Software Evaluation Via a Study of Deviations in Results of Manual and Computer-Based Step-Wise Method Calculations for Shell and Tube Heat Exchangers

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Abstract: In this study manual calculation for both shell and tube, sides of a single-phase shell and tube heat exchanger are conducted. The calculations are made using Aspen B-JAC software, results of step-wise method computations are compared with manual calculations. Results show that although more accurate methods like step-wise method include many details in calculations, difference between the yielded results and experimental-based algorithms like Bell-Delaware's method is acceptable.

Keywords: Shell and tube; heat exchanger; step-wise method; manual calculation.

1. Introduction

The possibility of applying shell and tube heat exchangers in a wide range of heat transfer surface, temperature, mass flow rate, and pressure has made them the most common heat exchanger type in industry; as we can see their applications as evaporators, condensers, process heaters and coolers, multiphase heat exchangers, etc.

The basic design and fabrication of shell and tube heat exchanger were introduced in early 1900s to fill the needs in high heat transfer surface, and high-pressure operating condition in power plants [1]. Grimmison conducted a systematic study of tube banks in 1937 [2]. Also the study of heat transfer and pressure drop calculations of baffled heat exchangers were introduced in 1940s [3, 4]. The first step to develop a method for calculations of shell and tube heat exchangers was taken by Kern in 1950 [5]. Meanwhile, Donohue developed a method for the calculation of shell and tube heat exchangers with its data limited to almost small heat exchangers [6]. Tinker developed a method, in which many parameters were included so that they made it complex and almost impossible for manual calculations [7]. However, Tinker's method was more accurate than kern's method.

Bell-Delaware's method was issued after 16 years of experimentation on shell side flow in lab. Nowadays, this method is commonly used in manual calculations [8]. This method does not have the complexities of Tinker's method; it is simple and reliable enough to be used in engineering applications. Since then many software have been developed based on Bell-Delaware method [9].

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117

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Compact formulation of the Bell-Delaware method for heat exchanger design and optimization was reported via a mathematical approach [10]. Moreover, new methods such as genetic algorithms utilized for optimizing the design of shell and tube heat exchangers in literatures [11-12].

One more accurate step is using step-wise method to discrete heat transfer length into some definite segments, where variation of some dependent parameters can be included. Aspen B-JAC is a newly developed software operating based on step-wise method. In this paper, software method, which breaks calculations in to the some separate steps, is called Step-wise method. Although development of numerical fluid dynamics and heat transfer methods has been a new more accurate approach to analysis and design of heat exchangers, numerical methods are not as simple and user-friendly as manual methods and take longer time as well as need a good knowledge of computational fluid dynamic (CFD).

The case study in this paper is dedicated to a shell and tube heat exchanger, which was installed as a feed water heater in Banias Power plant in Syria. In this study, both manual and computer based calculations for a typical TEMA (Tubular Exchanger Manufacturers Association) class R, BEM type are carried out. Class R attends to "*unfired shell and tube heat exchangers for the generally severe requirements of petroleum and related processing applications*". In addition, BEM addresses front-end type, shell type and rear end type of exchangers and in this case "*Bonnet-One pass shell-fixed tube sheet*" [13].

The aim of this research is to make more assurance that well-known B-JAC software is work properly and how much error emerges when manual calculation method is used. More ever the variation of some important heat transfer and hydrodynamic parameters in shell and tube heat exchanger is investigated.

2. Description of manual calculation

2.1. Shell side flow

This method uses ideal tube banks j_i and f_i then corrects them to account for the effects resulted from various leakage and bypass streams [1]. The j_i and f_i are defined as follows:

$$j_i \frac{\alpha_i}{c_p m} (Pr)^{\frac{2}{3}} (\Phi)^{-r} = f(Re, tube \, layot)$$
(1)

$$f_i \frac{\Delta p_i \rho}{2(m^{\prime})^2 N_c} (\Phi)^r = f(Re, tube \, layot)$$
⁽²⁾

where, Φ is the viscosity correction factor. The Reynolds number in the above equations is based on the minimum cross-sectional flow area at shell side stream.

According to the Bell-Delaware's method leakage and bypass stream effects must be included in actual heat transfer and pressure drop coefficients as follows:

$$\alpha_s = \alpha_i \left(J_c J_l J_b J_s J_r \right) \tag{3}$$

The shell side total pressure drop is considered as the summation of the pressure drops for the inlet and exit sections, the mid sections, and window sections. Pressure drop in mid section is given by

$$\Delta p_c = \Delta p_{bi} \left(N_b - 1 \right) R_l R_b \tag{4}$$

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where

$$\Delta p_{bi} = 2(10^{-3}) f_i N_{tcc} \frac{(m_s)^2}{\rho_s} (\Phi_s)^{-r}$$
(5)

also

$$\Delta p_{w} = \Delta p_{wi} N_{b} R_{l} \tag{6}$$

Where pressure drop in the windows is affected by leakage only. Bell-Delaware's method offers two other different correlations, one for turbulent and the other for laminar flow respectively as follows:

$$\Delta p_{w} = N_{b} \left[\left(2 + 0.6 N_{tcc} \right) \frac{(m_{w})^{2}}{2\rho_{s}} \left(10^{-3} \right) \right] R_{l}$$
(7)

$$\Delta p_{w} = N_{b} \left[26 \frac{\binom{\bullet}{m_{w}} \eta_{s}}{\rho_{s}} \left\{ \frac{N_{tcw}}{L_{tp} - D_{t}} + \frac{L_{bc}}{D^{2}_{w}} \right\} + \dots \left\{ 2(10)^{-3} \right) \frac{\binom{\bullet}{m_{w}}^{2}}{2\rho_{s}} \right\} \right] R_{l}$$
(8)

where

$$\dot{m}_{w} = \frac{\dot{M}_{s}}{\sqrt{s_{m}s_{w}}} \tag{9}$$

Pressure drop in inlet and outlet zones, Δp_e , becomes

$$\Delta p_e = \Delta p_{bi} \left(1 + \frac{N_{tcw}}{N_{tcc}}\right) R_b R_s \tag{10}$$

2.2. Tube side flow

In shell-and-tube heat exchanger, calculation of heat transfer coefficient of tube side flow is based on experimental correlations. For turbulent and fully developed flow of liquids, Sieder and Tate give a correlation considering heating and cooling features as follows

$$Nu = 0.022 \operatorname{Re}^{0.8} \operatorname{Pr}^{0.42} \left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$
(11)

where μ_b and μ_w are viscosity in bulk and tube wall temperature, respectively.

Tube side pressure drop appears in three zones as following:

2.2.1. Nozzles

Pressure drop at inlet nozzles results from two factors, sudden expansion from nozzle area to downstream and irreversibility of velocity gradient. Therefore,

$$\frac{p_n - p_D}{\rho} = \left[k_e + \left(\frac{s_n}{s_D}\right)^2 - I\right] \frac{v_n^2}{2}$$
(12)

where, S_n and S_D are nozzle and downstream cross sections. Irreversibility coefficient, k_e can be derived from "Borda-Carnot correlation"

$$k_e = \left(1 - \frac{s_n}{s_D}\right)^2; Re > 4000 \tag{13}$$

Outlet nozzles cause sudden contraction of flow in tube path from upstream cross section to nozzle area. Therefore, static head gradient can be calculated from,

$$\frac{p_U - p_n}{\rho} = \left[I - \left(\frac{s_n}{s_U}\right)^2 + \lambda k_c \right] \frac{\nu^2_n}{2}$$
(14)

where K_c and λ are given from [9].

2.2.2. Entrance and exit of tubes

Pressure drop in these sections are resulted from sudden flow contraction and expansion. Thus, the basic correlation for entrance and contraction are equations (14) and (12), respectively. In addition, for triangular tube pitch following correlation should be considered:

$$\frac{s_D}{s_U} = 0.907 \left(\frac{d}{\lambda_{tp}}\right)^2 \tag{15}$$

2.2.3. Tube core

This term of pressure drop is because of fluid passing through the tubes. Darcey Weisbach gives a well-known equation for frictional pressure drop

$$\Delta p = f \cdot \left(\frac{l}{d}\right) \cdot \left(\frac{\rho v^2}{2}\right) \tag{16}$$

where, f is the friction factor and is defined for fully developed and turbulent flow as follows:

$$f_i = 0.046 \, Re^{-0.2}; \, 3 \times 10^4 < Re < 10^6 \tag{17}$$

In above correlation, the effect of tube wall temperature considered as

$$\Delta p = f \cdot \left(\frac{l}{d}\right) \cdot \left(\frac{\rho v^2}{2\Phi}\right) \tag{18}$$

where

$$\Phi = (\frac{\mu_b}{\mu_w})^{0.14}$$

It should be mentioned that in all correlations physical properties are considered in bulk temperature unless they are explained.

3. Steps of calculations

Shell and tube heat exchanger characteristics required for thermal rating are given in Table 1.

Step 1.

Initial calculations: from Table 1. mass flow rate, specific heat, and fluid inlet and outlet temperatures are available. Therefore, heat transfer rate is determined as follows

 $Q = 140.7 \times 4389 \times (211.63 - 207.24) = 2710658w$

Applying mean temperature difference (MTD) leads to

GTTD =263.37 LTTD =43.71

According to tube pattern, shell diameter, and tube pitch, number of tubes are calculated as follows [1]:

Tube specification and geometrical parameters		Temperature and physical properties	
Symbol	Value	Symbol	Value
Ds	725mm	Tso	250.90
Dt	16 mm	Tti	207.24
Ltw	2mm	Tto	211.63
Dti	12mm	Ms	5.3531kg/s
$\lambda_{_{tw}}$	39.8 W/m.oc	$ ho_s$	7.355 kg/m3
Ltp	21mm	λ_s	0.0545 w/mk
$ heta_{tp}$	300	(Cp)s	2381 kg/kgk
Lta	1100mm	η_{s}	0.023 * 10-3mpa.s
Bc	24%	(Rf)o	0.00009 m2k/w
Lbc	245mm	Mt	140.7 kg/s
Lbi	162.12mm	ρ_t	857.485 kg/m3
Lbo	162.12mm	λ_t	0.635 w/mk
Nss	4	$\left(C_{p}\right)_{t}$	4389 KJ/kg
$\left(R_{f}\right)_{i}$	0.00009 m2k/w	$\left(R_{f}\right)_{o}$	0.000009 m2k/w
Kc	0.5	λ	1

Table 1. Characteristics of shell and tube heat exchanger

This information is based on the referenced heat exchanger and the thermodynamics data are extracted from the software. These data directly were used for manual calculation.

$$N_{tt} = \frac{0.78(640.3)^2}{0.688(21)^2} \approx 837$$

Step 2.

Heat transfer on tube side: according to Table1. Reynolds number becomes

M. H. Karimi Dona and M. R. Jalalirad

$$(Re)_{t} = \frac{857.48 \times 1.76 \times 0.012}{(0.1395 \times 10^{-3})} = 129821.3$$

A trial and error method needs to be applied to determine the wall temperature. Given the wall temperature, heat transfer coefficient is determined using equation 11 as follows:

$$Nu = 0.022(129821.3)^{0.8}(0.94)^{0.42}(\frac{0.14}{0.13})^{0.14} = 266.61$$
$$\alpha_t = \frac{266.61 \times 0.653}{0.012} = 14508.22 \text{ m/m}^2$$

. . .

Step 3.

Pressure drop on tube side: determining the Reynolds number in inlet nozzle and using equation (12) gives

$$p_n - p_D = [0.55 - 1 + (0.255)^2](860.04)(\frac{1.26^2}{2}) = -262.64 \, pa$$

It is evident that the pressure drop in inlet nozzle must be negative. Pressure drop for the outlet nozzle using equation (14) becomes

$$p_U - p_n = \frac{854.93 \times 1.27^2}{2} \times [1 + (1 \times 0.5) - 0.255^2] = 989.3 \, pa$$

Also, using equation (12) pressure drop at the entrance and exit of tubes becomes, respectively

$$p_{U} - p_{D} = (0.33 + 1 - .5265^{2}) \times 860.44 \times (\frac{1.76^{2}}{2}) 1403.0 \, pa$$
$$p_{U} - p_{D} = \frac{854.93 \times 1.76^{2}}{2} \times (0.2242 + 0.5265^{2} - 1) - 660.2 \, pa$$

Finally, pressure drop in tube core is calculated from equation (17) and [14] as follows:

$$f = 0.046(129821.38)^{-0.2} = 4.4 \times 10^{-3}$$
$$\Delta p = \left(\frac{4 \times (4.4 \times 10^{-3})(857.48) \times 1.76^2}{2 \times 0.012}\right) = 1594.5 \, pa$$

Then total pressure drop in tube path becomes

$$\Delta p_{tot} = 1594.5 + 1403.0 + 989.3 - 660.2 - 262.64 = 3064 \, pa$$

Step 4.

Heat transfer on shell side: the process of determination of heat transfer coefficient for shell side fluid includes two main points; first, calculating of ideal tube bank heat transfer coefficient, α_i and second, calculation of correction factors according to geometrical considerations. α_i is given by equation (2) as follows:

$$\alpha_i = 0.0045(1.00)^{-\frac{2}{3}}(2381)(98.88) = 1059 \frac{w}{m^2}$$

where j_i and correction factors are known from [14]. Since the flow regime is turbulent, adverse temperature gradient has no effect in heat transfer coefficient.

Heat transfer correction factors are given from [1] as follows:

 $J_{c} = 1.6$ $J_{l} = 0.65$ $J_{b} = 0.9$ $J_{s} = 1.1$ $J_{tot} = 1.6 \times 0.65 \times 0.9 \times 1.1 = 0.686$ $\alpha_{s} = 1059 \times 0.686 = 726.47$

Step 5.

Pressure drop on shell side: like the previous step, geometrical consideration correction factors are known from [1] as follows,

 $R_{l} = 0.421$ $R_{b} = 0.74$ $R_{s} = 4.2$

Friction factor, f_i is determined from [1] as well. Hence, pressure drops in each three zones can be calculated as follows:

$$\Delta p_{bi} = \frac{2(10^{-3})(0.096)(20.35)(98 - 88)^2}{7.33} = 5.2 \, kpa$$

$$\Delta p_c = (5.2)(3 - 1)(0.74)(0.42) = 3.23 \, kpa$$

$$\Delta p_w = 3 \times \{(2 + 0.6 \times 5.79)(\frac{101.13^2}{2 \times 7.33}) \times 10^{-3}\} \times 0.42 = 4.81 \, kpa$$

$$\Delta p_e = (5.2)(1 + \frac{5.79}{20.35})(0.74)(4.2) = 20.76 \, kpa$$

Furthermore, the net pressure drop in inlet and outlet nozzles on shell path is given by

$$\Delta p_{n,s} = (1.85)(7.33)(\frac{14.41^2}{2}) = 1.41 \, kpa$$

Finally, total pressure drop on shell side becomes

 $\Delta p_{tot} = 3.23 + 4.81 + 20.76 + 1.41 = 30.21 kpa$

In order to compare the manual results with step-wise method results, Aspen B-JAC software is run for the same inputs as the manual calculations. The results are compared in the Figure 1 to Figure 4.

4. Results and discussion

Herein, calculations results for manual and step-wise method are illustrated in Figure 1 to Figure 4. In continue, for results comparison, maximum differences between two methods are considered according to following formula:

$$\frac{Max\{(Step - wise) - (Manual)\}}{Step - wise} \times 100$$
(19)



Figure 1. Tube side heat transfer coefficient



Figure 2. Shell side heat transfer coefficient

As shown in Figure 1, that the tube side heat transfer coefficient increases along the flow direction. Since the sensitivity of dynamic viscosity to temperature for liquid is higher than that of the liquid density to temperature, Reynolds number increases along the tube side heated flow direction, and causes an increase for heat transfer coefficient. This figure also shows that there is only a 4% difference between step-wise and manual calculations.

Figure 2. illustrates the shell side heat transfer coefficient distribution along the flow direction. Since the shell side flow is being cooled, a continuous decrease in heat transfer coefficient occurs along the flow direction. As it can be seen, there is a 14% difference between step-wise and manual calculations. The reason for differences between shell side and tube side heat transfer coefficients in two methods is clear, manual calculation method assumes that heat transfer coefficient is constant along the heat exchanger, while step-wise method is equipped to calculate this parameter for several times with lots of details. However, its results are closer to reality.







Figure 3. shows shell side pressure drop given both by step-wise method and by manual method. There is only a 1% difference between results. In this case manual calculation has the best accuracy.

In Figure 4 distribution of tube side pressure drop is illustrated. As it can be seen almost all of the total pressure drop occurs in inlet and outlet heads. There is a 27% difference between step-wise and manual calculations. The reason for this difference may be addressed to complicated flow regime at entrance of tubes. Really, this research cannot declare which method is near to fact.

5. Conclusions

In this paper distribution of heat transfer coefficient and pressure drop in a shell and tube heat exchanger, is calculated manually and computationally. Results show that although step-wise method employs lots of details such as variable physical properties in terms of temperature, the average difference between the calculated heat transfer coefficients is 9% and the average difference between manual and computer-based calculated pressure drop is less than 14%.

Considering the benefits of manual calculations such as their simplicity, makes them one of the best criteria or reference for software evaluation for the shell and tube heat exchangers calculations in most common engineering applications.

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