Experimental convective heat transfer studies in a turbulent flow regime using alumina-water nanofluids

K.B. Anoop1,*, Sarit K. Das2, S. Kabelac3

ABSTRACT
The present work investigates forced convective heat transfer characteristics of alumina-water nanofluids in a turbulent flow regime. Nanofluids are dilute colloidal suspensions with nano-sized particles (<100 nm) dispersed in a basefluid. The thermal conductivity values are measured by a steady state method, using a guarded hot plate (GHP) apparatus customized for liquids. The forced convective heat transfer characteristics are evaluated with the help of a test loop maintained in a constant heat flux condition. Controlled experiments under a turbulent flow regime are carried out using four particle concentrations (0.5vol%, 1vol%, 2vol% and 4vol %). The experimental results show that, the thermal conductivity of nanofluids increases with an increase in particle concentration and closely follow effective medium theories. However, the enhancement of heat transfer coefficients in the turbulent regime is observed to be within the measurement uncertainty.

Keywords: guarded hot plate, thermal conductivity, viscosity, effective medium theory

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INTRODUCTION

Cooling of thermal equipment has been one of the vital problems faced since the dawn of industrialization. Constant efforts to improve the heat transfer capabilities of cooling fluids have been carried out since then. One strategy to improve the heat transfer capabilities of cooling fluids was to suspend solid particles in them. The use of micron-sized particles in a cooling fluid brought technical problems of sedimentation and clogging, which restricted their applicability. It was understood that the thermal conductivity of suspensions increased with the surface area to volume ratio of the particles used. This triggered the concept of applying nano-sized particles in suspensions, which efficiently coped with the technological ability of the 21st century to produce nano-sized particles with controlled material properties. The use of nano-sized particles for suspensions was initially proposed by Masuda et al. Extensive research followed by Choi and co-workers from Argonne National Laboratories, whose investigation on heat transfer characteristics, introduced a new class of engineered fluids called nanofluids. Nanofluids by definition are dilute colloidal suspensions with nano-sized particles (<100 nm) dispersed into a basefluid. Several experiments were conducted to investigate the heat transfer capabilities of nanofluids. Initial experimental observations on thermal conductivity enhancement in alumina-water nanofluids showed an enhancement of thermal conductivity by 30% for a nanoparticle concentration of 4.5–vol%. Whereas, for similar nanoparticle concentration for alumina-water nanofluids, Wang et al. in their study using a steady state parallel-plate apparatus, could get an enhancement of thermal conductivity by only 14%. A similar order of enhancement was observed by Das et al. in their studies using a Temperature Oscillation (TO) measurement technique. In addition, the majority of literature showed that the thermal conductivity augmentation follows an increasing trend with a decrease in particle size, an increase in particle loading and an increase in temperature. On the contrary to the above studies, which showed anomalous enhancement in thermal conductivity, a few studies have shown that the effective thermal conductivity of the nanofluids exhibited no anomalous enhancements and it could be predicted accurately by the model equation, such as effective medium theory for suspensions. Zhang et al. using short-hot-wire probes coated with a pure Al₂O₃ thin film (to prevent electrical leakage), also demonstrated that there was no anomalous enhancement. Similar observations were reported by Beck et al., who investigated the thermal conductivity of alumina-ethylene glycol nanofluids using a liquid metal transient hot wire apparatus. Enhancement of thermal conductivity with increase in temperature was not observed in above cases. Observations of International Nanofluid Property Benchmark Exercise (INPBE) on thermal conductivity of nanofluids also revealed parallel results.

Similar to observations of thermal conductivity enhancement, forced convective heat transfer characteristics of nanofluids also showed mixed performances. The majority of publications showed a heat transfer enhancement in the range of 10–14%. The first work on convective flow and heat transfer of nanofluids was presented by Pak and Cho, even before Choi and his group introduced the term nanofluids. They referred it as “dispersed fluid with sub-micron particles.” It was observed that in the turbulent flow regime the convective heat transfer coefficient of nanofluid was about 3–12% lower than that of pure water (when compared at the same average velocity of fluid in the pipe). On the contrary, Xuan and Li observed an increase in heat transfer coefficient by 40% at the same velocity for Cu-water nanofluid in a turbulent flow regime. Heris et al. conducted experiments using copper oxide-water and alumina-water nanofluids, in both laminar and turbulent flow regimes. Increased enhancement was reported for alumina nanofluids when compared to that with copper oxide nanofluid. They observed a 40% enhancement in heat transfer coefficient for a particle concentration of 3vol%. He et al. investigated the convective heat transfer of titania nanofluids and obtained a maximum enhancement ~12% for 1.18vol% in the heat transfer coefficient for laminar as well as turbulent flow regimes. Wen and Ding focused on the laminar heat transfer in the entrance region of a tube flow using alumina-water nanofluids. They observed a significant enhancement in heat transfer in the entrance region and it decreased with axial distance. Anoop et al. investigated the effect of particle size in convective heat transfer nature of a developing flow in a laminar regime. It was observed that the nanofluid with 45 nm particles exhibited higher heat transfer coefficient than that with 150 nm particles. Enhancement in the laminar developed region was shown by Hwang et al. Here convective heat transfer coefficient of the alumina-water nanofluid increased by 8% for a concentration of 0.3 vol%. Contradictory to the above observations, two recent studies from Massachusetts Institute of Technology claimed that there was no anomalous heat transfer enhancement when using nanofluids. Williams et al. in their studies with alumina-water and zirconia-water nanofluids in turbulent flow...
regime showed that if the measured temperature and loading dependent thermal conductivities and viscosities of the nanofluids were used, then conventional correlations would accurately reproduce the convective heat transfer and viscous pressure loss characteristics. They argued that nanofluids do not exhibit any anomalous heat transfer behaviour. Similar observations were made in a laminar regime by Rea et al.20

The rheological characteristics of nanofluids were observed to be higher than the theoretical predictions of classical viscosity models. The majority of publications show an increase in viscosity ranging from 5 to 14% with particle loading.21 The effective viscosity of nanofluids was observed to increase with a decrease in temperature and particle size.21 It was also observed that majority of water based nanofluids exhibited Newtonian characteristics, whereas oil-based nanofluids exhibited non-Newtonian nature.21

Several theoretical models were proposed to explain the anomalous heat transfer characteristics of nanofluids. The majority of these models tried to explain the enhancement in heat transfer through Brownian motion of nanoparticles in conjunction with aggregation and diffusion theories.12 The flattening of velocity profile, shear thinning behavior of nanofluids and thermo-phoretic forces were argued to be the probable reasons for enhanced convective heat transfer characteristics.12,22,23

From the literature review, it is clear that many controversies still exist in the heat transfer studies of nanofluids. Hence more experimental efforts are required to validate the real heat transfer characteristics of nanofluids.

The present work attempts to investigate the convective heat transfer characteristics of alumina nanofluids in turbulent flow regime. Alumina-water nanofluids are used for the investigation. Thermal conductivity is a crucial parameter in heat transfer evaluation, and is measured using a steady state method. The viscosity values of nanofluids are measured and included in the thermal evaluation.

METHODS

(a) Preparation of nanofluid

To conduct a proper evaluation of thermal characteristics of nanofluids, it is very important to have a stable and homogeneous suspension. In the present study, alumina-water nanofluids were prepared in a top-down approach by dispersing commercially purchased nanopowder (Sigma Aldrich powder (No: 544833) size: < 50 nm, with a surface area of 35–43 m²/g) into the basefluid (water). Four basic steps were involved in the formulation of nanofluids. (a) After mixing nanopowder into the basefluid the suspension was homogenized using a high performance disperser/stirrer (T25 digital ULTRA-TURRAX) for 30 minutes. (b) The above solution was kept in an ultrasonic bath (35 kHz) for 2 hours. (c) A high power ultrasonication using an ultrasonic disruptor (KLN Sys 587) was applied to the nanofluid solution for 1 hour. The disruptor consisted of a probe attached to a sonotrode, vibrating at 20 kHz and 50 µm amplitude. The temperature of the nanofluid sample was maintained at 22°C by external cooling during the process of sonication. (d) Further de-agglomeration was carried out by passing the suspension through a high pressure shearing process.24 For this, a high pressure static mixer (Combi-mixer (101-10001-F) Ehrfeld Mikrotechnik BTS GmbH), which consisted of a chamber made of micro-channels (with gap of 70 µm each), was used. Working pressure of about 100 bar was applied for shearing. Electro-static stabilization technique was applied to the suspension by keeping the pH value away from the Iso-Electric Point (IEP around pH 8.925). For all the nanofluid samples the pH value was maintained at 4.5. The suspension had a zeta potential value of 54 mV which assured good electrostatic stability.

The particle size distribution of the nanofluids prepared were measured using a Malvern Zetasizer (Nano-ZS90). The average particle diameter obtained was observed to be approximately 100 nm (Figure 1), which was larger than primary particle size claimed by the powder suppliers (which was 50 nm). This indicated that some agglomeration could have occurred while suspending particles in the basefluid.24 For the present study, nanofluids were prepared with varying concentrations from 0.5 – 6vol%.

(b) Thermal conductivity

A steady state parallel plate apparatus was used for thermal conductivity measurement. The setup was specially designed for liquid measurement and in close co-operation with the German Institute of Standards. A schematic of the equipment is shown in Figure 2. The fluid sample is placed between two parallel copper plates. At steady state, the top plate, termed the heater plate, provides the heat that
travels through the specimen. On the opposite side is a cold plate that provides the temperature difference driving heat through the sample. In order to maintain a one-dimensional heat transfer, the heat flux, from the hot plate has to flow only to the cold plate kept below. The heat losses from the top and to the sides of the heater plate are to be minimized. For this purpose, the hot plate is shielded towards the top by a top guard heater and towards the circumference by a heated guard ring. Typically, the top plate is heated to a temperature $2{}^\circ C$ above the lower cold plate temperature and is separated from each other by three small high precision quartz spacers (with very low thermal expansion coefficient), having a thickness of exactly 1 mm each. Heating from the top and the narrow gap between the plates helped in minimizing natural convection currents. The key point in successful running of a GHP system lies in maintaining controlled isothermal surfaces (checked prior to the experiments using an infrared camera), together with an accurate measurement of surface heat flux values. Isothermal surfaces were maintained by indirect heating of plates (flat electrical heater embedded in a rubber sheath and mounted over the plates). An optimized PID controlled heating of the main and guard heaters, controlled by Lab-Vi ew program aided in attaining a steady state faster, as well as acquiring sufficient data for the thermal conductivity evaluation. The setup was enclosed and immersed in a

Figure 1. Particle size distribution of nanoparticles based on number.

Figure 2. Guarded hot plate experimental setup for measurement of thermal conductivity.
constant temperature fluid. During experimentation, the temperatures of the guard heaters were matched as closely as possible to the temperature of the hot plate. Remaining deviations were measured and taken into account as corrections.

(c) Viscosity measurements
Viscosities of nanofluids were measured using a Physica UDS 200 Rheometer with a cone and plate geometry (cone diameter of 75 mm and a cone angle of 1°). All the experiments were conducted at a constant gap of 0.05 mm. An initial stabilization period of 2 minutes was given for achieving constant temperature, after which a variable shear rate, ranging from 10 to 1000 (1/sec) was applied. Since the torque applied during the experiment was in the range 1–150 µNm, the maximum percentage error in viscosity measurement, as specified for the equipment, was limited to 2%.

(d) Convective heat transfer coefficient
A laboratory test loop was built to investigate the convective heat transfer characteristics under turbulent flow condition for nanofluids. Figure 3(a) and (b) show a schematic and photograph of the test loop used. The circulation loop consists of a pulsation free vane pump (Speck Pump DS-300) followed by a cooling chamber, turbine flow meter (KEM, MH3E/4) and a collecting tank. The turbulent flow in the loop was maintained and varied with the help of a pump, whose flow rate was precisely controlled by a speed controller (Danfoss VLT 2800) attached to the pump. Pressure drop across the test section tube was measured using a pressure transducer (Contrans ASD800, in 3 different pressure ranges viz. ± 400mbar, ± 2.5 bar and 0–10 bar). The heater section was made of a thin steel tube, length 500 mm and 1.3 mm inner diameter. Constant heat flux was provided by a heater coil uniformly wound along the length of the tube. DC power supply was used as the power source for the heater. Six calibrated thermocouples (K-type) were brazed on the surface of the tube, which measured the surface temperatures along the length of the tube. An initial un-heated length of 50 mm was provided at the entrance in order to allow the flow to be hydro-dynamically developed. The inlet and outlet temperatures were measured by two thermocouples immersed in the mixing chambers provided at the inlet and exit of the test section. The fluid after passing through the heater section was collected in a tank and then it was pumped into a cooling unit, which was a brazed plate heat exchanger. The constant temperature fluid coming out of the heat exchanger was then passed through a calibrated turbine flow meter towards the test section. The wall temperatures, the flow rate and the electrical power supplied were acquired using a Keithley 2700 DAQ system, with the help of a Lab-View program. Quantitative evaluation of the heat transfer coefficient is made by measuring the heat supplied (which is the product of voltage and current supplied to the heater), wall temperatures and fluid temperatures. The average heat transfer coefficient occurring is evaluated as

$$h_{av} = \frac{q''}{\Delta T_{LMTD}}$$

where,

$$\Delta T_{LMTD} = \frac{(T_{w, out} - T_{f, out}) - (T_{w, in} - T_{f, in})}{\ln \left(\frac{(T_{w, out} - T_{f, out})}{(T_{w, in} - T_{f, in})}\right)}$$

$T_w$ and $T_f$ are the wall and fluid temperatures, respectively, and $q''$ is the heat flux applied.

The heat transfer coefficient and flow rates are non-dimensionlized as

$$Nu = \frac{h_{av}D}{k} \quad \text{and} \quad Re_D = \frac{\rho V D}{\eta}$$

The thermal conductivity value used is at the bulk mean temperature. The density and specific heat of the nanofluid are evaluated using averaged volume fraction ratio, which is generally accepted.

$$\rho_{nf} = (1 - \phi)\rho_{bf} + \phi\rho_p$$

$$\left(\rho C_p\right)_{nf} = (1 - \phi)\left(\rho C_p\right)_{bf} + \phi\left(\rho C_p\right)_{p}$$
The theoretical value for heat transfer coefficient in the turbulent flow regime for validation is evaluated using Gnielinski Correlation and is used for comparison.

\[
\begin{align*}
\Nu_{D,H} &= \frac{(f/8)(\Re_D - 1000)\Pr}{1 + 12.7(f/8)^{1/2}(\Pr^{2/3} - 1)} \\
&= \left( \frac{f}{8} \right) \frac{\Re_D - 1000}{1 + 12.7 \left( \frac{f}{8} \right)^{1/2} \left( \Pr^{2/3} - 1 \right)}
\end{align*}
\]

Where \( f = (0.79 \ln \Re_D - 1.64)^{-2} \) and \( \Pr = \left( \frac{\eta C_p}{k} \right) \)

During experimentation it is observed that the energy balance ratio, which is the ratio of power input to the heater tube to the heat taken away by the fluid, is always above 0.97.
RESULTS AND DISCUSSION

After preparing stable alumina-water nanofluids, their thermal conductivities are measured using a guarded hot plate (GHP) apparatus. During the measurements, there exists a vertical heat flow between the main heater and the top guard heater, and a horizontal heat flow between heater plate and the guard ring plate. These losses (systematic errors) have to be accounted and corrected for. After accounting for these losses along with corrections for thermo-elements, it was observed that the apparatus was able to measure thermal conductivity with a maximum uncertainty of 1%. Prior to the experiments with nanofluids, the effect of basefluid pH on thermal conductivity measurement was evaluated. It was noticed that basefluid pH did not influence the thermal conductivity, even though the electrical conductivity was highly dependent on pH. This fact is depicted in Figure 4. Since indirect electrical heating is applied in present GHP apparatus, the electrical conductivity of fluids will not influence the thermal conductivity measurements. Figure 5 shows the variation in thermal conductivity values at various concentrations and temperatures for alumina-water nanofluid. It is observed that with an increase in particle concentration the thermal conductivity of nanofluid also increases. It may be noticed that, with respect to temperature, the increase in thermal conductivity of nanofluids follows a similar trend to that of the basefluid.

Viscosity measurements of both water and alumina-water nanofluids exhibited Newtonian behavior in the shear rate range of 10 to 1000s⁻¹ at 20°C. Viscosity of water measured is observed to be approximately 1mPa.s, which practically matches with the theoretical value and it is shown in Figure 6(a). Except for the initial part, where the effects of inertial forces are high, both basefluid and nanofluids exhibit Newtonian characteristics. Figure 6(b) shows the variation of nanofluid viscosities with temperature at a constant shear rate of 200s⁻¹ (where initial inertial effects are minimum). It can be seen here that the viscosity of basefluid and nanofluids decrease with an increase in temperature. It is also noticed that at 50°C, the viscosity of 6 vol% alumina nanofluid becomes almost equal to that of water at 20°C. The above observation reveals that an inappropriate use of viscosity value could give rise to misleading results in pumping power and in turn affects the heat transfer analysis.

A comparison of increased viscosity and thermal conductivity, as a function of volume fraction, is shown in Figure 7. Here, the relative viscosity (ratio of viscosity of nanofluid to basefluid at a shear rate of 200s⁻¹) and relative thermal conductivity (ratio of thermal conductivity of nanofluid to basefluid) obtained at 20°C are plotted against volume fraction (φ). Classical prediction of thermal conductivity enhancement starting from Maxwell to Hamilton Crosser, all converge to a single form (effective medium theory), when the particle conductivity is much higher than basefluid (about 20 times), and with minimum interfacial resistance conditions. This defines the classical effective medium theory, where the relative thermal conductivity would be equal to $1 + 3\phi$. From Figure 7, it is shown that the thermal conductivity measured for alumina-water nanofluids follow the classical effective medium
theory. However, relative viscosity seems to be much higher than the conventional predictions. Classical prediction of relative viscosity of uncharged particle suspension should follow the Einstein equation, which is equal to $1 + 2.5\phi$. However, for the present nanofluids, linear fitting to the measured relative viscosity follow a relation $1 + 10\phi$, which is higher than the classical prediction. Combined contribution of particle agglomeration and electro-viscous effects due to electro-static stabilization procedure could be the reasons for the above increase in viscosity.

After having evaluated the thermo-physical properties, experiments are conducted in the turbulent flow regime to evaluate the heat transfer coefficients. Initially, experiments are conducted with the basefluid. This helped in validating the test loop and formed the basis for comparing the results with the nanofluids. Due to experimental limitations, only four volume concentrations (0.5%, 1%, 2% and 4%) were considered for convective heat transfer studies. Pressure drop studies were carried out prior to heat transfer experimentations. Figure 8 shows the parity plot between the measured and predicted pressure drop. Theoretical pressure drop was evaluated using equation 7.

\[ \Delta \rho = \frac{\rho R V^2}{2D} \]  

(7)

The Gnielinski correlation with thermo-physical properties of nanofluids was used to evaluate the friction factor. It is observed that the measured values of pressure drop lie within 10% of the predicted values, with all uncertainties included. The above fact reveals that the pressure drop that occurred in

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Figure 5. Thermal conductivity values measured using guarded hot plate apparatus.

Figure 6. (a) Viscosity variation with shear rate measured at 20°C. (b) Viscosity variation as a function of temperature at a shear rate of 200 s$^{-1}$. 

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the turbulent region could be predicted once the thermo-physical transport properties of nanofluids are known. This observation also vindicates the homogeneous nature of nanofluids used.

Convective heat transfer measurements using alumina-water nanofluids are then carried out. It is believed here that the real nature of heat transfer enhancement could be revealed by plotting dimensional results rather than non-dimensional ones. Both dimensional and non-dimensional plots are presented. The maximum experimental uncertainty in the present turbulent flow loop, in Re and Nu are estimated to be ±2.3% and ±2.6%, respectively.

Figure 9(a) shows the heat transfer coefficient variation plotted against the flow velocity and a non-dimensional plot of the same is given in Figure 9(b). Non-dimensionalization for heat transfer coefficient and flow velocity is carried out using the scaling factors depicted in equation (3). It is seen that the basefluid (water), moderately follows the theoretical trend as given by Gnielinski correlation. At lower particle concentrations, a slight enhancement in heat transfer is observed for nanofluids, while the enhancement deteriorates at higher concentrations. The enhancement values fall within the uncertainty range of experimentation and hence no major enhancement in heat transfer is observed. This is contrary to some observations in literature (Xuan and Li), however it is in similar to the observations made by Williams et al. No peculiar trend of anomalous enhancement/deterioration is observed, as the data lie scattered within the uncertainty limits of experimentation. Exact reasons for this cannot be ascertained, however given below are some plausible explanations.

Major transport mechanisms in a turbulent flow are by eddies and by energy dissipation of eddies. From an analysis, in line with Buongiorno, it may be seen that particles in a nanofluid cannot make an individual effect as they move along with the eddies. It may be seen that the smallest eddy would be much larger than the particle sizes involved and there would not be any major slip motion between particles and eddies, giving rise to an extra heat transfer effect. The largest eddy would be the size of the tube and smallest one, according to Kolmogorov’s scale, would be \( \sim 1.3 \text{ mm} \times (\text{Re})^{-3/4} \sim 1 \mu m \) in size. Hence the particles in the flow would be easily carried along with eddies.

Another evaluation based on particle stopping distance also reveals similar observations. Particle stopping distance, is the distance a particle can move by inertia once the eddies have come to rest. If these particles could penetrate into the laminar sub-layer, it could influence the sub-layer characteristics and thereby increase the heat transfer. For the present flow situation, based on definition given by Buongiorno, the particle stopping distance is of the order of 1 nm. The laminar sub-layer thickness near the wall (from definition) would be having a thickness of 3 \( \mu m \). Thus the particle stopping distance is far less than the laminar sub-layer thickness, and hence the particles cannot penetrate into the sub-layer to alter the rate of heat transfer.

From the above scaling analysis, it is observed that the nanoparticles individually cannot have an effect in the flow and the heat transfer characteristics would be overridden by the eddy mixing characteristics. Thus, in the turbulent regime, the overall effect on heat transfer is caused only by
thermo-physical property changes. The effect of thermo-physical properties on the heat transfer characteristics of nanofluids in turbulent flow regime is discussed next.

As observed from the measurements for alumina-water nanofluids, the thermal conductivity, viscosity and density values increases with volume fraction, whereas the heat capacity decreases with an increase in particle concentration. In addition to the above, increase in viscosity is observed to be much higher than the increase in thermal conductivity. From the fundamental knowledge of heat transfer, it is known that viscosity will not influence the heat transfer coefficient in a fully developed laminar flow, however, it will affect in the case of a turbulent flow conditions. The above fact can be easily understood by analyzing the two basic situations of fully developed laminar flow and fully developed turbulent flow at constant heat flux condition. For the above condition, the effect of thermo-physical properties on heat transfer coefficient would finally take the form as given by Equation 8 and Equation 9, respectively:

\[ Nu = \frac{4.36}{h \alpha k} \]  \hspace{1cm} (8)

\[ Nu = 0.023Nu^{0.8}Pr^{0.4} = h\alpha k^{0.6}C_{p}^{0.4} \rho^{0.8} \eta^{-0.4} \]  \hspace{1cm} (9)

Figure 8. Parity plot comparing measured pressure drop and estimated pressure drop for turbulent flow.

Figure 9. (a) Heat transfer coefficient variation with velocity. (b) Non-dimensionalized heat transfer coefficient variation, with Reynolds number.
It is noticed that the effect of increase in thermal conductivity is more pronounced in laminar flow than in turbulent flow. When the increase in viscosity is very high, it can result in the deterioration of heat transfer coefficient along with a reduction in the $C_p$ value. The interplay between increase in thermal conductivity and viscosity is depicted in Figure 10. Here the heat transfer coefficient ratio (ratio of heat transfer coefficient of nanofluid to basefluid) is evaluated from Equation 9 for alumina-water nanofluids. $C_k$ and $C_h$ are the coefficients of thermal conductivity enhancement and viscosity enhancement, respectively (Equations 10 and 11).27

$$\frac{k_{nf}}{k_{bf}} = 1 + C_k \phi$$

(10)

$$\frac{\eta_{nf}}{\eta_{bf}} = 1 + C_h \phi$$

(11)

When the $C_h$ value is less (eg. $C_h = 2.5$), a moderate increase in thermal conductivity ($C_k = 3$) gives enhancement in heat transfer coefficient. However, when $C_h$ is higher ($C_h = 10$) when compared to $C_k$ (eg. $C_k = 3$), which is similar to the present experimental case, slight deterioration in heat transfer coefficient is observed. Thus, it may be inferred that the experimentally observed deterioration (as in present case) is brought about by the changes in thermo-physical properties of nanofluids.

**CONCLUSION**

Evaluation of forced convective heat transfer coefficient under a constant heat flux condition is carried out in a turbulent flow regime with four particle concentrations (0.5vol%, 1vol%, 2vol% and 4vol %) of alumina-water nanofluids. The experimental observations showed that the enhancement of thermal conductivity with increase in particle concentration, which matches with the predictions of effective medium theories for suspensions. The nanofluids exhibited Newtonian behavior, however, the increase in viscosity is found to be much higher than classical predictions. No major trend in heat transfer enhancement is observed under turbulent flow conditions. The experimental results were scattered within the experimental uncertainty limits. The nanoparticles were not able to impart individual slip motion under turbulent flow. Hence, particles were moving with eddies and enhancements and were directly related to the changes in thermo-physical properties of nanofluids.

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Nomenclature

- $C_p$: Specific heat [J/kgK]
- $D$: Diameter of the tube [m]
- $f$: Friction factor
- $h$: Heat transfer coefficient [W/m²K]
- $k$: Thermal conductivity [W/m K]
- $Nu$: Nusselt number
- $P$: Pressure [Pa]
- $Pr$: Prandtl number
- $q^o$: Heat flux [W/m²]
- $Re$: Reynolds number
- $T$: Temperature [K]
- $\rho$: Density [kg/m³]
- $\eta$: Viscosity [Pa s]
- $\phi$: Volume fraction

Subscripts

- avg: average
- bf: basefluid
- f: fluid
- in: inlet
- LMTD: Log Mean Temperature Difference
- nf: nanofluid
- out: outlet
- p: particle
- w: wall

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