat source temperature and hence entropy will be more for higher heat flux. Whereas, the difference in pressure drop and pumping power with variation of heat flux are mainly due to property variation associated with change in operating condition. The thermal resistance decreased rapidly with the pumping power and then tended to reach a saturated and hence, some trade-off between these two parameters is needed to optimize the channel numbers.

Dissimilar to water, due to abrupt change in supercritical CO$_2$ properties, the thermal resistance and pumping power is highly dependent on CO$_2$ inlet pressure and temperature. Hence, the heat source temperature is a strong function of fluid pressure, temperature and heat flux. Results show that the MCHS with supercritical CO$_2$ yields lower thermal resistance compared to water for certain range of fluid inlet temperature (i.e. for heat flux of 300W/cm$^2$ and flow rate of 0.001kg/s, CO$_2$ can give better performance for the minimum inlet temperature of 22°C). However, CO$_2$ can be used for the inlet temperature up to critical temperature ($\approx$31°C) only. For the studied ranges, the maximum reduction of thermal resistance upto 30% can be achieved for optimum inlet temperature. Study shows that supercritical CO$_2$ is suitable for lower ambient temperature as heat transmission fluid in microchannel heat sink for electronic cooling.

Nomenclature

- $A_c$: channel cross-sectional area
- $A_r$: channel aspect ratio (= channel height/channel width)
- $c_{pf}$: fluid specific heat capacity, J/kgK
- $D_h$: channel hydraulic diameter, m
- $G_z$: Graetz number
- $H_{ch}$: channel height, mm
- $h_f$: fluid specific enthalpy, J/kg
- $k_{Cu}, k_{AlN}$: thermal conductivity, W/mK
- $k_f$: fluid thermal conductivity, W/mK
- $L_{hs}$: heat sink or channel length, mm
- $m_f$: fluid mass flow rate, kg/s
- $N_c$: number of micro-channels
- $N_u$: Nusselt number
- $P_{ch}$: channel pitch, mm
- $p_f$: fluid pressure, kPa
- $P_{P}$: pumping power, W
- $Pr$: Prandtl number
- $\dot{Q}$: heat transfer rate, W
- $q''$: heat flux, W/cm$^2$
- $R_{cap}$: capacitive thermal resistance, K/W
- $R_{conv}$: convective thermal resistance, K/W
- $R_{Cu}, R_{AlN}$: conductive thermal resistance, K/W
- $Re$: Reynolds number
- $s$: specific entropy, J/kgK
- $t_{Cu}, t_{AlN}$: thickness of interfacial layer, mm
- $t_w$: side wall thickness, mm
- $T_f$: fluid temperature, °C
- $T_s$: chip temperature, °C
- $W_{ch}$: channel width, mm
- $W_{fin}$: fin width or thickness, mm
- $W_{hs}$: heat sink width, mm
- $\alpha_f$: heat transfer coefficient, W/m$^2$K
- $\beta$: channel width ratio (= channel width/channel pitch)
- $\Delta L$: segmental length, mm
- $\eta_0$: overall fin efficiency
- $\eta_{pump}$: pump efficiency
- $\rho_f$: fluid density, kg/m$^3$
References


