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Effects of wavelength on the thermal performance of a plate heat exchanger

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ABSTRACT

The Plate Heat Exchanger(PHE) consists of a series of parallel plates that are corrugated both to increase turbulence and to give mechanical rigidity . They normally have flow ports in all four corners and are clamped together in a frame that carries bushes and nuzzles lined up with the plate ports and connected to the external pipework that carries the two liquid streams .The plates themselves are fitted with gaskets, which are shaped and located both to prevent external leakage and to direct the two liquids normally counter currently through the relatively narrow passage between alternate pairs of heat transfer plates. One of the important feature of corrugated plate heat exchangers is their wavy surfaces. This feature has many advantages and of courses some disadvantages. For example, it enhances total heat transfer coefficient, but increases pressure drops and naturally running costs. The effects of wavelength on thermal performance of the heat exchanger was investigated in this study. This paper also presents a study on calculating wavelength, which can be useful in optimization process in order to find optimum value of the wavelength.

Introduction

Plate Heat Exchangers have many applications in the food, petrochemical, power plant, oil and chemical industry. Compared to other types of heat exchangers, such as shell and tube, plate heat exchangers are commonly used because of their compactness, ease of production, sensitivity, easy care after set-up and efficiency. Basically, the plate heat exchanger is a series of individual plates that pressed between two heavy and covers. Depending on the application the heat exchanger, these plates are gasketed, welded or brazed together. The pressed pattern on each plate surface induces turbulence and minimizes stagnant areas and fouling. Unlike shell and tube heat exchangers, which can be custom-built to meet almost any capacity and operating conditions, the plates for plate and frame heat exchangers are mass-produced using expensive dies and presses. Although the plate heat exchangers are made from standard parts, each one is custom designed as variation in the chevron angle, flow path or flow gap can alter the number of transfer units in the heat exchangers. Decreasing the chevron angle from 90^{o}, the path becomes more tortuous and offers greater hydrodynamic resistance giving rise to high *NTU* (The number of transfer unit) characteristics. Also, it is possible to use a combination of different plates to create an intermediate *NTU* passage, which can be used to meet a specific *NTU* requirement.

Focke.W.W [1] suggested that one of the important parameters in the thermal performance of the plate heat exchangers is the inclination angle between plate corrugations and the overall flow direction. Mehrabian and Pouter [2] investigated the local hydrodynamic and thermal characteristics of the flow between two identical APV SR3 plates. They also studied the effect of corrugation angle on the thermal performance of the heat exchanger when plate spacing is fixed. Laminar periodically developed forced convection in sinusoidal corrugated-plate channels with uniform wall temperature and single-phase constant property flows was considered by Metwally and Mbanglik [3]. Gradeck et.al, [4] conducted experiments in order to study the effects of hydrodynamic conditions on the enhancement of heat transfer. They performed these experiments for a wide range of Reynolds numbers. Finally, they pointed out a strong relation between the wall velocity gradient and the Nusselt number. Bobbili and Sunden [5] conducted experimental investigations to find the flow and the pressure difference across the port to channel in plate heat exchangers for a wide range of Reynolds numbers $(1000 \le \mathrm{Re} \le 17000)$.

Nomenclature

- A_t : corrugated area
- a : amplitude
- D_h : hydraulic diameter
- f : friction factor
- H_p : height
- h : heat transfer coefficient
- K_f : thermal conductivity
- N_c : number of channels per each fluid
- N_p : number of channels per pass
- N_t : number of plates
- N_λ : number of wavelength per plate
- PHE : plate heat exchanger
- Re : Reynolds number
- W_P : width

Greek symbols

- μ : dynamic viscosity
- β : chevron angle
- ϕ : surface enlargement factor
- γ : surface waviness
- λ : wave length
- δ : thickness

Longo and Gasparella [6] used water as a working fluid in herringbone type plate heat exchanger with the chevron angle of 65^{o}, and developed Nusselt number correlations. Warnakulasuriya and Worek [7] studied heat transfer and pressure drop of a viscous absorbent salt solution in a commercial plate heat exchanger. They presented some correlations for calculating Nusselt number and friction factor.

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The hydrodynamic characteristics and distribution of flow in two cross-corrugated channels of plate heat exchangers was investigated by Tsai and Liu [8].

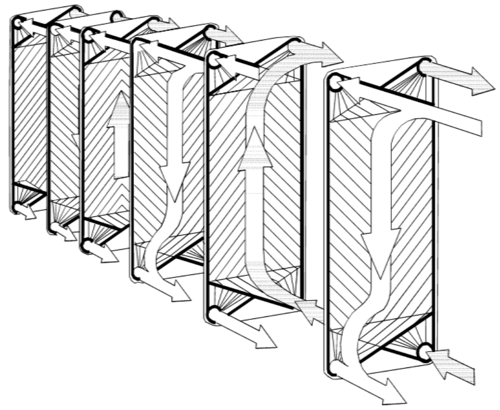

Fig.1. Nature of fluid flow through plate heat exchanger

Wang and Sunden [9] reported the optimal design of a PHE by adjusting corrugation patterns. An alternative design approach based on graphical representation, which facilitates the choice from the options calculated for the available plates was presented by Picon-Nunez, Polley and Jantes-Jaramillo [10]. They have estimated some correlations for heat transfer and hydraulic resistance from literature data. Also, Mehrabian [11] proposed a manual method for the thermal design of plate heat exchangers. Kanaris, Mouza and Paras [12] studied the parameters of Nusselt and friction factor correlations using CFD modeling of the flow in a PHE channel. The influence of varying plate corrugation pattern is achieved using combination of chevron plates with different corrugation inclination angle. The design approach and its advantages was offered by Marriot [13] for a one pass counter-current arrangement of PHE channels. Although the required heat ransfer load and pressure drops can be satisfied more efficiently by application of multi-pass arrangement of PHE's channels compared with one pass arrangement of channels, but this arrangement has many advantages in view of piping and maintenance. Kandlikar and Shah [14] studied different flow arrangements and proposed formulas for up to four passes, where the main assumptions made on driving such formulas are:

- Constant fluid properties
- Constant overall heat transfer coefficients
- Uniformity of fluid flow distribution between channels.
- Sufficiently large number of plates

This papers presents a study on the effects of the wavelength and number of plates on the PHE thermal performance. In the plate and frame heat exchanger due to some reasons Reynolds number, heat transfer coefficient, friction factor and especially heat transfer area are dramatically dependence to the wavelength. Here, to analyze more efficiently, a case study for cooling application has been done.

Development of mathematical modeling

One problem associated with the PHE design is the precise matching of both the thermal and hydraulic loads. It is difficult to accommodate the required thermal duty and at the same time fully utilize the available pressure drop, as the minimum cross-flow area and the surface area in a PHE are interdependent, unlike the in other types of heat exchanger. This limits the exchanger designed to be either pressure drop or heat transfer.

The wavelength λ of the chevron pattern is the corrugation pitch as shown in Figure.(1). The amplitude of the corrugation is denoted as $2a$, where a is the amplitude of the sinusoidal corrugation with the plate thickness δ. The number of wavelength per plate " N_λ " is calculated by dividing the width " W_P " by the wavelength " λ ".

$$N_\lambda = \frac{W_P}{\lambda} \tag{1}$$

Considering the front and end covers, it is possible to calculate the number of channels per each fluids.

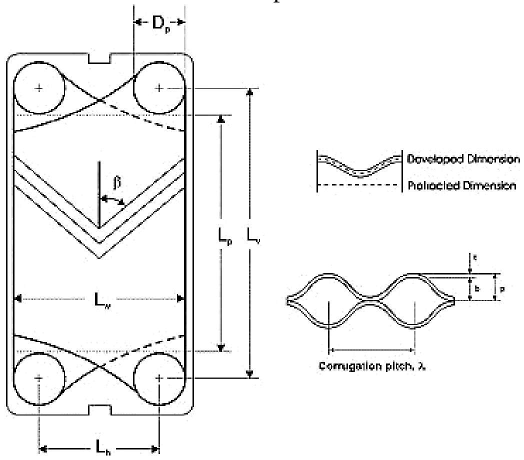

Fig. 2. Chevron-Type Plate

$$N_c = \frac{N_t + 1}{2N_P} \tag{2}$$

If N_t be introduced as total number of plates, then the amplitude " a " can be expressed in terms of the PHE height, " H_P ":

$$a = \frac{1}{2}\left(\frac{H_P}{N_t + 1} - \delta\right) \tag{3}$$

The surface waviness can be essentially represented by two dimensionless parameters, the corrugation aspect ratio " γ " and the surface enlargement factor " ϕ ". The corrugation aspect ratio " γ " is:

$$\gamma = \frac{4a}{\lambda} \tag{4}$$

It is noted that when $\gamma = 0$, a flat-parallel plate is obtained. So, $\gamma \rangle 0$ in indicative of surface area enlargement s well. Rising " γ " enlarges the surface area, but high " γ " may induce vortexes at the top and bottom of the channel which can trap the fluids locally and reduce the heat transfer. So, the plate heat exchanger design is commonly limited to $\gamma \langle 1$, depending on the Reynolds number.

The corrugated area is obtained using the path distance of the sinusoidal corrugation. It is noticeable that the chevron angle (β) has not any effects on the corrugated area. Therefore , the enlarged length per wavelength is :

$$L_\lambda = \int_0^\lambda \sqrt{1 + \left(\frac{2\pi a}{\lambda}\right)\cos^2\left(\frac{2\pi x}{\lambda}\right)} dx \tag{5}$$

So, the corrugated area for each fluid can be calculated by

$$A_t = 2L_\lambda N_\lambda L_P N_C \tag{6}$$

The free-flow area in a channel that is equivalent to a rectangular area for taking half of each fluid is calculated by,

$$A_c = 2aW_P N_c \tag{7}$$

The surface enlargement factor, ϕ, is expressed by

$$\phi = \frac{A_t}{A_c} = \frac{L_\lambda . N_\lambda}{W_p} \tag{8}$$

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

$$\phi \approx 0.16\left(1+\sqrt{1+x^2}+4\sqrt{1+\frac{x^2}{2}}\right) \tag{9}$$

In this formula, *x* is:

$$x = \frac{2\pi a}{\lambda} \tag{10}$$

$$D_h = \frac{4a}{\phi} \tag{11}$$

One of the important correlation for the friction factor is provided by Martin [3],

$$f = \left\{\frac{\cos\beta}{\left(0.045\tan\beta + 0.09\sin\beta + f_0 / \cos\beta\right)^{0.5}} + \frac{1-\cos\beta}{\sqrt{3.8 f_1}}\right\}^{-0.5} \tag{12}$$

Where,

$$f_0 = \begin{cases} \dfrac{16}{\mathrm{Re}} & for\ \mathrm{Re}\langle 2000 \\ (1.56\ln\mathrm{Re} - 3.0)^{-2.0} & for\ \mathrm{Re}\rangle 2000 \end{cases} \tag{13}$$

$$f_1 = \begin{cases} \dfrac{149.25}{\mathrm{Re}} + 0.9625 & for\ \mathrm{Re}\langle 2000 \\ \dfrac{9.75}{\mathrm{Re}^{0.289}} & for\ \mathrm{Re}\rangle 2000 \end{cases} \tag{14}$$

Here ,heat transfer coefficient is:

$$h = 0.205.\frac{K_f . L_\lambda . N_\lambda}{4a.W_P}.\mathrm{Pr}^{0.333}.\left(f.\mathrm{Re}^2 \sin 2\beta\right)^{0.374}.\left(\frac{\mu}{\mu_s}\right)^{1/6} \tag{15}$$

Where $10^\circ \langle \beta \langle 80^\circ$ and K_f is considered as thermal conductivity of the fluid. μ_s is also dynamic viscosity at the wall temperature.

Case study

To analyze the influence of the chevron angle on th thermal performance of corrugated plate heat exchanger a case study has been done. In the heat recovery application a cold water will be heated by wastewater using mentioned plate heat exchanger. The cold water with a flow rate of 130 kg/s enters the plate heat exchanger at 22 C°, and it will be heated to 42 C°. The hot wastewater enters at a flow rate of 140 kg/s and 65 C°. The maximum permissible pressure drop for each fluid is 70Kpa. The chevron plates rae single-pass with the chevron angle of 30°. The metal of the plates is stainless steel AISI 304.

Here, we face a sizing problem and then a rating problem to determine T, q, ε and ΔP.

Table.1.
Designation information

Description	Value
Number of passes,	1
Chevron angle	30°
Total number of plates	109
Plate thickness	0.6 mm
Corrugation pitch	9 mm
Port diameter	200 mm
Thermal conductivity	14.9 W/m.K

After designing process, the thermal and hydraulic quantities are according to the Table of (2). Table of (3) and (4) also show the Geometry properties and outlet temperature and pressure drops for designed heat exchanger respectively.

With increasing N_t, the surface area density increases while γ decreases. Consequently, the volume of the heat exchanger decreases with increasing the surface area density or number of plates.

Table.2.
Thermal results

Description	Value
Hydraulic diameter	5.734 mm
Effectiveness	0.465
Heat transfer rate	1.087×10^7 W
UA Value	4.581×10^5 W/K
Convective coefficient for hot stream	1.787×10^4 W/m^2.K
Convective coefficient for cold stream	1.242×10^4 W/m^2.K
Reynolds number for hot stream	1.501×10^4
Reynolds number for cold stream	7.642×10^3
Friction factor for hot stream	0.1
Friction factor for cold stream	0.102

Table.3
Geometry result

Description	Value
Number of passes	1
Number of plates	109
Corrugation pitch	9 mm
Plate thickness	0.6 mm
Chevron angle	30
Port diameter	200 mm
Corrugation aspect ratio	0.874
Surface enlargement factor	1.372
Surface area density	697.612 m^2/m^3
Heat transfer area for each fluid	80.821 m^2
Amplitude	1973 mm

Table.4.
Outlet temperatures and pressure drops

Description	Value
Cold water outlet temperature	42 C°
Hot wastewater outlet temperature	46.45 C°
Pressure drop for hot wastewater	70 Kpa
Pressure drop for cold water	60.857 Kpa

Analysis and discussion

Equation of (5) express a formula to calculate heat transfer area. From geometry view, the plates are sinusoidal curve, so calculating the length of this sinusoidal curve is necessary. The length of a curve is:

$$\int_0^a \sqrt{1+[f'(x)]^2}dx \tag{16}$$

Figure of (3) shows the sinusoidal curve of these plates.

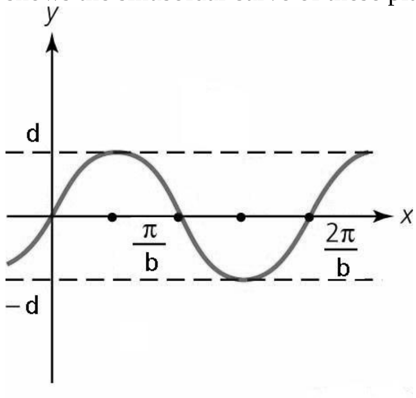

Fig .3. Sinusoidal curve of Plates

The equation of (5) is a elliptical integral, which its solution is:

$$L_\lambda = \int_0^\lambda \sqrt{1+\left(\frac{2\pi a}{\lambda}\right)\cos^2\left(\frac{2\pi x}{\lambda}\right)}dx = \frac{\sqrt{\left(\frac{2\pi a}{\lambda}\right)+1}}{\left(\frac{2\pi}{\lambda}\right)} E\left(\left(\frac{2\pi x}{\lambda}\right) \Big| \frac{\left(\frac{2\pi a}{\lambda}\right)}{\left(\frac{2\pi a}{\lambda}\right)+1}\right) \tag{17}$$

Terry.L Brown [15] offered a solution for incomplete elliptical integral of the first and second kind. Based on his research,

$$E(\phi|m) = \int_0^\phi \sqrt{1-m\sin^2(\theta)}d\theta = \frac{2\phi}{\pi}E(m) + \sin(\phi).\cos(\phi) \times[\frac{1}{2}A_2 m + \frac{1}{2.4}A_4 m^2 + \frac{1.3}{2.4.6}A_6 m^3 + ...] \tag{18}$$

$$A_2 = \frac{1}{2} \quad , A_4 = \frac{3}{2.4} + \frac{1}{4}\sin^2(\phi) \quad , \quad A_6 = \frac{3.5}{2.4.6} + \frac{5}{4.6}\sin^2(\phi) + \frac{1}{6}\sin^4(\phi) \tag{19}$$

Here, *E(m)* is the complete elliptical integral of second kind. However, A.Narayan (2012)suggested a numerical approximation methods , based on series expansions. Based on his research a estimation of the incomplete elliptical integral of second type is:

$$E(\phi|m) = \int_0^\phi \sqrt{1-m\sin^2(\theta)}d\theta = \int_{t_2}^{t_1} R(t).dt = R(t).(t_1 - t_2) \tag{20}$$

$$R(t) = \frac{\sqrt{(x_2-x_1)^2 + (y_2-y_1)^2}}{2.\sin\left(\frac{(t_1-t_2)}{2}\right)} \tag{21}$$

Figure of (4) demonstrates the function of heat transfer area versus the wavelength. Although we expect that heat transfer area has a reverse relation with the wavelength, but the results denote that with increasing the wavelength, the heat transfer area will increase dramatically. To investigate its reasons, it is necessary to notice that heat transfer area is a function of number of plates as well. Figure of (5) shows this function. Here, similar to previous figure, the heat transfer function has a direct relation with number of plates. However, pressure drops impose some limitations for selecting optimal wavelength and number of plats values. In fact, it depends on where PHE will be employed.

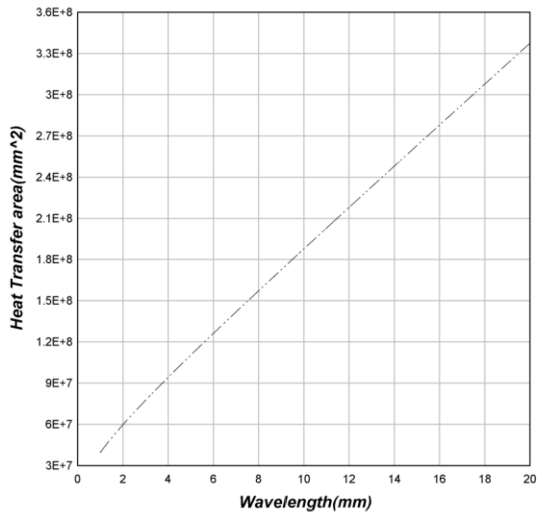

Fig.4. Heat transfer area versus Wavelength

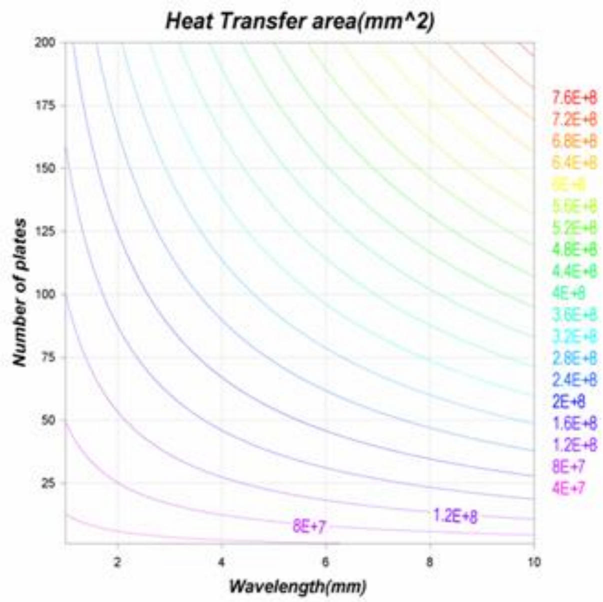

Fig.5. Heat transfer area versus Wavelength and number of plates

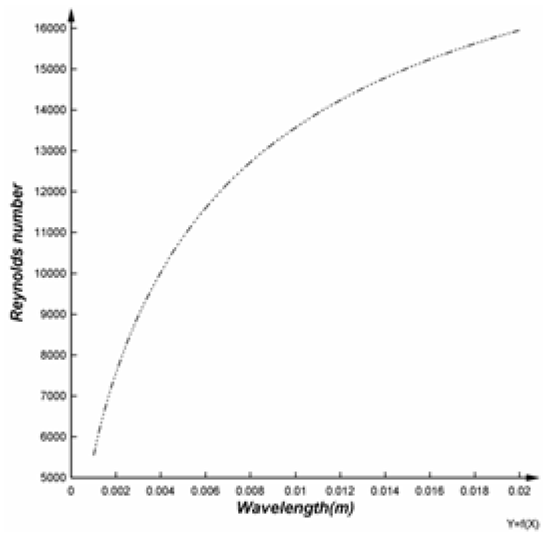

Fig.6. Effects of the wavelength on the Reynolds number

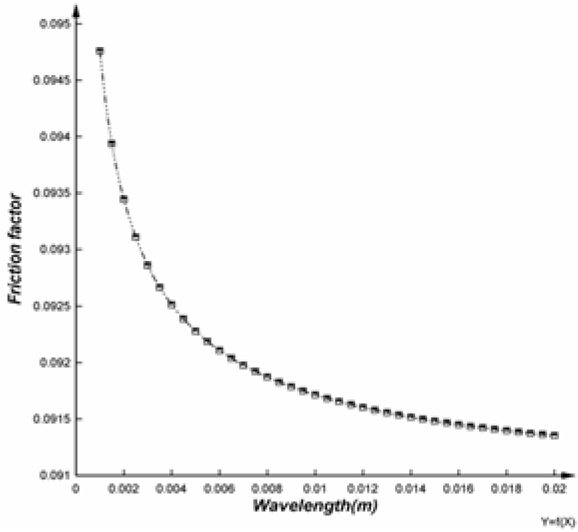

Fig.7. Effects of the wavelength on the Friction factor

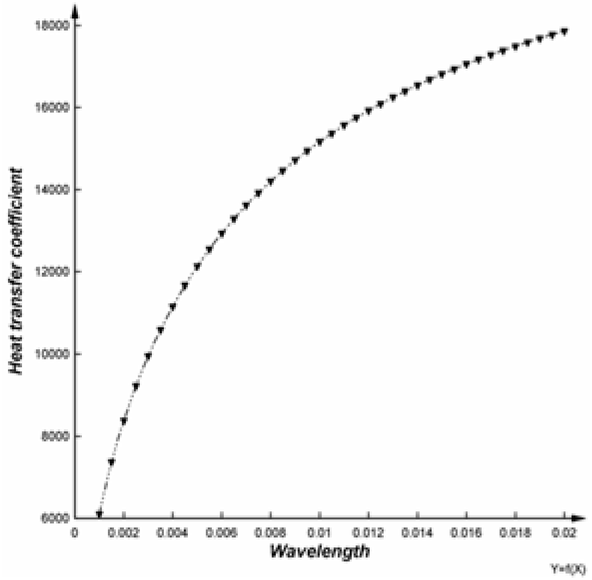

Fig.8. Effects of the wavelength on the heat transfer coefficient

As it is clear from the figure of (6) the Reynolds number will increase with growing the wavelength, consequently heat transfer coefficient have to rise this figure denotes that friction factor has increased with growing the wavelength . So, pressure drops for both fluids decrease. What it is important is that when the wavelength is *14 mm* and more, the friction factor is constant approximately. On the other hands, in this limit, heat transfer coefficient did not change very much. Therefore, it is reasonable that the value of *12-14 mm* for the wavelength is a optimal value(of course in this case study).

Conclusion

Plate and Frame heat exchangers are one of common types of the heat exchangers in industry. Many researchers studied about the effects of chevron angle, hydrodynamic conditions, and Nusselt numbers on the different thermal properties of the heat exchanger. However, the influence of wavelength on the Friction factor, heat transfer coefficients, heat transfer area and heat Reynolds number was remained unexplored. This matter when will be important that our objective is to optimize the heat exchanger and find optimum dimensions of the heat exchanger. The results showed that when the number of plates is below 150, the optimum value of the wavelength to have maximum heat transfer area is between 6 and 10 mm.

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