Proceedings of the ASME 2009 InterPACK Conference
IPACK2009
July 19-23, 2009, San Francisco, California, USA

InterPACK2009-89327

# ANALYSIS OF NANOFLUIDS IN LIQUID ELECTRONIC COOLING SYSTEMS

**N.A. Roberts and D.G. Walker***
Department of Mechanical Engineering
Vanderbilt University
Nashville, TN USA, 37235

## ABSTRACT

Nanofluids are solutions of a small fraction of suspended nanoparticles in a bulk fluid. Nanofluids have shown great promise as heat transfer fluids over typically used bulk fluids and fluids with micron sized particles. The nanoparticles do not settle in the fluid and do not cause clogging or damage to surfaces as with micron sized particles. In the current work we compare the performance of different volume loadings of water-based alumina nanofluids in a commercially available electronics cooling system to that of pure DI-water. The commercially available system is a water block used for liquid cooling of a computational processing unit. The size of the nanoparticles in the study is varied from 20 nm to 30 nm. Results show an enhancement in convective heat transfer, but not in the temperature increase through a heated tube or commercial cooling system in nanofluids with volume loadings of nanoparticles up to 2% by volume. The current nanofluids showed significant settling within an hour of preparation.

## NOMENCLATURE

$c_p$ specific heat (kJ/kg-K)
$d$ diameter (mm)
$l$ length (m)
$P$ pressure (kPa)
$Q$ heat dissipation (W)
$T$ temperature (K)
$\dot{V}$ flow rate (ml/min)
$V_{\text{heat}}$ heating voltage (V)
$\rho$ density (kg/m$^3$)
Re Reynolds number
Nu Nusselt number

## INTRODUCTION

Nanofluids are colloidal solutions containing a small fraction of nanoparticles in a bulk fluid. Recently there have been several studies that show enhancement of thermal energy transport over Maxwell's model [1] for fluids with particles. These studies have shown an increase in the thermal conductivity with a corresponding increase in the convection heat transfer coefficient. In 1995 Choi and Eastman [2] presented a theoretical model for the enhancement of thermal transport properties of nanofluids over bulk fluids using the two-component mixture model for the effective thermal conductivity developed by Hamilton and Crosser [3]. In 1999 Lee et al. [4] measured the thermal conductivity of fluids with oxide nanoparticles using the transient hot-wire method and found reasonable agreement with Hamilton and Crosser's model. In 2000 Xuan and Li [5] showed that the addition of copper nanophase powders to process fluids increased the fluid thermal conductivity with increasing volume loading. It has yet to be determined what causes the enhancements in thermal conductivity and convection. In 2002 Keblinski et al. [6] presented several potential mechanisms for the enhanced heat conduction and used molecular dynamics simulations to show that Brownian motion does not directly lead to the enhanced heat conduction observed in experiments. Keblinski showed theoretical evidence that particle clustering could explain the enhancement. In 2005 Prasher [7] performed an order-of-

*Address all correspondence to this author.

magnitude analysis of several mechanisms and found that Brownian movement was the only mechanism that could explain the enhancement in thermal conductivity.

Convective heat transfer is of even greater interest than the thermal conductivity. In 2003 and 2005 Xuan and Li [8, 9] showed an increasing Nusselt number, Nu, with increasing volume loading of Cu-water nanofluids and Reynolds number, Re. In 2008 Lai et al. [10] also showed an enhancement in convective heat transfer in water-based alumina nanofluids in a straight tube. Lai's results also showed increased enhancement with increased volume loading. In 2006 Buongiorno [11] analyzed several mechanisms that could be responsible for the observed enhancement in convective heat transfer in nanofluids which were included in previous models.

The current work investigates how nanofluids perform in commercially available liquid cooling systems for computational processing units (CPUs) when compared to deionized (DI) water.

## EXPERIMENTAL SETUP

The experimental setup consists of several different sections. First the experiment designed by Lai et al. [10] was repeated to verify that a similar enhancement in convection heat transfer in the nanofluids is observed in a straight tube with a constant heat flux boundary condition. The nanofluids were then compared to DI-water in the commercially available liquid cooling system (Thermaltake Big Water 760is). The nanofluids used in this study were all alumina based and the base fluid was DI-water. The size of the particles were 20-30 nm in diameter. The nanofluids were sonicated for 1 hour before any measurements were made to ensure any agglomerates were eliminated.

The single heated straight tube setup is used to compare the effectiveness of the current nanofluids to those presented by Lai et al. [10]. A schematic of the setup is shown in Figure 1. This setup consists of a single stainless steel tube with inner and outer diameters of 1.07 mm and 1.47 mm, respectively. A section of the tube is wrapped with a nichrome wire to act as a constant heat flux boundary condition. Thermocouples are inserted into the flow at the inlet and outlet of the tube and 6 additional thermocouples are bonded to the exterior of the tube using a conductive metal-filled epoxy to measure the temperature along the tube. The flow rate and pressure drop are recorded along with the voltage applied to the nichrome wire. The fluid is pumped using a peristaltic pump and the flow rates vary from $0.596\,\mathrm{mL/min}$ to $15.61\,\mathrm{mL/min}$ which corresponds to Reynolds numbers from 14 to 360 for DI-water.

The setup of the commercial system consists of a copper water block (ThermalTake Bigwater 760i system), a silicon strip heater (constant heat flux) and a heat exchanger to maintain a constant inlet temperature. In this setup the heat exchanger exit temperature, the water block inlet and outlet temperatures and the top and bottom temperatures of the heater are recorded and

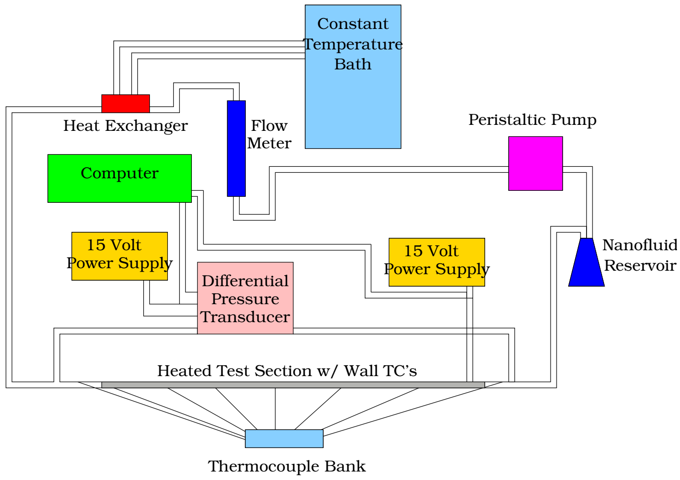


Figure 1. Schematic of the single straight heated tube setup

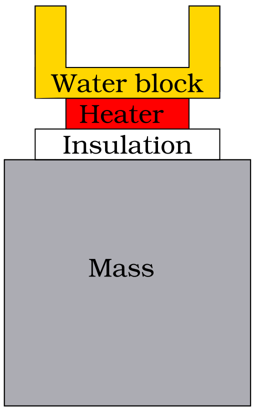


Figure 2. Schematic of the commercial setup around the water block

compared between DI-water and DI-water loaded with nanoparticles. The same pump is used for the working fluid and the same flow meter is used to measure the flow rate. Figure 2 shows the setup of just the area involving the water block. The rest of the setup is the same as in Figure 1 where the commercial system is replaced by the water block, heater, insulation and thermal mass.

## RESULTS

The experimental equipment was first validated for the flow of pure DI-water by measuring the pressure drop through the tube. Figure 3 shows the pressure drop of pure DI-water and the 1% and 2% volume loaded alumina nanofluids. The measured pressure drop showed good agreement with Poiseuille's law [12]

$$\Delta P = \frac{128\mu L\dot{V}}{\pi d_i^4}. \tag{1}$$

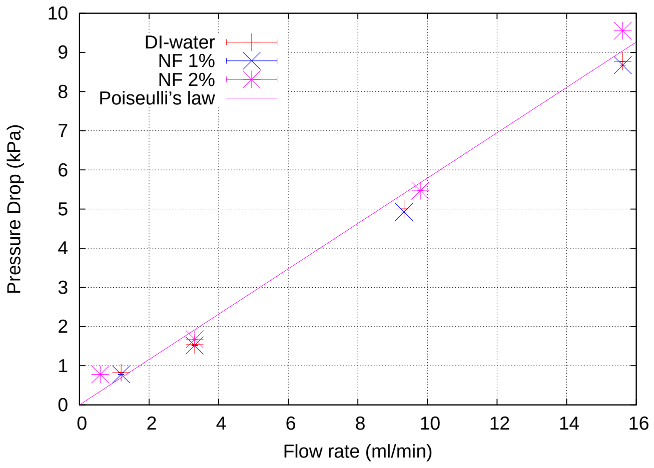


Figure 3. Pressure drop across tube for pure DI-water and the 1% and 2% volume loaded nanofluids as a function of the flow rate

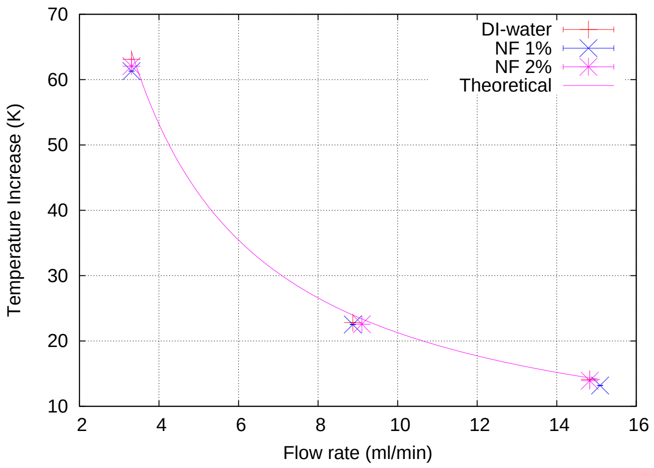


Figure 4. Temperature gain along heated tube for pure DI-water and DI-water with 20-30 nm alumina nanoparticles (1% and 2% by volume)

The second step in the validation process was the measurement of the temperature increase across the heated tube and the convection heat transfer coefficients for pure DI-water. Figure 4 shows the temperature increase as a function of volumetric flow rate for DI-water and the 1% and 2% volume loaded nanofluids. The temperature increase of the DI-water agreed well with that predicted from an energy balance [12]

$$\Delta T = \frac{Q}{\rho c_p \dot{V}}. \tag{2}$$

Good agreement between the data and model shows that the assumption of a constant heat flux boundary condition is valid.

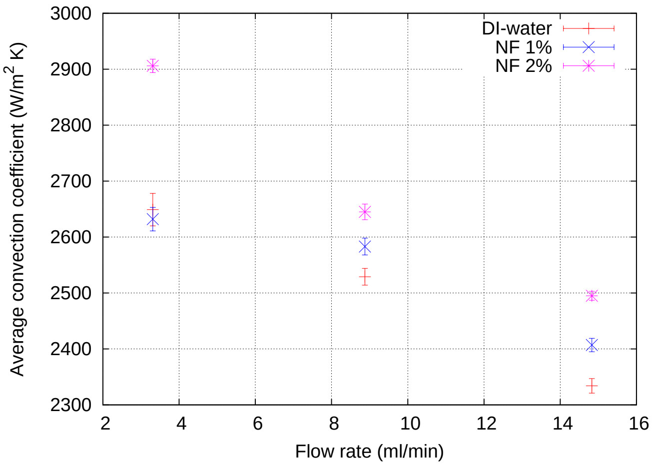


Figure 5. Average convection heat transfer coefficients for DI-water and the 1% and 2% volume loaded nanofluids

There was no difference observed in the temperature increase between the DI-water and the two nanofluids.

The convection coefficients were also calculated using the temperature measurements along the heated tube. Figure 5 shows the average convection heat transfer coefficients as a function of the flow rate for the DI-water and the 1% and 2% volume loaded nanofluids. The convection coefficients are greater in the nanofluids by as much as 2.5%. The enhancement of the average convection coefficient was of the same order as that calculated by Lai et al. [10]. A difference in the current result when compared to the results from Lai et al. [10] is the average convection coefficients dependence on flow rate. Lai et al. observed an increasing average convection coefficient with increasing flow rate while the current results show the opposite. The Nusselt number calculated here is close to the laminar value, but the trend of the average convection coefficient matches the Dittus-Boelter correlation for turbulent flow [12].

The average convection coefficients are calculated by averaging the local convection coefficients obtained from the measurements along the tube. Figure 6 shows the local convection coefficients along the heated section of the tube for a flow rate of 3.3 ml/min. There is uncertainty in the measurement of the temperature and amount of local heating that leads to noise in the data that can not be quantified at this time. The location of the thermocouples with respect to the heater wire and the consistency in the wire wrap around the tube are two obvious items that will result in some level of uncertainty, which is why the local convection coefficients are not consistent, but fluctuate when fully developed ( 2.4 cm). This distribution is consistent across test of different flow rates with the current setup. A different setup results in the same trend of decreasing local convection coefficient in the entry region and a nearly constant local convection coefficient in the developed reason.

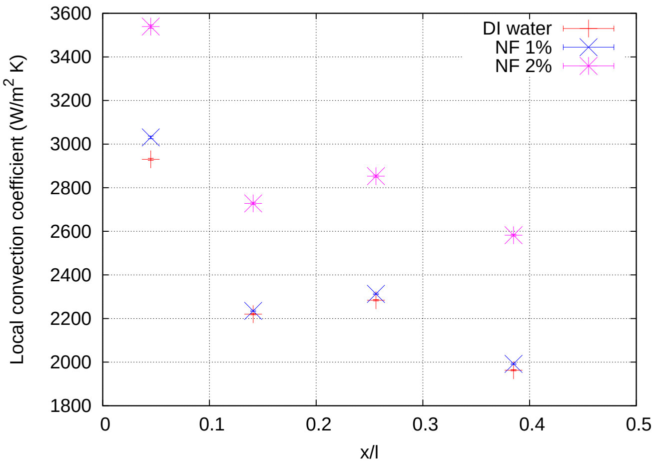


Figure 6. Local convection coefficients for DI-water and the 1% and 2% volume loaded nanofluids at a flow rate of 3.3 ml/min

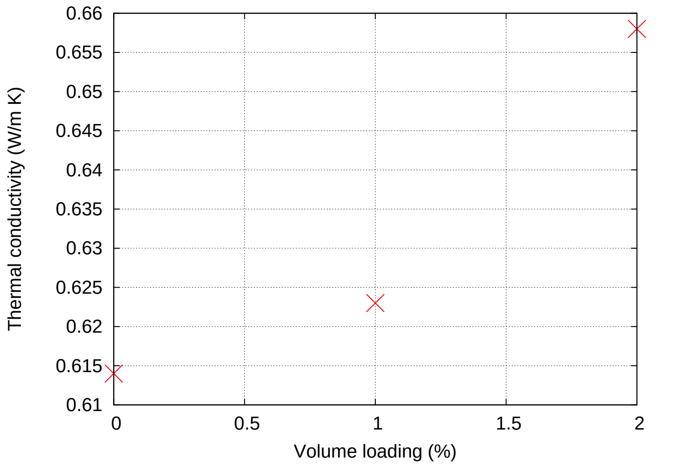


Figure 7. Calculated thermal conductivities of the nanofluids assuming $\mathrm{Nu} = 4.364$

Assuming the nanofluid maintains the same thermal conductivity which is likely not the case based on the literature [4, 5, 13–18] we calculate the Nusselt to be $\mathrm{Nu} = 5.062$ which is greater than the expected value of $\mathrm{Nu} = 4.364$ for a constant heat flux boundary condition within the laminar flow regime in the case where the flow rate is 3.3 ml/min. The Nusselt number was calculated using

$$\mathrm{Nu} = \frac{\mathrm{hd_i}}{\mathrm{k}}. \tag{3}$$

If we assume that the Nusselt number is 4.364 we can then calculate a thermal conductivity from the relationship in equation 3. Figure 7 shows the calculated thermal conductivity of the nanofluids based on the average of their convection coefficients. This calculation shows an enhancement in the thermal conductivity of 7.2% in the 2% volume loaded case.

At first glance one would assume that an enhancement in the convection coefficient would lead to an enhancement in the temperature increase across the heated section. Presumably if the convection coefficient is greater in the case of the nanofluids then the energy or heat flux into the fluid is also increased when under the same flow conditions. If this is the case, one would expect to see an increase in the temperature along the heated tube unless a change in another property (heat capacity) also occurred. If the heat capacity of the nanofluids is enhanced by roughly the same amount has the convection coefficient ( 2.5%) no difference in the temperature increase across the heated tube will be observed. Since the volume ratio of the solution is known we can calculate the heat capacity from a weighted ratio of the two components (water and nanoparticles) given by

$$(\rho c_p)_{nf} = n(\rho c_p)_{np} + (1-n)(\rho c_p)_w \tag{4}$$

where $n$ is the volume loading of the nanofluid. The impact of the addition of nanoparticles actually reduces the heat capacity (due to the lower heat capacity of the nanoparticles) of the fluid which should result in an increased temperature gain across the tube. This increase in the temperature gain across the tube is calculated to be a maximum of 0.14 degrees for the 2% volume loading case. This increase in temperature is within the noise of the measurements of the temperature gain along the heated tube.

Figure 8 shows the temperature increase through the commercial water block for pure DI-water and the DI-based alumina nanofluid with a 1% volume loading. Similar to the single straight tube there is no distinguishable difference in the temperature increase across the heated section between the pure DI-water and the nanofluid. We were unable to measure the convection coefficients with the current setup (Figure 2) because the commercial water block does not have an ideal location for measuring surface temperatures, the geometry of the block's channels is unknown and the temperature through the block is know accurately know as the only measurement made were the inlet and outlet temperatures; therefore the enhancement in heat transfer that occurs in nanofluids is difficult to determine when used in a commercially available cooling system.

## CONCLUSIONS

An enhancement of up to 2.5% in the convective heat transfer coefficients was observed in the straight heated tube in the nanofluids over that of the DI-water, but no noticeable difference in the temperature gain along the heated tube or in the commercial system was measured between the DI-water and the nanofluids. If no difference in temperature gain is observed one typically assumes that heat capacity of the fluid is not affected with the addition of nanoparticles. We have shown that the heat capacity is

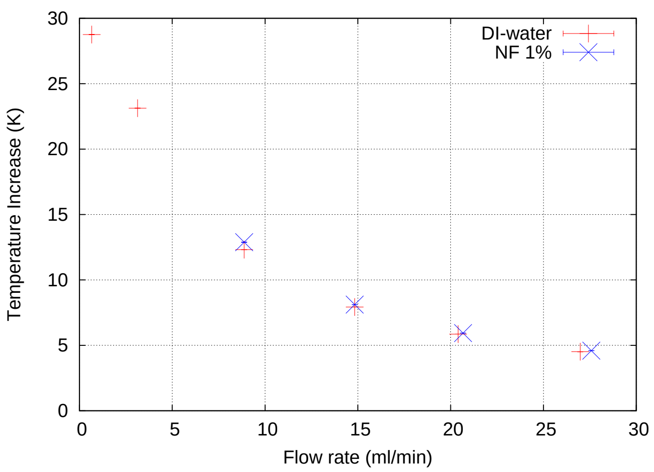


Figure 8. Temperature gain through the commercial water block for pure DI-water and DI-water with 20-30 nm alumina nanoparticles (1% by volume)

in fact reduced because of the addition of nanoparticles and that the only explanation for no observed different in the temperature gain is due to a reduction in the temperature gradient at the surface of the tube. The increase in the convective heat transfer coefficient could also result in an increase in the temperature gain across the tube, but this would require a greater amount of heat being absorbed by the fluid (less loss to the environment). An increase in the heat absorbed by the fluid in this case could only be a result of a more efficient heat absorbing fluid, but would still result in an increase of 2.5% (maximum of 0.75 degrees) which could also be within the uncertainty of the measurements. The current work was unable to determine whether nanofluids are effective in enhancing thermal transport in existing commercial liquid electronics cooling systems. Future work will include more detailed analysis of the impact of nanofluids in applications that require heat transfer fluids. Specifically a more involved study of liquid electronics cooling systems will be performed to learn what level of enhancement can be expected and how enhancement can be optimized using nanofluids.

## Acknowledgments

The authors would like to thank Oak Ridge National Laboratory for funding this project.

## REFERENCES

[1] Maxwell, J. C., 1881. *A Treatise on ELectricity and Magnetism*, 2 ed., Vol. 1. Clarendon Press, Oxford, UK.

[2] Choi, S. U. S., 1995. “Enhancing thermal conductivity of fluids with nanoparticles”. In *Developments and Applications of Non-Newtonian Fluids*, D. A. Siginer and H. P. Wang, eds. American Society of Mechanical Engineers, New York, pp. 99–105.

[3] Hamilton, R., and Crosser, O., 1962. “Thermal conductivity of heterogeneous two-component systems”. *I and EC Fundamentals*, **1**(3), pp. 187–191.

[4] Lee, S., Choi, S. U. S., Li, S., and Eastman, J. A., 1999. “Measuring thermal conductivity of fluids containing oxide nanoparticles”. *Journal of Heat Transfer*, **121**(2), pp. 280–289.

[5] Xuan, Y., and Li, Q., 2000. “Heat transfer enhancement of nanofluids”. *IJHFF*, **21**, pp. 58–64.

[6] Keblinski, P., Phillpot, S. R., Choi, S. U. S., and Eastman, J. A., 2002. “Mechanism of heat flow in suspensions of nano-sized particles (nanofluids)”. *International Journal of Heat and Mass Transfer*, **45**, pp. 855–863.

[7] Prasher, R., 2005. “Thermal conductivity of nanoscale colloidal solutions (nanofluids)”. *PRL*, **94**(025901), jan.

[8] Xuan, Y., and Li, Q., 2003. “Investigation on convective heat transfer and flow features of nanofluids”. *JHT*, **125**, pp. 151–155.

[9] Li, Q., Xuan, Y., Jiang, J., and Xu, J., 2005. “Experimental investigation on flow and convective heat transfer feature of a nanofluid for aerospace thermal management”. *Journal of Astronautics*, **26**, pp. 391–394.

[10] Lai, W., Phelan, P., Vinod, S., and Prasher, R., 2008. “Convective heat transfer for water-based alumina nanofluids in a single 1.02-mm tube”. In 11th Intersociety Conference on Thermal and Thermomechanical Phenomena in Electronic Systems.

[11] Buongiorno, J., 2006. “Convective transport in nanofluids”. *JHT*, **128**, mar.

[12] Incropera, F. P., and DeWitt, D. P., 2002. *Introduction to Heat Transfer*, 4 ed. John Wiley and Sons.

[13] Wang, X., and Xu, X., 1999. “Thermal conductivity of nanoparticle-fluid mixture”. *Journal of Thermophysics and Heat Transfer*, **13**(4), pp. 474–480.

[14] Eastman, J. A., Choi, S. U. S., Li, S., Yu, W., and Thompson, L. J., 2001. “Anomalously increased effective thermal conductivities of ethylene glycol-based nanofluids containing copper nanoparticles”. *Applied Physics Letters*, **78**(6), pp. 718–720.

[15] Patel, H., Das, S., Sundararajan, T., Nair, A. S., George, B., and Pradeep, T., 2003. “Thermal conductivities of naked and monolayer protected metal nanoparticle based nanofluids: Manifestation of anomalous enhancement and chemical effects”. *Applied Physics Letters*, **83**(14), oct, pp. 2931–2933.

[16] Das, S., Putra, N., Thiesen, P., and Roetzel, W., 2003. “Temperature dependence of thermal conductivity enhancement for nanofluids”. *Journal of Heat Transfer*, **125**, August, pp. 567–574.

[17] Jwo, C., Teng, T., Hung, C., and Guo, Y., 2005. “Research

and development of measurement device for thermal conductivity of nanofluids". In Journal of Physics: Conference Series, Vol. 13, pp. 55–58.

[18] Zhu, D., Li, X., Wang, N., Wang, X., Gao, J., and Li, H., 2009. "Dispersion behaviour and thermal conductivity characteristics of Al2O3-H2O nanofluids". *Current Applied Physics*, **9**, pp. 131–139.