EXPERIMENTAL INVESTIGATION OF BRAZED PLATE HEAT EXCHANGER FOR SMALL TEMPERATURE DIFFERENCE

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Abstract- *We performed experimental study on a Single phase flow Brazed plate heat exchanger with corrugated plates for the applications of small temperature difference and comparison of thermo-hydraulic properties with gasket plate heat exchanger for both counter flow and parallel flow arrangement. We performed the experiment on a Brazed Plate Heat Exchanger with corrugated plates with corrugation angle 45°, ten plates arranged in parallel connection making four hot and five cold channels alternatively. The per plate heat transfer area is 0.014 m². Water is used as working fluid for both hot and cold flow. We kept the temperature of hot water inlet and cold water inlet constant and flow rate of cold water also constant. Hot water flow rate is varied from 2 litre/min to 6 litre/min. We have observed that if we increase the flow rate of hot water, the heat transfer increases. It is due to turbulence enhanced by corrugation in plates at high flow velocities, which increases the heat transfer. We have shown the comparisons of the overall heat transfer coefficients, U, the temperature difference between hot and cold water flow streams at outlet. The finding from this experiment will increase the use of brazed plate heat exchangers over gasket type for small temperature difference when flow rate is also small. Here we also tried to investigate the change in thermo-hydraulic properties at the different flow rate of cold water. The reading from this experiment will enhance the current knowledge in BPHE for small temperature difference applications and also help in the validation in CFD codes.*

Key Words: BPHE, CFD, CWF, HWF.

1. INTRODUCTION

The exchange of heat between two media depends upon the difference of temperature between them. If the difference of temperature between them is more then heat flux will be more. The heat exchangers are the devices which exchange heat or thermal energy between two media. We have seen some popular applications of heat exchangers in power plants, food and chemical industries, air-conditioning and refrigeration industries, electronic industries, waste heat recovery systems and manufacturing industries.

In Plate heat exchangers a stack of plate comprises and form alternate channels for hot and cold fluids. Inlet and outlet ports are given for entry and exit of hot and cold fluids separately. Plates are made of stainless steel with corrugated or embossed pattern. The hot and cold fluid flow alternatively in the channel formed by plates.

According to Thulukkanam[20], the plate heat exchanger (PHE) was invented by Richard Seligman in 1923. In early applications, plate heat exchangers (PHEs) were utilized for hygienic reasons in the fields such as the dairy and paper/pulp industries. Later they found broader application in heating, ventilation, and air-conditioning systems. Again according to Thulukkanam, in the 1990s the brazed plate heat exchanger (BPHE) was invented. Adjacent plates are welded, brazed, or soldered together to form inner flow channels, instead of being sealed by gaskets. As with the PHE, the BPHE provides excellent thermal-hydraulic performance, low discharge operation, and it's highly compactness; moreover, the BPHE is capable of withstanding high pressure. The disadvantage of the BPHE compared with the PHE is that mechanically cleaning is much more difficult.

To date, both single-phase flow and two-phase flow in BPHEs have been studied. For single-phase flows, geometric parameters (such as chevron/herringbone angle, enlargement factor, and corrugation profile aspect ratio), working fluid, and flow maldistribution are three important factors that influence thermal-hydraulic characteristics. For two-phase flow, working conditions (such as vapour quality, mass flux, heat flux, and saturation temperature) also play significant roles in performance. Although much research have been completed over the past eighty years, there is still a dearth of design information available in the open literature like general heat transfer and pressure drop correlations, due to the proprietary nature of BPHE industry. Many results are important but applicable for very limited geometry, fluid, or operating conditions. There has been some work in which researchers focus on the generalization, using a theoretical approach, model predictions and empirical correlations, for design and optimization. However, the current design tools remain restricted, even for single-phase flows; the situation for two-phase flow is even less complete.

2. Nomenclature

Table-1

Q	heat transfer rate, (W)
U	overall heat transfer coefficient, (W/m2K)
h	convective heat transfer coefficient, (W/m2K)
m	mass flow rate, (Kg/s)
u	fluid flow velocity, (m/s)
Cp	specific heat, (J/KgK)
K	thermal conductivity, (W/mK)
N	number of corrugated plates
T	temperature of the fluid flow, °C
A	heat transfer area of heat exchanger, m2
b	gap between two plates, m
W	width of channel, m
L	length of plates, m
L_p	port to port length, m
Dh	hydraulic diameter of channel, m
Pc	corrugation pitch, m
Δx	
ΔT_{LMTD}	
Re	
Pr	
Nu	
f	

Greek Symbols

μ	dynamic viscosity, Pa.s
ρ	mass density, kg/m3
ν	volume concentration
ϕ	chevron angle of BPHE

Nowadays Heat exchangers have a comprehensive application in industries. The brazed plate heat exchanger is a compact heat exchanger with high efficiency, operability at higher pressure, small size and low cost rather than most other compact heat exchangers, which are used in refrigeration and heat pump systems, process water heating and domestic hot water system that shown in Fig. 1. Due to the limitation of fossil energy sources, reducing the thermal energy used to produce hot water in buildings– which comprise a large portion of domestic energy consumption– may lead to economization in energy. Some experimental studies on enhancement of plate heat exchanger heat transfer by changes in its physical specification such as the effect of changes in plate configuration and the effect of increment in plate roughness were performed by researchers.

In an analysis of PHE all the heat resistance combined into a single resistance, and an overall heat transfer coefficient, U, of the PHE determined. Phase change processes in heat exchangers have very high value of U due to high thermal conductivities. One of requirements is in ocean thermal energy conservation (OTEC) plants effective heat transfer with minimum pressure loss for small temperature difference between hot and cold fluids (27-33°C).

3. Literature Reviews

Warnakulasuriya and worek [1] investigated heat transfer and pressure drop of a viscous absorbent salt solution in a commercial plate heat exchanger. Overall heat transfer coefficient and Nusselt number are reported to increase with Reynolds number while friction factor decreased. Based on the experimental data, correlations for Nusselt number and friction factor were proposed.

Pinto and Gut [2] developed the optimization method for determining the best configuration of gasketed plate heat exchangers. The main objective is to select the configuration with the minimum heat transfer area that still satisfies constraints on the number of channels, the pressure drop of fluids, the channel flow velocities and the exchanger thermal effectiveness. The configuration of the exchanger is defined by six parameters, which are as follows: the number of channels, the numbers of passes on each side, the fluid locations, the feed positions and the type of flow in the channels. The resulting configuration optimization problem is formulated as the minimization of the exchanger heat transfer area and a screening procedure is proposed for its solution. Gradeck M. et al. [3] performed experiments to study effects of hydrodynamic conditions on the enhancement of heat transfer for single phase flow. These experiments have been conducted for a wide range of Reynolds numbers, [0 < Re < 7500] in order to obtain the different regimes from steady laminar to turbulent. Finally they have pointed out a strong relation between the wall velocity gradient and the Nusselt number. Further investigations will be made on two-phase and boiling flow. Nandan et al. [4] that even at moderate velocities plate heat exchanger can achieve high heat transfer coefficient, low fouling factor etc. Nusselt Number is found to be greatly depending upon the Reynolds Number and it increases with the increase in Reynolds Number.At the different possible conditions various correlations have been proposed for Nusselt Number, Reynolds Number, Prandtl Number, heat transfer coefficient, friction factors etc.

M.faizal and M.R.Ahmed [5] worked on gasket plate heat exchanger by varying the gap between two plates (6mm, 9mm and 12mm) and compare the temperature difference of hot and cold water, the difference of outlet temperature of hot and cold water, average heat transfer, overall heat transfer coefficient and the pressure loss at different flow rate of hot water after keeping the flow rate of cold water constant. They reached out the conclusion that the gap between the plate having 6mm was the most suitable for the applications which have small temperature difference. They also validate their result using CFD codes. Jie Yang .et al.[6] performed their experiment on eight BPHE of different geometry. Ethylene glycol and water mixture is taken as a working fluid. Individual correlations and a general correlation based on experimental data are presented. Then the empirical equations are compared to existing correlations from the open literature. Based on archival data and the experimental data presented in this work, a generalization of single-phase heat transfer performance of plate heat exchangers is given, relating the Nusselt number to heat exchanger geometry, Reynolds number, and fluid properties.

Barzegarian et al.[7] worked on the effect of using TiO_2-water nanofluid on heat transfer enhancement and pressure drop in a Brazed Plate Heat Exchanger used in domestic hot water system is investigated experimentally. TiO_2 nanoparticles with 20 nm diameter and 99+% purity are used for making the nanofluid at 0.3%, 0.8% and 1.5% weight concentration of suspended nanoparticles, in such experiments. This study is done on the brazed plate heat exchanger under turbulent flow condition. The effects of Reynolds number and weight concentration of nanoparticles on the heat transfer characteristics are investigated experimentally. This results in a significant increase in convective heat transfer coefficient through adding nanoparticles to distilled water.

4. Objective:

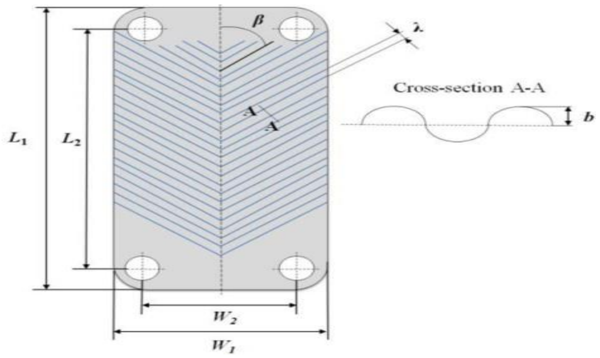
From the previous research carried on heat transfer enhancement, it is obvious that wavy corrugations for plate heat exchangers are an attractive option. On the basis of the above finding, the present work is aimed at experimentally studying the heat transfer characteristics (with pressure drops) for corrugated brazed plate type heat exchangers for use in small temperature difference applications. The results from this work will also be useful for the design of heat exchangers for OTEC applications where the objective is same maximum heat transfer between two fluids having a temperature difference of 20–35 ° C keeping the pressure loss at a minimum. The current design is chosen based on the enhancement of heat transfer characteristics due to the incorporation of wavy configurations in plate exchangers. . The traditional geometry of the wavy configurations is retained to reduce the number of variables in the present work and to study the effect of variable flow rate of hot water.

5. Experimental setup and procedure:

The experiments were carried out in the thermo-fluids laboratory at the B.I.T. Sindri. Experiments were performed on a BPHE single corrugation pattern on 10 plates arranged in parallel. The space between the plates is 2mm. Water was used on both the hot and the cold channels as a working fluid. In the parallel flow arrangement bot the hot and cold fluid entered the BPHE from the bottom. Due to this water fully fill the BPHE channels before existing into the atmosphere. Due to this full area of plate is utilised for the heat transfer and chances for forming hydraulic diameter is prevented. In the parallel flow arrangement hot water interning from Upper port and cold water entering from lower port. We have done our experiment after keeping the cold water constant and varying the hit water flow rate from 2lpm to 6lpm. But here we keep the flow rate of cold water fixed at 2lpm in 1st set of reading and 4lpm in the second set of reading for the both counter flow and parallel flow arrangement. The inlet temperature of both hot and cold water is constant at 65°C and 31°C respectively. This means there was a temperature difference of 34°C created at the inlet of BPHE. The plates used are corrugated galvanized sheets, with a thickness of 0.4mm. The other geometric details of plate and the BPHE is given in the table-2. For maintaining a constant temperature of 65°C in the hot water tank a steam generator is used. A centrifugal pump of rated capacity of 40lpm and a total head of 10m, driven by 0.5 HP variable speed motor is used for hot water flow in BPHE. A digital temperature indicator with a resolution of 0.1°C and a temperature range of - 50°C to +250°C is used for measuring the temperature at inlet and outlet of port of hot and cold water. Two Rotameters of capacity 10lpm were mounted at the inlet of hot and cold water used for measuring the discharge or flow rate of water. The repeatability of the temperature measurement was within 3% and flow measurement is 2%. The accuracy of measurement or estimation of ρ, Cp, μ, k and temperature were taken into consideration for estimating the uncertainty in Q. The value of U is directly found from Q. The maximum error in estimation of Q is found to be 2.4%.

BPHE Geometry details	Dimension
Plate height L_1	208 mm
Plate Width W_1	73 mm
No. of plates	10
No. of hot channels	4
No. of cold channels	5
Total area	0.014 mm^2
Space between plates	2.5 mm
Chevron angle	46.5°
Plate thickness	0.3 mm
Port to port height L_2	173 mm
Port to port width W_2	40 mm

Table-2 : BPHE geometry



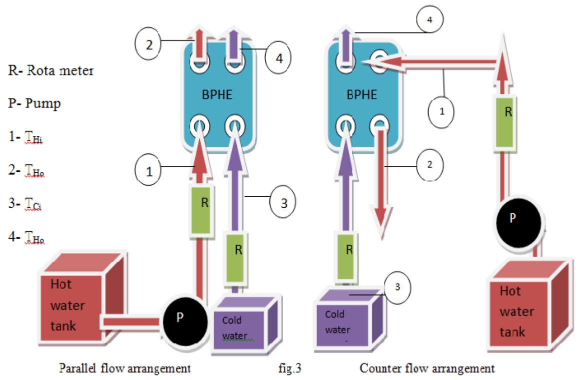


Fig-2: Schematic diagram of setup

6. RESULT AND DISCUSSION:

The results are presented and discussed in this section. Chart-1, 2 shows change in the temperature of hot and cold fluid (i.e. the difference of inlet and outlet temperature of hot and cold fluid.) with varying flow rate of hot water from 2lpm to 6lpm for both counter and parallel flow arrangement.

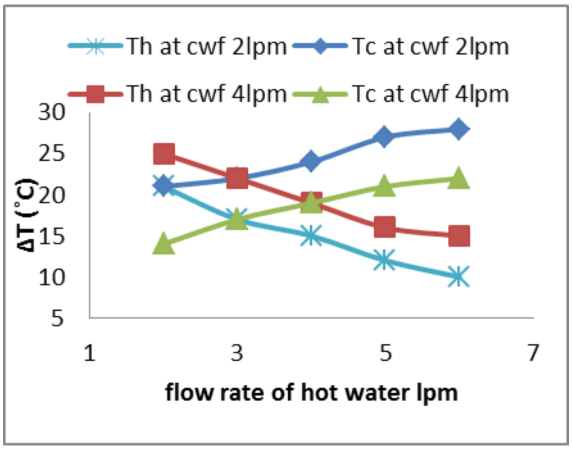


Chart-1: Counter flow arrangement

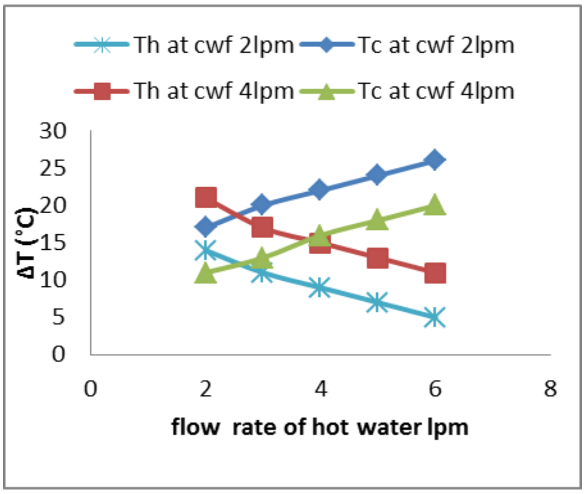


Chart-2: parallel flow arrangement

The ΔT_{hw} decreases with increasing flow rate, and is a minimum at highest flow rate. The ΔT_{hw} is a maximum at the lowest flow rate because hot water gets more time to exchange heat with cold water. When flow rate of cold water increased to 4lpm then temperature difference of cold water stream is lower and hot water stream is higher than that of when cold water flow rate is 2lpm. Similar trends are observed from both counter and parallel flow arrangement. But in parallel flow arrangement lower temperature difference obtained as compare to counter flow arrangement. Now in chart-3 we draw a graph of temperature difference at outlet of BPHE (i.e. the difference between the T_c outlet and T_h outlet) against varying flow rate of hot water. For counter flow arrangement the temperature difference at outlet when the flow rate of cold water is constant at 2lpm is initially decreases and then slightly increases and again decreases but for the for cold water flow rate constant at 4lpm initially it is constant, then decreases and become constant at higher flow rates.

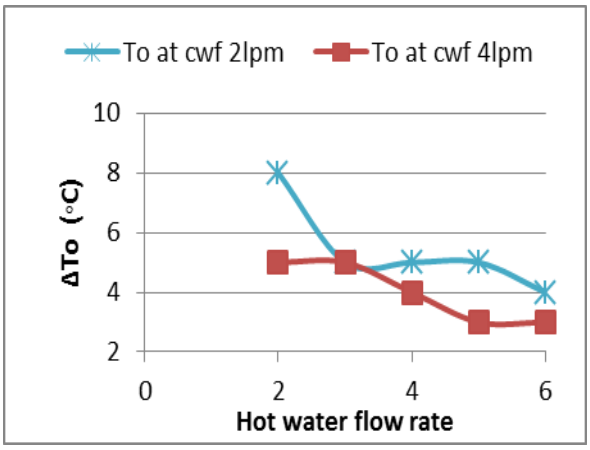


Chart-3: Counter flow arrangement

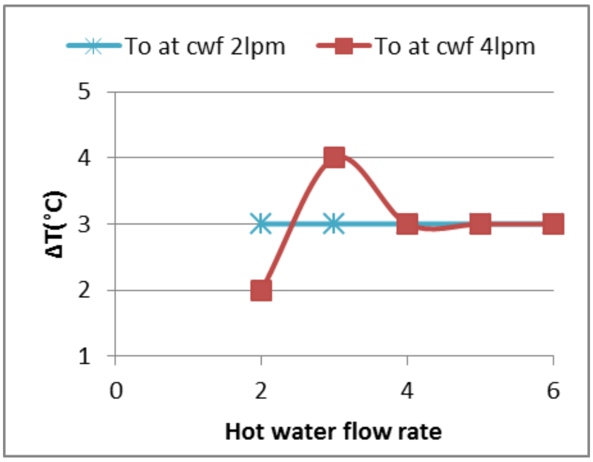


Chart-4: parallel flow arrangement

For parallel flow arrangement it's constant throughout when cold water flow rate constant at 2lpm. But for cold water flow rate constant at 4lpm the initially increases then decreases and become constant. The similar trends found for the gasket plate heat exchanger in the journal of M. Faizal and M.R. Ahmad {13}.

In chart-5,7 the average heat transfer between two streams have shown. We have calculated average heat transfer as:

$\dot{Q}_{hw} = \varrho_{hw} C_{Phw} \dot{V}_{hw} (\Delta T_{hw})$ ……………… (1)

$\dot{Q}_{cw}=\varrho_{cw}C_{Pcw}\dot{V}_{cw}(\Delta T_{cw})$ ……………….. (2)

$\dot{Q}_{Average}=(\frac{\dot{Q}hw+\dot{Q}cw}{2})$ ……………. (3)

Where $\dot{Q}_{cw}$ and $\dot{Q}_{hw}$ are heat transferred by cold and hot water streams respectively. $\dot{Q}_{Average}$ is the average heat transfer between two streams.
As seen from Chart-5,6 $\dot{Q}_{Average}$ increases with increasing the flow rate of hot water. Higher turbulence at higher velocities, cause a much higher heat transfer.

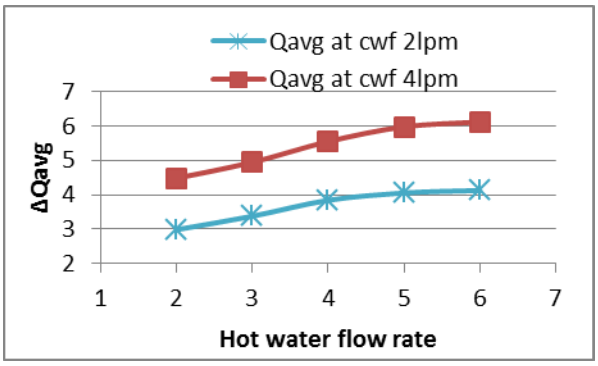


Chart-5: Counter flow arrangement

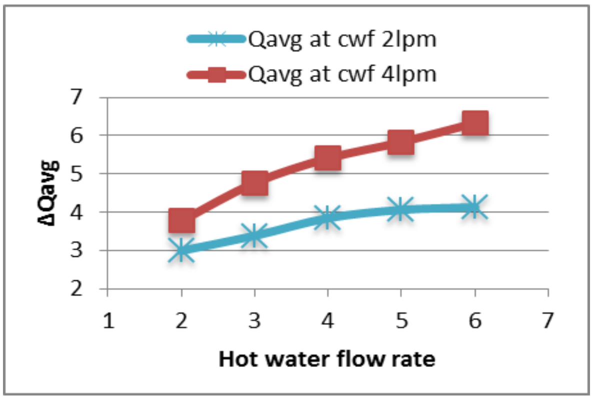


Chart-6: Parallel flow arrangement

Here we have find that for cold water flow at 4lpm have higher value of heat transfer as compare to $\dot{V}_{cw}$ at 2lpm for the both arrangement. But the highest value of $\dot{Q}_{Average}$ is for counter flow arrangement. The variations of overall heat transfer co-efficient, U, for different $\dot{V}_{hw}$ are shown in Chart-7, 8. The U is calculated as:

$$U = \frac{QAverage}{A\Delta Tm}$$

$$\Delta T_m = \frac{(Thwi-Tcwi)-(Thwo-Tcwo)}{\log\frac{(Thwi-Tcwi)}{(Thwo-Tcwo)}}$$ .... For parallel flow arrangement

$$\Delta T_m = \frac{(Thwi-Tcwo)-(Thwo-Tcwi)}{\log\frac{(Thwi-Tcwo)}{(Thwo-Tcwi)}}$$ .... For counter flow arrangement

Where, $\dot{Q}_{Average}$ is the arithmetical mean of $\dot{Q}_{hw}$ and $\dot{Q}_{cw}$. A is the total heat transfer area and ΔT_m is LMTD (logarithmic mean Temperature difference). U takes into account all the resistance that are present in the path of the heat transfer. As shown in Chart-7, 8 U increases with $\dot{V}_{hw}$ for both type of flow arrangements. In parallel flow arrangement there is a slight decrease in value of U, it is due to $T_{LMTD} = T_{AMTD}$. Here values of U is higher because of we have a very small gap between the plates. That's why the flow velocity is higher and more turbulence is created.

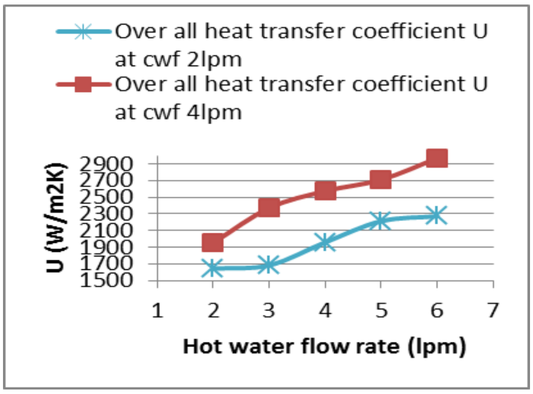


Chart-7: Counter flow arrangement

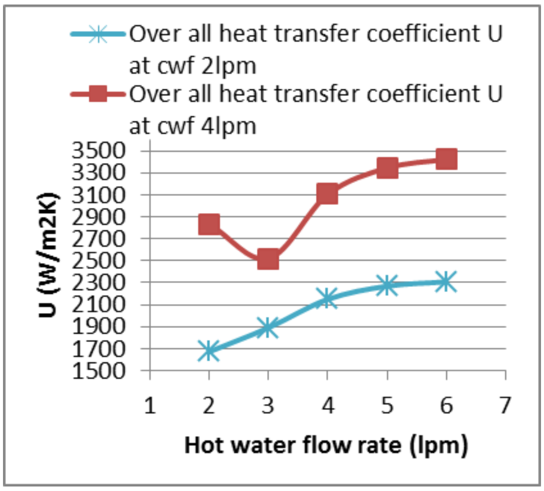


Chart-8: Parallel flow arrangement

A similar trend for U vs $\dot{V}_{hw}$ has been reported in Refs.[11]. Sparrow and Comb found that heat transfer coefficient for larger plate spacing is slightly lower than small plate spacing, but pressure drop was also lower.

Now the variations of thermal length $\theta_{average}$, varying $\dot{V}_{hw}$ are shown in Chart-9,10. Thermal length represents the performance and is the relationship between the temperature difference between one stream and the T_{LMTD}. Higher thermal length means heat transfer and pressure drop is large and at lower thermal length lower the value overall heat transfer coefficient and heat transfer. The thermal length is calculated as:

$$\theta_{hw} = \frac{\Delta T hw}{\Delta T m}$$

$$\theta_{cw} = \frac{\Delta T cw}{\Delta T m}$$

$$\theta_{average} = (\theta_{hw} + \theta_{cw})/2$$

Her θ_{cw} and θ_{hw} are the thermal length of cold and hot water channels respectively, and $\theta_{average}$ is the arithmetic mean of these two. As from Chart-9,10 $\mathbf{\Theta_{average}}$ for parallel flow arrangement is constant for cold water flow rate 2lpm. In other cases initially it decreases and then become constant. If we increase the flow rate then the value of $\mathbf{\Theta_{average}}$ will increase. Smooth plates are not so effective because once the hydraulic boundary layer is fully developed, the central region of the fluids do not receive much heat from the adjacent channel compared to the fluid elements close to the wall. Also, as the wall spacing is increased, the heat received by the central region decreases. In contrast, corrugations on the plate surface lead to continuous disruptions in the boundary layer across the length of a channel from inlet to exit. The secondary flow causes turbulent mixing of the flu- ids in the channels from one wall to another. This allows almost all the fluid elements to have effective heat transfer from adjacent channels.

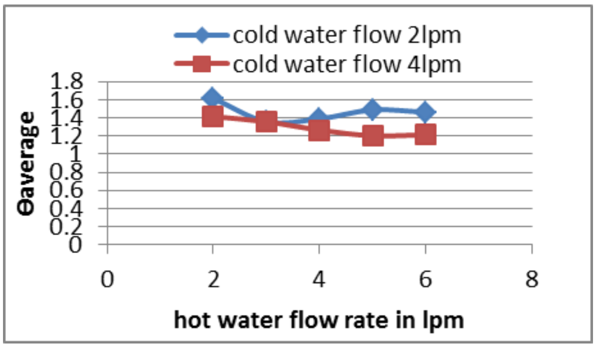


Chart-9: Counter flow arrangement

7. Conclusions:

Heat transfer and various properties in a corrugated plate BPHE with variable hot water flow rate, keeping flow rate of cold water constant at 2lpm and 4lpm have been studied for both parallel and counter flow arrangement, on the basis of temperature measurement at inlet and outlet of brazed plate heat exchanger.

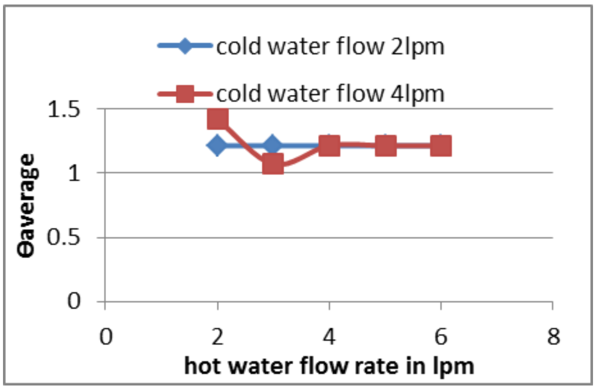


Chart-10: Parallel flow arrangement

We found that at higher flow rate of hot water and cold water $\Delta\dot{Q}_{average}$, U and pressure drop is higher in both arrangements but counter flow arrangement have higher value than parallel flow arrangement. But the average thermal length is almost constant for shorter size of BPHE. It is due to very small channel width. As we increase the size of BPHE the average thermal length will also increase. On the basis of these comparisons we can say that for smaller size BPHE there is not a large difference in results found between counter and parallel flow arrangement. But as the size will increase then counter flow arrangement is better choice at lower flow rate of cold water. The corrugations on the plate surfaces induce secondary flows in the channels and cause turbulent mixing which allows all the fluid elements in a particular channel to have effective heat transfer with the adjacent channels. The results from this work will be useful for the design of heat exchangers for small temperature difference applications, irrespective of whether any phase change is involved or not.

8. Future scope:

Concerning to our experimental work on BPHE, there is still a lot of work to do. We performed our work after taking water as a working fluid, many of fluids and nano fluids can be taken as working fluid for increasing the performance of BPHE. For compact type heat exchanger we can extend our work for two phase flow. Also we can vary the dimensions of BPHE and choose the optimum one for any particular operation. Change of plate material of BPHE for better heat transfer and minimum loss. Experimental investigation by varying the corrugation angle and corrugation pitch of BPHE's plate is not yet done. We can also perform these experiments numerically by using CFD codes.

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