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Insights into Design Parameters to Improve Gasketed-Plate Heat Exchanger Performance

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The current study presents the effects of various plate parameters, such as; type and plate geometries. Chevron plate characteristics such as chevron angle, port diameter, and channel spacing were presented. The plate fabrication material was also examined based on its thermal conductivity, and the effect of the fluid's physical properties was investigated through an industrial case study. PHEx® software was used as a computational tool to analyse and show each parameter's effect through the simulation of an industrial case study. The selection of the ratio of vertical and horizontal lengths (L_v/L_h) is a key variable in the design and any optimization stage. The varying port diameter is not a viable option for reducing pressure drop. The plate fabrication material, in terms of increasing thermal conductivity, was ineffective in reducing pressure drop but effective in reducing the number of plates. A better understanding of the role of each parameter on the overall exchanger design was achieved to attain the required target in the allowable pressure drop of the exchanger system.

1. Introduction

Plate heat exchangers are highly favoured due to their large surface area per unit volume and high heat transfer efficiency. Several extensive reviews were conducted on the developments and thermal and hydraulic enhancements of such types of exchangers (Abu-khader, 2012; Kumar et al., 2022; Arsenyeva et al., 2023). Delgado-García et al. (2022) developed new generalized correlations for various plate heat exchager design options. Abou Elmaaty et al. (2017) studied the heat exchanger's best performance when there is a high heat rate at low pumping power with a minimum cost. Experimental and numerical analyses were conducted by Martins, et al., (2022) to evaluate the structural behaviour of plate heat exchangers with 316 L stainless steel plates. An exergy-economic approach using the Harris Hawks optimization method was implemented on gasketed plate heat exchangers to increase their efficiency by 30 % (Alavi et al., 2023). The plates have marked corrugated designs that offer a greater surface area for effective heat transfer. They alter the flow dynamics from laminar to turbulent to improve thermal-hydraulic performance. Intermating and chevron corrugations are the two available types in use. However, the chevron plates are commonly used in industry due to the greater heat transfer enhancement at increased pressure drops. Such types of exchangers are efficient, robust, and easy to clean (Wang, et al. 2007). Due to the large variety of enhancements, it is quite difficult to be fully supported by commercial and academic simulators. In addition, plate manufacturers will never release their critical exclusive data (Hesselgreaves, et al. 2016). It is important to remember that adding extra plates is not always a simple way out or has a beneficial effect on targeting the required exchanger duty. Even though adding plates provides higher heat transfer more flowing channels reduce velocities and lower exchanger performance. Furthermore, the higher fluid velocity causes an elevated pressure drop. Such behaviour forms a complex trade-off relationship with the overall capital cost. Therefore, understanding the geometry and design parameters is a vital issue. The chevron plate modifications play an important role in meeting application specifications, and the required performance and duty. The main objective of the present study is to gain better insights into the various plate parameters, such as; chevron angles, channel spacing, plate heights and type on heat transfer and pressure drop calculations.

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2. Plate Heat Exchanger Design

Due to the complexity of the design process of heat exchangers, there is a need for subjective decisions at each intermediate step. Also, the design process involves several steps and the use of tentative data until the design targets are met. Generally, the design process of a heat exchanger includes the following elements: heat transfer to achieve the required duty, the capital cost, the actual physical size, and the overall pressure drop. In the present work, PHEx® software is used. It is a computational tool for either rating or sizing plate heat exchangers. The tool can conduct an important sensitivity analysis before going into an optimization stage to solve a specific problem. The software illustrates the effect of several plate parameters' effect on transferred heat and pressure drop when using the proper correlations for various flow conditions (Palmeira et al. 2014).
The tool methodology assumes either the total effective area or the number of plates as variables for design program calculations. In addition, fouled and clean values are assumed for heat transfer coefficient calculation. The allowable maximum pressure drop must be considered. The surface area is then determined to calculate the required heat duty. This is done by employing the given temperature difference and not exceeding the maximum pressure drops allowed in the design specification. The exchanger pressure drop is the total sum of the pressure drop from the plate channel, which is a function of plate geometry, both fluids used and the pressure drop from the port.
A typical industrial unit of plate heat exchanger has a size range from 1540 to 2500 m^2 containing up to 700 plates with port sizes up to 39 cm. Each plate has a thickness range of 0.5 to 1.2 mm and a size range of 0.03 to 2.2 m with a plate spacing of 1.5 to 5.0 mm. Also, it has a corrugation depth of 3 to 5 mm. The plates are classified based on their chevron angle (β) from 30 up to 65. Plates with chevron angles ≤ 45 are characterized as soft plates. Whereas, plates with < 45 angles are hard plates (Abu-Khader, 2007).
Polley and Abu-Khader (2005) emphasized that the end plate effects are significant when the exchanger has many plates. The middle-packed plates are characterized as follows (Wang et al. 2007): a) very small spacing between plates to maximize thermal contact, b) high vertical and horizontal lengths to provide effective surface area and high heat transfer rate, c) the material of construction of plates plays an important role in thermal performance, and, d) high turbulence flow generates high local heat transfer coefficients and reduces fouling. The fabricated material has a direct effect on heat transfer calculations. Plate material differs from one manufacturer to another. There is a wide variety of metals used in the fabrication of these plates, such as AISI304 SS, AISI316SS, Inconel 600, Titanium, Nickel 600, Brass, and others. Besides being highly corrosive resistant, these materials have a wide range of thermal conductivities ranging from 16 to 100 W/m K keeping in mind that carbon steel is rarely used due to its low corrosion resistance.
One of the rules of thumb is to maintain high fluid velocity within the exchanger but not more than 7 m/s through a pipe reducer to 200 mm at the hole inlet of the exchangers. High velocity promotes a high local heat transfer coefficient, giving the exchanger an elevated overall heat transfer coefficient. Also, there is a decreasing relationship between the cost of the plate heat exchanger and the overall pressure drop at fixed duty. For example, low cost is found at a high 100 kPa allowable pressure drop, and reducing the pressure drop to 30 kPa could increase the price up to 35%. This percentage differs based on the exchanger type, size, and manufacturer (Wang et al. 2007).
One of the important themes of the exchanger design is the flow arrangement. Counter-current flow is usually selected to obtain the highest logarithm temperature difference (LMTD) to perform the required duty. It is considered true counterflow, giving high heat transfer efficiency for the plate. The smaller the ratio ($\Delta T1/\Delta T2$) of the temperature difference ($\Delta T = T_{hi} - T_{co}$) between the inlet and outlet streams of both sides of the counter flow arrangement, the greater the difficulty of performing the duty of the exchanger. Therefore, a larger surface area is required for the lower temperature difference ratio. This leads to higher exchanger costs. The minimum temperature difference approach is related to the amount of heat recovery in the system, indicating the inefficiency of the exchange equipment. The cold process stream must possess a higher heat capacity to optimize the energy recovery from a hot process stream (Heggs, 1989).

3. Industrial Case Study

The heating of a cold raw water stream using a hot distilled water stream is carried out using a plate heat exchanger. The hot distilled water stream has a flow rate of 180000 kg/h that enters at a temperature of 305K and leaves at 298K. The cold stream has an unknown flow rate and its inlet/outlet temperatures are 293K and 295.3K respectively. The maximum allowable pressure drop for both fluid streams is 100 kPa. Table 1 provides the exchanger design specifications for determining thermal and hydraulic performances. Table 2 illustrates the outcome results of the distilled water/raw water system. Conversely, the pressure drop analyses showed a clear deviation in the cold stream where the calculated maximum allowable pressure drop was exceeded. Therefore, the next step is optimizing the design by adding more plates to increase the heat transfer area and correcting design errors. To overcome the problem of exceeding the allowable pressure drop in one of the streams, adding extra plates may help reduce the pressure drop to the required limits.

Table 1: Design parameters for the industrial case study (Kakaç & Liu 2002)

Plate Material	Stainless steel (SMO-254)	Port diameter, Dp (mm)	200
Plate thickness (mm)	0.6	Vertical port length, L_v (m)	1.55
Chevron angle (degrees)	45	Horizontal port width, L_h (m)	0.43
Total number of plates	105	Channel spacing (mm)	2.83
Enlargement factor	1.25	Plate material thermal conductivity (W/m^2.K)	17.5
Number of passes	1/1	Total effective area (m^2)	110

Table 2: Outcome results of the distilled water/raw water system

Stream Properties	Hot Stream		Cold Stream	
Name	Distilled water		Raw water	
Mass Flowrate (kg/h)	180000		547446.16	
Mass Flow per channel (kg/s)	0.96		2.92	
Velocity through port (m/s)	1.60		4.85	
Reynolds Number Re	2978.3		7578.59	
Prandtl Number Pr	5.64		6.80	
Nusselt Number Nu	122.47		249.86	
Port mass Velocity [kg/(m^2.s)]	1591.49		4840.50	
Number of Passes	1		1	
Heat Transfer Coefficient [W/(m^{2}K)]	16610.15		33336.23	
Velocity	Port	Channel	Port	Channel
	1.597 m/s	0.541 m/s	4.849 m/s	1.643 m/s
Pressure Drop	Allowable	Calculated	Allowable	Calculated
	100 kPa	57.070 kPa	100 kPa	437.450 kPa
Total Effective Area	110.000 m^2	Surface Area		47.36 m^2
Channel Flow area (m^2)	0.000178	Number of Channels per Pass		52
Heat Duty	1,462.545 kW			
LMTD Corrected	7.092 K			
Overall Heat Transfer Coefficient	Design		Clean	
	5,035.58 W/m^2 K		7,215.69 W/m^2 K	

All figures show that the increase in the number of plates reduces the total allowable pressure in both cold and hot streams and increases the exchanger's surface area. Furthermore, the selection of a certain chevron angle plays a crucial role. In Figure 1, a chevron angle of 30 promotes a higher total pressure drop in both streams. Whereas, the use of larger angle of 45 or 60 reduces the total pressure drop dramatically. When using chevron plates with 30, 45, and 60, the required plates to be added are 423, 256, and 172 respectively, to reach the allowable pressure drop in the cold stream. This concludes that the use of a high chevron angle has a high impact on lowering the high-pressure drop stream. Such an impact effect is reduced when using low-chevron angle plates. It is noticed that high-chevron angle plates are more sensitive than low-angle plates for pressure drop conditions. Therefore, the high-chevron angle should be named as soft plates and the low one as hard

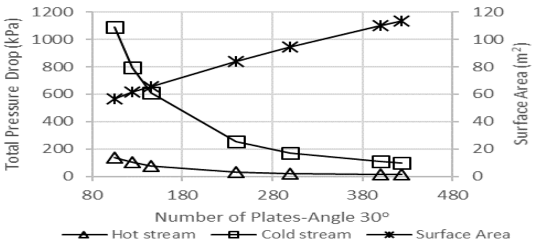
plates. Whereas, in heat transfer calculations, low-chevron angles become more sensitive plates (soft plates) in providing higher values of the overall heat transfer coefficients than high-chevron angles (hard plates).
The channel spacing is another important geometric property that has a direct impact on the plate heat exchanger's performance. The typical range used in the design is between 1.5 to 5 mm. Figure 2 illustrates the effect of channel spacing on hydraulic performance. The increase in channel spacing contributes significantly in reducing the total pressure drop in both streams.
Both vertical and horizontal distances (L_v and L_h) from port to port are vital in manipulating the stream pressure drop. Each of these parameters has the opposite effect on the system. The increase in the vertical promotes an increase in the total pressure drop of both streams especially the cold one as shown in Figure 3. Conversely, the increase in the horizontal length effectively reduces the total pressure drop of both streams, as shown in Figure 4. The ratio of L_v/L_h can be used as an effective tool to regulate the total pressure drop of both streams. For example, a ratio of 2.15/1.2 in the presented case effectively reduced the pressure drop to the allowable limits. The variation in plate port diameter was ineffective in overcoming the high total pressure drop where the cold stream could not reach the allowable pressure drop limits of 100 kPa even when using the maximum port diameter of 400 mm (Figure 5).
In Figure 6, fabrication materials with different thermal conductivity have a direct effect on reducing the required surface area, but with no impact on the pressure drop of both streams. It is interesting to notice that the use of materials with a thermal conductivity of 16 to 45 W/m k has a noticeable reduction in the surface area reaching 10%. Whereas, conductivity ranges from 45 to 100 W/m.k the reduction reaches 4%. A critical trade-off appears between the cost of fabricated material and the cost of reduced surface area. An optimised search is needed.
The fluid physical properties play a vital role in the exchanger design. Therefore, the raw water (the cold stream) was replaced with an organic liquid of Ethylene glycol. The calculated results are shown in Table 3.

Table 3 Calculated results with the cold side as Ethylene Glycol.

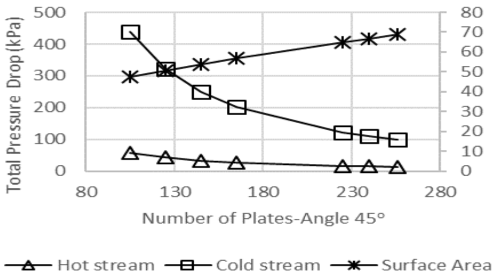
Allocation	**Hot Side**		**Cold Side**	
Fluid Name	Distilled water		ETHYLENE GLYCOL	
Mass Flowrate	180000 kg/hr		867121.5 kg/hr	
Temperatures (In/Out)	305 K	298 K	293 K	295.3 K
Density	996.41 kg/m^3		1079.98 kg/m^3	
Heat Capacity	4178.7 J/kg K		2640 J/kg K	
Viscosity	0.00082 kg/m s		0.00392 kg/m s	
Thermal Conductivity	0.6144 W/m K		0.258 W/m K	
Fouling Factors	0.00005 m^2 K/W		0.00001 m^2 K/W	
Heat Transfer Coefficient	1.12 x10^4 W/m^2 K		5.13 x10^3 W/m^2 K	
Velocity	Port	Channel	Port	Channel
	1.597 m/s	0.541 m/s	7.099 m/s	2.406 m/s
Pressure Drop	Allowable	Calculated	Allowable	Calculated
	100 kPa	57.07 kPa	100 kPa	1,220.21 kPa
Area (Effective)	110.000 m^2		Area (Surface)	78.057 m^2
Heat Duty	1,462.545 kW			
LMTD Corrected	7.092 K			
Overall Heat Transfer Coefficient	Design		Clean	
	2,641.99 W/m^2 K		3,139.69 W/m^2 K	

Figure 7 illustrates the effect of using two different cooling fluids. Raw water (light stream) could reach the allowable pressure drop with much fewer chevron plates around 260 plates. On the other hand, Ethylene glycol (viscous stream) as a cold stream took more than 520 plates to reach pressure drop limits. In Figure 8, more than 500 chevron plates of 30 and 45 angles, and a much smaller number of plates for chevron angle 60 are added to reach the allowable pressure drop of 100 kPa. Also, sudden drop when adding 430 plates to the system without reaching the allowable pressure drop. The sudden drop is due to an unexpected drop in the exchanger's performance (effectiveness).

a)



b)



c)

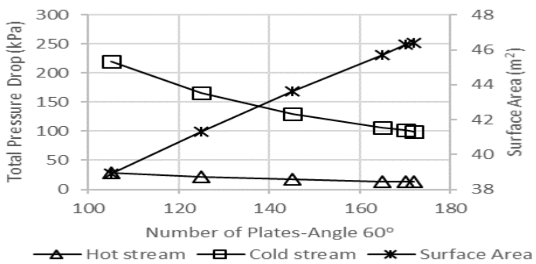


Figure 1 Number of plates with chevron angles 30, 45,60 (a,b,c)

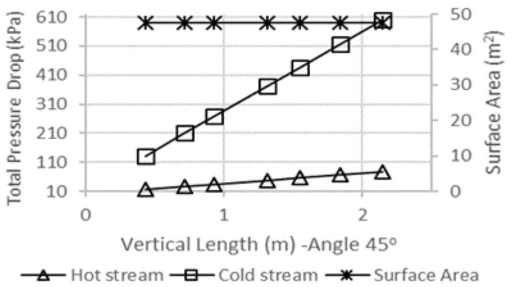


Figure 3 Vertical length effect with an angle 45 and horizontal length (L_h = 0.43 m)

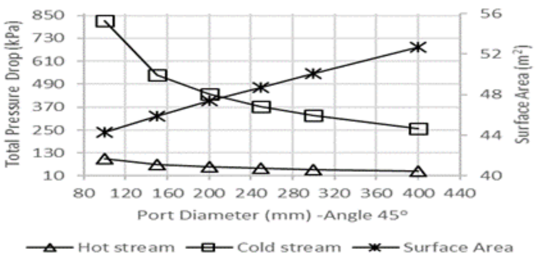
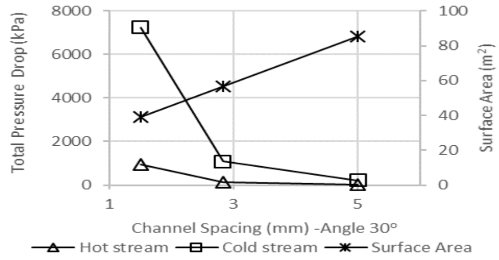
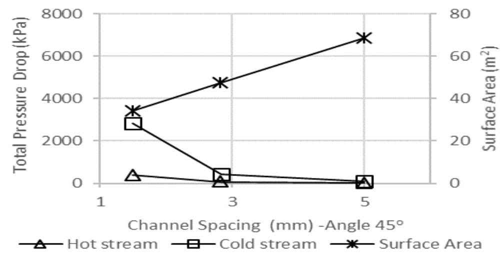


Figure 5 Port diameter effect with chevron angles 45.

a)



b)



c)

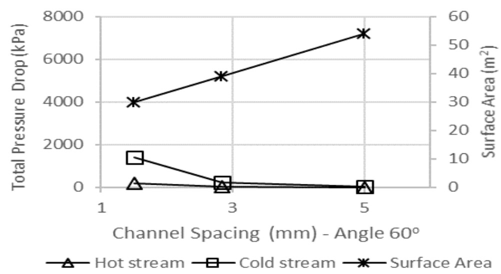


Figure 2 Channel spacing effect with chevron angles 30,45,60 (a,b,c)

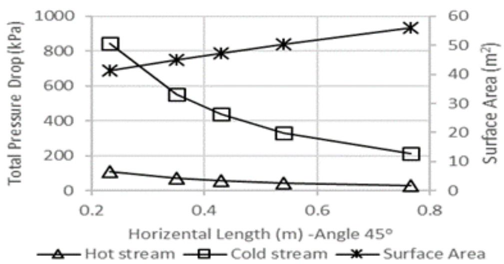


Figure 4 Horizontal length effect with an angle 45 and vertical length (L_v = 1.55 m)

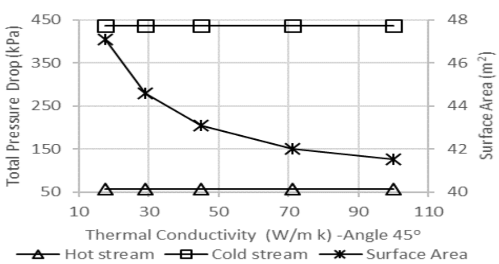


Figure 6 Thermal conductivity effect with chevron angle 45

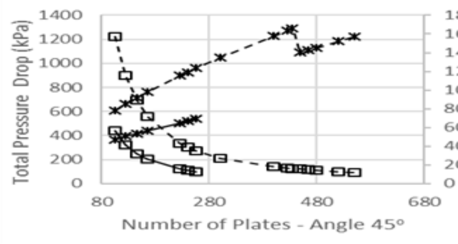


Figure 7 Number of plates effect (E.G. vs. raw water)

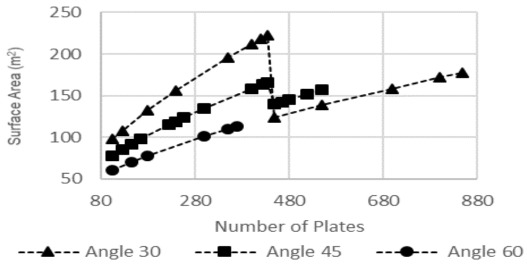


Figure 8 Number of plates effect with chevron angles 30,45, 60 (E.G. case)

4. Conclusions

In the selected case study, several physical variables are involved in plate heat exchangers that sensitively interact and affect the exchanger's performance. Low-chevron angles can be considered soft plates for thermal performance, hard plates for hydraulic performance, and vice versa for high-chevron angle plates. This was clear when trying to reduce system pressure drop to allowable limits a smaller number of plates are added when using a chevron angle plate of 60 and more plates are required when using a 30 plate. The increase in channel spacing from 1.5 mm to 5 mm reduces the pressure drop of the streams, especially when using a high-chevron plate angle. The vertical and horizontal lengths have an opposite effect on the system pressure drop. The vertical length (L_v) from 0.5 up to 2.2 m has an increasing effect, while the horizontal length (L_h) from 0.25 up to 0 .77 mm has a decreasing effect. It is interesting to notice that a better ratio of L_v/L_h can be used in any optimization stage. It can be concluded that varying port diameter is not a viable option for reducing pressure drop. The change in the plate fabrication material, in terms of increasing thermal conductivity, was ineffective in reducing pressure drop, even though it was the only parameter that reduced the number of plates. It is better to have good exchanger performance rather than a large amount of heat transfer area.

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