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EXPERIMENTAL INVESTIGATION OF BRAZED PLATE HEAT EXCHANGER (CORRUGATED PLATE) IN SERIES

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ABSTRACT

The main aim behind development and enhancement of heat transfer equipment is to save energy and minimise production investment by minimising the cost energy or material. A good H.E is one that transfer high amount of energy at low cost. Experimental studies were performed on a B.P.H.E (Brazed plate heat exchanger) with corrugation in series. (Two B.P.H.E with 10 plate arrangement in each) having area per plate 0.014 m^2 *. water is used on both the channel with flow being parallel and counter . The hot water flow rate were varied while cold water flow rate is kept constant .it is found that that the average heat transfer between the two liquid increases with increase in hot water flow rate. The corrugation on the plate enhance turbulence at higher velocity which increases heat transfer. The overall heat transfer coefficient U, the temperature difference between two stream at outlet are also present The result of this work would enhance the current knowledge of BPHE with corrugation in series for small temperature difference application and will also help to verify CFD codes.*

Keywords: B.P.H.E, Corrugated Plat , Heat Transfer.

I. INTRODUCTION

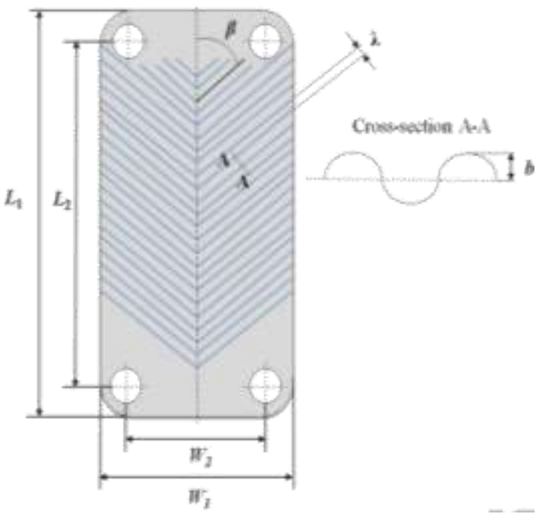
Heat exchanger is a device that transfer heat between two or more media and the transfer of heat is totally dependent on temperature difference of two media without any use of external force or energy some of application of heat exchanger are in air conditioning, refrigeration ,chemical and food industries , power plant , nuclear reactor ,process industries heat recovery units etc. heat exchanger are of many different types like shell and tube (vertical/horizontal), plate heat exchanger (corrugated or flat/ gasket or brazed) ,parallel counter or cross flow type heat exchanger direct contact or open heat exchanger (i .e the two media between which heat is exchanged are in indirect contact e. g cooling tower) ,indirect contact heat exchanger (i. e the two media between which heat is exchanged is separated by wall). In the analysis of heat exchanger all the thermal resistance in the transfer of heat from one fluid to another fluid is combined into single resistance and overall heat transfer heat transfer coefficient U is determined. One of the requirements of OTEC plant is effective heat transfer with minimum pressure loss for small temperature difference between hot and cold fluids (20-25°C).

Pressure loss in heat exchanger will affect pumping power of the pumps in OTEC plants .studies reported by researcher shows that pressure drop increase significantly with flow rate.

NOMENCLATURE

A	heat transfer area (m^2)
M_{cw}	coldwater flow rate(lpm)
M_{hw}	hot water flow rate(lpm)
$Q_{avwrage}$	average heat transfer between hot and cold water(W)
T_{CWO}	outlet cold water temperature(^oc)
T_{HWO}	outlet hot water temperature(^oc)
ΔT_{CW}	temperature change of cold water(^oc)
ΔT_m	logarithmic mean temperature difference(LMTD)
ΔX	plate spacing (mm)
θ	temperature difference at any point
n	no of channel either hot or cold
n_h	no of channel per plate for hot side
C_C	heat capacity of cold water
Cp_{cw}	specific heat of cold water (kj/kg°c)
Cp_{hw}	specific heat of hot water (kj/kg°c)
Q_{cw}	heat transferred by cold water(W)
Q_{hw}	heat transferred by hot water(W)
T_{CWI}	inlet cold water temperature(^oc)
T_{HWI}	inlet hot water temperature(^oc)
U	overall heat transfer coefficient(w/m^2k)
ΔT_{HW}	temperature change of hot water(^oc)
ρ_{cw}	density of cold water(kg/m^3)
ρ_{hw}	density of hot water(kg/m^3)
N_C	total no of channel per plate
n_c	no of channel per plate for hot side
C_h	heat capacity of hot water
C_{min}	lowest value of C_h or C_C

BRAZED PLATE HEAT EXCHANGER (BPHE)



DIMENSION OF BPHE	VALUE
Length of heat exchanger(L_1)	208mm
Port to port length(L_2)	173mm
Width of heat exchanger(W_1)	73mm
Port to port width(W_2)	40mm
No of corrugated plate	10
No of channel(N_C)	9
No of channel for hot side	4
No of channel for cold side	5
Chevron angle(β)	46.5
Plate thickness (t)	0.3mm
Enlargement factor(ϕ)	1.16
Corrugation pitch(λ)	8mm
Effective area or projected area(A)	0.014m^2
Corrugation depth (b)	2mm

Fig 1 Corrugated Plate

Table 1 Plate Dimensions

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BPHE is also known as compact brazed plate heat exchanger. It has similar geometry to gasket plate heat exchanger but the gasket are completely removed. In this unit plates are brazed together with copper or nickel alloy and exchanger is completely sealed and leak tight. BPHE have typical design temperature of -160 to 190°C and design pressure upto 30 bar

II BACKGROUND

Corrugation on plate improves the heat transfer by 20-30%. By enhancing the heat tramafer area and by enhancing turbulence at flow rate. Corrugation on plate also improves the mechanical strength of the plate. Many enhanced geometry are used in plate heat exchanger and their main objective is to increase heat transfer coefficient without much increase in pressure loss.

M.Faizal studied heat transfer and pressure drop in corrugated plate heat exchanger with various plate spacing and various flow rate of hot water. and found that for a given plate spacing with increase in hot water flow rate the average heat transfer between the two stream increases due to high turbulence at high velocity. The overall heat transfer coffecient the pressure loss and the average thermal length are found to increase with increase in flow rate of hot water.

Lyytikainen et al performed numerical study by varying corrugation angle and corrugation length he found that both heat transfer and pressure drop increases as the corrugation angle is increased and he stated that it is not easy to find optimum geometry with low pressure drop and high heat transfer simultaneously.

Warnakulasuriya and worek 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 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.

Rao et al. presented the effect of flow distribution to the channels on the thermal performance of a plate heat exchanger. The study indicates the importance of considering the heat-transfer coefficient inside the channels as a function of flow rate through that particular channel. This eliminates the contradictory proposition of unequal flow rates but an equal heat-transfer coefficient. A wide range of parametric study have been presented, which brings out effects such as those of the heat-capacity rate ratio, flow configuration, number of channels and correlation of heat transfer.

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III. OBJECTIVES

From the previous research carried on heat transfer it is found that wavy corrugated plate is better alternative for heat transfer in comparison to another. On the basic of above finding the present work is aimed at experimentally studying the heat transfer characteristic(without pressure drop) for corrugated plate type heat exchanger for use in small temperature difference. The result of this work will be useful in design of heat exchanger for OTEC application where the objective is same maximum transfer of heat between the two fluid having a temperature difference of 20-25°C with keeping the pressure loss at minimum.. the traditional geometry of wavy configuration is kept that is tit is not altered to reduce the no of variables in the present work and to study the effect of the flow rate . the focus of the experiment is to measure the temperature of two fluids at inlet and exist of heat exchanger and check its variation with flow rate.

IV. EXPERIMENTAL SETUP

The experiments were carried out in the hydrodynamics laboratory of BIT sindri dhanbad Jharkhand experiments were performed on the single corrugation pattern of two BPHE with 10 plate arrangement parallel on each BPHE. The spacing between the plate is 2.5mm. water was used in both the channel i.e hot and cold with parallel and counter flow arrangement. The flow rates $\dot{M}_{hw}$ is varied from 2 lpm to 6 lpm. While the cold water flow rate was kept constant at 2lpm . The inlet temperature of hot and cold fluid was constant at 65 °C and 31 °C throughout the experiment. The plate used are corrugated galvanised stainless steels E316 having plate thickness of 0.3mm the other geometry detail of heat exchanger are provided in Fig. 1and table 1. the hot water is directed into the heat exchanger with the help of pump with the rated capacity of 15L/min at total head of 21 m and driven by 0.5HP variable speed motor. Digital thermometer with a resolution of 0.5°C and a temperature range of 0to 750 °C were mounted at inlet and outlet of heat exchanger. The accuracy of measurements or estimation of ρ ,C_P, $\dot{M}$ and temperature were taken into consideration for the estimation of uncertainty of Q (heat transferred).the fluid exit to the atmosphere from the heat exchanger. The flow rate of a particular stream of water is equally divided in all the channel the pipe which carried water to and to and away from the channel had its ends equally divided this is done to achieve similar velocity and pressure of water in their respective channels. There is 4 channel of hot water and 5 channel for cold water per plate on total for my experimental investigation of 2 BPHE in series there is on total 8 hot water channel and 10 cold water channel.

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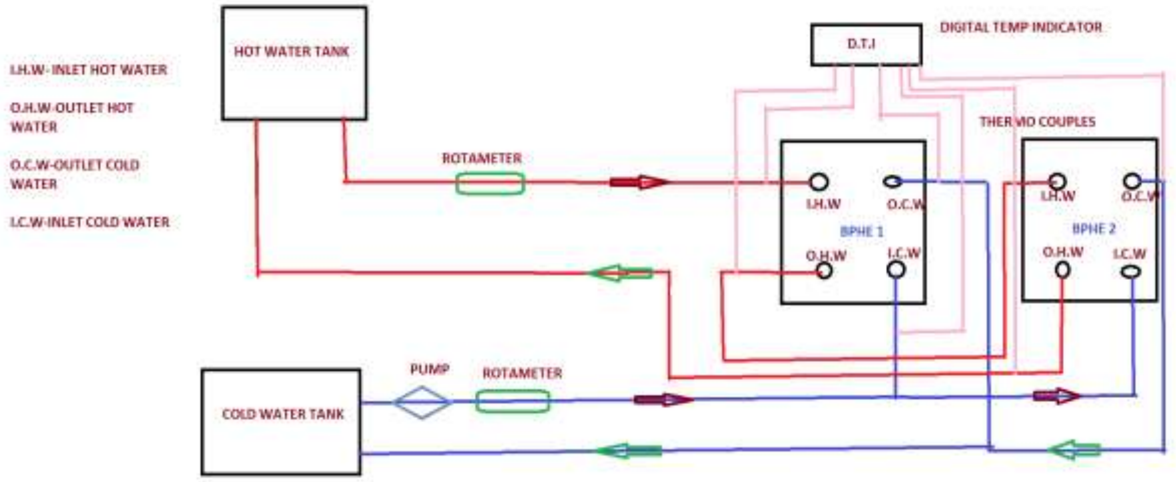

Fig 2. Schematic diagram of my experimental setup

V. OBSERVATION & CALCULATION

The results are presented and discussed in this section. figure 3 shows the change in change in temperature of hot and cold water(i.e. difference of inlet and outlet temperature of respective stream) with varying hot water flow rate. m, for ΔX=2.5mm and M_{CW}.=2LPM(constant). the ΔT_{HW} decreases with increase in flow rate ,and is minimum at highest flow rate. The ΔT_{HW} is maximum at lowest flow rate because hot water get more time to exchange heat with the cold water. The ΔT_{CW} is maximum at the maximum M_{hw} because the hot water stream continuously supplies heat energy to the cold water stream at highest rate without loosing much heat energy. At highest flow M_{HW} the temperature change of hot water from inlet to outlet is is very small. Therefore hot water act as a continous heat source to the cold water.

The average heat transfer between the two stream is shown in fig 5 the heat transfer is calculated as :

$Q_{HW} = \rho_{HW} C_{PHW} M_{HW}(\Delta T_{HW})$

$Q_{CW} = \rho_{CW} C_{PCW} M_{CW}(\Delta T_{CW})$

$Q_{AVERAGE} = ((Q_{HW} + Q_{CW})/2)$

Where Q_{HW} and Q_{CW} are heat transferred by hot and cold water streams respectively. . Q_{AVG} is the average heat transfer between the two streams. Q_{AVG} increases with increase in $M_{HW.}$ Which can be seen in fig 5 for M_{CW} is kept constant at 2LPM..

The variation of the overall heat transfser coefficient U for different U is shown in the fig 6 for M_{CW} is kept constant at 2LPM. The U value is calculated as:

$U = Q_{AVERAGSE}/(A\Delta T_{LMTDMEAN})$

$$\Delta T_{LMTD} = \frac{(\theta_2 - \theta_1)}{\ln(\theta_2/\theta_1)}$$

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Where $Q_{AVERAGE}$ is the arthemetic mean of Q_{HW} and $Q_{CW.}$ A is the total heat transfer area and ΔT_{LMTD} is the logarithmic mean temperature difference .the overall heat transfer coefficient takes into account all the resistance that is present in the path of heat flow. U increases with increases in mass flow rate of hot water .

The variation of average thermal length ,$\theta_{AVERAGE}$ for varying M_{HW} is shown in figure _____. The thermal length represent the performance and relationship between the temperature difference in one stream and LMTD. A higher thermal length means heat transfer is large where as lower thermal length means heat transfer is low . the thermal length is calculated as:

$$\Theta_{HW} = \frac{\Delta THW}{\Delta TLMTD}$$

$$\Theta_{CW} = \Delta T_{CW}/\Delta T_{LMTD}$$

$$\theta_{AVERAGE} = (\Theta_{HW} + \Theta_{CW})/2$$

where θ_{HW} and Θ_{CW} are the average thermal length of hot and cold water channel respectively and $\theta_{AVERAGE}$ is the average thermal length. Average thermal length increases with increases with increase in mass flow rate of hot water .and is shown in fig

Number of transfer unit

NTU is the measure is of effectiveness of heat exchanger.NTU methed or approach facilate the comparison between various type of heat exchanger.

$$NTU = ((N_C-1)AU)/C_{min}$$

Effectiveness

It is defined as the ratio of actual heat transfer to the maximum heat transfer.

$$\epsilon = \frac{actual\ heat\ transfer}{maximum\ heat\ transfer} = (C_h(t_{hwi}-t_{hwo}))/(C_{min}(t_{hwi}-t_{cwi}))$$

Assuming the correlations for both hot-side and cold-side nusselt number can be written as follows, with appropriate values for a and b

$$\mathbf{Nu = a\ Re^{b} Pr^{0.333}} \quad \text{...}\mathbf{1}$$

Further assuming the correlation for the hot-side is identical to the correlation for the cold-side for one BPHE, both the hot-side and the cold-side flows have the same values for the constants *a* and *b*. Neglecting fouling effects, the thermal resistance of the hot-side flow can be expressed as follows:

$$\frac{1}{hhot} = \frac{1}{U} - \frac{t}{K} - \frac{1}{hcold} \quad \text{...}\mathbf{2}$$

where t is the plate thickness, and k_w is thermal conductivity of the plate. Then

substituting Eq. (1) into (2) and using the definition of *Nu*:

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$$\left(\frac{1}{U} - \frac{t}{K}\right) Re_c^{\,b}\, Pr_c^{\,0.333}\, (K_C/D_h) = \frac{1}{a} + \frac{1}{a}\{(Re_c/Re_h)^b (Pr_C/Pr_h)^{0.333}(K_C/K_H)\} \text{ ....3}$$

where D_h is hydraulic diameter. In the present work, the hydraulic diameter is defined as :

$$D_h = 2b$$

where b is the corrugation depth.

. The Reynolds number is defined as:

$$Re = {}^{\rho V D_h}/_{\mu}$$

where   is density, V is the velocity inside a BPHE defined in the following form

$$V = {}^{\dot{M}}/_{\rho n W_1 b}$$

where n is the channel number for the corresponding side.

Eq. (3) can be written as:

$$YY = \frac{1}{a} * XX + \frac{1}{a} \text{ ....4}$$

the coefficient a and the exponent b on Reynolds number are obtained by minimizing the least-squared error of the experimental data with respect to Eq. Now, with a and b available, Eq. (1) can be rewritten as

$$\mathbf{Nu = a\,Re^{\,b} Pr^{0.333} = 0.0748 Re_h^{0.63975} Pr^{0.333}}$$

Number of transfer unit

NTU is the measure is of effectiveness of heat exchanger.NTU methed or approach facilate the comparison between various type of heat exchanger.

$$NTU = ((N_C-1)AU)/C_{min}$$

Effectiveness

It is defined as the ratio of actual heat transfer to the maximum heat transfer.

$$\epsilon = \frac{actual\ heat\ transfer}{maximum\ heat\ transfer} = (C_h(t_{hwi}-t_{hwo}))/(C_{min}(t_{hwi}-t_{cwi}))$$

VI. RESULT AND DISCUSSION

The results are presented and discussed is this section fig 5.1 and 5.2 shows the change in temperature of hot and cold water with varying hot water flow rates, $M_{hw.}$ The temperature change of hot water decreases with increase in flow rates and is minimum at highest flow rate. The ΔT_{HW} is maximum at lowest flow rates because water gets maximum time to exchange heat with the cold water. The ΔT_{CW} is maximum at maximum flow rates of hot water because the hot water stream continuously supplies heat energy to the cold water stream at a highest

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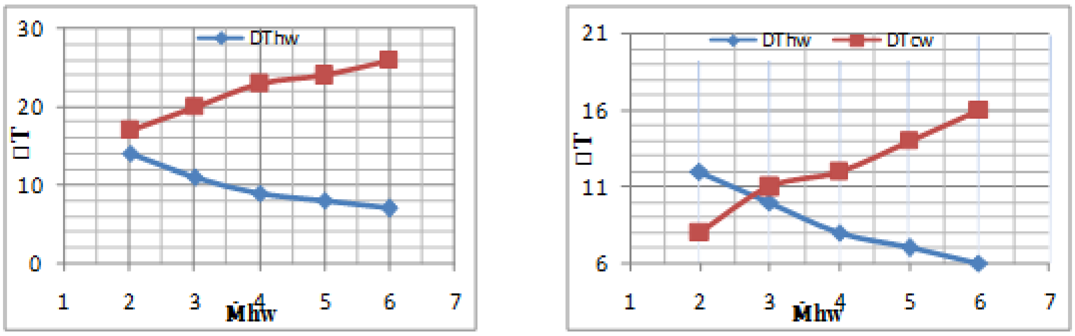
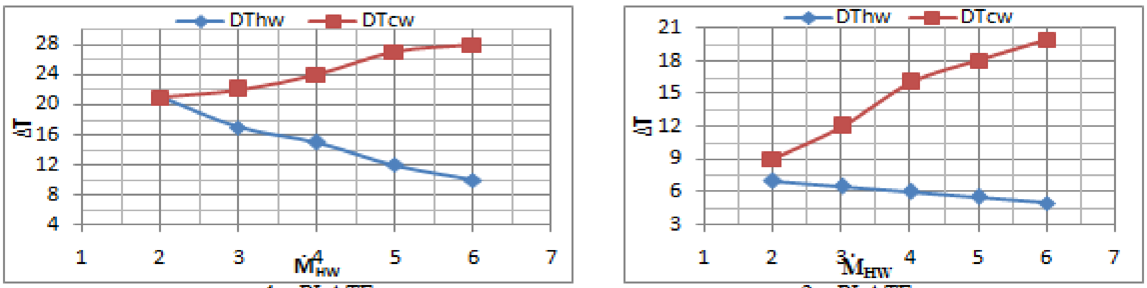
flow rate without losing much energy. At higher M_{hw} the temperature change of hot water from inlet to outlet is very small. Therefore hot water act as a continuous Heat source to cold water stream

Fig 3 experimental setup



1st Plate **2nd Plate**

Fig: 4 A temperature change of fluids (difference of inlet and outlet temperature of respective streams) with increase in flow rate of hot water for PARALLEL FLOW



1_{ST} PLATE 2_{ND} PLATE

Fig :5 A temperature change of fluids(difference of inlet and outlet temperature of respective streams) with increase in flow rate of hot water for COUNTER FLOW

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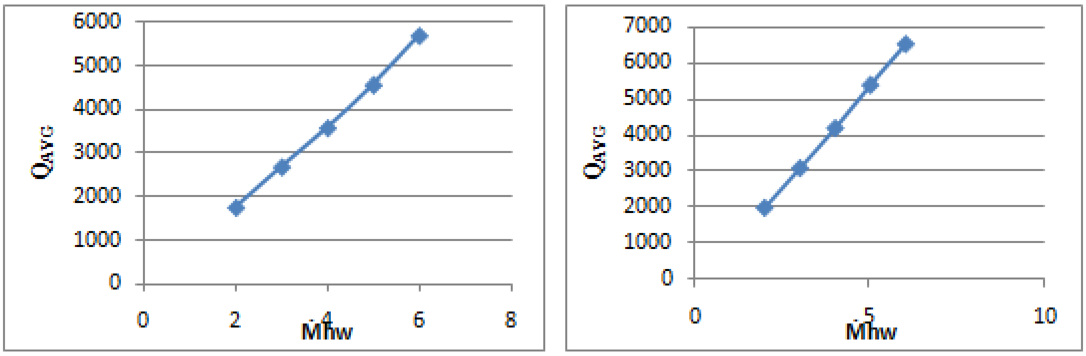

(a) Parallel flow (b) counter flow

Fig 6 The average heat transfer between hot and cold fluid hot and cold water at varying mass flow rate of hot water

The average heat transfer coefficient between two stream is shown in figure 5.3.the heat transfer is calculated as Where Q_{HW} and Q_{cw} are heat transferred by hot and cold water stream respectively. Where Q_{AVG} is the average heat transfer between the two streams.

Parallel **counter**

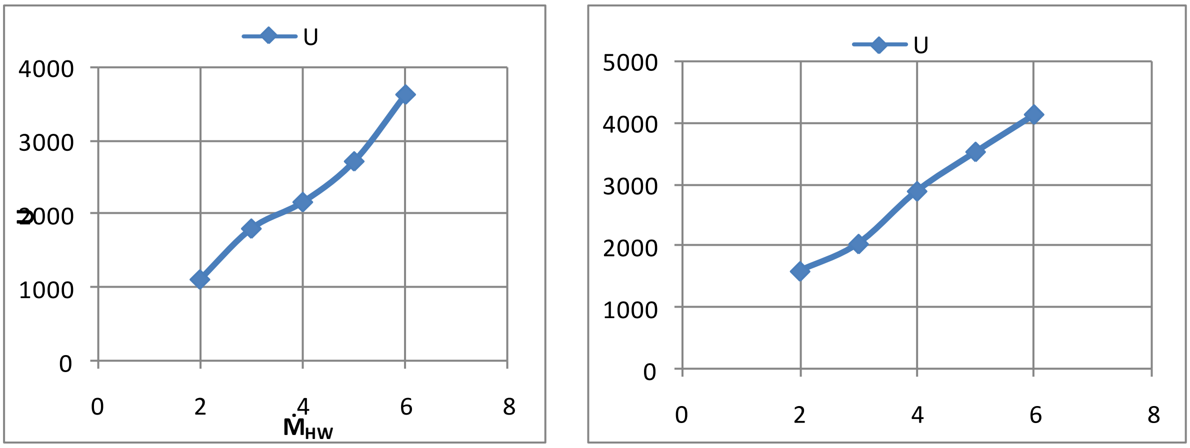

Fig 7 shows the variation of overall heat transfer coefficient with mass flow rate of hot water.

The overall heat transfer coefficient takes into account all the resistance that is present in the path of heat transfer. the overall heat transfer increases with increase in mass flow rate of hot water because of increment difference of hot fluid temperature and decrement in logarithmic mean temperature difference.

Variation of friction factor with renonlds number

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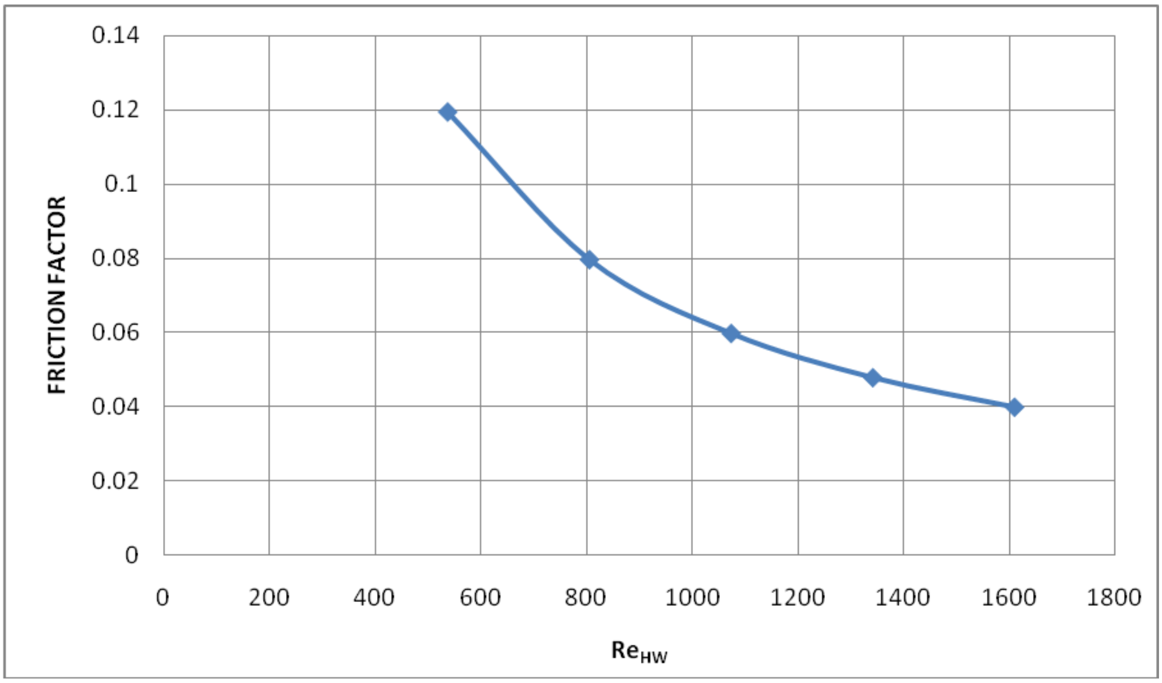

Fig 8 shows the variation of friction of friction factor with reynold number.

Figure 8 shows friction factor of hot water decreases with increase in reynold number .while the friction factor of cold water is 0.2683 constant as the cold fluid velocity is kept constant at 2 LPM, having reynold no 238.1454 The friction factor of cold water is higher than that of hot water this is because of viscocity effect.

Effect of NTU on effectiveness

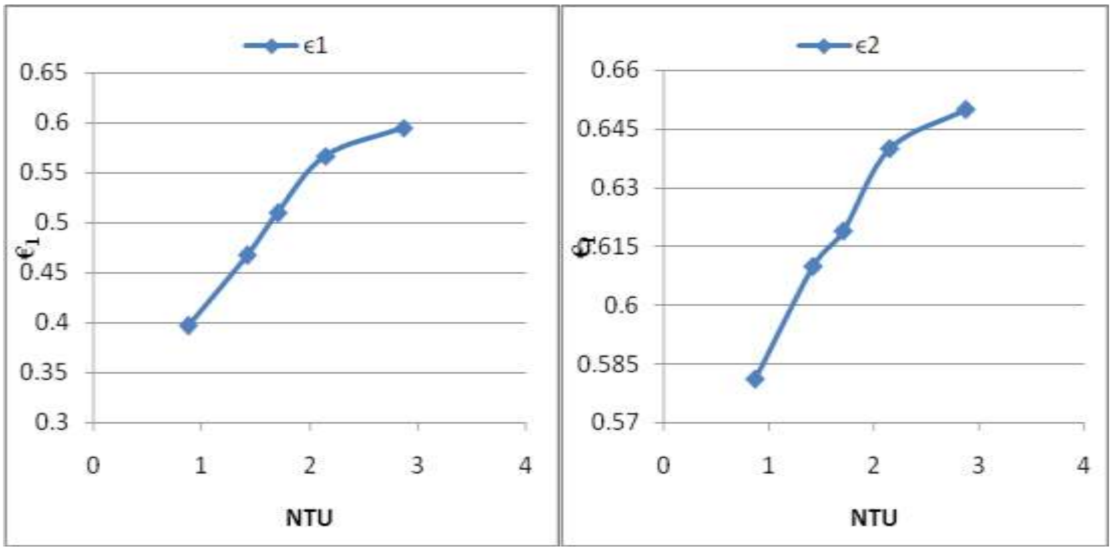

PLATE 1 **PLATE 2**

Fig 9 shows variation of NTU with effectiveness for parallel flow.

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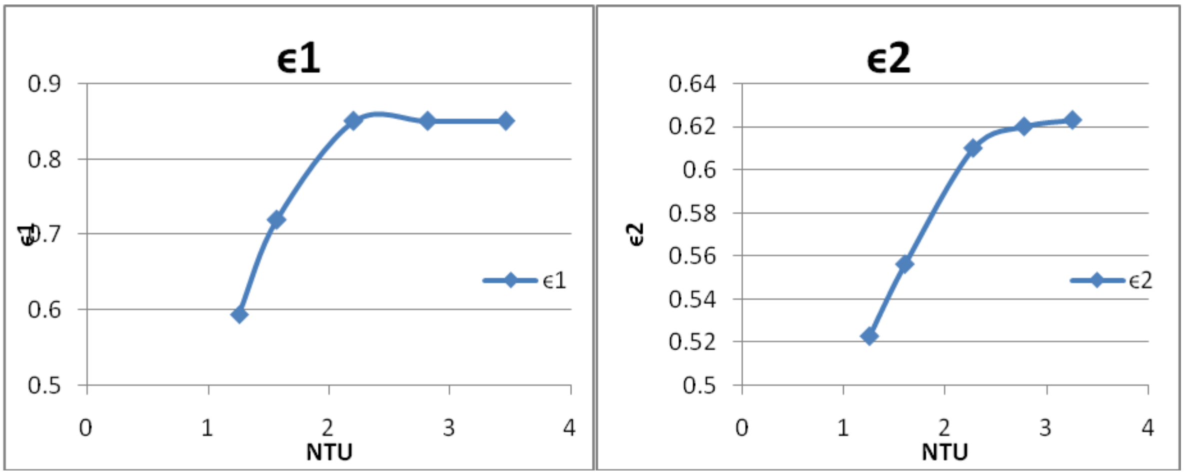

Fig 10 shows the variation of NTU and effectiveness for counter flow

Fig 9and 10 shows the variation of effectiveness with number of transfer unit in bphe for parallel and counter flow .it shows that effectiveness increases with increase in number of transfer unit due to increase in flow rate resulting in increase in overall heat transfer coefficient.

VI. CONCLUSION

The main focus of the experiment is to investigate experimentally the performance of brazed plate heat exchanger in series with counter and parallel flow arrangement with regard to heat exchange effectiveness , convective heat transfer coefficient overall heat transfer coefficient ,mass flow rate, and Reynolds number. The following are the result of experimental investigation.

- Convective heat transfer coefficient increases with increase in mass flow rate, also overall heat transfer coefficient increases with increase in Reynolds number. This can be attributed to more turbulent flow.
- Nusselt number increases with increase in Reynolds number because of higher flow rates in the channel. The higher flow rates promote the good mixing of fluid in the channel resulting in enhancement of the convective heat transfer.
- A Correlation has been developed among Nusslet Number, Reyonlds Number , and Prandlt Number. the predicted results obtained from the developed correlations are also in reasonable agreement with the experimental data
- Overall heat transfer coefficient increases with increase in mass flow rate. the mass flow rate of cold water is constant while the mass flow rate of hot water varies.
- Effectiveness of plate heat exchanger increases with number of transfer unit (NTU)
- Friction factor for cold tap water flow in channels is substantially higher than hot distilled water flow in their respective channels due to the viscosity effect

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VII FUTURE SCOPE

- The main aim of this investigation is to enhance the different types of thermal as well as hydraulic performances like effectiveness, heat transfer coefficient by using brazed plate heat exchanger . It could be extended to uses by using multipass brazed plate heat exchanger or by varying the chevron angle ,or by varying the spacing between the plates.
- In this experiment as water is used as the working fluid other fluids like oil, air, nano fluid and two or three different types of fluids may be used for investigate and compare the different performances.

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