Numerical study for heat transfer enhancement using CuO-H₂O nano-fluids through minichannel heat sinks for microprocessor cooling

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ABSTRACT

Water cooled heat sinks are becoming popular due to increased heat generation inside the microprocessor. Timely heat removal from microprocessor is the key factor for better performance and long life. Heat transfer enhancement is reached either by increasing the surface area density and/or by altering the base fluid properties. Nano-particles emerge as a strong candidate to increase the thermal conductivity of base fluids. In this research, the thermal performance of mini-channel heat sinks for different fin spacing (0.2mm, 0.5mm, 1mm and 1.5mm) was investigated numerically using CuO-H₂O nano-fluids with volumetric concentration of 1.5%. The numerical values computed were than compared with the literature and a close agreement is achieved. We recorded the minimum base temperature of chip to be 36.8°C for 0.2mm fin spacing heat sink. A reduction of 9.1% in base temperature was noticed using CuO-H₂O nano-fluids for 0.2mm fin spacing as compared to previously experimental estimated value using water [1]. The drop percentage difference in pressure between water and CuO-H₂O nano-fluids was 2.2 to 13.1 % for various fin spacing heat sinks. The percentage difference in thermal resistance between water and CuO-H₂O nano-fluids was computed 12.1% at maximum flow rate. We also observed uniform temperature distribution for all heat sinks.

KEY WORDS: Thermal management, heat transfer enhancement, nano-fluids

1. INTRODUCTION

Cooling of high heat flux electronic devices effectively is one of the challenging tasks. Developments in cooling system for electronic devices has been continuously improved to support advancement in high-tech industries but still with the modern trend towards compactness need further progress in this sector. Therefore, an efficient and effective cooling system is necessary to improve the reliability of these devices and for further development in the IT industry.

More and more contributions by researchers since last few decades, aims to enhance heat transfer rate by manufacturing efficient heat transfer devices. It is achieved either by increasing the surface area density or by using the working fluid having better thermo physical properties. The conventional fluids have reached to their limits of thermal
enhancement. Now studies are focused on working fluids that have suspended nano-particles aim to absorb more heat. These fluids have issues including stability, increased viscosity and pumping power.

Earlier, air cooling technique was the most prominent technique in electronic cooling. Sivasankaran et al. [2] in their study performed experimental and theoretical analysis on parallel plate heat sinks with optimum fan distance. Thanigaivelan et al. [3] investigated micro heat sinks with different material and coatings. Many researchers focused on geometrical design of minichannels to enhance the heat transfer rate. Mertzger et al. [4] in their study, explored circular cross-sectional area with different dimensions and spacing. Liao et al. [5] in their study on pin-fin arrangement, found that the Nusselt number decreases with increasing the ratio between lengths to diameter of pin-fins. Bianchini et al. [6] investigated arrangement of pin fins with circular cross section. Kelkar et al. [7] investigated the staggered pins on parallel plates forming a channel. They found higher pressure drop as compare to heat transfer. Yang et al. [8] addressed the various geometrical features of offset strip fins. Xie et al. [9] optimized the topology of pin fin for heat exchangers. Xie et al. [10] performed numerical study to examine water cooled mini channel heat sink performance in different arrangement of chips.

The nano-fluids have also been used to enhance the heat transfer rate by increasing the thermal conductivities of the base fluid using suspended nano-particles. Arjun and Kumar [11] performed numerical study to optimize the pin fin heat sink using water and carbon nanotube. Kumar et al. [12] reviewed progression in theoretical modeling of the nano-fluids. Bhogare et al. [13] in their study reviewed application and challenges of nano-fluids used in car radiators as a coolant. Mohammadian et al. [14] investigated nano-fluids performance with different sizes of nano-particles. Rafati et al. [15] investigated experimentally the thermal management of the mini-channels using nano-fluids. Wang et al. [16] estimated the thermal conductivity of nano-particles $\text{Al}_2\text{O}_3$ and CuO in different base fluids i.e. water, vacuum pump fluid, engine oil and ethylene glycol. Lee et al. [17] measured the thermal conductivity of nano-particles experimentally. Wang et al. [18] reviewed nano-fluids and their applications in detail. Sarit [19] in their study presented nano-fluids as a future cooling medium in his studies. Lee and Choi [20] presents the application of nano-particles in their study. Xuan and Li [21] in their study presented the enchantment in heat transfer using nano-fluids. Siddiqui et al. [22] performed experimental study on flat heat sink using $\text{Al}_2\text{O}_3$ and CuO nano-particles.

Chand et al. [23] investigated thermal instability of nano-fluids in a porous medium using Galerkin weighted residuals method. Ahmed et al. in their study of stretching tube in heat sink using nano-fluids investigated the effects of thermal conductivity and dynamic viscosity on heat transfer and boundary layer flow [24]. Hussein et al. [25] in their study of CuO-$\text{H}_2\text{O}$ nano-fluids in open enclosure investigated natural convection in magnetic field. Hussein et al. [26] performed numerical study to investigate natural convection in open cavity of parallelogrammic shape which was filled with Cu–$\text{H}_2\text{O}$ nano-fluids having heat source at the bottom. Mohammed et al. [27] perform a numerical study using different nano-fluids on buoyancy-opposing laminar mixed convection. Chand et al. [28] in their study of onset thermal convection on nano-fluids layer investigated the suspended particles effects in detail.

Saisorn and Wongwises [29] reviewed the study of pressure drop on two-phase flow boiling in micro-channels. Çebi et al. [30] reviewed the flow boiling in enhanced geometries (minichannel and microchannel). Awad et al. reviewed condensation heat transfer [31] and pressure drop [32] in minichannel and microchannel based on different parameters. Taner
and Wongwises performed parametric study of energy, exergy and thermoeconomic analyses on vapor compression and vapor absorption systems [33].

Tariq et al. performed numerical and experimental study on cellular structure to evaluate thermal performance using air, water and nano-fluids as a coolant. The lowest temperature they achieved in their experimental study was; 47.4°C for air [34], 32.3°C for water [35], 26.6°C for Al₂O₃-H₂O and 27.7°C for CuO-H₂O [36]. Bello-Ochende et al. [37] performed numerical study to optimize the micro-channel geometry. Xie, et al. [38] evaluated numerical study on mini channel heat sink using water. The optimized structure was able to dissipate heat flux of 350W/cm². Shoukat et al. [39] studied the stability of nano-fluids and noticed the heat transfer enhancement compared to water. Zunaid et al. [40] performed numerical study on rectangular and semi cylindrical microchannel heat sink and found semi cylindrical heat sink to be the best as compare to rectangular heat sink. Belhadj et al. [41] numerically investigate the microchannel using Al₂O₃-H₂O nano-fluids. Jajja et al. [1] in their experimental study on mini channel heat sink achieved the minimum base temperature of 40.5°C for 0.2mm fin spacing. Saeed et al. evaluated the thermal hydraulic performance of mini-channel heat sinks numerically using water [43] and Al₂O₃-H₂O nano-fluids [44].

Taner et al. investigated techno-economic analysis for drying plant [45], turbine power plant [46], power plant co-firing with olive pits [47], PEM fuel cell [48] and sugar factory in Turkey [49][50]. Dalkılıç et al. carried out thermo-economic analysis on vapour-compression cycle and calculate the payback period for different refrigerants [51].

The temperature distribution in these devices is also important along with maximum absolute value of temperature in continuous functioning. The numerical simulations have advantage over experiments that they can provide very refined/high resolution temperature distribution on the base of the heat sink. Through simulation, we can also check if there are any hot spots formed on the base of the heat sink and modify the design easily. Then, the size of the heat sink can also be optimized by analyzing the temperature and flow distribution. In this study, we first investigated the thermal performance of mini-channel heat sinks for microprocessor through numerical simulations using CuO-H₂O nano-fluids and then compared our numerical results with already published experimental results on same heat sinks using water at heating power of 325W [1].

2. NUMERICAL MODEL

ANSYS 16.0 commercial software was used to solve the numerical model of conjugate heat transfer problem. Following assumptions were taken during the modelling of the numerical solutions.
1. Steady state flow and in compressible
2. Fluid phase was considered as a single phase
3. Turbulent flow was considered
4. No heat generation inside the heat sink and no viscous heating
5. Thermal properties were considered to be constant throughout the flow

Governing equations on the basis of above assumptions were as follows for conservation of mass, momentum, energy, turbulence kinetic energy and turbulence dissipation rate:

\[
\frac{\partial}{\partial x_i} (\rho u_i) = 0
\]  \hspace{1cm} (1)

\[
\frac{\partial}{\partial x_i} (\rho u_i u_j) = - \frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_j} \left[ \left( \mu + \mu_t \right) \frac{\partial u_i}{\partial x_j} \right] + \frac{\partial}{\partial x_i} \left[ \left( \mu + \mu_t \right) \frac{\partial u_j}{\partial x_i} \right] \quad j=1,2,3
\]  \hspace{1cm} (2)
\[
\frac{\partial}{\partial x_i} (\rho u_i T) = \frac{\partial}{\partial x_i} \left[ \left( \frac{\mu_t}{\sigma_p} \frac{\partial T}{\partial x_i} \right) \left( \frac{\lambda}{c_p} + \frac{\mu_t}{\sigma_T} \frac{\partial T}{\partial x_i} \right) \right]
\]
(3)

\[
\frac{\partial}{\partial x_i} (\rho k u_i) = \frac{\partial}{\partial x_i} \left[ \left( \mu + \frac{\mu_t}{\sigma_k} \frac{\partial k}{\partial x_i} \right) + G_k - \rho \varepsilon \right]
\]
(4)

\[
\frac{\partial}{\partial x_i} (\rho \varepsilon u_i) = \frac{\partial}{\partial x_i} \left[ \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \frac{\partial \varepsilon}{\partial x_i} \right) + \frac{\varepsilon}{k} (c_1 G_k - c_2 \rho \varepsilon) \right]
\]
(5)

Where \( k \) is turbulence kinetic energy, \( \varepsilon \) is turbulence dissipation rate and \( G_k \) is generation of turbulence kinetic energy.

Boundary conditions for this study were as:
1. No slip velocity boundary condition at the solid walls
2. The uniform inlet velocity inside the heat sink
   at \( y = h_T, \ u = 0, v = 0, w = U_{in} \)
3. Whereas inside the solid region, the velocity was considered to be zero everywhere
   at solid regions, \( u = 0, v = 0, w = 0 \)
4. Heat flux supplied at the bottom of the heat sink
   at \( y = 0, -\lambda \frac{\partial T}{\partial y} = q \)
5. The right and left surfaces of the structure were considered to be adiabatic
   at \( x = 0, \frac{\partial T}{\partial x} = 0 \)
   at \( x = A, \frac{\partial T}{\partial x} = 0 \)
6. The inlet temperature of fluid was given constant
   at \( y = h_T, \ T = T_i \)

2.1. Mesh sensitivity analysis
A complete mesh study was carried out to sort out the mesh influence on the results. We analyse four different cases ranging from 2 million to 4 million numbers of elements. The solution was considered to be independent of the mesh when the relative temperature difference (\( T_{b} - T_{i} \)) noted less than 1%. It was observed that after 3.5 million numbers of elements, the results did not considerably change as shown in Fig. 1.
2.2. Data reduction

The Volume fraction ($\phi$) of nano-fluids was calculated using eq. (6)

$$\phi = \frac{w_{np} \rho_{bf}}{\rho_{np} (1-w_{np}) + \rho_{bf}}$$  \hspace{1cm} (6)

Density of nano-fluids was calculated using eq. (7)

$$\rho_{nf} = \phi \rho_{np} + (1-\phi) \rho_{bf}$$  \hspace{1cm} (7)

Specific heat capacity of nano-fluids was calculated using eq. (8)

$$C_{nf} = (\phi \rho_{np} C_{np} + (1-\phi) \rho_{bf} C_{bf})/\rho_{nf}$$  \hspace{1cm} (8)

Viscosity of nano-fluids was calculated using eq. (9)

$$\mu_{nf} = \mu_{bf}(1 + 2.5\phi + 6.2\phi^2)$$  \hspace{1cm} (9)

Thermal conductivity of nano-fluids was calculated using eq. (10) by formula for measuring thermal conductivity of nano-fluids by Wasp model [21]

$$K_{nf} = \frac{K_{np} + 2K_{bf} - 2\phi(K_{bf} - K_{np})}{K_{np} + 2K_{bf} + \phi(K_{bf} - K_{np})}$$  \hspace{1cm} (10)

Heat transfer rate was calculated using eq. (11)

$$\dot{Q} = \dot{m}C_{nf}(T_0-T_i)$$  \hspace{1cm} (11)

Log of mean temperature difference was calculated from eq. (12)

$$LMTD = \frac{(T_B - T_i) - (T_B - T_o)}{\ln \left(\frac{T_B - T_i}{T_B - T_o}\right)}$$  \hspace{1cm} (12)

Thermal resistance was calculated from eq. (13)

$$R_{th} = \frac{LMTD}{Q}$$  \hspace{1cm} (13)

2.3. Heat sinks

Heat sinks with different fin spacing are shown in Fig. 2. The heat sinks were designed on ANSYS Design Modeller. All the other parameters of minichannel heat sinks (fin height, fin width, fin length) were kept constant. The effect of fin spacing on thermal
performance was studied in detail. In the base of all heat sinks, there was a square chip of 28.7mm*28.7mm with 0.5mm thickness. Dimensions of all heat sinks are listed in TABLE 1. Label diagram of heat sink is shown in Fig. 3.

Fig. 2 Heat sinks with different spacing

TABLE 1 Heat sinks geometric specifications (mm)

<table>
<thead>
<tr>
<th>l</th>
<th>A</th>
<th>B</th>
<th>h</th>
<th>t</th>
<th>l_b</th>
<th>l_0</th>
<th>t_c</th>
</tr>
</thead>
<tbody>
<tr>
<td>45</td>
<td>55</td>
<td>10</td>
<td>3</td>
<td>1</td>
<td>3</td>
<td>28.7</td>
<td>0.5</td>
</tr>
</tbody>
</table>

The surface area responsible for heat transfer changes with varying fin spacing. Surface area is listed in TABLE 2 for different fin spacing of heat sinks.

TABLE 2 Surface area of heat sinks

<table>
<thead>
<tr>
<th>c (mm)</th>
<th>0.2</th>
<th>0.5</th>
<th>1</th>
<th>1.5</th>
</tr>
</thead>
<tbody>
<tr>
<td>A_s (mm²)</td>
<td>13931</td>
<td>11852</td>
<td>9773</td>
<td>8319</td>
</tr>
</tbody>
</table>
Fig. 3 Schematic of heat sink geometry

Flow propagation from inlet to outlet through minichannel heat sink is shown in Fig. 4. Flow stream lines are represented in multi-colour lines. Flow is coming inside along –ive y-axis while coming out from the channel along +ive y-axis.

Fig. 4 Flow propagation through heat sink

2.4. Temperature distribution on chip base

Temperature distribution on the base of chip for different fin spacing heat sinks is shown in Fig 3, 4, 5, 6. It can be seen clearly that red hot circular region shown in Fig. 8 is shorter as compare to Fig. 5. It is due to the reason that surface area for 0.2mm fin spacing is greater than 1.5mm fin spacing. Heat transfer is greater in case of 0.2mm fin spacing as
compare to 1.5mm fin spacing. The value of temperature at each point of the chip base is given in the legends. The maximum temperature value noted for each heat sink at the flow rate of 0.5 LPM was computed higher as compare to 1LPM volumetric flow rate.

Fig. 5 1.5mm fin spacing a) 0.5LPM, b) 0.75LPM, c) 1LPM

Fig. 6 1mm fin spacing a) 0.5LPM, b) 0.75LPM, c) 1LPM

Fig. 7 0.5mm fin spacing a) 0.5LPM, b) 0.75LPM, c) 1LPM

Fig. 8 0.2mm fin spacing a) 0.5LPM, b) 0.75LPM, c) 1LPM
3. RESULTS AND DISCUSSIONS

Thermal performance of heat sinks with different fin spacing using CuO-H₂O nanofluids computed numerically is presented in this section. Base temperature, pressure drop and thermal resistance with volumetric flow rate are discussed. The results are then compared with published data and a close agreement is found.

3.1. Base temperature with flow rate

Numerical and experimental results of chip base temperature with volumetric flow rate using CuO-H₂O nano-fluids and water, respectively is shown in Fig. 9. The minimum base temperature was recorded for 0.2mm fin spacing heat sink numerically using CuO-H₂O nano-fluids as 36.8°C and experimentally using water as 40.5°C at the flow rate of 1LPM. It is clear from the results that base temperature of heat sinks recorded for CuO-H₂O nano-fluids was lower than water. This fact is because of the higher thermal conductivity of CuO-H₂O nano-fluids as compare to water. It can also be seen that base temperature reduces by decreasing the fin spacing of heat sinks. This is because lower fin spacing heat sinks provides more active area as compare to higher fin spacing heat sinks. An inverse relation between base temperature of chip and volumetric flow rate was found. Increasing the flow rate tends to decrease the base temperature. The minimum base temperature of chip was recorded for 0.2mm fin spacing heat sink as 36.8°C at the flow rate of 1LPM. The experimental value estimated for 0.2mm fin spacing was 40.5°C using water at the same flow rate [1]. A reduction of 9.1% in base temperature was noticed using CuO-H₂O nano-fluids as compared to water at 1LPM. A reduction of 14.1% in base temperature was recorded by Tariq et al. [36] using CuO-H₂O nano-fluids in cellular structure as compare to water. The maximum value of base temperature was recorded as 53.7°C for 1.5mm fin spacing heat sink at the flow rate of 0.5LPM. Experimentally the recorded value for 1.5mm fin spacing was 57.1°C at the same flow rate. The red hot region is found slightly forward from the middle of heat sinks. The numerical data evaluated in this section using CuO-H₂O nano-fluids is following the similar trend of experimental data which was recorded by Jajja et al. [1] in their study using water as a coolant. The result shows a percentage difference of 7.6%, 3.0% and 1.4% for 0.5mm, 1mm and 1.5mm fin spacing respectively, with Saeed and Kim reported values at 1% volumetric concentration of Al₂O₃·H₂O nano-fluids at flow rate of 1LPM [44].

Fig. 9 Base temperature with volumetric flow rate
3.2. Pressure Drop with volumetric flow rate

Pressure drop across minichannel heat sink using water and CuO-H$_2$O nano-fluids with respect to volumetric flow rate is shown in Fig. 10. A direct relation was observed between pressure drop and volumetric flow rate. Maximum pressure drop was observed in case of using 0.2mm fin spacing heat sink while the minimum pressure drop was found for 1.5mm fin spacing heat sink. More over a minor pressure difference was seen between water and CuO-H$_2$O nano-fluids. The highest value of pressure drop using CuO-H$_2$O nano-fluids was noted 7240 Pa for 0.2mm fin spacing heat sink at 1LPM. The lowest value recorded was 100Pa using CuO-H$_2$O nano-fluids for 1.5mm fin spacing heat sink at 0.5LPM. The pressure drop difference was estimated 2.2 to 13.1 % between water and CuO-H$_2$O nano-fluids. Pressure drop noted values for 1mm and 1.5mm fin spacing were very close. A remarkable pressure drop difference was seen for 0.2mm fin spacing heat sink. This fact was eminent because reducing the fin spacing actually results in increase of number of fins in the vicinity. More pressure drop will require more pumping power. The values noted for water are lower as compare to CuO-H$_2$O. This fact is due to the increased viscosity by adding nano-particles in water for nano-fluids. Saeed et al. reported the same trend for pressure drop in their study [44]. They also reported that increasing the volumetric concentration of Al$_2$O$_3$-H$_2$O nano-fluids, pressure drop increases.

![Pressure drop with volumetric flow rate](image)

**Fig. 10** Pressure drop with volumetric flow rate

3.3. Thermal resistance with flow rate

Thermal resistance for different spacing heat sinks with volumetric flow rate is shown in Fig. 11. It can be seen that thermal resistance decreases with increasing the volumetric flow rate. Lowest thermal resistances was found for 0.2mm fin spacing while highest values noted for thermal resistances was for 1.5mm fin spacing heat sink. This fact is because of the higher heat transfer surface area provided by the 0.2mm fin spacing heat sink as compare to 1.5mm fin spacing. Thermal resistance is inversely related to rate of heat transfer. Lower thermal resistance value depicts the greater amount of heat transfer while higher value depicts low heat transfer. High thermal resistance value shows that heat transfer founds greater resistance to flow. The computed values using CuO-H$_2$O nano-fluids are lower as compare to the water which shows greater amount of heat transfer possible for CuO-H$_2$O nano-fluids as compare to
water. Maximum heat transfer is hereby possible by using 0.2mm fin spacing heat sink. Moreover, thermal resistance values using CuO-H₂O nano-fluids were found lesser as compare to water as shown in Fig. 11. The minimum computed value of thermal resistance for 0.2mm fin spacing heat sink was 0.033°C/W and 0.029°C/W using water and CuO-H₂O nano-fluids respectively. The percentage difference between water and CuO-H₂O nano-fluids was computed 12.1% at maximum flow rate. The maximum value of thermal resistance computed for 1.5mm fin spacing was 0.083°C/W and 0.075°C/W using water and CuO-H₂O nano-fluids respectively. The percentage difference was found 9.6% between water and CuO-H₂O nano-fluids at minimum flow rate. Saeed and Kim found the similar trend of thermal resistance with volumetric flow rate in their study using Al₂O₃-H₂O nano-fluid. They estimated the percentage difference for 0.5mm, 1mm and 1.5mm fin spacing as 10.2%, 6.7% and 10.8%, respectively for maximum flow rate at 1% volumetric concentration [44].

![Thermal resistance with volumetric flow rate](image)

**Fig. 11** Thermal resistance with volumetric flow rate

### 4. ECONOMIC ANALYSIS

Economic analysis for four different manufactured fin spacing heat sinks was carried out on the basis of initial cost and cost of electricity required for pump to flow the fluid in the closed loop. The details are given in TABLE 3.

<table>
<thead>
<tr>
<th>Fin spacing (mm)</th>
<th>Pumping power (W)</th>
<th>Cost/unit (PKR)</th>
<th>Initial cost (PKR)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.2</td>
<td>3.12</td>
<td>10.48</td>
<td>10000</td>
</tr>
<tr>
<td>0.5</td>
<td>3.02</td>
<td>10.14</td>
<td>9000</td>
</tr>
<tr>
<td>1</td>
<td>3.008</td>
<td>10.11</td>
<td>8000</td>
</tr>
<tr>
<td>1.5</td>
<td>3.006</td>
<td>10.10</td>
<td>7000</td>
</tr>
</tbody>
</table>

Pumping losses against auxiliary components were added for each fin spacing heat sink. The data is of full time run for 14 days. It was analysed that though 0.2mm fin spacing provides better heat transfer but it cost 3% higher PKR (Pakistani Rupee) per unit of electricity as compare to 1.5mm fin spacing. On the basis of pumping power, 1.5mm fin spacing was the best heat sink and save cost per unit of electricity.
5. CONCLUSION

The thermal performance of mini-channel heat sinks with fin spacing of 0.2mm, 0.5mm, 1mm and 1.5mm was investigated numerically using CuO-H₂O nano-fluids. We deduced following conclusions from this work:

- We found the minimum base temperature to be 36.8°C using CuO-H₂O nano-fluids for 0.2mm fin spacing. This value is 9.1% smaller as compared to previously reported value of 40.5°C using water [1].
- The minimum and maximum base temperature of chip was calculated to be 36.8°C and 53.7°C using 0.2mm and 1.5mm fin spacing heat sink respectively.
- The base temperature decreases with increasing the volumetric flow rate.
- Pressure drop was found increasing with increase in the volumetric flow rate.
- Maximum pressure drop using CuO-H₂O nano-fluids was estimated as 7240Pa for 0.2mm fin spacing heat sink while the lowest value was 100Pa for 1.5mm fin spacing heat sink.
- The pressure drop percentage difference was between 2.2 to 13.1 % for water and CuO-H₂O nano-fluids for varied fin spacing heat sinks.
- Higher thermal resistance was estimated for 1.5mm fin spacing while the minimum thermal resistance was offered for 0.2mm fin spacing heat sink.
- The thermal performance of CuO-H₂O nano-fluids was better as compare to water.
- A uniform temperature distribution was observed in all heat sinks.

NOMENCLATURE

A = Width of finned section, (mm)
Aₚ = Surface area, (mm²)
B = Un-finned length, (mm)
c = Fin spacing, (mm)
Cₙf = Specific heat of nano-fluids, kJ/kgK
Cₚ = Specific heat of nano-particles, kJ/kgK
Cₙbf = Specific heat of base fluid, kJ/kgK
h = Height of fins, (mm)
hT = Total height from bottom to inlet/outlet top surface, (mm)
Kₙf = Thermal conductivity of nano-fluids, W/mK
Kₚ = Thermal conductivity of nano-particles, W/mK
Kₙbf = Thermal conductivity of base fluid, W/mK
l = Length of fins, (mm)
lₒ = Square chip base length, (mm)
LMTD = Log of mean temperature difference, °C
LPM = Litres Per Minute
ṁ = Mass flow rate, kg/s
ΔP = Pressure Difference, Pa
Q = Heat transfer rate, W
Q = Volumetric flow rate, m³/s
q = Heat flux, W/cm²
Rₒ = Thermal resistance, °C/W
Tb = Base temperature, °C
t = Thickness of fins, (mm)
tb = Thickness of heat sink base plate, (mm)
tc = Chip thickness, (mm)
\[ T_i = \text{Fluid Inlet Temperature, } ^\circ\text{C} \]
\[ T_o = \text{Fluid Outlet Temperature, } ^\circ\text{C} \]
\[ U_{in} = \text{Inlet velocity} \]
\[ U,v,w = \text{Velocity in x,y,z respectively (m/s)} \]
\[ w_{np} = \text{Weight fraction of nano-particle} \]
\[ w_{bf} = \text{Weight fraction of base fluid} \]

Greek symbols:

\[ \alpha_{sf} = \text{Surface area density} \]
\[ \mu_t = \text{Turbulence viscosity (kg/ms)} \]
\[ \lambda = \text{Thermal conductivity (W/m}^0\text{C)} \]
\[ \rho_{bf} = \text{Density of base fluid, kg/m}^3 \]
\[ \rho_{np} = \text{Density of nano-particles, kg/m}^3 \]
\[ \rho_{nf} = \text{Density of nano-fluids, kg/m}^3 \]
\[ \mu_{nf} = \text{Dynamic viscosity of nano-fluids, kg/ms} \]
\[ \mu_{bf} = \text{Dynamic viscosity of base fluid, kg/ms} \]
\[ \phi = \text{Volume fraction} \]

**Bibliography**


