A Software for the Design of Shell and Tube Heat exchangers with Helical Baffles

Introduction

The concept of shell and tube heat exchangers with helical baffle (STHXHB) was developed for the first time in Czechoslovakia. The helical baffle heat exchanger (Aka HELIXCHANGER) is a good modification to the conventional, segmental baffle shell and tube heat exchangers. It provides the necessary characteristics to reduce flow dispersion and generate near plug flow condition. It also ensured a certain amounts of cross flow to the tubes to provide high heat transfer coefficient. The Shell side flow configuration offer a very high conversion of pressure drop to heat transfer. The STHXHB has improved shell side heat transfer coefficient, lesser pressure drop for given mass flow rates, reduced fouling in shell side, reduced bundle vibration. In reality, the STHXHB actually have discontinuous approximate helicoids. These are usually elliptical sector-shaped plates joined successively.

In this project, I have developed an Free and Open Source Software to aid in the design of Shell and Tube Heat exchangers with Helical Baffles.

Inputs to the program:

Shell side and Tube side Inlet temperature, outlet temperature, Specific Heat capacity, Fouling resistance, Prandtl Number, mass flow rate, allowable pressure drop, viscosity, thermal conductivity.

Modifiable parameters to the program are present in the top section of the program. This helps in in-depth customisation of the heat exchanger.

Design Process

The design process followed is based on the Bell-Delaware method used for design of conventional Shell and tube Heat exchangers with segmental baffles accounting for various correction factors that needs to be introduced when applying the method for STHXHB.

Step 1: To find the heat duty of each side

$$Q_s = M_s \times c_{ps} \times \left| t_{s,in} - t_{s,out} \right|$$

$$Q_t = M_t \times c_{pt} \times \left| t_{t,in} - t_{t,out} \right|$$

Step 2: Determine tube layout

There are four standard tube layouts

- 30^o layout - triangular - Fig.1(a)
- 45^o layout - rotated square - Fig.1(d)
- 60^o layout - rotated triangular - Fig.1(b)
- 90^o layout - square - Fig.1(c)

The 30^o, 45^o and 60^o patterns are called staggered whereas the 90^o is called in-line. Note that these angles are defined with respect to the direction of the flow.

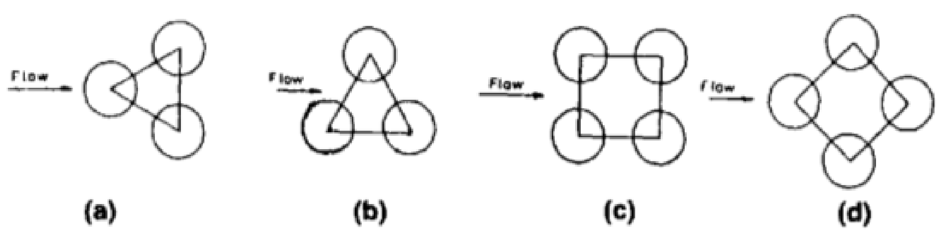


Figure 1

While determining tube layout, we try to get as many tubes as possible within the shell to achieve maximum heat transfer area. In addition to heat transfer area, tube layout is of practical significance in case of fouling fluids. Fouling fluids require require regular maintenance and cleaning. This is easily possible in case of the 90^o arrangement. access to the tubes for cleaning as required by process conditions. Four standard types of tube. For identical tube pitch and flow rates, the tube layouts in decreasing order of shell-side heat-transfer coefficient and pressure drop are 30^o, 45^o, 60^o, and 90^o.

Thus the 90^o layout has the least heat-transfer coefficient and pressure drop. Triangular and rotated triangular layouts (30^o and 60^o) provides a compact arrangement, better shell-side heat-transfer coefficients at the same time, it offers difficulty to cleaning as well as has a higher pressure drop. Only chemical cleaning or water jet cleaning is possible with triangular and rotated triangular layout. The 45^o layout is preferred for single-phase laminar flow or fouling service, and for condensing fluid on the shell side. If the pressure drop is a constraint on the shell side, then the 90^o layout is used for turbulent flow, since in turbulent flow the 90^o has superior heat-transfer rate and less pressure drop. For reboilers, the 90^o offers better stability since it provides lanes for vapours to escape.

One assumption which I had made is that, I assumed all the patterns to form regular shapes. i.e either squares or equilateral polygons.

Step 3: To gather all thermophysical properties of the fluids. Temperature dependant properties are found at the average of inlet and outlet temperature

Step 4: Provisional heat transfer area, A_o is calculated by assuming a value of heat transfer coefficient K_o. It is calculated using the following equation

$$A_o = \frac{Q}{K_o \times \Delta t_{lm}}$$

Where Δt_{lm} is the logarithmic mean temperature difference

Step 5: One of the TEMA recommended tube lengths (l) -(2438,3048,3657,4978,6096 mm) can used for the design. Default value of the tube length in the program is 2438 mm, however user can input the length of the tube as per his choice. Long tube lengths generally tend to be economical to manufacture, however they are tough to install, not economical to maintain and require more space.

For tube diameter selections also, TEMA recommended tubes with outer diameters(d_o) - (6.35,9.53,12.7,15.88,19.05,22.23,25.40,31.75,38.10,50.80mm). Default value used in the program is 15.88 mm. 6.35mm tubes can be used for clean fluids. However, for fouling fluids which require mechanical cleaning, minimum diameter of 19.05 mm is to be used.

Number of tubes required can now be determined from

$$N_t = \frac{A_o}{\pi d_o l}$$

From the number of tubes, N_t and using schlunder's relation

$$N_t = \frac{0.78\, D_i^2}{C_1 \times P_t^2}$$

Where P_t is the tube pitch. Conventionally, P_t is taken to be 2.25times the tube's outer diameter.

With the help of this relation, we can now find the tube bundle diameter, and using TEMA's standard clearances , we determine the shell inner diameter.

Step 6: Helical angle is chosen by default as 20^o, and the overlap ratio is taken as 0.8, but the user has an option to enter them in the GUI.

overlap ratio is the dimensionless radius of the contacting point of two successive helical baffles as shown in figure below.

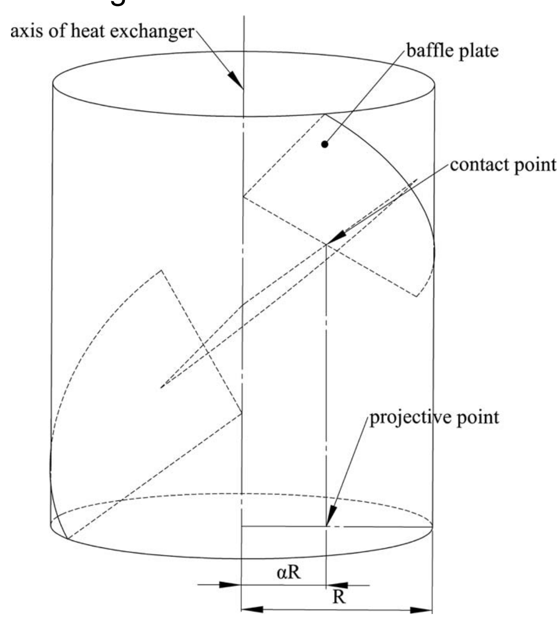


Fig. 2 Definition of overlap ratio

Step 7: Determination of Reynold's Number, Nusselt's Number and the correction factors for heat transfer and Pressure drop

Shell Side:

$$Nu_s = 0.62 \times (0.3 + \sqrt{Nu_{lam}^2 + Nu_{turb}^2}) \times Y_2 \times Y_3 \times Y_4 \times Y_7 \times Y_8 \times Y_9 \times Y_{10}$$

Where, Y_i are the correction factors

$$Nu_{lam} = 0.664\, Re^{0.5} Pr^{0.33}$$
$$Nu_{turb} = 0.037\, Re^{0.7} Pr\,(1 + 2.433\, Re^{-0.1}(Pr^{0.67} - 1))$$

Heat Transfer coefficient is given by,

$$h_s = \frac{Nu_s \times k_s}{l}$$

Where, $l = \frac{\pi d_o}{2}$

Here, Y_2 accounts for the thermal-physics properties effects
Y_3 accounts for the scale-up from a single tube row to a bundle of tubes
Y_4 accounts for the adverse temperature gradient
Y_7 accounts for the bundle-shell bypass streams
Y_8 accounts for the baffle spacing in inlet and outlet sections
Y_9 accounts for the change in the cross-flow characteristics in heat exchanger
Y_{10} accounts for the turbulent enhancement.

Pressure drop across the tube bundle per unit cycle without bypass flow is given by

$$\Delta P^* = 2 f n_r \rho u^2 Z_2 Z_6 Z_7$$

Where,
f is darcy's friction factor
n_r is no. of tube rows on centre streamline
Z_i are correction factors
Z_2 accounts for thermo-physical properties effects
Z_3 accounts for bundle-shell bypass streams
Z_5 accounts for baffle spacing inlet and outlet zones
Z_6 accounts for change in cross-flow characterstics
Z_7 accounts for turbulent enhancement

$$n_r = \frac{\sqrt{3}D_{tb}/2P_t + 1}{2}$$ for 30° and 60° layouts

$$n_r = \frac{D_{tb}}{\sqrt{2}P_t} + 1$$ for 45° and 90° layouts

D_{tb} is the diameter of the tube bundle

Pressure drop across the bundle with bypass flow is given by

$$\Delta P_1 = \Delta P^* l_{to} Z_3 / B$$

Pressure drop across inlet and outlet zones are calculated using

$$\Delta P_2 = \Delta P^* Z_5$$

Pressure drop across inlet and outlet nozzles are calculated using

$$\Delta P_3 = \xi \times 0.5 \times \rho v_{nozzle}^2$$

ξ is pressure drop coefficient, ξ is either 1.5 or 2. In this code it is taken as 2 for a conservative design

Total pressure drop = $\Delta P_1 + \Delta P_2 + \Delta P_3$

$Y_2 = \left(\frac{\eta_s}{\eta_{s,w}}\right)^{0.14}$ $\quad Z_2 = \left(\frac{\eta_s}{\eta_{s,w}}\right)^{-0.14}$

This effect is very negligible, hence Y_2 , Z_2 as well as Y_4 is assumed to be 1

$Y_3 = 1 + \frac{0.7 \times (b/a - 0.3)}{\varepsilon^{1.5}(b/q + 0.7)^2}$

where,

a is the ratio of distance between the tubes in the normal to flow direction to tube pitch

b is the ratio of distance between the tubes in the flow direction to tube pitch

$$\varepsilon = 1 - \frac{\pi}{4a}$$

Y_7 = $\exp[-1.343x(1-(2y)^{0.338})]$
Z_3 = $\exp[-3.56x(1-(2y)^{0.363})]$

Where,

$x = S_{ss}/S_{2z}$

$$S_{ss} = 0.5(B - S_p/\cos\beta)[D_1 - D_s - S_{tt}]$$

$$S_{2z} = 0.5(B - S_p/\cos\beta)\left[D_i - D_1 + \frac{D_1 - d_o}{t_t}(t_t - d_o)\right]$$

$y = P_t \times n_{pt}/D_i$

P_t is tube pitch
n_{pt} is no. of sealing strips . Note: this is assumed in a constant proportion to the no. of tube given by the variable “ssr” in the code

Y_8 = $1.079y^{0.0487} - 0.445y^{-0.301}x^{1.2}$
Z_5 = $(-0.0172 + 0.0899y)x^{-1.2}$

Where,

$x = (l_{tc} - l_{to})/l_{tc}$

l_{tc} is effective length of tube
l_{to} is baffled length of tube
l_{to} is approximated to be equal to the baffle spacing and the baffle spacing is taken to be one-fifth of the shell inner diameter as per TEMA recommendations

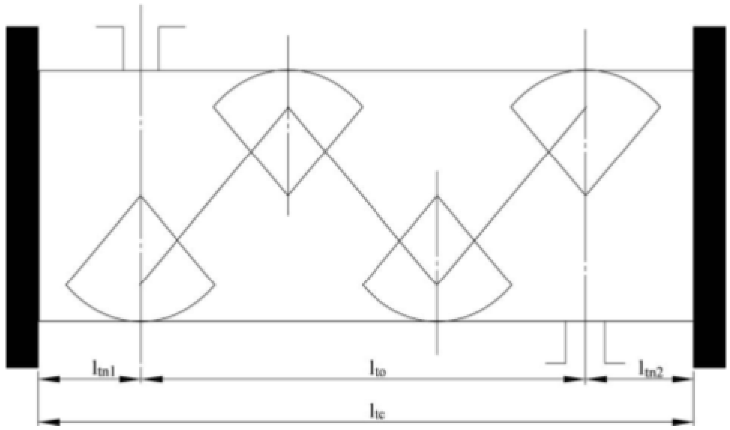


Fig. 3 parameters for calculation of Y_3

$y = B/D_i$

$B = \alpha n \cdot D_1 \sin\frac{\pi}{n} \cdot \tan\beta, \quad n \geq 2, \quad 0 < \alpha \leq 1$

B is the helical pitch of the baffles
n is the no. of baffles
α is overlap ratio
β is the helix angle

Parameters Y_9 , Y_{10} , Z_6 , Z_7 depend only upon the helix angle

$Y_9 = 0.977 + 0.00455x - 0.0001821x^2$, if $18^o < x < 45^o$
$Y_9 = 1$, otherwise

$Y_{10} = -56.39 + 8.28x - 0.46x^2 + 0.012x^3 - (1.64 \times 10^{-4})x^4 + (8.19 \times 10^{-7})x^5$, if $25^o<x<45^o$
$Y_{10} = 1$, otherwise

$Z_6 = 0.289 - 0.000506x - 0.0000453x^2$

$Z_7 = -5.411 + 0.379x - 0.00402x^2$, if $22^o<x<45^o$
$Z_7 = 1$, otherwise

here x represents the helix angle β in degrees

$$Re_s = \frac{G_s d_e}{\mu}$$

G_s - shell-side mass velocity

$$G_s = \frac{m}{A_s}$$

m_s - mass flow rate in shell side

$$A_s = \frac{(P_t - d_o) \times D_i \times l_B}{P_t}$$

d_e - Hydraulic diameter. It varies depending on the tube layout and is calculated with different formulas

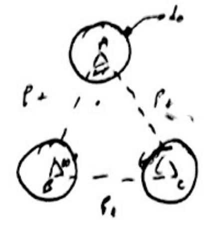
Determination of Hydrolic diameter

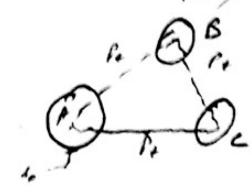
For Triangular layout (i.e.) (30°) (or) Rotated Triangular layout (60°)

Consider the ΔABC,

$AB = BC = CA = P_t$

Since we consider only regular layouts





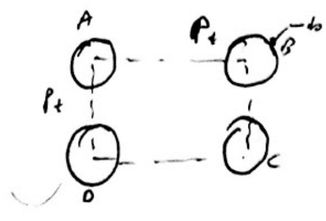
hydraulic Diameter, $D_h = \dfrac{4A}{P} = \dfrac{4\times\left[\frac{\sqrt{3}}{4}P_t^2 - \left(3\times\frac{60°}{360°}\times\frac{\pi d_o^2}{4}\right)\right]}{3\times\frac{60°}{360°}\times\pi d_o}$

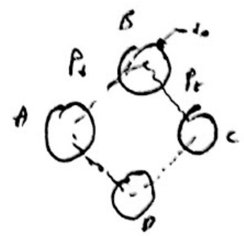
$$\boxed{D_h = \dfrac{4\left[\frac{\sqrt{3}}{4}P_t^2 - \frac{\pi d_o^2}{8}\right]}{\frac{\pi d_o}{2}}}$$

For Square layout (i.e.) 90° or rotated Square layout (45°)

Consider ABCD

$AB = BC = CD = DA = P_t$.





$D_h = \dfrac{4A}{P} = \dfrac{4\left[P_t^2 - \frac{\pi d_o^2}{4}\times\left(\frac{4\times 90°}{360°}\right)\right]}{\frac{4\times 90°}{360°}\times\pi d_o}$

$$\boxed{D_h = \dfrac{4\left[P_t^2 - \frac{\pi d_o^2}{4}\right]}{\pi d_o}}$$

Tube side:

Tube side nusselt's number is calculated with the help of mill's correlation

$$Nu = \frac{(f/8)(\mathrm{Re}-1{,}000)\mathrm{Pr}}{1+12.7(f/8)^{1/2}(\mathrm{Pr}^{2/3}-1)}$$

Darcy's friction factor f, is calculated with help of petukhov relation

$$f = \frac{1}{[0.790\ln(\mathrm{Re})-1.64]^2}$$

It is assumed in this code that the heat exchanger is a single pass heat exchanger with no pressure drop at inlet and exit nozzle of the tubes. Therefore pressure drop expression simplifies to

$$\Delta P = f\frac{L}{d}\frac{1}{2}\rho v^2$$

Step 8: Heat transfer coefficient is calculated with the help of nusselt's number of both the sides and the thermal conductivity of the tubes

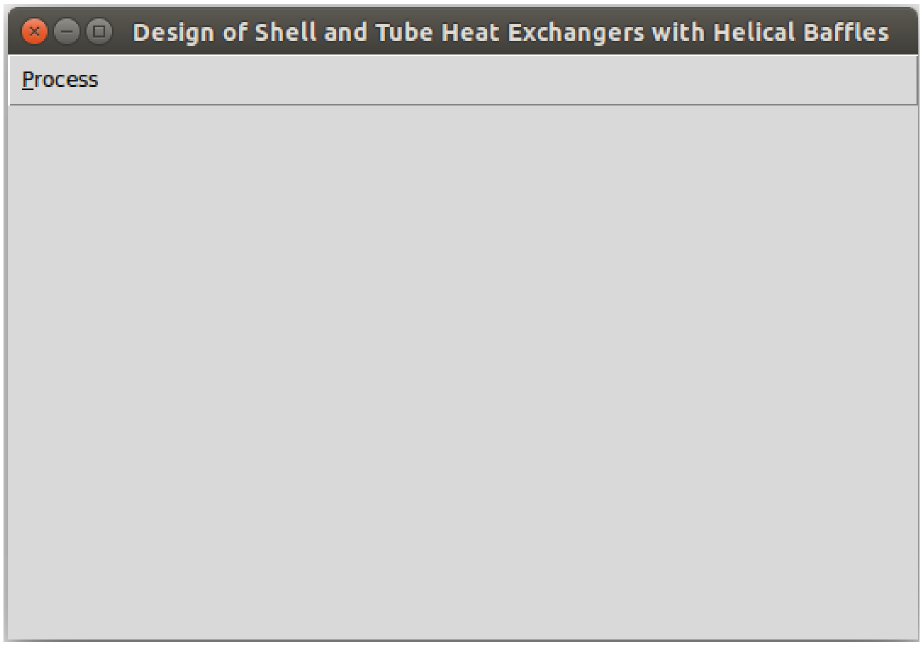
Average overall heat transfer coefficient, K_{o2} is calculated as

$$\frac{1}{k_{o2}} = \frac{1}{h_t}\frac{d_o}{d_i} + \frac{d_o}{2k_w}ln(\frac{d_o}{d_i}) + \frac{1}{h_s}$$

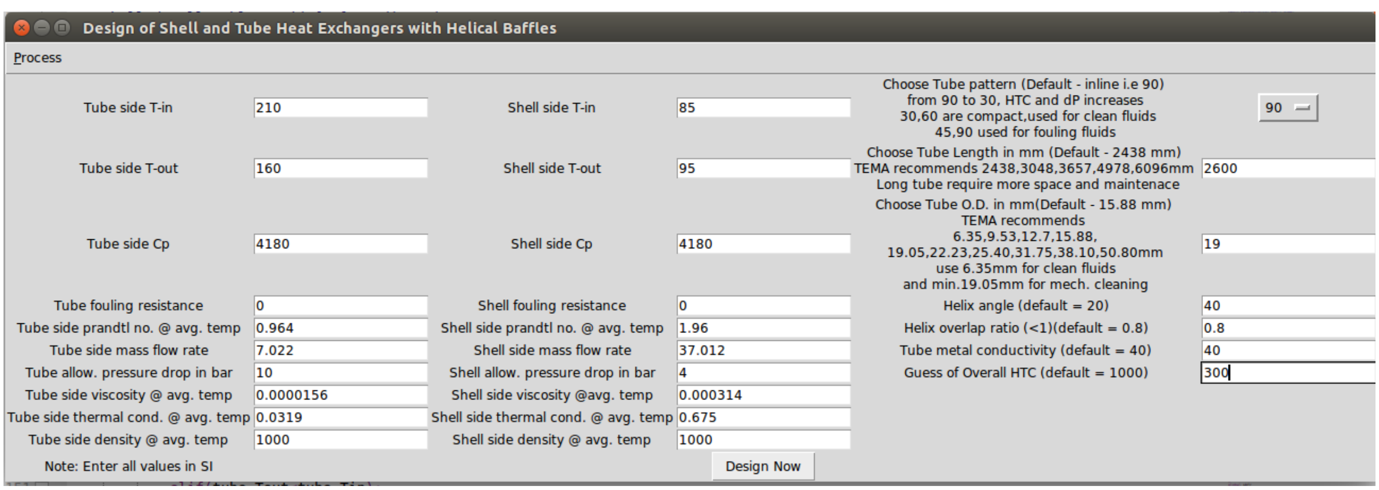
Step 9: Steps 4-8 is iterated with new assumed value of K_o till K_{o2} is greater than K_o with 15% of K_o

Program interface:

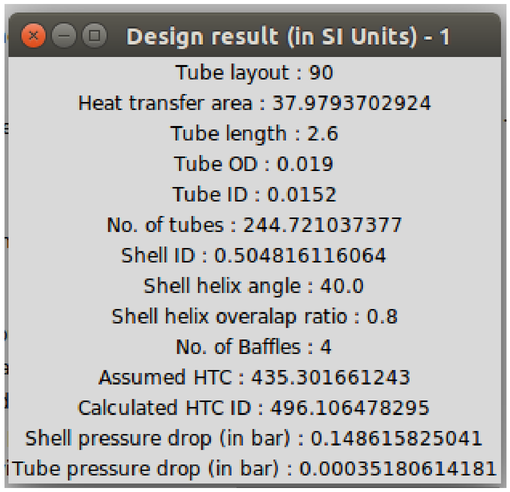
1. Open the program, select the “Process” tab, and select “Sizing”



2. Now enter the required parameters and click “Design Now” . Enter all values in SI units, unless otherwise mentioned.



3. Design results are shown as below



Validation of results : A shell and tube heat exchanger with helical baffles, for a similar purpose was designed by Jian et. Al[5] which yielded similar results.

References:

1. TEMA Guidelines
2. “Heat Exchanger Design Handbook” by Schlunder (1983 edn)
3. “Process Heat Transfer” By. S.K.Das
4. “Helical Baffles in Shell and Tube Heat Exchangers, Part I: Experimental Verification” by Kral, D.,Stehlik, P., Van Der Ploey, H.J. and Master, B.I.
5. “A Design and rating method for shell and tube heat exchangers with helical baffles” by Jian-Fei Zhang, Ya-Ling He, Wen-Quan Tao
6. “Heat Exchanger Design Handbook” by Kuppan. T (2000 edn)
7. “Shell-side characteristics of Shell and Tube Heat Exchangers”, by Tinker T