NUMERICAL STUDY FOR HEAT TRANSFER ENHANCEMENT USING CuO-WATER NANOFLUIDS THROUGH MINI-CHANNEL HEAT SINKS FOR MICROPROCESSOR COOLING

by

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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. Nanoparticles 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 (0.2 mm, 0.5 mm, 1 mm, and 1.5 mm) was investigated numerically using CuO-water nanofluids 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 0.2 mm fin spacing heat sink. A reduction of 9.1% in base temperature was noticed using CuO-water nanofluids for 0.2 mm fin spacing as compared to previously experimental estimated value using water [1]. The drop percentage difference in pressure between water and CuO-water nanofluids was 2.2-13.1% for various fin spacing heat sinks. The percentage difference in thermal resistance between water and CuO-water nanofluids 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, nanofluids*

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.

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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 nanoparticles 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 mini-channels 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 nanofluids have also been used to enhance the heat transfer rate by increasing the thermal conductivities of the base fluid using suspended nanoparticles. Arjun and Kumar [11] performed numerical study to optimize the pin fin heat sink using water and CNT. Kumar *et al.* [12] reviewed progression in theoretical modelling of the nanofluids. Bhogare *et al.* [13] in their study reviewed application and challenges of nanofluids used in car radiators as a coolant. Mohammadian *et al.* [14] investigated nanofluids performance with different sizes of nanoparticles. Rafati *et al.* [15] investigated experimentally the thermal management of the mini-channels using nanofluids. Wang *et al.* [16] estimated the thermal conductivity of nanoparticles Al_2O_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 nanoparticles experimentally. Wang *et al.* [18] reviewed nanofluids and their applications in detail. Sarit [19] in their study presented nanofluids as a future cooling medium in his studies. Lee and Choi [20] presents the application of nanoparticles in their study. Xuan and Li [21] in their study presented the enchantment in heat transfer using nanofluids. Siddiqui *et al.* [22] performed experimental study on flat heat sink using Al_2O_3 and CuO nanoparticles.

Chand *et al.* [23] investigated thermal instability of nanofluids in a porous medium using Galerkin weighted residuals method. Ahmed *et al.* [24] in their study of stretching tube in heat sink using nanofluids investigated the effects of thermal conductivity and dynamic viscosity on heat transfer and boundary-layer flow. Hussein *et al.* [25] in their study of CuO-water nanofluids 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-water nanofluids having heat source at the bottom. Mohammed *et al.* [27] perform a numerical study using different nanofluids on buoyancy-opposing laminar mixed convection. Chand *et al.* [28] in their study of onset thermal convection on nanofluids 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. Cebi *et al.* [30] reviewed the flow boiling in enhanced geometries (mini-channel and micro-channel). Awad *et al.* [31, 32] reviewed condensation heat transfer and pressure drop in mini-channel and micro-channel based on different parameters. Dalkilic *et al.* [33] performed parametric study of energy, exergy and thermoeconoimc analyses on vapour compression and vapour absorption systems.

Tariq *et al.* [34-36] performed numerical and experimental study on cellular structure to evaluate thermal performance using air, water and nanofluids as a coolant. The lowest temperature they achieved in their experimental study was; 47.4 °C for air, 32.3 °C for water, 26.6 °C for Al_2O_3-water and 27.7 °C for CuO-water. 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 350 W/cm^2. Shoukat *et al.* [39] studied the stability of nanofluids and noticed the heat transfer enhancement compared to water. Zunaid *et al.* [40] perform numerical study on rectangular and semi cylindrical micro-channel heat sink and found semi cylindrical heat sink to be the best as compare to rectangular heat sink. Belhadj *et al.* [41] performed numerical study on heat sink with different cross-section cavities inside micro-channel. Belhadj *et al.* [42] numerically investigate the micro-channel using Al_2O_3-water nanofluids. Jajja *et al.* [1] in their experimental study on mini-channel heat sink achieved the minimum base temperature of 40.5 °C for 0.2 mm fin spacing. Saeed and Kim [43, 44] evaluated the thermal hydraulic performance of mini-channel heat sinks numerically using water and Al_2O_3-water nanofluids.

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

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 analysing 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-water nanofluids and then compared our numerical results with already published experimental results on same heat sinks using water at heating power of 325 W [1].

Numerical model

The 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:
– Steady-state flow and in compressible.
– Fluid phase was considered as a single phase.
– Turbulent flow was considered.
– No heat generation inside the heat sink and no viscous heating.
– Thermal properties were considered to be constant throughout the flow.

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

$$\frac{\partial}{\partial x_i}\left(\rho u_i\right) = 0 \tag{1}$$

$$\frac{\partial}{\partial x_i}\left(\rho u_i u_j\right) = -\frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_i}\left[\left(\mu+\mu_t\right)\frac{\partial \mu_j}{\partial x_i}\right] + \frac{\partial}{\partial x_i}\left[\left(\mu+\mu_t\right)\frac{\partial \mu_i}{\partial x_j}\right], \quad j=1,2,3 \tag{2}$$

$$\frac{\partial}{\partial x_i}\left(\rho u_i T\right) = \frac{\partial}{\partial x_i}\left[\left(\frac{\lambda}{c_p}+\frac{\mu_t}{\sigma_T}\right)\frac{\partial T}{\partial x_i}\right] \tag{3}$$

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

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

where k is turbulence kinetic energy, ε – turbulence dissipation rate, and G_k – the generation of turbulence kinetic energy.

Boundary conditions for this study were as:

– No slip velocity boundary condition at the solid walls.
– The uniform inlet velocity inside the heat sink at $y = h_T$, $u = 0$, $v = 0$, and $w = U_{in}$.
– Whereas inside the solid region, the velocity was considered to be zero everywhere at solid regions, $u = 0$, $v = 0$, and $w = 0$.
– Heat flux supplied at the bottom of the heat sink at $y = 0$, $-\lambda(\partial T/\partial y) = q$.
– The right and left surfaces of the structure were considered to be adiabatic at $x = 0$, $\partial T/\partial x = 0$ and at $x = A$, $\partial T/\partial x = 0$.
– The inlet temperature of fluid was given constant at $y = h_T$, $T = T_i$.

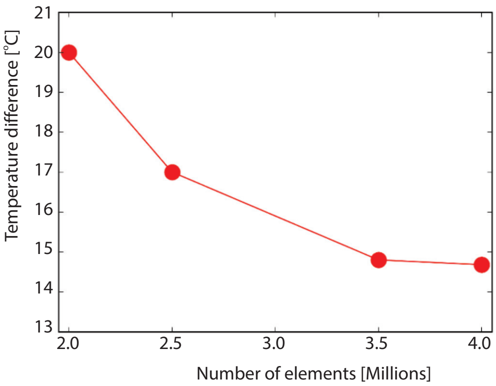

Figure 1. Temperature difference with number of elements

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 2-4 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.

Data reduction

The volume fraction, ϕ, of nanofluids was calculated:

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

Density of nanofluids was calculated:

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

Specific heat capacity of nanofluids was calculated:

$$C_{\mathrm{nf}} = \frac{[\phi\rho_{\mathrm{np}}C_{\mathrm{np}} + (1-\phi)\rho_{\mathrm{bf}}C_{\mathrm{bf}}]}{\rho_{\mathrm{nf}}} \tag{8}$$

Viscosity of nanofluids was calculated:

$$\mu_{\mathrm{nf}} = \nu_{\mathrm{bf}}(1 + 2.5\phi + 6.2\phi) \tag{9}$$

Thermal conductivity of nanofluids was calculated using eq. (10) by formula for measuring thermal conductivity of nanofluids by wasp model [21]:

$$\frac{K_{\mathrm{nf}}}{K_{\mathrm{bf}}} = \frac{K_{\mathrm{np}} + 2K_{\mathrm{bf}} - 2\phi\left(K_{\mathrm{bf}} - K_{\mathrm{np}}\right)}{K_{\mathrm{np}} + 2K_{\mathrm{bf}} + \phi\left(K_{\mathrm{bf}} - K_{\mathrm{np}}\right)} \tag{10}$$

Heat transfer rate was calculated:

$$\dot{Q} = \dot{m}C_{\mathrm{nf}}(T_{\mathrm{o}} - T_{\mathrm{i}}) \tag{11}$$

Log of mean temperature difference was calculated:

$$LMTD = \frac{(T_{\mathrm{B}} - T_{\mathrm{i}}) - \left(T_{\mathrm{B}} - T_{\mathrm{o}}\right)}{\ln\left(\dfrac{T_{\mathrm{B}} - T_{\mathrm{i}}}{T_{\mathrm{B}} - T_{\mathrm{o}}}\right)} \tag{12}$$

Thermal resistance was calculated:

$$R_{\mathrm{th}} = \frac{LMTD}{\dot{Q}} \tag{13}$$

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 mini-channel 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 28.7 mm × 28.7 mm with 0.5 mm thickness. Dimensions of all heat sinks are listed in tab. 1. Label diagram of heat sink is shown in fig. 3.

Table 1. Heat sinks geometric specifications [mm]

l	A	B	h	t	t_b	l_b	t_c
45	55	10	3	1	3	28.7	0.5

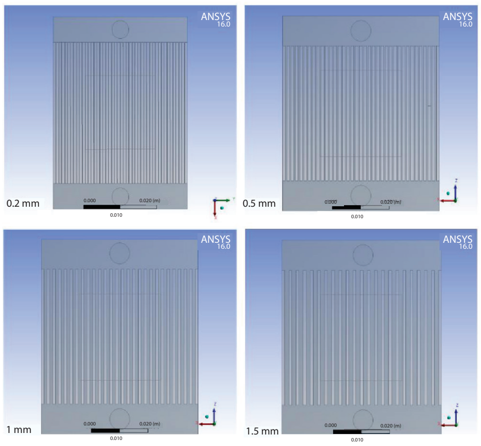

Figure 2. Heat sinks with different spacing

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

Table 2. Surface area of heat sinks

c [mm]	0.2	0.5	1	1.5
A_s [mm^2]	13931	11852	9773	8319

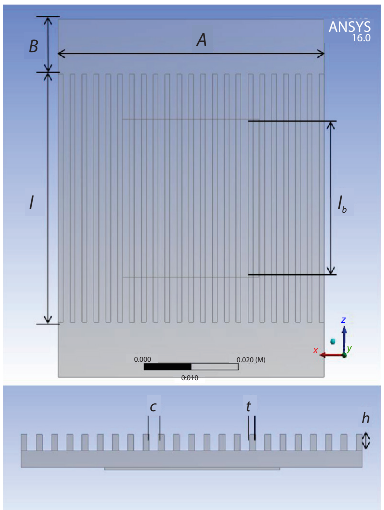

Figure 3. Schematic of heat sink geometry

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

Temperature distribution on chip base

Temperature distribution on the base of chip for different fin spacing heat sinks is shown in figs. 5-8. 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 0.2 mm fin spacing is greater than 1.5 mm fin spacing. Heat transfer is greater in case of 0.2 mm fin spacing as compare to 1.5 mm 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 1 Lpm volumetric flow rate.

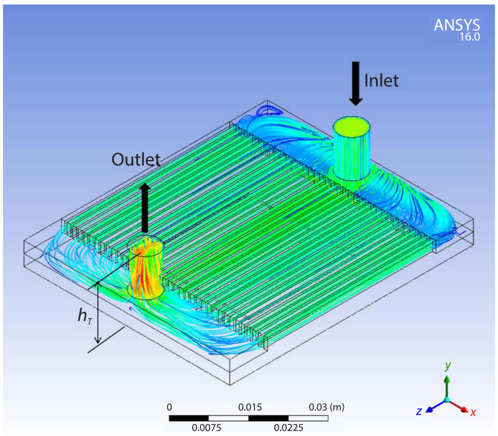

Figure 4. Flow propagation through heat sink

Results and discussions

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

Base temperature with flow rate

Numerical and experimental results of chip base temperature with volumetric flow rate using CuO-water nanofluids and water, respectively is shown in fig. 9. The minimum base temperature was recorded for 0.2 mm fin spacing heat sink numerically using CuO-wa-

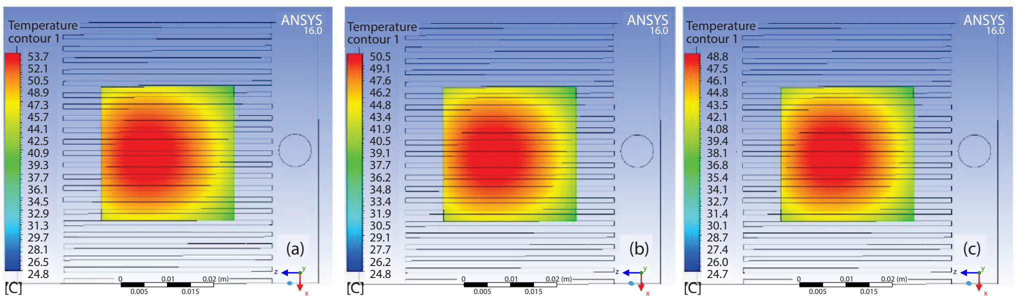

Figure 5. The 1.5 mm fin spacing; (a) 0.5 Lpm, (b) 0.75 Lpm, and (c) 1 Lpm

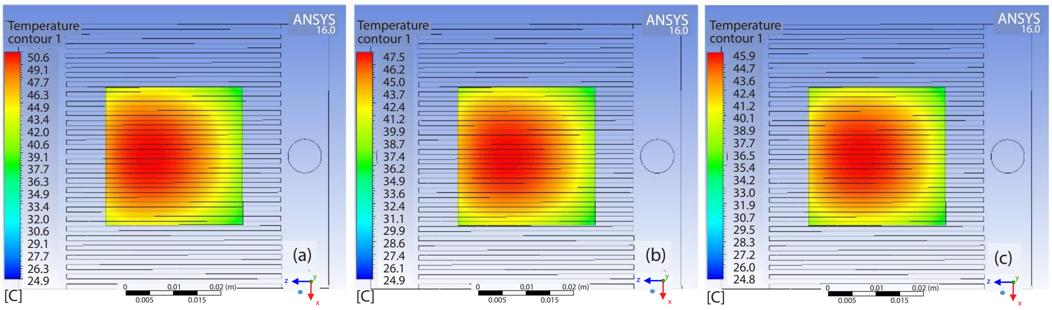

Figure 6. The 1 mm fin spacing; (a) 0.5 Lpm, (b) 0.75 Lpm, and (c) 1 Lpm

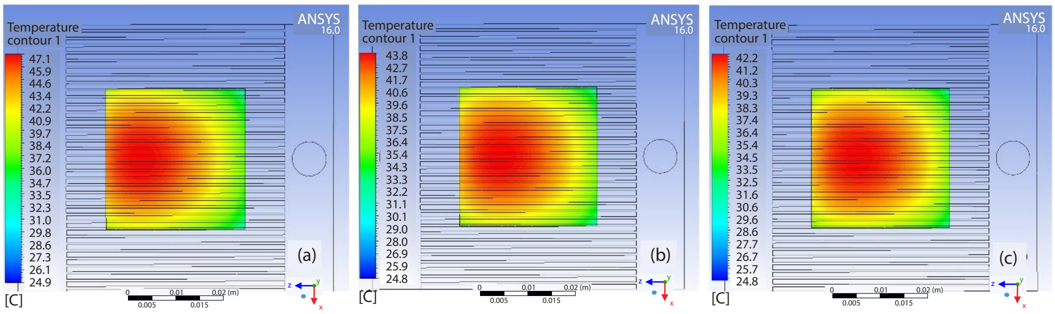

Figure 7. The 0.5 mm fin spacing; (a) 0.5 Lpm, (b) 0.75 Lpm, and (c) 1 Lpm

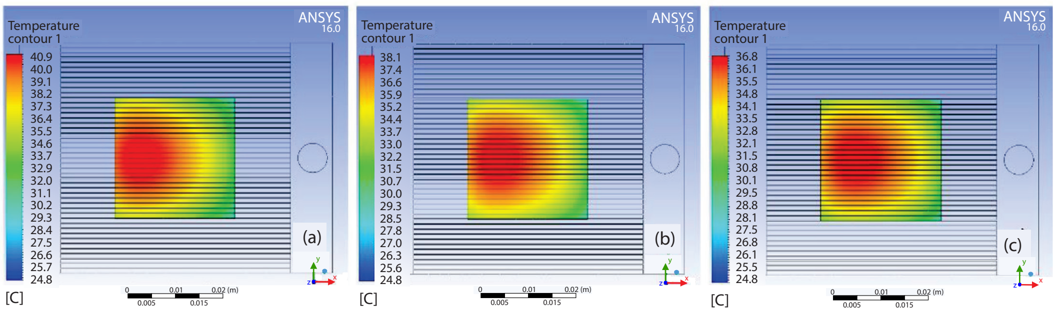

Figure 8. The 0.2 mm fin spacing; (a) 0.5 Lpm, (b) 0.75 Lpm, and (c) 1 Lpm

ter nanofluids as 36.8 °C and experimentally using water as 40.5 °C at the flow arte of 1 Lpm. It is clear from the results that base temperature of heat sinks recorded for CuO-water nanofluids was lower than water. This fact is because of the higher thermal conductivity of CuO-water nanofluids 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 0.2 mm fin spacing heat sink as 36.8 °C at the flow rate of 1 Lpm. The experimental value estimated for 0.2 mm 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-water nanofluids as compared to water at 1 Lpm. A reduction of 14.1% in base temperature was recorded

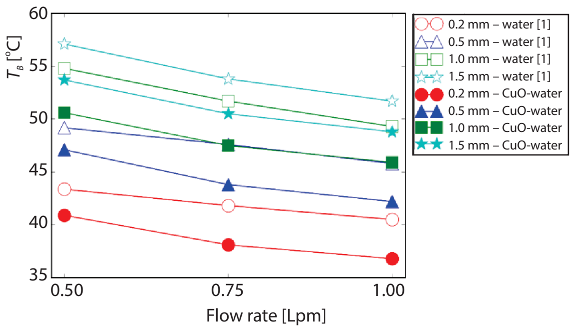

Figure 9. Base temperature with volumetric flow rate

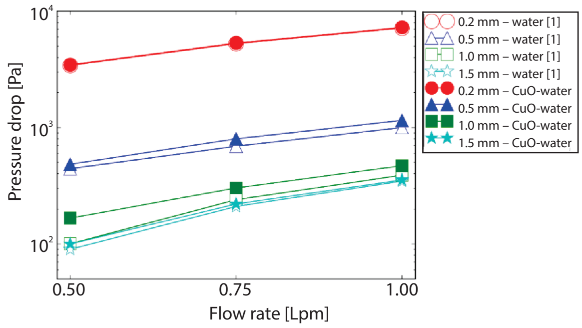

Figure 10. Pressure drop with volumetric flow rate

by Tariq *et al.* [36] using CuO-water nanofluids in cellular structure as compare to water. The maximum value of base temperature was recorded as 53.7 °C for 1.5 mm fin spacing heat sink at the flow rate of 0.5 Lpm. Experimentally the recorded value for 1.5 mm 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-water nanofluids 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 0.5 mm, 1 mm, and 1.5 mm fin spacing, respectively, with Saeed and Kim [44] reported values at 1% volumetric concentration of Al_2O_3-water nanofluids at flow rate of 1 Lpm.

Pressure drop with volumetric flow rate

Pressure drop across mini-channel heat sink using water and CuO-water nanofluids 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 0.2 mm fin spacing heat sink while the minimum pressure drop was found for 1.5 mm fin spacing heat sink. More over a minor pressure difference was seen between water and CuO-water nanofluids. The highest value of pressure drop using CuO-water nanofluids was noted 7240 Pa for 0.2 mm fin spacing heat sink at 1 Lpm. The lowest value recorded was 100 Pa using CuO-water nanofluids for 1.5 mm fin spacing heat sink at 0.5 Lpm. The pressure drop difference was estimated 2.2-13.1% between water and CuO-water nanofluids. Pressure drop noted values for 1 mm and 1.5 mm fin spacing were very close. A remarkable pressure drop difference was seen for 0.2 mm 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-water. This fact is due to the increased viscosity by adding nanoparticles in water for nanofluids. Saeed and Kim [44] reported the same trend for pressure drop in their study. They also reported that increasing the volumetric concentration of Al_2O_3-water nanofluids, pressure drop increases.

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 0.2 mm fin spacing while highest val-

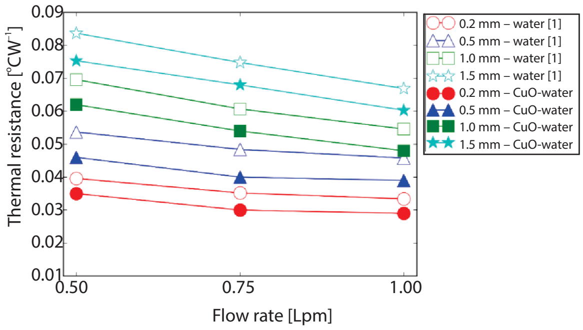

Figure 11. Thermal resistance with volumetric flow rate

ues noted for thermal resistances was for 1.5 mm fin spacing heat sink. This fact is because of the higher heat transfer surface area provided by the 0.2 mm fin spacing heat sink as compare to 1.5 mm 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-water nanofluids are lower as compare to the water which shows greater amount of heat transfer possible for CuO-water nanofluids as compare to water. Maximum heat transfer is hereby possible by using 0.2 mm fin spacing heat sink. Moreover, thermal resistance values using CuO-water nanofluids were found lesser as compare to water as shown in fig. 11. The minimum computed value of thermal resistance for 0.2 mm fin spacing heat sink was 0.033 °C/W and 0.029 °C/W using water and CuO-water nanofluids, respectively. The percentage difference between water and CuO-water nanofluids was computed 12.1% at maximum flow rate. The maximum value of thermal resistance computed for 1.5 mm fin spacing was 0.083 °C/W and 0.075 °C/W using water and CuO-water nanofluids, respectively. The percentage difference was found 9.6% between water and CuO-water nanofluids at minimum flow rate. Saeed and Kim found the similar trend of thermal resistance with volumetric flow rate in their study using Al_2O_3-water nanofluid. They estimated the percentage difference for 0.5 mm, 1 mm, and 1.5 mm fin spacing as 10.2%, 6.7%, and 10.8%, respectively for maximum flow rate at 1% volumetric concentration [44].

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 tab. 3.

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 0.2 mm fin spacing provides better heat transfer but it cost 3% higher PKR (Pakistani Rupee, 1 PKR = 0.0061 $) per unit of electricityas compare to 1.5 mm fin spacing. On the basis of pumping power, 1.5 mm fin spacing was the best heat sink and save cost per unit of electricity.

Table 3. Economic analysis

Fin spacing [mm]	Pumping power [W]	Cost/unit [PKR]	Initial cost [PKR)]
0.2	3.12	10.48	10000
0.5	3.02	10.14	9000
1	3.008	10.11	8000
1.5	3.006	10.10	7000

Conclusions

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

- We found the minimum base temperature to be 36.8 °C using CuO-water nanofluids for 0.2 mm 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 0.2 mm and 1.5 mm 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-water nanofluids was estimated as 7240 Pa for 0.2 mm fin spacing heat sink while the lowest value was 100 Pa for 1.5 mm fin spacing heat sink.
- The pressure drop percentage difference was between 2.2-13.1 % for water and CuO-water nanofluids for varied fin spacing heat sinks.
- Higher thermal resistance was estimated for 1.5 mm fin spacing while the minimum thermal resistance was offered for 0.2 mm fin spacing heat sink.
- The thermal performance of CuO-water nanofluids was better as compare to water.
- A uniform temperature distribution was observed in all heat sinks.

Nomenclature

A – width of finned section, [mm]
A_s – surface area, [mm^2]
B – un-finned length, [mm]
C_{nf} – specific heat of nanofluids, [kJkg^{-1}K^{-1}]
C_{np} – specific heat of nanoparticles, [kJkg^{-1}K^{-1}]
C_{bf} – specific heat of base fluid, [kJkg^{-1}K^{-1}]
c – fin spacing, [mm]
h – height of fins, [mm]
h_T – total height from bottom to inlet/outlet top surface, [mm]
K_{nf} – thermal conductivity of nanofluids, [Wm^{-1}K^{-1}]
K_{np} – thermal conductivity of nanoparticles, [Wm^{-1}K^{-1}]
K_{bf} – thermal conductivity of base fluid, [Wm^{-1}K^{-1}]
l – length of fins, [mm]
l_b – square chip base length, [mm]
$LMTD$ – log of mean temperature difference, [°C]
$\dot{m}$ – mass-flow rate, [kgs^{-1}]
$\dot{Q}$ – heat transfer rate, [W]
q – heat flux, [Wcm^{-2}]
R_{th} – thermal resistance, [°CW^{-1}]
T_B – base temperature, [°C]
T_{i} – fluid inlet temperature, [°C]
T_{o} – fluid outlet temperature, [°C]
t – thickness of fins, [mm]
t_b – thickness of heat sink base plate, [mm]
t_c – chip thickness, [mm]
U_{in} – inlet velocity
u, v, w – velocity in x, y, z, respectively [ms^{-1}]
w_{np} – weight fraction of nanoparticle
w_{bf} – weight fraction of base fluid

Greek symbol

μ_{bf} – dynamic viscosity of base fluid, [kgm^{-1}s^{-1}]
μ_{nf} – dynamic viscosity of nanofluids, [kgm^{-1}s^{-1}]
μ_t – turbulence viscosity [kgm^{-1}s^{-1}]]
λ – thermal conductivity [Wm^{-1} °C^{-1}]
ρ_{bf} – density of base fluid, [kgm^{-3}]
ρ_{np} – density of nanoparticles, [kgm^{-3}]
ρ_{nf} – density of nanofluids, [kgm^{-3}]
ϕ – volume fraction

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