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RESEARCH ARTICLE

HEAT TRANSFER ENHANCEMENT OF TUBE IN TUBE HEAT EXCHANGER USING CuO/H_2O NANOFLUID UNDER TURBULENT FLOW

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ABSTRACT

An experimental system is built to investigate convective heat transfer and flow characteristics of the nanofluid in a concentric tube heat exchanger. Both the convective heat transfer coefficient and friction factor of Cu-water nanofluid for the turbulent flow are measured. The effects of such factors as the volume fraction of suspended nanoparticles and the Reynolds number on the heat transfer and flow characteristics are discussed in detail. The experimental results show that the suspended nanoparticles remarkably increase the convective heat transfer coefficient of the base fluid and show that the friction factor of the sample nanofluid with the low volume fraction of nanoparticles is almost not changed. Compared with the base fluid, for example, the convective heat transfer coefficient is increased about 60% for the nanofluid with 2.0 vol% Cu nanoparticles at the same Reynolds number. Considering the factors affecting the convective heat transfer coefficient of the nanofluid, a new convective heat transfer correlation for nanofluid under single-phase flows in concentric tube heat exchangers is established. Comparison between the experimental data and the calculated results indicate that the correlation describes correctly the energy transport of the nanofluid.

Keywords : Nanofluid, Convective Heat Transfer, Correlation, Friction Factor.

1. INTRODUCTION

With increasing heat transfer rate of the heat exchange equipment, the conventional process fluid with low thermal conductivity can no longer meet the requirements of high-intensity heat transfer. Low thermal property of heat transfer fluid is a primary limitation to the development of high compactness and effectiveness of heat exchangers. An effective way of improving the thermal conductivity of fluids is to suspend small solid particles in the fluids[1]. Traditionally, solid particles of micrometer or millimeter magnitudes were mixed in the base liquid. Although the solid additives may improve heat transfer coefficient, practical uses are limited because the micrometer and/or millimeter-sized particles settle rapidly, clog flow channels, erode pipelines and cause severe pressure drops. The concept of nanofluids refers to a new kind of heat transport fluids by suspending nano-scaled metallic or non-metallic particles in base fluids. Some experimental investigations[2-5] have revealed that the nanofluids have remarkably higher thermal conductivities than those of conventional pure fluids and shown that the nanofluids have great potential for heat transfer enhancement. Nanofluids are more suits for practical application than the existing techniques for enhancing heat transfer by adding millimeter and/or micrometer-sized particles in fluids. It incurs little or no penalty in pressure drop because the nanoparticles are so small that the nanofluid behaves like a pure fluid.

To apply the nanofluid to practical heat transfer processes, more studies on its flow and heat transfer feature are needed. Pak and Cho[6] performed experiments on turbulent friction and heat transfer behaviour of two kinds of the nanofluids. In their study, γ -Al2O3 with mean diameter of 13 nm and TiO2 with mean diameter of 27 nm were dispersed in water, and the experimental results showed that the suspended nanoparticles increased convective heat transfer coefficient of the fluid. The Nusselt number of the nanofluids was found to increase with the increasing volume fraction and Reynolds number. Lee and Choi[7] applied the nanofluid as the coolant to a micro channel heat exchanger for cooling crystal silicon mirrors used in high-intensity X-ray sources and pointed out that the nanofluid dramatically enhanced cooling rates compared with the conventional water-cooled and liquid-nitrogen-cooled micro channel heat exchangers. The cooling capability of 30MW·m-2 was achieved. It is expected that the heat transfer enhancement of the nanofluids may result from intensification of turbulence or eddy, suppression of the boundary layer as well as dispersion or back mixing of the suspended nanoparticles, besides substantial augmentation of the thermal conductivity of the fluid. Therefore, the convective heat transfer coefficient of the nanofluids is a function of properties, dimension and volume fraction of suspended nanoparticles, and the flow velocity. The conventional convective heat transfer correlation of the pure fluid is not applicable to the nanofluid. This paper is aimed at investigating convective heat transfer performances of the nanofluid and establishing heat transfer correlation for single-phase flows in tubes.

2. EXPERIMENTAL SETUP

In order to study above objectives in a double pipe counter flow heat exchanger an experimental set-up is designed as shown in the figure 4.1. The experimental set-up consisted of a flow loop containing several sections such as temperature, pressure and flow rate measuring units, heating and cooling sections and flow controlling system. Experimental setup consists of two flow circuit through which heat transfer takes place by thermal difference. These flow circuits are nanofluid flow circuit through inner copper tube and cooling water circuit through annulus of double pipe heat exchangers. Test section consisted of 1.03 m long double pipe counter flow heat exchanger which is constructed of 11 mm diameter inner copper tube with 0.5 mm thickness and outer GI pipe of 27 mm inner diameter. The outer GI pipe is insulated by foam insulations to reduce loss of heat to surrounding by convection and radiation.

- Nanofluid Circuit
- Cooling Water Circuit
- Measuring Instruments
- Temperature Measurements.
- Pressure Drop Measurements.
- Flow Rate Measurements.

3. HEAT TRANSFER DATA REDUCTION

The heat transfer rate from the heating fluid (nanofluid) is calculated from:

$$Q_{nf} = m_{nf}.Cp_{nf}.(T_1 - T_2)_{nf} \quad \ldots (3.1)$$

Where, Q_{nf} is the heat transfer rate of the hot nanofluid, m_{nf} is the mass flow rate of the hot nanofluid, Cp_{nf} is the specific heat of nanofluid, T1 is inlet temperature of nanofluid into the heat exchanger and T2 is the outlet temperature of nanofluid from the heat exchanger. Mass flow rate of nanofluid is obtained by multiplying volume flow rate obtained from nanofluid rotameter to density of nanofluid at that temperature.

The heat transfer rate to the cooling water (heat gain by cooling water) is calculated as:

$$Q_w = m_w.Cp_w.(T_4 - T_3)_w \quad \ldots (3.2)$$

Where, Q_w is the heat transfer rate to the cooling water and m_w is the mass flow rate of the cooling water through annulus. Cp_w is specific heat of cooling water. T3 is inlet temperature of cooling water into the heat exchanger and T4 is the outlet temperature of cooling water from the heat exchanger. Mass flow rate of cooling water is obtained by multiplying volume flow rate obtained from flow rotameter to density of water at that temperature.

The average heat transfer rate (heat flux) between the nanofluid and cooling water is calculated as follows:

$$Q_{avg} = \frac{Q_{nf} + Q_w}{2} \quad \ldots (3.3)$$

Where Q_{avg} is the average heat transfer rate between the cooling water and the nanofluid.

In this study, the energy differences between heat loss by nanofluid and heat gain by cooling water is maximum 3% which mainly due to heat loss through the wall of geyser, inlet tubing to heat exchanger, outlet tubing from heat exchanger, radiation heat loss and heat loss through the nanofluid reservoir . These heat losses are minimized by providing foam insulation over copper tube, test sections at hot zones.

The experimental overall heat transfer coefficient based on outer surface of copper tube is calculated as:

$$U_{o(\exp)} = \frac{Q_{avg}}{A_o.\theta_m} \quad \ldots (3.4)$$

Where,

$A_o = \pi d_o L$ = outer heat transfer surface of copper tube,
do= Copper tube outer diameter, = 0.012 m,
L= test section length, = 1.03 m,
$\theta_m = \frac{\theta_1 - \theta_2}{\ln(\frac{\theta_1}{\theta_2})}$ = log mean temperature difference of counter flow heat exchanger,
θ_1= (T1-T4), and
θ_2=(T2-T3).

The nanofluid heat transfer coefficient is calculated from Uo(exp) as below:

$$h_{nf} = \frac{d_o}{d_i\left(\frac{1}{U_{o(\exp)}} - \frac{d_o.\ln(d_o/d_i)}{2.K_{co}} - \frac{1}{h_o}\right)} \quad \ldots (3.5)$$

Where Kco = Thermal conductivity of copper tube material = 385 W/m2 oC. do and di are outer and inner diameters of copper tube, ho is outer heat transfer coefficient of water through annulus and calculated from the Dittus-Boelter equation as below:

$$Nu_o = 0.023(Re)^{0.8}(Pr)^{0.4} \quad \ldots (3.6)$$

Where 'Re' and 'Pr' are the Reynolds number and Prandtl's number of annulus flow at mean bulk temperature.

The outer heat transfer coefficient is calculated from the Nusselt number as below:

$$h_o = \frac{Nu_o.K_w}{D_e} \quad \ldots (3.7)$$

Where D_e is the equivalent diameter of annulus for heat

flow = $\frac{4.Ac}{Pw}$ = Di-do

Nanofluids Thermal and Flow Properties : The thermal and flow properties of water like Reynolds number, Prandtl's number, conductivity are calculated at bulk mean temperatures. The thermal and flow properties of nanofluid are calculated using different available correlations as below:

Thermal conductivity using Timofeeva correlations as below:

$$K_{nf} = [1 + 3\phi]K_w \quad \ldots (3.8)$$

Viscosity of nanofluid using Drew and Passman correlations as below:

$$\mu_{nf} = [1 + 2.5\phi]\mu_w \quad \ldots (3.9)$$

The density and specific heat using Pak and Cho correlations as below :

$$\rho_{nf} = \phi\rho_{np} + (1-\phi)\rho_w \quad \ldots (3.10)$$

$$Cp_{nf} = \phi Cp_{np} + (1-\phi)Cp_w \quad \ldots (3.11)$$

The Reynolds number and Prandtl's number of nanofluid are calculated using following relations:

$$\mathrm{Re}_{nf} = \frac{\rho_{nf}.V_{nf}.d_i}{\mu_{nf}} \quad \ldots (3.12)$$

$$\mathrm{Pr}_{nf} = \frac{\mu_{nf}.Cp_{nf}}{K_{nf}} \quad \ldots (3.13)$$

Where, Vnf is the average velocity of nanofluid flowing through the copper tube.

Nusselt number of the nanofluid flow is computed from the following equation:

$$Nu_{nf} = \frac{h_{nf}.d_i}{K_{nf}} \quad \ldots (3.14)$$

Pressure Drop Data Reduction : Theoretical Head loss due to friction in the copper tube is calculated using Darcy Weisbach's equations as below:

$$H_{nf} = \frac{4.f_{nf}.L.V_{nf}^2}{g.d_i} \quad \ldots (3.15)$$

Equivalent nanofluid head loss is measured from mercury monomer as below:

$$H_{nf} = h_{hg}\left[\frac{S_{hg}}{S_{nf}} - 1\right] \quad \ldots (3.16)$$

Where h_{hg} is difference in the mercury levels in the manometer.

$\frac{S_{hg}}{S_{nf}}$ is the ratio of specific gravities of mercury and nanofluid.

Nanofluid friction factor is calculated from the above equation as below:

$$f_{nf} = \frac{.H_{nf}.g.d_i}{4.L.v_{nf}^2} \quad \ldots (3.17)$$

Loss in the pressure is calculate using the equation,

$$\Delta P_{nf} = \rho_{nf}.g.H_{nf} \quad \ldots (3.18)$$

4. EXPERIMENTAL SETUP VALIDATION

Before evaluating the heat transfer performance of the nanofluids, experimental set up is validated by conducting trial with water to water. The distilled water is used as the working fluid for estimating the reliability and accuracy of the experimental system. The experimental values of overall heat transfer coefficient are compared with that obtained by theoretical correlations for water. Tests are conducted with hot water instead of nanofluid flowing through the copper tube closed loop and cooling water through annulus of test section.

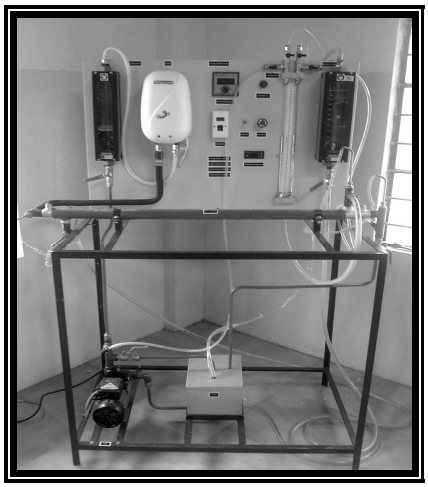

Fig. 4.1 : Photograph of Experimental Set Up

Validation for Heat Transfer Study : The heat transfer rate from the hot water fluid is calculated as

$$Q_{hw} = m_{hw}.Cp_{hw}.(T_1 - T_2)_{hw} \quad \ldots (4.2)$$

Where, Q_{hw} is the heat transfer rate of the hot water, m_{hw} is the mass flow rate of the hot water, Cp_{hw} is the specific heat of hot water, T1 is inlet temperature of hot water into the heat exchanger and T2 is the outlet temperature of hot water from the heat exchanger. Mass flow rate of hot water is obtained by multiplying volume flow rate obtained from

hot water rotameter to density of hot water at that temperature.

The heat transfer rate to the cooling water (heat gain by cooling water) is calculated as:

$$Q_{cw} = m_{cw}.Cp_{cw}.(T_4 - T_3)_{cw} \quad \ldots (4.3)$$

Where, Q_{cw} is the heat transfer rate to the cooling water and m_{cw} is the mass flow rate of the cooling water through annulus. Cp_{cw} is specific heat of cooling water. T3 is inlet temperature of cooling water into the heat exchanger and T4 is the outlet temperature of cooling water from the heat exchanger. Mass flow rate of cooling water is obtained by multiplying volume flow rate obtained from flow rotameter to density of water at that temperature.

The average heat transfer rate (heat flux) between hot water and cooling water is calculated as follows:

$$Q_{avg} = \frac{Q_{hw} + Q_{cw}}{2} \quad \ldots (4.4)$$

Where Q_{avg} is the average heat transfer rate between the cooling water and the hot water.

The experimental overall heat transfer coefficient based on outer surface of copper tube is calculated as:

$$U_{o(\exp)} = \frac{Q_{avg}}{A_o.\theta_m} \quad \ldots (4.5)$$

Where,

$A_o = \pi doL$ = outer heat transfer surface of copper tube,
do = Copper tube outer diameter, = 0.012 m,
L= test section length, = 1.03 m,
$\theta_m = \frac{\theta 1 - \theta 2}{\ln(\frac{\theta 1}{\theta 2})}$ = log mean temperature difference of counter flow heat exchanger,
$\theta 1$= (T1-T4), and
$\theta 2$= (T2-T3).

Theoretical overall heat transfer coefficients are calculated using following equation :

$$Uo_{(th)} = \frac{1}{\left(\frac{do}{di.hi} + \frac{d_o.\ln(d_o/d_i)}{2.K_{co}} + \frac{1}{h_o}\right)} \quad \ldots (4.6)$$

The internal heat transfer coefficient is calculated from the Nusselt number as below:

$$h_i = \frac{Nui.K_w}{di} \quad \ldots (4.7)$$

Where internal flow Nusselt number is calculated from Dittus-Boelter equation as below:

$$Nu_i = 0.023(Re)^{0.8}(Pr)^{0.4} \quad \ldots (4.8)$$

Also the outer heat transfer coefficient is calculated from the Nusselt number as below:

$$h_o = \frac{Nu_o.K_w}{D_e} \quad \ldots (4.9)$$

Where D_e is the equivalent diameter of annulus for heat flow = $\frac{4.Ac}{Pw}$ = Di-do

The thermal and flow properties of hot and cold water like Reynolds number, Prandtl's number, and conductivities are calculated at bulk mean temperatures.

$$\mathrm{Re}_w = \frac{\rho_w.V_w.d_i}{\mu_w} \quad \ldots (4.10)$$

$$\mathrm{Pr}_w = \frac{\mu_w.Cp_w}{K_w} \quad \ldots (4.11)$$

Energy loss due to friction is calculated by using mercury manometer. This gives experimental head loss across test section. This is converted into equivalent water column as below

$$H_w = h_{hg}\left[\frac{S_{hg}}{S_w} - 1\right] \quad \ldots (4.12)$$

Where
h_{hg} is difference in the mercury levels in the manometer.
$\frac{S_{hg}}{S_w}$ is the ratio of specific gravities of mercury and water.

Darcy coefficient of friction factor for the water flow through copper tube is calculated from the above equation as below:

$$f_{(\exp t)} = \frac{.H_w.g.d_i}{4.L.v_w^2} \quad \ldots (4.13)$$

Loss in the pressure is calculate using the equation,

$$\Delta P_w = \rho_w.g.H_w \quad \ldots (4.14)$$

Theoretical Head loss due to friction in the copper tube is calculated using Darcy Weisbach's equations as below:

$f_{th} = \frac{0.0791}{Re^{0.25}}$ = Darcy friction factor and theoretical head loss as below

$$H_{th} = \frac{4.f_{th}.L.V_w^2}{g.d_i} \quad \ldots (4.15)$$

Theoretical loss in the pressure is calculated using the equation,

$$\Delta P_{ch} = \rho_w \cdot g \cdot H_{ch.} \quad \ldots (4.16)$$

5. RESULTS AND DISCUSSION

The values of overall heat transfer coefficient increases with increase in flow rate of hot fluid i.e. Reynolds number of flow. Overall heat transfer coefficients for nanofluids are more than that of base fluid water. It indicates additions of nanoparticles in the base fluid increases heat transfer rate of base fluid. Overall heat transfer coefficient of nanofluid increases with increase in its concentrations. This increase in heat transfer rate is mainly contributed by increase in thermal conductivity of nanofluid by addition of nanoparticles. Higher the concentrations of nanoparticles in the base fluid it increases its thermal conductivity and hence less resistance for heat convection. Maximum overall heat transfer coefficient is found to be at 44973.33 Reynolds number and at 2.00 wt % of Copper oxide-Water Nanofluid. This value increases from 2500.84 W/m2.0C for water at same flow rate to 3801.03 W/m2.0C for 2.00 wt % of Copper oxide-Water nanofluid. A 51.99 % increase in overall heat transfer coefficient as compare to water is found in this case by 2.00 wt % of Copper oxide-Water nanofluid.

The values of convective heat transfer coefficient increases with increase in flow rate of nanofluid i.e. Reynolds number of flow. Convective heat transfer coefficients for nanofluids are more than that of base fluid water. It indicates additions of nanoparticles in the base fluid increases heat transfer rate of base fluid. Convective heat transfer coefficient of nanofluid increases with increase in its concentrations. This increase in heat transfer rate is mainly contributed by increase in thermal conductivity of nanofluid by addition of nanoparticles and increased turbulence in the flow. Higher the concentrations of nanoparticles in the base fluid it increases its thermal convection and hence less resistance for heat transfer. Maximum convective heat transfer coefficient is found to be at 44973.33 Reynolds number and at 2.00 wt % of Copper oxide-Water Nanofluid. This value increases from 10799.58 W/m2.0c for water at same flow rate to 22376.37W/m2.0c for 2.00 wt % of Copper oxide-Water nanofluid. A 63.59 % increase in overall heat transfer coefficient as compare to water is found in this case by 2.00 wt % of Copper oxide-Water nanofluid. This large increase in the values of convective heat transfer coefficient is mainly because of increased convection due to nanoparticles, which has higher thermal conductivity as compared to base fluid molecules. During the flow nanoparticles increases heat transfer by collisions with each other and collisions with heat transfer surface. Increased concentrations of nanoparticles in the base fluid increases these collisions in nanofluid and thereby rate of heat flux. Increased flow rate also increases molecular movements, which increases molecular collisions with each other and with the heat transfer surface. This increases heat transfer by conduction from nanoparticles to heat transfer surface i.e. copper concentric tube heat exchanger. Thus nanoparticles act as energy carrier from hotter zone to colder zone and boost the rate of heat transfer as compared to base fluid molecules. This reduces thermal resistance and increases heat transfer rate.

The values of Nusselt number of flow increases with increase in flow rate of nanofluid i.e. Reynolds number of flow. Nusselt numbers of flow for nanofluids are more than that of base fluid water. Nusselt number of flow of nanofluid increases with increase in its concentrations. Maximum Nusselt number of flow is found to be at 44973.33 Reynolds number and at 2.00 wt % of Copper oxide-Water Nanofluid. This value increases from 236.19 W/m2.0c for water at same flow rate to 360.38 W/m2.0c for 1.25 wt % of Copper oxide-Water nanofluid. A 52.58 % increase in overall heat transfer coefficient as compare to water is found in this case by 2.00 wt % of Copper oxide-Water nanofluid.

With increase in flow rate pressure drop increases. Increase in flow rate increases the turbulence in the flow and thereby energy loss due to friction in the fluid. Pressure loss also increases with increase in nanofluid concentrations this indicates additions of nanoparticles in the base fluid increases fluid friction. This is mainly because of increases fluid viscosity. An addition of nanoparticles increases fluid viscosity by molecular cohesion. Pressure drop of nanofluids are little more than base fluid water. This indicates addition of nanoparticles in the base fluid increases small the pumping energy loss. Maximum pressure drop of flow is found to be at 44973.33 Reynolds number and at 2.0 wt % of Copper oxide-Water Nanofluid. This value increases from 7.33 kPa for water at same flow rate to 8.98 kPa for 2.0 wt % of Copper oxide-Water nanofluid. A 22.51 % increase in pressure drop as compare to water is found in this case by 2.0 wt % of Copper oxide-Water nanofluid.

With increase in flow rate friction factor decreases. Friction factor increases with increase in nanofluid concentrations. This indicates additions of nanoparticles in the base fluid increases fluid friction. This is mainly because of increases fluid viscosity. An addition of nanoparticles increases fluid viscosity by molecular cohesion. Data of friction factor shows small variations and random oriented. This is because of small inaccuracies in the measurements. But it shows that addition of nanoparticles in the base fluid at low concentration has very little effect on the friction factor of fluid.

Table 1 : Thermal Properties of Water to Water

Sr. No	Flow Rate	Water to water					
		Re (w)	Uo(w)	hi(w)	Nu(w)	ΔP (w)	f(w)
	(lph)		W/m$^{2.O}$C	W/m$^{2.O}$C		kPa	
1	350	19983	2302.2	6077.1	75.77	0.49	0.006
2	400	22838	2327.6	6530.4	90.06	0.74	0.006
3	500	28065	2430.1	7025.2	120.9	1.97	0.005
4	600	33119	2496.7	8111.3	150.1	2.57	0.005
5	700	38009	2500.8	10799.6	170.9	3.89	0.004

Table 2 : Thermal Properties of Copper oxide-Water Nanofluid (0.25 wt %)

Sr. No.	Flow Rate	Copper oxide-Water Nanofluid (0.25 wt %)					
		Re	Uo	hnf	Nu	ΔP	f
	(lph)		W/m2.OC	W/m2.OC		kPa	
1	350	22490	2479.1	6054.3	140.7	1.59	0.006
2	400	25702	2611.2	7102.7	139.2	2.49	0.006
3	500	31119	2694.1	8674.3	174.9	4.42	0.005
4	600	36208	2717.3	10894.7	216.6	6.98	0.005
5	700	41596	2820.5	13126.3	235.9	8.91	0.005

Table 3 : Thermal Properties of Copper oxide-Water Nanofluid (0.5 wt %)

Sr. No.	Flow Rate	Nanofluid (0.50 wt %)					
		Re	Uo	hnf	Nu	ΔP	f
	(lph)		W/m2.OC	W/m2.OC		kPa	
1	350	22323	2700.5	6031.5	144.0	0.75	0.006
2	400	25097	2758.5	7675.5	143.7	2	0.006
3	500	30876	2897	9323.4	173.9	3.76	0.006
4	600	36465	2988.6	10678.1	213.7	6.51	0.005
5	700	41912	3109.3	13453.5	247.4	10.7	0.005

Table 4 : Thermal Properties of Copper oxide-Water Nanofluid (1 wt %)

Sr. No.	Flow Rate	Nanofluid (1 wt %)					
		Re	Uo	hnf	Nu	ΔP	f
	(lph)		W/m2.OC	W/m2.OC		kPa	
1	350	22963	2710.6	8525.9	103.7	0.9	0.0069
2	400	26243	2718.2	8433.6	132.0	2.31	0.0067
3	500	32331	2927.7	10327	177.9	3.98	0.0063
4	600	38165	3106.8	12604	236.1	7.45	0.006
5	700	43817	3420.2	14565.	250.9	11.0	0.0058

Table 5 : Thermal Properties of Copper oxide-Water Nanofluid (1.5 wt %)

Sr. No.	Flow Rate	Nanofluid (1.5 wt %)					
		Re	Uo	hnf	Nu	ΔP	f
	(lph)		W/m2.OC	W/m2.OC		kPa	
1	350	23653	2798.5	8581.62	116.9	1.94	0.0072
2	400	27032	2915.5	8552.17	151.6	3.35	0.0068
3	500	32804	3106.5	10576.4	213.4	5.7	0.0066
4	600	39365	3271.5	13080.2	252.6	7.75	0.0064
5	700	45926	3610.5	20247.7	286.9	11.0	0.0063

Table 6 : Thermal Properties of Copper oxide-Water Nanofluid (2 wt %)

Sr. No.	Flow Rate	Nanofluid (2 wt %)					
		Re	Uo	hnf	Nu	ΔP	F
	(lph)		W/m2.OC	W/m2.OC		kPa	
1	350	24651	2998.5	10424.1	182.6	3.81	0.008
2	400	28750	3113.4	12233.6	210.5	5.42	0.008
3	500	33668	3285.1	14411.4	249.6	7.25	0.007
4	600	39172	3437.5	18376.4	290.3	10.33	0.007
5	700	44973	3801	22466	320.2	11.78	0.006

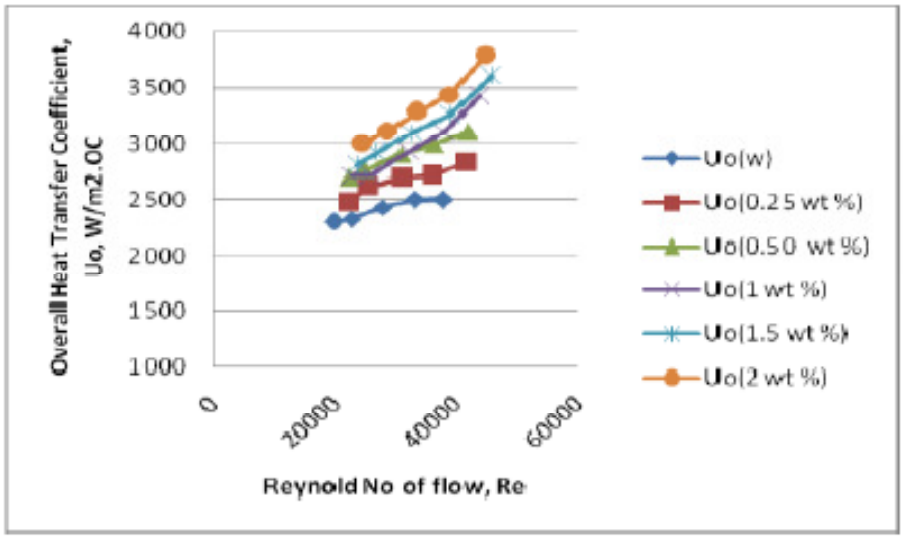

Fig. 1: Uo Vs Re number for Copper oxide-Water Nanofluid

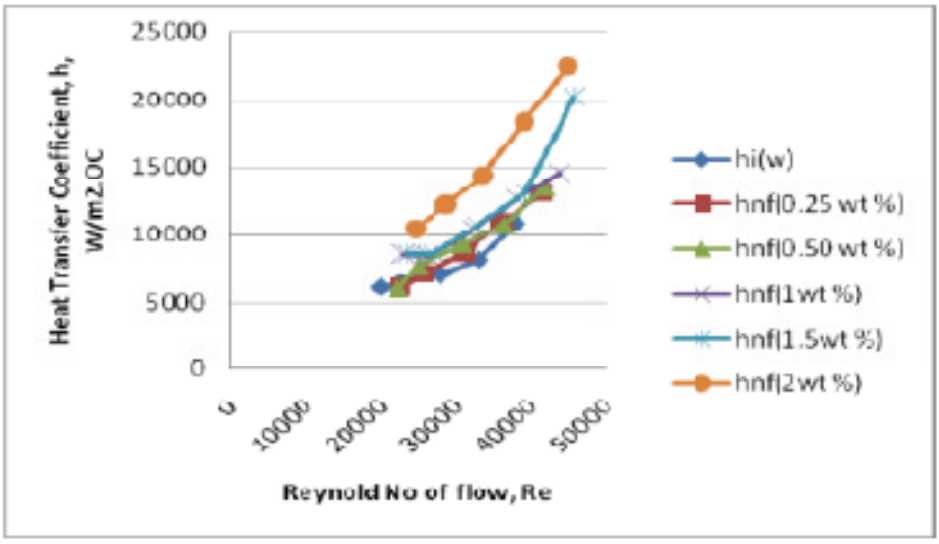

Fig. 2 : hnf Vs Re for Copper oxide-Water Nanofluid

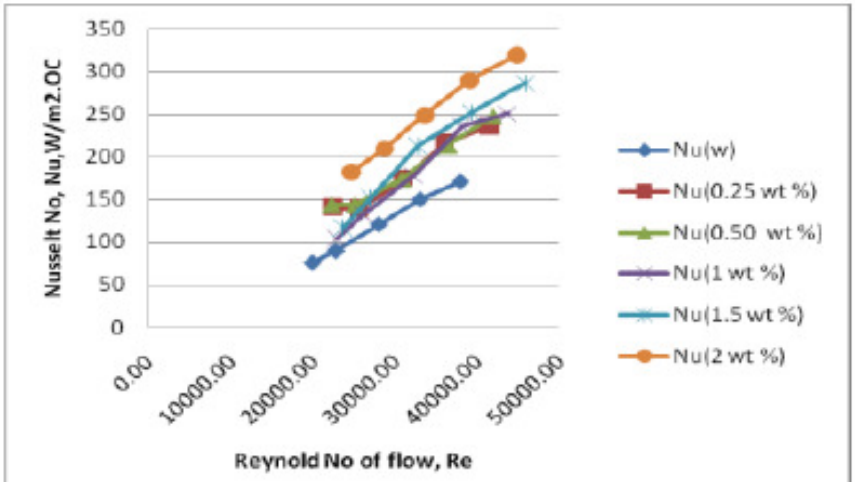

Fig. 3 : Nunf Vs Re for Copper oxide-Water Nanofluid

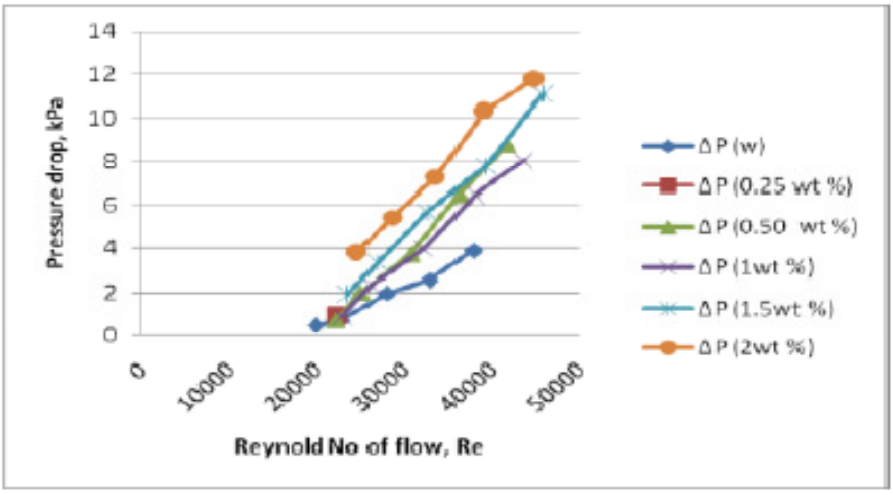

Fig. 4 : ΔPnfVs Re for Copper oxide-Water Nanofluid

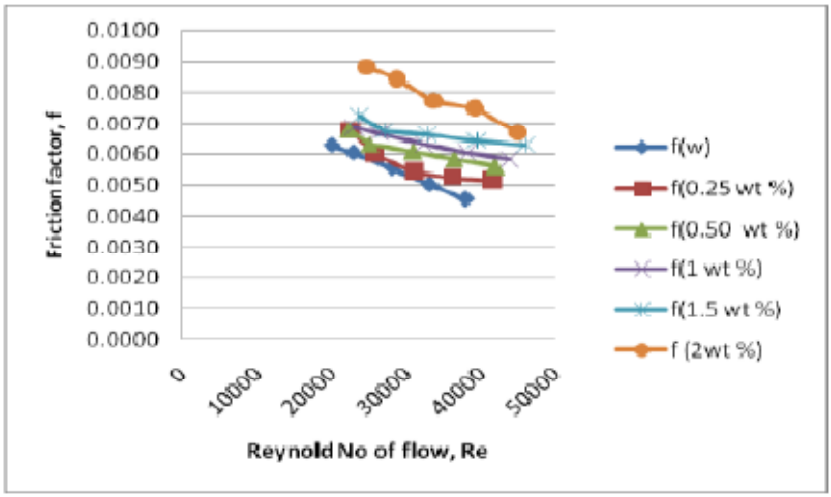

Fig. 5 : fnf Vs Re for Copper oxide-Water Nanofluid

6. ACKNOWLEDGEMENT
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7. CONCLUSION
The convective heat transfer performance and flow characteristic of copper oxide-water nanofluid flowing in a horizontal double concentric tube heat exchanger counter-flow heat exchanger was experimentally investigated.

- Experiments were carried out under turbulent flow conditions, and with five concentrations of copper oxide-water nanofluid viz. 0.25 wt %, 0.50 wt %, 1.0 wt %, 1.50 wt % and 2.00 wt %. The effects of particle concentrations and the Reynolds number on the heat transfer performance and pressure drops of nanofluids are determined. Important conclusions have been obtained and are summarized as follows:
- Dispersion of the nanoparticles into the base liquid increases the thermal conductivity and viscosity of the nanofluids, and this augmentation increases with increasing particle concentrations.
- At a particle concentration of 1.00 wt %, the use of copper oxide-water nanofluid increases convective heat transfer coefficient from 7% to 58% than that of water at same flow conditions. Also at a particle concentration of 2.0 wt %, the use of copper oxide-water nanofluid increases convective heat transfer coefficient from 10% to 70% than that of water at same flow conditions.
- This significant increase in convective heat transfer coefficient is mainly contributed by increased thermal conductivity of nanofluid and increased convection by molecular collisions. Other parameters like particle size, particle shape, agglomerations and nanolayer also augments heat transfer by convection.
- Nusselt number of the flow also increases from 7% to 68% for copper oxide-water nanofluid for concentrations 0.25 wt % to 2.00 wt %.
- The pressure drop of nanofluids increases with increasing Reynolds number and there is a small increase with increasing particle concentrations. This is caused by increase in the viscosity of nanofluids, and it means that nanofluids incur little penalty in pressure drop.

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