Experimental Investigation of Convective Heat Transfer Coefficient of CNTs Nanofluid under Constant Heat Flux

F. Rashidi, N. Mosavari Nezamabad

Abstract—This study is concerned with the heat transfer behavior of aqueous suspension of multi-walled carbon nanotubes flowing through a horizontal tube under constant heat flux. We measured heat transfer coefficient of CNTs nanofluid in laminar flow regime. Results show considerable enhancement in convective heat transfer coefficient of nanofluids. The increment was particulary significant in entrance length and it depends on the CNTs concentration and flow condition (Reynolds number). The enhancement in convective heat transfer is a function of axial distance from inlet and it has a decreasing trend.

***Index Terms*—Convective heat transfer coefficient, Nanofluids, Multi-walled carbon nanotube, Constant heat flux.**

I. INTRODUCTION

Nanofluids are a new class of heat fluids consisting of nanometer-sized particles (less than 100 nm) dispersed in convectional fluids. The advantages of nanofluids are: (1) higher thermal conductivity than that predicted by currently available macroscopic models, (2) excellent stability, and (3) little penalty due to an enhancement in pressure drop and pipe wall erosion experienced by suspensions of micrometer or millimeter particles [1].

Such advantages of nanofluid offer important benefits for numerous applications in many fields such as petrochemical, refining, electronic, transportation, medicine, heating, and air- conditioning.

Several experimental studies on nanofluid heat transfer have been reported in the literature. Pak and Cho [2] studied TiO_2-water and γAl_2O_3-water nanofluids in turbulent convective heat transfer in tubes. Yang et al. [3]studied laminar convective heat transfer of graphite nanofluids in a horizontal tube heat exchanger. Xuan and Roetzel [4] investigated a heat transfer correlation for nanofluids to study the effect of energy transport by particle dispersion. Xuan and Li [5] considered convective heat transfer and flow features of CuO-water nanofluids. Wen and Ding [6] focused on entry region under laminar flow condition using nanofluids containing γ- Al_2O_3 nanoparticles. Ding et al. [1] reported significant enhancement in convective heat transfer of multi-wall carbon nanotube dispersion in water and found that amount of enhancement depends on the Reynolds number, carbon nanotube concentration, and PH. Heris et al. [7], [8] investigated the effects of alumina and copper oxide nanofluids on laminar convective heat transfer in a circular tube under constant wall temperature boundary condition. They found heat transfer coefficient enhances for both nanofluids with increasing peclect number as well as nanofluid concentrations, and observed higher enhancement in Al_2O_3-water nanofluid than CuO-water nanofluid. Lee et al. [9] studied Convective heat transfer for alumina nanofluid in micro channel. Hwang et al. [10] considered convective heat transfer coefficient and pressure drop of

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water-based Al_2O_3 nanofluid in fully developed laminar flow regime. They reported enhancement of convective heat transfer coefficient is more than the thermal conductivity enhancement. He et al. [11] studied the heat transfer of titania nanofluids in both laminar and turbulent flow regimes under constant heat flux boundary condition, and found that the heat transfer enhancement increased with concentration of nano particles, and reducing particle size. Garg et al. [12] inspected the effect of ultrasonication on viscosity and heat transfer performance of multiwall carbon nanotube-based aqueous nanofluids. Also, they found an optimum time for ultrasonication.

Based on the aforementioned, it could be stated that little published literature exists regarding MWCNTs in water and their applications as heat transfer fluid. Also, there is inconsistency in the few reported studies on the convective heat behavior of CNT-nanofluids. Therefore, it was decided to study the convective heat transfer behavior of MWCNTs in water in laminar flow regimes.

II. EXPERIMENTAL

A. Sample Preparation

De-ionized (DI) water, Gum Arabic (GA), and multiwall carbon nanotubes (MWCNT) were used to produce nanofluids. The carbon nanotubes (CNTs) were supplied from Nanostructured & Amorphous Materials, Inc., and nanotubes had an average outside diameter of 30-50 nm, inside diameter of 5-15 nm, and length 10-20 μm and purity 95%, produced by Catalytic Chemical Vapor Deposition process. GA was dissolved in DI water using a magnetic stirrer, followed by the addition of MWCNT to solution. The resulting composition was agitated by magnetic stirrer for 30 min, and then was ultrasonicated for 2 hours at 100% amplitude using a 200W, 24 KHz ultrasonication probe (Hielscher Ultrasonics GmbH, Germany).

Three samples were prepared with specifications as follows:

- nanofluid 1: 0.5wt% MWCNT, 0.25wt% GA
- nanofluid 2: 1wt% MWCNT, 0.5wt% GA
- nanofluid 3: 2wt% MWCNT, 1wt% GA

CNT nanofluids made in this way were found to be stable for over 3 weeks with no visible sedimentation or settling. After 3 weeks, if agitated with magnetic stirrer for 30 minutes it becomes stable again and doesn't need to undergo ultrasonic again.

Fig.1. shows a TEM image of dispersed sample. It can be seen that carbon nanotubes have largely been disentangled.

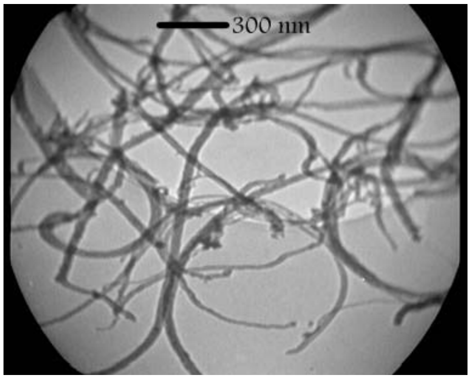

Fig. 1.TEM images of aqueous suspensions of 2 wt% MWCNT, 1 wt% GA at 300 nm scale.

B. Heat Tansfer Coefficient

The experimental system for measuring the convective heat transfer coefficient is shown schematically in Fig.2. It mainly consists of a test section, a pump, a reservoir tank, and a cooling part.

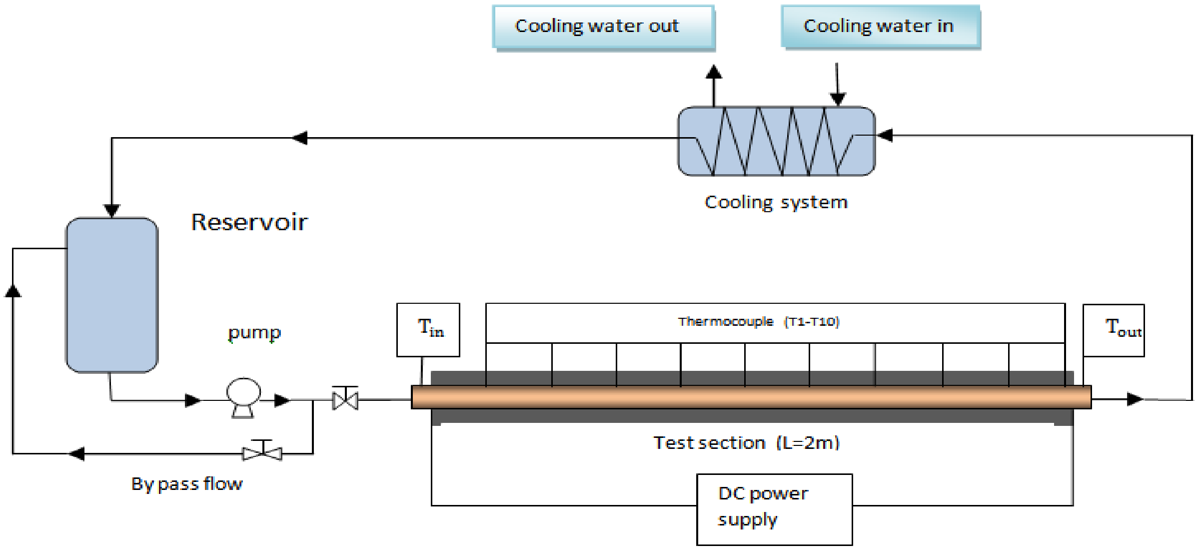

Fig. 2 Experimental system for the measurement of local convective heat transfer coefficient.

A straight copper tube of 2100 mm length and 6 mm inner diameter is used as the test section. A heater wire was adopted to supply a constant heat flux condition along the test section. The heater wire was wrapped around the tube in a noninductive manner to avoid magnetic effects during experiments. The noninductive swathing is done by doubling the heater wire and then bending in around the tube like a solenoid. Therefore, all along the wire there are equivalent currents passing in both directions that induce the same magnetic fields with different directions. The induced magnetic fields neutralize the effect of each other. The heater was linked to a DC power supply. The power supply was adjustable and had a maximum power supply of 250 W. There was a thick thermal isolating layer surrounding the heater to obtain a constant heat flux condition along the test section. Ten PT-type thermocouples were mounted on the test section at axial positions 200 mm apart from the inlet of the test section to measure the wall temperatures. Additionally, two thermocouples were inserted into the flow at the inlet and outlet of the heat transfer section. The magnet pump was used for running nanofluid into the loop. The nanofluid flow rate was evaluated by the time taken for a given mass of nanofluid to be discharged. The mass flow rate of fluid varied in a range from 1.5 to 2.7 g/s. To preserve a constant temperature at the inlet of test section, the heated fluid passes through a cooling system, and then returns to the reservoir tank.

The convective heat transfer coefficient *(h(x))* at any axial distance $'x'$ from inlet is defined as:

$$h(x) = \frac{q_s''}{T_s(x) - T_b(x)} \qquad (1)$$

where q_s'' is the heat flux, $T_s(x)$ is measured wall temperature at a distance $'x'$ from the inlet, and $T_b(x)$ is the fluid bulk temperature at a distance $'x'$ from the inlet.

From the energy balance equation, the bulk temperature of the fluid ($T_b(x)$) at an axial distance $'x'$ can be found using:

$$T_b(x) = T_{bi} + \frac{q_s'' p}{mc} x \qquad (2)$$

Where c is specific heat, m is mass flowrate of fluid, p is perimeter of copper tube, $\mathrm{T_{bi}}$ is the inlet fluid bulk temperature, and x is distance where the bulk temperature of fluid is calculated.

The heat flux applied to the fluid ($\mathrm{q_s''}$) can be found using:

$$\mathrm{q_s''} = \frac{\mathrm{mc}\,(\mathrm{T_{bo}} - \mathrm{T_{bi}})}{\mathrm{A}} \qquad (3)$$

Where $\mathrm{T_{bo}}$ is the outlet fluid bulk temperature.

The convective heat transfer coefficient, *h(x)*, in Eq. (1) is usually expressed in the form of Nusselt number *(Nu(x))* as:

$$Nu(x) = h(x) D_i / \mathrm{k} \qquad (4)$$

Where $\mathrm{D_i}$ and k are the inside diameter of the copper tube and the thermal conductivity of the test fluid, respectively.

Xuan and Roetzel [4], by assuming thermal equilibrium between the particle and surrounding fluid suggested an equation to determine the specific heat as follows:

$$c = \frac{\phi \rho_p c_p + (1-\phi)\rho_f c_f}{\rho} \qquad (5)$$

where $\phi, \rho, \rho_p, \rho_f, c_p, c_f$ are volume fraction, the nanofluid density, density of carbon nanotubes, density of base fluid, specific heat of CNTs, and specific heat of base fluid.

The density of the nanofluids is calculated by use of the Pak and Cho [2] correlations, which is defined as follows:

$$\rho = \phi \rho_p + (1-\phi)\rho_f \qquad (6)$$

Brinkman [13] suggested an equation for calculating the viscosity of the suspension, which is defined as follows:

$$\mu_{nf} = \frac{\mu_f}{(1-\emptyset)^{2.5}} \qquad (7)$$

Where μ_f, μ_{nf} are viscosity of base fluid, and viscosity of nanofluid.

Since the loading analysis was obtained in weight percent (w); the conversion between weight and volume fraction can be obtained using:

$$\phi = \frac{w \rho_f}{\rho_p (1-w) + w \rho_f} \qquad (8)$$

III. RESULT AND DISCUSSION

A. Convective Heat Transfer Coefficient of Distillated Water

In order to establish the reliability and accuracy of experimental measurements, initially some experiments were performed for DI water. For internal forced convective heat transfer, the results of the Shah equation [14] for laminar flow under constant heat flux conditions were compared with our experimental results.

Shah equation: (9)

$$Nu_x = \begin{cases} 1.302\, x_*^{-\frac{1}{3}} - 1 & x_* \le 0.00005 \\ 1.302\, x_*^{-\frac{1}{3}} - 0.5 & 0.00005 \le x_* \le 0.001 \\ 4.364 + 8.68(10^3 x_*)^{-0.506} e^{-41 x_*} & x_* \ge 0.001 \end{cases}$$

Where $Nu_x = h(x) D_i / k,\ x_* = \left[\frac{\frac{x}{D_i}}{Re\ Pr}\right]$.

The heat transfer coefficient in laminar region for DI water is shown in Fig.3. For all tests the measured error in heat transfer coefficient in comparison with predicted value from the Shah equation is around 7%. This seems presumably good because Shah equation was derived for large channels.

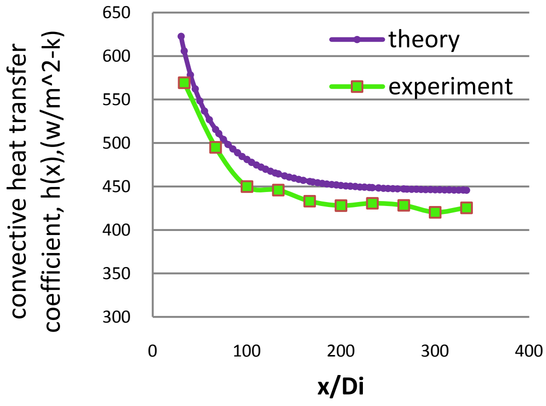

Fig.3 Axial variation of heat transfer coefficient of water at Re = 483

B. Convective Heat Transfer Coefficient of Nanofluid

Fig.4 exhibits the local convective heat transfer coefficients of nanofluid 3 versus axial distance, x/D$_i$, for different Reynolds numbers. As expected, the local heat transfer coefficient, $h(x)$, enhances by increasing Reynolds number.

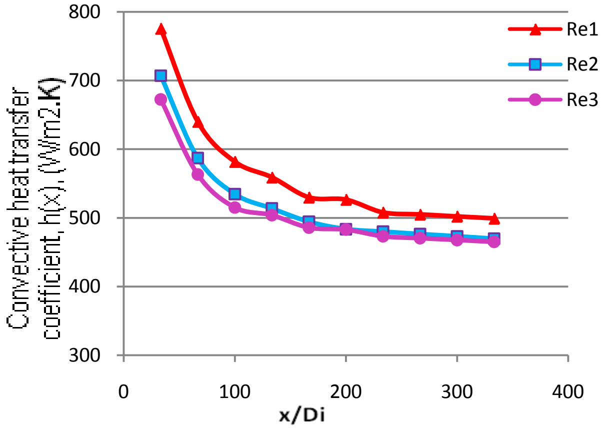

Fig.4 Axial variation of heat transfer coefficient for nanofluid2 (1wt% CNT) at Re1 = 616, Re2 =526, and Re3=472

Fig. 5 shows the effect of CNTs concentration on the local convective heat transfer coefficient at various axial distance. It's clearly seen, the local convective heat transfer coefficient increases with increasing concentration of carbon nanotubes, and a remarkable enhancement in heat transfer coefficient is observed in CNT nanofluid samples compared with DI water. As suggested by other researchers [1], [6], [15] the reasons for the heat transfer enhancement of nanofluids include many issues, such as the mixing effects of particles near the wall, Brownian motion of particles, thermal conductivity increment, particle shape, particle migration and, reduction of boundary layer thickness, and delay in boundary layer development.

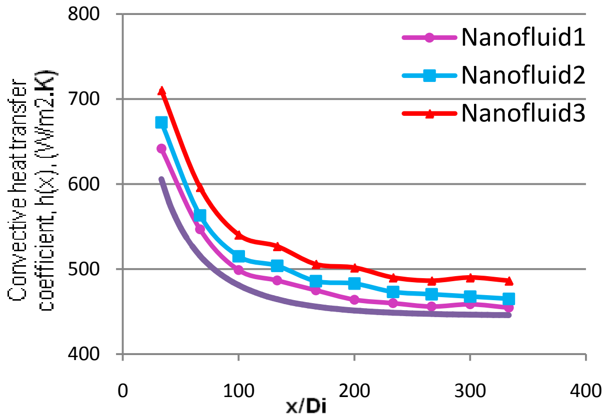

Fig.5 Axial variation of heat transfer coefficient at Re=472± 10

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Fig. 6 displays the enhancement of convective heat transfer coefficient, with reference to DI water, versus axial distance for nanofluid3 (2wt%). The maximum enhancement is in an early entrance, and then increment of convective heat transfer coefficient decreased along the tube length.

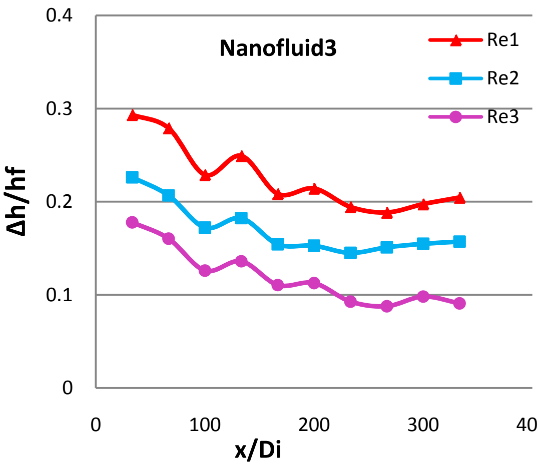

Fig.6 enhancement of convective heat transfer coefficient versus axial distance for nanofluid3 at Re1=603, Re2 =554, and Re3=472

The local convective heat transfer coefficient, h(x), can be approximately given as k/δ_t , where δ_t , k are the thickness of thermal boundary layer and thermal conductivity. Therefore, it can be said that two parameters k and δ_t are the most important factors that are responsible for increasing the local convective heat transfer coefficient in laminar flow. As regards to thermal conductivity is constant along the tube length for the test fluid, it can be deducted that developing thermal boundary layer due to enhancement of thickness boundary layer (δ_t), is a crucial factor on reducing increment in the local heat transfer coefficient along tube length. So, in laminar flow, delay in developing thermal boundary layer is an important factor for increasing of convective heat transfer coefficient in entry region.

IV. CONCLUSION

The convective heat transfer coefficient of CNTs nanofluids in a single 6 mm inner diameter, constant heat flux copper tube was investigated for laminar flow in both developing and fully developed region. It is concluded that addition of MWCNTs to water increases the local convective heat transfer coefficient. The enhancement decrease with developing thermal boundary layer.

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