Experimental Investigation of single phase Chevron Type Gasket Plate Heat Exchanger

Vishal R. Naik, V.K. Matawala

Abstract*—corrugated plate heat exchangers have larger heat transfer surface area and increased turbulence level due to the corrugations. In this study, experimental heat transfer data will obtain for single phase flow (oil-to-water) configurations in a corrugated plate heat exchanger for different chevron angle plates. The effect of variation of chevron angles with other geometric parameter on the heat transfer coefficient will be study. Reynolds number ranging from 50 to 10000 and Prandtl number ranging from 3 to 75 will be taken for given experiment. Based on the experimental data, a correlation will estimate for Nusselt number as a function of Reynolds number, Prandtl number and chevron angle.

***Index Terms*—Chevron angle, Corrugated plate heat exchangers, Heat transfer coefficient, Nusselt number, Prandtl number, Reynolds number, Single phase flow.**

I. INTRODUCTION

Plate-type heat exchangers are usually built of thin plates (all prime surfaces). The plates are either smooth or have some form of corrugation, and they are either flat or wound in an exchange. Generally, these exchangers cannot accommodate very high pressures, temperatures, or pressure and temperature differences. Plate heat exchangers (PHE) can be classified as gasket, welded or brazed depending on the leak tightness required. [1]

A. Plate Heat Exchanger

Gasketed plate heat exchangers (PHEs) are widely used in dairy and food processing plants, chemical industries, power plants and central cooling systems. They exhibit excellent heat transfer characteristics, which allows a very compact design, and can be easily dismounted for maintenance, cleaning or for modifying the heat transfer area by adding or removing plates.

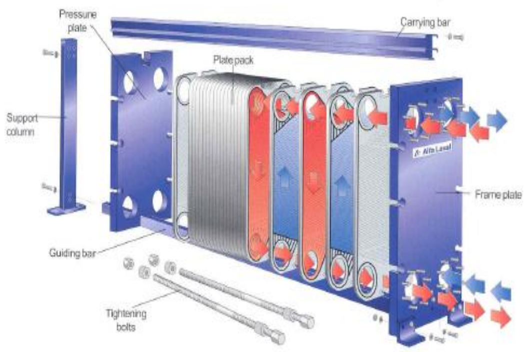

Fig. 1 Gasket plate heat exchanger assemblage and parts

The PHE consists of a pack of thin corrugated metal plates with portholes for the passage of the fluids, as shown in Fig. 1.Each plate contains a bordering gasket, which seals the channels formed when the plate pack is compressed and mounted on a frame. The hot and cold fluids flow in alternate channels and the heat transfer takes place between adjacent channels. The corrugation of the plates promotes turbulence inside the channels and improves the mechanical strength of the plate pack.

B. Working Principle

Channels are formed between the plates and the corner ports are arranged so that the two media flow through alternate channels. The heat is transferred through the plate between the channels, and complete counter-current flow is created for highest possible efficiency. The corrugation of the plates provides the passage between the plates supports each plate against the adjacent one and enhances the turbulence, resulting in efficient heat transfer.

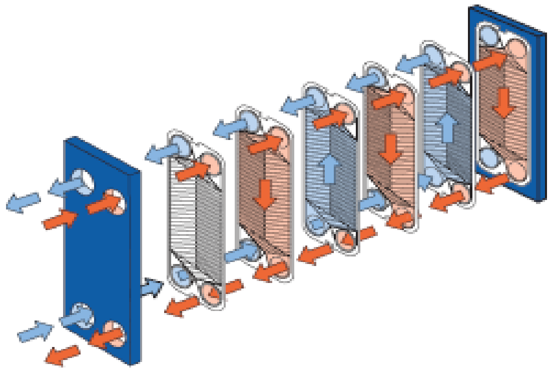

Fig. 2 Fluid Flow in Plate Heat Exchanger

When a package of plate are pressed together, the holes at the corners form continuous tunnels or manifolds, leading the media from the inlet into the plate package, where they are distributed into narrow passages between the plates. Because of the gasket arrangement on the plates, and the placing plates alternate passages, e.g. the warm liquid between even number passages. Thus the media are separated by a thin metal wall. In most cases the liquids flow in opposite directions. The warmer medium will give some of its heat energy to the thin wall, which instantly looses it again to the colder medium on the other side. The warmer medium drops in temperature, while the colder one is heated up. Finally, the media are led into similar hole-tunnels at the other end of the plates and discharged from the heat exchanger.

C. Basic Component of Gasket Plate Heat Exchanger

The basic elements of the gasket plate heat exchanger are plate, gasket, frame and connectors.

(1) PLATES

The basic elements of the plate pack is plate, a sheet of metal, precision pressed into a corrugated pattern as shown in

Manuscript received on April, 2013.
Vishal R. Naik, P.G student at Shri S'ad Vidya Mandal Institute of Technology, Bharuch, India.
Dr. V. K. Matawala, H.O.D. at Shri S'ad Vidya Mandal Institute of Technology, Bharuch, India.

Fig. 3. The largest single plate is of the order of 4.3 m × 1.1 m wide. The heat transfer area for a single plate lies in the range 0.01 to 3.6 m2. It should be assured that the fluid is equally distributed over the full width of the plate. To avoid poor distribution of the fluid across the width, the minimum (length/width) ratio is of the order of 1.8. Plate thickness range between 0.5 and1.2 mm and are spaced with nominal gaps of 2.5 to 5 mm, yielding hydraulic diameter for the flow channels of 4 to 10 mm. The most common materials are shown below on table.

I. PLATE MATERIALS [2], [3], [5]

Material	**Thermal conductivity(w/m^{2}k)**
Stainless steel (316)	16.5
Titanium	20
Inconel 600	16
Incolay 825	12
Hastelloy C-276	10.6
Monel 400	66
Nickel 200	66
9/10 Cupronickel	52
70/30 Cupronickel	35

A wide range of corrugation types are available in practical applications shown in figure. Although most modern plate heat exchangers are the chevron type.

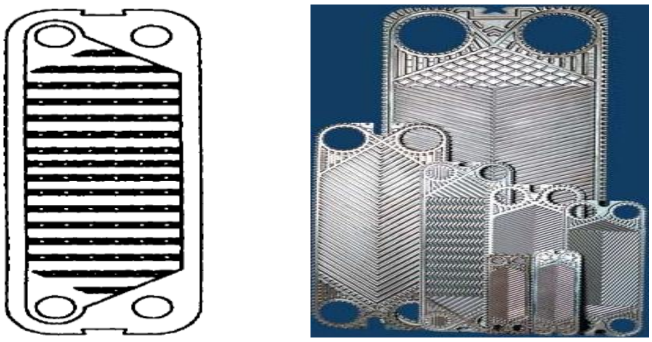

Fig. 3 Two most widely used corrugation types. (a) Intermating troughs or washboar pattern (b) chevron or herringbone pattern.

In the chevron type, adjacent plates are assembled such that the flow channel provides swirling motion to the fluids; the corrugated pattern has an angle β that is referred to as Chevron angle. In the herringbone pattern, the corrugations are pressed to the same depth as the plate spacing. The chevron angle is reversed on adjacent plates so that when the plates are clamped together the corrugations cross one another to provide numerous contract points. The herringbone type therefore has greater strength than the washboard type, which enables it to withstand higher pressures with smaller plate thickness. It is the most common type in use today. The contact arrangements of the plates are shown in Fig. 4. The corrugation depth generally varies from 3 to 5 mm.

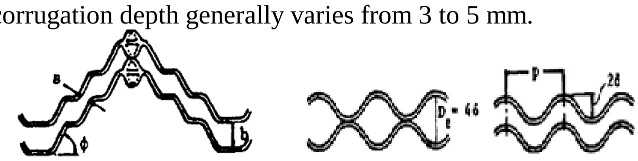

Fig. 4 Cross-section of the (a) Intermating troughs and (b) and (c) chevron troughs

In the intermating-type plate shown in Fig. 4.Is based on corrugations that are pressed to a depth greater than the compressed gasket depth. When the machine is closed, the corrugations fit into one another. A cross section at right angles to the flow direction is shown in Fig 4. This gives a constant change in direction and cross-sectional area in the direction of flow, so that turbulence is induced by a continuing variation in liquid velocity. The maximum flow gap b varies typically from 3 to 5 mm, with the minimum flow gap c being between 1.5 and 3 mm. Liquid velocities in the turbulent regime range from approximately 0.2 to 3 m/s depending on the pressure drop required.[4], [5]

(2) GASKETS

The leakage from the channels between the plates to the surrounding atmosphere is prevented by the gasketing around the exterior of the plate as shown in fig.1.4.3. The two media are effectively kept apart by the ring and field gaskets. To prevent intermixing of the media in the corner areas where field and ring gaskets are very close to each other, the link pieces have a number of slots opening the area between the field and ring gaskets to atmosphere. Any leakage of media across either gasket therefore escapes from the heat exchanger through the slots.

It is important that these openings are not permitted to plug. If that should happen, there is a risk that in case of a leakage in that region of the plate, there might be a local pressure buildup, which could let one medium leak over and mix the other. The below figure show such opening.

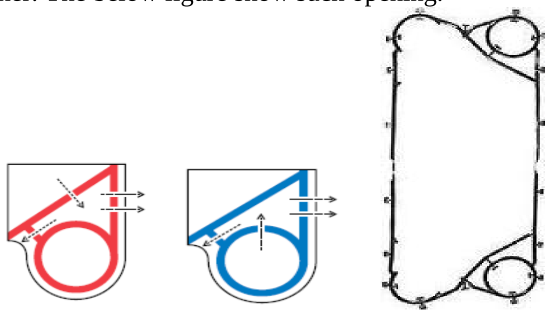

Fig 5.Shows the Different Arrangement of Gasket on P.H.E.

When selecting the gasket material, the important requirements to be met are chemical and temperature resistance coupled with good sealing properties and shape over an acceptable period of life. Much work has been done to develop elastomer formulations that increase the temperature range and chemical resistance of gaskets. Typical gasket materials and their maximum operating temperature are given in Table.

II. GASKET MATERIALS USED IN PLATE [4], [5]

Gasket Material	Maximum operating temperature	Application
Natural rubber	70	Oxygenated solvent, acids, alcohols
SBR (styrene butadiene)	80	General-purpose aqueous, alkalies, acids and oxygenated solvent
Neoprene	70	Alcohols, alkalies, acids, aliphatic hydrocarbon solvents
Nitrile	100-140	Dairy, fruit juices, pharmaceutical and biochemical applications, oil,

		gasoline, animal and vegetable oils, alkalies.
Butyl	120-150	Alkalies, acids, animal and vegetable oils, phenols, and some esters.
Ethylene propylene rubber	140	Alkalies, oxygenated solvents
Silicone rubber	140	General low temperature use, alcohols, sodium hypochlorite.
Fluorinated rubber	175	High temperature aqueous solutions, mineral oils and gasoline, organic solvents, animal and vegetable oils.

II. LITERATURE REVIEW

F.Akturk et al. [6] carried out experimental investigation to find characteristics of a chevron type Gasket plate heat exchanger. They are measure the temperatures, volumetric flow rates at all ports and the pressure drops between inlet and outlet ports at different channel Reynolds numbers (450-5250).In their study, different number of plates with city water as the working fluid for both hot and cold sides is utilized. Results are used to develop new correlations for the heat transfer coefficient and the friction factor for pressure drop calculations for the chevron plates tested. Obtained correlations are compared with correlations in literature.

Masoud Asadi et al. [7] studied the thermodynamically optimization a plate and frame heat exchanger for micro turbine applications. In this study analysis the manual designing the plate and frame heat exchanger with Thermal Analysis and Thermodynamic Optimization is calculated and compare these results with the plate-fin heat exchanger. The results show the higher performance of plate and frame heat exchanger, after modifying mass flow rates based on thermodynamical optimization, outlet air temperature have increased about 6 ^{o}C.

Bobbili Prabhakara Rao et al. [8] carried out experimental investigation to find the flow and the pressure difference across the port to channel in plate heat exchangers for a wide range of Reynolds number,(1000-17000). The port flow maldistribution is caused by port pressure variation in a given pass so that the flow distribution among channels is determined by pressure profile at the inlet port, outlet port and in the channels. They are recorded overall pressure drop for various flow rates and the results indicated that the flow maldistribution increases with increasing overall pressure drop in plate heat exchanger.

Warnakulasuriya and worek [9] 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.

A Bhanu Prakash et al.[10] taken the plant data available for the existing shell and tube heat exchanger it is found that the approach temperature is of The order of 16 ^{o}C, which is considerably high and the heat recovery is low. With the existing plant data, calculations are made to obtain a better heat recovery by selecting a corrugated plate heat exchanger for low approach temperatures ranging from 2 to 6 ^{o}C with different chevron angles of 300, 450 and 500. By going through the economic analysis it is concluded that, the maximum savings result around an approach temperature of 4.5 ^{o}C, with a chevron angle 300, when a plate heat exchanger with an area of 750 m^{2} is operated with the plant data. so the results show that plate type heat exchanger have the advantage over the shell and tube heat exchangers for the heat recovery, as large areas can be provided in smaller space.

L.Wang & B.Suden [11] developed design method of plate heat exchanger with and without pressure drop specifications. In the case of the design with pressure drop specification, only one stream can fully utilize the allowable pressure drop. In the case of no pressure drop specification, allowable pressure drops can be determined through economical optimization. Compared to the previous design method, the proposed method does not require many trial iterations. Instead, all heat exchanger parameters, including plate size, number of passes, path, fluid velocity, etc., are determined in a straightforward way.

Mohammad S. Khan et al. [12] performed experimental investigation of evaporation heat transfer and pressure drop of ammonia in a 30^{o} chevron plate heat exchanger. They are used U- type counter flow arrangement with liquid ammonia evaporating in upward flow direction and glycol/ water solution simulating the heat load in counter flow. Two counter flow channels were formed in the plate heat exchanger by three chevron plates of commercial geometry. Saturation temperature of ammonia was varied from -25 ^{o}C to -2 ^{o}C. The heat flux ranged from 21 kW m^{-2} to 44s kW m^{-2} while equivalent Reynolds number is varied from 1225 to 3000. The results show significant effects of saturation temperature, heat and exit vapor quality on heat transfer coefficient and pressure drop. Two phase Nusselt number and friction factor correlations are proposed.

P. Narataruksa & R. Ponpai [13] proposed the new thermal design structure of three- stream plate and frame heat exchangers with two thermal Communications. They are used six port plates with one hot stream transfers heat to two colder streams in which there is no direct heat transfer between two cold streams. A new thermal design methodology and design of heat recovery between the streams are developed. To estimate the required number of channels for each stream, the relationship of the heat exchanger effectiveness and the number of transfer unit for counter flow can be employed, in which the constraint of uniform U.

A.A. Fahmy et al. [14] studied the performance of heat exchanger under normal and abnormal operational conditions are of great importance relevant to the economic and operational safety in power plants. Fouling and scale formation in heat exchangers could have serious impacts on the operating conditions of the nuclear reactors. This study aims at the simulation of fouling crystallization process in plate-type heat exchanger in MTR reactor by developing an Engineering Equation Solver (EES) Program. The finding of this work would enable us to evaluate the thickness and fouling rate in plate-type heat exchanger in MTR reactor. The crystallization fouling of calcium sulphate (CaSO4) in plate heat exchanger was also investigated. Also, the effect of fluid velocity on fouling resistance and the rate of deposit thickness were studied in each of primary and secondary circuits.

J.M.Pinto & Jorge.A.W.Gut [15] 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.

Khan T. S. et al. [16] carried out experiment for single phase flow [water-to-water] configurations in a commercial plate heat exchanger for symmetric 30˚/30˚, 60˚/60˚, and mixed 30˚/60˚ chevron angle plates having Reynolds number ranging from 500 to 2500 and Prandtl number from 3.5 to 6.5. Based on the experimental data, a correlation to estimate Nusselt number as a function of Reynolds number, Prandtl number and chevron angle has been proposed.

Using the Buckingham Pi theorem, Lin J.H. et al. [12] derives dimensionless correlations to characterize the heat transfer performance of the corrugated channel in a plate heat exchanger. The experimental data are substituted into these correlations to identify the flow characteristics and channel geometry parameters with the most significant influence on the heat transfer performance.

III. EXPERIMENTAL ANALYSIS

Material required for plate heat exchanger parts:

- Plate material - 316 stainless steel
- Gasket material –Nitrial Butadiene Rubber (NBR)
- Nozzle material - 316 stainless steel

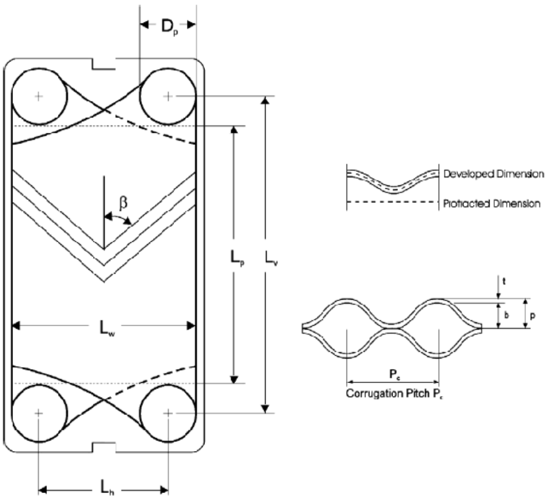

Fig. 6 Basic geometry of chevron plate

III. GEOMETRY OF CHEVRON PLATES IN PRESENT STUDY

Plate width, L_w (mm)	95
Vertical distance between centers of ports, L_v (mm)	357
Port diameter, D_p (mm)	35
Plate thickness, t (mm)	0.5
Effective area of single plate, A (m^2)	0.0357
Corrugation Pitch, P_c (mm)	2.9
Surface enlargement factor, φ	1.167

A. *Geometric Parameter Affecting Plate Heat Exchanger*

Chevron Angle, β: Typically varying from 20° to 65°, β is the measure of softness (small β, low thermal efficiency and pressure drop) and hardness (large β, high thermal efficiency and pressure drop) of thermal and hydraulic characteristics of plates. Some authors define "Π/2- β" as the chevron angle.

Surface Enlargement Factor, φ: φ is the ratio of developed area [based on corrugation pitch, Pc, and plate pitch, p] to the projected area (viz. $L_w \times L_p$, $L_w = L_h + D_p$ and $L_p = L_v - D_p$).

Corrugation Depth or Mean Channel Spacing, b: b = p–t, the difference between plate pitch, p and the plate thickness, t

Channel Flow Area, A_{ch}: Ach is the minimum flow area between plates and is estimated as product of plate corrugation depth and width of plate (i.e., $A_{ch} = b \times L_w$)

Channel Hydraulic Diameter, D_h: D_h is defined as four times ratio of minimum flow area to wetted perimeter,

$$D_h = \frac{2b}{\varphi} \quad \text{Since } b \ll L_w.$$

B. *Physical Parameters Affecting Plate Heat Exchanger*

The six most important parameters are as follows:

- The amount of heat to be transferred (heat load).
- The inlet and outlet temperatures on the primary and secondary sides.
- The maximum allowable pressure drop on the primary and secondary sides.
- The maximum operating temperature.
- The maximum operating pressure.
- The flow rate on the primary and secondary sides.

Temperature Program: This means the inlet and outlet temperatures of both media in the heat exchanger.

Heat Load (Q): Disregarding heat losses to the atmosphere, which are negligible, the heat lost (heat load) by one side of a plate heat exchanger is equal to the heat gained by the other. The heat load is expressed in kW or kcal/h.

Logarithmic Mean Temperature Difference: Logarithmic mean temperature difference (LMTD) is the effective driving force in the heat exchanger.

Density: Density (ρ) is the mass per unit volume and is expressed in kg/m^3 or kg/dm^3.

Flow Rate: This can be expressed in two different terms, either by weight or by volume. The units of flow by weight are in kg/s or kg/h, the units of flow by volume in m3/h or l/min. To convert units of volume into units of weight, it is necessary to multiply the volume flow by the density.

Pressure Drop: Pressure drop (Δp) is in direct relationship to the size of the plate heat exchanger. If it is possible to increase the allowable pressure drop, and incidentally accept higher pumping costs, then the heat exchanger will be smaller and less expensive. As a guide, allowable pressure drops between 20 and 100 kPa are accepted as normal for oil/water duties.

Specific Heat: Specific heat (c_p) is the amount of energy required to raise 1 kg of a substance by one degree centigrade. The specific heat of oil at 70 ^{o}c is 2.05 KJ/kg ^{o}c and for water specific heat at 32 ^{o}c is 4.18 KJ/kg ^{o}c.

Viscosity: Viscosity is a measure of the ease of flow of a liquid. The lower the viscosity, the more easily it flows. Viscosity is expressed in centipoises (cP) or centistokes (cSt).

For hot side is 4.62 cP and for cold side 0.767 cP.

Overall Heat Transfer Coefficient: Overall heat transfer coefficient (U) is a measure of the resistance to heat flow, made up of the resistances caused by the plate material, amount of fouling, nature of the fluids and type of exchanger used. Overall heat transfer coefficient is expressed as W/m^2 °C or kcal/h, m^2 °C.

C. Heat Transfer Methodology

$$Q_h = m_h \times c_{ph} (T_{h1} - T_{h2}) \quad (1)$$

$$T_{c2} = \frac{Q_h}{m_c c_{pc}} + T_{c1} \quad (2)$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}} \quad (3)$$

Here, $\Delta T_1 = T_{h1} - T_{c2}$ & $\Delta T_2 = T_{h2} - T_{c1}$

Channel Reynolds number can be considering to characterize the flow dominantly which is found by channel mass velocity, equivalent diameter and dynamic viscosity.

$$G_{ch} = \frac{\dot{m}}{bN_{cp}L_w} \quad \& \quad D_e = 2b \quad (4)$$

Where, Number of channel per pass;

$$N_{cp} = \frac{N_t - 1}{2N_p} \quad (5)$$

Hot and Cold fluid Reynolds numbers are

$$Re = \frac{G_{ch} D_e}{\mu} \quad (6)$$

j_{Nu} Nusselt-type j- factor for hot side,

$$j_{Nu} = \left\{0.335 - 0.105\sin\left[3.8(\beta - 41)\right]\right\} R_e^{0.6} \quad (7)$$

Hot Fluid heat transfer coefficient, h_h

$$h_h = j_{Nu} \left(\frac{K}{D_e}\right) P_r^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.17} \quad (8)$$

Overall heat transfer coefficient [U]:

$$\frac{1}{U} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{t}{k_w} \quad (9)$$

The friction factor is defined by the following equation.

$$f = \frac{k_p}{R_e^{\,m}} \quad (10)$$

Total pressure drop for both sides is the channel pressure drop and port pressure drop

$$(\Delta p_t)_h = (\Delta p_c)_h + (\Delta p_p)_h \quad (11)$$

$$(\Delta p_t)_c = (\Delta p_c)_c + (\Delta p_p)_c \quad (12)$$

Where, pressure drop of channel $[\Delta p_c]$

$$(\Delta p_c) = 4.f.\frac{L_v.N_p}{D_e}.\frac{G_c^2}{2\rho} \quad (13)$$

The pressure drop in port ducts; $[\Delta p_p]$

$$(\Delta p_p) = 1.4.N_p.\frac{G_c^2}{2\rho} \quad (14)$$

IV. EXPERIMENTAL RESULTS

Fig. 7 Experimental setup

The experimental analysis is to be done in this present work. Take the reading from the experiment, and to be measured the different mass flow rate of cold side as well as hot side and also measured the temperature of hot side as well as cold side.

Considering the temperature of oil Th1 = 70oc, outlet temperature of oil Th2 = 52.5oc and inlet temperature of water Tc1 = 32.1oc, outlet temperature of water Tc2 = 36.53 is keep constant.

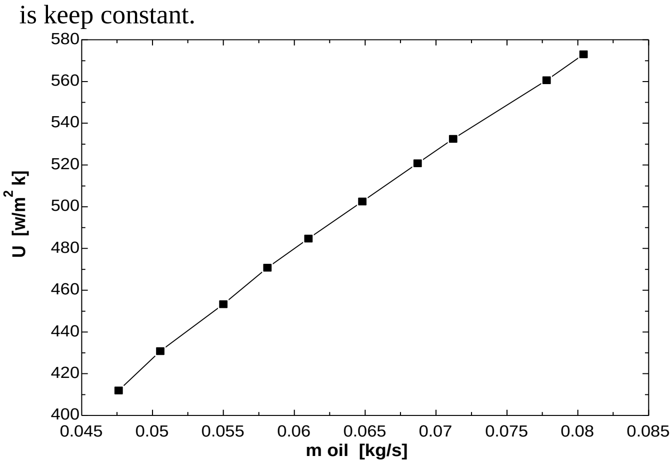

Fig. 8 overall heat transfer co-efficient vs. mass flow rate of oil

Fig. 8 shows that as mass flow rate of oil increase the overall heat transfer co-efficient also increased. Higher the mass flow rate of oil higher value of overall heat transfer co-efficient. This increased the effectiveness of gasket plate heat exchanger.

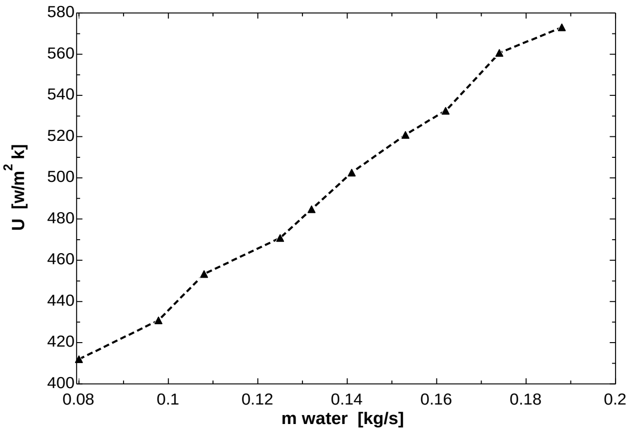

Fig 9 overall heat transfer co-efficient vs. Mass flow rate of water

Fig. 9 shows that as mass flow rate of water increased the overall heat transfer co-efficient also increased. Considering the temperature of oil and water constant.

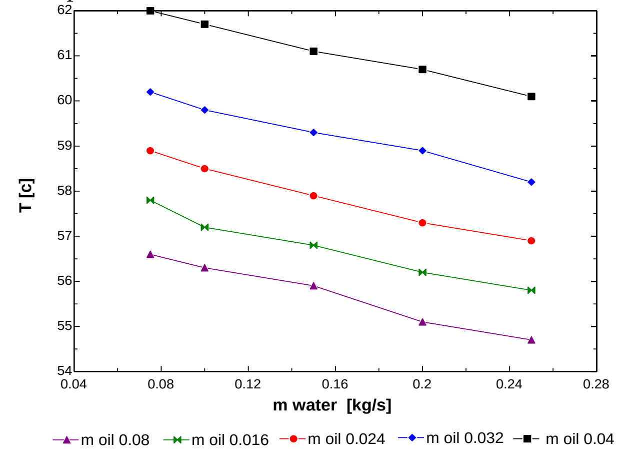

Fig. 10 Outlet Temperature of Oil [^{o}c] vs. Mass Flow Rate of Water [kg/s]

Figure 10 shows the effect of variation of mass flow rate of water on outlet temperature of oil for different mass flow rate of oil. It is clear from the Figure 10 that as the mass flow rate of water increases, the outlet temperature of oil decreases because increase in flow rate of water increase the heat transfer and more cooling water is available. It is observed that as the mass flow rate of oil increases at particular mass flow rate of water, the outlet temperature of oil increases. The initial rise in temperature is more and that become smaller as flow rate of oil increases.

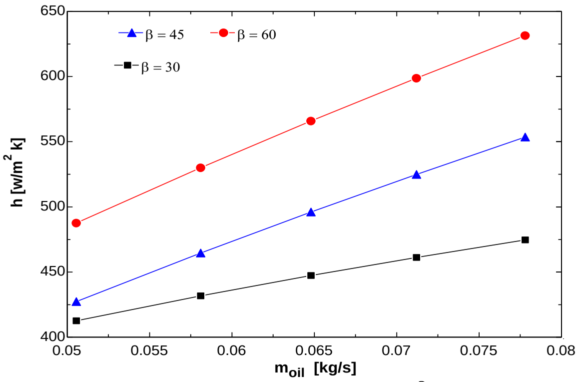

Fig. 11 heat transfer co-efficient of oil [w/m^{2}k] vs. mass flow rate of oil

Figure 11 shows the mass flow rate of oil increase the heat transfer co-efficient of hot side also increase by keeping the mass flow rate of water constant. Comparing the tested 60^{o} chevron plate by changing the chevron angle 45^{o} and 30^{o} the result shows tested 60^{o} chevron plate better heat transfer co-efficient at wide range of Reynolds number.

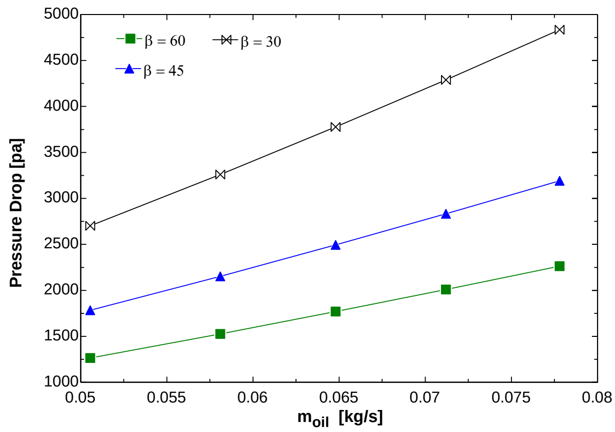

Fig. 12 Frictional pressure drop Δp_c [pa] vs. mass flow rate of oil [kg/s]

Figure 12 shows the mass flow rate of oil increase at constant mass flow rate of water the pressure drop in channel increase but comparing the tested plate with 30^{o} and 45^{o} chevron angle plate the pressure drop in tested plate is lower than 30^{o} and 45^{o} plate. So result indicates that 60^{o} chevron plate has better characteristic in particular this application.

V. SOME NUSSELT CORRELATION IN LITERATURE AND THEIR APPLICATION RANGES

Ref.	β	Re	Correlation
Okada et al.	60	400-15000	$Nu=0.317*Re^{0.65}*p_r^{0.4}$
Maslov et al.	60	50-20000	$Nu=0.78*Re^{0.5}*P_r^{0.333}$
Resenblad et al.	60	60-2415	$Nu=0.289*Re^{0.697}*P_r^{0.333}$
Muly et al.	60	Re>700	$Nu=0.109*Re^{0.7}*P_r^{0.3}$
Kumar et al.	60	Re>400	$Nu=0.108*Re^{0.703}*P_r^{0.333}$
Experimental	60	$50<Re>10^5$	$Nu=j_{Nu}*P_r^{0.333}(\mu/\mu_w)$

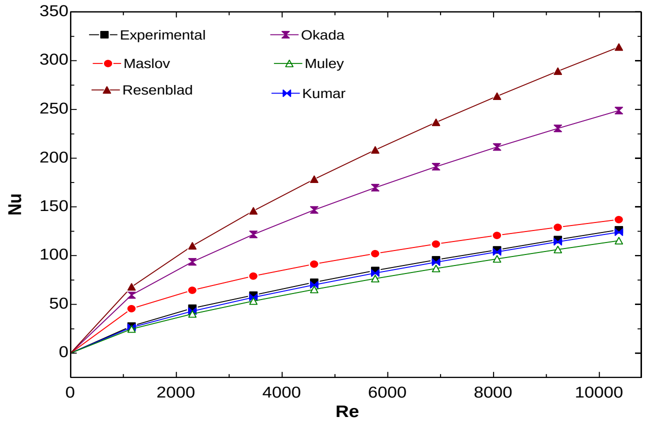

Fig. 13 comparison of experimental correlation with existence

Fig. 13 illustrates that the plate tested has similar behavior with the plates used by okada, kumar, muly, maslov and kovalenko, resenblad.

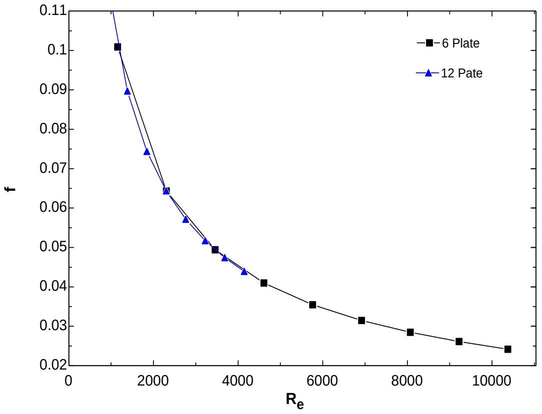

Fig. 14 friction factor characteristic of the plate tested

Fig. 14 illustrates that increasing Reynolds number results in lower friction factor. However, increasing Reynolds number can be achieved with high flow rates which will increase pressure drop dominantly more than f factor because of G2c component of Equation 13.

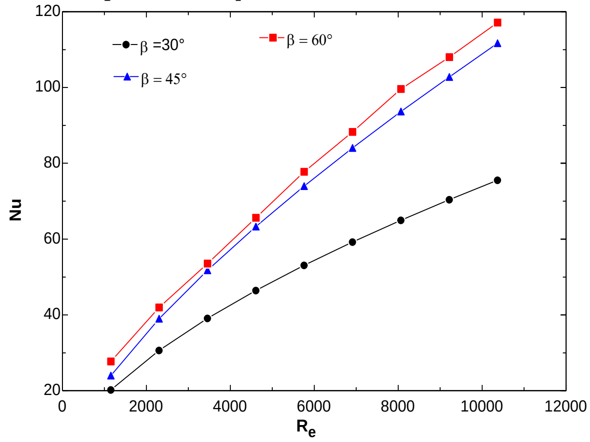

Fig. 15 comparison of Nusselt Number Data for three plate configuration with respective Nusselt Number correlation

Fig. 15 illustrates Experimental heat transfer data are obtain for single phase flow configuration in a commercial plate heat exchanger for symmetric 30^o /30^o, mixed 30^o /60^o , 60^o /60^o chevron plates.

VI. CONCLUSION

Experiments have been performed to investigate characteristics of chevron type heat exchanger with different chevron angles and wide range of Reynolds number. The experimental set-up designed and constructed to determine the characteristics of gasket plate heat exchanger with chevron plates. Experiments are performing to measure the temperature and mass flow rates at all port with varying flow condition.

1. By changing the mass flow rate of oil and water at constant temperature overall heat transfer co-efficient increasing with increasing mass flow rate of both fluid.
2. Outlet temperature of oil decreasing with increasing mass flow rate of water at constant mass flow rate of oil.
3. Mass flow rate of oil increase the heat transfer co-efficient of hot side also increase by keeping the mass flow rate of water constant. Comparing the tested 60^o chevron plate by changing the chevron angle 45^o and 30^o the result shows tested 60^o chevron plate better heat transfer co-efficient at wide range of Reynolds number.
4. Increasing mass flow rate of oil also increasing frictional pressure drop. By comparing the tested 60^o chevron plate by changing the chevron angle 45^o and 30^o the result shows tested 60^o chevron plate has lower pressure drop at same mass flow rate.
5. Nusselt number is found increase with increasing Reynolds number. By comparing the present Nusselt number correlation with existent ones in literature. Result shows that the plate tested has similar behavior with plate used by okada et al., Kumar, Muley and Manglik, maslov and kovalenko, Rosenblad and Kullendorff.
6. Increasing Reynolds number results in lower friction factors.
7. Comparing tested 60^o chevron plate with 30^o and 45^o plates by changing chevron angle result shows higher Nusselt number compare to other at wide Reynolds number range.

VII. NOMENCLATURE

A Total heat transfer area [m2]
b mean mass channel gap [m]
C_p specific heat capacity [$Jkg^{-1}K^{-1}$]
D_e channel equivalent diameter [m]
D_h channel hydraulic diameter [m]
G_{ch} mass velocity of a channel [$kgm^{-2}s^{-1}$]
h heat transfer coefficient [$Wm^{-2}K^{-1}$]
k thermal conductivity [$Wm^{-1}K^{-1}$]
L_w Plate width inside gasket [m]
L_{eff} Effective flow length between ports [m]
m mass flow rate [kgs^{-1}]
N_{cp} number of channels per pass
N_p number of passes
N_u Nusselt number
P pressure [Pa]
ΔP Pressure drop [Pa]
p_r Prandtl number
Q heat transfer rate [W]
R_e Reynolds number
t plate thickness [m]
T temperature [°C]
ΔT_{lm} logarithmic mean temperature [°C]
U overall heat transfer coefficient [$Wm^{-2}K^{-1}$]
β Chevron angle [°]
Ø Surface enlargement factor
ρ density [kgm^{-3}]

Subscripts

c cold water
ch channel

REFERENCES

[1] Ramesh K.Shah, D. P. (2003). “Fundamental of Heat Exchanger Design”. New Jersey: John Wiley& Sons.
[2] S. Kakac, H. Liu, “Heat Exchanger Selection, Rating and Thermal Design”. Second Edition, United State of America, CRC press, 2002.
[3] L.Wang, B. Sunden, R.M.Manglik.“ Plate heat exchangers Design, Application and Performance”. Billerica: United State of America, WIT Press, 2007.
[4] T. Kuppan. (2000), “Heat Exchanger Design Handbook”. New York: United State of America.

[5] Anthony Cooper and J. Dennis Usher,“ Heat Exchanger Design Handbook”. Hemisphere Publishing Corporation, 1983.Washington: United State of America.

[6] F. Akturk, G. Gulben, S. Aradag, N. Sezer Uzol, S. Kakac, “Experimental investigation of the characteristics of a chevron type Gasketed plate heat exchanger”, 6th International Advanced Technologies Symposium (IATS’11), 16-18May2011, Elazıg, Turkey.pp.172-178.

[7] Masoud Asadi, Dr Ramin Haghighi Khoshkho, “Thermodynamical optimization a plate and frame heat exchanger for microturbine applications”, Journal of American Science 2012;8(9) (ISSN: 1545-1003).pp. 716-723.

[8] Prabhakara Rao Bobbili , Bengt Sunden, Sarit K. Das,“An experimental investigation of the port flow maldistribution in small and large plate package heat exchangers”, Applied Thermal Engineering 26 (2006).pp. 1919–1926.

[9] F.S.K.Warnakulasuriya, W.M. Worek, “Heat transfer and pressure drop properties of high viscous solutions in plate heat exchangers”, International Journal of Heat and Mass Transfer 51 (2008).pp.52–67.

[10] A Bhanu Prakash, B Sridhara Rao and Prof. R.C. Sastry, “Heat Recovery Studies of Process Fluids in Corrugated Plate Heat Exchangers”, International Conference on Chemical, Biological and Environment Sciences (ICCEBS'2011) Bangkok Dec., 2011.pp.324-328.

[11] Lieke Wang, Bengt Sunden, “Optimal design of plate heat exchangers with and without pressure drop specifications”, Applied Thermal Engineering 23 (2003).pp. 295–311.

[12] Mohammad S. Khan, Tariq S. Khan, Ming-C. Chyu, Zahid H. Ayub, “Experimental investigation of evaporation heat transfer and pressure drop of ammonia in a 30o chevron plate heat exchanger”, International Journal of refrigeration 35(2012).pp. 1757-1765.

[13] P.Naratarukasa, R.Ponpai,“ Thermal design Algorithm of three-stream plate and frame heat exchanger with two thermal communications”, The Journal of KMITNB.,vol.13,No.3,Jul-sep,2003.

[14] A.A. Fahmy, M.M. EL Fawal and B.M. Taher, “Prediction of Thickness and Fouling Rate in Plate Heat Exchanger of MTR Reactor”, Journal of American Science, 2012; 8(3) (ISSN: 1545-1003).pp. 377-383.

[15] Jorge A.W. Gut, J. M. Pinto, “Optimal configuration design for plate heat exchangers”, International Journal of Heat and Mass Transfer 47 (2004).pp. 4833–4848.

[16] Khan T.S., Khan M.S., Chyu Ming-C., Ayub Z.H.,(2010)“ Experimental investigation of single phase convective heat transfer coefficient in a corrugated plate heat exchanger for multiple plate configurations”, Applied Thermal Engineering 30 ,pp1058–1065.

[17] Lin J.H., Huang C.Y., Su C.C., 2007 Dimensional analysis for the heat transfer characteristics in the corrugated channels of plate heat exchangers, International Communications in Heat and Mass Transfer 34 ,pp 304–312.

My Self Naik Vishal R., P.G student at Shri S’ad Vidya Mandal Institute of Technology, Bharuch, India.