Chapter 14

Application of Nanofluids in Heat Transfer

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1. Introduction

A wide variety of industrial processes involve the transfer of heat energy. Throughout any industrial facility, heat must be added, removed, or moved from one process stream to another and it has become a major task for industrial necessity. These processes provide a source for energy recovery and process fluid heating/cooling.

The enhancement of heating or cooling in an industrial process may create a saving in energy, reduce process time, raise thermal rating and lengthen the working life of equipment. Some processes are even affected qualitatively by the action of enhanced heat transfer. The development of high performance thermal systems for heat transfer enhancement has become popular nowadays. A number of work has been performed to gain an understanding of the heat transfer performance for their practical application to heat transfer enhancement. Thus the advent of high heat flow processes has created significant demand for new technologies to enhance heat transfer

There are several methods to improve the heat transfer efficiency. Some methods are utilization of extended surfaces, application of vibration to the heat transfer surfaces, and usage of micro channels. Heat transfer efficiency can also be improved by increasing the thermal conductivity of the working fluid. Commonly used heat transfer fluids such as water, ethylene glycol, and engine oil have relatively low thermal conductivities, when compared to the thermal conductivity of solids. High thermal conductivity of solids can be used to increase the thermal conductivity of a fluid by adding small solid particles to that fluid. The feasibility of the usage of such suspensions of solid particles with sizes on the order of 2 millimeters or micrometers was previously investigated by several researchers and the following significant drawbacks were observed (Das and Choi, 2006).

1. The particles settle rapidly, forming a layer on the surface and reducing the heat transfer capacity of the fluid.
2. If the circulation rate of the fluid is increased, sedimentation is reduced, but the erosion of the heat transfer devices, pipelines, etc., increases rapidly.

3. The large size of the particles tends to clog the flow channels, particularly if the cooling channels are narrow.
4. The pressure drop in the fluid increases considerably.
5. Finally, conductivity enhancement based on particle concentration is achieved (i.e., the greater the particle volume fraction is, the greater the enhancement—and greater the problems, as indicated above).

Thus, the route of suspending particles in liquid was a well known but rejected option for heat transfer applications.

However, the emergence of modern materials technology provided the opportunity to produce nanometer-sized particles which are quite different from the parent material in mechanical, thermal, electrical, and optical properties.

1.1. Emergence of nanofluids

The situation changed when Choi and Eastman in Argonne National Laboratory revisited this field with their nanoscale metallic particle and carbon nanotube suspensions (Choi and Eastman (1995); Eastman et al. (1996)). Choi and Eastman have tried to suspend various metal and metal oxides nanoparticles in several different fluids (Choi (1998); Choi et al. (2001); Chon et al. (2005); Chon et al. (2006); Eastman et al. (2001); Eastman et al. (1999); Eastman et al. (2004)) and their results are promising, however, many things remain elusive about these suspensions of nano-structured materials, which have been termed "nanofluids" by Choi and Eastman.

Nanofluid is a new kind of heat transfer medium, containing nanoparticles (1–100 nm) which are uniformly and stably distributed in a base fluid. These distributed nanoparticles, generally a metal or metal oxide greatly enhance the thermal conductivity of the nanofluid, increases conduction and convection coefficients, allowing for more heat transfer

Nanofluids have been considered for applications as advanced heat transfer fluids for almost two decades. However, due to the wide variety and the complexity of the nanofluid systems, no agreement has been achieved on the magnitude of potential benefits of using nanofluids for heat transfer applications. Compared to conventional solid–liquid suspensions for heat transfer intensifications, nanofluids having properly dispersed nanoparticles possess the following advantages:

- High specific surface area and therefore more heat transfer surface between particles and fluids.
- High dispersion stability with predominant Brownian motion of particles.
- Reduced pumping power as compared to pure liquid to achieve equivalent heat transfer intensification.
- Reduced particle clogging as compared to conventional slurries, thus promoting system miniaturization.
- Adjustable properties, including thermal conductivity and surface wettability, by varying particle concentrations to suit different applications.

The first test with nanofluids gave more encouraging features than they were thought to possess. The four unique features observed are listed below (Das and Choi, 2006).

- *Abnormal enhancement of thermal conductivity*. The most important feature observed in nanofluids was an abnormal rise in thermal conductivity, far beyond expectations and much higher than any theory could predict.
- *Stability*. Nanofluids have been reported to be stable over months using a stabilizing agent.
- *Small concentration and Newtonian behavior*. Large enhancement of conductivity was achieved with a very small concentration of particles that completely maintained the Newtonian behavior of the fluid. The rise in viscosity was nominal; hence, pressure drop was increased only marginally.
- *Particles size dependence*. Unlike the situation with microslurries, the enhancement of conductivity was found to depend not only on particle concentration but also on particle size. In general, with decreasing particle size, an increase in enhancement was observed.

The above potentials provided the thrust necessary to begin research in nanofluids, with the expectation that these fluids will play an important role in developing the next generation of cooling technology. The result can be a highly conducting and stable nanofluid with exciting newer applications in the future.

2. Thermo physical properties of nanofluids

Thermo physical properties of the nanofluids are quite essential to predict their heat transfer behavior. It is extremely important in the control for the industrial and energy saving perspectives. There is great industrial interest in nanofluids. Nanoparticles have great potential to improve the thermal transport properties compared to conventional particles fluids suspension, millimetre and micrometer sized particles. In the last decade, nanofluids have gained significant attention due to its enhanced thermal properties.

Experimental studies show that thermal conductivity of nanofluids depends on many factors such as particle volume fraction, particle material, particle size, particle shape, base fluid material, and temperature. Amount and types of additives and the acidity of the nanofluid were also shown to be effective in the thermal conductivity enhancement.

The transport properties of nanofluid: dynamic thermal conductivity and viscosity are not only dependent on volume fraction of nanoparticle, also highly dependent on other parameters such as particle shape, size, mixture combinations and slip mechanisms, surfactant, etc. Studies showed that the thermal conductivity as well as viscosity both increases by use of nanofluid compared to base fluid. So far, various theoretical and experimental studies have been conducted and various correlations have been proposed for thermal conductivity and dynamic viscosity of nanofluids. However, no general correlations have been established due to lack of common understanding on mechanism of nanofluid.

2.1. Thermal conductivity

A wide range of experimental and theoretical studies were conducted in the literature to model thermal conductivity of nanofluids. The existing results were generally based on the definition of the effective thermal conductivity of a two-component mixture. The Maxwell (1881) model was one the first models proposed for solid–liquid mixture with relatively large particles. It was based on the solution of heat conduction equation through a stationary random suspension of spheres. The effective thermal conductivity (Eq.1) is given by

$$k_{eff} = \frac{k_p + 2k_{bf} + 2\varphi\left(k_p - k_{bf}\right)}{k_p + 2k_{bf} - \varphi\left(k_p - k_{bf}\right)} k_{bf} \tag{1}$$

Where k_p is the thermal conductivity of the particles, k_{eff} is the effective thermal conductivity of nanofluid, k_{bf} is the base fluid thermal conductivity, and ϕ is the volume fraction of the suspended particles.

The general trend in the experimental data is that the thermal conductivity of nanofluids increases with decreasing particle size. This trend is theoretically supported by two mechanisms of thermal conductivity enhancement; Brownian motion of nanoparticles and liquid layering around nanoparticles (Ozerinc et al, 2010). However, there is also a significant amount of contradictory data in the literature that indicate decreasing thermal conductivity with decreasing particle size.

Published results illustrated neither agreement about the mechanisms for heat transfer enhancement nor a unified possible explanation regarding the large discrepancies in the results even for the same base fluid and nanoparticles size. There are various models available for the measurement of effective thermal conductivity of nanofluids (Wang and Mujumdar, 2007) but there exists large deviations among them. Currently, there are no theoretical results available in the literature that predicts accurately the thermal conductivity of nanofluids.

2.2. Viscosity

Compared with the experimental studies on thermal conductivity of nanofluids, there are limited rheological studies reported in the literature for viscosity. Different models of viscosity have been used by researchers to model the effective viscosity of nanofluid as a function of volume fraction. Einstein (1956) determined the effective viscosity of a suspension of spherical solids as a function of volume fraction (volume concentration lower than 5%) using the phenomenological hydrodynamic equations (Eq.2). This equation was expressed by

$$\mu_{eff} = \left(1 + 2.5\varphi\right)\mu_{bf} \tag{2}$$

Where μ_{eff} is the effective viscosity of nanofluid, μ_{bf} is the base fluid viscosity, and ϕ is the volume fraction of the suspended particles.

Later, Brinkman (1952) presented a viscosity correlation (Eq.3) that extended Einstein's equation to suspensions with moderate particle volume fraction, typically less than 4%.

$$\mu_{eff} = \mu_{bf} \frac{1}{(1-\phi)^{2.5}} \tag{3}$$

The effect of Brownian motion on the effective viscosity in a suspension of rigid spherical particles was studied by Batchelor (1977). For isotropic structure of suspension, the effective viscosity was given by(Eq.4):

$$\mu_{eff} = \left(1 + 2.5\varphi + 6.2\phi^2\right)\mu_{bf} \tag{4}$$

2.3. Specific heat and density

Using classical formulas derived for a two-phase mixture, the specific heat capacity (Pak and Cho, 1998) and density (Xuan and Roetzel, 2000) of the nanofluid as a function of the particle volume concentration and individual properties can be computed using following equations(Eqs 5,and 6) respectively:

$$\rho_{eff} = (1-\varphi)\rho_{bf} + \phi\rho_p \tag{5}$$

$$\left(\rho C_p\right)_{eff} = (1-\varphi)\left(\rho C_p\right)_{bf} + \varphi\left(\rho C_p\right)_p \tag{6}$$

3. Applications of nanofluids

The novel and advanced concepts of nanofluids offer fascinating heat transfer characteristics compared to conventional heat transfer fluids. There are considerable researches on the superior heat transfer properties of nanofluids especially on thermal conductivity and convective heat transfer. Applications of nanofluids in industries such as heat exchanging devices appear promising with these characteristics. Kostic reported that nanofluids can be used in following specific areas:

- Heat-transfer nanofluids.
- Tribological nanofluids.
- Surfactant and coating nanofluids.
- Chemical nanofluids.
- Process/extraction nanofluids.
- Environmental (pollution cleaning) nanofluids.
- Bio- and pharmaceutical-nanofluids.
- Medical nanofluids (drug delivery and functional tissue–cell interaction).

Nanofluids can be used to cool automobile engines and welding equipment and to cool high heat-flux devices such as high power microwave tubes and high-power laser diode arrays. A nanofluid coolant could flow through tiny passages in MEMS to improve its efficiency. The measurement of nanofluids critical heat flux (CHF) in a forced convection loop is useful for nuclear applications. Nanofluids can effectively be used for a wide variety of industries, ranging from transportation to energy production and in electronics systems like microprocessors, Micro-Electro-Mechanical Systems (MEMS) and in the field of biotechnology. Recently, the number of industrial application potential of nanofluids technology and their focus for specific industrial applications is increasing. This chapter deals the some of the important application of nanofluids in the field of heat transfer.

4. Heat transfer applications

The increases in effective thermal conductivity are important in improving the heat transfer behavior of fluids. A number of other variables also play key roles. For example, the heat transfer coefficient for forced convection in tubes depends on many physical quantities related to the fluid or the geometry of the system through which the fluid is flowing. These quantities include intrinsic properties of the fluid such as its thermal conductivity, specific heat, density, and viscosity, along with extrinsic system parameters such as tube diameter and length and average fluid velocity. Therefore, it is essential to measure the heat transfer performance of nanofluids directly under flow conditions. Researchers have shown that nanofluids have not only better heat conductivity but also greater convective heat transfer capability than that of base fluids. The following section provides the wide usage and effective utilization of nanofluids in heat exchangers as heat transfer fluids.

4.1. Tubular (circular pipe) heat exchangers

Pak and Cho (1998) investigated experimentally the turbulent friction and heat transfer behaviors of dispersed fluids (i.e., ultrafine metallic oxide particles suspended in water) in a circular pipe. Two different metallic oxide particles, γ-alumina (Al_2O_3) and titanium dioxide (TiO_2) with mean diameters of 13 and 27 nm, respectively, were used as suspended particles. In their flow loop, the hydrodynamic entry section and the heat transfer section was made using a seamless, stainless steel tube, of which the inside diameter and the total length were 1.066 crn and 480 crn, respectively. The hydrodynamic entry section was long enough (i.e., x/D = 157) to accomplish fully developed flow at the entrance of the heat transfer test section. They observed that the Nusselt number for the dispersed fluids increased with increasing volume concentration as well as Reynolds number. But at constant average velocity, the convective heat transfer coefficient of the dispersed fluid was 12% smaller than that of pure water.

They proposed a new correlation (Eq.7) for the Nusselt number under their experimental ranges of volume concentration (0-3%), the Reynolds number (10^4 - 10^5), and the Prandtl number (6.54 - 12.33) for the dispersed fluids γ-Al_2O_3 and TiO_2 particles as given below

$$Nu = 0.021 \mathrm{Re}^{0.8} \mathrm{Pr}^{0.5} \tag{7}$$

Xuan and Li (2003) built an experimental rig to study the flow and convective heat transfer feature of the nanofluid flowing in a tube. Their test section was a straight brass tube of the inner diameter of 10 mm and the length of 800 mm. Eight thermocouples were mounted at different places of the heat transfer test section to measure the wall temperatures and other two thermocouples were respectively located at the entrance and exit of the test section to read the bulk temperatures of the nanofluid. They investigated convective heat transfer feature and flow performance of Cu-water nanofluids for the turbulent flow. The suspended nanoparticles remarkably enhance heat transfer process and the nanofluid has larger heat transfer coefficient than that of the original base liquid under the same Reynolds number. They found that at fixed velocities, the heat transfer coefficient of nanofluids containing 2.0 vol% Cu nanoparticles was improved by as much as 40% compared to that of water. The Dittus–Boelter correlation failed to predict the improved experimental heat transfer behavior of nanofluids. The heat transfer feature of a nanofluid increases with the volume fraction of nanoparticles.

They have proposed the following correlation (Eq.8) to correlate the experimental data for the nanofluid. The Nusselt number Nu for the turbulent flow of nanofluids inside a tube are obtained as follows

$$Nu_{nf} = 0.0059(1.0 + 7.6286\varphi^{0.6886} Pe_d^{0.001}) \mathrm{Re}_{nf}^{0.9238} \mathrm{Pr}_{nf}^{0.4} \tag{8}$$

They found good coincidence between the results calculated from this correlation and the experimental ones.

The Peclet number Pe describes the effect of thermal dispersion caused by micro convective and micro diffusion of the suspended nanoparticles. The case $c_2 = 0$ refers to zero thermal dispersion, which namely corresponds to the case of the pure base fluid. The particle Peclet number Pe_d, Re_{nf} and Pr_{nf} in (Eq.9) are defined as

$$\begin{aligned}
&\text{Particle Peclet number } Pe_d = \frac{u_m d_p}{\alpha_{nf}} \quad \text{(i)} \\
&\text{The Reynolds number of the nanofluid } \mathrm{Re}_{nf} = \frac{u_m D}{\upsilon_{nf}} \quad \text{(ii)} \\
&\text{The Prantdl number of the nanofluid } \mathrm{Pr}_{nf} = \frac{\upsilon_{nf}}{\alpha_{nf}} \quad \text{(iii)} \\
&\alpha_{nf} = \frac{k_{nf}}{(\rho c_p)_{nf}} = \frac{k_{nf}}{(1-\varphi)\left(\rho c_p\right)_f + \varphi\left(\rho c_p\right)_d} \quad \text{(iv)}
\end{aligned} \tag{9}$$

The thermal diffusivity of the nanofluid in Eq.8 is defined as Eq 8.iv

They defined the friction factor (Eq.10) as

$$\lambda_{nf} = \frac{\Delta P_{nf} D}{L} \frac{2g}{u_m^2} \tag{10}$$

It should be noted that, correlations developed by Pak and Cho (1998) and Xuan and Li (2003) were of a form similar to that of well known Dittus - Boelter formula. In both the works, the nanofluid was treated as a single phase fluid for the calculation of nanofluid Nusselt number

Wen and Ding (2004) were first to study the laminar entry flow of nanofluids in circular tubes. A straight copper tube with 970 mm length, 4.5 mm inner diameter, and 6.4 mm outer diameter was used as the test section. The whole test section was heated by a silicon rubber flexible heater. Their results showed a substantial increase in the heat transfer coefficient of water-based nanofluids containing γ-Al_2O_3 nanoparticles in the entrance region and a longer entry length is needed for the nanofluids than that for water. They concluded that the enhancement of the convective heat transfer could not be solely attributed to the enhancement of the effective thermal conductivity. Particle migration is proposed to be a reason for the enhancement, which results a non-uniform distribution of thermal conductivity and viscosity field and reduces the thermal boundary layer thickness.

Yang et al., (2005) measured the convective heat transfer coefficients of several nanofluids under laminar flow in a horizontal tube heat exchanger. A small circular tube of inner diameter 0.457 cm, outside diameter of 0.635 cm and length 45.7 cm was used as test section. The whole system was heavily insulated to reduce heat loss. Pipes were wrapped with insulation material, and plastic fittings were attached at both ends of the test area to thermally isolate the connection. The average diameter of the disk-shaped graphite nanoparticles used in this research was about 1 to 2μm, with a thickness of around 20 to 40 nm.

They applied the correlations for the convective heat transfer of the single-phase fluid to predict heat transfer coefficient of a nanofluid system, if the volume fraction of particles is

very low. They used the following correlation (Eq.11) to identify the impact of Reynolds number on the heat transfer coefficient

$$Nu.\mathrm{Pr}^{-1/3}\left(\frac{L}{D}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{-0.14} = 1.86\,\mathrm{Re}^{1/3} \tag{11}$$

Their results indicated that the increase in the heat transfer coefficient of the nanofluids is much less than that predicted from a conventional correlation. Near-wall particle depletion in laminar shear flow is one possible reason for the phenomenon. However, there is a doubt whether this work falls in the category of nanofluids at all because the particle diameter is too large for the particles to be called nanoparticles.

Maiga et al., (2005) presented the numerical study of fully developed turbulent flow of Al_2O_3 - water nanofluid in circular tube at uniform heat flux of 50 W/cm^2. The classical k-ε model was used for turbulence modeling and their study clearly showed that the inclusion of nanoparticles into the base fluids has produced a considerable augmentation of the heat transfer coefficient that clearly increases with an increase of the particle concentration. However, the presence of such particles has also induced drastic effects on the wall shear stress that increases appreciably with the particle loading. Among the mixtures studied, the ethylene glycol γ-Al_2O_3nanofluid appears to offer a better heat transfer enhancement than water– γ-Al_2O_3. The following correlations(Eqs 12 and 13) have been proposed for computing the averaged Nusselt number for the nanofluids considered for both the thermal boundary conditions, valid for $Re \leq 1000$, $6 \leq Pr \leq 7.53$ and $\phi \leq 10\%$

$$Nu_{nf} = 0.086\,\mathrm{Re}_{nf}^{0.55}\,\mathrm{Pr}_{nf}^{0.5} \text{ for constant wall flux} \tag{12}$$

$$Nu_{nf} = 0.28\,\mathrm{Re}_{nf}^{0.35}\,\mathrm{Pr}_{nf}^{0.36} \text{ for constant wall temperature} \tag{13}$$

Maiga et al., (2006) studied the hydrodynamic and thermal behavior of turbulent flow in a tube using Al_2O_3 nanoparticle suspension at various concentrations under the constant heat flux boundary condition. Assuming single-phase model, governing equations were solved by a numerical method of control volume. The following correlation (Eq.14) was proposed to calculate the heat transfer coefficient in terms of the Reynolds and the Prandtl numbers, valid for $10^4 \leq Re \leq 5x10^5$, $6.6 \leq Pr \leq 13.9$ and $0 \leq \phi \leq 10\%$.

$$Nu_{nf} = 0.085\,\mathrm{Re}_{nf}^{0.71}\,\mathrm{Pr}_{nf}^{0.35} \tag{14}$$

Ding et al., (2006) were first to study the laminar entry flow of water-based nanofluids containing multiwalled carbon nanotubes (CNT nanofluids). The experimental system for measuring the convective heat transfer coefficient was similar to the one reported by Wen and Ding (2004). Significant enhancement in the convective heat transfer was observed in relation to pure water as the working fluid. The enhancement depends on the flow condition, CNT concentration and the pH level, and the effect of pH is observed to be small. They stated that the enhancement in convective heat transfer is a function of the axial distance from the inlet of the test section. This enhancement increases first, reaches a maximum, and then decreases with increasing axial distance. For nanofluids containing only 0.5 wt% CNTs, the maximum enhancement in the convection heat transfer coefficient reaches over 350% at Re = 800. Such a high level of enhancement could not be attributed purely to enhanced thermal conductivity. They proposed possible mechanisms such as particle rearrangement, reduction of thermal boundary layer thickness due to the presence of nanotubes, and the very high aspect ratio of CNTs. They also concluded that, the observed large enhancement of the convective heat transfer could not be attributed purely to the enhancement of thermal conduction under the static conditions. Particle re-arrangement, shear induced thermal conduction enhancement, reduction of thermal boundary layer thickness due to the presence of nanoparticles, as well as the very high aspect ratio of CNTs are proposed to be possible mechanisms.

Heriz et al., (2006) investigated laminar flow convective heat transfer through circular tube with constant wall temperature boundary condition for nanofluids containing CuO and

Al_2O_3 oxide nanoparticles in water as base fluid. The experimental apparatus consisting of a test chamber constructed of 1 m annular tube with 6 mm diameter inner copper tube and with 0.5 mm thickness and 32 mm diameter outer stainless steel tube. Nanofluid flows inside the inner tube while saturated steam enters annular section, which creates constant wall temperature boundary condition. The fluid after passing through the test section enters heat exchanger in which water was used as cooling fluid. The experimental results emphasized that the single phase correlation with nanofluids properties (Homogeneous Model) was not able to predict heat transfer coefficient enhancement of nanofluids. The comparison between experimental results obtained for CuO/ water and Al_2O_3 / water nanofluids indicated that heat transfer coefficient ratios for nanofluid to homogeneous model in low concentration were close to each other but by increasing the volume fraction, higher heat transfer enhancement for Al_2O_3/water was observed. They concluded that heat transfer enhancement by nanofluid depends on several factors including increment of thermal conductivity, nanoparticles chaotic movements, fluctuations and interactions.

The flow and heat transfer behavior of aqueous TiO_2 nanofluids flowing through a straight vertical pipe was carried out by He et al., (2007) under both the laminar and turbulent flow conditions. Their experimental system consisted of a flow loop, a heating unit, a cooling unit, and a measuring and control unit. The test section was a vertically oriented straight copper tube with 1834 mm length, 3.97 mm inner diameter, and 6.35 mm outer diameter. The tube was heated by two flexible silicon rubber heaters. There was a thick thermal isolating layer surrounding the heaters to obtain a constant heat flux condition along the test section. Two pressure transducers were installed at the inlet and outlet of the test section to measure the pressure drop. They investigated the effects of nanoparticles concentrations, particle size, and the flow Reynolds number. They reported that, addition of nanoparticles into the base liquid enhanced the thermal conduction and the enhancement increased with increasing particle concentration and decreasing particle size. Their results also showed that the convective heat transfer coefficient increases with nanoparticle concentration in both the laminar and turbulent flow regimes and the effect of particle concentration seems to be more considerable in the turbulent flow regimes for the given flow Reynolds number and particle size. Pressure drop of nanofluids was very close to that of the base liquid for given flow Reynolds number. Predictions of the pressure drop with the conventional theory for the base liquid agree well with the measurements at relatively low Reynolds numbers. Deviation occurs at high Reynolds numbers possibly due to the entrance effect.

Kulkarni et al., (2008) investigated heat transfer and fluid dynamic performance of nanofluids comprised of silicon dioxide (SiO_2) nanoparticles suspended in a 60:40 (% by weight) ethylene glycol and water (EG/water) mixture. The heat transfer test section was a straight copper tube with outside diameter of 4.76 mm, inside diameter of 3.14 mm, and a length of 1 m. The wall temperature was measured by means of six thermocouples mounted on the tube surface along the length. The inlet and outlet temperatures of the nanofluid were measured using two thermowells at the inlet and outlet of the test section. Two plastic fittings at inlet and outlet section of the copper tube provide a thermal barrier to axial heat conduction. The test section was heated electrically by four strip heaters to attain the constant heat flux boundary condition. The test section was insulated by 10 cm of fiber glass to minimize the heat loss from the heat transfer test system to ambient air. A four-pass shell and tube counter flow heat exchanger cools the nanofluids to keep the inlet fluid temperature constant using shop water. The effect of particle diameter (20 nm, 50 nm, 100 nm) on the viscosity of the fluid was investigated. They performed experiments to investigate the convective heat transfer enhancement of nanofluids in the turbulent regime by using the viscosity values measured. They observed increase in heat transfer coefficient due to nanofluids for various volume concentrations and loss in pressure was observed with increasing nanoparticle volume concentration.

Hwang et al., (2009) investigated flow and convective heat transfer characteristics of water-based Al_2O_3 nanofluids flowing through a circular tube of 1.812 mm inner diameter with the constant heat flux in fully developed laminar regime. Water-based Al_2O_3 nanofluids with various volume fractions ranging from 0.01% to 0.3% are manufactured by the two-step method. They also measured physical properties of water-based Al_2O_3 nanofluids such as the viscosity, the density, the thermal conductivity and the heat capacity. They presented that the nanoparticles suspended in water enhance the convective heat transfer coefficient in the thermally fully developed regime, despite low volume fraction between 0.01 and 0.3 vol%. Especially, the heat transfer coefficient of water-based Al_2O_3 nanofluids was increased by 8% at 0.3 vol% under the fixed Reynolds number compared with that of pure water and the enhancement of the heat transfer coefficient is larger than that of the effective thermal conductivity at the same volume concentration. Based on their experimental results, it was shown that the Darcy friction factor of water-based Al_2O_3 nanofluids experimentally measured has a good agreement with theoretical results from the friction factor correlation for the single-phase flow ($f = 64/Re_D$).

Sharma et al., (2009) conducted experiments to evaluate heat transfer coefficient and friction factor for flow in a tube and with twisted tape inserts in the transition range of flow with Al_2O_3 nanofluid. Hydro dynamically and thermally developed heat transfer test section is having 1.5 m long with an L/D ratio of 160. The tube was heated uniformly for a length of 1.5 m by wrapping with two nichrome heaters of 1 kW electrical rating. Their twisted tapes were made from 1 mm thick and 0.018 m width aluminum strip. The two ends of the strip are held on a lathe and subjected to 180° twist by turning the chuck manually and obtained twist ratios of 5, 10 and 15. The results showed considerable enhancement of convective heat transfer with Al_2O_3 nanofluids compared to flow with water. They found that the effect of inclusion of twisted tape in the flow path gives higher heat transfer rates compared to flow in a plain tube. They also observed the equation of Gleninski(1976) applicable in transitional flow range for single-phase fluids exhibited considerable deviation when compared with values obtained with nanofluid. The heat transfer coefficient of nanofluid flowing in a tube with 0.1% volume concentration was 23.7% higher when compared with water at number of 9000.

Heat transfer coefficient and pressure drop with nanofluid were experimentally determined with tapes of different twist ratios and found to deviate with values obtained from equations(Eqs 15 and 16) developed for single-phase flow. The data of Al_2O_3 nanofluid for flow in plain tube and with twisted tape insert is fit to a regression equation with average deviation of 4.0% and standard deviation of 5.0%.

$$Nu = 3.138\times10^{-3}\left(\mathrm{Re}\right)\left(\mathrm{Pr}\right)^{0.6}\left(1.0+H/D\right)^{0.03}\left(1+\varphi\right)^{1.22} \tag{15}$$

$0 < H/D < 15$, $3500 < Re < 8500$, $4.5 < Pr < 5.5$ and $35 < T_b < 40$.

The data of friction factor for flow of fluids a plain tube and with tape insert is also subjected to regression with the assumption that nanofluid behaves as single-phase fluid in the low volume concentration given by

$$f = 172\,\mathrm{Re}^{-0.96}\,(1.0+\varphi)^{2.15}\left(1.0+H/D\right)^{2.15} \tag{16}$$

Valid for water ($\phi = 0$) and nanofluid of $\phi < 0.1$ volume concentration

Yu et al., (2009) measured the heat transfer rates in the turbulent flow of a potential commercially viable nanofluid consisting of a 3.7% volume of 170-nm silicon carbide particles suspended in water. Their test facility was a closed-loop system with major components consisting of a pump with variable speed drive, pre heater, horizontal tube test section, heat exchanger (cooler), and flow meter. The test section itself was a stainless steel circular tube with dimensions of 2.27-mm inside diameter, 4.76-mm outside diameter, and 0.58-m heated length. Heat transfer coefficient increase of 50–60% above the base fluid water was obtained when compared on the basis of constant Reynolds number. This enhancement is 14–32% higher than predicted by a standard single-phase turbulent heat transfer correlation pointing to heat transfer mechanisms that involve particle interactions. The data were well predicted by a correlation modified for Prandtl number dependence although experiments in the present study did not support the postulated mechanisms of Brownian diffusion and thermophoresis. The pumping power penalty of the SiC/water nanofluid was shown to be less than that of an Al_2O_3/water nanofluid of comparable particle concentration. The two nanofluids were compared using a figure of the merit(Eq.17) consisting of the ratio of heat transfer enhancement to pumping power increase.

$$\textit{Figure of merit} = \frac{h_{nanofluid} / h_{base\ fluid}}{Power_{nanofluid} / Power_{base\ fluid}} \tag{17}$$

The merit parameter was 0.8 for the SiC/water nanofluid compared to 0.6 for the Al_2O_3/water nanofluid, which is favorable to the SiC/water nanofluid for applications that are pumping power sensitive.

Torii and Yang (2009) studied the convective heat transfer behavior of aqueous suspensions of nanodiamond particles flowing through a horizontal tube heated under a constant heat flux condition. Their experimental system consisting of a flow loop, a power supply unit, a cooling device, and a flow measuring and control unit. The flow loop includes a pump, a digital flow meter, a reservoir, a collection tank, and a test section. A straight seamless stainless tube with 1000 mm length, 4.0 mm inner diameter, and 4.3 mm outer diameter was used as the test section. The whole test section was heated with the aid of the Joule heating method through an electrode linked to a dc power supply. They reported that (i) significant enhancement of heat transfer performance due to suspension of nanodiamond particles in the circular tube flow was observed in comparison with pure water as the working fluid, (ii) the enhancement was intensified with an increase in the Reynolds number and the nanodiamond concentration, and (iii) substantial amplification of heat transfer performance is not attributed purely to the enhancement of thermal conductivity due to suspension of nanodiamond particles.

Effect of particle size on the convective heat transfer in nanofluid by Anoop et al., (2009) in the developing region of pipe flow with constant heat flux showed that the enhancement in heat transfer coefficient was around 25% whereas for the 150 nm particle based nanofluids it was found to be around 11%. The heated test section was made of copper tube of 1200 mm length and 4.75 ± 0.05 mm inner diameter and the thickness of the tube was around 1.25 mm. Electrically insulated nickel chrome wire was uniformly wound along the length giving a maximum power of 200 W. They found that, with increase in particle concentration and flow rate the average heat transfer coefficient value was increased. They also observed that at the developing region the heat transfer coefficient is more than that at nearly developed region. It was further observed that the nanofluid with 45 nm particles shows higher heat transfer coefficient than that with 150 nm particles. For instance at x/D = 147, for 45 nm particle based nanofluid (4 wt%) with Re = 1550, the enhancement in heat transfer coefficient was around 25% whereas for the 150 nm particle based nanofluids it was found to be around 11%. After conducting sufficient number of experiments, they proposed the following correlation (Eq.18)

$$Nu_x = 4.36 + \left[6.219 \times 10^{-3}\, x_+^{1.1522} \left(1 + \varphi^{0.1533}\right).\exp^{\left(-2.5228 x_+\right)}\right] \left[1 + 0.57825 \left(\frac{d_p}{d_{ref}}\right)^{-0.2183}\right] \tag{18}$$

Where, d_{ref} = 100 nm and x_+ is the dimensionless distance.

Rea et al., (2009) investigated laminar convective heat transfer and viscous pressure loss for alumina–water and zirconia–water nanofluids in a flow loop. The vertical heated test section was a stainless steel tube with an inner diameter (ID) of 4.5 mm, outer diameter (OD) of 6.4 mm, and length of 1.01 m. The test section had eight sheathed and electrically insulated T-type thermocouples soldered onto the outer wall of the tubing along axial locations of 5, 16, 30, 44, 58, 89, 100 cm from inlet of the heated section. Two similar T-type thermocouples were inserted into the flow channel before and after the test section to measure the bulk fluid temperatures. The heat transfer coefficients in the entrance region and in the fully developed region were found to increase by 17% and 27%, respectively, for alumina–water nanofluid at 6 vol % with respect to pure water. The zirconia–water nanofluid heat transfer coefficient increases by approximately 2% in the entrance region and 3% in the fully developed region at 1.32 vol %. The measured pressure loss for the nanofluids was in general much higher than for pure water and in good agreement with the traditional model predictions for laminar flow

Garg et al., (2009) used a straight copper tube of 914.4 mm length, 1.55 mm inner diameter and 3.175 mm outer diameter. The whole section was heated by an AWG 30 nichrome 80 wire wound on the tube. Both ends of the copper tube were connected to well-insulated plastic tubing to insulate the heat transfer section and fluid from axial heat conduction, and to avoid heat losses. The experiments were run under constant heat flux conditions using a current of 0.2 A. The test section was insulated to prevent loss of heat to the surroundings. Four surface-mount thermocouples were mounted on the test section at axial positions of 19 cm, 39.5 cm, 59 cm and 79 cm from the inlet of the section to measure wall temperatures. Additionally, two thermocouples were mounted on individual, unheated, and thermally insulated, short copper tubes located before and after the heat transfer section to measure the fluid bulk temperature at the inlet and outlet of the heat transfer section. De-ionized (DI) water, Gum Arabic (GA) and multi-walled carbon nanotubes (MWCNT) were used to produce aqueous suspensions. The nanotubes procured had a specified average outside diameter of 10–20 nm, length of 0.5–40 lm and purity of 95%. They observed a maximum percentage enhancement of 32% in heat transfer coefficient at Re - 600 ± 100. This percentage enhancement in heat transfer coefficient was found to continuously increase with axial distance. The percentage enhancement in heat transfer coefficient was found to continuously increase with axial distance. The reason behind the phenomenon is explained by the contribution from a considerable increase in thermal conductivity with an increase in bulk temperature with axial distance.

Lai et al., (2009) experimentally investigated the convection heat transfer performance of 20-nm, γ Al_2O_3 water-based nanofluids in a single 1.02-mm inner diameter, and constant heat flux stainless steel tube for laminar flow in both the developing and fully developed regions. Overall experimental results showed that the heat transfer coefficient increases with volume flow rate and nanoparticle volume fraction. In the developing region, the heat transfer coefficient enhancement decreased with increasing axial distance from the test section entrance. These results also showed that the higher the volume fraction, the longer is the thermal entrance length.

Chandrasekar et al., (2010) carried out experimental investigations on convective heat transfer and pressure drop characteristics of Al_2O_3/water nanofluid in the fully developed laminar region of pipe flow with constant heat flux with and without wire coil inserts. Their test loop consisting of a pump, calming section, heated test section, cooling section, a collecting station and a reservoir. Calming section of straight copper tube 800 mm long, 4.85 mm inner diameter, and 6.3 mm outer diameter was used to eliminate the entrance effect and to ensure fully developed laminar flow in the test section. A straight copper tube with 1200 mm length, 4.85 mm inner diameter, and 6.3 mm outer diameter was used as the test section. The test section was first wound with sun mica to isolate it electrically. Then, ceramic beads coated electrical SWG Nichrome heating wire giving a maximum power of 300W was wounded over it. Over the electrical winding, thick insulation consisting of layers of ceramic fiber, asbestos rope, glass wool and another layer of asbestos rope at the outer surface was provided to prevent the radial heat loss. The test section was isolated thermally from its upstream and downstream sections by plastic bushings to minimize the heat loss resulting from axial heat conduction. Two types of wire coil inserts were used which were fabricated using 0.5 mm stainless steel wire having a coil diameter of 4.5 mm and coil pitch ratio (defined as the ratio of pitch of the coil to diameter of tube) of 2 and 3. Dilute 0.1% Al_2O_3/water nanofluid increased the Nusselt number by 12.24% at Re = 2275 compared to that of distilled water. Further enhancements in Nusselt numbers was observed when Al_2O_3/water nanofluid is used with wire coil inserts. Nusselt numbers were increased by 15.91% and 21.53% when Al_2O_3/water nanofluid was used with their two types of wire coil inserts respectively at Re = 2275 compared to those of distilled water.

The Nusselt number and friction factor experimental results have been correlated by the following equations (Eqs 19 and 20).

$$Nu = 0.279\left(\mathrm{Re\,Pr}\right)_{nf}^{0.558}\left(\frac{p}{d}\right)^{-0.447}\left(1+\varphi\right)^{134.65} \tag{19}$$

$$f = 530.8\,\mathrm{Re}_{nf}^{-0.909}\left(\frac{p}{d}\right)^{-1.388}\left(1+\varphi\right)^{-512.26} \tag{20}$$

The regression equation coefficients were assessed with the help of classical the least square method and the correlation is valid for laminar flow with Re < 2300, dilute Al_2O_3/water nanofluid with volume concentration $\phi = 0.1\%$ and wire coil inserts with $2 \le p/d \le 3$. They also found that, when compared to the pressure drop with distilled water, there was no significant increase in pressure drop for the nanofluid.

Amrollahi et al., (2010) measured the convective heat transfer coefficients of water-based functionalized multi walled nano tubes (FMWNT) nanofluid under both laminar and turbulent regimes flowing through a uniformly heated horizontal tube in entrance region. The straight copper tube with 11.42 mm inner diameter and 1 m length was used as the test section. The tube surface is electrically heated by an AC power supply to generate constant heat 800W and was insulated thermally by about 150 mm thick blanket to minimize the heat loss from the tube to the ambient. Five thermocouples were soldered on at different places along the test section to measure the wall temperature of the copper tube and the mean temperature of the fluids at the inlet, and two thermocouples were inserted at the inlet and outlet of the test section. They compared effective parameters to measure the convective heat transfer coefficients for functionalized MWNT suspensions such as Re, mass fraction and temperature altogether in entrance region for the first time. Their experimental results indicated that the convective heat transfer coefficient of these nanofluids increases by up to 33–40% at a concentration of 0.25 wt. % compared with that of pure water in laminar and turbulent flows respectively. Their results also showed that, increasing the nanoparticles concentration does not show much effect on heat transfer enhancement in turbulent regime in the range of concentrations studied. Also the ratio of heat transfer coefficient decreased with increasing Reynolds number. It was observed that the wall temperature of the test tube decreased considerably when the nanofluid flowed in the tube. Furthermore, this coefficient of nanofluids at the entrance of the test tube increases with Reynolds number, contrary to the fully developed laminar region that is constant.

Xie et al., (2010) reported on investigation of the convective heat transfer enhancement of nanofluids as coolants in laminar flows inside a circular copper tube with constant wall temperature. Nanofluids containing Al_2O_3, ZnO, TiO_2, and MgO nanoparticles were prepared with a mixture of 55 vol. % distilled water and 45 vol. % ethylene glycol as base fluid. It was found that the heat transfer behaviors of the nanofluids were highly depended on the volume fraction, average size, species of the suspended nanoparticles and the flow conditions. MgO, Al_2O_3, and ZnO nanofluids exhibited superior enhancements of heat transfer coefficient with the highest enhancement up to 252% at a Reynolds number of 1000 for MgO nanofluid. They also demonstrated that these oxide nanofluids might be promising alternatives for conventional coolants.

Fotukian and Esfahany (2010a) experimentally investigated the CuO/water nanofluid convective heat transfer in turbulent regime inside a tube. The test section was constructed of 1 m annular tube with inner copper tube of 5 mm inner diameter and 0.5 mm thickness and 32mm diameter outer stainless steel tube. Nanofluid flowed inside the inner tube while saturated steam entered annular section. They used dilute nanofluids with nanoparticles volume fractions less than 0.3%. They got excellent agreement between the measured heat transfer coefficients of pure water and the Dittus–Boelter predictions. They observed that heat transfer coefficients for nanofluids were greater than that of water and increasing the nanoparticle concentration showed a very weak effect on heat transfer coefficient. In such low concentrations of nanofluid investigated, the augmentation of heat transfer coefficient could not be attributed to the increase of thermal conductivity. The heat transfer coefficient increased about 25% compared to pure water. They concluded that, increasing nanoparticles concentration does not show much effect on heat transfer enhancement in turbulent regime in their studied range of concentrations. Also, the ratio of convective heat transfer coefficient of nanofluid to that of pure water decreased with increasing Reynolds number. It was also reported that the wall temperature of the test tube decreased considerably when the nanofluid flowed in the tube.

Fotukian and Esfahany (2010b) investigated turbulent convective heat transfer and pressure drop of γ Al_2O_3 /water nanofluid inside a circular tube, the same as described previously. The volume fraction of nanoparticles in base fluid was less than 0.2%. Their results indicated that addition of small amounts of nanoparticles to the base fluid augmented heat transfer remarkably. Increasing the volume fraction of nanoparticles in the range studied did not show much effect on heat transfer enhancement. Their experimental measurements showed that pressure drop for the dilute nanofluid was much greater than that of the base fluid.

Experimental investigations on convective heat transfer and pressure drop characteristics of three different concentration of CuO/water nanofluid was carried out by Suresh et al., (2010) in the fully developed turbulent region of pipe flow with constant heat flux. Experiments were done with a dimpled tube having dimensions of 4.85 mm diameter and 800 mm length. They reported that i) the relative viscosity of nanofluids increase with an increase in concentration of nanoparticles. ii) The thermal conductivity of nanofluid increases nonlinearly with the volume concentration of nanoparticles. iii) The convective heat transfer coefficient increases with increasing Reynolds number and increasing volume concentration in plain tube, and increases further with a dimpled tube. The Nusselt number and friction factor experimental results of nanofluids with dimpled tubes have been correlated by the following expressions (Eqs 21 and 22) using the least squares regression analysis

$$Nu = 0.00105\,\mathrm{Re}^{0.984}\,\mathrm{Pr}^{0.4}\left(1+\varphi\right)^{-80.78}\left(1+\frac{p}{d}\right)^{2.089} \tag{21}$$

$$f = 0.1648\,\mathrm{Re}^{0.97}\left(1+\varphi\right)^{107.89}\left(1+\frac{p}{d}\right)^{-4.463} \tag{22}$$

Pathipakka and Sivashanmugam (2010) numerically estimated the heat transfer behavior of nanofluids in a uniformly heated circular tube fitted with helical inserts in laminar flow. They used Al_2O_3 nanoparticles in water of 0.5%, 1.0% and 1.5% concentrations and helical twist inserts of twist ratios (ratio of length of one twist to diameter of the twist) 2.93, 3.91 and 4.89 for the simulation. Assuming the nanofluid behave as a single phase fluid, they investigated three dimensional steady state heat transfer behavior using Fluent 6.3.26. They concluded that the heat transfer increases with Reynolds number and decrease in twist ratio with maximum for the twist ratio 2.93. The increase in Nusselt number was 5%_31% for helical inserts of different twist ratio and nanofluids of different volume concentrations. The heat transfer enhancement was 31% for helical tape insert of twist ratio 2.93 and Al_2O_3volume concentration of 1.5% corresponding to the Reynolds number of 2039.

Suresh et al., (2011) presented a comparison of thermal performance of helical screw tape inserts in laminar flow of Al_2O_3/water and CuO/water nanofluids through a straight circular duct with constant heat flux boundary condition. Their experimental set up consists of a test section, calming section, pump, cooling unit, and a fluid reservoir. Both the calming section and test sections were made of straight copper tube with the dimension 1000 mm long, 10 mm ID and 12 mm OD. The calming section was used to eliminate the entrance effect. The test section tube was wounded with ceramic beads coated electrical SWG Nichrome heating wire. Over the electrical winding a thick insulation is provided using glass wool to minimize heat loss. They used three types of helical screw tape inserts with various twist ratio (1.78, 2.44, and 3) was made by winding uniformly a copper strip of 3.5 mm width over a 2.5 mm copper rod. The twist ratio 'Y', defined as the ratio of length of one twist (pitch, P) to diameter of the twist.

They used their experimental results of heat transfer to derive the following correlations(Eqs 23 and 24) of Nusselt number using least square method of regression analysis. The correlations are valid for laminar flow (Re < 2300) of 0.1% volume concentration of Al_2O_3/water and CuO/water nanofluids and for helical screw tape inserts of twist ratio ranging from 1.78 to 3.

$$\text{For } Al_2O_3 \text{ / water nanofluid; } Nu = 0.5419\left(\mathrm{Re}\,\mathrm{Pr}\right)^{0.53}\left(\frac{P}{D}\right)^{0.594} \tag{23}$$

$$\text{For CuO / water nanofluid; } Nu = 0.5657\left(\mathrm{Re}\,\mathrm{Pr}\right)^{0.5337}\left(\frac{P}{D}\right)^{0.6062} \tag{24}$$

Their results showed thermal performance factor of helical screw tape inserts using CuO/water nanofluid is found to be higher when compared with the corresponding value using Al_2O_3/water.

The experimental results on convective heat transfer of non-Newtonian nanofluids flowing through a horizontal uniformly heated tube under turbulent flow conditions by Hojjat et al., (2011a) states that convective heat transfer coefficient and Nusselt number of nanofluids are remarkably higher than those of the base fluid. Their experimental setup consists of a flow loop comprised of three sections: cooling unit, measuring and control units. The test section consists of a straight stainless steel (type 316) tube, 2.11-m long, 10-mm inner diameter, and 14-mm outer diameter. The test section was electrically heated by an adjustable DC power supply in order to impose a constant wall heat flux boundary condition. Ten K-type thermocouples were mounted on the tube outside wall to measure the wall temperature at different axial locations. The locations of the thermocouples were placed at the following axial positions from the test section inlet: 100, 150, 200, 350, 550, 800, 1100, 1400, 1700, and 2000 mm. The test section was thermally insulated from the upstream and downstream sections by thick Teflon bushings in order to reduce the heat loss along the axial direction. Two K-type thermocouples were also inserted in the calming chamber and the mixing chamber to measure the inlet and outlet bulk temperatures of the nanofluid, respectively. The whole test section including the calming and mixing chambers were heavily insulated. Three different types of nanofluids were prepared by first dispersing γ-Al_2O_3, TiO_2 and CuO nanoparticles in deionized water. The solution were subjected to ultrasonic vibration to obtain uniform suspensions, and then appropriate amounts of concentrated Carboxy Methyl Cellulose (CMC) solution were added to the suspension and mixed thoroughly with a mechanical mixer to achieve homogeneous nanofluids with the desired concentration. Average sizes of γ-Al_2O_3, TiO_2 and CuO nanoparticles were 25, 10, and 30-50 nm, respectively. Their results showed that Convective heat transfer coefficient and Nusselt number of nanofluids are remarkably higher than those of the base fluid. These enhancements of nanofluids were directly proportional to the particle concentration and Peclet number. Since the enhancement of heat transfer coefficient of nanofluids was much higher than that attributed to the improvement of the thermal conductivity, it was expected that the enhancement of heat transfer coefficient of nanofluids was affected by some other factors. Based on the experimental results, they proposed the following empirical correlation (Eq.25) to predict the heat transfer coefficients of non-Newtonian nanofluids.

$$Nu = 7.135 \times 10^4\, \mathrm{Re}^{0.9545}\, \mathrm{Pr}^{0.913}\left(1 + \varphi^{0.1358}\right) \tag{25}$$

$2800 < \mathrm{Re} < 8400$; $40 < \mathrm{Pr} < 73$.

Hojjat et al., (2011b) experimentally investigated the forced convection heat transfer of non-Newtonian nanofluids in a circular tube with constant wall temperature under turbulent flow conditions. Three types of nanofluids were prepared by dispersing homogeneously γ-Al_2O_3, TiO_2 and CuO nanoparticles into the base fluid. An aqueous solution of carboxymethyl cellulose (CMC) was used as the base fluid. Nanofluids as well as the base fluid show shear-thinning (pseudoplastic) rheological behavior. The test section consists of two 2-m long concentric tubes. The internal diameter of inner tube was 10 mm and a thickness 2 mm. The internal diameter of outer tube was 48 mm. Both tubes were made of stainless steel (type 316). The nanofluid flows through the inner tube whereas hot water was circulated through the annular section at high flow rates in order to create constant wall temperature boundary condition. Results indicated that the convective heat transfer coefficient of nanofluids is higher than that of the base fluid. The enhancement of the convective heat transfer coefficient increases with an increase in the Peclet number and the nanoparticle concentration. The increase in the convective heat transfer coefficient of nanofluids was greater than the increase that would be observed considering strictly the increase in the effective thermal conductivity of nanofluids. Experimental data were compared to heat transfer coefficients predicted using available correlations for purely viscous non-Newtonian fluids. Results showed poor agreement between experimental and predicted values. Hence they proposed a new correlation(Eq.26) to successfully predict Nusselt numbers of non-Newtonian nanofluids as a function of Reynolds and Prandtl numbers.

$$Nu = 0.00115 \mathrm{Re}^{1.050} \mathrm{Pr}^{0.693} (1+\varphi^{0.388}) \tag{26}$$

Mahrood et al., (2011) experimentally investigated free convection heat transfer of non Newtonian nanofluids under constant heat flux condition. Two different kinds of non-Newtonian nanofluids were prepared by dispersion of Al_2O_3 and TiO_2 nanoparticles in a 0.5 wt. % aqueous solution of carboxy methyl cellulose (CMC). Experimental investigation of natural convection heat transfer behavior of non- Newtonian nanofluids in a vertical cylinder was attempted. Test section was a vertical cylindrical enclosure made up of PTFE (Poly Tetra Fluoro Ethylene). Fluid in the test section was heated from below by a heating system which consists of an aluminum circular plate and an electrical heater. In order to achieve a constant wall heat flux, the heater was placed between the aluminum plate and a thick PTFE circular plate. The PTFE plate also acts as insulation. Their results showed that the heat transfer performance of nanofluids is significantly enhanced at low particle concentrations. Increasing nanoparticle concentration has a contrary effect on the heat transfer of nanofluids, so at concentrations greater than 1 vol. % of nanoparticles the heat transfer coefficient of nanofluids is less than that of the base fluid. Indeed it seems that for both nanofluids there exists an optimum nanoparticle concentration that heat transfer coefficient passes through a maximum. The optimum concentrations of Al_2O_3 and TiO_2 nanofluids are about 0.2 and 0.1 vol. %, respectively. It is also observed that the heat transfer enhancement of TiO_2 nanofluids is higher than that of the Al_2O_3 nanofluids. The effect of enclosure aspect ratio was also investigated and the heat transfer coefficient of nanofluids as well as the base fluid increases by increasing the aspect ratio as expected.

Corcione et al., (2012) theoretically studied the heat transfer of nanoparticle suspensions in turbulent pipe flow. Both constant pumping power and constant heat transfer rate have been investigated for different values of the Reynolds number of the base fluid in the range between 2300 and 5×10^6, the diameter of the suspended nanoparticles in the range between 25 nm and 100 nm, the length-to-diameter ratio of the pipe in the range between 50 and 1000, the nanofluid bulk temperature in the range between 303 K and 343 K, as well as for three different nanoparticle materials (i.e., CuO, Al_2O_3, and TiO_2) and two different base liquids (i.e., water and ethylene glycol). The significant findings of their study was the existence of an optimal particle loading for either maximum heat transfer at constant driving power or minimum cost of operation at constant heat transfer rate. In particular, for any assigned combination of solid and liquid phases, they found that the optimal concentration of suspended nanoparticles increases as the nanofluid bulk temperature is increased, the Reynolds number of the base fluid is increased, and the length-to-diameter ratio of the pipe is decreased, while it is practically independent of the nanoparticle diameter.

4.2. Double pipe heat exchanger

Chun et al., (2008) experimentally reported the convective heat transfer of nanofluids made of transformer oil and three kinds of alumina nanoparticles in laminar flow through a double pipe heat exchanger system. The experimental system consisted of two double-pipe heat exchangers for heating and cooling of nanofluid and was made of a non-corrosive stainless steel. Their experimental data showed that the addition of nanoparticles in the fluid increases the average heat transfer coefficient of the system in laminar flow. By non-linear regression of experimental data, the correlation (Eq.27) for heat transfer coefficient was decided as follows

$$h_i = \frac{k}{D} \times 1.7\,\mathrm{Re}^{0.4} \tag{27}$$

The surface properties of nanoparticles, particle loading, and particle shape were key factors for enhancing the heat transfer properties of nanofluids. They stated that these increases of heat transfer coefficients may be caused by the high concentration of nanoparticles in the wall side by the particle migration.

Duangthongsuk and Wongwises (2009) experimentally studied the heat transfer coefficient and friction factor of a nanofluid consisting of water and 0.2 vol. % TiO_2 flowing in a horizontal double-tube counter flow heat exchanger under turbulent flow conditions. Their test section was a 1.5 m long counter flow horizontal double-tube heat exchanger with nanofluid flowing inside the tube while hot water flows in the annular. The inner tube is made from smooth copper tubing with a 9.53 mm outer diameter and an 8.13 mm inner diameter, while the outer tube is made from PVC tubing and has a 33.9 mm outer diameter and a 27.8 mm inner diameter. The test section was thermally isolated from its upstream and downstream section by plastic tubes in order to reduce the heat loss along the axial direction.

They investigated the effects of the flow Reynolds number and the temperature of the nanofluid and the temperature and flow rate of the heating fluid on the heat transfer coefficient and flow characteristics. Their results showed that the convective heat transfer coefficient of nanofluid is slightly higher than that of the base liquid by about 6 -11%. The heat transfer coefficient of the nanofluid increased with an increase in the mass flow rate of the hot water and nanofluid, and increased with a decrease in the nanofluid temperature, and the temperature of the heating fluid had no significant effect on the heat transfer coefficient of the nanofluid. They also concluded that Gnielinski correlation for predicting the heat transfer coefficient of pure fluid is not applicable to a nanofluid. But, the Pak and Cho correlation (Eq. (7)) for predicting the heat transfer coefficient of a nanofluid agreed better with their experimental results than the Xuan and Li correlation (Eq. (8)).

Duangthongsuk and Wongwises (2010) experimentally studied the heat transfer coefficient and friction factor of the TiO_2-water nanofluids flowing in a horizontal double tube counter-flow heat exchanger under turbulent flow conditions. Their test fluid was TiO_2 nanoparticles with diameters of 21 nm dispersed in water with volume concentrations of 0.2 - 2 vol. %. The heat transfer coefficient of nanofluids was approximately 26% greater than that of pure water and the results also showed that the heat transfer coefficient of the nanofluids at a volume concentration of 2.0 vol.% was approximately 14% lower than that of base fluids for given conditions.

Their results showed that the Pak and Cho correlation (Eq. (7)) can predict the heat transfer coefficient of nanofluids and gives results that corresponded well only with the experimental results for the volume concentration of 0.2%. However, for the volume concentrations of 0.6% and 1.0%, the Pak and Cho equation fails to predict the heat transfer performance of the nanofluids. For the pressure drop, their results showed that the pressure drop of nanofluids was slightly higher than the base fluid and increases with increasing the volume concentrations.

New heat transfer and friction factor correlations(Eqs 28 and 29) for predicting the Nusselt number and friction factor of TiO2-water nanofluids were proposed in the form of

$$Nu = 0.074 \mathrm{Re}^{0.707} \mathrm{Pr}^{0.385} \varphi^{0.074} \tag{28}$$

$$f = 0.961 \varphi^{0.052} \mathrm{Re}^{-0.375} \tag{29}$$

The majority of the data falls within ±10% of the proposed equation. These equations are valid in the range of Reynolds number between 3000 and 18,000 and particle volume concentrations in the range of 0 and 1.0 vol. % for Nusselt number and 0 and 2.0 vol. % for friction factor.

Asirvatham et al., (2011) investigated the convective heat transfer of nanofluids using silver – water nanofluids under laminar, transition and turbulent flow regimes in a horizontal 4.3 mm inner-diameter tube-in-tube counter-current heat transfer test section. The volume concentration of the nanoparticles were varied from 0.3% to 0.9% in steps of 0.3% and the effects of thermo-physical properties, inlet temperature, volume concentration, and mass flow rate on heat transfer coefficient were investigated. Experiments showed that the suspended nanoparticles remarkably increased the convective heat transfer coefficient, by as much as 28.7% and 69.3% for 0.3% and 0.9% of silver content, respectively. Based on the experimental results a correlation (Eq.30) was developed to predict the Nusselt number of the silver–water nanofluid, with ±10% agreement between experiments and prediction.

$$Nu_{nf} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.3} + \left(0.617\varphi - 0.135\right)\mathrm{Re}^{\left(0.445\varphi - 0.37\right)}\,\mathrm{Pr}^{\left(1.081\varphi - 1.305\right)} \tag{30}$$

4.3. Plate heat exchanger

Zamzamian et al., (2011) used nanofluids of aluminum oxide and copper oxide in ethylene glycol base fluid. They investigated the effect of forced convective heat transfer coefficient in turbulent flow, using a double pipe and plate heat exchangers. The inner pipe of the double pipe heat exchanger was made of copper, 12 mm in diameter and 1 mm in thickness, with a heat exchange length of 70 cm. The shell was made of green pipes, 50.8 mm in diameter. The flow inside the double pipe heat exchanger was arranged in opposite directions. The plate heat exchanger was a small, particularly manufactured model of common home radiators, 40 cm in height and 60 cm in length, exchanging heat freely with the ambience through four fins. The forced convective heat transfer coefficient of the nanofluids using theoretical correlations also calculated in order to compare the results with the experimental data. The effects of particle concentration and operating temperature on the forced convective heat transfer coefficient of the nanofluids were evaluated. The findings indicated considerable enhancement in convective heat transfer coefficient of the nanofluids as compared to the base fluid, ranging from 2% to 50%. Moreover, the results indicated that with increasing nanoparticles concentration and nanofluid temperature, the convective heat transfer coefficient of nanofluid increases.

4.4. Shell and tube heat exchanger

Farajollahi et al., (2010) measured the heat transfer characteristics of γ Al_2O_3 /water and TiO_2/water nanofluids in a shell and tube heat exchanger under turbulent flow condition. Water was allowed to flow inside the shell with 55.6 mm inside diameter and the nanofluid was passed through the 16 tubes with 6.1 mm outside diameter, 1 mm thickness, and 815 mm length. The tube pitch is 8 mm and the baffle cut and baffle spacing are 25% and 50.8 mm, respectively. The heat exchanger and pipe lines are thermally insulated to reduce heat loss to the surrounding. The effects of Peclet number, volume concentration of suspended nanoparticles, and particle type on heat transfer characteristics were investigated.

The observed the overall heat transfer coefficient of nanofluids increases significantly with Peclet number. For both nanofluids the overall heat transfer coefficient at a constant Peclet number increases with nanoparticle concentration compared to the base fluid. The experimental results for the Nusselt number of γ Al_2O_3/water and TiO_2/water nanofluids were compared with the prediction of Xuan and Li correlation (Eq. (6)). Results show that at

0.5 vol. % of γ Al_2O_3 nanoparticles and at 0.3 vol. % of TiO_2 nanoparticles a good agreement exists between the experimental results and the predicted values by Eq. (6) especially at higher Peclet numbers. They observed that the correlation is almost valid for the prediction of Nusselt number at low volume concentrations.

They reported that, adding of nanoparticles to the base fluid causes the significant enhancement of heat transfer characteristics. They experimentally obtained two different optimum nanoparticle concentrations for both the nanofluids. the heat transfer behavior of two nanofluids were compared and the results indicated that at a certain Peclet number, heat transfer characteristics of TiO_2/water nanofluid at its optimum nanoparticle concentration are greater than those of γ Al_2O_3 /water nanofluid while γ Al_2O_3 /water nanofluid possesses better heat transfer behavior at higher nanoparticle concentrations.

The emergence of several challenging issues such as climate change, fuel price hike and fuel security have become hot topics around the world. Therefore, introducing highly efficient devices and heat recovery systems are necessary to overcome these challenges. It is reported that a high portion of industrial energy is wasted as flue gas from heating plants, boilers, etc. Leong et al., (2012) focused on the application of nanofluids as working fluids in shell and tube heat recovery exchangers in a biomass heating plant. Heat exchanger specification, nanofluid properties and mathematical formulations were taken from the literature to analyze thermal and energy performance of the heat recovery system. It was observed that the convective and overall heat transfer coefficient increased with the application of nanofluids compared to ethylene glycol or water based fluids. In addition, 7.8% of the heat transfer enhancement could be achieved with the addition of 1% copper nanoparticles in ethylene glycol based fluid at a mass flow rate of 26.3 and 116.0 kg/s for flue gas and coolant, respectively.

4.5. Multi channel heat exchanger (MCHE)

Jwo et al., (2010) employed Al_2O_3 /water nanofluid to electronic chip cooling system to evaluate the practicability of its actual performance. Their experimental variables included nanofluids of different weight concentrations (0, 0.5, and 1.0 wt. %) and the inlet water temperature at different flow values. To determine if the addition of nanoparticles has any effects on overall heat transfer performance, they conducted a comparative experiment with water first. The control variables of their study were the mass flow rate, inlet water temperature, and heating power. Having completed the control experiment with water, nanofluids of different concentrations were used to carry out the same experiment. Using the same control variables, the ratio of the overall heat transfer performance of nanofluid to the overall heat transfer performance of water was calculated, and then acquired the overall heat transfer coefficient ratios under different conditions. Based on the collected temperature data for different mass flow rates, electric input powers, and nanofluid concentrations, the overall heat transfer coefficient ratio (r_U) of the MCHE (Eq.31) can be written as follows:

$$r_U = \frac{U_{nanofluid}}{U_{water}} = \frac{\left(T_{wall} - T_m\right)_{water}}{\left(T_{wall} - T_m\right)_{nanofluid}} \tag{31}$$

Where, $T_m = (T_{liq.in} + T_{liq.out})/2$ is the averaged temperature of liquid traversing the MCHE.

Results showed that the overall heat transfer coefficient ratio was higher at higher nanoparticle concentrations. In other words, the overall heat transfer coefficient ratio was higher when the probability of collision between nanoparticles and the wall of the heat exchanger were increased under higher concentration, confirming that nanofluids have considerable potential for use in electronic chip cooling systems. These results confirmed that nanofluid offers higher overall heat transfer performance than water, and a higher concentration of nanoparticles provides even greater enhancement of the overall heat transfer coefficient ratio.

4.6. Radial flow and electronic cooling devices

Gherasim et al., (2009) presented an experimental investigation of heat transfer enhancement capabilities of coolants with suspended nanoparticles (Al_2O_3 dispersed in water) inside a radial flow cooling device. Steady, laminar radial flow of a nanofluid between a heated disk and a flat plate with axial coolant injection has been considered. An experimental test rig was built consisting of the space between the two coaxial disks with central axial injection through the lower, high-temperature resistant PVC disk and the upper disk was machined from aluminum stock piece. They investigated the influence of disk spacing on local Nusselt number and proved that the local Nusselt number increases with a decrease in gap spacing. This behavior is obviously due to the increase of convection effects. They also analyzed the influence of particle volume fraction and Reynolds number on mean Nusselt number and found that the local Nusselt number increases with particle volume fraction. Their results showed that heat transfer enhancements are possible in radial flow cooling systems with the use of nanofluids. In general, it was noticed that the Nusselt number increases with particle volume fraction and Reynolds number and decreases with an increase in disk spacing.

Nguyen et al., (2007) investigated the heat transfer enhancement and behavior of Al_2O_3 nanoparticle - water mixture, for use in a closed cooling system that was destined for microprocessors or other heated electronic components. Their experimental liquid cooling system was a simple closed fluidic circuit which is mainly composed of a 5 l open reservoir and a magnetically driven pump that ensures a forced recirculation of liquid. An electrically heated block (aluminum body) was considered which simulates heat generated by a microprocessor. On top of this heated block, water-block (copper body) was installed. A thin film of high thermal conductivity grease was applied to minimize the thermal contact resistance at the interface junction between the heated block and the water-block. The assembly of heated block and water block has been thermally very well insulated with respect to the surrounding environment by means of fiberglass. Their data showed clearly that the inclusion of nanoparticles into distilled water produced a considerable enhancement of the cooling convective heat transfer coefficient. For a particular particle volume concentration of 6.8%, the heat transfer coefficient was found to increase as much as 40% compared to that of the base fluid. They observed that an increase of particle volume concentration has produced a clear decrease of the heated block temperature. Their experimental results also shown that a nanofluid with 36 nm particle size provides higher convective heat transfer coefficients than the ones given by nanofluid with 47 nm particles.

Gherasim et al., (2011) carried out a numerical investigation for heat transfer enhancement capabilities of coolants with suspended nanoparticles (Al_2O_3 dispersed in water) inside a confined impinging jet cooling device. They considered a steady, laminar radial flow of a nanofluid in an axis-symmetric configuration with axial coolant injection. A single phase fluid approach was adopted to numerically investigate the behavior of nanofluids. Good agreement was found between numerical results and available experimental data. Results indicated that heat transfer enhancement is possible in this application using nanofluids. In general, it was noticed that the mean Nusselt number increases with particle volume fraction and Reynolds number and decreases with an increase in disk spacing.

4.7. Double tube helical heat exchangers

G. Huminic and A. Huminic (2011) numerically studied heat transfer characteristics of double-tube helical heat exchangers using nanofluids under laminar flow conditions. CuO and TiO_2 nanoparticles with diameters of 24 nm dispersed in water with volume concentrations of 0.5–3 vol. % were used as the working fluid. The effect of particle concentration level and the Dean number on the heat transfer characteristics of nanofluids and water are determined. The mass flow rate of the nanofluid from the inner tube was kept and the mass flow rate of the water from the annulus was set at either half, full, or double the value. They showed the variations of the nanofluids and water temperatures, heat transfer rates and heat transfer coefficients along inner and outer tubes.

The effect of the nanoparticle concentration level on the heat transfer enhancement was calculated for different nanofluids and mass flow rate of the water. The results of the CFD analysis were used to estimate of the heat transfer coefficients and of the Dean number.

The numerical heat transfer coefficients of the nanofluid and water and Dean number were computed from the following equations (Eqs 32 and 33)

$$h_{nf} = \frac{q_{ave}}{(T_{nf,in} - T_{nf,out})}, \quad De_{nf} = \mathrm{Re}_{nf}\left(\frac{d_i}{2R_o}\right)^{0.5} \tag{32}$$

$$h_w = \frac{q_{ave}}{\left(T_{w,out} - T_{w,in}\right)}, \quad De_{nf} = \mathrm{Re}_{nf}\left(\frac{D_i - d_o}{2R_o}\right)^{0.5} \tag{33}$$

Where the average heat transfer rate is defined as $q_{ave} = \dfrac{q_{nf} + q_w}{2}$

Their results showed that for 2% CuO nanoparticles in water with the same mass flow rate in inner tube and annulus, the heat transfer rate of the nanofluid was approximately 14%

greater than that of pure water. They also showed that the convective heat transfer coefficients of the nanofluids and water increased with increasing of the mass flow rate and with the Dean number.

5. Conclusion

A detailed description of the state-of-the-art nanofluids research for heat transfer application in several types of heat exchangers is presented in this chapter. It is important to note that preparation of nanofluids is an important step in experiments on nanofluids. Having successfully engineering the nanofluids, the estimation of thermo physical properties of nanofluids captures the attention. Great quanta of attempts have been made to exactly predict them but large amount of variations were found. Research works on convective heat transfer using nanofluids is found to increase exponentially in the last decade. Almost all the works showed that the inclusion of nanoparticles into the base fluids has produced a considerable augmentation of the heat transfer coefficient that clearly increases with an increase of the particle concentration. It was reported by many of the researchers that the increase in the effective thermal conductivity and huge chaotic movement of nanoparticles with increasing particle concentration is mainly responsible for heat transfer enhancement. However, there exists aplenty of controversy and inconsistency among the reported results. The outcome of all heat transfer works using nanofluids showed that our current understanding on nanofluids is still quite limited. There are a number of challenges facing the nanofluids community ranging from formulation, practical application to mechanism understanding. Engineering suitable nanofluids with controlled particle size and morphology for heat transfer applications is still a big challenge. Besides thermal conductivity effect, future research should consider other properties, especially viscosity and wettability, and examine systematically their influence on flow and heat transfer. An in-depth understanding of the interactions between particles, stabilizers, the suspending liquid and the heating surface will be important for applications.

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6. References

[1] S.K Das, S.U.S. Choi, H.E. Patel, Heat Transfer in Nanofluids-A Review, Heat Transf. Eng. 27 (10) (2006) 3-19.

[2] S.U.S. Choi, Enhancing thermal conductivity of fluids with nanoparticles, in: The Proc. 1995 ASME Int. Mech. Eng. Congr. Expo, San Francisco, USA, ASME, FED 231/MD 66, 1995, pp. 99-105.

[3] J.A. Eastman, S. U. S. Choi, S. Li, , L. J. Thompson, , and S. Lee, "Enhanced thermal conductivity through the development of nanofluids. Fall Meeting of the Materials Research Society (MRS), Boston, USA, 1996.

[4] S.U.S. Choi, Nanofluid technology: current status and future research. Korea-U.S. Technical Conference on Strategic Technologies, Vienna, VA, 1998.

[5] S.U.S. Choi, Z.G. Zhang, W. Yu, F.E. Lockwood, E.A. Grulke, Anomalous thermal conductivity enhancement in nanotube suspensions. Appl. Phys. Lett., 79(14) (2001), 2252-2254.

[6] C.H. Chon, K.D. Kihm, S.P. Lee, S.U.S. Choi, "Empirical correlation finding the role of temperature and particle size for nanofluid (Al_2O_3) thermal conductivity enhancement." Appl. Phys. Lett., 87(15), (2005), 153107-1531.

[7] C.H. Chon, S.W. Paik, J.B. Tipton, K.D. Kihm, Evaporation and Dryout of Nanofluid Droplets on a Microheater Array. J. Heat Transfer, 128(8), (2006) 735.

[8] J.A. Eastman., S. U. S. Choi, S. Li, W. Yu , L. J. Thompson, Anomalously increased effective thermal conductivities of ethylene glycol-based nanofluids containing copper nanoparticles." Appl. Phys. Lett., 78(6) (2001), 718-720.

[9] J.C. Maxwell, A Treatise on electricity and magnetism, second ed., Clarendon Press, Oxford, UK, 1881.

[10] S.Özerinç, S.Kakaç, A.G. Yazıcıoğlu, Enhanced Thermal Conductivity of Nanofluids: A State-of-the-Art Review, Microfluid. Nanofluid, **8**(2), (2010), 145-170.

[11] X.Q. Wang, A.S. Mujumdar, Heat transfer characteristics of nanofluids: a review, Int. J. Therm. Sci. 46 (2007) 1-19.

[12] Einstein, Investigation on the theory of brownian motion, Dover, New York, 1956.

[13] H.C. Brinkman, The viscosity of concentrated suspensions and solutions, J Chem Phys, 20 (1952), pp. 571–581.

[14] G. Batchelor, The effect of Brownian motion on the bulk stress in a suspension of spherical particles, J. Fluid Mech. 83 (1977) 97–117.

[15] Kostic, www.kostic.niu.edu/DRnanofluids; 2009 [14.11.2009].

[16] Y. Xuan, W. Roetzel, (2000). Conceptions for heat transfer correlation of nanofluids, International Journal of Heat and Mass Transfer, 43 (2000), 3701-3707

[17] B.C. Pak, Y.I. Cho, Hydrodynamic and heat transfer study of dispersed fluids with submicron metallic oxide particles, Exp. Heat Transf. 11 (1998) 151-170.

[18] Y. Xuan, Q. Li, Investigation on convective heat transfer and flow features of nanofluids, J. Heat Transf. 125 (2003) 151-155.

[19] D. Wen, Y. Ding, Experimental investigation into convective heat transfer of nanofluids at the entrance region under laminar flow conditions, Int. J. Heat Mass Transf 47 (2004) 5181-5188.

[20] Y. Yang, Z.G. Zhang, E.A. Grulke, W.B. Anderson, G. Wu, Heat transfer properties of nanoparticle-in-fluid dispersions (nanofluids) in laminar flow, Int. J. Heat Mass Transf. 48 (2005) 1107-1116.

[21] S.E.B. Maiga, S.J. Palm, C.T. Nguyen, G. Roy, N. Galanis, Heat transfer enhancement by using nanofluids in forced convection flows. Int. J. Heat Fluid Flow, 26 (2005) 530 - 546.

[22] S.E.B. Maiga, C.T. Nguyen, N. Galanis, G. Roy, T. Maré, M. Coqueux, Heat transfer enhancement in turbulent tube flow using Al_2O_3 nanoparticle suspension. Int. J. Numerical Methods Heat Fluid Flow, 16 (2006) 275 - 292.

[23] Y. Ding, H. Alias, D. Wen, R.A. Williams, Heat transfer of aqueous suspensions of carbon nanotubes (CNT nanofluids), Int. J. Heat Mass Transf. 49 (2006) 240 - 250.

[24] S.Z. Heris, S.G. Etemad, M.N. Esfahany, Experimental investigation of oxide nanofluids laminar flow convective heat transfer, Int. Commun. Heat Mass Transf. 33 (2006) 529-535.

[25] Y. He, Y. Jin, H. Chen, Y. Ding, D. Cang, H. Lu, Heat transfer and flow behavior of aqueous suspensions of TiO_2 nanoparticles (nanofluids) flowing upward through a vertical pipe, Int. J. Heat Mass Transf. 50 (2007) 2272-2281.

[26] D.P. Kulkarni, P.K. Namburu, H.E. Bargar, D.K. Das, Convective heat transfer and fluid dynamic characteristics of SiO_2-ethylene glycol/water nanofluid, Heat Transf. Eng. 29 (12) (2008) 1027-1035.

[27] K.S. Hwang, S.P. Jang, S. U. S. Choi, Flow and convective heat transfer characteristics of water-based Al_2O_3 nanofluids in fully developed laminar flow regime, Int. J. Heat Mass Transf. 52 (2009) 193-199.

[28] A. Sharma, S. Chakraborty, Semi-analytical solution of the extended Graetz problem for combined electro osmotically and pressure-driven microchannel flows with step-change in wall temperature, Int. J. Heat Mass Transf. 51 (2008) 4875-4885.

[29] W. Yu, D.M. France, D.S. Smith, D. Singh, E.V. Timofeeva, J.L. Routbort, Heat transfer to a silicon carbide/water nanofluid, Int. J. Heat Mass Transf 52 (2009) 3606-3612.

[30] S.Torii, W.J. Yang, Heat transfer augmentation of aqueous suspensions of nanodiamonds in turbulent pipe flow, J. Heat Transf. 131 (2009) 043203-1 - 043203-5.

[31] K.B. Anoop, T. Sundararajan, S.K. Das, Effect of particle size on the convective heat transfer in nanofluid in the developing region, Int. J. Heat Mass Transf. 52 (2009) 2189-2195.

[32] U. Rea, T. McKrell, L.W. Hu, J. Buongiorno, Laminar convective heat transfer and viscous pressure loss of alumina-water and zirconia-water nanofluids, Int. J. Heat Mass Transf. 52 (2009) 2042-2048.

[33] P. Garg, J. L. Alvarado, C. Marsh, T.A. Carlson, D.A. Kessler, K. Annamalai, An experimental study on the effect of ultrasonication on viscosity and heat transfer performance of multi-wall carbon nanotube-based aqueous nanofluids, Int. J. Heat Mass Transf. 52 (2009) 5090-5101.

[34] W.Y. Lai, S. Vinod, P.E. Phelan, P. Ravi, Convective heat transfer for water based alumina nanofluids in a single 1.02-mm tube, J. Heat Transf. 131(2009) 112401-1 - 112401-9.

[35] M. Chandrasekar, S. Suresh, A. Chandra Bose, Experimental studies on heat transfer and friction factor characteristics of Al_2O_3/water nanofluid in a circular pipe under laminar flow with wire coil inserts, Exp. Therm. Fluid Sci. 34 (2010) 122-130.

[36] A. Amrollahi, A.M. Rashidi, R. Lotfi, M.E. Meibodi, K. Kashefi, Convection heat transfer of functionalized MWNT in aqueous fluids in laminar and turbulent flow at the entrance region, Int. Commun. Heat Mass Transf. 37 (2010) 717- 723.

[37] H. Xie, Y. Li, W. Yu, Intriguingly high convective heat transfer enhancement of nanofluid coolants in laminar flows, Phys Lett. A 374 (2010) 2566-2568.
[38] S.M. Fotukian, M. Nasr Esfahany, Experimental study of turbulent convective heat transfer and pressure drop of dilute CuO/water nanofluid inside a circular tube, Int. Commun. Heat Mass Transf. 37 (2010a) 214–219.
[39] S.M. Fotukian, M. Nasr Esfahany, Experimental investigation of turbulent convective heat transfer of dilute γ -Al_2O_3/water nanofluid inside a circular tube, Int. J. Heat Fluid Flow 31 (2010b) 606-612.
[40] S. Suresh, M. Chandrasekar, S. Chandra sekhar, Experimental Studies on Heat Transfer and Friction Factor Characteristics of CuO/Water Nanofluid under Turbulent Flow in a Helically Dimpled Tube, Exp. Therm. Fluid Sci. 35 (2010) 542-549.
[41] G. Pathipakka, P. Sivashanmugam, Heat transfer behaviour of nanofluids in a uniformly heated circular tube fitted with helical inserts in laminar flow, Superlattices Microstruct. 47 (2010) 349 -360.
[42] S. Suresh, K.P. Venkitaraj, P. Selvakumar, Comparative study on thermal performance of helical screw tape inserts in laminar flow using Al_2O_3/water and CuO/water nanofluids, Superlattices Microstruct. 49 (2011) 608–622.
[43] M. Hojjat, S. Gh. Etemad, R. Bagheri, J. Thibault, Convective heat transfer of non-Newtonian nanofluids through a uniformly heated circular tube, Int. J. Therm. Sci. 50 (2011a) 525-531.
[44] M. Hojjat, S. Gh. Etemad, R. Bagheri, J. Thibault, Turbulent forced convection heat transfer of non-Newtonian nanofluids, Exp. Therm Fluid Sci. 35 (2011b) 1351–1356.
[45] M.R.K. Mahrood, S.G. Etemad, R. Bagheri, Free convection heat transfer of non Newtonian nanofluids under constant heat flux condition, Int. Commun. Heat Mass Transf. 38 (2011) 1449–1454.
[46] M. Corcione, M. Cianfrini, A. Quintino, Heat transfer of nanofluids in turbulent pipe flow, Int. J. Therm. Sci. 56 (2012) 58-69.
[47] B.H. Chun, H.U. Kang, S.H. Kim, Effect of alumina nanoparticles in the fluid on heat transfer in double-pipe heat exchanger system, Korean J. Chem. Eng. 25 (5) (2008) 966-971.
[48] W. Duangthongsuk, S. Wongwises, Heat transfer enhancement and pressure drop characteristics of TiO2–water nanofluid in a double-tube counter flow heat exchanger, Int. J. Heat Mass Transf. 52 (2009) 2059-2067.
[49] W Duangthongsuk, S Wongwises, An experimental study on the heat transfer performance and pressure drop of TiO2-water nanofluids flowing under a turbulent flow regime, Int. J.Heat Mass Transf. 53 (2010) 334-344.
[50] L.G. Asirvatham, B. Raja, D.M. Lal, S. Wongwises, Convective heat transfer of nanofluids with correlations, Particuology 9 (2011) 626– 631.
[51] A. Zamzamian, S.N. Oskouie, A. Doosthoseini, A. Joneidi, M. Pazouki, Experimental investigation of forced convective heat transfer coefficient in nanofluids of Al2O3/EG and CuO/EG in a double pipe and plate heat exchangers under turbulent flow, Exp. Therm Fluid Sci. 35 (2011) 495–502.

[52] B. Farajollahi, S.G. Etemad, M. Hojjat, Heat transfer of nanofluids in a shell and tube heat exchanger, Int. J. Heat Mass Transf. 53 (2010) 12-17.

[53] K.Y. Leong, R. Saidur, T.M.I. Mahlia, Y.H. Yau, Modeling of shell and tube heat recovery exchanger operated with nanofluid based coolants, Int. J. Heat Mass Transf. 55 (2012) 808–816

[54] C.S. Jwo, L.Y. Jeng, T.P. Teng, C.C. Chen, Performance of overall heat transfer in multi-channel heat exchanger by alumina nanofluid, J. of Alloy. Compd. 504 (2010) s385-s388.

[55] Gherasim, G. Roy, C.T. Nguyen, D. Vo-Ngoc, Experimental investigation of nanofluids in confined laminar radial flows, Int. J. Therm. Sci. 48 (2009) 1486-1493.

[56] C.T. Nguyen, G. Roy, C. Gauthier, N. Galanis, Heat transfer enhancement using Al_2O_3-water nanofluid for an electronic liquid cooling system, Appl. Therm. Eng. 27 (2007) 1501-1506.

[57] Gherasim, G. Roy, C.T. Nguyen, D. Vo-Ngoc, Heat transfer enhancement and pumping power in confined radial flows using nanoparticle suspensions (nanofluids), Int. J. Therm. Sci. 50 (2011) 369 - 377

[58] G. Huminic, A. Huminic, Heat transfer characteristics in double tube helical heat exchangers using nanofluids, International Journal of Heat and Mass Transfer 54 (2011) 4280–4287.

[59] V. Gnielinski, New equations for heat and mass transfer in turbulent pipe and channel flow, International Chemical Engineering 16 (1976) 359–368.