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Advantages and Disadvantages Concerning the Heat Transfer in Tube in Tube Heat Exchangers versus Shell and Tube Heat Exchangers

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Abstract

This study is based on the technological analysis of shell and tube heat exchangers carrying out the vapors condensing from the top of propane-propylene fractionating column. By knowing operating conditions for a shell and tube heat exchanger, were sized a tube in tube heat exchangers batteries. The quantities compared between the two apparatus are represented by partial and overall heat transfer coefficients, required heat transfer areas and pressure drops. The comparative results of these quantities, good maintenance and operating flexibility are the main criteria for choosing the optimal device in terms technical and economic for the condensation of propylene according to practical data.

Key words: *heat transfer, tubular heat exchanger, heat transfer coefficient*

Introduction

The tubular heat exchangers have a major role in achieving the heat transfer processes of heating, preheating, cooling, condensation, evaporation, etc. in process units from petroleum refineries. This category of devices includes shell and tube heat exchangers and tube in tube heat exchangers. The shell and tube heat exchangers are commonly used. The advantages of using shell and tube heat exchangers are simple design, low pressure loss and easy maintenance. The tube in tube heat exchangers have the advantage of working in countercurrent, which means a better heat transfer can be operated at high temperatures, have good resistance to high pressure and as needed their surface can be modified by increasing or decreasing the number of elements. The disadvantage of their use can be given by large outline marker dimensions, making it voluminous and heavy in relation to the heat transfer area [1-9].

The aim of this paper is to highlight the advantages and disadvantages of condensation process in a tube in tube heat exchanger compared to a shell and tube heat exchanger. The operating conditions, as practical data, taken from propane - propylene fractionation unit of Catalytic Cracking platform were used for this study. Figure 1 shows a schematic diagram of propane - propylene fractionation unit which is composed of: fractionating column, reboiler, air-cooled, condensers with water, reflux vessel and water coolers. In fractionating column is separated the propane – propylene mixture, in bottom the propane and propylene at the top. After leaving the column, the propylene flow in vapor phase is cooled in an air cooler, and then is condensed in a tubular heat exchangers using recirculated water as thermal agent. Then, the propylene flow in liquid phase is passed into the reflux vessel. From the reflux vessel a part of the propylene is recirculated to top of column, and another part is cooled and circulated to the output of unit.

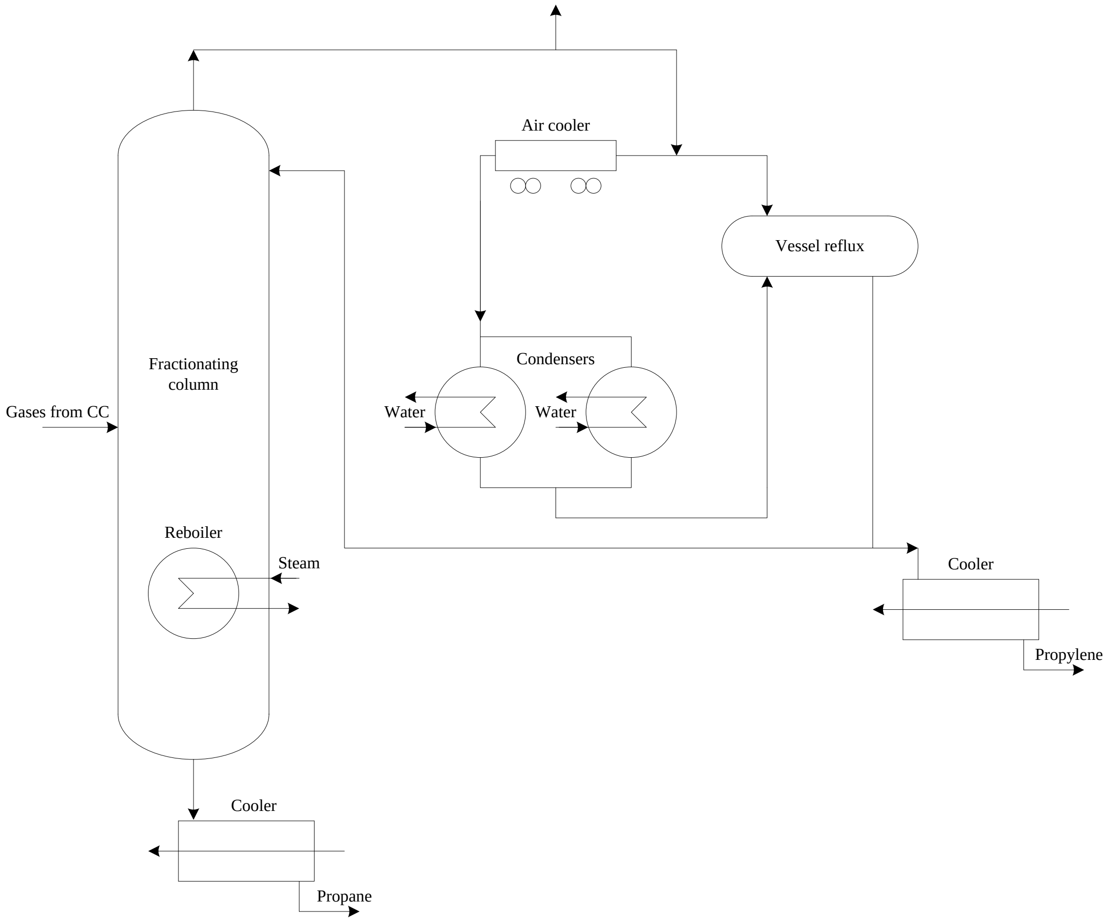

Fig. 1. Propane - propylene fractionation unit

When the working parameters were taken only one of the two condensers was functionally.
The used condenser is a shell and tube heat exchanger with movable head. By knowing operating conditions and the configuration of propane - propylene fractionation unit, was done the technological analysis of existing apparatus and was dimensioned a tube in tube heat exchangers system. The compared functional parameters between the two types of devices are the partial and global heat transfer coefficients, heat transfer areas and pressure drops. Therefore, may be establish the main performance indicators and could be established the advantages and disadvantages of each apparatus.

The Shell and Tube Heat Exchanger

The existing shell and tube heat exchanger (STHE) is with one pass through the shell and four passes through the tubes (n_p = 4), fluid flow being performed in cross-flow. The construction of the apparatus (fig. 2) consists of a tubes bundle (1), fixed to the tube ends into the tubesheet holes (2) and enclosed in a cylindrical shell (3), provided with input and output fittings for propylene. The recirculated water flows through the inside of tubes, and propylene flows through the intertubular space. This space is provided with transversal baffles (4). The tubular space is closed at the ends with caps (5) provided with input and output fittings. Between the tubesheet and caps is formed the distribution chamber (6), place where is done the collection of recirculated water.

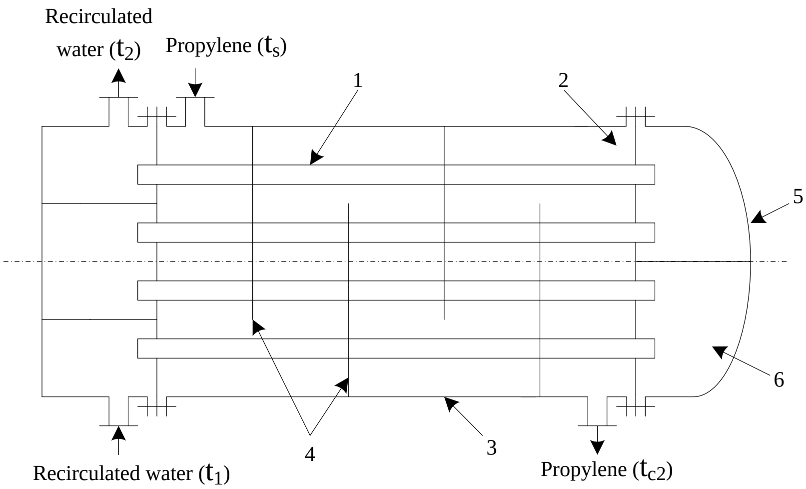

Fig. 2. Shell and tube heat exchanger 1 - 4:
1 - tube bundle; 2 – tubesheet; 3 - shell;
4 - baffle; 5 - cap; 6 – distribution chamber.

The condensation of 8.4 t/h propylene (m_c) is achieved in apparatus. The saturation temperature of the propylene vapors is t_s = 44 °C at 18 bar, and outlet the liquid propylene has temperature t_{c2} = 42.6 °C. The recirculated water has flow rates m_r = 60.5 t/h, inlet temperature t_1 = 25 °C (pressure 5 bar) and outlet temperature t_2 = 35 °C. In the shell with inner diameter D_i = 0.9 m 469 tubes (n_t) are placed. The tubes are arranged in an equilateral triangle shape with step s = 32 mm, they have outer diameter d_e = 20 mm, inner diameter d_i = 16 mm, length L = 6 m and they are manufactured of carbon steel with thermal conductivity λ_o = 52 W/m/°C.

For the calculation of partial heat transfer coefficients criterial relations existing in the literature [3, 4] were used.

The partial heat transfer coefficient on the inside of tube, α_i (W/m^2/°C) can be calculated with the equation:

$$\alpha_i = (1415 + 21 \cdot t_m) \cdot \frac{w^{0.8}}{d_i^{0.2}}, \tag{1}$$

where t_m is the average temperature (estimated as arithmetic mean of inlet and outlet temperatures) and w is the flow velocity (m/s). These relations are recommended to water flow in turbulent regime temperatures, through circular spaces and temperatures between 0 and 100°C.

The partial heat transfer coefficient on outside of the tube, α_e (W/m^2/°C) can be calculated using Kern method with relation:

$$\alpha_e = 0.36 \cdot \frac{\lambda}{d_{ec}} \cdot Re^{0.55} \cdot Pr^{1/3} \cdot \left(\frac{\mu}{\mu_p}\right)^{0.14}, \tag{2}$$

where Re represents Reynolds number, Pr represents Prandtl number, λ is thermal conductivity (W/m/°C), d_{ec} is the equivalent diameter (m), μ is dynamic viscosity (kg/m/s), and the p index is for wall temperature.

$$Re = \frac{d_{ec} \cdot G_{fa}}{\mu}, \tag{3}$$

$$d_{ec} = \frac{1.1 \cdot s^2}{d_e} - d_e, \tag{4}$$

$$G_{fa} = \frac{m_c}{S_{fa}}, \tag{5}$$

$$S_{fa} = s_s \cdot \left[D_i - d_f + \frac{\left(d_f - d_f\right) \cdot \left(2 \cdot s_t - d_e\right)}{2 \cdot s_t} \right], \tag{6}$$

In the last two equations G_{fa} represents the mass velocity in bundle (kg/m^2/s), S_{fa} represents the flow area of the bundle (m^2), s_s is distance between two baffles, s_s =0.4 m, d_f is the diameter of the bundle (m) and s_t is transversal step, $s_t = 0.5 \cdot s$.

$$Pr = \frac{c_p \cdot \mu}{\lambda}, \tag{7}$$

The overall heat transfer coefficient, k (W/m^2/°C) is calculated from the equation:

$$Q = k \cdot A_e \cdot \Delta t_{ml}, \tag{8}$$

$$k = \frac{Q}{A_e \cdot \Delta t_{ml}}, \tag{9}$$

where Q represents the thermal flow (W), A is the heat transfer area on the outside of the tubes (m^2) and Δt_{ml} is the logarithmic mean temperature difference.

$$Q = m_c \cdot r, \tag{10}$$

where r is latent heat of condensation.

Considering the tubesheet thickness (δ_{pt}) 100 mm, the effective length of the tubes ($L_{t,ef}$) through which the heat exchange is achieved is $L_{t,ef} = L - 2 \cdot \delta_{pt}$.

The effective heat transfer area of the apparatus, A_{ef} is:

$$A_{ef} = n_t \cdot \pi \cdot d_e \cdot L_{t,ef} \quad , \tag{11}$$

For simplicity, in the apparatus was considered countercurrent flow therefore the expression of Δt_{ml} is following:

$$\Delta t_{ml} = \frac{\left(t_s - t_2\right) - \left(t_{c2} - t_1\right)}{\ln \dfrac{t_s - t_2}{t_{c2} - t_1}}, \tag{12}$$

The expression of overall heat transfer coefficient for the dirty apparatus, k_{ed}, (W/m^2/°C) is:

$$k_{ed} = \left[\frac{d_e}{\alpha_i \cdot d_i} + R_{mi} + \frac{d_e}{2 \cdot \lambda_p} \ln \frac{d_e}{d_i} + R_{me} + \frac{1}{\alpha_e} \right]^{-1}, \tag{13}$$

In equation (13) the deposition resistances are for recirculated water R_{mi} = 0.0003 m$^2 \cdot$°C/W and for propylene R_{me} = 0.0001 m$^2 \cdot$°C/W [3].

The total pressure drop of the fluid from tubes (water), ΔP_i, can be calculated with the equation:

$$\Delta P_i = \frac{n_p \cdot G_i^2}{\rho} \cdot \left(\frac{f \cdot L}{2 \cdot d_i} + 2 \right), \tag{14}$$

where G_i represents the mass velocity in the tubes (kg/m^2/s), ρ is the density and f is the load loss coefficient.

$$G_i = \frac{m_r}{N_{t,p} \cdot \frac{\pi}{4} \cdot d_i^2}, \tag{15}$$

where $N_{t,p}$ is the number of tubes per step.

From the construction of apparatus resulting 2 passes in bundle each with 116 tubes, a pass with 118 tubes and a pass with 119 tubes. For the calculation of the pressure loss is used flow velocities through the passage with the minimum tubes number.

Load loss coefficient depends on value of the number *Re*, where:

$$Re = \frac{d_i \cdot w_i \cdot \rho_i}{\mu_i}, \tag{16}$$

For *Re* > 2300, *f* can be calculated with the equation:

$$f = 0.16 \cdot \mathrm{Re}^{-0.16}, \tag{17}$$

The pressure drop in the intertubular space, ΔP_e, can be calculated with the equation:

$$\Delta P_e = \frac{f \cdot G_{fa} \cdot D_i \cdot (n_s + 1)}{2 \cdot d_{ec} \cdot \rho}, \tag{18}$$

where *f* can be calculated with the equation:

$$f = 1.84 \cdot \mathrm{Re}^{-0.2}, \tag{19}$$

In equation (18) n_s is the number of baffles and can establish with the relation [3, 4]:

$$n_s = \frac{L_{t,ef} - (s_{si} + s_{se})}{s_s} + 1, \tag{20}$$

where the distance from the baffle to tubesheet in inlet nozzle zone of fluid in shell s_{si}, will be considered equal with output zone, s_{se}, namely s_{si} = s_{si} = 0.6 m.

The Tube in Tube Heat Exchanger

The tube in tube condenser is constituted as a system of tube in tube heat exchangers (TTHE), composed of several elements which in turn may constitute one or more batteries (fig. 3). Recirculated water flows through central tube, and propylene flows through the annular space created between the central tube and the outer tube.

Sizing the TTHE was performed by using the same working parameters presented before for the STHE. The main steps in the sizing computation of the tube in tube condenser are presented below:

1. Estimation of overall heat transfer coefficient, *k*, from the literature data[3, 8];
2. The calculation of heat transfer area, *A* (m^2) for estimated *k*;

$$A = \frac{Q_C}{k \cdot \Delta t_{ml}}, \tag{21}$$

3. The choice of apparatus dimensions:

   Tubes from the same material as the tubes of shell and tube heat exchanger and the dimensions $d_{e,1}/d_{i,1}$ = 114.3/109.3 mm for the central tube, $d_{e,2}/d_{i,2}$ = 168.3/163.3 mm for outer tube and the length *L* = 10 m have been chosen. Then, batteries placed in parallel with the same distribution of fluid flow rates have been chosen. The number of batteries n_b and the number of elements of a battery n_e so as to obtain flow velocities (implicit flow regimes) desired in apparatus had been established.

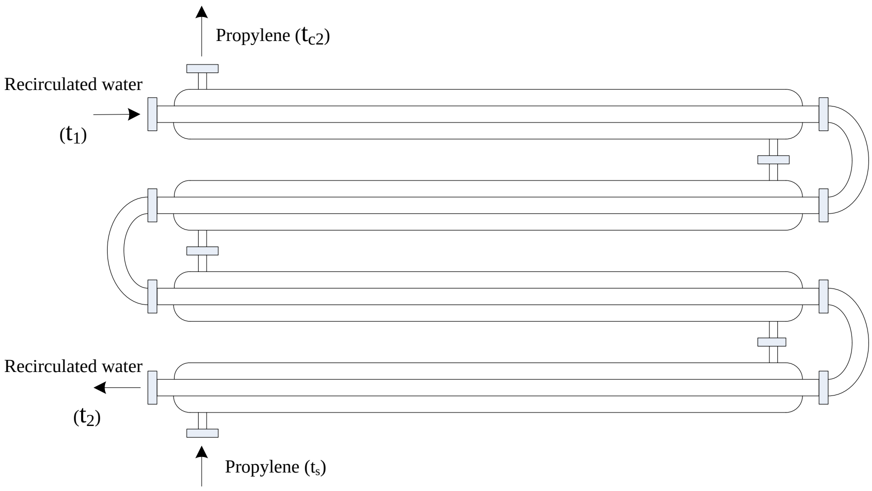

Fig. 3. Tube in tube heat exchangers battery

At the flowing through the central tube the flow velocity w_i has been calculated using the equation:

$$w_i = \frac{4 \cdot m_r}{n_b \cdot \rho_i \cdot \pi \cdot d_{1,i}^2}, \tag{22}$$

and at the flowing through annular space the flow velocity w_e has been calculated using the equation:

$$w_e = \frac{4 \cdot m_c}{n_b \cdot \rho_e \cdot \pi \cdot \left(d_{2,i}^2 - d_{1,e}^2\right)}, \tag{23}$$

The effective heat transfer area area A_{ef} of the apparatus chosen had been calculated:

$$A_{ef} = \pi \cdot d_{1,e} \cdot L \cdot n_e \cdot n_b, \tag{24}$$

4. Verification of the apparatus chosen by calculating the partial and overall heat transfer coefficients, the heat transfer area and the oversize:

   The partial heat transfer coefficients α_i and α_e have been calculated using the same relations used to calculate these quantities for shell and tube heat exchanger, with the specification that in the calculation of α_e the equivalent diameter d_{ec} in equation (2) is the equivalent heated diameter d_t, expressed by the relation:

$$d_t = \frac{d_{2,i}^2 - d_{1,e}^2}{d_{1,e}}, \tag{25}$$

   and the *Re* numbar in the annular space has been calculated with equation (15) and and wherein the d_i is replaced by the equivalent hydraulic diameter d_h:

$$d_h = d_{2,i} - d_{1,e}, \tag{26}$$

   After the calculation of k_{ed} with equation (13) can be obtained the necessary heat transfer area A_{nec} using the equation (21).

5. The calculation of the oversize apparatus, OA

$$OA = \frac{A_{ef} \quad A_{nec}}{A_{nec}} \quad 100\,, \tag{27}$$

The in the design calculations shall be accepted OA << 20 % [3, 4].

The pressure in the central tube and in the annular space has been calculated with the following equation [3, 8]:

$$\Delta P_{i,e} = \frac{f_{i,e} \cdot G_{i,e}^2 \cdot L}{2 \cdot g \cdot \rho_{i,e} \cdot l_c}\,, \tag{28}$$

where l_c represents the characteristic length. At the flowing through the central tube l_c is d_i, and at the flowing through the annular space l_c is equivalent hydraulic diameter.

Results and Discussion

The latent condensation heat of propylene at the saturation temperature is r = 296,380 kJ/kg. The physical properties at the average temperature of condensation film, t_m = 43.3 °C (admitted), are: density ρ = 471 kg/m^3, thermal conductivity λ = 0.0929 W/m/°C and dynamic viscosity μ = 0.00008 kg/m/s. The physical properties of water at the average temperature, t_m = 30 °C, are: ρ = 996 kg/m^3, λ = 0.6155 W/m/°C and μ = 0.0008 kg/m/s. In Table 1 the values of main quantities used in the sizing of TTHE and the technological analysis of STHE are presented.

The chosen geometrical data for the sizing of TTHE are standardized [8]. The flow velocities have acceptable values for the two types of apparatus. For water the flow regime is turbulent in both apparatus, and for propylene the flow regime is turbulent in TTHE and transitory in STHE. Table 2 shows the comparison of the performance indicators obtained for the two apparatus.

The large difference between the partial heat transfer coefficients on the inside, α_i of TTHE and STHE is given by significant difference between the inner diameters of the tubes of the two type apparatus, as shown in the Tables 1 and 2.

Conclusions

In this paper have been presented the results obtained from the analysis technology of the shell and tube heat exchanger that condenses the vapors from the top of propane-propylene fractionating column. By knowing operating conditions for this apparatus which is the current system, were sized a tube in tube heat exchangers batteries. The aims were to investigate the possibility of replacing the current system with tube in tube heat exchangers batteries.

As a result of the performed study, the following comments can be made:

- The partial heat transfer coefficient for the recirculated water is higher in the STHE than in the TTHE since the inner diameter of the central tube is greater than at the tubes of STHE;
- The partial heat transfer coefficient for propylene is higher in the TTHE than in the STHE since the flow velocity of propylene in the TTHE is higher than in the STHE;
- The overall heat transfer coefficient in the TTHE is higher than in the STHE since the value of the lowest partial heat transfer coefficient influences the values of overall heat transfer coefficient, and in the present case the partial heat transfer coefficient for propylene at the TTHE is greater than at the STHE;
- The necessary heat transfer area of TTHE is lower than that of STHE;
- The oversize of TTHE is acceptable;
- Pressure drops are performs within acceptable limits.

Table 1. The values of main quantities used in the sizing of THE and the technological analysis of STHE

Parameter	TTHE		STHE	
	Central tube	Annular space	Tubes	Intertubular space
m, kg/s	4.2	0.6	16.8	2.3
t_i, °C	25.0	44.0	25.0	44.0
t_e, °C	35.0	42.5	35.0	42.5
P, bar	5	18	5	18
Q, W	701924			
k, W/m^2/°C	300		240	
Δ_{tml}, °C	12.78		12.78	
d_i, mm	109.3	163.3	20	900
d_e, mm	114.3	168.3	25	-
L, m	10		6	
n_t	52		469	
n_b	4		-	-
n_e	13		-	-
w, m/s	0.45	1.39	0.46	0.67
Re	62327	401434	11620	7477
Pr	5.2	2.7	5.2	2.7
f	0.0274	0.0203	0.03578	0.3091
d_{ec}, m	0.1093	0.049 (d_h) 0.119 (d_t)	0.020	0.0201
s_t	-	-	-	0.016
S_{fa},m^2	-	-		0.078125
G, m/s/m^2	448	55	462	29.8
n_s	-	-	12	

Table 2. The quantities compared for TTHE and STHE

Parameter	TTHE		STHE	
	Central tube	Annular space	Tubes	Central tube
α, W/m^2/°C	1680	473	2395	314
k_{ed}, W/m^2/°C	314		240	
A_{nec}, m^2	175		214	
OA, %	6.7		-	
Δp, N/m^2	253	1.3	6310	5.9

The previously mentioned advantages at replacing STHE with sized TTHE Do not completely counteract the disadvantage which the TTHE has in terms of space on that this system of heat exchangers can occupy in the propane-propylene unit. In this paper, the implications of economic aspects at replacing STHE with a TTHE were not analyzed, which would be a more rigorous technical and economic study, his approach following to be done into another paper.

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Avantaje și dezavantaje la realizarea transferului de căldură în schimbătoare de căldură tub în tub versus schimbătoare de căldură fascicul tubular în manta

Rezumat

Acest studiu se bazează pe analiza tehnologică a schimbătoarelor de căldură fascicul tubular în manta în care se realizează condensarea vaporilor de la vârful coloanei de fracţionare propan-propilenă. Cunoscând condiţiile de operare pentru un schimbător de căldură fascicul tubular în manta au fost dimensionate baterii de schimbătoare de căldură tub în tub. Mărimile comparate între cele două tipuri de aparate sunt coeficienţii parţiali şi globali de transfer de căldură, ariile de transfer de căldură şi căderile de presiune. Rezultatele comparative ale acestor mărimi, întreținerea uşoară și flexibilitatea în operare sunt principalele criterii pentru alegerea aparatului optim din punct de vedere tehnic și economic la condensarea propilenei potrivit datelor practice de operare.