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Exploring Plate Heat Exchanger Design Options Using Generalised Correlations

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The thermohydraulic performance of the surfaces used in plate heat exchangers are directly related to the surface geometrical features which have an impact on the size of a heat exchanger. In the case of chevron corrugations, the variables are the chevron angle, pitch, and height. Over the years, generalised correlations for plate surfaces have been produced but still there are differences in the predictions when compared to experimental data. There is not a single correlation that suits all the applications. This situation justifies the development of new correlations based on the model of limiting solutions that incorporates the laminar and turbulent regions into a single expression to be used in design. In this work, new alternative generalised correlations for the Colburn and friction factor including the chevron angle effect are derived and compared with experimental data to assess its accuracy; average absolute errors of 3.98 % and 4.50 % for *f* and *j* are obtained in the range of 30 ° ≤ β ≤ 80 °. These correlations are implemented in a shortcut design approach where a graphical representation of the design options, referred to as parameter plot, is used as an aid in the design process. Two feasible designs are obtained: for β = 35 ° an area of 35.37 m^2 with a pressure drop of 12.05 kPa, and for β = 40 ° an area of 34.69 m^2 and 18.68 kPa where it can be appreciated that choice of the final design depends mostly on the operational costs.

1. Introduction

Plate Heat Exchangers (PHE) are a heat exchanger technology that finds wide applications due to their geometrical and operational features. They exhibit large heat transfer area per unit volume which makes them small and light for a given application. As its thermal and hydraulic performance depends on the corrugation geometry of the plates, there is not much information on the heat transfer and friction performance in the open literature due to the proprietary nature. Some of the research carried out is the work of Nilpueng et al. (2018), who experimentally studied the thermal performance based on the chevron angle and the roughness of the surface, finding that the highest thermal performance was with a Chevron angle of 30 ° and the highest roughness at low Reynolds. Based on experimental data, he proposes correlations for the Nusselt number and friction factor. Other works include the one carried out by Zhu and Haglind (2020) who studied the friction factor in crossed corrugated channels and determined a correlation for the friction factor in the laminar and turbulent regime. They also presented expressions to identify the zone of separation regime. Qingchan et al. (2021) developed a numerical simulation using plates with wavy channels with three corrugation angles and found that the thermal performance of the lower plate is higher than that of the upper one due to the presence of vorticity. Optimum performance for all three corrugated channels occurs for an offset angle of 324 °. Ham et al. (2021) numerically studied the phenomenon of temperature distribution and the phenomenon of stratification in channels. Korobiichuk et al. (2022) also performed a numerical study of a new geometry for the heat exchanger plate with conical stampings. He carried out the optimization of the new surface, finding that the optimal geometric parameters are a cone height of 1.5 mm and a cone inclination of 55 degrees. Few studies on the development of generalised correlations for plate heat exchangers using the approach of limiting solutions have been reported. Kapustenko et al. (2011) used the Reynolds analogy for tubes and predicted the heat transfer coefficients of different plate corrugations. The errors found do not exceed 15%. Arsenyeva et al. (2011) studied

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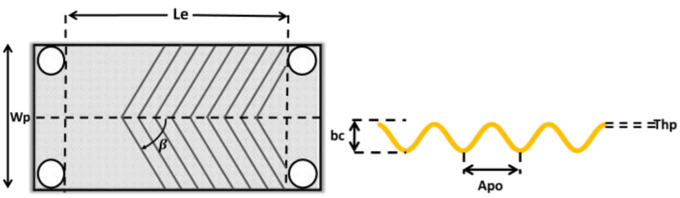
the effect of corrugation geometry using expressions for straight tubes in criss-cross flow channels for a wide range of corrugation parameters. Although research has been focused on improving the thermohydraulic performance of heat transfer plates, there is a need for techniques that allow the development of better prediction models of the Colburn factor (j) and friction factor (f) in plate heat exchangers, as well as design techniques based on the use of generalized correlations. This work focuses on the development of alternative generalized expressions based on reported experimental data. The methodology uses the limiting solutions model to generate correlations that give improved predictions of the friction and Colburn factors by incorporating the thermohydraulic data of the laminar and turbulent region for heat transfer plates into a single conjugated expression. The correlations are presented as a function of the Reynolds number and the Chevron angle in the range between 30 ° and 80 °. An alternative design methodology for plate and frame exchangers is presented and applied to a case study.

2. Generalized correlations

This section introduces the methodology for the development of new generalised correlations for plate heat exchangers. The enlargement factor ϕ is a dimensionless parameter used to characterize the corrugation and includes the height and corrugation pitch, the expression to calculate is proposed by Zhu and Haglind (2020). The geometric parameters of the plates and the expression for the elongation factor are shown in Figure 1.



$$\phi = \frac{1}{6}\left(1 + \sqrt{1 + \left(\frac{b_c \pi}{A_{po}}\right)^2} + 4\sqrt{1 + \left(\frac{b_c \pi}{A_{po}}\right)^2}\right)$$

L_e = Effective Length (m)
W_p = Plate width (m)
b_c = Corrugation height (m)
A_{po}= Corrugation pitch (m)
β = Chevron angle (°)
T_{hp} = Plate thickness (m)

Figure 1: Plate main geometrical parameters

2.1 Performance prediction model

A model that has demonstrated to produce accurate predictions in processes characterised by asymptotic behaviour is the one proposed by Churchill (2000). The model known as limiting solutions, applies well to heat transfer processes. The model predicts the value of the function *y*, based on the *nth* root of the sum of the *nth* powers of the limit solutions for small (y_0) and large (y_∞) values of the independent variable. This can be expressed as:

$$y = [(y_0)^n + (y_\infty)^n]^{1/n} \tag{1}$$

The model is applied for the analysis of the friction factor *f*, as follows:

$$f = [(f_{lam})^n + (f_{tur})^n]^{1/n} \tag{2}$$

The terms f_{lam} and f_{tur} are the friction factor in the laminar and turbulent zones, *n* is an arbitrary prediction exponent. Similarly, the Colburn factor can be studied using the proposed prediction model:

$$j = [(j_{lam})^n + (j_{tur})^n]^{1/n} \tag{3}$$

The study developed by Focke et al. (1985) contains experimental information on the thermohydraulic performance of commercial plates. The methodology used in this work is applied to these experimental data to determine generalised expressions. The experimental data is presented in the entire flow region, so it is necessary to divide the laminar and turbulent zones. The expression proposed by Zhu and Haglind (2020) is used to calculate the critical Reynolds.

$$Re_c = 954\text{Cos}(\beta^4) + 53 \tag{4}$$

2.2 Development of conjugated approach

The expressions that best fit the experimental data in the laminar and turbulent regions are determined. Then, the conjugate model is applied, and a solution is obtained in the full range of the Reynolds number. Since potential expressions are found to be suitable mathematical formulations to describe the thermal performance of secondary surfaces, this work also uses these types of expressions. The experimental data is fitted to:

$$f_{lam} = a_L Re^{b_L} \tag{5}$$

The variable f_{lam} is the friction factor in the laminar zone, Re is the Reynolds number, a_L and b_L are the coefficient and exponent of the fitted expression and depend on the chevron angle. Similarly, for the friction factor in the turbulent zone a potential fit is proposed:

$$f_{turb} = a_T Re^{b_T} \tag{6}$$

The terms a_T and b_T are the coefficient and exponent of the fitted expression in the turbulent region and they depend on the geometry of the plate. The accuracy of the predictions can be determined calculating the simple absolute error of the prediction for the friction and Colburn factor as follows:

$$e_{fp} = \left|\frac{(f_{Exp} - f_M)}{f_{Exp}}\right| \tag{7}$$

$$e_{jp} = \left|\frac{(j_{Exp} - j_M)}{j_{Exp}}\right| \tag{8}$$

In the expressions above e_{fp} and e_{jp} are the average absolute error of the prediction for the friction and Colburn factors. The *Exp* subscript corresponds to the experimental value and *M* represents the proposed prediction model. Figure 2 shows the fit to the experimental data taken from Focke et al. (1985) in the laminar and turbulent region for an angle β = 30 °.

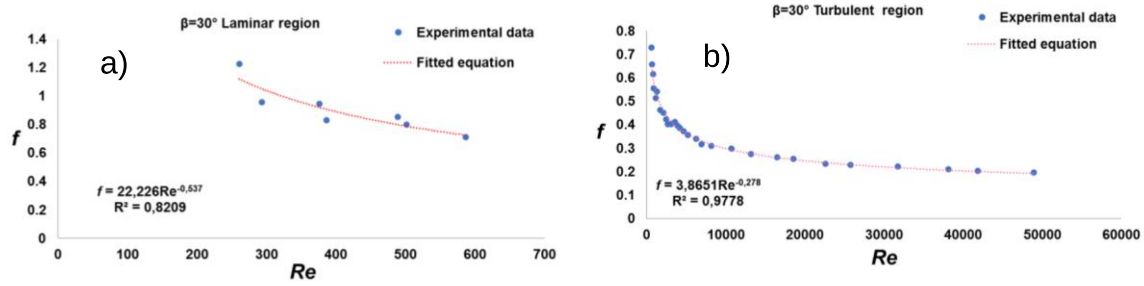


Figure 2: f vs Re for heat transfer plate with $\beta = 30$ ° a) laminar region, b) turbulent region

Developing a similar analysis, a solution for the Colburn factor in the laminar j_{lam} and turbulent j_{tur} region is proposed as shown in Eq(9) and Eq(10).

$$j_{lam} = c_L Re^{d_L} \tag{9}$$

$$j_{tur} = c_t Re^{d_t} \tag{10}$$

In the expressions above c_L, c_T, d_L and d_T are the adjustment coefficients and exponents. Figure 3 shows the proposed fits for the thermohydraulic data in the laminar and turbulent regions.

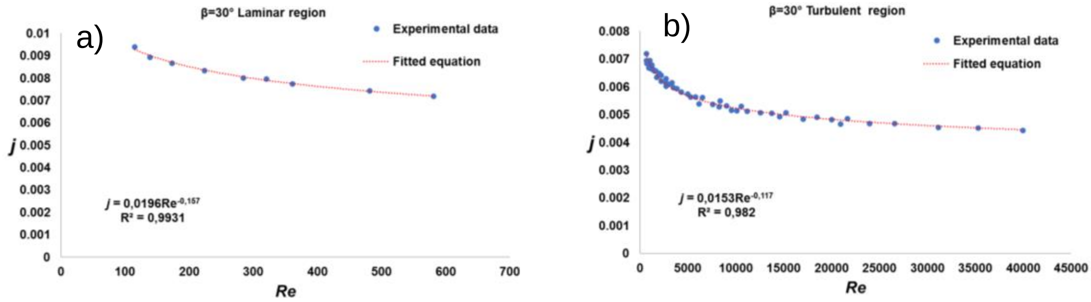


Figure 3: j vs Re for heat transfer plate with $\beta = 30$ ° a) laminar region, b) turbulent region

Applying the model of limit solutions for both regions, the expression of the resulting Colburn factor is:

$$j = [(0.0196Re^{-0.157})^{15} + (0.0153Re^{-0.117})^{15}]^{1/15} \tag{11}$$

The prediction exponent (*n*) plays an important role in the accuracy of the model. For the friction factor, when *n*=1 the adjustment to the experimental data presents an absolute average error of 73.27 %, when *n*=15 the error is only 4.81 %. For the Colburn factor, the exponent *n*=15 presents a better prediction with respect to the experimental data, with an average absolute error of 5.26 %. Figure 4 shows the comparison between the experimental data and the generalized expressions for β=30 °. Extending the value of n beyond 15 does not improve on the predictions, as the average error starts to reach a minimum value when n=15.

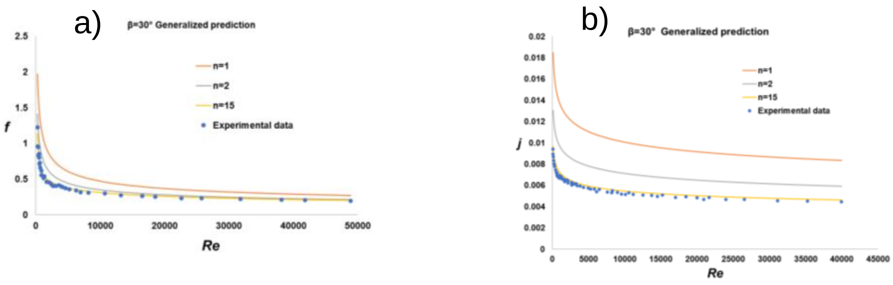


Figure 4: Thermohydraulic prediction using the proposed model for heat transfer plate with $\beta = 30°$ for overall flow regime a) f vs Re b) j vs Re

3. Prediction as a function of chevron angle

The prediction model is extended to heat transfer plates with corrugation angles between β=30 ° and β=80 °. The predictions are compared with the experimental data obtained by Focke et al. (1985). The proposed model gives good predictions of f and j. Table 1 shows the mean absolute errors and the prediction exponent for each plate relative to the experimental data. The angles chosen were the same that were specifically reported by Focke et al. (1985) for a plate pitch of 0.010 m and corrugation height of 0.005 m.

Table 1: Absolute prediction errors for heat transfer plates for $30° \leq \beta \leq 80°$

Chevron angle (β)	Friction factor error (e_{fp})	Colburn factor error (e_{jp})	Prediction exponent (n)
30 °	4.81 %	5.26 %	15
45 °	5.27 %	2.79 %	15
60 °	3.02 %	3.79 %	15
72 °	4.47 %	5.24 %	15
80 °	2.32 %	5.41 %	15
Overall absolute error	3.98 %	4.50 %	

As can be seen in Table 1, the prediction of *j* and *f* for the family of plates present an average absolute error in the whole range of 3.98 % for the friction factor and 4.50 % for the Colburn factor. Due to the relatively small mean absolute error, it can be established that the prediction model is valid for heat transfer plates in that range.

3.1 Development of a generalized correlation

A generalized correlation is developed for the prediction of *j* and *f* as a function of Reynolds number and plate corrugation angle (*β*). According to the proposed model, the constants and exponents a, b, c, d for the laminar (*L*) and turbulent (*T*) regimes can be correlated in terms of the corrugation angle. For angles *β* > 60 ° it is observed that there is no data in the laminar region. For this reason, the laminar component of the generalized expression in the range of 30 °≤ *β* ≤ 60 ° is also used for the full range 30° ≤ *β* ≤ 80 °. The turbulent component is determined in each region. The final generalised expressions are given in Table 2.

4. Implementation of design approach.

The proposed methodology is implemented using a case study (Picón-Núñez et al. 2003). Table 3 presents the process data and physical properties. A design methodology is proposed to satisfy the thermal load and fully utilise the allowable pressure drop. The pressure drop of the stream with the lowest allowable value is to be fully utilised. This consideration allows to develop a design approach. The free flow area is calculated using the available pressure drop, then compared to the free flow area that transfers the thermal load. The procedure is iterative until the difference between the two free flow areas is less than an error. Subsequently, the friction and Colburn factor are calculated using the generalized expressions. Then the number of thermal plates and the size of the unit are determined. Different plate sizes are used to produce a plot that gives an overall view of the design space. The plate spacing and plate thickness are 0.0029 m and 0.0006 m; the thermal conductivity of material is 15.06 (W/m°C) and the elongation factor used was 1.21. Different plates are analysed and referred to as plates number 1 to 8. The length (*L*) and width (*W*) of each plate are as follows: Plate numbers 1,6,7,8: 1 m x 0.225 m, 0.719 m x 0.334 m, 0.694 m x 0.216 m, 0.661 m x 0.210 m (Arsenyeva et al, 2011); plate number 2: 0.904 m x 0.354 m (Picón-Núñez et al., 2003); plate numbers 3,4,5: 0.875 m x 0.386 m, 0.802 m x 0.271 m, 0.783 m x 0.318 m (Kelvion, 2022).

Table 2: Generalized expressions for friction and Colburn factor for heat transfer plate for $30\,° \le \beta \le 80\,°$

Expression		Range of validity
Friction factor		
$f = \left[\left((0.5803\beta^2 - 35.119\beta + 553.49)Re^{(0.0003\beta^2 - 0.0362\beta + 0.281)}\right)^{15} + \left((0.0318\beta^2 - 2.2718\beta + 43.37)Re^{(-0.0002\beta^2 + 0.0207\beta - 0.714)}\right)^{15}\right]^{\frac{1}{15}}$	(28)	$30\,° \le \beta \le 60\,°$ $44 \le Re \le 49{,}000$
$f = \left[\left((0.5803\beta^2 - 35.119\beta + 553.49)Re^{(0.0003\beta^2 - 0.0362\beta + 0.281)}\right)^{15} + \left((-0.1289\beta^2 + 22.496\beta - 864.18)Re^{(0.0003\beta^2 - 0.0451\beta + 1.43)}\right)^{15}\right]^{\frac{1}{15}}$	(29)	$60\,° < \beta \le 80\,°$ $44 \le Re \le 19{,}700$
Colburn factor		
$j = \left[\left((-0.00003\beta^2 + 0.0032\beta - 0.0495)Re^{(-0.00007\beta^2 + 0.0071\beta - 0.307)}\right)^{15} + \left((0.00002\beta^2 + 0.0021\beta - 0.029)Re^{(0.00007\beta^2 + 0.0075\beta - 0.279)}\right)^{15}\right]^{1/15}$	(30)	$30\,° \le \beta \le 60\,°$ $44 \le Re \le 49{,}000$
$j = \left[\left((-0.00003\beta^2 + 0.0032\beta - 0.0495)Re^{(-0.00007\beta^2 + 0.0071\beta - 0.307)}\right)^{15} + \left((-0.0003\beta^2 + 0.1059\beta - 5.24889)Re^{(0.0009\beta^2 - 0.142\beta + 5.2)}\right)^{15}\right]^{1/15}$	(31)	$60\,° < \beta \le 80\,°$ $44 \le Re \le 19{,}700$

Table 3: Process data and physical properties for case of study

Process data	Hot stream	Cold stream
Mass flow rate (kg/s)	13.6	13.6
Pressure drop (Pa)	39,310	-
Inlet temperature (°C)	80	20
Outlet temperature (°C)	40	60
Physical properties		
Density (kg/m^3)	983.2	992.2
Heat capacity (J/kg°)	4,185	4,178
Thermal conductivity (W/m°C)	0.6536	0.6316
Viscosity (kg/m s)	0.000467	0.00065
Heat load (kW)	2,273	
Fouling factor (m^2°C/W)	0.0000103	0.000052

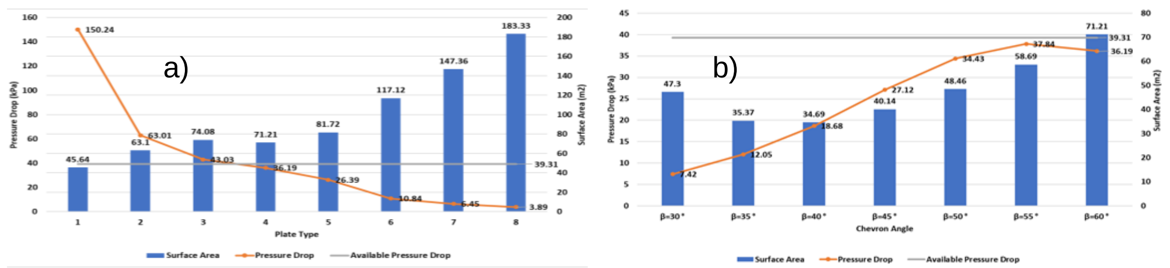


Figure 5: Design of plate heat exchangers: a) Different plate dimensions with a chevron angle of 60 °,b) Design with plate No. 4 for different chevron angles

Figure 5a shows that plate number 4 gives a pressure drop near the allowable value. This design is further explored in a second stage where different chevron angles are analysed. The results are shown in Figure 5b. The second stage of the design approach reveals two other interesting design options. With a chevron angle of 35 ° an area of 35.37 m^2 is reached with a pressure drop of 12.05 kPa, while for a chevron angle of 40 ° an area of 34.69 m^2 and 18.68 kPa. The decision between these two designs lies in the total operating costs. In the long run, the design with lower pressure drop will render the lower operating costs. Two other variables that provide other degrees of freedom for design are the plate spacing and the corrugation height. The stages where the effect of these variables in design are analysed are no discussed here due to space limitations.

5. Conclusions

A methodology to derive alternative generalized correlations for friction and Colburn factors have been developed for plate heat exchangers using the limiting solutions model. These expressions cover the full range of Reynolds number incorporating the laminar and turbulent region into a conjugated model and are presented as a function of the chevron angle β in the range between 30 ° to 80 °. The main conclusions of the work are:

- The prediction model results in average absolute errors of 3.98 % and 4.50 % for *f* and *j* in the range of 30 ° ≤ β ≤ 80 ° compared with the experimental data
- The approximation of the predictions depends on the value of the prediction exponent (*n*). For the specific case of plate data under study, a value of n = 15 gave accurate results. Values larger than 15 did not improve the predictions.
- The derived correlations represent a practical means of finding suitable exchanger geometries in design. Heat exchangers with the smallest surface area and lower pressure drop are readily identified in an alternative design approach based on the graphical display of the design options.
- Further work leading to optimisation studies is underway.

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