International Journal of Engineering and Technology Innovation, vol. 5, no. 2, 2015, pp. 99-107

Effect of Corrugation Angle on Heat Transfer Studies of Viscous Fluids in Corrugated Plate Heat Exchangers

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Received 10 June 2014; received in revised form 18 December 2014; accepted 13 January 2015

Abstract

In the present investigation heat transfer studies are conducted in corrugated plate heat exchangers (PHEs) having three different corrugation angles of 30^0, 40^0 and 50^0. The plate heat exchangers have a length of 30 cm and a width of 10 cm with a spacing of 5 mm. Water and 20% glycerol solution are taken as test fluids and hot fluid is considered as heating medium. The wall temperatures are measured along the length of exchanger at seven different locations by means of thermocouples. The inlet and outlet temperatures of test fluid and hot fluid are measured by means of four more thermocouples. The experiments are conducted at a flowrate ranging from 0.5 lpm to 6 lpm with the test fluid. Film heat transfer coefficient and Nusselt number are determined from the experimental data. These values are compared with different corrugation angles. The effects of corrugation angles on heat transfer rates are discussed.

Keywords: corrugation angle, corrugated plate heat exchanger, Nusselt number, viscous fluids.

1. Introduction

Nowadays, plate heat exchangers are universally used in many fields such as heating and ventilation, breweries, dairy, food processing, pharmaceuticals and fine chemicals, power generation plants petroleum and petrochemical industries, pulp and paper industries, etc. The details of design of plate heat exchangers are given in reference [1]. The design of sinusoidal wavy type arrangement results in more complex flow structure and that improves the heat transfer rates by two/three times that of conventional straight channels [2-3]. The corrugated plate heat exchanger improves heat transfer performance by improving heat transfer area and promoting formation of vortex in the flow [4-5]. The reason for the wide spread application of PHE compared to shell and tube heat exchanger in industry today is not only due to the ease of maintenance, but also because high turbulence can be achieved at low fluid velocities, which results in high overall heat transfer coefficient. With plate heat exchangers heat can be recovered with comparatively lower temperature difference between the fluids [6]. The corrugated walls of corrugated PHE are effective on heat transfer enhancement by breaking and destabilizing the thermal boundary layer, therefore corrugated or wavy surfaces operate as turbulence promoters to increase local heat transfer coefficients [7]. The heat transfer is enhanced by a factor of 2.5 [8] and by a factor of 3 [9] as compared to straight duct.

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Heggs et al. [10] developed electrochemical mass transfer techniques to calculate values of local heat transfer coefficients within corrugated PHE channels. They performed experiments over 30^0, 45^0, 60^0, and 90^0 corrugation angles. They analyzed that the mass transfer coefficient results revels that particularly at low corrugation angle (30^0 corrugation angle), pure laminar flow does not occur for the Reynolds number range of 150 to 11500. Pandey and Nema [11] conducted experiments to determine the heat transfer characteristics for fully developed flow of air and water flowing in alternate corrugated ducts. A test section was formed by three identical corrugated channels having corrugation angle of 30^0 with cold air flowing in the middle one and hot water equally divided in the adjacent channels. He obtained various correlations of Nusselt number for water and air. Lin et al. [12] used Buckingham Pi theorem to study dimensionless correlation to characterize the heat transfer performance of the corrugated channel in a plate heat exchanger for the range of Reynolds number 300 to 7000. They performed experiments over 15^0 to 45^0 corrugation angles. They substituted the experimental data into these correlations to identify the flow characteristics and channel geometry parameters with the most significant influence on the heat transfer performance. They observed that the dimensional analysis reveals local Nusselt number (Nux) is determined primarily by Re, R/Dh (geometry), x/Dh (location) and θ (corrugation angle), but the effect of Pr and ΔT (temperature difference between flow and wall) do not have significant effect on heat transfer performance. From the correlation it is seen that, the Nusselt number is increases with increase in corrugation angle. Brien et al. [8] conducted experiments to determine forced convection, heat transfer coefficients and friction factors for flow in a corrugated duct and they have taken the corrugation angle as 30^0 and inter-wall spacing equal to the corrugation height. They observed that the results of Reynolds number based on hydraulic diameter varied from 1500-25000 and the Prandtl number was varied from 4 to 8 and also they derived the Nusselt numbers which will give high turbulence level compared to the conventional parallel plate channels. Veli et al. [13] studied numerically the flow field and heat transfer of water in two dimensional sinusoidal and rectangular corrugated wall channels for the range of Reynolds number 100 to 1000. They studied the effect of Reynolds number, spacing and wall geometry on the flow characteristics, local heat transfer coefficient and heat transfer rates. They observed that the heat transfer through corrugated walled geometries is always higher than that of the flat plate.

From the literature, it is observed that the work on corrugated PHEs is limited to water or air as the test fluid and with one corrugation angle only. Therefore, it is planned to undertake investigations in corrugated PHEs with different test fluids (having different viscosities) and also with different corrugation angles. It is also planned to study the heat transfer rates for different viscous fluids and to study the effect of corrugation angles on the rates of heat transfer.

2. Experimental Setup

Table 1 Dimensions of PHE

Sr. No.	Specification	Dimension
1	Length of each plate	30 cm
2	Width of each plate	10 cm
3	Plate spacing	0.5 cm
4	Corrugation angle	30^o, 40^o & 50^o

The experiments have been conducted on a custom manufactured plate heat exchanger unit shown in Fig. 1. The setup consists of a test box, test fluid tank, test fluid collection tank and hot water tank. Each test box consists of three sinusoidal corrugated plates welded together to form two horizontal channels. The sinusoidal plate heat exchanger shown in Fig. 2 has the dimensions shown in Table-1, three test boxes of having three different corrugation angles of 30^0, 40^0 and 50^0 are considered here. The flow through these two channels is controlled using rotameters. The flow pattern adopted is countercurrent manner. The seven thermocouples are fixed on the top of the middle plate along the length of the section to measure the temperatures at different locations. The fixing of thermocouples is shown in Fig. 3. Plate heat exchanger units of 30^0, 40^0 and 50^0 corrugation angles are shown in Figs. 4, 5 and 6 respectively.

Fig. 1 Experimental setup of plate heat exchange

Fig. 2 Sinusoidal test section

Fig. 3 Top view of thermocouples fixing on middle plate

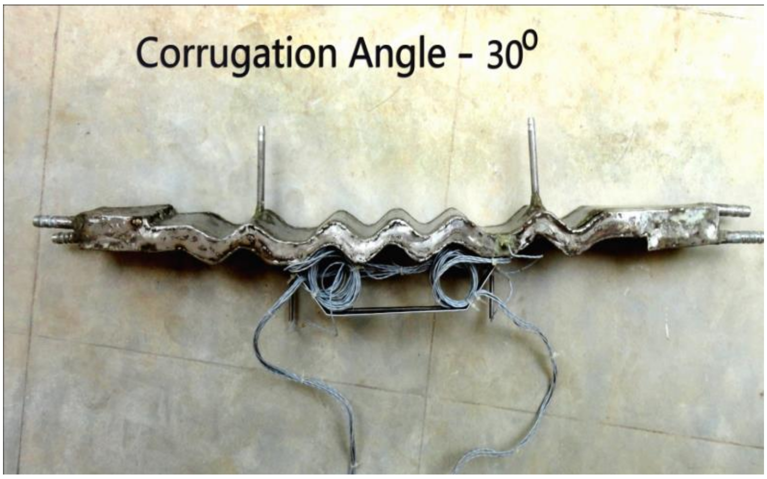

Fig. 4 PHE unit with 30^0 corrugation angle

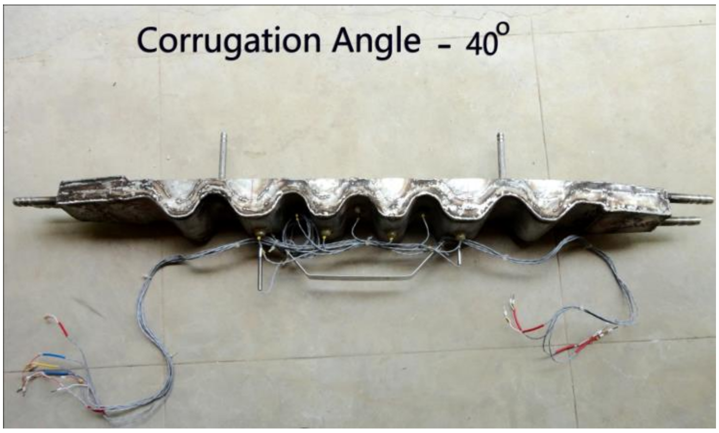

Fig. 5 PHE unit with 40^0 corrugation angle

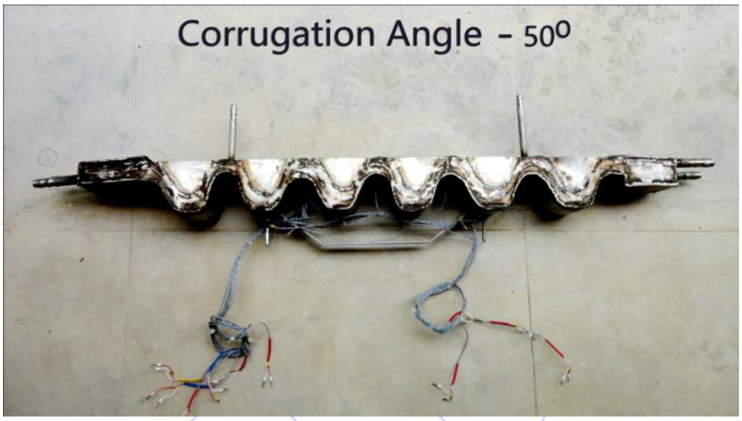

Fig. 6 PHE unit with 50^0 corrugation angle

Three plates of corrugation configuration, forming two channels, one for the test fluid (bottom) and another for the hot fluid (top). The corrugation angle is taken with reference to the horizontal plane and shown in the Fig. 7.

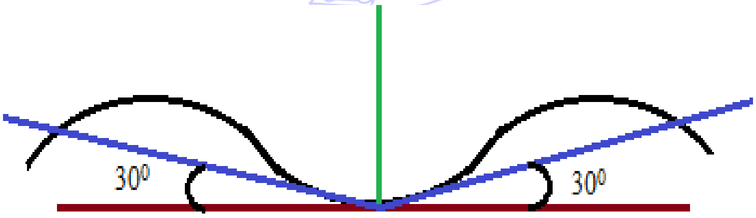

Fig. 7 Corrugated angle of corrugated plate

3. Existing Mechanism

The experiments were carried out in the plate heat exchanger having 5 mm spacing. The corrugation angles used are 30^0, 40^0 and 50^0. The experiments were carried out with the water of viscosity 0.7284 cp at 35^0C as test fluid. For all the experiments hot water at 70^0C at constant flow rate was used for heating the test fluids. For each experimental reading, the inlet and outlet temperatures of the fluid as well as the wall temperatures on the heat exchanger plate at seven different locations were noted by means of thermocouples welded at these locations and read through the digital temperature indicator. These temperatures were used for the analysis of heat transfer. The average temperature of wall is found in the range of 43.6^0C to 68.4^0C. For making the heat transfer studies, the hot fluid flow rate was maintained constant. The test fluids were pumped into the bottom channel through the calibrated rotameter from 0.5 to 6 lpm. The sufficient length at inlet and outlet is provided to have end effects. The middle plate is fitted with 7 thermocouples, along the length and breadth of the plate, to measure the wall temperatures. Four more thermocouples are inserted into the bulk fluid to measure the inlet and outlet temperatures of hot and cold fluids. These thermocouples are connected to a digital temperature indicator having an accuracy of 0.1^0C. For each flow rate the inlet and outlet temperatures as well as the wall temperatures were noted from the temperature indicator, when it shows a constant value. For all the heat transfer studies the inside film heat transfer coefficient (hi) is calculated by making an energy balance with log mean temperature difference (LMTD). The viscosity and specific gravity of the fluids are determined experimentally by Redwood Viscometer no. 1 and hydrometer respectively.

The specifications of measuring devices i.e. thermocouples and rotameters in present study for the experimental setup with their range and error associated with it are presented in table 2.

Table 2 Specifications of measuring device utilized in the experiment with the range of parameters and error.

Measured Parameter	Measuring Device	Range	Error (%)
Fluid temperature	Thermocouple	20-69^{0}C	±2%
Fluid flow rate	Rotameter	0.5-6 lpm	±5%

4. Calculation Procedure

The effect of Reynolds number on Nusselt number is more significant than that of any of the other parameters [9]. The arithmetic mean temperature of wall measured at seven different locations is Tw,avg, using this temperature the log mean temperature difference (LMTD) is calculated using equation (1). The rate of heat transfer is calculated using equation (2) applied for cold fluid (test fluid). The average heat transfer coefficient is calculated using equation (3).

$$LMTD = \frac{(T_{avg} - T_{c,in}) - (T_{avg} - T_{c,out})}{\ln \frac{T_{avg} - T_{c,in}}{T_{avg} - T_{c,out}}} \tag{1}$$

$$Q = MFR \times C_P \times \Delta T_{cold} \tag{2}$$

$$Q = hA(LMTD) \tag{3}$$

Where $T_{c,in}$ and $T_{c,out}$ are inlet and outlet temperatures of cold fluid (test fluid). Where MFR is mass flowrate of water in kg/s. C_p is specific heat capacity of water in J/kgK. All the temperatures are measured in ^{0}C. Once heat transfer coefficient is known, the Nusselt number were calculated using equation (4).

$$Nu = \frac{hD_H}{k} \tag{4}$$

Here D_h is hydraulic diameter of channel and this was calculated by use of equation (5) and Reynolds number was calculated using equation (6).

$$D_h = \frac{4A}{p} = \frac{2Wx}{W+x} \tag{5}$$

$$Re = \frac{D_h v \rho}{\mu} \tag{6}$$

5. Results and Discussion

All the experimental data collected for a system along with the different corrugation angles are plotted on graphs are shown in Figs. 8 to 19. Figs. 8 to 10 represents the variation of heat transfer coefficient with the Reynolds number for 30^0, 40^0 and 50^0 corrugation angle for water and 20% glycerol solution as test fluids. It is observed from these graphs that the heat transfer coefficient is high for high viscous fluids. This is due to the increase in thickness of laminar sub-layer for increase in viscosity of fluid. The similar trends and results were found for the Nusselt number versus Reynolds number plots as shown in Figs. 11 to 13.

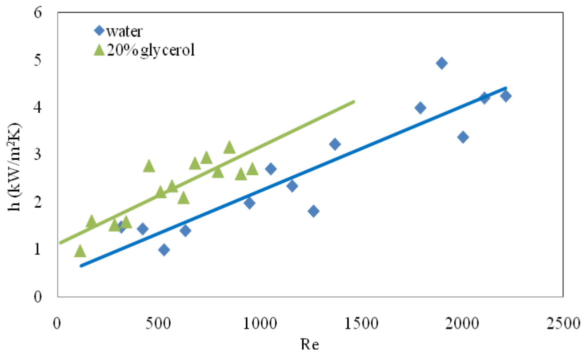

Fig. 8 Heat transfer coefficient Vs Reynolds number for 30^0 corrugation angle with water and 20% glycerol solution as test fluids

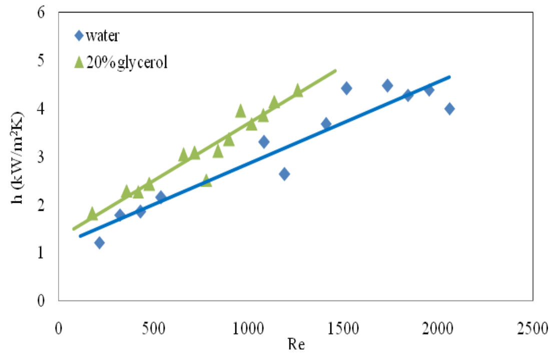

Fig. 9 Heat transfer coefficient Vs Reynolds number for 40^0 corrugation angle with water and 20% glycerol solution as test fluids

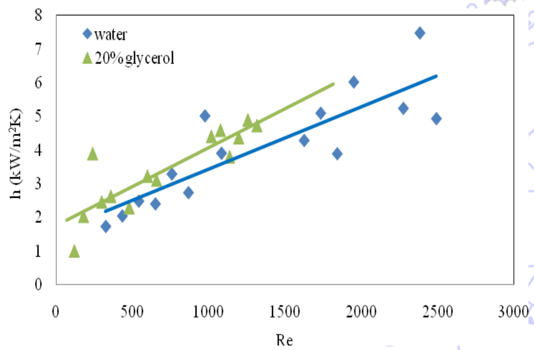

Fig. 10 Heat transfer coefficient Vs Reynolds number for 50^0 corrugation angle with water and 20% glycerol solution as test fluids

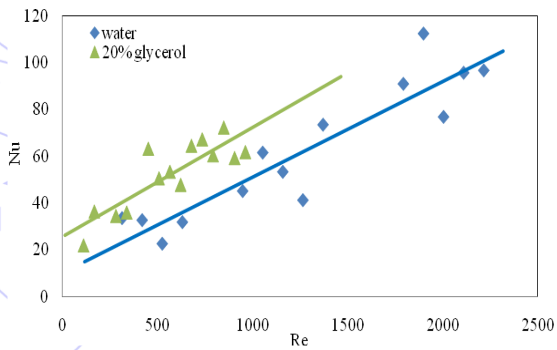

Fig. 11 Nusselt number Vs Reynolds number for 30^0 corrugation angle with water and 20% glycerol solution as test fluids

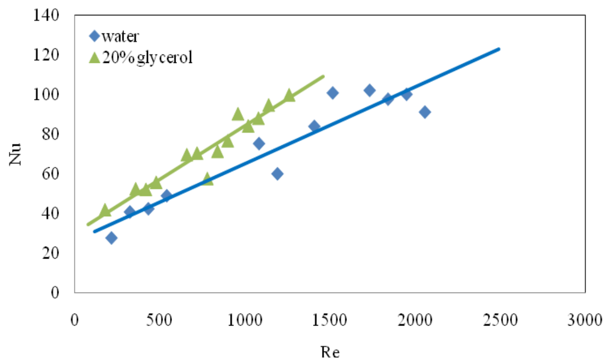

Fig. 12 Nusselt number Vs Reynolds number for 40^0 corrugation angle with water and 20% glycerol solution as test fluids

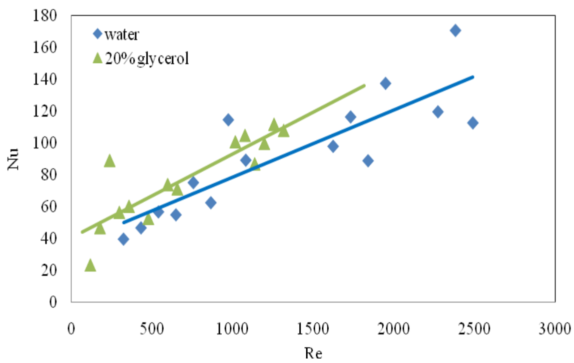

Fig. 13 Nusselt number Vs Reynolds number for 50^0 corrugation angle with water and 20% glycerol solution as test fluids

The variation of heat transfer coefficient with the Reynolds number for three different corrugation angles for water and 20% glycerol solution as test fluids are shown in Figs 14 and 15 respectively. It is observed from these figures that the heat transfer coefficient is higher for higher corrugation angle. Therefore the corrugated PHE with 50^0 corrugation angle will give more heat transfer performance. The trends given in Figs. 14 to 15 are in agreement with the reported work in literature (J H Lin (2007)).

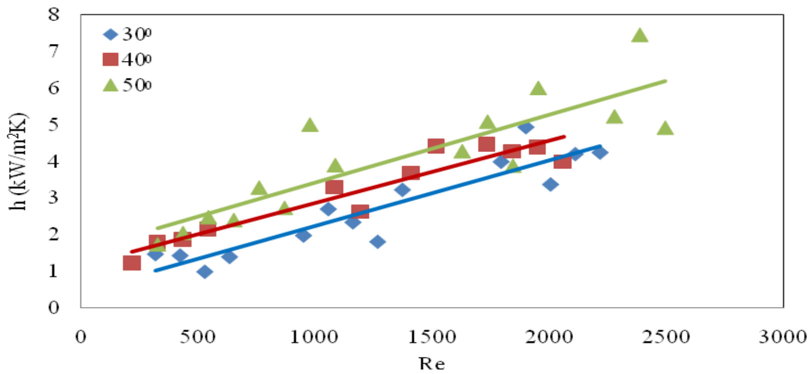

Fig. 14 Heat transfer coefficient Vs Reynolds number for 30^0, 40^0 and 50^0 corrugation angles with water as test fluid

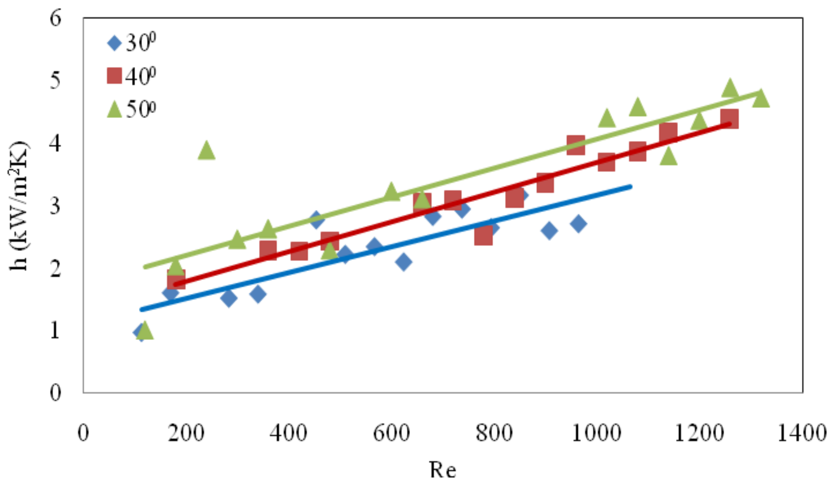

Fig. 15 Heat transfer coefficient Vs Reynolds number for 30^0, 40^0 and 50^0 corrugation angles with 20% glycerol solution as test fluid

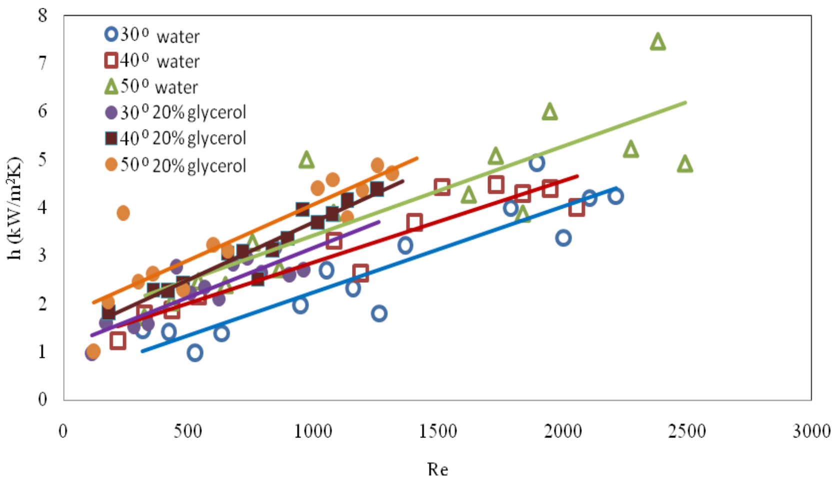

Fig. 16 Heat transfer coefficient Vs Reynolds number for 30^0, 40^0 and 50^0 corrugation angles with water and 20% glycerol solution as test fluids

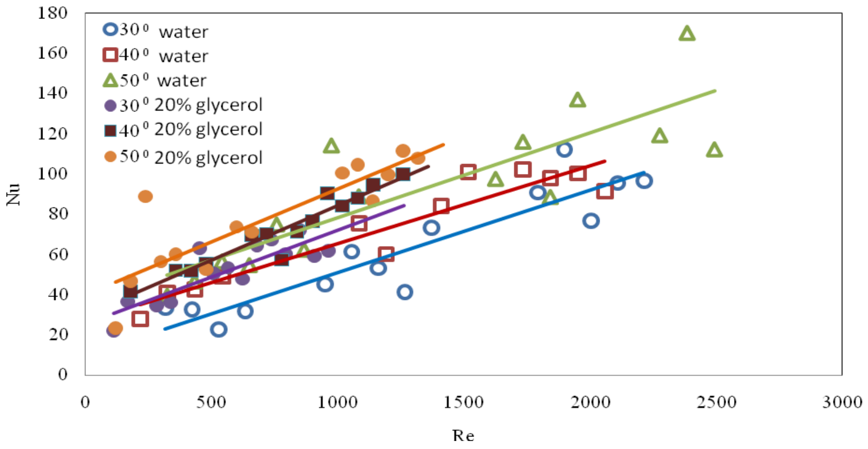

Fig. 17 Nusselt number Vs Reynolds number for 30^0, 40^0 and 50^0 corrugation angles with water and 20% glycerol solution as test fluids

The combined effect of corrugation angle and effect of viscosity of a test fluid are shown in Figs. 16 and 17. Fig. 16 shows the variation of heat transfer coefficient with the Reynolds number. It is observed from this figure that the heat transfer coefficient at giving Reynolds number is more for 50^0 corrugation angle and 20% glycerol solution as test fluid when compared with other systems. Because high turbulence is generated for higher corrugation angles and heat transfer rates are more for more viscous fluids. The similar results were found for the variation of Nusselt number with the Reynolds number as shown in Fig. 17. The average percentage increase of heat transfer coefficient or Nusselt number of water and 20% glycerol solution for 30^0 to 40^0 corrugation angle is 20%, for 40^0 to 50^0 corrugation angle is 9% and for 30^0 to 50^0 corrugation angle is 32%. Therefore, 50^0 corrugated plate heat exchanger will give more heat transfer performance with high viscous fluid.

6. Conclusions

Heat transfer coefficients are experimentally determined for different corrugation angles and their results are compared. It is observed from the experiments that the heat transfer coefficients and Nusselt number are higher at a given Reynolds number for 50^0 corrugation angle as compared to 30^0 and 40^0 corrugation angles. For higher corrugation angle, the heat transfer rates are higher due to the generation of higher turbulence. As the corrugation angle increases, there will be significant improvement in the Nusselt number or heat transfer rates for high viscous fluids compared to low viscous fluids. It is also found that the heat transfer rates are high for high viscous fluids. The mean percentage increase of heat transfer coefficient or Nusselt number of water and 20% glycerol solution for 30^0 to 40^0 corrugation angle is 20%, for 40^0 to 50^0 corrugation angle is 9% and for 30^0 to 50^0 corrugation angle is 32%.

Acknowledgements

Authors are grateful to the authorities of NIT Warangal and CBIT Hyderabad for providing facilities and technical support.

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LIST OF SYMBOLS AND NOMENCLATURE

A	Heat transfer area, m2	Tc,out	Outlet cold fluid temperature, K
CP	Specific heat capacity, J/kgK	v	Velocity, m/s
Dh	Hydraulic diameter, m	x	Thickness of wall, m
h	Heat transfer coefficient, W/mK	ΔT	Temperature difference between flow and wall
k	Thermal conductivity, W/m2K	μ	Dynamic viscosity of fluid
Nu	Nusselt number	ρ	Density of fluid
Pr	Prandtl number	θ	Corrugation angle
Q	Rate of heat transfer, W	cp	Centipoise
R	Radius of Pipe, m	LMTD	Logarithmic Mean Temperature Difference
Re	Reynolds number	lpm	Liter Per Minute
Tavg	Average wall temperature, K	PHE	Plate Heat Exchanger
Tc,in	Inlet cold fluid temperature, K	TEMA	Tubular Exchanger Manufacturers Association