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# Augmentation of the heat transfer in a compact heat exchanger with asymmetric herringbone wavy-fin

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**Abstract.** The passive heat transfer intensification technique is a successful approach in thermal engineering applied to the heat exchanger. Present work evaluates through numerical modeling a compact fin-and-tube heat exchanger with asymmetric herringbone fins and aligned/ staggered circular tubes arrangement. Numerical modeling is performed considering a steady-state and turbulent flow. Heat transfer and pressure loss are evaluated by Colburn (j) and Friction (f) factors. A wave amplitude growth ratio of 0.8 and 1.2 were evaluated for a range of Reynolds number between 800 and 5000. The density mesh analysis is ensured by Grid Convergence Index methodology (GCI) and the numerical robustness is verified through experimental comparison. Colburn factor is increased of 36% and 13% for the configuration with amplitude growth ratio of 1.2, for aligned and staggered cases, respectively, in relation to their reference cases. Friction factor is increased of 35.3% and 12.7% for the aligned and staggered cases, respectively, for amplitude ratio of 1.2. The main flow phenomenology found is related to the generation of downstream secondary flow, boundary layer separation at the waves tip and reduction of the wake behind of the tubes. For higher wave amplitude, mixtures between cold and hot streams decrease the recirculation zones and increase the local heat transfer coefficient.

**Keywords:** Compact heat exchanger, Asymmetric wavy fin, Heat transfer augmentation, Passive technique, Computational Fluid Dynamics.

## 1 Introduction

The growing industrial and technological development in the material science area has created a necessity of cleaner sources of energy for the maintenance and improvement of industrial processes, in order to obtain a more efficient and sustainable production and consumption chain [1]. Research centers, universities and companies invest time and money to achieve performance developments in industrial procedures for obtaining and transferring energy, by many means. Heat exchanger devices are one of the means used to execute the task of transferring energy between two or more fluids, using corrugated fins, which applications range from complex industrial processes, with or without phase change, to equipment for personal and daily use [2]. Another important application of a heat exchanger is the thermal comfort of environments, in which the strict application requirements for daily needs are increasingly demanding efficiency improvements and systems that are easy to maintain, allowing for an increased benefit in terms of their utility and performance.

The necessity to increase heat transfer in heat exchangers has stimulated, therefore, the development of several intensification techniques. In general, the increase in the heat transfer rate is accompanied by a drop in fluid pressure, a fact that results in unwanted increases in the needed pumping power, used to maintain the mass flow constant in the system. In this way, it is desirable to develop and achieve a performance intensification technique that has the minimum penalty for pressure loss, with the best possible gains in terms of heat transfer capabilities, in order to make the best combination and not to require increments in the pumping power, to compensate a possible great pressure lost penalty for the primary fluid.

The methods of heat transfer intensification can be classified as active and passive methods [3]. For active methods, the application of external energy is necessary for an increase in the convective heat transfer rate to occur. These methods are costly and complex due to the difficulty in controlling the external energy to be applied and, therefore, they are used only when strict temperature control is required in the system. Among the active methods, some options can be mentioned, such as the techniques of vibration of the heat transfer surface, pulsating flows, application of electric field and the injection of compressed air stand out [4]. In contrast, passive methods do not require any external energy source, as they generally use modified surfaces and/or inserts of elements that promote turbulence in the flow. These devices modify the flow dynamics, increasing the convective heat transfer coefficient [5]. One of the main objectives of these passive heat transfer intensification devices is to interrupt the development of the thermal and dynamic boundary layers, increasing the mixing of the fluid streams, increasing the heat transfer rate.

The rate of heat transfer can also be increased through the use of extended surfaces named fins. As mentioned by [6], these surfaces increase the rate of heat transfer and are used to improve the thermal performance of heat exchangers. This alternative is highly widespread, considering that the contact surface directly influences the heat transfer process, increasing heat transfer rates both by increasing the value of the total contact area and by the ability to change the flow dynamics through geometric changes in the fins, which may even provide the transition from laminar to turbulent flow regime [7]. In terms of possible geometric changes in the extended surfaces, the wavy fins stand out as an alternative that presents advantages in terms of the capacity to modify the flow dynamics.

Associated with these passive techniques, the study of the thermal and dynamic behavior of the heat transfer intensification process in compact heat exchangers has been possible by the development of high-performance computers, which employ advanced techniques for a numerical solving approach on complex engineering problems, with a special treat on flows. Computational Fluid Mechanics (CFD), applied in this research project, is an area on the rise in universities, research centers and major multinational companies, especially for its versatility of application and the possibility of extrapolating the results.

In this research, modeling and numerical simulation of a heat transfer augmentation process are carried out in a wavy fin-and-tube compact heat exchanger, using Computational Fluid Dynamics (CFD), considering geometric changes in amplitudes of the fin waves, in aligned and staggered circular tube arrangements. The amplitudes of the waves will be modeled individually, a fact that characterizes an asymmetric profile of the wavy fin. The operating conditions of the compact heat exchangers for Reynolds numbers between 800 and 5000 will be investigated, a range that includes applications in several refrigeration systems.

## 2 Governing equations and thermal-hydraulic parameters

For this work, the hypothesis adopted for the numerical modeling of the dynamic flow and heat transfer characteristics is incompressible, tridimensional, steady-state and turbulent flow, according to [8]. Considering a Newtonian fluid, with constant properties, the conservation of mass, momentum and energy of the flow is described by the equations 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, $\tau_{ij}$ is tension tensor, $\rho$ is density, $k$ is thermal conductivity and $T$ is temperature.

In this work, a finite volume-based commercial software [9] was used to solve the governing equations for flow and heat transfer. The parameters to calculate the heat transfer and pressure drop in heat exchangers depending on the geometry and the flow conditions. The flow condition can be characterized by Reynolds number, Colburn (j) and Friction (f) factors, which are represented below [10].

$$Re = \frac{\rho u D_c}{\mu} \tag{4}$$

$$f = \frac{\Delta p}{\frac{1}{2}\rho u_{máx}^2} \frac{A_c}{A_t} \tag{5}$$

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

The total heat transfer, pressure loss and log-mean temperature differences are defined by the equations below, according to [11].

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

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

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

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 will be considered turbulent, even though the application range of this research project is located in low Reynolds number values, considering the occurrence of instability effects in the secondary flow, introduced by the geometrical modifications on the fins and in the tube shape, which can promote the separation of and the formation of wake regions.

The turbulence model used herein is the k-omega Shear-Stress Transport (SST) [12], which is a modification of the k-omega model, proposed by [13]. The SST k-omega turbulence model characteristics make this model more accurate and robust when high adverse pressure gradients are present in the flow. A robust algorithm called Coupled Algorithm was used to perform the pressure–velocity coupling, according to [14].

### 2.1. Computational domain and boundary conditions

In order to define the computational domains and boundary conditions in this study, it is defined that x is the streamwise direction, y is the spanwise direction and z stands for the fin pitch direction, as indicated by [5].

The geometry is symmetric and periodic in y and z directions. Fig.1 shows the computational domain divided into three parts: the upstream-extended region, the fin region and the downstream-extended region. The upstream and downstream parts of this representation contain a plain extension of the main region. The upstream region was extended one time of the main domain to ensure the inlet velocity uniformity. The downstream region was extended seven times of the main domain, in its turn, so that the outflow boundary condition could be used at the outlet and to avoid reversed flow. The no-slip condition was assumed on fins and tubes. In these two regions, a constant temperature value was assumed. In the downstream and upstream regions, symmetry boundary conditions were applied, and also on the side planes of the fin region.

The tridimensional geometry model of this project was designed with a herringbone fin. The geometry was based on the model presented by [10], with modifications in the inclination of upstream and downstream parts, in order to obtain calculation stability.

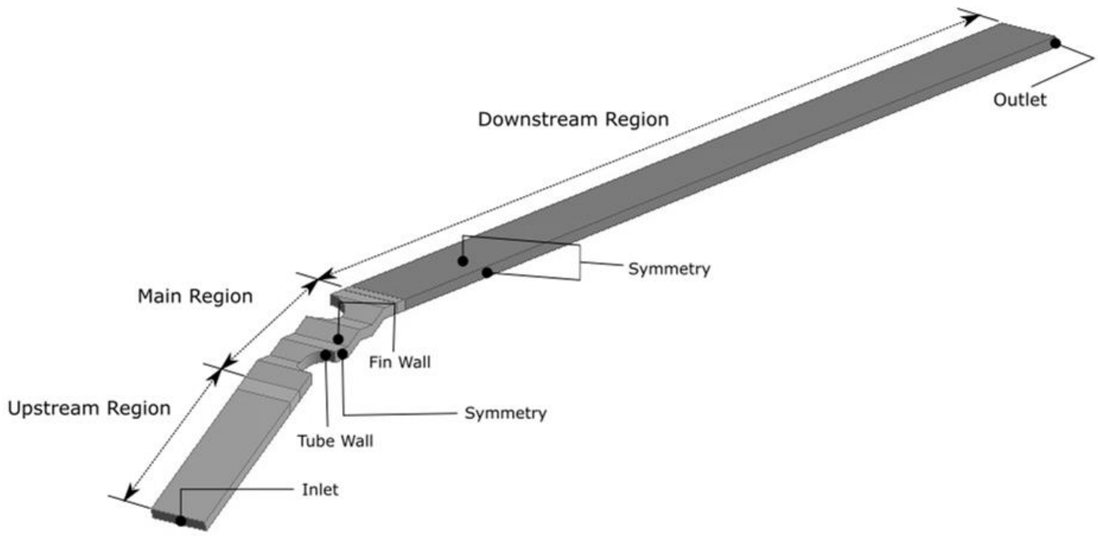


Figure 1. Computational domain and boundary condition of the compact heat exchanger model.

The proposed amplitude modification of the waves for the asymmetric heat exchanger is based on a multiplicative ratio of the amplitude value of the previous wave ($r_A$), for the subsequent waves. This value was established to be 0.8, when decreasing amplitudes, 1.0, for the reference case, or 1.2, for the case of increasing amplitudes, as the eq. (13) shows.

$$A_n = r_A \cdot A_{n-1}\,. \tag{13}$$

# 3 Validation and grid independence

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

In Table 1, the highest GCI value obtained was 4.50% for Factor Colburn for Re 4000, which is a small value considering the density mesh factor r. Therefore, the mesh density analysis is satisfactory and the Grid 2 is used for further analysis.

Moreover, an important metric to assure the numerical robustness is the y-plus of the grid, which must be close to unity to meet the k-omega SST turbulence model requirement. This parameter indicates the region in which the governing equations will be solved. Table 2 shows these values for each grid, which indicate that all grids evaluated in density mesh step are adequate for the use of k-omega SST turbulence model [9].

Table 1. GCI calculation reports.

| Cells number (main domain) | | Value $r$ | Re 800 | | Re 4000 | | GCI (%) | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | j | f | j | f | j | | f | |
| | | | | | | | Re 800 | Re 4000 | Re 800 | Re 4000 |
| Grid 1 (h1) | 6.6 10+5 | - | 0.0244 | 0.0806 | 0.0120 | 0.0529 | | | | |
| Grid 2 (h2) | 1.4 10+6 | 1,30 | 0.0253 | 0.0821 | 0.0125 | 0.0537 | 0.007 | 4.50 | 0.33 | 0.09 |
| Grid 3 (h3) | 3.1 10+6 | 1,30 | 0.0253 | 0.0826 | 0.0128 | 0.0538 | | | | |

Table 2. Grid y-plus values.

| | | |
|---|---|---|
| Fine Mesh (h3) | $y^+$ $(Re = 4000)$ | 1.05 |
| | $y^+$ $(Re = 800)$ | 0.33 |
| Intermediate Mesh (h2) | $y^+$ $(Re = 4000)$ | 1.07 |
| | $y^+$ $(Re = 800)$ | 0.30 |
| Coarse Mesh (h1) | $y^+$ $(Re = 4000)$ | 1.05 |
| | $y^+$ $(Re = 800)$ | 0.30 |

Therefore, the intermediate mesh (Grid 2) is used for the continuity of the analyzes. The numerical validation of the model was conducted by comparing the results of the numerical simulation with the values obtained from the correlations in [6]. The correlations were used to obtain values of the Colburn j-factor and Friction f-factor for six different Reynolds numbers. The validation comparison is shown in Figure 2, which a great agreement among results can be verified. The maximum difference was 11.2% for j and 12.3% for f. Thus, the assumptions adopted in the numerical modeling prove to be robust, indicating that the numerical results are consistent.

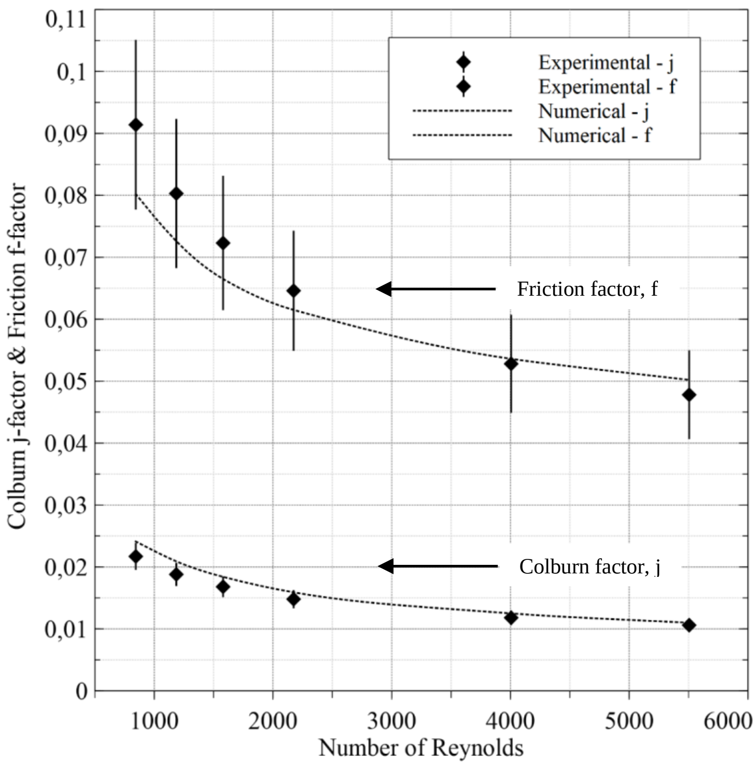


Figure 2. Comparison of the experimental and the numerical results.

# 4 Results and discussions

The Fig.3 shows a predominance tendency of increasing of j for transversal ratios of 1.2. For aligned and staggered tubes, both wave amplitude growth configurations of 1.2 are located among the three best cases, for j, as well as for the reference case of staggered tubes. The influence of staggered tubes is notable for the increase of flow mixtures, which is positive for the performance of j, in comparison with the reference case. It can be verified increments of 36% in Colburn factor, for Reynolds 4000, in the case of an amplitude growth ratio of 1.2 for aligned tubes, and 13% for Reynolds 2000, in case of an amplitude transversal growth ratio of 1.2 for staggered tubes, both compared to their symmetrical reference cases, respectively. Concerning the friction f-factor values, it is noted that the cases of transversal growth ratio of 1.2, aligned and staggered, as well as the staggered reference case, previously mentioned, are those in which cold and hot current mixtures is enhanced, with greater intensity, increasing the values of friction factor in these cases. Increases in the level of occurrence of flow boundary layer detachments are noted in cases where there is a progressive increase in wave height, with greater mixtures of cold and hot streams in the flow, as mentioned, generating vortices that is persistent on downstream, promoting reductions of the wake behind the tubes. These positive points are counterbalanced with the increase in the flow pressure losses, a fact confirmed by the graphics Fig.3 (a) and Fig.3 (b). Increments of pressure loss are noted in the order of 35.3% for the wave growth ratio case of 1.2 for aligned tubes, and 12.7% for the staggered wave growth ratio case of 1.2, both for Reynolds 2000, compared to their reference cases. About these two characteristics of j and f, the inferior behaviors of the aligned configurations and those from the wave growth ratio of 0.8 cases are remarkable. The aligned tubes cause fewer disturbances on the flow, as well as larger transversal recirculation zones, which do not favor the heat transfer in the heat exchanger system.

The performance of the cases was analyzed through the $j/f$ criterion, which is a direct cost-benefit relationship between heat transfer and flow pressure loss. Analyzing the Fig.3(c), it is noted that the cases of aligned tubes do not present themselves as beneficial, in comparison with the other alternatives. This is due to the smaller capacity of this configuration to offer the advantage of better heat transfer, as well as the high-pressure loss of the aligned wave growth ratio of 1.2 configuration, as shown in Fig.3(b). In fact, the analysis based on $j/f$ privileges the flow pressure loss in the analysis, and, for this reason, it is noted that the staggered reference case results in the best case for this criterion, especially for values of Reynolds greater than 3000, in detriment of the staggered wave growth ratio of 1.2 configuration, in which it obtains superiority of performance of approximately 3%, in Reynolds 4000 and 5000. Both mentioned configurations have the same behavior pattern to lower number of Reynolds. The slight superiority of the reference case since the pressure loss supplants the direct benefit of the heat transfer augmentation in this configuration and analysis.

For the analysis of the criterion $j/f^{\frac{1}{3}}$, which is based on the Performance Evaluation Criterion (PEC) defined by [16], there is a predominance of importance for colburn factor j, in detriment of friction factor f, so that the benefit of the changes in terms of heat transfer. Thus, there is a clear performance benefit for the configuration of staggered tubes and a wave amplitude growth ratio amplitude of 1.2, which stands out for a maximum performance superiority of 8.6%, for Reynolds 2000, concerning the staggered reference configuration, this being the configuration that provides greater mixtures of the flow and greater propagation of longitudinal vortices, as it can be seen by the analysis of the flow dynamics visualizations, as shown in Fig.4 and Fig.5. The aligned tube configurations do not perform this well in $j/f^{\frac{1}{3}}$, due to their relatively lower heat transfer capacity. The tube arrangement influence becomes clear when analyzing the maximum performance increase of 36%, for Reynolds 4000, in benefit of staggered tubes, in comparison between the aligned and staggered reference cases.

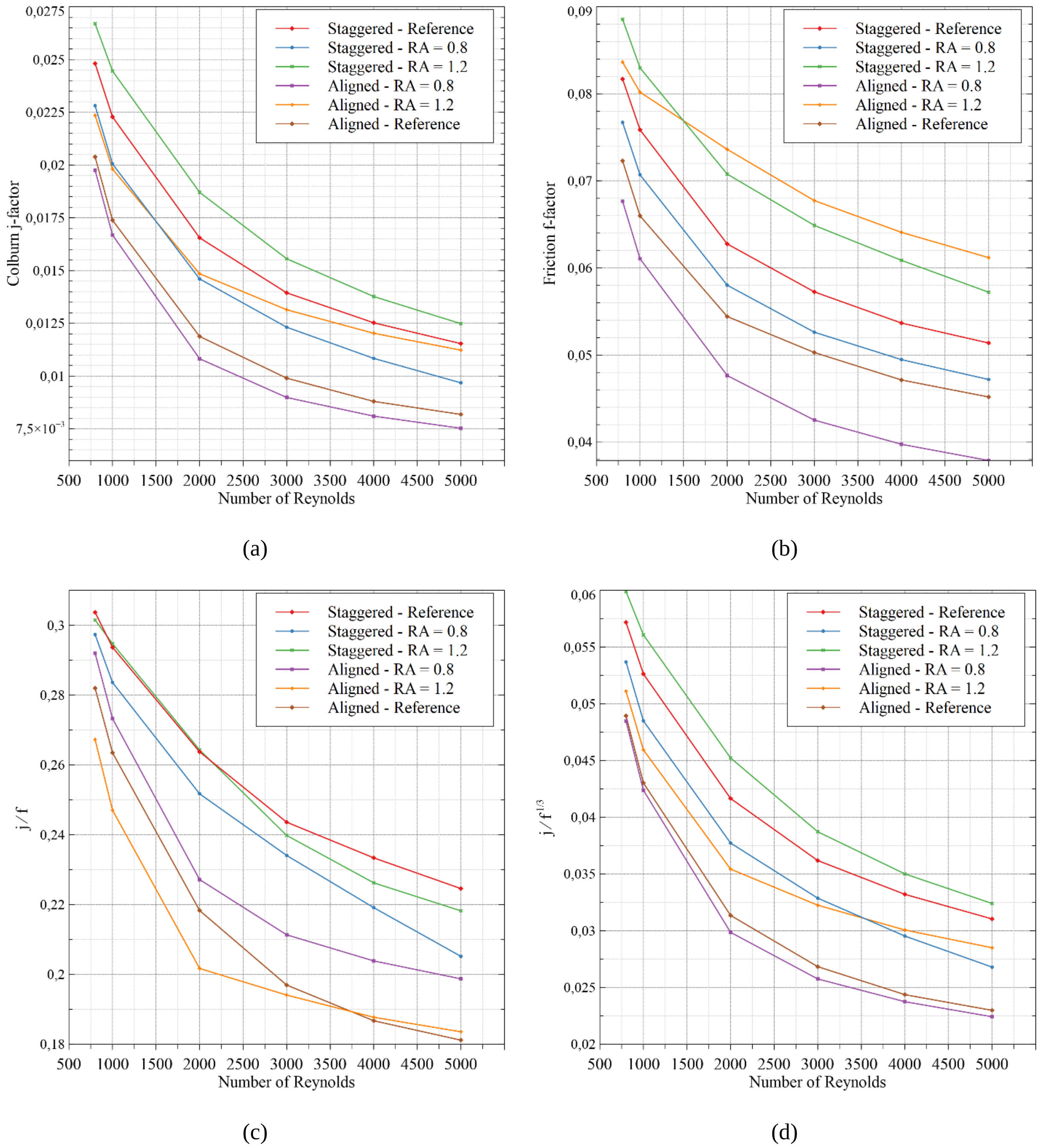


Figure 3. Results for several umber of Reynolds for: a) Colburn j-factor. b) Friction f-factor. c) $j/f$. d) $j/f^{\frac{1}{3}}$.

Fig.4 and Fig.5 present a comparison between the staggered amplitude growth ratio of 1.2 and the staggered reference case, for Reynolds Number of 5000, in order to identify the phenomenology which increases the thermal efficiency of the asymmetric heat exchangers. The intensification of cold and hot streams in the flow is evident, which causes the reduction of the recirculation zones behind the tubes, as well as the intensification of the secondary flow, which promotes strong mixtures in the flow on downstream.

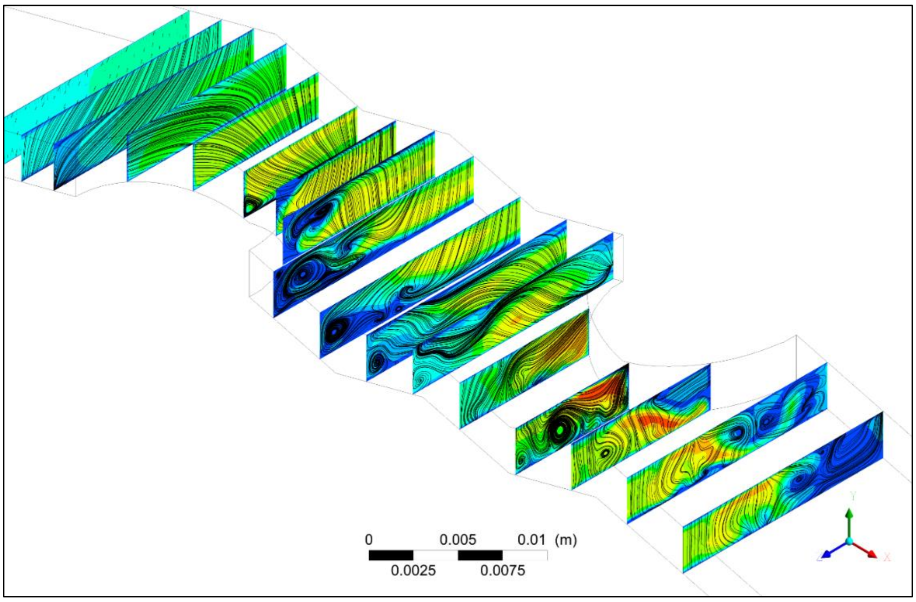


Figure 4. Velocity-contour of staggered reference configuration for Reynolds 5000.

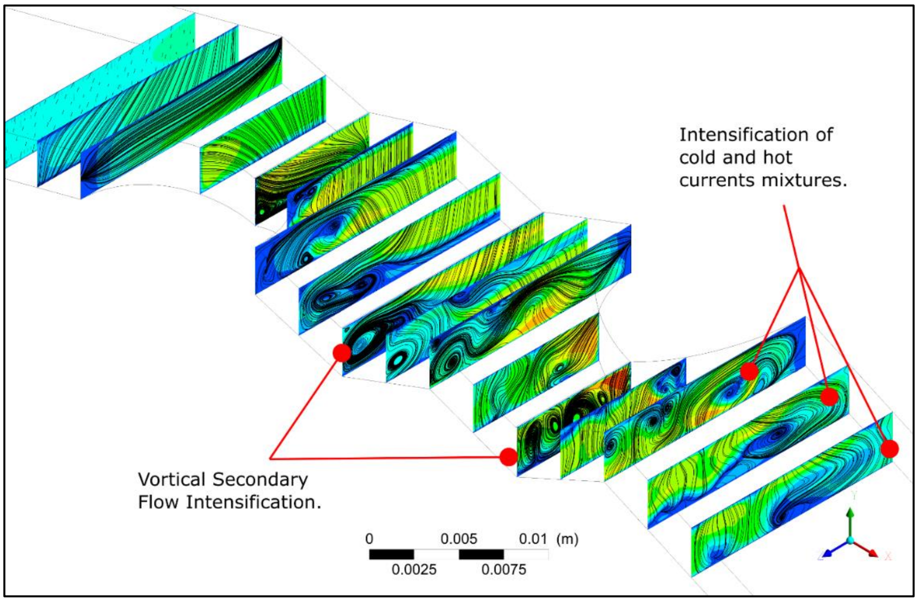


Figure 5. Velocity-contour of staggered amplitude growth ratio of 1.2 configuration for Reynolds 5000.

# 5 Conclusions

In this work, a numerical simulation study was performed regarding heat transfer enhancement through asymmetric wavy fin compact heat exchanger fins. A compact heat exchanger usually applied to refrigeration systems was chosen, with operation conditions between Re 800 and Re 5000, calculated based on the fin pitch. The major conclusions found are indicated below:

- Maximum heat transfer of 36% and 13% were noted for the cases of amplitude growth ratio of 1.2, for aligned and staggered cases, respectively, with an increase of 35.3% and 12.7% on friction factor.
- The analysis of j/f for staggered tube arrangement has higher heat transfer enhancement than aligned tubes since its mixtures of the cold and hot flow streams is more intense.

- For $j/f^{\frac{1}{3}}$, asymmetrical 1.2 ratio amplitude improved the thermal performance in the order of 8.6%.
- For staggered configurations, an amplitude ratio of 0.8 is not advantageous under any conditions. Amplitude ratio modifications of 1.2 are advantageous for applications where the main goal is to increase heat transfer with high ponderation on pressure loss penalty.
- For aligned configurations separately, amplitude ratio changes of 0.8 are interesting only for $j/f$, with a maximum performance increased of 9.7% for Reynolds 5000. The modification amplitude ratio of 1.2 is extremely beneficial for $j/f^{\frac{1}{3}}$, with a maximum performance increase of 24% for Reynolds 5000.

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