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Flat Plate Heat Exchanger Design for MTR Reactor Upgrading

Adel Alyan Fahmy
Atomic Energy Authority, Reactors Department, 13759 Cairo, Egypt
adelalyan@yahoo.com

Abstract: The purpose of the present study is to investigate the performance of plate heat exchanger used in a research reactor for updating it and raise its power to 10 (MW).The objective also to extend the present knowledge of the thermo-hydraulic performance of PHE with consideration of channel geometries (Chevron angles). In this study the surface area density, the total number of corrugated plates and the pressure drop will be study at different Chevron angles, also the corrugation pitch. The effect of variation of Chevron angle on the heat transfer coefficient will be study. The Chevron angle will vary from (30º to 60º).This study aims to obtain an applicable model which is able to design flat plate heat exchanger for many applications.

Index Terms— Compact heat exchanger, corrugated plates, friction factor, forced convection.

1. Introduction

A Compact heat exchanger consists of plates instead of tubes to separate the hot and cold fluids. Because each of the plate has very large surface area, the plates provide each of the fluids with an extremely large heat transfer area. Due to the high heat transfer efficiency of the plates, plate type heat exchanger is very compact when compared to a shell and tube heat exchanger with the same heat transfer capacity. In the 1930's PHEs were introduced to meet the hygienic demands of the diary industry. Today the PHE is universally used in many fields; heating and ventilating, breweries, dairy, food processing, pharmaceuticals and fine chemicals, petroleum and chemical industries, power generation, offshore oil and gas production, onboard ships, pulp and paper production etc. plate heat exchanger also find application in water closed circuit cooling water systems using potentially corrosive primary cooling water drawn from sea, river, lake, or cooling tower to cool non-corrosive secondary liquid flowing in a closed circuit.

2- Chevron Corrugations

Geometrical features of the chevron type of corrugation are in Figure.4. A single plate comprises four corner ports and the corrugation area. The corrugation pattern has a chevron angle β, evaluated as the angle between the corrugation troughs to the vertical axis. In a PHE unit plates are installed with the apex of the chevron pointing in opposite directions. The chevron design brings four main effects, it

1. Increases flow turbulence level,
2. Increases the effective heat transfer area (typically by a factor of 1.1 – 1.25),
3. Increases the stiffness of the plate pack,
4. Increases turbulence and high wall shear force which reduces fouling.

The chevron angle of commercially available plates varies between the extremes of a about 25 to 65º, and is perhaps the most important geometrical parameter of PHE's relating to thermal and hydraulic performance[1]. Conventional plates have approximately sinusoidal profiles. Corrugations that are asymmetrical in profile are not common and found to be less efficient [2].The surface enlargement factor ,Φ, is another important parameter, which is defined as the ratio of actual heat transfer area to the projected area. Most commercial plates have enlargement factors usually in a relatively small range of 1.1 – 1.25 [3]. Φ can be calculated approximately, for a sinusoidal corrugation profile, from a three-point integration formula [4].

3- Flow Arrangements

The flow arrangement in a PHE can be very flexible. There are basically four types of configuration that be used, as will be shown below in Figure1., which are related to flow distributed inside or between channels

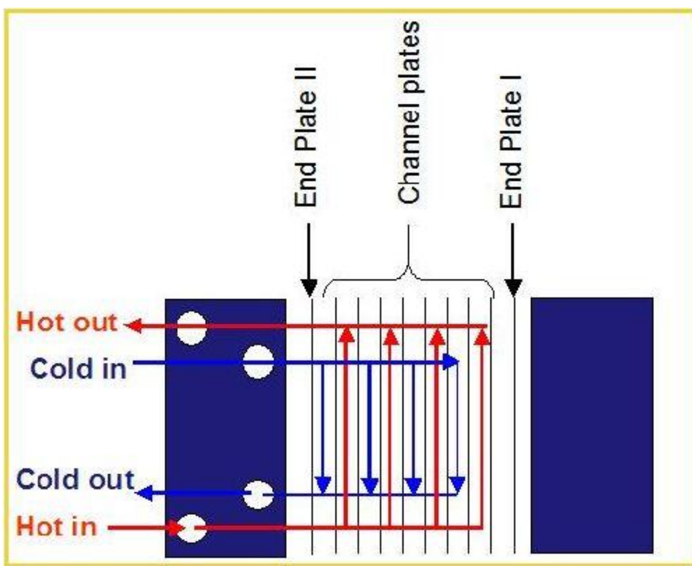

Figure.1. Flow distribution inside channels

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3.1. Inside Single Channel

In a single channel, the flow of one fluid can be either diagonal or parallel, depending on the sealing arrangement on the ports. A diagonal arrangement refers to the flow that enters and leaves the channel at diagonally opposite ports, whereas in the parallel (or vertical, same-side) arrangement the flow enters and leaves on the same side.

3.2. Pass

Pass refers to a group of channels in which the flow is in the same direction. The two streams in a PHE unit can have different pass arrangements, single or multiple, which the latter consisting of passes connected in series.

3.3. U and Z Type Arrangement in Single-pass Flows

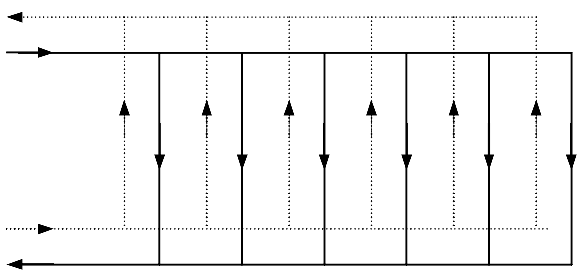

Figure.2. U arrangement of Plates

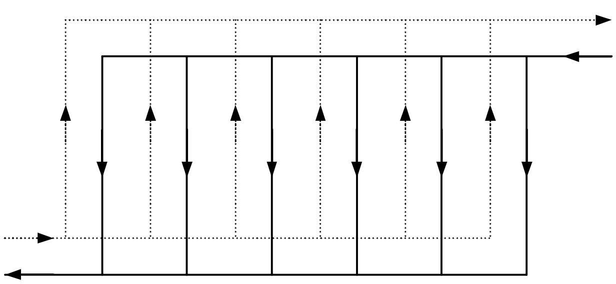

Figure.3. Z arrangement of Plates

The so-called U and Z type arrangements are found in single-pass flows. For any one fluid stream if the inlet and outlet ports are on the same side of the exchanger unit, it is called a U type arrangement; otherwise it is a Z type, as shown in Figures 2 and 3.

The surface enlargement factor can be calculated approximately for sinusoidal corrugation by:

$$\phi \approx \frac{1}{6}\left(1+\sqrt{1+X^2}+4\sqrt{1+\frac{X^2}{2}}\right) \quad (1)$$

Where:

$$X = \frac{2.\pi.a}{\lambda} \quad (2)$$

The hydraulic diameter of the plate heat exchanger is obtained as follows:

$$D_h = \frac{4.A_c.L_p}{P_{wet}.L_p} = \frac{4.A_c.L_p}{A_t} = \frac{4\left(2.aW_p.N_c\right).L_p}{2.L_\lambda.N_\lambda.N_p.N_c} \quad (3)$$

4.Friction factor

One of the correlations for the friction of a plate heat exchanger with Chevron pattern is provided by Martin [7]. The fanning friction factor is:

$$f = \left[\frac{\cos\beta}{\left(0.045.\tan\beta+0.09.\sin\beta+\frac{f_o}{\cos\beta}\right)^{\frac{1}{2}}}+\frac{1-\cos\beta}{\sqrt{(3.8).f_1}}\right]^{-2} \quad (4)$$

Where:

$$f_o = \frac{16}{\mathrm{R_e}} \qquad \text{For Re < 2000} \quad (5)$$

$$f_o = \left(1.56.Ln(\mathrm{R_e})-3.0\right)^{-2} \qquad \text{For Re>2000} \quad (6)$$

$$f_1 = \frac{149.25}{\mathrm{R_e}}+0.9625 \qquad \text{For Re <2000} \quad (7)$$

$$f_1 = \frac{9.75}{\mathrm{R_e}^{0.289}} \qquad \text{For Re>2000} \quad (8)$$

5. Nusselt Number

Martin[1] also provided the Nusselt number correlation for a plate heat exchanger with Chevron pattern as:

$$Nu = \frac{h.D_h}{K_f} \quad (9)$$

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$$Nu = 0.205.P_r^{\frac{1}{3}}\left(f\,.\mathrm{R}_e^2.\sin 2.\beta\right)^{0.374}\left(\frac{\mu}{\mu_s}\right)^{\frac{1}{6}} \tag{10}$$

Where $10^o < \beta < 80^o$ and Kf is the thermal conductivity of the fluid and μs is the dynamic viscosity at the wall temperature,(μ / μs)=1 may be used with the assumption the μ changes moderately with temperature.

6. Pressure drop

The total pressure drop in a plate heat exchanger is composed of the friction pressure drop of the channels (ΔPf) and the port pressure drop (ΔPp). We assumed that the pressure drop due to the elevation (gravity)change is negligible. The frictional pressure drop is calculated using the following equation:

$$\Delta P_p = \frac{2.f\,.L}{D_h}.\frac{G^2}{\rho}.N_p \tag{11}$$

The mass velocity for each fluid at the port of the plate heat exchanger is defined by:

$$G = \frac{4.\dot{m}}{\pi.D_p^2} \tag{12}$$

Where : Dp is the port diameter. The port pressure drop for each fluid is then calculated by:

$$\Delta P_p = \frac{1.5.N_P.G_p^2}{2.\rho} \tag{13}$$

The total pressure drop is the sum of equation (11) and (13).

$$\Delta P_t = \Delta P_f + \Delta P_p \tag{14}$$

The port diameter is designed such that the port pressure drop is usually less than 10 percent of the total pressure drop, but it may be as high as 25 to 30 percent in some designs.

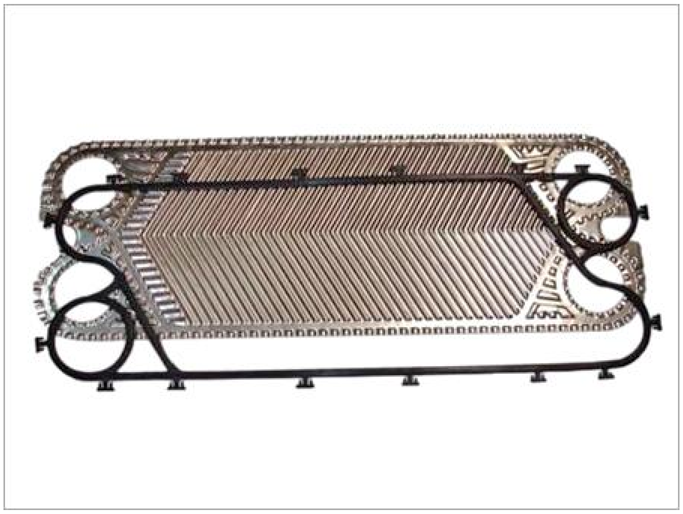

Figure.4. Plates with chevron-type corrugation pattern for a plate heat exchanger.

The hot and cold water properties at the average fluid temperatures are obtained as follows:

$$T_{hot-water} = \frac{(T\_hot\_1)+(T\_hot\_2)}{2} \tag{15}$$

$$T_{cold-water} = \frac{(T\_cold\_1)+(T\_cold\_2)}{2} \tag{16}$$

Table.1 The hot and cold water properties:

Hot water subscript 1	cold water subscript 2
$\rho_1 = 991.9\left(Kg/m^3\right)$	$\rho_2 = 994.7\left(Kg/m^3\right)$
$Cp_1 = 4182\left(J/Kg.K\right)$	$Cp_2 = 4183\left(J/Kg.K\right)$
$k_1 = 0.6188\left(W/m.K\right)$	$k_2 = 0.6075\left(W/m.K\right)$
$0.0006448\left(N.s/m^2\right)$	$0.0007513\left(N.s/m^2\right)$
$\mathrm{Pr}_1 = 4.358$	$\mathrm{Pr}_2 = 5.174$
$\dot{m}_1 = 275.5\left(Kg/s\right)$	$\dot{m}_2 = 414.5\left(Kg/s\right)$

The thermal conductivity of the plate (Stainless steel AISI 304) (kw) 14.9 (W/m.K).

Table.2. Geometric Parameters:

Character	Definition
$N_p = 1$	Number of passes.
$\beta = 30^o$	Corrugation inclination angle(Chevron angle).
$N_{TOT} = 160$	Total number of plates.
$D_p = 20$	Port
$\delta = 0.6\left(mm\right)$	Thickness of plates.
$X = 9\left(mm\right)$	Corrugation Pitch (Wave length).

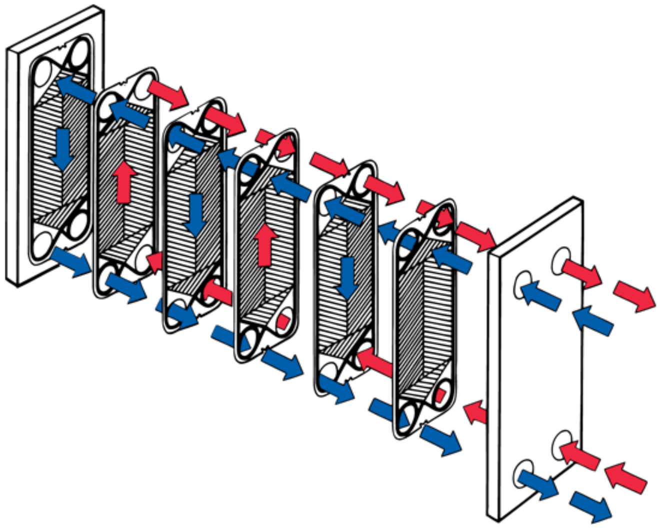

Figure.5. Flat Plate Heat Exchanger Channels.

The number of wave length per single a plate is obtained by dividing the PHE width (Wp) by the corrugation pitch (X).

$$N_X = \frac{w_p}{X} \tag{17}$$

The number of channels for each fluid is calculated by dividing the total number of channel (NTOT +1) by two and the number of passes Np as shown in the following equation:

$$N_{01} = \frac{N_{TOT} + 1}{2.N_p} \tag{18}$$

The number of channels per one fluid is the same as for the other fluids.

$$N_{01} = N_{02} \tag{19}$$

The corrugation amplitude is given by the following equation, the amplitude is expressed as a function of the PHE height,(Hp).

$$a = \left(\frac{1}{2}\right).\left(\frac{Hp}{N_{TOT}+1} - \delta\right) \tag{20}$$

The corrugation ratio (γ) is defined as:

$$\gamma = 4.\frac{a}{X} \tag{21}$$

The plate (PHE) design is commonly limited to (γ<1). The corrugation length is given using the following equation:

$$L_X = \int_0^X 1 + \left(\frac{2.\pi.a}{X}\right)^2 .\cos\left(\frac{2.\pi}{X} - X\right)^2 .dX \tag{22}$$

The heat transfer area for each fluid is calculated considering two surfaces per channel as follows:

$$A_1 = 2.L_X.N_X.L_p.N_{01} \tag{23}$$

$$A_1 = A_2 \tag{24}$$

The projected area for the plate is calculated by:

$$A_{p1} = 2W_p.L_p.N_{01} \tag{25}$$

So,

$$A_{p1} = A_{p2} \tag{26}$$

$$\Phi = \frac{L_X.N_X}{W_p} \tag{27}$$

Where is the enlargement factor.

$$D_h = \frac{4a}{\Phi} \tag{28}$$

Where, Dh is the hydraulic diameter.
The flow area Ac for each fluid is obtained using the following equation:

$$A_{C1} = 2aW_p.N_{c1} \tag{29}$$

$$A_{C1} = A_{C2} \tag{30}$$

The mass velocity, velocity and Reynolds number are defined using the general definition as:

$$G1 = \frac{\dot{m}1}{Ac1} \tag{31}$$

$$V_1 = \frac{G_1}{\rho_1} \tag{32}$$

$$\mathrm{Re}_1 = \frac{G_1.Dh}{\mu_1} \tag{33}$$

After updating the final sizing, the velocity may be checked against the reasonable values.

Martin's [7] developed with the Darcy friction factor is modified with the fanning friction factor as follows:

$$F_{01} = 1.56.(LN)(\mathrm{R_{e1}} - 3.0)^{-2} \tag{34}$$

Otherwise,

$$F_{01} = \frac{16}{\mathrm{R_{e1}}} \tag{35}$$

$$F_{02} = 1.56.(LN)(\mathrm{R_{e2}} - 3.0)^{-2} \tag{36}$$

Otherwise,

$$F_{02} = \frac{16}{\mathrm{R_{e2}}} \tag{37}$$

$$F_{m1} = \frac{9.75}{(\mathrm{R_{e1}})^{0.289}} \quad \text{for Re1} \geq 2000 \tag{38}$$

Otherwise,

$$F_{m1} = \frac{149.25}{\mathrm{R_{e1}}} + 0.9625 \tag{39}$$

$$F_{m2} = \frac{9.75}{(\mathrm{R_{e2}})^{0.289}} \quad \text{for Re2} \geq 2000 \tag{40}$$

Otherwise,

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$$F_{m2}=\frac{149.25}{R_{e2}}+0.9625 \quad (41)$$

The fanning friction factor for each fluid is given using the following equation:

$$F_1=\left[\frac{\cos(\beta)}{\left(0.045.\tan(\beta)+0.09.\sin.(\beta)+\frac{F_{01}}{\cos(\beta)}\right)^{0.5}}+\left(\frac{1-\cos(\beta)}{\sqrt{3.8.F_{m1}}}\right)\right]^{-2} \quad (42)$$

$$F_2=\left[\frac{\cos(\beta)}{\left(0.045.\tan(\beta)+0.09.\sin.(\beta)+\frac{F_{02}}{\cos(\beta)}\right)^{0.5}}+\left(\frac{1-\cos(\beta)}{\sqrt{3.8.F_{m2}}}\right)\right]^{-2} \quad (43)$$

7. Overall thermal-hydraulic performance

PHE'S are usually much more thermally efficient their shell- tube counterparts, particularly for liquid/liquid duties. Film coefficients can be two to four times those for tubular units of the same duty, at the same or even lower pressure drops [4].At normal working ranges the overall heat transfer coefficient U can be expected to be 2300 – 5800 (W/m2.K), depending on plate corrugation and flow conditions [5]. The highest U value that could be achieved by a PHE was reported up to 8500 (W/m2.K), making it capable of working with film coefficient three to five times high than tubular or spiral-plate design [6]. The augmented heat transfer performance of a PHE is due to several enhancement mechanisms, which directly result from the complex plate surface characteristics. These surface effects include disruption and reattachment of boundary layers, swirling motion of the fluids, continuous change in flow directions and velocity, combining to promote early transition to turbulence and produce exceptionally high film coefficients of heat transfer.

The heat transfer coefficient of the hot fluid is given by:

$$h_1=\left(\frac{k_1}{D_h}\right).\left[0.205.\left(\Pr_1\right)^{\frac{1}{3}}.(1)^{\frac{1}{3}}.F_1.\left(\mathrm{Re}_1\right)^{2}.\sin(2.\beta)^{0.374}\right] \quad (44)$$

The heat transfer coefficient of the hot fluid is given by:

$$h_2=\left(\frac{k_2}{D_h}\right).\left[0.205.\left(\Pr_2\right)^{\frac{1}{3}}.(1)^{\frac{1}{3}}.F_2.\left(\mathrm{Re}_2\right)^{2}.\sin(2.\beta)^{0.374}\right] \quad (45)$$

The overall heat transfer coefficient is obtained by using the following equation:

$$U.A=\frac{1}{\left[\left(\frac{1}{h_1A_1}\right)+\left(\frac{\delta}{K_W}\right)+\frac{1}{h_2A_2}\right]} \quad (46)$$

8.ε- NTU Method:

The heat capacity for both fluids is defined, and then the minimum and maximum heat capacities are obtained.

$$C_1=\dot{m}1.Cp_1 \quad (47)$$

$$C_2=\dot{m}2.Cp_2 \quad (48)$$

$$C_{\min}=\min.\left(C_1.C_2\right) \quad (49)$$

$$C_{\max}=\max.\left(C_1.C_2\right) \quad (50)$$

The heat capacity ratio Cr is defined as:

$$Cr=\frac{C_{\min}}{C_{\max}} \quad (51)$$

So, the NTU can be calculated from the following equation:

$$NTU=\frac{U.A}{C_{\min}} \quad (52)$$

The effectiveness of the PHE for counter flow is obtained using the following equation:

$$\varepsilon=\frac{1-e^{[-NTU.(1-Cr)]}}{1-Cr.e^{[-NTU.(1-Cr)]}} \quad (53)$$

The heat transfer rate for the PHE is obtained as:

$$q=\varepsilon.C_{\min}.\left(T_{hot}i-T_{cold}i\right) \quad (54)$$

9.Pressure drop:
9.1Pressure Drop Components

Pressure drop in a PHE consists of three contributions:

1. frictional pressure drop within the plate passages
2. pressure drop due to elevation change
3. Pressure drop in inlet and outlet manifolds (port).

The frictional channel pressure drops for both fluids are obtained using the following equations:

$$\Delta P_{f1}=\frac{2f_1.L_P}{D_h}.\frac{\left(G_1\right)^2}{\rho_1}.N_p \quad (55)$$

$$\Delta P_{f2}=\frac{2f_2.L_P}{D_h}.\frac{\left(G_2\right)^2}{\rho_2}.N_p \quad (56)$$

The connection and port pressure drops are obtained using the following equations:

$$G_{P1}=\frac{4.\dot{m}_1}{\pi.D_p^{2}} \quad (57)$$

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$$G_{P2} = \frac{4.\dot{m}_2}{\pi.D_p{}^2} \tag{58}$$

$$\Delta P_{p_1} = 1.5.Np.\frac{G_{P1}{}^2}{2.\rho_1} \tag{59}$$

$$\Delta P_{p_2} = 1.5.Np.\frac{G_{P_2}{}^2}{2.\rho_2} \tag{60}$$

10.Results:

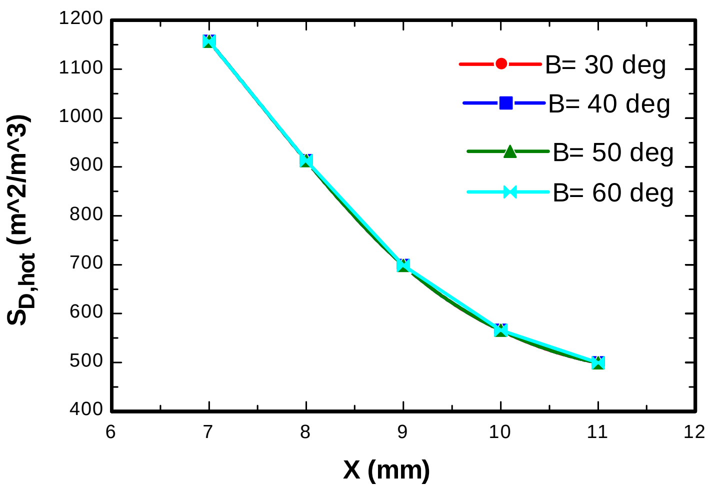

Figure.6. Surface area density and corrugation pitch.

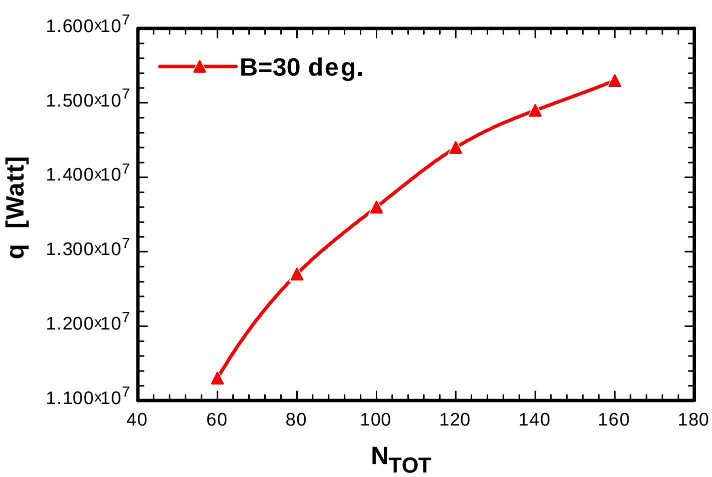

Figure.7. Effect of total number of plates on the compact heat exchanger heat transfer.

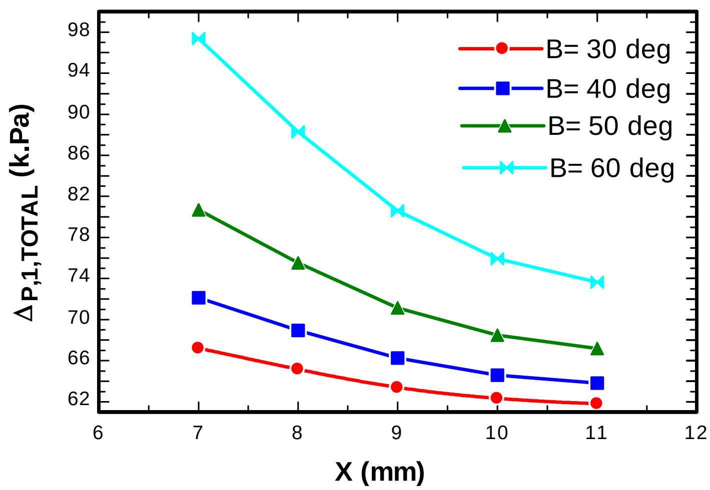

Figure.8. Effect of Chevron angle on the hot fluid pressure drop.

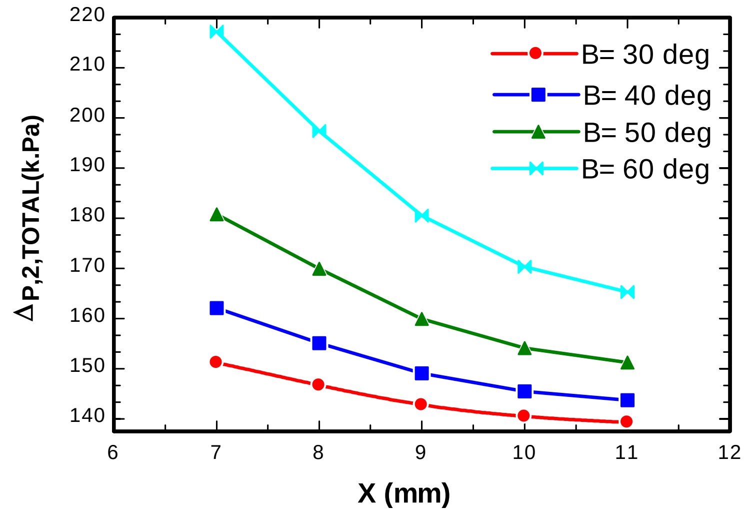

Figure.9. Effect of Chevron angle on the cold fluid pressure drop.

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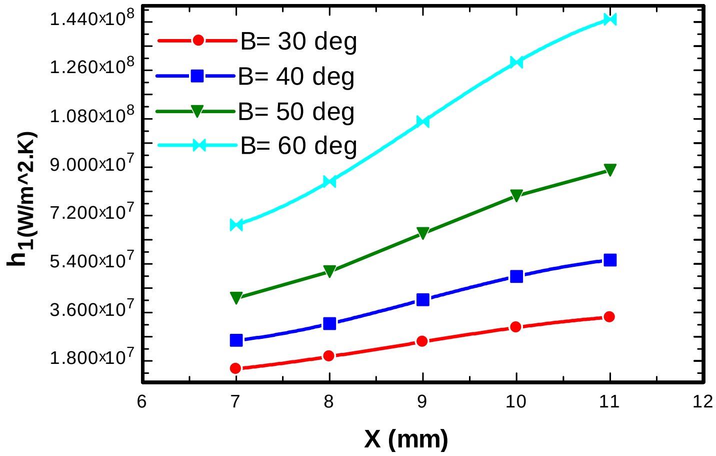

Figure.10. Effect of Chevron angle on the hot fluid heat transfer coefficient.

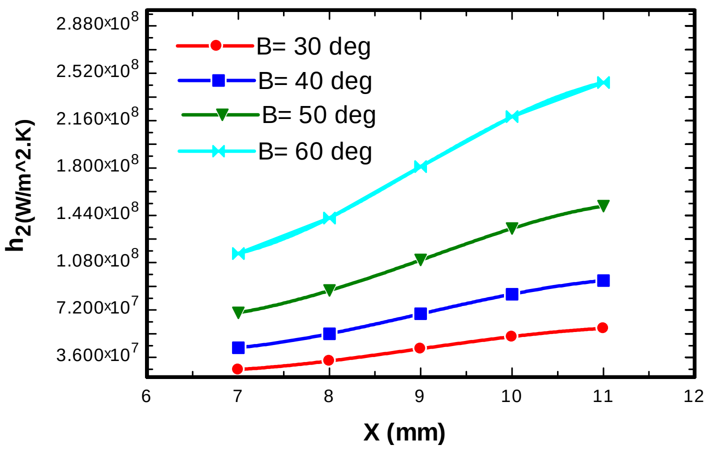

Figure.11. Effect of Chevron angle on the cold fluid heat transfer coefficient.

Fig.(6) shows the relation between the surface area density and the corrugation pitch. The surface area increases with decreasing the corrugation pitch. The chevron angle has no effect on the surface area density

Fig.(7) depicts the effect of the total number of plates on the compact heat exchanger heat transfer. The heat transfer of the compact heat exchanger increases with increasing the number of heat exchanger plates due to increasing of the effective heat transfer area.

Fig.(8,9) illustrate the relation between the pressure drop and corrugation pitch at different chevron angle .with decreasing the corrugation pitch the pressure drop increases due to increasing of fluid velocity between the corrugations. At constant corrugation pitch, the pressure drop increases with increasing of the Chevron angle due to increasing of fluid turbulence also the passage become more tortuous and offers greater hydrodynamic resistance.

Fig.(10,11) shows the relation between the heat transfer coefficient and the corrugation pitch at different Chevron angle .The corrugated pattern on the thermal plate induced a highly turbulent fluid flow. The high turbulent in the plat heat exchanger leads to an enhanced heat transfer.

4-Conclusion

Theoretical model have been introduced to investigate heat transfer characteristic of a commercial corrugated plate heat exchanger with different Chevron angles and corrugation pitch. The effect of Chevron angles variation on the heat transfer coefficient is simulated. It found that, as the corrugation angle is reduced from (90º), the flow passage become more tortuous and offers greater hydrodynamic resistance. The heat transfer carried by the fluid in a corrugated plate heat exchanger increased by increasing the total number of the corrugated plates due to the increase of the effective heat transfer area. With increasing the Chevron angle the friction factor increase, hence increasing the pressure drop.

Nomenclature:

β	Chevron angle.	deg.
Φ	Surface enlargement factor.	___
PHE	Plate heat exchanger.	___
X	Corrugation pitch.	mm
a	Corrugation amplitude.	mm
D_h	Hydraulic diameter.	mm
Ac	Free flow area.	m2
Lp	Heat exchanger length.	m
Pwet	Perimeter length.	mm
At		
Wp	Heat exchanger	m

	width.	
Nc	Number of cold channel.	—
f	Fanning friction factor.	—
Re	Reynolds number.	—
Nu	Nusselt number.	—
h	Heat transfer coefficient.	W/m^2.K
μ	dynamic viscosity.	N.s/m^2
ΔPp	Port frictional pressure drop.	kPa
ΔP_t	Total pressure drop.	kPa
ΔPf	Frictional pressure drop of the channels.	kPa
G	Mass velocity.	Kg/m^2.s
N_p	Number of passes.	—
$T_{hot-water}$	The average of hot water temperature.	ºC
$T_{cold-water}$	The average of cold water temperature.	ºC
T_hot_1	The temperature of hot water inlet	ºC
T_hot_2	The temperature of hot water outlet.	ºC
T_cold_1	The temperature of cold water inlet	ºC
T_cold_2	The temperature of cold water outlet.	ºC
N_X	Number of wave length.	—
N	Number of channels.	—
γ	Corrugation ratio	—
L_X	The corrugation length.	M
A	The heat transfer area	m^2
A_{p1}	Plate projected area.	m^2
F_{m1}	Modified fanning friction factor.	—
NTU	Number of Transfer UNIT	—
Cr	Heat capacity ratio	—
C	Heat capacity	m^2.kg/K.S^3
ε	The effectiveness of the PHE.	—
q	The heat transfer rate for the PHE.	W
ρ	The density	Kg/m^3
Cp	The specific heat.	J/kg.K
k	Thermal conductivity.	W/m.K
$\dot{m}$	The mass flow rate.	Kg/s
Pr	Prantle number	—

Subscripts:

t	total.
f	channel.
P	passes.
1	hot water.
2	cold water.
01	hot channels.
02	cold channels.

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