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COMPARITIVE STUDY ON PARALLEL FLOW AND DIAGONAL FLOW ARRANGEMENTS IN PLATE TYPE HEAT EXCHANGER

Uday J. Sahasrabuddhe[1], D. S. Patil[2]
G.H. Raisoni College of Engineering
Wagholi, Pune

Abstract: A number of experimental and analytical studies have been conducted to study the heat transfer characteristics of conventional heat exchangers. Since there is a very limited work is found in literature regarding heat transfer performance of plate heat exchangers. Because of commercial non-disclosure, there is no clarity of design information about plate type heat exchangers. Most of the openly available single-phase heat transfer correlations for plate heat exchangers are geometry specific and are applicable for a certain range of Reynolds (Re) and Prandtl (Pr) numbers. Problem arises there as majority of previous studies provide partial information about the operating conditions and the plate geometry. For practical and industrial applications, it is required to have that Nusselt (Nu) number correlations with complete constant numbers are available. Plate type heat exchangers are available in two different configurations viz. Diagonal and Parallel. End users opt for having existing piping connections without going for changes. However PHE with diagonal and parallel connections can arise with different thermo-hydraulic performance. This paper gives review about the prospect of having the comparison on diagonal and parallel flow arrangements of Plate Heat Exchanger.
Keywords: Plate Heat Exchanger (PHE), Reynolds number, Diagonal flow, Parallel flow

I. INTRODUCTION

For being easy to clean, compact and efficient, the Gasketed plate heat exchangers (PHE) are widely used in the food, chemical and pharmaceutical process industries. The PHE consists of a bundle of corrugated metal plates, gasketted together in a thick frame (see Fig. 1). The gaskets on each of the plates form a flow channels, where the fluids flows alternately and exchanges heat through the thin metal plates. The number of plates, their corrugation angles, the chevron type and level of the gaskets and the location of the inlet and outlet connections at the frames characterize the PHE configuration, which further defines the flow distribution inside the plate pack [3]. The flow arrangement can be parallel, diagonal or any of their various possible combinations. The majority of plate corrugations in PHEs are in the form of chevron because of its relatively simple manufacture. Two neighboring plates are stacked and then some network-shaped contact points are formed, as is shown in Fig.4. These contact points are acting as supporting plates and forming flow channels for working fluid between plates. In the chevron PHE, some intersections can be generated when the working fluid is flowing along the corrugation direction in the space formed by two neighboring plates. These intersections and contact points are distributed in a cross way, which can on one hand enhance heat transfer through increasing the strength of flow turbulence and on the other hand increase the flow resistance. This is the reason why the chevron type PHE is often assigned an obviously large flow resistance than that of the others.

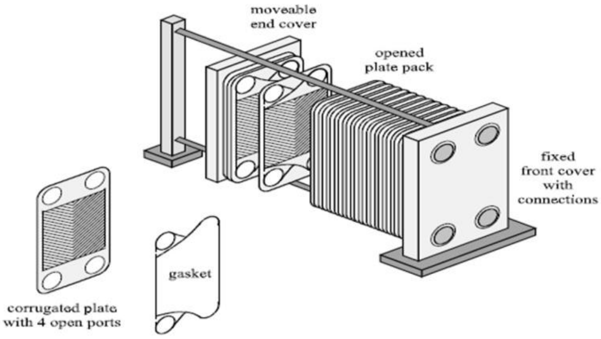


Fig. 1 Assembly of PHE

Usually the main parameters in evaluating the performance of corrugated PHEs are the efficiency of heat transfer [6], flow resistance, and the pressure capacity. It is generally believed that chevron corrugated PHE has high efficiency, large fluid resistance, and large pressure capacity, which is mainly because the flow channel cross section between the plates changes in a very complicated way, likely to cause turbulence. Such a flow, with repeated expansion and contraction of the flow cross section will consume more pumping power. The heat transfer plate of plate heat exchanger is designed for parallel flow OR diagonal flow arrangements. End user always demands for connection arrangement as per available piping at the site irrespective of deign of heat transfer plate. This change in connection style does have impact on the hydraulic and thermal performance of the plate heat exchanger. For the design of PHEs, the pumping power is an important constraint, which is proportional to the pressure drop. Moreover, PHEs have pressure limitations due to the extensive use of gaskets; consequently, the maximum pressure is also a design constraint that depends on the pressure drop. The pressure drop associated with the plate pack consists mainly of three components:

1. Pressure drop within the channels due to friction and flow contraction and expansion
2. Pressure drop associated with the distribution ducts inside the PHE
3. Pressure drop due to an elevation change (static head).

II. OBJECTIVE

A number of experimental and analytical studies have been conducted to study the heat transfer characteristics of conventional heat exchangers. Since there is a very limited work is found in literature regarding heat transfer performance of plate heat exchangers. Because of commercial non-disclosure, there is no clarity of design information about plate type heat exchangers. Most of the openly available single-phase heat transfer correlations for plate heat exchangers are geometry specific and are applicable for a certain range of Reynolds and Prandtl numbers. Problem arises there as majority of previous studies provide partial information about the operating conditions and the plate geometry. For practical and industrial applications, it is required to have that Nusselt number correlations [1] with complete constant numbers are available. The objective of this study is to investigate the thermal performance of a commercially available plate heat exchanger with a mixed theta plate configuration. The effect of Reynolds number on the Nusselt number correlation as well as heat transfer correlations for a diagonal flow and parallel flow configurations are investigated. Experiments were conducted and correlations were developed using modified Wilson plot technique.

III. OVERVIEW OF PLATE HEAT EXCHANGER

The plate-and-frame or gasketed plate heat exchanger (PHE) consists of a number of thin rectangular metal plates sealed around the edges by gaskets and held together in a frame as shown in Fig. 1. The frame usually has a fixed end cover fitted with connecting ports and a movable end cover. In the frame, the plates are suspended from an upper carrying bar and guided by a bottom carrying bar to ensure correct alignment. For this purpose, every plate is notched at the center of its bottom and top edges. The plate pack with fixed and movable end covers is clamped together by tightening bolts, thus compressing the gaskets and forming a sealing of the channels. The carrying bars are longer than the plate packs, so that when the movable end cover is removed, plates may be move along the guide bars for inspection and cleaning. Each plate is made by stamping or embossing a corrugated (or wavy) surface pattern on thin sheets (preferably SS). On one side of each plate, grooves are provided along the periphery of the plate and around the ports for a gasket, as indicated by the dark lines in Fig. 2. Typical plate geometries (corrugated patterns) are shown in Fig. 3, and over 60 different patterns have can be observed with different manufacturers. Alternate plates are assembled such that the corrugations on successive plates contact or cross each other to provide mechanical support to the plate pack through a large number of contact points. The resulting flow channels are narrow, highly turbulent, and tortuous, and enhance the heat transfer rate and decrease fouling resistance by increasing the shear stress, and increasing the level of turbulence. The corrugations also improve the mechanical strength of the plates and form the desired plate spacing. Plates are designated as high theta or low theta, depending on whether they generate a high or low intensity of turbulence.

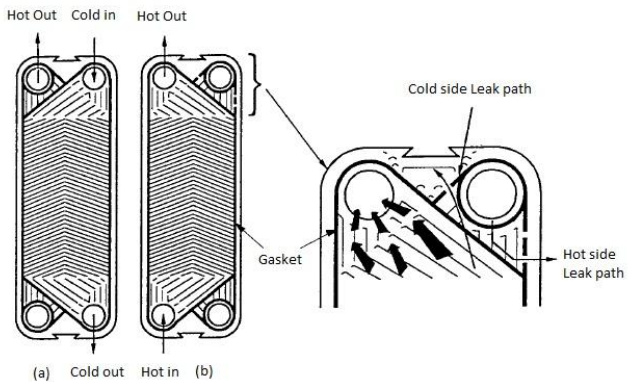


Fig. 2 Typical Heat Transfer Plate

IV. CHARACTERISTICS OF PHE

Plate heat exchangers have very good characteristics due to several reasons:

1. Large heat transfer surface / volume
PHEs are very compact and have a very large heat transfer surface area per apparatus volume compared to other heat exchangers, which gives economic advantages.

2. Thin barriers between the media
The plates separating the media are pressed very thin (down to 0.4 mm), which makes the heat transfer resistance very low. This of course yields excellent k-values.

3. Easy to change the capacity
In order to increase or decrease the capacity one needs only to add or remove a number of plates if the frame is big enough. This procedure can even be done by the user, without the involvement of the manufacturer.

4. Easy to assemble
It is as easy to clean a PHE as it is to change the capacity. Flushing it with water can usually do this but if necessary it can be opened and rinsed plate by plate.

5. High transfer coefficient
Higher heat transfer coefficient due to higher turbulence.
There are two major reasons why PHEs have not taken over the heat exchanger market completely. The first is the gasket between the plates. There are different gasket materials but they normally only resist temperatures up to about 150 °C, while welded PHEs go up to about 300 °C. Chemical resistance is also a factor affecting the choice of gasket. The second reason is that one cannot increase the capacity for a certain exchanger to whatever the size. In order to get up to a certain size one must connect several PHEs in a series. The limit is set by the PHE s inlet, i.e. the area of it. Large flows yield high velocities. Extreme cases give a pressure profile those results in an almost complete halt of flow in some areas of the exchanger. This problem can unfortunately not be solved just by increasing the inlet because that reduces the heat transfer area. There are other minor reasons limiting the use of PHEs. Media with very high viscosities should not be

used because of the large pressure drop that they result in. This raises great demands on the pump used in the process. If a sufficiently powerful pump is not available one must lower the velocity and by that loose some efficiency. A possible way to deal with this problem is to use short plates with a low NTU (θ) (see fig. 5). No erosion or corrosion whatsoever is allowed in a PHE because of the thinness of the plates, which would be destroyed very quickly. A medium flowing through a PHE cannot contain large particles because of the small spacing between the plates. This can be compensated for by increasing the pressing depth but then one loses some of the efficiency in heat exchange. An effect of all the factors mentioned above can be that a less efficient device must be used that what is theoretically (thermally) available.

V. DESIGN BASIS OF PHE

Depending on the size of the flow through the heat exchanger one chooses an exchanger with different inlet and outlet areas. For example a condenser has a greater inlet and bigger spacing between the plates on its steam side than on its water side. One thing that must be considered is that larger inlet and outlet areas give smaller heat transfer surfaces. The frame of the exchanger must be adapted to the working pressure. This can lead to very thick framing.
What matters the most is the design of the plate itself (Fig. 3):

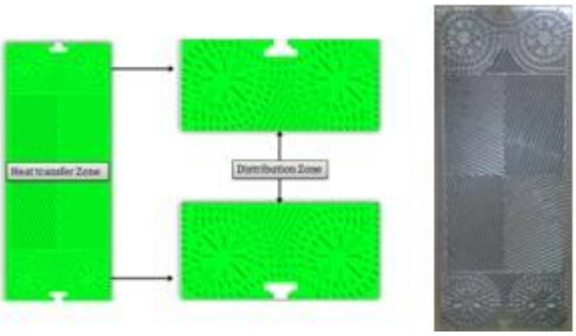


Fig. 3 Typical Heat Transfer Plate zones

In general heat transfer plates of PHE are constructed using chevron or conventional herringbone pattern. The plates we are going to study here are constructed with unique ultra-flex design.

a) *Length and width*
The ratio between the length and the width determines the plate efficiency. In case of an intercrossing temperature profile or a small difference in temperatures of the media, a larger thermal length is required and is accomplished by increasing the length/width ratio. This, of course, is done without the change of the necessary heat transfer surface. High NTU implies small width and long channels while low NTU implies large width and short channels.

b) Port holes
In general PHE plates has four holes punched at the corners, called as port holes. The primary and secondary fluid flows through these holes. These holes are located nearer to the distribution area formed on the plate. The hot or cold fluid enters through the port hole area and distributed towards the heat transfer zone.

c) Pressing depth
The pressing depth (Fig. 4) is important as a better heat transfer is obtained with smaller pressing depth. This must be put against the disadvantage of the pressure drop brought on by the small spacing. Low-density media require a lot of space why one must sometimes use large channels. Heating a fluid with steam sometimes requires asymmetric plates. This implies that the pressing depth is bigger on the steam side than on the fluid side.

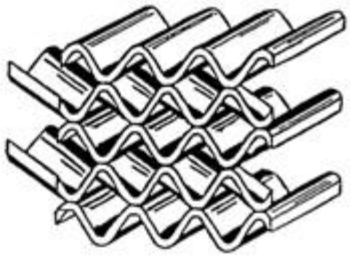
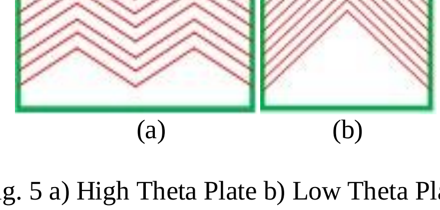


Fig. 4 Corrugations on Heat Transfer Plate

d) The pattern of the plate
The pattern of the plate is what determines the distribution of the medium on the plate and for the most part the pressure drop over it. When a medium enters a channel it follows the line of the least resistance. The distribution pattern immediately after the inlet must see to that all of the fluid doesn't go straight to the outlet but is distributed over the whole plate. This is accomplished by designing the plate so that the resistance is smaller in the parts of the plate channel leading to the farther part of the plate. The working section of the plate is the heat transfer area. For plates with high NTU this is where most of the pressure drop occurs (plates with low NTU have most of their pressure drops in the distribution areas) why it is of crucial importance that the design fits the purpose. The herring bone pattern (Fig. 3) allows for adaptation of the angle as means of increasing or decreasing the efficiency, the greater the angle (larger θ) the greater the efficiency.

e) The chevron angle of the plate
There are typically two types of heat transfer plates available according to their chevron angle θ. High theta ($\theta > 90°$) and low theta ($\theta < 90°$). The conventional heat transfer plate is designed as a 'W' pattern or 'V' pattern as shown below.



(a) (b)

Fig. 5 a) High Theta Plate b) Low Theta Plate

The unique pattern for heat transfer zone that we are going to analyze over here is ultra-flex pattern. In this design the heat transfer zone is divided into four quadrants and the chevron is formed as below (Fig. 6).

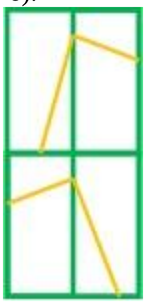
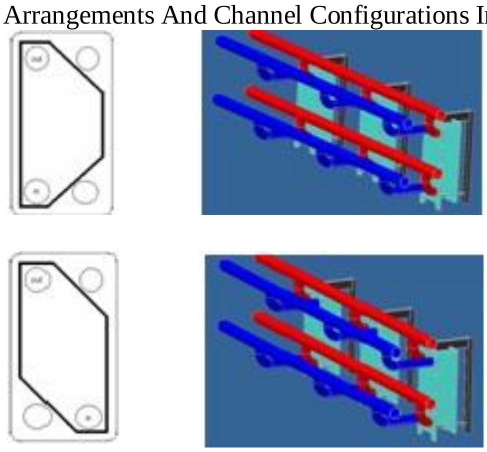


Fig. 6 Chevron angles on Heat Transfer Plate

Flow Arrangements And Channel Configurations In PHE



(a)

(b)

Fig. 7 (a) Parallel Flow Arrangement
(b) Diagonal flow arrangement

Depending on the design of the channel a medium can take two different paths across the plate, a diagonal or a parallel (Fig.7) one. The diagonal used to be the preferred one but the parallel one is used nowadays because of the simplicity of manufacturing the plates. Only one kind of plates is needed for the parallel pattern while the diagonal pattern requires two kinds of plates.

VI. SET UP ARRANGEMENT

The Plate Heat Exchangers are usually tested at elevated temperatures with varying mass flow rates. Below is the sketch showing the laboratory set up of the arrangement used for testing the PHE.

Fig. 8 Experimental set up for PHE tests

The hot water temperature is maintained with auxiliary heat exchanger with the help of steam whereas cold water temperature is maintained by the chiller. The RTD are connected at all four ports i.e. inlet and outlet of hot and cold side. The digital pressure transmitters are connected across the inlet and outlet of process fluid. The mass flow meters are connected to the supply lines which measure volumetric as well as mass flow rates of the hot water and cold water. The readings are then transferred to the data acquisition system. The digital recording system records all the experimental data.

VII. EXPERIMENTAL READINGS AND TEST OUTCOMES

The tests are carried out for diagonal and parallel arrangement of PHE. The flow rates were maintained between 10 to 30 m3/hr resulting into the pure turbulent flow. The Reynolds number varied in the range of 2500 to 7000.

Below are readings for diagonal flow PHE.

HOT SIDE			COLD SIDE				
Flow rate	T_in	T_out	Flow arte	T_In	T_out	Re_avg	ΔP
m3/hr	°C	°C	m3/hr	°C	°C		kPa
28.64	34.67	22.44	28.66	12.06	24.39	6698	91.70
26.01	35.11	22.50	26.05	12.22	24.94	6120	75.84
22.10	34.50	20.72	22.05	10.11	24.11	5064	55.16
17.97	35.78	20.33	17.87	9.39	25.11	4133	37.92
12.54	35.39	20.39	12.49	11.28	26.50	2926	19.31

Also the tests are carried out for the parallel flow PHE. Below are the readings for the same.

HOT SIDE			COLD SIDE				
Vol_1	T1_in	T1_out	Vol_2	T2_In	T2_out	Re_avg	ΔP
m3/hr	°C	°C	m3/hr	°C	°C		kPa
27.48	34.50	21.78	27.55	11.28	24.11	6,375	96.53
25.37	35.06	22.06	25.37	11.72	24.89	5,935	81.36
21.21	34.28	20.44	21.33	10.28	24.22	4,872	57.23
16.99	35.11	20.83	17.10	11.39	25.72	3,975	37.23
11.63	35.83	19.72	11.58	10.50	26.83	2,705	17.93

VIII. CORRELATIONS FOR PERFORMANCE

Heat transfer implies a quantity of energy to be transferred from a medium with a higher energy value to a medium with a lower energy value. Every medium has a heat transfer coefficient (*h*-value) determined by its phase (solid, liquid or gaseous), its temperature and its flow velocity solids cannot be used for continuous heat transfer but as a barrier between the media). The measure of how well a material transfers heat (thermal conductivity *k*) is normally determined experimentally and can be found in various tables.

The Energy Balance

The energy balance describes that the energy given up from one of the media during the heat transfer must either be taken up by the other medium or the surroundings. For simplicity's sake we will disregard from the lost to the surroundings even if it is present in all heat exchangers to a certain degree.

$Q = (\acute{m}\, Cp\, \Delta T)_{cold} = (\acute{m}\, Cp\, \Delta T)_{hot}$

Where,

Q = Total amount of energy transferred per unit time [*W*]

ḿ = Mass flow rate [kg/s]

Cp = Specific heat of fluid [kJ/kgK]

ΔT = Temperature difference between inlet and outlet temperature of fluid [°C]

Heat Transfer Coefficient (U-value)
In order to determine how well the energy can be transferred a closer look must be taken into heat transport within a medium. Convection is defined as the process of heat transfer within the medium. There are two kinds of convection: natural and forced. Natural convection is heat transport brought on by the density differences within the medium, i.e. heat input reduces the density of one part of the media, which then starts to move and by that transfer heat. Forced convection implies an influence of an outer force making the media move pass the heat transfer surface with a greater velocity. The heat transfer is described by:

$Q = U . A . \theta m$
Where ,
U = Overall heat transfer coefficient [W/m^2K]
A = Surface area of heat exchanger plates [m^2]
θm = Log mean temperature difference [k]

The value of U is determined by a number of factors
v --- The velocity of the medium [m/s]
ρ --- The density of the medium [kg/m^3]
C_p --- The specific heat capacity of the medium [J/kgK]

Generally for single phase heat transfer Nusselt number correlation is given by and equation form.

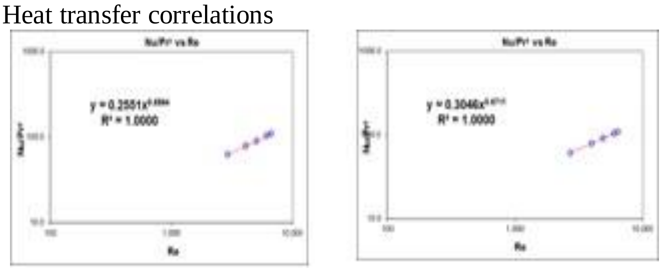
$Nu = C * Re^p Pr^n$
Nu in the above expression represents the Nusselt number. Whereas C , p and n are the constants independent of the nature of the fluid used. Nu can also be derived as below.

$Nu = h *D/k$
Further the non-dimensional number is derived as 'j' for the heat transfer correlation. It is represented as

$j = f (Nu / Pr^{0.33})$
The curves are then plotted for the 'j' factor taking care of heat transfer correlation and friction factor '*f*'.

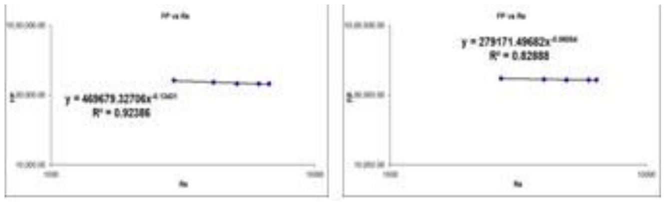
IX. PERFORMANCE CURVES

Heat transfer correlations



(a) (b)
a) j = 0.2551 x Re0.6884 b) j = 0.3046 x Re0.6711
Fig. 9 (a) Diagonal Flow Arrangement
(b) Parallel Flow Arrangement

Friction factor (Pressure drop) correlations



(a) (b)
a) f = 469679.32 x $Re^{-0.13431}$ b) f = = 279171.49 x $Re^{-0.06054}$
Fig. 10 (a) Diagonal Flow Arrangement
(b) Parallel Flow Arrangement

X. CONCLUSION

Plate Heat Exchangers (PHE) are very unique type of heat exchangers as far as the thermal and hydraulic design is considered. Plenty study and research is done on the performance evaluation of PHE. Researchers have worked out on establishing the correlations for conventional chevron corrugated plates. Also attempts are made to explore the performance impacts due to change in corrugation methodology.
However industrial applications are constrained due to below points.

- Existing piping at the site.
- Heat transfer duties.
- Energy consumed in pumping.

It is thus necessary to study and find out the optimum performance produced by diagonal and parallel flow patterns of PHE. It can be seen clearly from the correlations plotted above that the diagonal arrangement of the PHE has better heat transfer and pressure drop performance. It is observed that for the turbulent flow the pressure drop across the connections is lesser for the diagonal flow. This effect may increase in case of wider plates.

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NOENCLATURE

PHE – Plate Heat Exchanger
Nu – Nusselts Number
Re – Reynolds Number
Pr – Prandtls Number
θ - Included angle of Heat Transfer Plates
NTU – Number of units transferred
β - Chevron angle of Heat transfer plate
f - Friction factor constant
b - Depth of the corrugation
l - Pitch of corrugations
ASME – American Society For Mechanical Engineers
j – Heat transfer coefficient
f - Friction factor coeccicient
Q = Total amount of energy transferred per unit time
ḿ = Mass flow rate
Cp = Specific heat of fluid
ΔT = Temperature difference between inlet and outlet temperature of fluid
U = Overall heat transfer coefficient
A = Surface area of heat exchanger plates
θm = Log mean temperature difference
v --- The velocity of the medium
ρ --- The density of the medium
C_p --- The specific heat capacity of the medium

