Experimental study with analytical validation of thermo-hydraulic performance of plate heat exchanger with different chevron angles

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Abstract— The Plate heat exchanger is prominent among all alternatives to the conventional shell and tube heat exchanger because of simplicity in operation, maintenance, and high surface area density. Currently lot of research is going on to optimize the design by considering thermo-hydraulic performance. In the present study, an extensive analytical model is developed using MATLAB® to study the thermo-hydraulic characteristics of Plate heat exchanger with high beta (60°/60°), low beta (30°/30°) and thermally mixed (30°/60°) plate configurations. Further, the experimental investigation is carried out to validate the analytical model and it reveals that the maximum deviation of ±10% considering both hot and cold fluid outlet temperature with respect to different flow rates in case of all the three plate configurations. To incorporate the effect of rise in pressure drop with different chevron plate configurations, the term thermo-hydraulic effectiveness is introduced. In all cases, it is found to be highest for high beta plates and least for low beta plates. Whereas for thermally mixed plates, the values lies between high and low beta plates being more closer towards high beta plates. This indicates that the thermal performance of thermally mixed plates is nearly match with the high beta plates but the pressure drop being lesser than the latter. A generalized correlation to estimate the Nusselt number has been developed with a relative standard error of ±18%. The Nusselt number predicted from the developed correlation is found to be in good agreement with experimental results compared to the values predicted from different correlation in the literature.

***Keywords*—**Chevron angle, correlation, Thermal mixing, thermo-hydraulic effectiveness.

I. INTRODUCTION

PLATE heat exchangers (PHE) are built of thin plates which are either smooth or have some form of corrugations on the surface. In case of gasketed PHE, number of thin rectangular metal plates sealed around the edges by gaskets and held together in a frame. The major advantages of PHE is its simplicity in cleaning and change of heat duty, less fouling due to high turbulence etc. But the limitation of gasketed PHE is narrow working range of pressure and temperature. These are most commonly used in food processing industries, synthetic rubber industries, paper mills, coolers and closed circuit cooling system for large petrochemical and power plants. Despite of its better performance, continuous research is ongoing to optimize the thermal and hydrodynamic characteristics by means of plate geometry modification, flow distribution, use of nanofluids and thermal mixing.

The heat transfer and pressure drop characteristics of three different plate structures viz. smooth parallel plates, inclined discrete ribs and inclined continuous ribs were compared experimentally in air to air PHE under counter flow conditions by Li et al. [1]. For the flow range of $800 < Re < 1500$, the inclined discrete rib plates showed an enhancement in heat transfer of 110% to 310% and 5% to 15% compared to smooth parallel plates and inclined continuous rib plates respectively. The enhancement in heat transfer in inclined discrete rib plates is attributed to the high degree of turbulence and generation of longitudinal vortices due to zig-zag flow arrangement of fluid in the channel. Whereas the pressure drop for inclined discrete rib plates is less than the inclined continuous rib plates due to reduced blockage effects. Nilpueng and Wongwises [2] have performed a comparison between heat transfer and pressure drop characteristics of PHE ($\beta = 25°$) having different surface roughness (0.936μm, 1.189μm, 1.378μm and 3.312μm) at cold fluid side for $1300 < Re < 3200$ with water as the working fluid. It was observed that the heat transfer coefficient is proportional to the surface roughness with a rise of 4.46%, 8.18% and 17.95% for three higher roughness plates compared to plate roughness of 0.936μm. This is due to the fact that the surface roughness increases turbulence and also causes recirculation at the trough. The corresponding rise in pressure drop was noted as 3.9%, 8.25% and 19.24%. Separate Nu correlations for hot water flowing over smooth surface with a deviation of ±1.61% and cold fluid flowing over rough surface (±1.21%), and a common correlation for f (±0.95%) were developed.

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Visual basic based program was developed to evaluate thermal and hydrodynamic parameters for a given corrugated PHE by Gulben et al. [3]. In this program, the correlations (Tovazhyanski et al. [4], Chisholm and Wannairachchi [5], Talik and Swanson [6], Kumar [7] and, Muley and Manglik [8]) were used to calculate effective area, effective number of plates and total pressure drop (both sides). Later the code was validated using a sample PHE problem from Kakac and Hongtan [9]. Martin [10] developed a separate correlation for *f* and Nu for $300 \leq Re \leq 10000$ by referring the experimental results of Focke et al. [11]. The correlations were developed by considering Re, pressure drop (fRe^2) and β. The results for the range of $0° \leq \beta \leq 80°$ were comparable with the experimental values of Focke et al. [11]. The correlations were also compared with the Leveque equation [12] and a deviation of ± 20% (for *f*) and ±10% (for Nu) was noticed for the same flow conditions and range of β.

Han et al. [13] have performed an experimental and numerical analysis of a corrugated PHE (β = 30°) in counter-current mode in the flow range of 50 lph to 90 lph using water as working fluid. In the CFD analysis, the three dimensional temperature, pressure and velocity fields were obtained for the model built using GAMBIT©. Further, experiments were conducted and a deviation of 6% for water exit temperature and 35% in pressure drop was observed over the flow range as compared to the numerical results which is due to the presence of dead zone where the fluid temperature will be same as inlet temperature and also uneven distribution of flow occurs. Since uniform flow distribution was considered in CFD, considerable deviation in pressure drop is observed. Thus the performance of PHE is significantly influenced by this phenomenon. The heat transfer and pressure drop characteristics of corrugated channels with V-shaped two plates (β = 20°, 40° and 60°) placed in staggered arrangement were studied by Naphon [14] using air as working fluid for the range $2000 < Re < 9000$. The experimental results of V-shaped plates were compared with parallel plate, which showed the highest temperature for parallel plate being 36% greater than plates with β = 60°. A numerical and experimental study of heat transfer and fluid flow in a counter flow corrugated PHE (14 plates with β = 60°) for $400 < Re < 1300$ was studied by Jain et al. [15] with water as working fluid. The CFD analysis was carried under the similar conditions. A correlation was developed for estimating Nu and *f* and the error observed was 13% and 14.5% respectively. The deviation of 18% in Nu and 33% in *f* was noticed in the CFD analysis. The higher deviation in numerical value of *f* and Nu was due to the exclusion of port and flow distribution areas in the analysis.

Experimental performance of mixed chevron plates (β = 30°/60°) PHE was studied by Muley et al. [16] with vegetable oil ($130 < Pr < 230$) as hot fluid and water ($2.4 < Pr < 4.50$) as cold fluid in the range of $2 < Re < 6000$. The test was carried out for 12, 20 and 24 plate packs with U-type counter flow arrangement of fluid. It was observed that the mixed plate configuration enhanced Nu by 1.5 to 5.2 times and *f* by 1.7 to 37 times compared to parallel plate PHE. The mixed plate had 32% more heat transfer area as compared to parallel plate PHE. It was also observed that higher heat transfer and pressure drop was obtained due to the plate surface generated swirl flow and the periodic disruption and starting of boundary layers. From the results obtained, correlation for Nu and *f* were developed which shows a deviation of ± 14% and ± 5% for $2 < Re < 1000$ and ± 4% and ± 7.5% for $Re \geq 1000$.

Experimental study of commercial PHE for symmetric configuration ($30^0/30^0$ and $60^0/60^0$) and mixed configuration ($30^0/60^0$) of three plates (Ø = 1.17) were carried out by Khan et al. [17] for the range $500 < Re < 2500$ and $3.5 < Pr < 6.5$ with U-type flow arrangement. From the results, the Nu was found to increase with β being nearly four to nine times greater than parallel plates. For the particular case of Re = 2000, the Nu was found to be highest for β = 60°/60° being 36%, 57% and 87% greater than 30°/60°, 30°/30° and parallel plates respectively. A correlation for Nu was developed for three different cases and corresponding error obtained were of ±2%, ±1.8% and ±4% respectively. A similar study was conducted by Khan et al. [18] for mixed plate configuration (30°/60°) with water as the working fluid for U-type counterflow arrangement. Heat transfer coefficient was calculated using modified Wilson's plot (a general correlation for convective coefficient where mass flow rate is varied as a function of Re and Pr instead of velocity) and a correlation for Nu was developed based on the same with a deviation of ±10%.

In-detail literature review revealed that the performance analysis is based on plate configuration, flow direction and flow intensity. Further, as an outcome of experimental study, plenty of correlations are made available which is based on particular chevron angle. Hence the present work is focused on development of an analytical model to ascertain the thermo-hydraulic performance of PHE as well as to develop a single correlation to take into account various chevron angle plates.

II. ANALYTICAL ANALYSIS

The detailed analytical analysis to determine the energy parameters of the plate heat exchanger for different input conditions is presented using a MATLAB® program. The schematic of the corrugated plate of PHE is shown in Figure 1. The mean channel spacing is,

$$b = \left(L_c / N_t\right) - t \tag{1}$$

The effective area available for heat transfer of a single plate,

$$A_{ht} = L_p \times L_w \times \phi \tag{2}$$

The total effective area available for heat transfer,

$$A_e = A_{ht} \times \left(N_t - 2\right) \tag{3}$$

The hydraulic diameter of the plates,

$$D_h = \frac{4 \times A_c}{P_w} \tag{4}$$

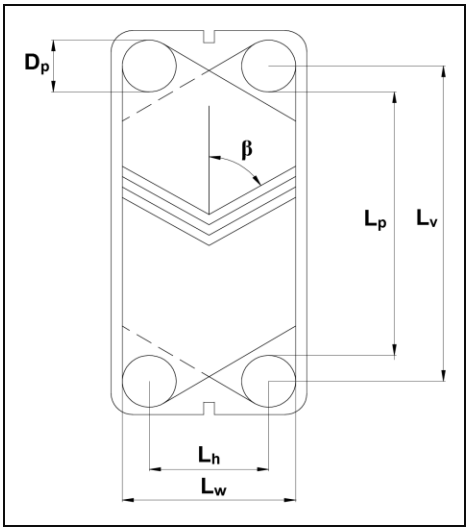


Fig. 1 Schematic of chevron plate

Where, 'N_t' is the total number of plates, 't' is thickness of the plate (m), 'L_c' PHE pack length (m), 'ϕ' is enlargement factor, 'A_c' is flow area (m^2) and 'P_w' is the wetted perimeter (m).

The Nusselt number and friction factor is determined using different correlations (Kakac et al. [9]).

The overall heat transfer coefficient,

$$U = \frac{1}{\frac{1}{h_h} + \frac{1}{h_c} + \frac{t}{k_w}} \tag{5}$$

The thermal effectiveness,

$$\varepsilon = \frac{1 - \exp\left(-NTU \times (1 - C)\right)}{1 - C \times \exp\left(-NTU \times (1 - C)\right)} \tag{6}$$

Where, 'h_h' and 'h_c' are the convective coefficient of hot and cold fluid (W/m^{2}K), 'k_w' is the thermal conductivity of plate material (W/mK) and 'C' is the ratio of heat capacity. The total pressure drop is the sum of channel side and port side pressure drop.

$$\Delta p_c = \frac{4 \times f \times L_v \times G_c^2}{2 \times \rho} \tag{7}$$

The port side pressure drop is:

$$\Delta p_p = \frac{1.4 \times N_p \times G_p^2}{2 \times \rho} \tag{8}$$

By considering total pressure drop, the thermo-hydraulic effectiveness is calculated as:

$$\varepsilon_{th} = \frac{Q - W_p}{Q_{\max}} \tag{9}$$

Where, 'G_c' and 'G_p' are the channel side and port side mass velocity (kg/m^{2}s), 'Q' and '$Q_{\max}$' are the actual and maximum heat transfer (W) and 'W_p' is the pumping power (W). Based on the above mentioned procedure, a code was developed.

The inlet temperatures, mass flow rates and the chevron angle of the plate configuration are provided as inputs to the model. Since, it is an iterative process, appropriate initial assumptions are made for the hot and cold fluid outlet temperatures. The heat transfer coefficient and performance parameters are obtained as outlined in the flow chart (Figure 2).

III. EXPERIMENTAL ANALYSIS

The details of experimental setup, procedure and methodology for the development of correlation and the experimental uncertainty analysis are discussed here.

A. Experimental setup

The experimental setup consists of a plate heat exchanger with 20 corrugated (30°-30°/30°-60°/60°-60°) stainless steel plates. Water is used as the working fluid and the flow pattern is a single pass U type counter flow. The schematic diagram of the experimental setup is as shown in Figure 3. The different sensors used are pre-calibrated. The inlet and outlet temperatures of the hot and cold fluid are measured using four K-type thermocouples positioned at the inlet and outlet ports of hot and cold fluid. The hot water obtained by heater is

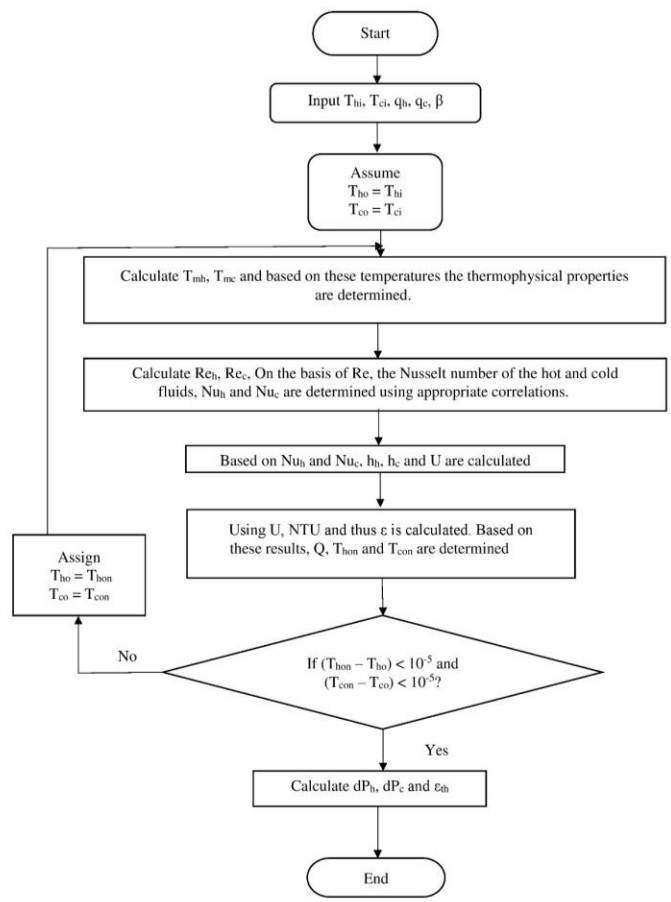


Fig. 2 Flowchart of analytical model

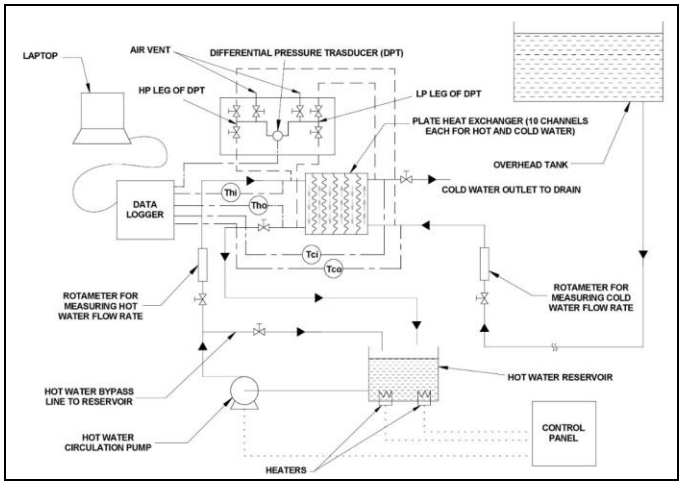


Fig. 3 Schematic of plate heat exchanger test setup

recirculated in the loop by pump and the cold water flow is once through basis which is supplied from overhead tank. The pressure drop across the hot and cold fluid is measured using a differential pressure transducer. The flow rates of both the fluids are measured using two separate rotameters. The temperature and pressure drop readings are monitored continuously using a real time data logging system.

B. Experimental procedure

The experimental procedure for determining thermal and hydrodynamic parameters is explained below:

1) The cold water supply is started and once the hot water in the tank reaches required temperature, it is circulated using pump. Both the flow rates are adjusted to required value.
2) The data logger is started as soon as the hot and cold water flow is initialized. The temperature and pressure drop variations are observed in the transient graph.
3) Once the steady state is reached, both pressure and temperature reading are captured.
4) Above set of procedure is repeated 4 times to check the repeatability in case of different hot and fluid flow combinations, varied hot water inlet temperature and chevron angle.

The different plate configurations used in the experimental analysis are shown in Figure 4. The error involved in measurement of different parameters are given in Table 1.

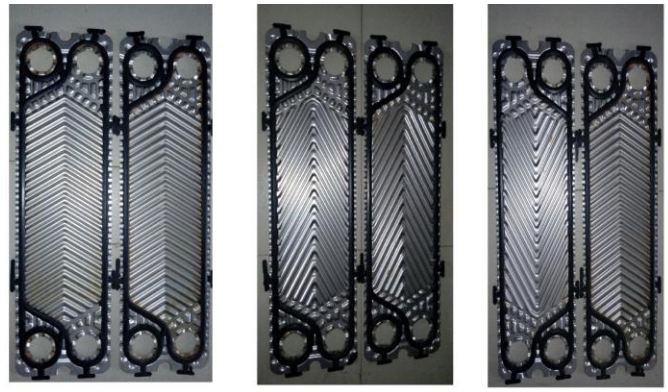
C. Correlation development

The terms involved in the Nu correlation are Reynolds number, Prandtl number and sine of chevron angle of the plate combination. The general form of the correlation is:

$$Nu = C \times \mathrm{Re}^{m} \times \mathrm{Pr}^{n} \times \left(\sin\theta\right)^{q} \tag{10}$$

A total of 169 data points are used and the analysis is done in MS EXCEL®. The final form of correlation is:

$$Nu = 0.05\,\mathrm{Re}^{0.52}\,\mathrm{Pr}^{1.99}\left(\sin\theta\right)^{0.21} \tag{11}$$



(a) high beta (b) low beta (c) thermal mixing

Fig. 4 Different Plate configurations used for experiments

TABLE I
PROBABLE ERRORS IN INSTRUMENTS

Instrument	Error
thermocouple	± 0.01 ^{0}C
flow meter	±0.01 lpm
differential pressure transducer	±0.001 Pa

IV. RESULTS AND DISCUSSION

The validation of analytical model by considering different plate configurations (30°-30°/30°-60°/60°-60°) and the goodness of fit of the developed correlation have been discussed in this section.

A. Validation of analytical model

The experimental outlet temperatures of the hot and cold water are compared with the output of the analytical model as a part of validation. For each cold water flow rate (5, 10, 15 and 20 lpm), the hot water flow rate is varied from 5 to 35 lpm in steps of 5 lpm. The plate configuration used in this study are high beta ($\beta = 60^0/60^0$), low beta ($\beta = 30^0/30^0$) and mixed plate configuration ($\beta = 30^0/60^0$) respectively. The experiment is carried out under variable conditions of hot and cold water inlet temperatures.

Figure 5 shows the variation of experimental and analytical outlet temperature for the high beta plate configuration with respect to the variable hot and cold water flow rate. The maximum deviation noticed is 10% in case of cold water flow rate of 10 lpm/hot water flow rate of 5 lpm. The minimum deviation is 0.1% for cold water flow rate of 5 lpm/hot water flow rate of 35 lpm. In the present experiment, the range of Reynolds number is 198<Re<2194. However, the correlations available are based on a wide range of Re. Hence deviation in analytical values is observed at multiple points. For the low beta plate configuration, the maximum and minimum deviation noticed are 5% and 0.8% respectively as shown in Figure 6. The random variation in the analytical and experimental values is due to the variable input temperatures. Thermal performance comparable with high beta plates but lower pressure drop can be achieved by thermal mixing. This is clearly evident from Figure 7.

Similarly, the analytical and experimental values of cold water outlet temperatures for different configurations are found to be in well agreement with ±10% deviation as depicted in Figure 8, Figure 9 and Figure 10.

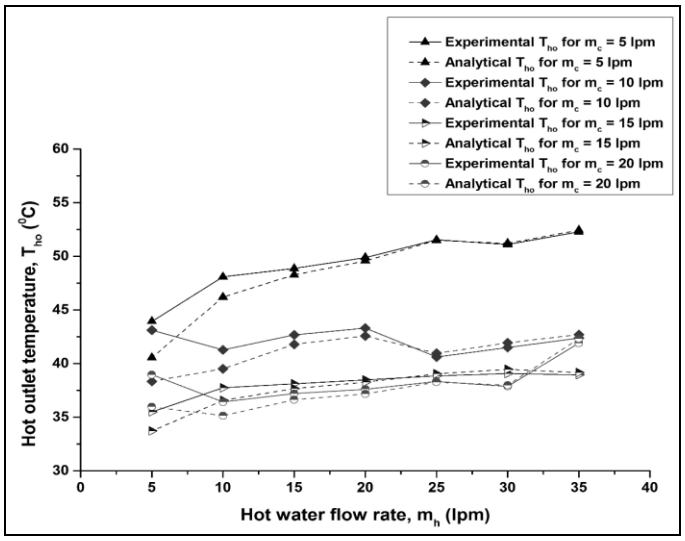


Fig. 5 Hot water outlet temperature for high beta configuration

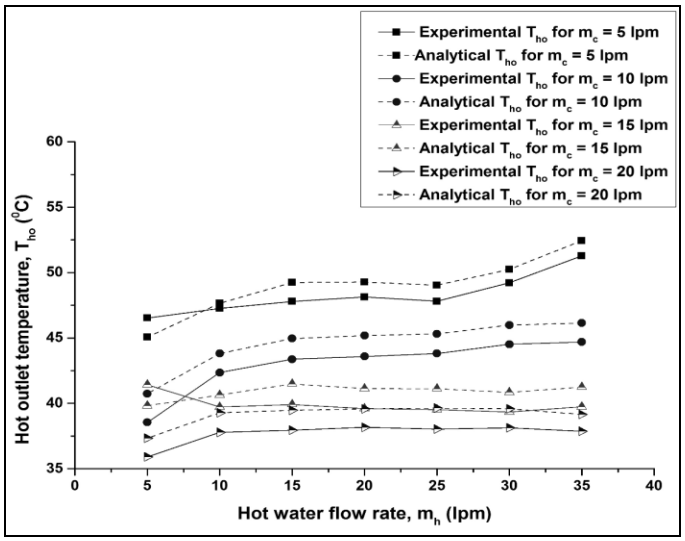


Fig. 6 Hot water outlet temperature for low beta configuration

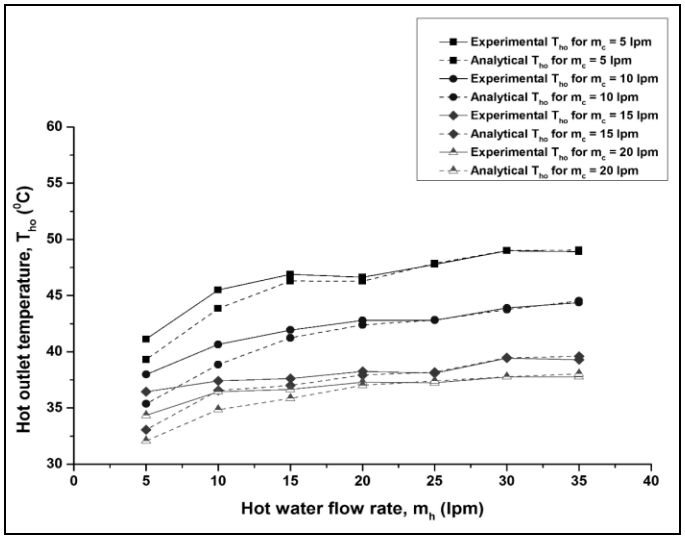


Fig. 7 Hot water outlet temperature for mixed plate configuration

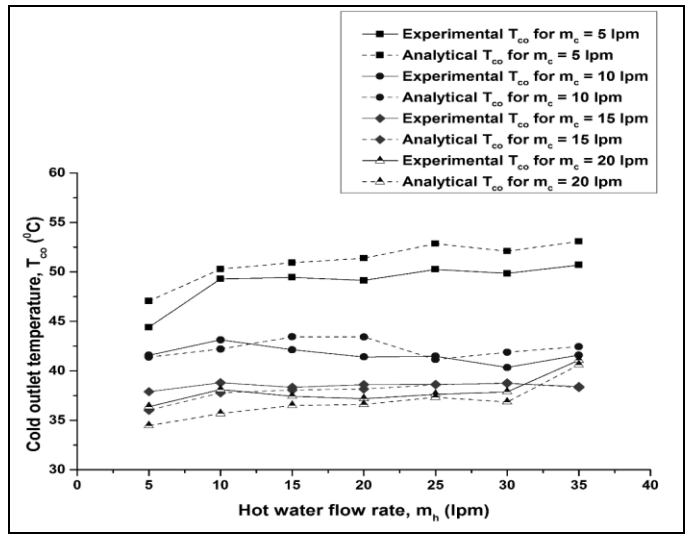


Fig. 8 Cold water outlet temperature for high beta configuration

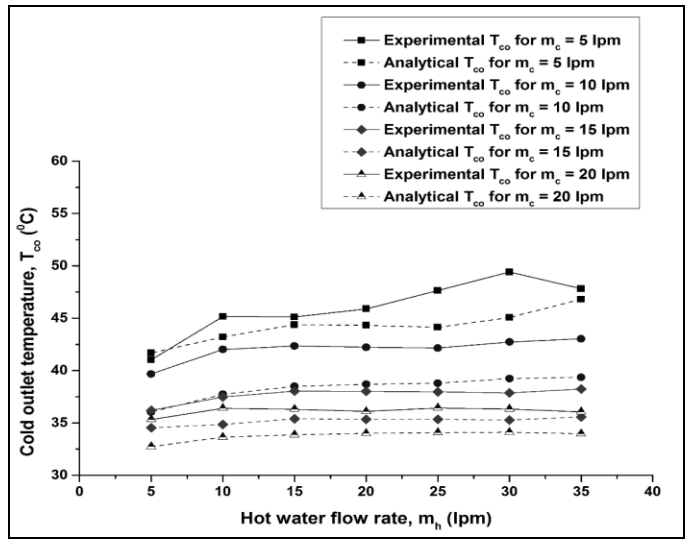


Fig. 9 Cold water outlet temperature for low beta configuration

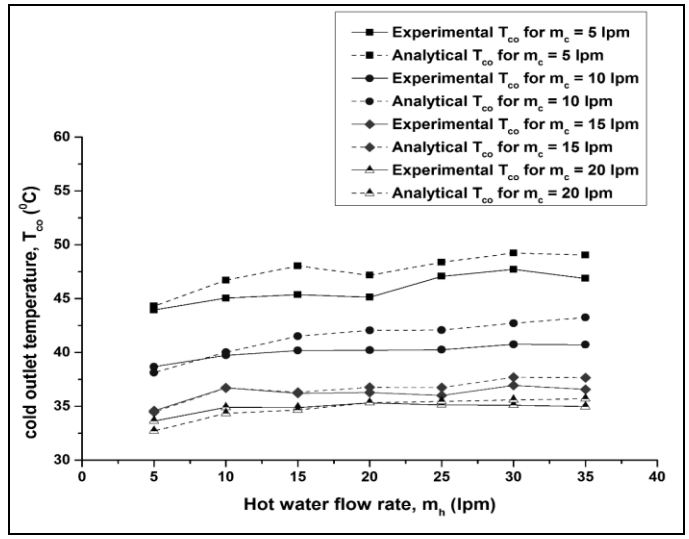


Fig. 10 Cold water outlet temperature for mixed plate configuration

Figure 11 shows the variation in analytical thermo-hydraulic effectiveness for the cold water flow rate of 5 lpm corresponding to the hot water flow rate from 5 lpm to 35 lpm. As discussed earlier, the thermo-hydraulic characteristics of thermal mixed plates lies near that of high beta plates. It is also observed that the condition of equal hot and cold heat capacity rate, thermo-hydraulic effectiveness is minimum because of LMTD. The trend of thermo-hydraulic effectiveness is also studied for the cold water flow rate of 10 lpm, 15 lpm and 20 lpm considering the same conditions of hot water flow rate. As shown in the figure the highest value of thermo-hydraulic effectiveness is observed in case of high beta, followed by thermally mixed and lowest in case of low beta plate configuration. Subsequently the dip in the value of effectiveness is observed in all cases for the condition of equal heat capacity rates.

B. Validation of developed correlation

In the previous discussion, three sets of correlations based on chevron angle and for each configuration, based on Re value, constants of correlations are changed. Instead of using multiple correlations and constants, a single correlation which includes chevron angle as an additional variable for determining Nusselt number is developed. The validation of experimentally determined Nusselt number vs Nusselt number of present correlation is depicted in parity plot (Figure 12). It is observed that the maximum number of points lies within the standard relative error (±18%). For the developed correlation, the coefficient of correlation was found to be 0.8124.

With respect to experimental results, maximum deviations of analytical and present work correlations should be compared. Figure 13, Figure 14 and Figure 15 shows the variation of results of experimental, analytical and developed correlation for different flow rates and different plate configuration. The trend of the developed correlation follows the trend of the experimental results while the trend of the analytical model shows deviation at low flow rates. Hence it can be concluded that the developed correlation has good agreement with the experimental data.

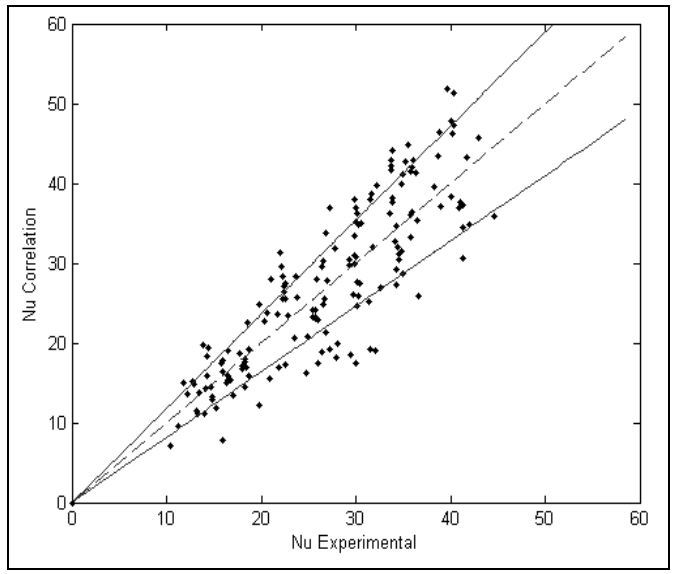


Fig. 12 Parity Plot

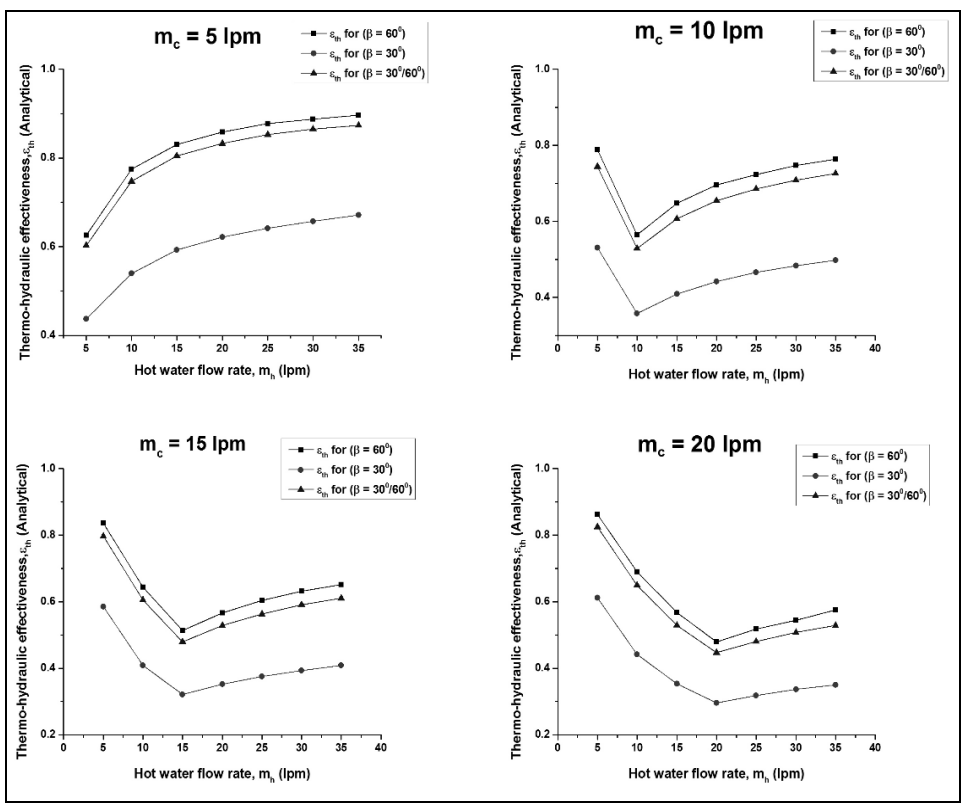


Fig. 11 Thermo-hydraulic effectiveness at constant cold water flow rates

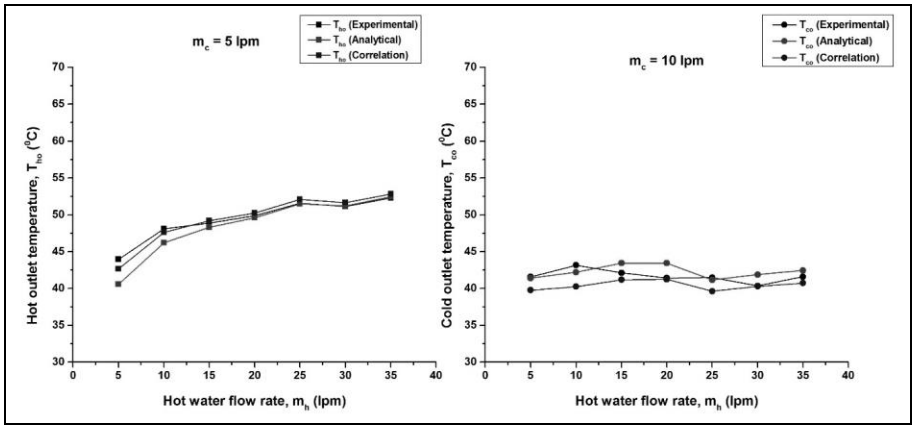


Fig. 13 Outlet temperature comparison for high beta configuration

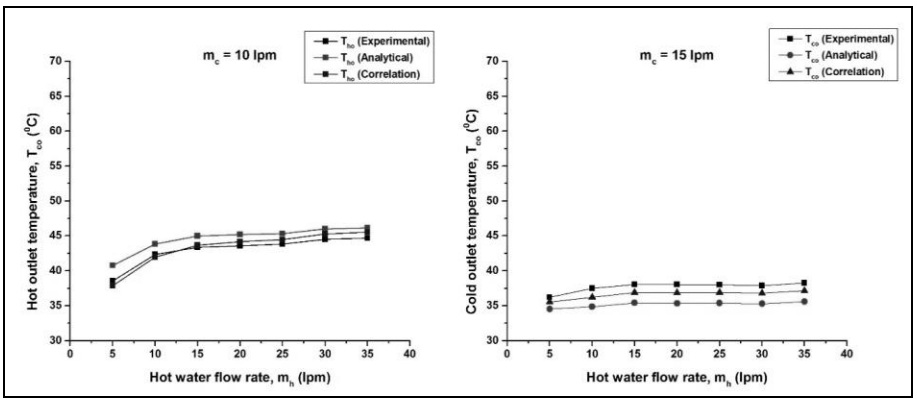


Fig. 14 Outlet temperature comparison for low beta configuration

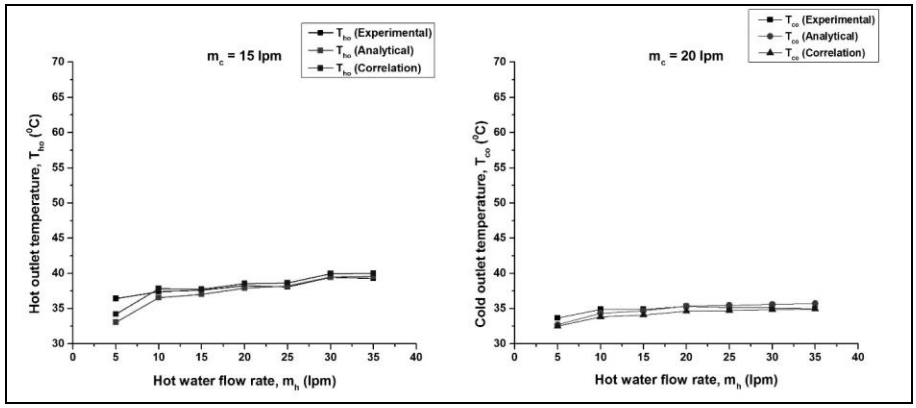


Fig. 15 Outlet temperature comparison for mixed plate configuration

V. CONCLUSION

Based on the intensive study, the following conclusions have been derived:

1. Analytical model is found to be in well agreement with the experimental results with a maximum and minimum deviation of ±10% and ±1% respectively in temperature.
2. The thermo-hydraulic effectiveness initially decreases with increase in flow rate and attains least value when the hot and cold water flow rates are the same. This is because at this point the value of LMTD is least. Further it increases with flow rate.
3. A generalized correlation for Nusselt number has been developed with a standard relative error of ±18%.

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