Experiments on mixed convection heat transfer and performance evaluation of MWCNT–Oil nanofluid flow in horizontal and vertical microfin tubes

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ARTICLE INFO

Article history:
Received 1 July 2014
Received in revised form 5 November 2014
Accepted 8 November 2014
Available online 15 November 2014

Keywords:
Performance evaluation
Mixed convection heat transfer
Microfin tube
Smooth tube
MWCNT
Nanofluid

Abstract

The mixed convection heat transfer characteristics of MWCNT (multi walled carbon nano tube) heat transfer oil based nanofluid were experimentally investigated inside smooth and microfin tubes. The tubes were submitted to uniform wall heat flux over outer surface which were produced by an electrical coil heater. Data were acquired for the horizontal and upward vertical laminar flows in the thermal entrance region. Pure heat transfer oil and nanofluids with particle weight concentrations of 0.05%, 0.1% and 0.2% were utilized as the working fluids. The experiments were performed in the range of Grashof number between $10^2$ and $10^4$, Reynolds number between 10 and 150 and Richardson number between 0.1 and 10. The experimental results indicate that the heat transfer coefficient increases slightly with the increase of particle weight concentration from 0% to 0.2% in microfin and smooth tubes under a given Grashof number. Moreover, it is revealed that Nusselt number increases notably with Richardson number while it augments slightly by increasing Reynolds number in both horizontal and vertical tubes. In addition, results show that using microfin tube instead of plain one, causes an enhancement of heat transfer in each particle concentrations. Also, two empirical correlations are developed to predict the Nusselt number in the thermal entrance region in horizontal and vertical microfin tubes for the range of data used in the study. Finally, the overall performance of the tested microfin and smooth tubes were assessed based on the performance index. The performance evaluation of the two enhanced heat transfer techniques studied in this work shows that applying nanofluids instead of the pure oil is a more effective way to enhance the convective heat transfer coefficient compared to using microfin tube.

1. Introduction

Several applications in the area of heat transfer involve mixed free and forced convection in ducts with various cross-sections and orientations. Examples of these applications include solar energy systems, cooling of electronic devices, compact heat exchangers and the cooling core of nuclear reactors. Free convection can aid the forced flow or acts in opposition to it. The knowledge of the heat-transfer characteristics under both conditions can guide the design of devices used in these applications. Several investigators performed studies on mixed convection heat transfer in smooth and enhanced tubes. Barozzi et al. [1] reported experimental data of combined forced and free convection in horizontal and inclined tubes for the Reynolds number range of $200 < Re < 2300$, Grashof number range of $3 \times 103 < Gr < 5 \times 105$ and $0 < \theta < 60$. They noted that the local Nusselt number, first decreases along the heated length until reaches a minimum, and then increases to the fully-developed value. The minimum is due to a balance between entrance and free-convection effects. Variation of Nusselt number with $\theta$ was found to be very small which is probably due to the small values of Grashof numbers used in the study. A detailed experiment was reported by Maughan and Incropera [2] for laminar air flow between parallel plates ($30.5 \times 30.8$ mm cross-section) heated uniformly from below. They used the horizontal and upward inclinations up to $\theta = 30^\circ$. The reported variation of the local Nusselt number along the heated length is similar in trend to the ones reported in work of Barozzi et al. [1]. Also, the data show that the local Nusselt number increases with both $Gr$ and $\theta$. Buseda and Soliman’s [3] investigation was designed to examine the effects of buoyancy (both aiding and opposed) on laminar heat transfer in the thermal entrance region of an inclined semicircular duct. These effects were to be examined over a range of the independent parameters. It was found that, for the upward inclinations, Nusselt number...
increased with Grashof number and the inclination angle (up to 208), while the effect of Reynolds number was found to be small. For the downward inclination, Reynolds number has a strong effect on Nusselt number and the manner by which it varies with Grashof number. Mohammed and Salman [4] performed an experimental investigation on combined convection heat transfer for thermally developing aiding flow in an inclined circular cylinder with constant heat flux. They observed an increase in the Nusselt number values as the heat flux increases and as the angle of cylinder inclination moves from $\theta = 60^\circ$ inclined cylinder to $\theta = 0^\circ$ horizontal cylinder. Also, the average Nusselt numbers have been correlated with the (Rayleigh numbers/Reynolds numbers) in empirical correlations by Mohammed and Salman [4].

On the other hand, Nanotechnology has been widely used in traditional industry. Materials with the grain size of nanometers possess unique optical, chemical and electrical properties. In recent developments, nanoparticles could be dispersed in conventional heat transfer fluids such as water, glycol and oil to produce a new class of high efficiency heat exchange media. Many experimental studies reporting on the effects of nanofluid flow inside tubes on the heat transfer and pressure drop have been published. Mansour et al. [5] studied experimentally laminar mixed convection flow of $\text{Al}_2\text{O}_3$ water nano-fluid in a uniformly heated inclined tube. They found that the presence of nano-particles intensifies the buoyancy-induced secondary flow, especially in the developing region. Their results also show an augmentation of the heat transfer coefficient and a decrease of the wall friction when using nano-fluids. Feng and Li [6,7] experimentally studied laminar mixed convection of nanofluid flow for large Prandtl number. They found that due to the existence of natural convection, the measured average Nusselt number is higher than that predicted by the pure forced convection correlation and it increases with the increasing of the Reynolds number and Grashof number. Also, it was found that by the inclusion of the nanoparticles, the contribution of natural convection to the overall convective heat transfer can be either deteriorated under the same heat flux or enhanced under a given Grashof number. In fact, such seemingly paradoxical phenomena boil down to the different comparison criteria, i.e., heat flux and/or Grashof number. Adding nano particles enhance the thermo-physical properties of nanofluids such as fluid viscosity, thermal conductivity but increase of fluid viscosity is more than thermal diffusivity, thus fluid viscosity is the governing parameter. While experiments are done under the same heat flux, Grashof number decreases since the fluid viscosity increases and is dominant, but the Prandtl number increases as a result Nusselt number deteriorates since the Grashof number is dominant. While experiments is under the a given Grashof number, only Prandtl number increases as a result of fluid viscosity, thus, Nusselt number enhances. Also, many researches have been done with oil based working fluid. Since the thermal conductivity of pure oil is low, adding nano particles to it, cause a significant enhancement in the value of thermal conductivity of nanofluid in comparison to the base fluid. This enhancement with oil based fluids is rather higher than other based fluids such as water. Thus, nano particles are used as additive for industrial oil such as engine oil, heat transfer oil and lubricating oil in order to remove heat from high heat flux surfaces, as a result, using oil-based nanofluids has many applications in industry such as engines, lubrications, heat exchangers and cooling systems.

Fakoor pakdaman et al. [8] studied the thermo-fluid properties and performance evaluation of MWCNT heat transfer oil nano fluid flow in vertical helically coiled tube. Their experimental results reveal enhancement of heat transfer coefficient by increasing nanofluid particles. They proposed correlations to predict the thermal properties of MWCNT–Oil nano fluid. Also, Razi et al. [9] performed experiments on the forced convective heat transfer and pressure drop of CuO/oil nano-fluid in flattened tube with constant heat flux. They observed that there is a noticeable enhancement in both heat transfer and pressure drop when they use nanofluid instead of base fluid, also flattening the tube increases pressure drop as well as heat transfer. Also, they observed that pressure drop increase with Reynolds number, however, it does not vary significantly with particle fractions.

Furthermore, another way of heat transfer augmentation is extending the heat transfer surfaces and the use of helical micro-fin tubes is one the most prevalent passive enhancement device
in use today. These tubes promote significant increments in heat transfer though not affecting in the same proportion the pressure drop. These characteristics have promoted their widespread application in the industries. Ghazvini et al. [10] carried out an empirical investigation of the forced convective heat transfer and pressure drop of nano diamond/oil fluid in microfin tube and observed that using microfin tube instead of plain tube increases heat transfer as well as pressure drop. Also, the effects of particle fractions on heat transfer and pressure drop was reported the same as those reported in [8,9].

This paper summarizes important results from an investigation of developing laminar mixed convection of a particular nanofluid, CNT/Oil based mixture. The purpose of this investigation is an examination of nanofluid behavior in microfin and plain tubes during laminar mixed convection. The data are collected for horizontal and vertical upward flows while tubes are submitted to a uniform heat flux on outside surface. The effects of various parameters on the mixed convective heat transfer performance of this nanofluid are reported and analyzed. In addition, Two empirical correlation are developed to predict the Nusselt number in the thermal entrance region of horizontal and vertical tubes.

2. Experimental apparatus and procedure

Fig. 1 shows a schematic illustration of the experimental set-up. The working fluid flows through heat exchanger (water cooler) and enters the heated section. This section has total heated length of 1.05 m. Two tubes were tested in this study; a smooth tube with 9.525 mm OD and 9.025 ID and a microfin tube with 9.52 mm OD, 55 fins and 15° helix angle. The geometrical parameters of microfin tube are the same as the one used in [10]. The nanofluid flowing inside the tube was heated by an electric heated coil with maximum enduring electrical power of 2 kW. In order to reduce the heat losses, two thick layers of fiber–glass insulation (2 cm) are wrapped around the heated element. The corresponding axial positions of the thermocouples on the surface of microfin and smooth tubes were 150(T2), 300(T3), 450(T4), 600(T5), 750(T6) and 900(T7) mm from the inlet of the test section. The flow leaves test section and enters to the flow measuring system. The working fluid then flows into a reservoir. A copper coil was located inside the reservoir to pre-cool the working fluid which was necessary to keep the inlet temperature the same among different tests. In order to control the flow rate a bypass line with a control valve was used to guide a portion of the flow back to the reservoir without flowing through the test section. A stand was used to change the inclination of the tube. The uncertainty of the major heat transfer parameters has been conducted based on the method proposed by Kline and Mcclintock [11] and presented in Table 1.

A Heat transfer oil (IRANOL HTB) was selected as the base fluid and Multi-Walled Carbon Nano-Tubes (MWCNTs) were utilized as the additive. These additives also have shown anti wear and anti friction characteristics due to their spherical shapes. Also, to study on the behavior of MWCNT nanoparticles more effectively, a type of oil with no additives is used. This type of oil is the basic component of the industrial oils. It is apparent that the effect of nanoparticles on heat transfer performance of the specified oil can be generalized to the mentioned industrial oils for the sake of heat transfer enhancement. The suspentions of nano-particles were prepared with three different weight fractions of 0.05%, 0.1% and 0.2%. The method used to prepare the nanofluids and measure
the properties of nanofluids and also the properties of the applied nanoparticles are the same as discussed in [8].

The Eq. (1) is used to calculate the convection heat transfer coefficient in which \( T_w, T_b, \) and \( h \) are function of \( X \).

\[
q^* = h(T_w - T_b)
\]

In the Eq. (1) \( T_w \) and \( T_b \) are the average wall temperatures of three thermocouples at each location and fluid bulk temperatures, respectively. \( T_b \) can be calculated from energy balance equation.

\[
Q = mC_p\Delta T = q^* \times p \times X
\]

Rearranging Eq. (2) result in

\[
T_b(X) = T_i + \frac{q^* \times p}{m C_p} X
\]

where \( T_i \) is the inlet temperature of the flow, \( p \) is the perimeter of tube and \( X \) is the distance from the tube inlet. \( m \) and \( C_p \) are the mass flow rate and specific heat, respectively. To calculate the mean convection heat transfer coefficient, the Eq. (4) is used. Finally Nusselt number was calculated with the Eq. (5).

\[
\bar{h} = \frac{1}{L} \int_0^L h(X) dX
\]

\[
Nu = \bar{h} \cdot D/k
\]

3. Results and discussion

Experiments were carried out using MWCNT/Oil mixtures, with particles of average diameter 36 nm. The data were acquired for the power supply, \( P \), range of 200–600 W and the particle weight fraction, \( \phi \), range of 0–0.2%. The results and discussion presented here demonstrate the effects of particle volume concentration on the flow and heat transfer behavior of the nanofluid in the entrance regions. These effects on both plain and microfin tubes are discussed.

3.1. Validation of results

First of all, the integrity of experimental set-up has been established by comparison of experimental results with theoretical predictions for horizontal flow in plain tube. In Fig. 2 such a comparison has been made taking theoretical Nusselt number as abscissa and experimental Nusselt number as ordinate. From Fig. 2, it is observed that the experimental Nusselt numbers are within an error range of –15% to +18% and it is revealed that experimental results are in agreement with those obtained by the theory. Theoretical Nusselt number is calculated from the Eq. (6). This equation is developed for laminar thermal entry length assisting mixed flow [12,13].

\[
Nu = A(G_{z} + \Phi)^{1/3} \left( \frac{\mu_w}{\mu_b} \right)^{-0.14}
\]

In this equation \( G_z \) is the Graetz number, \( \Phi \) is free convection effect which is function of Grashof number and Prandtl number, \( A \) is a constant equal to 2.11 for uniform wall heat flux condition and \( (\mu_w/\mu_b)^{-0.14} \) is the correction coefficient for the temperature dependent properties in which \( \mu_w \) and \( \mu_b \) are wall and bulk fluid viscosity [12,13]. Fig. 3a and b show a comparison of experimental mean Nusselt number of present study with other experimental models for the base fluid flow in horizontal [6,7,14,15] and vertical [16] plain tubes, respectively. It is found that the maximum deviation between Nu number of present study with those obtained by Oliver [14] is about 18% and occurs at the \( \text{Ra} = 3.6 \times 10^3 \). Also, the maximum deviation between Nusselt number of present study with those obtained by Mac Gregor and Emery [16] is about 8% which occurs at the \( Gr = 1640 \).

3.2. Heat transfer results

The effect of particle concentration on temperature field and Nusselt number along the inlet length is demonstrated in Fig. 4a and b. Fig. 4a shows the axial development of the average wall temperature (at each axial location), \( T_w \), and the fluid bulk temperature, \( T_b \), for two different particle weight concentrations, a power supply of \( P = 200 \) W and a mass flow rate of \( m = 3.5 \) g/s inside horizontal plain tube at \( Re = 15 \) and \( Gr = 1125 \). It is found from Fig. 4a that the bulk temperatures increase with increasing particle concentration while wall temperature decreases. Fig. 4a also shows that the difference between fluid and wall temperature, \( T_w - T_b \), decreases slightly with the increase of \( \phi \), which indicates that there is a slight increase of the heat transfer coefficient and Nusselt number. Also, as the heat flux along the tube length is constant the bulk temperature distribution is linear, and as it can be seen in Fig. 4a, the variation of wall temperature in the entrance region is in agreement with the results expected from theory [12,13]. Fig. 4b shows the variation of local Nusselt number versus inlet length from the heated section for different particle weight fractions in horizontal tube at \( P = 200 \) W, \( m = 3.5 \) g/s, \( Re = 15 \) and \( Ri = 5 \). Nusselt number decreases along the tube as it could be observed in Fig. 4b. It is found from this figure that the local Nusselt number increases when concentration of the particles increases. The enhancement in the Nusselt number is 10% for increasing particle weight fraction up to 0.2%.

The variation of the mean Nusselt number versus Reynolds number is indicated in Fig. 5 for horizontal and vertical upward flows inside the microfin tube. The Grashof number is fixed at \( Gr = 5000 \). It is found from Fig. 5 that for the horizontal and vertical
tubes under assisted flow, the Nusselt number increases slightly with the Reynolds number. The same result is also reported in the previous studies [4,6,17]. The same behavior is observed for the other particle weight fractions. Fig. 5 also demonstrates that the Nusselt number augments slightly by increasing particle weight fractions up to 0.2 wt.%. In this study, test experiments were done to obtain the same Grashof number when adding nano-particles to the base fluid by raising the heat flux at higher nano-particle fractions at the same mass flow rate. Therefore, an enhancement of heat transfer was observed when using nanofluids instead of base fluid which is due to enhancement of Prandtl number as a result of an increase in the value of dynamic viscosity [6,7,18,19].

Fig. 6 has been drawn to show the variation of mean Nusselt number with Richardson number and as it could be observed, the Nusselt number in horizontal and vertical tube follow the same trend. It is revealed from the Fig. 6 that by increasing inclination, Nusselt number decreases. This is due to the fact that the secondary flow and near wall acceleration are depends on the tube inclination. For the enhancement of heat transfer the component of buoyancy force in tube cross section is dominant and this component is maximum when tube is in horizontal position which leads
to the most value for secondary flow that causes an enhancement in the value of Nusselt number [4,17–19]. For example, the Nusselt number decreases about 30% in the vertical flow in comparison with the horizontal flow for base fluid flow at \( R_i = 4 \) in microfin tube. In addition, another point that could be observed in Fig. 6 is the enhancement of Nusselt number in microfin tube in comparison with the plain one. The most enhancements due to tube surface extension is about 15% for base oil flow inside horizontal tubes. Although, the enhancement of heat transfer in the microfin tube is superior in comparison with plain tube in each fraction, yet the effect of these particles in the plain tube is about 5% more than microfin tube. These results are in agreement with those reported in [10,20]. However, by considering effects of both tube surface extension and nanoparticles addition, the maximum enhancement occurs in horizontal microfin tube at \( \phi = 0.2 \text{ wt.\%}, P = 600 \text{ W and } R_i = 10 \). It can be observed that increasing Richardson number causes an enhancement in the value of Nusselt number, similar results were also reported in [4–7,17–19]. In addition, it can be seen that effect of inclination increases when there is an increase in the value of Richardson number.

The heat transfer behavior of the nanofluid is characterized by various factors such as thermal conductivity, heat capacity, thermal expansion coefficient, viscosity and particle volume concentration. Using the experimental data, Eq. (6) and Oliver model for free convection effect [14], the following correlation was developed to predict the mean Nusselt number in a horizontal microfin tube in the thermal entrance region, a new correlation was developed using Eq. (6) and MacGregor and Emery model [16] for predicting free convection effects. The following correlation, Eq. (8), expresses the mean Nusselt number as a function of the Grashof, Graetz and Prandtl numbers and particle weight fraction (\%) in the same range used for the horizontal tube.

\[
\overline{Nu} = 2.11(1 + \phi)^{0.63}(Gz + 0.135(GrPr^{0.75}(D/L)^{0.9})(\mu_w/\mu_p)^{-0.14}) (7)
\]

The above correlation given in Eq. (8) predicts the experimental data within an error range of –10% to +8%.

### 3.3. Performance evaluation

In this section, we are going to assess the thermal performance of the system based on the performance index which simultaneously considers the heat transfer enhancement besides the increase in pressure drop. Two heat transfer enhancement techniques are used in the present study. Microfin tubes have been used instead of smooth tubes and MWCNT/heat transfer oil nanofluids are utilized rather than conventional fluids. However, it was also observed that the mentioned methods caused the pressure of the working fluids to drop significantly along the test section. In order to find the optimum work conditions, the overall performance of the enhancement techniques considering simultaneous effects of heat transfer and pressure drop increment are studied here based on the performance index. This parameter is defined as follow:

\[
\eta = \frac{h' / h_{bf}}{\Delta P' / \Delta P_{bf}} \tag{9}
\]

where \( h' \) and \( \Delta P' \) represent mean heat transfer coefficient and pressure drop of the flow resulted by applying heat transfer enhancement techniques, respectively. In addition, \( h_{bf} \) and \( \Delta P_{bf} \) are the mean heat transfer coefficient and pressure drop of the pure oil flow inside the smooth tube, respectively. Apparently, when the performance index is greater than 1, it implies that the technique is more in the favor of heat transfer enhancement rather than in the favor of pressure drop increasing. Therefore, the heat transfer methods with performance indexes greater than 1 would be feasible choices in practical applications [8,9].

Fig. 8a illustrates the variation of performance index versus Richardson number inside horizontal and vertical plain tubes for two nano particle weight fractions. As it can be seen in this Fig. 8a values of performance index increases by increasing Richardson number, also, these values are higher for horizontal tube than the vertical one. The performance index is more than 1 for horizontal tube in the all range of the Richardson numbers, however, for vertical tube this value is more than 1 in the range of \( R_i > 1 \). The maximum performance index of 1.19 was found at \( R_i = 8 \) for the nanofluid 0.2 wt.\% inside plain tube. Fig. 8b demonstrates the variation of performance index versus Richardson number inside horizontal and vertical microfin tubes for the base fluid. In fact, this figure presents the performance of the second technique which is using microfin tube instead of plain one for the pure oil. It can be seen in this figure that, values of performance index increase with Richardson number, also these values are greater in horizontal tube than the vertical one. Furthermore, values of performance index are less than 1 inside horizontal and vertical microfin tubes. The maximum value of performance index is 0.92 which occurs at \( R_i = 8 \) and horizontal microfin tube. It was found from Fig. 8 that applying the nanofluids instead of base fluid is a more effective way than the second technique which is using microfin tube instead of plain one.
4. Conclusion

The following conclusions have been drawn from the present investigation:

1. It has been found that a higher particle weight concentration clearly induces an augmentation of the Nusselt number in both horizontal and vertical flows. There is an enhancement of about 10% by increasing nanoparticles' concentration up to 0.2 wt.% in horizontal plain tube for \( R_i \) up to 10 and \( Re \) up to 150.

2. Mean Nusselt numbers increases slightly in both horizontal and vertical tubes when Reynolds number increases up to \( Re = 150 \) while it augments notably by increasing Richardson number in all tested tubes for \( R_i = 01–10 \). Also, Nusselt number is higher in horizontal tube than the vertical one, where the average decrease of 30% occurs in vertical microfin tube in comparison to the horizontal one for the base fluid; In addition, the effect of inclination is increased by increasing \( R_i \).

3. Heat transfer coefficient augments when microfin tube is used instead of plain tube. There is an enhancement of 15% in the value of Nusselt number in horizontal microfin tube in comparison with plain one for base oil, thus by considering effect of all parameters, the maximum enhancement is achieved in horizontal microfin tube at \( \varphi = 0.2 \) wt.%, \( P = 600 \) W and \( R_i = 10 \).

4. Two new correlations were proposed to predict the mean Nusselt number in horizontal and vertical microfin tubes. These correlations predict the present experimental data within an error band of −6% to +4% and −10% to +8% for the horizontal and vertical microfin tubes, respectively.

5. Performance index was evaluated using two techniques; it was found that using nanofluid instead of base fluid is a more effective way to enhance the heat transfer rather than using microfin tube instead of plain one. The maximum performance index of 1.19 was found at \( R_i = 8 \) for the nanofluid of 0.2 wt.% flow inside the plain tube in comparison to the base fluid flow.

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