Optimization of corrugation angle for the heat transfer performance of corrugated plate heat exchangers

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Abstract - *The corrugated plate heat exchanger (PHE) is used to carry out the experiments for water as both hot and cold fluids. The experiments were conducted for 30^0 corrugation angle with 15 mm channel spacing hot fluid and 5 mm spacing of cold fluid. The Realizable k-ε model was used for CFD simulation of PHE to validate given model with experimentation. The thermal analysis and pressure drop calculation were done for different corrugation angles of 10^0, 20^0, 30^0, 40^0, 50^0, 55^0, 60^0 and 65^0 with fixed channel spacing of 7.5 mm. The CFD analysis was done to find optimal corrugation angle which will have high heat transfer to pressure drop ratio. It was found that heat transfer rate and pressure drop decreases as channel spacing increases, also heat transfer rate and pressure drop increases as corrugation angle increases.*

Key Words: Corrugated plate heat exchanger, corrugation angle, channel spacing, heat transfer analysis.

1. INTRODUCTION

There is current interest among various industries for the development of advanced heat exchangers that can effectively exchange heat between two fluid streams having a low temperature difference. Corrugated plate heat exchangers (PHEs) are widely used for such purposes and are particularly useful for heat recovery and energy conversion-based applications. As noted in [1], one of the earliest references to a corrugated plate heat exchanger can be found in a U.S. patent by Drache in Germany in 1878. During the past five decades rapid advances in engineering technology related to nuclear energy, fossil energy, electric power generation, ink-jet printers, and electronic chips cooling, dairy industries have expedited research in a variety of subjects related to heat transfer. Among the subjects, many engineering systems include problems related to heat transfer enhancement in corrugated plate heat exchangers (PHE). Accordingly, various experimental studies for enhancement of the PHE heat transfer have been proposed and studied. In addition to the fundamental nature of the corrugation patterns, these studies incorporated important parameters such as corrugation amplitude, corrugation wave length, corrugation inclination angle and flow rate [1-3].

In the literature [4-5], it is shown that the use of corrugated channel results in a more complex flow structure and improves the heat transfer by as much as two or three times compared to flat plate geometry. The inclination angle of the crests and furrows of the sinusoidal corrugation heat exchanger pattern relative to the main flow direction has been shown to be the most important design parameters with respect to fluid friction and heat transfer [6]. Modeling and simulation of corrugated PHE have been studied by many authors; a mathematical model is developed in algorithmic form for the steady state simulation of gasket PHE with generalized configurations [6], with this simulation the temperature profile in all channels, thermal effectiveness, distribution of the overall heat transfer coefficient and pressure drops could be calculated [7,8].

From the literature [7], realizable k-ε turbulence model has been most widely employed for the plate heat exchanger design optimization. The simulations generally yield results within good agreement with the experimental studies ranging from 2% to 10%.

J.E.O. Brien and E. M. Sparrow [4] performed experiments to determine forced convection heat transfer coefficients and friction factors for flow in a corrugated duct for a corrugation angle 30^0 and the inter-wall spacing equal to the corrugation height. J. H. Lin et al. [8] developed various correlations to determine local as well as average Nusselt number over 15^0 to 45^0 corrugation angles using experimental data. S. D. Pandey and V. K. Neema [9] conducted experiments to determine the heat transfer characteristics for fully developed flow of air and water flowing in alternate corrugated ducts with an inter-wall spacing equal to the corrugation height. The 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.

From the literature, it was seen that there is limited work on different corrugation angle with different channel spacing's. Also there is limited work on optimization of corrugation angle which gives optimal heat transfer rate with minimal pumping cost. In this report, the realizable k-ε model with standard wall functions was used for CFD simulation. This model have validated by conducting experiments for 30^0 corrugation angle with 15 mm hot channel spacing and 5 mm cold channel spacing. Simulations were done for the different corrugation angles of 10^0, 20^0, 30^0, 40^0, 50^0, 55^0, 60^0 and 65^0 with fixed channel spacing of 7.5 mm. The channel spacing has already been optimized in our previous work [10]. The optimal channel spacing was found out by analysis of heat transfer to pressure drop ratio.

2. Experimental Method

2.1 Experimental setup and procedure

An experimental setup has been developed to investigate the heat transfer and pressure drop characteristics of the corrugated channels under different flow conditions as shown in Fig- 1. The custom designed experimental setup is manufactured using stainless steel to prevent corrosion during operation. The primary component of the experimental setup is plate heat exchanger (PHE) unit. It is fabricated using 3 stainless steel plates welded in a well specified manner to form two adjacent channels. The spacing provided for the top channel is 15 mm and that for the bottom test fluid channel is 5 mm, length of test section used is 300 mm and thickness of middle thin plate is 1 mm. The experimental setup consists of auxiliary components like cold fluid storage tank, hot fluid storage tank, U-tube manometer, rotameters, and centrifugal pumps. The plate heat exchanger unit, cold fluid tank, hot fluid tank and piping were thermally insulated properly to minimize heat loss. The experimental setup used here is shown in Figure 1. The setup is fitted with a U-tube manometer to measure the pressure changes and a total of 13 PT-100 type temperature sensors have been used to measure the temperature at different locations of test section. One each temperature sensor is fitted at the inlet and outlet of each channel and also at both storage tanks and remaining seven sensors is fitted on the top surface of the heat transfer plate (middle plate) which can be used to measure temperatures at different locations. The inputs from the thermocouples are fed to the digital temperature indicator (DTI) and desired temperature of hot and cold fluid inlet to the PHE has been controlled through temperature controller (on–off control mechanism). The flow pattern adopted here is countercurrent flow. Water is used as both hot fluid as well as cold fluid. Both the fluids are pumped by means of centrifugal pumps to maintain constant flowrates.

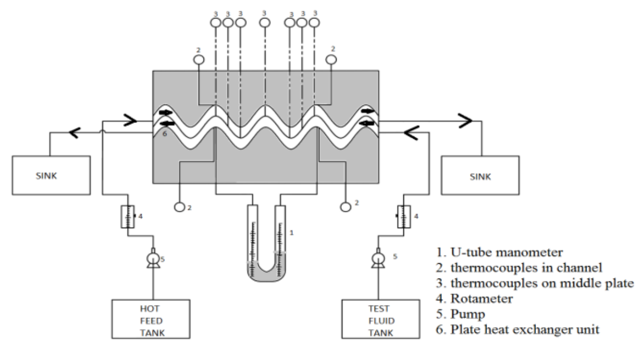

Fig- 1: The experimental setup of corrugated plate heat exchanger.

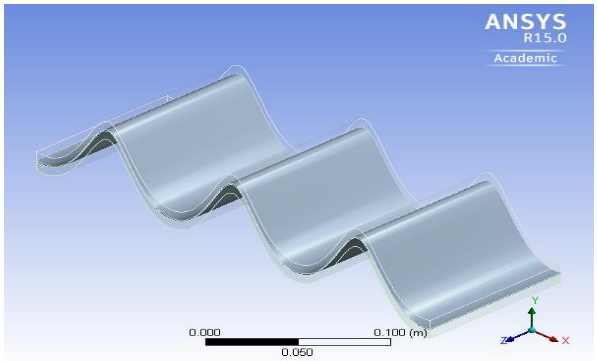

Fig- 2: Geometry created for 30^0 corrugation angle with 7.5 mm spacing of both hot and cold channel.

It is noted that the angle of corrugations have been defined in different manners. Abdulsayid [11] and Nema et al [9] have defined corrugation angle with respect to horizontal whereas, Kanaris et al [12], in their study of chevron plate exchangers have defined angles with vertical. It can said that as the corrugation angle changes from 0^0 to 90^0, the channel pattern changes from simple rectangular duct or flat plate to a square corrugated channel [11]. The corrugation angle (θ) in this study has been measured with respect to the horizontal. The geometry of 30^0 corrugated plate with 7.5 mm channel spacing's of both hot and cold channel is given in Fig- 2.

The inlet flowrate of hot fluid kept constant at 3 lpm with temperature of 60^0C and inlet flowrate of cold fluid was varied from 1 lpm to 6 lpm with constant temperature of 30^0C. A typical time required to achieve steady state was approximately 60 minutes and in order to establish if such steady-state conditions were reached, the temperatures were constantly monitored. Once the thermal equilibrium conditions were obtained, the flow rate and temperature readings were recorded. Each measurement was compared to the obtained average value and estimated errors were deducted.

2.2 Data analysis

The experimental data were used to calculate the heat transfer and pressure drop characteristics. For a corrugated channel (rectangular duct), the term equivalent diameter is considered as hydraulic diameter. The hydraulic diameter (D_h) is a commonly used term when handling flow in noncircular tubes and channels. This hydraulic diameter is defined as four times cross-sectional area (A_{cs}) divided by wetted perimeter (p).

$$D_h = \frac{4A_{cs}}{p} = \frac{4(w \times x)}{2(w+x)} \quad (1)$$

Where, w is the width and x is the channel spacing of corrugated PHE unit. Reynolds number can be calculated based on channel velocity and hydraulic diameter of the channel as follows:

$$Re = \frac{D_h v \rho}{\mu} \quad (2)$$

The heat removed from the hot fluid, Q_h and the heat absorbed by the cold fluid, Q_c are calculated by Eq. (3) and (4) using the measured temperature and mass-flow rate.

$$Q_h = \dot{m}_h C_{p,h}(T_{h,i} - T_{h,o}) \quad (3)$$

$$Q_c = \dot{m}_c C_{p,c}(T_{c,o} - T_{c,i}) \quad (4)$$

The logarithmic mean temperature difference, LMTD is which is given by:

$$\Delta T_{LMTD} = \frac{(T_{h,i}-T_{h,o})-(T_{c,o}-T_{c,i})}{\ln\left(\frac{T_{h,i}-T_{h,o}}{T_{c,o}-T_{c,i}}\right)} \quad (5)$$

The heat transfer coefficient is calculated using following equation.

$$h = \frac{Q}{A \times \Delta T_{LMTD}} \quad (6)$$

3. CFD methodology

3.1 Mathematical model

Any type of fluid flow can be represented mathematically by the equations governing continuity, momentum or Navier-Stokes and energy which have been derived on the principle of conservation of mass, momentum and energy [13]. All the fundamental equations together with a set of appropriate initial and boundary conditions have been solved to study the flow field. For an incompressible and steady state flow through corrugated channels, the mass continuity, momentum and energy equations can respectively be written as:

$$\frac{\partial(\rho.\bar{u}_i)}{\partial x_i} = 0 \quad (7)$$

$$u_j \frac{\partial(\rho \bar{u}_i)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left((\mu + \mu_T).\frac{\partial(\rho.\bar{u}_i)}{\partial x_j}\right) \quad (8)$$

$$C_p.u_j \frac{\partial(\rho T)}{\partial x_j} = \frac{\partial}{\partial x_j}\left((k + k_T).\frac{\partial T}{\partial x_j}\right) \quad (9)$$

The realizable k-ε turbulence model proposed by Bhutta et. al [7] is one of the most popular models because of its simplicity, robustness, reasonable accuracy and fast convergence [14]. However in the present investigation realizable, k-ε turbulence model has been used with the appropriate coefficients. The realizable k-ε model differs from the standard k- ε model in two important ways:

i. The realizable k-ε model contains a new formulation for the turbulent viscosity.
ii. A new transport equation for the dissipation rate (epsilon) has been derived from an exact equation for the transport of the mean-square vorticity fluctuation.

The transport equations for realizable k- ε model are [13]:

$$\frac{\partial}{\partial t}(\rho K) + \frac{\partial}{\partial t}(\rho K v_j) = \frac{\partial}{\partial t}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial K}{\partial t}\right] + G_k + G_b - \rho\epsilon - Y_M + S_k \quad (10)$$

$$\frac{\partial}{\partial t}(\rho\epsilon) + \frac{\partial}{\partial t}(\rho\epsilon v_j) = \frac{\partial}{\partial t}\left[\left(\mu + \frac{\mu_t}{\sigma_\epsilon}\right)\frac{\partial\epsilon}{\partial t}\right] + \rho C_1 - \rho C_2 \frac{\epsilon^2}{k+\sqrt{v\epsilon}} + C_{1\epsilon}\frac{\epsilon}{k}C_{3\epsilon}G_b + S_\epsilon \quad (11)$$

Where,

$$C_1 = \max\left[0.43, \frac{\eta}{\eta+5}\right] \qquad \eta = S\frac{K}{\varepsilon} \qquad S = \sqrt{2S_{ij}S_{ij}} \quad (12)$$

The first transported variable determines the energy in the turbulence and is called turbulent kinetic energy (k). The second transported variable is the turbulent dissipation (ε epsilon) which determines the rate of dissipation of the turbulent kinetic energy. Here, G_b is generation of turbulence kinetic energy due to buoyancy, G_k is generation of turbulence kinetic energy due to the mean velocity gradients, μ_t is turbulent viscosity of fluid, S_k & S_ε is user-defined source terms, S is surface area.

3.2. Solution Methodology

The governing equations have been solved using a commercial CFD package (ANSYS/Fluent 15.0) with the following simplifying assumptions:

i. The PHE operates under steady-state conditions.
ii. Heat transfer surface has been assumed to be free from fouling.
iii. The fluids remain in single phase along the channels.
iv. There is no mal-distribution of flow.
v. The outer walls of the PHE are considered insulated (adiabatic).
vi. Realizable k-ε model with standard wall functions has been used in the CFD simulation.

3.3. Physical model

The plate heat exchanger consists of one solid plate and two fluid bodies (hot fluid stream and cold fluid stream). Top fluid body was considered as hot fluid and bottom fluid body was considered as cold fluid. The geometry of 30^0 corrugated plate with 7.5 mm channel spacing of both fluids is given in Figure 2.

3.4 Mesh Topology

The three zones (1 plate and 2 fluid bodies) initially connected in the design modeler generated hexahedron type of mesh as shown in Fig- 3. The hexahedron type of mesh will have high accuracy solution for same cell amount [14].

Mesh generation was done in ANSYS workbench with options as follows.

i. Physics reference: CFD
ii. Solver reference: Fluent
iii. Relevance: 10
iv. Advanced size function: On: curvature
v. Maximum face size of element: Varied
vi. Maximum element size: Varied
vii. Relevance center: Fine
viii. Smoothening: High
ix. Transition: Fast
x. Span angle center: Fine
xi. Growth rate: 1.80

A mesh convergence analysis was carried out to determine the effect of the mesh elements size. Successive simulations were performed, altering the mesh element density to get a mesh independent solution as depicted in Fig- 4. For an optimum element size of 0.095 mm, number elements are found to be 20,30,188.

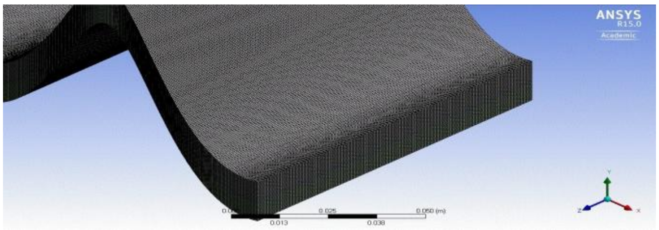

Fig- 3 Magnified view of mesh generated for 30^0 corrugation angle with 7.5 mm spacing of both hot and cold channels.

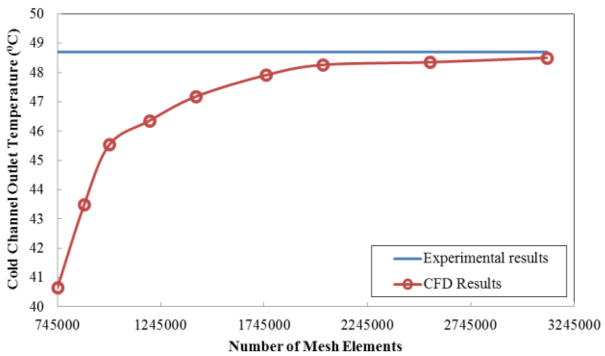

Fig- 4 The outlet temperatures readings for different mesh densities.

3.5 Boundary conditions

Inlet: Velocity specification method was used with absolute reference frame and velocity magnitude normal to the boundary. With 6.0% Turbulence Intensity, hydraulic diameter calculated using Eq. (1) for different channel spacing's. Constant inlet temperature of 30^{0}C and 60^{0}C for cold and hot fluid respectively, was specified in thermal boundary condition.

Outlet: Constant Gauge Pressure was zero Pascal. Backflow direction specification method was used with default settings. Intensity and hydraulic diameter specifications were same as that of the inlet.

Wall: Stationary wall with no slip and coupled thermal condition were used with default settings.

3.6 Solving and post-processing

ANSYS Fluent 15.0 was used as the solver to perform the computation. This code is based on the finite volume method discretization, consisting in the discrete approximation of the volume and surface integrals of the Naviere-Stoke's equations in steady state, applied to each control volume, in whose center, a computational node is present. Pressure based steady solver was applied and energy equations were enabled. Realizable k-ε model with standard wall functions was used with default settings. Both hot and cold fluids were assigned the default properties of water. SIMPLEC was used as pressure-velocity coupling with skewness correction 0.7. The solution was iterated for around 3000-4000 iterations with a relative residual of 10^{-4} for continuity, momentum, k and ε equations and 10^{-6} for energy has been achieved to get a completely converged solution. CFD Post (in ANSYS 15.0 Workbench) was used as post-processing tool for visualization of velocity vectors and temperature contours and also to report the results.

4. Results and discussions

4.1 Validation with experimental results

The realizable k-ε model was first validated using experimental data and then simulations were performed using Fluent in ANSYS Workbench 15.0. Flow in channels is primarily affected by two parameters, the corrugation angle and the channel spacing. As channel spacing changes the rate of heat transfer changes.

Fluent 15.0 was used as the solver to perform the computation. The outlet temperature of the fluid bodies was chosen as the parameter for comparing the CFD results with the experimental data since the outlet temperature is of significant importance in heat exchange operations as the heat transfer is solely dependent on temperature change and also the temperature is the measured parameter. The comparison between the experimental and CFD results of the outlet temperatures as a function of volume flow rate for the hot and cold fluids are illustrated in Fig- 5 and 6 respectively. The comparison of experimental and simulated results of pressure drop is illustrated in Fig- 7. It can be seen that the outlet temperature of hot and cold water decreases when cold channel inlet flow rate of plate heat exchanger increases. The trends in the experimental results are basically the same as those of simulation values. Results also shows that the simulated values of hot channel outlet temperature are more than experimental and this is because of heat loss during experimentation. The maximum difference between the experimental and CFD data for cold channel outlet temperature and hot channel outlet temperature was 0.986% and 1.72% respectively, and maximum difference between experimental and simulated data for pressure drop was 9.8%, which can be qualified as quite satisfactory.

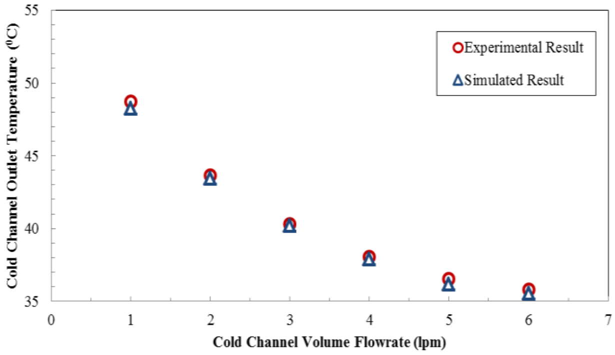

Fig- 5 Comparison of cold water temperatures.

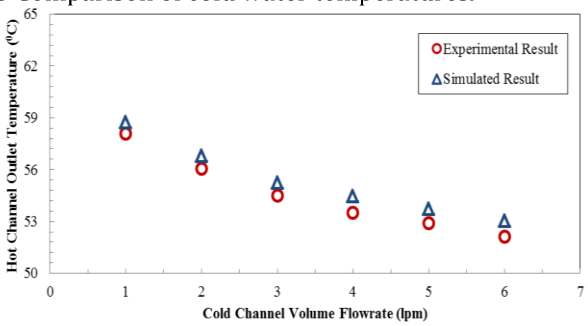

Fig- 6 Comparison of hot water temperatures.

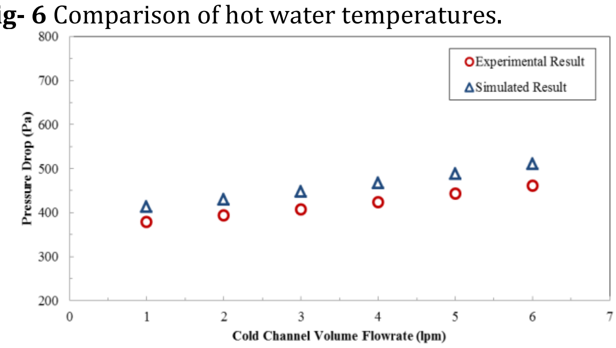

Fig- 7 Comparison of pressure drop.

4.2 Effect of channel spacing

The optimal value (7.5 mm) of channel spacing is used now for both hot channel and cold channel to optimize the corrugation angle. To optimize corrugation angle, heat transfer and pressure drop analysis was done for flat plate and different corrugation angles such as 10^0, 20^0, 30^0, 40^0, 50^0, 55^0, 60^0 and 65^0. Heat transfer coefficient is plotted against Reynolds number for different corrugation angles is shown in Fig. 8. It has been observed from Fig. 8 that for a given Reynolds number heat transfer rate is high for higher value of corrugation angle and increasing order of heat transfer coefficient is from flat to 65^0 corrugation angle. The percent increase in heat transfer coefficient over flat plate was observed lower for lower corrugation angle and higher for higher corrugation angle. Maximum percent increase of heat transfer coefficient over flat plate for 10^0, 20^0, 30^0, 40^0, 50^0, 55^0, 60^0 and 65^0 corrugation angles were observed to 14.09%, 27.50%, 47.33%, 58.74%, 69.42%, 75.69%, 82.19% and 86.93% respectively. The improvement of the heat transfer potential with lower channel spacing makes their use in heat exchangers an attractive option, leading to better system performance and the resulting advantage in energy efficiency.

As the decision for selection of spacing is not only based on heat transfer performance, pressure drop inside the channel is calculated. The variation of pressure drop with the Reynolds number is given in Fig. 9 for different corrugation angles. It was found that the heat transfer performance of PHE is enhanced using higher corrugation angle (Fig. 8), but the Pressure drop increases with increase in corrugation angle and that causes increase in pumping power. Therefore, judicious decision should be taken while selecting the geometry of corrugated plate heat exchangers. Maximum percent increase of pressure drop over flat plate for 10^0, 20^0, 30^0, 40^0, 50^0, 55^0, 60^0 and 65^0 corrugation angles were observed to 1.63%, 3.7%, 9.44%, 18.23%, 38.85%, 54.22%, 71.71% and 88.19% respectively.

Therefore, to have optimal corrugation angles, the heat transfer rate to pressure drop ratio ($Q/\Delta P$) was plotted against Reynolds number shown in Fig. 10. Point at which heat transfer to pressure drop ratio is high will be the optimal point. For low Reynolds number range (<992), 60^0 corrugation angle gives the optimal results and for Reynolds number ranger >992, 50^0 corrugation angle gives optimal results. From the results it was found that 3 lpm is the optimal flowrate at which there will be high heat transfer to pressure drop ratio.

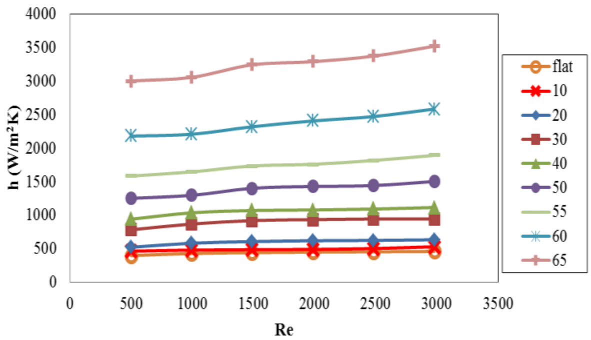

Fig- 8 Comparison of heat transfer coefficient for different corrugation angles.

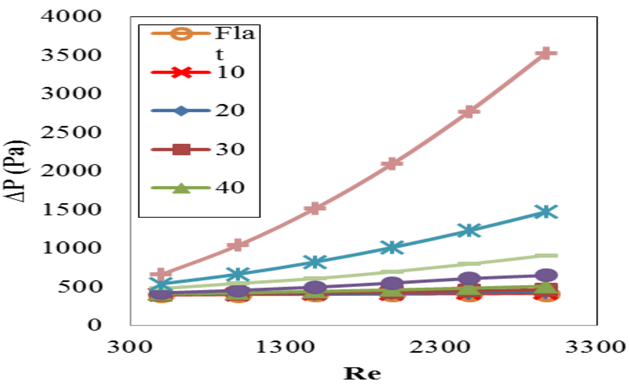

Fig- 9 Comparison of pressure drop for different corrugation angles.

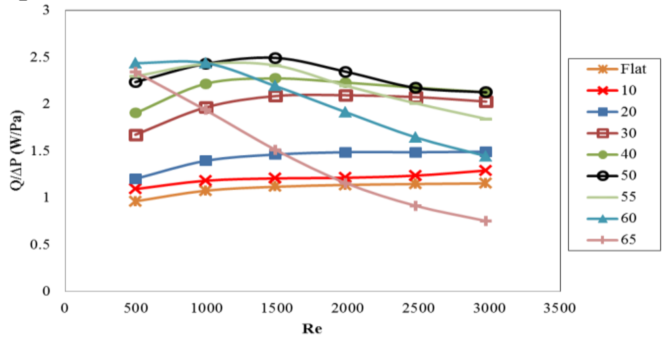

Fig- 10 Comparison of heat transfer rate per pressure drop for various corrugation angles.

5. Conclusions

In this paper, the model validation was done from the experimental results. After validation of model CFD simulations were carried out for different geometries (i.e. different corrugation angles of 10^0, 20^0, 30^0, 40^0, 50^0, 55^0, 60^0 and 65^0) to study the heat transfer performance as well as pressure drop and also to optimize different geometries by calculating heat transfer to pressure drop ratio. The optimal corrugation angle was 50^0 for which the percent increase in heat transfer coefficient and pressure drop were 69.42% and 38.85% respectively over flat plate geometry. For a Reynolds number less than 992, the optimal value of corrugation angle was 60^0 and for Reynolds number greater than 992, the optimal value of corrugation angle was 50^0.

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