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# Thermo-Hydraulic Design of Single and Multi-Pass Helical Baffle Heat Exchangers

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Helical baffle heat exchangers offer and attractive option in terms of reduced heat transfer area for the same pressure drop and heat duty compared to conventional exchangers. Besides, the flow patterns inside the helical shell promote higher heat transfer coefficients and even reduced fouling. This type of technology has successfully been used in crude processing. Open literature design methodologies for helical baffle exchangers make use of the Bell-Delaware approach and applications deal with the design single shell and single tube pass arrangements. In certain applications the fluid velocity and the pressure drop can be increased by increasing the number of passes and even the number of units in series, thus giving rise to complex thermal arrangements. In such applications a correction factor for the logarithmic mean temperature difference must be determined. Figure 1 shows a two shell pass and one tube pass unit. As the hot fluid enters and flows along the length of the exchanger, it encounters the opposite fluid flowing in parallel flow fashion whereas as the shell fluid enters the outer section of the shell it flows counter-currently. A shortcut design approach based on the simplified determination of the correction factor of the log mean temperature difference is developed and demonstrated on case studies.

## 1. Thermo-hydraulic model

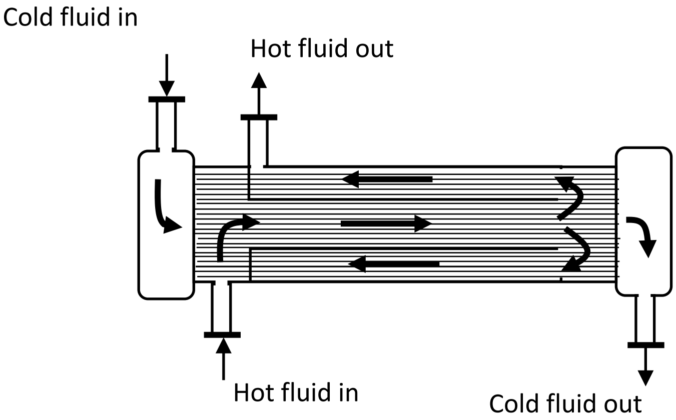


*Figure 1: 2-1 Pass helical baffle heat exchanger.*

Helical baffle heat exchangers also called “helixchangers” exhibit a more uniform flow distribution on the shell side compared to conventional shell and tube exchangers for the same pressure drop (Sivarajan et al,. 2013). They are also capable of reducing tube vibrations and fouling. Even though their manufacturing costs are higher the benefits in terms of reduced maintenance and operating costs make them superior in the long term (Movassag et al., 2012). The main geometrical parameters that define this type of

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technology are: helical pitch (distance between two consecutive baffles); the helical angle (the angle formed between the helix and the vertical) and the shell diameter (Zhang et al., 2010). See Figure 2.

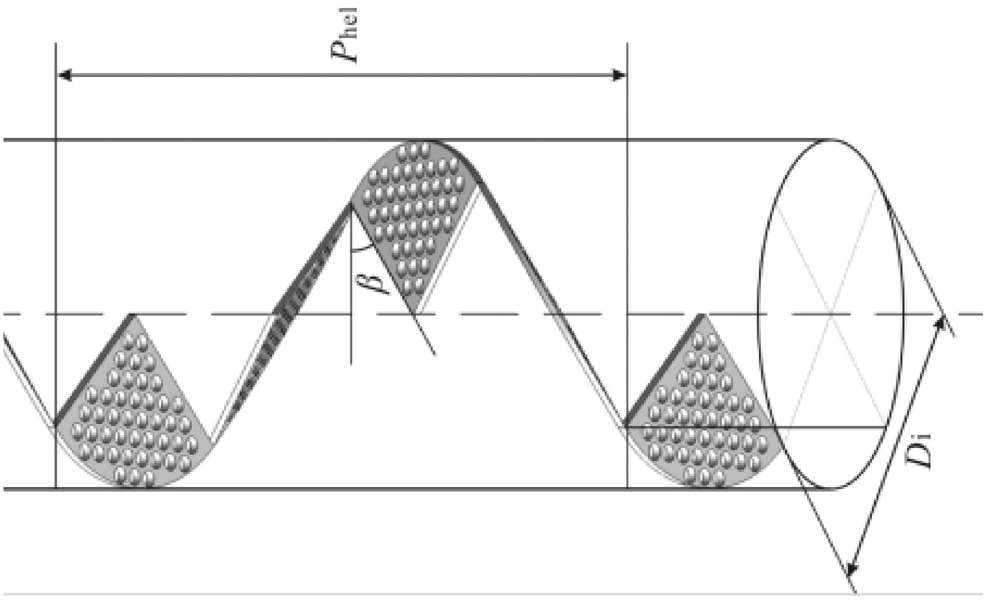


*Figure 2: Main geometrical features of a helical baffle exchanger.*

It has been demonstrated that the highest thermal performance in this type of geometry is achieved when the baffle angle is 40° (Zhang et al., 2013). Sivarajan et al. (2013) carried out a 3D analysis and found that the heat transfer rates in helical baffle exchangers is higher than in conventional segmental baffle exchangers from 9 % to 23 %. The performance of multiple shell passes has been analysed in a numerical way by Chen et al. (2010). Also experimental studies to determine the improved performance of this type of units has been reported by Zhang et al. (2013). Amidst the limited information on the thermal performance of helical baffle exchangers is the correlation presented by Wang et al. (2010). The expression and the corresponding definition of variables involved are:

$$\mathrm{Nu_s} = 0.62\left(0.3 + \sqrt{\mathrm{Nu_{lam}^2} + \mathrm{Nu_{turb}^2}}\right) \tag{1}$$

$$\mathrm{Nu_{lam}} = 0.664\ \mathrm{Re_s^{0.5}}\ \mathrm{Pr_s^{0.33}} \tag{2}$$

$$\mathrm{Nu_{turb}} = \frac{0.037\ \mathrm{Re_s^{0.7} Pr_s}}{1+2.443\mathrm{Re_s^{-0.1}}(\mathrm{Pr_s^{0.67}}-1)} \tag{3}$$

$$\mathrm{Pr} = \frac{\mu \mathrm{C_P}}{\mathrm{k}} \tag{4}$$

$$\mathrm{Re_s} = \frac{\mathrm{u_s}\rho_\mathrm{s}\mathrm{D_{eq}}}{\mu_\mathrm{s}} \tag{5}$$

$$\mathrm{u_s} = \frac{\dot{m}}{\rho A_{sec}} \tag{6}$$

$$\mathrm{D_{eq}} = \frac{4\mathrm{p_t}^2 - \pi \mathrm{d_{out,t}}^2}{\pi \mathrm{d_{out,t}}} \tag{7}$$

$$A_{sec} = \frac{1}{2}B\left[D_{int,s} - D_1 + \frac{D_1 - \mathrm{d_{out,t}}}{\mathrm{p}_t}(\mathrm{p}_t - \mathrm{d_{out,t}})\right] \tag{8}$$

$$\mathrm{B} = 2\sqrt{2}\alpha \mathrm{D_{int,s}}\tan\beta \tag{9}$$

Where α is the overlap rate of the helical baffle. In this work α = 1. In the case of the tube side, an expression for the determination of the heat transfer rate is:

$$\mathrm{Nu_t} = \frac{(\xi/8)(\mathrm{Re_t}-1000)\mathrm{Pr_t}}{1+12.7\left(\mathrm{Pr_t^{2/3}}-1\right)\sqrt{\xi/8}} \cdot \left[1 + \left(\frac{\mathrm{d}}{\mathrm{L}}\right)^{2/3}\right] \cdot \left(\frac{\mathrm{Pr}}{\mathrm{Pr}_w}\right)^{0.11} \tag{10}$$

$$\xi = (1.82 \cdot \log \mathrm{Re_t} - 1.64)^{-2} \tag{11}$$

The pressure drop through the core of the exchanger is the fraction of the total pressure drop that is directly related to the rate of heat transfer. The expression to determine it on a tubular geometry is:

$$\Delta \mathrm{P_t} = \frac{1}{2}\xi \rho_t v^2{}_{t,noz} + \frac{1}{2}\rho_t v^2{}_t \left[\frac{f_t l_{tc}}{d_{int,t}}\frac{1}{(\phi_t)^r} + k_c + k_e + 4\right] N_p \tag{12}$$

$$\mathrm{f_t} = 0.4137\mathrm{Re_t^{-0.2585}} \tag{13}$$

Now, the pressure drop across the core of the exchanger can be determined from:

$$\Delta \mathrm{P_{t0}} = 2\mathrm{f_s \rho_s u_s}^2 \frac{\mathrm{l_{t0}}}{\mathrm{B}} \tag{14}$$

$$\mathrm{f_s} = 3.5\left\{\left[1.33\left(\frac{\mathrm{d_{out,t}}}{\mathrm{p}_t}\right)\right]^e\right\}\mathrm{Re_s^{-0.476}} \tag{15}$$

$$e = \frac{6.59}{\left[1 + \left(0.14\mathrm{Re_s^{0.52}}\right)\right]} \tag{16}$$

The above equations constitute the thermos-hydraulic model that is used for design. To complement the model, it is outstanding the determination of the correction factor of the logarithmic mean temperature difference. This parameter can be obtained from the expression:

$$\mathrm{F} = \frac{\mathrm{Ntu_{counter-current}}}{\mathrm{Ntu_{other\ arrangement}}} \tag{17}$$

The term Ntu (number of heat transfer units) for a heat exchanger can be obtained from the expressions that relate this parameter to: the thermal effectiveness (ε) and the heat capacity-mass flow rate ratio ($C=CP_{min}/CP_{max}$). Each of these parameters can be expressed as a function of the terminal temperatures of the exchanger. The thermal effectiveness is defined as the ratio of the temperature change that the process stream with the lowest heat capacity-mass flow rate experiences to the maximum temperature difference existing in the unit.

$$\varepsilon = \frac{\Delta \mathrm{T_{CPmin}}}{\mathrm{T_{in,hot\ fluid}} - \mathrm{T_{out,cold\ fluid}}} \tag{18}$$

For a counter-current flow arrangement, the thermal effectiveness (ε) is expressed as:

$$\varepsilon = \frac{1 - \mathrm{e}^{[\mathrm{NTU}(1-\mathrm{C})]}}{1 - \mathrm{Ce}^{[\mathrm{NTU}(1-\mathrm{C})]}} \tag{19}$$

The value of C can be obtained from:

$$\mathrm{C} = \frac{\Delta \mathrm{T_{CPmax}}}{\Delta \mathrm{T_{CPmin}}} \tag{20}$$

For the case shown in Figure 1, it can be seen that overall the exchanger performs in counter-current fashion; however, locally, it is clear that the units exhibits a combination of two flow arrangements, namely: parallel-counter flow. This combination results in a combined effect resulting in the overall performance of the unit. An expression to determine the overall performance of a complex arrangement of individual unit that make up an overall counter flow exchanger is:

$$\varepsilon_\mathrm{g} = \frac{1 - \prod_1^\mathrm{i}\left(\frac{1-\varepsilon_\mathrm{i}\mathrm{C}}{1-\varepsilon_\mathrm{i}}\right)}{\mathrm{C} - \prod_1^\mathrm{i}\left(\frac{1-\varepsilon_\mathrm{i}\mathrm{C}}{1-\varepsilon_\mathrm{i}}\right)} \tag{21}$$

Developing Eq(21) for the arrangement under discussion the resulting equation is:

$$\varepsilon_\mathrm{g} = \frac{1 - \left(\frac{1-\varepsilon_1\mathrm{C}}{1-\varepsilon_1}\right)\left(\frac{1-\varepsilon_2\mathrm{C}}{1-\varepsilon_2}\right)}{\mathrm{C} - \left(\frac{1-\varepsilon_1\mathrm{C}}{1-\varepsilon_1}\right)\left(\frac{1-\varepsilon_2\mathrm{C}}{1-\varepsilon_2}\right)} \tag{22}$$

Where $\varepsilon_g$ is the overall thermal effectiveness that is computed from Eq(18). The terms $\varepsilon_1$ and $\varepsilon_2$ are the thermal effectiveness for the parallel and counter flow units. From Eq(19), Ntu for a counter-current arrangement is:

$$\mathrm{Ntu_{cc}} = \left(\frac{1}{1-\mathrm{C}}\right)\ln\left(\frac{1-\mathrm{C}\varepsilon}{1-\varepsilon}\right) \tag{23}$$

For the case of parallel flow, Ntu is:

$$\mathrm{Ntu_p} = -\frac{\ln[1-\varepsilon(1+\mathrm{C})]}{1+\mathrm{C}} \tag{24}$$

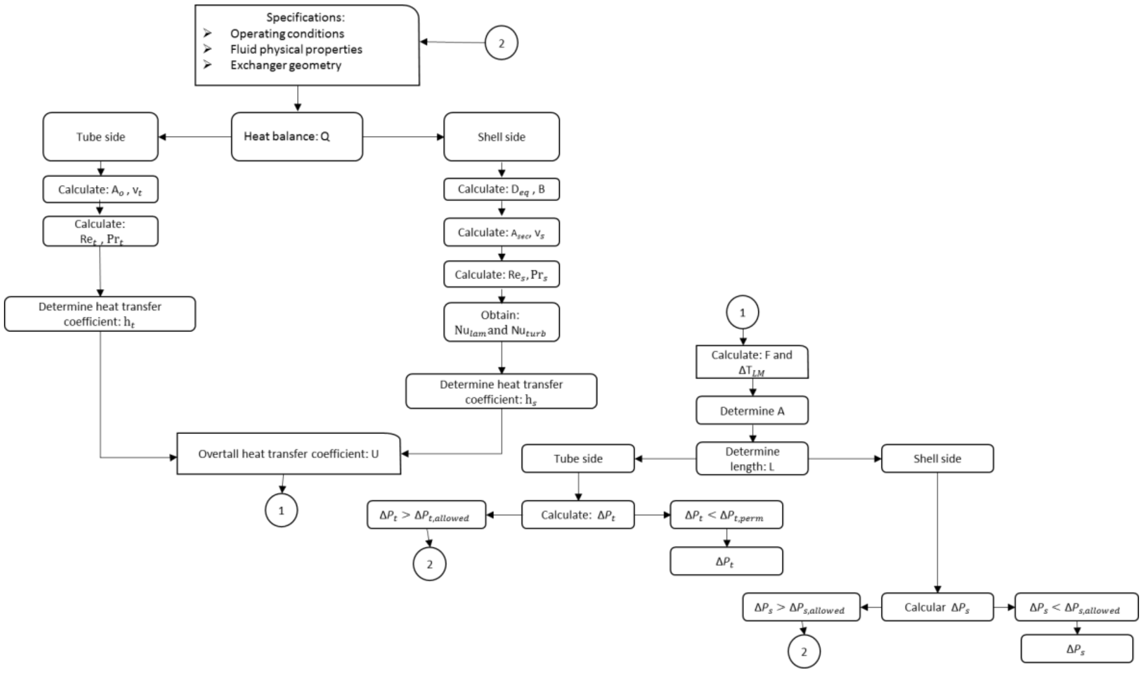


*Figure 3: Flow diagram of the design approach.*

Assuming that both units exhibit the same heat transfer coefficients and the same heat transfer area, then the Ntu of each unit can be equated:

$$-\frac{\ln[1-\varepsilon_1(1+C)]}{1+C} = \left(\frac{1}{1-C}\right)\ln\left(\frac{1-C\varepsilon_2}{1-\varepsilon_2}\right) \tag{25}$$

Solving for ε1 from Eq(25) results in:

$$\varepsilon_1 = \left(\frac{1}{1+C}\right)\left[1-\left(\frac{1-C\varepsilon_2}{1-\varepsilon_2}\right)^{\frac{1+C}{C-1}}\right] \tag{26}$$

Substitution of Eq(26) into Eq(22) results and an expression with $\varepsilon_2$ as a single variable. With the value of $\varepsilon_2$, $\varepsilon_1$ can be obtained from Eq(26) and in turn, $Ntu_{cc}$ and $Ntu_p$ from Eq(23) and Eq(24). The summation of these two terms given $Ntu_{other\text{-}arrangement}$ which along with $Ntu_{cc}$ are substituted in Eq(17) to give the value of F. The heat exchanger surface area is then calculated from the design equation:

$$A = \frac{Q}{U\,F\,\Delta T_{ln}} \tag{27}$$

A flow diagram that shown the design approach is in Figure 3.

## 2. Case study

A case study from the open literature is considered. The design using helical baffles is compares to the design using conventional segmental baffles. Table 1 shows the operating data and the physical properties for the case study. Table 2 shows the results.

*Table 1: Operating conditions for case study*

| | Kerosene | Crude oil |
|---|---|---|
| Mass flow rate (kg/h) | 19,867.24 | 67,584.91 |
| Inlet temperature (°C) | 198.9 | 37.78 |
| Outlet temperature (°C) | 93.3 | 76.67 |
| Allowable pressure drop (Pa) | 68,947.6 | 68,947.6 |
| Flow rate ($m^3$/h) | 24.4 | 79.1 |
| Fouling factor ($m^2$°C/W) | 0.00015 | 0.00015 |

*Table 2: Geometrical data for case study*

| | Shell side | Tube side |
|---|---|---|
| Inlet diameter (m) | 0.54 | 0.0205 |
| Outer diameter (m) | 0.57 | 0.0254 |
| Pitch (m) | | 0.0318 |
| Tube thickness | | 13 bwg |
| Tube arrangement | | Square 45° |

*Table 3: Design results for case study*

| | Segmental baffles | Helical baffles with same geometry | Helical baffles with changed geometry |
|---|---|---|---|
| Shell side pressure drop (Pa) | 68,947.5 | 3,893.1 | 29,350.3 |
| Tube side pressure drop (Pa) | 68,947.5 | 16025.56 | 49,264.5 |
| Heat duty (W) | 1,494,662.67 | 1,497,790.49 | 1,487,662.49 |
| Overall heat transfer coefficient ($W/m^2K$) | 317.98 | 182.31 | 524.48 |
| Tube length (m) | 4.877 | 10.26 | 4.6 |
| Heat transfer area ($m^2$) | 61.5 | 129.4 | 44.8 |
| Baffle angle | - | 15° | 15° |
| Number of baffles | 36 | 25 | 36 |
| Shell side heat transfer coefficient ($W/m^2K$) | 919.88 | 519.04 | 878.79 |
| Tube side heat transfer coefficient ($W/m^2K$) | 687.07 | 385.89 | 3,015.3 |
| No. shell passes | 1 | 2 | 2 |
| No. of tube passes | 4 | 1 | 2 |
| No. of tubes | 158 | 158 | 61 |

The use of helical baffles provides additional degrees of freedom that provide the designer with more options for improving the design. For instance in the case study above, suing the very same geometry as in the conventional segmental baffle exchanger, the helical baffle results in a larger unit with a pressure drop much lower than the original design. For a fair comparison, the design has to be based on a similar pressure drop use. If the shell side internal diameter of the helical baffle exchanger is reduced from 0.54 to 0.33 m and the number of passes on the tubes increased to two, the final design, although exhibiting still a lower pressure drop, results in a heat transfer area of 44.8 $m^2$. This means that with a 57 % reduction in pressure drop, the exchanger is 27 % smaller compared to conventional units, as shown in Table 3. This opens up the door for an optimization design strategy.

## 3. Conclusions

Helical baffle shell and tube heat exchangers offer improved performance for the same pressure drop compared to conventional segmental exchangers. This work presents a quick design methodology for the sizing of these types of units. Rigorous design requires the knowledge of accurate information on the thermal and friction performance. With the information available to date in the open literature, the approach presented in this work can readily be implemented to quickly and accurately carry out preliminary designs at the decision making stage in process design.

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