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Experimental Convective Heat Transfer Studies on Graphene Nanofluid for the Cooling of Next Generation Electronic Components

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Abstract

Experimental work on convective heat transfer coefficient of graphene-water nanofluid flowing through a circular stainless steel tube of diameter 4mm and 1m in length, is subjected to a uniform wall heat flux boundary conditions on laminar regime was presented. Graphene nanosheets concentrations were varied from 0.2 to 0.6% volume in water and the Reynolds number was varied from 500 to 2000 for laminar flow regime. The dependence of heat transfer coefficient on Reynolds number, nanoparticle concentration and pressure drop were investigated. The enhancement of convective heat transfer coefficient of 56% is observed and the pressure drop of 28.7% is obtained for 0.6 volume %.

Keywords: Convective heat transfer coefficient, Graphene, Nanofluids, laminar flow**.**

INTRODUCTION

Failure of machinery due to excessive heat is one of the main concerns in engineering field. Particularly machinery used in power generation, electronic components, transportation and chemical production sectors are the major concerns. Coolants used in the machines are usually fluids. They have significantly lesser heat transfer capability than solids. Fluids are prepared over solids because of flow-ability nature rather than for its heat transfer properties. Fluids which are suited better for industrial purpose are usually not good for heat transfer applications. Researchers have been investigating to develop an efficient fluid which will be suited for both problems. Earlier research showed that suspending fine solid particles changes the thermophysical properties of the fluid [1-3] and thus improve cooling. But they experienced issues like channel clogging, poor suspension stability and wall erosion. Effect of these issues depended on the size of solid particles suspended. Recent advances in nanotechnology helped in overcoming these problems. Suspending nanosized particles give high suspension stability, improves thermal conductivity than predicted by macro sized models, less pressure drop and wall erosion [4]. These justifies that nanofluids could be the potential replacement for conventional heat transfer fluids. For this reason, more numbers of research works are being carried out in this field. Low thermal conductivity fluids can be enhanced by dispersing a solid nanoparticles with the size in order of 1–100 nm. Such fluids are termed as nanofluids. Need for an experimental investigation to find this enhancement is necessary since numerical models failed to predict the enhancement [4-5]. Various experimental heat transfer studies have been carried out on nanofluids. Most of them focus on the thermal conductivity enhancement. The convective heat transfer characteristics and viscosity of nanofluid were studied by few of the researchers. Dongsheng Wen [4] studied convective heat transfer character of γ-Al_2O_3/water nanofluid and observed a maximum of 47% enhancement in local heat transfer coefficient in laminar regime. M. Saeedinia [6] investigated base oil contains CuO nanoparticles and found that a optimum increase of 12.7% in heat transfer coefficient for 2 wt.% nanoparticles at Re 110. S.M. Fotukian [7] observed the optimal increase of 48% in heat transfer coefficient compared to pure water for volume concentration of 0.054% of Al_2O_3 nanoparticles in turbulent regime. M. Saeedinia [8] studied convective heat transfer character of CuO nanofluid flowing through wire coil inserted smooth tube under uniform heat flux, 45% increase in heat transfer coefficient was achieved. S. M. Fotukian [9] observed an average 25% increase in heat transfer coefficient for CuO/water nanofluid.

The enhancement in heat transfer properties was found to be dependent on several factors like thermal conductivity of the particle added, specific heat, and concentration of the mixture and size of the particle added. Ulzie Rea [10] studied convective heat transfer of alumina/water and zirconia/water nanofluid in laminar flow regime. Heat transfer coefficient of alumina/water nanofluid was found to be 27% higher than that of water while for Zirconia/water nanofluid the enhancement was only 3%. Decrease in specific heat than the small increase in thermal conductivity was found to be the reason for the inferior behaviour of zirconia nanofluid. K.B.Anoop [11] studied the effect of nano particle concentration on the heat transfer enhancement, results show that enhancement is higher for smaller nanoparticles since 45 nm particles exhibited higher heat transfer coefficient than that with 150 nm particles. Cong Tam Nguyen [12] predicted that, less diameter size particle dispersed in nanofluid provides higher heat transfer coefficients than with larger size diameter particles. On contradictory increasing the nanoparticles concentration did not show much effect on heat transfer enhancement for the experiments reported by S.M. Fotukian [7]. M. Hojjat [13] studied convective heat transfer coefficient and Nusselt number of nanofluids and observed that the heat transfer enhancement in nanofluids are directly proportional to the particle concentration and Peclet number. S. Zeinali Heris [14] also observed the same, the heat transfer coefficient of Al_2O_3 nanofluid increased with Peclet number as well as particle concentration. Yurong He [15] observed Al_2O_3 nanofluid with concentration of 1%, enhance the heat transfer efficiency up to 45% in comparison with pure water was reported by S.M. Peyghambarzadeh [16] and the heat transfer enhancement was found to be dependent on the amount of nanoparticle added to pure water. Fluid viscosity also contributes to the convective heat transfer as expected it is higher for nanofluid than base fluid. M. Chandrasekar [17] reported that the viscosity increase of nanofluid is considerably greater than the increase in thermal conductivity. M. Fotukian [9] observed 20% increases in pressure drop for CuO/water nanofluid while M. Saeedinia [8] observed 63% rise in pressure drop for CuO/Base oil nanofluid flowing through coiled wire inserted tube.

Experimental studies show that performance of nanofluids is significantly better than the conventional heat transfer fluids in all aspects for practical applications since the penalty in pressure drop could be considered insignificant when compared with the increase in heat transfer rate. Amirhossein Zamzamian [18] experimentally studied the usage of nanofluids in heat exchangers. In double pipe heat exchanger, the maximum heat transfer enhancement was 26% for Al_2O_3/ethylene glycol and 37% for CuO/ethylene glycol and in the plate heat exchanger, the enhancement was 38% and 49%, respectively. Cong Tam Nguyen [12] studied that for 6.8% particle volume concentration of CuO nanofluid, the heat transfer coefficient increase as much as 40% compared to that of the base fluid. A.T.Utomo et al [19] studied the heat transfer coefficients of identical alumina, titania and CNT nanofluids. Nanoparticle to liquids only enhances the heat transfer coefficients by not more than 10% at equal mass flow rate or velocity. H. Akhavan-Zanjani et al.[20] investigated the addition of upto 0.02% of concentration of graphene to water increases thermal conductivity by 10.3% and heat transfer coefficient by 14.2% at the Reynolds number of 1850.

The convective heat transfer coefficient of nanofluids increases with increase in the mass flow rate and volume concentrations is observed from the previous literatures. Higher concentration of oxide nanoparticles increases the viscosity and leads to high pumping power, pressure drop. Only few works has been carried out with graphene nanosheets with less than 1% volume concentration and moreover no work has been carried out with less diameter test section. Therefore in the present work, an experimental study is carried out to investigate the forced convection of graphene-water nanofluid, inside a horizontal tube submitted to a uniform heat flux boundary condition on its outside surface in the laminar regime. The effect of mass flow rate, heat flux and particle volume concentration (0.2%-0.6%) on the convective heat transfer coefficient and pressure drop characteristics are experimentally studied and analyzed. The Reynolds number is varied from 500 to 2000, which is suited for all types of heat exchanger applications in the thermal equipment and systems and design industries. The heat flux applied is 10000 W/m^2 which is useful for electronic cooling applications. The results obtained from the present study are compared with existing published literatures, analyzed and presented.

EXPERIMENTATION

An experimental setup has been designed and fabricated as shown in Figure.1. The test section is made of Stainless Steel (SS) with inner diameter of 4 mm, 1m length and thickness of 1mm. In order to get the shorter entry length, at the inlet and outlet of the test section a 0.1m length, 19mm OD and thickness of 1.5mm of SS Tube are welded. K-Type thermocouples with an accuracy of $\pm 0.1^0$C are soldered to the outer wall of the test section at six equal intervals along the axial location. Two K-type thermocouples are inserted in flow channel before and after the heated test section to measure the bulk temperature of the fluid. Eight thermocouples are connected to a data logger (YOKOGAWA – N 200, Model – DA 100-13-1F) for the measurement of temperature.

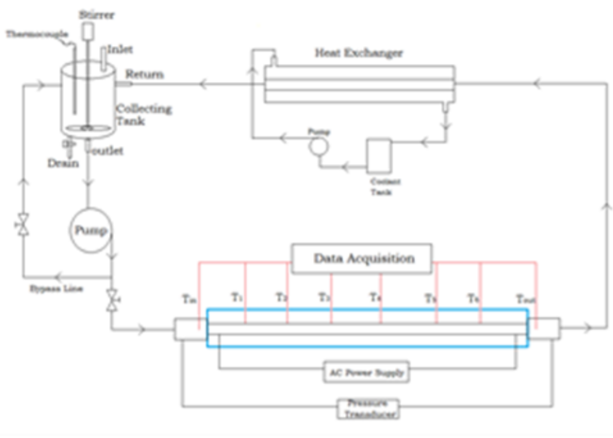

Figure 1: Schematic diagram of Experimental layout

The tube is heated by resistance heater (maximum capacity – 300 W) coiled over the test section and the heat load is varied by a variable transformer. Tube along with the heater is insulated outside for providing a constant heat flux to the tube wall. The fluid is supplied to the test section by a centrifugal pump, which can provide a maximum head of 15m. The hand shut off valves are used for controlling the mass flow rate and the flow rate is measured by conventional weighing method with ± 1% accuracy. A heat exchanger is provided in order to maintain the constant temperature in the collecting tank. Pressure loss was measured by two pressure transducers with an accuracy of 0.001% placed before and after the test section.

Nanofluid Preparation

The experiment should be conducted with stable nanofluid without aggregation and sedimentation. Graphene nanoparticles purchased from SkySpring Nanomaterials, Inc., USA is used for preparing the nanofluid by two step method. De-ionized water is used as base fluid. Surfactant, sodium dodecyl benzene sulfonate (SDBS) of about 50% of the mass of graphene nanoparticle is added to keep the nanoparticles stable in the base fluid. Nanofluids with 0.2%, 0.4% and 0.6% volume of graphene nanoparticles are prepared. The dispersant (SDBS) was first added to the DI water and sonicated for about 15min. Sonication is continued when nanoparticles were added at a very slow rate. After the desired amount of particles, the mixture is sonicated for 45 minutes. The nanofluid prepared by this method is found to be stable without sedimentation.

Characterization of the sample is done by Scanning Electron Microscope (SEM) and X-Ray diffraction (XRD). The SEM image of the graphene nanoparticles at 20000X Magnification is shown in Figure 2. XRD Spectra of the Graphene nanofluid is shown in figure.3. The Crystal Size, D is calculated from Scherrer's equation.

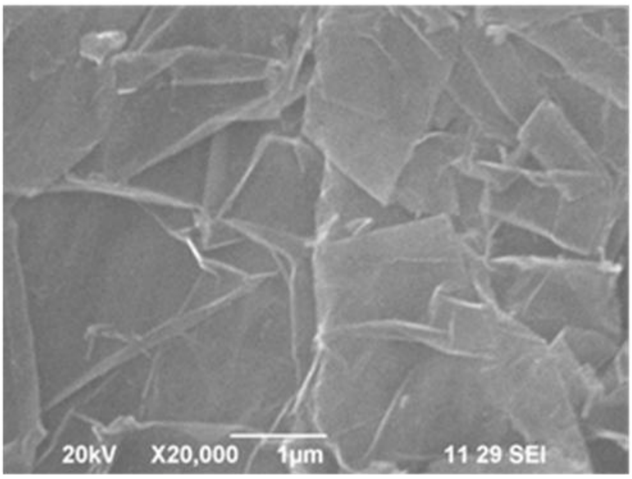

Figure 2: SEM image of Graphene Nanopowder

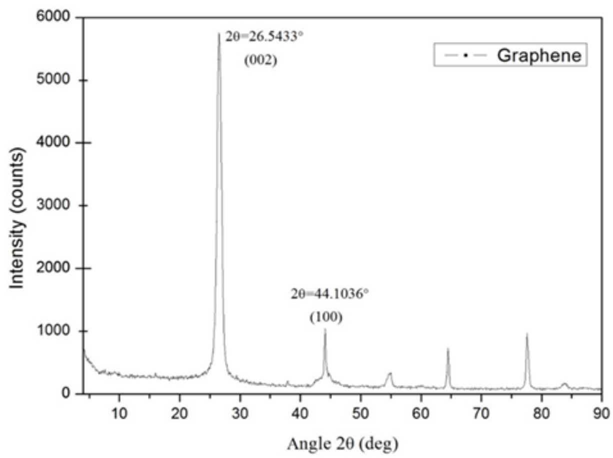

Figure 3: XRD Spectra of Graphene Nanopowder

Thermophysical Properties

The thermophysical properties of graphene nanofluid are calculated for mean bulk temperature of the fluid by the following correlations.

$$\rho_{nf} = \nu.\rho_s + (1-\nu)\rho_{bf}$$

$$\mu_{nf} = \mu_{bf}(1+2.5\nu)$$

$$k_{nf} = \left[\frac{k_s + 2k_w + 2(k_s - k_w)(1+\beta)^3 \nu}{k_s + 2k_w - (k_s - k_w)(1+\beta)^3 \nu}\right] k_w$$

$$C_{P_{nf}} = \frac{\nu\left(\rho_s.C_{P_s}\right) + (1-\nu)\left(\rho_w.C_{P_{bf}}\right)}{\rho_{nf}}$$

Data Processing

Convective heat transfer coefficient is calculated from the experimental data. To evaluate the local convective heat transfer coefficient, the following equation is used.

$$h(x) = \frac{q}{t_{iwx} - t_{fx}}$$

q is the heat flux in W/m^2

Inner wall temperature t_{iwx} was calculated from the conduction equation.

$$t_{iwx} = t_{owx} - \Delta t_x$$

Where, $\Delta t_x = Q.\frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi k L_x}$

Fluid temperature t_{fx} at a given point is approximated as average value between inlet bulk temperature of the fluid and wall temperature at the local point.

$$t_{fx} = t_{in} + \left(t_{out} - t_{in}\right)\left(\frac{x}{L}\right)$$

The corresponding local Nusselt number was expressed as,

$$Nu(x) = \frac{h(x)D}{k_f}$$

The experiment setup is validated by conducting experiments on water. The experimental results obtained are compared with existing theoretical correlations for forced convection of single phase flow inside cylinder with constant wall heat flux.

For Laminar regime, (Shah Equation)

$$Nu = \begin{cases} 1.953\left(\mathrm{Re}\,\mathrm{Pr}\dfrac{D}{x}\right)^{\frac{1}{3}}, & \left(\mathrm{Re}\,\mathrm{Pr}\dfrac{D}{x}\right) \geq 33.3 \\ 4.364 + 0.0722\,\mathrm{Re}\,\mathrm{Pr}\dfrac{D}{x}, & \left(\mathrm{Re}\,\mathrm{Pr}\dfrac{D}{x}\right) < 33.3 \end{cases}$$

The Scherrer's equation for finding the Crystal Size, D is shown below.

D= (λ. 0.94) / (β Cos θ)

Where λ = Wavelength of X-ray = (1.54 x 10^{-10}), Correction factor=0.94, β = (Full width half max. x 3.14) / 180 and θ = Incident light angle.

RESULTS

The experimental setup has been validated with water as a reference fluid. The values of theoretical Nusselt number are calculated from the shah correlations for Re=1250 and heat flux of 10000 W/m^2. The results were compared as shown in Figure.4. The theoretical and experimental values are in good agreement with only ±4% deviation.

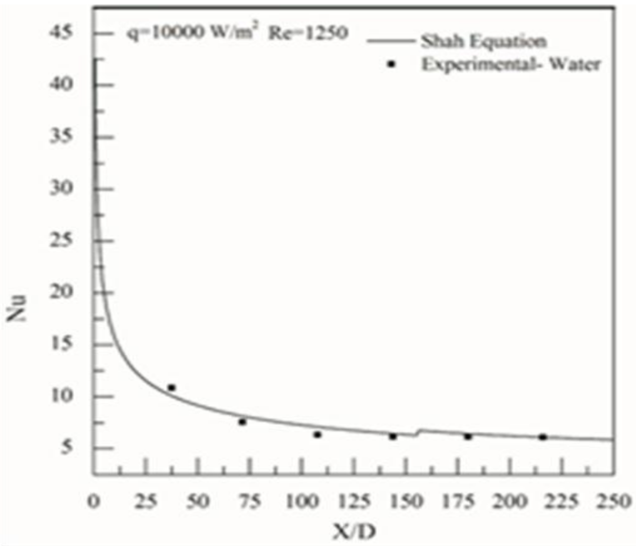

Figure 4: Comparison of the local Nusselt number with Shah Correlation at Re=1250

The temperature distribution of the graphene-water nanofluid, along the axial length of the test section in the laminar regime for Reynolds number ranging from 500 to 2000 is shown in Figure.5. The volume concentrations of 0.2%, 0.4% and 0.6 % and the average Reynolds number of 1250 the average wall temperatures are 52.73˚C, 49.71˚C, 48.4˚C, and 47.64˚C respectively. It is observed that as the particle volume concentration is increased the wall temperature values are decreased resulting in the increment of heat transfer coefficient.

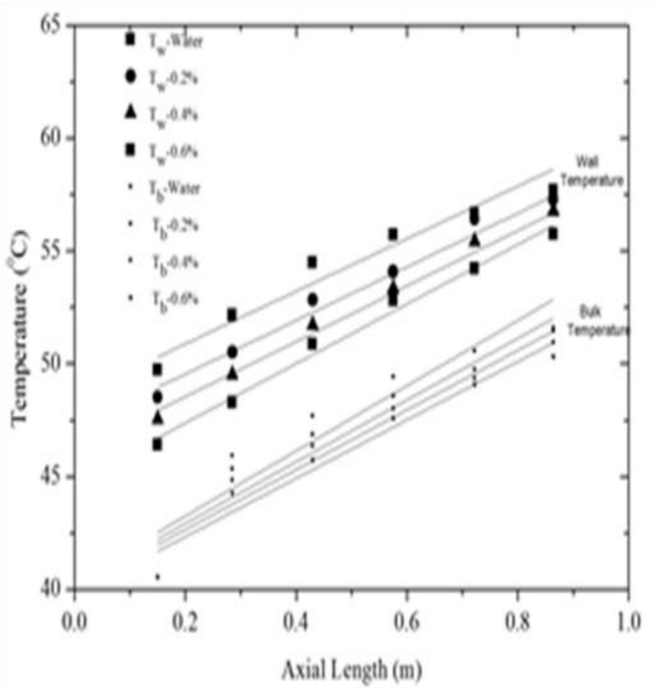

Figure 5: Effect of Particle concentration on axial development of wall and bulk temperature

Experimental Nusselt number which signifies the dominant mode of heat transfer is calculated from the measured wall temperature, for a given Re and constant heat flux is shown in Figure 6. Nusselt number decreases along the axial distance; of course this is due to the wall temperature profile development which depends on the development of flow itself. Local Nusselt number is very high at the entrance and keeps on decreasing as the distance from the entry increases. From the result it is clear that the enhancement due to the use of nanofluid is convection than conduction since Nusselt number is high for the nanofluid with various concentrations.

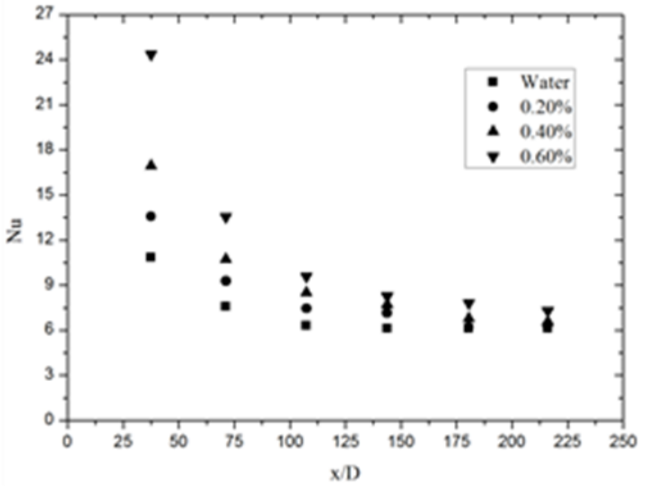

Figure 6: Effect of particle concentration on Nusselt number

The variation of local heat transfer coefficient on axial length in the laminar regime with average Reynolds number 1250 and different volume concentrations of graphene-water nanofluid is shown in Figure 7. It has been seen that the local heat transfer coefficient in the entrance region is high and decreases with the increase in the axial length of the test section. Donsheg Wen's [4] study explains one possible mechanism of enhancement in local convective heat transfer coefficient (CHTC) is due to increase in thermal conductivity and decrease in thermal boundary layer thickness.

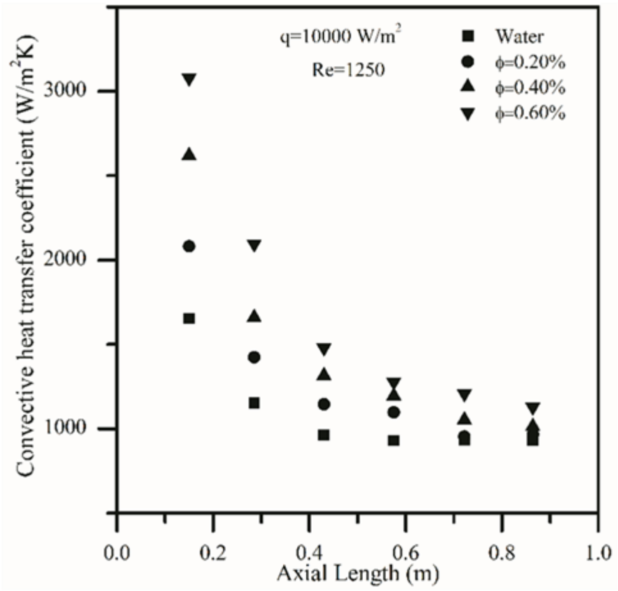

Figure 7: Effect of particle concentration on axial development of heat transfer coefficient

Enhancement in thermal conductivity from the previous studies show that this is due to mechanisms such as Brownian motion, clustering which are observed in nanofluid flow. For the same mass flow rate, the local heat transfer coefficient increases at the entrance by 125% with 0.6% particle volume concentration. In case of laminar flow, the percentage increase rapidly decreases as the dimensionless length increases and the values are almost same once the flow becomes fully developed. Choi et al. found an enhancement of 8% in convective heat transfer coefficient for 0.3% particle volume concentration of Al_2O_3 in a straight tube whereas in this study 31.56% average enhancement is observed for 0.6% volume concentration of graphene nanoparticles.

CONCLUSION

This paper is concerned with the study of convective heat transfer coefficient of graphene nanofluid. Convective heat transfer coefficient of the base fluid significantly increases by adding graphene nanoparticles. The enhancement of average CHTC of up to 56% and the pressure drop of 28.7% is obtained. The enhancement of the CHTC can be increased further by eliminating surfactant and maintain the stability without changing the thermophysical properties of the nanofluids.

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