Experimental Thermal and Fluid Science 75 (2016) 173–182

Contents lists available at ScienceDirect

Experimental Thermal and Fluid Science

journal homepage: www.elsevier.com/locate/etfs

Experimental studies on parallel wavy channel heat exchangers with varying channel inclination angles

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ARTICLE INFO

Article history:
Received 9 June 2015
Received in revised form 12 February 2016
Accepted 14 February 2016
Available online 18 February 2016

Keywords:
Heat transfer
Wavy channel heat exchanger
Channel inclination angle
Heat transfer correlations

ABSTRACT

Experimental studies were performed on corrugated plate heat exchangers. The corrugated plates' channel inclination angles for the parallel plates were varied from 0° to 80° in increments of 20° in order to experimentally determine the configuration with the optimum heat transfer rate. The channel spacing was kept constant. The media used in both the hot and the cold channels were water. The temperatures of the two media between which heat was exchanged were 25.4 °C for cold water and 38 °C for hot water. The flow in the heat exchangers was parallel with both the fluids entering from the same side. The flow rate of the hot water was kept constant while that of the cold water was varied. The heat exchangers consisted of ten channels, five for hot water and five for cold water. It was found that the rate of heat transfer increases for all the plate inclination angles as the flow rate of cold water is increased. The overall heat transfer coefficient was observed to be the highest for the 20° inclination angle. The low flow rates of the hot and cold fluids make it a unique study, as the two fluids get more time for heat transfer. The 20° channel inclination angle also has the highest average thermal length. Based on the results, correlations between Nusselt number and Reynolds number were developed.

1. Introduction

Heat exchangers are devices which are used to transfer heat between two media. A plate heat exchanger, which is an indirect heat exchanger, consists of a stack of metal plates with inlet and outlet ports and seals to direct the flow in alternate channels, which are formed by adjacent plates [1]. Plate heat exchangers have numerous applications in industries such as petrochemical, pharmaceutical, HVAC, dairy, power, electronics, food and beverages.

The corrugated plate heat exchanger has many advantages compared to other heat exchangers as it gives maximum heat transfer; there is an increase in area due to the presence of corrugations which gives rise to turbulence and this helps in its usage in high viscosity applications as turbulence is also induced at low flow rates [2]. Some of the many advantages as stated by [3,4] include: it can be used for high viscosity applications, have high thermal effectiveness, large heat transfer per unit volume, low weight, ease of maintenance and a compact design.

The rate of heat transfer increases significantly (by 20–30%) when channels of a heat exchanger are constructed using corrugated sheets [5,6]. These improved geometries and corrugations of plates allow mixing of fluid layers within each channel that generates secondary flows known as Gortler vortices [7]. The flow through a flat plate channel is very smooth and the layers of fluid do not normally mix around. The heat is only transferred near the walls of the heat exchanger. The fluids at the center of the channel exchange only a very little amount of heat as they do not come into contact with the walls. Thus, the heat exchanged by the particles only near the wall is high. However, this is not the case when the channels are provided with corrugations. The convex and concave geometries produce very complex flows by constantly modifying the boundary layer. The flow starts to form vortices in the concave parts of the channel. When the flow near the wall approaches the convex surface, it is somewhat attached to the wall and thus the boundary layer is thin. The flow separates as it leaves the convex surface of the channel and is pushed to the middle of the channel by the high pressure gradient created by the formation of vortices and this increases the boundary layer thickness. This mixing constantly brings the fluid from the center close to the wall

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http://dx.doi.org/10.1016/j.expthermflusci.2016.02.009
0894-1777/

Nomenclature

A	total heat transfer area, m^2
Cp_{CW}	specific heat at average cold water temperature, kJ/kg K
Cp_{HW}	specific heat at average hot water temperature, kJ/kg K
Dh	hydraulic diameter, m
h	convective heat transfer coefficient, W/m^2 K
k	thermal conductivity, W/m K
Nu	average Nusselt number
Pr	Prandtl number
$\dot{Q}_{CW}$	heat transferred by cold water, W
$\dot{Q}_{HW}$	heat transferred by hot water, W
$\dot{Q}_{Average}$	average heat transfer between hot and cold water, W
Re_D	Reynolds number based on hydraulic diameter
t	thickness of the plate
T_{CWI}	cold water temperature at inlet, °C
T_{CWO}	cold water temperature at outlet, °C
T_{HWI}	hot water temperature at inlet, °C
T_{HWO}	hot water temperature at outlet, °C
U	overall heat transfer coefficient, W/m^2 K
$\dot{V}_{CW}$	cold water flowrate, L/s
$\dot{V}_{HW}$	hot water flowrate, L/s
ΔP_C	pressure loss of cold water, kPa
ΔP_H	pressure loss of hot water, kPa
ΔT_{CW}	temperature change of cold water, °C
ΔT_{HW}	temperature change of hot water, °C
ΔT_m	log mean temperature difference (LMTD)
ΔT_{outlet}	temperature difference of hot and cold water at the outlet of heat exchanger, °C
ΔX	plate spacing, mm
ρ_{CW}	water density at average cold water temperature, kg/m^3
ρ_{HW}	water density at average hot water temperature, kg/m^3
θ_{CW}	thermal length of the cold water channels
θ_{HW}	thermal length of the hot water channels
$\theta_{Average}$	average thermal length

and takes it back to the center. This process continues throughout the channel providing an increase in heat transfer [7].

2. Background

Several studies have been carried out on heat transfer enhancement using corrugated plate heat exchangers. Turns [5] indicated that the wavy geometries are known to improve the heat transfer by breaking and destabilizing the thermal boundary layer. He further stated that corrugated surface serves as turbulence promoter to increase the heat and mass transfer.

From extensive experiments done by Picon-Nunez et al. [8], it was concluded that there is an increase of 50–70% in the heat transfer coefficient for wavy geometry when compared to a plain fin counterpart. They studied the design of compact heat exchangers and the study was focused on the surface selection of the heat exchangers based on the volume performance index. They considered plain fins and louvered fins in their study. Furthermore, it has been observed experimentally and through mathematical model that the Reynolds number and the angle of cross-corrugation also significantly contribute to the optimization and enhancement of heat and mass transfer in the corrugated heat exchangers [9]. Studies have also been conducted on how the channel spacing in corrugated plate heat exchangers affects the overall heat transfer coefficient of the heat exchanger. Elshafei et al. [10] conducted studies on channel spacing and its effects on the heat transfer. They compared their results with conventional plate heat exchangers and their major conclusion was that corrugations increase heat transfer. Tauscher and Mayinger [11] carried out both numerical and experimental studies on heat transfer enhancement in plate heat exchangers with rib-roughened surfaces. They covered a wide range of variables from shape, width, height, groove angle, spacing, angles, and arrangement patterns. They concluded that the ribs show their best effects in regions where they can induce turbulence. They concluded that in the transition region from laminar to turbulent flow, the turbulence promoters show best performance. Faizal and Ahmed [7] studied the effect of plate spacing and found that a spacing of 6 mm gives optimum heat transfer compared to larger spacings.

Kan et al. [12] designed corrugated plate heat exchangers with inclination angles of 30°, 45° and 60° and studied the heat exchanger effectiveness at three different mass flow rates. They found that an angle of 30° gives the optimum heat transfer. Xie et al. [13] numerically investigated the heat transfer and flow characteristics in a wavy channel by changing the wave height, wave length, pitch, and channel width. They found that overall Nusselt number increases with increasing Reynolds number. Awad and Muzychka [14] presented the results of modeling of heat transfer and pressure drop for wavy fins. Rush et al. [15] performed experiments on sinusoidal wavy passages and investigated the local heat transfer and flow behavior. They varied the wave amplitude, phase angle, and wall-to-wall spacing and paid special attention to detect the onset of macroscopic mixing in the flow. Lin et al. [16] performed experimental trials and applied the Buckingham Pi theorem to develop a set of non-dimensional correlations between local and average Nusselt numbers to the flow conditions and the geometric parameters of the corrugated channel. They compared the experimental values of local and average Nusselt numbers with the results obtained using the correlations. They found that the average Nusselt number increases as the corrugation angle is increased from 30° to 45°. Nilpueng and Wongwises [17] studied the flow pattern and pressure drop in an upward single phase flow of liquid and two-phase (air–water) flow in sinusoidal wavy channels. They also studied the effect of phase shift at three different angles of 0°, 90° and 180°. They observed bubbly, slug, churn, and dispersed bubbly flows in the experiments and concluded that the phase shift of the channel walls have a strong influence on the flow patterns and pressure drop. Yin et al. [18] studied the effects of phase shift and *Re* on the flow and heat transfer numerically. They studied a set of sinusoidal wavy channels with different phase shifts of 0°, 30°, 60°, 90° and 180°. In another work, Nilpueng and Wongwises [19] studied the flow pattern and the pressure drop for an air–water two phase flow. The plates had a V-shaped corrugation with asymmetric corrugation angles of 55° and 10°. There found bubbly flow, bubble recirculation flow, annular-liquid bridge flow, slug flow, annular-liquid bridge flow/air-alone flow, and annular-liquid bridge flow for different cases. The authors also studied the surface roughness [20] on the heat transfer and pressure drop in a plate heat exchanger and found that increase in surface roughness increases the heat transfer coefficient. However, this came at the cost of a higher pressure loss.

A computational study done by Paras et al. [3] showed that there is an improvement in both the heat transfer and flow distribution with the use of corrugated plates for the heat exchanger. They also conducted studies on closely packed cross-corrugated plates; they discovered that the Nusselt number of the heat exchanger increases as the angle between the corrugated plates is increased. The corrugations in the plates produced complex

swirling flows. It was reported by them that the corrugations in the heat exchanger increase the heat transfer accompanied by pressure drop. Kanaris et al. [21] found out that the Reynolds numbers were higher at the crest of the plates and lower at the furrows. Their results confirmed that the corrugations play a major role in the flow distribution and increase in heat transfer. Hanpeng et al. [6] conducted numerical studies on the cross-corrugated plate type heat exchangers using CFD. They used FLUENT and studied fluid temperature, velocity, heat transfer coefficient and other parameters. They compared their data from corrugated surface with that of non-corrugated surfaces. Through their study, they concluded that the secondary spiral flows generated are able to bring the low temperature center fluid close to the wall and disturb the boundary layer.

Shailendra et al. [22] conducted research on the heat transfer analysis of corrugated plate heat exchanger of different plate geometries. Their major focus was on the variation of chevron angle and its effect on the overall heat transfer coefficient of the heat exchanger. They obtained data for single phase flow (water to water) configuration in a corrugated plate heat exchanger for symmetric 45°/45°, 45°/75° chevron angle plates. They also studied other variables such as Reynolds number, Prandtl number and Nusselt number. Staseik et al. [23] conducted experimental studies on the heat transfer in corrugated passages. Their major focus was on cross-corrugated geometry. The angle in their case varied from 0° to 144°. Effects of corrugation angle, Reynolds number, friction factor and geometry were all investigated in their experiments. They discovered that Nusselt number of heat exchanger increases as the angle between the corrugated plates is increased. They also discovered that as the angle between the plates increased, the friction factor increased as well. Increase in angle between plates would mean a decrease in the angle of inclination, as the lowest inclination angles of cross-corrugated would have the highest value of angle between plates.

Lyytikäinen et al. [24] from their numerical studies for varying corrugation angles and corrugation lengths found that both heat transfer and pressure drop increase as the corrugation angle is increased. They concluded that it is not easy to find a specific geometry that provides both a low pressure drop and a high heat transfer simultaneously. Pehlivan et al. [25] from their experimental study concluded that corrugated channels are a good alternative for high heat flux applications. They also found that increasing the corrugation angle gives rise to a higher heat transfer rate. Aslan et al. [26] in a numerical work studied the effect of two types of wavy peaks (a) sharp wavy peak and (b) rounded wavy peak on the heat transfer characteristics using finite volume simulation. The heat transfer through such wavy channels takes place differently compared to parallel plate channels – the main flow direction is parallel to the channel axis, but the local flow direction is always changed due to channel waviness. There is periodic interruption to the thermal boundary layer on the wall by flow recirculation, separation and reattachment which improves the heat transfer; however, the pressure loss also increases. The computed values of heat transfer coefficient were in good agreement with the experimental results for the wavy channels [26].

Blomerius et al. [27] performed numerical investigation of the flow field and heat transfer in sine-wave cross-corrugated ducts. They compared the results of varying the angle between the main stream and the corrugations from a number of previous works. They observed that, for small inclination angles of less than 30°, the fluid primarily follows the main stream direction. The flow structure becomes complex for 45° since both types of flow trajectories are equally prominent. Focke et al. [28] from their experimental results also concluded that the transfer rates are almost uniform across the plate width for this angle. Lee and Lee [29] carried out large-eddy simulation (LES) to examine the characteristics of chevron type plate heat exchangers. They compared their heat transfer and pressure drop data with experimental results and found reasonably good agreement. Nishimura's group on their extensive experiments [30–32] observed flow separation and recirculation zones in the wave troughs at low *Re*. They also observed that, in a channel with nine wavelengths, the flow becomes three dimensional at *Re* of about 100, with spanwise and streamwise vortices. On a geometry with 180° phase shift, they observed large recirculation zones in the troughs of the channel. The flow structures were much larger than the separation bubbles observed for the case with no phase shift. For this case, with a Reynolds number of about 350, the flow was found to become unsteady as these vortices interacted with the core flow. Wang and Vanka [33] numerically studied the rates of heat transfer for flow through a periodic array of wavy passages. They found that the flow is steady till a Reynolds number of 180 after which it became oscillatory. In the transitional flow regime, the enhancement in heat transfer was found to be higher by a factor of 2.5.

3. Objectives

The demand for heat exchangers that are efficient, compact and less expensive has increased of late due to numerous and increasing applications in industries. According to Ohman [34], research on heat exchangers for low temperature power applications is not easily available despite the demand in industries. The present work focuses on determining the effect of wavy channel inclination angle on the heat transfer and pressure loss through experimentation. It is clear from a detailed literature search above that such work is not done in the past. It proposes to find out which inclination angle gives the optimum heat transfer. Findings from this study can be applied and adopted for small temperature difference and pressure applications where phase change is not involved. The low flow rates of the hot and cold fluids make it a unique study, as the fluids get more time to exchange heat.

4. Experimental setup and procedure

The geometric details of the wavy plates are shown in Fig. 1. The material used was galvanized iron with a thickness of 0.4 mm. All the plates had a height of 600 mm and width of 250 mm. Also shown is a photograph of the sheet which was cut for testing the 0° channel inclination angle heat exchanger.

A schematic diagram of the experimental set up for testing the heat exchanger is shown in Fig. 2. The channel spacing was kept constant at 6 mm for all the heat exchangers based on the finding from our previous work [7].

The orientation of the heat exchangers during testing was such that the inlet port was at the bottom and the outlet port was at the top. The fluid of both media entered the heat exchanger form the bottom and flowed to the top against gravity making it a parallel flow heat exchanger. The inlet and exit of the hot water into and out of the heat exchanger are shown by red arrows, while those of the cold water are shown by blue arrows in Fig. 2. The upward flow allowed for the full utilization of the whole channel for heat transfer and guaranteed the channel to be fully filled and prevented formation of hydraulic diameters [35].

The experiments were conducted in the Thermo-fluids (Mechanical Engineering) laboratory of the University of the South Pacific. The experimental set-up included a heater connected to normal tap water for continuous supply of water to the heater, four CABAC T6201 digital thermometers with a resolution of 0.1 °C and a temperature range of −50 °C to +250 °C and two WIKA EN 837-1 pressure gauges with pressure range of 0−100 kPa with an accuracy of ±1% and a temperature range of −20 °C to +60 °C. The

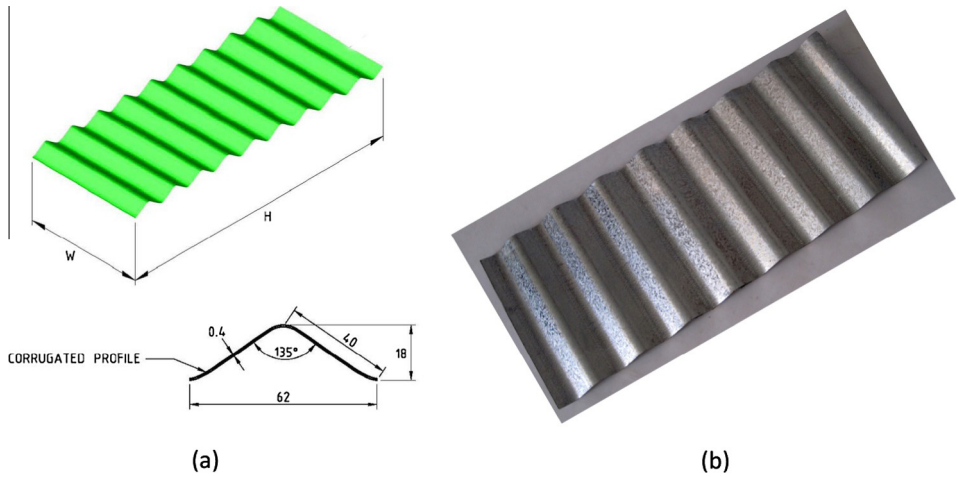

Fig. 1. (a) Geometric details of the heat exchanger plates and (b) a photograph of one of the plates.

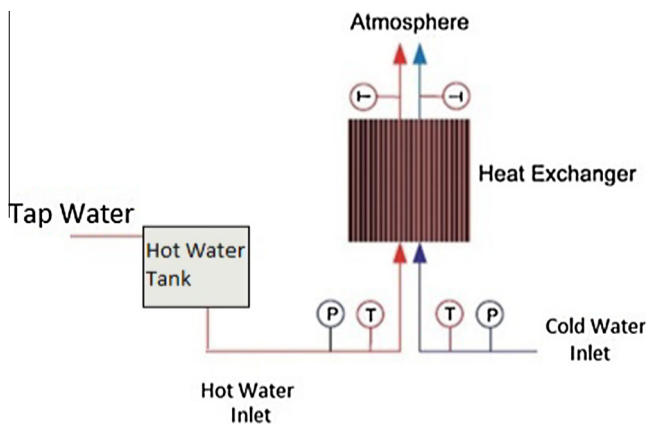

Fig. 2. Schematic diagram of the experimental setup.

two WIKA EN pressure gauges and two thermometers were mounted at the entrance of the heat exchanger while the remaining thermometers were mounted on the discharge end of the heat exchangers. The outlet pressure was taken as atmospheric for the purpose of analysis. The pressure and temperature gauges in the setup are denoted by P and T respectively in Fig. 2.

Designs of all the five heat exchangers are shown in Fig. 3 with their inclination angles. The flow direction of fluid is from bottom to the top as shown by the red arrows in Fig. 3. As the inclination angle is increased the fluid flow direction is getting closer to becoming parallel with the corrugation inclination angle. The 0° inclination angles are perpendicular to the fluid flow while the 80° inclination angle corrugations are almost parallel of the fluid flow. The heat exchangers were fabricated in the workshop of the School.

The hot water and the cold water flow rates were measured by collecting a known quantity of water in a flask and timing it. The cold water flow rate was then increased while the hot water flow rate was kept constant at 0.046 L/s. The cold water flow rate was varied form 0.057 L/s to 0.076 L/s as given in Table 1. The inlet and outlet temperatures and pressures were measured and recorded at the different flow rates of cold water. The inlet temperatures of hot and cold water were maintained at 38 °C and 25 °C respectively. Experiments on each heat exchanger were conducted for five different cold water flow rates. The heat exchangers were tested one after the other. Three trials were done in order to verify the results and to reduce experimental errors as much as possible. All the variables of the experiment are given in Table 1.

Fig. 4(a) shows the channel spacing ΔX which is kept constant for all the cases studied. Wave length is denoted by λ, and w is the amplitude. Fig. 4(b) shows how the corrugated sheets were rotated in order to achieve the desired corrugated plate inclination angles.

5. Method of analysis

The Log Mean Temperature Difference (LMTD) method was used to analyze the results as it uses the inlet and the outlet temperatures of the two media. LMTD method is appropriate for our case since there is no phase change involved during the heat transfer process and uses all the four temperatures to give a mean change in temperature which would otherwise be hard to find [1,35]. Each set of inlet and outlet temperatures for heat exchangers was recorded and the temperatures were used to calculate the LMTD. The heat energy gained by the cold water and the heat energy lost by the hot water were calculated in order to calculate the average change in energy. The heat transfer between the two streams was calculated as:

$$\dot{Q}_{HW} = \rho_{HW} c_{P_{HW}} \dot{V}_{HW} (\Delta T_{HW}) \tag{1}$$

$$\dot{Q}_{CW} = \rho_{CW} c_{PCW} \dot{V}_{CW} (\Delta T_{CW}) \tag{2}$$

$$\dot{Q}_{Average} = \left(\frac{\dot{Q}_{HW} + \dot{Q}_{CW}}{2} \right) \tag{3}$$

where $\dot{Q}_{HW}$ and $\dot{Q}_{CW}$ are heat transferred by hot and cold water streams respectively, $\dot{Q}_{Average}$ is the average heat transfer between the two streams.

This average change in energy was used along with the LMTD and area to calculate the overall heat transfer coefficient. The overall heat transfer coefficient (U) was calculated using the following equations:

$$U = \frac{\dot{Q}_{Average}}{A \Delta T_M} \tag{4}$$

$$\Delta T_M = \frac{(T_{HWI} - T_{CWI}) - (T_{HWO} - T_{CWO})}{\ln \frac{(T_{HWI} - T_{CWI})}{(T_{HWO} - T_{CWO})}} \tag{5}$$

where A is the total heat transfer area and ΔT_m is the LMTD. The overall heat transfer coefficient takes into account all the resistances that are present in the path of the heat transfer. Same

Fig. 3. AutoCAD drawings of different corrugation inclination angles. The figure on the far left is the 0° inclination angle followed by 20°, 40°, 60° and lastly 80°.

Table 1
Variables in the experiment.

Details		Dimensions
Plate height	H	600 mm
Plate width	W	250 mm
Total area	A	2.1 m^2
Spacing	ΔX	6 mm
Inclination angles	θ	0°, 20°, 40°, 60°, 80°
Number of plates	N	11
Number of channels (hot)	N_H	5
Number of channels (cold)	N_C	5
Wave length (λ)	λ	80 mm
Amplitude (w)	w	8 mm
Hot water flow-rate	V_{HW}	0.046 L/s
Cold water flow-rate	V_{CW}	0.057–0.076 L/s
Hot water inlet temperature	T_{HWI}	38 °C
Cold water inlet temperature	T_{CWI}	25.4 °C

process was followed in order to calculate the overall heat transfer coefficient for all the heat exchangers. The results were then analyzed. The change in temperature of both hot and cold water was plotted against the different flow rates of cold water for each inclination angle (heat exchangers). The pressure loss against the different flow rate of cold water was also plotted for different inclination angles. After comparing the results of all three trials, it was observed that the variation in the data was very little and the trends of all the trials were similar. It was observed that good heat transfer was occurring between the two fluids. The repeatability of the temperature measurements was within ±4% and that of the pressure measurements was within ±2.8%. The accuracies of measurement or estimation of ρ, c_p, volume flow rate and temperatures were taken into consideration for estimating the uncertainty of heat transfer rate, considering the fact that always the temperature change was used for estimating heat transfer rate. All uncertainties were estimated following the procedure of Moffat [36]. From Eqs. (1) and (2), the heat transfer rate is a function of density, specific heat, flow rate and temperature difference. The uncertainty in the heat transfer rate is

$$\frac{d\dot{Q}}{\dot{Q}} \times 100 = \pm\frac{1}{\dot{Q}}\left[\left(\frac{\partial\dot{Q}}{\partial\rho}\right)^2(d\rho)^2 + \left(\frac{\partial\dot{Q}}{\partial c_p}\right)^2(dc_p)^2 + \left(\frac{\partial\dot{Q}}{\partial\dot{V}}\right)^2(d\dot{V})^2 + \left(\frac{\partial\dot{Q}}{\partial\Delta T}\right)^2(d\Delta T)^2\right]^{0.5} \times 100 \quad (6)$$

Substituting the uncertainties and values of the individual parameters in Eq. (6), the uncertainty in the estimation of heat transfer rate was found to be ±4.2%. Similarly, taking into account the uncertainties of the contributing terms of the overall heat transfer coefficient, the uncertainty in the overall heat transfer coefficient was estimated to be ±4.7%.

6. Results and discussion

The results are presented and discussed in this section. The changes in temperature of the hot and cold water from inlet to outlet are presented in Fig. 5. It was noted that as the volume flow rate of the cold water was increased, the change in temperature of cold water decreased while at the same time the change in temperature of the hot water increased.

The change in temperature of the hot water is seen to be increasing as the flow rate of cold water is increasing and this is due to the fact that as the flow of cold water increases, the heat carrying capability of the cold water from the hot water, which gets enhanced due to corrugations, increases thus giving a high value of heat transfer. However, the opposite trend is noticed for cold water as the flow rate is increased. This is because of the higher flow rate of the cold water. So, as the particles traveling through the cold channels increase, there are more number of particles coming in contact with the walls in the same amount of time and as each carries some amount of heat, the change in tempera-

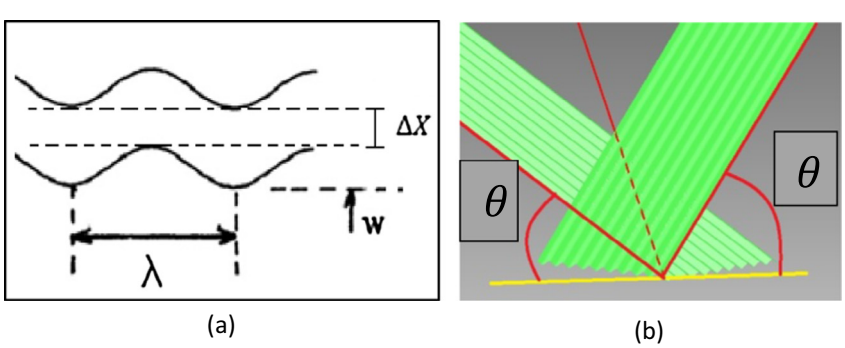

Fig. 4. (a) Channel spacing, wavelength and amplitude and (b) inclination angle channel before and after rotation.

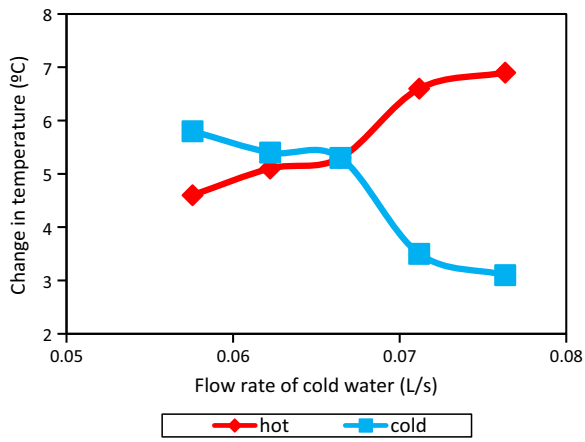

Fig. 5. Change in temperature from inlet to outlet for the 60° inclination heat exchanger.

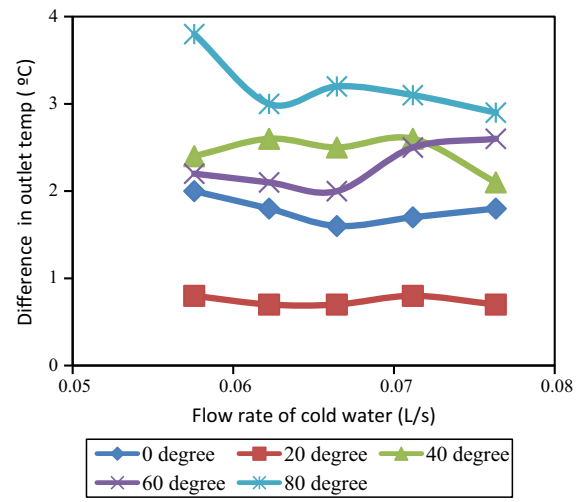

Fig. 6. Difference in outlet temperatures of the two streams for all the heat exchangers with varying cold water flow rate.

ture decreases with an increasing number of particles of cold water (increase in flow rate of cold water). Similar results were observed for the other channel inclination angles. It is believed that such significant changes in temperature are unique to the wavy plate heat exchangers which induce turbulence and mixing irrespective of the small temperature difference at the inlet and a small heat exchanger used in the present studies. This will be further discussed later.

The difference in outlet temperatures of the hot and cold streams of the heat exchangers were observed to be decreasing for all designs as the cold water flow rate was increased. This is depicted in Fig. 6. The 80° heat exchanger has the largest difference between the two outlet temperatures suggesting a lower rate of heat being exchanged whereas the 20° heat exchanger has the smallest difference between the two outlet temperatures suggesting a high rate of heat transfer. The outlet temperature difference was compared to the inlet temperature of both the 20° and the 80° designs. The inlet temperature difference was the same at about 13 °C for all the cases studied in this work. The outlet temperature difference of 20° design had an average value of 0.74 °C whereas that for the 80° design the value was around 3.2 °C suggesting a higher heat transfer in the 20° design. The results also show that all the heat exchangers in the present work have good heat transfer between the two streams due to the corrugations and the resulting turbulence and mixing.

The pressure loss of the cold water increases as the volume flow rate increases for all the heat exchangers as shown in Fig. 7. The pressure loss for the cold water is highest for the 0° design for all the cold water flow rates. This is because the corrugations in a 0° heat exchanger are perpendicular to the direction of fluid flow providing maximum resistance to the flow whereas the 80° inclination angle has the least amount of pressure losses as the inclination angle is such that the corrugations are almost parallel to the direction of fluid flow providing the least amount of resistance to the flow. Similar results were reported by Focke et al. [28] with the highest pressure loss occurring when the flow angle was at about 10° to the corrugations.

The flow through an 80° heat exchanger is more of "free flow" with the corrugations forming a pipe-like geometry. It was also observed that the variation in the pressure loss of the hot water was fairly constant (varying from 13.5 kPa to 15 kPa), as shown in Fig. 8. This is because the flow rate of the hot water was kept constant.

After finding out the average rate of heat transfer for all the heat exchangers, it was seen that the average rate of heat transfer increased as the cold water flow rate is increased for all the heat exchangers. This is shown in Fig. 9.

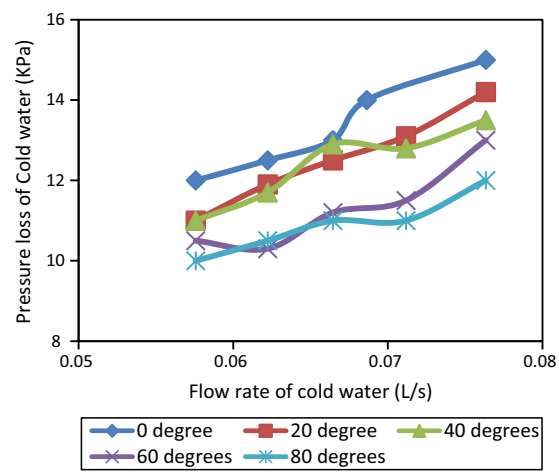

Fig. 7. Pressure loss of cold water with varying cold water flow rate.

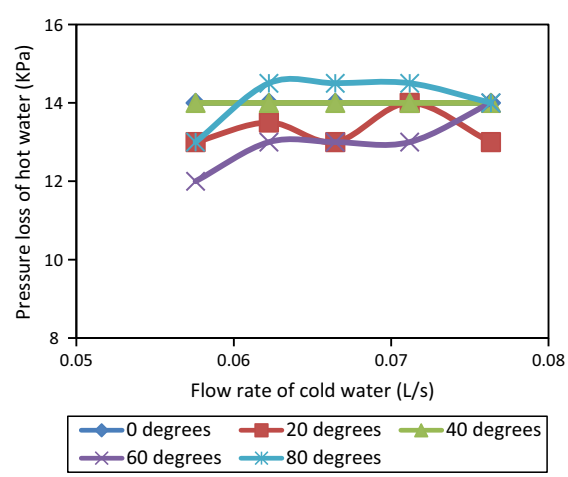

Fig. 8. Pressure loss of hot water with varying cold water flow rate.

The average rate of heat transfer generally increases for all the angles indicating that the hot water is transferring more heat to the cold water. However, the average rate of heat transfer is high-

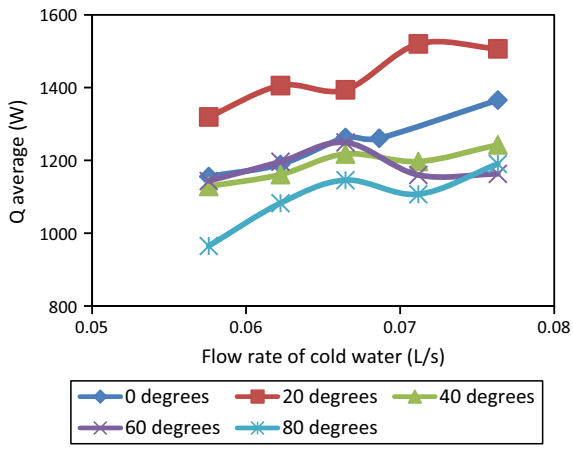

Fig. 9. Average rate of heat transfer for all the heat exchangers with varying cold water flow rate.

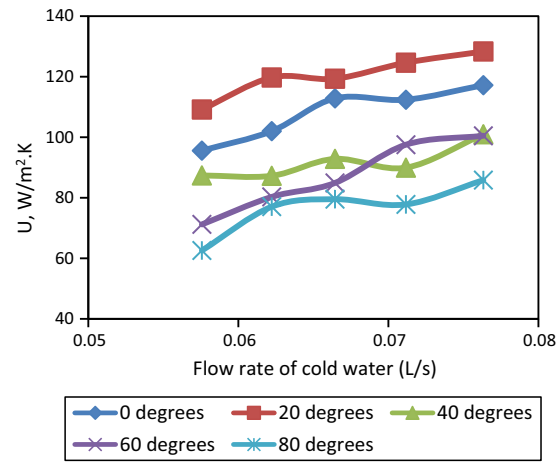

Fig. 10. Overall heat transfer coefficient for all the heat exchangers with varying cold water flow rate.

est for the 20° heat exchanger for all the flow rates compared to other heat exchangers. This is in agreement with the earlier presented results of the difference in outlet temperatures in Fig. 6. The 80° inclination angle heat exchanger has the lowest value of average rate of heat transfer for almost all the flow rates. This is because, as the inclination angle is increased further and approaches 80°, the corrugations are almost parallel (shown in Fig. 3) to the direction of fluid flow and moving more toward free flow. This effect starts to be felt from the 40° case, when the average heat transfer does not increase significantly beyond a certain flow rate. For the 60° angle, there is actually a drop in the heat transfer for the last two flow rates. The measurements showed that the heat gained by the cold water drops significantly for all the angles from 40° to 80°. This could probably be due to the flow tending to become more parallel to the channels and the higher flow rate does not allow enough transfer of heat to the cold water.

Similar results were reported by Focke et al. [28] who found the maximum heat transfer between flow angles of 10–20° with the horizontal.

The variations of the overall heat transfer coefficient for all the heat exchangers with cold water flow rate are shown in Fig. 10. As seen from Fig. 10, the overall heat transfer coefficient of all the heat exchangers increases as the flow rate of cold water increases indicating that more heat is transferred when the flow rate of the cold water is increased. It can also be noted that the 20° inclination angle heat exchanger has the highest value of heat transfer coefficient for all the flow rates while the 80° design has the lowest value of coefficient of heat transfer among all the heat exchangers.

The plot of the overall heat transfer coefficient against the corrugation inclination angle gives a clear picture of the performance of the heat exchangers in terms of rate of heat transfer (Fig. 11). The overall coefficient of heat transfer increases from 0° to 20° after which it gradually decreases. This is again explained by the inclination of the corrugations with regards to the flow direction of the fluid. It is seen in Fig. 11 that as the corrugation inclination angle increases from 0° to 20°, the overall heat transfer coefficient increases and falls afterward as the inclination angles increases from 20° to 80°. At 0°, where the corrugation are perpendicular to the flow direction, the rate of heat transfer is not that high as the fluid flow is predominantly two-dimensional, but as the angle is increased the heat transfer increases and is maximum at 20°. At this angle, the resistance to flow is high and at the same time, the third flow component comes into picture as some flow is forced to take place parallel to the corrugations. This results in generation of more turbulence and mixing and hence, a higher rate of heat transfer. As the inclination angle approaches 80°, the corrugation become almost parallel to the fluid flow direction and the overall heat transfer coefficient decreases. This is because the generation of turbulence is less for this angle and hence less effective heat transfer. Similar behavior was also observed by Stasiek et al. [23]. They observed that as the angle between the corrugations increased, the heat transfer increased. Same is observed in our case as our inclination angle is inversely related to the angle in-between the plates.

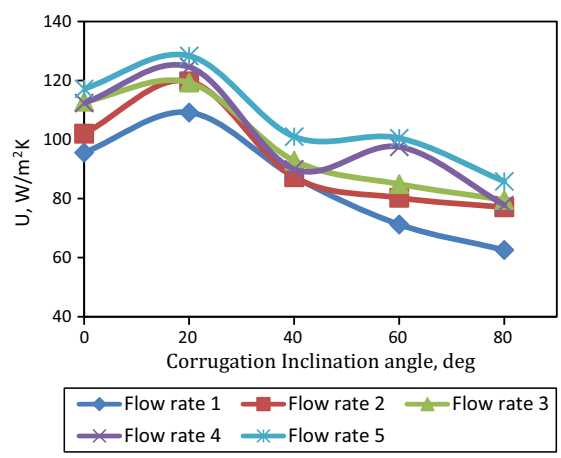

Fig. 11. Variation of the overall coefficient of heat transfer (U) with the inclination angle.

The performance of the heat exchanger is assessed by comparing the pressure loss against the overall heat transfer coefficient. It is observed that as the heat transfer rate increases, so does the pressure loss. This is depicted in Fig. 12. As shown in Fig. 12, the 20° heat exchanger has the greatest value of heat transfer while the 80° design has the lowest value of heat transfer. It can also be seen that the angle which has a higher value of heat transfer also tends to have a higher value of pressure loss (as in the case of 20° design) whereas the heat exchangers which have a lower value of heat transfer also have a lower value of pressure loss. However there are some cases for which a heat exchanger can have a high value of heat transfer while maintaining the same value of pressure loss at which another design has a lower value of heat transfer. This is shown by the circle on the plot, the two points are for the 0° design and 20° design; it is observed that the 20°

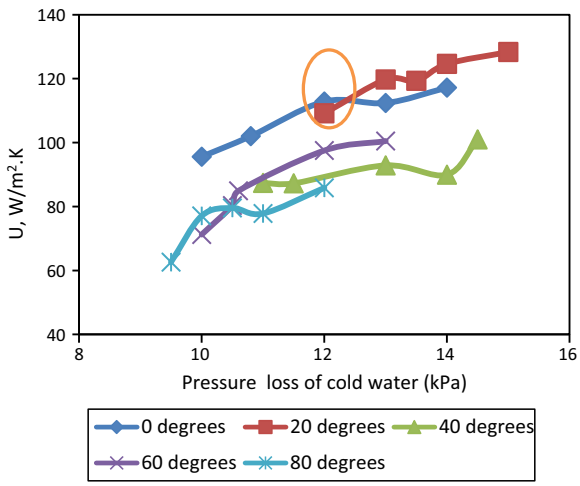

Fig. 12. Overall heat transfer coefficient of all heat exchangers against the pressure loss of cold water.

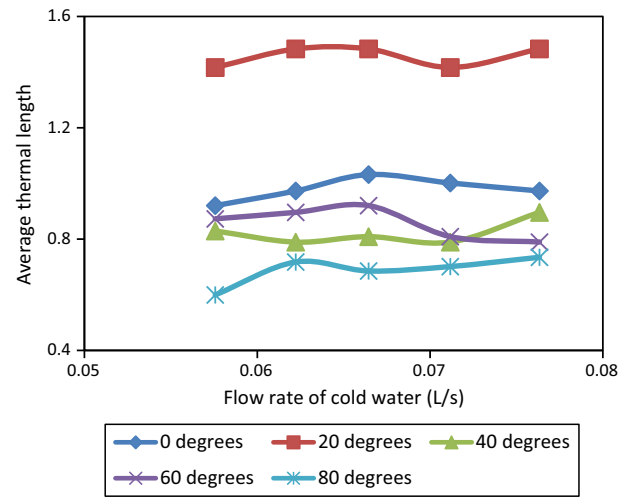

Fig. 13. Variation of the average thermal length with cold water flow rate.

design has a higher value of heat transfer than the 0° design while both of them have the same pressure loss. There is an increase in the overall heat transfer coefficient with increasing pressure loss, with the 20° heat exchanger giving the highest U value as well as the highest pressure loss. Therefore, this heat exchanger would have higher operational costs. However, this design is appropriate because of significant heat transfer coefficients and effective heat transfer, even though the pressure loss is high. The operational cost could be higher due to high pressure losses, but the main objective is to obtain effective heat transfer rate between the two streams for such a low temperature difference. However, for an efficient design, the selection of a suitable configuration is always a trade-off between the overall heat transfer coefficient, which has an influence on the surface area and hence the capital cost and pressure drop, which dictates the pumping power requirement and hence the operational cost.

The variations of the average thermal length, $\theta_{Average}$, for varying cold water flow rates and channel inclination angles are shown in Fig. 13. The thermal length represents the performance and is the relationship between the temperature difference in one stream and the LMTD. A higher thermal length means that the heat transfer and the pressure drop are large, whereas a lower thermal length means that heat transfer and pressure drops are low [37]. The thermal lengths are calculated as:

$$\theta_{HW} = \frac{\Delta T_{HW}}{\Delta T_m} \tag{7}$$

$$\theta_{CW} = \frac{\Delta T_{CW}}{\Delta T_m} \tag{8}$$

$$\theta_{Average} = \frac{\theta_{HW} + \theta_{CW}}{2} \tag{9}$$

It can clearly be seen from Fig. 13 that for the 20° heat exchanger, the thermal length is the highest, which again proves the superiority of the 20° design. Fig. 14 shows the variation of average thermal length with pressure loss of cold water. The average thermal length, which is a measure of the performance, is much higher for the 20° design. Surprisingly, the 0° case, again demonstrates a better performance, although it also means a higher pressure loss.

Interestingly, the pressure loss of cold water is higher for the 40° angle compared to the 0° case. However, it is still less than the 20° inclination angle. The 80° heat exchanger has the lowest value of pressure loss as well as the average thermal length, because the flow direction is almost parallel to the channel angle.

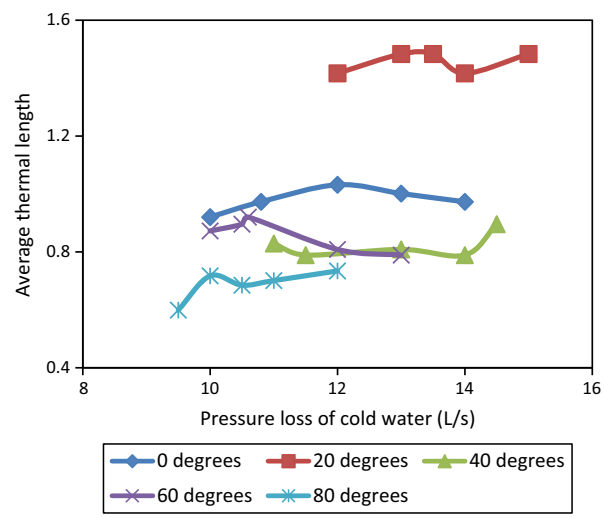

Fig. 14. Variation of the average thermal length with the pressure loss of cold water.

At an angle of 18°, the authors of [28] found the near-optimum heat transfer which increased slightly for 10°.

Fig. 15 shows the relation between the overall heat transfer coefficient and the average thermal length for all the channel inclination angles. It is very clear that the 20° angle gives the best performance with the highest heat transfer coefficient and the highest thermal length. Again, the 0° angle has a high value of U as well as average thermal length.

Since the Reynolds number was nearly the same for the low flow rate of cold water as that of the hot water and also since the temperature difference between the two streams was not large it was assumed that the convective heat transfer coefficient is same on both the sides of the plate [35,38].

The value of h was calculated using the equation

$$\frac{1}{h} = \frac{1}{2}\left[\frac{1}{U} - \frac{t}{k}\right] \tag{10}$$

The Nusselt number was calculated using the relation

$$Nu = \frac{hD_h}{k} \tag{11}$$

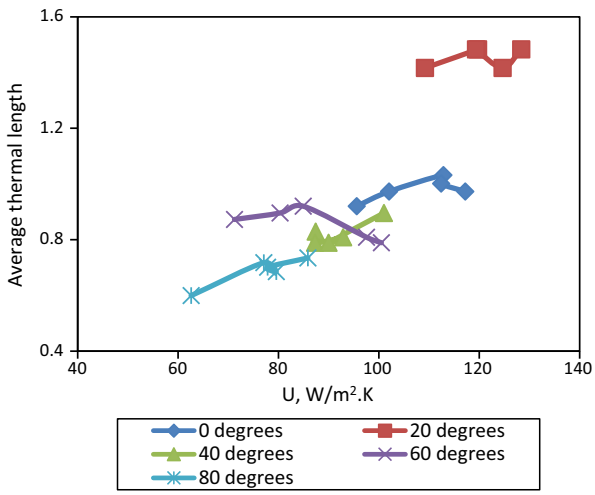

Fig. 15. Variation of the average thermal length with the overall heat transfer coefficient.

In the present work, the Reynolds number ranged from 450 to 600 because of the low flow rates. The Nusselt number range was 12–26. The uncertainty in the estimation of *Nu* was found to be ±7.8%. Fig. 16 shows the variation of the Nusselt number with Reynolds number for different corrugation inclination angles. The trends clearly suggest that the Nusselt number increases linearly with Reynolds number. Also, the 20° angle has the highest Nusselt number for all the Reynolds numbers followed by the 0° case.

For single phase heat transfer, *Nu* is normally represented by the following correlation:

$$Nu = C\,Re^{a}Pr^{b} \tag{12}$$

Using the experimental data, the values of *C* and *a* were obtained with the value of *b* assumed to be constant at 1/3 [20,29,35].

For the 20° inclination angle, the following correlation was obtained:

$$Nu = 0.11Re^{0.78}Pr^{1/3} \tag{13}$$

For the 0° inclination angle, the following correlation was obtained:

$$Nu = 0.11Re^{0.75}Pr^{1/3} \tag{14}$$

For the 40° inclination angle, the following correlation was obtained:

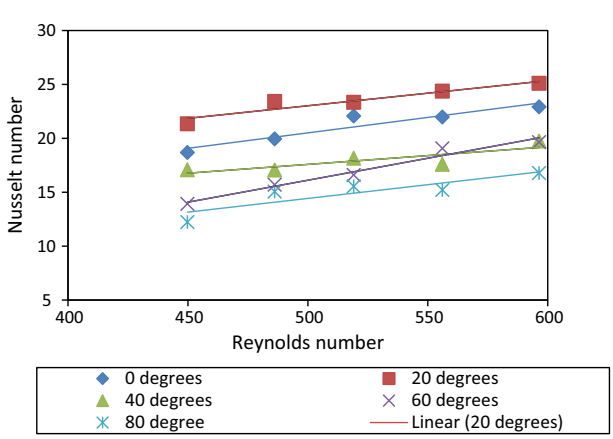

Fig. 16. Variation of Nusselt number with Reynolds number for different corrugation inclination angles.

$$Nu = 0.11Re^{0.73}Pr^{1/3} \tag{15}$$

For the 60° inclination angle, the following correlation was obtained:

$$Nu = 0.11Re^{0.71}Pr^{1/3} \tag{16}$$

For the 60° inclination angle, the following correlation was obtained:

$$Nu = 0.11Re^{0.69}Pr^{1/3} \tag{17}$$

Considering the smaller range of *Re* in the present work and the small temperature difference, it is not surprising that the *C* value is same for all the inclination angles and the value of *a* is the highest for 20° case, which showed superior performance of this heat exchanger compared to all other heat exchangers. For the 40° angle, the correlation was similar to the one obtained by Khan et al. [35] for their 30°/30° chevron angle. Kan et al. [12] tested three inclination angles of 30°, 45° and 60° and found the 30° angle to be the most effective. Also, interestingly, the correlation for the 20° angle was similar to the correlation obtained for the 60° chevron angle obtained by Lee and Lee using the Wilson plot method [29].

The *Nu* for this *Re* range is higher than that of a plate heat exchanger [39]. The enhancement in heat transfer is due to increased effective area and higher turbulence caused by the wavy channels. As discussed above, the present results also agree well with the 30°/30° and mixed 30°/60° chevron angle plate results of Khan et al. [35] and the *Nu* values in the present work were very similar to their values at the corresponding *Re* values for the inclination angles of 0° and 20° [35].

In the present study, both the streams are single phase flows which transfer heat by forced convection and conduction. Both hydrodynamic and thermal boundary layers begin to form as the fluids enter the channels. The corrugations (convex and concave surfaces) cause instabilities in the flow which enhance turbulence even at these low flow rates. The velocity gradient has a constant sign across the boundary layer making the flow over the convex surface more stable. Any fluid element that gets displaced outward to a higher velocity region gets pushed back to a lower radius region due to higher radial pressure gradient [7,40,41]. However, the velocity gradient changes sign in the boundary layer over the concave surface, as the flow is unstable. Any fluid element that gets displaced to a greater radius moves into a region of low velocities where the pressure gradient is too low to push it back to a lower radius (Görtler instability) [41]. The secondary flows resulting from the corrugations result in a partial restart of the boundary layer and prevent it from getting fully developed. This complex flow over the convex and concave surfaces continuously causes flow separation and reattachment resulting in mixing of recirculated flows with the core flows [32] and hence heat transfer from and to almost all the particles even at low Reynolds number of the flow. The 20° heat exchanger further enhances the mixing by inducing a third component of the flow and results in the highest heat transfer.

7. Conclusions

Experiments were conducted on cross-corrugated plate heat exchangers to investigate the heat transfer characteristics and pressure drop with different channel inclination angles of 0°, 20°, 40°, 60°and 80° at five different $\dot{V}_{CW}$ and at a constant $\dot{V}_{HW}$. From the experimental results, the following conclusion can be drawn:

- The change in temperature at the outlet for the cold and the hot water decreases with an increase in the flow rate of cold water implying that the heat transfer increases with an increase in flow rate. The 20° has the lowest change in outlet temperature difference.
- The average heat transfer between cold and hot water is optimum at the 20° inclination heat exchanger.
- Average heat transfer between hot and cold water and the overall heat transfer coefficient U is maximum for the 20° heat exchanger and minimum for the 80° one.
- The pressure loss did not vary much for hot water as the flow rate was kept constant but there was variation in the cold water pressure loss as the flow rate was varied. Pressure loss also increased with increasing rate of heat transfer.
- The average thermal length was considerably higher for the 20° inclination angle compared to the other cases.
- Correlations of non-dimensional numbers were developed from the experimental results and compared with some of the previous works.
- Finally, it can be concluded that the 20° plate heat exchanger gave the optimum heat transfer with a higher value of pressure loss. However there existed some flow rates which gave a reasonably high pressure loss when compared to other heat exchanger designs.

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