Mixed convection of MWCNT—heat transfer oil nanofluid inside inclined plain and microfin tubes under laminar assisted flow

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ABSTRACT

This study describes an experimental investigation on the heat transfer characteristics of MWCNT (Multi walled Carbon Nano-tube)—heat transfer oil based nanofluid during mixed convection inside inclined plain and microfin copper tubes under uniform heat flux condition. Data were acquired for the laminar flow in the thermal entrance region and for tube inclination angles of 0, 45° and 90°. Pure heat transfer oil and nanofluids with nanoparticles weight concentrations of 0.05%, 0.1% and 0.2% were used as the working fluids. Effects of nanoparticles concentration, heat flux, tube inclination and free convection on the development of the thermal field are studied under buoyancy assisted flow condition for Grashof number, Reynolds number and Richardson number between 500 and 10^4, 10–100, and 0.1–10, respectively. Results show that Nusselt number increases slightly with an increase of nanoparticles weight concentration from 0 to 0.2% under a given Grashof number. Moreover, increasing Grashof number can raise the heat transfer coefficient in horizontal tube more than the inclined tubes under assisted flow condition. The effect of nanoparticles on heat transfer enhancement decreases by increasing of inclination and Richardson number, also, this effect is more pronounced in the plain tube rather than the microfin one.

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1. Introduction

The use of nanotechnology has been increased in the recent years because nanoscale materials possess unique optical, electrical and chemical properties. Recent developments made it possible to disperse nanoparticles in conventional heat transfer fluids such as water, ethylene glycol and engine oil to produce a new class of heat exchange fluids with high efficiencies. Fluids formed of saturated liquid and suspended nanoparticles are called nano fluids. Despite a wide range of studies on the thermal conductivity of nano fluids, relatively few investigations have been performed to measure the rheological properties nano fluids; in addition, mixed convection in ducts with various cross-sections and orientations is one of the common applications in the area of heat transfer including solar energy systems, cooling of electronic devices and compact heat exchangers. Knowledge of heat transfer characteristics under this condition can guide the design of devices used in these applications. Sparrow et al. [1] carried out an analysis of boundary layer equations for combined free and forced convection flows. They reported that the flow can be classified by Gr/Re^2, while their study was conducted for Pr = 0.7. Mixed flow condition was obtained when 0.3 < Gr/Re^2 < 16 for the aiding flow. Ramachandran et al. [2] provided a correlation for the laminar mixed convection adjacent to an inclined flat plate where this correlation had been obtained by the concept of Churchill [3], which was correlating the mixed convection Nusselt number by using the pure forced convection (Nu_f) and pure free convection (Nu_n) then this expression was combined via Nu^n = Nu^n_f + Nu^n_n, where n = 3 and plus sign is used for assisted flow and minus for opposed one. Their work was in agreement with predictions for assisted flow when 0.1 < Gr/Re^2 < 7. Mohammed and Salman [4] experimentally investigated combined convection heat transfer for thermally developing aiding flow in horizontal and 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 θ = 60° inclined cylinder to θ = 0° horizontal cylinder. Also, the average Nusselt numbers have been correlated with the (Rayleigh numbers/Reynolds numbers) in empirical correlations. Mohammed [5] empirically studied the laminar mixed convection heat transfer in vertical circular tube under buoyancy assisted and
opposed flows, he found that Nusselt number for opposed flow is lower than that for assisted one. Empirical correlations were also provided for Nusselt number in terms of Grashof and Reynolds numbers. Feng and Li [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. 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. The huge increasing of the viscosity and Prandtl number were proposed to be the major reason for the observed deterioration and enhancement, respectively. Ben Mansour et al. [8] experimentally studied the laminar mixed convection flow of Al2O3 water nanofluid in uniformly heated inclined and horizontal tubes. They found that the presence of nanoparticle intensifies the buoyancy-induced secondary flow, especially in the developing region. Their results also showed a slight decrease of the heat transfer coefficient and a decrease of the wall friction when using nanofluids. For the horizontal and vertical tubes, new correlations were proposed to predict the heat transfer coefficient and wall friction factor in the fully developed region where Grashof number was ranging from 10^5 to 10^7 and nanoparticles volume concentrations up to 7%. Meyer et al. [9] investigated the forced convective heat transfer enhancement, pressure drop, and performance evaluation of multi-walled carbon nanotubes-water nanofluid flowing through a straight horizontal tube experimentally for the transitional flow regime. They found that heat transfer was enhanced when comparing the data on a Reynolds–Nu graph. However, when comparing the data at the same velocity, it was shown that heat transfer did not go up. They also observed that the tested carbon nanotubes are not an ideal fluid for heat transfer enhancement in the range of data studied. Fakoor Pakdaman et al. [10] experimentally studied the thermo-physical properties and performance evaluation of MWCNT–heat transfer oil nanofluid flow inside vertical helically coiled tube. They measured the thermo-physical properties of MWCNT–oil nanofluid and provided data for the properties of this nanofluid at particle weight fractions of up to 0.4%. They also provided correlations for the thermo-physical properties of MWCNT–heat transfer oil nanofluid with particle weight fractions of up to 0.4%. Furthermore, another way of enhancing heat transfer is extending the heat transfer surfaces; it provides a better contact between the fluid and the career. Microfin tube became popular in the industry because of its capability of enhancing the heat transfer and low pressure drop in comparison with other extended surfaces. Derakhshan et al. [11,12] investigated experimentally, mixed convection heat transfer, pressure drop, rheological and fluid characteristics, and performance evaluation of MWCNT–Oil nanofluid flow in inclined microfin and smooth tubes. They observed that using microfin tube instead of a smooth one increases heat transfer as well as pressure drop. They also found 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 based on the overall performance of the tested microfin tubes. They also provided two empirical correlations to predict the Nusselt number in the thermal entrance region in horizontal and vertical microfin tubes.

In the present work, thermal developing laminar mixed convection of MWCNT/Oil based nanofluid is experimentally investigated inside both inclined microfin and plain tubes under uniform wall heat flux during the buoyancy assisted flow condition. The effects of various parameters on the mixed convective heat transfer performance of this nanofluid are reported and analyzed.

2. Nanofluid preparation

A Heat transfer oil (HTB type manufactured by IRANOL company) was selected as the base fluid. Multi-Walled Carbon Nano-Tubes (MWCNTs) were utilized as the additive. Preparing stable

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### Nomenclature

- $C_p$: specific heat capacity [J/kg K]
- $D$: tube inside diameter [mm]
- $D_o$: tube outside diameter [mm]
- $D_h$: hydraulic diameter [mm]
- $e$: fin height [mm]
- $f$: fin pitch [mm]
- $Gr$: Grashof number $[-]$, $Gr = \rho^2g\alpha q^2D^8_h/k\alpha^2$
- $Gz$: Graetz number $[-]$, $Gz = \pi Re Pr D_h/4L$
- $h$: heat transfer coefficient $[W/m^2 K]$
- $k$: fluid thermal conductivity $[W/m K]$
- $m$: mass flow rate $[g/s]$
- $Nu$: Nusselt number $[-]$, $Nu = hD_h/k$
- $p$: power supply $[W]$
- $Pr$: Prandtl number $[-]$, $Pr = C_p\mu/k$
- $\rho$: tube cross section perimeter $[m]$
- $\rho^\prime$: heat flux $[W/m^2]$
- $Q$: heat transfer rate $[W/s]$
- $Re$: Reynolds number $[-]$, $Re = \nu D_h/\mu$
- $Ri$: Richardson number $[-]$, $Ri = Gr/Re^2$
- $Ra$: Rayleigh number $[-]$, $Ra = GrPr$
- $T$: temperature $[^\circ C]$
- $T_w$: wall temperature $[^\circ C]$
- $T_b$: bulk temperature $[^\circ C]$
- $t$: wall thickness [m]
- $v$: fin tip angle [degree]
- $wt$: weight fraction
- $X$: axial coordinate [m]
- $X/D$: dimensionless axial inlet length

### Greek symbols

- $\theta$: inclination angle of tube [degree]
- $\beta$: fluid thermal expansion coefficient [1/K]
- $\mu$: fluid dynamic viscosity [kg/m s]
- $\nu$: fluid kinematic viscosity $[m^2/s]$
- $\rho$: fluid density $[kg/m^3]$
- $\gamma$: helix angle [degree]
- $\phi$: particle weight concentration $[-]$
- $\Phi$: free convection function $[-]$

### Subscripts

- $b$: bulk
- $bf$: base fluid (oil)
- $f$: fluid
- $i$: inlet
- $nf$: nanofluid
- $p$: particle
- $w$: wall
suspension of nanoparticles in the base liquid is the first step in applying nanofluids for heat transfer enhancement. In order to prepare nanofluids by dispersing the nanoparticles in a base fluid, proper mixing and stabilization of the particles are required. Normally, there are three effective methods used to attain stable suspension of the nanoparticles, which are controlling the pH value of the suspensions, adding surface activators or surfactants and using ultrasonic vibration. All of these techniques aim at changing the surface properties of suspended nanoparticles and suppress the formation of clustering particles in order to obtain stable suspensions. In this study, no surfactant is used since it might affect the physical and/or thermal property of the fluid. Nanofluids with particle weight concentrations of 0.05%, 0.1% and 0.2% were prepared by dispersing a specific amount of MWCNTs in the heat transfer oil by using an ultrasonic processor (Hielscher company, Germany) for 6 h generating pulses of 400 W at 24 ± 3 kHz. This ultrasonic device is used to disperse the nanoparticles into the base oil and to break large agglomerates of nanoparticles in the fluid and make stable suspension. It should be noted that each time before dispersing the nanoparticles by the ultrasonic device, the MWCNTs were dispersed in the fluid by means of an electrical blender for 1 h. It was observed with SEM image that the sedimentation started after about 24 h of the time that the applied nanofluid was made uniform by the ultrasonic device. Therefore, each day before starting the experiments, the working fluid was made uniform by the mentioned ultrasonic device. The properties of applied nanoparticles and the method used to prepare the nanofluids and to measure the properties of nanofluids are the same as those discussed in Refs. [10–12].

3. Experimental apparatus

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 9.025ID and a microfin tube with 9.52 mm OD, 55 fins and 15° helix angle. The microfin tube and geometrical parameters of it are shown in Fig. 2. The hydraulic diameter of microfin tube was defined as the subtraction of the fin height from the tube inside diameter and it was computed 8.67 mm. The experimental set-up and the geometrical parameters of microfin tube are the same as those used in Refs. [11,12]. The nanofluid flowing inside the tube was heated by an electric heated coil with the 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. Even after that, heat loss is measured to be 7, 11, and 15% of the total power of 200, 400 and 600 W, respectively. Those heat losses were then subtracted from the initial heating power, therefore, the net exerted heat fluxes were 6400, 12,300 and 17,600 W/m² for plain and microfin tubes. A precise potentiometer with the accuracy of ±5% was used to adjust, the output voltage supplied to the electrical heater. Two calibrated RTD PT 100 type temperature sensors (measuring temperatures at points T1 and T8) with ±0.1 °C accuracy with digital indicators were used to measure the bulk inlet and outlet temperatures. These two thermocouples were immersed in the mixing chambers provided at the inlet and outlet. The mentioned mixing chambers were mixing the working fluid in
the heated section which led to a convenient measurement of the mean bulk fluid temperature. Wall temperatures were measured by 18 T-Type thermocouples at six axial locations along the heated section at top, middle and bottom of each location with the accuracy of ±0.1 °C. The average temperature of three readings at each location was considered as the temperature of that location. For all the tested microfin and plain tubes, the distance of the first and last thermocouples (T2 and T7 respectively) was 150 mm from the inlet and outlet of the test section, respectively. Other thermocouples were mounted at equal distances on the tube surface. The corresponding axial positions of the thermocouples on the surface of microfin and plain 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. This system consisted of 1000 cm³ glass vessel with a valve in bottom. Flow rate was measured directly from the time required to fill the glass vessel. A digital chronometer with the accuracy of 0.01 s was used to measure the required time to fill the glass vessel. The working fluid then flows into a reservoir. A copper coiled-tube with tap water running through was located inside the reservoir to pre-cool the working fluid which was necessary to keep the inlet temperature constant 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. The flow then passes through 1 HP gear pump connected to a three phase electromotor. The gear pump (ZDF, Czech) has a nominal power of 20 L/min. and finally goes through the water cooling heat exchanger. A framework stand was used to change the inclination of the tube in order to perform the experiments in three inclinations of θ = 0, 45° and 90°. When all the experiments associated with a fluid were conducted, we opened the mentioned flow loop. It was first washed by base fluid and then cleaned by means of an air compressor so that the inner wall of the tested tube was completely cleaned especially by releasing air in quick bursts. This process prevented from the deposition of MWCNTs on the inner wall of the tube before starting experiments on the other working fluids. The uncertainty of the major heat transfer parameters has been conducted based on the method proposed by Kline and Mcclintock [13] and is listed in Table 1. More details of the method were also presented in Refs. [10–12].

4. Nanofluid properties and data reduction

In this study four different nanofluids with MWCNT (Multi walled Carbon NanoTube) as the nanoparticles, and heat transfer oil (IRANOL HTB) as the base fluid were used. Each prepared nanofluid was given to the Iranian Research Institute of Petroleum Industry (RIPI) to evaluate the properties of such fluids. The suspensions of nanoparticles were prepared with three different weight fractions of 0.05, 0.1 and 0.2%. The method used to measure the thermo-physical and rheological properties of nanofluids and their properties are the same as those discussed in Refs. [10–12]. The relevant thermo-physical properties of different phases of nanofluid used in this study and their range of application for base fluid, nanofluids with particle weight concentrations of 0.05%, 0.1% and 0.2% and temperature between 40 °C and 80 °C are summarized in Table 2.

Equation (1) is used to calculate the convection heat transfer coefficient in which, \( T_w \) and \( T_b \) are local values and hence function of \( X \) and \( q' \) is the net heat flux.

\[
q' = h(T_w - T_b) \tag{1}
\]

In Eq. (1) \( T_w \) and \( T_b \) are the average wall temperatures of three thermocouples at each location and fluid bulk temperatures, respectively. The parameter \( T_b \) can be calculated from energy balance equation. Regarding to the low thickness of wall tube and high thermal conductivity of copper, the amount of temperature lost in wall tube is considerably gentle. Hence, the temperature lost in wall tube is neglected and so the measured outside surface temperature is considered as equal with inside surface of wall tube.

\[
Q = \dot{m}C_P\Delta T = q'pX \tag{2}
\]

Rearranging equation (2) results in

\[
T_b(x) = T_i + \frac{q'}{\dot{m}C_p}X \tag{3}
\]

Equation (3) is appropriate for calculating axial fluid bulk temperature of a circular tube under uniform wall heat flux with a negligible wall heat transfer in which, \( T_i \) is the inlet temperature of the flow, \( p \) is the perimeter of tube, \( \dot{m} \) and \( C_p \) are mass flow rate and specific heat of the fluid, respectively and \( X \) is the distance from the tube inlet.

To calculate the mean convection heat transfer coefficient, equation (4) is used. Finally Nusselt number was calculated with equation (5) in which \( D_h \) is the hydraulic diameter of the tested tube.

\[
\bar{h} = \frac{1}{l} \int_0^l h(X) dX \tag{4}
\]

\[
Nu = \frac{hD_h}{k} \tag{5}
\]

5. Results and discussion

Experiments were carried out using MWCNT/Oil nanofluid, with nanoparticles with average diameter of 8 nm and for the following ranges of the governing parameters: the power supply (\( P \)) between 200 W and 600 W, the nanoparticles weight fraction (\( \omega \)) from 0 to 0.2% and tube inclinations from 0° to 90°. The effects of nanoparticles weight concentration on the heat transfer behavior of the nanofluid in the entrance region inside plain and microfin tubes are presented and discussed here.

5.1. Validation of results

The integrity of experimental set-up has been established by comparing experimental results with theoretical predictions for horizontal flow in plain tube. It was observed that the experimental Nusselt numbers are within an error range of −15% to +18% which reveals decent agreement between experimental results and those obtained by the theory [12]. The complicated analytical results for the average Nusselt number are usually approximated by the following simplified algebraic equation [14,15].
\[ \mathrm{Nu} = A(Gz + \Phi)^{1/3} \left( \frac{\mu_w}{\mu_b} \right)^{-0.14}, \quad Gz > 26.2 \] (6)

In this equation \( \Phi \) 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. Our experimental data can be comparable to solutions which have been obtained for thermal entrance length condition. Fig. 3a and b shows a comparison of mean Nusselt number of this study with former experimental mixed convection studies in the literature for the base fluids in horizontal \([7,16-18]\) and vertical (upward) \([19]\) plane tubes, respectively. Results show that the maximum deviation of Nusselt number of this study with that obtained by Oliver \([16]\) is about 18% and occurs at the \( Ra = 3.6 \times 10^7 \) inside horizontal plain tube and with Ref. \([19]\) is about 8% for vertical plain tube which occurs at the \( Gr = 1640 \). As it can be seen in these figures, results of present study are in agreement with those obtained in Refs. \([7,16-19]\).

### 5.2. Heat transfer results

Effects of tube inclination angle on the average wall temperature, bulk temperature and local Nusselt number are illustrated and compared in Figs. 4 and 5 respectively at \( \dot{m} = 3.5 \text{ g/s}, q'' = 6.4 \text{ kW/m}^2, Re = 15, Gr = 1125 \). Fig. 4 exhibits the variations of wall and bulk temperatures against distance along the tube and it is observed that temperature distribution along the tube length decreases as the tube inclination angle changes from \( \theta = 90^\circ \) to \( \theta = 0^\circ \) for pure oil. This can be attributed to the effect of free convection (buoyant motion) which is more dominant in the horizontal tube compared with the vertical one. This could be explained by the fact that in the horizontal case the buoyant motion acts perpendicular to the forced motion. As a result, this enhances the fluid mixing and the secondary flow pattern as \( \theta \) moves from \( \theta = 90^\circ \) to \( \theta = 0^\circ \). This secondary flow assists the main flow in removing heat from the wall and as a result;
the wall temperature of the tube reduces by decreasing the tube inclination angle from vertical position to the horizontal one as seen in Fig. 4. This can be attributed to the fact that only one component of the net body force is driving the cross sectional secondary flow due to inclination which results in a weaker secondary flow current and less variation of the wall and bulk temperatures inside vertical and inclined tubes than horizontal one. 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. 4, the variation of wall temperature in the entrance region is in agreement with the results expected from theory [14,15]. Fig. 5 illustrates local Nusselt number versus dimensionless length of tube inside plain tube. According to Fig. 5, local Nusselt number decreases along the tube for pure oil and nanofluid with the weight fraction of 0.2% in three different tube inclinations. Also, Fig. 5 indicates that by increasing inclination, a decrease in the value of Nusselt number is observed for the base fluid and nanofluid 0.2 wt.%. This can be explained by the fact that the magnitude of  and decreases as the tube inclination is changed from vertical to horizontal (see Fig. 4). This is due to the decrease of buoyancy force, natural convection effect and the secondary flow by increasing inclination. Thus, this results in higher convection heat transfer coefficient and Nusselt number values for horizontal tube compared with the inclined and vertical tubes as seen in Fig. 5. In addition by observing Fig. 5, an enhancement in the value of Nusselt number is seen when nanofluid 0.2 wt.% is used as the working fluid instead of pure oil. This enhancement is due to test experiments that were conducted under a given Grashof number when nanoparticles were used as the working fluid. In order to obtain the same  the heat flux needs to be higher for higher nanoparticle fractions at the same mass flow rate. This results in the enhancement of Prandtl number due to the intensified Brownian motion of nanoparticles [7,17,20]. As a result, the enhanced  leads to more contribution of natural convection to overall convection heat transfer. Thus, Nusselt 60 inside microfin tube. It can be observed that the Nusselt number increases considerably with an augmentation of Grashof number in both tubes especially for the horizontal inclination which is in agreement with Refs. [4–8]. This behavior indicates that natural convection (the buoyant motion) is more dominant in the horizontal tube compared with the vertical one. This could be explained by the fact that in the horizontal case the buoyant motion acts perpendicular to the forced motion. As a result, this enhances the fluid mixing and the secondary flow which results in higher heat transfer [20]. Moreover, it can be observed in Fig. 7 that utilizing nanofluid instead of base fluid leads to an enhancement of Nusselt number. This is attributed to test experiments that were performed under a given Grashof number when nanofluids were used as the working fluid. In order to obtain the same  the heat flux needs to be higher for higher nanoparticle fractions at the same mass flow rate. This results in the enhancement of Prandtl number. Mean Nusselt number against Grashof number is presented in Fig. 7 for two tube inclination angles namely, 0° and 90° at the

**Fig. 5.** Variation of local Nusselt number versus dimensionless inlet length for plain tube. $q'' = 6400 \text{ W/m}^2$, $Re = 15$, $Ri = 5$.

**Fig. 6.** Effect of heat flux on local Nusselt number in horizontal plain tube for pure oil, $Re = 15$ and $Ri = 5$.

**Fig. 7.** Variation of mean Nusselt number versus Grashof number in microfin tube for two weight fractions (%) at $Gz = 60$. 
The ratio of mean Nusselt number of nanofluids to that of base fluid in two fractions versus Richardson number is displayed in Fig. 8 for both horizontal microfin and plain tubes. As shown in Fig. 8, this ratio is more than one for all fractions and Richardson numbers, which means utilizing nanoparticle in the base fluid, enhances the Nusselt number. According to Fig. 8, mean Nusselt number ratio of nano fluid to base fluid is decreased by increasing Ri which means that the effect of nanoparticles on Nusselt number enhancement is decreased by increasing Richardson number. In other words, increasing the nanoparticles mass fraction causes deterioration in values of buoyancy forces and natural convection effects due to an increase of dynamic viscosity. Adding nanoparticles boosts the suppression of the natural convection effects by increasing Richardson number that finally leads to deteriorating the effect of nanoparticles on heat transfer enhancement. Therefore, a decrease in values of mean Nusselt number ratio of nano fluids to base fluid can occur in mixed and natural convection flow by raising Ri [7,8]. Also it can be observed from this figure that the values of Nusselt number ratio for plain tube is more than that of the microfin one at the same Richardson number, which means that the effect of nanoparticles on heat transfer enhancement in plain tube is more than microfin tube. This is due to fine numerous fins inside the microfin tube that cause a disturbance in random molecular motion of nano fluids and their movements (Brownian motions) when nano fluids interact with these fins. Thus, it leads to a decrease in the value of thermal conductivity of the nano fluid in microfin tube which results in deteriorating the effect of nanoparticles on heat transfer enhancement. Thus, a decrease in the value of Nusselt number ratio in microfin tube can be observed in comparison with the plain tube at the same Ri. However, the enhancement of heat transfer in the microfin tube is more than that of the plain one in each nanoparticles mass fraction [12]. Nusselt number ratio is about 22% at the Ri = 0.1 for plain tube which happens at the maximum mass fraction of 0.2 wt.% This ratio in plain tube is about 5% more than the corresponding value for microfin tube at the same Ri. Thus, in order to have the maximum effect of nanoparticles on heat transfer enhancement during laminar mixed convection of buoyancy assisted flow, adding nanoparticles in plain horizontal tube at Richardson numbers lower than 1 is recommended.

Fig. 9 demonstrates the ratio of mean Nusselt number of nano fluid with 0.2 wt.% fraction to that of base fluid for three inclinations inside microfin tube. As observed in Fig. 9, effect of nanoparticles on Nusselt number enhancement decreases by increasing inclination from $\theta = 0^\circ - 90^\circ$. This can be explained by the fact that when the inclination increases the flow is under buoyancy assisted flow (upward) with a weaker secondary flow. As a result, the random molecular motion of nanoparticles decreases which leads to deteriorating movements of nanoparticles and their Brownian motions. This results in a decrease in the value of thermal conductivity of nano fluids which leads to decreasing values of convection heat transfer coefficient and Nusselt number enhancement. Thus, the value of Nusselt number ratio decreases by increasing inclination at a fixed $R_i$. This ratio is maximum about 18% in horizontal microfin tube at $R_i = 0.1$. Furthermore, as shown in Fig. 9, the effect of inclination is increased by raising Richardson number. Also from this figure it could be concluded that the effect of nanoparticles is decreased by increasing Richardson number. Therefore, the effect of nanoparticles is more in low inclinations and low Richardson numbers. As a result, in order to have the maximum effect of nanoparticles, it is recommended to make use of them in inclinations lower than 45° and Richardson numbers lower than 1.

6. Conclusion

In this study, developing laminar mixed convection flow of MWCNT (carbon nano-tube) – oil based nano fluid with different tube inclinations in plain and microfin tubes has been experimentally investigated under uniform heat flux and buoyancy assisted flow conditions. The results have clearly shown the following conclusions:

1. Nusselt number decreases by increasing of inclination from $\theta = 0^\circ - 90^\circ$ in tubes under the assisted flow condition. Thus, the most values of Nusselt number occur in the horizontal inclination as the effect of buoyancy forces, natural convection and secondary flow is in the maximum value in this inclination. In addition, the effect of inclination is more pronounced in higher Richardson numbers.
2. Nusselt number augments by increasing of Grashof number for all the tested tubes with inclination from $\theta = 0^\circ$ to $90^\circ$ and nanofluid weight fractions from 0 to 0.2%. Increasing of the Grashof number results in buoyancy forces enhancement. It has been found that the maximum nanoparticles weight fraction of this study (0.2 wt.%) clearly induces an augmentation of the Nusselt number for all the inclinations under a given Grashof number. However, increasing the nanoparticles mass fractions causes deterioration in the natural convection effects. Effects of nanoparticles on enhancement of $Nu$ are decreased by increasing inclination and Richardson number. Therefore, in order to have the maximum effect of nanoparticles on enhancement of heat transfer, using horizontal plain tube, at Richardson numbers lower than 1 and inclinations lower than $45^\circ$ is recommended.

3. When only the effect of nanoparticles is considered, plain tube is better than the microfin one. Thus with the aim of heat transfer augmentation by using nanoparticles, plain tube is recommended but if both tube and nanoparticles are of notice to have the maximum enhancement, then microfin tube is recommended. The maximum enhancement of the mean Nusselt number due to presence of nanoparticles is about 22 and 18% for horizontal plain and microfin tubes respectively which occurs at the nanoparticles weight fraction of 0.2% and $Re = 0.1$.

References