Vol. 36, No. 04, pp. 524-550/Septiembre 2023

ISSN-E 1995-9516
Universidad Nacional de Ingeniería

http://revistas.uni.edu.ni/index.php/Nexo
https://doi.org/10.5377/nexo.v36i04.16768

Thermo-hydraulic rating of a shell and tube heat exchanger for acetone cooling using bell-delaware method

Evaluación térmico-hidráulica de un intercambiador de calor de tubo y coraza para el enfriamiento de acetona usando el método de bell-delaware

Amaury Pérez Sánchez [1,*], Alicia Maria Gómez Julián [1], Nadia Corpas Rodriguez [1], Elizabeth Ranero González [1], Eddy Javier Pérez Sánchez [2]

[1] University of Camagüey "Ignacio Agramonte Loynaz", Faculty of Applied Sciences, Camagüey, Cuba.
[2] Company of Automotive Services S.A., Commercial Department, Ciego de Ávila, Cuba.
[*]amaury.perez84@gmail.com

(recibido/received: 15-junio-2023; aceptado/accepted: 1-agosto-2023)

ABSTRACT

Shell and tube heat exchangers are becoming the most popular devices for the transfer of heat in industrial process applications. In the present work, the thermo-hydraulic rating of a shell and tube heat exchanger proposed to cool an acetone stream was carried out using the Bell-Delaware methodology. An overall heat transfer coefficient value of 426.70 W/m^2.K, a calculated heat transfer area of 121.46 m^2, and a percent excess area of 20.83% were obtained. Both the pressure drop of acetone and water had values of 1,622.36 Pa and 3,020.46 Pa, respectively. The proposed shell and tube heat exchanger can be used satisfactorily for the required application, since the percent excess area does not exceed the 25%, and the pressure drops of both fluid streams are below the limit values established by the process.

Keywords: Acetone; Bell-Delaware method; Percent excess area; Pressure drop; Rating; Shell and tube heat exchanger.

RESUMEN

Los intercambiadores de calor de tubo y coraza se están convirtiendo en los dispositivos más populares para la transferencia de calor en aplicaciones de procesos industriales. En el presente trabajo se efectuó la evaluación térmico-hidráulica de un intercambiador de calor de tubo y coraza propuesto para enfriar una corriente de acetona usando la metodología de Bell-Delaware. Se obtuvo un valor del coeficiente global de transferencia de calor de 426,70 W/m^2.K, un área de transferencia de calor calculada de 121,46 m^2, y un porciento de área en exceso de 20,83%. Tanto la caída de presión de la acetona como del agua tuvieron valores de 1622,36 Pa y 3020,46 Pa, respectivamente. El intercambiador de calor de tubo y coraza propuesto puede emplearse satisfactoriamente para la aplicación requerida, ya que el porciento de área en exceso no supera el 25%, y las caídas de presión de ambas corrientes de fluido se encuentran por debajo de los valores límites establecidos por el proceso.

Palabras claves: Acetona; Método Bell-Delaware; Porciento de área en exceso; Caída de presión; Evaluación; Intercambiador de calor de tubo y coraza.

NOMENCLATURE

a'_t	Flow area of a tube	m^2
a_t	Total flow area of the tubes	m^2
A	Baffle central angle	º
A_{calc}	Calculated heat transfer area	m^2
A_{real}	Real area of the heat exchanger	m^2
$\%A_{exc}$	Percent excess area	%
B	Baffle spacing	m
BC	Baffle cut	m
c	Clearance between tubes	m
Cp	Specific heat	J/kg.K
d_e	External diameter of the tube	m
d_i	Internal diameter of the tube	m
d_N	Nominal diameter of tubes	in
D_B	Baffle diameter	m
D_{BT}	Diameter of the baffle hole	m
D_s	Internal shell diameter	m
f	Friction factor	-
F_{BP}	Bypass fraction	-
F_t	LMTD correction factor	-
G	Mass velocity	$kg/m^2.s$
h	Film heat transfer coefficient	$W/m^2.K$
h_L	Film heat transfer coefficient of a tube bank with leakage	$W/m^2.K$
h_{LN}	Film heat transfer coefficient of a tube bank without leakage	$W/m^2.K$
h_0	Film heat transfer coefficient for the shell-side fluid	$W/m^2.K$
h_{to}	Film heat transfer coefficient of the tube-side fluid referred to the external area	$W/m^2.K$
j	Colburn coefficient	-
k	Thermal conductivity	W/m.K
L_t	Tube length	m
m	Mass flowrate	kg/s
LMTD	Logarithmic Mean Temperature Difference	ºC
n	Number of tube passes	-
n_s	Number of shell passes	-
N	Number of tubes	-
N_B	Number of baffles	-
N_{BT}	Number of tubes crossing the baffle	-
N_c	Number of tube rows between baffles borders	-
N_{CL}	Number of tubes in the central row	-
N_s	Number of pairs of sealing strips	-
N_{tw}	Number of tubes in window	-
N_W	Effective number of tube rows in a window	-
Pr	Prandtl number	-
P_t	Tube pitch	m
P_x	Parameter for F_t calculation	-

Δp	Pressure drop	Pa
Δp_{BP}	Pressure drop in a cross-flow section without leakage	Pa
Δp_{L}	Pressure drop for an exchanger section considering leakage	Pa
Δp_{NL}	Pressure drop for an exchanger section with no leakage	Pa
Δp_{r}	Pressure drop in return headers	Pa
Δp_{s}	Total pressure drop of the shell-side fluid through the heat exchanger	Pa
Δp_{t}	Pressure drop in straight tube length	Pa
Δp_{T}	Total pressure drop of the tube-side fluid through the heat exchanger	Pa
Δp_{W}	Pressure drop through window	Pa
ΔP_m	Maximum allowable pressure drop	Pa
Q	Heat exchanged	W
r	Parameter	-
R	Fouling resistance	$m^2.K/W$
R	Parameter for F_t calculation	-
Re	Reynolds number	-
S	Parameter for F_t calculation	-
S_L	Total leakage area	m^2
S_m	Cross-flow area	m^2
S_{SB}	Baffle-shell leakage area	m^2
S_{TB}	Tube-baffle leakage area	m^2
S_W	Flow area at window	m^2
t	Temperature of the cold fluid	ºC
T	Temperature of the hot fluid	ºC
$\bar{t}$	Mean temperature of the cold stream	ºC
$\overline{T}$	Mean temperature of the hot stream	ºC
ΔT	Effective mean temperature difference between both streams	ºC
U	Overall heat transfer coefficient	$W/m^2.K$
V_z	Mean velocity	m/s

Greek symbols

α	Constant	-
μ	Viscosity	Pa.s
ρ	Density	Kg/m^3
ϕ	Viscosity correction factor	-
λ_h	Constant for film heat transfer coefficient calculation	-
$\lambda_{\Delta p}$	Constant for pressure drop calculation	
ξ_h	Bypass correction coefficient for film heat transfer coefficient calculation	-
$\xi_{\Delta p}$	Bypass correction coefficient for pressure drop calculation	-
ϕ	Correction factor	-

Subscripts

1	Inlet
2	Outlet
c	Cold fluid
h	Hot fluid

w	At the tube wall temperature
s	Shell-side fluid
t	Tube-side fluid

1. INTRODUCTION

Heat exchangers (HE) are one of the most important devices of mechanical systems in modern society. Most industrial processes involve the transfer of heat and more often, it is required that the heat transfer process be controlled (Kulkarni *et al*., 2014).

A HE is a device used to transfer heat between a solid object and a fluid, or between two or more fluids. The fluids may be separated by a solid wall to prevent mixing or they may be in direct contact (Sharma & Dewangan, 2017).

Heat exchangers are important heat transfer apparatuses in food industry, environmental protection, electric power generation, (Shinde *et al*., 2012), refrigeration, air conditioning, chemical and petrochemical plants, petroleum refineries, natural gas processing and sewage treatment (Sharma & Dewangan, 2017).

The most common type of heat exchangers utilized in industrial fields is the shell and tube heat exchanger according to their low costs, structural simplicity, and design flexibility (Yousufuddin, 2018).

Among different types of heat exchangers, shell and tube heat exchangers (STHXs) are relatively easy to manufacture and have multipurpose application possibilities for gaseous as well as liquid media in large temperature and pressure ranges (Bichkar *et al*., 2018).

STHX is an indirect contact type heat exchanger (Bayram & Sevilgen, 2017), and is widely used in many industrial areas. More than 35 - 40% of heat exchangers are of this type due to their robust geometry construction, easy maintenance, possible upgrades (Zhang *et al*., 2010), easy cleaning, lower cost and more flexible adaptability compared with other heat exchanger (Sharma & Dewangan, 2017), as well as long-term usage, operational capacity in wide temperature and pressure intervals, and higher reliability (Alperen *et al*., 2019).

In the shell and tube heat exchanger there are two main fluid flows, one fluid flow that goes through the tubes while the other flows on the shell side. Heat is transferred from one fluid to the other through the tube walls, either from tube side to shell side or vice versa. The fluids can either be liquids or gases on either the shell or the tube side. In order to transfer heat efficiently, a large heat transfer area is used, leading to the use of many tubes. This is an efficient way to use energy and avoid wastage of thermal energy (Bichkar et al., 2018). Figure 1 shows one of the simplest configurations of a STHX. This is a one-shell and two-pass through the tube's configuration, also referred to as STHX (1-2) (Honrubia *et al*., 2021).

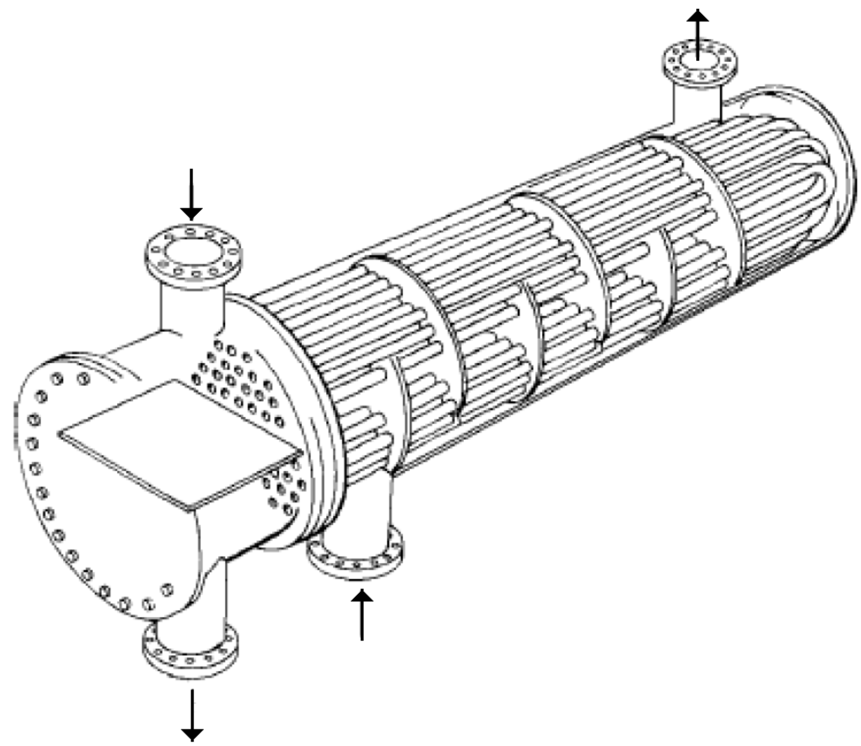

Figure 1. Shell and tube heat exchanger type 1-2

Source: Adapted from (Honrubia et al., 2021)

In a baffled STHX, the baffles are used for different reasons such as providing support for tubes, obtaining a desirable fluid velocity to be maintained for the shell-side fluid flow, and preventing the tubes from vibrating. They also direct the shell side flow to enhance the heat transfer coefficient but their usage conversely produces an increased pressure drop (Bayram & Sevilgen, 2017).

The most-commonly used baffle is the segmental baffle, which forces the shell-side fluid going through in a zigzag manner, hence, improves the heat transfer with a large pressure drop penalty. This type of heat exchanger has been well-developed and probably is still the most-commonly used type of the STHXs (Zhang *et al*., 2010).

The usual problems in heat exchanger design are rating and sizing. The rating problem consists in evaluating the thermo-hydraulic performance of a fully specified exchanger. The rating program determines the heat transfer rate and the fluid outlet temperatures for prescribed fluid flow rates, inlet temperatures, and the pressure drop for an existing heat exchanger; therefore the heat transfer surface area and the flow passage dimensions are available. Thus, rating is the computational process in which the inlet flow rates and temperatures, the fluid properties, and the heat exchanger parameters are taken as input and the outlet temperatures and thermal duty (if the exchanger length is specified) or the required length/area of the heat exchanger are calculated as output. In either case, the pressure drop of each stream will also be calculated.

In the STHX, the thermo-hydraulic performance of the HE can be evaluated in detail by calculating heat transfer coefficients (HTCs) and pressure losses (PLs) separately for each of the fluids flowing through the shell and through the tubes. Tube side calculations can be carried out with commonly known methods such as internal flow calculations. However, the direction of the shell flow changes due to the usage of baffle plates, and the use of baffle plates requires specially developed methods, since the flow field on the shell side is the area between the small diameter tubes. Two of the most commonly used methods for HTC and PL calculations are known in the literature as the Kern method and the Bell-Delaware method (Alperen *et al*., 2019).

In 1950 at Delaware University, a research program in shell-and-tube heat exchangers was developed with the sponsorship of the Tubular Exchanger Manufacturers Association (TEMA) and the American Society of Mechanical Engineers (ASME). Many researchers worked in this program and published their conclusions in several reports over that decade. The final synthesis of the study was published as a heat exchanger design method by Kenneth Bell (Bell, 1963) in 1963. This is now known as Bell or Delaware method (Cao, 2010).

This method accounts for leakage and bypass flow paths on the shell side of the exchanger and, consequently, provides more realistic estimates of the shell-side pressure drops and heat transfer coefficients than the Kern method (Serna *et al*., 2007).

Several authors have applied the Bell-Delaware method to design or evaluate STHXs. In this sense (Serna & Jiménez, 2005) reported an analytical expression that relates the pressure drop, the exchanger area and the film heat transfer coefficient for the shell side of a STHX, where the equation obtained was developed based on the Bell–Delaware method, thus aiding significantly in tasks such as heat exchanger design and optimization procedures. Also, (Serna *et al*., 2007) presented an improved methodology for generating feasible regions for STHX design, taking into account geometric and operational constraints. The approach used is based on the Bell-Delaware method to describe the shell-side flow with no simplification; this approach, therefore, can incorporate the entire range of geometric parameters of practical interest. Similarly, (Zhang *et al*., 2010) developed a method for the design and rating of STHX with helical baffles, based on the public literatures and the Bell–Delaware method for STHX with segmental baffles. Likewise (Shinde et al., 2012) modified the existing Bell-Delaware method used for conventional heat exchanger, taking into consideration the helical geometry of Helixchanger. These authors carried out thermal analysis to study the impacts of various baffle inclination angles on fluid flow and heat transfer of heat exchangers with helical baffles. In other study, (Kulkarni *et al*., 2014) accomplished a comparative analysis of a water to water STHX to study and analyze the heat transfer coefficient and pressure drops for different mass flow rates and inlet and outlet temperatures, using Kern, Bell and Bell-Delaware methods. Moreover, (Toledo *et al*., 2014) analyzed and upgraded the Bell-Delaware method using correction factors which take into account the undesirable currents of the mean flow, in order to explore different design alternatives to find the optimal solution to each proposed problem. The results of this work was a simple software that can perform calculations with the introduction of parameters depending only on the geometry of the heat exchanger, temperature and fluid characteristics, thus eliminating the human errors and increasing the calculations speed and accuracy. In addition, (Bayram & Sevilgen, 2017) used both numerical and theoretical analysis to investigate the effect of the variable baffle spacing on the thermal characteristics of a small STHX. The numerical study was performed by using a three dimensional computational fluid dynamics (CFD) method. The computations were performed under steady-state conditions, while the theoretical calculations were run using the Bell-Delaware and Kern methods. Furthermore, (Alperen *et al*., 2019) compared the results obtained from HTRI Xchanger software, as well as from Kern and Bell-Delaware methods, with experimental data obtained during the thermo-hydraulic evaluation of an ASME E type STHX. Finally, (Gonçalves *et al*., 2019) presented a rigorous reformulation of the Bell-Delaware model for the design optimization of STHX, in order to obtain a linear model. The linear character of the formulation allows the identification of the global optimum, even using conventional optimization algorithms. The proposed mixed-integer linear programming formulation with the Bell-Delaware method is able to identify feasible solutions for the design of heat exchangers at a lower cost than those obtained through conventional design formulations in the literature.

Several other authors (John et al., 2016; Jamil et al., 2020; Omar et al., 2021) have employed the Bell-Delaware approach for STHX design or rating.

In a Cuban chemical processing plant there is a need to cool a liquid stream of pure acetone using chilled water, and for what an existing shell and tube heat exchanger is proposed for this heat exchange service. Consequently, in the present work the thermo-hydraulic rating of this shell and tube heat exchanger is carried out by applying the calculation methodology known as Bell-Delaware approach. Several important parameters of the heat exchanger such as the percent excess area, overall heat transfer coefficient, heat transfer area and the pressure drop of both streams are calculated.

2. MATERIALS AND METHODS

2.1. Problem definition

A proposed shell and tube heat exchanger needs to be evaluated to cool 50,000 kg/h of an acetone stream from 80 ºC to 30 ºC, using 47,000 kg/h of chilled water at an inlet temperature of 2 ºC. The proposed heat exchanger is of TEMA L type, it is equipped with 492 tubes of ¾ in BWG 16 arranged in a triangular pattern, and with a tube pitch of 0.0254 m. In addition, it has 17 baffles with a spacing of 0.234 m, while the tube length is 5.2 m and the shell internal diameter is 0.635 m. The flow arrangement of the exchanger will be countercurrent, and has one pass through the shell and two passes through the tubes, that is, it is of 1-2 type. The rest of the physical parameters of the proposed heat exchanger are presented below, whose geometric dimensions and tubes layout are shown in Figure 2.

- Number of tubes in window (N_{tw}): 75.
- Number of tubes in the central row (N_{CL}): 21.
- Number of tubes crossing the baffle (NBT): 339.
- Baffle diameter (D_B): 0.6043 m.
- Baffle cut (BC): 0.1522 m.
- Baffle central angle (A): 120º
- Diameter of the baffle hole (D_{BT}): 0.0198 m.
- Clearance between tubes (c): 0.0640 m.
- Number of pairs of sealing strips (N_s): 2.
- Number of tube rows between baffles borders (N_c): 13.

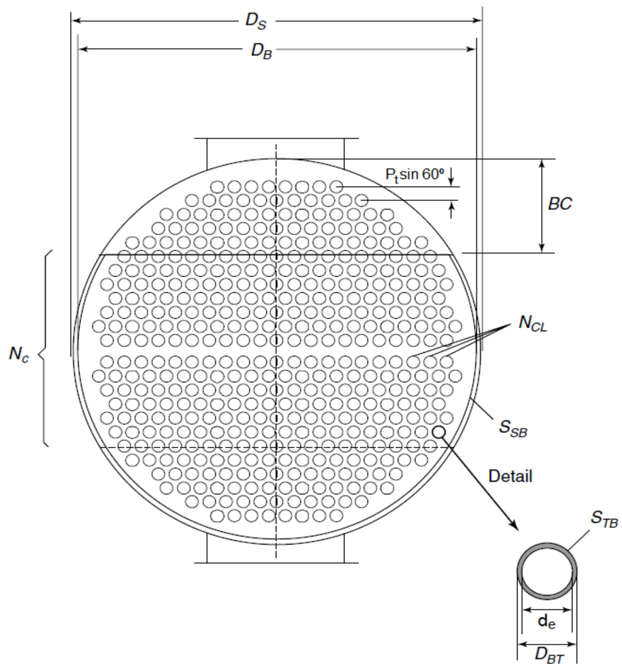

Figure 2. Geometric dimensions and tubes layout of the proposed shell and tube heat exchanger.
Source: Adapted from (Cao, 2010)

It is desired that the pressure drop of both the acetone and chilled water streams does not exceed 2,000 Pa and 4,000 Pa, respectively, and that the percent excess area does not surpass 25%.

2.2. Calculation methodology

To perform the rating of the proposed shell and tube heat exchanger, the calculation methodology known as Bell-Delaware was used, whose equations and correlations are published in (Cao, 2010). By means of this methodology, key parameters of the heat exchanger will be determined, such as the percent excess area and the pressure drop of both fluids, while the results obtained from these parameters will be compared with the maximum limits established by the process, to determine if the proposed heat exchanger is feasible to use for the required heat transfer service.

Percent excess area:

Step 1. Definition of the initial data of the streams involved:

Table 1 shows the initial data defined for each of the streams involved in the heat exchange system.

Table 1. Initial data required for each stream involved.

Parameter	Unit	Hot fluid (Acetone)	Cold fluid (Water)
Mass flowrate	kg/s	m_h	m_c
Inlet temperature	ºC	T_1	t_1
Outlet temperature	ºC	T_2	-
Maximum allowable pressure drop	Pa	$\Delta P_{m(h)}$	$\Delta P_{m(c)}$
Fouling resistance	m^2.K/W	R_h	R_c

Source: Own elaboration.

Step 2. Specification of the initial data of the proposed heat exchanger:

Table 2 presents the initial data that needs to be specified for the evaluated shell and tube heat exchanger.

Table 2. Required initial parameters for the proposed heat exchanger.

Parameter	Unit	Symbol
Tube nominal diameter	in	d_N
Tube pattern	-	$\Delta/\square$
Tube pitch	m	P_t
Internal shell diameter	m	D_s
Number of tubes	-	N
Number of tube passes	-	n
Number of shell passes	-	n_s
Tube length	m	L_t
Baffle spacing	m	B
Number of baffles	-	N_B
Number of tubes in window	-	N_{tw}
Number of tubes in the central row	-	N_{CL}
Number of tubes crossing the baffle	-	N_{BT}
Baffle diameter	m	D_B
Baffle cut	m	BC
Baffle central angle	º	A
Diameter of the baffle hole	m	D_{BT}
Clearance between tubes	m	c
Number of pairs of sealing strips	-	N_s
Number of tube rows between baffles borders	-	N_c

Source: Own elaboration.

Step 3. Definition of the internal and external diameter of the tubes depending on the nominal diameter and the caliber:

Table 3 represents the external and internal diameters that must be defined for the tubes depending on their nominal diameter and caliber.

Table 3. External and internal diameters to be defind for the tubes depending on their nominal diameter and caliber.

Parameter	Unit	Symbol
Internal diameter of a tube	m	d_i
External diameter of a tube	m	d_e

Source: Own elaboration.

Step 4. Average temperature of the hot stream ($\overline{T}$):

$$\overline{T} = \frac{T_1 + T_2}{2} \tag{1}$$

Step 5. Specific heat of the hot stream (Cp_h) at the average temperature determined in the previous step.

Step 6. Heat exchanged (Q):

Taking into account the initial data for the hot stream (acetone):

$$Q = m_h \cdot Cp_h \cdot (T_1 - T_2) \tag{2}$$

Step 7. Specific heat of the cold stream (Cp_c) at its inlet temperature t_1.

Step 8. Outlet temperature of the cold stream (t_2):

$$t_2 = t_1 + \frac{Q}{m_c \cdot Cp_c} \tag{3}$$

Step 9. Average temperature of the cold stream ($\bar{t}$):

$$\bar{t} = \frac{t_1 + t_2}{2} \tag{4}$$

Step 10. Physical properties for both fluids at the average temperature determined in the previous steps:

Table 4 displays the physical properties required by the rating methodology for both fluids at the average temperature determined in the previous steps.

Table 4. Physical properties required for both fluids.

Physical property	Unit	Hot fluid (Acetone)	Cold fluid (Water)
Density	kg/m^3	ρ_h	ρ_c
Viscosity	Pa.s	μ_h	μ_c
Thermal conductivity	W/m.K	k_h	k_c

Source: Own elaboration.

Step 11. Fluids allocation inside the shell and tube heat exchanger.

Step 12. Flow area of a tube (a_t'):

$$a_t' = \frac{\pi \cdot d_i^2}{4} \tag{5}$$

Step 13. Total flow area for the tube-side fluid (a_t):

$$a_t = \frac{a_t' \cdot N}{n} \tag{6}$$

Step 14. Mass velocity of the tube-side fluid (G_t):

$$G_t = \frac{m_t}{a_t} \tag{7}$$

Step 15. Reynolds number of the tube-side fluid (Re_t):

$$\mathrm{Re}_t = \frac{d_i \cdot G_t}{\mu_t} \tag{8}$$

Step 16. Prandtl number of the tube-side fluid (Pr_t):

$$\mathrm{Pr}_t = \frac{Cp_t \cdot \mu_t}{k_t} \tag{9}$$

Step 17. Film heat transfer coefficient of the tube-side fluid (h_t):

- For laminar flow ($\mathrm{Re}_t \leq 2{,}100$):

$$h_t = 1.86 \cdot \frac{k_t}{d_i} \cdot \left(\mathrm{Re}_t \cdot \mathrm{Pr}_t \cdot \frac{d_i}{L_t} \right)^{0.33} \cdot \left(\frac{\mu_t}{\mu_{tw}} \right)^{0.14} \tag{10}$$

- For transition zone ($2{,}100 < \mathrm{Re}_t < 10{,}000$):

$$h_t = 0.116 \cdot Cp_t \cdot G_t \cdot \left(\frac{\mathrm{Re}_t^{0.66} - 125}{\mathrm{Re}_t} \right) \cdot \left[1 + \left(\frac{d_i}{L_t} \right)^{0.66} \right] \cdot \mathrm{Pr}_t^{-0.66} \cdot \left(\frac{\mu_t}{\mu_{tw}} \right)^{0.14} \tag{11}$$

- For turbulent region ($\mathrm{Re}_t \geq 10{,}000$):

$$h_t = 0.023 \cdot \frac{k_t}{d_i} \cdot \mathrm{Re}_t^{0.8} \cdot \mathrm{Pr}_t^{0.33} \cdot \left(\frac{\mu_t}{\mu_{tw}} \right)^{0.14} \tag{12}$$

Step 18. Film heat transfer coefficient of the tube-side fluid referred to the external area (h_{to}):

$$h_{to} = h_t \cdot \frac{d_i}{d_e} \tag{13}$$

Step 19. Cross-flow area (S_m):

$$S_m = \left(D_s - N_{CL} \cdot d_e \right) \cdot B \tag{14}$$

Step 20. Mass velocity of the shell-side fluid (G_s):

$$G_s = \frac{m_s}{S_m} \tag{15}$$

Step 21. Reynolds number of the shell-side fluid (Re_s):

$$\mathrm{Re}_s = \frac{d_e \cdot G_s}{\mu_s} \tag{16}$$

Step 22. Colburn coefficient (j).

Step 23. Bypass fraction (F_{BP}):

$$F_{BP} = \frac{\left[D_s - \left(N_{CL} - 1\right) \cdot P_t - d_e\right] \cdot B}{S_m} \tag{17}$$

Step 24. Constant λ_h

Step 25. Bypass correction coefficient for film heat transfer coefficient calculation (ξ_h):

$$\xi_h = \exp\left[-\lambda_h \cdot F_{BP} \cdot \left(1 - \sqrt[3]{\frac{2 \cdot N_s}{N_c}} \right) \right] \tag{18}$$

Step 26. Parameter r:

$$r = \frac{N_{tw}}{N} \tag{19}$$

Step 27. Flow area at window (S_W):

$$S_W = \frac{\pi \cdot D_s^2}{4} \cdot \frac{A}{360} - \left(\sin \frac{A}{2} \right) \cdot \left(\frac{D_s}{2} \right) \cdot \left(0.5 \cdot D_s - BC \right) - \frac{\pi \cdot N_{tw} \cdot d_e^2}{4} \tag{20}$$

Step 28. Correction factor (ϕ):

$$\phi = 1 - r + 0.524 \cdot r^{0.32} \cdot \left(\frac{S_m}{S_w} \right)^{0.03} \tag{21}$$

Step 29. Tube-baffle leakage area (S_{TB}):

$$S_{TB} = N_{BT} \cdot \frac{\pi}{4} \cdot \left(D_{BT}^2 - d_e^2\right) \tag{22}$$

Step 30. Baffle-shell leakage area (S_{SB}):

$$S_{SB} = \frac{360 - A}{360} \cdot \frac{\pi}{4} \cdot \left(D_s^2 - D_B^2\right) \tag{23}$$

Step 31. Total leakage area (S_L):

$$S_L = S_{TB} + S_{SB} \tag{24}$$

Step 32. Factor $\left(1 - \frac{h_L}{h_{NL}}\right)_0$:

$$\left(1 - \frac{h_L}{h_{NL}}\right)_0 = 0.45 \cdot \frac{S_L}{S_m} + 0.1 \cdot \left[1 - \exp\left(-30 \cdot \frac{S_L}{S_m}\right)\right] \tag{25}$$

Step 33. Factor $\left(1 - \frac{h_L}{h_{NL}}\right)$:

$$\left(1 - \frac{h_L}{h_{NL}}\right) = \left(1 - \frac{h_L}{h_{NL}}\right)_0 \cdot \frac{S_{TB} + 2 \cdot S_{SB}}{S_L} \tag{26}$$

Step 34. Factor $\left(\frac{h_L}{h_{NL}}\right)$:

$$\left(\frac{h_L}{h_{NL}}\right) = 1 - \left(1 - \frac{h_L}{h_{NL}}\right) \tag{27}$$

Step 35. Prandtl number of the shell-side fluid (Pr_s):

$$\mathrm{Pr}_s = \frac{Cp_s \cdot \mu_s}{k_s} \tag{28}$$

Step 36. Film heat transfer coefficient for the shell-side fluid (h_0):

$$h_0 = \left[j \cdot Cp_s \cdot G_s \cdot \mathrm{Pr}_s^{-2/3} \cdot \left(\frac{\mu_{sw}}{\mu_s} \right)^{-0.14} \right] \cdot \phi \cdot \xi_h \cdot \left(\frac{h_L}{h_{NL}} \right) \tag{29}$$

Step 37. Overall heat transfer coefficient (U):

$$U = \left(\frac{1}{h_{to}} + \frac{1}{h_o} + R_h + R_c \right)^{-1} \tag{30}$$

Step 38. Logarithmic mean temperature difference (LMTD):

$$LMTD = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln \frac{(T_1 - t_2)}{(T_2 - t_1)}} \tag{31}$$

Step 39. Parameter R:

$$R = \frac{T_1 - T_2}{t_2 - t_1} \tag{32}$$

Step 40. Parameter S:

$$S = \frac{t_2 - t_1}{T_1 - t_1} \tag{33}$$

Step 41. Parameter P_x:

$$P_x = \frac{1 - \left(\frac{R \cdot S - 1}{S - 1} \right)^{1/n_s}}{R - \left(\frac{R \cdot S - 1}{S - 1} \right)^{1/n_s}} \tag{34}$$

Step 42. LMTD correction factor (F_t):

$$F_t = \frac{\sqrt{R^2+1}}{R-1} \cdot \frac{\ln\left[\frac{(1-P_x)}{(1-R\cdot P_x)}\right]}{\ln\left[\frac{\left(\frac{2}{P_x}\right)-1-R+\sqrt{R^2+1}}{\left(\frac{2}{P_x}\right)-1-R-\sqrt{R^2+1}}\right]} \tag{35}$$

Step 43. Effective mean temperature difference (ΔT):

$$\Delta T = LMTD \cdot F_t \tag{36}$$

Step 44. Calculated heat transfer area (A_{calc}):

$$A_{calc} = \frac{Q}{U \cdot \Delta T} \tag{37}$$

Step 45. Real heat transfer area (A_{real}):

$$A_{real} = \pi \cdot d_e \cdot N \cdot L_t \tag{38}$$

Step 46. Percent excess area ($\%A_{exc}$):

$$\%A_{exc} = \frac{A_{real} - A_{calc}}{A_{calc}} \tag{39}$$

Pressure drop:

Step 47. Friction factor of the tube-side fluid (f_t):

Laminar flow ($\mathrm{Re}_t \leq 2100$):

$$f_t = \frac{16}{\mathrm{Re}_t} \tag{40}$$

Turbulent flow ($\mathrm{Re}_t > 2100$):

$$f_t = 1.2 \cdot \left(0.0014 + 0.125 \cdot \mathrm{Re}_t^{-0.32}\right) \tag{41}$$

Step 48. Pressure drop of the tube-side fluid in straight tube length (Δp_t):

$$\Delta p_t = 4 \cdot f_t \cdot n \cdot \frac{L_t}{d_i} \cdot \frac{G_t^2}{2 \cdot \rho_t} \cdot \left(\frac{\mu_t}{\mu_{tw}} \right)^{\alpha} \tag{42}$$

Where α is – 0.25 for laminar regime and – 0.14 for turbulent regime.

Step 49. Pressure drop of the tube-side fluid in return headers (Δp_r):

$$\Delta p_r = 4 \cdot n \cdot \frac{G_t^2}{2 \cdot \rho_t} \tag{43}$$

Step 50. Total pressure drop for the tube-side fluid through the heat exchanger (Δp_T):

$$\Delta p_T = \Delta p_t + \Delta p_r \tag{44}$$

Step 51. Friction factor of the shell-side fluid (f_s):

Step 52. Constant $\lambda_{\Delta p}$

Step 53. Bypass correction coefficient for pressure drop calculation ($\xi_{\Delta p}$):

$$\xi_{\Delta p} = \exp\left\{ - \lambda_{\Delta p} \cdot F_{BP} \cdot \left[1 - \left(\frac{2 \cdot N_s}{N_c} \right)^{1/3} \right] \right\} \tag{45}$$

Step 54. Pressure drop in a cross-flow section without leakage (Δp_{BP}):

$$\Delta p_{BP} = \frac{4 \cdot f_s \cdot N_c \cdot G_s^2 \cdot \left(\frac{\mu_{sw}}{\mu_s} \right)^{0.14} \cdot \xi_{\Delta p}}{2 \cdot \rho_s} \tag{46}$$

Step 55. Mean velocity (V_z):

$$V_z = \frac{m_s}{\rho_s \cdot \sqrt{S_m \cdot S_W}} \tag{47}$$

Step 56. Effective number of tube rows in a window (N_W):

$$N_W = 0.8 \cdot \frac{BC}{P_t \cdot \sin 60°} \tag{48}$$

Step 57. Pressure drop through window (Δp_W):

$$\Delta p_W = (2 + 0.6 \cdot N_W) \cdot \frac{\rho_s \cdot V_z^2}{2} \tag{49}$$

Step 58. Factor $\left(1 - \frac{\Delta p_L}{\Delta p_{NL}}\right)_0$:

$$\left(1 - \frac{\Delta p_L}{\Delta p_{NL}}\right)_0 = 0.57 \cdot \frac{S_L}{S_m} + 0.27 \cdot \left[1 - \exp\left(-20 \cdot \frac{S_L}{S_m}\right)\right] \tag{50}$$

Step 59. Factor $\left(1 - \frac{\Delta p_L}{\Delta p_{NL}}\right)$:

$$\left(1 - \frac{\Delta p_L}{\Delta p_{NL}}\right) = \left(1 - \frac{\Delta p_L}{\Delta p_{NL}}\right)_0 \cdot \frac{S_{TB} + 2 \cdot S_{SB}}{S_L} \tag{51}$$

Step 60. Factor $\left(\frac{\Delta p_L}{\Delta p_{NL}}\right)$:

$$\left(\frac{\Delta p_L}{\Delta p_{NL}}\right) = 1 - \left(1 - \frac{\Delta p_L}{\Delta p_{NL}}\right) \tag{52}$$

Step 61. Total pressure drop of the shell-side fluid through the heat exchanger (Δp_s):

$$\Delta p_s = 2 \cdot \Delta p_{BP} \cdot \left(1 + \frac{N_W}{N_c}\right) + \left[(N_B - 1) \cdot \Delta p_{BP} + N_B \cdot \Delta p_W\right] \cdot \frac{\Delta p_L}{\Delta p_{NL}} \tag{53}$$

3. RESULTS AND DISCUSSION

3.1. Percent excess area

Shown below are the results obtained stepwise to determine the percent excess area of the proposed shell and tube heat exchanger.

Step 1. Definition of the initial data of the streams involved:

Table 5 presents the values of the initial data defined for each fluid stream involved in the heat exchange system.

Table 5. Initial data defined for each fluid stream involved in the heat exchange system:

Parameter	Unit	Hot fluid (Acetone)	Cold fluid (Water)
Mass flowrate	kg/s	13.89	13.06
Inlet temperature	ºC	80	2
Outlet temperature	ºC	30	-
Maximum allowable pressure drop	Pa	2,000	4,000
Fouling resistance†	$m^2.K/W$	0.0002	0.0003

† Reported by (Sinnott, 2005)
Source: Own elaboration.

Step 2. Specification of the initial data of the proposed shell and tube heat exchanger:

Table 6 shows the specified initial data for the proposed shell and tube heat exchanger.

Table 6. Initial data specification for the proposed shell and tube heat exchanger:

Parameter	Unit	Symbol	Value
Tube nominal diameter	in.	d_N	¾
Tube pattern	-	Δ	Triangular
Tube pitch	m	P_t	0.0254
Internal shell diameter	m	D_s	0.635
Number of tubes	-	N	492
Number of tube passes	-	n	2
Number of Shell passes	-	n_s	1
Tube length	m	L_t	5.0
Central baffle spacing	m	B	0.234
Number of baffles	-	N_B	17
Number of tubes in window	-	N_{tw}	75
Number of tubes in the central row	-	N_{CL}	21
Number of tubes crossing the baffle	-	N_{BT}	339
Baffle diameter	m	D_B	0.630
Baffle cut	m	BC	0.1522
Baffle central angle	º	A	120
Diameter of the baffle hole	m	D_{BT}	0.0198
Clearance between tubes	m	c	0.0640
Number of pairs of sealing strips	-	N_s	2
Number of tube rows between baffles borders	-	N_c	13

Source: Own elaboration.

Step 3. Definition of the internal and external diameter of the tubes depending on the nominal diameter and the caliber:

Table 7 indicates the external and internal diameters that must be defined for the tubes depending on their nominal diameter and caliber.

Table 7. External and internal diameters to be specified for the tubes depending on their nominal diameter and caliber:

Parameter	Unit	Symbol	Value
Internal diameter of a tube	m	d_i	0.0157
External diameter of a tube	m	d_e	0.0190

Source: Own elaboration.

Step 4. Average temperature of acetone ($\bar{T}$):

$$\bar{T} = \frac{T_1 + T_2}{2} = \frac{80 + 30}{2} = 55\,^{\circ}C$$

Step 5. Specific heat of acetone (Cp_h) at the mean temperature determined in the previous step.

According to (Green & Southard, 2019), the specific heat of acetone at $\bar{T}$ = 55 ºC is Cp_h = 2279,88 J/kg.K.

Step 6. Heat exchanged (Q):

Taking the data of the hot fluid (acetone):

$$Q = m_h \cdot Cp_h \cdot (T_1 - T_2) = 13.89 \cdot 2{,}279.88 \cdot (80 - 30)$$
$$Q = 1{,}583{,}376.66\, W$$

Step 7. Specific heat of water (Cp_c) at the inlet temperature t_1.

According to (Green & Southard, 2019), the specific heat of water at the inlet temperature t_1 = 2 ºC is Cp_c = 4,221,93 J/kg.K.

Step 8. Outlet temperature of chilled water (t_2):

$$t_2 = t_1 + \frac{Q}{m_c \cdot Cp_c} = 2 + \frac{1{,}583{,}376.66}{13{,}06 \cdot 4{,}221.93} = 28.72\,^{\circ}C$$

Step 9. Average temperature of chilled water ($\bar{t}$):

$$\bar{t} = \frac{t_1 + t_2}{2} = \frac{2 + 28.72}{2} = 15.36\,^{\circ}C$$

Step 10. Physical properties of both fluids at the average temperature determined in previous steps:

Table 8 establishes the physical properties required for both fluids at the average temperature determined in the previous steps.

Table 8. Physical properties required for both fluids

Physical property	Unit	Hot fluid (Acetone)	Cold fluid (Water)
Density	kg/m^3	751.32	998.87
Viscosity	Pa.s	0.0002374	0.001113
Thermal conductivity	W/m.K	0.1477	0.5937

Source: Own elaboration.

Step 11. Location of the fluids inside the heat exchanger:

Following recommendations and suggestions reported in (Sinnott, 2005), the hot fluid (acetone) will be located inside the tubes, while the chilled water will circulate through the shell.

Table 9 describes the results obtained for the parameters included in steps 12-16.

Table 9. Results of the parameters included in steps 12-16.

Step	Parameter	Symbol	Value	Unit
12	Flow area of a tube	a_t'	0.000193	m^2
13	Total flow area for the tube-side fluid	a_t	0.0475	m^2
14	Mass velocity of acetone	G_t	292.42	$kg/m^2.s$
15	Reynolds number of acetone	Re_t	19,338.64	-
16	Prandtl number of acetone	Pr_t	3.66	-

Source: Own elaboration.

Step 17. Acetone film heat transfer coefficient (h_t):

Since acetone flows under turbulent regime (Re_t > 10,000), equation (12) will be used to determine the film heat transfer coefficient of this fluid. In this case, it will be considered that the factor $\left(\mu_t / \mu_{tw}\right)^{0.14}$ = 1 (Cao, 2010). Then:

$$h_t = 0.023 \cdot \frac{k_t}{d_i} \cdot \mathrm{Re}_t^{0.8} \cdot \mathrm{Pr}_t^{0.33} \cdot \left(\frac{\mu_t}{\mu_{tw}}\right)^{0.14}$$

$$h_t = 0.023 \cdot \frac{0.1477}{0.0157} \cdot (19{,}338.64)^{0.8} \cdot (3.66)^{0.33} \cdot 1$$

$$h_t = 891.64\, W/m^2.K$$

Step 18. Film heat transfer coefficient of the acetone referred to the external area (h_{to}):

$$h_{to} = h_t \cdot \frac{d_i}{d_e} = 891.64 \cdot \frac{0.0157}{0.0190} = 736.77\, W/m^2.K$$

Step 19. Cross-flow area (S_m):

$$S_m = (D_s - N_{CL} \cdot d_e) \cdot B = (0.635 - 21 \cdot 0.0190) \cdot 0.234$$
$$S_m = 0.0552\ m^2$$

Step 20. Mass velocity of the chilled water (G_s):

$$G_s = \frac{m_s}{S_m} = \frac{13.06}{0.0552} = 236.59\ kg/m^2.s$$

Step 21. Reynolds number of the chilled water (Re_s):

$$\mathrm{Re}_s = \frac{d_e \cdot G_s}{\mu_s} = \frac{0.0190 \cdot 236.59}{0.001113} = 4{,}038.82$$

Step 22. Colburn coefficient (j):

As reported by (Cao, 2010), since Re_s > 3,000 and the tube pattern is triangular with a ratio P_t/d_e = 1,33, the Colburn coefficient will be determined by the following equation:

$$j = 0.275 \cdot \mathrm{Re}_s^{-0.38} = 0.275 \cdot (4{,}038.82)^{-0.38} = 0.01172$$

Step 23. Bypass fraction (F_{BP}):

$$F_{BP} = \frac{[D_s - (N_{CL} - 1) \cdot P_t - d_e] \cdot B}{S_m}$$
$$F_{BP} = \frac{[0.635 - (21-1) \cdot 0.0254 - 0.0190] \cdot 0.234}{0.0552}$$
$$F_{BP} = 0.458$$

Step 24. Parameter λ_h:

As stated by (Cao, 2010), the parameter λ_h will has a value of 1,35.

Table 10 shows the results of the parameters included in steps 25-35.

Table 10. Results of the parameters included in steps 25-35.

Step	Parameter	Symbol	Value	Unit
25	Bypass correction coefficient for film heat transfer coefficient calculation	ξ_h	0.818	
26	Parameter r	r	0.152	
27	Flow area at window	S_W	0.0388	m^2

28	Correction factor	ϕ	1.137	
29	Tube-baffle leakage area	S_{TB}	0.00825	m^2
30	Baffle-shell leakage area	S_{SB}	0.00331	m^2
31	Total leakage area	S_L	0.01156	m^2
32	Factor $\left(1-\frac{h_L}{h_{NL}}\right)_0$	$\left(1-\frac{h_L}{h_{NL}}\right)_0$	0.1941	
33	Factor $\left(1-\frac{h_L}{h_{NL}}\right)$	$\left(1-\frac{h_L}{h_{NL}}\right)$	0.2496	
34	Factor $\left(\frac{h_L}{h_{NL}}\right)$	$\left(\frac{h_L}{h_{NL}}\right)$	0.7504	
35	Prandtl number of chilled water	$\Pr_s$	7.91	

Source: Own elaboration.

Step 36. Film heat transfer coefficient for chilled water (h_0):

$$h_0 = \left[j \cdot Cp_s \cdot G_s \cdot \Pr_s^{-2/3} \cdot \left(\frac{\mu_{sw}}{\mu_s} \right)^{-0.14} \right] \cdot \phi \cdot \xi_h \cdot \left(\frac{h_L}{h_{NL}} \right)$$

Where it is considered that the factor $\left(\mu_{sw} / \mu_s \right)^{-0.14}$ = 1 (Cao, 2010). Then:

$$h_0 = \left[0.01172 \cdot 4{,}221.93 \cdot 236.59 \cdot 7.91^{-2/3} \cdot 1 \right] \cdot 1.137 \cdot 0.818 \cdot 0.7504$$

$$h_0 = 2{,}056.48\, W / m^2.K$$

Table 11 displays the results of the parameters included in steps 37-46.

Table 11. Results of the parameters included in steps 37-46.

Step	Parameter	Symbol	Value	Unit
37	Overall heat transfer coefficient	U	426.70	$W/m^2.K$
38	Logarithmic mean temperature difference	LMTD	38.48	ºC
39	Parameter R	R	1.871	
40	Parameter S	S	0.343	
41	Parameter P_x	P_x	0.343	
42	LMTD correction factor	F_t	0.794	
43	Effective mean temperature difference	ΔT	30.55	ºC
44	Calculated heat transfer area	A_{calc}	121.46	m^2
45	Real heat transfer area	A_{real}	146.76	m^2
46	Percent excess area	$\%A_{exc}$	20.83	%

Source: Own elaboration.

3.2. Pressure drop:

Step 47. Acetone friction factor (f_t):

Since Re_t > 2,100 is achieved, equation (41) will be used to determine the friction factor of acetone (f_t). Thus:

$$f_t = 1.2 \cdot \left(0.0014 + 0.125 \cdot \mathrm{Re}_t^{-0.32}\right)$$
$$f_t = 1.2 \cdot \left(0.0014 + 0.125 \cdot 19{,}338.64^{-0.32}\right)$$
$$f_t = 0.00805$$

Step 48. Pressure drop of acetone in straight tube length (Δp_t):

Since acetone flows under a turbulent regime, it will be considered that α = -0.14. Thus equation (42) will be:

$$\Delta p_t = 4 \cdot f_t \cdot n \cdot \frac{L_t}{d_i} \cdot \frac{G_t^2}{2 \cdot \rho_t} \cdot \left(\frac{\mu_t}{\mu_{tw}}\right)^{-0.14}$$

However, it will be considered that the factor $\left(\mu_t / \mu_{tw}\right)^{-0.14}$ = 1 (Cao, 2010). Then:

$$\Delta p_t = 4 \cdot 0.00805 \cdot 2 \cdot \frac{5}{0.0157} \cdot \frac{292.42^2}{2 \cdot 751.32} \cdot 1 = 1167.11\ Pa$$

Step 49. Pressure drop of acetone due to return headers (Δp_r):

$$\Delta p_r = 4 \cdot n \cdot \frac{G_t^2}{2 \cdot \rho_t} = 4 \cdot 2 \cdot \frac{292.42^2}{2 \cdot 751.32} = 455.25\ Pa$$

Step 50. Total pressure drop of acetone through the heat exchanger (Δp_T):

$$\Delta p_T = \Delta p_t + \Delta p_r = 1167.11 + 455.25 = 1622.36\ Pa$$

Step 51. Friction factor of the chilled water (f_s):

Since it is achieved that Re_s > 3,000, and that the tube pattern is triangular with a ratio P_t/d_e = 1.33, the following equation will be used (Cao, 2010):

$$f_s = \exp\left[5.293 - 1.864 \cdot \ln \mathrm{Re}_s + 0.1584 \cdot (\ln \mathrm{Re}_s)^2 - 0.00472 \cdot (\ln \mathrm{Re}_s)^3\right]$$
$$f_s = 0.140$$

Step 52. Parameter $\lambda_{\Delta p}$:

According to (Cao, 2010), the parameter $\lambda_{\Delta p}$ is 4.0.

Table 12 exhibits the results of the parameters included in steps 53-61.

Table 12. Results of the parameters included in steps 53-61.

Step	Parameter	Symbol	Value	Unit
53	Bypass correction coefficient for pressure drop calculation	$\xi_{\Delta p}$	0.552	-
54	Pressure drop in a cross-flow section without leakage	Δp_{BP}	112.59	Pa
55	Mean velocity	V_z	0.28	m/s
56	Effective number of tube rows in a window	N_W	5.54	
57	Pressure drop through window	Δp_W	208.46	Pa
58	Factor $\left(1 - \frac{\Delta p_L}{\Delta p_{NL}}\right)_0$	$\left(1 - \frac{\Delta p_L}{\Delta p_{NL}}\right)_0$	0.385	
59	Factor $\left(1 - \frac{\Delta p_L}{\Delta p_{NL}}\right)$	$\left(1 - \frac{\Delta p_L}{\Delta p_{NL}}\right)$	0.495	
60	Factor $\left(\frac{\Delta p_L}{\Delta p_{NL}}\right)$	$\left(\frac{\Delta p_L}{\Delta p_{NL}}\right)$	0.505	
61	Total pressure drop of the water through the entire unit	Δp_s	3,020.46	Pa

Source: Own elaboration.

Taking into account the results obtained, the mass velocity of acetone (292.42 kg/m^2.s) was 1.24 times greater than the mass velocity of chilled water (236.59 kg/m^2.s). The Reynolds number of acetone had a value of 19,338.64, while the value of this parameter for chilled water was 4,038.82, so both fluids flow under turbulent regime. In this case, the Reynolds number of acetone is 4.79 times higher than the Reynolds number of chilled water, which is mainly due to the higher mass flow rate (13.89 kg/s) and lower viscosity (0.0002374 Pa.s) reported by acetone with respect to the mass flow rate (13.06 kg/s) and viscosity (0.001113 Pa.s) of chilled water.

Likewise, the Prandtl number of acetone was 3.66, while the value of the Prandtl number of chilled water was 7.91. That is, the Prandtl number of chilled water is 2.16 times higher than the Prandtl number of acetone, which is mainly due to the higher value of specific heat and the lower value of viscosity that the chilled water presents with respect to the acetone.

The film heat transfer coefficient of acetone (h_{to}) was 736.77 W/m^2.K, while the value of this parameter for chilled water (h_0) was 2,056.48 W/m^2.K, the last being 2.79 times greater than h_{to} .

The global heat transfer coefficient (U) reached a value of 426.70 W/m^2.K, which is within the range reported by (Sinnott, 2005) of 250-750 W/m^2.K for a heat exchange systems like this, whereas a value of 38.48 ºC was obtained for the LMTD.

The value of the real heat transfer area was 146.76 m^2, which is 1.21 times higher than the value of the calculated heat transfer area (121.46 m^2).

The percent excess area was 20.83%, which is below the limit value established by the process (25%). The pressure drops of both acetone (1,622.36 Pa) and chilled water (3,020.46 Pa) are lower than the maximum permissible value established by the system, which are 2,000 Pa and 4,000 Pa for acetone and chilled water, respectively.

Considering the values obtained from the main rating parameters, it is concluded that the proposed shell and tube heat exchanger can be successfully applied for the required heat transfer service, since the percent excess area is not greater than 25%, and the pressure drops of both fluids do not exceed the maximum allowable limits set by the process.

4. CONCLUSIONS

1. The use of the Bell-Delaware calculation methodology made it possible to successfully evaluate a shell and tube heat exchanger for the cooling of an acetone stream.

2. A heat exchange rate of 1,583,376.66 W, an overall heat transfer coefficient of 426.70 W/m^2.K and a calculated heat transfer area of 121.46 m^2 were obtained.

3. The percent excess area had a value of 20.83%, while the pressure drops of both the acetone and the chilled water reached values of 1,622.36 Pa and 3,020.46 Pa, respectively.

4. The proposed shell and tube heat exchanger can be used for the required heat transfer service, since the percent excess area does not exceed 25%, and the pressure drops of both streams do not exceed the maximum limits established by the process.

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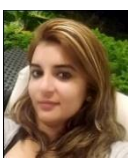
SEMBLANCE OF THE AUTHORS

Amaury Pérez Sánchez: Obtained a degree in Chemical Engineering from the University of Camagüey, Cuba, in 2009, where he is currently an Instructor Professor and assistant researcher. At the moment he is studying a Master in Biotechnology at the Center for Genetic Engineering and Biotechnology in Camagüey. He works in research lines related fundamentally with the design of heat and mass transfer equipment, simulation and optimization of processes and operations in the chemical industry using simulators such as SuperPro Designer® and ChemCAD®, and the techno-economic evaluation. of processes and biotechnological plants.

Alicia Maria Gómez Julián: Obtained a degree in Chemical Engineering from the University of Camagüey, Cuba, in 2021. Currently she works as an instructor professor and researcher at this university, where she is also a PhD student in Chemical Engineering. Her research areas involve the production of bioethanol and cellulolytic enzymes from agro-industrial wastes, with emphasis on the design and evaluation of heat and mass transfer equipment, chemical and fermentative reactions engineering, as well as the optimization and simulation of processes with MATLAB and SuperPro Designer® software.

Nadia Corpas Rodriguez: Obtained a degree in Chemical Engineering from the University of Camagüey, Cuba, in 2017, where she is currently an Instructor Professor. Obtained a Master's Degree in Chemical Processes Analysis from the University of Camagüey in 2022. She works in research lines related with the design of biological reactors and the fermentation of oleaginous microorganisms, the simulation and optimization of processes and operations of the chemical industry through the use of simulators such as SuperPro Designer®, as well as the techno-economic evaluation of chemical plants. Currently she is a PhD student.



Elizabeth Ranero González: Obtained a degree of Chemical Engineering from the University of Camagüey, Cuba, in 2016. She works professionally as Instructor Professor at the University of Camagüey, Cuba. Her research area includes the design and rating of heat and mass transfer equipment and processes, thermodynamic evaluation of chemical processes, and the simulation of biotechnological processes and systems.

Eddy Javier Pérez Sánchez: Obtained a degree in Chemical Engineering from the University of Camagüey, Cuba, in 2016. He works professionally at Empresa de Servicios Automotores S.A., in the Department of Commercial Management. His research lines include the design, rating and/or operation of heat and mass transfer equipment, as well as the simulation of petrochemical and biotechnological processes.