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Forced Convective Heat Transfer of Nanofluids in Minichannels

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1. Introduction

Nanofluids are a new class of heat transfer fluids which are engineered by dispersing nanometer-sized metallic or non-metallic solid particles or tubes in conventional heat transfer fluids such as water, ethylene glycol, and engine oil. This is a rapidly emerging interdisciplinary field where nanoscience, nanotechnology, and thermal engineering meet. Since this novel concept of nanofluids was innovated in the mid-last decade (Choi, 1995), this research topic has attracted tremendous interest from researchers worldwide due to their fascinating thermal characteristics and potential applications in numerous important fields such as microelectronics, transportation, and biomedical.

With an ever-increasing thermal load due to smaller features of microelectronic devices and more power output, cooling for maintaining desirable performance and durability of such devices is one of the most important technical issues in many high-tech industries. Although increased heat transfer can be achieved creating turbulence, increasing heat transfer surface area and other way, the heat transfer performance will ultimately be limited due to the low thermal properties of these conventional fluids. If extended heating surface is used to obtain high heat transfer, it also undesirably increases the size of the thermal management system. Thus, these conventional cooling techniques are not suitable to meet the cooling demand of these high-tech industries. There was therefore a need for new and efficient heat transfer liquids that can meet the cooling challenges for advanced devices as well as energy conversion-based applications and the innovation of nanofluids has opened the door to meet those cooling challenges.

In the field of heat transfer, all liquid coolants currently used at low and moderate temperatures exhibit very poor thermal conductivity and heat storage capacity resulting in their poor convective heat transfer performance. Although thermal conductivity of a fluid plays a vital role in the development of energy-efficient heat transfer equipments and other cooling technologies, the traditional heat transfer fluids possess orders-of-magnitude smaller thermal conductivity than metallic or nonmetallic particles. For example, thermal conductivities of water and engine oil are about 5000 times and 21000 times, respectively smaller than that of multi-walled carbon nanotubes (MWCNT) as shown in Table 1 which provides values of thermal conductivities of various commonly used liquids and nanoparticle materials at room temperature. Therefore, the thermal conductivities of fluids

that contain suspended metallic or nonmetallic particles or tubes are expected to be significantly higher than those of traditional heat transfer fluids. With this classical idea and applying nanotechnology to thermal fluids, Steve Choi from Argonne National Laboratory of USA coined the term “nanofluids” to designate a new class of heat transfer fluids (Choi, 1995). From the investigations performed thereafter, nanofluids were found to show considerably higher conductive, boiling, and convective heat transfer performances compared to their base fluids (Murshed et al., 2005, 2006, 2008a, 2008b & 2011; Das et al., 2006, Murshed, 2007; Yu et al., 2008). These nanoparticle suspensions are stable and Newtonian and they are considered as next generation heat transfer fluids which can respond more efficiently to the challenges of great heat loads, higher power engines, brighter optical devices, and micro-electromechanical systems (Das et al., 2006; Murshed et al., 2008a). Although significant progress has been made on nanofluids, variability and controversies in the heat transfer characteristics still exist (Kebblinski et al., 2008; Murshed et al., 2009).

Conventional Fluids and Materials	Thermal Conductivity (W/m ·K)	Reference
Deionized water (DIW)	0.607	Kaviany, 2002
Ethylene glycol	0.254	Kaviany, 2002
Engine oil	0.145	Kaviany, 2002
Fe ₃ O ₄	7.2	Slack,1962
TiO ₂	8.4	Masuda et al., 1993
CuO	13.5	Lide, 2007
Al ₂ O ₃	40	Slack,1962
Al	237	Lide, 2007
Cu	401	Lide, 2007
Ag	429	Lide, 2007
MWCNT	3000	Kim et al., 2001

Table 1. Thermal conductivities of commonly used liquids and materials at room temperature

As the heat transfer performance of heat exchangers or cooling devices is vital in numerous industries, the impact of nanofluids technology is expected to be great. For example, the transport industry has a need to reduce the size and weight of vehicle thermal management systems and nanofluids can increase thermal transport of coolants and lubricants. When the nanoparticles are properly dispersed, nanofluids can offer numerous benefits besides their anomalously high thermal conductivity. These benefits include improved heat transfer and stability, microchannel cooling without clogging, miniaturized systems and reduction in pumping power. The better stability of nanofluids will prevent rapid settling and reduce clogging in the walls of heat transfer devices. The high thermal conductivity of nanofluids translates into higher energy efficiency, better performance, and lower operating costs. They can reduce energy consumption for pumping heat transfer fluids. Miniaturized systems require smaller inventories of fluids where nanofluids can be used. In vehicles, smaller components result in better gasoline mileage, fuel savings, lower emissions, and a cleaner

environment (Murshed et al., 2008a). In addition, because heat transfer takes place at the surface of the particles, it is desirable to use particles with larger surface area. The much larger relative surface areas of nanoparticles compared to micro-particles, provide significantly improved heat transfer capabilities. Particles finer than 20 nm carry 20% of their atoms on their surface, making them instantaneously available for thermal interaction (Choi et al., 2004). Fig. 1 demonstrates that nanoparticles are much better than microparticles in applications (Murshed, 2007). With dispersed nanoparticles, nanofluids can flow smoothly through mini- or micro-channels. Because the nanoparticles are small, they weigh less and chances of sedimentation are also less making nanofluids more stable. With the aforementioned highly desirable thermal properties and potential benefits, nanofluids are considered to have a wide range of industrial and medical applications such as transportation, micromechanics and instrumentation, heating-ventilating and air-conditioning systems, and drug delivery systems. Details of the enhanced thermophysical properties, potential benefits and applications of nanofluids can be found elsewhere (Choi et al., 2004; Das et al., 2006; Yu et al., 2008; Murshed, 2007; Murshed et al., 2008a). As of today, researchers have mostly focused on anomalous thermal conductivity of nanofluids. However, comparatively few research efforts have been devoted to investigate the flow and convective heat transfer features of nanofluids which are very important in order to exploit their potential benefits and applications.

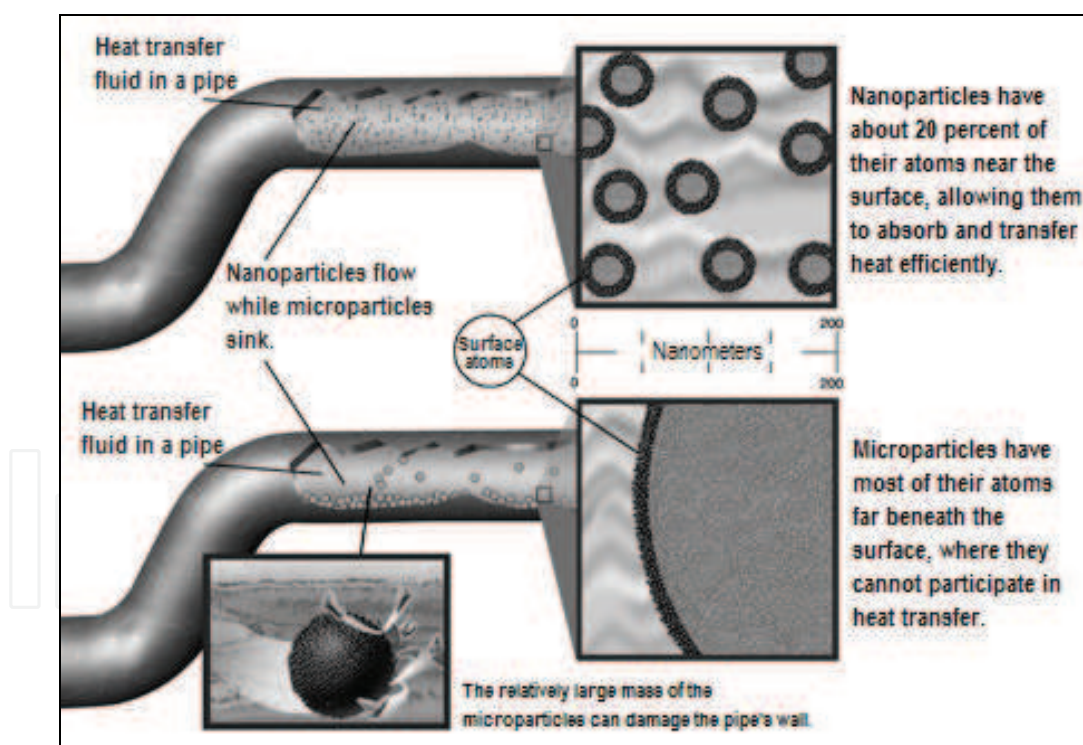


Fig. 1. Comparison between nanoparticles and microparticles flowing in channel

The aim of this chapter is to analyze experimental findings on forced convective heat transfer with nanofluids from literature together with representative results from our experimental investigation on heat transfer characteristics of aqueous TiO_2 -nanofluids flowing through a cylindrical minichannel. Effects of Reynolds number and concentration of nanoparticles on the heat transfer performance are also reported and discussed.

2. Literature survey on convective heat transfer with nanofluids

As mentioned before compared to the studies on thermal conductivity, efforts to investigate convective heat transfer of nanofluids are still scarce. For example, according to ISI Web of Knowledge searched results, only 222 convective heat transfer-related publications out of 1363 recorded publications on nanofluids appeared (publications including journal and conference papers, patent, news and editorial and searched by topic “nanofluids” and refined by topic “convective heat transfer” on 12 April 2011). However, the practical applications of nanofluids as advanced heat transfer fluids or coolants are mainly in flowing systems such as mini- or micro-channels heat sinks and miniaturized heat exchangers. A brief review of forced convective studies (experimental and theoretical) with nanofluids is presented in this section.

The first experiment on convective heat transfer of nanofluids ($\gamma\text{-Al}_2\text{O}_3/\text{water}$ and $\text{TiO}_2/\text{water}$) under turbulent flow conditions was performed by (Pak & Cho, 1998). In their study, even though the Nusselt number (Nu) was found to increase with increasing nanoparticle volume fraction (ϕ) and Reynolds number (Re), the heat transfer coefficient (h) of nanofluids with 3 volume % loading of nanoparticles was 12% smaller than that of pure water at constant average fluid velocity condition. Whereas, (Eastman et al., 1999) later showed that with less than 1 volume % of CuO nanoparticles, the convective heat transfer coefficient of water increased more than 15%. The results of (Xuan & Li, 2003) illustrated that the Nusselt number of Cu/water -based nanofluids increased significantly with the volumetric loading of particles and for 2 volume % of nanoparticles, the Nusselt number increased by about 60%. Wen and Ding investigated the heat transfer behavior of nanofluids at the tube entrance region under laminar flow conditions and showed that the local heat transfer coefficient varied with particle volume fraction and Reynolds number (Wen & Ding, 2004). They also observed that the enhancement is particularly significant at the entrance region. Later (Heris et al., 2006) studied convective heat transfer of CuO and $\text{Al}_2\text{O}_3/\text{water}$ -based nanofluids under laminar flow conditions through an annular tube. Their results showed that heat transfer coefficient increases with particle volume fraction as well as Peclet number. In their study, $\text{Al}_2\text{O}_3/\text{water}$ -based nanofluids found to have larger enhancement of heat transfer coefficient compared to CuO/water -based nanofluids.

An experimental investigation on the forced convective heat transfer and flow characteristics of TiO_2 -water nanofluids under turbulent flow conditions is reported by (Daungthongsuk & Wongwises, 2009). A horizontal double-tube counter flow heat exchanger is used in their study. They observed a slightly higher (6–11%) heat transfer coefficient for nanofluid compared to pure water. The heat transfer coefficient increases with increasing mass flow rate of the hot water as well as nanofluid. They also claimed that the use of the nanofluid has a little penalty in pressure drop.

In microchannel flow of nanofluids, the first convective heat transfer experiments with aqueous CNT-nanofluid in a channel with hydraulic diameter of $355\text{ }\mu\text{m}$ at Reynolds numbers between 2 to 17 was conducted by (Faulkner et al., 2004). They found considerable enhancement in heat transfer coefficient of this nanofluid at CNT concentration of 4.4%. Later, a study was performed on heat transfer performance of $\text{Al}_2\text{O}_3/\text{water}$ -based nanofluid in a rectangular microchannel under laminar flow condition by (Jung et al., 2006). Results showed that the heat transfer coefficient increased by more than 32% for 1.8 volume% of nanoparticles and the Nu increases with increasing Re in the flow regime of $5 < Re < 300$.

An up-to-date overview of the published experimental results on the forced convective heat transfer characteristics of nanofluids is given in Table 2. A comparison of results of Nusselt number versus Reynolds number for both laminar and turbulent flow conditions from various groups is also shown in Fig. 2. Both Table 2 and Fig. 2 demonstrate that the results from various groups are not consistent.

Nanofluids	Geometry/Flow Nature	Findings	Reference
Al ₂ O ₃ /water and TiO ₂ /water	tube/turbulent	At 3 vol. %, h was 12% smaller than pure water for a given fluid velocity.	Pak & Cho, 1998
Cu /water	tube/turbulent	A larger increase in h with ϕ and Re was observed.	Xuan & Li, 2003
Al ₂ O ₃ /water	tube/laminar	h increases with ϕ and Reynolds number.	Wen & Ding, 2004
CNT/water	tube/laminar	At 0.5 wt. %, h increased by 350% at $Re = 800$.	Ding et al., 2006
Al ₂ O ₃ /water and CuO /water	tube/laminar	h increases with ϕ and Pe . Al ₂ O ₃ shows higher increment than CuO.	Heris et al., 2006
Al ₂ O ₃ /DIW	tube/laminar	Nu increased 8 % for $\phi = 0.01$ and $Re = 270$.	Lai et al., 2006
Al ₂ O ₃ /water	rectangular microchannel/ laminar	h increased 15 % for $\phi = 0.018$.	Jung et al., 2006
Al ₂ O ₃ and ZrO ₂ /water	tube/turbulent	h increased significantly.	Williams et al., 2008
Al ₂ O ₃ /water	tube/laminar	h increased only up to 8% at $Re = 730$ for $\phi = 0.003$.	Hwang et al., 2009
Al ₂ O ₃ ,ZnO, TiO ₂ and MgO/water	tube/laminar	h increased up to 252% at $Re = 1000$ for MgO/water.	Xie et al., 2010
MWCNT/water	tube/laminar and turbulent	h increased up to 40% at concentration of 0.25 wt.%.	Amrollahi et al., 2010

Table 2. Summary of forced convection heat transfer experimental studies of nanofluids

Although a growing number researchers have recently shown interest in flow features of nanofluids (Murshed et al., 2011), there is not much progress made on the development of rigorous theoretical models for the convective heat transfer of nanofluids. Researchers investigating convective heat transfer of nanofluids mainly employed existing conventional single-phase fluid correlations such as those attributed to Dittus-Boelter and Shah (Bejan, 2004) to predict the heat transfer coefficient. Some researchers also proposed new correlations obtained by fitting their limited experimental data (Pak & Cho, 1998; Xuan & Li, 2003; Jung et al., 2006). However, none of these correlations were validated with wide range

of experimental data under various conditions and thus are not widely accepted. In an attempt to establish a clear explanation of the reported anomalously enhanced convective heat transfer coefficient of nanofluids, (Buongiorno, 2006) considered seven-slip mechanisms and concluded that among those seven only Brownian diffusion and thermophoresis are the two most important particle/fluid slip mechanisms in nanofluids. Besides proposing a new correlation, he also claimed that the enhanced laminar flow convective heat transfer can be attributed to a reduction of viscosity within and consequent thinning of the laminar sublayer.

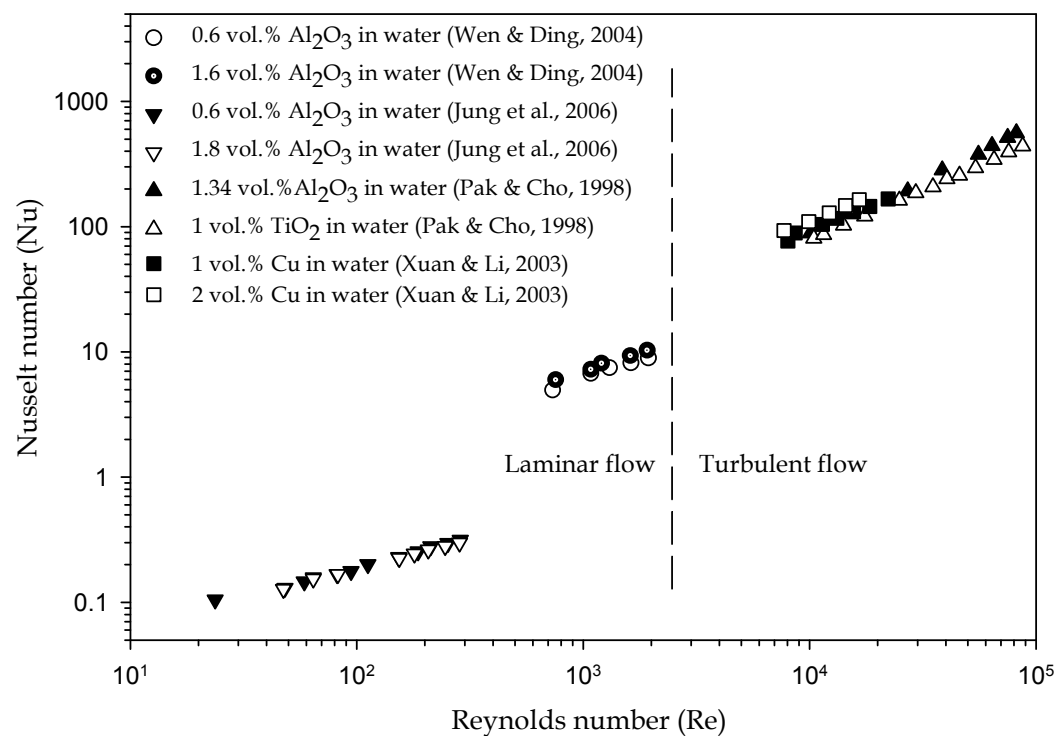


Fig. 2. Convective heat transfer data of nanofluids from various research groups

3. Present laminar flow heat transfer study with nanofluids

The forced convective heat transfer of TiO₂/DIW-based nanofluids flowing through a minichannel under laminar flow and constant heat flux conditions was experimentally studied (Murshed et al., 2008c) and some representative results on nanoparticles concentration and Reynolds number dependence of the convective heat transfer features of this nanofluid are presented together with the exposition on experimental and theoretical bases.

3.1 Experimental

Sample nanofluids were prepared by dispersing different volume percentages (from 0.2% to 0.8%) of TiO₂ nanoparticles of 15 nm diameter in deionized water. To ensure proper dispersion of nanoparticles, sample nanofluids were homogenized by using an ultrasonic dismembrator and a magnetic stirrer. Cetyl Trimethyl Ammonium Bromide (CTAB) surfactant of 0.1mM concentration was added to nanofluid as a dispersant agent to ensure better dispersion of nanoparticles.

An experimental setup was established to conduct experiments on convective heat transfer of nanofluids at laminar flow regime in a cylindrical minichannel. The schematic of experimental facilities used is shown in Fig. 3. The experimental facility consisted of a flow loop, a heating unit, a cooling system, and a measuring and control unit. The flow loop consisted of a pump, a test section, a flow meter, dampener, and a reservoir. The measuring and control unit includes a HP data logger with bench link data acquisition software and a computer. A straight copper tube of 360 mm length, 4 mm inner diameter (D_i) and 6 mm outer diameter (D_o) was used as flowing channel. A peristaltic pump (Cole-Parmer International, USA) with variable speed and flow rate was employed to maintain different flow rates for the desired Reynolds number. In order to minimize the vibration and to ensure steady flow, a flow dampener was also installed between the pump and flow meter. An electric micro-coil heater was used to obtain a constant wall heat-flux boundary condition. Voltmeter and ammeter were connected to the loop to measure the voltage and current, respectively. The heater was connected with the adjustable AC power supply. In order to minimize the heat loss, the entire test section was thermally insulated. Five K-type thermocouples were mounted on the test section at various axial positions from the inlet of the test section to measure the wall temperature distribution. Two thermocouples were further mounted at the inlet and exit of the channel to measure the bulk fluid temperature. A tank with running cold water was used as the cooling system and test fluid was run through the copper coils inside the tank before exiting. During the experiment, the pump flow rate, voltage, and current of the power supply were recorded and the temperature readings from the thermocouples were registered by the data acquisition system. By making use of these temperature readings and supplied heat flux into appropriate expressions, the heat transfer coefficients (h and Nu) were then calculated. Details of the experimental facilities and procedures are reported elsewhere (Murshed et al., 2008c) and will not be elaborated further. Instead formulations for obtaining the heat transfer coefficient of the sample fluids are provided in the subsequent section.

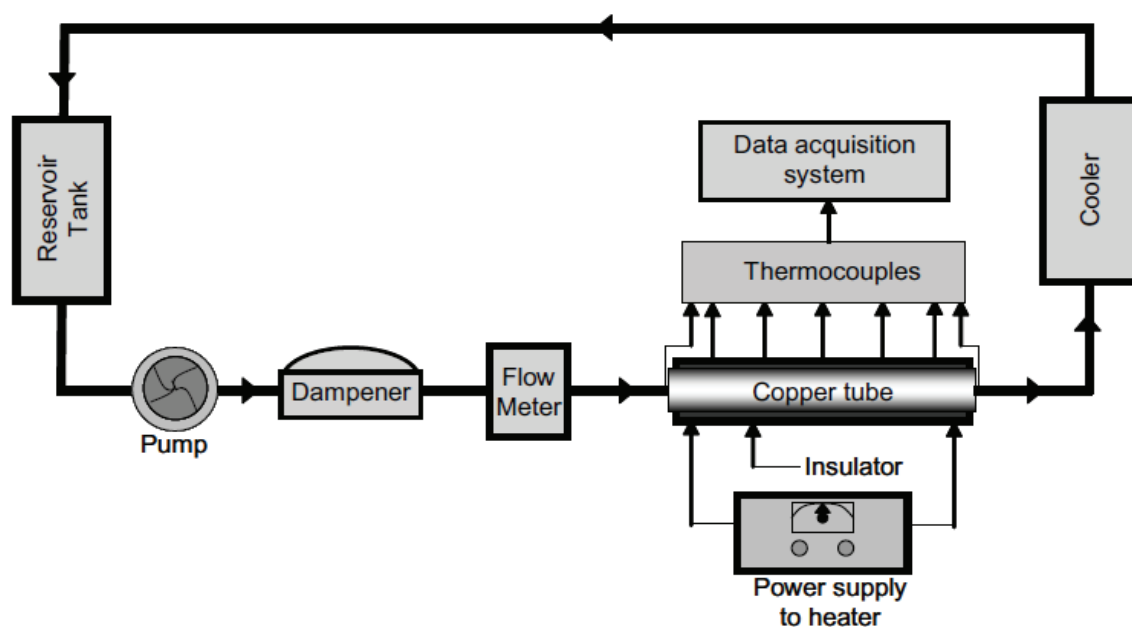


Fig. 3. Schematic of convective heat transfer experimental setup

3.2 Theoretical

The heat transfer feature of sample fluids was determined in terms of the following local convective heat transfer coefficient:

$$h_x = \frac{q''}{T_{i,w}(x) - T_m(x)} \quad (1)$$

where h_x is the local heat transfer coefficient (W/m²K) of sample fluids, $q'' = \dot{m}c_p(T_{out} - T_{in}) / (\pi D_i L)$ is the heat flux (W/m²) of the heat transfer test section, D_i is the inner diameter of the tube (also hydrodynamic diameter), L is the length of the test section, $\dot{m} (= \rho u A_c)$ is the mass flow rate (kg/s), c_p is the specific heat of the fluid, T_{in} and T_{out} are the inlet and outlet fluid temperature, respectively and $T_{i,w}(x)$ and $T_m(x)$ are the inner wall temperature of the tube and the mean bulk fluid temperature at axial position x , respectively.

Since the inner wall temperature of the tube, $T_{i,w}(x)$ could not be measured directly for an electrically heated tube, it can be determined from the heat conduction equation in the cylindrical coordinates as given (Pak et al., 1991)

$$T_{i,w}(x) = T_{o,w}(x) - \frac{q[2D_o^2 \ln(D_o / D_i) - (D_o^2 - D_i^2)]}{4\pi(D_o^2 - D_i^2)k_s x} \quad (2)$$

where $T_{o,w}(x)$ is the outer wall temperature of the tube (measured by thermocouples), q is the heat supplied to the test section (W), k_s is the thermal conductivity of the tube i.e. copper tube, D_o is the outer diameter of the tube, x represents the longitudinal location of the section of interest from the entrance.

The mean bulk fluid temperature, $T_m(x)$ at the section of interest can be determined from an equation based on energy balance in any section of the tube for constant surface heat flux condition. From the first law (energy balance) for the control volume of length, dx of the tube with incompressible liquid and for negligible pressure, the differential heat equation for the control volume can be written as

$$dq_{conv} = q''pdx = \dot{m}c_p dT_m \quad (3)$$

where perimeter of the cross section, $p = \pi D_i$ and dT_m is the differential mean temperature of the fluid in that section. Rearranging equation (3)

$$dT_m = \frac{q''\pi D_i}{\dot{m}c_p} dx \quad (4)$$

The variation of T_m with respect to x is determined by integrating equation (4) from $x = 0$ to x and after simplifying using q'' term and $T_m(x = 0) = T_{in}$, we have

$$T_m(x) = T_{in} + \frac{(T_{out} - T_{in})}{L} x \quad (5)$$

Applying equation (2) and equation (5) into equation (1), the following expression for the local heat transfer coefficient of flowing fluid is obtained

$$h_x = \frac{q''}{\{T_{o,w}(x) - \frac{q[2D_o^2 \ln(D_o/D_i) - (D_o^2 - D_i^2)]}{4\pi(D_o^2 - D_i^2)k_s x}\} - \{T_{in} + \frac{(T_{out} - T_{in})}{L}x\}} \quad (6)$$

Once the local heat transfer coefficient is determined from equation (6) and the thermal conductivity of the medium is known, the local Nusselt number is calculated from

$$Nu_x = \frac{h_x D_i}{k_f} \quad (7)$$

where k_f is the thermal conductivity of fluids. The classical Hamilton-Crosser model is used for the determination of effective thermal conductivity of nanofluids (k_{nf}) which is given by (Hamilton & Crosser, 1962)

$$k_{nf} = k_f \left[\frac{k_p + (n-1)k_f - (n-1)\phi(k_f - k_p)}{k_p + (n-1)k_f + \phi(k_f - k_p)} \right] \quad (8)$$

where k_f and k_p are the thermal conductivities of the base liquid and the nanoparticles, respectively, ϕ is the volume fraction of nanoparticles and n is the empirical shape factor, which has a value of 3 for spherical particle.

The Nusselt number can also be determined from the existing correlations. The well-known Shah's correlation for laminar flows under the constant heat flux boundary conditions is used and reproduced as (Bejan, 2004)

$$Nu = 1.953 \left(\text{RePr} \frac{D_i}{x} \right)^{1/3} \quad \text{for} \quad \left(\text{RePr} \frac{D_i}{x} \right) \geq 33.3 \quad (9)$$

This correlation is popular and commonly used for thermal entrance region.

For steady and incompressible flow of nanofluids in a tube of uniform cross-sectional area, the Reynolds number (Re) and Prandtl number (Pr) are defined as follows

$$\text{Re} = \frac{4\dot{m}}{\pi D_i \mu_{nf}} \quad \text{and} \quad \text{Pr} = \frac{c_{p-nf} \mu_{nf}}{k_{nf}} \quad (10)$$

where μ_{nf} and c_{p-nf} are the viscosity and specific heat of nanofluids, respectively.

The specific heat of nanofluids is calculated using the following volume fractioned-based mixture rule (Pak & Cho, 1998; Jung et al., 2006)

$$c_{p-nf} = \phi c_{p-p} + (1 - \phi) c_{p-f} \quad (11)$$

The viscosity of nanofluids is determined from Batchelor's model given by (Batchelor, 1977)

$$\mu_{nf} = \mu_f (1 + 2.5\phi + 6.2\phi^2) \quad (12)$$

where μ_f is the base fluid viscosity. It is noted that other classical models for calculating the viscosity of mixture also yield similar results (Murshed, 2007; Murshed et al., 2008b).

3.3 Results and discussion

Fig. 4 demonstrates the local heat transfer coefficient of DIW for various concentrations of TiO_2 nanoparticles against the axial distance from the entrance of the test section at Reynolds numbers of 1100 and 1700 (Murshed et al., 2008c). The results show that nanofluids exhibit considerably enhanced convective heat transfer coefficient which also increases with volumetric loadings of TiO_2 nanoparticles. For example, at 0.8 volume % of nanoparticles and at position $x/D_i = 25$ (where tube diameter $D_i = 4$ mm), the local heat transfer coefficient of this nanofluid was found to be about 12% and 14% higher compared to deionized water at Re of 1100 and 1700, respectively. The enhancement in heat transfer coefficients of nanofluids with particle loading is believed to be because of the enhanced effective thermal conductivity and the acceleration of the energy exchange process in the fluid resulting from the random movements of the nanoparticles. Another reason for such enhancement can be the migration of nanoparticles in base fluids due to shear action, viscosity gradient and Brownian motion in the cross section of the tube. For higher Reynolds number ($Re = 1700$), the heat transfer coefficients (h) of nanofluids of all concentrations showed almost linear decrease along the axial distance from the channel entrance (Fig.4b) while at lower Reynolds number (e.g., $Re = 1100$) clear non-linear trends of decreasing the heat transfer coefficients with axial distance are observed (Fig.4a). The reasons for such paradoxical trends of heat transfer coefficients are not clear at this stage.

Fig. 5 compares experimentally determined Nusselt numbers with the predictions by classical Shah's correlation i.e., equation (9) along the axial distance for DIW at Reynolds numbers of 1100 and 1700 (Murshed et al., 2008c). It is noted that in order to calculate Nusselt numbers at various axial positions by Shah's correlation the values of viscosity and thermal conductivity of DIW at mean temperature between the inlet and outlet (i.e., $T_m = (T_{out} + T_{in})/2$) were used. Although Shah's correlation slightly over-predicts the present results, both the experimental and the prediction data of Nusselt number as a function of axial distance show quite similar trends (Fig. 5). The difference in tube size may be one of the reasons for such over prediction. A relatively small tube was used in our experiment, whereas the Shah's equation was developed on the basis of laminar flow in large channel (Bejan, 2004). Nevertheless, for pure water and at about the same Reynolds numbers ($Re = 1050$ and 1600), similar over prediction of Nusselt number by Shah's equation was also reported by (Wen and Ding, 2004).

The effect of Reynolds number on Nusselt number at a specific axial location ($x/D_i = 25$) is shown in Fig. 6 (Murshed et al., 2008c). It can be seen that the measured Nusselt numbers for nanofluids are higher than those of DIW and they increase remarkably and non-linearly with Reynolds number. Again, this trend of increasing Nusselt number with increasing Reynolds number is similar to that observed by (Wen and Ding, 2004) from a similar experimental study with Al_2O_3 /water nanofluids. It was also found that the enhancement in heat transfer coefficient was particularly significant at the entrance region. The observed enhancement of the Nusselt number could be due to the suppression of the boundary layer, viscosity of nanofluids as well as dispersion of the nanoparticles. As expected, the Nusselt number of this nanofluid also increases with the particle concentration (Fig. 6).

Fig. 7 shows that at $x/D_i = 25$ and $Re = 1100$ the Nusselt number of this nanofluid increases almost linearly with the particle volume fraction (Murshed et al., 2008c). This is not surprising as the higher the particle concentration in base fluids, the larger the number

density of nanoparticles and this can intensify the particle migration as well as Brownian force related mechanisms in the base fluids contributing to increase heat transfer rate.

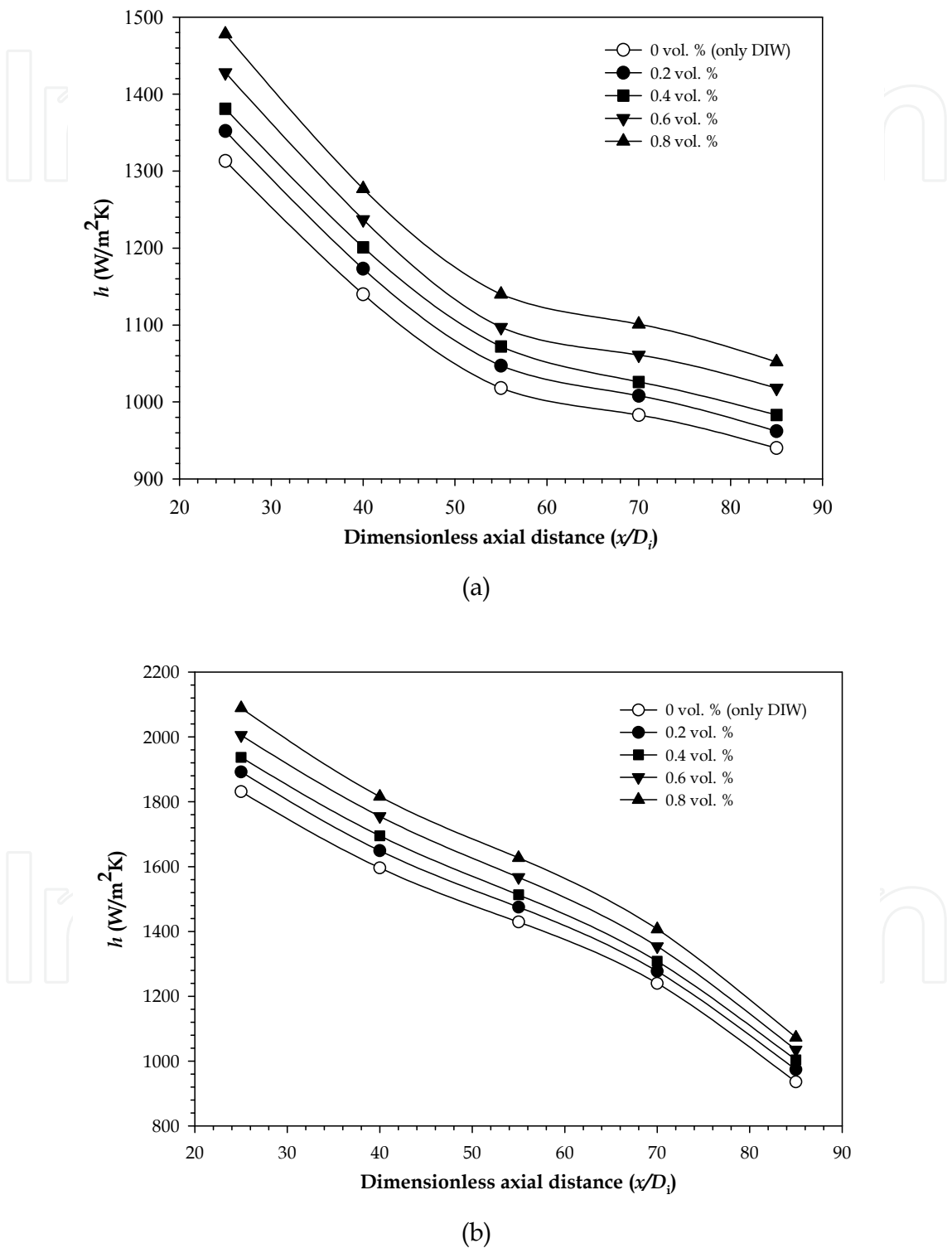


Fig. 4. Axial profiles of local heat transfer coefficient of DIW for various volumetric loadings of TiO_2 nanoparticles at (a) $Re = 1700$ and (b) $Re = 1700$

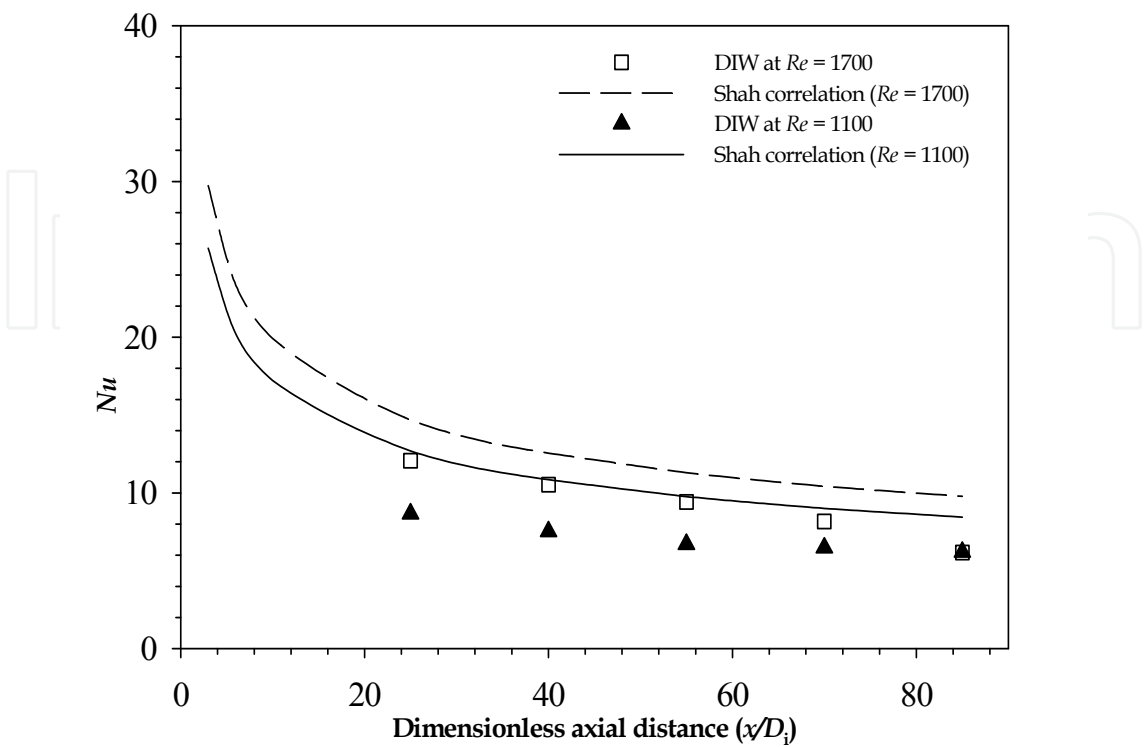


Fig. 5. Comparison with Shah’s correlation along axial distance at $Re = 1100$ and 1700

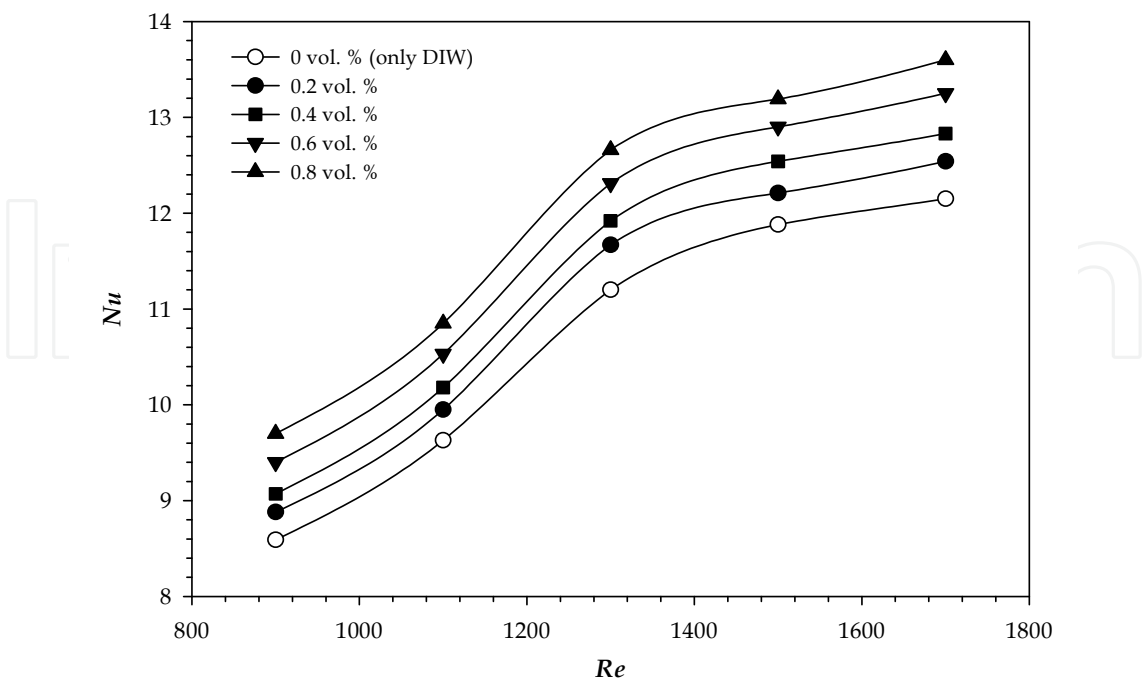


Fig. 6. Effect of Reynolds number and TiO_2 nanoparticle concentrations on Nusselt number

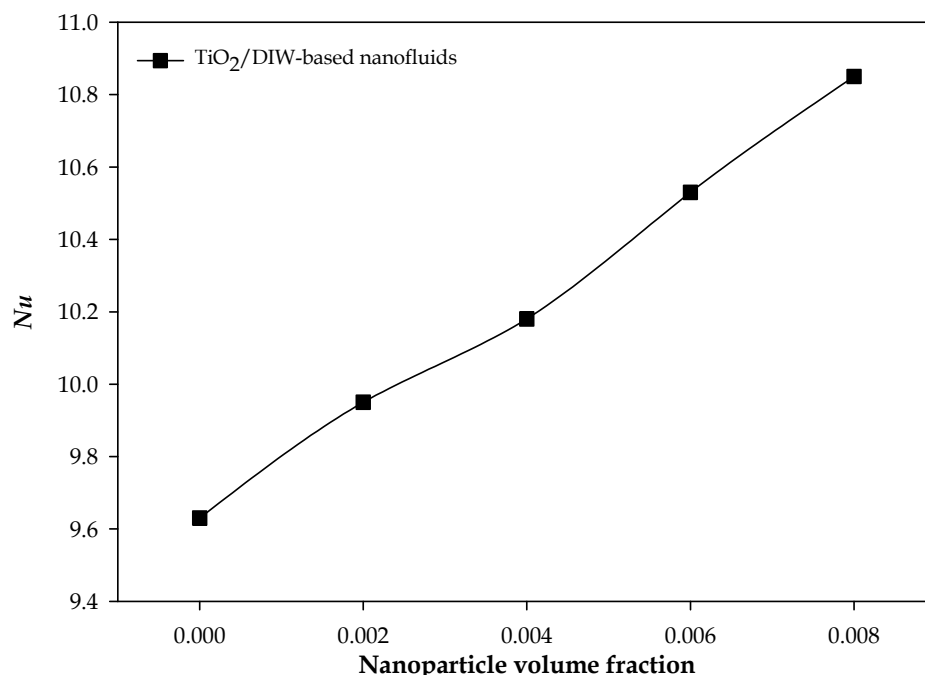


Fig. 7. Nanoparticle concentration-dependent Nusselt number of nanofluids at $Re = 1100$

4. Conclusions

The review of the experimental studies on forced convective heat transfer with nanofluids together with the representative results obtained from our investigation on laminar flow heat transfer features of a nanofluid are presented in this chapter.

Results demonstrate that like most of the literature data, aqueous- TiO_2 nanofluids exhibit enhanced heat transfer coefficient compared to its base fluid. The heat transfer coefficient found to increase significantly with increasing Reynolds number i.e., flow rate. It also shows a strong and linear dependent on the concentration of nanoparticles.

The review showed a considerable chaos and randomness in the reported data on convective heat transfer coefficient from various research groups. Despite of such chaotic and inconsistent data, the applications of nanofluids as advanced heat transfer fluids appear to be promising due to their overall enhanced heat transfer performance in flowing as well as static conditions. However, the advancement toward concrete understanding the mechanisms for the observed heat transfer features of nanofluids remain challenging. There is also not much progress made on the development of rigorous theoretical models for convective heat transfer of nanofluids. Therefore, more careful and systematic investigations on nanofluids preparation and measurements of their heat transfer features as well as rigorous theoretical analysis are needed.

5. Acknowledgments

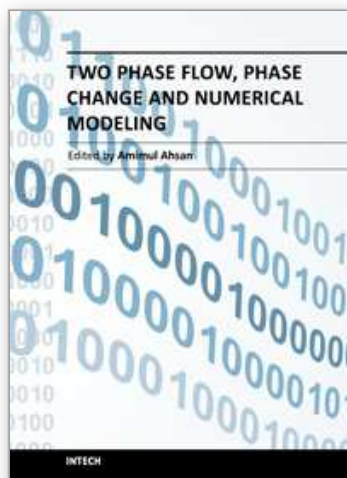
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The heat transfer and analysis on laser beam, evaporator coils, shell-and-tube condenser, two phase flow, nanofluids, complex fluids, and on phase change are significant issues in a design of wide range of industrial processes and devices. This book includes 25 advanced and revised contributions, and it covers mainly (1) numerical modeling of heat transfer, (2) two phase flow, (3) nanofluids, and (4) phase change. The first section introduces numerical modeling of heat transfer on particles in binary gas-solid fluidization bed, solidification phenomena, thermal approaches to laser damage, and temperature and velocity distribution. The second section covers density wave instability phenomena, gas and spray-water quenching, spray cooling, wettability effect, liquid film thickness, and thermosyphon loop. The third section includes nanofluids for heat transfer, nanofluids in minichannels, potential and engineering strategies on nanofluids, and heat transfer at nanoscale. The forth section presents time-dependent melting and deformation processes of phase change material (PCM), thermal energy storage tanks using PCM, phase change in deep CO₂ injector, and thermal storage device of solar hot water system. The advanced idea and information described here will be fruitful for the readers to find a sustainable solution in an industrialized society.

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