Impact of wave inclination on enhancement heat transfer for a wavy-fin compact heat exchanger with circular and elliptical tubes for staggered tube arrangement

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Abstract. With the increasing industrial development, the improvement of several processes and devices represents a great need, especially for thermal devices. The present work evaluates the enhancement of heat transfer in a compact heat exchanger with wavy-fins, operating at Reynolds number of 180 to 900, through modified inclination of wavy-fin in relation to the main flow direction. A 3D numerical modeling considers incompressible, turbulent and steady-state flow. The compact heat exchanger is evaluated for staggered circular and elliptical tubes. The wavy inclination investigated are -25 °, -20 °, -15 °, -10 °, 10 °, 15 °, 20 ° and 25 °. The heat transfer is performed by the Colburn factor (j) and the pressure loss penalty is performed by the friction factor (f). Compared to the conventional heat exchanger (θ = 0 °), the results showed an increase in the heat transfer of 13.5% with elliptical tubes and 7.8% with circular tubes at wavy inclination of -25 °. For the friction factor, all inclinations led to a decrease, with a reduction of 18.8% with circular tubes and 15.1% with elliptical tubes. Overall, the wavy inclination proposed is an effective alternative to both increase the heat transfer and reduce the pressure losses.

Keywords: Compact Heat Exchangers; Wavy Fin; Numerical Simulation, Enhancement Heat Transfer.

1 Introduction

There is a wide range of applications for heat exchangers, being present in industrial processes and equipment until domestic devices daily used. In the industrial applications, they are utilized directly for heat and cooling processes in product manufacturing and also as condensers and evaporators, according to Incropera et al [1]. Damavandi et al [2] stated that the need for efficient processes and devices to avoid waste, led to the development of several intensification techniques to improve the heat transfer with low/moderate pressure drop penalty.

Heat transfer intensification techniques can be classified as passive and active. Passive techniques do not use any external energy source, it uses geometrical modifications and or insertion of elements to impact the flow field, creating distortions, altering the flow dynamics, and therefore, increasing the convective heat transfer coefficient as stated in the work of Wang et al [3]. These passive technique devices and geometrical modifications interrupt the dynamic and the thermal boundary layer growth, which is one of the main goals of these techniques, since it increases the mixture between heat and cold streams, increasing the heat transfer.

The utilization of extended surfaces, known as fins, represents a way to increase the heat transfer since it improves the heat flux and also increases the performance of the heat exchanger, especially when its necessary to reduce the high thermal resistance according to Wang et al [3]. Thus, these surfaces are widely used due to the direct influence on the heat transfer rate, where the fins not only increase the contact area and the heat transfer, but also have a high influence on the flow dynamics. Wang et al and Ke et al [4, 5] stated that the main objective in designing efficient heat exchangers is to obtain the greatest heat transfer with the least possible pressure loss and material consumption, seeking for lighter, cheaper and more compact designs. However, the alternatives for increasing the heat transfer coefficient associated with a compact heat exchanger project are many, since geometric changes to the insertion of accessories. In this context, wavy fins have been used, combined with staggered and inline tube arrangements, using different tube profiles, and in some cases combined with the indroduction of vortex generators (VG's).

According to Kim et al [6], the wavy fins, represents a technique widely explored in the scientific community, since they promote heat transfer increase in an effective way. Kim et al [7] define a corrugated fin as a surface capable of periodically changing the direction of the main flow and intensifying the mixing of the fluid. The increase in the heat transfer rate would then be a function of this change in flow dynamics and also of the increase in the area, but also resulting in an increase in the pressure drop.

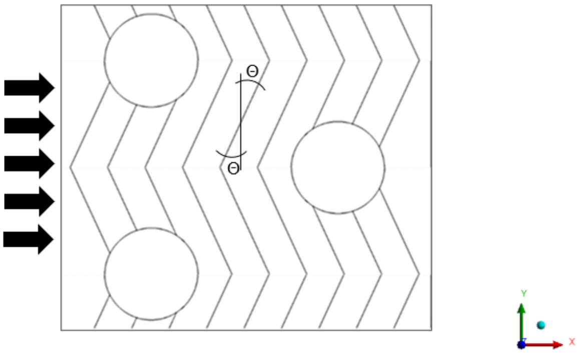


Figure 1. Top view of the Heat exchanger showing the flow direction with θ = -25°.

From extensive experiments done by Picon-nuñez et al [8] . It was concluded that there is a 50 to 70% increase in the heat transfer coefficient for wavy geometries when compared to a flat fin geometry. Elshafei et al [9], conducted studies on channel spacing and its effects on heat transfer. Their results were compared to those of conventional flat-finned exchangers and their conclusion was that waviness improves heat transfer. Xu et al [10] carried out an experimental work aiming at heat transfer improvement using corrugated fins and showed that if the Webb [11] PEC criterion, which evaluates the thermo-hydraulic performance of the equipment, is greater than 1.0, there is a significant increase in heat transfer in relation to the associated head loss.

Paras et al [12] showed that there is an improvement in heat transfer and flow distribution using corrugations in the fins of the heat exchanger. They also conducted studies on compact corrugated cross plates; they found that the Nusselt number increases as the angle between the corrugated plates is increased. The ripples in the plates produce swirling complex flows and generate an increase in pressure drop. Paras et al [13] found that the Reynolds numbers were higher in the regions of crests and lower in the regions of valleys. Their results confirmed that the ripples play an important role in the flow distribution and in the increase of heat transfer, similar results were also shown by Damavandi et al [2], who used a comparison between a wavy-fin heat exchanger and a flat-fin heat exchanger, both with elliptical and staggered tubes. They observed that the presence of the corrugations had a greater influence on the Nusselt distribution in relation to the tube arrangement, which had a greater role in the flat fin heat exchanger. Lyytikäinen et al [14] from their numerical studies of different corrugation angles and spacing found that heat transfer increases as the corrugation angle increases. They concluded that it is not easy to find a specific geometry that provides less pressure loss with greater heat transfer simultaneously.

The inclination of the wave propagation direction in relation to the flow direction is not reported in many studies, and Singh et al [15] , experimentally studied the effect of the change in the direction of the waves using several plates with corrugations in different directions, ranging from 0 ° to 80 °, using crossed water currents, for a Reynolds range of 400 to 600, and found that the optimum value is 20 °.

Many researchers proposed the use of fin corrugations to enhance the heat transfer, though to increase of the total area and to change the flow dynamics. However, few works have been investigated the influence of the wavy inclination in relation to the main flow direction on heat transfer rate. Therefore,from the extensive literature analyzed, in the present work, the impact of wavy inclination has been numerically studied and the main characteristics of the thermal and dynamic flow are investigated in detail considering staggered circular and elliptical tubes. The Reynolds number range is considered from 180 to 900, based on Fin pitch, which is commonly associated to refrigeration applications. Fig.1 shows the modifications in the waves used for this work, where the angle θ is initially at 0° inclination, wich is the standard wavy-fin compact heat exchanger, and is modified to 10°, 15°, 20°, 25°, -10°, -15°, -20° and -25° in relation to the main flow direction, indicated by an arrow.

2 Governing equations and thermal-hydraulic parameters

For the present work, a numerical modeling is considered under the hypothesis of tridimensional, steady-state, turbulent and incompressible flow in accordance to Salviano et al [16]. For Newtonian fluid with constant properties, the equations of conservation of mass, momentum and energy are presented as stated by Salviano et al [17]:

$$\frac{\partial\left(\rho u_j\right)}{\partial x_j} = 0 \tag{1}$$

$$\frac{\partial}{\partial x_j}\left(\rho u_j u_i - \tau_{ij}\right) = -\frac{\partial p}{\partial x_i} \tag{2}$$

$$\frac{\partial}{\partial x_j}\left(\rho u_j h - k\frac{\partial T}{\partial x_j}\right) = -u_j\frac{\partial p}{\partial x_j} + \tau_{ij}\frac{\partial u_i}{\partial x_j} \tag{3}$$

where: u is the velocity component, h is the convective heat transfer coefficient, x_{i} and x_{j} are generalized coordinates, p is the pressure, τ_{ij} is the tension tensor, ρ is the density, k is the thermal conductivity and T is the temperature.

To solve the governing equations a finite volume-based commercial software (ANSYS Fluent 19.0) is used, as showed in Launder et al [18]. The turbulent flow is considered for the present simulation since even though the application range is at low Reynolds number, there are instability effects generated by the secondary flow introduced by the geometrical modifications such as the corrugations, inclination and tube shape. The turbulence closure method used is the k-omega SST (Shear- Stress Transport), which considers the enhanced wall treatment as default, this method is one of the most wide used by researchers as stated by Menter et al [19], which is a model robust for flows with high adverse pressure gradients as mentioned by Wilcox [20]. Coupled Algorithm is adopted to perform the pressure–velocity coupling, this method is widely used for this application, as stated by Rezaeiha et al [21]. The advective terms are interpolated by a second order discretization method. Finally, the computational convergence is ensured by the residuals being lower than 10^{-5} for the continuity and momentum equations and 10^{-7} for the energy equation.

The thermal-hydraulic parameters to calculate heat transfer and pressure drop are based on Colburn factor (j) and the Friction factor (f), respectively, and the flow is characterized by the Reynolds number (Re). The Reynolds number, Colburn factor (j) and the Friction factor (f) are listed below:

$$Re = \frac{\rho U_{in} F_p}{\mu} \tag{4}$$

$$f = \frac{\Delta P}{\frac{1}{2}\rho U_{in}^2}\frac{F_p}{L} \tag{5}$$

$$j = \frac{h}{\rho u_{máx} c_p} Pr^{2/3} \tag{6}$$

To compare circular and elliptical tubes, the Reynolds number is calculated using the Fin Pitch (F_p) ,in accordance to the work of Deepakkumar et al [22], since the minimum flow area is a function of the tube eccentricity. Thus, the total heat transfer, pressure loss and heat transfer coefficient are defined as mentioned in the work of Salviano et al [17].

$$Q = \dot{m} c_p \Delta T \quad = \dot{m} c_p (\bar{T}_{out} - \bar{T}_{in}). \tag{7}$$

$$\Delta p = \bar{p}_{in} - \bar{p}_{out}. \tag{8}$$

$$h = \frac{Q}{A_t \, \Delta T_{ln}} \tag{9}$$

where:

$$\Delta T_{lin} = \frac{(T_W - \bar{T}_{in}) - (T_W - \bar{T}_{out})}{ln[\frac{(T_W - \bar{T}_{in})}{(T_W - \bar{T}_{out})}]}. \tag{10}$$

$$\bar{p} = \frac{\iint_A \ p dA}{\iint_A \ dA}. \tag{11}$$

$$\bar{T} = \frac{\iint_A \ uT dA}{\iint_A \ u dA}. \tag{12}$$

2.1 Computational domain and boundary conditions

Numerical simulations could take a huge of computational time to solve all governing equations for large computational domain. To avoid excessive expenses for the numerical modeling, it is considered a compact wavy-fin heat exchanger with two tube rows in staggered arrangement under symmetry conditions. Fig. 1 shows this computacional domain with three parts, extended upstream and downstream regions and the main domain. The upstream region is extended one times the main domain length to ensure unifom flow at inlet and also to avoid flow instability. The downstream region is extended 7 times the main domain length to ensure the Neumann's boundary condition. The non-slip conditions and constant temperature is assumed on fins and tubes. The model is based in the heat exchanger proposed by Damavandi et al [2], with modifications in the main domain in order to model the wave inclinations in relation to the main flow direction. Circular and elliptical tubes parameters were modeled to have the same perimeter in order to ensure constant heat transfer area.

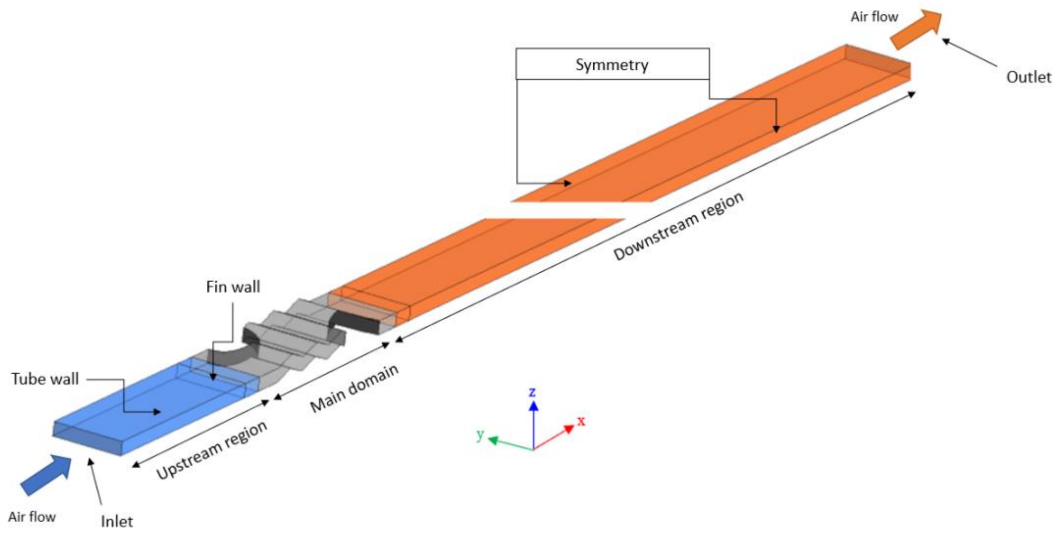


Figure 2. Computational domain of the compact heat exchanger model.

3 Numerical validation and grid independence study

For the grid independence study, the methodology proposed by Celik et al [23] is performed, considering the Grid Convergence Index (GCI), considering three different mesh densities. Tab.1 shows the results for Colburn factor (j) and Friction factor (f) for both lower and upper Reynolds number investigated.

Table 1. GCI results for three evaluated mesh refinements.

	Cells number (Main Domain)	Refinement index, r	GCI_{32} (%)			
			Re = 800		Re = 4000	
			j	f	j	f
Grid 1 (h1)	462.510	-	1.35	3.38	1.53	1.04
Grid 2 (h2)	1.431.136	1.457				
Grid 3 (h3)	3.182.755	1.305				

For the GCI analysis and the validation, the Reynolds number is based on the collar diameter, D_c, in a different approach than the proposed in Eq.4.

The recommendations proposed by the method were achieved, with the highest uncertainty being 3.38% for the Friction factor (f) at Re = 800. Therefore, the mesh density study showed that the intermediate mesh can be used.

Another important parameter to check is the average and maximum y-plus values according to Launder et al [18], as shown in Tab.2. As ca be seen, those lower values are adequate to the model SST-K-omega, which should around the unity as mentioned by Salviano et al [16].

Table 2. Grid y-plus values.

Meshes	Average y^+		Maximum y^+	
	Re = 800	Re = 4000	Re = 800	Re = 4000
Refined (h3)	0.5	1.2	0.7	1.4
Intermediate (h2)	0.3	1.1	0.4	1.2
Coarse (h1)	0.2	0.8	0.3	1.0

The numerical modeling validation was made by comparing the numerical results with values from correlations proposed by Wang et al [3]. These correlations describe 100% of the Colburn factor with a deviation of 15%; while the friction factor described 99% of experimental data with a deviation of 20%.

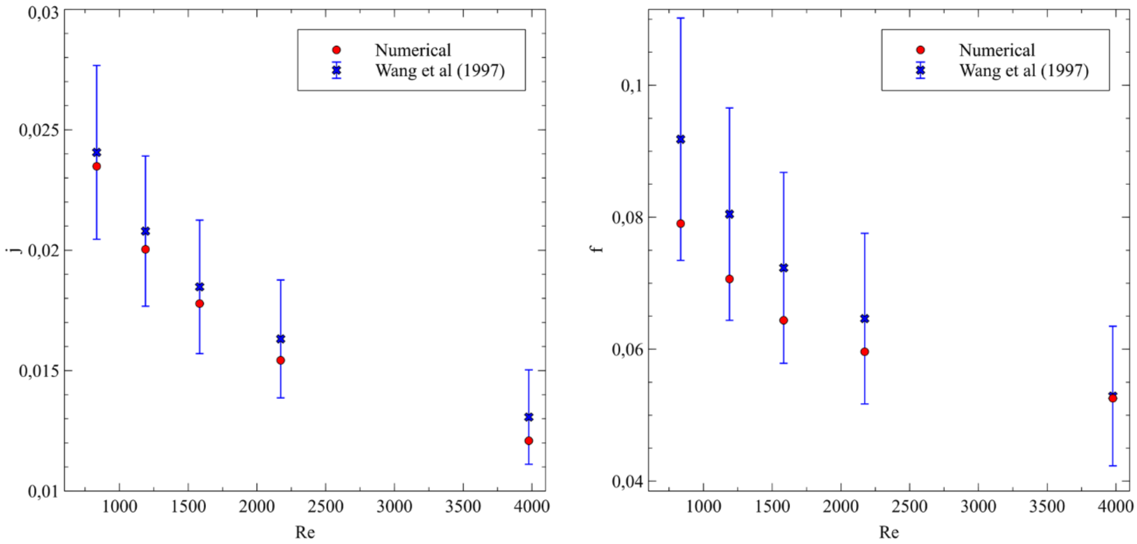


Figure 3. Validation comparison between the numerical results and Wang's correlations.

Fig. 3 shows that the maximum deviations were found to be 14% for Friction factor (f) at lower Reynolds number and maximum deviations of 7.5% for the Colburn factor (j) at higher Reynolds number. Thus, the numerical results are within the experimental uncertainties, showing that the numerical approach is suitable for the further analysis.

4 Results and discussions

The results of the numerical simulations performed on the model presented in Fig.2 are shown in Fig.4. The results are divided for circular tube and elliptical tubes, for wavy inclinations as shown in Fig 1. The Reynolds number range is 180 to 900 based on Fin Pitch (F_p).

Fig. 4(a-b) indicates that the highest Colburn factor (j) values are verified for negative wavy inclinations. In this aspect, the highest Colburn factor (j) is found for wavy inclination of -25° for both circular and elliptical tube, although the highest difference occurred for the elliptical tube, showing increases up to 16%. In other hand, the 25° showed the lowest values for both tube profiles, remaining under the conventional case (θ = 0 °) for all the Reynolds number investigated range.

Moreover, it is observed increases in the Colburn factor(j) up to 8% for the wavy inclination of -25° for circular tubes and 16% for elliptical tubes, and decreases in the friction factor(f) up to 10% for the wavy inclination of 25° for circular and 8% for elliptical tubes.

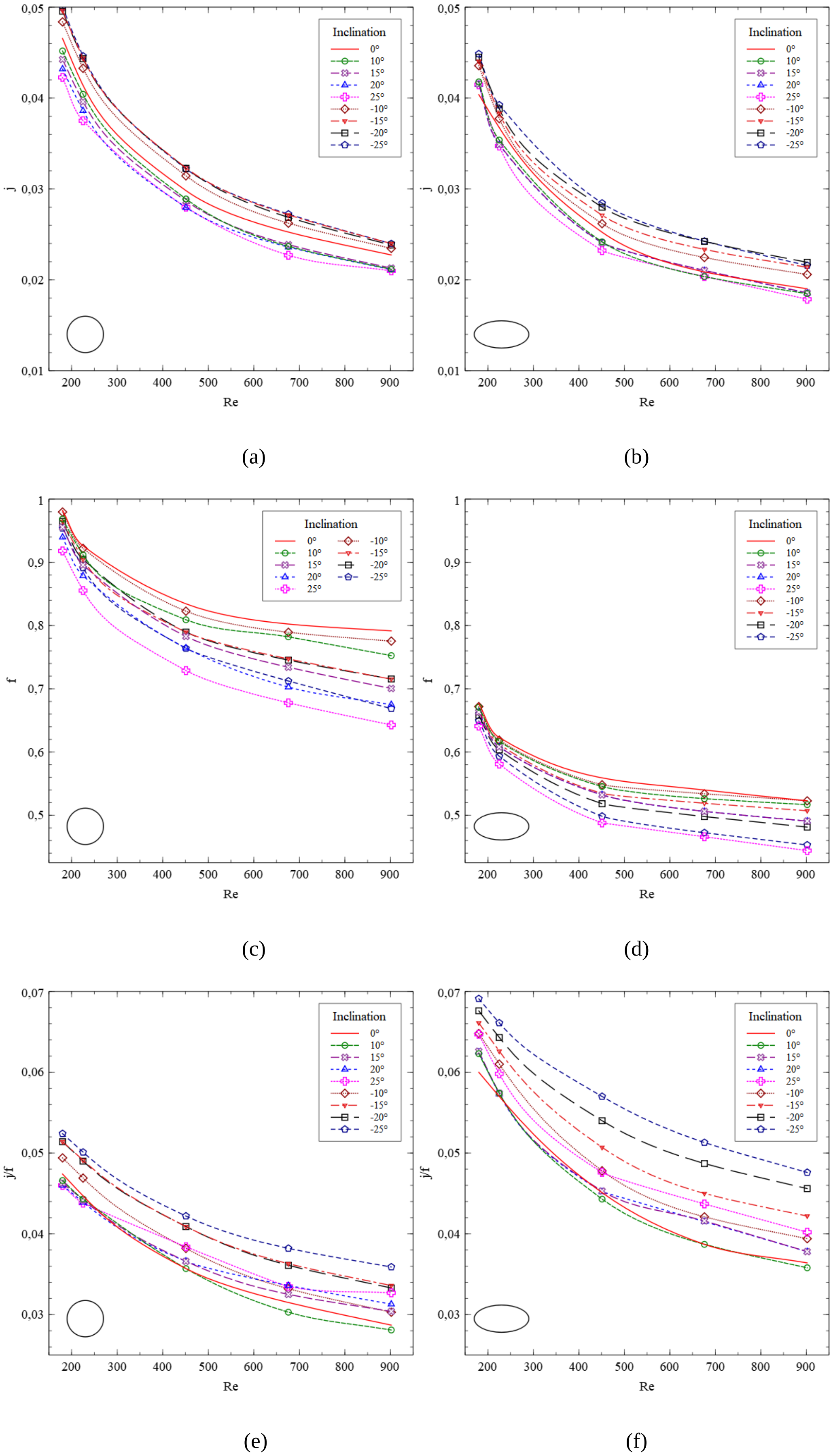
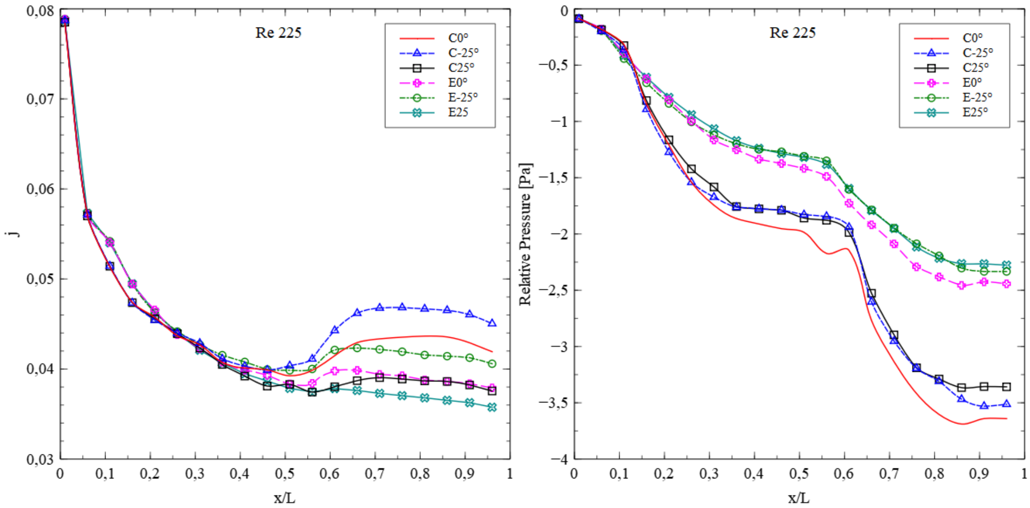


Figure 4. Graphics of the Colburn j-factor (a,b), Friction f-factor (c,d) and j/f-ratio (e,f).

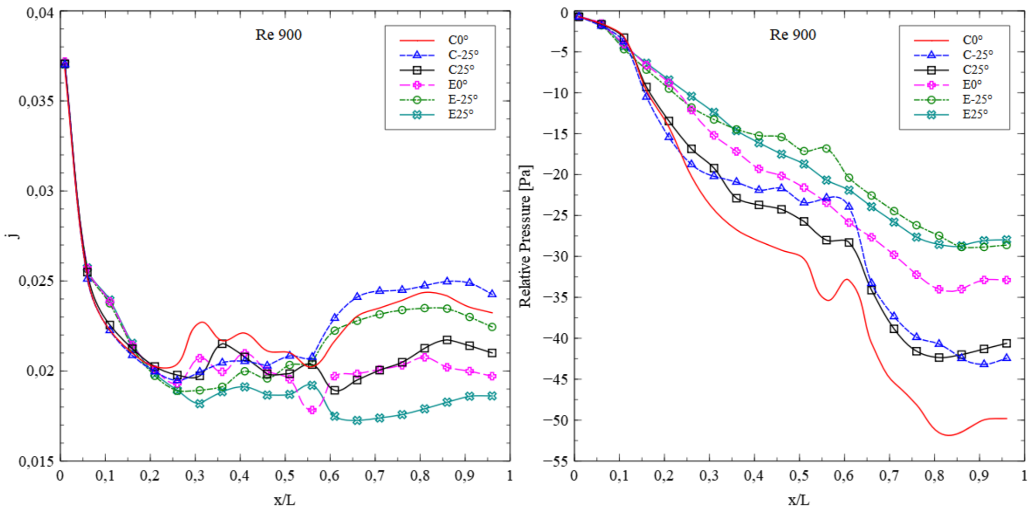
Fig. 4(c-d) shows that all wavy inclinations provided decreases in the Friction factor (f), especially for wavy inclinations of 25° and -25°. This behavior shows that the higher the wavy inclinations, the higher is the decrease in the flow losses. This is due to the fact for the conventional case (θ = 0°) the waves are in a perpendicular condition in relation to the main flow direction, representing the highest shape resistance. According to Singh et al [15], for inclinations closer to θ = 90°, the waves become almost parallel to the flow direction, forming tube like structures, decreasing the pressure losses, since no additional resistance is imposed to the flow passage. However, heat transfer rate is also decreased due to the insignificant fluid mix cause by this kind of structure.

The lowest values related to friction factor (f) can be seen for wavy inclination of 25°, that decreases up to 19% for circular tubes and up to 15% for elliptical tubes, in accordance to the lower Colburn factor values presented for this same angulation in Fig. 4(a-b). For the negative wavy inclintation of -25°, decreases up to 15% for circular tubes and up to 13% for elliptical tubes are observed. For the j/f-ratio in Fig. 4(c-d), it can be observed that the wavy inclination of 25° had the lowest Colburn Factor (j), but also the lowest Friction factor (f), resulting in a j/f-ratio up to 14% higher than the conventional wavy-fin heat exchanger for circular tubes and up to 13% higher for elliptical tubes. This represents the best performance of the positive wavy inclinations. For the negative wavy inclinations, the -25° inclination is highlighted. At this inclination, j/f-ratio increases up to 25% for circular tubes and up to 33% for elliptical tubes. Overall, it can be noted that the elliptical tubes showed lower levels of the friction factor, and higher j/f performance ratio, since these tubes are known for generating smaller wake regions than their circular counterparts, however this fact leads to lower fluid mix capabilities and heat exchange levels. Therefore, after this analysis, it is evident that the inclinations of 25° and -25° showed interesting results, with the highest friction factor (f) decrease occurring for 25° and the highest Colburn factor (j) increase occurring at -25°, where both inclinations also increased the j/f performance ratio for both tube profiles at low and high Reynolds number.

Based on previous discussion, additional analysis is carried out for the wavy inclination of -25° and 25°, since they showed interesting behaviors among the inclinations adopted for positive and negative directions.The Colburn factor profiles (j) and the relative pressure along longitudinal direction *x/L* of the computational domain is shown in Fig. 5 for both circular and elliptical tubes. The prefix C and E are related to circular and elliptical tubes, respectively, at Reynolds numbers of 225 and 900.



(a)



(b)

Figure 5. Colburn factor (j) and Relative pressure in the main domain, for Re = 225 (a) and Re = 900 (b).

It can be seen that for wavy inclination of -25° is capable to enhance the heat transfer for both tube profiles, especially at high Reynolds, which is in agreement with the global results shown in Fig. 4(a-b). However, it is also verified that the enhancement heat transfer is persistent on downstream after *x/L = 0.5*, showing an increase in the region that the conventional case has its highest heat transfer behavior. At high Reynolds number, the effects in the region after the first tube at *x/L = 0.1* to the middle at *x/L = 0.5*, the conventional case showed better levels of the Colburn factor (j), indicating that in this region the -25° is not so effective. On the other hand, the wavy inclination of 25° shows the same behavior, but with lower levels in the present peaks in relation to the conventional case. However, for the wavy inclination of -25° these difficulties is compensated atthe final region of the heat exchanger, showing higher Colburn factor (j) levels than the peaks present in the conventional heat exchanger, with increases up to 6% for circular tubes and 16% for elliptical tubes at high Reynolds. For low Reynolds, the wavy inclination of -25° showed the best performance with increases up to 8% for circular tubes and 7.5% for elliptical tubes, showing that this case was able to reduce the difficulties to enhance the heat transfer at low Reynolds number.

For the relative pressure profiles, the behaviors are the same for both Reynolds numbers. The elliptical tube profile cases showed the lowest levels of the pressure penalty, since it has an advantage compared to circular tubes. The behavior shown in Fig. 5 indicated that the both wavy inclinations showed pressure drop attenuation when compared to the conventional heat exchanger. Moreover, for the wavy inclination of -25° is slighter superior than the 25° for low Reynolds. For circular tubes at high Reynolds, in the region from *x/L=0.3* to *x/L=0.6*, the pressure drop for the 25°inclination is up to 23% higher than the -25° incination, representing the highest difference between this cases. At low Reynolds in Fig 5(a), for circular tube cases the highlight is the region downstream of *x/L= 0.35*, wich shows decreases in the pressure drop up to 15%, and for elliptical tube cases the highlight is the region downstream of *x/L=0.15*, with decreases in the pressure drop up to 9%. At high Reynolds in Fig 5(b), for circular tube cases the highlight is the region downstream of *x/L= 0.35*, wich shows decreases in the pressure drop up to 35%, and for elliptical tube cases the highlight is the region downstream of *x/L=0.15*, with decreases in the pressure drop up to 28%.

Overall, it is possible to observe that both wavy inclinations are interesting to decrease the pressure loss penalty, according to show in Fig. 4(c-d). However, the wavy inclination of -25° showed higher heat transfer enhancement capabilities for both tube profiles, especially for the elliptical ones. Furthermore, this can be underlined by flow dynamics, which can be seen in Fig. 6 and Fig. 7, that show the velocity contour at XY plane at *z=1.55 mm* and the temperature contour with streamlines at transversal planes along the flow direction.

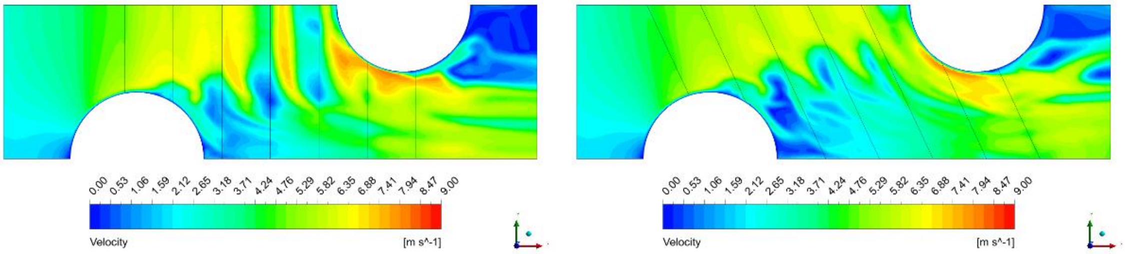


Figure 6. Velocity field contours taken along the waves at *z=1.55 mm* for Re = 900 in standard and -25° inclination configuration in circular tube heat exchangers.

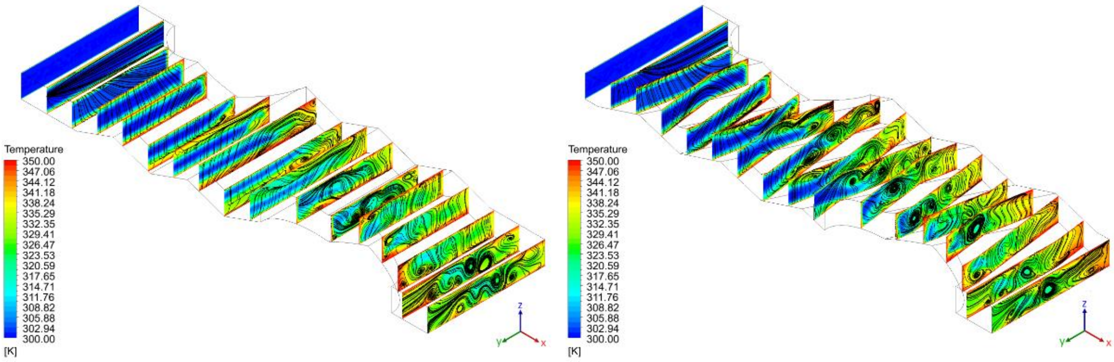


Figure 7. Temperature field contours with streamlines taken at plane intersections along the main domain for Re = 900 in standard and -25° inclination configuration in elliptical tube heat exchangers.

Fig. 6 shows two wake zones after the tubes in the conventional heat exchanger, which is a known behavior for circular tubes. For wavy inclination of -25°, high momentum fluid is direct into the recirculation zones, reducing it and consequently decreasing the pressure losses along the main domain and increase the local heat transfer coefficient. Fig. 7 shows the effects of the wavy inclination of -25° compared to the conventional heat exchanger, showing large amount of vortical structures along the main domain, especially after the first tube, which is much more present than the conventional case. The temperature contourn show a slight reduction in the performance in the initial half, and a huge increase after *x/L = 0.5*, showing that this vortical structures increase the fluid mix, increasing the heat exchange.

5 Conclusions

In this work, a numerical simulation of a compact heat exchanger was conducted toward heat transfer enhancement trough wave inclination combined with circular and elliptical tubes. The Reynolds number from 180 to 900 (based in the fin pitch) was investigated which similar those found in refrigeration applications. From the thermal-hydraulic analysis, the following conclusions can be made:

- The maximum heat transfer is observed for wavy inclination of -25° for both tubes;
- The wavy inclination of -25° increased the heat transfer up to 8% for circular tubes and up to 16% for elliptical tubes;
- All the wavy inclinations investigated decrease the Friction factor(f) for both circular and elliptical tubes.
- The wavy inclination of 25° has the minimum friction factor, decreasing it in 10% for circular tubes and 8% for elliptical tubes;

- The wavy inclination of -25° inclination presented the highest performances, increasing the j/f-ratio of 25% for circular tubes and 30% for elliptical tubes.
- Further analysis showed that the wavy inclination of -25° impacted the velocity contourn, introducing high momentum fluid into the wake zones behind tubes.
- The wavy inclination of -25° also impacted the temperature contourn, due to vortical structures along the heat exchanger, mixing cold and hot fluid streams.
- The passive technique investigated in this work showed that the wavy inclination is a suitable techniques to enhance the heat transfer in wavy-fin compact heat exchangers for both tube profiles in staggered configurations.

Authorship statement.

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