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Thermo-hydraulic design of an unfinned and finned double pipe heat exchanger for tomato juice cooling. Part 1 - Unfinned heat exchanger

Diseño térmico-hidráulico de un intercambiador de calor de doble tubo sin y con aletas para el enfriamiento de jugo de tomate. Parte 1 – Intercambiador de calor sin aletas

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Abstract.
The double pipe heat exchanger (DPHE) is commonly used in the food industry due to its simplicity in mechanical design, minimal maintenance, ease of cleaning and also the possibility to operate at high temperatures and pressures. The objective of this paper is to design an unfinned DPHE form the thermo-hydraulic point of view, to determine its feasibility to cool down a stream of hot tomato juice using chilled water as coolant. The main design parameters calculated were the total number of hairpins (32), the cleanliness factor (0.708), the percent over surface (41.29%), and the frictional pressure drop and pumping power for both fluids. Likewise, it will be required a mass flowrate of cooling water of 6.50 kg/s, with a heat load of 951.77 kW. The designed DPHE cannot be successfully applied to the requested heat transfer service because the high values obtained for the total number of hairpins, percent over surface and the frictional pressure drop (16.56 atm) for the annulus fluid (chilled water), which is quite higher than the maximum allowable value set by the process (0.296 atm) for this stream, thus increasing also the pumping power of this fluid to a significant value (13.67 kW). The designed unfinned DPHE will cost around USD $ 165,500 based on Jan, 2026.

Keywords.
Double pipe heat exchanger, design, total number of hairpins, percent over surface, pressure drop, pumping power.

Resumen.
El intercambiador de calor de doble tubo (ICDT) es comúnmente usado en la industria alimenticia debido a su simplicidad en el diseño mecánico, mínimo mantenimiento, facilidad de limpieza y también la posibilidad de operar a elevadas temperaturas y presiones. El objetivo de este artículo es el de diseñar un ICDT desde el punto de vista térmico-hidráulico, para determinar su factibilidad para enfriar una corriente de jugo de tomate caliente usando agua fría como agente de enfriamiento. Los principales parámetros de diseño calculados fueron el número total de horquillas (32), el factor de limpieza (0,708), el porcentaje de sobre superficie (41,29%), y la caída de presión friccional y potencia de bombeo para ambos fluidos. Asimismo, se requerirá un caudal másico de agua fría de 6,50 kg/s, con una carga de calor de 951,77 kW. El ICDT diseñado no puede ser aplicado satisfactoriamente para el servicio de transferencia de calor requerido, debido a los elevados valores obtenidos para el número total de horquillas, porcentaje de sobre superficie y caída de presión friccional (16,56 atm) para el fluido del ánulo (agua fría), el cual es muy superior al valor mínimo permisible fijado por el proceso (0,296 atm) para esta corriente, incrementado por tanto también la potencia de bombeo de este fluido hacia un valor significativo (13,67 kW). El ICDT sin aletas diseñado costará alrededor de USD $ 165 500 basado en Enero del 2026.

Palabras clave.
Intercambiador de calor de doble tubo, diseño, número total de horquillas, porcentaje de sobre superficie, caída de presión, potencia de bombeo.

1. Introduction.

Heat exchanger equipment transfers heat from a hot fluid to a cold fluid, where both fluids are separated by means of a solid wall. Typically, most heat exchangers are two-fluid heat exchangers, although three-fluid heat exchangers are becoming popular. In terms of construction, heat exchangers can be classified as shell and tube, concentric tube and finned tube heat exchangers. The choice of a heat exchanger for a given application is dependent on the application itself, available resources, space, economics, materials selection, and other factors. Whatever may be the choice of a heat exchanger, it is very essential that a heat exchanger be designed such that it delivers the required heat transfer while occupying less space, has pressure drops under the allowable limits set by the process, and yet be cost competitive [1].

Double pipe heat exchanger (DPHE) is regarded as one of the simplest types of tubular heat exchanger. It is made up of concentric pipes that are separated by mechanical closures. Double pipe heat exchangers are relatively cheap to construct and require minimal maintenance. They are used in high temperature and high pressure applications [2].

DPHE (also called hairpin heat exchangers) are a widely used configuration in heating and industrial systems.

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DPHEs can be used in the chemical, food, oil and gas industries, solar and geothermal energy applications, air-conditioning applications, reheating, preheating, and digester heating processes. Moreover, DPHE is still widely used for the research and development of experimental facilities in energy engineering laboratories. One advantage is the low cost of design and maintenance. This type of heat exchanger is simple in structure and easy to install, clean, maintain, and modify, which allows a significant extension of life and service [3].

DPHE consists of one or more pipes placed concentrically inside another pipe of a larger diameter with appropriate fittings to direct the flow from one section to the next. One fluid flows through the inner pipe (tube-side), and the other flows through the annular space (annulus). The inner pipe is connected by U-shaped return bends enclosed in a return-bend housing. DPHE can be arranged in various series and parallel arrangements to meet pressure drop requirements. The major use of the DPHE is the sensible heating or cooling process of fluids where small heat transfer areas (up to 50 m^2) are required. This configuration is also very suitable for one or both of the fluids at high pressure because of the smaller diameter of the pipes. They can be used when one stream is a gas, viscous liquid, or small in volume, and also under severe fouling conditions because of the ease of cleaning and maintenance [4].

According to [5], if the stream contains solids in suspension, DPHE may also be a better alternative, because they can be built with an inner tube with larger diameter to avoid plugging. In addition, double pipe heat exchangers are easily cleaned, and the longitudinal flow avoids the existence of stagnation regions, which in shell and tube exchangers are prone to fouling. DPHEs have also the benefit of flexibility due to its modular structure, which allows an easier adaptation to process modifications.

Tomato (*Lycopersicon esculentum* Mill) is one of the most widely consumed cultivated horticultural crops globally and ranks second in global production among vegetable crops. Tomato undergoes a variety of different treatments aiming to transform it into well-known commercial products such as sauce, ketchup and juice, canned whole tomatoes, and tomato paste/concentrate. For example, for preservation after harvesting, tomatoes are thermally treated with several techniques, such as pasteurization and sterilization, and drying and evaporation at high temperatures, which help to ensure the microbiological safety and stability of the resulting products [6].

The technology for the production of tomato juice drink is basically the same with that used for the production of tomato paste. The process involves selection of ripen and wholesome fruits, followed by washing, chopping, removal of seeds, hot break, pre-heating demineralized water, and then screened over 0.3-0.4 mm meshes to extract finer juice with less suspended solids and to remove free fatty acids and oily matter. The juice obtained may be concentrated by heating under vacuum. Salt, sugar and spices may be added to taste. The finished product is usually stored in compatible packaging materials made of glass or aluminum coated containers [7].

As stated by [8], two methods are used commercially to produce tomato juice: hot break and cold break. "Hot break" typically refers to a chopping temperature of 85 to 90 ºC, which causes inactivation of enzymes important to aroma and viscosity. Pectinmethylesterase and polygalacturonase break down chains of pectin in tomato tissue. By inactivating these enzymes in the hot break process, a more viscous product can be achieved, which is usually desirable. Conversely, lipoxygenase is the initiating enzyme involved in the creation of important fresh aroma compounds through the breakdown of unsaturated fatty acids, and when it is inactivated in the hot break, fewer aroma compounds are present. These authors also mention that "cold break" refers to a temperature below 70 ºC to increase enzyme activity. A characteristic of the cold break process is a decrease in viscosity. Advantages of cold breaking over hot breaking are that the final product is said to have a more natural color and a fresher flavor. The hot break method is used to produce the majority of tomato products to maintain a high viscosity. However, an increase in the use of the cold break method should produce tomato products with a fresher aroma.

In [6] is indicated that conventional heat treatments are currently the most common ones in the food industry, but due to the heat transfer mechanism (conduction and convection), these processes have some drawbacks such as high temperature, loss of nutritional value, and sensory changes. In addition, the combustion of the fuel used to generate heat results in economic energy losses. These disadvantages can be avoided with modern technologies such as ohmic heating, which is a thermal treatment that heats food by passing an alternating electric current through it; as a consequence, this method has a large capacity for rapid and homogeneous heating. In addition, this technique provides safe and high-quality food products while preserving their nutritional value and is characterized by high energy allowing reducing heating time by 90–95%. These authors developed an ohmic-vacuum combination heating system for producing tomato paste at temperatures of 70, 80, and 90 ºC and pressure of 0.3, 0.5, and 0.7 bar and electric field of 1.82, 2.73, and 3.64 V/cm using a central composite design. The effects of heating conditions on the quality and sensory evaluation of tomato paste were also evaluated in this study, while each combination of temperature, pressure, and the electric field was quantified for specific energy consumption, energy efficiency, and productivity.

Few studies are reported related to the thermo-hydraulic design or assessment of heat transfer equipment to process tomato products. In this sense, [9] applied Computational Fluid Dynamics (CFD) to design a shell and tube heat exchanger on an industrial scale and to simulate heat transfer in order to visualize the pasteurization process of tomato paste with the objective of presenting it to the industry, where the behavior of viscosity in the pasteurization process of tomato paste was explained using the Herschel-Bulkley model. Likewise, [10] aimed to survey possible improvements in thermophysical properties of nanofluids when they are used as heating mediums for time reduction and energy saving in food industries for the first time. Accordingly, three different variables of temperature (70, 80, and 90 °C), alumina nanoparticle concentration (0, 2, and 4 %), and time (30, 60, and 90 s) were selected for thermal processing of tomato juice by a shell and tube heat exchanger. Similarly, [11] evaluated the thermal performance of a triple-tube heat exchanger equipped with staggered fins for tomato concentrate processing. In this study, challenges were encountered due to the high viscosity of non-Newtonian fluids treated in the food industry, which lead to laminar flow and limited thermal penetration. Numerical simulations, implemented within an ANSYS® Fluent environment, were used to solve the equations governing the thermofluid dynamics of a triple-tube heat exchanger with staggered fins in the internal annulus. The rheological behavior of tomato concentrate was modeled by means of a power-law model, coupled with the Arrhenius law to introduce temperature dependency of the consistency index. Experimental validation was adopted to verify simulations accuracy. Despite all the studies mentioned above, to the best of the authors' knowledge, the design of a DPHE for tomato juice cooling has not been reported yet.

Some of the authors of the present study recently published an article [12] where an unfinned DPHE was designed from the thermo-hydraulic point of view, to cool a stream of liquid cow's milk using chilled water as coolant. The design methodology employed in this research was that reported by [13], obtaining as the main result that the designed unfinned DPHE cannot be applied satisfactorily in the requested heat transfer service because the quite high value obtained for the tube-side fluid pressure drop.

In a Cuban food processing facility is desired to cool a stream of tomato juice using chilled water as coolant, and two DPHEs have been proposed for this thermal system, the first one unfinned and the second with longitudinal fins in the inner tube (finned). Therefore, the present paper aims to design the unfinned DPHE from the thermo-hydraulic point of view, in order to know the suitability of this heat exchanger for this heat exchange service. The design methodology employed in the present paper was that reported by [13], where several important design parameters are calculated such as the total number of hairpins, the cleanliness factor, the percent over surface, as well as the pressure drop and pumping power of both liquid streams. Also, the purchase cost of the unfinned DPHE is calculated. In a second paper, a finned DPHE will be designed in order to calculate the same design parameters previously mentioned, while the design results of the present paper will be compared with those of the finned DPHE designed in the second paper, with the purpose of selecting the most appropiate and suitable DPHE from the thermal and hydraulic points of view to perform this heat exchange service.

2. Materials and methods.

2.1. Problem statement.

In a Cuban food processing facility is required to cool down 14,000 kg/h of a stream of tomato juice from 90 ºC to 30 ºC using chilled water as coolant available at 5 ºC, while the outlet temperature of this chilled water must not exceed 40 ºC. The selected construction material of the DPHE is carbon steel, while the pressure drop of the tomato juice and chilled water streams should not be higher than 300,000 Pa. Design a suitable unfinned DPHE from the thermo-hydraulic point of view to satisfy this heat transfer service, using the following inlet data for the tubes:

- Tube length: 4.0 m.
- Annulus internal diameter: 0.07790 m.
- Internal diameter of inner tube: 0.0525 m.
- External diameter of inner tube: 0.0603 m.
- Thermal conductivity of carbon steel: 54 W/m.K [13].

2.2. Design methodology

Total number of hairpins and percent over surface

Step 1. Definition of the initial parameters for the streams: Table 1 presents the initial parameters that must be defined for both fluids.

Table 1. Initial parameters that must be defined for the two fluids.

Parameter	Hot fluid	Cold fluid	Units
Mass flowrate	m_h	m_c	kg/h
Inlet temperature	T_1	t_1	ºC
Outlet temperature	T_2	t_2	ºC
Maximum allowable pressure drop	ΔP_{mh}	ΔP_{mc}	Pa
Fouling factor	R_h	R_c	m^2.K/W

Source: Own elaboration.

Step 2. Definition of the geometric data for the hairpins: Table 2 shows the geometric data that should be defined for the hairpins.

Table 2. Geometric data to be defined for the hairpins.

Parameter	Symbol	Units
Length	L_t	m
Internal diameter annulus	D_i	m
Internal diameter inner tube	d_i	m
External diameter inner tube	d_e	m

Thermal conductivity metallic material of the inner pipe	k_m		W/m.K

Source: Own elaboration.

Step 3. Definition of the flow arrangement inside the double-pipe heat exchanger:

- Counterflow
- Parallel

Step 4. Allocation of fluids inside the double-pipe heat exchanger

Step 5. Consider insulation of the double-pipe heat exchanger against heat losses.

Step 6. Average temperature of both streams:

- Hot fluid (T):

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

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

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

Step 7. Physical parameters of both fluids at the average temperature:
Table 3 exhibits the physical parameters that are required to be considered for both fluids at the average temperature calculated in the previous step.

Table 3. Physical parameters that should be considered for both fluids.

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

Source: Own elaboration.

Step 8. Heat load (Q):

- Using the data for the hot fluid:

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

- Using the data for the cold fluid:

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

Where both m_h and m_c are given in kg/s.

Step 9. Mass flowrate of one stream:

- Mass flowrate of the hot fluid:

$$m_h = \frac{Q}{Cp_h \cdot (T_1 - T_2)} \tag{5}$$

- Mass flowrate of the cold fluid:

$$m_c = \frac{Q}{Cp_c \cdot (t_2 - t_1)} \tag{6}$$

Step 10. Tube wall temperature (T_w):

$$T_w = \frac{1}{2} \cdot (\bar{T} + \bar{t}) \tag{7}$$

Step 11. Viscosity of both fluids at the tube wall temperature:

- Hot fluid (μ_{hw}) [Pa.s].
- Cold fluid (μ_{cw}) [Pa.s].

Step 12. Net free flow area of the inner tube (A_{ct}):

$$A_{ct} = \frac{\pi \cdot d_i^2}{4} \tag{8}$$

Step 13. Velocity of the tube-side fluid (v_t):

$$v_t = \frac{\frac{m_t}{3{,}600}}{\rho_t \cdot A_{ct}} \tag{9}$$

Step 14. Reynolds number of the tube-side fluid (Re_t):

$$Re_t = \frac{\rho_t \cdot v_t \cdot d_i}{\mu_t} \tag{10}$$

Step 15. Prandtl number of the tube-side fluid (Pr_t):

$$Pr_t = \frac{Cp_t \cdot \mu_t}{k_t} \tag{11}$$

Where Cp_t is given in J/kg.K.

Step 16. Nusselt number for the tube-side fluid (Nu_t):

- Laminar flow ($Re_t < 2{,}300$)

$$Nu_t = 1.86 \cdot (Re_t \cdot Pr_t)^{0.33} \cdot \left(\frac{d_i}{L_t}\right)^{0.33} \cdot \left(\frac{\mu_t}{\mu_{tw}}\right)^{0.14} \tag{12}$$

Valid for smooth tubes for the following conditions:
$0.48 < Re_t \cdot Pr_t < 16{,}700$

$0.0044 < \left(\frac{\mu_t}{\mu_{tw}}\right)^{0.14} < 9.75$

$\left(Re_t \cdot Pr_t \cdot \frac{d_i}{L_t}\right)^{0.33} \left(\frac{\mu_t}{\mu_{tw}}\right)^{0.14} \geq 2$

- Turbulent flow ($2{,}300 < Re_t < 10^4$) [Gnielinski's correlation]:

$$Nu_t = \frac{\left(\frac{f_t}{2}\right) \cdot (Re_t - 1{,}000) \cdot Pr_t}{1 + 12.7 \cdot \left(\frac{f_t}{2}\right)^{0.5} \cdot \left(Pr_t^{2/3} - 1\right)} \tag{13}$$

Where f_t is the Fanning friction factor for the tube-side fluid and is calculated using the following correlation:

$$f_t = (1.58 \cdot lnRe_t - 3.28)^{-2} \tag{14}$$

- Turbulent flow ($10^4 < Re_t < 5 \times 10^6$) [Prandtl's correlation]:

$$Nu_t = \frac{\left(\frac{f_t}{2}\right) \cdot Re_t \cdot Pr_t}{1 + 8.7 \cdot \left(\frac{f_t}{2}\right)^{0.5} \cdot (Pr_t - 1)} \quad (15)$$

Valid for $Pr_t > 0.5$.
Where:

$$f_t = (1.58 \cdot lnRe_t - 3.28)^{-2} \quad (14)$$

Step 17. Heat transfer coefficient for the tube-side fluid (h_t):

$$h_t = \frac{Nu_t \cdot k_t}{d_i} \quad (16)$$

Step 18. Net free flow area of the annulus (A_{ca}):

$$A_{ca} = \frac{\pi \cdot (D_i^2 - d_e^2)}{4} \quad (17)$$

Step 19. Velocity of the annulus fluid (v_a):

$$v_a = \frac{\frac{m_a}{3{,}600}}{\rho_a \cdot A_{ca}} \quad (18)$$

Step 20. Hydraulic diameter (D_h):

$$D_h = D_i - d_e \quad (19)$$

Step 21. Reynolds number of the annulus fluid (Re_a):

$$Re_a = \frac{\rho_a \cdot v_a \cdot D_h}{\mu_a} \quad (20)$$

Step 22. Prandtl number of the annulus fluid (Pr_a):

$$Pr_a = \frac{Cp_a \cdot \mu_a}{k_a} \quad (21)$$

Where Cp_a is given in J/kg.K.

Step 23. Nusselt number for the annulus fluid (Nu_a):

- Laminar flow ($Re_a < 2{,}300$)

$$Nu_a = 1.86 \cdot (Re_a \cdot Pr_a)^{0.33} \cdot \left(\frac{D_h}{L_t}\right)^{0.33} \cdot \left(\frac{\mu_a}{\mu_{aw}}\right)^{0.14} \quad (22)$$

Valid for smooth tubes for the following conditions:
$0.48 < Re_a \cdot Pr_a < 16{,}700$
$0.0044 < \left(\frac{\mu_a}{\mu_{aw}}\right)^{0.14} < 9.75$
$\left(Re_a \cdot Pr_a \cdot \frac{D_h}{L_t}\right)^{0.33} \left(\frac{\mu_a}{\mu_{aw}}\right)^{0.14} \geq 2$

- Turbulent flow ($2{,}300 < Re_a < 10^4$) [Gnielinski's correlation]:

$$Nu_a = \frac{\left(\frac{f_a}{2}\right) \cdot (Re_a - 1{,}000) \cdot Pr_a}{1 + 12.7 \cdot \left(\frac{f_a}{2}\right)^{0.5} \cdot (Pr_a^{2/3} - 1)} \quad (23)$$

Where f_a is the Fanning friction factor for the annulus fluid and is calculated using the following correlation:

$$f_a = (1.58 \cdot lnRe_a - 3.28)^{-2} \quad (24)$$

- Turbulent flow ($10^4 < Re_a < 5 \times 10^6$) [Prandtl's correlation]:

$$Nu_a = \frac{\left(\frac{f_a}{2}\right) \cdot Re_a \cdot Pr_a}{1 + 8.7 \cdot \left(\frac{f_a}{2}\right)^{0.5} \cdot (Pr_a - 1)} \quad (25)$$

Valid for $Pr_a > 0.5$.
Where:

$$f_a = (1.58 \cdot lnRe_a - 3.28)^{-2} \quad (24)$$

Step 24. Equivalent diameter for heat transfer (D_e):

$$D_e = \frac{D_i^2 - d_e^2}{d_e} \quad (26)$$

Step 25. Heat transfer coefficient for the annulus fluid (h_a):

$$h_a = \frac{Nu_a \cdot k_a}{D_e} \quad (27)$$

Step 26. Fouled overall heat transfer coefficient based on the outside area of the inner tube (U_f):

$$U_f = \frac{1}{\frac{d_e}{d_i \cdot h_t} + \frac{d_e \cdot R_t}{d_i} + \frac{d_e \cdot ln\left(\frac{d_e}{d_i}\right)}{2 \cdot k_m} + R_a + \frac{1}{h_a}} \quad (28)$$

Step 27. Log-mean temperature difference (ΔTm):

- For parallel flow:

$$\Delta Tm = \frac{(T_1 - t_1) - (T_2 - t_2)}{ln\frac{(T_1 - t_1)}{(T_2 - t_2)}} \quad (29)$$

- For counterflow:

$$\Delta Tm = \frac{(T_1 - t_2) - (T_2 - t_1)}{ln\frac{(T_1 - t_2)}{(T_2 - t_1)}} \quad (30)$$

Step 28. Heat transfer surface area (A_o):

$$A_o = \frac{Q \cdot 1{,}000}{U_f \cdot \Delta Tm} \quad (31)$$

Where Q is given in kW.

Step 29. Heat transfer area per hairpin (A_{hp}):

$$A_{hp} = 2 \cdot \pi \cdot d_e \cdot L_t \quad (32)$$

Step 30. Number of hairpins (N_h):

$$N_h = \frac{A_0}{A_{hp}} \quad (33)$$

Step 31. Clean overall heat transfer coefficient based on the outside heat transfer area (U_c):

$$U_c = \frac{1}{\frac{d_e}{d_i \cdot h_t} + \frac{d_e \cdot ln\left(\frac{d_e}{d_i}\right)}{2 \cdot k_m} + \frac{1}{h_a}} \quad (34)$$

Step 32. Cleanliness factor (CF):

$$CF = \frac{U_f}{U_c} \quad (35)$$

Step 33. Total fouling (R_{ft}):

$$R_{ft} = \frac{1 - CF}{U_c \cdot CF} \quad (36)$$

Step 34. Percentage over surface (OS):

$$OS = 100 \cdot U_c \cdot R_{ft} \quad (37)$$

Pressure drop and pumping power

Step 35. Frictional pressure drop of the tube-side fluid (Δp_t):

$$\Delta p_t = 4 \cdot f_t \cdot \frac{2 \cdot L_t}{d_i} \cdot N_h \cdot \frac{\rho_t \cdot v_t^2}{2} \quad (38)$$

Where for laminar flow (Re_t < 2,300):

$$f_t = \frac{16}{Re_t} \quad (39)$$

Correction of the Fanning friction factor for laminar flow (f_{ct}):

$$f_{ct} = f_t \cdot \left(\frac{\mu_t}{\mu_{tw}}\right)^m \quad (40)$$

Where m = - 0.58 for heating and - 0.50 for cooling under laminar flow.

Step 36. Pumping power for the tube-side fluid (P_t):

$$P_t = \frac{\Delta p_t \cdot m_t}{\eta_p \cdot \rho_t} \quad (41)$$

Where m_h is given in kg/s and η_p is the isentropic efficiency of the pump.

Step 37. Frictional pressure drop of the annulus fluid (Δp_a):

$$\Delta p_a = 4 \cdot f_a \cdot \frac{2 \cdot L_t}{D_h} \cdot \rho_a \cdot \frac{v_a^2}{2} \cdot N_h \quad (42)$$

Where for laminar flow (Re_a < 2,300):

$$f_a = \frac{16}{Re_a} \quad (43)$$

Correction of the Fanning friction factor for laminar flow (f_{ct}):

$$f_{ca} = f_a \cdot \left(\frac{\mu_a}{\mu_{aw}}\right)^m \quad (44)$$

Where m = - 0.58 for heating and - 0.50 for cooling under laminar flow.

Step 38. Pumping power for the annulus fluid (P_a):

$$P_a = \frac{\Delta p_a \cdot m_a}{\eta_p \cdot \rho_a} \quad (45)$$

Where m_a is given in kg/s and η_p is the isentropic efficiency of the pump.

2.3. Purchase cost of the designed DPHE

According to [14], the purchase cost of a unfinned DPHE can be calculated using the following correlation on a U.S. Gulf Coast Basis referred to Jan. 2007:

$$C_{2007} = 1{,}600 + 2{,}100 \cdot A_0^{1.0} \quad (46)$$

Where A_0 - Heat transfer surface area calculated using equation (31), which must range from 1.0 to 80.0 m^2 [14].

Since the unfinned DPHE purchase cost calculated by equation (46) is based on Jan, 2007, this cost must be updated to 2026 using the following correlation [15]:

$$C_{2026} = C_{2007} \frac{CEPCI_{2026}}{CEPCI_{2007}} \quad (47)$$

Where:
C_{2007} – Purchase cost of the unfinned DPHE based on Jan, 2007, calculated using equation (46).
$CEPCI_{2026}$ – Chemical Engineering Plant Cost Index based on Jan, 2026 = 840.9 [16].
$CEPCI_{2007}$ – Chemical Engineering Plant Cost Index based on Jan, 2007 = 509.7 [14].
C_{2026} – Purchase cost of the unfinned DPHE based on Jan, 2026.

3. Results.

3.1. Total number of hairpins and percent over surface

Following next are the numerical results of the several key design parameters calculated by the methodology to size the unfinned DPHE, specifically the total number of hairpins, the cleanliness factor and the percent over surface.

Step 1. Definition of the initial parameters for the streams: Table 4 exposes the numerical values of the initial parameters that must be defined for each fluid.

Table 4. Numerical values of the initial parameters for both fluids.

Parameter	Hot fluid (tomato juice)		Cold fluid (chilled water)		Units
Mass flowrate	m_h	14,000	m_c	-	kg/h
Inlet temperature	T_1	90	t_1	5	ºC
Outlet temperature	T_2	30	t_2	40	ºC
Maximum allowable pressure drop	ΔP_{mh}	300,000	ΔP_{mc}	300,000	Pa

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Fouling factor[1]	R_h	0.0003	R_c	0.000176	m².K/W

[1] The values of the fouling factors for both fluids were taken from suggestions reported by [13].
Source: Own elaboration.

Step 2. Definition of the geometric dimensions for the hairpins:
Table 5 indicates the values of the geometric data for the hairpins.

Table 5. Values of the geometric data for the hairpins.

Parameter	Symbol	Value	Units
Length	L_t	4.0	m
Internal diameter annulus	D_i	0.00779	m
Internal diameter inner tube	d_i	0.0525	m
External diameter inner tube	d_e	0.0603	m
Thermal conductivity metallic material of the inner pipe	k_m	54	W/m.K

Source: Own elaboration.

Step 3. Definition of the flow arrangement inside the double-pipe heat exchanger:
Taking into account suggestions reported by [13][17][18], the chosen flow arrangement for this heat exchange system is counterflow.

Step 4. Allocation of fluids inside the double-pipe heat exchanger.
In [3] is stated that it is appropriate to use a DPHE when the heated medium is placed in the annulus with a relatively small cross section area and the heating medium (e.g., steam, hot water, hot process stream) flows in the inner pipe. Following the suggestions of this study, as well as recommendations reported by [14], the hot fluid (tomato juice) will be located in the inner pipe, while the cold fluid (chilled water) will be located in the annulus.

Step 5. Consider insulation of the double-pipe heat exchanger against heat losses.
The designed DPHE will be insulated to prevent unwanted heat losses in the equipment.

Step 6. Average temperature of both streams:

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

$$\bar{T} = \frac{T_1 + T_2}{2} = \frac{90 + 30}{2} = 60\ ºC \quad (1)$$

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

$$\bar{t} = \frac{t_1 + t_2}{2} = \frac{5 + 40}{2} = 22.5\ ºC \quad (2)$$

Step 7. Physical parameters of both fluids at the average temperature:
Table 6 presents the values of the physical parameters that are defined for both the tomato juice and chilled water at the average temperature calculated in the previous step, which were taken from [19] for the tomato juice, and [20] for the chilled water.

Table 6. Values of the physical parameters defined for both fluids.

Parameter	Hot fluid (tomato juice)		Cold fluid (chilled water)		Units
Density	ρ_h	1,006.3	ρ_c	997.658	kg/m³
Viscosity	μ_h	0.021	μ_c	0.000943	Pa.s
Heat capacity	Cp_h	4.079	Cp_c	4.1825	kJ/kg.K
Thermal conductivity	k_h	0.630	k_c	0.6029	W/m.K

Source: Own elaboration.

Step 8. Heat load (Q):

- Using the data for the hot fluid:

$$Q = m_h \cdot Cp_h \cdot (T_1 - T_2) \quad (3)$$

$$Q = \frac{14{,}000}{3{,}600} \cdot 4.079 \cdot (90 - 30) = 951.77\ kW$$

Step 9. Mass flowrate of the cold fluid (chilled water):

$$m_c = \frac{Q}{Cp_c \cdot (t_2 - t_1)} = \frac{951.77}{4.1825 \cdot (40 - 5)} = 6.50\ kg/s \quad (6)$$

Step 10. Tube wall temperature (T_w):

$$T_w = \frac{1}{2} \cdot (\bar{T} + \bar{t}) = \frac{1}{2} \cdot (60 + 22.5) = 41.25\ ºC \quad (7)$$

Step 11. Viscosity of both fluids at the tube wall temperature:
For a value of the tube wall temperature of 41.25 ºC, the tomato juice (hot fluid) and chilled water (cold fluid) will present the following values for the viscosities [19][20]:

- Tomato juice (μ_{hw}) = 0.03018 Pa.s.
- Chilled water (μ_{cw}) = 0.00064 Pa.s.

Since the tomato juice was located in the inner tube and the chilled water was located in the annulus, the following new nomenclature presented in Table 7 will be applied for the flowrates, physical parameters and fouling factors of both streams.

Table 7. New nomenclature to be applied for both streams.

Parameter	Hot fluid (tomato juice)	Cold fluid (chilled water)

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	Former nomenclature	New nomenclature	Former nomenclature	New nomenclature
Flowrate	m_h	m_t	m_c	m_a
Density	ρ_h	ρ_t	ρ_c	ρ_a
Viscosity	μ_h	μ_t	μ_c	μ_a
Heat capacity	Cp_h	Cp_t	Cp_c	Cp_a
Thermal conductivity	k_h	k_t	k_c	k_a
Fouling factor	R_h	R_t	R_c	R_a

Source: Own elaboration.

Table 8 shows the results of the parameters calculated in steps 12 to 25.

Table 8. Results of the parameters calculated in steps 12 to 25.

Step	Parameter	Symbol	Value	Units
12	Net free flow area of the inner tube	A_{ct}	0.00216	m^2
13	Velocity of the tube-side fluid (tomato juice)	v_t	1.785	m/s
14	Reynolds number of the tube-side fluid	Re_t	4,492	-
15	Prandtl number of the tube-side fluid	Pr_t	135.97	-
16	Fanning friction factor for the tube-side fluid	f_t	0.00998	-
16	Nusselt number for the tube-side fluid[1]	Nu_t	99.45	-
17	Heat transfer coefficient for the tube-side fluid	h_t	1,194	W/m^2.K
18	Net free flow area of the annulus	A_{ca}	0.00191	m^2
19	Velocity of the annulus fluid (chilled water)	v_a	3.411	m/s
20	Hydraulic diameter	D_h	0.0176	m
21	Reynolds number of the annulus fluid	Re_a	63,521	-
22	Prandtl number of the annulus fluid	Pr_a	6.542	-
23	Fanning friction factor for the annulus fluid	f_a	0.00496	-
23	Nusselt number for the annulus fluid[2]	Nu_a	303.17	-
24	Equivalent diameter for heat transfer	D_e	0.0403	m
25	Heat transfer coefficient for the annulus fluid	h_a	4,531.13	W/m^2.K

[1] Since 2,300 < Re_t < 10^4, Gnielinski's correlation was employed to calculate Nu_t.
[2] since 10^4 < Re_a < 5 x 10^6, Prandtl's correlation was utilized to calculate Nu_a.
Source: Own elaboration.

Table 9 breakdowns the results of the parameters calculated in steps 26-34.

Table 9. Results of the parameters calculated in steps 27-40.

Step	Parameter	Symbol	Value	Units
26	Fouled overall heat transfer coefficient based on the outside area of the inner tube	U_f	561.46	W/m^2.K
27	Log-mean temperature difference[1]	ΔTm	36.07	ºC
28	Heat transfer surface area	A_o	47	m^2
29	Heat transfer area per hairpin	A_{hp}	1.515	m^2
30	Number of hairpins	N_h	32	-

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31	Clean overall heat transfer coefficient based on the outside heat transfer area	U_c	793.33	W/m^2.K
32	Cleanliness factor	CF	0.708	-
33	Total fouling	R_{ft}	0.000521	m^2.K/W
34	Percent over surface	OS	41.29	%

[1] Calculated for counterflow arrangement.
Source: Own elaboration.

3.2. Pressure drop and pumping power

Table 10 presents the results of the parameters calculated in steps 35-38, which comprise the calculation of the pressure drop and pumping power for both fluids.

Table 10. Results of the parameters calculated in steps 35-38.

Step	Parameter	Symbol	Value	Units
35	Frictional pressure drop of the tube-side fluid	Δp_t	312,558	Pa
			3.08	atm
36	Pumping power for the tube-side fluid	P_t	1.51	kW
37	Frictional pressure drop of the annulus fluid	Δp_a	1,678,128	Pa
			16.56	arm
38	Pumping power for the annulus fluid	P_a	13.67	kW

Source: Own elaboration.

3.3. Purchase cost of the designed DPHE

Using correlation (46), and for a value of A_o of 47 m^2, the purchase cost of the unfinned DPHE based on Jan, 2007 is:

$$C_{2007} = 1{,}600 + 2{,}100 \cdot A_0^{1.0} \quad (46)$$

$$C_{2007} = 1{,}600 + 2{,}100 \cdot 47^{1.0} = USD\ \$\ 100{,}300$$

Then, the purchase cost of the unfinned DPHE based on Jan, 2026 is:

$$C_{2026} = C_{2007} \cdot \frac{CEPCI_{2026}}{CEPCI_{2007}} \quad (47)$$

$$C_{2026} = 100{,}300 \cdot \frac{840.9}{509.7} \approx USD\ \$\ 165{,}500$$

4. Discussion

The heat load of the heat exchange system was 951.77 kW, while it was required a mass flowrate of 6.50 kg/s of chilled water at 5 ºC, to cool down 14,000 kg/h of tomato juice from 90 ºC to 30 ºC. In [12] an unfinned DPHE was designed to cool down 4,320 kg/h of a stream of liquid cow's milk using chilled water as coolant available at 2 ºC. In this study, the hot fluid (cow's milk) was located in the annulus, while the cold fluid (chilled water) was located in the inner pipe, thus obtaining a heat load of 235.14 kW and a required mass flowrate of chilled water of 9.32 kg/s.

According to the results of Table 8, the velocity of the tomato juice (tube-side fluid) was 1.785 m/s, which is within the range proposed by [14] of 1-2 m/s for process fluids flowing in tubular heat exchangers, while the velocity of the chilled water (annulus fluid) was 3.411 m/s, which is above the range proposed by [14] of 1.5-2.5 m/s for water flowing in tubular heat exchangers. The velocity of the annulus fluid was 1.91 times higher than the velocity of the tube-side fluid. In [12] the calculated velocities of the tube-side fluid (chilled water) and the annulus fluid (liquid cow's milk) are 16.64 m/s and 0.92 m/s, respectively.

The Reynolds number of the tube-side fluid (tomato juice) had a value of 4,492, thus indicating that this fluid flows in the transition zone, while the annulus fluid (chilled water) flows in the turbulent zone because the value of its Reynolds number (63,521) is above 10,000 [13]. The Reynolds number of the chilled water was 14.14 times higher than the Reynolds number of the tomato juice. The tomato juice had this low value of the Reynolds number mainly because the relatively high value of its viscosity (0.021 Pa.s) and also because the low value of the internal diameter of the inner tube (0.0525 m). In [12], the Reynolds number of the tube-side fluid (chilled water) and the annulus fluid (liquid cow's milk) are 291,629 and 16,796, respectively, therefore both fluids flowing under turbulent regime.

The calculated value of the Prandtl number for the tomato juice was 132.97, and was 20.78 times higher than the Prandtl number of the chilled water (6.542). This difference is largely due to the higher value that presents the viscosity of the tomato juice (0.021 Pa.s) compared to the viscosity of the chilled water (0.000943 Pa.s). In [12], the Prandtl number of the tube-side fluid (chilled water) and the annulus fluid (liquid cow's milk) are 11.19 and 7.16, respectively.

The Nusselt number of the tube-side fluid (tomato juice) was 99.45, while the value of this parameter for the annulus fluid (chilled water) was 303.17. The Nusselt number for the chilled water was 3.05 times higher than the Nusselt number for the tomato juice, which is because the higher number of the Reynolds number that presented the chilled water (63,521) compared to the Reynolds number of the tomato juice (4,492). However, it should be mentioned that the equations employed to calculate the Nusselt numbers for both fluid were different, i.e. the Gnielinski's correlation was employed to calculate the Nusselt number for the tomato juice, while the Prandtl's correlation was utilized to calculate the Nusselt number for the chilled water, based on the values of the Reynolds number for both fluids. In [4] an unfinned DPHE was designed to cool down 5 kg/s of a stream of hot engine oil (annulus fluid) from 65 to 55 ºC using sea water (tube-side fluid) as coolant. The calculated

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Nusselt number for the tube-side fluid (cold sea water) fluctuated from 422.0330 to 798.6098 for a range of the tube-side friction factor of 0.002-0.004, depending on the set of correlation employed to calculate these parameters, while the calculated Nusselt number for the annulus fluid (hot engine oil) was 34.693 for a value of the friction factor of 0.024. In [12], the Nusselt number of the tube-side fluid (chilled water), which was calculated using the Prandtl's correlation and amounted 1,237.84, is 12.44 times higher than the Nusselt number of the annulus fluid (liquid cow´s milk), which was also calculated using the Prandtl's correlation and amounted 99.49.

The heat transfer coefficient for the annulus fluid (4,531.13 $W/m^2.K$) was 3.78 times higher than the heat transfer coefficient for the tube-side fluid (1,194 $W/m^2.K$). This occurred mainly because the higher value obtained of the Nusselt number for the water compared to the value of this parameter for the tomato juice. In [4] the heat transfer coefficient for sea water flowing inside the tubes ranged from 12,885 $W/m^2.K$ to 24,382 $W/m^2.K$, depending on the correlations employed to calculate the Nusselt number. Likewise, the heat transfer coefficient for the engine oil flowing in the annulus for the unfinned DPHE is 64.549 $W/m^2.K$. In [12], the heat transfer coefficient of the tube-side fluid (chilled water) is 26,531.78 $W/m^2.K$, while the heat transfer coefficient of the annulus fluid (liquid cow's milk) is 1,175.24 $W/m^2.K$, that is, the heat transfer coefficient for chilled water is 22.58 higher than the heat transfer coefficient for the liquid cow's milk.

The values of the fouled and clean overall heat transfer coefficients were 561.46 $W/m^2.K$ and 793.33 $W/m^2.K$, respectively, and were within the range reported by [21] of 280-850 $W/m^2.K$; the range reported by [14] of 350-900 $W/m^2.K$, while the fouled overall heat transfer coefficient was within the range proposed by [13] of 370-750 $W/m^2.K$, whereas the clean overall heat transfer coefficient was higher than the range reported by these authors. The values of the overall heat transfer coefficient reported by the literature are for sensible heat transfer with no phase change using water as coolant, fitting adequately to our heat transfer system. In [12] the fouled and clean overall heat transfer coefficients have values of 774.31 $W/m^2.K$ and 1,030.11 $W/m^2.K$, respectively.

The calculated log-mean temperature difference had a value of 36.07 ºC, with a heat transfer surface area of 47 m^2, a heat transfer area per hairpin of 1.515 m^2, thus resulting in a total number of hairpins of 32, which can be considered high and atypical for a DPHE [13]. We consider that the reasons behind this high value for the total number of hairpins were the high value of the heat load (951.77 kW), and the relatively low values obtained for the fouled overall heat transfer coefficient (561.46 $W/m^2.K$) and the log-mean temperature difference (36.07 ºC). Likewise, the flowing of the tube-side fluid (tomato juice) under the transition zone, a zone that must be avoided in designing heat exchangers [17], could have influenced in obtaining such a high value for the total number of hairpins, because the low value obtained for the Reynolds number of the tomato juice reduced the value of the overall heat transfer coefficient under fouling conditions (U_f), therefore increasing the heat transfer surface area (A_o) and consequently increasing the total number of hairpins. In [4], the total number of hairpins for an engine oil/sea water unfinned DPHE are 85 and 87 under clean and fouled conditions, respectively. In [12], the log-mean temperature difference, the heat transfer surface area, and the heat transfer per hairpin are 23.51 ºC, 12.92 m^2 and 0.629 m^2, respectively, thus obtaining a total number of hairpins of 21.

The cleanliness factor, which is a term developed for the steam power industry that relates the overall heat transfer coefficient when the heat exchanger is fouled to when it is clean [13], had a value of 0.708, a result that can be considered below the value proposed by [13] of 0.85 for typical tubular heat exchangers. This occurred because the relatively high value for the clean overall heat transfer coefficient (793.33 $W/m^2.K$). In [4] the cleanliness factor had a value of 0.9819 for a range of the Nusselt number of 422.0330 – 798.6098 during the design of an unfinned DPHE. In [12], the cleanliness factor has a value of 0.752 for an unfinned DPHE.

The percent over surface was 41.29%, which could be considered high compared to the average value recommended by [13] of not be more than about 30 %. Thus, this is a rather large heat exchanger, which could be reduced to satisfy the process specifications with more frequent cleaning. It may be preferable to use a 30% over surface design to decrease the investment cost, and then the cleaning scheduling could be arranged accordingly. Therefore, the proposed design may be modified and rerated [13]. In [4] the percent over surface fluctuated from 1.8400 % to 1.8455 % for a range of the Nusselt number of 422.0330 – 798.6098 during the design of an unfinned DPHE. In [12], the value of the percent over surface is 32.96%.

The frictional pressure drop for the tube-side fluid (tomato juice, 312,558 Pa or 3.08 atm) slightly surpassed the maximum allowable limit set by the heat exchange system (300,000 Pa), while the frictional pressure drop of the annulus fluid (chilled water) had a quite high value (1,678,128 Pa or 16.56 atm), which was 5.59 times higher than the maximum allowable limit set by the process for this stream (300,000 Pa). This high value obtained for the frictional pressure drop of the chilled water is principally because the high value of the chilled water velocity 3.411 m/s, the high number of hairpins required (32) and the low value of the hydraulic diameter (0.0176 m). In [17] it is stated that, if the calculated pressure drop is excessive, it will be necessary to increase the flow area, either by increasing the tube diameters or installing more branches in parallel. The frictional pressure drop of the annulus fluid was 5.37 times higher than the pressure drop of the tube-side fluid. The results of our study agree with those reported

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by [4], where the pressure drops of the annulus fluid (hot engine oil) are higher than the pressure drops of the tube-side fluid (sea water), for both fouled and clean conditions. However, the results of our study are not consistent with those reported by [12], where the frictional pressure drop of the tube-side fluid (chilled water) is 122.51 times higher than the frictional pressure drop of the annulus fluid (liquid cow's milk).

Finally, the pumping power of the annulus fluid (13.67 kW) was 9.05 times higher than the pressure drop of the tube-side fluid (1.51 kW), primarily because the quite high value of the pressure drop obtained for chilled water compared to the value of this parameter for the tomato juice. This result agrees with those reported by [4], where the value of the pumping power for the annulus fluid is higher than the pumping power for the tube-side fluid, both for the unfinned clean and unfinned fouled DPHE designed in this study. However, the results of our study regarding this parameter disagrees with those reported by [12], where the pumping power of the tube-side fluid (chilled water) is 110.5 kW, while the pumping power for the annulus fluid (liquid cow's milk) is 114.58 W.

The purchase cost of the unfinned DPHE based on Jan, 2026 was around USD $ 165,500 for a heat transfer surface area (A_o) of 47 m^2. In [12] the purchase cost of an unfinned DPHE designed to cool down a stream of liquid cow's milk is USD $ 45,600, based on May 2025 for a heat transfer surface area of 12.92 m^2 and applying the same correlation reported by [14].

5. Conclusions.

In the present paper, an unfinned DPHE was designed from the thermo-hydraulic point of view, to cool down a stream of hot tomato juice using chilled water as coolant and by applying the design methodology reported by [13]. Several key design parameters were calculated such as heat load (951.77), the required mass flowrate of chilled water (6.50 kg/s), the fouled overall heat transfer coefficient (561.46 W/m^2.K), the total number of hairpins (32), the cleanliness factor (0.708) and the percent over surface (41.29%). Likewise, the frictional pressure drop and pumping power were also calculated for both fluids. It is concluded that the designed DPHE cannot be implemented in the requested heat transfer service because the high values obtained for the total number of hairpins and percent over surface, which is mainly due to the flowing of the tube-side fluid (tomato juice) in the transition zone. This conclusion can be corroborated taking into account the quite high value obtained for the frictional pressure drop of the annulus fluid (16.56 atm), which is 5.59 times higher than the maximum allowable limit set by the process for this stream, which in turn increased the pumping power for this fluid to a significant value (13.67 kW). The designed DPHE will cost around USD $ 165,500 based on Jan, 2026. It's recommended to increase the diameter of both pipes and redesign the unfinned DPHE to decrease the pressure drop of the annulus fluid, as well as to avoid designing the unfinned DPHE with one of the fluids flowing in the transition zone.

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8. Appendix.

Nomenclature.

A_o	Heat transfer surface area	m^2
A_{ca}	Net free flow area of the annulus	m^2
A_{ct}	Net free flow area of the inner tube	m^2
A_{hp}	Heat transfer area per hairpin	m^2
Cp	Heat capacity	kJ/kg.K
CF	Cleanliness factor	-
d_e	External diameter inner tube	m
d_i	Internal diameter inner tube	m
D_e	Equivalent diameter for heat transfer	m
D_h	Hydraulic diameter	m
D_i	Internal diameter annulus	m

f	Fanning friction factor	-
f_c	Corrected Fanning friction factor	-
h	Heat transfer coefficient	$W/m^2.K$
k	Thermal conductivity	W/m.K
k_m	Thermal conductivity metallic material of the inner pipe	W/m.K
L_t	Tube length	m
m	Mass flowrate	kg/h
N_h	Number of hairpins	-
Nu	Nusselt number	-
OS	Percent over surface	%
P	Pumping power	kW or W
Pr	Prandtl number	-
Δp	Frictional pressure drop	Pa
ΔP_m	Maximum allowable pressure drop	Pa
Q	Heat load	kW
R	Fouling factor	$m^2.K/W$
Re	Reynolds number	-
R_{ft}	Total fouling	$m^2.K/W$
t	Temperature cold fluid	ºC
$\bar{t}$	Average temperature cold fluid	ºC
T	Temperature hot fluid	ºC
T_w	Tube wall temperature	ºC
$\bar{T}$	Average temperature hot fluid	ºC
ΔTm	Log-mean temperature difference	ºC
U_c	Clean overall heat transfer coefficient	$W/m^2.K$
U_f	Fouled overall heat transfer coefficient	$W/m^2.K$
v	Velocity	m/s

Greek symbols

ρ	Density	kg/m^3
μ	Viscosity	Pa.s
μ_w	Viscosity of the fluid at the tube wall temperature	Pa.s
η_p	Isentropic efficiency of the pump	-

Subscripts

1	Inlet
2	Outlet
a	Annulus fluid
c	Cold fluid
h	Hot fluid
t	Tube side fluid

9.- References.

[1] K. Silaipillayarputhur, T. A. Mughanam, A. A. Mojil, and M. A. Dhmoush, "Analytical and Numerical Design Analysis of Concentric Tube Heat Exchangers – A Review," *IOP Conf. Series: Materials Science and Engineering*, vol. 272, p. 012006, 2017. https://doi.org/10.1088/1757-899X/272/1/012006

[2] N. Nwokolo, P. Mukumba, and K. Obileke, "Thermal Performance Evaluation of a Double Pipe Heat Exchanger Installed in a Biomass Gasification System," *Journal of Engineering*, pp. 1-8, 2020. https://doi.org/10.1155/2020/6762489

[3] J. Havlik, T. Dlouhy, and J. Krempasky, "Heat Transfer in Double Pipe Heat Exchangers With Small Tube Spacing," *Journal of Fluids Engineering*, vol. 147, pp. 1-10, 2025.

[4] C. Ezgi and Ö. Akyol, "Thermal Design of Double Pipe Heat Exchanger Used as an Oil Cooler in Ships: A Comparative Case Study," *Journal of Ship Production and Design*, vol. 35, no. 1, pp. 12-18, 2019. https://doi.org/10.5957/JSPD.170009

[5] A. Peccini, J. C. Lemos, A. L. H. Costa, and M. J. Bagajewicz, "Optimal Design of Double Pipe Heat Exchanger Structures," *Industrial & Engineering Chemistry Research*, vol. 58, pp. 12080-12096, 2019. https://doi.org/10.1021/acs.iecr.9b01536

[6] Z. T. Alkanan, A. B. Altemimi, A. R. S. Al-Hilphy, F. Cacciola, and S. A. Ibrahim, "Application and Effects of Ohmic-Vacuum Combination Heating on the Quality Factors of Tomato Paste," *Foods*, vol. 10, p. 2920, 2021. https://doi.org/10.3390/foods10122920

[7] G. I. Igile, O. O. Ekpe, N. M. Essien, S. C. Bassey, and M. A. Agiang, "Quality Characteristics of Tomato Juice Produced and Preserved with and without its Seeds," *Donnish Journal of Food Science and Technology*, vol. 2, no. 1, pp. 001-009, 2016.

[8] C. L. Goodman, S. Fawcett, and S. A. Barringer, "Flavor, Viscosity, and Color Analyses of Hot and Cold Break Tomato Juices," *Journal of Food Science*, vol. 67, no. 1, pp. 404-408, 2002.

[9] S. Asadbeigi, E. Ahmadi, M. Goodarzi, and A. Sagharichian, "Analyzing and simulating heat transfer and designing a shell and tube heat exchanger for the pasteurization process of tomato paste: A CFD study," *Heliyon*, vol. 9, p. e21593, 2023. https://doi.org/10.1016/j.heliyon.2023.e21593

[10] S. M. Jafari, S. S. Jabari, D. Dehnad, and S. A. Shahidi, "Heat Transfer Enhancement in Thermal Processing of Tomato Juice by Application of Nanofluids," *Food Bioprocess Technol*, vol. 10, pp. 307-316, 2017. https://doi.org/10.1007/s11947-016-1816-9

[11] L. Pagliarini, U. Neviani, and S. Rainieri, "Numerical modelling and experimental validation of heat transfer enhancement and thermal mixing in a triple-tube heat exchanger for food processing," *International Journal of Numerical Methods for Heat & Fluid Flow*, vol. 35, no. 11, pp. 3749-3774, 2025.

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e – ISSN: 3028-8533
ISSN – L: 3028-8533

[12] A. P. Sánchez, L. d. l. C. A. Águila, H. V. González, M. I. L. R. Veliz, and Z. M. S. Rodríguez, "Thermo-hydraulic design of an unfinned and finned double pipe heat exchanger for milk cooling. Part 1 - Unfinned heat exchanger," *Ingeniería Química y Desarrollo,* vol. 08, no. 01, pp. 88-101, 2026. https://doi.org/10.53591/fv3x4n54
[13] S. Kakaç, H. Liu, and A. Pramuanjaroenkij, *Heat Exchanger - Selection, Rating, and Thermal Design*, 3rd ed. Boca Raton, USA: CRC Press, 2012.
[14] R. Sinnott and G. Towler, *Chemical Engineering Design*, 6yh ed. Oxford, United Kingdom: Butterworth-Heinemann, 2020.
[15] J. R. Couper, D. W. Hertz, and F. L. Smith, "9. Process Economics," in *Perry's Chemical Engineers' Handbook*, D. W. Green and M. Z. Southard, Eds. 9th ed. New York, USA: McGraw-Hill Education, 2019.
[16] S. Jenkins, "Economic Indicators," *Chemical Engineering*, vol. 135, no. 3, p. 40, 2026.
[17] E. Cao, *Heat transfer in process engineering*. New York, USA: The McGraw-Hill Companies, Inc., 2010.
[18] A. M. Flynn, T. Akashige, and L. Theodore, *Kern's Process Heat Transfer*, 2nd ed. New Jersey, USA: John Wiley & Sons, Inc., 2019.
[19] Y. Choi and M. R. Okos, "The Thermal Properties of Tomato Juice Concentrates," *Transactions of the ASAE,* vol. 26, no. 1, pp. 305-311, 1983.
[20] M. Z. Southard, R. L. Rowley, and W. V. Wilding, "2. Physical and Chemical Data," in *Perry's Chemical Engineers' Handbook*9th ed. New York, USA: McGraw-Hill Education, 2019.
[21] S. M. Hall, *Rules of Thumb for Chemical Engineers*, 6th ed. Amsterdam, Netherlands: Elsevier, 2018.