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Design and Operability of Multi-Stream Heat Exchangers for Use in LNG Liquefaction Processes

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Among the diverse types of heat exchanger technology, the geometrical features of plate-fin exchangers provide the required flexibility for the incorporation of large number of streams in a single unit. The main design and operating issues related to multi-stream exchangers are linked to the complex thermal flows that take place inside the unit. Several simplifications have been proposed to reduce this complexity and derive design methodologies. One of these simplifications is the uniform channel heat load which gives rise to the uniform wall temperature assumption. This simplification assumes that at any plane cut normal to the flow of the fluids inside the exchanger, the temperature of the separating walls is the same. One way of practically achieving such condition is by appropriate surface selection. The thermal performance of secondary surfaces has been extensively studied and vast information on their thermo-hydraulic performance has been published. A design criterion that has emerged in the last decade is to engineer secondary surfaces to achieve the required thermal performance. Such approach is taken advantage of in this work so that the concept of uniform wall temperature in design can be effectively met. This paper shows the derivation of a surface design methodology. The approach is demonstrated on a case study using triangular shape fins.

1. Introduction

In plate-fin multi-stream heat exchangers, fluids flow through channels separated by metal walls. Between the metal walls, secondary surfaces are placed to carry out three main functions: a) provide mechanical resistance between plates; b) increase the heat transfer surface area, and c) increase the heat transfer coefficient. In multi-stream exchangers, complex thermal flow paths take place inside the exchanger (Picón-Núñez, 2002). A way to simplify the analysis and design of multi-stream heat exchangers and eliminate these complex thermal flow paths is by means of the uniform channel heat load that gives rise to the uniform wall temperature assumption (Chato et al.,1971). This simplification assumes that at any cross-sectional area along the length of a multi-stream unit, the temperature of any separating wall is the same. For this assumption to be valid, each channel must exhibit a like thermal performance. On the operating side, multi-stream heat exchangers as any other type of heat recovery network, are likely to be subject the temperature and flow rate disturbances giving rise to complex interactions between the streams within the unit. Reduction of the adverse effects upon the most critical variables is achieved through flow passage arrangement. In turn, flow passage arrangement becomes a design variable if uniform heat load per channel is achieved. Such condition is intimately linked to an appropriate surface selection or, as it is shown in this work, by surface design.

Given the different physical properties of the streams involved in a multi-stream application, the different flow rates and different pressure drops they are likely to have, uniform heat load per channel can be achieved by manipulation of the thermal performance of the streams as they flow through the channels. Since the thermal performance is dictated by the type of secondary surface, a key issue in the design of plate fin heat exchangers is their appropriate specification. There are various types of secondary surfaces available for design; among them are: plain fin, louvered fin, off-set strip fin and wavy fin. Within each type of surface there are a certain number of specific geometries (Kays and London, 1984). The selection of a surface for a given application is made based on its thermal-hydraulic performance that refers to the way the heat transfer coefficient and the friction factor vary with respect to the Reynolds number. “Surface selection” is a process whereby a given surface is chosen from among the available pool of options whereas, “surface design” consists in the

identification of the surface geometrical features that will meet a design criterion. The underlying assumption in surface design is that geometrical variables become continuous over a practical range of values.
In the design of multi-stream heat exchangers, it would be desirable to engineer the type of secondary surface required for a given application (Kunpeng et al., 2015). This can be achieved if accurate generalized correlations are available for each type of surface. These correlations must be able to predict the thermal-hydraulic performance through all flow regimes, namely; laminar, transition and turbulent (Picón-Núñez et al., 2009). In this work, the use of a generalized correlation for triangular shape fins is explored and applied to the design multi-stream exchangers.

2. Uniform channel heat load

The concept of uniform channel heat load that gives rise to the uniform wall temperature condition is graphically represented in Figure 1. An assumption in the derivation of this model is that the effect of end plates is neglected. An end plate is the one that has no contact with other stream on the other side of the wall. In Figure 1, the term q_1, represents the heat load transferred by the channel occupied by stream H1; q_2 and q_3 are the heat loads transferred by stream H2 and H3. From this diagram, it follows that:

$$q_1 = (\eta hA)_1(T_{h1} - T_{w1}) \quad (1)$$

$$q_2 = (\eta hA)_2(T_{h2} - T_{w2}) \quad (2)$$

Where η is the fin thermal efficiency, h is the heat transfer coefficient (W/m^2°C) and A is the total surface area. Within an enthalpy interval the following condition applies:

$$T_{h1} = T_{h2} \quad (3)$$

For the common wall temperature to apply, it is assumed that:

$$T_{w1} = T_{w2} \quad (4)$$

Therefore, from Eq(1) and Eq(2) it follows that for the hot stream and for cold streams:

$$(\eta hA)_{h1} = (\eta hA)_{h2} \quad (5)$$

$$(\eta hA)_{c1} = (\eta hA)_{c2} \quad (6)$$

For a single stream, the term (ηhA) is expressed as:

$$(\eta hA) = (\eta h)\beta V \quad (7)$$

Where V is the product between the length (L), width (W) and height (H) and since the length and width are the same for all streams, an effective thermal performance per stream can be expressed as:

$$(\eta hA)_{effective} = (\eta h\beta\delta) \quad (8)$$

Where β is the fin density (m^2/m^3) and δ is the plate spacing(m) and both are surface geometrical properties.

3. Surface design

Plate-fin heat exchangers are referred to as low Reynolds heat transfer surfaces since the Reynolds number is expressed in terms of the hydraulic diameter which normally has a small value. Experimental heat transfer data are not readily available for all flow regimes; so, in most cases, analytical solutions for the limiting Nusselt number in the laminar region can be solved. For a heat transfer condition of constant wall temperature and fully developed flow, Shah and Bhatti (1987) proposed a generalized expression to determine the limiting Nusselt number as a function of the characteristic angle:

$$Nu = 0.943\,[\alpha^5 + 5.3586\alpha^4 - 9.2517\alpha^3 + 11.9314\alpha^2 - 9.8035\alpha + 3.3754]/\alpha^5 \quad (9)$$

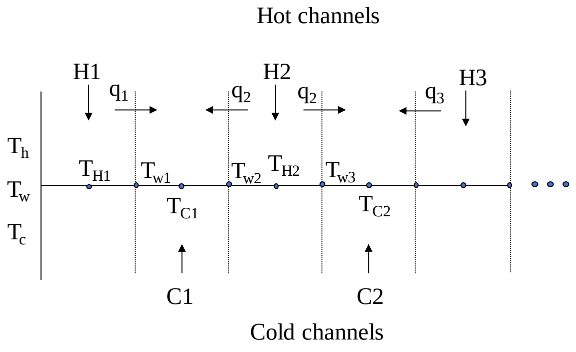

Figure 1: Graphical representation of the uniform heat load per channel concept

Where α is the aspect ratio given by δ/2a. Figure 2 shows the main geometrical features of triangular fins where F_{th} is the fin thickness and 2θ is the characteristic angle.

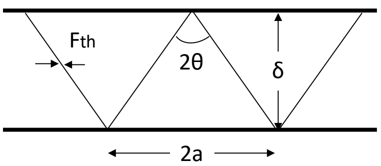

Figure 2: Characteristic angle (2θ) for isosceles triangular ducts

The frictional performance of triangular surfaces as a function of the angle (2θ) is determined from the following expression (Carreón, 2008):

$$f = \frac{24 - 0.178(2\theta)}{Re} \tag{10}$$

The frictional pressure drop across the core of the heat exchanger can be calculated form:

$$\Delta P = \frac{2fL\,m^2}{\rho d_h A_c^2} \tag{11}$$

Where *m* is the mass flow rate (kg/s), *ρ* is the fluid density (kg/m^3), d_h is the hydraulic diameter (m) and *Ac* is the free flow area (m^2). For the specification of a triangular fin surface, Eq(12) to (14) show the physical dimensions as a function of the surface parameters. For the free flow area, it follows that:

$$A_C = \frac{1}{2}(2a - 2F_{th})(\delta - F_{th}) \tag{12}$$

Where F_{th} is the thickness of the fin (m). The hypotenuse is expressed as:

$$hyp = \frac{\delta - 2F_{th}}{Cos\ \theta} \tag{13}$$

While the wetted perimeter is:

$$P_w = (2a - 2F_{Thickness}) + 2(\delta - F_{th})/cos\theta \tag{14}$$

The fin hydraulic diameter is defined as four times the free flow area divided by the wetted perimeter. So, the expression for the hydraulic diameter is:

$$d_h = \frac{2(a - F_{th})(b - F_{th})}{(a - F_{th}) + (\delta - F_{th})/cos\theta} \tag{15}$$

The fin efficiency can be determined from:

$$\eta = 1 + f_s\left\{\frac{tanh\left[(2h/k\tau)^{1/2}(\delta/2)\right]}{\left[(2h/k\tau)^{1/2}(\delta/2)\right]} - 1\right\} \tag{16}$$

The term f_s is the ratio of the secondary surface area to that of the total surface area and can be expressed as:

$$f_s = \frac{(\delta - F_{th})/cos\theta}{[(a - F_{th}) + (\delta - F_{th})/cos\theta]} \quad (17)$$

For the triangular surface used in this work, the product (βδ) is:

$$\beta\delta = \frac{2[(a - F_{th}) + (\delta - F_{th})/cos\theta]}{a} \quad (18)$$

4. Design methodology

The methodology proposed by Picón-Núñez et al. (2002) for the design of plate-fin multi-stream heat exchangers is used in this work. The methodology assumes that the hot and cold temperature fields at any sectional section are uniform. Such assumption ensures that temperature driving force is efficiently used in the unit and eliminates the distortion of temperature fields and any adverse heat flow in the exchanger except for longitudinal heat conduction (Chato et al., 1971). In a multi-stream unit both hot and cold streams enter the unit at different temperatures and are heated or cooled to different temperature span. In physical terms, this means that the streams must enter and leave the unit at different points along the length of the exchanger. These points correspond to the slope changes in the Composite Curves. Drawing a vertical line at any slope-changing point determines the Enthalpy Intervals. In turn, Enthalpy Intervals indicate the number of sections that compose a whole multi-stream exchanger. In consequence, the design of a multi-stream exchanger is broken down into various smaller problems. The design challenge is to physically size each of the individual sections in a way that they are consistent with one another in terms of dimensions. For instance, the different sections must have the same width and height. Within each section, the number of hot channels must equal the number of cold channels. The methodology starts by choosing secondary surfaces for the initial calculation of the term (ηhA). One hot stream and one cold stream are chosen as reference streams. The purpose is to set the reference streams as the basis for the homologation of the (ηhA) value. Once these reference values are known, the surfaces of the hot streams are engineered so that all of them will exhibit the same (ηh*A*) value; the same applies for all cold streams. In the case of the triangular surfaces used in this work the limiting Nusselt number is calculated; from this value, the characteristic angle is determined from Eq(9). The new friction factor is obtained from Eq(10) and the new pressure drop from Eq(11). If the new pressure drop is within the permitted limits for the stream in that enthalpy interval, the hydraulic diameter and the number of fins per inch of the new fin are determined from Eq(12) to (15).

5. Case study

The case study refers to the design of multi-stream exchanger for a LNG liquefaction process operating with a nitrogen recompression refrigeration system. The fin thickness ($F_{Thickness}$) used in the problem is 0.0003 m and a plate spacing (δ) of 0.0065 m.

Table 1: Process data for case study

Stream type	ṁ [kg/s]	C_p [kJ/kg]	T_{in} [°C]	T_{Out} [°C]	Q [kW]
H_1 (NG)	5	2.214	25	-152.7	1967.14
H_2 (REF)	10.75	1.04	30	22.63	82.40
C_1 (REF)	10.75	1.04	-160	24.27	-2060.14

Table 1 presents the operating data of the problem. The Composite Curves of the process are shown in Figure 3 for a 5°C minimum temperature difference and Figure 4 shows the stream population on a Grid Diagram (Klemeš et al., 2014). From this diagram it can be observed that three streams need to exchange heat in interval II. The rest of the intervals contain only two-stream problems and will not be considered in this study. The pressure drop per stream for design purposes is distributed linearly across the enthalpy intervals as given in Table 2. After a first calculation round using the approach by Picón-Núñez et al. (2002) and choosing a triangular surface 16.96T for streams H1 and H2 and for the cold stream; the resulting block dimensions are determined as shown in Table 3. The detail design of Enthalpy Interval II is shown in Table 4. The (ηhA) value of stream C1 is 8.78 (W/°C). This value is taken as a reference and from it, new triangular surfaces are engineered such that the other two hot streams exhibit the same value.

An iterative approach for the design of the new surfaces is as follows: a) For each stream, the value of h is determined from the original η and A values and a target (ηhA) of 253.3 (W/°C); b) with the value of h, a Nu_{lim} is calculated from $Nu_{lim}= d_h$ h/k; c) with the value of Nu_{lim}, the characteristic angle (2θ) is determined from Eq(9); d) the value of the base of the triangle (2a) is determined from: Tanθ=a/(δ-F_{th}); e) the pressure drop associated to the new surface is determined from Eq(10) and (11). If the resulting value is within limits, the surface hydraulic diameter (d_h) and number of fins per inch (1/2a) are determined; f) the new (ηhA) is calculated. If the value is equal or less than an error with respect to the reference value, the surface has been designed. Otherwise, the process is repeated until convergence.

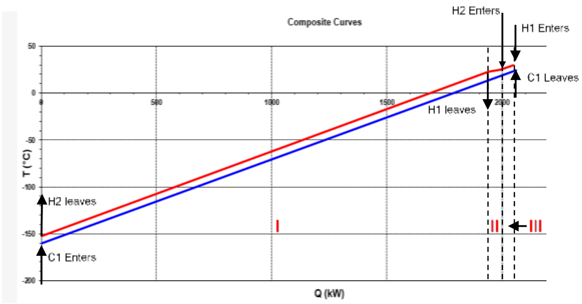

Figure 3: Composite curves for a 5 °C ΔT_{min}

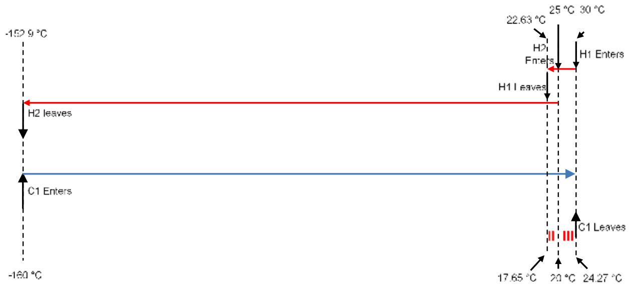

Figure 4: Stream population per enthalpy interval for case study

Table 2: Pressure drop distribution

Stream	I	II	III	Total ΔP (kPa)
$H1$ (NG)	29.6	0.4		30
$H2$ (REF)		8.0	17.0	25
$C1$ (REF)	24.1	0.3	0.6	25

Table 3: Block dimensions

Interval	Length (m)	Width (m)	Height (m)	Volume (m^3)
I	0.762	2	2.125	3.24
II	0.012	2	2.135	0.57
III	0.021	2	2.158	0.09

Table 4: Initial detailed design of section (II)

Stream	No. of passages	Surface Type	Plate spacing (m)	Hydraulic diameter (m)	Fins per inch	(ηhA) (W/°C)	Re	ΔP (kPa)
H1	119.6	16.96T	0.0065	0.001722	16.96	7.063	128.4	0.27
H2	75.7	16.96T	0.0065	0.001722	16.96	10.13	718.8	1.33
C1	195.3	16.96T	0.0065	0.001722	16.96	8.78	1014	2.5

The resulting surfaces and the final block performance data are shown in Table 5. Since the Reynolds number of Plate-fin heat exchangers tends to be low as it is calculated using the hydraulic diameter that for this type of exchangers has a small value, the use of the Nu_{lim} value for laminar flow is justifiable.

Table 5: Final detailed design of section (II) after four iterations

Stream	No. of passages	New surface	Plate spacing (m)	Hydraulic diameter (m)	Fins per inch	(ηhA) (W/°C)	Nu_{lim}	2θ	f	New ΔP (kPa)	Permitted ΔP (kPa)
H1	426.7	28.69T	0.0065	0.0002751	28.69	8.58	1.623	2.64	1.295	0.10	0.53
H2	260.9	25.39T	0.0065	0.0003808	25.39	8.80	1.809	3.7	0.0866	1.62	8.0
C1	687.6	21.17T	0.0065	0.0005565	21.17	8.78	1.71	5.54	0.0222	2.40	0.39

The results show the case of the design using triangular surfaces; however, more work is needed to derive the appropriate generalized correlations for other types of surfaces. This concept can be implemented within an optimization approach to target minimum exchanger volume and cost. Such work is underway.

6. Conclusions

The engineering of secondary surfaces to meet specific heat transfer duties offers new design options in plate-fin multi-stream heat exchangers. Although the work is at a theoretical level, there are two potential applications at the design stage that can be tackled with this approach: the sizing of the unit to required dimensions and the flow channel distribution. This paper has covered only the sizing aspect, however, channel distribution despite being a design output, it must be refined by operating issues at a later stage. The concept of designing heat exchangers to meet specified dimensions is not new but is a challenging aspect in the design of plate-fin exchangers. This is an area that has received little attention but in theory, subject to the limitations imposed by pressure drop, any shape and dimensions could be achieved by surface engineering. The work presented in this paper demonstrates that traditional design simplifications in multi-stream heat exchangers such as uniform channel heat load and common wall temperature can be readily attainable if the thermal performance of the streams becomes a design objective rather than a design constraint. As mentioned, this can be achieved by surface engineering or surface design. Broadly speaking, surface design refers to the specification of the surface geometrical parameters to meet specific thermal performance represented by the term (ηhA). The underlying assumption that drives the design of secondary surfaces to meet a specific thermal performance is that most internal thermal flow paths in multi-stream heat exchangers can be eliminated; for instance, uneven heat load distribution, heat load bypassing between streams, etc., the only exception being longitudinal heat conduction. One important condition that must be met for effective surface design is that reliable and generalized thermal-hydraulic correlations for secondary surfaces must be available. This work has shown how a single and simple type of surface can be used to meet the design specification and thermal performance in multi-stream heat exchangers. The application to larger and more complex problems requires the development of the appropriate correlations for other types of surfaces.

References

Chato J.C., Laverman R.J., Shah J.M., 1971, Analyses of parallel flow, multi-stream heat exchangers, Journal of Heat Mass Transfer, 14, 1691-1703.

Carreón C.E., 2008, Development of procedures for the design of multi-stream heat exchangers. PhD Thesis, University of Guanajuato, Guanajuato, Mexico.

Kays W.M. and London A. L., 1984, Compact heat exchangers, Mac Graw Hill, New York, USA.

Klemeš J.J, Varbanov P. S., Alwi S. R.W., Manan Z. A., 2014, Process integration and intensification saving energy, water and resources, De Gruyter, Berlin, Germany.

Kunpeng G., Shang, N. Smith R., 2015, Optimization of fin selection and thermal design of counter-current plate-fin heat exchangers, Applied Thermal Engineering, 78, 491-499.

Picón-Núñez M., Carreon C.E. and Polley G.T., 2009, The engineering of compact exchangers to required dimensions, Proceedings of the 1st Annual Gas Processing Symposium, Qatar. H. Alfadala, G.V. Rex Reklaitis and M. M. El-Halwagi (Eds.), Elsevier B.V., 88- 95.

Picón-Núñez M., Polley G.T. Medina-Flores M., 2002, Thermal design of multi-stream heat exchangers, Applied Thermal Engineering, 22, 1643, 1660.

Shah R.K., Bhatti M.S., 1987, Laminar convective heat transfer in ducts, Chapter in: Kakac S., Shah R.K., Aung W. (Eds.), Handbook of single phase convective heat transfer, Wiley, New York, 123-158.