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Thermo-hydraulic design of a gasketed-plate heat exchanger for liquid cow's milk cooling.

Diseño térmico-hidráulico de un intercambiador de calor con placa con junta para refrigeración líquida de leche de vaca.

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Abstract.
Plate heat exchangers offer greater compactness compared to tubular exchangers. The plate configuration enhances heat exchange by creating an extensive and fully compact area that allows for the efficient heat transfer between two fluids. The present paper aims to design, from the thermo-hydraulic point of view, a gasketed-plate heat exchanger to cool down a stream of hot liquid cow's milk using chilled water as coolant. Several important parameters were determined such as the total number of plates (3), the heat load (163.79 kW), the required mass flowrate of chilled water (5,638 kg/h), the required surface area (2.21 m^2) and the overall heat transfer coefficient calculated (2,194.06 $W/m^2.K$). Likewise, the values of the pressure drops for the water (48,558 Pa) and milk (14,720 Pa) streams are below the maximum permissible values set by the process. The designed plate heat exchanger will cost USD $ 2,692 and can be successfully implemented in this heat transfer service from the thermo-hydraulic perspective.

Keywords.
Gasketed-plate heat exchanger; area; overall heat transfer coefficient; pressure drop; purchase cost.

Resumen.
Los intercambiadores de calor de placas ofrecen una mayor compactación comparado con los intercambiadores tubulares. La configuración de la placa mejora el intercambio de calor mediante la creación de un área extensiva y completamente compacta que permite la transferencia de calor eficiente entre dos fluidos. El presente artículo aspira a diseñar, desde el punto de vista térmico-hidráulico, un intercambiador de calor de placas con juntas para enfriar una corriente de leche de vaca liquida caliente usando agua fría como agente de enfriamiento. Varios parámetros importantes fueron determinados tales como el número total de placas (3), la carga de calor (163,79 kW), el caudal másico requerido de agua fría (5 638 kg/h), el área superficial requerida (2.21 m^2) y el coeficiente global de transferencia de calor calculado (2 194,06 $W/m^2.K$). Asimismo, los valores de las caídas de presión de las corrientes de agua (48 558 Pa) y la leche (14 720 Pa) están por debajo de los valores máximos permisibles fijados por el proceso. El intercambiador de placas diseñado costará USD $ 2 692 y puede ser implementado satisfactoriamente en este servicio de transferencia de calor desde la perspectiva térmico-hidráulica.

Palabras clave.
Intercambiador de calor de placas con juntas; área; coeficiente global de transferencia de calor; caída de presión; costo de adquisición.

1. Introduction

Heat exchangers (HX) consist of devices designed to transfer thermal energy between two fluids as a result of a temperature difference. The primary categories of HX are divided based on their structural geometries, which include tubular, plate, and extended surface types [1].

A plate heat exchanger (PHE) is a compact type of heat exchanger that utilizes multiple thin plates for transferring heat between two fluids. There are primarily four types of PHE: gasketed, brazed, welded, and semi-welded. The gasketed or plate-and-frame heat exchanger is composed essentially by of a series of thin rectangular plates bordered by gaskets and secured together within a frame. Initially designed for milk pasteurization in 1923, plate heat exchangers are now widely utilized in various industries, including chemicals, petroleum, HVAC systems, refrigeration, dairy production, pharmaceuticals, beverages, liquid food processing, and health care. This widespread use arises from the distinct benefits offered by PHEs, like adaptable thermal configurations (where plates can be easily added or removed to adjust for varying thermal requirements), simplicity of cleaning necessary for maintaining high hygiene standards, effective temperature regulation (essential for cryogenic uses), and improved heat transfer efficiency [2]. Similarly, plate heat exchangers are preferred for their high surface area relative to volume and superior heat transfer rates [3].

A typical PHE is made up of a set of corrugated plates designed to enhance heat transfer, featuring gaskets positioned in a way that seals off a pathway between the plates when they are compressed within a framework. These pathways enable fluids, which can enter from the same or opposite directions within the apparatus, to transfer heat as they move through the plates in either parallel or

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counterflow setups. As a result, a PHE can accommodate a variety of flow arrangements, such as single, multiple passes, series, parallel, and their various combinations [1].

Because the design process of heat exchangers is complicated, it requires subjective choices at each design step. Additionally, the design methodology consists of multiple stages and relies on provisional information until the objectives are achieved. Typically, a heat exchanger's design encompasses these components: heat transfer to meet the necessary performance, total expenses, the actual geometrical dimensions, and the overall pressure drop [3].

As noted in [4], a lot of the design information related to plate heat exchangers is kept proprietary. A step-by-step approach for calculating the size and internal structure of the exchanger from available process information is not commonly found. Existing commercial software does not allow users to access the underlying mathematical models, and engineers typically lack familiarity with the specific terms and configurations of these exchangers. This reference also emphasizes that experimental findings in the literature regarding heat transfer and pressure drop are limited. Nonetheless, there are dimensionless correlations available for heat transfer coefficients as well as pressure drop within the channels of plate heat exchangers. Recommendations for constant and exponents values in the correlating equations are based on limited data and insights from manufacturers. Proper sizing of a plate heat exchanger relies on the required thermal duty and the characteristics of the exchanger itself. Its adaptability and operational benefits are accompanied by the challenge of creating a model for its steady flow behavior [1].

A considerable amount of studies has been carried out so far to investigate the characteristics of heat transfer and pressure drop in plate heat exchangers, which are continuously being improved and developed by scholars and technologists [5].

Various researchers have explored and evaluated the design of plate heat exchangers. In this regard, [3] conducted an investigation aimed at obtaining a clearer understanding of various plate characteristics, like Chevron angles, channel spacing, plate heights, and type on heat transfer and pressure drop calculations, employing PHEx® software as a computational resource to assess and illustrate the impact of each parameter through the simulation of an industrial case study. In [6], an experimental arrangement was developed and built to examine the influence of using nanofluids within a plate heat exchanger. The tests involved three distinct working fluids: tap water and nanofluids containing 1 and 0.5 wt. % Al_2O_3 in water, during a hot cycle, with flow rates between 100 to 450 L/h in every case. Additionally, [1] conducted a performance assessment supported by the principles of the first and second laws of thermodynamics for various operational arrangements of viable gasketed-plate heat exchangers. To ensure this, 40 simulations were performed utilizing the distributed-U differential model reported by various researchers, applying an adaptive damped secant shooting technique. The effectiveness of heat and exergy transfer, dimensionless entropy generation, potential entropic losses, and energy efficiency indices were computed when both fluids were either above or below ambient temperature, as well as when at least one fluid crossed the room temperature threshold.

In [7], the efficiency of a transformed corrugated plate heat exchanger was analyzed numerically through ANSYS-Fluent 20R1. A pressure-based transient model was implemented for the analysis. The k-ω SST turbulence model was utilized for this study. A nanofluid composed of water mixed with metallic oxide nanoparticles (Al_2O_3) was employed to improve thermal conductivity, and a broad range of Reynolds numbers ranging from 1,000 to 12,000 was considered. In another investigation [8], the researchers aimed to enhance the heat transfer efficiency between plates and minimize the pressure loss during fluid movement within the system. The numerical simulations conducted enabled the assessment of thermal flow within the heat exchanger, as well as the pressure drop and overall performance while altering the flow speeds and the spacing of the plates. Other authors [9] explored various methods to increase the thermal efficiency of plate heat exchangers utilized in processing vegetable oils by conducting multiple calculations. This research initiated from a baseline scenario where vegetable oils were cooled by water within plate heat exchangers, all featuring a Chevron angle of 30º along with varying channel numbers and plate surface areas. Similarly, in [10], the numerical study examined convective heat transfer, energy efficiency, and pressure drop of γ-Al_2O_3/water nanofluid in a gasketed plate heat exchanger across a varied concentration range of particles (0% to 6%), while the thermo-physical characteristics of γ-Al_2O_3/water nanofluid were obtained from established empirical relationships.

Similarly, [5] carried out the initial design of gasketed plate heat exchangers for single-phase flow using MATLAB as a computational platform. Subsequently, a software application was created for performing thermal and hydraulic calculations of gasketed plate heat exchangers, relying on established correlations found in existing research. The developed design program was then evaluated for precision and dependability compared to several approved designs of gasketed plate heat exchangers. In [4], a straightforward design approach for plate heat exchangers was introduced, which emphasized the use of uniform plates while neglecting various factors such as heat conduction along the plates and in flow passages, along with fluid properties that change with temperature. In [11], a design optimization for multi-pass plate-and-frame heat exchangers utilizing a mixed arrangement of plates was explored, where the approach was structured as a mathematical problem to determine the minimum value of an implicit nonlinear discrete/continuous objective function constrained by inequalities. The optimizing parameters assessed in this research included the number of passes for

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both fluid streams, the numbers of plates featuring different corrugation types in each pass, and the type and size of the plates.

In [12], advancements in the design principles of plate heat exchangers were examined, focusing on how they can enhance heat recovery and improve energy efficiency, while evaluating the ideal arrangement of a multi-pass plate-and-frame heat exchanger featuring mixed plate configurations. The variables considered for optimization in this analysis included the number of passes for each fluid stream, the quantity of plates with varying corrugation designs in every pass, as well as the type and dimensions of the plates. A mathematical model was created to estimate the value of the objective function within the optimization variable space for the plate heat exchanger. In [13], a plate and frame system was developed to reduce the temperature of a slurry stream, for which multiple parameters like the heat transfer rate and the necessary number of plates for the PHE were calculated, and cost optimization for the designed PHE were also examined. Other researchers [14] introduced a straightforward CAE approach for quickly designing and optimizing the dimensions of plate heat exchangers aimed at heat recovery. In this investigation, the flow dynamics and heat transfer processes in an air-to-air recuperative counter-flow plate heat exchanger were analyzed using numerical methods, while the pressure drop and effectiveness were assessed based on inlet velocity for three different sizes of actual heat exchangers.

Finally, [15] introduced an innovative and comprehensive methodology for the ideal design of gasket and welded plate heat exchangers, accommodating various plate shapes and flow patterns. This method combines a new design strategy with an optimization system aimed at achieving the best solution that minimizes the overall transfer area by creating a series of relationships between the temperatures in each single-pass block while using known inlet and outlet temperatures from the process streams. A MINLP mathematical model was consequently established in this research to determine the optimal combination of flow pass configurations and commercially available plate shapes while adhering to feasible design limitations. The distinctions in the design strategies for gasket and welded PHEs were then emphasized.

In a certain Cuban dairy factory it is desired to cool down 2,500 kg/h of a liquid cow's milk stream from 85 ºC to 25 ºC using chilled water as a coolant available at 5 ºC. Accordingly, a gasketed plate heat exchanger was proposed to carry out this heat transfer service. In this context, the objective of this study is to design a gasketed plate heat exchanger from the thermo-hydraulic point of view by using the design methodology reported by [16], where several important design parameters such as the total number of plates, heat load, overall heat transfer coefficient, surface area and the pressure drops of both fluids were calculated. Also, the purchase cost of the designed gasketed plate heat exchanger was estimated and updated to 2025 year.

2. Materials and methods.

2.1. Problem statement.

It's required to cool down 2,500 kg/h of a hot liquid cow's milk stream from 85 ºC to 25 ºC using chilled water at 5 ºC. The values for the effective plate, effective length and effective width are 0.75 m^2, 1.5 m and 0.5 m, respectively, while the plate spacing, plate thickness and plate material are 0.003 m, 0.0006 m and stainless steel, respectively. A maximum permissible pressure drop of 50,000 Pa and 20,000 Pa are set for the water and milk streams, respectively. Design, from the thermo-hydraulic point of view, a suitable gasketed-plate heat exchanger for this heat transfer service having a 1:1 flow arrangement and using the methodology reported by [16].

2.2 Design methodology.

Preliminary design

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

Table 1 presents the initial data that must be defined for the two fluids.

Table 1. Initial data to be defined for the two fluids.

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:

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

Where the unit of Q is kW.

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

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

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).
The overall heat transfer coefficient will be assumed based on values reported by [16] for plate heat exchangers.

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. Number of transfer units (NTU):

$$NTU = (T_1 - T_2)/\Delta Tlm \quad (6)$$

Step 9. Log mean temperature correction factor (F_t):

The log mean temperature correction factor will be selected based on a figure reported by [16] based on the value of NTU and the flow arrangement.

Step 10. Corrected mean temperature difference (ΔTm):

$$\Delta Tm = \Delta Tlm \cdot F_t \quad (7)$$

Step 11. Surface area required (A_0):

$$A_0 = \frac{Q \cdot 1{,}000}{U_0 \cdot \Delta Tm} \quad (8)$$

Where Q is given in kW and U_0 is given in W/m^2.K.

Step 12. Selection of the several parameters for the plates:

- Effective plate area (A_p)
- Effective length (L_p)
- Effective width (W_p)

Step 13. Number of plates required (N_0):

$$N_0 = \frac{A_0}{A_p} \quad (9)$$

Step 14. Flow arrangement and number of passes (N_p):

Step 15. Number of channels per pass (N_T):

$$N_T = \frac{N_0 - 1}{2} \quad (10)$$

Step 16. Assumption of the plate spacing (b).

Step 17 . Cross-sectional area (A_f):

$$A_f = b \cdot W_p \quad (11)$$

Step 18. Equivalent (hydraulic) mean diameter (d_e):

$$d_e = 2 \cdot b \quad (12)$$

- Hot fluid:

Step 19. Channel velocity for the hot fluid (v_{ph}):

$$v_{ph} = \frac{m_h}{N_T \cdot \rho_h \cdot A_f} \quad (13)$$

Where m_h is given in kg/s.

Step 20. Reynolds number for the hot fluid (Re_h):

$$Re_h = \frac{\rho_h \cdot v_{ph} \cdot d_e}{\mu_h} \quad (14)$$

Step 21. Prandtl number for the hot fluid (Pr_h):

$$Pr_h = \frac{(Cp_h \cdot 1{,}000) \cdot \mu_h}{k_h} \quad (15)$$

Step 22. Nusselt number for the hot fluid (Nu_h):

$$Nu_h = 0.26 \cdot (Re_h)^{0.65} \cdot (Pr_h)^{0.4} \cdot \left(\frac{\mu_h}{\mu_{hw}}\right)^{0.14} \quad (16)$$

Where the viscosity correction factor $\left(\frac{\mu_h}{\mu_{hw}}\right)^{0.14} = 1$ according to [16].

Step 23. Heat-transfer coefficient for the hot fluid (h_h):

$$h_h = \frac{Nu_h \cdot k_h}{d_e} \quad (17)$$

- Cold fluid:

Step 24. Channel velocity for the cold fluid (v_{pc}):

$$v_{pc} = \frac{m_c}{N_T \cdot \rho_c \cdot A_f} \quad (18)$$

Where m_c is given in kg/s.

Step 25. Reynolds number for the cold fluid (Re_c):

$$Re_c = \frac{\rho_c \cdot v_{pc} \cdot d_e}{\mu_c} \quad (19)$$

Step 26. Prandtl number for the cold fluid (Pr_c):

$$Pr_c = \frac{(Cp_c \cdot 1{,}000) \cdot \mu_c}{k_c} \quad (20)$$

Step 27. Nusselt number for the cold fluid (Nu_c):

$$Nu_c = 0.26 \cdot (Re_c)^{0.65} \cdot (Pr_c)^{0.4} \cdot \left(\frac{\mu_c}{\mu_{cw}}\right)^{0.14} \quad (21)$$

Where the viscosity correction factor $\left(\frac{\mu_c}{\mu_{cw}}\right)^{0.14} = 1$ according to [16].

Step 28. Heat-transfer coefficient for the cold fluid (h_c):

$$h_c = \frac{Nu_c \cdot k_c}{d_e} \quad (22)$$

Step 29. Select the plate thickness (X_p):

Step 30. Select the plate material and, therefore, its thermal conductivity (k_p):

Step 31. Overall heat transfer coefficient calculated (U_C):

$$U_C = \frac{1}{\frac{1}{h_c} + \frac{1}{h_h} + \frac{1}{R_h} \cdot \frac{1}{R_c} + \frac{X_p}{k_p}} \quad (23)$$

The calculated value of the overall heat transfer coefficient must be compared with the assumed overall heat transfer coefficient of Step 6. If the percentage error calculated through equation (24) is between -0% and +10%, the design is satisfactory, and then the designer should proceed to calculate the pressure drop of both fluids.

$$\%Error = \frac{U_C - U_0}{U_C} \cdot 100 \quad (24)$$

Pressure drop:

Step 32. Define port diameter (d_{pt}):

Step 33. Port area (A_{pt}):

$$A_{pt} = \frac{\pi \cdot d_{pt}^2}{4} \quad (25)$$

- Hot fluid:

Step 34. Friction factor for the hot fluid (j_{fh}):

$$j_{fh} = 0.6 \cdot (Re_h)^{-0.3} \quad (26)$$

Step 35. Plate pressure drop for the hot fluid (ΔP_{ph}):

$$\Delta P_{ph} = 8 \cdot j_{fh} \cdot \left(\frac{L_p}{d_e}\right) \cdot \frac{\rho_h \cdot v_{ph}^2}{2} \quad (27)$$

Step 36. Velocity through port for the hot fluid (u_{pth}):

$$u_{pth} = \frac{m_h}{\rho_h \cdot A_{pt}} \quad (28)$$

Step 37. Port pressure drop for the hot fluid (ΔP_{pth}):

$$\Delta P_{pth} = 1.3 \cdot \frac{(\rho_h \cdot u_{pth}^2)}{2} N_p \quad (29)$$

Step 38. Total pressure drop for the hot fluid (ΔP_{Th}):

$$\Delta P_{Th} = \Delta P_{ph} + \Delta P_{pth} \quad (30)$$

- Cold fluid:

Step 39. Friction factor for the cold fluid (j_{fc}):

$$j_{fc} = 0.6 \cdot (Re_c)^{-0.3} \quad (31)$$

Step 40. Plate pressure drop for the cold fluid (ΔP_{pc}):

$$\Delta P_{pc} = 8 \cdot j_{fc} \cdot \left(\frac{L_p}{d_e}\right) \cdot \frac{\rho_c \cdot v_{pc}^2}{2} \quad (32)$$

Step 41. Velocity through port for the cold fluid (u_{ptc}):

$$u_{ptc} = \frac{m_c}{\rho_c \cdot A_{pt}} \quad (33)$$

Step 42. Port pressure drop for the cold fluid (ΔP_{ptc}):

$$\Delta P_{ptc} = 1.3 \cdot \frac{(\rho_c \cdot u_{ptc}^2)}{2} N_p \quad (34)$$

Step 43. Total pressure drop for the cold fluid (ΔP_{Tc}):

$$\Delta P_{Tc} = \Delta P_{pc} + \Delta P_{ptc} \quad (35)$$

2.3. Purchased cost of the designed gasketed-plate heat exchanger

According to [16], the purchase cost of a stainless steel gasketed-plate and frame heat exchanger can be calculated using the following correlation [16]:

$$C_{(2007)} = 1{,}350 + 180 \cdot A^{0.95} \quad (36)$$

Where:

- $C_{(2007)}$ - Purchased equipment cost referred to January 2007.
- A - Area of the plate heat exchanger [m^2].

Once the purchase cost of the plate heat exchanger is calculated for January 2007 using equation (36), it was then updated to March 2025 using the following equation:

$$C_{(2025)} = C_{(2007)} \cdot \frac{CE\ Index_{(2025)}}{CE\ Index_{(2007)}} \quad (37)$$

Where:

- $C_{(2025)}$ – Purchased equipment cost referred to March 2025.
- $CE\ Index_{(2025)}$ – Chemical Engineering Cost Index in March 2025 = 791.6 [17].
- $CE\ Index_{(2007)}$ – Chemical Engineering Cost Index in January 2007 = 509.7 [16].

3. Analysis and Interpretation of Results.

3.1. Preliminary design.

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

Table 3 shows the values of the initial data for the two fluids.

Table 3. Values of the initial data for the two fluids.

Parameter	Units	Water	Milk
Mass flowrate	kg/h	-	2,500
Inlet temperature	ºC	5	85
Outlet temperature	ºC	30	25
Maximum permissible pressure drop	Pa	50,000	20,000
Fouling factor	W/m^2.ºC	8,000	1,000

Source: Own elaboration.

Step 2. Average temperature of both streams:

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

$$\bar{t} = \frac{t_1 + t_2}{2} = \frac{5 + 30}{2} = 17.5\ ºC \quad (1)$$

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

$$\bar{T} = \frac{T_1 + T_2}{2} = \frac{85 + 25}{2} = 55\ ºC \quad (2)$$

Step 3. Physical properties of both fluids at the average temperature:
Table 4 displays the values of the physical properties for both fluids at the average temperature calculated in Step 2, which were taken from data reported by [18] for the milk, and from [19] for the water.

Table 4. Values of the physical properties for both fluids.

Property	Units	Water	Milk
Density	kg/m^3	998.7	1,015.4
Viscosity	Pa.s	0.00107	0.002127
Heat capacity	kJ/kg.ºC	4.184	3.931
Thermal conductivity	W/m.K	0.599	0.559

Source: Own elaboration.

Step 4. Heat load (Q):

- For the hot fluid:

$$Q = \frac{m_h}{3{,}600} \cdot Cp_h \cdot (T_1 - T_2) \quad (3)$$
$$= \frac{2{,}500}{3{,}600} \cdot 3.931 \cdot (85 - 25)$$
$$= 163.79\ kW$$

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

$$m_c = \frac{Q}{Cp_c \cdot (t_2 - t_1)} = \frac{163.79}{4.184 \cdot (30 - 5)} \quad (4)$$
$$= 1.5659\ kg/s$$

Step 6. Assumption of the overall heat transfer coefficient (U_0).
Taking into account the values reported by [16] between the range of 2,000-4,500 W/m^2.K, it was assumed a preliminary value of 2,200 W/m^2.K for U_0.

Table 5 presents the values of the parameters included in steps 7-18.

Table 5. Values of the parameters included in steps 7-11.

Step	Parameter	Value	Units
7	Log mean temperature difference	34.60	ºC
8	Number of transfer units	1.73	-
9	Log mean temperature correction factor[1]	0.975	-
10	Corrected mean temperature difference	33.73	ºC
11	Surface area required	2.21	m^2

[1] As reported by [16].
Source: Own elaboration.

Step 12. Selection of several parameters for the plates:
Based on suggestions reported by [16] for typical plate dimensions, it was selected the following values for several parameters of the plates:

- Effective plate area (A_p) = 0.75 m^2.
- Effective length (L_p) = 1.5 m.
- Effective width (W_p) = 0.5 m.

Step 13. Number of plates required (N_0):

$$N_0 = \frac{A_0}{A_p} = \frac{2.21}{0.75} = 2.95 \sim 3 \quad (9)$$

Step 14. Flow arrangement and number of passes (N_p):
The flow arrangement will be 1:1, with a number of passes (N_p) of 1.

Step 15. Number of channels per pass (N_T):

$$N_T = \frac{N_0 - 1}{2} = \frac{3 - 1}{2} = 1 \quad (10)$$

Step 16. Assumption of the plate spacing (b):
It was assumed a plate spacing of 3 mm = 0.003 m, a typical value according to [16].

Step 17 . Cross-sectional area (A_f):

$$A_f = b \cdot W_p = 0.003 \cdot 0.5 = 0.0015\ m^2 \quad (11)$$

Step 18. Equivalent (hydraulic) mean diameter (d_e):

$$d_e = 2 \cdot b = 2 \cdot 0.003 = 0.006\ m \quad (12)$$

Table 6 displays the results of the parameters included in steps 19-28, where the heat transfer coefficients are calculated for each fluid.

Table 6. Results of the parameters included in steps 19-28.

Parameter	Milk	Water	Units
Channel velocity	0.456	1.045	m/s
Reynolds number	1,306	5,852	-
Prandtl number for the hot fluid	14.96	7.47	-
Nusselt number	81.34	163.36	-
Heat-transfer coefficient	7,578	16,309	W/m^2.K

Source: Own elaboration.

Step 29. Select the plate thickness (X_p):
A value of 0.0006 m was selected for the plate thickness.

Step 30. Select the plate material and, therefore, its thermal conductivity (k_p):
It was selected stainless steel for the plate material, therefore k_p = 16 W/m.K [16].

Step 31. Overall heat transfer coefficient calculated (U_C):

$$U_C = \frac{1}{\frac{1}{h_c} + \frac{1}{h_h} + \frac{1}{R_h} \cdot \frac{1}{R_c} + \frac{X_p}{k_p}} \quad (23)$$

$$U_C = \frac{1}{\frac{1}{16{,}309} + \frac{1}{7{,}578} + \frac{1}{1{,}000} + \frac{1}{8{,}000} + \frac{0.0006}{16}}$$

$$U_C = 2{,}194.06\ W/m^2.K$$

Percentage error

$$\%Error = \frac{U_C - U_0}{U_C} \cdot 100 \quad (24)$$

$$\%Error = \frac{2{,}194.06 - 2{,}200}{2{,}194.06} \cdot 100$$

$$\%Error = -0.27\% \sim 0\%$$

3.2. Pressure drop.

Step 32. Define port diameter (d_{pt}):
The select value for the port diameter (d_{pt}) was 0.1 m.

Step 33. Port area (A_{pt}):

$$A_{pt} = \frac{\pi \cdot d_{pt}^2}{4} = \frac{3.14 \cdot (0.1)^2}{4} = 0.00785\ m^2 \quad (25)$$

Table 7 shows the results of the parameters included in steps 34-43 for each fluid:

Table 7. Results of the parameters included in steps 34-43.

Parameter	Milk	Water	Units
Friction factor	0.0697	0.0445	-
Plate pressure drop	14,716.33	48,532	Pa
Velocity through port	0.087	0.1997	m/s
Port pressure drop	4.996	25.888	Pa
Total pressure drop	14,720	48,558	Pa

Source: Own elaboration.

3.3. Purchase cost of the designed gasketed-plate heat exchanger.

By using equation (36), where A – surface area required = 2.21 m^2, the purchase cost of the plate heat exchanger, referred to January 2007, is:

$$C_{(2007)} = 1{,}350 + 180 \cdot A^{0.95} \quad (36)$$

$$C_{(2007)} = 1{,}350 + 180 \cdot 2.21^{0.95}$$

$$C_{(2007)} = USD\ \$\ 1{,}733$$

Then, to update this purchase cost to March 2025, equation (37) was used:

$$C_{(2025)} = C_{(2007)} \cdot \frac{CE\ Index_{(2025)}}{CE\ Index_{(2007)}} \quad (37)$$

$$C_{(2025)} = 1{,}733 \cdot \frac{791.6}{509.7}$$

$$C_{(2025)} = USD\ \$\ 2{,}692$$

4. Discussion

According to the results, the heat load (Q) had a value of 163.79 kW, thus requiring a mass flowrate for the cooling water (m_c) of 1.5659 kg/s (5,637.24 kg/h). Also, the surface area required was 2.21 m^2, with a corrected mean temperature difference of 33.73 ºC and a required number of plates of 3. This low quantity of plates is because the relatively low value of the heat load and the high value of the assumed overall heat transfer coefficient (2,200 W/m^2.K), which influences then in the low value of the calculated surface area, and thus, in the required number of plates. In the 1:1 plate heat exchanger designed in [16] in order to cool 27.8 kg/s of a methanol stream from 95 ºC to 40 ºC using brackish water at 25 ºC, the heat duty is 4,340 kW, the required mass flowrate of brackish water is 68.9 kg/s and the required surface area is 72.92 m^2, therefore needing 97 plates.

The heat-transfer coefficient for the cooling water (16,309 W/m^2.K) was 2.15 times higher than the value of this parameter for the milk (7,578 W/m^2.K), which is due to the fact that the mass flowrate of the cooling water (5,637.24 kg/h) is 2.25 times higher than the mass flowrate for the milk (2,500 kg/h). This influences then in that the channel velocity for the water (1.045 m/s) is higher than the channel velocity for the milk (0.456 m/s), thus obtaining that the Reynolds number for the water (5,852) is 4.48 times higher than the Reynolds number for the milk (1,306), which influences in this difference. This agrees with the reported by [16], where the heat transfer coefficient for the brackish water (16,439 W/m^2.K) is 3.37 times higher than the heat transfer coefficient for the methanol (4,870 W/m^2.K). The values of the Reynolds number obtained in the present study agrees with the reported by (Mehrabian, 2009), where it is indicated that the fluid flow in plate heat exchanger channels is usually at low Reynolds numbers, and at the same time in turbulent regime.

A value for the calculated overall heat transfer coefficient of 2,194.06 W/m^2.K was obtained, which agrees very close with the assumed overall heat transfer coefficient (2,200 W/m^2.K), while a calculated percentage error of -0.27% was obtained that corresponds with the range proposed by [16] for this parameter, thus indicating that the design is satisfactory, there is no need to perform additional iterations and that we must proceed to calculated the pressure drops for both fluids. In the plate heat exchanger designed in [16], the initial value assumed for the overall heat transfer coefficient was 2,000 W/m^2.K.

Regarding the pressure drops, the value of the friction factor for the milk (0.0697) was 1.57 times higher than the friction factor for the water (0.0445), which is because the lower value obtained for Reynolds number of the milk compared to the Reynolds number of the water. The plate pressure drop for the water (48,532 Pa) was 3.29 times higher than the value of this parameter for the milk, which is largely due to the higher value obtained for the channel velocity of the water (1.045 m/s) as compared to the channel velocity of the milk (0.456 m/s). Likewise, the velocity through port is higher for the water (0.1997 m/s) as compared to the value of this parameter for the milk (0.087 m/s) because water has a higher mass flowrate, while the port pressure drop for the water (25.888 Pa) is 5.18 times higher than the port pressure drop for the milk (4.996 Pa) mainly because the water has a higher value of the velocity through port. The total pressure drop for water (48,558 Pa) is 3.29 times higher than the total pressure drop for the milk (14,720), because both the plate pressure drop and the port pressure drop are higher for the water as compared to the values of these parameters for the milk.

The above agrees with the results of the gasketed-plate heat exchanger designed in [16], where the plate pressure drop (26,547 Pa), the port pressure drop (50,999 Pa) and the total pressure drop (77,546 Pa) are higher for the cold fluid (water) compared to the value of the plate pressure drop (5799 Pa) the port pressure drop (10,860 Pa) and the total pressure drop (16,659 Pa) for the hot fluid (methanol). Lastly, in the heat exchange service studied in this paper the calculated values of the total pressure drops for both fluids are below the maximum pressure drops set by the process, which are 50,000 Pa for water and 20,000 Pa for milk. Thus it is concluded that the designed plate heat exchanger in this study is suitable and appropriate from the thermo-hydraulic point of view, and can be successfully implemented in the requested heat transfer application of cow's milk cooling.

In [13] a plate heat exchanger was designed to cool down 231,000 kg/h of a slurry stream from 86.6 ºC to 66 ºC using cooling water at 34 ºC. In this study, the total number of plates was 108, the area of the plate heat exchanger was 110.377 m^2, the heat load was 1,132,500 kcal/h and the overall heat transfer coefficient was 327.17 kcal/h.m^2.°C.

The purchase cost of the gasketed-plate heat exchanger, referred to January 2007, was USD $ 1,733, while the purchase cost of the same gasketed-plate heat exchanger updated to March 2025 was USD $ 2,692.

5. Conclusions.

A gasketed-plate heat exchanger was designed to carry out the cooling of a hot milk stream using chilled water as coolant. Several important design parameters were computed, being the most important the heat load, the required mass flowrate of chilled water, the surface area and the number of plates. Similarly, the heat transfer coefficients for both fluids were estimated based on well-established correlations, as well as the overall heat transfer coefficient. Finally, the pressure drops of both fluid streams were also calculated and compared to the maximum values set by the heat exchanger process. The designed heat exchanger will present three plates, a flow arrangement of 1:1, a surface area of 2.21 m^2, a heat load of 163.79 kW, a required mass flowrate of chilled water of 1.5659 kg/s (5,638 kg/h) and a calculated overall heat transfer coefficient of 2,194.06 W/m^2.K. Both the total pressure drop of chilled water (48,558 Pa) and milk (14,720 Pa) are below the maximum permissible values set by the process, i.e. 50,000 Pa for the water and 20,000 Pa for the milk. It is concluded that the designed PHE will cost USD $ 2,692 and could be satisfactorily implemented, from the thermo-hydraulic point of view, in the heat transfer service.

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

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9. Project management: Not applicable.
10. Resources: Not applicable.
11. Software: Not applicable.
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7.- Appendix

Nomenclature.

A_0	Surface area required	m^2
A_f	Cross-sectional area	m^2
A_p	Effective plate area	m^2
A_{pt}	Port area	m^2
b	Plate spacing	-
Cp	Heat capacity	kJ/kg.ºC
d_e	Equivalent (hydraulic) mean diameter	m
d_{pt}	Port diameter	m
k	Thermal conductivity	W/m.K
F_t	Log mean temperature correction factor	-
h	Heat-transfer coefficient	W/m^2.K
j_f	Friction factor	-
k_p	Plate thermal conductivity	W/m.K
L_p	Effective length	m
m	Mass flowrate	kg/h
N_0	Number of plates required	-
N_p	Number of passes	-
N_T	Number of channels per pass	-

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NTU	Number of transfer units	-
Nu	Nusselt number	-
Pr	Prandtl number	-
ΔP_p	Plate pressure drop	Pa
ΔP_{pt}	Port pressure drop	Pa
ΔP_T	Total pressure drop	Pa
Q	Heat load	kW
R	Fouling factor	W/m^2.ºC
Re	Reynolds number	-
t	Temperature cold fluid	ºC
$\bar{t}$	Average temperatura cold fluid	ºC
T	Temperature hot fluid	ºC
T	Average temperatura hot fluid	ºC
ΔTlm	Log mean temperature difference	ºC
ΔTm	Corrected mean temperature difference	ºC
u_{pt}	Velocity through port	m/s
U_C	Overall heat transfer coefficient calculated	W/m^2.K
U_0	Overall heat transfer coefficient assumed	W/m^2.K
v_p	Channel velocity	m/s
W_p	Effective width	m
X_p	Plate thickness	m

Greek symbols

ρ	Density	kg/m^3
μ	Viscosity	Pa.s
μ_h	Viscosity of the fluid at the wall temperature	Pa.s

Subscripts

1	Inlet
2	Outlet
c	Cold fluid
h	Hot fluid

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