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Thermo-hydraulic design of a shell and tube heat exchanger for acrylic acid cooling.

Diseño térmico-hidráulico de un intercambiador de calor de tubo y coraza para el enfriamiento de ácido acrílico.

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Abstract.
Shell and tube heat exchangers (STHE) in their various manifestations are undoubtedly the most widely and commonly used heat transfer equipment in the chemical processing industries. The objective of the present work is to design, from the thermo-hydraulic point of view, a 1-2 STHE to cool 50,000 kg/h of an acrylic acid stream from 97 to 40 ºC using water as coolant at an inlet temperature of 25 ºC. The proposed STHE will present a heat transfer area of 284.29 m^2, an overall heat transfer coefficient of 364.26 $W/m^2.K$, a number of tubes of 702, a bundle diameter of 975.62 mm, and a shell diameter of 1,047.62 mm. The selected type of STHE is split-ring floating head, the heat load has a value of 1,733.59 kW, it will be required 20.74 kg/s (74,664 kg/h) of cooling water to carry out the heat transfer service, while the values of the pressure drop of both the water (402.54 Pa) and the acrylic acid (2,479.27 Pa) are below the maximum allowable limits set by the heat exchange process, which are 1,000 Pa and 3,000 Pa for the water and acrylic acid, respectively. The designed STHE will has a purchase cost of USD $ 101,209.

Keywords.
Design; Shell and Tube Heat Exchanger; Area; Pressure Drop, Purchase Cost.

Resumen.
Los intercambiadores de calor de tubo y coraza (ICTC) en sus varias manifestaciones son indudablemente los equipos de transferencia de calor más ampliamente y comúnmente usados en las industrias de procesamiento químico. El objetivo del presente trabajo es diseñar, desde el punto de vista térmico-hidráulico, un ICTC 1-2 para enfriar 50 000 kg/h de una corriente de ácido acrílico desde 97 hasta 40 ºC usando agua como refrigerante a una temperatura de entrada de 25 ºC. El ICTC propuesto presentará un área de transferencia de calor de 284,29 m2, un coeficiente global de transferencia de calor de 364,26 W/m2.K, un número de tubos 702, un diámetro del haz de 975,62 mm, y un diámetro de la coraza de 1 047,62 mm. El tipo de ICTC seleccionado es de cabezal flotante de anillo hendido, la carga de calor tiene un valor de 1 733,59, se requerirán 20,74 kg/s (74 664 kg/h) de agua de enfriamiento para llevar a cabo el servicio de transferencia de calor, mientras que los valores de la caída de presión de tanto el agua (402,54 Pa) como el ácido acrílico (2 479,27 Pa) están por debajo de los límites máximos permisibles fijados por el proceso de intercambio de calor, los cuales son 1 000 Pa y 3 000 Pa para el agua y el ácido acrílico, respectivamente. El ICTC diseñado tendrá un costo de adquisición de USD $ 101 209.

Palabras clave.
Diseño; Intercambiador de Calor de Tubo y Coraza; Área; Caída de Presión; Costo de Adquisición.

1.- Introduction

Heat transfer is the field that focuses basically on the rate heat is exchanged between hot and cold objects, referred to as the source and the receiver, respectively. The devices used to facilitate this heat transfer are known as heat exchangers [1].

Heat exchangers function on the concept of transferring thermal energy between a fluid at a higher temperature and one at a lower temperature. They work by enabling the hot fluid to come into contact with the cooler fluid either directly or indirectly. This mechanism allows for heat to transfer from the hotter fluid to the cooler one, leading to a reduction in the temperature of the first fluid and a rise in the temperature of the second fluid. The direction of heat transfer is determined by whether heating or cooling is required for the particular system [2].´

The transfer of heat primarily occurs through conduction and convection. Heat exchangers are typically categorized based on the number of fluids involved, the characteristics of the surface elements, design aspects, fluid flow patterns, and their heat transfer techniques [3].

Among the various categories of heat exchangers, shell and tube heat exchangers (STHEs) are reasonably simple to assemble and offer a wide range of applications for both gases and liquids across extensive temperature and pressure levels [3].

In STHE, two fluids with varying temperatures flow through the system. One fluid travels inside the tubes (known as the tube side) while the other circulates around the tubes within the shell (referred to as the shell side). Thermal energy is exchanged between the fluids via the walls of the tubes, moving from the tube side to the shell side or vice versa. These fluids can be in liquid or gas state,

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whether on the shell side or the tube side. To facilitate effective heat transfer, a considerable heat transfer area is required, leading to the utilization of numerous tubes.

STHEs can be specially designed while considering factors such as functionality, ease of maintenance, adaptability, and safety, resulting in a highly durable heat exchanger that encourages its extensive application across various sectors. It is projected that over 35-40% of heat exchangers used in contemporary engineering sectors are of the shell and tube type, thanks to their reliable structural design, easy maintenance, and potential for upgrades. For optimal heat transfer efficiency, shell and tube heat exchangers should aim for a minimal pressure drop, elevated mass flow velocity on the shell side, a high heat transfer coefficient, and minimal to negligible fouling, among other essential features [3].

STHEs facilitate the exchange of large quantities of heat efficiently and cost-effectively, offering a low-cost tube surface while minimizing the area needed on the floor, the volume of liquid, and the overall weight, while they are available in diverse sizes and lengths [4].

These heat exchangers are prevalent across various sectors, such as power generation facilities where they act as condensers, and in chemical and petrochemical sectors for preheating or cooling functions [5]. They are also employed in refrigeration, climate control, and the food production industry, among others [3]. Common uses often include the heating or cooling of relevant fluid streams and the condensation or evaporation of fluid mixtures. Furthermore, certain applications aim to recover or reject heat or carry out sterilization, pasteurization, fractionation, distillation, concentration, crystallization, or thermal adjustment of process fluids [6].

The thermo-hydraulic design of a shell and tube heat exchanger generally involves calculating the heat transfer surface area, amount of heat transferred, overall heat transfer efficiency, tube quantity, tube dimensions, arrangement, number of passes for the shell and tube, type of heat exchanger (like fixed tube sheets or removable tube bundles), tube spacing, quantity and specifications of baffles, as well as pressure drops on both the shell and tube sides, among other factors [4].

Numerous investigations have been documented involving the design of a STHE. In this context, [5] introduced a detailed design approach for STHE influenced by the analysis of flexibility indices. This approach aims to mitigate challenges like possible design inefficiencies or inadequate functioning of entire process systems. This research incorporates a genetic algorithm with stringent constraints for optimizing the design of the STHE. Furthermore, [4] provided insights into the calculations required for designing heat exchangers of the shell and tube variety, outlining a methodical process for determining designs, intending to serve as a standardized guide for performing these calculations systematically for STHE design. Similarly, [7] focused on designing an STHE intended for applications related to nanofibril cellulose production, adhering to the TEMA standards, and executed parameter calculations manually through the Microsoft Excel program. Likewise, [8] designed a shell and tube heat exchanger for Diesel Locomotives employing the Bell Delaware technique to derive various dimensions, including shell, tubes, and baffles. Subsequently, a thermal analysis was executed using COMSOL, applying various thermal loads while adjusting the number of baffles. Additionally, [2] highlighted the design and evaluation of shell and tube heat exchangers by examining different materials and their heat transfer capabilities from surfaces, while also studying baffle spacing and its influence on heat transfer through Computational Fluid Dynamics (CFD) analysis. The findings were contrasted with theoretical models. The design and simulation of the heat exchanger was completed using PTC Creo Parametric and ANSYS Fluent for CFD analysis, considering materials such as copper, aluminum, and steel.

In [9], a counter-current shell and tube heat exchanger constructed for a nitric acid manufacturing facility was presented, where the design was conducted with the target processing capacity of 100 tons of nitric acid per day. This project employed two distinct methodologies, Kern's approach and Bell's approach, during the design process. It was determined that Bell's approach provided more precise results, as the overall heat transfer coefficient derived from this method closely matched the predicted value. Additionally, the design included auxiliary components of the heat exchanger such as flanges, gaskets, bolts, supports, and saddles. In another study [10], researchers designed and assessed the effectiveness of a shell and tube heat exchanger utilizing both Kern's approach and Ansys software, employing CFD to analyze the temperature and flow rate within the tubes and shell, reaching the conclusion that the heat transfer along the tube length varies.

In [11], a straightforward method for designing a shell and tube heat exchanger for applications in the beverage and process industries was described; this design process addressed both thermal and structural aspects. The thermal design aspect involved calculating the necessary effective surface area (which refers to the number of tubes) and determining the logarithmic mean temperature difference, while the mechanical design involved designing the shell to withstand both internal and external pressures, along with the design of tubes, baffles, gaskets, etc. The design process adhered to the ASME/TEMA standards.

In [12], a shell-and-tube heat exchanger featuring a single shell pass along with two tube passes was developed to function as a water heater, utilizing sulfur water as the heating agent. The construction materials chosen for the heat exchanger included stainless steel 304 for the shells and copper for the tubes. Likewise, in [13], a design and rating approach for STHEs equipped with helical baffles was introduced, which was based from existing public sources and the prevalent Bell-Delaware technique for STHEs utilizing segmental baffles. This method replaced various curve-type factors from the literature with mathematical formulas to simplify engineering design, thereby detailing the calculation process for the proposed approach. Finally, [14] explores into the fundamental principles of thermal design for STHEs, discussing elements such as the components of STHEs; their classification based on construction and operation; necessary data for thermal design; tube-side design; shell-side design, incorporating tube arrangement, baffling, pressure drop on the shell side; and the mean temperature difference. It emphasizes the use of essential equations related to heat transfer and pressure loss on both the tube side and shell side for the optimal design of the STHE.

Several books [1] [15]-[18] describe useful calculation methodologies to design STHE form the thermal and hydraulic point of view, which are modern adaptations or versions of the classic Kern's method, as well as describing the Bell-Delaware method.

In certain chemical plant is desired to cool down 50,000 kg/h of a stream of acrylic acid produced at the bottom of a distillation column prior to be stored, and for that a shell and tube heat exchanger was proposed. In this context, the aim of this study is to design a STHE to cool this acrylic acid stream from 97 ºC to 40 ºC using cooling water at an inlet temperature of 25 ºC. To design the STHE the calculation methodology reported in [17], which is based on Kern's approach, was applied due to its simplicity and innovative features. This methodology allows calculating several design parameters for the STHE such as heat exchange area, number of tubes, shell diameter, overall heat transfer coefficient, as well as the pressure drop of both streams. Also, the purchase cost of the designed STHE will be calculated and updated to 2024 year.

2.- Materials and methods.

2.1. Problem statement

It's required to cool down 50,000 kg/h of an acrylic acid stream coming from the bottom of distillation column from 97 ºC to 40 ºC using cooling water at an inlet temperature of 25 ºC. For this heat transfer service a horizontal shell and tube heat exchanger is proposed working as a cooler. The outlet temperature of the cooling water must not exceed 45 ºC for safety issues, while the pressure drop of the acrylic acid and cooling water streams should not exceed 5,000 Pa and 1,000 Pa respectively. The heat exchanger must operate under countercurrent arrangement and will be of 1-2 type, i.e. with one shell pass and two tube passes. To design the proposed 1-2 shell and tube heat exchanger the methodology reported by [17] will be used, which is based on Kern's approach. Also, the purchase cost of the heat exchanger will be calculated by using the correlation published in [17], which depends on the calculated heat exchange area.

2.2. Design methodology.

The calculation methodology applied to design the shell and tube heat exchanger from the thermo-hydraulic point of view is shown below.

Preliminary design

Step 1. Definition the initial data available for the two streams:

Table 1 shows the initial data available for the two streams.

Table 1. Initial data available for the two streams.

Parameter	Units	Cold fluid	Hot fluid
Mass flowrate	kg/h	m_c	m_h
Inlet temperature	ºC	t_1	T_1
Outlet temperature	ºC	t_2	T_2
Maximum permissible pressure drop	Pa	$\Delta P_{c(p)}$	$\Delta P_{h(p)}$
Fouling factor	W/m^2.ºC	R_c	R_h

Source: Own elaboration.

Step 2. Average temperature of both streams:

- Cold fluid ($\bar{t}$):

$$\bar{t} = \frac{t_1 + t_2}{2} \tag{1}$$

- Hot fluid ($\bar{T}$):

$$\bar{T} = \frac{T_1 + T_2}{2} \tag{2}$$

Step 3. Physical properties of both fluids at the average temperature:

Table 2 presents the physical properties that must be defined for both fluids at the average temperature calculated in the previous step.

Table 2. Physical properties to be defined for both fluids.

Property	Units	Cold fluid	Hot fluid
Density	kg/m^3	ρ_c	ρ_h
Viscosity	Pa.s	μ_c	μ_h
Heat capacity	kJ/kg.ºC	Cp_c	Cp_h
Thermal conductivity	W/m.K	k_c	k_h

Source: Own elaboration.

Step 4. Heat load (Q):

- For the hot fluid:

$$Q = \frac{m_h}{3{,}600} \cdot Cp_h \cdot (T_1 - T_2) \tag{3}$$

Where the unit of Q is kW.

Step 5. Required mass flowrate of the cold fluid (cooling water) (m_c):

$$m_c = \frac{Q}{Cp_c \cdot (t_2 - t_1)} \tag{4}$$

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Where Q is given in kW and Cp_c is given in kJ/kg.K.

Step 6. Assumption of the overall heat transfer coefficient (U_0).

Step 7. Log mean temperature difference (ΔTlm):

- For a countercurrent arrangement:

$$\Delta Tlm = \frac{(T_1 - t_2) - (T_2 - t_1)}{ln \frac{(T_1 - t_2)}{(T_2 - t_1)}} \quad (5)$$

Step 8. Factor R:

$$R = \frac{(T_1 - T_2)}{(t_2 - t_1)} \quad (6)$$

Step 9. Factor S:

$$S = \frac{(t_2 - t_1)}{(T_1 - t_1)} \quad (7)$$

Step 10. Temperature correction factor (F_t):

- For a 1 shell: 2 tube pass heat exchanger:

$$F_t = \frac{\sqrt{(R^2+1)} \cdot ln\left[\frac{(1-S)}{(1-R\cdot S)}\right]}{(R-1)\cdot ln\left[\frac{2-S\cdot\left[R+1-\sqrt{(R^2+1)}\right]}{2-S\cdot\left[R+1+\sqrt{(R^2+1)}\right]}\right]} \quad (8)$$

Step 11. True temperature difference (ΔTm):

$$\Delta Tm = \Delta Tlm \cdot F_t \quad (9)$$

Step 12. Provisional heat transfer area (A_0):

$$A_0 = \frac{Q \cdot 1{,}000}{U_0 \cdot \Delta Tm} \quad (10)$$

Where Q is given in kW.

Step 13. Select the following data for the tubes:

- Nominal diameter.
- Material.
- Length (L_t).

Step 14. Area of one tube (a_1):

$$a_1 = \pi \cdot L_t \cdot d_0 \quad (11)$$

Where L_t and d_0 are given in m.

Step 15. Number of tubes (N_0):

$$N_0 = \frac{A_0}{a_1} \quad (12)$$

Step 16. Tube arrangement:
Triangular or square pitch.

Step 17. Selection of the constants K_1 and n_1 depending on the tube arrangement (triangular or square) and the number of tube passes.

Step 18. Bundle diameter (D_b):

$$D_b = d_0 \cdot \left(\frac{N_0}{K_1}\right)^{1/n_1} \quad (13)$$

Where d_0 is given in mm.

Step 19. Select the type of shell and tube heat exchanger:

- Pull-through floating head.
- Split-ring floating head.
- Outside packed head.
- Fixed and U-tube.

Step 20. Shell-bundle clearance (C_{sb}) in mm.

Step 21. Shell diameter (D_s):

$$D_s = D_b + C_{sb} \quad (14)$$

Where D_b and C_{sb} are given in mm.

Step 22. Fluids allocation inside the heat exchanger.

Tube side coefficient

Step 23. Tube cross-sectional area (a_t):

$$a_t = \frac{\pi \cdot d_i^2}{4} \quad (15)$$

Where d_i is given in m.

Step 24. Number of tubes per pass (N_{tp}):

$$N_{tp} = \frac{N_0}{n_p} \quad (16)$$

Where n_p – number of tube-side passes = 2.

Step 25. Total flow area (a_T):

$$a_T = N_{tp} \cdot a_t \quad (17)$$

Step 26. Mass velocity of the tube-side fluid (G_t):

$$G_t = \frac{m_t}{a_T} \quad (18)$$

Where m_t is given in kg/s.

Step 27. Linear velocity of the tube-side fluid (v_t):

$$v_t = \frac{G_t}{\rho_t} \quad (19)$$

Step 28. Reynolds number of the tube-side fluid (Re_t):

$$Re_t = \frac{\rho_t \cdot v_t \cdot d_i}{\mu_t} \quad (20)$$

Step 29. Prandtl number of the tube-side fluid (Pr_t):

$$Pr_t = \frac{(Cp_t \cdot 1{,}000) \cdot \mu_t}{k_t} \quad (21)$$

Where Cp_t is given in kJ/kg.K.

Step 30. Ratio L_t/d_i, where both L_t and d_i are given in m.

Step 31. Tube-side heat-transfer factor (j_{h1}), depending on the ratio L_t/d_i and Reynolds number.

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Step 32. Tube-side heat-transfer coefficient (h_i):

$$h_i = \frac{k_t}{d_i} \cdot j_{h1} \cdot Re_t \cdot Pr_t^{0.33} \cdot \left(\frac{\mu_t}{\mu_{tw}}\right)^{0.14} \tag{22}$$

Where d_i is given in m.

For water flowing in pipes, the following correlation could be used:

$$h_i = \frac{4{,}200 \cdot (1.35 + 0.02 \cdot \bar{t}) \cdot v_t^{0.8}}{d_i^{0.2}} \tag{23}$$

Where:
$\bar{t}$ – Average temperature of water (ºC).
v_t – Water velocity (m/s).
d_i – Tube inside diameter (mm).

Shell-side coefficient:
Step 33. Baffle spacing (l_B):

$$l_B = D_s \cdot \varphi \tag{24}$$

Where φ = 0.2 – 0.5 [17] and D_s is given in mm.

Step 34. Tube pitch (p_t):

$$p_t = 1.25 \cdot d_o \tag{25}$$

Where d_o is given in m.

Step 35. Cross-flow area of the shell-side fluid (A_s):

$$A_s = \frac{(p_t - d_0)}{p_t} \cdot D_s \cdot l_B \tag{26}$$

Where all the parameters are given in m.

Step 36. Mass velocity of the shell-side fluid (G_s):

$$G_s = \frac{\frac{m_s}{3{,}600}}{A_s} \tag{27}$$

Where m_s and A_s are given in kg/h and m^2, respectively.

Step 37. Shell-side equivalent diameter (hydraulic diameter) (d_e):

- Square pitch:

$$d_e = \frac{1.27}{d_0} \cdot (p_t^2 - 0.785 \cdot d_0^2) \tag{28}$$

- Triangular pitch:

$$d_e = \frac{1.10}{d_0} \cdot (p_t^2 - 0.917 \cdot d_0^2) \tag{29}$$

Where p_t and d_0 are given in m.

Step 38. Reynolds number of the shell-side fluid (Re_s):

$$Re_s = \frac{G_s \cdot d_e}{\mu_s} \tag{30}$$

Step 39. Prandtl number of the shell-side fluid (Pr_s):

$$Pr_s = \frac{(Cp_s \cdot 1{,}000) \cdot \mu_s}{k_s} \tag{31}$$

Where Cp_s is given in kJ/kg.K.

Step 40. Selection of the baffle cut (%).

Step 41. Shell-side heat-transfer factor (j_{h2}), depending on the baffle cut and Reynolds number.

Step 42. Shell-side heat-transfer coefficient (h_0):

$$h_0 = \frac{k_s}{d_e} \cdot j_{h2} \cdot Re_s \cdot Pr_s^{0.33} \cdot \left(\frac{\mu_s}{\mu_{sw}}\right)^{0.14} \tag{32}$$

Where d_e is given in m.

Overall heat transfer coefficient calculated
Step 43. Thermal conductivity of the tube material (k_W).

Step 44. Overall heat transfer coefficient calculated (U_C):

$$U_C = \frac{1}{\frac{1}{h_0} + \frac{1}{R_s} + \frac{d_0 \cdot ln\left(\frac{d_0}{d_i}\right)}{2 \cdot k_W} + \frac{d_0}{d_i} \cdot \frac{1}{R_t} + \frac{d_0}{d_i} \cdot \frac{1}{h_i}} \tag{33}$$

Pressure drop
Step 45. Friction factor for the tube-side fluid (j_{f1}).

Step 46. Pressure drop of the tube-side fluid (ΔP_t):

$$\Delta P_t = n_p \cdot \left[8 \cdot j_{f1} \cdot \left(\frac{L_t}{d_i}\right) \cdot \left(\frac{\mu_t}{\mu_{tw}}\right)^{-m} + 2.5\right] \cdot \frac{\rho_t \cdot v_t^2}{2} \tag{34}$$

Where m = 0.25 for laminar flow (Re_t < 2,100) and = 0.14 for turbulent flow (Re_t > 2,100), while L_t and d_i are given in m, ρ_t and v_t are given in kg/m^3 and m/s, respectively.

Step 47. Friction factor of the shell-side fluid (j_{f2}).

Step 48. Linear velocity of the shell-side fluid (v_s):

$$v_s = \frac{G_s}{\rho_s} \tag{35}$$

Step 49. Pressure drop of the shell-side fluid (ΔP_s):

$$\Delta P_s = 8 \cdot j_{f2} \cdot \left(\frac{D_s}{d_e}\right) \cdot \left(\frac{L_t}{l_B}\right) \cdot \frac{\rho_s \cdot v_s^2}{2} \cdot \left(\frac{\mu_s}{\mu_{sw}}\right)^{-0.14} \tag{36}$$

Where D_s, d_e, L_t and l_B are given in m.

Purchase cost of the heat exchanger
To calculate the purchase cost of the proposed heat exchanger, the following correlation was used [17]:

$$C_{exch(2007)} = a + b \cdot A^n \tag{37}$$

Where:
a = 24,000
b = 46.
n = 1.2
A – Heat exchanger area, which must be in the range of 10 – 1,000 m^2.

The purchase cost calculated by eq. (37) for the designed heat exchanger is referred to January 2007. To update the purchase cost of the shell and tube heat exchanger to May 2024, the following correlation was used:

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$$C_{exch(2024)} = C_{exch(2007)} \cdot \frac{CE\ Index\ (2024)}{CE\ Index\ (2007)} \quad (38)$$

Where:

$C_{exch(2024)}$ – Cost of the shell and tube heat exchanger in May 2024.

$C_{exch(2007)}$ – Cost of the shell and tube heat exchanger in January 2007, calculated by eq. (37).

$CE\ Index\ (2024)$ - Chemical Engineering Index in May 2024 = 800.0 [19].

$CE\ Index\ (2007)$ - Chemical Engineering Index in January 2007 = 509.7 [17].

3.- Analysis and Interpretation of Results.

3.1. Preliminary design.

Shown below are each step implemented in the methodology to design the shell and tube heat exchanger for acrylic acid cooling.

Step 1. Definition of the initial data available for the two streams:
Table 3 shows the initial data available for the two streams.

Table 3. Initial data available for the two streams.

Parameter	Units	Cooling water	Acrylic acid
Mass flowrate	kg/h	-	50,000
Inlet temperature	ºC	25	97
Outlet temperature	ºC	45	40
Maximum allowable pressure drop	Pa	1,000	5,000
Fouling factor	W/m².ºC	1,000	3,000

Source: Own elaboration.

Step 2. Average temperature of both streams:

- Cold fluid ($\bar{t}$):

$$\bar{t} = \frac{t_1 + t_2}{2} = \frac{25 + 45}{2} = 35\ ºC \quad (1)$$

- Hot fluid ($\bar{T}$):

$$\bar{T} = \frac{T_1 + T_2}{2} = \frac{97 + 40}{2} = 68.5\ ºC \quad (2)$$

Step 3. Physical properties of both fluids at the average temperature:
According to [20], both fluids present the physical properties displayed in Table 4 at the average temperatures calculated in the previous step.

Table 4. Physical properties defined for both fluids.

Property	Units	Cooling water	Acrylic acid
Density	kg/m³	994.033	995.54
Viscosity	Pa.s	0.000719	0.0005696
Heat capacity	kJ/kg.ºC	4.179	2.1897
Thermal conductivity	W/m.K	0.6233	0.1449

Source: Own elaboration.

Step 4. Heat load (Q):
Using the initial data for the hot fluid:

$$Q = \frac{m_h}{3{,}600} \cdot Cp_h \cdot (T_1 - T_2) \quad (3)$$

$$Q = \frac{50{,}000}{3{,}600} \cdot 2.1897 \cdot (97 - 40)$$

$$Q = 1{,}733.59\ kW$$

Step 5. Required mass flowrate of the cold fluid (cooling water) (m_c):

$$m_c = \frac{Q}{Cp_c \cdot (t_2 - t_1)} = \frac{1{,}733.59}{4.179 \cdot (45 - 25)} \quad (4)$$

$$m_c = 20.74\ kg/s$$

Step 6. Assumption of the overall heat transfer coefficient (U_0).
Taking into account the range reported by [17] for coolers that use water to cool organic solvents, it was assumed a value for the overall heat transfer coefficient (U_0) of 300 W/m².K.

Step 7. Log mean temperature difference (ΔTlm):

- For a countercurrent arrangement:

$$\Delta Tlm = \frac{(T_1 - t_2) - (T_2 - t_1)}{ln\frac{(T_1 - t_2)}{(T_2 - t_1)}} \quad (5)$$

$$\Delta Tlm = \frac{(97 - 45) - (40 - 25)}{ln\frac{(97 - 45)}{(40 - 25)}}$$

$$\Delta Tlm = 29.76\ ºC$$

Step 8. Factor R:

$$R = \frac{(T_1 - T_2)}{(t_2 - t_1)} = \frac{(97 - 40)}{(45 - 25)} = 2.85 \quad (6)$$

Step 9. Factor S:

$$S = \frac{(t_2 - t_1)}{(T_1 - t_1)} = \frac{(45 - 25)}{(97 - 25)} = 0.278 \quad (7)$$

Step 10. Temperature correction factor (F_t):

- For a 1 shell: 2 tube pass heat exchanger:

$$F_t = \frac{\sqrt{(R^2 + 1)} \cdot ln\left[\frac{(1 - S)}{(1 - R \cdot S)}\right]}{(R - 1) \cdot ln\left[\frac{2 - S \cdot \left[R + 1 - \sqrt{(R^2 + 1)}\right]}{2 - S \cdot \left[R + 1 + \sqrt{(R^2 + 1)}\right]}\right]} \quad (8)$$

$$F_t = 0.683$$

Step 11. True temperature difference (ΔTm):

$$\Delta Tm = \Delta Tlm \cdot F_t = 29.76 \cdot 0.683 = 20.326\ ºC \quad (9)$$

Step 12. Provisional heat transfer area (A_0):

$$A_0 = \frac{Q \cdot 1{,}000}{U_0 \cdot \Delta Tm} = \frac{1{,}733.59 \cdot 1{,}000}{300 \cdot 20.326} \quad (10)$$

$A_0 = 284.29\ m^2$

Step 13. Selection of the following data for the tubes:

- Nominal diameter: ¾ in, 40ST. Thus, according to [20]:
  Outside diameter (d_0) = 0.0267 m.
  Inside diameter (d_i) = 0.0209 m.
- Material: Stainless steel (18/8).
- Length (L_t) = 4.83 m.

Step 14. Area of one tube (a_1):

$$a_1 = \pi \cdot L_t \cdot d_0 = 3.14 \cdot 4.83 \cdot 0.0267 \quad (11)$$

$$a_1 = 0.4049\ m^2$$

Step 15. Number of tubes (N_0):

$$N_0 = \frac{A_0}{a_1} = \frac{284.29}{0.4049} = 702.12 \approx 702 \quad (12)$$

Step 16. Tube arrangement:
The triangular pitch was selected in order to give higher heat transfer rates, even at the expense of higher pressure drops [17], because the pressure drop is not an important parameter to consider in this heat transfer service according to the supervisors of the industry where this STHE will be installed. However, the pressure drop will be calculated for both fluid streams in this design methodology, and the values obtained will be compared to the maximum allowable limits set by the process.

Step 17. Selection of the constants K_1 and n_1:

According to [17], for a triangular tube arrangement and a number of tube passes (n_p) of 2, the values of these constants are:

- K_1 = 0.249.
- n_1 = 2.207.

Step 18. Bundle diameter (D_b):

$$D_b = d_0 \cdot \left(\frac{N_0}{K_1}\right)^{1/n_1} = 26.67 \cdot \left(\frac{702}{0.249}\right)^{1/2.207} \quad (13)$$

$$D_b = 975.62\ mm$$

Step 19. Select the type of shell and tube heat exchanger:
The selected type of shell and tube heat exchanger is split-ring floating head for efficiency and ease of cleaning [17].

Step 20. Shell-bundle clearance (C_{sb}):
As referred by [17], the shell-bundle clearance for a value of the bundle diameter (D_b) of 975.62 mm and a split-ring floating head type, is 72 mm.

Step 21. Shell diameter (D_s):

$$D_s = D_b + C_{sb} = 975.62 + 72 = 1{,}047.62\ mm \quad (14)$$

Step 22. Fluids allocation inside the heat exchanger.
Taking into account suggestions reported by [17], the cold fluid (cooling water) will be located on the tube side, while the hot fluid (acrylic acid) will be located on the shell side.

3.2. Tube side coefficient.
Due to the allocation of the cold fluid on the tubes and the hot fluid on the shell, the nomenclature of some parameters will be corrected to agree with the nomenclature of the equations that will be used hereafter.

Table 5 indicates the initial and corrected nomenclature of the parameters employed in the upcoming equations.

Table 5. Original and corrected nomenclature of the parameters used in the upcoming equations.

Parameter	Original nomenclature	Corrected nomenclature	Units
Hot fluid flowrate	m_h	m_s	kg/h
Cold fluid flowrate	m_c	m_t	kg/h
Hot fluid density	ρ_h	ρ_s	kg/m³
Cold fluid density	ρ_c	ρ_t	kg/m³
Hot fluid viscosity	μ_h	μ_s	Pa.s
Cold fluid viscosity	μ_c	μ_t	Pa.s
Hot fluid heat capacity	Cp_h	Cp_s	kJ/kg.K
Cold fluid heat capacity	Cp_c	Cp_t	kJ/kg.K
Hot fluid thermal conductivity	k_h	k_s	W/m.K
Cold fluid thermal conductivity	k_c	k_t	W/m.K

Source: Own elaboration.

Table 6 depicts the results of the parameters calculated in the steps 23 to 32, in order to determine the tube-side heat-transfer coefficient.

Table 6. Results of the parameters calculated in steps 23-32.

Step	Parameter	Symbol	Value	Units
23	Tube cross-sectional area	a_t	0.00034	m²
24	Number of tubes per pass	N_{tp}	351	-
25	Total flow area	a_T	0.1193	m²
26	Mass velocity of the tube-side fluid	G_t	173.85	kg/s.m²
27	Linear velocity of the tube-side fluid	v_t	0.175	m/s
28	Reynolds number of the tube-side fluid	Re_t	5,056.57	-
29	Prandtl number of the tube-side fluid	Pr_t	4.82	-

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30	Ratio L_t/d_i	-	231.10	-
31	Tube-side heat-transfer factor[1]	j_{h1}	0.0041	-
32	Tube-side heat-transfer coefficient[2]	h_i	1,162.11	W/m^2.K

[1]For a value for Re_t and L_t/d_i of 5056.57 and 231.10, respectively.
[2]Equation (23) was employed to calculate this parameters since water flows in the pipes.
Source: Own elaboration.

3.3. Shell-side coefficient.
Table 7 presents the results of the parameters calculated in steps 33-42, to determine the shell-side heat transfer coefficient.

Table 7. Results of the parameters calculated in steps 33-42.

Step	Parameter	Symbol	Value	Units
33	Baffle spacing[1]	l_B	209.52	mm
34	Tube pitch	p_t	0.0334	m
35	Cross-flow area of the shell-side fluid	A_s	0.0440	m^2
36	Mass velocity of the shell-side fluid	G_s	315.65	kg/s.m^2
37	Shell-side equivalent diameter[2]	d_e	0.0191	m
38	Reynolds number of the shell-side fluid	Re_s	10,584.47	-
39	Prandtl number of the shell-side fluid	Pr_s	8.61	-
40	Selection of the baffle cut	-	25%	-
41	Shell-side heat-transfer factor	j_{h2}	0.0058	-
42	Shell-side heat-transfer coefficient[3]	h_0	947.66	W/m^2.K

[1]A value of 0.2 was selected for φ to calculate the baffle spacing.
[2]Equation (29) was employed to calculate the shell side equivalent diameter due to the selection of the triangular pitch arrangement.
[3]The viscosity correction term $(\mu_s/\mu_{sw})^{0.14}$ was not considered because both fluids have low viscosity [17].
Source: Own elaboration.

3.4. Overall heat transfer coefficient calculated.
Step 43. Thermal conductivity of the tube material (k_W).
Because the material selected for the tubes is stainless steel 18/8, the thermal conductivity of this material is 16 W/m.K [17].

Step 44. Overall heat transfer coefficient calculated (U_C):

$$U_C = \frac{1}{\frac{1}{h_0} + \frac{1}{R_s} + \frac{d_0 \cdot ln\left(\frac{d_0}{d_i}\right)}{2 \cdot k_W} + \frac{d_0}{d_i} \cdot \frac{1}{R_t} + \frac{d_0}{d_i} \cdot \frac{1}{h_i}} \quad (33)$$

$$U_C = \frac{1}{\frac{1}{947.66} + \frac{1}{5,000} + \frac{0.0267 \cdot ln\left(\frac{0.0267}{0.0209}\right)}{2 \cdot 16} + \frac{0.0267}{0.0209} \cdot \frac{1}{3,000} + \frac{0.0267}{0.0209} \cdot \frac{1}{1162.11}}$$

$$U_C = 364.26\ W/m^2.K$$

3.5. Pressure drop
Step 45. Friction factor for the tube-side fluid (j_{f1}).
According to [17], for a Reynolds number of the tube-side fluid (cooling water) of 5,056.57, the friction factor (j_{f1}) has a value of 0.0058.

Step 46. Pressure drop of the tube-side fluid (ΔP_t):

$$\Delta P_t = n_p \cdot \left[8 \cdot j_{f1} \cdot \left(\frac{L_t}{d_i}\right) \cdot \left(\frac{\mu_t}{\mu_{tw}}\right)^{-0.14} + 2.5\right] \cdot \frac{\rho_t \cdot v_t^2}{2} \quad (34)$$

$$\Delta P_t = 2 \cdot \left[8 \cdot 0.0058 \cdot \left(\frac{4.83}{0.0209}\right) \cdot 1 + 2.5\right] \cdot \frac{994.033 \cdot 0.175^2}{2}$$

$$\Delta P_t = 402.54\ Pa$$

Where $(\mu_t/\mu_{tw})^{-0.14} = 1$ as suggested by [17] because water is not considered a highly viscous fluid.

Step 47. Friction factor of the shell-side fluid (j_{f2}).
According to [17], for a Reynolds number of the shell-side fluid (acrylic acid) of 10,584.47 and a baffle cut of 25%, the friction factor (j_{f2}) has a value of 0.0049.

Step 48. Linear velocity of the shell-side fluid (v_s):

$$v_s = \frac{G_s}{\rho_s} = \frac{315.65}{995.54} = 0.317\ m/s \quad (35)$$

Step 49. Pressure drop of the shell-side fluid (ΔP_s):

$$\Delta P_s = 8 \cdot j_{f2} \cdot \left(\frac{D_s}{d_e}\right) \cdot \left(\frac{L_t}{l_B}\right) \cdot \frac{\rho_s \cdot v_s^2}{2} \cdot \left(\frac{\mu_s}{\mu_{sw}}\right)^{-0.14} \quad (36)$$

$$\Delta P_s = 8 \cdot 0.0049 \cdot \left(\frac{1.04762}{0.0191}\right) \cdot \left(\frac{4.83}{0.20952}\right) \cdot \frac{995.54 \cdot 0.317^2}{2} \cdot 1$$

$$\Delta P_s = 2,479.27\ Pa$$

3.6. Purchase cost of the heat exchanger.
For a value of the heat exchange area of 207.47 m^2, the purchase cost of the designed shell and tube heat exchanger is:

$$C_{exch(2007)} = a + b \cdot A^n \quad (37)$$
$$C_{exch(2007)} = 24,000 + 46 \cdot 284.29^{1.2}$$
$$C_{exch(2007)} \approx USD\ \$\ 64,483$$

Since the purchase cost calculated by equation (37) is for January 2007, the purchase cost of this equipment referred to May 2024 is:

$$C_{exch(2024)} = C_{exch(2007)} \cdot \frac{CE\ Index\ (2024)}{CE\ Index\ (2007)} \quad (38)$$

$$C_{exch(2024)} = 64{,}483 \cdot \frac{800.0}{509.7}$$

$$C_{exch(2024)} = USD\ \$\ 101{,}209$$

4.- Discussion

A shell and tube heat exchanger with one shell pass and two tube passes was designed to cool a stream of acrylic acid, originated at the bottom of a distillation column, from 97 to 40 ºC by means of cooling water at an inlet temperature of 25 ºC, and using the design methodology reported by [17], which is based on Kern's approach. The cooling water was allocated to flow inside the tubes, while the acrylic acid was assigned to flow on the shell.

The calculated value of the heat load for this heat exchanger service was 1,733.59 kW, while it will be required a flowrate of 20.74 kg/s (74,664 kg/h) for the selected heat transfer agent (cooling water). The log mean temperature difference had a value of 29.76 ºC, while the values of the temperature correction factor and the true temperature difference were 0.683 and 20.326 ºC, respectively.

The mass velocity and linear velocity of the cooling water were 173.85 kg/s.m^2 and 0.175 m/s, respectively, while the calculated Reynolds number for this fluid was 5,056.57, thus indicating that the cooling water will flow under the transition regime. The calculated heat transfer coefficient of the tube-side fluid was 1,162.11 W/m^2.K.

The values of the mass velocity and linear velocity of the acrylic acid were 315.65 kg/s.m^2 and 0.317 m/s, respectively. The calculated Reynolds number for the acrylic acid was 10,584.47, thus stating that this fluid will flow under turbulent regime in the designed heat exchanger. The calculated shell-side heat transfer coefficient was 947.66 W/m^2.K.

The heat transfer coefficient of the tube-side fluid is about 1.23 times higher than the shell-side heat transfer coefficient, which agrees with the results of the shell and tube heat exchanger designed in [17], where the heat transfer coefficient of the tube-side fluid (brackish water) is 3,852 W/m^2.K, while the heat transfer coefficient for the shell-side fluid (methanol) is 2,740 W/m^2.K (i.e. about 1.40 times higher).

The calculated pressure drop of the tube-side fluid, i.e. cooling water (402.54 Pa) is about 6.16 times lower than the pressure drop of the shell-side fluid, i.e. acrylic water (2,479.27 Pa). This result agrees with the results of the pressure drop calculated during the design of a shell and tube heat exchanger in [17], where the value of the pressure drop (7.2 kPa) of the brackish water used as a coolant (tube-side fluid) is lower than the value of the pressure drop (272 kPa) of the shell-side fluid (methanol). The values of the calculated pressure drop in the present study for both fluids are below the maximum allowable limits set by the heat exchange service.

A calculated value of the overall heat transfer coefficient of 364.26 W/m^2.K was obtained, which is above the assumed value (300 W/m^2.K) in step 6, thus indicating that the design has adequate area for the duty required [17].

Accordingly, the designed shell and tube heat exchanger in this study will present the following design data:

- Type: Split-ring floating head.
- Heat transfer area (A): 284.29 m^2.
- Number of tubes (N): 702.
- Bundle diameter (D_b): 975.62 mm.
- Shell diameter (D_s): 1,047.62 mm.

The shell and tube heat exchanger designed in [17] in order to cool 100,000 kg/h of a methanol stream by means of brackish water, has the following design parameters:

- Type: Split-ring floating head.
- Heat transfer area (A): 278 m^2.
- Number of tubes (N): 918.
- Bundle diameter (D_b): 826 mm.
- Shell diameter (D_s): 894 mm.

In [9] a shell and tube heat exchanger was designed to cool 0.827 kg/s of nitric oxide stream from 150 ºC to 50 ºC, using water at a supply temperature of 35 ºC. The parameters of the shell and tube heat exchanger designed in this study are shown below:

- Heat transfer area (A): 8.98 m^2.
- Number of tubes (N): 60.
- Bundle diameter (D_b): 240.049 mm.
- Shell diameter (D_s): 251.049 mm.
- Overall heat transfer coefficient (U): 405.62 W/m^2.K.
- Shell side pressure drop: 82.93 kPa.

In this study, the nitric oxide was allocated on the shell-side, while the cooling water was allocated on the tubes. However, the value of tube side heat transfer coefficient (1,059.197 W/m^2.K) is 1.51 times lower than the value of the shell-side heat transfer coefficient (1,601.63 W/m^2.K), which differs with the results of our study.

Other authors [7] carried out the design of a shell and tube heat exchanger for nanofibril cellulose production applications. The results of the performance parameters obtained during the design of this STHE are shown below:

- Heat transfer rate (Q): 167,720 W.
- Area of heat transfer (A): 16.87 m^2.
- Number of tubes (N_t): 53.
- Bundle shell (D_b): 1.85 m.
- Convection heat transfer coefficient in the tube (h_i): 135.34 W/m^2.K.
- Convection heat transfer coefficient in shell (h_0): 0.5934 W/m^2.K.

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- Overall heat transfer coefficient actual (U_{act}): 0.5932 W/m^2.K.
- Effectiveness (ε): 89.21%.

Likewise, in [4] a STHE is designed to cool 1.5 kg/s of an oil stream from 107 ºC to 27 ºC using 1.72 kg/s of cooling water with an inlet temperature of 27 ºC. In this study, the hot fluid is allocated on the shell side while the cold fluid is located on the tube side, which is similar to the conditions of our study. Several parameters are calculated in this work, some of which are presented below:

- Energy transferred (Q): 129,660 W.
- Heat transfer area (A): 3.43 m^2.
- Number of tubes (N_t): 26.
- Heat transfer coefficient on the tube side (h_i): 126.63 W/m^2.K.
- Heat transfer coefficient on the shell side (h_i): 182.65 W/m^2.K.
- Overall heat transfer coefficient assumed (U): 800 W/m^2.K.
- Effectiveness (ε): 50.01%.

5.- Conclusions.

A shell and tube heat exchanger with on shell pass and two tube passes was designed from the thermo-hydraulic point of view, using a well-known design methodology based in Kern's approach, in order to cool down 50,000 kg/h of an acrylic acid stream from 97 ºC to 40 ºC using cooling water at an inlet temperature of 25 ºC. Several parameters were determined such as heat load (1,733.59 kW); overall heat transfer coefficient (364.26 W/m^2.K); heat transfer area (284.29 m^2); number of tubes (702); and shell diameter (1,047.62 mm). The mass flowrate of cooling water required to cool this acrylic acid stream is 20.74 kg/s (74,664 kg/h). The selected shell and tube heat exchanger type was split-ring floating head, while the pressure drop of the water (402.54 Pa) and the acrylic acid (2,479.27 Pa) are lower than the maximum allowable pressure drop set by the service. The purchase cost of the designed shell and tube heat exchanger is USD $ 101,209.

6.- Author Contributions (Contributor Roles Taxonomy (CRediT))

1. Conceptualization: Amaury Pérez Sánchez.
2. Data curation: Laura Thalía Alvarez Lores, Lizthalía Jiménez Guerra.
3. Formal Analysis: Amaury Pérez Sánchez, Laura Thalía Alvarez Lores, Laura de la Caridad Arias Aguila.
4. Acquisition of funds: Not applicable.
5. Research: Amaury Pérez Sánchez, Laura Thalía Alvarez Lores, Laura de la Caridad Arias Águila, Lizthalía Jiménez Guerra.
6. Methodology: Amaury Pérez Sánchez, Laura de la Caridad Arias Águila.
7. Project management: Not applicable.
8. Resources: Not applicable.
9. Software: Not applicable.
10. Supervision: Amaury Pérez Sánchez.
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12. Display: Not applicable.
13. Wording - original draft: Laura Thalía Alvarez Lores, Laura de la Caridad Arias Águila, Lizthalía Jiménez Guerra.
14. Writing - revision y editing: Amaury Pérez Sánchez.

7.- Appendix.

Nomenclature.

a	Constant to use in equation (37)	-
a_1	Area of one tube	m^2
a_t	Tube cross-sectional area	m^2
a_T	Total flow area	m^2
A	Heat exchanger area to use in equation (37)	m^2
A_0	Provisional heat transfer area	m^2
A_s	Cross-flow area of the shell-side fluid	m^2
b	Constant to use in equation (37)	-
Cp	Heat capacity	kJ/kg.K
C_{sb}	Shell-bundle clearance	mm
d_e	Shell-side equivalent diameter (hydraulic diameter)	m
d_i	Tube inside diameter	m
d_0	Tube outside diameter	m
D_b	Bundle diameter	m
D_s	Shell diameter	mm
F_t	Temperature correction factor	-
G	Mass velocity	kg/s.m^2
h_i	Tube-side heat-transfer coefficient	W/m^2.K
h_0	Shell-side heat-transfer coefficient	W/m^2.K
j_{f1}	Friction factor for the tube-side fluid	-
j_{f2}	Friction factor of the shell-side fluid	-
j_{h1}	Tube-side heat-transfer factor	-
k	Thermal conductivity	W/m.K
k_W	Thermal conductivity of the tube material	W/m.K
K_1	Constant to use in equation (13)	-
l_B	Baffle spacing	mm
L_t	Tube length	m
m	Mass flowrate	kg/h
n	Constant to use in equation (37)	-
n_1	Constant to use in equation (13)	-
n_p	Number of tube-side passes	-
N_0	Number of tubes	-
N_{tp}	Number of tubes per pass	-
p_t	Tube pitch	m
Pr	Prandtl number	-
ΔP_t	Pressure drop of the tube-side fluid	Pa
Q	Heat load	kW

R	Factor	-
Re	Reynolds number	-
S	Factor	-
t	Temperature cold fluid	ºC
T	Temperature hot fluid	ºC
$\bar{t}$	Average temperature cold fluid	ºC
$\bar{T}$	Average temperature hot fluid	ºC
ΔTlm	Log mean temperature difference	ºC
ΔTm	True temperature difference	ºC
U_0	Overall heat transfer coefficient assumed	$W/m^2.K$
U_C	Overall heat transfer coefficient calculated	$W/m^2.K$
v	Linear velocity	m/s

Greek symbols

φ	Factor	-
ρ	Density	kg/m^3
μ	Viscosity	Pa.s

Subscripts

1	Inlet
2	Outlet
c	Cold fluid
h	Hot fluid
s	Shell side fluid
t	Tube side fluid

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