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Effect of Chevron Angles on the Thermal Performance of Corrugated Plate Heat Exchanger (CPHEs)

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Abstract:

Corrugated plate heat exchangers (CPHEs) possess a greater surface area for heat transfer and a heightened level of turbulence as a result of the corrugations. This study presents an experimental analysis comparing the thermal performance of a flat plate heat exchanger and a corrugated plate heat exchanger (CPHE) with varied corrugation angles. The analysis was conducted using water as the test fluid. The comprehensive testing has been carried out on CPHEs with different chevron angles (β) of 30°/30°, 45°/45°, and 60°/60°. Experimental data is collected under steady-state conditions, with a single phase (water-water) system, covering Reynolds numbers (Re) ranging from 900 to 1250. The test fluid's mass flow rate, ranging from 0.05 to 0.16 kilograms per second, is measured together with the accompanying steady state temperatures. Each plate was equipped with four thermocouple sensors to detect the bulk temperature of the cold and hot fluids at the intake and output. The performance was evaluated using both parallel and counterflow configurations. Experimental observations are utilized to estimate the temperature difference (ΔT) between the inlet and output streams, the logarithmic mean temperature difference (LMTD), and the exergy (E). The measured ΔT and E values for corrugation angles (30°, 45°, and 60°) of CPHE were compared to those of flat plate heat exchangers. For corrugation angles of 30°, 45°, and 60°, the values of ΔT and E (efficiency) increase as the mass flow rate of the fluid increases. An increase in the corrugation angle leads to an increase in turbulence in the flow, which in turn enhances heat transmission. In addition, the thermal effectiveness (ε) was calculated using the NTU method and compared for all of the plates.

Key Words: Corrugated plate heat exchanger, Thermal performance, Chevron angle, heat transfer coefficient.

1. Introduction

Heating and cooling devices have been widely adopted in industrial and resident applications. And these applications bring huge amount of energy consumption. What's more, huge low-grade heat is rejected to the surroundings as waste, which results in huge waste of energy and leads to environmental destruction. Absorption system (like heat exchanger) has been proved to be a highly efficient heat recovery technology. Besides, absorption systems can benefit the environment by reducing the emission of global warming gas such as carbon dioxide. A heat exchanger is a device that is used to transfer thermal energy (enthalpy) between two or more fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperatures and in thermal contact. In heat exchangers, there are usually no external heat and work interaction. The CPHE is the most efficient comparing to other conventional type heat exchangers. The efficiency of the heat exchanger increases in fluid contact surfaces, pressure drop and mass flow rates due to enhanced heat transfer to the fluid [1]. Many researches have studied different approaches to enhance flow mixing and heat transfer by introducing passive techniques to control the energy dissipation rate and enhance turbulence intensity [2-4]. Corrugated Plate Heat Exchanger (CPHE) are compact, provides high thermal effectiveness and close approach temperatures (up to 2°C difference) can obtained [5]. CPHEs are lighter and require less space. So, they have been employed in food, power, HVAC and many other applications [6]. Corrugated plate heat exchanger belongs to such category because it can retrieve effective heat at small temperature due to high turbulence created at low velocities [7, 8]. In CPHEs, the turbulent flow can be achieved at low Reynolds Number (Re), Re > 400 [9].

Many experimental [10-21] shows that the rate of heat transfer enhance with using of corrugated plate heat exchanger (CPHE). The experimental investigation on CPHE for single phase flow of Reynolds number ranging from 500 to 2500 had shown an enhancement in heat transfer coefficient and they derived correlations between Nussels number and Reynolds number [10]. In CPHE with parallel flow using water as fluid at small temperature difference by varying the space between the plates and observed that the optimal heat transfer possible at a minimum spacing of 6 mm [11]. The corrugated plate of corrugation angle increases (30°, 40° and 50°), heat transfer rate also increases because it offers higher surface area of contact as well as turbulence in the flow [12-14]. Furthermore, as the angle of corrugations is reaches to 50°, creates more turbulence in the flow regimes which ultimately improves heat transfer rate also severely pressure drop [8, 15-17]. Moreover, the plate design of CPHE, additionally improves the mechanical strength to heat exchanger [18].

Okada et al. [22], investigated the impact of different β on CPHE's thermal performance, where β was considered with horizontal line, but it was common β to considered with respect to vertical line. Later Muley and Manglik [23] studied heat transfer characteristics (HTC) in CPHE for β =30°/30°, 60°/60° and 30°/60°. Never the less each correlation to be separated, for the whole study were incorporated in one formula. Khan et. al. [24] carried out a one-one pass, water-water fluids for the same β's, and found out that, the same Re on the both sides of the CPHE does not imply heat transfer co-efficient (h) will be the same, as h depends on many other factors such as fluid viscosity, fluid density, fluid velocity and many other parameters. The experimental and numerical studies have concluded that the flow inside the CPHE is non-uniform and tends to flow towards the lateral edges of the plate but does not consider the thermal performance [25]. A two symmetric β = 30°/30° and 60°/60° CPHEs were numerically studied using CFD by Asif et al. [26].

While most studies focus on CPHE are limited to either air or water as the test fluid [11, 27-29]. This paper focuses to optimize the efficacy of CPHE using different chevron angles. Hence, the study investigates the heat transfer enhancement in the corrugated plate heat exchanger and thermal effectiveness of corrugated plate heat exchanger for different corrugated angles. Moreover, these values are compared with flat plate heat exchanger.

2. Methodology

2.1 Experimental Setup and Procedure

The photograph of the experimental setup, fabricated with 22 gauge GI sheets to investigate the heat transfer characteristics of the plate heat exchanger channels for same flow conditions with different inlet hot water temperatures are shown in (Fig. 1 and Fig. 2). It includes a hot water loop, two coolant loop and a measurement system. The hot water loop comprises a water tank, a heater, and a water pump. The cold water loop comprises a water tank, and a water pump. A digital temperature indicator with thermocouples is used to measure temperatures at inlet and exit of the hot and cold streams. The flow rate is measured by noting down time for collection of fixed volume of the fluid. The whole system is thermally insulated to minimize the energy loss. Specification details are as follows: Length of the test section =100 cm Width of the test section = 10 cm Height of a flow channel, i.e. gap between two successive corrugated plates = 5 cm. Chevron angle of the plate = 30°, 45° and 60°.
Hot water was made to flow through the central corrugated channel to maintain the channel surfaces at approximately constant temperature. Cold water is made to flow in the upper and lower channels. Thermocouples were first calibrated using thermometer and then inserted in the inlet and exit of the hot and cold streams, were used to record the

corresponding fluid temperatures. Experiments were conducted for 40, 45, 50, 55, 60 °C inlet temperature of hot water in parallel and counter flow arrangement. The hot and cold water flow rate is maintained constant for all inlet hot water temperatures and for both parallel and counter flow arrangements. Mass flow rate of hot fluid (water) is maintained at 0.05 kg/s and that of the cold fluid (water) is 0.16 kg/s.

Figure 1 Corrugated plates of Heat Exchanger

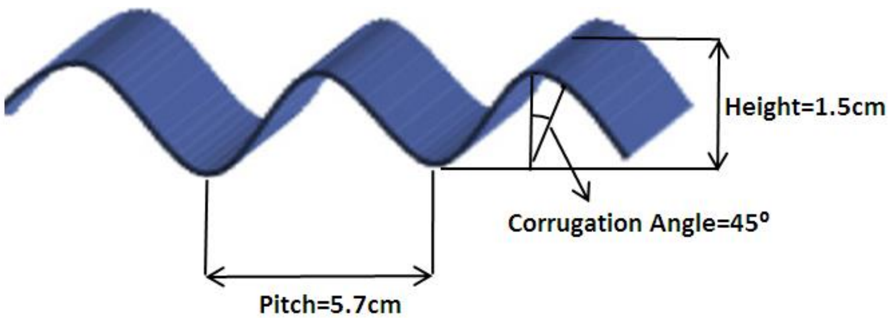

Figure 2 Typical geometry of Plate Segment

2.2 Numerical methods:

2.2.1 Heat Transfer Coefficient

The heat transfer coefficient or film coefficient, in thermodynamics and in mechanics is the proportionality coefficient between the heat flux and the thermodynamic driving force for the flow of heat (i.e., the temperature difference, ΔT):

$h = (q/\Delta T)$

where

q : heat flux, W/m^2 i.e., thermal power per unit area, $q = dQ/dA$

h : heat transfer coefficient, W/(m^{2}K)

ΔT : difference in temperature between the solid surface and surrounding fluid area, K

2.2.2 Overall heat transfer coefficient (U)

In general case we would like to have the heat transfer given by a simple relation of the form Q = U.A.LMTD

In the recuperative type heat exchanger, the hot and cold fluid a are separated by a solid wall, like the cylindrical wall as in the case of tubes in shell and tube heat exchangers.

, where is applied to composite walls and pipes.

The heat flow, Q, through the wall of the pipe of length L, is given by

$$\frac{Q}{L}=2\pi h_i r_i (T_i - T_1) = -2\pi rk\frac{dT}{dr} = 2\pi h_0 r_0 (T_2 - T_0)$$

From equating the first and third terms

$$dT = -\left(\frac{Q}{2\pi kL}\right)\frac{dr}{r}$$

Which gives, after integration,

$$T_1 - T_2 \frac{Q}{2\pi kL} In(r_0 / r_i)$$

From the first and second terms

$$T_1 - T_1 \frac{Q}{2\pi Lh_i r_i}$$

Likewise, from the first and last terms

$$T_2 - T_0 = \frac{Q}{2\pi Lh_0 r_0}$$

$$T_1 - T_0 = \frac{Q}{2\pi L}\left\{\frac{In(r0/ri)}{K} + \frac{1}{h_i r_i} + \frac{1}{h_0 r_0}\right\}$$

And rearranging

$$\frac{Q}{2\pi Lr_0} = \frac{T_i - T_0}{\frac{r_0}{k} In\left(\frac{r_0}{r1}\right) + \frac{r_0}{h_1 r_1} + \frac{1}{h_0}}$$

$$= U_0 (T_i - T_0)$$

Where by dividing by the area of the outer surface, the overall heat transfer coefficient U_0 is said to be referred to the outer surface and is given by

$$U_o = \left\{\left[\frac{r_o}{k} In\left(\frac{r_o}{r_i}\right)\right] + \left[\frac{1}{h_i}\right] + \left[\frac{ri}{r_o h_o}\right]\right\}^{-1}$$

It can be useful to think of the analogy and Ohm's law (I=V/R). In the context of heat exchange by convection and conduction, the temperature difference (Ti-T0) provides the potential to 'drive' a current, in this case the heat flux ($Q/2\pi Lr_0$), and overcome the thermal resistance, $1/U_0$.

Similarly, the overall heat transfer coefficient U_i referred to the inside surface is

$$U_i=\left\{\left[\frac{r_0}{k} In\left(\frac{r_0}{r_i}\right)\right]+\left[\frac{1}{h_i}\right]+\left[\frac{ri}{r_0 h_o}\right]\right\}^{-1}$$

Some typical values of overall heat transfer coefficient they are presented as guidelines only; it is always preferable to calculate values using standard convention correlations for plain tubes and other passageways or the data of Kays and London (1984), for more geometrically complex designs.

Very often $r_0/r_i \approx 1$ and the tubes are made from a good conductor whose thermal resistance can be neglected,

$$= \frac{h_0 h_1}{h_0 + h_i} = U$$

An important practical implication that if hi >> ho, then $U_0 \approx h_0$, or if $h_0 >>$ hi, then $U_0 \approx$ hi, which may be summarized as the heat transfer will be controlled by the lower of the two heat transfer coefficients. To illustrate this, consider a water-to-air heat exchanger, where the heat transfer coefficient on the water side is 1000 W/m^2K and that on the air side $40W/m^2K$. Neglecting the thermal resistance of the tube walls, the overall heat transfer coefficient will be

$$U = \frac{h_0 h_1}{h_0 + h_i}$$

$$= \frac{4x10^4}{1040} = 38.46 W / m^2 K$$

Doubling the water-side heat transfer coefficient will increase the value of the overall heat transfer coefficient to an increase of 2%. Doubling the air-side hat transfer coefficient (lower value and hence controlling) will lead to an overall heat transfer coefficient of

$$U = \frac{h_0 h_i}{h_0 + h_i} = \frac{8x10^4}{1080} = 74.07 W / m^2 K$$

an increase of 93%. Clearly, if the overall heat transfer coefficient needs to be increased, then it is more worthwhile to focus attention on the side with the lower value of heat transfer coefficient.

In addition, the thermal resistance will be increased owing to the action of deposits formed from salts, solid particles, chemical reactions, corrosion and biological organisms. The general term for this action is fouling and, in practice, fouling is handled by an extra resistance term known as the fouling factor or fouling resistance (Rf, I for the inside, Rf, o for the outside).

If such fouling resistances are included,for the overall heat transfer coefficient become.

$$U_0=\left\{\left[\frac{r_0}{k} In\left(\frac{r_0}{r_i}\right)\right]+\left[\frac{r_0}{r_i} R_{f,i}\right]+\left[\frac{r_0}{h_1 r_i}\right]+\left[\frac{1}{h_0}\right]+R_f.o\right\}^{-1}$$

$$U_0=\left\{\left[\frac{r_i}{k} In\left(\frac{r_0}{r_i}\right)\right]+\left[R_{f\cdot i}\right]+\left[\frac{1}{h_1}\right]+\left[\frac{r_i}{r_0 h_0}\right]+\left[\frac{r_i}{r_0} R_f.o\right]\right\}^{-1}$$

The above analysis has been developed for circular tubes. For a heat exchanger where the channels are constructed from plates of thickness t and thermal conductivity, k, with the outside and inside areas being equal, it is relatively easy to show that the overall heat transfer coefficient is given by

$$U=\left\{\left[\left(\frac{t}{k}\right)\right]+R_{f,i}+\left[\frac{1}{h_1}\right]+\left[\frac{1}{h_o}\right]+R_{f,o}\right\}^{-1}$$

The overall heat transfer coefficient is related to the total thermal resistance of the system as follows :Where the suffix 1, 2 referred to the inner and outer surface area.

$$U_1A_1=U_2A_2=\frac{1}{\sum Rth}$$

$$U_2A=\frac{1}{\frac{1}{h_1A_1}+\frac{\ln(r_2/r_1)}{2\pi kL}+\frac{1}{h_2A_2}}$$

2.2.3 Effectiveness of heat exchanger

When the area of a heat exchanger is known and the outlet temperatures of both streams are to be determined, an iterative calculation is required, but by using the heat exchanger effectiveness the iterative procedure can be avoided. The effectiveness is defined as the ratio of he actual rate of heat transfer in a given exchanger to the maximum possible rate of heat transfer. The maximum possible heat transfer occurs in a counter-flow exchanger having infinite heat-transfer area. When one of the fluid Streams will gain or loss heat until its outlet temperature equals the inlet temperature of the other stream.

The fluid that experiences the maximum temperature change is the one having the smaller value of $C = mc_p$

By definition effectiveness is :

$$\varepsilon=\frac{Q}{Q_{\max}}=\frac{Q}{C_{\min}\,\Delta T_{\max}}$$

It can be seen that the effectiveness for a given type of heat exchanger depends on only two parameter r and NTU, where :

$$r=\frac{C_{\min}}{C_{\max}}$$

NTU = UA/C_{min}

Here, NTU is number of transfer units.

3. Results and Discussions:

The experimental analysis is to be done in this present work. Take the reading from the experiment, and to be measured the different mass flow rate of cold side as well as hot side and also measured the temperature of hot side as well as cold side. Considering the temperature of fluid, T_{hi} = 50°C, Outlet temperature of fluid, T_{ho} = 46°C and inlet temperature of water T_{ci} = 31°C, outlet temperature of water T_{co} = 32°C is kept constant. In this study, the mass flow rate adopted for hot fluid is 0.05 kg/sec and for cold fluid is 0.16 kg/sec. Reynolds number for counter and parallel flow arrangements was adopted as 1100.

3.1 Heat Transfer Coefficient (h):

Figure 3 shows a decreasing trend in heat transfer coefficient plot with the increase in the chevron angle. The heat transfer co-efficient of hot side kept on decreasing by keeping the mass flow rate of water constant. Comparing the tested 60° chevron plate by changing the chevron angle 45° and 30°, the result shows tested 30° chevron plate better heat transfer co-efficient.

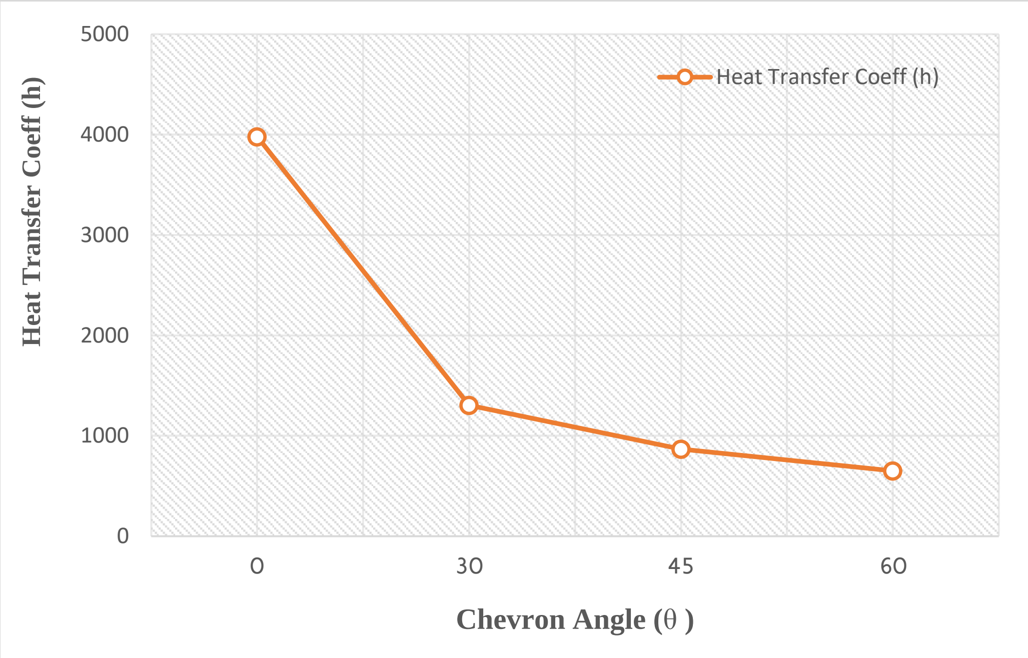

Fig. 3: Effect of Chevron angle on Heat transfer coefficient of CPHEs for counter and parallel flow arrangements

3.2 Effectiveness (ε):

Figure 4 indicates a trend behavior of effectiveness with respect to varying chevron angle for parallel as well as counter flow arrangements system. In general, the plot for effectiveness has a decreasing trend with the increase in corrugated/chevron angle, and indicates a better performance result at 30° corrugation in plate. The performance of CPHEs in terms of effectiveness reveals appreciable outcome when the heat exchanger system follows counter flow arrangements.

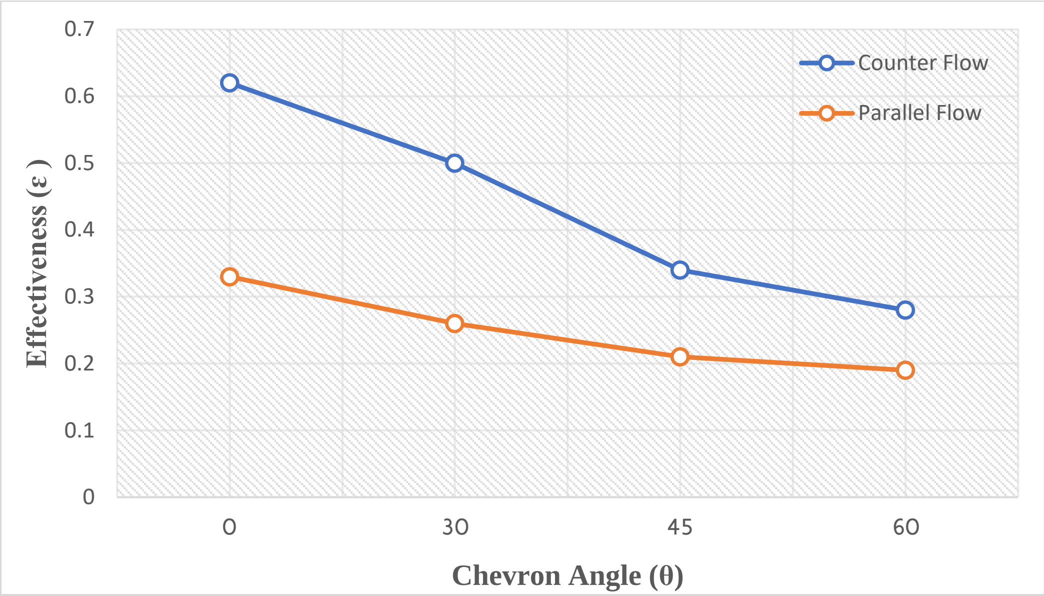

Fig. 4: Impact of Chevron angle on effectiveness of CPHEs for counter and parallel flow arrangements

4. Conclusion

Experiments have been performed to investigate characteristics of chevron type heat exchanger with different chevron angles. The experimental set-up designed and constructed to determine the characteristics of gasket plate heat exchanger with chevron plates. The thermal performance of corrugated plate heat exchanger was measured in terms of heat transfer coefficient and effectiveness of CPHEs.

I. The heat transfer coefficient decreases with the increase in the chevron angle of plate irrespective of flow arrangements.
II. The effectiveness of plate exchanger also shows a decreasing trend with the increase in the chevron angle. However, a marginal change (increasing) in the effectiveness is observed in case of counter flow arrangements.

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