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Experimental Study of Convective Heat Transfer in Miniature Double Tube Hair-Pin Heat Exchanger

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ABSTRACT

In this paper, we present experimental investigation of convective heat transfer characteristic in the inner tube fluid of miniature concentric double tube Hair-pin heat exchanger with constant heat flux. This work is carried out for distilled water and a mixture of distilled water and ethylene glycol (10%, 20% and 25% by volume) as heat transfer cold fluid flowing in inner tube to carry away heat from the annulus hot fluid. The obtained Friction factor and heat transfer coefficient are compared with the existing conventional correlations. The present experimental results show that mixing of ethylene glycol in distilled water enhances the thermal performance of the miniature heat exchanger considerably for both laminar and turbulent flow in counter flow arrangements. From the experimental results obtained it's also found that thermal performance is better in laminar region than turbulent region.

Keywords- *Convective heat transfer coefficient, Laminar flow, Nusselt number, Turbulent flow.*

I. INTRODUCTION

Cooling is one of the most important technical challenges facing numerous industries such as automobiles, electronics and manufacturing. Present scenario of the miniature heat exchangers is very concern to many small devices where required heat transfer area is low. Application of the miniature size double tube hair pin heat exchangers are in cars' radiators, radiators of electric transformers, air conditioners, and refrigerators etc. New technological developments are increasing thermal loads and requiring faster cooling. The conventional methods in increasing the cooling rate (fins and micro-channels) are already stretched to their limits. Hence, there is an urgent need for new and innovative coolants to achieve this high performance cooling.

Traditional heat transfer fluids such as water, oil, and ethylene glycol mixtures are inherently poor heat transfer fluids. With increasing global demand, competition, industries have a strong need to develop advanced heat transfer fluids with significantly higher thermal conductivity than water. The thermal conductivity of heating or cooling fluid is a very important property in the development of energy efficient heat transfer systems. At the same time, in all processes involving heat transfer, the thermal conductivity of the fluid is one of the basic properties taken account in designing and controlling the process.

Tsai et al., (1998) tried the heat transfer analysis for the tube heat exchanger by three dimensional finite volume method and **Jang et al., (1997)** also tried numerical heat transfer and fluid flow analysis of a tube heat exchanger. They also performed all these local heat transfer analysis of the tube heat exchangers of complicated shapes numerically. **Wang et al., (1999)** investigated the heat transfer and friction correlation for compact louvered fin and tube heat exchangers in the field of experimental study and result showed a significant enhancement in convective heat transfer coefficient.

The use of additives in the base fluid like water or ethylene glycol is one of the techniques applied to augment the heat transfer. Recently an innovative nanometer sized particles have been dispersed in the base fluid in heat transfer fluids. The fluids containing the solid nanometer size particle dispersion are called 'nanofluids'. The dispersed solid metallic or nonmetallic nanoparticles change the thermal properties like thermal conductivity, viscosity, specific heat, density, heat transfer and friction factor of the base fluid. Nanofluids are having high thermal conductivity and high heat transfer coefficient compared to single phase fluids **(Sundar and Singh, 2013)**.

Enhancement in heat transfer was tried also, with the help of suspended micro-particles;

Ahuja (1975, 1982) conducted experiments on the enhancement of heat transport in the laminar flow of water with micro-sized polystyrene suspension. The results showed a significant enhancement in the Nusselt number and heat exchanger effectiveness compared to that of a single phase liquid. **Pak and Cho (1998)** presented an experimental investigation of the convective turbulent heat transfer characteristics of nanofluids (γAl_2O_3– water and TiO_2–water) with 1–3 volume percent. The Nusselt number for the nanofluids increased with an increasing volume concentration and Reynolds number. **Eastman et al., (2001)** showed that Cu–ethylene glycol (nanoparticles coated with thioglycolic acid) with volume fraction, φ = 0.3% gave a 40% increase in thermal conductivity. Recently, an attempt at the Indira Gandhi Centre for Atomic Research (IGCAR) was made, to align magnetic nanoparticles (Fe_3O_4 coated with Oelic acid) in a base fluid (hexadecane) in a linear chain using a magnetic field, which was applied to increase the thermal conductivity by 300%. **Ding et al., (2006)** investigated the heat transfer performance of Carbon nano-tube (CNT) nanofluids flowing through a tube with a 4.5 mm inner diameter. Their results showed that the heat transfer coefficient of CNT nanofluids is much larger than that of pure water and the enhancement depends on the flow conditions, CNT concentration and PH value.

Heris et al., (2007) presented an investigation of the laminar flow convective heat transfer of Al_2O_3–water under constant wall temperature with 0.2 to 2.5 vol.% of nanoparticle for Reynolds number varying between 700 and 2050. They presented again the Nusselt number for the nanofluid which is greater than the base fluid. **Choi et al., (2008)** showed that nanofluids have the potential to be the next generation of coolants for vehicle thermal management due to their significantly higher thermal conductivities. Several researchers showed that the convective heat transfer coefficient increases substantially for nanofluids. The heat rejection requirements of automobiles and trucks are continually increasing due to trends toward more powerful outputs. **Ho et al. (2010)** conducted an experiment for cooling in horizontal tube in laminar flow of Al_2O_3–water at 1 and 2 vol.% concentrations and concluded the interesting enhancement of 51% in heat transfer coefficient. **Duangthongsuk and Wongwise (2010)** studied on the heat transfer coefficient and the friction factor of the TiO_2-Water nanofluids in a horizontal double tube counter-flow heat exchanger at Reynolds number ranging from 3000 to 18000. Titanium dioxide nanoparticles with diameters of 21 nm dispersed in water with volume concentrations of 0.2 – 2 vol.%. The heat transfer coefficient of nanofluids with 1 vol.% was approximately 26% greater than that of base fluids, while for volume concentration of 2.0 vol.% was approximately 14% lower than that base fluids. The pressure drop of nanofluids was slightly higher than the base fluid and increases with increasing the volume concentrations. **Xie et al., (2010)** reported the convective heat transfer enhancement of nanofluids as coolants in laminar flows inside a circular copper tube with constant wall temperature. Different nanofluids consisting of Al_2O_3, ZnO, TiO_2, and MgO nanoparticles were prepared with a mixture of 55 vol.% distilled water and 45 vol.% EG as base fluid. MgO, Al_2O_3, and ZnO nanofluids exhibited superior enhancements of heat transfer coefficient, with the highest enhancement up to 252% at a Reynolds number of 1000 for MgO nanofluid. **Salman et al., (2013)** reported that SiO2–EG nanofluid has the highest Nusselt number, followed by ZnO–EG, CuO–EG, Al2O3–EG, and lastly pure EG. The Nusselt number for all cases increases with the volume fraction but it decreases with the rise in the diameter of nanoparticles. The maximum Nusselt number is the main target of such research. Many other researchers such as **Nguyen et al., (2007), Sharma et al., (2009), Fotukian and Nasr Esfahny (2010)** have also studied on the convective heat transfer performance and pressure drop of using various size and concentration of nanoparticles flowing in various size and dimension of double tube heat exchangers. Measurements showed that heat transfer coefficient increases significantly with different size and concentration of nanoparticles.

It was observed that only a few studies have been done by using mixtures of low cost base fluids for better enhancement of heat transfer than water. High production cost of nanofluids is main reason that may hinder the application of using nano- fluids in industry. Therefore, in this present study, an experimental investigation has been carried out to study in a double tube Hair-pin heat exchanger.

II. EXPERIMENTAL SETUP

Fig. 1 shows a schematic diagram of the experimental set up. The set up consists of a test section (hairpin double-tube heat exchanger), hot water reservoir, cold fluid reservoir, manometer and two centrifugal pumps. The effective length of double tube Hair-pin heat exchanger across which heat was being transferred is a 45 cm long. The perimeter of U-type return bend is 9 cm. Horizontal double-tube heat exchanger operating in counter flow manners, with cold stream flowing inside the tube while hot water flows in the annulus. The inner tube is made of copper material with 9.5 mm outer diameter (d_o) and 8.43 mm inner diameter (d_i), while the outer tube is made of Galvanized Iron has 22 mm outer diameter (D_o) and 19.05 mm inner diameter (D_i). Installed double tube Hair-pin heat exchanger and the experimental setup in laboratory are shown in fig. 2 and fig. 3 respectively.

Table 1: Dimensions of designed double tube Hair-pin heat exchanger:

Outer pipe specification	**Inner tube specification**
Galvanized Iron pipe	Copper tube of U bend
I.D. of shell = 19.05 mm	I.D. of tube = 8.4 mm
Galvanized Iron pipe	Copper tube of U bend
I.D. of shell = 19.05 mm	I.D. of tube = 8.4 mm
O.D. of shell = 22 mm	O.D. of tube = 9.5 mm
Center to center distance is taken	Wall thickness = 0.55 mm

1.5 - 1.8 times of outer dia. of shell.	Thermal conductivity of wall = 385 w/m^{2}K
Length of each G.I. pipe = 22.86cm	Effective length of copper tube through which heat transfer could take place = 45cm
Total length of the copper tube = straight part (51cm) + U-shaped bend part (9cm) =60cm	

The test section is thermally isolated from its upstream and downstream section by using cotton rope and asbestos mud on it, in order to reduce the heat loss of radial outward direction. The differential pressure transmitter was mounted to measure pressure drop consequently the frictional loss across the inner tube ends, and mercury thermometers were mounted on each of the ends of the test section for both the stream to measure the temperatures of the streams at inlet and outlet. Hot water bath was used as hot water reservoir, from which hot water was continuously supplied as hot stream at constant temperature. The temperature of the tank was maintained by inbuilt PID controller which ensures the constant temperature inside the tank at set point. A common simple manometer consists of a U shaped tube of glass filled with mercury was also used for pressure measurement.

Hot water at 70°C was continuously supplied at constant flow rate of 3 lit/min in annulus. Cold stream at 30°C was supplied at different flow rate to observe rise in temperature of the stream. Friction losses and heat recovery was analyzed with varying flow rate of the cold stream. Cold stream was taken firstly pure distilled water and then a mixture of water and ethylene glycol (10%, 20% and 25% by volume). A coil made of copper material with dimension 9.5mm outer diameter was merged in a cold water reservoir which is a chiller with temperature controller set at needed temperature. The cold stream coming out from the heat exchanger was passed through the coil to lose the energy, consequently come to the inlet temperature of the cold stream.

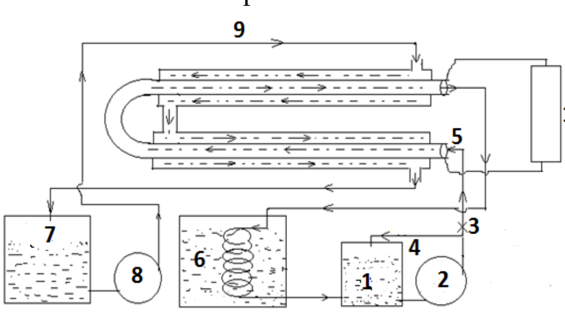

Fig 1: Schematic diagram of experimental setup.
1. Cold fluid reservoir. 2. Pump. 3. Control valve.
4. Bypass. 5. Cold stream. 6. Coolant. 7. Hot water reservoir. 8. Pump. 9. Hot stream. 10. Mercury manometer.

Fig 2: Installed double tube Hair-pin heat exchanger in laboratory.

Fig 3: Experimental setup in laboratory. 1. Cold fluid reservoir. 2. Pump. 3. Rotameter. 4. Cooling system with PID controller. 5. Double tube Hair-pin heat exchanger. 6. Mercury thermometer. 7. Mercury manometer. 8. Hot water reservoir with PID temperature controller.

III. HEAT TRANSFER ANALYSIS

This resemble the counter flow study in double tube heat exchanger using hot water supplying in annulus as heating medium to heat the wall of the inner tube where constant heat flux was maintained and this heat was carried away by supplying the cold water as cooling medium in tube side.

Rate of heat transferred from the hot fluid to outer wall of the tube,

$$q_h = \dot{m}_h C_{ph}(T_{hi} - T_{ho}) \quad (1)$$

Where, $\dot{m}_h$ C_{ph}, T_{hi} and T_{ho} are mass flow rate, heat capacity, inlet and outlet temperature of the hot stream respectively.

Heat gained by the cold stream from wall is given by

$$q_c = \dot{m}_c C_{pc}(T_{co} - T_{ci}) \quad (2)$$

Where, $\dot{m}_c, C_{pc}, T_{ci}$ and T_{co} are mass flow rate, heat capacity, inlet and outlet temperature of the cold stream respectively.

In ideal case the heat energy lost by the hot stream is equal to the heat energy gained by the cold stream when heat exchanger is well insulated but in the real sense it is not equal. The heat gained by the cold stream is the actual heat transferred between the hot and cold streams. We can observe from the experiment that the value of q_h is more than the value of q_c, it means the amount of heat i.e., $q_h - q_c$ is lost into the environment through radiation mode.

Overall heat transfer coefficient on the basis of inner diameter of inner tube can be evaluated as

$$\frac{1}{U_i} = \frac{1}{h_i} + \left(\frac{x}{k}\right)\frac{A_i}{A_m} + \left(\frac{A_i}{A_o}\right)\frac{1}{h_o} \quad (3)$$

Where, x and k are the wall thickness and thermal conductivity of inner tube respectively. A_i and A_o are cross-sectional areas of the inner tube on the basis of inner and outer diameter respectively. h_i and h_o are convective heat transfer coefficients of inner and outer side of the inner tube respectively. A_m is log mean cross-sectional area.

In the above equation (3) the middle term of the right side has negligible value because of high value of thermal conductivity of the copper material. Hence, this term can be neglected.

The heat transfer coefficient on the basis of inner side of the inner tube is given by the following equation,

$$\frac{1}{h_i} = \frac{1}{U_i} - \left(\frac{A_i}{A_o}\right)\frac{1}{h_o} \quad (4)$$

Where, U_i is the overall heat transfer coefficient on the basis of inner diameter of the inner tube and it can be calculated experimentally from the following equation,

$$q_c = U_i A_h \Delta T_{lmtd} \quad (5)$$

Where, A_h is the heat transfer area which is the area of curved surface of the inner tube and it can be calculated as

$$A_h = \pi d_o l \quad (5a)$$

Where, d_o is the outer diameter of the inner tube, and l is the effective length of heat exchanger along which heat is being transferred from hot stream to cold stream.

And, h_o is the heat transfer coefficient on the shell side (annulus) which can be calculated from a valid empirical correlation.

ΔT_{lm} is the log mean temperature difference given by

$$\Delta T_{lm} = \frac{(T_{ho} - T_{ci}) - (T_{hi} - T_{co})}{ln\frac{(T_{ho} - T_{ci})}{(T_{hi} - T_{co})}} \quad (5b)$$

Theoretical Nusselt number can be calculated from a valid correlation and is given by ***Hausen (1943)*** for laminar flow through circular tubes,

$$N_u = 3.66 + \frac{0.0668\left(\frac{d_i}{l}\right)R_e P_r}{1+0.04\left\{\left(\frac{d_i}{l}\right)R_e P_r\right\}^{(2/3)}} \quad (6)$$

This correlation is valid for laminar regime (R_e < 2100)

Where, d_i is inner diameter of the inner tube through which coolant is flowing and l is the effective length of the inner tube.

Theoretical heat transfer coefficient can be calculated from a valid empirical correlation and is given by *Gnielinski correlation* **(Celata et al., 2007)** for turbulent flow in tubes.

$$N_u = \frac{\left(\frac{f}{8}\right)(R_e - 1000)P_r}{1+12.7\left(\frac{f}{8}\right)\left(P_r^{\left(\frac{2}{3}\right)} - 1\right)} \quad (7)$$

$$f = (0.79\ln(R_e) - 1.64)^{-2} \quad (7a)$$

This correlation is valid for $3000 \le Re \le 5*10^6$
$0.5 \le Pr \le 2000$

$$R_e = \frac{\rho_f U d_i}{\mu_f} \quad (7b)$$

$$P_r = \frac{\mu_f C_{pf}}{K_f} \quad (7c)$$

$$N_u = \frac{h_i d_i}{K_f} \quad (7d)$$

Where, ρ_f, μ_f, C_{pf} and K_f are density, viscosity, heat capacity and thermal conductivity of the fluid respectively. U is average velocity of the fluid flowing in inner tube.

IV. RESULTS AND DISCUSSIONS

The experiments are conducted for single phase flow of water and ethylene glycol mixtures in the inner tube of double tube Hair-pin heat exchanger within the flow rate ranges from 1-5 litre/minute. The inner diameter of inner tube is 8.4 mm. The obtained Reynolds number are in the ranges from 800 to 18000. A fully developed flow condition is assumed for numerical calculations. Experimentally measured Nusselt number are compared against the values obtained numerically by Hausen and Gnielinski correlations for laminar and turbulent flow respectively as given in above equations.

Fig. 4 represents comparison of experimental and numerical Nusselt number for water and ethylene glycol mixtures (0%, 10%, 20% and 25% by volume) with Reynolds number. It can be easily observed that experimental results show good agreement with the numerical results for lower Reynolds number ($Re \le 2100$). But for higher Reynolds number ($Re \ge 3000$) experimental Nusselt number is lesser than the numerical nusselt number. For flow rate ranges from 1-5 litre/minute, the fluid is in turbulent flow only for distilled water and 10% ethylene glycol-water mixture but it is in laminar flow for lower flow rate and turbulent flow for higher flow rate for both the 20% and 25% ethylene glycol-water mixtures.

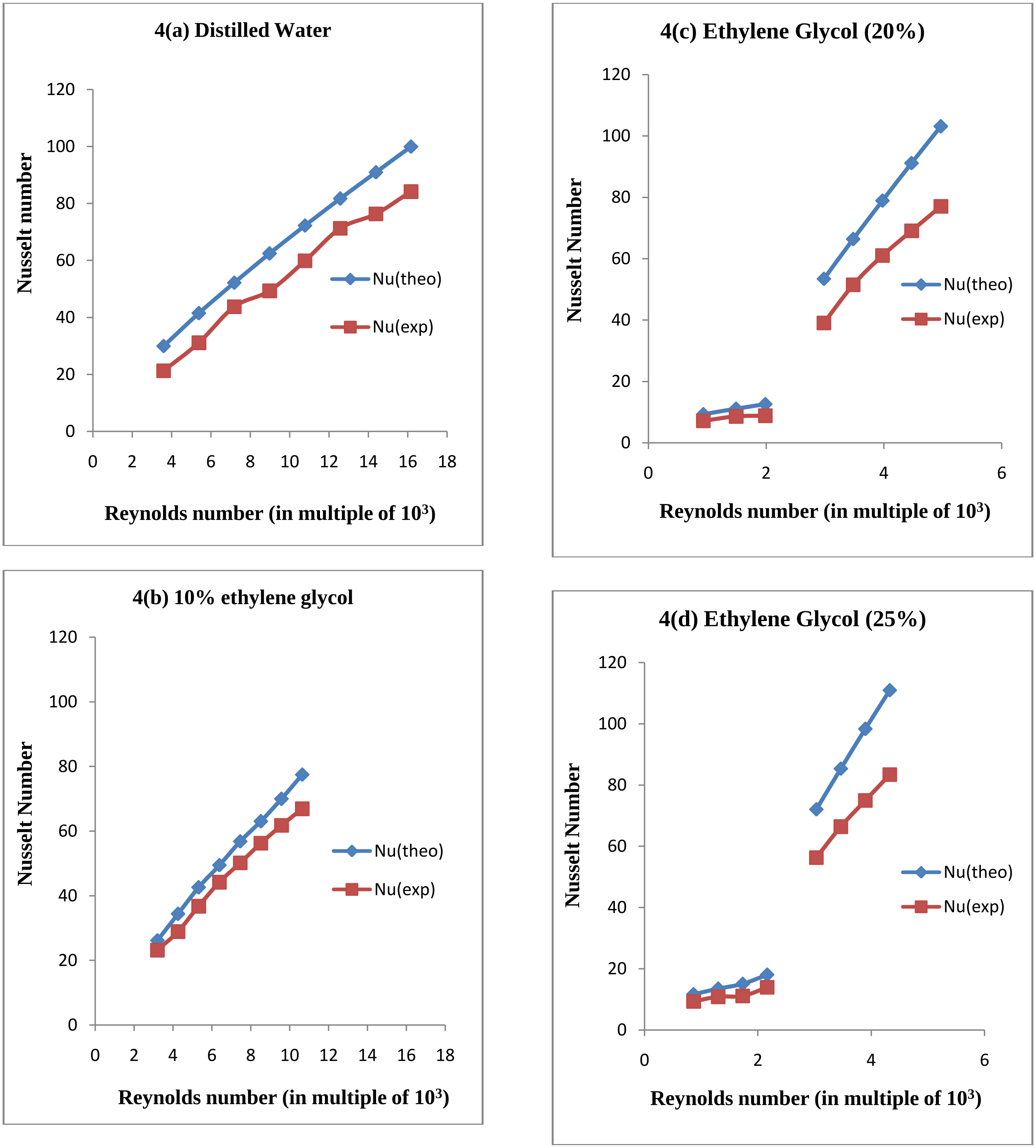

Fig. 4 Comparative study of experimental and theoretical Nusselt number with Reynolds number for water-ethylene glycol concentrations (0%, 10%, 20% and 25%)

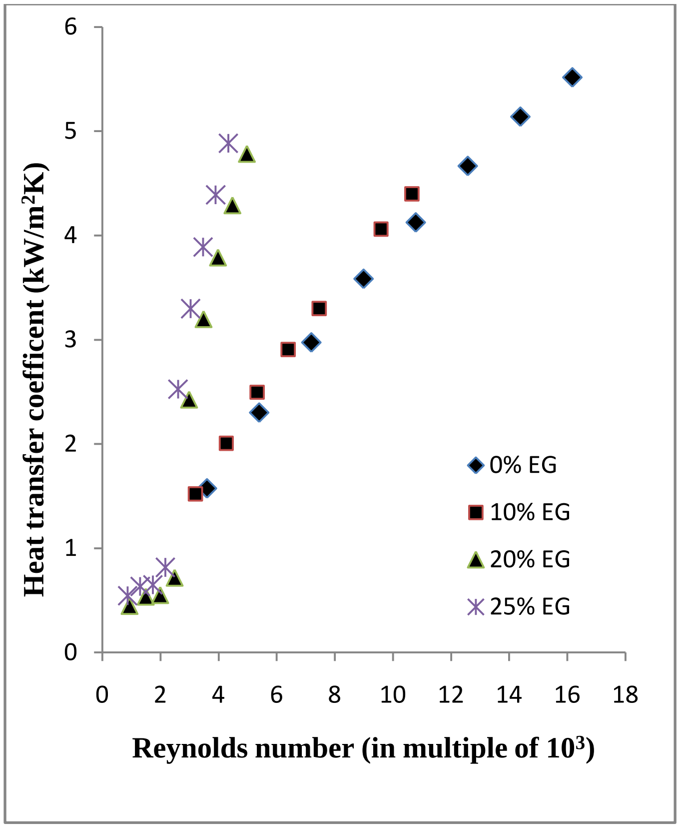

Fig. 5: Variation of Heat transfer coefficient with Reynolds number for various composition of Ethylene glycol in distilled water

Fig. 5 represents comparison of experimental heat transfer coefficient against Reynolds number for distilled water and mixture of distilled water and ethylene glycol (10%, 20% and 25% by volume). We can analyze from the following curve that if we increase the percentage composition of ethylene glycol heat transfer coefficient increases rapidly even for lesser Reynolds number. For the same value of Reynolds number, the heat transfer coefficient for higher concentration of ethylene glycol was found much larger than that for lower concentration of ethylene glycol. It means high concentration of ethylene glycol enhances the heat recovery thereby increasing the convective heat transfer coefficient.
Since the heat transfer coefficient is function of Reynolds number and prandtl number. In heat transfer study viscosity and thermal conductivity of the fluid plays more important role than the density of the fluid. In this study a mixture of ethylene glycol and water is taken as heat transfer fluid which has more density and viscosity and less thermal conductivity than pure distilled water, therefore Prandtl number found is increased with increasing the concentration of ethylene glycol. Higher Prandtl number shows a very low thermal boundary layer, it means convection mode of heat transfer for same temperature driving force becomes more dominant that is why convective heat transfer coefficient enhances. From experimental study the thermal performance of miniature heat exchanger was found better for laminar flow only. Since, the empirical relations for Nusselt number have been generated for the conventional heat exchangers. In laminar flow the heat transfer coefficient curve approaches near the empirical curve but in turbulent flow it diverges from the empirical curve.

V. CONCLUSIONS

Experiment was performed for the laminar and turbulent flow of varying composition of ethylene glycol in water. The experimental results indicate that heat transfer coefficient of a mixture of ethylene glycol and water increases with Reynolds number as well as ethylene glycol concentration. This is because of enhanced fluid properties. Factors such as thermal conductivity, density and viscosity, especially in high Prandtl number may cause the augmentation of heat transfer coefficient. A slight increment in the flow rate of the cold stream shows more augmentation in heat transfer coefficient for the concentration of 25% ethylene glycol mixture than that for lesser concentration.

Considering the economy and mechanical stability of the heat exchanger, the concentration of ethylene glycol cannot be increased above the optimal limit, because more concentration provides higher pressure drop and pumping cost. Hence, the Experimental investigation shows that the fabricated Double tube Hair-pin heat exchanger is more suitable for laminar flow.

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