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Alternative design approach for spiral plate heat exchangers

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This work presents an alternative design approach for the sizing of spiral heat exchangers in single phase counter-current applications. In this approach, pressure due to friction can be directly related to the heat transfer coefficient through the exchanger geometry, thus resulting in a sizing methodology that maximises pressure drop and results in the design of the smallest dimensions. In counter-current arrangement both fluids have the same flow length. A degree of freedom in design is the plate spacing that may be changed in order for both streams to maximise its allowable pressure drop. The approach is compared with design results presented in the open literature and the results show that the method is reliable and easy to implement.

Key words: Spiral plate heat exchangers, pressure drop, thermo-hydraulic model.

Introduction.

Spiral plate heat exchangers consist of plates rolled together forming a spiral. The space between the plates is kept by welding bolts to form the channels for the flow of the fluids. In single phase applications, it is common for the hot stream to enter the exchanger through the central part of the exchanger and to exit at the periphery. The cold fluid, on the other hand, enters the unit from the outermost part of the unit and circulates to eventually exit the exchanger from the centre as shown in Figure 1.

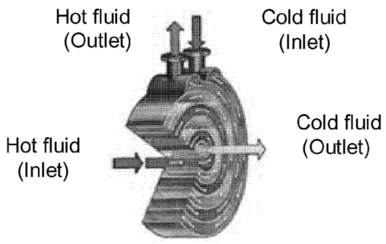

Figure 1. Flow pattern for counterflow arrangement in spiral heat exchangers.

One of the main advantages of spiral plate exchangers with respect to other exchanger technologies is its capacity to handle dirty fluids, exhibiting lower tendency to fouling. This is due to the particular geometry that creates a constant change in the direction increasing local turbulence that eliminates fluid stagnant zones. The use of this type of exchanger is common in the paper, petrochemical, food and sugar industry with applications in evaporation and condensation (Trom, 1995). Due to the counter-current flow pattern, in single phase processes this exchanger can be used in duties that involve high thermal effectiveness. In addition, its geometrical features make them suitable to accommodate a large heat transfer area in a relatively small volume. The operating pressures of the fluids impose a limitation to the application of this technology. For instance, the maximum plate thickness that can be rolled is 0.013 m, which limits the maximum allowable pressure difference between the fluids to 27.57903 bar, (Rodríguez 2000).

Bes y Roetzel (1992) developed an analytical rating study to determine the temperature profile within the plates. They also studied the influence of various geometrical parameters in the design and thermal performance. In later work, Bes y Roetzel (1993) determined the thermal performance using a dimensionless approach that gave better results as the number of spirals in the unit increases. A thorough description of the exchanger geometry was published by Dongwu (2003); he provides the expressions to calculate spiral diameter, number of turns and length of the semicircles.

In this work we introduce a new methodology for the design of spiral plate exchangers based on the maximisation of pressure drop. In this approach plate width and plate length are taken as a continuous variable.

Methodology

The following assumptions are made in the development of the model: the effects of the change of velocity at the entrance and exit of the exchanger are neglected; pure counter-flow arrangements is assumed; steady state; the heat transfer coefficient is constant along the length of the exchanger; losses to ambient are negligible; the bolts separating the plates are considered not to affect the fluid movement.

The basic concept of the methodology is the development of a thermo-hydraulic model that relates the heat transfer coefficient with the pressure drop through the geometry of the exchanger and the physical properties (Picón et al., 1999 and Picón et al., 2006). A set of equations are solved simultaneously to find the exchanger geometry such as spiral diameter and plate width. The initial information is the operating data and the stream physical properties. Next a plate width (H), spiral diameter (Ds), plate spacing and plate thickness are fixed.

The choice of the right expression for heat transfer and pressure drop depends on the values of the Reynolds and the critical Reynolds number (Eqs. 1 and 2)

$$\mathrm{Re} = \frac{10000\left(\frac{F}{1000}\right)}{H\mu} \tag{1}$$

$$\mathrm{Re}_c = 20000\left(\frac{D_e}{D_h}\right)^{0.32} \tag{2}$$

Where F is the mass flow rate, H is the plate width, μ is the viscosity, D_e is the equivalent diameter and D_h is the hydraulic diameter.

The length of the exchanger (L) is calculated using the allowable pressure drop from corresponding equation (3,4 or 5).

For single phase, Re > Re_c

$$\Delta P = 0.001\frac{L}{s}\left[\frac{F}{d_s H}\right]^2\left[\frac{1.3\mu^{\frac{1}{3}}}{(d_s+0.125)}\left(\frac{H}{F}\right)^{\frac{1}{3}}+1.5+\frac{16}{L}\right] \tag{3}$$

Without phase change, 100 < Re < Re_c

$$\Delta P = 0.001\frac{L}{s}\left[\frac{F}{d_s H}\right]\left[\frac{1.035\mu^{\frac{1}{2}}}{(d_s+0.125)}\left(\frac{\mu_f}{\mu_b}\right)^{0.17}\left(\frac{H}{F}\right)^{\frac{1}{2}}+1.5+\frac{16}{L}\right] \tag{4}$$

Without phase change, Re < 100

$$\Delta P = \frac{Ls\mu}{3385(d_s)^{2.75}}\left(\frac{\mu_f}{\mu_b}\right)^{0.17}\left(\frac{H}{F}\right) \tag{5}$$

Where ΔP is the fluid pressure drop, s is the specific gravity, d_s and is the plate spacing.

Once the plate length is known, the real pressure drop is calculated according to equations (3, 4 or 5) as applicable. This allows for the calculation of the velocity of the fluid using equation (6).

$$Vf = \frac{F}{Ac} \tag{6}$$

Where Ac is the free flow area.

The Prandtl number on the hot and cold side can be obtained from equation (7).

$$\Pr = \frac{Cp\,\mu}{k} \tag{7}$$

Where, Cp is the heat capacity and k is the thermal conductivity of the fluid.

The correlations for the calculation of the heat transfer coefficients as reported by Minton (1970) are given by equations (8 and 9):

Liquid no phase change, Re > Re_c

$$h = \left(1+3.54\frac{D_e}{D_h}\right)0.023CpV_f\,\mathrm{Re}^{-0.2}\,\Pr^{-\frac{2}{3}} \tag{8}$$

Liquid no phase change, Re < Re_c

$$h = 1.86CpV_f\,\mathrm{Re}^{-\frac{2}{3}}\,\Pr^{-\frac{2}{3}}\left(\frac{L}{D_e}\right)^{-\frac{1}{3}}\left(\frac{\mu_f}{\mu_b}\right)^{-0.14} \tag{9}$$

Where, h is the heat transfer coefficient, μ_f is the fluid viscosity at wall temperature conditions and μ_b is the fluid bulk viscosity. Knowing these values, the overall heat transfer coefficient can be calculated from equation (10):

$$U = \frac{1}{\frac{1}{h_1} + \frac{x0.0833}{k_w A_p} + \frac{1}{h_2}} \tag{10}$$

Where, U is the overall heat transfer coefficient, k_w is the thermal conductivity of the material of construction and A_p is the surface area of the rolled plate.

The logarithmic mean temperature difference is obtained from equation (11):

$$\Delta T_{LM} = \frac{(T_{c1} - T_{f2}) - (T_{c2} - T_{f1})}{In \frac{(T_{c1} - T_{f2})}{(T_{c2} - T_{f1})}} \tag{11}$$

Where, T_{c1} and T_{c2} are the hot stream inlet and outlet temperature and T_{f1} y T_{f2} are the cold stream inlet and outlet temperature respectively. The heat transfer area is calculated from equation (12):

$$A_T = \frac{q}{U \Delta T_{LM}} \tag{12}$$

Where, q is the heat load. Finally, the plate width and the spiral diameter are calculated from equations (13) and (14):

$$H_{calc} = \frac{A_T}{2L} \tag{13}$$

$$D_{s\,calc} = \left[15.36L(ds_f + ds_c + 2p) + C^2\right]^{\frac{1}{2}} \tag{14}$$

If the value of these variables is different from the initial guess, the process starts again using the calculated values in place of the initial ones until convergence is achieved.

Results and Discussion

The proposed model is validated using the case study reported by Minton (1970). The physical properties for the case study are shown in Table 1. The algorithm was implemented on a Math-cad code; an iterative approach permits the adjustment the plate width and the spiral diameter simultaneously. For the case study, a plate spacing of 6.35E-3 m, an internal diameter of 0.203 m and a plate thickness of 3.175E-3 m were chosen.

In the design presented by Minton (1970), the allowable pressure drop on the hot and cold side is 6.894 bar. However, his approach does not fully utilise available pressure drop thus his results present a design with a pressure drop of 4.1851 on the hot side and 5.936 on the cold side. The heat transfer area is 15.515 m^2; a plate length of 12.741 m and a plate width of 0.61 m. Using these same pressure drops, the new algorithm gives a heat transfer area of 15.45 m^2, a plate length of 12.68 m and a plate width of 0.61 m. The results are almost identical.

Table 1. Stream data for case study.

	Hot stream	Cold stream	
Flow rate	2.824E+3	2.688E+3	kg/hr
Inlet temperture	473.15	333.15	K
Outlet temperatura	393.15	273.15	K
Heat capacity	2.973	2.763	kJ/kg K
Termal conductivity	0.348	0.322	W/m K
Specific gravity	0.0581228	0.0581228	bar
Pressure drop	6,894757	6,894757	bar
Viscosity	3.35E-3	8E-3	kg/m s
Plate thickness	3.175E-3		m
Internal diameter	0.203		m
Plate spacing	6.35E-3		m

Table 2. Comparison of results between new method and Minton's.

	Minton Design	*New method*	
Re (Hot side)	3839	4390	
Re (Cold side)	1530	1750	
Heat load	1.865E+5	1.865E+5	W
Heat transfer area	15.515	14.772	m^2
Heat transfer coefficient. Hot side	-	455.17	W/m^2 K
Heat transfer coefficient. Cold side	-	467.15	W/m^2 K
Overall heat transfer coefficient	220.31	230.53	W/m^2 K
Plate length	12.741	13.81	m
Plate width	0.61	0.5334	m
Spiral diameter	0.593	0.61468	m
Pressure drop. Hot side	4.18E-02	4.895277	bar
Pressure drop. Cold side	5.93E-02	6.89E-02	bar
Plate spacing	6.35E-3	6.35E-3	m

When the new algorithm is run to fully utilise the full pressure drop of 6.89E-02 bar, the results are shown in Tale 2, where as it is expected, the new design is smaller. Plate spacing is kept the same for both fluids so only the cold stream is allowed to maximise its pressure drop.

Conclusions

This work makes use of the concept of pressure drop maximisation to develop a design methodology of spiral plate heat exchangers. The application of the approach results in a simple, quick and easy to implement methodology. For the same duty, this type of exchanger gives less heat transfer area compared to shell and tube exchangers which, in some cases, has some advantages in terms of weight, volume and cost. The geometrical features of spiral plate heat exchangers allow for the incorporation of large heat transfer area into a relatively small volume compared to conventional units. Besides, the continuous change of direction of the fluids favours the generation of high heat transfer coefficients.

References

Bes, TH. y Roetzel, W., 1992, Distribution of heat flux density in spiral heat exchangers, International Journal of Heat and Mass Transfer, Vol. 35 no.6, pp. 1331-1347.

Bes, TH. y Roetzel, W., 1993, Thermal Theory for Spiral Heat Exchanger, International Journal of Heat and Mass Transfer, Vol. 36 no. 3 pp. 765-773.

Dongwu, W., 2003, Geometric calculations for the Spiral Heat Exchanger, Chemical Engineering Technology, 26, 592-598.

Minton, P. E., 1970 Designing Spiral Heat Exchangers, Chemical Engineering, pp. 103-112

Picón-Núñez, M., Polley, G. T., Torres-Reyes, E., Gallegos-Munoz, A., 1999, Surface selection and design of plate-fin heat exchangers, Applied Thermal Engineering, 19, 917-931.

M. Picón-Núñez, J.L. López-Robles and G. Martínez-Rodríguez, 2006, Alternative design approach for multi-pass and multi-stream plate heat exchangers for use in heat recovery systems, Heat Transfer Engineering, Vol. 27, No. 6.

Rodríguez, J., A., 2000, Intercambiadores de placas en espiral, Introducción a la Termodinámica con Algunas Aplicaciones Ingeniería, Ed. 1, 692-696.

Trom, L., 1995. Use spiral plate exchangers for various applications, Hydrocarbon Processing, v.74, n.5, pp. 73-81.