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THE NEW CORRELATIONS FOR HEAT TRANSFER IN THE COOLING PROCESS OF AL_2O_3 – WATER NANOFLUIDS IN PIPE

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The aim of this experiment was to investigate convection heat transfer in the cooling processes of Al2O3-water nanofluid in concentrations of 0.15%, 0.25% and 0.5% by volume, respectively. The test section was a 1.25 m brass pipe with a 4.9 mm inner diameter and outer pipe of 38.5 mm diameter of a counter flow double-pipe heat exchanger. The temperatures at the nanofluid inlet in the inner pipe were a constant 40oC, and 23oC for the water at the inlet of the outer pipe. The results of this study indicated a more enhanced coefficient convective heat transfer of the cooling process than that of the heating process. The new equation of the Nusselt number obtained in the cooling process was $Nu_{nf}=0.75R_e^{0.846}\ P_r^{-2.28}\ \varphi^{0.03}$ *at particle volume concentrations of 0.15%, 0.25% and 0.5%. The maximum ratio enhancement of the nanofluid heat transfer in the cooling process was 45.2% at a particle volume concentration of 0.25%, while for the heating process nanofluid heat transfer was same as that for distilled water.*

Key words: Nanofluid, Cooling process, Convective heat transfer, Nusselt number

INTRODUCTION

The rapid technological development of heat transfer resulting from smaller size products (miniaturization) with high heat fluxes have sharply increased high heat flux which cause increases in temperature that can lead to malfunction or even severe damage to equipment. So, cooling is vitally important for efficient operations and to keep the components working optimally. In addition, heat transfer directly affects engine performance, fuel efficiency, material selection and emissions. The benefits of improved heat exchangers and heat transfer devices are as follows: reduced weight, improved fuel economy; smaller components, which take up less room under the hood and allow for greater latitude in aerodynamic styling; more effective cooling, and increased component life.

There are several ways to enhance convective heat transfer, including increasing heat transfer areas, altering boundary conditions, changing flow regimes and by using rough surfaces or grooves and ribs. But the traditional heat transfer fluid, such as water, oil and ethylene glycol have poor heat transfer characteristics which prevent improvement in energy efficiencies, and one way to improve heat transfer rate is to increase the thermal conductivity of these fluids. More than a century ago Maxwell [11], a scientist and engineer, made great efforts by suspending microsized particles in fluids. However, heat transfer fluids containing these micro/millimeter sized particles suffered from numerous drawbacks, such as erosion of the components through abrasive action, clogging in small passages, the settling of particles and increased pressure drop.

Recent improvements in nanotechnology have made it possible to produce solid particles with diameters smaller than 100 nm and nanosized particles under 100 nm in diameter which when suspended in fluids are called nanofluids [01] that have overcome the above problems. An important feature of nanofluids is that since nanoparticles are very small, they behave like fluid molecules and it is even possible to use nanofluids in micro channels. Nanofluid as a new class of heat transfer fluid exhibits superior thermal properties than those of base fluids and nanoparticles used in nanofluids have been made of various materials such as oxides, ceramic nitrides, carbide, metal, semiconductors and carbon nanotubes. Also, many types of fluids, such as water, ethylene glycol, oil and refrigerants have been used as base fluids.

The presence of the nanoparticles in the fluids increases the effectiveness of thermal conductivity

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and consequently enhances heat transfer characteristics, as shown by the many studies into their thermal conductivity [02, 13]. Moreover, various mechanisms and models have been proposed for explaining the enhanced thermal conductivity of nanofluids using various assumptions, and some have proposed that the enhancement is due to the ordered layering of liquid molecules near the solid particles [21, 19]. As a new working fluids, they have drawn a great deal of attention since their emergence and the amount of published work dealing with nanofluids has increased rapidly—growing at an average of around 32% per year for the past five years.

Experimentally, the enhancement of laminar flow convection coefficient of Al_2O_3-water nanofluids under constant wall temperatures in heating process is much higher than that predicted by single phase heat transfer correlation used in conjunction with the nanofluids properties, as in [06]. It was also concluded that the heat transfer enhancement of nanofluids is not merely due to the thermal conductivity increase of nanofluids which means other factors may contribute to this phenomenon. Investigations into laminar flow forced convection heat transfer of Al_2O_3-water nanofluid inside a circular tube at constant wall temperatures in the heating processes, found that the addition of nanoparticles in a 2.5% volume concentration could enhance average heat transfer coefficient by up to 40% [07]. The results of an experiment investigating the convective heat transfer coefficient of graphite nanoparticle dispersed in a fluid for laminar flow in a horizontal tube heat exchangers, indicated that the heat transfer coefficient increased with increasing Reynold number and particle volume concentration [20]. Duangthongsuk and Wongwises [04] have presented the heat transfer and flow characteristics of a nanofluid consisting of water and TiO2 nanoparticles at a 0.2% volume concentration in a double-tube heat exchanger for heating. The results showed that the convective heat transfer coefficient of the nanofluid was only slighty higher than that of the base fluid, by about 6-11%, and had a small increase in pressure-drop. It should be noted that of all of the investigations found in literature none are concerned with the cooling process, i.e. tube wall temperatures being higher than the average temperature of the fluid. Based on the above-mentioned literature review, the objective of this paper was to find the effects of the cooling process in forced convective heat transfer under a laminar flow regime.

EXPERIMENT SET-UP

Nanofluid Preparation

The γ-Al_2O_3 nanoparticles of 20-50 nm produced by Zhejiang Ultrafine Powder&Chemical Co, Ltd China were used. The nanofluid was prepared by dispersing Al_2O_3 nanoparticles in different volume concentrations in distilled water as the base fluid. The mechanical mixer (magnetic-stirring) was used for dispersing the nanoparticles. The solutions of water-alumina nanoparticles were prepared by the equivalent weight of nanoparticles according to their volume and was measured and gradually added to distilled water while being agitated in a flask. No sedimentation was observed after 5 hours in the low concentrations used in this study.

Experiment Apparatus

The experiment system used for this study is shown schematically in Figure 1, comprised of two flow circuits, cold water and hot nanofluid and consisted of a flow loop, a heating unit, a cooling unit, a flow measuring and pressure drop measuring units and a flow loop containing a reservoir, a pump, a valve for controlling the flow rate and a test section.

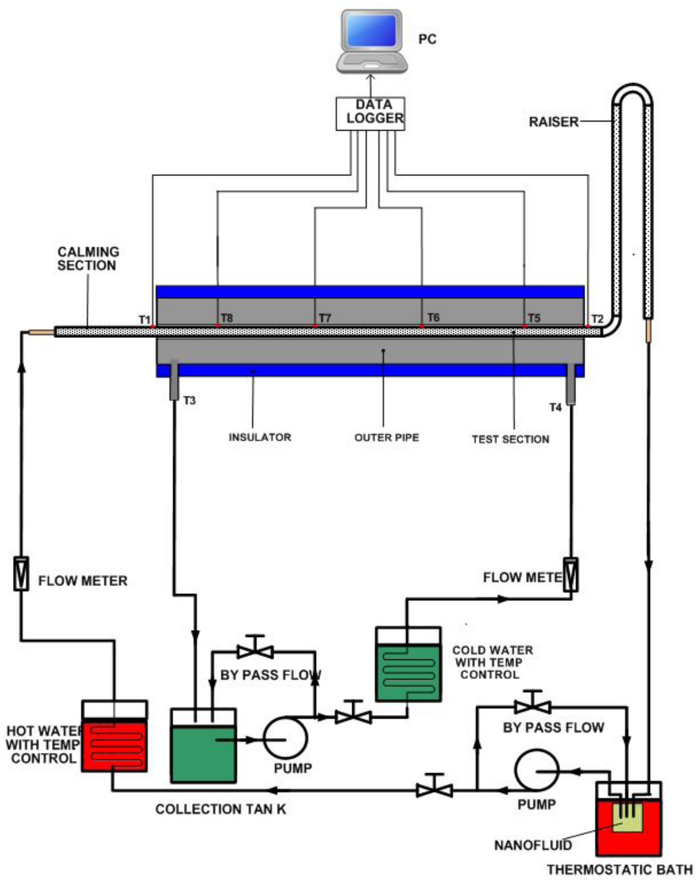

Figure 1: Schematic diagram of the experiment set-up

The test section was a 1.25 m long 1 mm thick pipe with a 4.9 mm inner diameter, was 1 mm thick and a 38.5 mm outer diameter of a counter flow double- pipe heat exchanger. The nanolfluid used in this studied was Al_2O_3 with distilled water as the base fluid in three particle volume concentrations, namely 0.15%; 0.25% and 0.5%, respectively. The temperatures at the nanofluid inlet in the inner pipe were a constant 40°C, and 23 °C for the water at the inlet of the outer pipe. An electric heater with a PID controller and cooling tank with a thermostat were installed to keep the temperature of the nanofluid constant. The nanofluid flow rate and cold water were controlled by adjusting the bypass flow valve, and measured by rotameter (DAKOTA ACRYLIC FLOW METER) type 6B7100-01B (0.1–1.0 LPM) and type 6B7100-03C (0.75–7.5 LPM) with accuracy 5%. The nanofluid flow regime was laminar and the fluid flow at the annulus was made constant. Temperature measurements at steady state were taken using a thermocouple data logger USB TC-08 with 8 channels, four K-type thermocouples were mounted at the inlet and outlet fluid flow and a further four mounted on the walls of the tubes. While, pressure drops were obtained by measurements using a model 409-005DWU5V Brand Omega PX wet-wet differential pressure transducer. To ensure that all data concerning the fully developed flow region were collected, a 60 cm long calming section was installed and to keep the nanofluid within the pipe, a 100 cm high raiser was mounted at the outlet.

DATA REDUCTION

Thermo-Physical Properties

Nanofluid properties such as density, specific heat, viscosity and thermal conductivity were calculated by using the following published correlations. Nanofluid density was calculated from Pak and Cho [14] correlations, which are defined as follows:

$$\rho_{nf}=\varphi\ \rho_p+(1-\varphi)\ \rho_b \qquad (1)$$

where, ρ_{nf} is the density of the nanofluid,

- φ is the particle volume concentrations,
- ρ_p is the density of the particles and
- ρ_b is the density of the base fluids.

Nanofluid heat capacity was calculated by Xuan and Roetzel's [18] correlations, as follows:

$$Cp_{nf} = \frac{(1-\varphi)\rho_b\, Cp_b + \varphi(\rho_p\, Cp_p)}{\rho_{nf}} \qquad (2)$$

where, Cp_{nf} Cp_p and Cp_b are nanofluid, particles and base fluids heat capacity, respectively.

Nanofluid viscosity was calculated from Maiga et al [14] correlations,

$$\mu_{nf}=\mu_b\ (1+7{,}3\ \varphi+123\varphi^2\ ) \qquad (3)$$

where, μ_{nf} and μ_b are nanofluid and base fluids dynamic viscosity.

Thermal conductivity of nanofluid was calculated from Li et al [09] correlations by considering Brownian motion and clustering of nanoparticles. He proposed an equation to predict the thermal conductivity of nanofluids:

$$k_{nf} = \frac{k_p + 2k_b - 2(k_b - k_p)\varphi}{k_p + 2k_b + (k_b - k_p)\varphi}\, k_b \ + \frac{\rho_p\, \varphi\, Cp_p}{2\, k_b} \sqrt{\frac{k_B T}{3\,\pi\, r_{cl}\, \mu_b}} \qquad (4)$$

where, k_{nf}, k_p and k_b are nanofluid, particles and base fluids thermal conductivity, respectively, k_B is the Boltzmann constant and r_{cl} is the apparent radius of the nanoparticle clusters. The first term on the right-hand side of equation (4) is the Maxwell model for the thermal conductivity and the second term on the right-hand side of equation (4) adds the effect of the random motion of the nanoparticle into account.

Nanofluid Heat Transfer

The heat transfer performance of the nanofluids through a tube was defined in terms of the convective heat transfer coefficient and calculated as follows:

Heat transfer rate from cooling or heating nanofluid was calculated from,

$$Q_{nf} = \dot{m}_{nf}\, Cp_{nf}(T_{nf.i} - T_{nf.o}) \qquad (5)$$

where, Q_{nf} is the heat transfer rate, $\dot{m}_{nf}$ is the mass flow rate, $T_{nf.i}$ is nanofluid temperature at the inlet and $T_{nf.o}$ is nanofluid temperature at the outlet, respectively.

Logarithmic mean temperature difference was calculated as follows:

$$\Delta T_{LMTD} = \frac{(T_{w.i} - T_{nf.o}) - (T_{w.o} - T_{nf.i})}{ln\left(\frac{T_{w.i} - T_{nf.o}}{T_{w.o} - T_{nf.i}}\right)} \qquad (6)$$

where, ΔT_{LMTD} is the logarithmic mean temperature difference, $T_{w.i}$ is the chilled water inlet temperature inlet and $T_{w.o}$ is chilled water outlet temperature.

Heat transfer coefficient of nanofluid was calculated, as flollows:

$$U_i A_i = \frac{1}{\frac{1}{h_i A_i} + \frac{1}{k}\, ln\frac{D_o}{D_i} + \frac{1}{h_o A_o}} \qquad (7)$$

where, U_i is the overall heat transfer coefficient, h_i=h_{nf} is the heat transfer coefficient of the inner diameter/nanofluid, h_o is the heat transfer coefficient of chilled water at the annulus section, A_i and A_o are the inner and outer cross sections of the tube, D_i and D_o are the inner and outer diameters of the tube and k is the thermal conductivity of the tube.

The average Nusselt number of the nanofluid was calculated, as follows:

$$\overline{Nu_{nf}} = \frac{\bar{h}_{nf}\, d}{k_{nf}} \tag{8}$$

where, $\overline{Nu}_{nf}$ is the average Nusselt number of the nanofluid,

d is the inner diameter of the tube

k_{nf} is the thermal conductivity of the nanofluid.

RESULTS AND DISCUSSION

As a comparison for the results of the convective heat transfer using nanofluids, similar experiments were done using distilled water as the base fluid. Figure 2 shows the experiment results of the distilled water in a laminar flow regime based on the predictions from the Seider and Tate equation [16], defined as:

$$\overline{Nu_{nf}}(th) = 1.86\left(Re\,Pr\,\frac{d}{L}\right)^{1/3}\left(\frac{\mu_s}{\mu}\right)^{0.14} \tag{9}$$

The above equation shows very good agreement between experimental data and Seider- Tata equation results which emphasizes the accuracy and reliability of the experiments.

As shown in Figure 3, the heat transfer coefficient increases with increases in the Reynold number as well as in the particle volume concentrations. It can be clearly seen that the heat transfer coefficient of the nanofluid is higher than that of the base fluid (pure water) at a given Reynold number. For example, at Reynold number 2000 the Nusselt number increases from 6.12 to 8.6 (40.5%), for volume concentration 0% (pure water) to 0.5%. The thermal conductivity enhancement according equation (4) is about 1.9% for the same volume concentration.

It can be concluded from this figure that other mechanism beside thermal conductivity increase can be responsible for heat transfer enhancement.

The convection heat transfer caused by the presence of nanoparticle is speculatively attributed to the interactions of nanoparticles with the wall as well as with the surrounding fluids. It is considered by [05] that the interactions between nanoparticles and solid walls play an important role in the convection heat transfer of nanofluids. The nanoparticles, serving as 'heat carriers', frequently collide with the tube walls. With the increase in the nanoprticles concentrations, the interaction and collisions between nanoprticles and the wall become more frequent, and causing a much higher heat transfer and the Nusselt number.

The New Nusselt Number Correlation

Dittus and Boelter [03] have shown that the correlation of the Nusselt number of pure water is a function of Reynold number and Prandtl number, and Pak and Cho [14] presented nanofluid correlation of the Nusselt number as a function of both the Reynold number and Prandtl number. Xuan and Li [19] also presented their correlation in which the Nusselt number was a function of particle concentration, particle size, Reynold and Prandtl numbers. However, their correlation was only for heating processes.

The Nusselt number Nu by definition is a function of the heat transfer coefficient h and thermal conductivity k of the nanofluid. Both h and k vary with the particle volume concentration. Therefore, the Nusselt number Nu must be a function of the particle volume concentration.

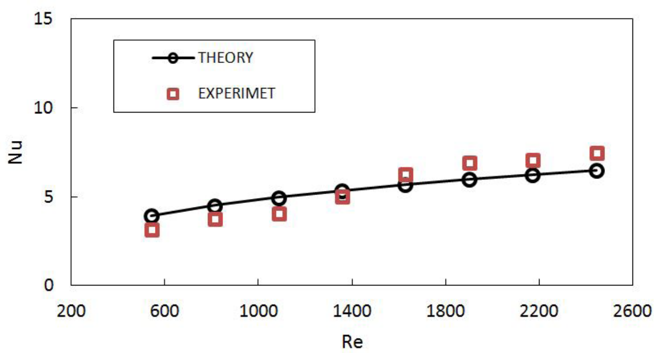

Figure 2: A comparison of the experimental Nusselt number with data obtained by Seider dan Tata equation vs Reynold number

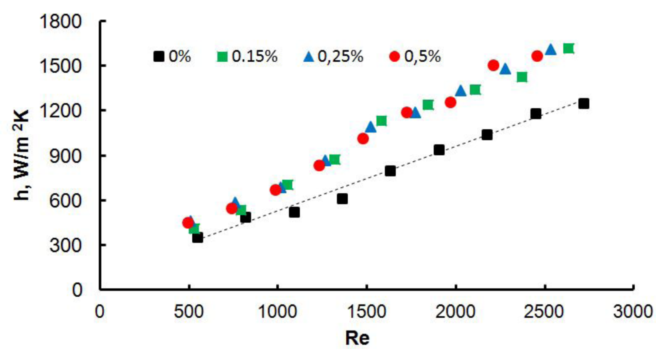

Figure 3: Heat transfer coefficient of Al_2O_3-water nanofluid under the cooling process

From our experiment observation on laminar flows of nanofluids inside the tube in the cooling processes, the equation obtained was:

$$Nu_{nf}=0.75\ Re^{0,846}\ Pr^{-2,28}\ \varphi^{0,03} \qquad (12)$$

The equation (12) is a new correlation for nanofluid heat transfer under a cooling process at particle volume concentrations of φ= 0.15% - 0.5%. Figure 4, depicts the curve of the theoretical predictions of the nanofluids Nusselt number from correlation (12) and the experimental data. Obviously, there exists good agreement between the results calculated from these correlations. The maximum ratio enhancement of the nanofluid heat transfer in the cooling process was 45.2% at a particle volume concentration of 0.25%, and the deviations between the experiment data and new correlations were reduced to within – 16.4 % to 16.5%, respectively.

The pressure drop of the nanofluid at low volume concentrations i.e. 0.15% and 0.25% under the cooling process there was good agreement for the traditional single phase correlation, but, at higher volume concentrations i.e. 0.5%, the traditional single phase correlation failed to predict nanofluid pressure drop. Based on the experiment data, new correlations was proposed for calculating nanofluid friction factors as a function of the Reynold number [16]. The pressure drop from the heating process for all nanofluid volume concentrations was almost the same as for pure water.

Experiment Uncertainty

Uncertainty of experiment results was determined by measuring deviation in the parameters, including weight, temperature, flow rate and pressure drop. The weight (W) of nanoparticles was measured by a precise electronic balance with the accuracy of ± 0.001 g, the precision temperature data acquisitions (T) was ± 0.1 oC, flow rate (V) was measured by a Dakota-rotameter with the full scale accuracy of ±5%.The uncertainty of heat transfer experimental results could be expressed as follows [20]:

$$u_C = \pm\left[\left(\frac{\Delta V}{V}\right)^2 + \left(\frac{\Delta W}{W}\right)^2 + \left(\frac{\Delta T}{T}\right)^2\right]^{1/2}$$

Therefore, the uncertainty of the experiment was less than ± 6.0%

CONCLUSION

The results of this study, showed that heat transfer coefficient of nanofluids in the cooling process differed from those of the heating process. Enhancement of the coefficient convective heat transfer increased with increasing Reynolds number and mass flow rates. The new equations of Nusselt number obtained in the cooling process at particle volume concentrations of 0.15%, 0.25% and 0.5%, was $Nu_{nf}=0.75\ Re^{0.846}\ P_r^{-2.28}\ \varphi^{0.03}$ with a maximum deviation within the range of -16.4% to + 16.5%. The maximum ratio enhancement of the nanofluid heat transfer in the cooling process was 45.2% at a particle volume concentration of 0.25%, while for the heating process nanofluid heat transfer was same as that for distilled water.

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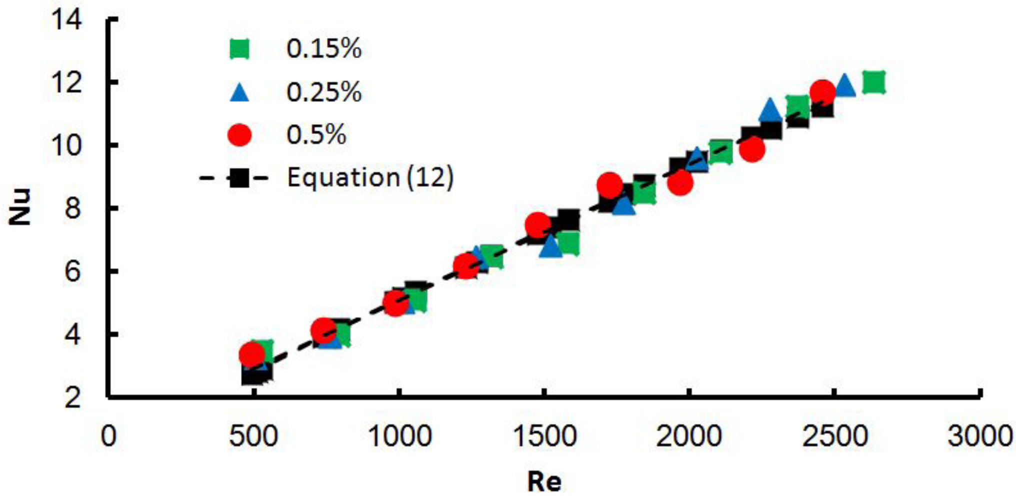

Figure 4: Comparison of experimental data with equation (12)

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