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Chapter

Designing Spiral Plate Heat Exchangers to Extend Its Service and Enhance the Thermal and Hydraulic Performance

Lázaro Canizalez Dávalos, Edilberto Murrieta Luna, Mario Alberto Rodríguez Ángeles and Víctor J. Cruz Delgado

Abstract

Spiral plate heat exchangers are well suitable for handling fluids with features linked to fouling, high viscosities, fluids with suspended fragments of solids and process streams with tough heat transfer targets. Correlations to describe the thermal and hydraulic performance are a function of the geometrical configuration of the equipment. The present work shows procedures to design spiral plate heat exchangers as a function of the fluid arrangements, government flow, as well whether the thermal equipment is for condensing, cooling or heating duty. An additional study looking for determining the geometrical variables that allow to enhance and improve the thermal and hydraulic performance is presented. Moreover, computational fluid dynamics to validate the thermal and hydraulic method is performed.

Keywords: design, thermal and hydraulic performance, spiral plate heat exchanger, single- and two-phase flow, computational fluid dynamics

1. Introduction

A spiral heat exchanger is assembled by two metallic plates separated by studs. They are welded on the plate surface. The objective is to maintain a constant spacing between plates at the time the plates are rolled up. Thus, the first turn represents one flow section; at this point, the second flow section initiates. Fluids pass in spiral plate heat exchangers by two arrangements, cross-flow and spiral flow, and these configurations are shown in **Figure 1**. Since these patterns were stated, manufacture companies and researchers have proposed more understandings regarding designs, thermal and hydraulic analyses and ways to provide more worth to these heat exchangers.

Spiral plate heat exchangers have important industrial applications; particularly, they are suitable for dirty fluids and viscous fluids. However, most of the correlations and methods explain the single-phase liquid-liquid, and as a consequence, this is not sufficiently to describe heat transfer and hydraulic behaviour, i.e., two-phase flow in spiral heat exchangers liquid–gas and liquid-vapour. Sathiyan et al. [1, 2] presented a study to evaluate a new equation to approximate the Nusselt number of

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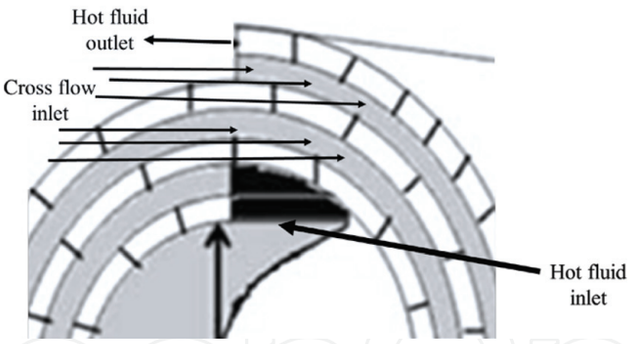


Figure 1.
Cross-flow and spiral flow arrangement.

an immiscible mixture using a countercurrent spiral heat exchanger for two-phase flow. The new correlation was based on the experimental data, and the results were agreement with the theoretical correlation. Khorshidi and Heidari [3] fabricated a spiral heat exchanger geometry to study the performance, the examination showed that spiral heat exchanger is an excellent option to transfer heat especially from fouling fluids and also a computational fluid dynamic was presented to determine a previous design. Ramachandran et al. [4] determined the heat transfer behaviour for a system of two fluids by implementing a countercurrent spiral plate heat exchanger; the data were obtained from varying mass fraction inlets and demonstrated efficient results between the experiment and the correlation. Maruyama et al. [5] measured the thermal effectiveness of a cross-flow spiral plate heat exchanger; the aim was to convert radiation energy from a combustion chamber. Wang [6] analysed the thermal performance of a spiral plate heat exchanger used as an adsorber in a refrigeration process, the flows were configured to follow a spiral trajectory and the spiral exchanger resulted appropriated for a refrigeration system. Bahiraei et al. [7, 8] presented a study to evaluate the thermal and hydraulic performance of a spiral plate heat exchanger under a turbulent flow of a nanofluid. The experimental procedure was to determine the effects due to the spiral geometry varying the flow rate to define the optimal operational conditions.

The thermal and hydraulic concept is an innovative tool to achieve designs for heat exchangers and applies to all types of heat exchangers. Previously, researches have been using this procedure [9, 10]. Compact and conventional exchangers are employed to develop two duties heating and cooling. Nevertheless, they behave differently from each other due to their geometrical configuration, effectiveness, outlet temperature, pressure drop and heat transfer area [11–15].

The current study is organised to describe four main purposes: 1. To present two design methods by a thermal and hydraulic procedure. The first design is for a cooler using a cross-flow arrangement (liquid-gas) without phase change, to evaluate if a spiral plate heat exchanger can take part as a radiator of the cooling system car. The second approach is to size a vertical spiral heat exchanger condenser for a cryogenic operation. 2. To extend the operational activities of spiral plate heat exchangers. 3. To improve the spiral thermal and hydraulic performance by modifying the spacing plate. 4. A numeric analysis applying computational fluid dynamics to validate the method.

2. Empirical thermal and hydraulic model

The method to design spiral plate heat exchangers includes two main equations, the film heat transfer coefficient and the pressure drop, which both are functions of the fluid properties, heat load, geometrical standard parameters, flow section area and

metal construction characteristics. Thermal and hydraulic model is a key relationship which reduces the calculations to sizing and performs heat exchangers. Commonly, five iterations are needed to reach the balance between pressure drop and heat transfer. Finally, the heat transfer area is determined. The present procedure primarily introduces a hydraulic equation which is a function of pressure drop, the spacing between plates, flow rate and spiral plate length. The correlation is solved iteratively for the length; if the calculated length does not satisfy the pressure drop design, the spacing plate could adjust to maximise the use of permissible pressure drop.

2.1 Hydraulic equations

The hydraulic equations were presented by Minton [16]. These correlations are a function of a flow along the spiral channel which is separated by studs to give support to the plates. Factor 1.5 in Eq. 1 (**Table 1**) supposes 17 studs per square foot. Every stud has a diameter of 0.3125, and, then, in every 0.118 in^2, a stud is installed [16]. Eq. 2 (**Table 1**) has the same approach; however, the pressure drop is negligible because the fluid flows across the plate width, and value close to zero represents minor influence even by installing the studs [16]. Calculating the Reynolds number and the critical Reynolds number values is possible to select the correct equation to describe the hydraulic operation of the spiral plate heat exchangers. The equations are developed for the three flow regimes: laminar, transition and turbulent. The hydraulic correlations used in this study are presented in **Table 1**.

2.2 Thermal equations

The thermal model equations were introduced by Minton, although Sander [17] proposed the heat transfer correlation earlier.

Eq. (3) describes the heat transfer coefficient for a liquid fluid flowing by the spiral side. Similarly, Minton presented 11 mechanisms to determine the heat transfer coefficient as a function of flow configuration, type of service (condensing and heating) and a vertical nucleate boiling. Eq. (4) is for gas fluid where the Reynolds number is higher than 10,000. Even when this number is above critical Reynolds number, gases have low heat capacity, and they have poor heat transfer coefficient values (**Table 2**).

Flow configuration	Empirical pressure drop correlation
For spiral flow without phase change $\mathrm{Re} > \mathrm{Re_c}$	(1) $\Delta P = 0.001\frac{L}{s}\left[\frac{F}{d_s H}\right]^2\left[\frac{1.3\mu^{1/3}}{(d_s+0.125)}\left(\frac{H}{F}\right)^{1/3}+1.5+\frac{16}{L}\right]$
For axial flow without phase change $\mathrm{Re} > 10,000$	(2) $\Delta P = \frac{4x10^{-5}}{sd_s{}^2}\left[\frac{F}{L}\right]^{1.8}\left[0.0115\mu^{0.2}\left(\frac{H}{d_s}\right)+1+0.03H\right]$

Table 1.
Correlations for pressure drop [16].

Flow configuration	Empirical heat transfer coefficient correlation
For spiral flow without phase change (liquid fluid) $Re > Re_c$	(3) $h = \left(1+3.54\frac{D_e}{D_H}\right)0.023cGRe^{-0.2}Pr^{-2/3}$
For axial flow without phase change (gas fluid) $\mathrm{Re} > 10{,}000$	(4) $h = 0.0144cG^{0.8}D_e^{-0.2}$

Table 2.
Correlations for heat transfer coefficient [16].

The Reynolds number and critical Reynolds number are represented by Eqs. (5) and (6):

$$Re = 10,000\left(\frac{F}{H\mu}\right) \tag{5}$$

$$Re_c = 20,000\left(\frac{D_e}{D_H}\right)^{0.32} \tag{6}$$

where F is the flow rate in lb./hr., H is the plate width in inches, μ is the viscosity in cp, De is the equivalent diameter in ft. and D_H is the spiral diameter in ft.

Eq. (7) describes the equivalent diameter:

$$D_{e=\left[\frac{(d_s H)^{0.625}{}_{1.3}}{(d_s+H)^{0.25}}\right]} \tag{7}$$

where ds is the channel spacing between plates.

2.3 Geometric additional equations

Auxiliary equations are needed to complete the thermal and hydraulic spiral plate model, for instance, the spiral outside diameter is determined by Eq. (8):

$$D_s = [15.36L(d_{sc} + d_{sh} + 2x)]^{0.5} \tag{8}$$

L is the plate length; d_{sc} and d_{sh} are the cold and cold spacing cannel, respectively; and x is the plate thickness.

The total heat transfer area is defined by the two spiral plates and is shown in Eq. (9):

$$A = H(2L) \tag{9}$$

Dongwu presented an entire description to calculate the number of spiral turns. This equation is based on plate length L, spiral semicircles, plate spacing, core diameter d and plate thickness. These geometric values are presented in Eq. (10). Due to a constant spacing of the two rolled passages, the number of turn equation gives an effective accuracy [18]:

$$N = \frac{-\left(d - \frac{t}{2}\right) + \sqrt{\left(d - \frac{t}{2}\right)^2 + \frac{4tL}{\pi}}}{2t} \tag{10}$$

where d is the core diameter of the first turn at the centre of the spiral heat exchanger and t is composed by Eq. (11):

$$t = d_{sh} + d_{sc} + 2x \tag{11}$$

The improvement of the spiral heat exchangers depends on geometrical variables. For instance, the spacing plate can increase the pressure drop if the separation between the spiral plates decreases. That means the heat exchanger needs more pumping energy, besides this, improves the heat transfer coefficient and the thermal effectiveness. The variation of the spacing channel allows to enhance the heat transfer and optimise the thermal and hydraulic performance.

Table 3 shows some recommended spacing plate values, and the values depend on the calculated plate width.

Plate spacing (m)	Plate width (m)
0.00476	0.101
	0.152
	0.304
	0.304
0.00635	0.457
	0.457
	0.609
	0.609
	0.762
	0.914
	1.219
0.00793	1.524
0.00952	1.778
0.0127	For more than 1.778 m
0.0158	
0.0190	
0.0254	

Table 3.
Recommended spacing plate values for standard plate widths [16].

2.4 Thermal performance

The validation of the method was carried out by two forms: the calculation of the NTU method and a numerical simulation with the commercial software for computational fluid dynamics ANSYS Fluent. The number of transfer units was implemented to calculate the thermal effectiveness of the spiral plate exchanger, as shown in Eq. (12):

$$NTU = \frac{UA}{CP_{min}} \tag{12}$$

where U is the overall heat transfer coefficient, A is the total heat transfer area and CP_{min} is the minimum CP for the stream with minimum heat capacity times the flow rate.

The overall heat transfer coefficient was calculated by the following equation:

$$U = \frac{1}{\left(\frac{1}{h_h} + \frac{x}{kA} + \frac{1}{h_c}\right)} \tag{13}$$

The CP ratio was set by Eq. (14) based on the hot stream $CP_h = F_h c_{ph}$ and the cold stream $CP_c = F_c c_{pc}$:

$$R = \frac{CP_{min}}{CP_{max}} \tag{14}$$

Bes and Roetzel [12, 19] reported an analytical equation to calculate thermal effectiveness. Correlation 15 contains the number of transfer units, the number of turns and the CP ratio:

$$\varepsilon = \frac{1 - e^{(R-1)NTU}}{1 - Re^{(R-1)NTU}} \tag{15}$$

The next expressions were used to calculate the hot outlet temperature and the cold outlet temperature. These equations depend on the CP ratio, effectiveness and maximum temperature difference. The procedure to calculate the hot temperature at the exit of the spiral flow is as follows:

If the $CP_h > CP_c$, then the correct form of the equation is

$$T_{hout} = T_{hin} - \varepsilon(T_{hin} - t_{cin}) \tag{16}$$

If the $CP_h < CP_c$, then the equation takes the next form:

$$T_{hout} = T_{hin} - \varepsilon R(T_{hin} - t_{cin}) \tag{17}$$

Next, the procedure to calculate the cold temperature at the exit of the axial flow is shown in Eqs. (18) and (19).

If the $CP_h < CP_c$, then the equation takes the next form:

$$t_{cout} = t_{cin} + \varepsilon(T_{hin} - t_{cin}) \tag{18}$$

If the $CP_h > CP_c$, then the correct equation is

$$t_{cout} = t_{cin} + \varepsilon R(T_{hin} - t_{cin}) \tag{19}$$

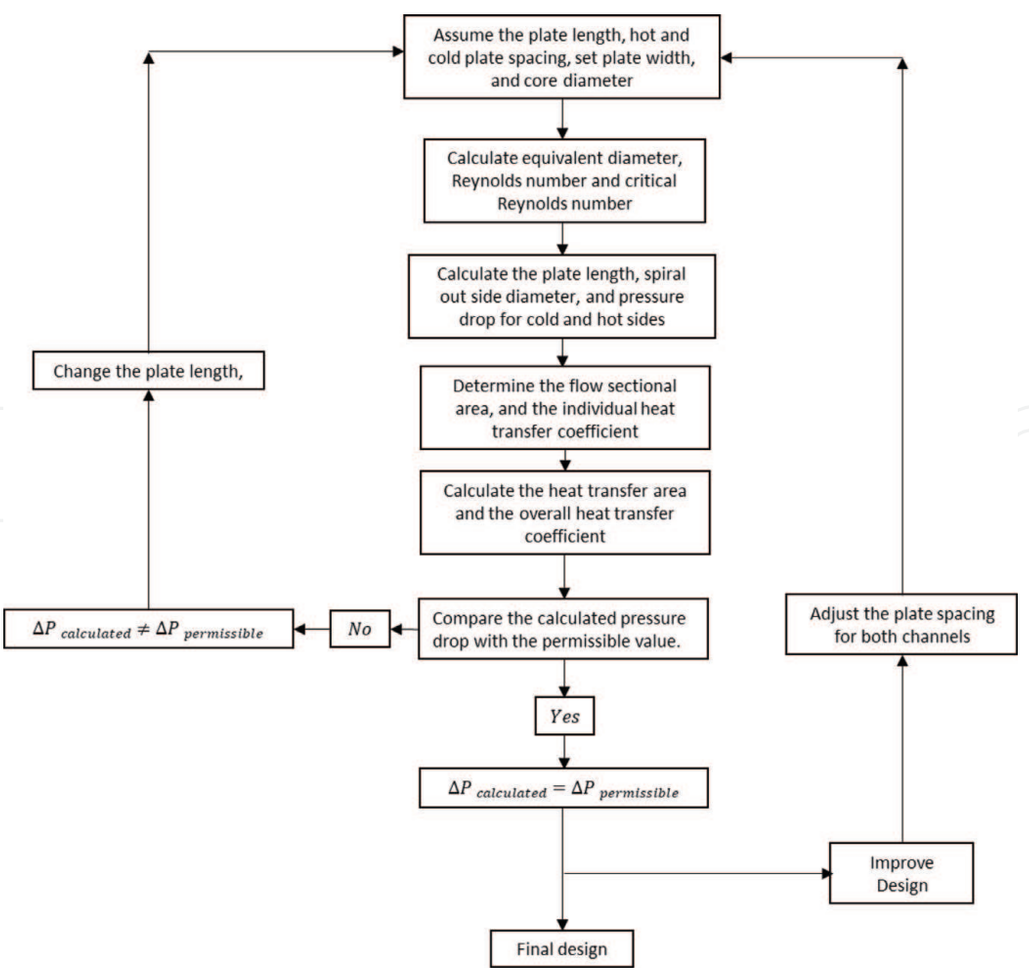


Figure 2.
Flow chart to design the cross-flow spiral heat exchanger.

	Water	Air
Total heat transfer area (m^2)	11.57	
Thermal effectiveness	0.84	
Outlet temperature (°C)	80.95	86.75
Mass flow (kg/h)	4200	5200
Inlet temperatures (°C)	98	20
Height (m)	0.41	
Length (m)	0.54	
Width (m)	0.028	

Table 4.
Car radiator design (car radiator) [20].

Figure 2 shows the sequential steps to design the spiral plate heat exchanger. A visual basic programming code was used to achieve these calculations.

2.5 Computational fluid dynamics (CFD)

The numerical simulation was performed by the software ANSYS Fluent. The mathematical model κ-ε solved the three balances: mass, energy and momentum.

Conservation of mass:

$$\nabla \cdot (\rho v) = 0\nabla \tag{20}$$

Conservation of momentum:

$$\nabla \cdot (\rho vv) = -\nabla P + \nabla \cdot (\mu \nabla v) \tag{21}$$

Conservation of energy:

$$\nabla \cdot (\rho vcpT) = \nabla \cdot (k \nabla T) \tag{22}$$

where k is the thermal conductivity, ρ is the density, μ is the dynamic viscosity, cp is the specific heat, v is the velocity, T is the temperature and P is the pressure (**Table 4**).

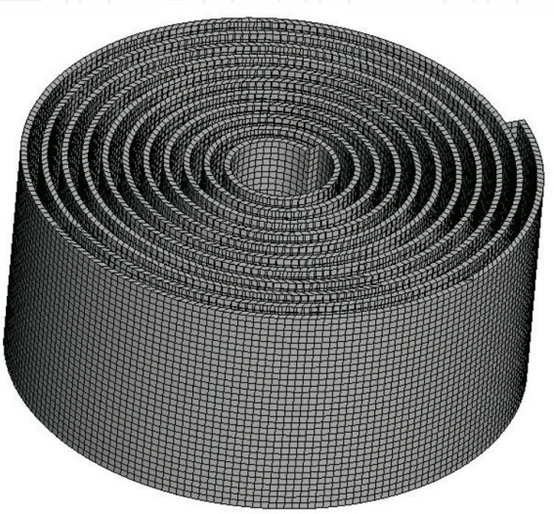


Figure 3.
Mesh of the spiral plate heat exchanger.

	Water	Air
Flow (kg/h)	4200	5200
Inlet temperatures (°C)	98	20
Viscosity (cp)	0.3	0.2
Maximum pressure drop (kPa)	6.89	6.89
Thermal conductivity (W/m K)	0.6	0.0259
Heat capacity (J/Kg K)	4270.53	544.28
Thermal conductivity (aluminium, W/m K)	205	
Plate width (m)	0.152	0.152
Core diameter (m)	0.0508	
Specific gravity	0.97	1
Channel spacing (m)	0.0043	0.0088
Plate thickness (m)	0.0787	

Table 5.
Second case study.

Figure 3 shows the virtual spiral plate heat exchanger designed by the software workbench, the mesh was structured with 97,250 control volumes and the geometrical features (plate width, plate spacing, plate length, thickness, etc.) are shown in **Tables 5** and **8**.

The boundary conditions such as the inlet velocity, inlet temperatures, fluid properties, flow rates and metal properties were considered from the results shown in **Tables 5** and **8**.

3. Case study

In order to demonstrate the rating and the design method, a case study was proposed. The example consists of two streams, water (hot stream) and air (cold stream), where the hot fluid flows by a spiral channel and the cold fluid flows through a cross-flow arrangement. **Table 5** explains the operational conditions for the first case study. This data was taken from a normal operation of a car radiator where they proposed to design an equipment to cool down a hot stream by 17 degrees. **Table 4** shows the operational data for the car radiator.

Table 5 summarizes the operational conditions. The plate width is set by a numeric value of 0.1524 m. The initial plate spacing for the hot stream is 0.0043 m and for the cold stream is 0.0088 m, and the core diameter is 0.0508 m.

The second case study consists of designing a condenser for a cryogenic process. The data were taken from a case study to design a compact heat exchanger [13].

The hot temperature must cool down by 9°C to transfer a latent heat to the cold fluid. The cold temperature is 15°C. The initial core spiral diameter was 0.5 m. A plate width of 0.076 m was assumed to find the optimal plate length for the spacing plates of 0.00254 and 0.00635 m for cold and hot stream, respectively.

4. Results and discussions

The new spiral was sized by considering the same instal space than the actual radiator. The features were 0.4 m of height, 0.15 m of depth and 0.4 m of width.

A suggested car cooling system performance must cool down the hot fluid 10°C or maximum 20°C. The results were achieved by implementing the thermal and hydraulic model, where one of the fundamental variables to measure was the hot outlet temperature. The difference between hot inlet and hot outlet was 6.4°C. The thermal and hydraulic method is an option to design heat exchangers. One of its goals is to find the lowest heat transfer area because the permissible pressure drop was fixed as a parameter to use completely. Then, from this value, the method seeks for some geometrical configuration to satisfy the required pressure drop. The method determined the heat transfer area, and the numeric value was of 2.04 m^2 (**Tables 6** and 7).

The permissible pressure drop was set by 1 psi for the two passages. The simulation calculated a value of 0.975 psi for the hot channel. The value for cold channel was 0.00013 psi because the cold channel has a length of 0.15 m. This flow section was considered as an open channel. This hydraulic behaviour was determined by the spiral diameter of 0.387 m, a plate length of 6.7 m, 10 spiral turns and the plate

	Water	**R134a**
Flow (kg/s)	1	0.38
Inlet temperatures (°C)	15	42.14
T_{out} (C)	31.33	33.65
Viscosity (cp)	1.1	0.175
Maximum pressure drop (kPa)	150	950
Thermal conductivity (k, W/m K)	0.61	0.0763
Heat capacity (J/kg K)	4528.02	1563.76
Thermal conductivity of steel (W/m K)	13	
Plate width (m)	0.0762	
Core diameter (m)	0.508	
Specific gravity	0.97	
Channel spacing (m)	0.00254	0.00635
Plate thickness (m)	0.003175	

Table 6.
Second case study [13].

	Water		**Air**
T_{out} (°C)	6.72		60.7
Pressure drop (kPa)	0.975		0.00068
Area (m^2)		2.04	
Diameter (m)		0.378	
HTC (W/m^2 K)	12227.62		305
U (W/m^2 K)		52.41	
Effectiveness		0.52	
Number of rounds		10	
Plate length (m)	6.7		6.7

Table 7.
Results for case study 1.

spacing for both channels. To achieve a pressure drop close to the ideal value, it should be necessary to increase the plate length, but the spiral diameter would be higher than the delimited value of 0.4 m. The thermal performance remained with no significantly variations.

For the improvement of the spiral design, only the plate spacings for the cold stream and the hot stream were reduced to 4 and 4.2 mm, respectively. The results are shown in **Table 8**.

The results in **Table 8** show that the thermal and hydraulic performance was improved, by decreasing the plate gap, because the flow distribution along the channel enhances the heat transfer, and the hot outlet temperature increased the difference by 9.4°C. The available pressure drop for the hot stream was used fully, while the cold pressure drop presented the variation of 0.00072 psi. The heat transfer area continued as 2.04 m, the number of turns raised by 1.5 rounds and the spiral outside diameter was modified to 0.32 and 0.06 m lower than the first case study.

The results for the second design were compared with the compact heat exchanger design, and they are shown in **Table 9**.

The results are similar regarding the hot outlet temperature; however, there was a significant difference between the heat transfer areas. The compact heat exchanger was designed to use fins to increase the thermal effectiveness.

	Water	Air
T_{out} (°C)	88.6	79.4
Maximum pressure drop (kPa)	6.89	0.004826
Area (m^2)	2.04	
Diameter (m)	0.32	
HTC (W/m^2 K)	12577.35	628.41
U (W/m^2 K)	105.37	
Effectiveness	0.76	
Number of rounds	11.5	
Plate length (m)	6.7	6.7

Table 8.
Results for an improvement design.

	R134a	Water	Compact HE
T_{out} (°C)	31.72	16.7	32.85
Pressure drop (kPa)	0.875	5.38	84.5
Area (m^2)	0.0929		1.46
Diameter (m)	0.120		
HTC (W/m^2 K)	69098.78	18091.41	
U (W/m^2 K)	3185.5		
Effectiveness	0.38		
Number of rounds	2.43		
Plate length (m)	0.609		

Table 9.
Results for a condenser design.

The compact heat exchanger was implemented to handle three stages: superheated phase, condensed phase and subcooling phase. The spiral plate heat exchanger has the same purpose, which is to condensate the refrigerant and cooling until the target temperature is reached. Though the compact exchanger has more heat transfer area than spiral exchanger, the compact used completely its maximum area to achieve the duty, but the spiral heat exchanger achieved the service using less heat transfer area. This feature is demonstrated by a pressure drop for R134a, because the maximum pressure drop was fixed to 6.89 kPa. If the pressure drop increased, the result will be a larger spiral exchanger. The hot outlet temperature could be minor than 31.72°C. These results validate the use of spiral plate heat exchangers as a part of a cryogenic process.

Spiral plate heat exchangers have a potential participation in the cryogenic process. Heat exchangers are 30% approximately of the total cost of the cryogenic plant [14, 15]. Only two types of heat exchangers are considered for cryogenic applications: tubes (concentric tubes and coil wounded tubes) and plates (perforated plates and plate-fin) [14]. The principal duty that must fill the heat exchangers in the refrigeration process is the high thermal efficiency. Then, spiral plate heat exchangers are capable of dealing with that situation. The thermal and hydraulic model presented in this work showed a suitable precision since the procedure does not support the traditional design such as shell and tube heat exchanger. It is possible to size spiral plate exchangers even with poor flow distribution and axial thermal behaviour and think forward to design a multi-stream spiral plate heat exchanger for cryogenic challenges.

The numerical and computational results were collected by colours which they represent a profile of the measured variables, the inlet and outlet temperatures mainly. Colour red means the hottest temperature and colour blue represents the coldest temperature. **Figure 4** describes the temperature profile for the water and the air and the arrangement of the inlet and the outlet streams. The hot liquid (water) enters at the centre of the spiral moving out along the plate, and the cold stream (air) crosses the plate width.

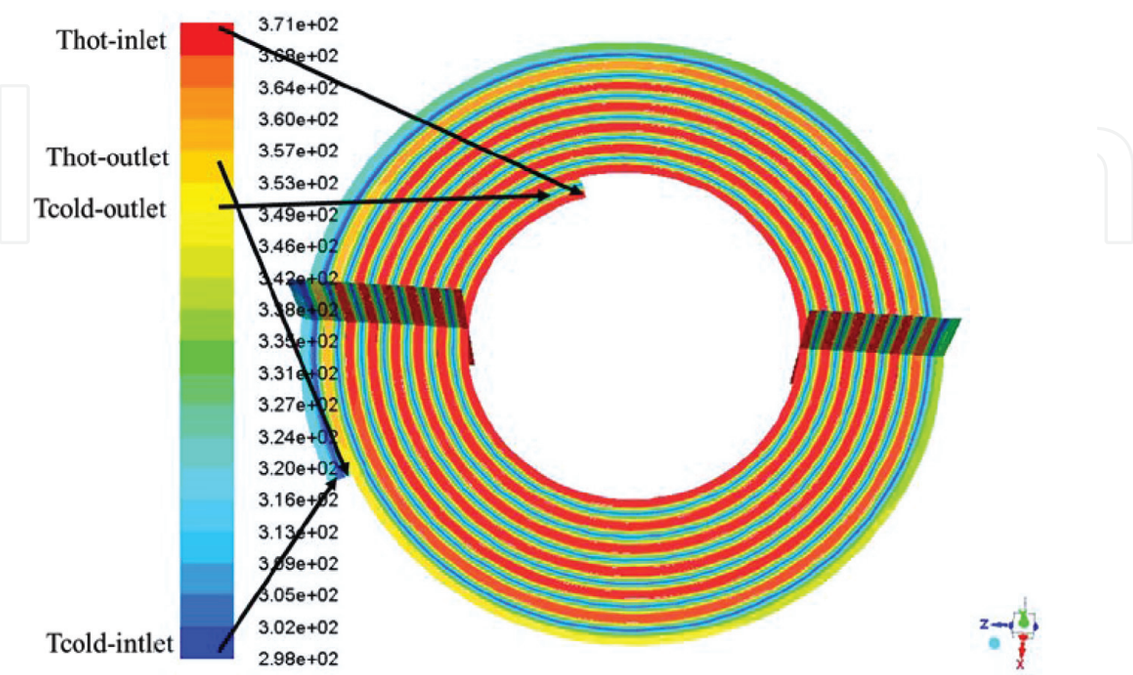


Figure 4.
The temperature profile for the cold and the hot streams.

	CFD		Method	
	Water	**Air**	**Water**	**Air**
T_{in}	98	20	98	20
T_{out}	86.85	75.85	88.6	79.4
Pressure drop	17.23	60	6.89	0.004826

Table 10.
Thermal and hydraulic performance.

The numerical study determined an appreciable accuracy between the method design and the computational simulation. **Table 10** shows the approximation of the outlet temperatures. The hydraulic performance was measured by calculating the outlet pressure for both streams. The results determined that the maximum pressure drop was observed at the hot section. The design method calculated a pressure drop of 6.89 kPa, and the numerical result was 17.23 kPa, because the spiral flow section has the less wide spacing between the plates. The minimum pressure drop was expected for the cold stream. The method reported 0.004826 kPa. The simulation determined a value of 3.4 kPa, due to the cross-flow section that has a spacing wider than the spiral section and furthermore because the channel is open.

5. Conclusions

This work presented two new methods to design cross-flow spiral plate heat exchangers: the first design was to compare the thermal behaviour of the spiral plate heat exchanger versus a car radiator. The radiator is an option to remove the excess of heat; however, this device needs to increase the heat transfer area by installing fins to dissipate the heat, specifically when a gas phase is involved. The constant spiral movement promotes an effective heat transfer, even when a laminar regimen is observed. An additional thermal performance can be proposed by decreasing the spacing plates. The spiral outer diameter will be reduced, and the flow velocity will increase; nevertheless, more energy to pumping flows will be necessary. The studs have not represented a constraint for the flows, but more studs could be installed with a higher diameter to increase heat transfer area and optimise the subcooling zone in spiral plate condensers. The spiral plate heat exchanger has many industrial applications, and this study contributed to expanding the usages by implementing new simple methodologies for cross-flow arrangement particularly for a condenser and for a cooler.

Computational fluid dynamics is a robust instrument to simulate and validate the empirical methods. The results between CFD simulation and the design method approve the accuracy of the method. It allows to extend the service of the spiral plate heat exchangers, as a part of the industrial process, cooling systems, heat networks and recover energy.

Conflict of interest

All authors have contributed in (1) the proposal and design or analysis and interpretation of the data, (2) drafting the article or revising it critically for important intellectual content and (3) the approval of the final version.

This manuscript has not been submitted to, nor is under review at, another journal or other publishing institutions.

Nomenclature

A_c	free flow area (m^2)
A_p	plate area (m^2)
A	heat transfer area (m^2)
C	core diameter (m)
C_p	heat capacity (J/Kg K)
D_e	equivalent diameter (m)
D_h	helix diameter (m)
d_s	spacing plate (m)
F	flow (kg/hr)
h	film heat transfer coefficient (W/m^2 K)
H	plate width (m)
k	thermal conductivity (W/m K)
L	plate length (m)
Pr	Prandtl number
q	heat load (W)
Re	Reynolds number
Re_c	critical Reynolds number
s	specific gravity
T	temperature (°C)
U	overall heat transfer coefficient (W/m^2 K)
G	fluid velocity (kg/hr. m^2)
x	plate thickness (m)

Greek symbols

ΔP	pressure drop (kPa)
μ	viscosity (cp)

Subindices

h	hot side
c	cold side
in	inlet
out	outlet
f	film fluid properties
b	bulk fluid properties

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