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Optimisation of Fin Selection and Thermal Design of Plate-Fin Heat Exchangers

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Small minimum temperature approach, large surface area per volume, high heat transfer efficiency and possibility of handling several streams of multi-stream plate-fin heat exchangers lead to more heat recovery and smaller heat exchanger size. Therefore, plate-fin heat exchangers have been applied extensively in sub-ambient processes. However, one major difficulty in optimising the design of plate-fin heat exchangers is the large number of discrete combinations of standardised fin parts and types involved in the thermal design. A new approach is proposed to systematically optimise fin selection and thermal design of plate-fin heat exchangers simultaneously. Continuous correlations are developed between basic fin geometry parameters and thermal performance of plate-fin heat exchangers based on published data. Validation study shows a good agreement between published results and prediction from the developed correlations. Such correlations can then be incorporated into an overall design procedure, in which the problem is solved as a continuous NLP problem. Compared with previous published design results, 20 % heat exchanger volume could be saved with the new design method. Furthermore, the minimum temperature approach optimisation is also integrated into the design model to minimise the total cost for better trade-offs between energy cost and capital cost.

1. Introduction

Plate-fin heat exchanger is a type of compact heat exchangers that consist of a stack of plates and fins. The particular geometrical features make plate-fin heat exchangers suitable for sub-ambient processes due to smaller minimum temperature approach, larger surface area per volume and higher heat transfer efficiency. Compared with normal heat exchangers, plate-fin heat exchangers have several distinctive advantages. Firstly, the minimum temperature approach can be as low as 1 ºC in plate-fin heat exchangers, which can contribute to more heat recovery and less utility requirement. 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 heat exchanger number (Lee, 2002).

Fin selection is introduced in the design of plate-fin heat exchangers. There are four fin categories (plain fin, louver fin, offset strip fin and wavy fin) and approximately 60 standardised fin types (Kays and London, 1984). For a two-stream plate-fin heat exchanger, there are 3,600 combinations at the design stage, much more when designing a multi-stream plate-fin heat exchanger. Lee (2002) proposed two new concepts (identical-fin concept and Z-Y graph) to select fin types in the design stage. Picon-Nunez (2005) developed a new term “Volume Performance Index (VPI)” and conducted sensitive analysis to determine fin types based on assumed Reynolds number (Re). Hao et al. (2008) took fin geometry parameters as variables and minimised the total heat exchanger weight or annual cost to select fin types and design a cross-flow plate-fin heat exchanger. Since each standardised fin part has its corresponding Colburn factor (j) and friction factor (f) correlations, the design problem is discrete and difficult to converge in the optimization stage. However, there are few general methodologies that select fin types and consider the imposed constraints simultaneously.

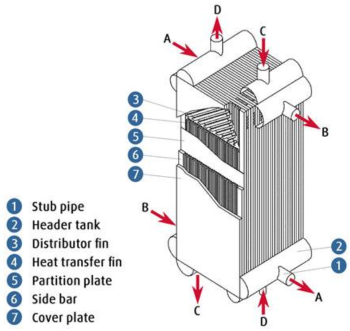

Figure 1: Plate-fin heat exchanger (source: Linde Engineering)

In this study, fin parameters (plate spacing b, fin pitch c, fin length x and fin thickness t_f) are taken as variables to minimize the heat exchanger volume. Continuous correlations of friction factor, Colburn factor and fin characteristic parameters (d_h, f_s, β) are developed based on published data in terms of plain fin (Zhu et al., 2008), louvered fin and wavy fin (Shah, 2003), and offset strip fin (Manglik, 1995) and integrated into a thermal design model. Therefore, the new modified design methodology can select fin types and take detailed heat transfer into consideration simultaneously. Moreover, the determination of the minimum temperature approach is introduced in the design methodology.

2. Methodology

The new methodology is divided into several steps: minimum temperature approach determination, fin selection and optimisation, checking and recalculation of heat exchanger dimensions. The design is developed based on the following assumptions: steady-state operation, single-phase heat transfer, constant fluid properties, constant heat transfer coefficients, common wall temperature, counter-current arrangement, and identical fin category for both sides.

2.1 Minimum Temperature Approach Determination

Smaller minimum temperature approach gives more heat recovery and less utility requirement, which can decrease the operating cost but increase the capital cost inversely. By trading off total cost, the optimum minimum temperature approach can be determined.

Based on assumed minimum temperature approach, Pinch Technology (Linnhoff et al., 1982) is employed to represent the process stream graphically and determine the utility requirement.

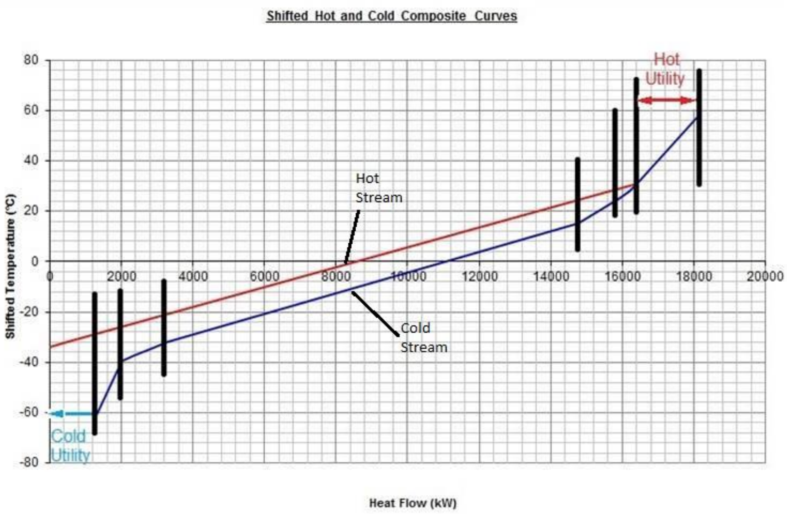

Figure 2: Process Composite Curves

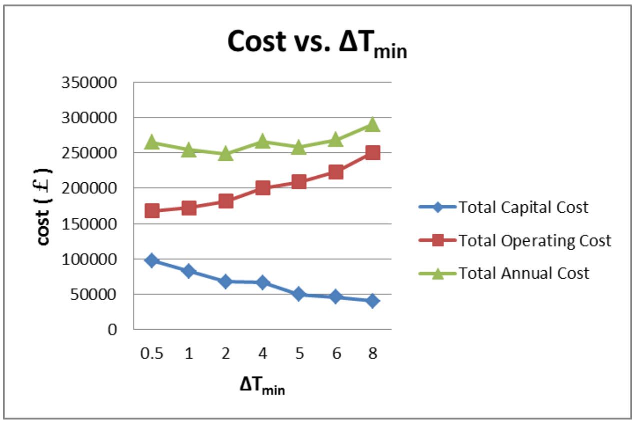

Figure 3: Graph of total cost vs minimum temperature approach

To avoid difficulties in finding heat transfer area and overall heat transfer coefficient, the approximate capital cost can be estimated by C-value method (Geoff, 2007) in terms of the cost of per unit ($Q/\Delta T_{LM}$).

$$T_{cap,a} = \frac{\dot{Q}}{\Delta T_{LM}} * C * xi \tag{1}$$

$$C = exp\left\{lnC_1 + \frac{\ln{(C_1/C_2)}\ln{[(Q/\Delta T_{LM})/(Q/\Delta T_{LM})_1]}}{\ln{[(Q/\Delta T_{LM})_1/(Q/\Delta T_{LM})_2]}}\right\}$$ 1. (2)

$$xi = \frac{i(1+i)^n}{(1+i)^n - 1}$$ 2. (3)

The first term of Eq(1) presents the annual capital cost of plate-fin heat exchangers, and cost per unit coefficient C can be obtained from EDSU (1997). ΔT_{LM} is the logarithmic mean temperature difference, and Q is heat duty. The factor xi is annual factor, i is interest rate and n is the number of years.
For a low-temperature process, hot utility can be satisfied by hot water and cold utility can only be satisfied with the help of a refrigeration cycle. The operating cost can be estimated by published data (Smith, 2005).
The total annual cost can be obtained by adding total annual capital cost and operating cost. A graph of the total annual cost vs the minimum temperature approach can be drawn to determine the minimum temperature approach ΔT_{min}.

2.2 Fin selection

For each fin category - strip-fin (Manglik, 1995), louver fin (Davenport, 1983), plain fin (Bala Sundar Rao, 2013) and wavy fin (Dong, 2007), the Colburn factor and friction factor can be expressed as a function of Reynolds number (Re) and basic fin geometry parameters (plate spacing b, fin pitch c, fin length x and fin thickness t_f) to avoid discrete problem in optimisation. These literatures claimed that these correlations agreed well with experimental data within ±15 %. Take the strip-fin as an example (Manglik, 1995):

$$\mathrm{j} = 0.6522\mathrm{Re}^{-0.5403}\left(\frac{c}{b}\right)^{-0.1541}\left(\frac{t_f}{x}\right)^{0.1499}\left(\frac{t_f}{c}\right)^{-0.0678} \times \left[1 + 5.269\times10^{-5}\mathrm{Re}^{1.34}\left(\frac{c}{b}\right)^{0.504}\left(\frac{t_f}{x}\right)^{0.456}\left(\frac{t_f}{c}\right)^{-1.055}\right]^{0.1} \tag{4}$$

$$\mathrm{f} = 9.6243\mathrm{Re}^{-0.7422}\left(\frac{c}{b}\right)^{-0.1856}\left(\frac{t_f}{x}\right)^{0.3053}\left(\frac{t_f}{c}\right)^{-0.2659} \times [1 + 7.669\times10^{-8}\mathrm{Re}^{4.429}\left(\frac{c}{b}\right)^{0.920}\left(\frac{t_f}{x}\right)^{3.767}\left(\frac{t_f}{c}\right)^{0.236}]^{0.1}$$ 3. (5)

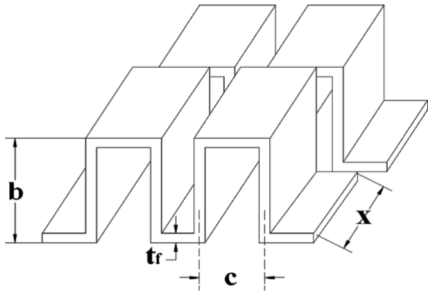

Figure 4: Strip fin basic geometry (Sepehr and Hassan, 2010)

Similarly, 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 can also be described as a function of basic fin geometry parameters Shah (2003) and later Zhu et al., (2008). The continuous geometric expressions of these parameters of strip fin are listed as follow:

$$\beta = \frac{2(b-t_f)x + 2(c-t_f)x + 2(b-t_f)t_f + ct_f}{bcx} \tag{6}$$

$$f_s = \frac{2(b-t_f)x + 2(b-2t_f)t_f + ct_f}{2(b-t_f)x + 2(c-t_f)x + 2(b-t_f)t_f + ct_f} \tag{7}$$

4.

$$d_h = \frac{4(c-t_f)(b-t_f)x}{2\left((c-t_f)x + (b-t_f)x + t_f(b-t_f)\right) + t_f(c-t_f) - {t_f}^2} \tag{8}$$

5.

These correlations can be integrated into the plate-fin heat exchanger thermal design model proposed by Picon-Nunez (2005) to convert the discrete design problem to continuous. The objective function in the optimisation is minimising the total heat exchanger volume, and the problem is then optimised by CONOPT solver in GAMS. The design variables are basic fin geometries (plate spacing b, fin pitch c, fin length x and fin thickness t_f). Due to the standardised fin parts, the optimum fin geometry may not exist. Compared with fixed fin parts sizes, the “relative difference” ER should be calculated to select the closest fin types. After fin selection, the heat exchanger dimensions and heat transfer coefficient should be recalculated.

$$ER = \left|\frac{b-b_{st}}{b_{st}}\right| + \left|\frac{c-c_{st}}{c_{st}}\right| + \left|\frac{x-x_{st}}{x_{st}}\right| + \left|\frac{t_f-t_{f,st}}{t_{f,st}}\right| \tag{9}$$

3. Case study

A two-stream plate-fin heat exchanger designed by Picon-Nunez (1999) is revisited in this study. The process information and physical properties are listed in Tables 1 and 2. To verify the feasibility of the new design methodology, the minimum temperature approach is assumed as 20 ºC, which is the same as the previous work.

Table 1: Process information

	Stream 1	Stream 2
Mass flow rate (kg/s)	49.0	49.0
Allowed pressure drop (Pa)	8,800	8,800
Inlet temperature (ºC)	524	290
Outlet temperature (ºC)	313	501

Table 2: Physical properties

	Stream 1	Stream 2
Density (kg/m^3)	0.55	0.55
Heat capacity (J/kg ºC)	1,059	1,059
Thermal conductivity (W/m^2 ºC)	0.0780	0.0789
Viscosity (cP)	0.0509	0.0509

Table 3: Fin type selection result comparison

	Base design		New design	
	Stream 1	Stream 2	Stream 1	Stream 2
Fin types	Strip-fin 1/10-19.35	Strip-fin 1/9-24.12	Strip-fin 1/10-19.74	Strip-fin 1/10-19.74
Fin pitch c (mm)	1.312	1.053	1.287	1.287
Plate spacing b (mm)	1.91	1.91	1.29	1.29
Fin length x (mm)	2.54	2.8	2.54	2.54
Fin thickness t_f (mm)	0.102	0.102	0.051	0.051

Table 4: Plate-fin heat exchanger dimension and performance comparison

	Base design	New design
Core volume (m^3)	3.18	2.563
Block length (m)	0.303	0.247
Block width (m)	3.24	3.24
Block height (m)	3.24	3.20
Number of passage-1	767	1,243
Number of passage-2	767	1,243
Film coefficient-1 (W/m^2 ºC)	340.2	318.9
Film coefficient-2 (W/m^2 ºC)	317.5	321.4
Pressure drop-1 (Pa)	8,800	8,800
Pressure drop-2 (Pa)	737	503

The thermal conductivity of fin is set as 90.0 W/m^2 ºC. Based on the GAMS optimisation result, the optimum fin pitches for both sides are 1 mm and 1 mm, fin lengths are 2 mm and 2 mm, and plate spacing are 0.9 mm and 1.1 mm, respectively. Compared with the standardised fin parts sizes, the closest fin type is strip-fin 1/10-19.74 for both sides by calculating ER value. The detail design result and comparison are presented as follow in Tables 3 and 4.

To determine the block dimensions, the width is assumed as 3.24 m in both the base and the new design. The length of block of heat exchanger in the new design is 0.247 m, which is smaller than that of the base design. Due to the fin type, the number of layers is much larger. However, the film coefficients are almost the same as the base design around 320 W/m^2 ºC. It is also clear from tables that the core volume is decreased by approximately 20 % in the new design. And the pressure drops of both sides in the new design are no more than those of the base design.

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 can not only provide a better design, but also improve the job efficiency, especially for multi-stream plate-fin heat exchanger design.

4. Conclusions

Integrating the continuous relationship of friction factor f, Colburn factor j, and some design parameters (d_h, f_s, β) with basic fin geometries into the thermal-hydraulic design model can convert the discrete plate-fin heat exchanger design problem into a continuous problem. Meanwhile, fin selection and imposed constraints could be optimised simultaneously in the design procedure. Furthermore, the minimum temperature approach can be determined by minimising the estimated total annual cost at the early design stage.

However, the design methodology employed in this study is based on using the same fin category for both sides and the constant physical properties. In practical, four fin categories are mixed and matched in the plate-fin heat exchanger. And verified physical properties induced by phase change are common in the low temperature process. Therefore, mixing and matching fin categories and dealing with verified physical properties in the plate-fin heat exchanger design should be considered in the future work.

Nomenclature

b – plate spacing, m;
b_{st} – standardised fin plate spacing, m;
c – fin pitch, m;
c_{st} – standardised fin pitch, m;
C – capital cost per unit $Q/\Delta T_{LM}$, £/(W/K);
d_h – hydraulic diameter, m;
ER – relative difference;
f_s – ratio of secondary surface area to total surface area;
i – interest rate;
n – number of years;
Q – heat duty of heat exchanger, W;
Re – Reynolds number;
t_f – fin thickness, m;
$t_{f,st}$ – standardised fin thickness, m;
$T_{cap,a}$ – total annual capital cost, £/a;
x – fin length, m;
x_{st} – standardised fin length, m;
xi – annual factor;
ΔT_{LM} – minimum temperature difference, ºC;
β - ratio of transfer area to volume of one side.

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