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Computation of temperature field by cell method and comparing with commercial software

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Computation of temperature field by cell method and comparing with commercial software

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Abstract. This paper deals with the temperature field of the shell and tube heat exchanger with segmental baffles. Two different types of shell and tube heat exchangers were analysed by a numerical model for thermal-hydraulic rating called the cell method. The cell method is a numerical computational model for calculating of temperature field of a shell and tube heat exchanger with segmental baffles. A huge benefit of the cell method is especially its simplicity. The computation of temperature field by the cell method is very fast and without the necessity of powerful hardware accessories. For analyses, two different types of shell and tube heat exchangers with segmental baffles were used. First, a co-current flow heat exchanger with a floating head and second a counter-current flow heat exchanger with a fixed tubesheet. Both analysed heat exchangers are horizontal, have one tube and one shell pass and segmental baffles. The results from cell method were compared with results from the commercial software for thermal-hydraulic rating HTRI, which is one of the most widely used commercial software for solving thermal-hydraulic rating of heat exchangers. The scope of this paper is to assess how exact the cell method is and if its results are useful for a mechanical design of shell and tube heat exchanger with segmental baffles.

1. Introduction

Heat exchangers are used in wide range of industries to facilitate heat transfer between two fluids at different temperatures [1]. The heat transfer is usually forced through heat transfer area by convention, conduction, radiation or combination of these phenomena [2]. Many types of the heat exchangers are used in a great number of industry branches (shell and tube, compact, plate, etc.) [3].

For proper functionality a proper design of the heat exchanger is required. The heat exchanger design may be divided into the several steps. The first step is a design specification, when client and manufacturer discuss the client needs and manufacturer possibilities. The second step is a data collection, when the designer has to collect all process data. Generation of possible design solutions is the third step of heat exchanger design. Designer is finding a proper configuration, relying on his or her previous experience. Fourth step of design is an evaluation and selection [4].

Thermal-hydraulic rating and assessment of temperature field is very important step during heat exchanger design. The proper accomplishment is crucial for the correct function of heat exchanger, especially for executing correct heat duty and then for proper choice of design options of the heat exchanger. It is possible to use a great number of analytical models or commercial software for thermal hydraulic rating of heat exchangers. The selection of calculation model depends mainly on the type of analysed heat exchanger [4]. Nowadays, CFD (computational fluid dynamics) represents the state of the art. CFD simulations can yield very accurate results, but they can be very time consuming. CFD

approach allows us to include simulation of phenomena such as condensing, evaporation or fouling of the heat exchanger which are hard to solve by analytical methods [5–7].

The thermal stresses can cause grand failure of the shell and tube heat exchanger. These eventual failures can cause operating problems or problems with safety of plant. So, it is very important to have some utility for calculation of temperature field, which is crucial for thermal stresses calculation. Unfortunately, the thermal stress is often neglected during the shell and tube heat exchanger mechanical design due to high cost of software for calculation of temperature field. This paper deals with a simple method for calculation of temperature field for shell and tube heat exchangers. Results from this method are compared with commercial software for computation of the thermal hydraulic rating HTRI. This article also highlights the problems with initial data for cell method, especially with correct determination of the overall heat transfer coefficient.

2. Cell method description

This method for the temperature field calculation is useful mainly for a single-phase single pass or multi pass shell-and-tube heat exchanger with segmental baffles [8]. The part of heat exchanger between baffles makes cells (figure 1). There is a cross flow in the space between baffles, or between tube sheets and baffles. In reality, the situation is slightly different. There are a leakage and bypass streams in the shell side, which are ignored in this model [3].
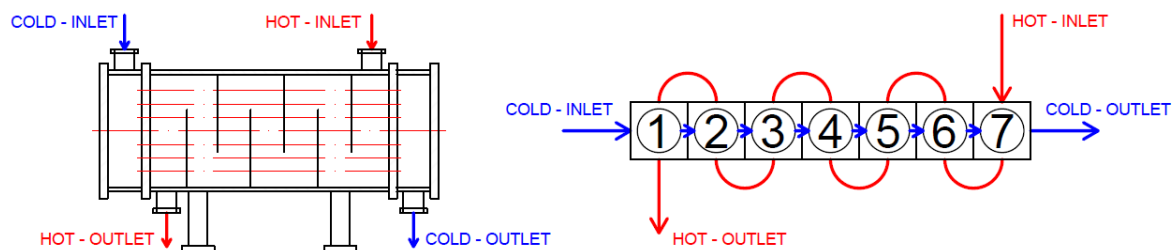

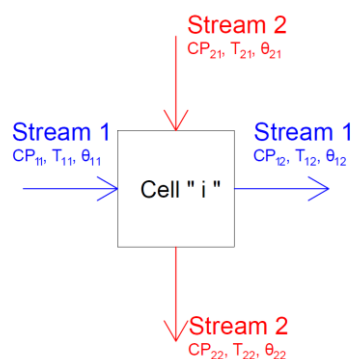**Figure 1.** Heat exchanger conversion for cell method.

Figure 2. Cell scheme [3].

The scheme of cell is shown in figure 2. The stream 1 flows through tube side of heat exchanger, stream 2 flows across tubes in the shell side of the heat exchanger. For this description the lower heat capacity of stream 1 then stream 2 is expected. If the heat capacity premise is executed, it is possible to write system of three equations [3]:

$$\dot{Q}_{cell} = CP_1 \cdot (T_{12} - T_{11}) \tag{1}$$

$$\dot{Q}_{cell} = -CP_2 \cdot (T_{22} - T_{21}) \tag{2}$$

$$\dot{Q}_{cell} = E_C \cdot CP_1 \cdot (T_{21} - T_{11}) \tag{3}$$

where E_c is thermal effectiveness of the cell defined as [3]:

$$E_C = \frac{T_{12} - T_{11}}{T_{21} - T_{11}} \tag{4}$$

It is convenient to use the dimensionless temperatures defined as [3]:

$$\theta = \frac{T - T_{1i}}{T_{2i} - T_{1i}} \tag{5}$$

Where T_{1i} is inlet temperature of stream 1, T_{2i} is inlet temperature of stream 2.

The inlet cell temperatures for each stream are identical with the exit temperatures of the preceding cells [1]. For all cells $0 \le \Theta \le 1$ and the dimensionless inlet temperatures are $\Theta_{1i} = 0$ and $\Theta_{2i} = 1$ for tube side and shell side respectively [3].

After assessment of dimensionless temperatures, equations (1) – (3) and (5) give the dimensionless equations which must be applied for all cells [3]:

$$\theta_{12i} = a \cdot \theta_{11i} + b \cdot \theta_{21i} \tag{6}$$

$$\theta_{12i} = e \cdot \theta_{21i} + f \cdot \theta_{11i} \tag{7}$$

where:

$$a = 1 - E_C \tag{8}$$

$$b = E_C \tag{9}$$

$$e = 1 - R \cdot E_C \tag{10}$$

$$f = R \cdot E_C \tag{11}$$

$$R = \frac{CP_1}{CP_2} \tag{12}$$

The cell effectiveness E_c may be expressed by [3]:

$$E_C = f(NTU_C, R, cell\ flow\ condition) \tag{13}$$

where NTU_C is number of transfer unit and it is defined as [3]:

$$NTU_C = \frac{U \cdot A_C}{CP_1} \tag{14}$$

where U (W m^{-2} K^{-1}) is overall heat transfer coefficient, A_C (m^2) is heat transfer area of cell and CP_1 (J kg^{-1} K^{-1})) is the heat capacity of stream 1. It is possible to calculate only one thermal effectiveness for whole analysed heat exchanger but for this paper, it was calculated for each cell separately. Just as the thermal effectiveness, values of heat capacity are calculated for each cell.

The solution was obtained by an iteration method. Just as thermal effectiveness, each number of transfer unit is calculated for each cell separately for more precise results. For calculating of number of transfer unit, the heat transfer coefficient is essential. Many methods can be used for evaluation of overall heat transfer coefficient. The easiest and fastest way is to use Kern method, but many more sophisticated methods can be used. For this paper Kern method was used [4]. Cell area is derived from

Kern method where it is defined as total heat transfer area, which is divided by number of heat exchanger cells for obtain a cell area.

3.   Analysed heat exchangers

As mentioned above, this article is focused on the shell and tube heat exchangers with segmental baffles. Two cases of heat exchanger are solved by cell methods as well as commercial software for thermal hydraulic ratings HTRI. Results from both methods will be compared.

Case 1

The co-current shell and tube heat exchanger with floating head from TEMA database [9] was analysed first (figure 3). It has one shell and one tube pass and six segmental baffles. Number of tubes is 78 and their longitudinal is 1.77 m. Cold water flows through tubes, hot air flows in the shell side. The inlet temperature of cold water is 73 °C, the air inlet temperature is 276 °C and it is being cooled down to 121 °C. The water is heating up to 77 °C. Mass flow rate of hot stream (air) is 4680 kg h^{-1}. Geometry data are shown in table 1. Input data for thermal hydraulic rating are showed in table 2.
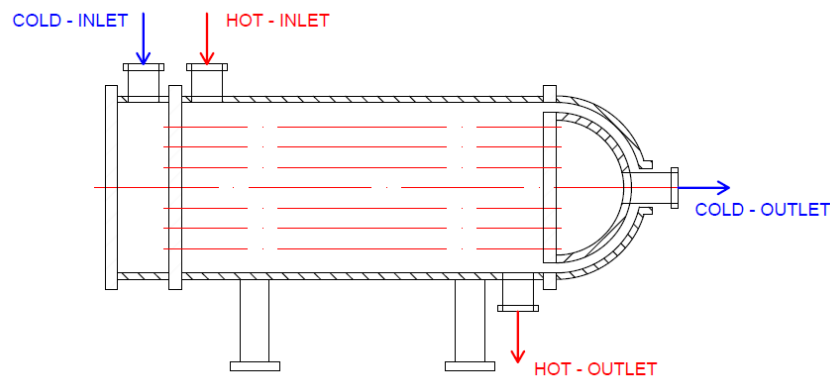

Figure 3. Analysed heat exchanger – Case 1.

Table 1. Analysed heat exchanger data – Case 1.

Quantity	Value
Flow configuration	Co-current
Tube side	
Tube side stream	Cold
Tube side fluid	Water
Tube arrangement	Triangle 30°
Tube pitch	19.844 mm
Tube length	1829 mm
Outer tube diameter	15.875 mm
Wall thickness	1.651 mm
Number of tubes	78
Shell side	
Shell side stream	Hot
Shell side fluid	Air
Shell diameter	257.287 mm
Bundle diameter	206.872 mm

Table 2. Initial data for thermal-hydraulic rating – Case 1.

Quantity	Notation	Value
Number of tube passes	n_T	1
Number of shell passes	n_S	1
Number of baffles	n_B	6
Baffle cut	Baf_{cut}	40 %
Hot stream inlet temperature	TH_{in}	276 °C
Hot stream outlet temperature	TH_{out}	121 °C
Cold stream inlet temperature	TC_{in}	73 °C
Cold stream outlet temperature	TC_{out}	77 °C
Hot stream mass flow rate	m_H	4680 kg h^{-1}

Initial data from table 2 are required for cell method, and for estimate of overall heat transfer coefficient U (W m^{-2} K^{-1}) by Kern method. Just as overall heat transfer coefficient, the heat transfer coefficient h (W m^{-2} K^{-1}) on shell side and tube side was calculated by Kern method. Kern method results for the first case of calculated heat exchangers are listed in table 3.

Table 3. Kern method results – Case 1.

Quantity	Notation	Value
Mean temperature hot stream	TH_{mean}	198.5 °C
Mean temperature cold stream	TC_{mean}	75.0 °C
Logarithmic mean temperature	dT_{lm}	103.99 °C
Heat rate	Q	209.20 kW
Heat transfer area	A_{ht}	8.03 m^2
Overall heat transfer coefficient	U	250.26 W m^{-2} K^{-1}
Tube side		
Mass flow rate	m_C	12.47 kg s^{-1}
Fluid velocity	u_C	1.32 m·s^{-1}
Reynolds number	Re_C	42841.28
Tube heat transfer coefficient	h_C	9232.77 W m^{-2} K^{-1}
Shell side		
Mass flow rate	m_H	1.32 kg s^{-1}
Fluid velocity	u_H	17.05 m s^{-1}
Reynolds number	Re_H	43029.34
Shell heat transfer coefficient	h_H	303.29 W m^{-2} K^{-1}

Analysed heat exchanger has rather small number of tubes and relatively small dimensions. The overall heat transfer coefficient for case 1 is only around 250 W m^{-2} K^{-1}. The heat transfer area is 8 m^2 which reflects tube parameters. Relatively high value of the Reynolds number, especially in tube side, may indicate susceptibility to vibrations. On the other hand, the high speed of water in tubes, which causes high value of Reynolds number, causes higher heat transfer coefficient on tube side.

When parameters of heat transfer coefficients and geometry are known, it is possible to use the cell method to calculate the temperature field. Temperature field is very useful not only during the thermal-hydraulic rating, but also for structural design. Thermal stress could have detrimental effect on structure, mainly on tubes, tube sheet or shell and could cause accident. Unequal heating of flange and bolts could cause flange leakage. These are reasons why the correct calculation of temperature field is very important during the heat exchanger design. There are listed all input data for cell method in table 4. In figure 3 is shown scheme of heat exchanger which is converted to cells for cell methods.

Table 4. Cell method input data – Case 1.

Quantity	Notation	Value
Number of baffles	N_b	6
Tube side passes	N_t	1
Number of tubes	N_{tube}	78
Mass flow rate of cold stream	m_C	1.32 kg s^{-1}
Mass flow rate of hot stream	m_H	12.47 kg s^{-1}
Inlet temperature of cold stream	TC_{in}	276 °C
Inlet temperature of hot stream	TH_{in}	73 °C
Tube length	L_t	1.829 m

For proper calculation, the heat capacities of both process fluids are needed. Process fluid in tube side is water. Its initial specific heat capacity is 52285 J kg^{-1} K^{-1}. In the shell side, process fluid is air and its mass heat capacity is 1370.8 J kg^{-1} K^{-1}. As written above, mass heat capacity changes in each iteration step according a temperature. Just as mass heat capacity, the effectiveness of cells is changing during the iteration steps.

There is list of process fluid temperatures in cells along the analysed heat exchanger in the table 5. In figure 4, there is a plot of temperature profile along the heat exchanger. Red colour represents temperature profile in shell side, blue colour represents temperature profile in tube side.

Table 5. Temperature field results (°C) – Case 1.

Tubes							
Tin	Tcell1	Tcell2	Tcell3	Tcell4	Tcell5	Tcell6	Tout
276.0	248.30	224.49	204.0	186.39	171.24	158.21	146.77
Shell							
Tin	Tcell1	Tcell2	Tcell3	Tcell4	Tcell5	Tcell6	Tout
73.0	73.73	74.35	74.88	75.35	75.74	76.09	76.39
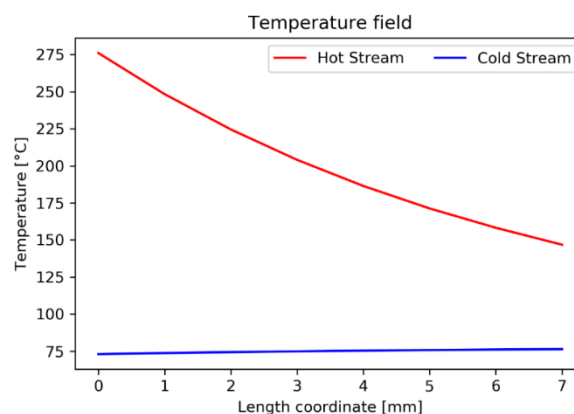
Figure 4. Heat exchanger temperature field – Case 1.

Results shows that outlet temperature in tube side is rather similar to inlet temperature which was expected (table 3). However, the shell side outlet temperature indicates more than 20 °C difference in compare with initial data. It could be caused by using Kern method for estimation of the shell side condition. Kern method does not reflect bypass streams and leakage in shell side which can influence results negatively.

The graph of temperature field is shown in figure 4. Red line represents hot stream, blue line represents cold stream. Hot stream shows the massive decrease trend which is opposite to cold stream. The cold stream trend is almost constant along the analysed heat exchanger.

Case 2

The second analysed case is a shell and tube heat exchanger with segmental baffles. It has 61 tubes welded in the fixed tubesheet. Flow configuration is counter-current. Process fluid is water in the tube side as well as in the shell side. The inlet temperature of water to shell side is 95 °C, the outlet temperature is 88 °C. In the tube side, the water is heated up from 32 to 37 °C. Other heat exchanger data are listed in table 6.
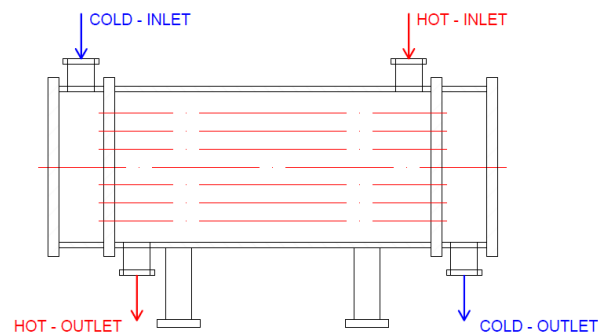

Figure 5. Analysed heat exchanger – Case 2.

Table 6. Analysed heat exchanger data – Case 2.

Quantity	Value
Flow configuration	Co-current
Tube side	
Tube side stream	Cold
Tube side fluid	Water
Tube arrangement	Triangle 30°
Tube pitch	26.0 mm
Tube length	954.0 mm
Outer tube diameter	20.0 mm
Wall thickness	2.0 mm
Number of tubes	61
Shell side	
Shell side stream	Hot
Shell side fluid	Air
Shell diameter	284.24 mm
Bundle diameter	232.37 mm

Table 7. Initial data for thermal-hydraulic rating – Case 2.

Quantity	Notation	Value
Number of tube passes	n_T	1
Number of shell passes	n_S	1
Number of baffles	n_B	4
Baffle cut	Baf_{cut}	45 %
Hot stream inlet temperature	TH_{in}	95 °C
Hot stream outlet temperature	TH_{out}	88 °C
Cold stream inlet temperature	TC_{in}	32 °C
Cold stream outlet temperature	TC_{out}	37 °C
Hot stream mass flow rate	m_H	7200 kg h^{-1}

Initial data for calculation of the overall heat transfer coefficient are listed in table 7. For cell method, it is crucial to estimate overall heat transfer coefficient. In this case by Kern method, its results are listed in next table 8.

Table 8. Kern method results – Case 2.

Quantity	Notation	Value
Mean temperature hot stream	TH_{mean}	91.5 °C
Mean temperature cold stream	TC_{mean}	34.5 °C
Logarithmic mean temperature	dT_{lm}	56.99 °C
Heat rate	Q	58.91 kW
Heat transfer area	A_{ht}	2.08 m^2
Overall heat transfer coefficient	U	497.25 W m^{-2} K^{-1}
Tube side		
Mass flow rate	m_C	2.82 kg s^{-1}
Fluid velocity	u_C	0.23 m·s^{-1}
Reynolds number	Re_C	5054.67
Tube heat transfer coefficient	h_C	1231.78 W m^{-2} K^{-1}
Shell side		
Mass flow rate	m_H	2.0 kg s^{-1}
Fluid velocity	u_H	0.19 m s^{-1}
Reynolds number	Re_H	8457.88
Shell heat transfer coefficient	h_H	2325.43 W m^{-2} K^{-1}

From Kern method results, it is shown that case 2 has significantly smaller heat transfer area, which has great impact on overall heat transfer coefficient. However, these cases are not comparable, because of the different process fluids, flow rates and velocities. In table 8, it is shown that mass flow rate is relatively small just as the velocity of process fluids. Temperature field data obtained using cell method are given in table 9.

Table 9. Cell method input data – Case 2.

Quantity	Notation	Value
Number of baffles	N_b	5
Tube side passes	N_t	1
Number of tubes	N_{tube}	61
Mass flow rate of cold stream	m_C	2.0 kg s^{-1}
Mass flow rate of hot stream	m_H	2.82 kg s^{-1}
Inlet temperature of cold stream	TC_{in}	95 °C
Inlet temperature of hot stream	TH_{in}	32 °C
Tube length	L_t	0.954 m

It was operated equally to case 1 with physical data of process fluids. The initial mass heat capacity of water in tube side was 4178 J kg^{-1} K^{-1}, for shell side 4212 J kg^{-1} K^{-1}. Cell methods results are listed in table 10. In figure 6 is shown temperature field of analysed heat exchanger.

Table 10. Temperature field results (°C) – Case 2.

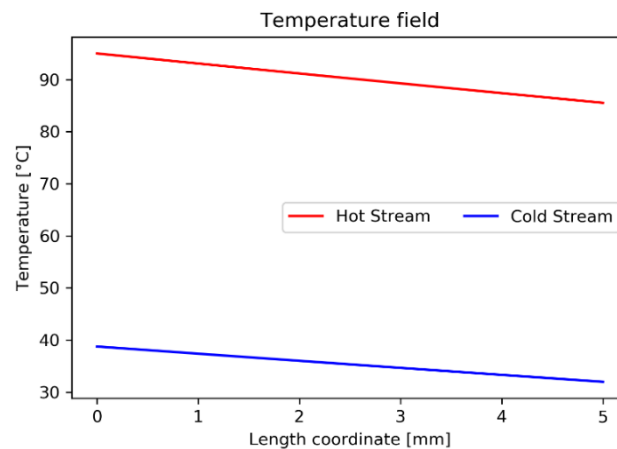| Tubes					
Tin	Tcell1	Tcell2	Tcell3	Tcell4	Tout
32.0	33.33	34.67	36.03	37.39	38.77
Shell					
Tin	Tcell1	Tcell2	Tcell3	Tcell4	Tout
95.0	93.07	91.16	89.26	87.39	85.53

Figure 6. Heat exchanger temperature field – Case 2.

Results of case 2 indicated negligible outlet temperature difference compared to initial data. Especially in shell side, it is very satisfactory results in case of case 1. When the velocity of stream is low, the bypass streams in shell side are not noticeable and has not got great influence to temperatures.

4.   Comparing of solutions results

The commercial software for thermal-hydraulic rating HTRI was used for validation of cell method results. This software is one of the most used software for thermal analyses of the great number heat exchanger configurations. The biggest problem is a high price, which cannot be rentable for everyone. For simple heat exchanger configurations deals with in this paper is HTRI needlessly sophisticated. This means that, cell method could be fair alternative for temperature field rating of simple configurations heat exchanger.

Case 1

Comparison of results from cell method and HTRI for case 1 are listed in table 11, where the temperatures of hot and cold stream along tube length of the analysed heat exchanger are listed. Length coordinate from HTRI is also used for cell method. Course of temperatures from both methods is shown in figure 7.

Table 11. Cell method and HTRI comparison – Case 1.

	T1 (°C)	T2 (°C)	T3 (°C)	T4 (°C)	T5 (°C)	T6 (°C)	T7 (°C)	T8 (°C)
Length coordinate	0.0	190.23	507.50	761.50	1015.50	1269.50	1586.67	1776.90
Hot stream – shell side								
HTRI	276.0	249.60	207.92	180.83	159.60	142.93	128.30	121.0
Cell	276.0	255.26	224.54	204.04	186.42	171.27	155.39	146.77
Difference (%)	0.0	3.65	10.72	14.97	17.31	18.28	17.48	16.66
Cold stream – tube side								
HTRI	73.0	73.69	74.77	75.47	76.01	76.44	76.81	77.0
Cell	73.0	73.54	74.35	74.89	75.35	75.75	76.16	76.39
Difference (%)	0.0	3.58	10.44	14.46	16.56	17.33	16.27	15.33

The difference between results from cell method and HTRI software is calculated for inlet and outlet temperature differences from HTRI for both streams. From table 11, it is obvious that the largest difference is on the outlet temperatures of both streams. Through whole length of the shell side the huge difference is evident between results from HTRI and cell method. These differences are caused by calculation of shell side heat transfer coefficient by Kern method, which is not sophisticated enough.

This method, for example, does not compute with bypass streams in the shell side, which has negative impact to shell side heat transfer coefficient and consequently to stream temperatures.
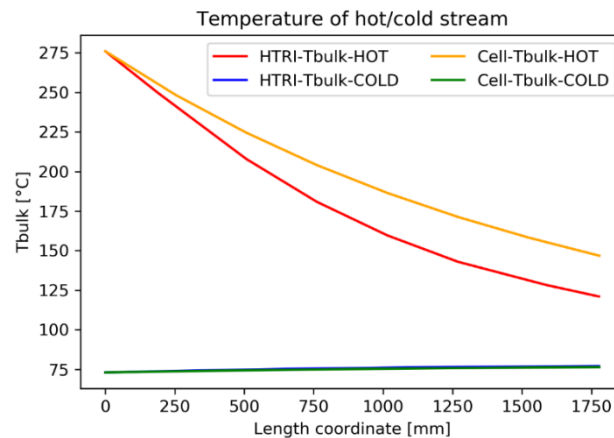

Figure 7. Comparison of temperature field form cell method and HTRI – Case 1.

The course of hot streams (red – HTRI; orange – cell method) and cold streams (blue – HTRI; green – cell method) is shown in figure 7. Difference of hot stream, which was described in previous paragraph is less than 20 % in the worst case. On the other hand, it possible to watching almost the same trend of cold stream curves.

Case 2

Similar comparison as for the case 1 is done for the case 2. In the table 12 are listed data from HTRI and from cell method.

Table 12. Cell method and HTRI comparison – Case 2.

	T1 (°C)	T2 (°C)	T3 (°C)	T4 (°C)	T5 (°C)	T6 (°C)	T7 (°C)
Length coordinate	0.0	85.08	265.56	456.36	647.16	827.64	912.72
Hot stream – shell side							
HTRI	95.0	94.37	93.07	91.74	90.33	88.80	88.0
Cell	94.60	93.75	91.95	90.06	88.17	86.37	85.52
Difference (%)	5.85	8.85	15.97	23.98	30.91	34.62	35.29
Cold stream – tube side							
HTRI	37.0	36.55	35.62	34.67	33.66	32.57	32.0
Cell	38.77	38.17	36.89	35.53	34.18	32.90	32.29
Difference (%)	35.42	32.43	25.30	17.27	10.31	6.60	5.86

Results listed in table 12 have larger diversion in compare with previous case. The large difference between results from HTRI and cell method is obvious. The maximum difference is more than 35 % on both sides, which is not satisfactory. On the inlet of heat exchanger, the difference between cell method results and HTRI is around 6 % and the difference increase through whole length of heat exchanger.
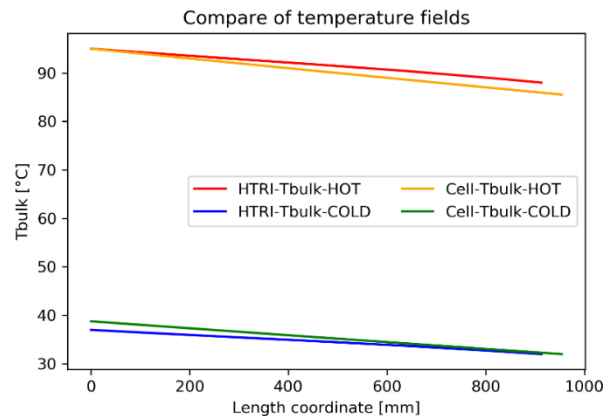
Figure 8. Comparison of temperature field form cell method and HTRI – Case 2.

The courses of both stream from HTRI and cell method are shown in figure 8. Temperature field from HTRI is represented by red curve (hot stream) and blue (cold stream). Temperature field from cell method is represented by orange curve (hot stream) and green curve (cold stream). It is obvious that curves of hot streams have similar trend as curves of cold streams which correspond with data from table 12.

5. Conclusion

The cell method can be used for calculation of temperature field of the shell and tube heat exchanger. The results from cell method has been compared with results from commercial software HTRI. The results from cell method are less accurate than from HTRI. The differences between temperatures from cell method and HTRI can be caused by used input data for the cell method. However, these results can be sufficient for calculation of temperature field for structural design of the shell and tube heat exchanger.

The most problematic input parameter for cell method is overall heat transfer coefficient. In this paper, Kern method has been used for calculation of cell method initial data. Other methods may provide more accurate result of overall heat transfer coefficient, since they are more complex, but on the other hand more time consuming. The next parameter, which has significant impact on results is fluid velocity especially in the shell side, because of bypass streams and leakage. This phenomenon can affect the overall heat transfer coefficient. Despite all of this, the results showed in this paper are satisfactory. The average difference between results from the cell method and HTRI is around 12 % for case 1 and around 20 % for case 2. This difference is probably caused by initial data, mainly overall heat transfer coefficient.

The differences of results are showed in the figure 9. The red colour, represents the hot stream difference between HTRI and cell method, the blue colour, represents the cold stream difference. It is obvious, that the difference increase with length of heat exchanger. In the figure 9 a) the difference goes down in the outlet of both streams. The difference is probably caused by more sophisticated calculation in the HTRI. The HTRI should use some more exact calculation for heat transfer coefficient rating or for temperature field calculation. Unfortunately, it is not possible to discover some HTRI calculation code.
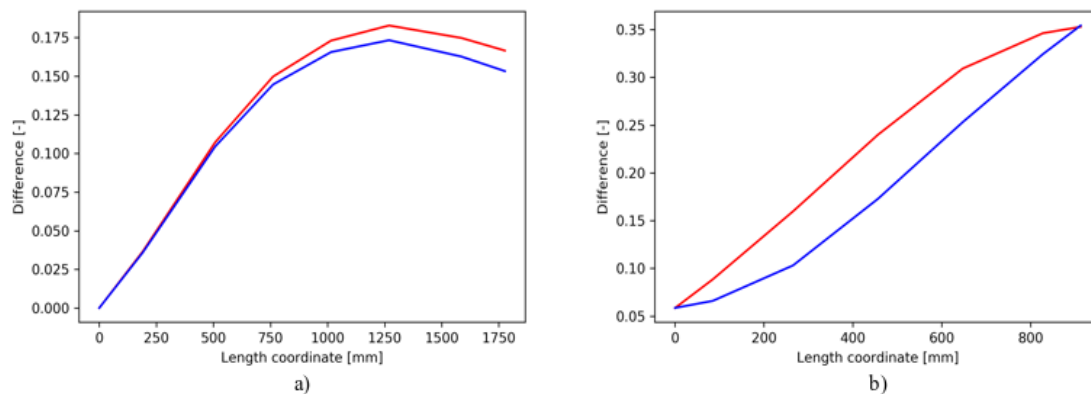
Figure 9. Difference between HTRI and Cell method: a) Case 1, b) Case 2.

This configuration of the cell method, which is described in this paper, can be used only for described type of heat exchangers (single phase shell and tube heat exchanger with segmental baffles). Since there are many types of heat exchangers in industry the future work will focus on multiphase heat exchangers and calculation of more accurate initial data including bypass streams or leakage.

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