Comparison of heat sink’s fin-spacing using CuO–H₂O-based nanofluids for high heat generating microprocessor: an experimental study

Ahmad Adnan Shoukat¹, M. Zubair Khan², Asif Israr² and Muhammad Anwar²,³

¹Department of Mechanical Engineering, Institute of Space Technology, Islamabad, Pakistan; ²Multiphysics Nanofluidics Laboratory, Sungkyunkwan University, Suwon, Republic of Korea; ³Faculty of Sciences, University of Nottingham, Nottingham, UK

ABSTRACT

Technological advancement in previous years has resulted in decrease in the size of microprocessors and thus, high heat generation. To address this problem, experiments were carried out to optimize the performance of nanofluid-cooled heat sinks at a power of 300 W for better thermal management. The optimization process involved varying both the concentration of the coolant (and by extension, its thermo-physical properties) and the fin spacing of the heat sink (to modify the convective and conductive heat transfer area) to achieve the lowest thermal resistance at different flow rates. Instead of an actual microprocessor, a copper block heated by a mica insulated clamp heater (rated at 285 W) was used to simulate the microprocessor. The evaluation of the heat sink performance was also based on the lowest base temperature achieved, the total heat rejected and the overall heat transfer coefficients for all iterations.

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1. Introduction

The rapid technological advancements in the field of information technology over the last decade or so have resulted in the development of microprocessors with massive processing power and a remarkable decrease in the chip size. This substantial increase in power coupled with the decrease in the size has resulted in a massive increase in the heat flux generated by the processor. For the proper functioning of the processor and to prevent thermal damage it is imperative to remove the heat generated. Conventional air-cooled heat sinks have almost maximized their potential to dissipate the heat generated and struggle to cope with the heat generated by the modern top of the line microprocessors and graphics processing units. The increasing demand for heat removal has given rise to numerous researches to develop efficient cooling systems. As a result, the dated air-cooled heat sinks are gradually being replaced by liquid-cooled heat sinks.

Amongst the resulting researches, the optimization of liquid-cooled heat sinks has been one of the primary focuses. Heat sinks can be optimized using several techniques, the most common of which is to vary the geometry and architecture of the heat sink to achieve enhanced heat transfer. Another technique which has seen prominence in recent researches is to vary the thermo-physical properties of the coolant to optimize the performance of the heat sink such as the use of nanofluids [1]. A nanofluid is a two-phased mixture which has solid Nano-sized particles suspended in a fluid. These suspended particles alter the properties of the base fluid and have prompted researches to observe the various effects of these changes on the fluids’ properties [2]. One of these modified properties and the most relevant to the scope of this study is the enhanced thermo-physical property of the original base fluid upon the addition of the nanoparticles.

Researches to optimize heat sinks gained popularity almost 35 years ago. Investigation of the effect of varying the hydraulic diameter of a micro-channel heat sink has on the heat transfer rate [3] and hence it is concluded that decreasing the effective hydraulic diameter results in an increase in the heat transfer rate [4]. Several other researches change the geometry of heat sink to study the heat transfer rate.

Numerical study of the effect of various dimensional configurations on the heat transfer rate of water-cooled mini-channel heat sinks may increase the channel height and by decreasing the bottom thickness and channel width a higher heat transfer rate can be achieved [5].

By comparing both numerical and experimental results at different Reynolds numbers it is observed that heat transfer effects the fin height of a water-cooled...
mini-channel heat sink [6]. Hence by this experiment and numerical investigation, it is concluded that the dimensions of the mini-channel heat sink substantially influence its thermal performance [7].

More experiments were done on the cooling ability of copper-based nanofluids [8]. On using a 0.3% volumetric loading of copper nanoparticles in ethyl glycol he reported a 40% increase in thermal conductivity [9].

Ho et al. compared the effect of nanofluids with water and concluded that a 1% volume of alumina nanoparticles in water resulted in a 70% enhancement in convective heat transfer as compared to water.

Das et al. [8] experimentally investigated the effect of temperature changes on the thermal capabilities of CuO–H2O. The said nanofluid had a volumetric concentration of 4% and upon an increase of 30°C in temperature they observed a 20% increase in the thermal conductivity of the fluid.

Two conclusions can be drawn from the above literature survey. Firstly, most of the researches done till now show that varying the geometry of the heat sink can significantly alter its thermal performance. Secondly, researches on nanofluids have shown that the presence of nanoparticles in various fluids has resulted in a significant increase in the heat transfer capabilities of the base fluid [10].

2. Experimental setup

Keeping in mind the conclusions drawn from the preceding literature review, the outline of the present study is to experimentally investigate and optimize the performance of a commercially available heat sink by decreasing the fin spacing [11]. The effect of varying the thermo-physical properties of the coolant will also be incorporated. Along with the changes in the channel width, 1% (\( \alpha = 0.12 \)) by weight of water based copper oxide nanofluid (CuO–H2O) will also be used in an attempt to increase the thermal performance of the heat sink.

2.1. Selected heat sinks

The selected heat sinks had their dimensions based on commercially available water blocks. The optimization process began at the heat sink with 2.0 mm fin spacing and then further experiments were carried out on sinks with a spacing of 1.5 mm and 0.2 mm [12]. The dimensions of the heat sinks are shown in Figure 1 where “a” is the channel width and the fin height and thickness is 1.4 mm and 0.5 mm for all heat sinks, respectively.

The selected heat sinks were heated by a 285 W mica-insulated clamp heater wound around a cylindrical copper (heating) block as shown in Figure 2. This heating block acted as a surrogate microprocessor and allowed more control over the experiment as compared to an actual microprocessor which has fluctuations in power due to varying workload. Thermal paste was applied on the sink-block interface to ensure even contact. The exposed surface of the heating block was insulated fibreglass wool to reduce heat loss to the surroundings. A 1 mm deep and 1.5 mm wide slot was milled radially across the face of the heating block to facilitate the insertion of a temperature probe [13].

2.2. Test rig

The test rig consisted of the standard components found in a liquid cooling system. A heat sink to absorb heat from the copper block, a radiator (EK-Cool Stream PE-240 dual fan) to reject the heat absorbed by the coolant and a reservoir-pump combo (EK-XRES DCP 2.2) to drive the coolant through the circuit. In addition to these components, a flow-meter (Dwyer VFA 0-3.1545 LPM) and a needle valve were added to the loop for measuring and altering the volumetric flow rate. A dual channel DC power supply (GW 3030D) was used to power the radiator and the pump. Three digital multimeters with K-type thermocouple probes were used to measure the inlet, outlet and base temperatures [14].
Figure 3 shows the picture of the experimental rig and Figure 4 shows the flow schematic of the rig.

Steady state conditions were assumed when the change in temperature was less than 1°C for five minutes. Each experiment was performed thrice and the obtained readings were averaged to reduce the amount of error involved.

3. Data reduction

3.1. Evaluation formulae

The following formulae are used to reduce the raw experimental readings into the parameters required for heat sink evaluation:

The heat rejected by the sink to the fluid [15]:

\[ Q = \dot{m}C_p(T_{\text{out}} - T_{\text{in}}) \]  

where \( \dot{m} \) is the mass flow rate, \( C_p \) is the specific heat capacity of the coolant and \( T_{\text{out}} \) and \( T_{\text{in}} \) are outlet and inlet temperatures of the coolant respectively.

The log-mean temperature difference (LMTD) is expressed as [16]:

\[ \text{LMTD} = \frac{(T_B - T_{\text{in}}) - (T_B - T_{\text{out}})}{\ln(T_B - T_{\text{in}})} \]  

where \( T_B \) is the base temperature.

The overall heat transfer coefficient where “\( h \)” is the heat transfer area of the heat sink and \( \dot{m} \) is the mass flow rate:

\[ U = \frac{\dot{m}C_p(T_{\text{out}} - T_{\text{in}})}{\text{LMTD} \cdot h}. \]

3.2. Nanofluid calculations

The following formulae were used to calculate the required properties of the nanofluids which are needed to calculate the evaluation parameters.

The densities, specific heat capacities and viscosities of the nanofluids are calculated at the mean temperature “\( T_m \)”: \[ T_m = \frac{T_{\text{out}} + T_{\text{in}}}{2} \]

The volumetric percentage “\( \alpha \)” of the nanofluid is calculated from the known weight percentage using the following formula [9]:

\[ \alpha = \frac{w_{\text{np}}\rho_{bf}}{(\rho_{np}(1 - w_{\text{np}}) + w_{\text{np}}\rho_{bf})} \]

where \( \rho_{np} \), \( w_{\text{np}} \) and \( \rho_{bf} \) are the densities of the nanoparticles, weight percentage of nanoparticles and density of base fluid respectively. The heat sink specs are shown in Table 1.

Table 1. Heat transfer area.

| Channel width (mm) | Heat transfer area mm² |
|-------------------|------------------------|
| 0.2               | 6414.58                |
| 1.5               | 3979.51                |
| 2.0               | 3692.20                |

Figure 4. Flow schematic.
Table 2. Nanofluids properties.

| Properties            | Values          |
|-----------------------|-----------------|
| Volumetric percentage | 1%              |
| Density               | 1027.91 kg/m³   |
| Specific heat capacity | 4021.69 J/kg K |

The density of the nanofluids is calculated by the following formula [6]:

$$\rho_{nf} = \rho_{np} + (1 - \varepsilon)\rho_{bf}. \quad (6)$$

The specific heat capacity is calculated by [17]:

$$C_{nf} = (\varepsilon\rho_{np}C_{np} + (1 - \varepsilon)\rho_{bf}C_{bf})/\rho_{nf}. \quad (7)$$

The viscosity $\mu_{nf}$ of the nanofluid is calculated by the formula [18]:

$$\mu_{nf} = \mu_{bf}(1 + 2.5\varepsilon + 6.2\varepsilon^2). \quad (8)$$

The values of volumetric percentage, density, specific heat capacity is shown in the Table 2.

4. Results and discussion

The following results were obtained after reducing the raw experimental data to the required evaluation parameters:

4.1. Base temperature

The base temperature is very important when evaluating a heat sink since it is analogous to the processor’s temperature [19]. Figure 5 shows how the base temperature varied with the geometry of each heat sink and with the change in the type of coolant at different flow rates. The lowest base temperature obtained was 45.4°C by the heat sink with 0.2 mm spacing and the nanofluid as the coolant (46.6°C for distilled water). The highest base temperature observed at a flow rate of 0.315451, LPM was 62°C for 2.0 mm fin spacing which subsequently decreased to 55°C upon increasing the flow rate to 1.104078 LPM. Upon the use of nanofluid, the reading at 1.104078 LPM dropped by 5.45% to 52°C.

4.2. LMTD

Figure 6 shows the variation of LMTD with a change in fin spacing and with the change in the type of coolant at different flow rates.

The lowest LMTD of 12.6°C was achieved by using nanofluids at 1.104078 LPM with 0.2 mm spacing. A 10.13% decrease in LMTD was achieved by using nanofluid as the coolant instead of water. The largest difference in LMTD (46.8%) at 1.104078 LPM was observed between the 2.0 mm heat sink with water as the coolant and the 0.2 mm heat sink with nanofluid as the coolant.

4.3. Heat rejected

The amount of heat rejected by the heat sink is also an important factor. A large difference between the processor’s heat output and the rate of heat removal by the heat sink can result in high base temperatures and consequently shorten the processor’s life span [20]. Figure 7 shows the amount of heat rejected by each heat sink.

The heat transfer lines did not always follow a fixed pattern. However, all six curves peaked at a particular flow rate after which the rate of heat transfer began to drop [21]. This is because although at high flow rates the mass flow rate increases, the contact time between the fluid and the heat sink (by extension the fluid temperature rise) decreases. When the decrease in temperature change outweighs the increase in mass flow rate, the heat transfer rate begins to decrease [22]. For 0.2 mm fin spacing, the values peaked at a flow rate of 0.946353
LPM after which the heat transfer rate began to tail. The highest heat transfer rate this heat sink achieved was 268.76 W by the nanofluids at 0.946353 LPM. Water achieved a maximum heat transfer rate of 255.67 W. The use of nanofluid improved the heat transfer rate by 4.72%. The heat sink was able to reject 94% of the provided heat [13].

4.4. Thermal resistance

The amount of thermal resistance also provides an important insight regarding the heat sink’s performance. Thermal resistance is a heat property and a measurement of a temperature difference by which an object or material resists a heat flow. A smaller thermal resistance means the heat flow will encounter less resistance. Figure 8 shows the thermal resistance of the heat sinks. The thermal resistance lowers when the fin spacing is lowered and when water is replaced with nanofluids. The effect of using nanofluids and increasing the flow rate was much less drastic on the heat sink with 0.2 mm spacing when compared to the other heat sinks. The lowest thermal resistance achieved by 0.2 mm spacing heat sink was 0.0498°C/W with nanofluid as the coolant. 18.2% more than the 0.0609°C/W thermal resistance achieved by distilled water. The heat sink with 2.0 mm spacing with water as the coolant had the highest thermal resistance (0.198°C/W).

4.5. Overall heat transfer coefficient ‘u’

The overall heat transfer coefficient is an important criterion for comparing the performances of the heat sink since it incorporates the inlet and outlet temperature as well as the base temperature of the heat sink. The decrease in the fin spacing results in a substantial increase in the overall heat transfer coefficient. The minimum value of U was found to be 842.7 J/m²°C at 2.0 fin spacing, almost four times less than the value achieved by 0.2 mm heat sink employing nanofluid as the coolant (3148.81 J/m²°C). The use of nanofluid resulted in a 1.2% increase in U (Figure 9).

5. Conclusion

Evident from the results discussed above it can be seen that by varying the spacing of the heat sink and by varying the thermo-physical properties of the cooling fluid the performance of a heat sink was optimized. At the optimum level of fin spacing (0.2 mm) and by using CuO–H₂O nanofluid the cooling system was able to reject most of the 285 W of heat provided to maintain the base temperature at 45.4°C which is well within the safe working threshold of the microprocessors. The nanofluid used showed great promise as a coolant; however, before nanofluids can be commercialized as CPU coolants, research needs to be done in order to increase their stability.

Disclosure statement

No potential conflict of interest was reported by the authors.

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