Heat transfer intensification of a compact heat exchanger through-composed passive techniques with wavy-fin and longitudinal vortices generators

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Abstract. Considering the current need for continuous research to develop heat exchangers with high efficiency with low-pressure drop penalty, aiming to cost reduction in materials and manufacturing process, the present work evaluates a compact heat exchanger with wavy-fin and longitudinal vortex generator. The numerical approach considers a heat exchanger with circular and elliptical tubes and delta-winglet and rectangular-winglet vortex generator with an aspect ratio of 2 and angle of attack angle of 45°, for low Reynolds number. The heat transfer is evaluated by Colburn's factor (j) and the pressure drop is analyzed by Friction factor (f). The flow is considered tridimensional, steady-state and turbulent. In comparison to the reference heat exchanger, the results showed a heat transfer improvement of 41% by using elliptical tubes associated with a rectangular-winglet vortex generator. For the friction factor, the configuration with rectangular-winglet vortex generators presented a higher pressure drop than the delta-winglet type and the arrangements of circular tubes showed a higher friction factor than elliptical tubes. Overall, the results showed that the composed passive technique is effective to enhance the heat exchange in compact heat exchangers.

Keywords: compact heat exchanger, vortice generator, heat transfer, passive technique, computational fluid dynamics.

1 Introduction

The application field of heat exchangers is very wide, presenting itself in most complex industrial equipment until domestic equipment. For the industrial segment, can be used direct ways of heat and cooling process in products, or non-directs manners, called utilities, as condensers and evaporators [1]. The need to improve heat transfer in heat exchangers stimulated the development of many intensification technics, where heat transfer improvement is generally coupled by pressure drop increasing, resulting in an increase of pumping potency. Thus, technics have been developed to improve the heat transfer with lower pressure drop penalty [2].

The heat transfer improvement techniques can be classified as actives and passives [3]. In actives methods, external energy is needed to improve the convective heat transfer. The actives methods are complex due to its difficult control of external energy applied. This is applied only is required a rigorous temperature control. Between the actives methods, it is detached the techniques of heat transfer surface vibration, pulsatile flux, electric field application, beyond others. In most of the applications, this method becomes commercial unfeasible. The passive methods, instead, do not need any energy external source, because it uses modified surfaces and/or the insertion of elements that promotes the flow distortion. According to Wang et al. [4], these devices alter the flow dynamics, increasing, consequently, the convective heat transfer coefficient. One of the main goals of these passives devices of heat transfer is to interrupt the dynamic and thermal boundary layer growth, increasing the mixture of heat and cold streams.

The heat flux rate also can be improved through the utilization of extended surfaces, known as fins. According to Wang et al. [4], theses surfaces improve the heat transfer rates and are used to increase the thermal performance of heat exchangers, where it is necessary to reduce the high thermal resistance. This alternative is widely defunded, considering that contact surfaces directly influences the heat transfer process. The fins, besides its proportionate the increasing of heat transfer and the contact area, also have a high influence on thermal and dynamic flow characteristics. Another important technique is heat transfer intensification is known as Vortex Generator (VG). This passive technique is intentionally inserted over the fins to induce and promote a secondary flow [5]. In this way, the VG not only perturbs the flow and interrupts the boundary limit layer growth, but also causes the heat transfer between the fluid and walls, leading to a heat transfer increasing.

In this research, the modeling and numerical simulation of a wavy-fin compact heat transfer with a longitudinal vortex generator, using Computational Fluid Dynamics (CFD), is analyzed considering circular and elliptical tubes. The operational range of this heat exchanger is similar to those found in refrigeration applications and air conditioning, which operates in a Reynolds number range from 150 to 600 (based on Fin Pitch).

2 Governing equations and thermal-hydraulic parameters

For this work, the hypothesis adopted for the numerical modeling is incompressible, tridimensional, steady-state and turbulent flow, according to [6]. Considering a Newtonian fluid, with constant properties, the governing equation of conservation of mass, momentum and energy are shown below.

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

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

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

where: u is the velocity component, h is the convection heat transfer coefficient, x_i and x_j are generalized coordinates, p is pressure, τ_ij is tension tensor, ρ is density, k is thermal conductivity and T is temperature.

A finite volume-based commercial software [7] was used to solve the governing equations. The thermal-hydraulic parameters to calculate the heat transfer and pressure drop in heat exchangers depending on the geometry and flow conditions. The flow condition can be characterized by Reynolds number, Colburn (j) and Friction (f) factors, which are represented below [8].

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

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

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

This way of calculating is adequate to compare circular and elliptical tubes since the minimum passage area is a function of elliptical tube eccentricity. Thus, the thermal-hydraulic parameters are calculated under the same reference, which in this case is the fin high (H) and computational domain length (L).

The total heat transfer, pressure loss and log-mean temperature differences are defined by the equations below, according to Salviano et al. [9].

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

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

$$\Delta T_{lin} = \frac{(T_W - \bar{T}_{in}) - (T_W - \bar{T}_{out})}{ln[\frac{(T_W - \bar{T}_{in})}{(T_W - \bar{T}_{out})}]}. \tag{9}$$

where:

$$\bar{p} = \frac{\iint_A pdA}{\iint_A dA}. \tag{10}$$

$$\bar{T} = \frac{\iint_A uTdA}{\iint_A udA}. \tag{11}$$

The convective heat transfer coefficient is calculated by eq. (12).

$$h = \frac{Q}{A_t \, \Delta T_{ln}}. \tag{12}$$

The flow is considered turbulent, even though the application range of the present simulation is for low Reynolds number since the occurrence of instability effects could appear due to generated secondary flow, introduced by the geometrical modifications on the fins and tube shape, which can promote the separation of flow and formation of wake regions.

The turbulence closure method used herein is the k-omega Shear- Stress Transport (SST) model, which considers the enhanced wall treatment as default [10]. SST k-omega turbulence model characteristics make this model more accurate and robust when high adverse pressure gradients are present in the flow [11]. Moreover, a robust algorithm called Coupled Algorithm to perform the pressure–velocity coupling was used [12]. This coupled algorithm solves the momentum and pressure-based continuity equations together. The full implicit coupling is conducted through an implicit discretization of pressure gradient terms in the momentum equations and an implicit discretization of the face mass flux. This coupled algorithm solves the momentum and pressure based continuity equations together [5]. Computational convergence is ensured making the residuals lower than $10{-}5$ for continuity and momentum equations and $10{-}7$ for energy equation.

2.1. Computational domain and boundary conditions

Considering computational time and cost associated with numerical simulation, the present modeling considers a compact heat exchanger under symmetry condition, with two tubes rows and a staggered-tube arrangement. Fig. 1 shows a domain illustration divided in three parts: the extended region at upstream, the main domain and the extended downstream region. The upstream region is considered in order to have flow uniformization, being extended one times of the main domain. The downstream region is extended 7 times of the main domain in order to ensure the boundary condition of Neumann. The non-slip and constant temperature is assumed on fins and tubes.

The present geometry is based on heat exchanger by Damavandi et al. [13], with slight modification on the inclination of the upstream and downstream extended-regions in order to obtain more stability in the numerical convergence.

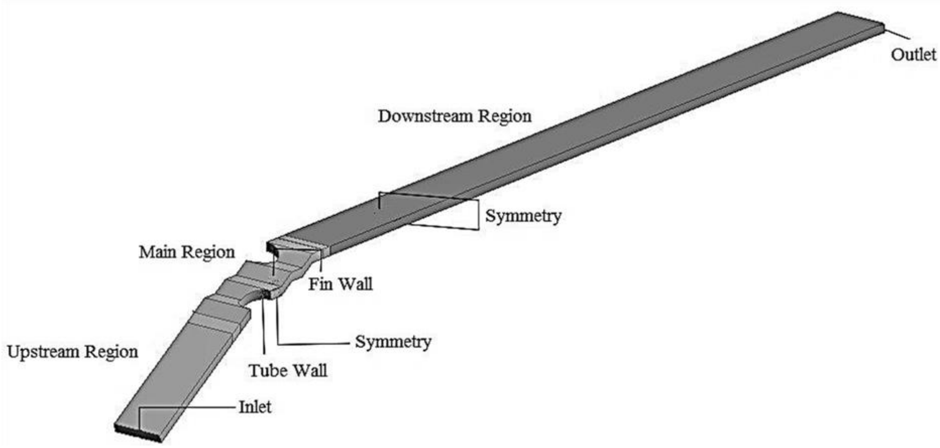


Figure 1. Computational domain of the compact heat exchanger model.

Circular and elliptical tubes were modeled in such a way that its perimeters were equal, ensuring the same heat transfer area. Frontal velocity was considered 0.5 to 2.5 m/s, which corresponds to a typical application in refrigeration systems. Vortices generators are welded on inferior wavy-fin, according to Lotfi et al. [14], and showed by Eq. (13) and (14). Vortex generators used are delta-winglet and rectangular-winglet type.

$$\Delta X = \pm R_a \cos\frac{\pi}{3}. \tag{13}$$

$$\Delta Z = \pm 2R_b \sin\frac{\pi}{3}. \tag{14}$$

where: ΔX is the distance from the center of the tube in the x-axis up to the center of the VG, ΔZ is the distance from the center of the tube in the z-axis up to the center of the VG, R_a is the semi-major diameter, R_b is the semi-minor diameter.

3 Validation and grid independence

The grid independence procedure was conducted according to Celik et al. [15], by the methodology of GCI (Grid Convergence Index). Three different mesh refinements were evaluated and the results are in Tab.1.

Table 1. GCI calculation reports.

Cells number (Main Domain)		Refinement index, r	GCI (%)			
			Re = 800		Re = 4000	
			J	f	J	f
Grid 1 (h1)	439.306	-				
Grid 2 (h2)	1.113.200	1,3	0.01	0.47	1.75	0.75
Grid 3 (h3)	2.451.768	1,3				

The Reynolds number for validation and GCI study is based on a different characteristic length (D_c), different that the information is shown in Eq.4.

The results obtained in Tab. 1 shows that the recommendation is achieved related to mesh density.

According to CGI values for Friction and Colburn Factors, the higher discretization uncertainty is 1.75% for the Colburn factor and Re = 4000. In this way, mesh density analysis is reached and intermediate mesh can be used for further analysis.

Moreover, the average values for the y-plus [7] are shown in Tab. 2. As can be seen, values are adequate in accordance with recommended for turbulence model SST “k- ω”, which should be near the unity.

Table 2. Grid y-plus values.

Meshes	y^+	
	Re = 800	Re = 4000
Refined (h3)	0.12	0.42
Intermediate (h2)	0.13	0.44
Coarse (h1)	0.14	0.46

Overall, the intermediate mesh is used for the continuity of analyzes. Validation of numerical modeling was conducted by a comparison of numerical results with values obtained from correlations proposed by Wang et al. [4]. These correlations proposed by Wang et al. [4] describes 94% of the Colburn factor with a deviation of 10%; while the friction factor described 95% of experimental data with a deviation of 15%.

Figure 2. Validation comparison of the correlation and the simulation values.

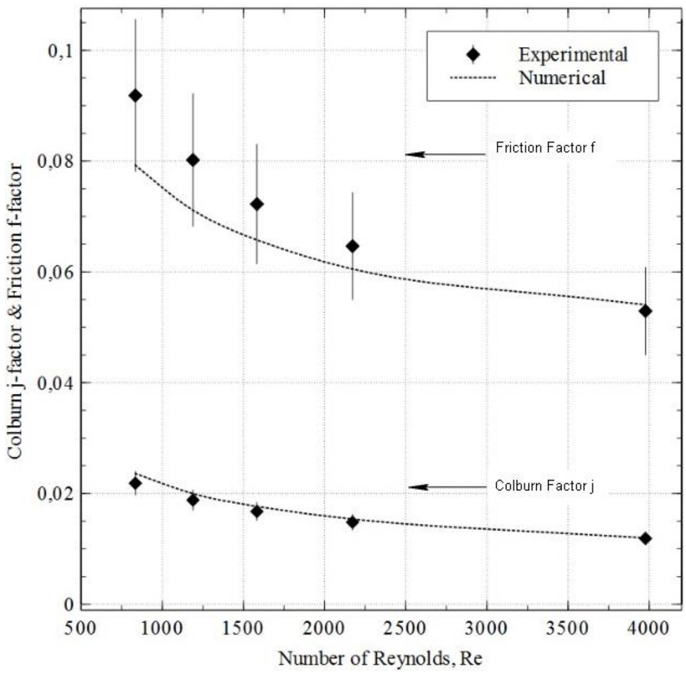
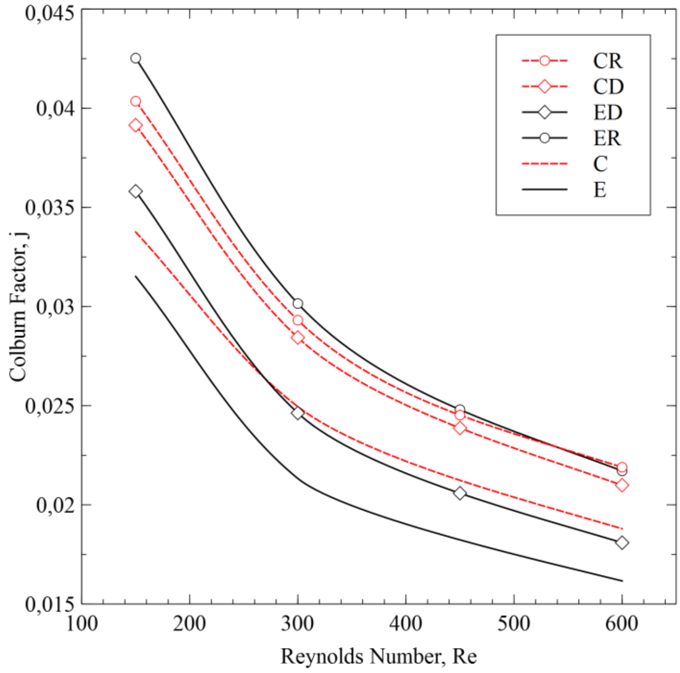


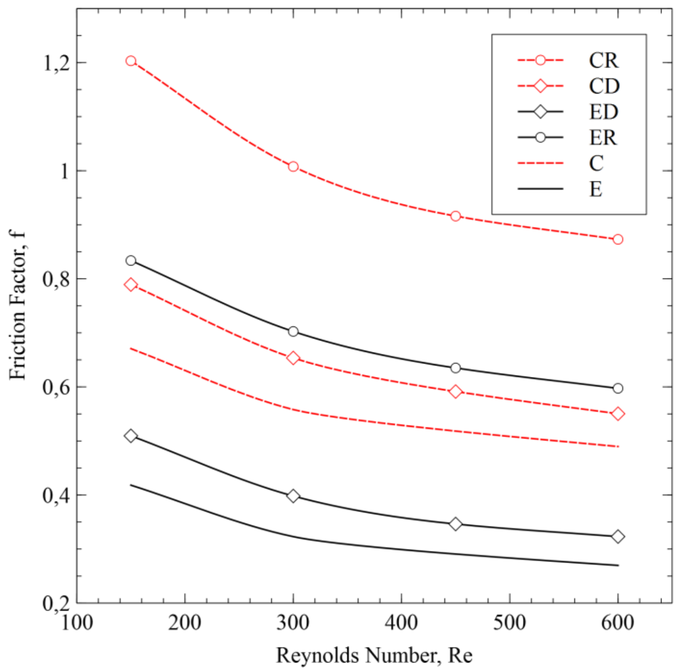
Fig. 2 shows a comparison between Wang’s correlation and numerical results. To friction factor and Re = 834 and Re = 3977, the differences are 13.7% and 2.14%, respectively, while for Colburn factor, the differences are 8.03% and 0.9%, respectively. Therefore, numerical results are within uncertainty deviation, showing that the present numerical approach is robust.

4 Results and discussions

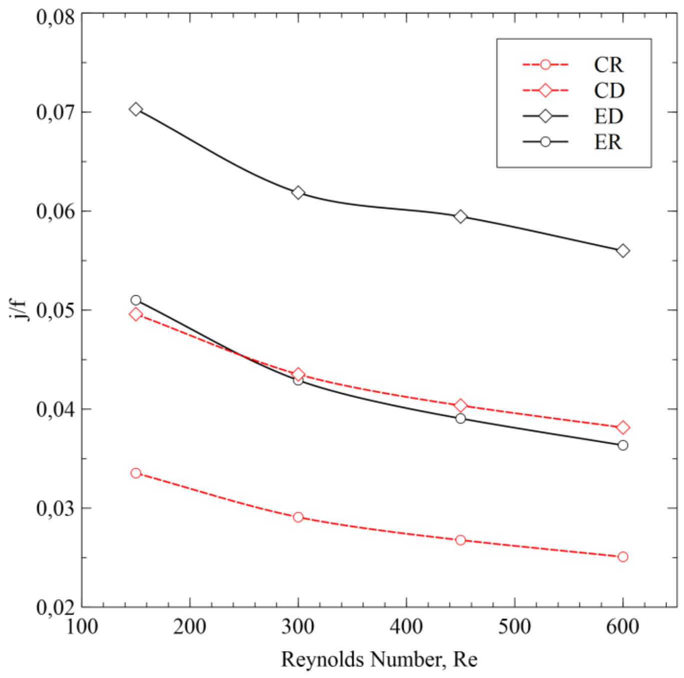
Fig.3 is based on results collected by calculating numerical computer simulations performed on the model. Taking into account four possible combinations, the elliptical tube with delta vortex generator (ED), the elliptical tube with rectangular vortex generator (ER), the circular tube with delta vortex generator (CD) and finally circular tube with rectangular vortex generator (CR), and reference cases are heat exchangers with elliptical tubes without a vortex generator (E) and circular tubes without a vortex generator (C).



(a)



(b)



(c)

Figure 3. Graphics of values of interest per Reynolds. a) Colburn j-factor. b) Friction f-factor. c) j/f.

Fig.3 (a) shows that a higher Colburn factor is verified for a combination of the elliptical tube with a rectangular-winglet vortex generator. The rectangular-winglet vortex generator increases of mixture flow between cold and hot streams. However, for rectangular-winglet vortex generator and higher Reynolds number influence of the tube is insignificant. This behavior is not observed for delta-winglet vortex generator and circular tube, in which the Colburn factor for the circular tube is higher than the elliptical tube for range evaluated. Overall, all combinations of vortex generator and tube increased heat transfer higher than reference configuration without vortex generator.

Moreover, it is observed improvements of 41% for Reynolds 300 with the configuration of elliptical tubes combined with rectangular vortex generators, and about 20% to Reynolds number of 150 with elliptical tubes combined with delta vortex generator, related with reference configuration. These positive achievements are counterbalanced with the pressure loss increasing, confirmed by Fig. 3(b-c).

Fig. 3(b) shows that configurations with rectangular vortex generator presented higher friction factor than reference configuration, which is justified by the higher intensity of the secondary flow generated due to the difference between the pressure in front and behind of the vortex generator. The delta vortex generators do not generate a large pressure gradient and, therefore, a small pressure penalty is verified. Similar behavior is observed for the elliptical tubes which de frontal area is smaller than for the circular tube. For analysis of the j/f, Fig. 3(c), it is possible to observe insignificant differences between the ER and CD configurations over the whole range of the number of Reynolds evaluated, the reference configuration with elliptical tube presented notable behavior compared to all configuration expect to configuration with elliptical combined to delta-winglet vortex generator for lower Reynolds number, which can be explained due to increase of the Friction factor shown in Fig. 3(b). Considering the configuration with a circular tube, the delta-winglet vortex generator has no impact on thermal-hydraulic performance j/f. For rectangular-winglet vortex generator combined to circular tube the j/f is also decreased.

The Colburn factor profiles (j) along of the x-axis of the computational domain for ER and ED configurations are shown in Fig. 4, considering the highest Reynolds number (Re = 600).

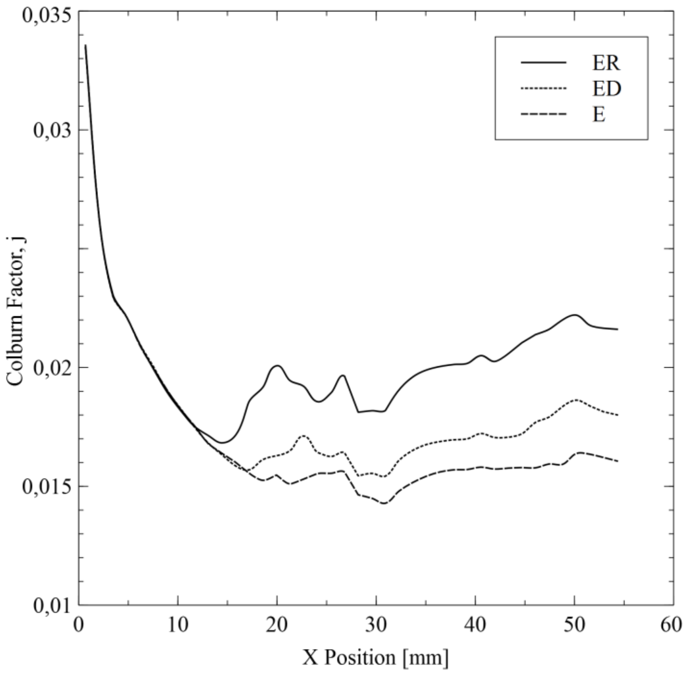


Figure 4. Colburn factor profiles (j) across the entire domain, with Re = 600.

It is possible to observe that the heat transfer is enhanced for configurations with vortex generators. These results corroborate with global results shown in Fig. 3a, where the heat transfer is higher for the winglet-rectangular vortex generator. Moreover, the first peak of heat transfer is due to the vortex generator which is located at x = 13.8 mm, and this enhancement heat transfer is persistent on downstream up to at the position of the second vortex generator, positioned at x = 41.3 mm, increasing the heat transfer up to the end of the computational domain. Phenomena already expected according to the authors, [16], [5] and [8].

5 Conclusions

In this work, a study of a numerical simulation of a compact heat exchanger was performed regarding the enhancement heat transfer through longitudinal vortex generators combined with wavy-fin and circular and elliptical tubes. The Reynolds number between 150 and 600 is similar to those verified for refrigeration applications. Based on Colburn and Friction factors, the following conclusions can be made:

- The maximum heat transfer is observed for the configuration with rectangular-winglet vortex generator for both circular and elliptical tubes;
- For circular tubes configurations with rectangular-winglet and delta-winglet vortex generator an increase of 20% and 16% on heat transfer is achieved;
- Suitable thermal-hydraulic (j/f) performance is observed for configuration with an elliptical tube with a delta-winglet vortex generator when faced with a circular tube with a rectangular-winglet vortex generator. Otherwise, it is observed slight performance between the elliptical reference configuration to the elliptical tube with delta-winglet vortex generator for Reynolds number from 300;
- The configurations with delta-winglet vortex generator presented lower heat transfer than rectangular-winglet vortex generator due to weaker mixture between the cold and hot streams, although the pressure drop penalty is also lower;
- CR configuration presented the worst thermo-hydraulic performance (j/f). Although the heat transfer is relevant the pressure drop penalty is higher, as evidenced in Fig. 3b. Thus, the use of the CR configuration is recommended only if the heat transfer requirement is desired;
- ED configuration is the highest value for the j/f criteria, where the low heat transfer is compensated with a lower friction factor.
- The passive technique with a longitudinal vortex generator is suitable to enhance the heat transfer with wavy-fin for both circular and elliptical tubes shape.

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