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Design of Heat Exchangers in Complex Arrangements for Use in Preheat Trains

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Welded plate heat exchangers such as compabloc units, have found wide application in heat recovery systems, particularly in crude preheat trains. Due to their construction features, these types of exchangers present a complex multipass arrangement with an overall counter-flow arrangement and a cross-flow arrangement between passes. In design, the number of passes is a degree of freedom that is used in order to increase the velocity and the pressure drop of the streams involved, with the number of passes per stream being equal or unequal. An important step in the design of heat exchangers is the determination of the correction factor of the logarithmic- temperature difference. In this paper the correction factor is calculated using the thermal effectiveness model and a general expression for the complex arrangements exhibited by this type of exchangers is developed and applied to design.

1. Introduction

Heat exchanger technologies such as compabloc units have had an important impact on applications where not only high temperature and pressure prevail but also where high fouling can occur (Andersson et al., 2009). The corrugated plates used in these units create high turbulence and wall shear stress which, apart from increasing the heat transfer rate, tend to mitigate fouling specially in cases where fouling is the result of chemical reaction. That is one of the reasons why this technology is finding great acceptation if refinery applications, particularly in crude preheat trains. Cases have been reported that even after eighteen months of operation in a crude processing plant, the performance remained as in the original stage. The benefits of using compabloc units in such applications are: less complicated handling compared to conventional exchangers due to the smaller size required for the same duty, and smaller downtime for maintenance (Anipko et al., 2006).
Further investigation of the use of compabloc heat exchangers in refinery applications show that better thermo-hydraulic performance is achieved when multi-passes are used (Tamakloe et al., 2013). In these units, a number of possible arrangements are possible giving rise to complex arrangements. In this work, the thermal effectiveness method is used to develop a sizing methodology for these units.
In the development of new heat exchanger technologies the goals sought after are: mechanical resistance to extreme operating conditions and high heat transfer coefficients compared to the amount of pressure drop being used. In the case of compabloc exchangers few technologies can compete against them and deliver the required heat duty and pressure drop in less surface area or volume. Thus the welded plate heat exchanger features important improvements over similar technologies such as the plate and frame heat exchanger (Picón et al., 2012). One advantage that allows these units to withstand higher pressures and temperatures is that the plates are welded instead of being sealed using polymer gaskets, feature that increases its mechanical resistance. A welded plate exchanger or compabloc, consists of stacks of corrugated plates which constitute the heat transfer area. The separation between the plates which are welded to one another form the channels that allows for the flow of the fluids. Inside the unit, streams flow in cross-flow fashion either in single or multiple passes. Overall, for maximum thermal performance, the system is arranged in counterflow fashion (Figure 1).

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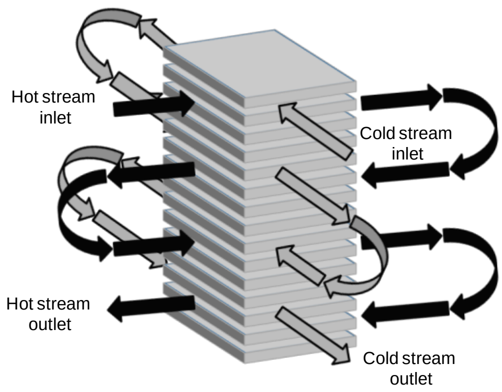

Figure 1: Schematic of a four pass compabloc heat exchanger with an overall counter-current arrangement

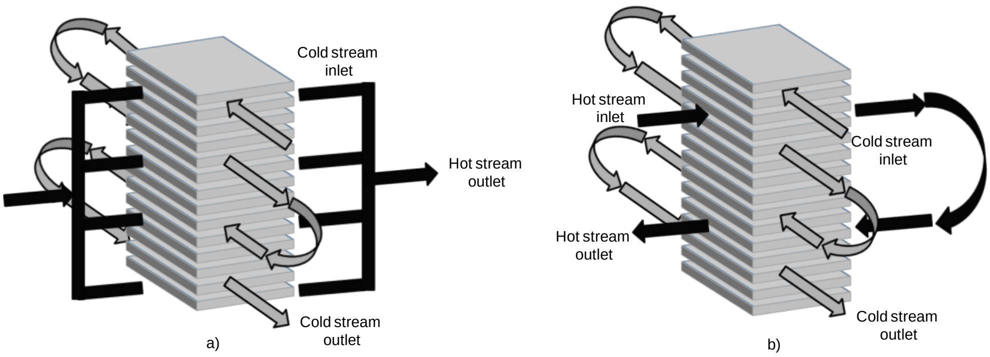

Figure 2: Exchanger flow arrangement: (a) four passes on the cold stream and one pass on the hot stream, (b) four passes on the cold stream and two passes on the hot stream

There are a large number of possible arrangement combinations in a compabloc unit. For instance, Figure 2 (a) shows a heat exchanger with four passes on the cold stream and one pass on the hot stream while Figure 2(b) shows an arrangement with four passes on the cold side and a two passes on the hot side.
The thermal design of single phase heat exchangers in complex arrangements require that a close attention be given to the determination of the heat transfer coefficients on both stream sides as well as the determination of the correction factor of the logarithmic temperature difference. According to Hesselgreaves (2001), the performance of the plates used in compabloc untis is similar to the performance of the type of plates used in plate and frame units. Typical heat transfer and friction correlations for plate and frame surfaces are the ones reported by Shah and Focke (1988):

$$N_u = \begin{cases} 0.729\,\mathrm{Re}^{1/3}\mathrm{Pr}^{1/3} & \text{for Re} \leq 7 \\ 0.380\,\mathrm{Re}^{2/3}\mathrm{Pr}^{1/3} & \text{for Re} \rangle\ 7 \end{cases} \tag{1}$$

$$f = \begin{cases} 17.0\,\mathrm{Re}^{-1} & \text{for Re} \langle\ 10 \\ 6.29\,\mathrm{Re}^{-0.57} & \text{for } 10 \langle \text{Re} \rangle\ 101 \\ 1.141\,\mathrm{Re}^{-0.20} & \text{for } 101 \langle \text{Re} \langle\ 855 \\ 0.58\,\mathrm{Re}^{-0.10} & \text{for Re} \rangle\ 855 \end{cases} \tag{2}$$

2. Analysis of complex heat transfer arrangements

In many situations, as in the case of the compabloc exchanger, the heat transfer duty can be achieved by a combination of a set of heat exchangers arranged in series. Each exchanger may exhibit a flow arrangement different from counter-flow, for instance: parallel flow, cross flow or a 1-2 pass arrangement. The performance of these associations of heat exchangers can be characterised by means of the overall thermal effectiveness as demonstrated by Domingos (1969). For the purposes of the present work, the overall thermal effectiveness is determined in terms of the individual thermal effectiveness and heat capacity rate ratio ($C=CP_{min}/CP_{max}$). The main assumptions made in the development of the expression are: steady state heat transfer, negligible heat losses to ambient, constant overall heat transfer on all exchangers, fluid perfectly mixed along the length of the channels and at the entrance and exit of each pass. For a set of four exchangers arranged in an overall counter-flow fashion (Figure 1), Domingos (1969) developed the following general expression o overall countercurrent arrangement:

$$\varepsilon_t = \frac{1-\prod_1^n \left(\frac{1-\varepsilon_i C}{1-\varepsilon_i}\right)}{C-\prod_1^n \left(\frac{1-\varepsilon_i C}{1-\varepsilon_i}\right)} \tag{3}$$

If the heat exchangers are all identical, the expression reduces to:

$$\varepsilon_t = \frac{\left(\frac{1-\varepsilon C}{1-\varepsilon}\right)^n - 1}{\left(\frac{1-\varepsilon C}{1-\varepsilon}\right)^n - C} \tag{4}$$

Where ε_t is the overall thermal effectiveness of the assembly, ε is the thermal effectiveness of the single exchanger and n is the number of exchangers in the assembly. CP is the product of the mass flow rate and the heat capacity. The term CP_{min} refers the stream that experiences the largest temperature change for the given heat load and C_{max} is the one that experiences the smallest temperature change of the two. The overall thermal efficiency can be regarded as a design parameter and can be calculated from the operating temperature conditions (inlet and outlet temperatures). For instance, if the hot stream is the CP_{min} stream we have that:

$$\varepsilon_t = \frac{T_{in} - T_{out}}{T_{in} - t_{in}} \tag{5}$$

The dimensionless number that relates the size of the unit and the heat transfer coefficient of the exchanger is the number of heat transfer units, represented by:

$$N_{tu} = \frac{UA}{CP_{\min}} \tag{6}$$

The thermal effectiveness and the number of heat transfer units are related to each other through an expression that depends on the flow arrangement. For instance, in the case of a cross-flow arrangement (Kays and London, 1982) we have:

$$\varepsilon = 1 - \exp\left[\left(\frac{1}{C}\right)(N_{tu})^{0.22}\left\{\exp\left[-C\left(N_{tu}\right)^{0.78}\right]-1\right]\right] \tag{7}$$

Kays and London (1984) show that the correction factor of the logarithmic mean temperature difference (F) can be expressed as:

$$F = \frac{N_{tu,countercurrent}}{N_{tu,other}} \tag{8}$$

The term $N_{tu,other}$, refers to the number of heat transfer units of an arrangement different from countercurrent and $N_{tu,countercurrent}$ refers to the number of heat transfer units if the arrangement is countercurrent. The expression for this arrangement is:

$$\varepsilon = \frac{1-\exp\left[-N_{tu}\left(1-C\right)\right]}{1-C\exp\left[-N_{tu}\left(1-C\right)\right]} \qquad \text{for C} \langle \ 1 \tag{9}$$

and

$$\varepsilon = \frac{N_{tu}}{1+N_{tu}} \qquad \text{for C} = 1 \tag{10}$$

For a given design problem, the overall thermal effectiveness can be determined form the inlet and outlet temperatures. For a given number of passes (n), the thermal effectiveness of the single cross flow exchanger can be calculated and the corresponding heat transfer units per pass ($N_{tu,pass}$) determined from Eq (7). The total number of heat transfer units is:

$$NTU_{other} = n \ \mathrm{NTU}_{\mathrm{pass}} \tag{11}$$

The number of thermal plates in a compabloc unit is calculated from the overall heat transfer area and the surface area per plate. The number of channels equals the number of thermal plates plus 1. From the general design equation for heat exchangers, the overall heat transfer area can be computed. To this end, the overall heat transfer coefficient and the correction factor of the logarithmic temperature difference are determined. As a first approach, a Re number is assumed and the appropriate heat transfer and friction factor correlations are used. The pressure drop across the unit can be determined from:

$$\Delta P = \frac{2nfLG^2}{\rho d_h} \tag{12}$$

Where n is the number of passes, G is the mass flow rate per unit area, L is the length of the plate and d_h is the hydraulic diameter given by:

$$d_h = \frac{2\delta}{\varphi} \tag{13}$$

Where δ is the spacing between plates and φ is the elongation factor. The cross sectional area to fully absorb the specified pressure drop can be determined from (Picon et al., 2010):

$$A_c = \left(\frac{2x\, \dot{m}^{2-y}\, \mu^y\, \mathrm{L}}{\mathrm{d_h}^{1+y}\, \rho\, \Delta P} \right)^{1/(2-y)} \tag{14}$$

And the number of thermal plates from:

$$N_{thermal,pass} = \frac{1}{(W\delta)} \left[\frac{2\, x\, \dot{m}^{2-y}\, \mu^y\, \mathrm{N_p}\, L}{\Delta P\, \rho\, \mathrm{d_h}^{1+y}} \right]^{\frac{1}{(2-y)}} \tag{15}$$

3. Case study

To demonstrate the approach, a case study is analysed (Arsenyeva et al., 2011). The physical properties and operating conditions are given in Table 1. The design approach starts by assuming a Reynolds number in order to identify the heat transfer coefficient and the friction factor correlation. Other assumptions are: the plate length, and the number of passes. The design methodology is summarised in Figure 3. The results of the application of the methodology for the case of a four pass and a five pass design are given in Table 2.

Table 1: Physical properties and operating conditions

	Cold stream	Hot stream
Mass flow rate (kg/s)	1.36	4.01
Core pressure drop (Pa)	10,000	10,000
Inlet temperature (°C)	28	95
Outlet temperature (°C)	79	90.5
Density (kg/m^3)	978.4	961.87
Heat capacity (J/kg°C)	3,180.0	4,209.2
Thermal conductivity (W/m °C)	0.66	0.6789
Viscosity (kg/m s)	0.0166	0.000297
CP	4,321.27	16,869.6
Thermal conductivity material of construction (W/m°C)	16.3	
C (CP_{min}/CP_{max})	0.26	
ΔTlm (°C)	19.15	
Heat load (kW)	269.9	
Thermal Effectiveness (ε)	0.9328	
Plate spacing (m)	0.005	
Plate thickness (m)	0.0020	
Elongation factor	1.15	

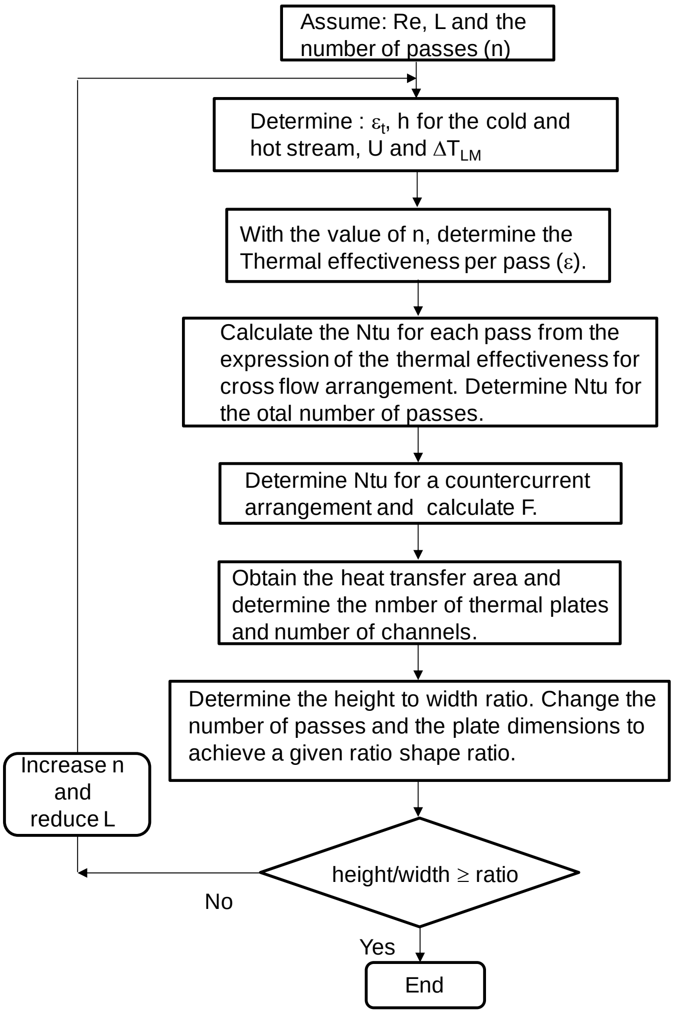

Figure 3: Flow diagram of the design methodology

Table 2: Results for four pass and five pass designs

	Four pass design	Five pass design
Heat transfer area (m^2)	7.73	9.7
Plate length (m)	0.30	0.20
Number of thermal plates	46	124
Number of channels per pass	6	12
Re cold stream	96	64.8
Re hot stream	16,102	10,776.8
h cold stream ($W/m^2 °C$)	2,295.5	1,787.7
h hot stream ($W/m^2 °C$)	20,500.0	15,813.0
U ($W/m^2 °C$)	1,874.4	1,488.8
NTU_{cc}	3.26	3.26
NTU_{pass}	0.84	0.67
NTU_{other}	3.3.6	3.34
Correction factor (F)	0.972	0.98

4. Conclusions

This paper shows the development of a design methodology for the sizing of welded plate heat exchangers with a complex flow arrangement. The construction features and thermo-hydraulic performance of these types of exchangers make them a suitable exchanger technology for applications subject to high fouling and operating conditions. In this type of technology, fluids experience high shear stress due to the small hydraulic diameter and also due to the possibility of increasing fluid velocity by manipulation of the number of passes. The use of the thermal effectiveness-number of transfer units model for the calculation of the correction factor of the logarithmic temperature difference is demonstrated on a case study. The number of passes becomes a design parameter that can be manipulated to determine the appropriate final block dimensions.

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