Numerical Investigation on the Role of Al2O3 Nanoparticle Diameter on Convective Cooling Enhancement of a Rectangular Fin

Mahdad Seyedi

MSc student, Tarbiat Modares University, Tehran, Iran
Mahdad.seyedi@modares.ac.ir

Mohammad Mahdi Heyhat

Assistant Professor of Mechanical Engineering, Tarbiat Modares University, Tehran, Iran
mmheyhat@modares.ac.ir

Majid Siavashi

Associate Professor of Mechanical Engineering, Iran University of Science and technology, Tehran, Iran
msiavashi@iust.ac.ir

Abstract

In this work, laminar flow forced convective heat transfer of Al_2O_3/water nanofluid over a rectangular fin with constant substrate temperature was investigated numerically. An empirical correlation was used for predicting the effective thermal conductivity and dynamic viscosity of nanofluids. The heat transfer coefficients of nanofluids were obtained for different nanoparticle concentrations as well as various diameters. Three different considered nanoparticle volume concentrations were 1%, 2% and 3%, while the nanoparticle diameters were 20nm, 40nm and 80nm. The results showed that heat transfer coefficient increased by increasing the concentration of nanoparticles and by decreasing the diameter of them. Numerical results emphasize the enhancement of 18% in heat transfer coefficient by using 20nm nanoparticles compared to 80nm nanoparticles.

Keywords: Heat Transfer Enhancement, Nanofluids, Rectangular Fin, Nanoparticle Diameter

1. Introduction

In the last two decades, many efforts have been made to enhance thermal properties of heat transfer fluids, as thermal conductivity. Particular attention has been given to liquid/solid suspensions, made with traditional heat transfer fluids and nanoparticles (<100 nm) of metal or metal oxide, that Choi[1] was the first to call nanofluids. The term 'nanofluid' refers to a two-phase mixture usually composed of a continuous liquid phase and dispersed nanoparticles in suspension (i.e. extremely fine metallic or non-metallic particles of size below 100 nm). Several nanoparticle dispersions of engineering interest are readily available from various commercially sources.

Generally, the addition of nanoparticles yields an enhancement of thermal conductivity that depends on volume fraction of solid phase, particle size, material etc., as demonstrated by several researchers. Xie et al.[2] carried out a comparison between experiments and theoretical models, showing that the measured thermal conductivity in nanofluids is much higher than the values calculated using theoretical correlation, indicating new heat transport mechanisms. Nanofluids enhance also convective heat transfer coefficient, as Wen et al.[3] demonstrated with Al2O3-water nanofluid, flowing in a copper pipe with diameter of 4.5 mm, in laminar

conditions: enhancement up to 47.0% at the entrance region has been obtained with a volume fraction of 1.6%. A convective heat transfer coefficient enhancement up to 8.0% with Al2O3-water nanofluid at 0.3 vol% has been measured in a stainless tube by Hwang et al.[4]. Al2O3-water nanofluid in a stainless tube with an inner diameter of 4.57 mm has been studied by Kim et al.[5]. They obtained an increase of heat transfer coefficient of 15.0% and 20.0% under laminar and turbulent flow conditions respectively. Gillot et al.[6] applied the single-phase and two-phase micro heat sinks for cooling of power components. Qu and Mudawar[7] experimentally and numerically investigated the pressure drop and heat transfer characteristics of a single-phase micro-channel heat sink. The results obtained from the numerical prediction had reasonable agreement with the measured data.
In the present work, we have numerically investigated the heat transfer behavior of a typical liquid cooling system, by replacing the base fluid, which is water, by a nanofluid that is composed of water and Al2O3 nanoparticles for three different particle average diameters and three volume concentrations. Some significant numerical results are presented and discussed.

2. Problem Description and Governing Equations

A schematic of the computational domain is shown in Figure 1. A single vertical fin is placed on a constant temperature copper substrate. The dimensions of the fin used were (0.5 ×1×0.01) inches. The dimension of the substrate was (1×1) inches. The substrate's temperature was held constant at 333 K. Grid independence tests were carried out using different hexahedral grids for steady state at Re=2000. As Figure 2 shows, Grid with 95×80×46 (350000) hexahedral elements with local bias around the substrate and the fin for capturing the boundary layer effects appropriately was found to be appropriate and for higher element sizes, the outlet average temperature changes could be ignored.

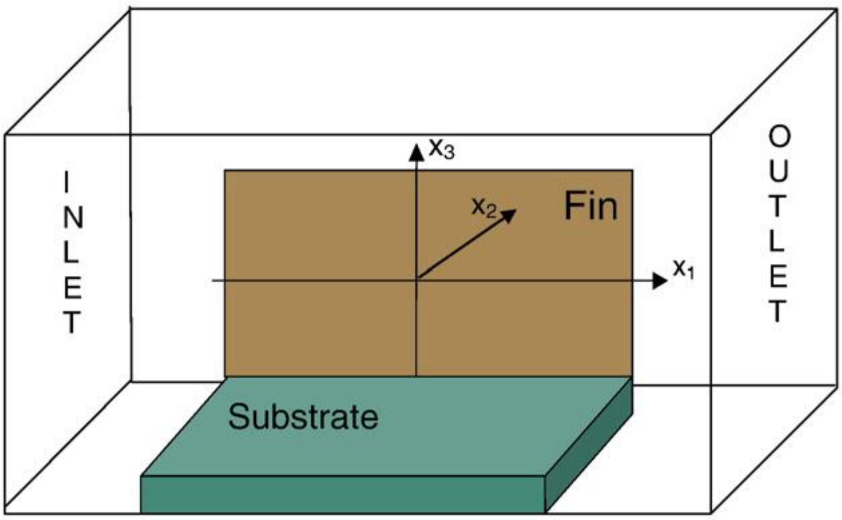

Figure 1: schematic of computational domain

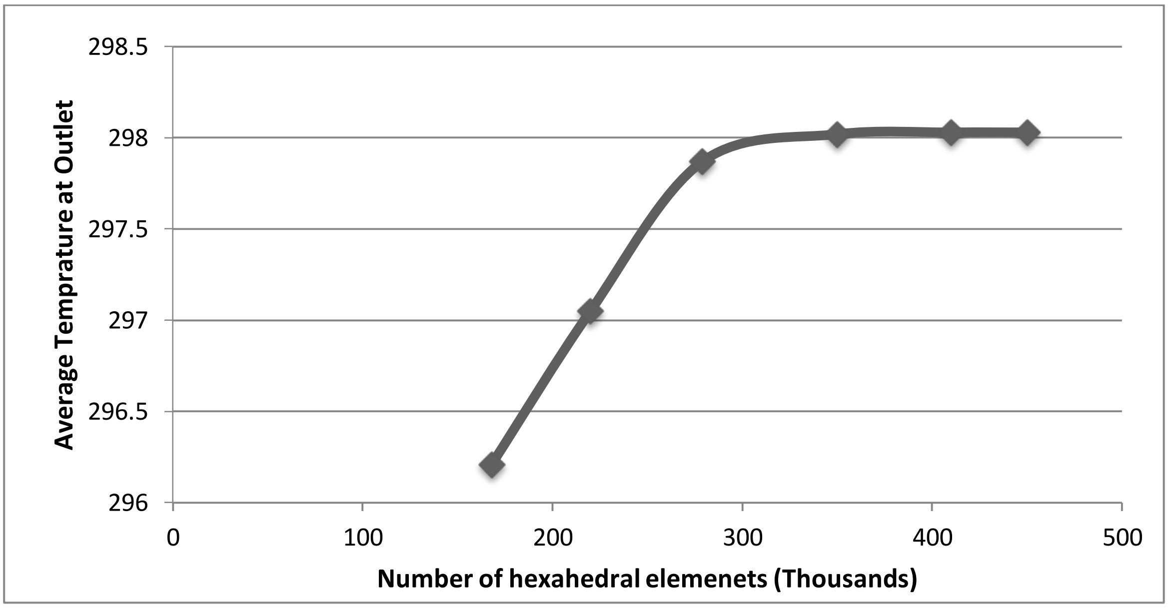

Figure 2: Average temperature at outlet for different element sizes

Flow was assumed to be incompressible and hence the continuity equation takes the form:

$$\frac{\partial u_i}{\partial x_i} = 0 \tag{1}$$

where u_i represents the velocity along the i direction while the coordinate is denoted by x_i . The balance of momentum along the i direction, assuming no body force, can be expressed as:

$$\frac{\partial(\rho_{nf} u_i)}{\partial t} + \frac{\partial(\rho_{nf} u_i u_j)}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}(\mu_{nf} \frac{\rho u_i}{\partial x_j}) \tag{2}$$

where ρ_{nf} is the density of the nanofluid, P is the pressure and μ_{nf} is the viscosity of the nanofluid. The energy conservation equation can be represented to obtain the governing differential equation for temperature (T) with negligible viscous dissipation, as:

$$\frac{\partial(\rho_{nf} C_{P_{nf}} T)}{\partial t} + \frac{\partial(\rho_{nf} C_{P_{nf}} u_j T)}{\partial x_j} = \frac{\partial}{\partial x_j}\left(k_{nf} \frac{\partial T}{\partial x_j}\right) + q^{"} \tag{3}$$

where k_{nf} and $C_{P_{nf}}$ are the thermal conductivity and the specific heat of nanofluid, respectively. According to Corcione[8] empirical correlations, the following formulas are used to calculate the thermophysical properties of the Al2O3 nanofluid.

$$\frac{k_{eff}}{k_f} = 1 + 4.4Re^{0.4}Pr^{0.66}(\frac{T}{T_{fr}})^{10}(\frac{k_p}{k_f})^{0.03}\varphi^{0.66} \tag{4}$$

$$\frac{\mu_{eff}}{\mu_f} = \frac{1}{1 - 34.87(\frac{d_p}{d_f})^{-0.3}\varphi^{1.03}} \tag{5}$$

$$C_{P_{nf}} = \frac{(1-\varphi)(\rho C_p)_f + \varphi(\rho C_P)_s}{(1-\varphi)\rho_f + \varphi\rho_s} \tag{6}$$

$$\rho_{nf} = (1-\varphi)\rho_f + \varphi\rho_s \tag{7}$$

where Re is the nanoparticle Reynolds number, Pr is the Prandtl number of the base liquid, T is the nanofluid temperature, T_{fr} is the freezing point of the base liquid, d_p is the nanoparticle diameter, k_p is the nanoparticle thermal conductivity, and φ is the volume fraction of the suspended nanoparticles. In more detail the nanoparticle Reynolds number is defined as:

$$Re = \frac{2\rho_f k_b T}{\pi \mu_f^2 d_p} \tag{8}$$

The term d_f in equation 5, is the equivalent diameter of the base fluid molecule, given by:

$$d_f = 0.1\left(\frac{6M}{N\pi\rho_{f_0}}\right)^{1/3} \tag{9}$$

where M is the molecular weight of the base fluid, N is the Avogadro number and ρ_{f_0} is the mass density of the base fluid calculated at T=293K.

Equations 1-3 are solved using SIMPLER method in ANSYS FLUENT with the single phase approach and the boundary conditions that are shown in table 1. Computations are considered to have converged when the residuals for continuity, momentum and energy are less than 1×10^{-5}, 1×10^{-5}, and 1×10^{-6}, respectively.

Table 1 : Boundary conditions

Boundary	**Boundary Conditions**
INLET	Velocity inlet
OULET	Pressure outlet
SUBSTRATE	Adiabatic
BOTTOM WALL	Slip
TOP & SIDES	Symmetry

2.1. Validation, results and discussion

In order to indicate the validity and precision of the solving method, figure 2 illustrates the temperature distribution on the fin surface with dimensionless axial position. According to Incropera[9] for a straight rectangular fin we have:

$$\frac{\theta}{\theta_b} = \frac{T - T_\infty}{T - T_b} = \frac{coshm(L-x) + (h/_{mk})sinhm(L-x)}{coshmL + (h/_{mk})sinhmL} \tag{10}$$

$$ml = \sqrt{\frac{hp}{kA_c}} \qquad (11)$$

where T_∞ is the ambient temperature, T_b is the base temperature, h is the heat transfer coefficient of the fluid, k is the thermal conductivity of the fin material and A_c is the area of the fin tip. By replacing the properties and geometrical specification of the considered case in to the equation 10, we have $ml = 2.4$.

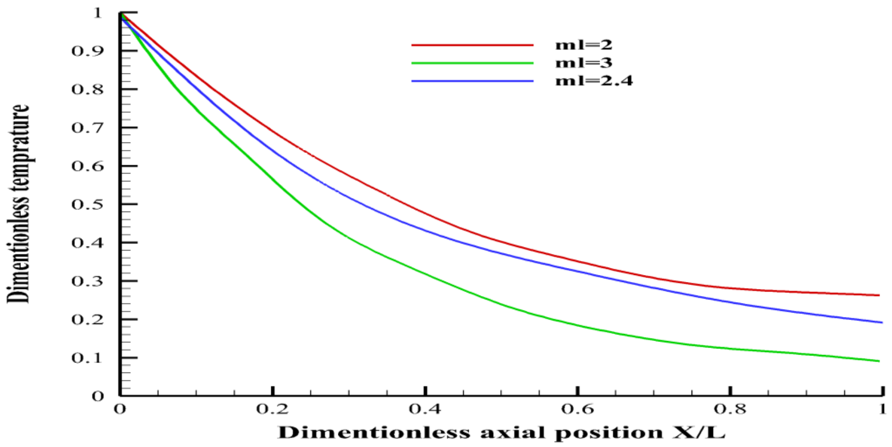

Figure 3: dimensionless temperature on the fin surface for ml=2.4

As we can see there is a good agreement between present modeling result (ml=2.4) and the analytical ones (ml=2 and ml=3). Maximum deviation from analytical approach is 3%.

The effect of AL2O3 nanoparticle volume fraction on the heat transfer coefficient is indicated in Figure 3. The diameter of nanoparticles is assumed to be 40nm. As this figure shows heat transfer coefficient improves with the increase of volume fraction. Table 3 shows the heat transfer enhancement caused by addition of nanoparticles in percent, compared to water.

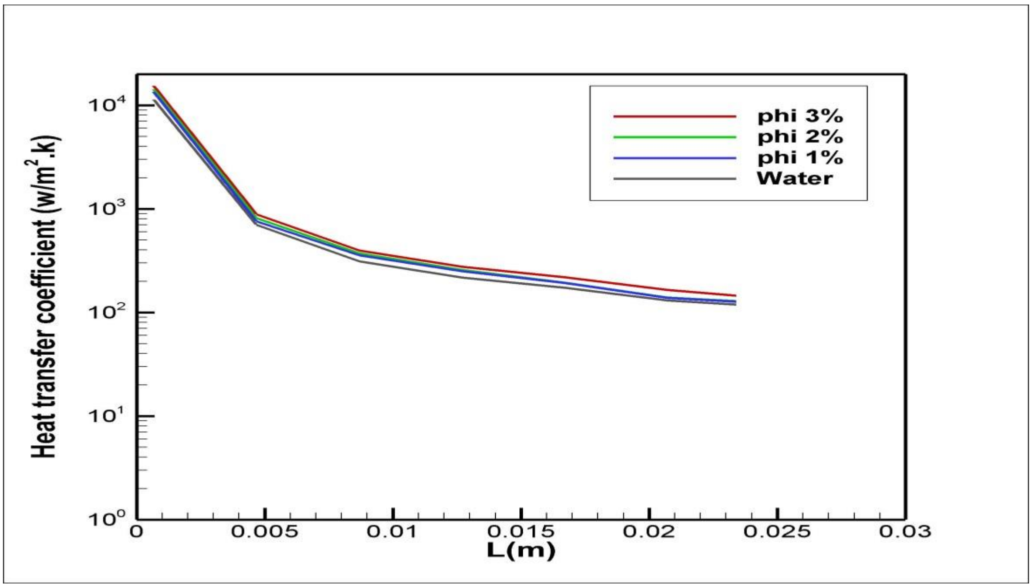

Figure 4: heat transfer coefficient for different volume fraction of Al2O3

Table 3: heat transfer coefficient enhancement compared to water

Volume concentration (%)	**Heat transfer coefficient enhancement (%)**
1	**18**
2	**25**
3	**35**

The thermal conductivity of AL2O3-water nanofluid is calculated at different volume fractions from 1 to 3% and particle sizes ranging from 20 to 80 nm. Thermal conductivity of nanofluids with various nanoparticle diameters and 1% volume concentration were captured by area weighted average properties report type of FLUENT and are shown in table 4.

Table 4: variation of thermal conductivity of AL2O3-water nanofluid obtained for 1% volume fraction

Nanoparticle size (nm)	Thermal conductivity (W/m.k)	Enhancement in thermal conductivity (%)
20	0.6482	8
40	0.6366	6.1
80	0.6285	4.7

These results indicate that the enhancement in thermal conductivities of Al2O3 nanoparticle suspension in water have decreased with increase in the particle size of nanoparticles.
Figure 4 (a-c) shows the variation of heat transfer coefficient with length for different nanoparticle diameters and volume fractions. It can be seen that with the decreasing nanoparticle diameter, the heat transfer coefficient tends to increase. In addition, it can be noted that the heat transfer coefficient improved about 18% by the decrease in volume fraction from 80nm to 20nm.

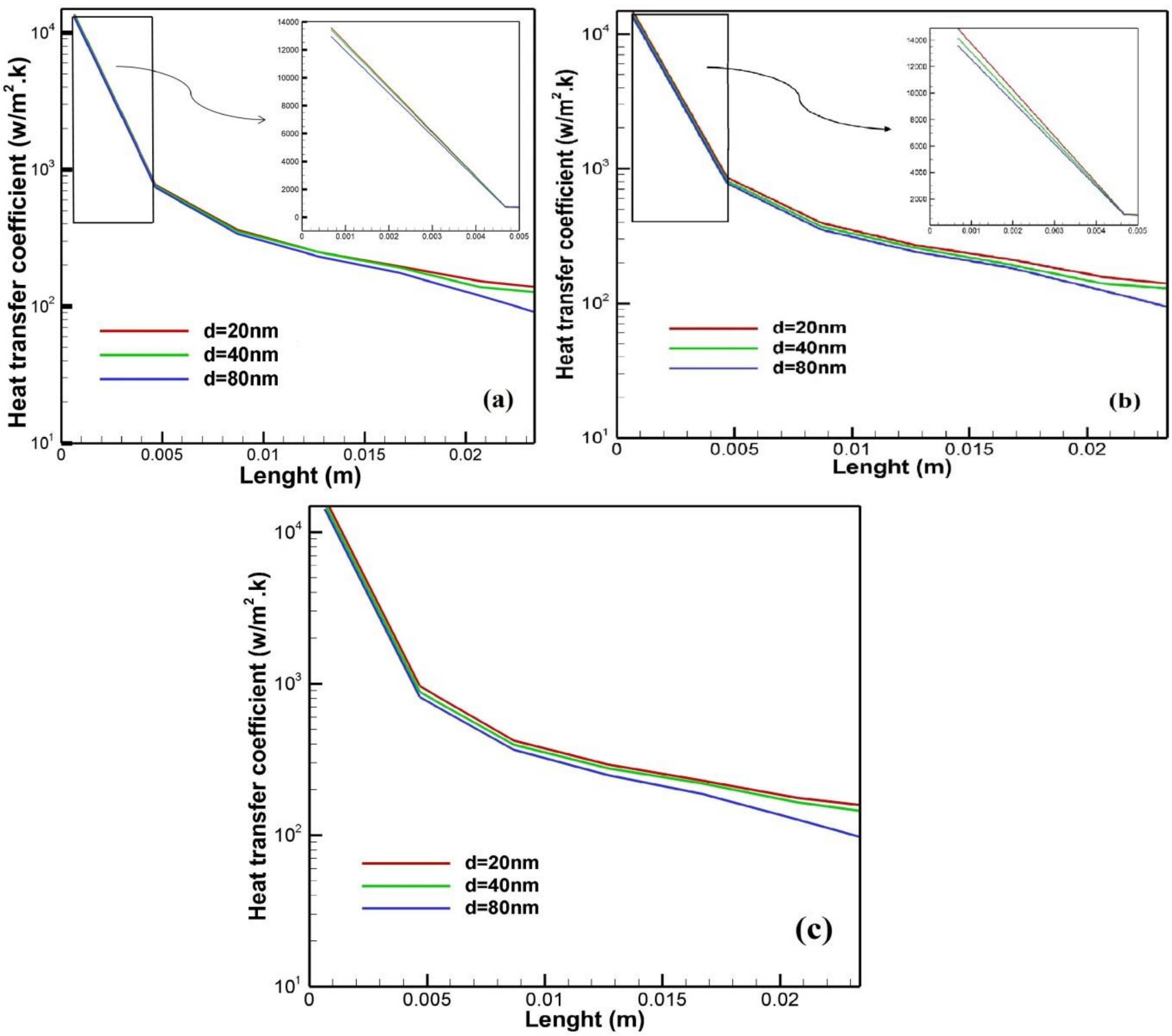

Figure 5: variations of heat transfer coefficient for different nanoparticle diameters: (a) $\boldsymbol{\varphi = 1}\%$ (b) $\boldsymbol{\varphi = 2}\%$ (c) $\boldsymbol{\varphi = 3}\%$

3. Conclusions

- Heat transfer coefficient of Al2O3-water nanofluid is directly related to volume fraction of nanofluid.
- The enhancement in thermal conductivities of Al2O3-water has decreased with increase in the particle size of nanoparticles.
- Heat transfer coefficient improved about 18% by the decrease in volume fraction from 80nm to 20nm.

References

[1]. Xuan, Y. & Li, Q. Heat transfer enhancement of nanofluids. *Int. J. Heat Fluid Flow* (2000). doi:10.1016/S0142-727X(99)00067-3

[2]. Xie, H., Wang, J., Xi, T. & Liu, Y. Thermal Conductivity of Suspensions Containing Nanosized SiC Particles. *Int. J. Thermophys.* (2002). doi:10.1023/A:1015121805842

[3]. Wen, D. & Ding, Y. Experimental investigation into convective heat transfer of nanofluids at the entrance region under laminar flow conditions. *Int. J. Heat Mass Transf.* (2004). doi:10.1016/j.ijheatmasstransfer.2004.07.012

[4]. Hwang, K. S., Jang, S. P. & Choi, S. U. S. Flow and convective heat transfer characteristics of water-based Al2O3 nanofluids in fully developed laminar flow regime. *Int. J. Heat Mass Transf.* (2009). doi:10.1016/j.ijheatmasstransfer.2008.06.032

[5]. Kim, D. *et al.* Convective heat transfer characteristics of nanofluids under laminar and turbulent flow conditions. *Curr. Appl. Phys.* (2009). doi:10.1016/j.cap.2008.12.047

[6]. Gillot, C., Bricard, A. & Schaeffer, C. Single- and two-phase heat exchangers for power electronic components. *Int. J. Therm. Sci.* (2000). doi:10.1016/S1290-0729(00)00278-7

[7]. Qu, W. & Mudawar, I. Experimental and numerical study of pressure drop and heat transfer in a single-phase micro-channel heat sink. *Int. J. Heat Mass Transf.* (2002). doi:10.1016/S0017-9310(01)00337-4

[8]. Corcione, M. Empirical correlating equations for predicting the effective thermal conductivity and dynamic viscosity of nanofluids. in *Energy Conversion and Management* (2011). doi:10.1016/j.enconman.2010.06.072

[9]. Bergman, T. L., Incropera, F. P., Lavine, A. S. & DeWitt, D. P. *Introduction to heat transfer*. (John Wiley & Sons, 2011).