Proceedings of the 3rd IASME/WSEAS Int. Conf. on HEAT TRANSFER, THERMAL ENGINEERING AND ENVIRONMENT, Corfu, Greece, August 20-22, 2005 (pp160-165)

Nanofluids Heat Transfer Performance for Cooling of High Heat Output Microprocessor

C. T. NGUYEN, G. ROY, P.-R. LAJOIE, S. E. B. MAÏGA
Department of Mechanical Engineering, Faculty of Engineering
Université de Moncton
Moncton, New Brunswick
CANADA E1A 3E9
http://www.umoncton.ca

Abstract: - The heat transfer performance of two nanofluids, namely water-γAl_2O_3 and Ethylene Glycol-γAl_2O_3, for cooling of high heat output microprocessors has been numerically investigated. Results have shown that with an increase of particle volume concentration and/or the flow Reynolds number, the heat transfer coefficient has considerably augmented, which results in a significant decrease of the maximum junction temperature. The combination of such use of nanofluids and the turbulent flow regime has been found particular advantageous as it can give very low level of junction temperature.

Key-Words: - Electronics cooling, Forced convection, Heat transfer augmentation, Heat transfer enhancement, Nanofluids, Nanoparticles, Numerical simulations.

1 Introduction

Thermal properties of heating or cooling fluids play a major role in the development of energy-efficient heat transfer equipments. Conventional heat transfer fluids such as water, ethylene glycol and engine oils have, in general, poorer heat transfer properties than most solids, in particular metals. Despite considerable research, major improvements in heat transfer capabilities, especially for cooling of high heat output electronics devices, have suffered a certain lacking [1]. As result, a clear need does still persist today to develop new strategies improving heat transfer behaviours of conventional fluids. Recent developments in nanotechnology have produced a new and very special class of fluids, called 'nanofluids', which appear highly potential [2]. The term nanofluid refers to a two-phase mixture constituted of a continuous liquid phase and very fine metallic particles called 'nanoparticles' [3]. These mixtures seem to possess thermal properties that appear well higher than those of the base fluid; in fact, some scarce experimental data have shown that even for low particle concentration, say from 1 to 5% in volume, the effective thermal conductivity of nanofluid can be increased by almost 20% over that of base fluid, see in particular [4]. Recent experimental work and numerical investigations [5-8] have confirmed the heat transfer enhancement of nanofluids in some confined flow situations as well. Such an enhancement can be attributed, in part, to the corresponding augmentation of nanofluids thermal properties. On the other hand, due to the random movement of particles, another interesting phenomenon, called 'the thermal dispersion', takes place within the mixture, which is believed to have a key role in the increase of energy exchange [9]. The nanofluids appear clearly a very interesting alternative for heat transfer applications, in particular for electronics cooling and micro-heat transfer applications [1, 2].

In the present work, we are interested to numerically investigate the enhancement effects due the use of such nanofluids in cooling of high heat output microprocessors.

2 Mathematical Formulation

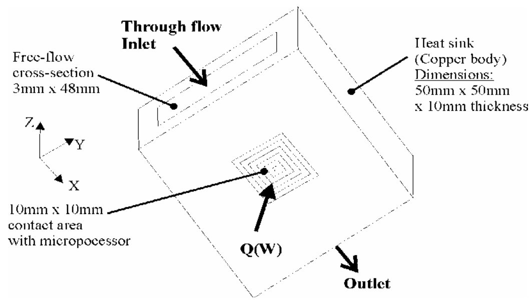

Fig. 1 Geometry of the heat sink considered

The problem consists of a forced convection flow of a nanofluid circulating through a heat sink installed on top of a high heat output microprocessor. The heat sink under consideration is of an all-copper rectangular slot-type having as overall dimensions, 50mm X 50mm X 10mm thickness with a free flow cross-section area of 3mm X 48mm, Figure 1. The flow is assumed to be steady and laminar, and the fluid enters with uniform profiles of axial velocity V_0 and temperature T_0.

2.1 Assumptions

Because of the extremely reduced size of particles, it was suggested that they may be easily fluidized. Furthermore, by assuming thermal equilibrium and negligible inter-phases slip, the liquid-solid particle mixture may reasonably be considered as a conventional single-phase fluid having properties that are function of those of constituents as well as of their respective concentrations [5,9]. Under such assumption, one may then apply the classical theory developed for conventional single-phase fluids to nanofluids. In particular, the conservation equations (mass, momentum and energy) can consequently be extended and employed for nanofluids as well. It is very interesting to mention that such a major hypothesis has been validated, to some extent and although indirectly, by some experimental works [5,6], in which correlations of form similar to that of the well-known Dittus and Boelter formula was obtained to characterize the turbulent heat transfer of some nanofluids within a tube.

2.2 Governing equations and boundary conditions

The governing equations are as follows for the fluid zone [10]:

$$\text{div}\,(\rho \mathbf{V}) = 0 \quad (1)$$
$$\text{div}\,(\rho \mathbf{V}\mathbf{V}) = -\,\text{grad}\,P + \mu \nabla^2 \mathbf{V} \quad (2)$$
$$\text{div}\,(\rho \mathbf{V} C_p T) = \text{div}\,(k\ \text{grad}\,T) \quad (3)$$

and for the solid zone that is the heat sink body:

$$\text{div}\,(k_s\ \text{grad}\,T) = 0 \quad (4)$$

where **V**, P and T are, respectively the fluid velocity vector, pressure and temperature; k_s is the thermal conductivity of solid; all fluid properties are evaluated at the reference temperature T_0.

In the present study, we have also attempted to investigate heat transfer and fluid flow under turbulent flow regime using the water-γAl_2O_3 mixture. The well-known and classical κ-ε turbulent model proposed by Launder and Spalding [11, 12] has been adopted for modelling the turbulence. Such a model appears appropriate for use with nanofluid flows in some confined situations [7].

As boundary conditions, we consider uniform profiles of axial velocity V_0 and temperature T_0 at the inlet. On the heat sink exterior surface, convective heat losses towards 20˚C ambient air are imposed with heat transfer coefficient fixed to be $10 W/m^2K$, except for a 10mm X 10mm contact area, Fig. 1, through which a total heat Q is imposed (Q represents in fact the microprocessor heat output). Inside the flow channel, the usual no-slip conditions prevail on all liquid/solid interfaces. Finally, at the outlet section, conditions approximating the fully-developed flow conditions are assumed.

2.3 Thermal properties of nanofluids

Assuming that Al_2O_3 nanoparticles (ρ_p= 3880kgm^{-3} and C_p= 773 Jkg^{-1}K^{-1}) are well mixed within the base fluids i.e. their concentration may be considered uniform throughout the domain, the effective thermal properties of nanofluids can be evaluated using formulas developed from the classical theory of two-phases mixtures [5,9]. The following formulas were employed in this study (indices '*nf*', '*bf*', '*p*' et '*r*' refer respectively to nanofluid, base fluid, particles and 'nanofluid-to-base fluid' ratio) :

$$\rho_{nf} = (1-\varphi)\rho_{fb} + \varphi\rho_p \quad (5)$$
$$(C_p)_{nf} = (1-\varphi)(C_p)_{fb} + \varphi(C_p)_p \quad (6)$$

$$\mu_r = \frac{\mu_{nf}}{\mu_{fb}} = 123\varphi^2 + 7.3\varphi + 1 \quad (7)$$

$$\mu_r = \frac{\mu_{nf}}{\mu_{fb}} = 306\varphi^2 - 0.19\varphi + 1 \quad (8)$$

$$k_r = \frac{k_{nf}}{k_{fb}} = 4.97\varphi^2 + 2.72\varphi + 1 \quad (9)$$

$$k_r = \frac{k_{nf}}{k_{fb}} = 28.905\varphi^2 + 2.8273\varphi + 1 \quad (10)$$

It is important to mention that for the nanofluids studied, namely water-γAl_2O_3 and Ethylene Glycol-γAl_2O_3, their dynamic viscosities are computed using empirical correlations, Equations (7, 8), which were obtained by curve fitting of some scarce experimental data available in the literature (details regarding the evaluation of the nanofluid properties have been discussed elsewhere, see [7]).

Proceedings of the 3rd IASME/WSEAS Int. Conf. on HEAT TRANSFER, THERMAL ENGINEERING AND ENVIRONMENT, Corfu, Greece, August 20-22, 2005 (pp160-165)

One can determine that the problem considered can be characterized by a set of dimensionless parameters, namely the flow Reynolds number $Re=V_0 D_h \rho/\mu$, the fluid Prandtl number $Pr=C_p\mu/k$, the particle volume concentration φ, the ratios k_p/k_{bf} and $(\rho C_p)_p/(\rho C_p)_{bf}$ and the aspect ratios L/D_h where L is the overall length of the channel.

2.4 Numerical method

The set of governing equations (1-4) has been successfully solved by employing a 'finite control volume' numerical method where the exponential scheme was used throughout to compute the convection-and-diffusion combined fluxes of heat and momentum [13, 14]. In order to ensure that results are independent regardless the number of grid points used, several grids were extensively tested. The 50 x 44 x 46 non-uniform grid – with respectively 50, 44 and 46 nodes along the spatial directions X, Y and Z (see again Fig. 1) – has been found quite appropriate for the problem under study. This grid possesses nodes that are highly packed in the vicinity of all solid/liquid interfaces, in the entrance region and also near the exit of the heat sink (by symmetry conditions, we consider only half of the physical domain).

As starting conditions, we have employed velocity and temperature fields that are corresponding to the cases without particles i.e. $\varphi=0$; and for subsequent simulations, a converged solution corresponding to a particular particle concentration was used in order to marching to a higher value of φ. For all the cases performed in this study, converged solutions were usually achieved when reaching a residue as low as 10^{-8} for all the conservation equations. The numerical simulations have been carried out for the two nanofluids mentioned earlier and covered several values of Q from 50W to 150W and the flow Reynolds number varied from 10 to 4000 (Re =4000 corresponds in fact to the turbulent flow regime).

3 Results and Discussion

3.1 Heat transfer of nanofluids

Results as obtained from our numerical simulations have eloquently shown that the inclusion of the nanoparticles within the base liquids has produced a tremendous augmentation of the convective heat transfer coefficient and as consequence, has greatly contributed to a more efficient cooling of the microprocessor itself. In fact, in term of T_{max}, which is the maximum temperature that may be reached at the junction between the microprocessor and its heat sink, it has been observed that T_{max} has considerably decreased with an increase of the particle concentration, thus indicating obviously a clear improvement of the cooling rate as given by the heat sink. For example, for the particular case with water-γAl_2O_3 mixture, Q =150W and Re =2000, T_{max} has decreased from 65.8ºC for φ=0 to 58.4ºC for φ=7.5%, Table 1; while the corresponding enhancement of the surface-averaged convective heat transfer coefficient h_{av} has been estimated to be 53% between these two cases, Fig. 2. It has been observed, however, that for lower levels of the microprocessor heat output, the corresponding decrease of T_{max} appears less pronounced, only from 29.4ºC to 26ºC for cases with Q=50W and φ augmenting from 0 to 7.5%. In general, the junction maximum temperature T_{max} has been found to increase considerably with the augmentation of the microprocessor heat output, given identical operating conditions of the heat sink. Such a behavior appears physically quite realistic.

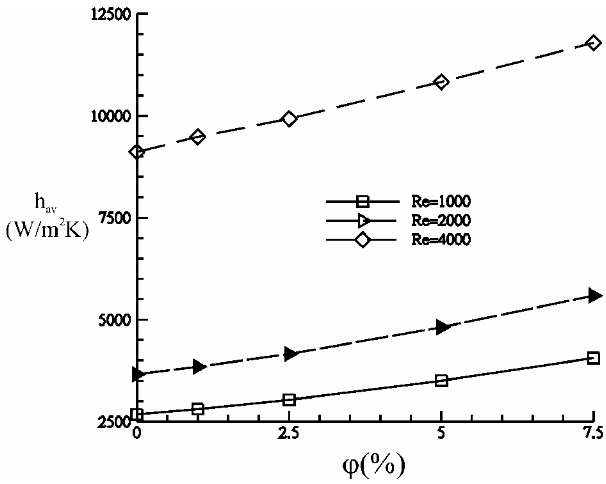

Fig.2 Effects of parameters φ and Re on h_{av} for water-γAl_2O_3 and Q = 150W

	Water-γAl₂O₃			Ethylene Glycol-γAl₂O₃	
	Re=1000	Re=2000	Re=4000	Re=10.5	Re=21
Φ = 0	71.7	65.8	51.4	91.9	82.9
φ= 1	70.7	64.9	50.7	90.9	82
φ= 2.5	69.1	63.5	49.9	87.5	79.2
φ= 5	66.3	60.9	48.5	80.4	74.2
φ= 7.5	63.6	58.4	47.1	73.4	67

Table 1. Effects of parameters φ and Re on T_{max} for nanofluids studied and Q= 150W

Similar behaviours with regard to the advantageous effects due to the use of nanofluids have also been found for other flow Reynolds numbers Re and levels of heat output Q considered in this study. Figures 3 and 4 show, in particular, the variation of T_{max} as function of both parameters Q and φ respectively for Re= 1000 and Re =2000.

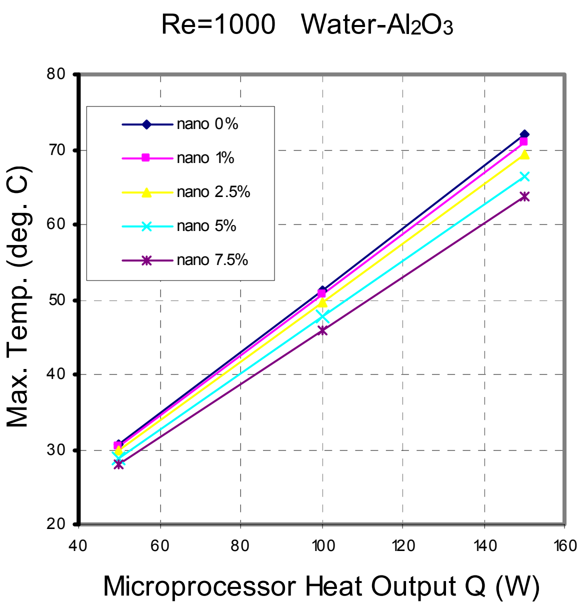

Fig. 3 Effects of φ and Q on T_{max} (Re = 1000)

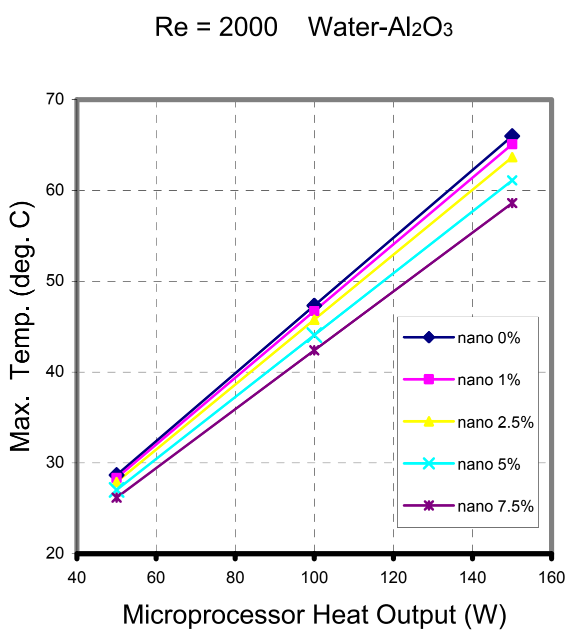

Fig. 4 Effects of φ and Q on T_{max} (Re = 2000)

We observe, here again, that the junction maximum temperature T_{max} decreases appreciably with an augmentation of the particle volume concentration. For a given value of parameter φ, T_{max} increases indeed with an augmentation of the microprocessor heat output Q. Also, as one may expect, the fact of increasing the flow Reynolds number (i.e. the mass flow rate) has produced a greater enhancement of heat transfer within the heat sink and consequently, a more important decrease of T_{max}. Figure 5 shows, in particular, the variation of the junction maximum temperature T_{max} as function of the parameters Q and φ for various cases using water-γAl_2O_3 and Re=4000 (turbulent flow regime). It is very interesting to see that such a combination of turbulent flow and nanofluid can provide quite high heat transfer rates. Thus, we can observe that a value of T_{max} as low as 48.5ºC has been obtained with a particle volume concentration of 5% and this, for a relatively high heat output level, say Q =150W (see again Fig. 2 and Table 1).

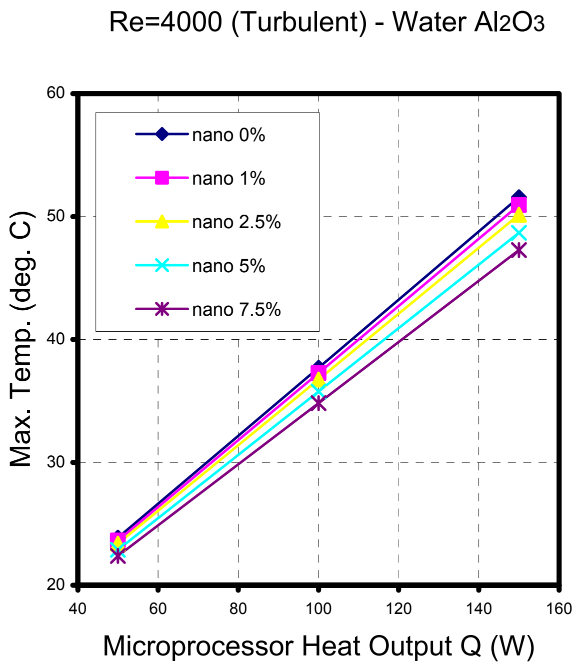

Fig. 5 Effects of φ and Q on T_{max} (Re = 4000)

Numerical results have also shown similar behaviours and trends regarding the heat transfer enhancement for the case of Ethylene Glycol-γAl_2O_3 mixture, see again Table 1. Thus, for the case Q=150W and Re=21 in particular, the maximum junction temperature T_{max} has decreased from 82.9ºC to nearly 67ºC for φ increasing from 0% to 7.5%; the corresponding decrease of T_{max} appears less pronounced for lower flow Reynolds numbers, as it has been previously observed for water-γAl_2O_3.

3.2 Wall Friction of Nanofluids

The heat transfer enhancement provided by the utilisation of nanofluids as shown and discussed previously is, unfortunately, accompanied by a drastic influence on the wall shear stress. Figure 6 shows, in particular, the results for the ratio τ/τ_0 - the 'nanofluid-to-base fluid' ratio of the surface-averaged wall shear stresses as obtained for the nanofluids studied. We can obviously observe that the inclusion of nanoparticles within base fluids has produced drastic augmentation of the wall friction. In general, the wall shear stress considerably increases with an increase of particle concentration. For the case of water-γAl_2O_3 and Re=1000 for example, the ratio τ/τ_0 has as values, 1.2, 1.5, 2.4 and 4.15 respectively for φ=1%, 2.5%, 5% and 7.5%; this indicates that for φ =7.5%, one would expect an increase of the wall friction of a factor more than four times of that of saturated water. For the case of Ethylene Glycol-γAl_2O_3 mixture, such an increase appears to be even more drastic. Such behaviour regarding the drastic influence of nanoparticles on the wall shear stress can be explained by the fact that with the presence of these particles, the apparent viscosity of the resulting mixture has considerably increased [7, 8] with respect to that of the base fluid.

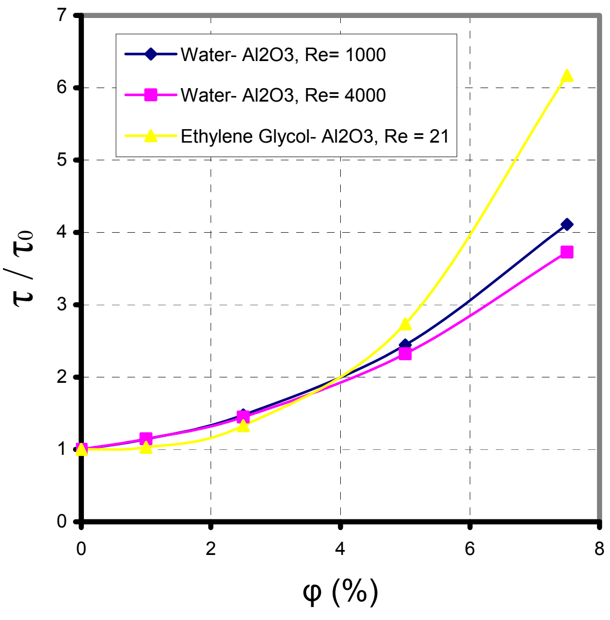

Fig. 6 Ratio τ/τ_0 as function of φ and Re

4 Conclusion

In this paper, we have numerically investigated the performance of a rectangular slot-type heat sink using two specific nanofluids, namely water-γAl_2O_3 and Ethylene Glycol-γAl_2O_3 mixtures, for cooling of high heat output microprocessors. Results have clearly shown that the use of these nanofluids appears particularly advantageous. Thus, the heat transfer coefficient has been found to increase considerably with an augmentation of the particle volume concentration as well as of the flow Reynolds number, which has consequently produced a greater decrease of the junction temperature between the microprocessor and its heat sink. It is also observed that the use of nanofluid under turbulent flow regime can provide interestingly high heat transfer rates. The presence of nanoparticles within the base fluids has, unfortunately, a rather drastic influence on the wall friction. In general, it is expected that the wall shear stress would increase appreciably with an augmentation of the particle volume concentration.

Acknowledgement

The authors wish to sincerely thank the '*Natural Sciences and Engineering Research Council of Canada*' and the Faculty of Graduated Studies and Research of the '*Université de Moncton'* for their financial support to the present research project as well as for the summer research grant that one of the authors, P.-R. Lajoie, has received. Thanks are also due to the Faculty of Engineering for allocating computing facilities.

References:

[1] K. Azar, Cooling technology options, parts 1 & 2, *Electronics Cooling*, Vol. 9, No. 3, 2003, pp. 10-14 & Vol. 9, No. 4, 2003, pp. 30-36.

[2] S. Lee, S. U.-S. Choi, Application of metallic nanoparticle suspensions in advanced cooling systems, ASME Publications PVP-Vol. 342/MD-Vol. 72, 1996, pp. 227-234.

[3] Nanophase Technologies, Romeoville, IL, USA, http://www.nanophase.com

[4] X. Wang, X. Xu, S. U. S. Choi, Thermal conductivity of nanoparticles-fluid mixture, *J. of Thermophysics and Heat Transfer*, Vol. 13, No. 4, 1999, pp. 474-480.

[5] B. C. Pak, Y. I. Cho, Hydrodynamic and heat transfer study of dispersed fluids with submicron metallic oxide particles, *Experimental Heat Transfer*, Vol. 11, No. 2, 1998, pp. 151-170.

[6] Q. Li, Y. Xuan, Convective heat transfer performances of fluids with nano-particles, *Proc. 12th Int. Heat Transfer Conference* (Grenoble, France, August 2002), pp. 483-488.

[7] S. E. B. Maïga, C. T. Nguyen, N. Galanis, G. Roy, Heat transfer behaviors of nanofluids in uniformly heated tube, *Superlattices and Microstructures*, Vol. 35, No. 3-6, 2004, pp. 543-557.

[8] G. Roy, C. T. Nguyen, P.-R. Lajoie, Numerical investigation of laminar flow and heat transfer in a radial flow cooling system with the use of nanofluids, *Superlattices and Microstructures*, Vol. 35, No. 3-6, 2004, pp. 497-511.

[9] Y. Xuan, W. Roetzel, Conception for heat transfer correlation of nanofluid, *Int. J. Heat Mass Transfer*, Vol. 43, 2000, pp. 3701-3707.

[10] Z. U. A. Warsi, *Fluid Dynamics Theoretical and Computational Approach,* 2nd Ed., CRC Press, Boca Raton, Florida, USA, 1999.

[11] B. E. Launder, D. B. Spalding, The numerical computation of turbulent flows, *Comp. Meth. Appl. Mech. Eng.*, Vol. 3, 1974, pp. 269-289.

[12] D. A. Anderson, J. C. Tannehill, R. H. Pletcher, *Computational Fluid Mechanics and Heat Transfer*, Hemisphere Pub. Co., New York, USA, 1984.

[13] S. V. Patankar, *Numerical Heat Transfer and Fluid Flow,* McGraw-Hill Book Co., New York, NY, USA, 1980.

[14] *Fluent 6 User's Guide*, Fluent Inc., NH, USA, 2002.