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RESEARCH ARTICLE - MECHANICAL ENGINEERING

Forced Convective Heat Transfer of MWCNT/Water Nanofluid Under Constant Heat Flux: An Experimental Investigation

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Abstract This research article investigates the effect of multi-walled carbon nanotubes (MWCNT)/water nanofluid on convective heat transfer in a uniformly heated copper tube under laminar flow regime. The MWCNT were synthesized using chemical vapor deposition method and characterized using transmission electron microscope. These nanoparticles were dispersed (with 0.05, 0.1, 0.3 and 0.5 % weight concentrations) in distilled water to form stable suspensions of nanofluids. The heat transfer coefficients (HTC) of nanofluids and distilled water (base fluid) were evaluated and compared using constant velocity basis. The thermophysical properties of nanofluids change with the addition of nanoparticles; thus, we have considered the constant velocity criteria which provide the true comparison in contrast to constant Reynolds number used by earlier researchers. The effect of flow velocity (0.166–0.232 m/s) and nanoparticles weight concentration on the HTC considering constant heat flux boundary conditions was studied. It is observed that with the increase in the weight concentration of nanoparticles or flow velocity, the HTC increases. Nanofluids show higher HTC with respect to distilled water at all the concentrations of nanoparticles. At 0.5 wt.% weight concentration and flow velocity of 0.232 m/s, the maximum HTC obtained is 77.60 % in comparison with distilled water.

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Keywords Nanofluids · Laminar flow · Heat transfer enhancement · Constant heat flux

List of symbols

c_p	Specific heat (J/kg K)
D	Diameter of copper tube (m)
h	Heat transfer coefficient (W/m^2 K)
λ	Thermal conductivity (W/m K)
L	Tube length (m)
m	Mass flow rate (kg/s)
Nu	Nusselt number
q''	Heat flux (W/m^2)
Pr	Prandtl number
Re	Reynolds number
T	Temperature (°C)
x	Distance from the pipe inlet (m)
A	Heat transfer area (m^2)
I	Current (A)
Q	Heat transfer (W)
U	Velocity (m/s)
V	Voltage (V)
ρ	Density (kg/m^3)
ϕ	Volume fraction
μ	Dynamic viscosity (Pa s)
ΔP	Pressure drop (Pa)

Subscripts

f	Fluid
i	Inlet
m	Bulk
nf	Nanofluids
bf	Base fluid
p	Particle
s	Surface
x	Local

1 Introduction

The traditional heat transfer fluids such as water, propylene glycol, oil, gear oil, ethylene glycol and paraffin are extensively used for heat transfer purposes in several industries such as electronics, transportation, chemical processes, power plant, air-conditioning, food processing and nuclear reactors. Numerous active and passive techniques are available in the literature which aids in enhancing the heat transfer characteristics of conventional fluids [1–7]. But the heat transfer augmentation through these techniques has reached a bottleneck. The conventional fluids used for heat transfer have poor thermal performance; hence, these are primary obstacles in the advancement of future energy efficient heat exchangers. In order to resolve this problem, there is a need to develop new cutting-edge heat transfer fluids which can provide better thermophysical properties leading to enhanced heat transfer. The solid materials have better thermal properties compared to fluids. Researchers have considered the suspension of solid particles in conventional heat transfer fluids in order to study their thermal properties. These dispersed millimeter- or micrometer-size solid particles in the base fluid result in alteration of thermophysical properties of the base fluids that eventually lead to heat transfer improvement [8]. Nevertheless, the use of these fluids is restricted due to some serious complications such as stability of suspensions, blockage, erosion of pipelines as well as increase in pressure drop. Today with the rapid growth in material technology, nanometer-sized particles can be produced. The suspension of this nanometer-sized (10^{-9} m) (metallic and nonmetallic) nanoparticles in a base fluid (water, ethylene glycol, oil etc.) is termed as nanofluids. This term was first used by Choi [9] at Argonne National Laboratory. The nanoparticles enjoy better heat transfer characteristics such as high thermal conductivity, better stability, homogeneity along with the insignificant blockage in flow passages due to their minute size and large specific area [10]. These advantages had motivated the researchers to perform studies on the heat transfer performance taking various nanofluids, i.e., different combination of nanoparticles and base fluids. The studies were conducted considering practical applications of convective heat transfer in laminar and turbulent flow with different boundary conditions [11–14]. Pak and Cho [15] were the first to report heat transfer performance considering alumina–water and titania–water nanofluids in a straight tube using constant heat flux (CHF) boundary conditions under turbulent flow regime. The experimental findings showed that the Nusselt number of nanofluids improved with growth in the Reynolds number and the volume concentration. But the convective heat transfer of nanofluids with 3 vol. % nanoparticles was 12 % lesser than water at the prescribed conditions.

Chen et al. [16] studied heat transfer and flow behavior of titanate nanotubes dispersed in water (nanofluids). The experimental results depicted a small improvement in thermal conductivity (~3 % at 25 °C and ~5 % at 40 °C) with 2.5 wt.% of titanate nanotubes (nanofluids). Although the improvement in thermal conduction enhancement was small, the augmentation in the convective heat transfer coefficient was found to be excellent. The results showed that the heat transfer phenomenon was not only dependent on the increase in the thermal conductivity but other factors also played an important role. Garg et al. [17] studied the influence of ultrasonication on viscosity as well as heat transfer performance of multi-wall carbon nanotube–water nanofluids under CHF boundary condition. The maximum improvement in thermal conductivity was found to be 20 %, and it showed considerable increase after 24 °C. Further, the maximum enhancement in heat transfer coefficient at a Reynolds number of 600 ± 100 was found to be 32 %. The percentage enhancement in heat transfer coefficient continuously improved along the axial distance. Amrollahi et al. [18] experimentally measured the convective heat transfer coefficients of water-based FWNT nanofluid through a uniformly heated horizontal tube in the entrance region considering both laminar and turbulent flow. The Reynolds number and mass fraction along with temperature were compared in the entrance region in order to compute the convective heat transfer coefficient (CHTC) of nanofluids (functionalized MWNT suspensions). The experimental results indicated that taking concentration of 0.25 wt.%, the convective HTC of these nanofluids augmented by up to 33–40 % with respect to base fluid (pure water) in laminar as well as turbulent flows, respectively (at 20 °C). Suresh et al. [19] performed experiments in a plain and helically dimpled tube to find the convective heat transfer and friction factor characteristics of CuO/water nanofluids under turbulent flow. The results presented that for dimpled tube with nanofluids having 0.1, 0.2 and 0.3 % volume concentrations, the corresponding value of Nusselt number was 19, 27 and 39 %, greater than obtained with plain tube and water under turbulent flow. Further results indicated that the dimpled tube friction factors were around 2–10 % greater than the plain tube of isothermal pressure drop for same flow conditions.

An empirical study was performed by Hashemi and Akhavan-Behabadi [20] in a horizontal helically coiled tube to find out the heat transfer and pressure drop characteristics of CuO–base oil nanofluid. It was observed that the nanofluids showed improved heat transfer characteristics flowing through helical tube instead of straight tube. The maximum heat transfer augmentation of 18.7 and 30.4 % was attained for nanofluid flow with 2 wt.% concentrations inside the straight tube and helical tube, respectively, as compared to base oil. Selvakumar and Suresh [21] showed the convective heat transfer performance of water-based nanofluid in an electronic heat sink. As the volume flow rate and nanoparticles volume fraction increased, the convective heat transfer coefficient of water block was found to be increased. The maximum rise of 29.63 % for the 0.2 % volume fraction was observed as compared to deionised water. Based on the pressure drop in the water block, pumping power for the deionised water and nanofluids were calculated and the average increase was 15.11 % for the nanofluid having 0.2 % volume fraction with respect to base fluid. Liu and Liao [22] experimentally studied the forced convective heat transfer characteristics of aqueous drag-reducing fluid by adding the carbon nanotubes. The new working fluid was an aqueous CTAC (cetyltrimethyl ammonium chloride) solution with CNTs. It has special effects of drag-reducing and heat transfer enhancement. Results indicated that there were no noticeable differences of the drag-reducing characteristics between conventional drag-reducing fluid and new drag-reducing nanofluid. However, there were evident differences of the heat transfer characteristics between both fluids. The heat transfer characteristics of new drag-reducing nanofluid have strong dependencies on the fluid temperature, the concentration of both nanoparticles and CTAC. Further Wang et al. [23] reported the heat transfer and pressure drop in a horizontal circular tube considering nanofluids containing carbon nanotubes (CNT). A substantial improvement in the average convective heat transfer was witnessed compared with the distilled water. At a Reynolds number of 120, the nanofluids with volumetric concentration (0.05 and 0.24 %), the heat transfer enhancement was found to be 70 and 190 %, respectively. The enhancement of thermal conductivity was found to be less than 10 %. It was concluded that the large heat transfer cannot be increase exclusively due to the superior thermal conductivity. Recently Gupta et al. [24] experimentally studied the heat transfer characteristics in a circular pipe using TiO_2/water nanofluids in laminar flow regime using constant flow velocity criteria.

Researchers have used nanofluids with different nanoparticles such as Al_2O_3, TiO_2, CuO, SiO_2, Fe_2O_3 Fe_3O_4. The motivation behind the present study is that very limited publications are reported on the convective heat transfer using CNT nanofluids (single-, double- or multi-walled nanotubes). Further, the carbon nanotubes display extraordinary thermal properties and structure [25]. The carbon nanotubes are considered as single-walled nanotubes and multi-walled nanotubes. This paper presents the effect of nanoparticles weight concentration in the range of 0.05–0.5 % on the heat transfer characteristics of MWCNT-distilled water nanofluid in laminar flow under CHF boundary conditions. The researchers in the previous studies on MWCNT nanofluids have used constant Reynolds number basis of comparison. This study has been performed considering the constant velocity criteria instead of traditionally used constant Reynolds number criteria.

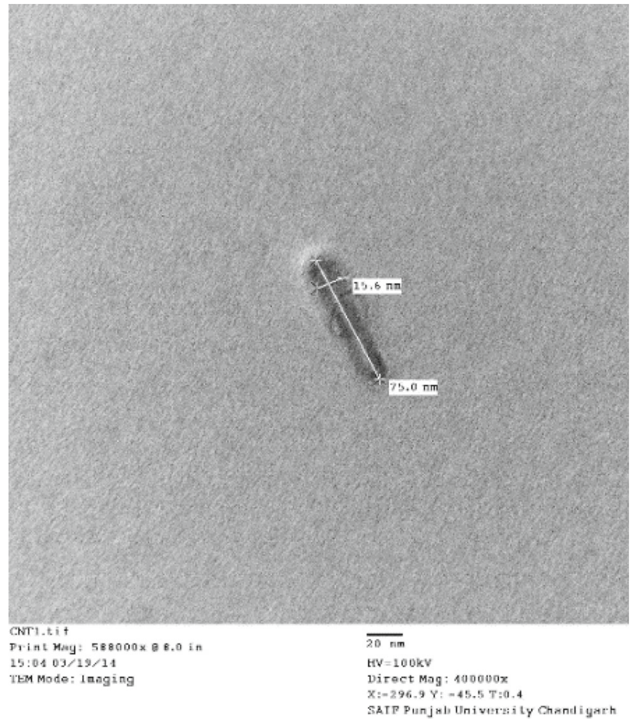


Fig. 1 TEM image of MWCNT nanoparticles

2 Synthesis and Characterization of Nanoparticles

Multi-walled carbon nanotubes (MWCNT) were prepared by the chemical vapor deposition (CVD) method at the Department of Bio-Nano Technology, GJUS&T, Hisar. The CVD system has a quartz tube, two- stage electric furnace, temperature controller and gas controller. The nanotubes were grown at 750 °C and at feed rate of 5 ml/h of a ferrocene and toluene solution (1:3). Argon gas was used as a carrier gas. First of all, the mixture was injected into the first-stage furnace and it was preheated at 350 °C to confirm that the solution is vaporized. After that, the nanotubes were grown at 750 °C in a second-stage furnace using carrier gas. After completing the growth process, the furnace was slowly cooled to room temperature.

The TEM (transmission electron microscopy) image of MWCNT nanoparticles is shown in Fig. 1. The nanoparticles appear in the form of rods in morphology. The diameter of these nanotubes is in the range of 7–20 nm and having a length of 75–88 nm.

3 Preparation of Nanofluids

In this study, two-step method was used to produce stable suspension of nanofluids by dispersing 0.05, 0.1, 0.3 and

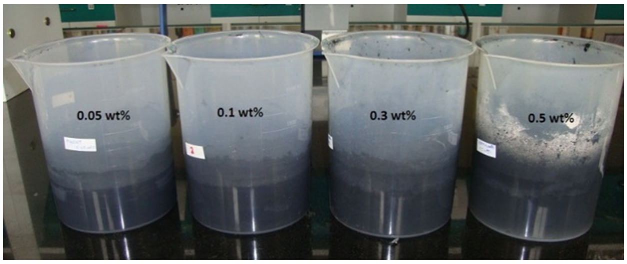


Fig. 2 MWCNT nanofluids at various particle concentrations

0.5 % weight concentrations of the MWCNT nanoparticles with an average size in distilled water. A digital electronic mass balance was used to weight the appropriate amount of nanoparticles. An ultrasonic vibrator (POWERSONIC 410, Hwashin Technology, Korea) producing ultrasonic pulses of 400 W was used for sonicating the mixture from 6–8 h. It is used to make stable suspension of nanoparticles and break down the agglomeration of nanoparticles in the fluid. No surfactant was used in the study as they may affect the thermal conductivity of nanofluids [22,25]. It was witnessed with bare eyes that during experiment work, there was sedimentation and nanofluids were uniformly dispersed for 10 h. The dispersed MWCNT nanoparticles in water base fluid are represented in Fig. 2.

4 Experimental Set Up

An experimental setup was assembled to investigate the convective heat transfer characteristics of MWCNT/water nanofluids flowing through a tube under constant heat flux boundary conditions. The experimental unit consists of a peristaltic pump, test section, a reservoir tank, cooling unit, DC heating section and thermocouples. The test section contains a straight copper pipe of 1.05 m length with 8 mm inner diameter and 1 mm thickness. Thereafter, the electrical nichrome heating wire having a high melting point of about 1400 °C is enfolded over it. Ten calibrated J-type thermocouples with 0.1 °C resolution, attached on the wall of copper tube at regular intervals to appraise the wall temperature. Two thermocouples are introduced into the flow at the inlet and outlet of the test section to measure bulk temperatures of nanofluids. Over the electrical winding, a thick thermal insulation consisting of glass wool is provided to avoid the radial heat loss. This heated nichrome wire is connected to a DC power supply which is tunable with a Variac transformer (range 0–240 V). The test section is heated by nichrome wire. A digital voltmeter and ammeter are attached to determine the power input to the nichrome heating wire. Digital temperature indicators with 0.1 °C resolution display the temperature of all thermocouples mounted in the test section. The flow rates were calculated by accumulating the fixed volume of fluid with the help of an accurate measuring jar (1000 ml) with 10-ml resolution and measuring the time with stop watch having 0.1-s resolution. The schematic of the experimental setup is shown in Fig. 3a, b.

The nanofluids having different weight fraction (0.05, 0.1, 0.3 and 0.5 %) of MWCNT in distilled water are pumped in the test section with the help of peristaltic pump. The working fluid is uniformly heated with the nichrome wire. After passing through the test section, nanofluid is passed through the cooling section where its temperature is reduced. Lastly, the flow is measured in the flow measuring section. During experimental test runs, the inlet and outlet temperature of nanofluids and the wall surface temperature at regular intervals were measured. Each reading was repeated minimum six times.

5 Data Reduction

Before performing the experiments, the MWCNT nanoparticles were dispersed in distilled water with 0.05, 0.1, 0.3 and 0.5 wt.% in order to produce nanofluids. The thermophysical properties of nanofluids were evaluated at the mean bulk temperature. The values of density, specific heat, viscosity and thermal conductivity for nanofluids were calculated using the Eqs. (1)–(4) [26–29].

$$\rho_{nf} = \phi\rho_p + (1-\phi)\,\rho_{bf} \tag{1}$$

$$\left(\rho C_p\right)_{nf} = \phi\left(\rho C_p\right)_p + (1-\phi)\left(\rho C_p\right)_{bf} \tag{2}$$

$$\mu_{nf} = (1+2.5\phi)\,\mu_{bf} \tag{3}$$

$$\lambda_{nf} = \frac{2\lambda_{bf} + \lambda_p + 2\phi\left(\lambda_p - \lambda_{bf}\right)}{2\lambda_{bf} + \lambda_p - \phi\left(\lambda_p - \lambda_{bf}\right)}\lambda_{bf} \tag{4}$$

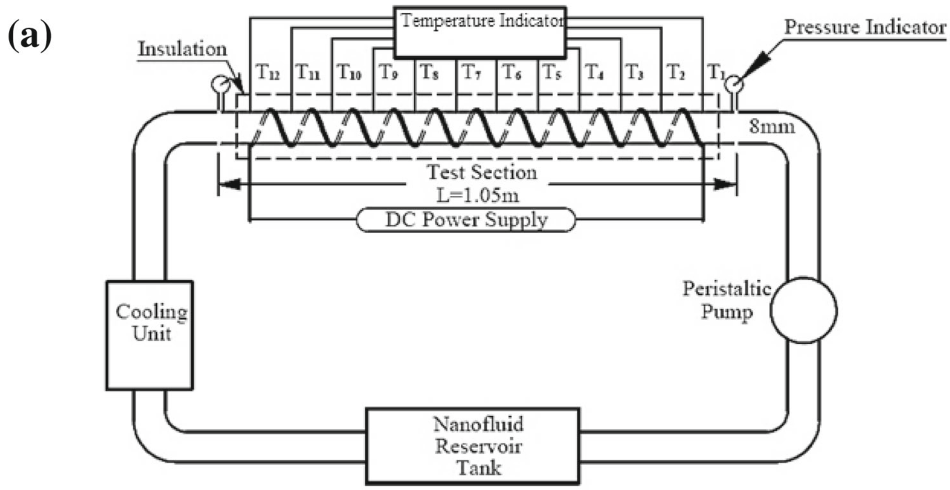
The thermophysical properties of MWCNT nanoparticles and distilled water used in the current study are given in Table 1. The thermophysical properties of nanofluids at different concentration using equations mentioned above are evaluated in Table 2.

The thermophysical properties are used to appraise the convective heat transfer coefficient and Nusselt number of base fluid and nanofluids at different weight concentrations. Considering the test section to be well insulated and neglecting heat loss, heat flow can be considered equal to the power input. Heat supplied to the test section can be calculated by:

$$Q_a = V \times I \tag{5}$$

Heat absorbed by the nanofluid can be appraised as:

$$Q_b = \dot{m}c_p\left(T_{out} - T_{in}\right) \tag{6}$$



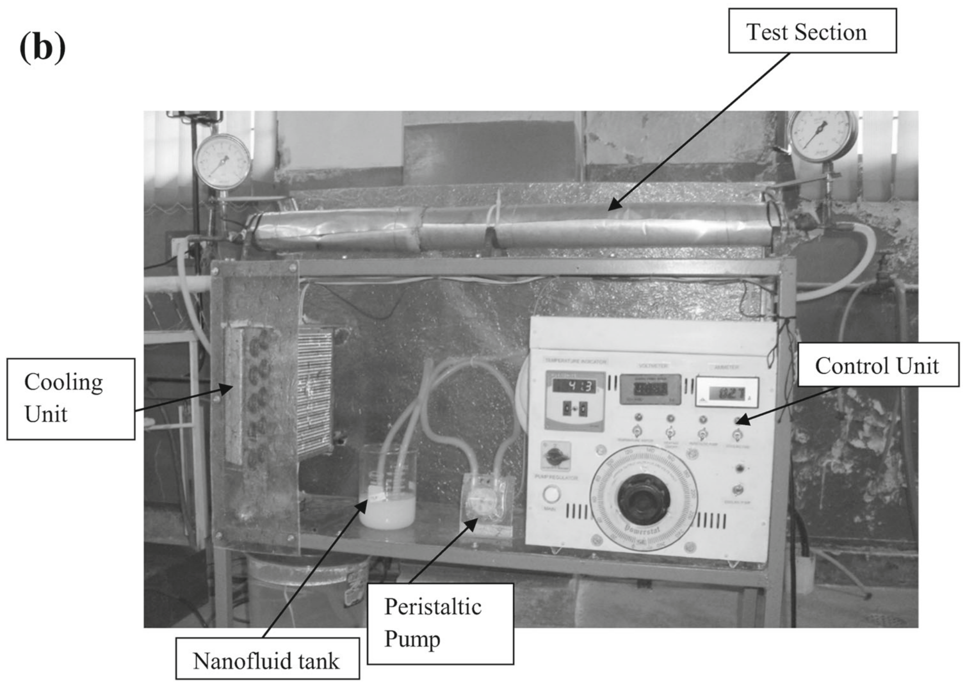


Fig. 3 **a** Schematic view of the experimental setup. **b** Experimental setup for investigation the convective heat transfer characteristics of nanofluids

Table 1 Thermophysical properties of distilled water and MWCNT nanoparticles at 30 °C

Substance/nanoparticles	Density (kg/m^3)	Thermal conductivity (W/m K)	Specific heat (kJ/kg K)	Viscosity (kg/ms)
Distilled water	1000	0.62	4.187	0.000798
MWCNT	2100	3000	0.702	–

Table 2 Thermophysical properties of nanofluids at different concentrations

Nanofluids	wt.%	Vol.%	Density (kg/m^3)	Thermal conductivity (W/m K)	Specific heat (kJ/kg K)	Viscosity (kg/ms)
MWCNT/water	0.05	0.02380	1026.2	0.6653	4017.3	0.00084548
MWCNT/water	0.1	0.047596	1052.4	0.7129	3856.0	0.00089295
MWCNT/water	0.3	0.142653	1156.9	0.9293	3284.6	0.0011
MWCNT/water	0.5	0.237529	1261.3	1.1990	2808.8	0.0013

where 'T_{out}' and 'T_{in}' are the temperatures at the outlet and entry of the test section, respectively. Heat flux is given by:

$$q'' = \frac{Q}{\pi DL} \tag{7}$$

where 'D' is the inner diameter of copper tube, 'L' is the length of the test section and 'Q' is the heat transfer rate which can be determined by Eq. (8).

$$Q = \frac{Q_a + Q_b}{2} \tag{8}$$

The heat transfer coefficient convective 'h' of the working fluids can be calculated by:

$$h = \frac{q''}{T_s - T_m} \tag{9}$$

where,

$$T_m = \frac{T_{in} + T_{out}}{2} \tag{10}$$

and

$$T_s = \frac{\left(\sum_{n=2}^{11} T_n\right)}{10} \tag{11}$$

where 'T_n' and 'T_s' are bulk and mean surface temperatures, respectively. The Nusselt number can be calculated by:

$$Nu = \frac{h \times D}{\lambda_{nf}} \tag{12}$$

Now in order to calculate the local heat transfer coefficient and local Nusselt number, replace 'h' by h(x) and Nu by Nu(x). The values can be calculated as

$$h(x) = \frac{q''}{T_s(x) - T_m(x)} \tag{13}$$

$$Nu(x) = \frac{h(x) \times D}{\lambda_{nf}} \tag{14}$$

where the value of $T_m(x)$ can be measured from:

$$T_m(x) = T_{m,i} + \frac{q'' P}{m \times C_p} x \tag{15}$$

The term 'P' is the perimeter, 'm' is the mass flow rate and 'x' is the axial distance from the entrance of the test section.

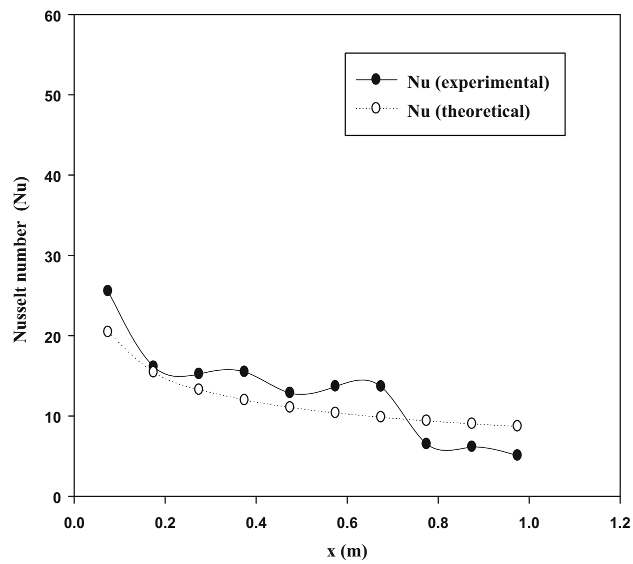


Fig. 4 Comparison of the measured local Nusselt number with the empirical correlation for distilled water

6 Result and Discussion

6.1 Validation of the Experimental Setup

In order to confirm the consistency and correctness of the experimental setup, preliminary experiments were performed using distilled water as the working fluid under laminar flow conditions. For the convective heat transfer (forced), the results predicted from Shah equation for laminar internal flow [30], under constant heat flux boundary conditions, were compared with our experimental results along the axial distance of the test section. The Nusselt number results were measured in many runs to obtain repeatability of the experiment. Figure 4 shows the evaluation of experimental results with Shah equation for local Nusselt number along the axial direction at a fixed Reynolds number (in the present study, it was kept 1995) under constant heat flux.

$$Nu(x) = \begin{cases} 1.953 \left(Re.\Pr.\frac{D}{x}\right)^{1/3} & \left(Re.\Pr.\frac{D}{x}\right) \geq 33.3 \\ 4.364 + 0.0722 \left(Re.\Pr.\frac{D}{x}\right) & \left(Re.\Pr.\frac{D}{x}\right) < 33.3 \end{cases} \tag{16}$$

It can be observed from Fig. 4 that there is a reasonable agreement between the experimental results and the calculated values for distilled water.

6.2 Heat Transfer

In the present investigation, the laminar forced convective heat transfer of MWCNT/distilled water nanofluids passing through a circular tube considering uniform heat flux bound-

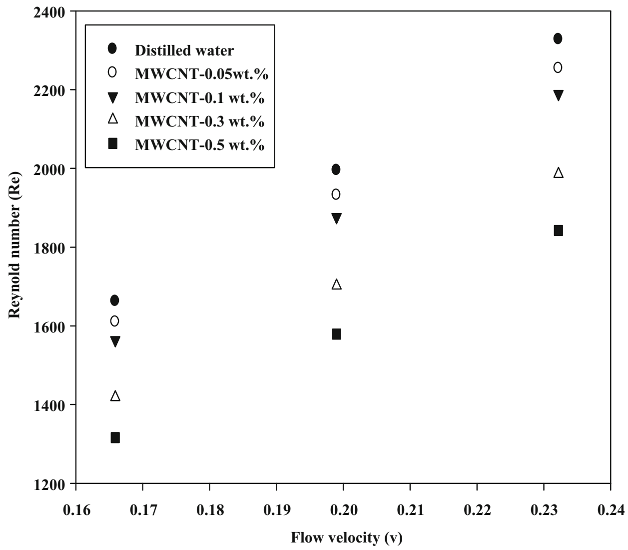


Fig. 5 Variation of Reynolds number with respect to the flow velocity

ary condition is experimentally measured. The MWCNT/distilled water nanofluids with four weight concentration (0.05, 0.1, 0.3, 0.5 wt.%) were used in the present study. The performance has been evaluated using the constant velocity criteria where the flow velocity is varied from (0.1666–0.232 m/s). In this velocity range the Reynolds number varies from 1300 to 2300.

In previous studies [13,22,23], the constant Reynolds number criteria have been used for comparing nanofluids with base fluid. As we know that Reynolds number is a function of thermophysical properties (density, velocity and dynamic viscosity), if the thermophysical properties of the fluid remain same or same fluid is used, the Reynolds number can be used as the basis of comparison. But the nanofluids with different weight concentrations have different thermophysical properties than base fluid (water in the present study). Figure 5 illustrates the variation of the Reynolds number for different concentrations of nanofluids with respect to flow velocity.

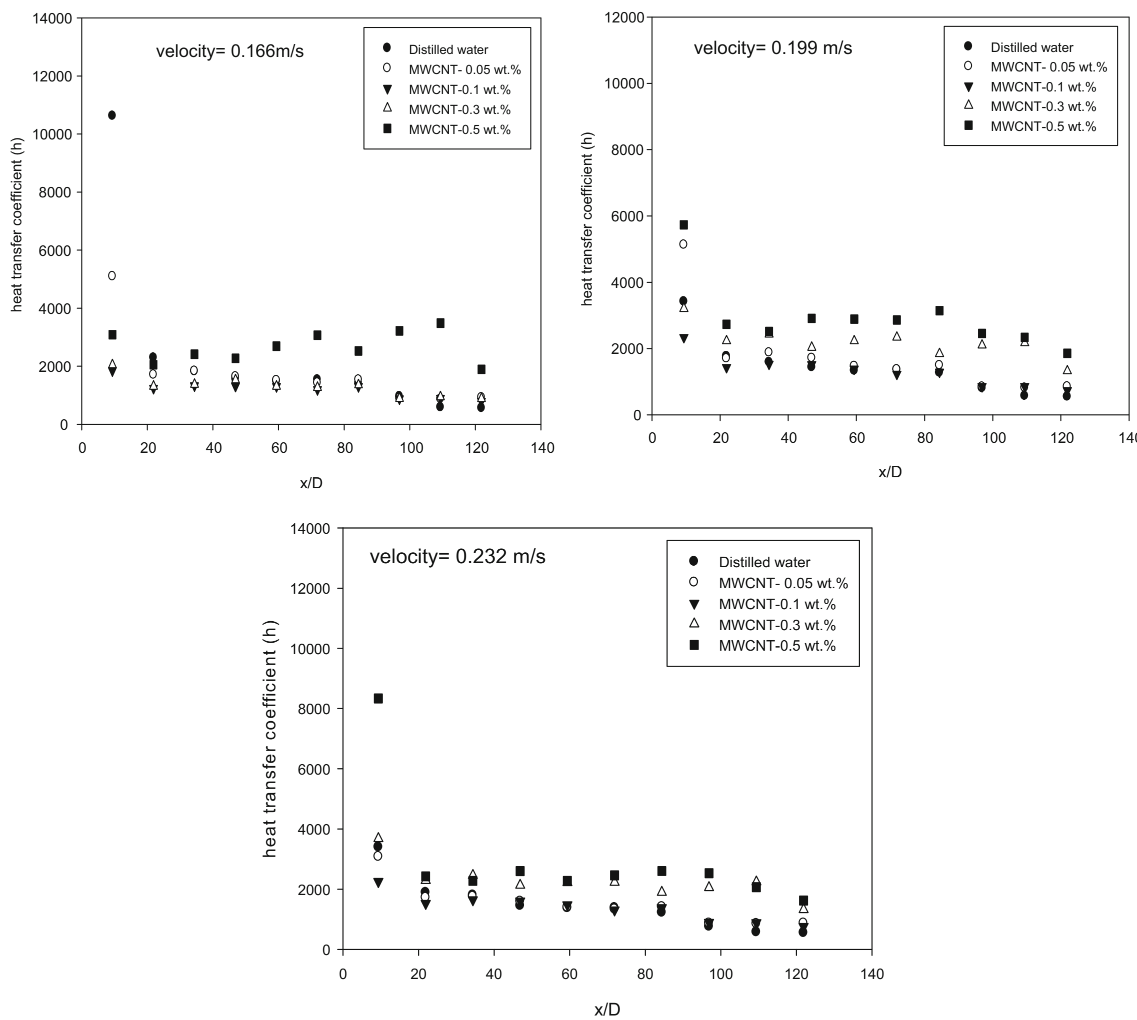


Fig. 6 Local heat transfer coefficient variation of nanofluids along non-dimensional axial distance at different flow velocities

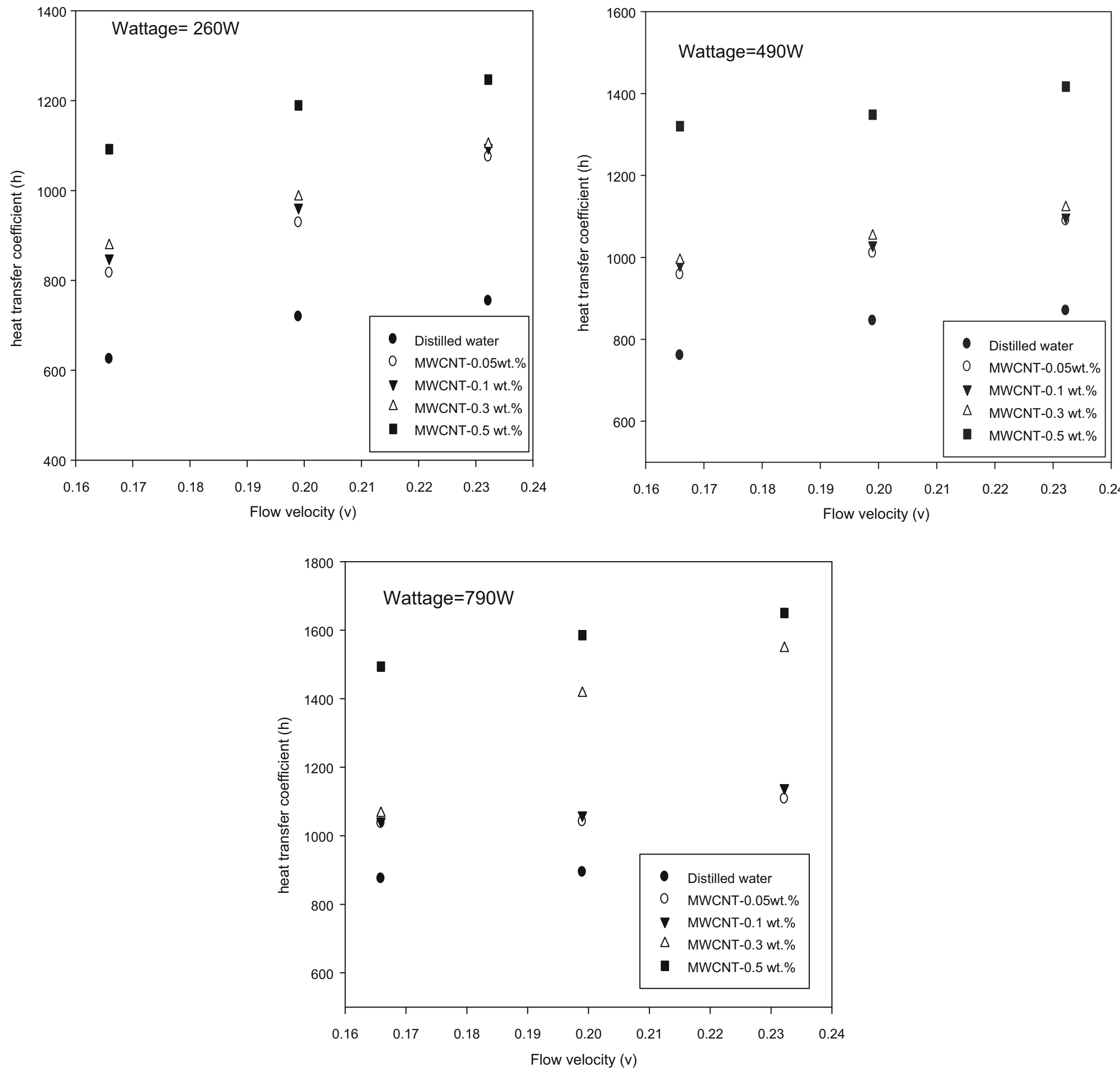


Fig. 7 Heat transfer coefficient variation of nanofluids versus flow velocity

It can be depicted that the Reynolds number increases with rise in the flow velocity for fixed concentration of nanofluids. However, substantial reduction in the Reynolds number can be observed at a fixed flow velocity, with the increase in concentration of MWCNT nanoparticles. The Reynolds number reduces from 1611 to 1316, from 1933 to 1579 and 2255 to 1842 at flow velocities 0.166, 0.199 and 0.232 m/s, respectively, as the concentration of nanoparticles increases from 0.05 to 0.5 wt.%. The results demonstrate that at 0.5 wt.% of MWCNT in base fluid, the Reynolds number decreases by about 26 % as compared to the base fluid. The main reason behind the reduction in Reynolds number can be the increase in viscosity along with the increase in particle concentrations.

Due to the reduction in the Reynolds number at a particular flow velocity with small addition of MWCNT nanoparticles, the Reynolds number cannot be suitable for comparing the convective heat transfer characteristics of nanofluids as compared to the base fluid [31,32]. Because if we take a fixed value of Reynolds number, the flow velocity of nanofluids will vary at different concentrations of MWCNT nanoparticles. The flow velocity is a function of convective heat transfer coefficient. Hence, in this paper, the constant flow velocity of nanofluids and base fluid, instead of the commonly used fixed Reynolds number comparison, is considered as the basis of comparison for convective heat transfer coefficient.

Figure 6 demonstrates the comparison of local heat transfer coefficient along non-dimensional axial distance for nanofluids and distilled water at constant flow velocity. The results show that the maximum enhancement rate is attained at the inlet of the test section and it diminishes as we go away in the axial direction from the inlet of the test section. This fact is based on the thermal boundary layer theory. The thermal boundary layer is small in the beginning of the test section. Hence, the thermal resistance is less which leads to

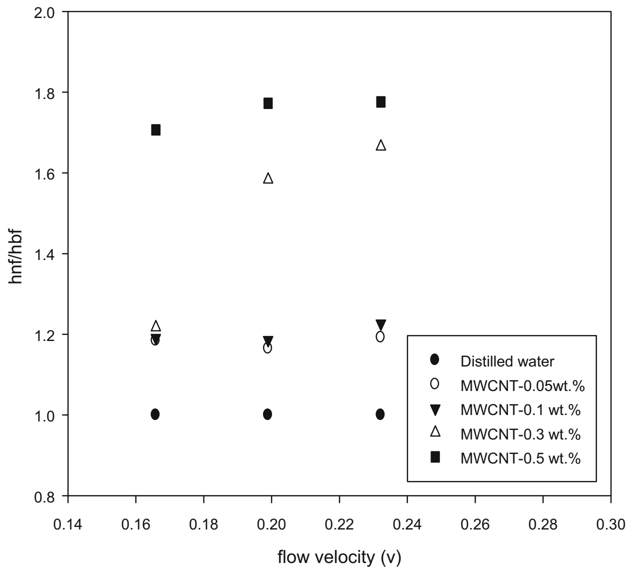


Fig. 8 Heat transfer coefficient ratio versus flow velocity

higher heat transfer coefficient at the inlet of the test section. Moving along the axial direction away from the inlet, the thermal boundary layer becomes developed and due to this fact, the thermal resistance increases and the heat transfer reduces. Further with the increase in the concentration, the heat transfer coefficient is higher than the distilled water. The values of heat transfer coefficient increase with the increase in the flow velocity from 0.166 to 0.232 m/s. This may be attributed to the increase in the mixing and further thermal dispersion of nanoparticles leading to the flattened temperature distribution and sharp temperature gradient between wall and the nanofluids.

Figure 7 shows the heat transfer coefficient of the nanofluids and distilled water at different weight concentrations for flow velocities varying from 0.166 to 0.232 m/s at different heat flux. The results show that the heat transfer for nanofluids is greater than distilled water and it increases with the increase in the weight concentration for a fixed flow velocity. Further, it increases with the increase in the flow velocity.

The presence of nanotubes powder provides higher heat transfer coefficients than distilled water. The nanoparticles when added improve the thermal conductivity of the base fluid [33]. The enhancement of thermal conductivity further increases with increase in the particle concentration. Previous researches [34–37] show that the improvement in thermal conductivity further improves the convective heat transfer. The Brownian motion, shrinking of boundary layer thickness and arbitrary movement of dispersed particles may also be the probable reasons for this enhancement. The disordered movement of the nanoparticles in flow disturbs the thermal boundary layer development on the surface of the wall. This disturbance leads to delay in thermal boundary layer. Thus, higher heat transfer coefficients of fluid are achieved. At higher weight concentrations, both the chaotic movement of the solid nanoparticles and the thermal conductivity increase. This may be the reason for higher heat transfer coefficient at higher concentrations. Further with the increase in the heat flux, heat transfer shows the augmentation. The maximum enhancement of convective heat transfer coefficient is found to be 77.60 % for the 0.5 wt.% of MWCNT-distilled water nanofluid at 790 W with flow velocity of 0.232 m/s compared to the distilled water.

Figure 8 shows the ratio of the convective heat transfer coefficient of nanofluid to that of distilled water at the same flow velocity for different weight concentrations of nanofluid. This figure is used to make comparison between the heat transfer performances of the nanofluid and base fluid. The flow velocity has a little effect on the heat transfer enhancement for a fixed weight concentration. The ratio of heat transfer coefficient varies from 1.1648 to 1.7760. The maximum enhancement of convective heat transfer coefficient is found to be 77.60 % for the 0.5 wt.% of MWCNT-distilled water nanofluid.

6.3 Uncertainty Analysis

The systematic error analysis in the measurement of experimental analysis is estimated following the procedure given

Table 3 Uncertainties of measuring instruments and thermal properties

Name of instrument	Variables measured	Least division (measuring instrument)	Max. values (measured in experiment)	Uncertainty %
Thermocouple	Wall temperature, T_w	0.1	120	0.0833
Thermocouple	Bulk temperature T_b	0.1	82	0.12195
Voltage	Voltage	0.1	220	0.04545
Current	Current	0.01	10	0.1
Volume of fluid	Volume	1 ml	1000 ml	0.1
Time taken	Time	0.01 s	60 s	0.01667
Thermal conductivity, viscosity				0.1

Table 4 Summary of uncertainty analysis

S. no.	Parameter	Uncertainties (%)
1	Discharge, q	0.101379
2	Mass flow rate, m	0.101379
3	Reynolds number, Re	0.1423997
4	Heat flux, q''	0.10984
5	Heat transfer coefficient, h	0.11644
6	Nusselt number, Nu	0.153487

by Beckwith et al. [38]. The uncertainties in the values estimated are summarized and presented in Tables 3 and 4.

7 Conclusions

This research article presents an investigation on the convective heat transfer characteristics of MWCNT/water nanofluids in laminar flow through a horizontal copper tube under CHF. Multi-walled nanotubes were synthesized using CVD method, characterized and scattered in the base fluid. The influence of the flow velocity, weight concentration and power supply on nanofluids is investigated. The study is performed taking constant velocity as the basis of comparison. The following conclusions are achieved based on the experimental results.

- The addition of a small weight concentration of MWCNT nanoparticles in base fluid at fixed velocity reduces the Reynolds number. By the addition of 0.5 wt. % of MWCNT in base fluid, the Reynolds number decreases about 26 % with respect to the base fluid. Thus, constant Reynolds number comparison does not provide true outcomes. At fixed Reynolds number, the flow rate of nanofluids varies due to increase in viscosity with particle loading.
- The local heat transfer coefficient depends on the axial distance from the inlet of the test section. The maximum augmentation rate is attained at the entry of the test section and then diminishes along the axial distance from the inlet of the test section.
- All nanofluids have a higher heat transfer coefficient compared to the base fluids. The augmentation of heat transfer coefficient increases with the increase in the weight concentration as well as flow velocity of nanofluids. The maximum enhancement of convective HTC is found to be 77.60 % for the 0.5 wt.% of MWCNT-distilled water nanofluid at flow velocity of 0.232 m/s compared to the distilled water.

The most precise basis of comparison is constant velocity which directly takes into account the effect of thermophysical properties. It is suggested that while evaluating and comparing results from the literature, the reader must be observant on how the results are presented in order to get true advantage of nanofluids over the base fluids for heat transfer. The work can further be extended using MWCNT along with some other nanoparticles to make hybrid nanofluids with better thermophysical properties as well as low cost.

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