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CONVECTIVE HEAT TRANSFER OF NANOFLUIDS IN MINICHANNEL UNDER LAMINAR FLOW CONDITIONS

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ABSTRACT

This paper reports an experimental investigation on force convective heat transfer of nanofluids flowing through a cylindrical minichannel under laminar flow and constant wall heat-flux conditions. Sample nanofluids were prepared by dispersing different volume percentage (i.e. 0.2% - 0.8%) of nanoparticles in deionized water. The present results showed that both the convective heat transfer coefficient and the Nusselt number of this nanofluid increase considerably with nanoparticle volume fraction and Reynolds number. The migration and Brownian motion of nanoparticles and the resulting disturbance of the boundary layer could be the reason for such enhancements of heat transfer coefficient of nanofluids.

Keywords: Nanofluids, Convective Heat Transfer, Laminar Flow, Minichannel.

1. INTRODUCTION

Over the last several decades, scientists and engineers have been attempting to develop fluids, which can offer better cooling or heating performance for variety of thermal systems compared to conventional heat transfer fluids. Applying nanotechnology to thermal engineering, the novel concept of a "*nanofluids*" which was coined at Argonne National Laboratory of USA by Choi in 1995 [1] has been proposed as a means of meeting these cooling challenges. This new class of heat transfer fluids i.e. nanofluids is engineered by dispersing nanometer-sized (one billionth of a meter) solid particles in traditional heat transfer fluids such as water, ethylene glycol, or engine oil. From the investigations in the past decade, nanofluids was found to exhibit significantly higher thermal properties, particularly effective thermal conductivity [2-8] and effective thermal diffusivity [9] compared to their base fluids. Aiming at the potential application of nanofluids in advanced cooling techniques for more efficient cooling of electronics and microelectromechanical systems (MEMS), studies on convective heat transfer of nanofluids are of great interest. However, compared to the reported research efforts on the effective thermal conductivity, only a handful were performed for the convective heat transfer of nanofluids. Since the concept of nanofluids is relatively new and its heat transfer characteristics is not well understood by the researchers, a review of reported studies particularly on force convection of nanofluids will be discussed here.

The first experiment on convective heat transfer of nanofluids (e.g. γ-Al_2O_3/water) under turbulent flow conditions was performed by Pak and Cho [10]. In their study, even though the Nusselt number (Nu) was found to increase with particle volume fraction and the Reynolds number, the heat transfer coefficient (h) actually decreased by 3-12%. On the other hand, Eastman *et al*. [11] showed that with less than 1 volume % of CuO nanoparticles, the convective heat transfer coefficient of water under dynamic flow conditions was increased more than 15%. The experimental results of Xuan and Li [12] illustrated that the convective heat transfer coefficient of Cu/water-based nanofluids varied significantly with the flow velocity and the volumetric loading of particles. For example, for 2 volume % of Cu nanoparticles in water, the Nusselt number increased by about 60% and the Dittus-Boelter correlation [13] was unable to predict such enhanced Nusselt number.

Wen and Ding [14] reported the convective heat transfer behaviour of nanofluids at the tube entrance region under laminar flow conditions. The experiments were conducted for $600 < Re < 2200$. Their results illustrated that the local heat transfer coefficient varied with particle volume fraction, ϕ and Reynolds number, Re. For the case of $\phi = 0.016$ and dimensionless axial distance $x/D \approx 63$ from the entrance, the local h was 41% higher for Re = 1050, and 47% higher for Re = 1600, compared with the results for pure water. They observed that the enhancement is particularly significant in the entrance region and decreased with axial distance. Recently, the same research group [15] investigated the convective heat transfer of CNT-based nanofluids in laminar flow and constant wall heat flux conditions. Surprisingly, the maximum increase in the convective heat transfer coefficient was found to be more than 350% at Re = 800 and at 0.5 weight % of CNT.

Heris *et al.* [16] investigated convective heat transfer of CuO and Al_2O_3/water-based nanofluids under laminar flow conditions through an annular copper tube. Their results showed an enhanced heat transfer coefficient, which increased with an increasing particle volume fraction as well as Peclet number. Al_2O_3/water-based nanofluids showed higher enhancement of heat transfer coefficient compared to CuO/water-based nanofluids.

Recently, Jung *et al.* [17] conducted heat transfer experiments for Al_2O_3/water-based nanofluid in a rectangular microchannel (50 μm × 50 μm) under laminar flow condition. The convective heat transfer coefficient increased by more than 32% with 1.8 volume% of nanoparticles. The Nusselt number increases with increasing Reynolds number in the laminar flow regime (5 >*Re*<300). Based on their experimental results, they proposed a new convective heat transfer correlation for nanofluids in microchannels.

The above review demonstrated that the results reported by various groups vary widely and most of the studies lack physical explanation for their observed results. There is therefore, a need for more research efforts on convective heat transfer of nanofluids. In this study, convective heat transfer of nanofluids in laminar flow condition is presented and the observed results are analyzed.

2. EXPERIMENTAL SETUP AND PROCEDURE

An experimental setup was established to conduct force convective heat transfer of nanofluids at laminar flow regime in a minichannel. 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 and a reservoir. The measuring and control unit includes a data acquisition HP data logger with bench link data acquisition software and a personal computer. The schematic of the experimental facility is shown in Fig.1.

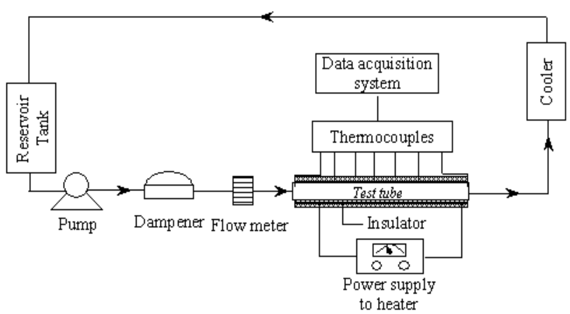


Fig 1: Schematic of experimental setup

In this study, a straight copper tube of 360 mm length, 4 mm inner diameter and 6 mm outer diameter was used as flowing channel. A peristaltic pump from Cole-Parmer International, USA with variable speed of 6 to 600 RPM and flow rate ranging from 0.36 to 3400 mL/min was employed to maintain different flow rate for the required Reynolds number. To minimize the vibration and to ensure steady flow, a flow dampener was also used between the pump and flow meter. An electric micro-coil heater (Chong Mei Heater Co. Ltd, Singapore) was used to obtain a constant wall heat-flux boundary condition. Voltmeter and ammeter were connected to the loop to measure voltage and current respectively. The heater (3.5kW maximum capacity) is connected with the adjustable AC power supply, which has a maximum power of 240V. In order to minimize the heat loss, the entire test section is thermally insulated. The hydrodynamic entry section is long enough to accomplish fully developed flow at the heat transfer test section. Five K-type thermocouples were mounted on the test section at axial positions of 100 mm (T_{w1}), 160 mm (T_{w2}), 220 mm (T_{w3}), 280 mm (T_{w4}), and 340 mm (T_{w5}) 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 copper tube to measure the bulk temperature of the nanofluids. A tank with running water was used as the cooling system and test fluid was run through the copper coils before exiting the water tank.

During the experiment, the pump flow rate (for Reynolds number), voltage and current of the power supply were recorded and the temperature readings from the thermocouples were registered by a data acquisition system. By making use of these temperature readings and supplied heat flux into appropriate expressions (see following section *Data Processing*), the convective heat transfer coefficients (*h* and *Nu*) were calculated.

3. DATA PROCESSING

The cooling heat transfer performance of nanofluids was defined in terms of the following local convective heat transfer coefficient:

$$h_{nf-x} = \frac{q''}{T_{i,w}(x) - T_m(x)} \tag{1}$$

where h_{nf-x} is the local heat transfer coefficient of nanofluids (W/m^{2}K), $q'' = \frac{\dot{m} c_p (T_{out} - T_{in})}{\pi D_i L}$ is the heat flux of the heat transfer test section, D_i is the inner diameter of the tube (also hydrodynamic diameter), $T_{i,w}$ (x) is the inner wall temperature of the tube, and T_m(x) is the mean bulk fluid temperature at axial position *x*.

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 given as [18]

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

where $T_{o,w}$ (x) is the outer wall temperature of the tube (measurable), *q* is the heat supplied to the test section (W), k_s is the thermal conductivity of the tube i.e. copper tube in this study, 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 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, we can write

$$dq_{conv} = q''pdx = \dot{m}c_p dT_m \tag{3}$$

where perimeter of the cross section, $p = \pi D_i$, $\dot{m}\ (= \rho u A_c)$ is the mass flow rate (kg/s), c_p is the specific heat of the fluid, and dT_m is the differential mean temperature of the fluid in that section.
Rearranging Eq. (3)

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

The variation of T_m with respect to x is determined by integrating from $x = 0$ to x and simplifying, we have

$$T_m(x) = T_{m,inlet} + \frac{q''\pi D_i}{\dot{m}c_p} x \tag{5}$$

Substituting Eqs. (2) and (5) into Eq. (1), the local heat transfer coefficient can be obtained from

$$h_{nf-x} = \frac{q''}{\{T_{o,w}(x) - \frac{q[2D_o^2 \ln(D_o/D_i) - (D_o^2 - D_i^2)]}{4\pi(D_o^2 - D_i^2)k_s x}\} - \{T_{m,inlet} + \frac{q''\pi D_i}{\dot{m}c_p} x\}} \tag{6}$$

By using the measured wall and fluid temperatures along with the heat flux into Eq. (6), the local heat transfer coefficient is determined.

Once the local heat transfer coefficient is determined and the thermal conductivity of the medium is known, a local Nusselt number is calculated from

$$Nu_{nf-x} = \frac{h_{nf-x} D_i}{k_{nf}} \tag{7}$$

where k_{nf} is the effective thermal conductivity of nanofluids. The classical Hamilton-Crosser model [19], is used for the determination of the effective thermal conductivity of nanofluids which is given by

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

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 given by $n = 3/\psi$ and ψ is the sphericity. For the spherical and cylindrical shape particle, the sphericity (ψ) is 1 and 0.5, respectively.

For laminar flow under constant heat flux boundary conditions, Nusselt number can also be determined from the existing correlations. The well-known Shah's correlation [20] for laminar flows under the constant heat flux boundary conditions is given as

$$Nu = 1.953\left(Re\,Pr\frac{D}{x}\right)^{1/3} \text{ for } \left(Re\,Pr\frac{D}{x}\right) \geq 33.3 \tag{9}$$

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

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

where $\dot{m}$ is the mass flow rate and μ_{nf}, c_{p-nf} and k_{nf} are the viscosity, specific heat and thermal conductivity of nanofluids, respectively.

The specific heat of nanofluids is calculated using the following volume fraction mixture rule

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

The viscosity of nanofluids can be obtained from Batchelor's [21] model given by

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

where ϕ is particle volume fraction and μ_f is the base fluid viscosity.

Experimentally determined Nusselt numbers are compared with the predictions by Shah's correlation.

4. SAMPLE PREPARATION

Nanofluid is not just a simple mixture of liquid and solid nanoparticles. Proper mixing and stabilization of the nanoparticles are essential to determine their heat transfer coefficients. The properties and behaviour of a suspension depend on the liquid, suspended particle size as well as the quality of dispersion of the particles in the liquid. For this study, sample nanofluids were prepared by dispersing different volume % (i.e. 0.2% - 0.8%) of titanium dioxide (TiO_2) nanoparticles (spherical shaped and 15 nm diameter) in deionized water (DIW). To ensure proper dispersion of nanoparticles, the sample nanofluid was homogenized by using an ultrasonic dismembrator (Fisher Scientific Model 500) and a magnetic stirrer. Cetyl Trimethyl Ammonium Bromide (CTAB) surfactant of 0.1mM concentration was used as a dispersant agent.

5. CALIBRATION AND UNCERTAINTY ANALYSIS

The significant errors that could influence the accuracy of the experimental data can be classified into two groups: systematic error and random error. The systematic errors are minimized with careful experimentation and the calibration operations. The precision error is the random component of the total error, which can be treated statistically.

Calibration of the thermocouples was done using a calibration bath. Before measuring the heat transfer coefficient of nanofluids, the experimental system was calibrated with based fluids i.e. deionised water.

The experimental uncertainties are estimated by guidelines described by Moffat [22]. The uncertainty for the measurements of heat transfer coefficient and Nusselt number can be estimated from the following equations, respectively

$$\frac{\delta h}{h} = \sqrt{(\frac{\delta T}{T})^2 + (\frac{\delta q''}{q''})^2 + (\frac{\delta \dot{m}}{\dot{m}})^2} \qquad (13a)$$

$$\frac{\delta Nu}{Nu} = \sqrt{(\frac{\delta T}{T})^2 + (\frac{\delta q''}{q''})^2 + (\frac{\delta \dot{m}}{\dot{m}})^2 + (\frac{\delta D}{D})^2 + (\frac{\delta k}{k})^2} \qquad (13b)$$

The thermocouples were calibrated and their accuracy of temperature readings was within $\pm$ 0.5K. The accuracy of the heat flux generated by a heater connected to power source was less than 2%. The pump performance was calibrated by a simple timed weighting method. The accuracy of the flow rate measured by a flow meter was within 0.2%. The accuracy of tube diameter (D) was less than 0.1%. Since the effective thermal conductivity of nanofluids was calculated from the classical Hamilton-Crosser model [Eq. (8)], its uncertainty was ignored.

From equation (13) the uncertainty of the measured heat transfer coefficient (h) and Nusselt number (Nu) were estimated to be $\pm$ 2.1% and $\pm$ 2.2%, respectively.

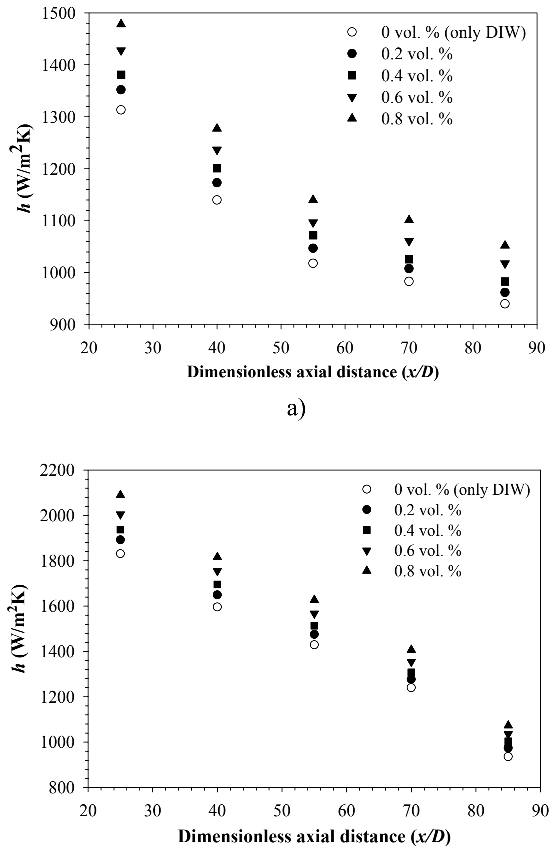
6. RESULTS AND DISCUSSION

The effects of axial position, nanoparticle concentration and Reynolds number on the heat transfer characteristics of TiO_2/deionized water based-nanofluids are investigated. The range of Reynolds numbers was 900 to 1700.

6.1 Axial Profiles of the Convective Heat Transfer Coefficient

Figures 2(a) and 2(b) illustrate the local heat transfer coefficient against the axial distance from the entrance of the test section at two Reynolds numbers, Re = 1100 and Re = 1700. The results clearly show that nanofluids considerably exhibits improved convective heat transfer coefficient and it increases with volumetric loading of nanoparticles. For example, nanofluids containing 0.8 volume % of nanoparticles and at position x/D = 25, the local heat transfer coefficients were found to be about 12% and 14% for Re = 1100 and 1700, respectively higher than those of deionized water (Fig. 2). These enhanced heat transfer coefficients of nanofluids could be because of the enhanced effective thermal conductivity and the acceleration of the energy exchange process in the fluid due to the random movements of the nanoparticles. Another reason for such enhanced convective heat transfer capability of nanofluids could be the migration of nanoparticles in base fluids due to shear action, viscosity gradient and Brownian motion in the cross section of the tube [14].

Figure 3 depicts the comparison between results by Shah's correlation i.e. Eq. (9) and measured Nusselt numbers along the axial distance. It can be seen that Shah's correlation slightly over-predicts the present results. The difference in tube size may be one of the reasons for such over prediction. A relatively small tube (4 mm diameter) was used for in this experiment, whereas the



a)

(b)

Fig 2: Axial profiles of local heat transfer coefficient- (a) Re = 1100 and (b) Re = 1700

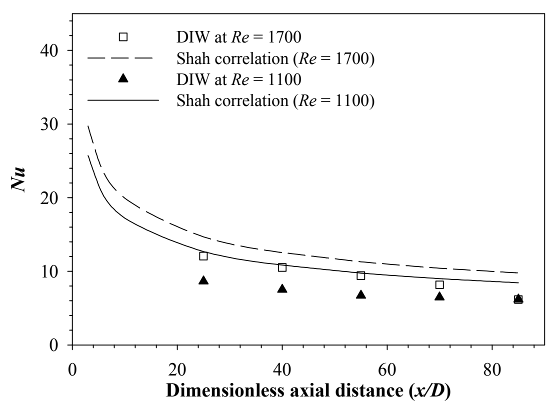


Fig 3 :Comparison with Shah's correlation along axial distance at Re = 1100 and Re = 1700

Shah's equation was developed on the basis of laminar flows in large channels. Nevertheless, similar over prediction by Shah's equation was also observed by Wen and Ding [14].

6.2 Effect of Reynolds Number on Nusselt Number

The effect of Reynolds number on Nusselt number is shown in Fig. 4. It is seen that the measured Nusselt numbers for nanofluids are higher than those of water and they increase remarkably with Reynolds number. The reasons of heat transfer enhancement (i.e. *Nu*) of the nanofluids could be due to the intensification of eddy, suppression of the boundary layer as well as dispersion of the nanoparticles

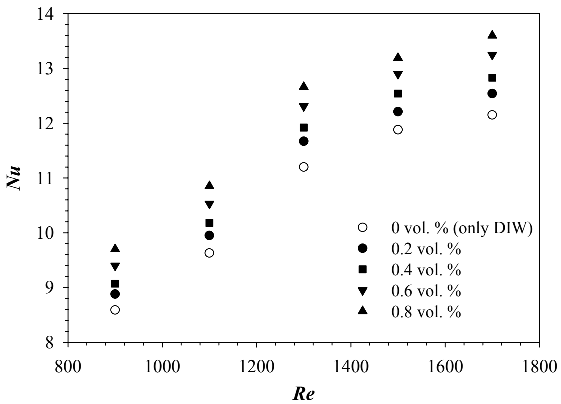


Fig 4: *Re* versus *Nu* at location $x/D = 25$

6.3 Effect of Nanoparticle Volume Fraction on Nusselt Number

Figure 5 demonstrates particle volume fraction dependence of Nusselt number. As can be seen, the Nusselt number of nanofluids increases almost linearly with the particle volume fraction.

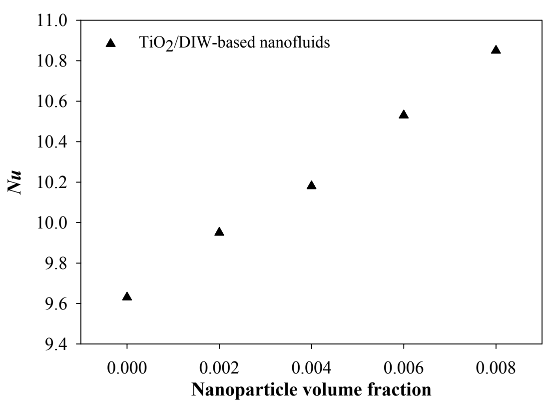


Fig 5: Particle volume fraction versus *Nu* at $x/D = 25$ and $Re = 1100$

The nanofluid behaves more like a fluid than a conventional solid-fluid mixture in which relatively larger particles of micrometer or millimeter orders are suspended. The effects of several factors such as gravity, Brownian force, and friction force between the fluid and the ultra-fine particles and dispersion may coexist in the main flow of nanofluids.

7. CONCLUSIONS

An experimental study on force convective heat transfer of a nanofluid flowing through a cylindrical minichannel under laminar flow conditions is performed. The present results showed that nanofluid exhibits an enhanced heat transfer coefficient compared to its base fluid. Both the heat transfer coefficient and Nusselt number increased significantly with nanoparticle volume fraction and Reynolds number. The migration of nanoparticles and the resulting disturbance of the boundary layer can be one of the reasons for such increase in convective heat transfer coefficient. More studies are worthwhile in order for better understanding the underlying mechanisms for enhanced heat transfer characteristics of nanofluids.

8. REFERENCES

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9. NOMENCLATURE

Symbol	Meaning	Unit
T	Temperature	(K)
D	Tube diameter	(m)
q	Heat energy	(W)
c_p	Specific heat	(kJ/kgK)
h	Heat transfer coefficient	(W/m^2K)
k	Thermal conductivity	(W/m-K)
x	Axial location	(m)
μ	Viscosity	(Pa.s)
$\dot{m}$	Mass flow rate	(kg/s)
ϕ	Particle volume fraction	-
Nu	Nusselt number	-
Re	Reynolds number	-
Pr	Prandtl number	-