Research paper

Optimisation of fin selection and thermal design of counter-current plate-fin heat exchangers

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HIGHLIGHTS

- Standardized basic fin geometries are treated as continuous variables.
- Considering thermal hydraulic performance of different fin types in optimisation.
- Fin types and corresponding design parameters are optimised simultaneously.
- The optimisation objective is to minimise the total volume.
- A case study shows a very promising result when compared with a published result.

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ABSTRACT

A major challenge in designing optimal multi-stream plate-fin heat exchangers is the large number of combinations of standardised fin geometries for various fin types to choose from, which adds discrete aspects to an already complicated design problem. In this work, a new design algorithm is proposed to address this issue. By treating basic fin geometries such as plate spacing, fin pitch, fin length and fin thickness as continuous variables for all the fin types, different fin types are characterised based on the work published by different researchers. Then by taking into account thermal hydraulic performance of different fin types, optimal fin types and their corresponding design parameters can be obtained simultaneously by minimising the total volume of heat exchanger. The design parameters can be rounded to the nearest standardised fin parts for a feasible design. A case study with a comparison of published results is carried out to demonstrate the effectiveness of the method.

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1. Introduction

A plate-fin heat exchanger is a type of compact heat exchangers that uses plates and finned chambers to transfer heat between fluids. The main structure of plate-fin heat exchangers, shown in Fig. 1, consists of nozzle (stub pipe), distributor and plate-fin. Typical materials are brazed aluminium, stainless steel and titanium. Fig. 2 presents a typical single chamber made up of plates (parting sheet), fins and side bar (edge seal). Fins are introduced in the design of compact heat exchangers because: A) as a secondary surface, fin can transfer heat to fluid streams through parting sheets; B) due to higher thermal conductivity, heat transfer is more efficient; C) fins can enlarge fluid turbulence and enhance the local convective heat transfer coefficient. Typically, there are four

categories of fins, namely, plain fin, louvered fin, serrated fin (offset strip fin) and wavy fin (herringbone fin) shown in Fig. 3.

Compared with shell and tube heat exchangers, plate-fin heat exchangers have several distinctive advantages. Firstly, the minimum temperature approach can be as low as 1 °C in plate-fin heat exchangers, which can contribute to more heat recovery, less utility requirement and operating cost. Secondly, due to higher heat transfer efficient and large surface area per unit volume, a heat exchanger unit can be much smaller and lighter. Furthermore, the possibility of handling several streams in one unit provides the opportunity of reducing the number of heat exchanger [4]. Based on these merits, plate-fin heat exchangers have been extensively used in low temperature process systems, such as cryogenics for separation, LNG plants and ethylene plants, etc [5].

However, there are still several barriers in the application of plate-fin heat exchangers [5]. First of all, fouling limits their application to clean non-fouling streams [6]. Secondly, materials

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Nomenclature

A	total heat transfer area of one side, m^2
A_c	free flow area of one side, m^2
A_{fr}	total heat exchanger front area, m^2
b	plate spacing, m
b_{st}	standardised plate spacing, m
c	fin pitch, m
c_{st}	standardised fin pitch, m
c_p	heat capacity, J/kg K
d_h	hydraulic diameter, m
e	wavy fin amplitude, m
ER	relative difference
f	friction factor
f_s	ratio of secondary surface area to total surface area for heat transfer
h	heat transfer coefficient, $\text{W}/\text{m}^2 \text{K}$
i	interest rate
j	Colburn factor
k	fluid conductivity, $\text{W}/\text{m K}$
k_f	thermal conductivity of fins, $\text{W}/\text{m K}$
l	wavy fin wavelength, m
L	flow length on one side, m
m	mass flow rate, kg/s
N	number of layers per stream
n	number of holes per unit length
ΔP	pressure drop, Pa
Pr	Prandtl number
Q	heat duty of heat exchanger, W
R	fouling resistance, $\text{m}^2 \text{K}/\text{W}$
Re	Reynolds number
St	Stanton number

t_f	fin thickness, m
$t_{f,st}$	standardised fin thickness, m
Δt	stream temperature difference, $^\circ\text{C}$
ΔT_{LM}	logarithmic temperature difference, $^\circ\text{C}$
U	overall heat transfer coefficient, $\text{W}/\text{m}^2 \text{K}$
V	the total exchanger volume, m^3
W	exchanger width, m
x	fin length, m
x_{st}	standardised fin length, m
ξ	annual factor

Greek letters

α	ratio of total transfer area of one side exchanger to total exchanger volume, m^2/m^3
β	ratio of total transfer area of one side of exchanger to volume between plates of that side, m^2/m^3
σ	ratio of free flow area of one side to total frontal area, m^2/m^2
μ	viscosity, $\text{W}/\text{m K}$
ρ	density, kg/m^3
η	fin efficiency
Ψ	ratio of the wavy fin passage length to the wavy fin length in the z-direction

Subscript

i	hot stream
j	cold stream
y	design sequence
k	interval
T	total
h	hot stream
c	cold stream

allow the operation condition only at moderate temperature and pressure [7]. Thirdly, the capital cost is another major hurdle [8]. More importantly, there are few general systematic design methods that consider fin selection and thermal design of plate-fin

heat exchangers simultaneously in the previous researches [9]. Therefore, the objective of this study is to develop a new design methodology and take fin selection and thermal design consideration simultaneously.

Kays and London [13] presented approximately sixty standardised fin parts. Therefore, the large number of discrete combinations of standardised fin parts and types involved in the thermal design is one major difficulty in optimising the design of plate-fin heat exchangers. Pau [4] proposed two new concepts: identical-fin concept and Z–Y graph, which means that the same fins are used for all streams in the early design stage and a Z–Y graph is developed to select fin types. Picon-Nunez et al. [6–9] came up with a new term “Volume Performance Index (VPI)”, plotted a

Fig. 1. 4-stream plate-fin heat exchanger [1]

Fig. 2. Plate-fin chamber [2]

Fig. 3. Typical fin types [3]

graph of VPI vs Reynolds number and conducted sensitivity analysis to select appropriate fins based on assumed Reynolds number. On the basis of the identical fin concept and assumed Reynolds number, fin selection could not provide sophisticated results. On top of that, the optimisation of design is a time-consuming procedure.

Hao et al. developed a new design methodology for a cross-flow plate-fin heat exchanger, taking fin height and fin pitch as variables to minimize the total exchanger weight [10] or annual cost by Genetic Algorithm combined with back propagation neural network [11]. Yousefi [12] optimised cross-flow plate-fin heat exchangers using an ϵ -NTU model, treating hot flow length, cold flow length, fin height, fin frequency and length, fin thickness and numbers of layers as variables to minimise the total heat transfer area by hybrid Genetic Algorithm with particle swarm solution. He [14] then used a self-adaptive penalty function scheme and employed an improved harmony search algorithm to minimise the total heat transfer area and total pressure drop. Each standardised fin has its corresponding Colburn factor j and friction factor f correlations, which makes the design problem discrete. The optimisation of fin selection cannot converge easily. Therefore, the priority should be given to a general fin selection method, which can select fin types and consider imposed constraints simultaneously.

Current plate-fin heat exchanger design methodologies divide the process streams into several sections or intervals by composite curves and analyse each interval in detail [4]. The main design method is based on a thermal-hydraulic model or a ϵ -NTU model that present the relationship between heat exchanger volume, heat transfer coefficients and pressure drop.

In general, due to the complex discrete nature of plate-fin heat exchanger design caused by a large number of standardised fin parameters, the design of plate-fin heat exchangers still contains a major element of trial and error, which often leads to sub-optimal design configurations. Therefore, the aim of this study is to develop a systematic design methodology for optimal multi-stream plate-fin heat exchangers that takes into account fin selection and pressure drop utilization simultaneously. A case study from Picon-Nunez [15] is redesigned and compared to verify the feasibility of the new algorithm.

2. Problem definition

Given physical properties and thermal information (mass flow rate, inlet and outlet temperature and pressure) of all process streams, as well as fin physical properties, the objective is to find the optimum design of a plate-fin heat exchanger. Total heat

exchanger volume, exchanger dimensions (height, width and length), number of layers per stream, heat transfer coefficients and pressure drop should all be taken into consideration simultaneously.

This design methodology is developed based on the following assumptions:

- > Steady-state operation
- > Single-phase heat transfer
- > Constant fluid physical properties
- > Constant heat transfer coefficients
- > Common wall temperature
- > Counter-current arrangement
- > Identical fin category

3. Characterisation of different fin types

For a two-stream plate-fin heat exchanger, there are approximately 3600 combinations that need to be screened. The number will be increased exponentially with more streams involved. Without an appropriate fin selection method, design would be extremely time-consuming especially for multi-stream plate-fin heat exchangers.

Based on the published data, pressure drop, heat transfer performance and design parameters can be expressed as continuous correlations in this model to convert the discrete design problem to a continuous problem. The Colburn Factor j and the Friction Factor f may be expressed as a function of Reynolds Number and basic fin geometry parameters. Similarly, several characteristic dimensions, hydraulic diameter d_h , ratio of transfer area to volume of one side β and ratio of secondary surface area to total surface area f_s , may be described as a function of basic fin geometry parameters.

The Colburn Factor j and the Fanning Friction Factor f for offset strip fin cores were regressed by Manglik [16] as a function of Reynolds Number and basic fin geometry, such as, fin pitch c , fin height b , fin length x and fin thickness t_f shown in Fig. 4. Manglik et al. [16] claimed that these correlations predicted the experimental data within 14%, which is much better than that in previous design work.

$$j = 0.6522 Re^{-0.5403} \left(\frac{c}{b}\right)^{-0.1541} \left(\frac{t_f}{x}\right)^{0.1499} \left(\frac{t_f}{c}\right)^{-0.0678} \times \left[1 + 5.269 \times 10^{-5} Re^{1.34} \left(\frac{c}{b}\right)^{0.504} \left(\frac{t_f}{x}\right)^{0.456} \left(\frac{t_f}{c}\right)^{-1.055} \right]^{0.1} \quad (1)$$

Fig. 4. Basic offset strip fin geometry.

$$f = 9.6243Re^{-0.7422} \left(\frac{c}{b}\right)^{-0.1856} \left(\frac{t_f}{x}\right)^{0.3053} \left(\frac{t_f}{c}\right)^{-0.2659} \times \left[1 + 7.669 \times 10^{-8} Re^{4.429} \left(\frac{c}{b}\right)^{0.920} \left(\frac{t_f}{x}\right)^{3.767} \left(\frac{t_f}{c}\right)^{0.236}\right]^{0.1} \quad (2)$$

Re is the Reynolds Number which is defined as a function of the surface hydraulic diameter:

$$Re = \frac{\dot{m}d_h}{\mu A_c} \quad (3)$$

Equations (1) and (2) are valid for $120 < Re < 10^4$, $0.134 < c/b < 0.997$, $0.012 < t_f/x < 0.048$ and $0.041 < t_f/c < 0.121$.

For plain fin, Diani et al. [17] claimed these correlations correctly predicted with a relative deviation of -0.4% and absolute deviation of 4.3% when the Reynolds Number is around 2700 to 10,100:

$$j = 0.233Re^{-0.48} \left(\frac{c}{b}\right)^{0.192} \left(\frac{t_f}{b}\right)^{-0.208} \quad (4)$$

$$f = 0.029Re^{-0.09} \left(\frac{c}{b}\right)^{0.034} \left(\frac{t_f}{b}\right)^{-0.169} \quad (5)$$

For louver fin, Davenport [18] derived these correlations and agreed with experimental data within 10% and 12% respectively when the Reynolds Number is around 300 to 4000:

$$j = 0.249Re^{-0.42} b^{0.26} \left(\frac{x}{b}\right)^{1.1} x^{0.33} \quad (6)$$

$$f = 0.494Re^{-0.39} c^{0.46} \left(\frac{x}{c}\right)^{1.1} \left(\frac{b}{c}\right)^{0.33} \quad (7)$$

For wavy fin, Dong [19] verified that the mean deviations of these correlations in the range of the Reynolds Number between 800 and 6500 for j and f factors were 4.4% and 5.1% respectively:

$$j = 0.0836Re^{-0.2309} \left(\frac{c}{b}\right)^{0.1284} \left(\frac{x}{l}\right)^{-0.326} \left(\frac{c}{2e}\right)^{-0.153} \quad (8)$$

$$f = 1.16Re^{-0.309} \left(\frac{c}{b}\right)^{0.3703} \left(\frac{x}{l}\right)^{-0.1152} \left(\frac{c}{2e}\right)^{-0.25} \quad (9)$$

To make the model continuous, for each category of fins, hydraulic diameter d_h , secondary surface ratio f_s and heat transfer surface area per unit volume β may be simply regressed as a function of basic fin geometry parameters. The continuous correlations for four fin categories are listed as below:

For offset strip-fin [20]:

$$\beta = \frac{2(b - t_f)x + 2(c - t_f)x + 2(b - t_f)t_f + ct_f}{bcx} \quad (10)$$

$$f_s = \frac{2(b - t_f)x + 2(b - 2t_f)t_f + ct_f}{2(b - t_f)x + 2(c - t_f)x + 2(b - t_f)t_f + ct_f} \quad (11)$$

$$d_h = \frac{4(c - t_f)(b - t_f)x}{2((c - t_f)x + (b - t_f)x + t_f(b - t_f)) + t_f(c - t_f) - t_f^2} \quad (12)$$

For plain fin [21]:

$$\beta = \frac{2(b + c)}{bc} \quad (13)$$

$$f_s = \frac{2b + 5c/3}{2b + 4c} \quad (14)$$

$$d_h = \frac{2b(c - t_f)}{b + c} \quad (15)$$

For louvered fin [20]:

$$\beta = \frac{2x(c - t_f) + 2bc + 2(\sqrt{c^2 + b^2} - t_f)}{bc} \quad (16)$$

$$f_s = \frac{2(\sqrt{c^2 + b^2} - t_f)}{2x(c - t_f) + 2bc + 2(\sqrt{c^2 + b^2} - t_f)} \quad (17)$$

$$d_h = \frac{4bc - t_f(\sqrt{c^2 + b^2} - t_f)}{2x(c - t_f) + 2bc + 2(\sqrt{c^2 + b^2} - t_f)} \quad (18)$$

For wavy fin [21]:

$$\beta = \frac{2\psi b + 2c}{bc} \quad (19)$$

$$f_s = \frac{2\psi b + c}{2\psi b + 2c} \quad (20)$$

$$d_h = \frac{2bc}{b + \psi c} \quad (21)$$

4. Modelling of thermal-hydraulic performance of plate-fin heat exchangers

The thermal-hydraulic model is based on the basic heat transfer design equation for a two-stream heat exchanger [6–9,20].

$$Q = UA\Delta T_{LM} \quad (22)$$

With the inclusion of surface fouling and fin effects, the overall heat transfer coefficient is defined as below [20]:

$$\frac{1}{UA} = \frac{1}{U_1A_1} = \frac{1}{U_2A_2} = \frac{1}{\eta_1A_1} \left(\frac{1}{h_1} + R_1\right) + \frac{1}{\eta_2A_2} \left(\frac{1}{h_2} + R_2\right) + R_w \quad (23)$$

Combining it with the basic equation gives

$$A_1 = \frac{Q}{\Delta T_{LM}} \left[\frac{1}{\eta_1} \left(\frac{1}{h_1} + R_1\right) + \frac{1}{\eta_2} \left(\frac{A_1}{h_2A_2} + R_2\right) + R_w \right] \quad (24)$$

where A_1 and A_2 represent the total heat transfer area of each side; h_1 and h_2 are the film heat transfer coefficients of each side; R_1 and R_2 are the thermal resistance due to fouling in each side. R_w is the wall thermal resistance.

Due to different fin types and densities for both streams, the heat exchanger volume will be more reasonable as the key indicator of sizing than heat transfer area in plate-fin heat exchanger design [13]. Kays and London [13] set a new parameter α that relates the total heat transfer area of one side of heat exchanger to the total heat exchanger volume. Based on the definition, the total heat transfer area of one side can be expressed as:

$$A_1 = \alpha_1 V_T \quad (25)$$

$$A_2 = \alpha_2 V_T \quad (26)$$

where α can be calculated by the heat exchanger geometry [13]:

$$\alpha_1 = \beta_1 \left(\frac{b_1}{b_1 + b_2} \right) \quad (27)$$

$$\alpha_2 = \beta_2 \left(\frac{b_2}{b_1 + b_2} \right) \quad (28)$$

where β is the ratio of total heat transfer area of one side to the volume between plates of that side and b is the plate spacing of each side respectively.

Therefore, by substituting Equations (25) and (26) into Equation (24), the heat exchanger volume expressed as [13]:

$$V_T = \frac{Q}{\Delta T_{LM}} \left[\frac{1}{\eta_1 \alpha_1} \left(\frac{1}{h_1} + R_1 \right) + \frac{1}{\eta_2 \alpha_2} \left(\frac{1}{h_2} + R_2 \right) \right] \quad (29)$$

The overall surface efficiency η_1 and η_2 can be calculated by Ref. [13]:

$$\eta = 1 + f_s \left\{ \frac{\tanh \left[\left(\frac{2h}{k_f t_f} \right)^{1/2} \frac{b}{2} \right]}{\left[\left(\frac{2h}{k_f t_f} \right)^{1/2} \frac{b}{2} \right]} - 1 \right\} \quad (30)$$

where k_f is the thermal conductivity of fin; t_f is the fin thickness.

The heat transfer performance is presented as the Colburn factor j [20],

$$j = St Pr^{2/3} \quad (31)$$

where Pr is the Prandtl Number, St is the Stanton Number which are given by Ref. [20]:

$$St = \frac{h A_c}{m C_p} \quad (32)$$

$$Pr = \frac{\mu C_p}{k} \quad (33)$$

Therefore, the heat transfer coefficient can be rearranged as [6]:

$$h = \frac{\dot{m} C_p j}{A_c Pr^{2/3}} \quad (34)$$

A similar procedure can be applied to express the pressure drop. The pressure drop across the core of a heat exchanger is [13]:

$$\Delta P = \frac{2 f L \dot{m}^2}{\rho d_h A_c^2} \quad (35)$$

5. Methodology for optimal fin selection

Based on the above models for different fin categories and the thermal-hydraulic model, the discrete design problem can be converted to a continuous NLP problem. Since Friction Factor, Colburn Factor and design parameters (d_h, f_s, β) can be expressed as a function of Reynolds Number Re and basic fin geometries, plate spacing b , fin pitch c , fin thickness t_f and fin length x are taken as continuous design variables in this methodology. The objective function, total exchanger volume can be minimised by the CONOPT solver in GAMS to find the optimum fin types.

By combining these two models together, heat transfer coefficient and pressure drop can also be related to basic geometry parameters. Therefore, fin selection can be carried out with consideration of imposed constraints, such as pressure drop and fluid flow types. In other words, there is no need to select the fin type blindly based on assumed Reynolds Number and regardless of pressure drop in the early design stage.

Due to standardised fin parts, the optimum fin geometry may not exist. Compared with fixed standardised fin geometries, the “relative difference” ER should be calculated to select the closet fin types. A smaller ER value indicates a better match.

$$ER = \left| \frac{b - b_{st}}{b_{st}} \right| + \left| \frac{c - c_{st}}{c_{st}} \right| + \left| \frac{x - x_{st}}{x_{st}} \right| + \left| \frac{t_f - t_{f,st}}{t_{f,st}} \right| \quad (36)$$

6. Overall design methodology of plate-fin heat exchangers

The whole general plate-fin heat exchanger design methodology is presented in Fig. 2.

Pinch technology [22] is employed to represent process streams directly in a graphical format. The composite curves include hot and cold composite curves, which depict the heat balance of process streams and give the minimum hot and cold utility requirements. Based on assumed constant fluid physical properties, the composite curve is formed by several straight lines. The kink points indicate certain process streams inlet or outlet points. Therefore, the composite curves can be divided into several enthalpy intervals by kink points shown in Fig. 3 [4–9].

With the interval decomposition, the flow rate, permissible pressure drop and heat load per stream are fixed in each enthalpy interval. The allowable pressure drop per stream in each interval is distributed linearly according to the fraction of heat load [6].

$$\Delta P_{i,k} = \Delta P_{i,total} \left(\frac{\Delta H_{i,k}}{\Delta H_{i,total}} \right) \quad (37)$$

where i is the stream number, k is the interval number.

In each interval, the entry and exit temperatures are fixed. The temperature driving force is uniform for all hot streams and cold streams. Therefore, each hot stream can match any cold stream in the same interval.

In this work, a superstructure based heat exchanger network is employed at the design stage. Each hot stream is spilt into several streams to match every cold stream. Conversely, each cold stream is spilt to allow a match with all the hot streams. Fig. 6 illustrates a simple superstructure example (Fig. 7).

The heat load distribution is calculated by.

$$Q_{i,j,k} = Q_{i,k} \times \frac{Q_{j,k}}{\sum_j Q_{j,k}} \quad (38)$$

Fig. 5. Overall optimum design algorithm.

Every match can be regarded as a two-stream plate-fin heat exchanger with a volume $V_{i,j,k}$. Each interval is a plate-fin heat exchanger network. Consequently, the volume of each interval is the sum of each match included. The whole plate-fin heat exchanger volume is the sum of volume of all intervals involved.

$$V_k = \sum_i \sum_j V_{i,j,k} \quad (39)$$

$$V = \sum_k V_k = \sum_k \sum_i \sum_j V_{i,j,k} \quad (40)$$

As mentioned above, each interval is a two-stream heat exchanger network. At the design stage, once a match is set and

optimised, the fin types for these two streams will be determined to ensure the unified pressure drop for the same stream in the same interval. Therefore, different design sequences will give different heat exchanger volumes. Fin types are determined based on the minimum interval volume.

Based on selected fin types, the heat exchanger volume should be recalculated and used to determine the interval and exchanger dimensions (width W , length L and height H) and number of layers per stream.

The frontal area A_{fr} can be obtained by free flow area A_c [13]:

$$\sigma = \frac{A_c}{A_{fr}} = \frac{\alpha d_h}{4} \quad (41)$$

Fig. 6. Composite curves and interval decomposition.

The length L can be determined by Ref. [13]:

$$L = \frac{V_T}{A_{fr}} \quad (42)$$

With the assumed width W , the height of exchanger H and the number of layers per stream can be deduced by Ref. [13]:

$$H = \frac{A_{fr}}{W} \quad (43)$$

$$N = \frac{A_{fr}}{W(b_1 + b_2)} \quad (44)$$

7. Case study

A four-stream plate-fin heat exchanger [8] is revisited in this work. The process stream information is listed below in Table 1. To test and verify the feasibility of the new design methodology, the

Fig. 7. Interval superstructure.

minimum temperature approach of 20 °C is assumed the same as in the previous work [8].

Fig. 5 presents the hot and cold composite curves and shows the interval decomposition in this case. The detailed information of each interval is shown in Table 2. The distribution of pressure drop per stream is listed in Table 3.

The optimisation is executed with GAMS version 23.4 on a 2.6 GHz 4th Intel Core i5 PC with 8 GB memory. The CONPT solver is used in this study and the computation time is 10–15 s. There are 9 variables and 8 nonlinear equations in the optimisation program.

7.1. Heat exchanger volume and fin selection

The use of identical fin categories is assumed in this case study. With higher heat transfer efficiency and larger pressure drop, offset strip fin and louvered fin cannot meet the heat exchanger requirement because of unrealistic heat exchanger size and pressure drop. Therefore, the plain fin is employed in this case in spite of lowest heat transfer efficiency. Take interval 1/H1–C1 match as an example, the optimum fin height, fin thickness and fin pitch for H1 side obtained from GAMS are 6.05 mm, 1.20 mm and 0.152 mm respectively. By calculating ER value, the most similar fin type is plain-fin 19.86. The optimum fin height, fin thickness and fin pitch for C1 side are 6.45 mm, 1.24 mm and 0.152 mm respectively. Among all standardised parts sizes, plain-fin 19.86 (fin height 6.35 mm, fin pitch 1.278 mm, fin thickness 0.152 mm) is the best choice for C1 side. The same method is employed in each match and interval. To determine the interval dimensions, the width in this

Table 1
Process data for case study [8]

Stream	T_s (°C)	T_t (°C)	Flow rate (kg/s)	ΔP (KPa)	ρ (kg/m ³)	C_p (J/kg °C)	μ (cP)	K (W/m °C)
H1	150	60	25.0	46	700	800	0.3	0.12
H2	90	60	106.7	60	700	750	0.4	0.12
C1	20	125	27.7	30	750	900	0.5	0.12
C2	25	100	37.5	86	750	800	0.5	0.12

Table 2
Interval decomposition results.

Interval	$T_{H,in}$ (°C)	$T_{H,out}$ (°C)	$T_{C,in}$ (°C)	$T_{C,out}$ (°C)	ΔT_{LM} (°C)	ΔH (kW)
1	65.3	64	20	25	42.13	124.65
2	90	65.3	25	70	29.00	2417.8
3	150	90	70	91.8	35.74	1200.0

Table 3
Pressure drop distribution.

Stream	ΔP distribution (kPa)		
	Interval 1	Interval 2	Interval 3
H1	0.72	12.62	30.66
H2	2.60	49.40	—
C1	1.43	12.86	6.23
C2	—	51.60	25.00

case is assumed to be 0.75 m as the basic design. The detailed design information is shown below in Tables 4–6.

From Tables 4–6, fin types employed (plain fin 15.08 and plain fin 11.94T) in the basic design cannot provide the minimum heat exchanger volume. Because the fin type selection in the basic design is based on the graph of VPI (Volume Performance Index) vs assumed Reynolds Number Re . Once the Reynolds Number is calculated, the fin type will be reselected. The iteration will not stop until the final Reynolds Number agrees well with the previous Reynolds Number within the accepted limit. But in the new

methodology, the fin selection and thermal design are optimised simultaneously, to avoid random selection and improve the precision. Using plain-fin 19.86 saves the block volume by almost 16.7%, 1% and 37.6% respectively. The number of layers in each block is changed in different directions correspondingly. Through the verification of this case study, the fin selection can be completed while designing the plate-fin heat exchanger, rather than drawing a graph of special terms vs Reynolds Number and selecting fin types based on assumed Reynolds Number at early design stage. The fin selection problem can be taken as a continuous problem through the integration of continuous expression of fin parameters in spite of dealing with standardised parts. Moreover, the new design methodology not only provides a better design, but also improves the job efficiency, especially for multi-stream plate-fin heat exchanger design.

7.2. Pressure drop sensitivity analysis

Pressure drop is an important constraint, which should be considered in the plate fin heat exchanger design process. In the new design methodology, the allowed pressure drop of a critical stream is maximised or fully utilised, while pressure drop values for other streams are calculated consequently. But all pressure drops should be less than allowed maximum pressure drops.

Table 7 lists the allowed maximum pressure drops, pressure drop of each stream in each interval in the basic case and the new design. The total pressure drop for each stream is also listed in Table 8.

Table 4
Interval 1 design details.

Interval 1	Interval dimensions				Fin types			Number of layers		
	Volume (m ³)	Length (m)	Width (m)	Height (m)	H1	H2	C1	H1	H2	C1
Basic Case	0.012	0.053	0.750	0.313	PF15.08	PF11.94T	PF15.08	4.9	19.7	24.6
New Design	0.010	0.170	0.750	0.715	PF19.86	PF19.86	PF19.86	1.2	4.7	5.9

*PF – plate fin.

Table 5
Interval 2 design details.

Interval 2	Interval dimensions				Fin types				Number of layers			
	Volume (m ³)	Length (m)	Width (m)	Height (m)	H1	H2	C1	C2	H1	H2	C1	C2
Basic Case	0.204	0.868	0.750	0.313	PF15.08	PF15.08	PF15.08	PF15.08	9.0	36.2	20.5	24.7
New Design	0.204	1.008	0.750	0.272	PF19.86	PF19.86	PF19.86	PF19.86	4.5	16.8	9.7	11.6

Table 6
Interval 3 design details.

Interval 3	Interval dimensions				Fin types			Number of layers		
	Volume (m ³)	Length (m)	Width (m)	Height (m)	H1	C1	C2	H1	C1	C2
Basic Case	0.178	0.780	0.750	0.313	PF15.08	PF15.08	PF15.08	20.4	9.2	11.1
New Design	0.111	0.530	0.750	0.282	PF19.86	PF19.86	PF19.86	22.2	10	12.2

Table 7
Pressure drop details in each interval.

	Interval 1			Interval 2				Interval 3		
	H1	H2	C1	H1	H2	C1	C2	H1	C1	C2
Allowed	0.72	2.60	1.43	12.62	49.40	12.86	51.60	30.66	6.23	25.00
Basic Case	0.243	0.48	0.17	7.52	9.83	12.92	15.77	0.481	7.4	13.82
New Design	0.46	0.91	0.322	9.22	10.07	12.70	16.12	1.15	6.23	7.81

*All pressure drops are in kPa.

Table 8
Comparison of total pressure drop.

Stream	Allowed	Basic case	New design
H1	44	8.25	10.83
H2	52	10.31	10.98
C1	20.52	20.49	19.25
C2	76.6	29.59	23.93

*All pressure drops are in kPa.

Fig. 8. Pressure drop sensitive analysis.

Compared with plain fin 15.08, plain fin 19.86 provides a smaller heat exchanger but generates higher pressure drop. Actually, a fin type has an effect on the rate of heat transfer and pressure drop utilization. Higher fin density gives a larger heat transfer area, which can enhance heat transfer efficiency and decrease heat exchanger size, but consequently increases the pressure drop. In the optimal design, not only pressure drop of a critical stream is maximised, but also the pressure drop of other streams should be under the allowed pressure drops.

In the existing method, provided that the total pressure drop is under the allowed value, small tolerance in any interval can be accepted. Based on assumed Reynolds numbers, the previous methodology needs iteration to converge after obtaining real Reynolds numbers in the initial design. Therefore, the verified pressure drop could violate the allowed maximum pressure drop in some cases, as shown in Table 7. On the other hand, the new design methodology considers fin selection and pressure drop simultaneously that can avoid any pressure drop violation.

To investigate the influence of pressure drop on fin selection, a sensitive analysis is studied by changing the allowed pressure drop of the critical stream C1 in the interval 3 and keeping other pressure drops constant. The fin selection results and corresponding heat exchanger volumes are shown in Fig. 8.

The higher density fin type gives higher heat transfer coefficient and pressure drop. Clearly, when the allowed pressure drop of the critical stream increases, higher density fin types can be employed for a smaller heat exchanger volume. Hence, there is a trade-off between pressure drop and heat exchanger volume. Therefore, fin selection and pressure drop optimisation for every single stream should be considered simultaneously.

8. Conclusions

A new design methodology for multi-stream plate-fin heat exchangers is proposed to minimise heat exchanger volume. The selection of fin types and imposed constraints can be considered

simultaneously in the optimal design process by regressing Friction Factor f , Colburn Factor j and some design parameters (d_h, f_s, β) as a function of basic fin geometries such as fin height b , fin pitch c , fin length x and fin thickness. The discontinuous optimisation problem caused by discrete standardised fin types can be converted to a continuous problem to overcome computation difficulties. For a large scale multi-stream plate-fin heat exchanger, the design quality could be greatly improved.

However, this design methodology is developed based on assumed constant fluid physical properties and identical fin category for both sides. In practical, especially in low temperature systems, various physical properties, induced by phase and/or temperature change, should be taken into consideration in the design methodology as well as mixing and matching fin categories in the plate-fin heat exchanger. Further research work is needed to tackle these challenges.

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