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Experimental study of forced convective heat transfer from a vertical tube conveying dilute Ag/DI water nanofluids in a cross flow of air

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Abstract

Forced convective heat transfer from a vertical circular tube conveying deionized (DI) water or very dilute Ag-DI water nanofluids (less than 0.02% volume fraction) in a cross flow of air has been investigated experimentally. Some experiments have been performed in a wind tunnel and heat transfer characteristics such as thermal conductance, effectiveness, and external Nusselt number has been measured at different air speeds, liquid flow rates, and nanoparticle concentrations. The cross flow of air over the tube and the liquid flow in the tube were turbulent in all cases. The experimental results have been compared and it has been found that suspending Ag nanoparticles in the base fluid increases thermal conductance, external Nusselt number, and effectiveness. Furthermore, by increasing the external Reynolds number, the external Nusselt number, effectiveness, and thermal conductance increase. Also, by increasing internal Reynolds number, the thermal conductance and external Nusselt number enhance while the effectiveness decreases.

Keywords: Ag/DI water nanofluids, Forced convective heat transfer, Cross flow of air, Thermal conductance, Effectiveness

Background

Prediction of forced convective heat transfer from a circular tube conveying a hot nanofluid is a basic but important problem in the field of heat transfer enhancement in recent years. Researchers and engineers are confronted with this problem in a wide variety of industries ranging from transportation, HVAC, heat exchangers, and textiles. All of these industries are limited by heat transfer; hence, they have a strong need for improved fluids that can transfer heat more efficiently.

Adding small particles such as metallic or metal oxide particles with high conductivity in micro and nano sizes to the heat transfer fluids such as water, ethylene glycol, and oils is an effective method for heat transfer enhancement [1-3]. However, using nanoparticles is much better than using microparticles because of their suspension stability and large heat transfer specific surface area [4]. An interesting feature of nanofluids is that they have anomalously high thermal conductivities even at very low nanoparticle concentrations [4-7]. Nanoparticles have approximately20%of their atoms near the surface, allowing them to absorb and transfer heat more efficiently [4]. However, many researchers have been intrigued by the anomalous behavior of nanofluids in thermal conductivity and convection heat transfer coefficient [5]. Traditional conductivity theories of solid/liquid suspensions, such as Maxwell [8] or other macroscale approaches [9], cannot explain why nanofluids have these intriguing features. It has been found that at nanoscales, the effects of different factors such as particles size, distribution of particles, nanolayering, and aggregation of nanoparticles, and Brownian motion [1-3,10] on thermal conductivity enhancement become more important, requiring a different approach to understand energy transport in nanofluids.

In the past decade, many researchers have experimentally and numerically demonstrated that the use of micro and nano size particles in a specific base fluid leads to higher heat transfer performance. For example, Lee and Choi [11] studied convective heat transfer of laminar

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flow of unspecified nanofluids in microchannels. They observed that nanofluids approximately are able to dissipate heat power three times more than pure water. Eastman et al. [12] found that the application of nanoparticles provides an effective way of improving heat transfer characteristics of fluids. They experimentally showed that significant increase in thermal conductivity can be obtained for a nanofluid consisting of water and CuO nanoparticles. Xuan and Li [13] experimentally investigated flow and convective heat transfer characteristics of Cu-water nanofluids through a straight brass tube with a constant wall heat flux. Their experimental results illustrated that the convective heat transfer coefficient of the nanofluids is higher than that of base fluid and varies with the flow velocity and volume fraction. Fotukian and Esfahany [14] experimentally studied turbulent convective heat transfer performance and pressure drop of very dilute(less than 0.24% volume) CuO/water nanofluid in a circular tube. They observed 25% increase in heat transfer coefficient and 20% increase in pressure drop. Wen and Ding [15] investigated convective heat transfer of γ-Al_2O_3-water nanofluids flowing through a copper tube in laminar flow regime. The results clearly showed that the use of nanofluids significantly improves the convective heat transfer specifically at higher Reynolds numbers. Yang et al. [16] measured the convective heat transfer coefficient of graphite nanofluids under laminar flow in a horizontal tube heat exchanger. The experimental results showed that the nanoparticles increase the heat transfer coefficient of the fluid system, but the increase was much less than that predicted by current correlation based on static thermal conductivity measurements. Zeinali et al. [17] investigated the laminar flow forced convection heat transfer of Al_2O_3-water nanofluids inside a circular tube with constant wall temperature. The experimental results indicated that heat transfer coefficient of nanofluids increases with Peclet number as well as nanoparticle concentration. The increase in heat transfer coefficient due to the presence of nanoparticles was much higher than the predictions of single phase heat transfer correlations used with nanofluids properties. Chein and Chuang [18] employed CuO-water nanofluid with various volume concentrations in a microchannel heat sink. The results showed greater energy absorption by nanofluid with respect to water. Furthermore, with increasing flow rate, both the thermal resistance and the temperature difference between the inlet and outlet of the microchannel heat sink decreased. Li and Kleinstreuer [19] investigated numerically the thermal performance of a trapezoidal microchannel with pure water and CuO-water with volume fractions of 1% and 4% as working fluid. They used a temperature-dependent model for thermal conductivity that accounts for the fundamental role of Brownian motion. Their results showed that nanofluids measurably enhance the thermal performance of microchannel mixture flow with small increase in pumping power, and with increasing volume fraction, the thermal performance increases. Chun et al. [20] investigated experimentally the convective heat transfer of nanofluids made of transformer oil and alumina nanoparticles in laminar flow through a circular pipe and showed that the addition of nanoparticles in the fluid increases the average heat transfer coefficient of the system in laminar flow. Hwang et al. [21] studied the flow and convective heat transfer characteristics of water-based Al_2O_3 nanofluids flowing through a circular tube with the constant heat flux in fully developed laminar regime. They clearly presented that the nanoparticles suspended in water enhance the convective heat transfer coefficient in the thermally fully developed regime. Pantzali et al. [22] studied experimentally the efficacy of nanofluids as coolants and showed that the type of flow (laminar or turbulent) inside the heat exchanging equipment plays an important role in the effectiveness of nanofluids. When the heat exchanging equipment operates under conditions that promote turbulence, the use of nanofluids is beneficial if and only if the increase in their thermal conductivity is accompanied by a marginal increase in viscosity. On the other hand, if the heat exchanger operates under laminar conditions, the use of nanofluids seems advantageous. Seyf and Mohammadian [23] investigated the thermal and hydrodynamic performance of a counter flow microchannel heat exchangers with and without nanofluids. They used nanofluids in both hot and cold channels, and showed that the nanofluid enhances the efficiency of the system; with increasing volume fraction and Reynolds number, the influence of Brownian motion on effectiveness increases. Seyf and Feizbakhshi [24], for the first time, studied the effect of particle size on thermal and hydrodynamic performance of micropin-fin heat sinks. They reported that for Al_2O_3-water nanofluid with decreasing particle size, the Nusselt number increases, while the trend is reverse for CuO-water nanofluid.

The effects of Ag nanoparticle concentration on heat transfer enhancement from a vertical circular steel tube conveying very dilute Ag-DI water nanofluids in a cross flow of air have not been studied yet. Therefore, in this paper, a series of experiments have been done to obtain forced convective heat transfer properties of the tube at different internal and external Reynolds numbers and Ag-DI water nanofluid concentrations.

Methods

Sample preparation

Preparing the nanofluids by dispersing the nanoparticles in a base fluid needs proper mixing and stabilization of the particles. To attain stability of the suspension against sedimentation of nanoparticle, there are three effective methods which aim at changing the surface properties of the suspended nanoparticles and suppressing the formation of clusters of particles in order to obtain stable suspensions. These techniques are the (1) use of ultrasonic vibration, (2) control of the PH value of suspensions, and (3) addition of surface activators or surfactants.

In this study, dilute nanofluids including DI water as base fluid and Ag nanoparticles with three different concentrations (0.005%, 0.01%, and 0.02% volume fraction) were prepared. Both surfactant and ultrasonic vibrator methods were used for dispersing the nanoparticles into the base fluid. In order to produce nanofluid of desired volume fraction, equivalent weight of nanoparticles according to their volume was measured and gradually added to DI water while agitated in flask. Suspension was then sonicated continuously for 10 h using an ultrasonic mixer. The transmission electron microscope (TEM) was used to monitor the dispersion of the nanoparticles in the DI water as well as approximating the size of nanoparticles, as shown in Figure 1a,b,c. The figure shows that the primary shape of the nanoparticles is approximately spherical, and a little agglomeration was observed 3 h after the nanofluid was sonicated.

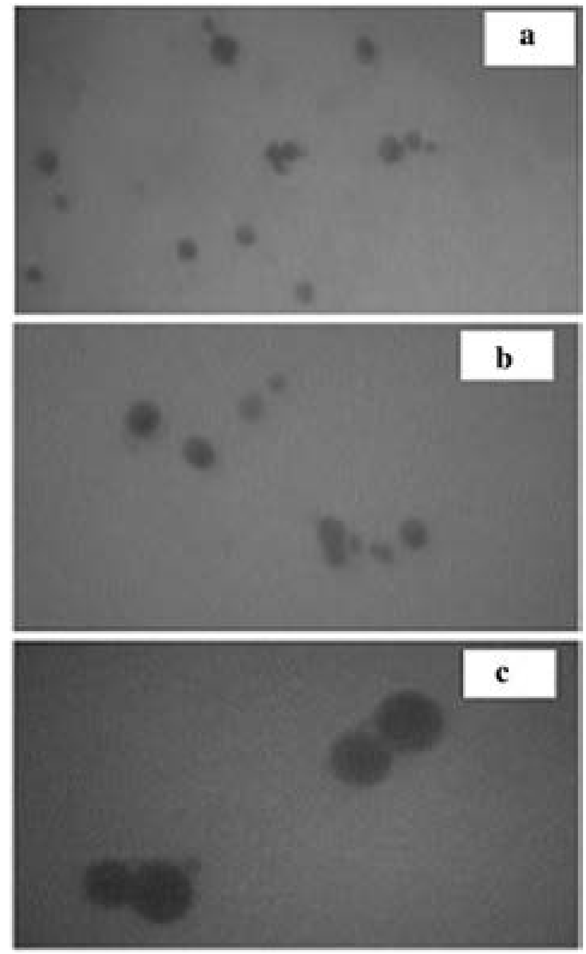

Figure 1 TEM images of the Ag-DI water nanofluid. (**a**) Silver nanoparticles have a size ranging from 25 to 60 nm. (**b**) Spread of nanoparticles without agglomeration. (**c**) Spherical shape of nanoparticles.

Experimental setup and test

The experimental setup is shown schematically in Figure 2. It consisted of a flow loop, heating system, flow controlling system, and measuring system. The flow loop included a pump, a reservoir, flow bypass line, and test section. The fluid became hot in a reservoir which was made by glass. It was located in a wooden box and completely isolated with glass wool. Two 1-kW heaters were located in the reservoir to increase the fluid temperature and to keep the inlet fluid temperature to the test section constant. At the test section of the wind tunnel, a circular steel tube with 500-mm length, 10-mm inner diameter, and 12-mm outer diameter conveying fluid was located vertically. The low-speed wind tunnel used in the experiments had circular test section with 800-mm length and 500-mm diameter of test section. The test section was insulated with fiberglass to avoid heat loss to the environment. To ensure a constant air intake and to minimize the effect of air flow turbulence maldistribution in the experiment, a wire screen mesh with 2 × 2-cm grid was attached at the entrance of the wind tunnel.

The ambient airflow was driven to the test section by a centrifugal fan. The rotations per minute of the wind tunnel fan and the air speed were controlled by an inverter which provided frequency control with different accuracies: 0.01 and 0.1 Hz. The velocity of the cross flow of air was measured by a pitot-static tube. For measuring the bulk temperatures at inlet and outlet of the tube, two T type thermocouples with accuracy of 0.1°C were calibrated and inserted into the flow. To prevent the heat transfer between the tube and the outer environment, the steel tube outside of the test section was completely isolated with glass wool. Also, other T type thermocouples were inserted in the reservoir and test section, which controlled the performance of the heaters and measured the air temperature at the inlet and outlet of the test section. The flow measuring system was a calibrated orifice which was connected to a manometer. The rate of flow which passed through the orifice was calibrated by the orifice equation presented in ASME MFC-14 M-2001. For calibrating the orifice, the cylindrical gauge was used and the orifice calibrated with 2% error. A flow bypass line was used to adjust the flow rate through the test section.

Ag-water nanofluids with different concentrations including 0.005%, 0.01%, and 0.02% volume fractions were

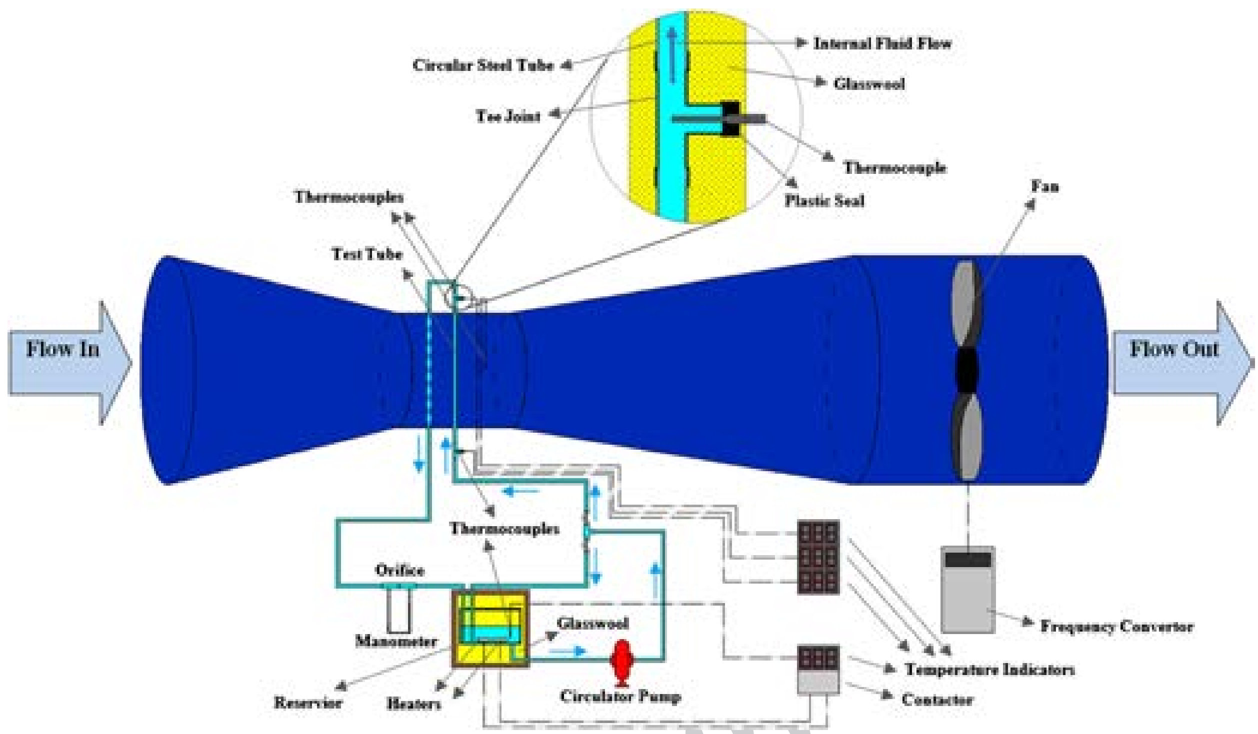

Figure 2 Schematic of the experimental setup.

used, and their results were compared with those ones for base fluid. The external Reynolds number (air flow Reynolds number) was varied between 8,000 and 25,000 so that it was turbulent.

Prediction procedure and validation of experimental setup

In this section, the heat transfer characteristics around the tube will determine the thermal measurement inside and outside of the tube. The amount of heat transfer q from internal fluid to air for a circular tube is governed by the equation:

$$q = \frac{\Delta T_{\mathrm{LMTD}}}{\left(\frac{1}{h_{out}A_{out}}\right) + \left(\frac{D_{out}}{2k_f A_{out}}\right) \ln \left(\frac{A_{out}}{A_{in}}\right) + \left(\frac{1}{h_{in}A_{in}}\right)} \quad (1)$$

where h is the convective heat transfer coefficient, A is surface area of the tube, D is the diameter, and indices 'in' and 'out' indicate the inside and outside of the tube, respectively. ΔT_{LMTD} is the log mean temperature difference between the internal fluid and the cross-flow of air.

$$\Delta T_{\mathrm{LMTD}} = \frac{\left(T_{nf,o} - T_{a,i}\right) - \left(T_{nf,i} - T_{a,o}\right)}{\ln \frac{\left(T_{nf,o} - T_{a,i}\right)}{\left(T_{nf,i} - T_{a,o}\right)}} \quad (2)$$

where $T_{\mathrm{nf,o}}$, $T_{\mathrm{nf,i}}$, $T_{\mathrm{a,o}}$, and, $T_{\mathrm{a,i}}$ are the bulk temperatures of nanofluid at the outlet and inlet of the tube in the test section as well as the outlet and inlet temperature of cross flow of air, respectively. The internal convective heat transfer coefficient for the hot fluid h_{in} can be calculated using the Gnielinski correlation [25]. For a steady state case, the amount of heat transfer from nanofluid to air, q, is evaluated as follows:

$$q = \dot{m}_{\mathrm{nf}} C_{p,\mathrm{nf}} \left(T_{\mathrm{nf},i} - T_{\mathrm{nf},o}\right) \quad (3)$$

where q is the heat transfer rate from the tube, $\dot{m}_{nf}$ is the nanofluid mass flow rate in the tube, and $C_{\mathrm{p,nf}}$ is the specific heat of nanofluid.

Therefore, the air-side convective heat transfer coefficient for the tube h_{out} can be found from Equation 1; consequently, the mean Nusselt number for the tube, $\mathrm{Nu}_{\mathrm{ext}} = h_{\mathrm{out}} D_{\mathrm{out}} / k_{\mathrm{air}}$, can be evaluated. It is worth mentioning that in the present work for air side $Gr/\mathrm{Re}^2_{D,\mathrm{out}} \ll 1$; hence, the effect of the natural convection can be neglected according to [25]. The thermophysical properties of the air are evaluated at the average temperature between the internal fluid and air temperatures. All internal fluid properties were evaluated at T_{m} where

$$T_m = \frac{T_{\mathrm{nf},i} + T_{\mathrm{nf},o}}{2} \quad (4)$$

The properties of DI water were obtained as [26]. The effective density and specific heat capacity of the nanofluids are calculated using the classical models for very dilute suspension available in literature [27,28] and are expressed as

$$\rho_{nf} = \alpha.\rho_\rho + (1-\alpha).\rho_w \tag{5}$$

$$C_{p,\mathrm{nf}} = \frac{\alpha.\left(\rho_p.C_{p,p}\right) + (1-\alpha).\left(\rho_w.C_{p,w}\right)}{a\rho_{\mathrm{nf}}} \tag{6}$$

where α is the nanoparticle volume fraction. Indices 'w' and 'p' refer to DI water and Ag nanoparticles, respectively. Due to very low volume fraction of the nanoparticles (very dilute suspension) used in this study, conventional models such as Brinkman equation [29] can be used to compute viscosity of nanofluid. The Brinkman equation [29] for viscosity of nanofluid is as follows:

$$\mu_{\mathrm{nf}} = \frac{\mu_f}{(1-\alpha)^{2.5}} \tag{7}$$

The thermal conductivity of the nanofluids was calculated from Yu and Choi [30], using the following equation:

$$k_{\mathrm{nf}} = \left[\frac{k_p + 2k_w + 2\left(k_p - k_w\right)(1 + \beta)^3\alpha}{k_p + 2k_w - \left(k_p - k_w\right)(1 + \beta)^3\alpha}\right]k_w \tag{8}$$

where k_{nf} is the thermal conductivity of the nanofluids, k_{p} is the thermal conductivity of the nanoparticles, k_{w} is the thermal conductivity of DI water, and β is the ratio of the nanolayer thickness to the original particle radius. Normally, a value of $\beta = 0.1$ is used to calculate the thermal conductivity of the nanofluids [30]. For very dilute suspensions similar to the present work, many researchers used this model for thermal conductivity of nanofluid, for instance, see [31-33]. The system effectiveness is the ratio of actual heat transfer to the maximum possible heat that can be transferred:

$$\varepsilon = \frac{q}{q_{\max}} \tag{9}$$

where

$$q_{\max} = C_{\min}\left(T_{\mathrm{nf},i} - T_{a,i}\right) \tag{10}$$

where $C_{\min}$ is equal to C_{nf} or C_{air}, whichever is smaller. C_{air} and C_{nf} are air and nanofluid heat capacity rates, respectively, and can be expressed as

$$C_{\mathrm{nf}} = \dot{m}_{\mathrm{nf}}C_{p,\mathrm{nf}} \tag{11a}$$

and

$$C_{\mathrm{air}} = \dot{m}_{\mathrm{air}}C_{p,\mathrm{air}} \tag{11b}$$

$\dot{m}_{nf}$ and $\dot{m}_{air}$ are mass flow rates through tube and test section, respectively. Substituting Equation (3) and Equation (10) into Equation (9), the effectiveness can be calculated as

$$\varepsilon = \frac{C_{\mathrm{nf}}\left(T_{\mathrm{nf},i} - T_{\mathrm{nf},o}\right)}{C_{\min}\left(T_{\mathrm{nf},i} - T_{a,i}\right)} \tag{12}$$

The thermal conductance can be calculated as follows:

$$UA = \frac{q}{\Delta T_{\mathrm{LMTD}}} \tag{13}$$

To the knowledge of the authors, there is no correlation and experimental study on investigating the thermal performance of circular tube conveying nanofluid in a cross flow of air. Therefore, to validate our experimental setup, we compare our results with Morgan [34] correlation for a circular cylinder in a cross-flow of air. Figure 3 illustrates the comparison of measurement results of present tests and Morgan correlation. As seen in this figure, the measurement points are very close to the correlation. Therefore, the present experimental setup is reliable.

The significant errors that could influence the accuracy of the experimental data can be classified into two groups: systematic errors and random errors. While systematic errors are minimized with careful experimentation and the calibration operations, the precision error can be treated statistically. The uncertainty of experimental results may be originated from measuring errors of parameters such as flow rates, temperatures, and etc. Using a method described by Taylor and Kuyatt [35], the uncertainty of the measured thermal conductance, effectiveness, and external Nusselt number were estimated to be ±4.1%, ±3.7%, and ± 4.3%, respectively. It is

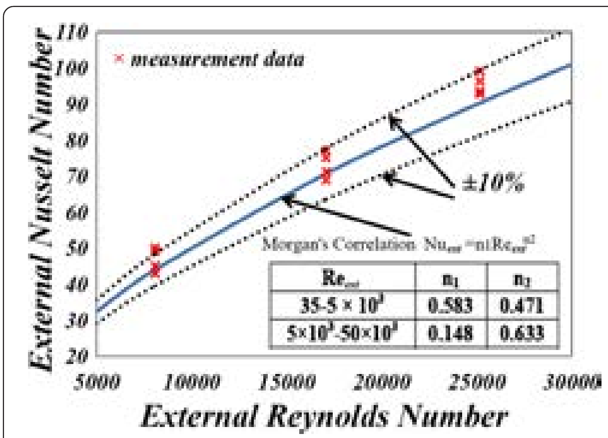

Figure 3 Check of the measuring procedure and facilities.

important to note that, in order to check the repeatability of the experiments, duplicate tests were performed for certain operating conditions, and the measured temperatures were found virtually identical within the measurement uncertainties.

Results and discussion

In this section, the results of experiments are presented for different internal Reynolds numbers, various volume fractions of Ag/DI water nanofluid (i.e., 0.005%, 0.01%, and 0.02%), as well as different external Reynolds numbers. In general, the influence of nanoparticles elucidates two opposing effects on the heat transfer coefficient. First is a favorable effect that is driven by the presence of high thermal conductivity of nanoparticles. In other words, the addition of nanoparticles to the base fluid enhances the thermal conduction and with increasing particle volume fraction, the enhancement increases. The enhancement of the thermal conduction should increase the convective heat transfer coefficient. Second is an undesirable effect promoted by high level of viscosity experienced at high volume fractions of nanoparticles. The nanofluid viscosity increases with increasing particle volume fraction which results in a reduction of velocity and consequently thicker boundary layer thicknesses on the tube surface. The growth in thermal boundary layer thickness is responsible for the lesser temperature gradients at the surface of the tube which results lower convective heat transfer coefficient accordingly.

Figure 4 demonstrates the effects of the internal and external Reynolds numbers as well as nanoparticle volume fraction on the thermal conductance. It can be seen that adding very low volume fraction of nanoparticles (0.005% to 0.02%) to the base fluid will lead to a significant increase in thermal conductance. This is an indication that the presence of nanoparticles clearly enhances heat transfer of the fluid. In general, this enhancement is due to higher thermal conductivity of nanoparticles and the role of Brownian motion of nanoparticles on enhancement of thermal conductivity which is due to larger surface area of nanoparticles for molecular collisions. The higher mass concentrations of nanoparticles compared to the base fluid molecules have higher momentum; this momentum carries and transfers thermal energy more efficiently inside the base fluid before releasing the thermal energy in colder regions of the fluid (small packets of energy) [24]. It can also be seen that with increasing internal Reynolds number, the thermal conductance increases in both cases with and without nanofluids. It is obvious that with increasing Reynolds number, the average bulk temperature of fluid increases as well as the heat transfer rate between the fluid and tube surface. This is due to the heat transfer that occurs inside the tube which comprised two mechanisms:

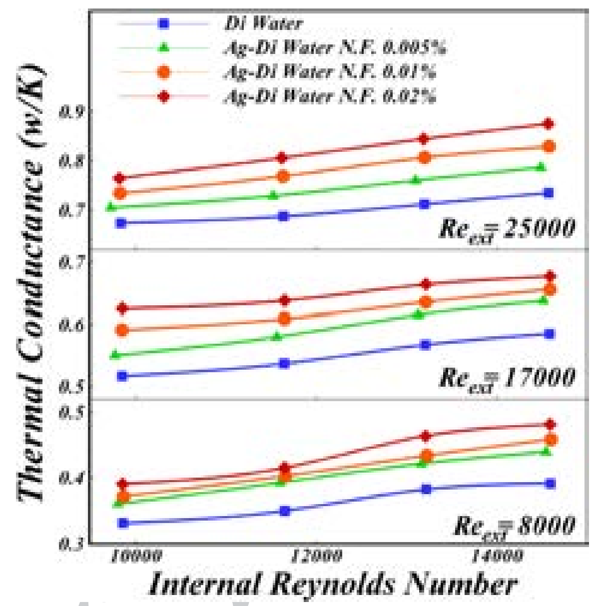

Figure 4 Effect of volume fraction on the thermal conductance of the circular tube.

energy transfer due to the bulk motion of the fluid and energy transfer due to diffusion in the fluid. At low Reynolds number, the fluid mean velocity is low and the fluid has more time to absorb and spread heat; therefore, diffusive heat transfer is the dominant player, causing the fluid to obtain a lower bulk temperature. On the other hand, as Reynolds number increases, the mean fluid velocity increases and forced convection plays a higher role in the heat transfer, thus, transferring more heat without much decrease in outlet temperature. With increasing internal Reynolds number, the heat transfer coefficient increases. This indicates that under the condition of this work, the energy transfer due to the bulk motion of the fluid outweighs the energy transfer due to diffusion in the fluid. Also, with increasing external Reynolds number, the thermal boundary layer thickness on the tube decreases. Therefore, the heat transfer from the tube increases.

Experimental results of the effectiveness obtained for the circular tube using different Reynolds numbers and Ag nanoparticles volume fractions are presented in Figure 5. From this figure, it can be seen that at a constant internal Reynolds number, the circular tube with nanofluids as working fluid has higher values of effectiveness with respect to DI water. As seen, with increasing internal Reynolds number, effectiveness decreases. This is because with increasing internal Reynolds number, the fluid mean velocity increases and the fluid does not have enough time to spread heat; therefore, the effectiveness decreases and the outlet temperature increases. Another point to be noted in this figure is that

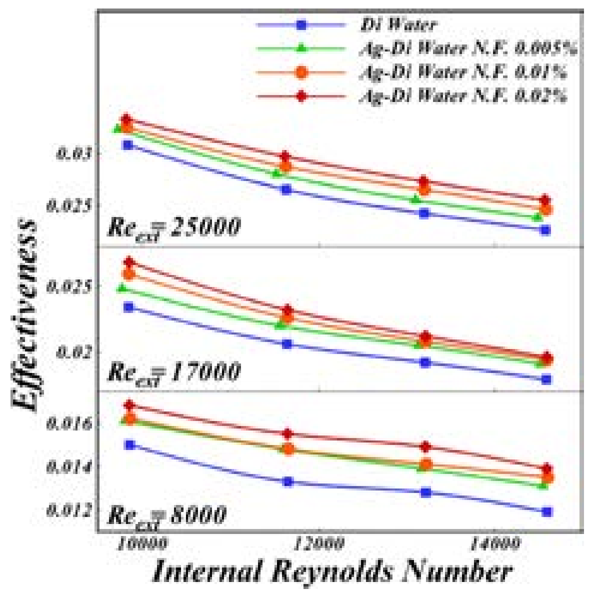

Figure 5 **Effect of volume fraction on the effectiveness of the circular tube.**

with increasing external Reynolds number, effectiveness increases. This is in complete agreement with fundamental principle of convective heat transfer, because with increasing external Reynolds numbers, the thermal boundary layer thickness on the tube decreases and consequently the heat transfer rate increases.

Figure 6 exemplifies the effect of volume fraction as well as internal and external Reynolds number on

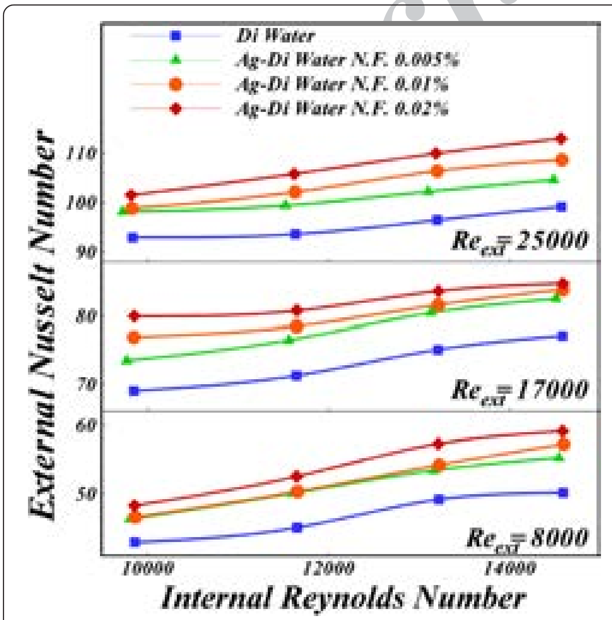

Figure 6 **Effect of volume fraction on the external Nusselt number.**

external Nusselt number. It can be seen that at a constant internal Reynolds number, the external Nusselt number enhances significantly with increasing nanoparticle volume fraction. This is due to the fact that with increasing volume concentration, the thermal conductivity of nanofluid increases which leads to higher internal heat transfer coefficient and consequently higher energy transfer between internal fluid and external air. Another reason for increasing Nusselt number is that with increasing nanoparticle volume fraction, the interaction and collision of nanoparticles with the wall of tube increase, transferring more thermal energy to the air in the test section and consequently increasing external heat transfer coefficient. The remarkable point is the high values of Nusselt number for very dilute Ag/DI water nanofluid used in this study with respect to DI water which shows the potential of Ag/water nanofluids in heat transfer enhancement even for very dilute suspensions. Moreover, as expected with increasing internal Reynolds number, the external Nusselt number increases. This is because with increasing Reynolds numbers, the level of augmentation of thermal energy transfer from the wall to the air increases. Another point that can be noted is the enhancement of external Nusselt number when the external Reynolds number increases. This is due to the fact that with increasing external Reynolds numbers, the thermal boundary layer thickness on the tube decreases and consequently the heat transfer rate increases.

Conclusion

Forced convective heat transfer from a vertical circular tube, conveying DI water and very dilute Ag-DI water nanofluids, in cross flow of air has been investigated experimentally. Some experiments have been performed in a wind tunnel, and the thermal characteristics of the system have been measured at different air speeds and Ag/DI water concentrations. The experimental results have been compared, and it is proven that the use of Ag nanoparticles in DI water leads to increasing thermal conductance, external Nusselt number, and effectiveness. Also, at the same external Reynolds number, the thermal conductance and Nusselt number increase with increasing internal Reynolds number while effectiveness decreases. Furthermore, it is concluded that while the internal Reynolds number increases with increasing external Reynolds number, the thermal conductance, effectiveness, and external Nusselt number increase as well.

Abbreviations

A: area of the tube; C_p: specific heat of the liquid flow in the tube; *D*: diameter of the tube; *Dr*: internal diameter of the test section in the wind tunnel; *Gr*: Grashof number; *h*: convective heat transfer coefficient; *k*: thermal conductivity; *L*: length of the tube; $\dot{m}$: liquid mass flow rate in the tube; *Nu*: Nusselt number; Pr: Prandtl number; *q*: heat transfer rate; Re: Reynolds number; T: temperature; UA: thermal conductance; V: average fluid velocity in the tube; ΔT_{LMTD}: log mean temperature.

Greek symbols

α: volume fraction of nanoparticles; β: ratio of the nanolayer thickness to the original particle radius; ρ: density; μ: viscosity.

Subscripts

a: air; f: internal liquid fluid; i: inlet of the tube; in: internal/inside; m: average; nf: nanofluid; o: outlet of the tube; out: outside; ext: external; p: Ag nanoparticle; w: DI water (base fluid).

Competing interests

The authors declare that they have no competing interests.

Authors' contributions

SKM gave the idea of the research work, designed and constructed an experimental setup for the research with the aid of ML, carried out the experimental work, and wrote the article. ML supervised the whole research work, checked the grammar of the whole article, and helped in editing. MH provided the nanofluids for the experiments and helped in the nanofluid preparation and tests. All authors read and approved the final manuscript.

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