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IMPACT OF THE DOUBLE LONGITUDINAL VORTEX GENERATOR ON ENHANCEMENT HEAT TRANSFER FOR A WAVY-FIN COMPACT HEAT EXCHANGER WITH CIRCULAR AND ELLIPTICAL TUBES

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Abstract. *Combined passive heat transfer enhancement techniquesare a successful approach in thermal engineering. The present work investigates the impact of double longitudinal vortex generator in a compact wavy-fin heat exchanger operating at a low Reynolds number, related to refrigeration and air conditioning systems, by combining passives techniques of wavy fin and longitudinal vortices generators. The computational modeling considers a tridimensional model, for an incompressible, steady-state and turbulent flow. The heat exchanger with wavy fins is evaluated considering circular and elliptical tubes for staggered arrangement combined with vortex generator of delta-winglet and rectangular-winglet with an aspect ratio of 2 and angles of attack of 15º, 30º and 45º. In comparison with the reference cases, the results show that there is no need to use the second vortex generator for all evaluated angles, taking into account the elliptical tube with the delta-winglet generator. As for the cases with circular tube and rectangular-winglet vortex generators for all angles of attack analyzed, the second vortex generator has an important contribution. Overall, the results showed that in some cases, not using the second vortex generator does not generate heat transfer loss, allowing, among other things, the reduction of materials for the fabrication process and manufacturing.*

Keywords: *compact heat exchanger, longitudinal vortex generator, computational fluid dynamics, passive techniques.*

1. INTRODUCTION

Heat exchangers play a crucial role in heat transfer between fluids of different temperatures. These devices find widespread application across various industries, including process and chemical industries, transportation, air conditioning, and refrigeration (Pis'mennyi et al. 2016) Utilizing high-efficiency heat exchangers can effectively reduce material costs, energy consumption, and mitigate environmental impacts (Incropera and Witt 2008).

Over the past few decades, significant advancements have been made to enhance the thermal efficiency of heat exchangers (Alam and Kim 2018). These advancements can be categorized into two main techniques: Active and Passive. Active techniques involve the utilization of external power to augment the heat transfer rate, whereas passive techniques leverage inherent properties without requiring external power to enhance heat transfer performance (Bhuiyan and Islam 2016).

Passive techniques have emerged as the predominant approach to optimize performance and meet the growing demand for high-efficiency and cost-effective heat exchanger devices. These techniques employ specialized heat transfer surfaces strategically designed to intermittently disrupt the dynamic and thermal boundary layers along the flow direction (Wang, Chang, and Lee 1999). Notable examples of such surfaces include wavy fins and longitudinal vortex generators, which have proven to be exceptionally effective in enhancing heat transfer within compact heat exchangers (Tian et al. 2009). These mechanisms primarily operate by impeding the growth of the boundary layer, intensifying turbulence, and inducing secondary flows such as eddies or vortices (Fiebig 1998). It is important to acknowledge that while the heat transfer rate increases, there is typically a concomitant rise in pressure drop and pumping power (Wang et al. 2002). For a visual representation, please refer to Fig. 1, which illustrates the waved geometry of a compact heat exchanger.

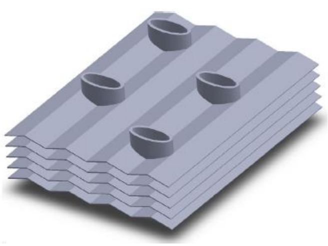


Figure 1. Geometry of the wavy fin of the heat exchanger. (Darvish Damavandi, Forouzanmehr, and Safikhani 2017)

In the present research, a modeling and numerical simulation of a wavy-fin compact heat transfer with a longitudinal vortex generator is performed, using Computational Fluid Dynamics (CFD), considering a wavy-fin compact heat exchanger with circular and elliptical tubes combined with vortex generator. Vortex generator type delta-winglet and rectangular-winglet with aspect ratio of 2 and angles of attack of 15º, 30º and 45º are investigated for staggered arrangement tubes. The operational range is similar to found in refrigeration applications and air conditioning, corresponding to Reynolds number 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 (Aslam Bhutta et al. 2012). Considering a Newtonian fluid, with constant properties, the governing equation of conservation of mass, momentum and energy are shown below.

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

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

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

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 (Ansys fluent 2013) 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 Deepakkumar and Jayavel (2017).

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

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

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

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, Dezan, and Yanagihara (2016).

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

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

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

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 characterized as turbulent, despite the simulation being intended for low Reynolds numbers. This is due to the potential occurrence of instability effects caused by geometric modifications to the fins and tube shape, which can lead to flow separation and the formation of wake regions.

In this study, the turbulence closure method employed is the k-omega Shear-Stress Transport (SST) model, with the enhanced wall treatment being the default (Menter 1993). The SST k-omega turbulence model offers improved accuracy and robustness, particularly in the presence of high adverse pressure gradients in the flow (Menter 1993). To ensure computational stability, a reliable algorithm known as the Coupled Algorithm was utilized (Ansys fluent 2013). This algorithm simultaneously solves the momentum and pressure-based continuity equations.

The coupling between momentum and pressure equations is achieved through full implicit discretization of pressure gradient terms in the momentum equations and implicit discretization of face mass flux. By employing this coupled algorithm, computational convergence is guaranteed, with residuals reduced below 10-5 for continuity and momentum equations and below 10-7 for the energy equation.

2.1 Computational domain and boundary conditions

In this study, the computational domains and boundary conditions are defined according to (Salviano, Dezan, and Yanagihara 2015). The streamwise direction is denoted by x, the spanwise direction by y, and the fin pitch direction by z. To optimize computational time and cost, a compact heat exchanger under symmetry conditions is considered. It consists of two rows of tubes arranged in a staggered manner. Figure 2 illustrates the division of the computational domain into three sections: the upstream-extended region, the fin region, and the downstream-extended region.

To ensure inlet velocity uniformity, the upstream region is extended by one times the size of the main domain. To prevent reversed flow, the downstream region is extended by seven times the size of the main domain. The fins and tubes are assumed to have a no-slip condition and a constant temperature. Symmetry boundary conditions are applied to the downstream and upstream regions, while an outflow (Neumann condition) boundary condition is specified at the outlet.

The geometry employed in this study is based on the heat exchanger design proposed by Darvish Damavandi, Forouzanmehr, and Safikhani (2017), with slight modifications made to the inclination of the upstream and downstream extended regions to enhance numerical convergence stability.

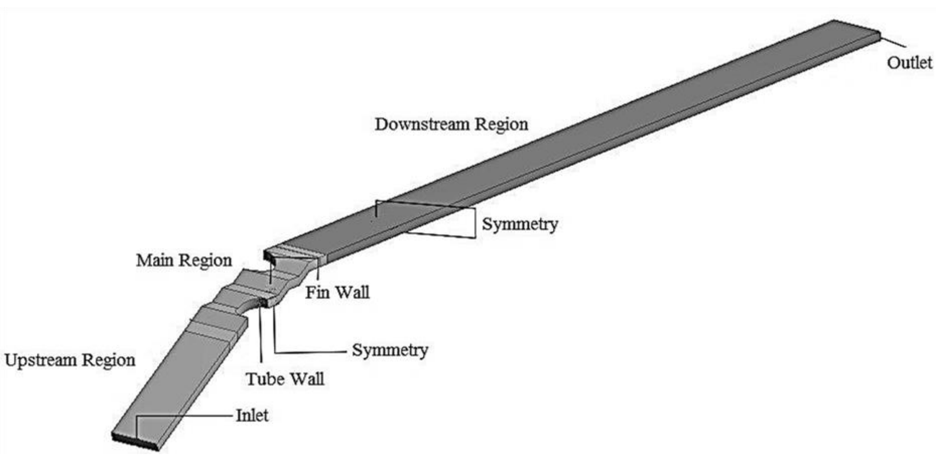


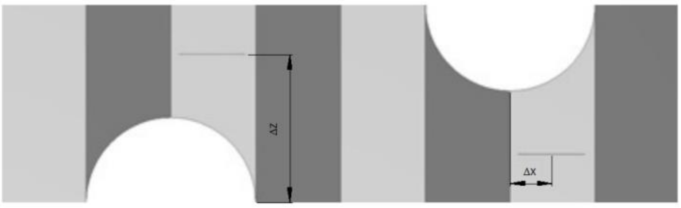
Figure 2. Computational domain of the compact heat exchanger model.

Circular and elliptical tubes were modeled with equal perimeters to ensure the same heat transfer area. The frontal velocity was varied between 0.5 and 2.5 m/s, representing a typical range for refrigeration system applications. Vortex generators, as described by Lotfi, Sundén, and Wang (2016) and expressed in Eq. (13) and (14), were welded onto the inferior wavy-fin. The vortex generators used in this study include delta-winglet and rectangular-winglet types

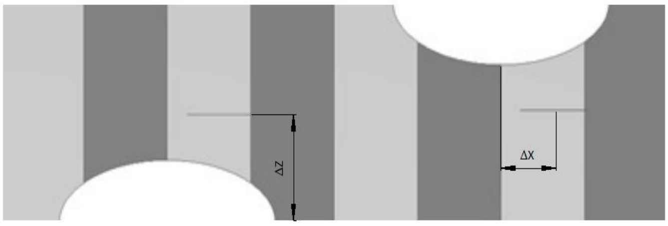
$$\Delta X = \pm R_a \cos\frac{\pi}{3}. \tag{13}$$

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

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.



a) Circular tube



b) Elliptical tube

Figure 3. Top view of heat exchanger with vortex generators with 0° attack angle

3. VALIDATION AND GRID INDEPENDENCE

The grid independence procedure was conducted according to Celik et al. (2008), by the Grid Convergence Index Methodology (GCI). Three different mesh refinements were evaluated and the results are shown in Tab.1.

Table 1. GCI calculation reports.

Cells number (Main Domain)		Refinement index, r	GCI_32 (%)			
			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 tube diameter. Table 1 shows the total number of cells for each grid, refinement factor r and GCI values. According to GCI_32 values for Friction and Colburn Factors, the higher discretization uncertainty is 1.75% for the Colburn factor and Re = 4000, which is considered small for the present work. Thus, mesh density analysis is reached and intermediate mesh can be used for the further analysis. Moreover, the average values for the y-plus (Ansys fluent 2013) are shown in Tab. 2, which indicates adequate values as recommended for the turbulent model (should be close to unit).

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

The validation of the numerical modeling was performed by comparing the numerical results with the values obtained from correlations proposed by Wang, Fu, and Chang (1997). These correlations have a standard deviation of 10%; while the friction factor with a deviation of 15%. Fig. 4 shows that for friction factor and Re = 800 and Re = 4000, the differences are 13.7% and 2.14%, respectively, while for Colburn factor the differences are 8.03% and 0.9%, respectively. Thus, numerical modeling adopted herein could be considered robust and reliable.

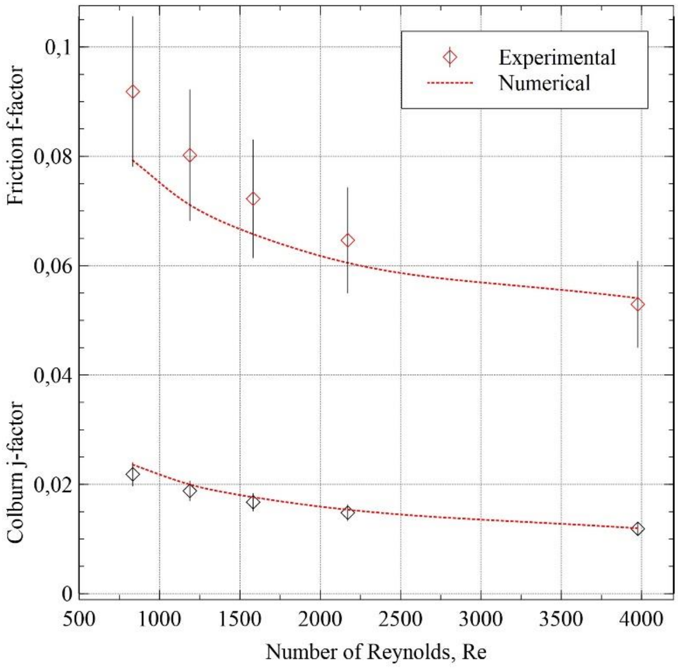


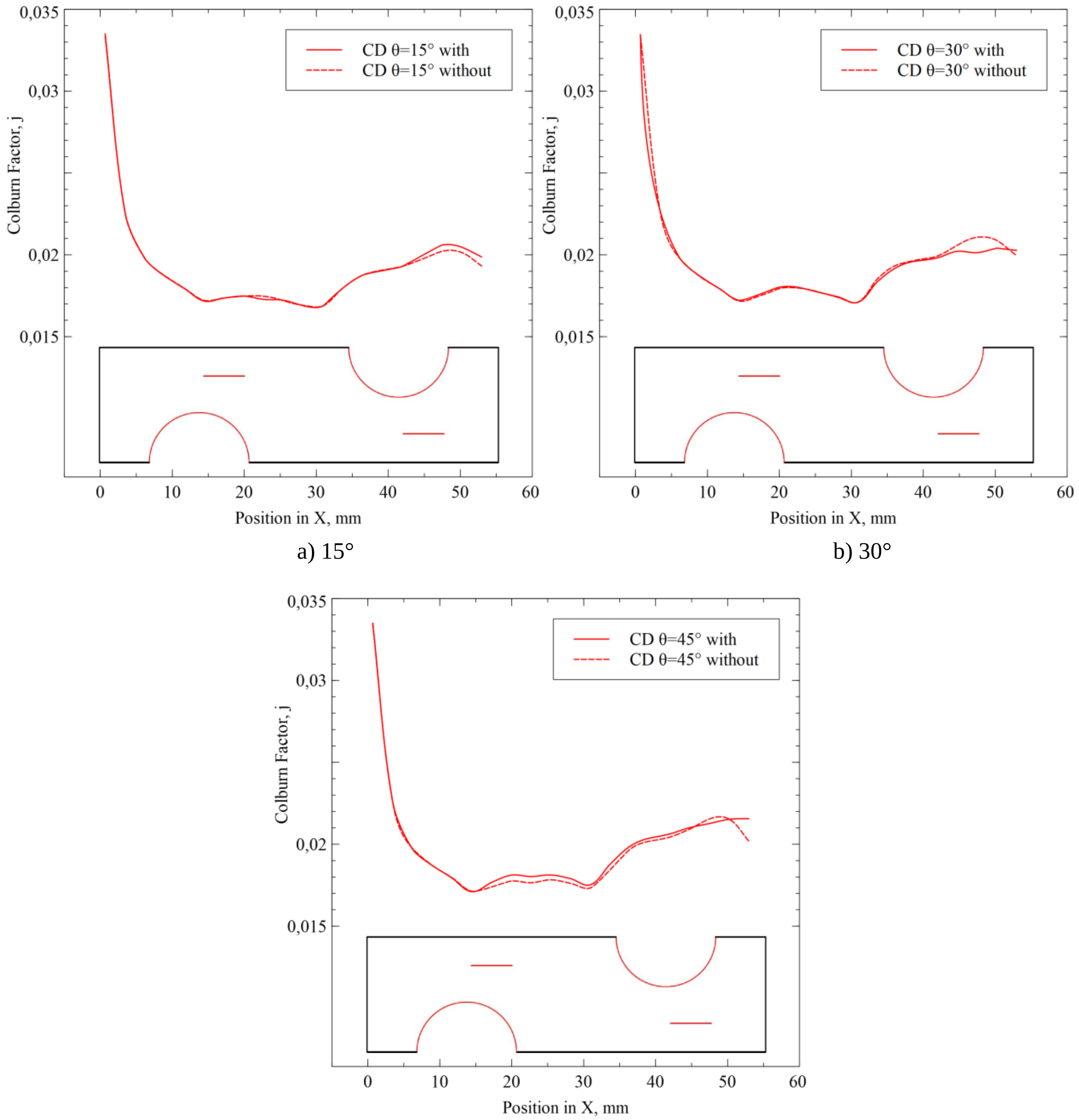
Figure 4. Validation comparison of the correlation and the simulation values.

4. RESULTS AND DISCUSSIONS

Based on previous works Bandini and Salviano (2021) and Bandini and Salviano (2020), it was noted that the second vortex generator has little or no effect on heat transfer intensification. The second vortex generator, located at position x = 41.8 mm, did not exhibit any apparent change in behavior compared to the reference case. Therefore, to confirm this fact, we present a sequence of results by eliminating the second vortex generator from the numerical simulation. Additionally, the effects were more pronounced for cases with higher Reynolds numbers, thus the analysis will focus on a Reynolds number of 600.

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 the circular tube with rectangular vortices (CR). , and cases (E) and (C), are heat exchangers without the second vortex generator, being elliptical and circular, respectively.

The graphs in Fig. 5 show a comparison of the Colburn factor (j) profile across the main domain for cases with and without the second vortex generator for the CD configuration, where a) corresponds to an attack angle of 15°, b) 30°, and c) 45°.



a) 15° b) 30° c) 45°

Figure 5. Colburn factor profile for the CD case: a) 15° attack angle, b) 30° attack angle, and c) 45° attack angle.

From the graphs in Fig. 5, it can be observed that for the 15° and 30° attack angles (a and b), the effects of the second vortex generator can be neglected. However, for the case with a 45° attack angle (c), there is an effect that should be taken into account. In the configuration with a 45° attack angle, an improvement in heat transfer is noticeable with the presence of the second vortex generator.

Similarly, Fig. 6 shows a comparison of the Colburn factor profile across the main domain for cases with and without the second vortex generator in the CR configuration, where a) corresponds to a 15° attack angle, b) 30° attack angle, and c) 45° attack angle.

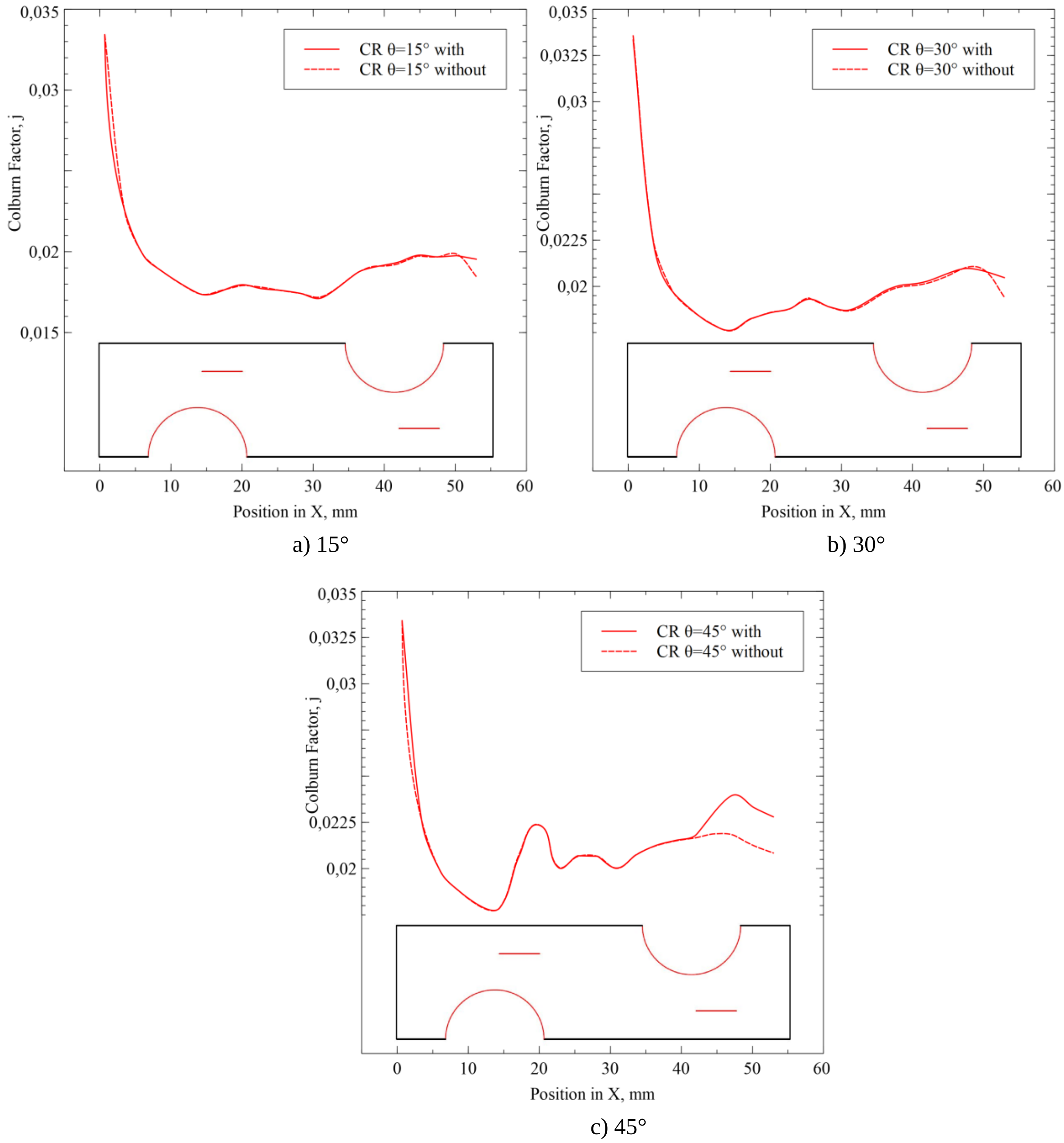


Figure 6. Colburn factor profile for the CR case: a) 15° attack angle, b) 30° attack angle, and c) 45° attack angle.

For the CR case, Fig. 6 shows a greater difference between the cases with and without the second vortex generator, being more evident for the 45° angle of attack. However, contrary to the conclusion obtained when using a delta-winglet vortex generator, when a rectangular vortex generator is used, the second vortex generator should not be excluded from the analysis.

Identically, we repeat the previous procedure for the ED configuration, as shown in Fig. 7, with a) representing a 15° lead angle, b) 30° lead angle, and c) 45° lead angle.

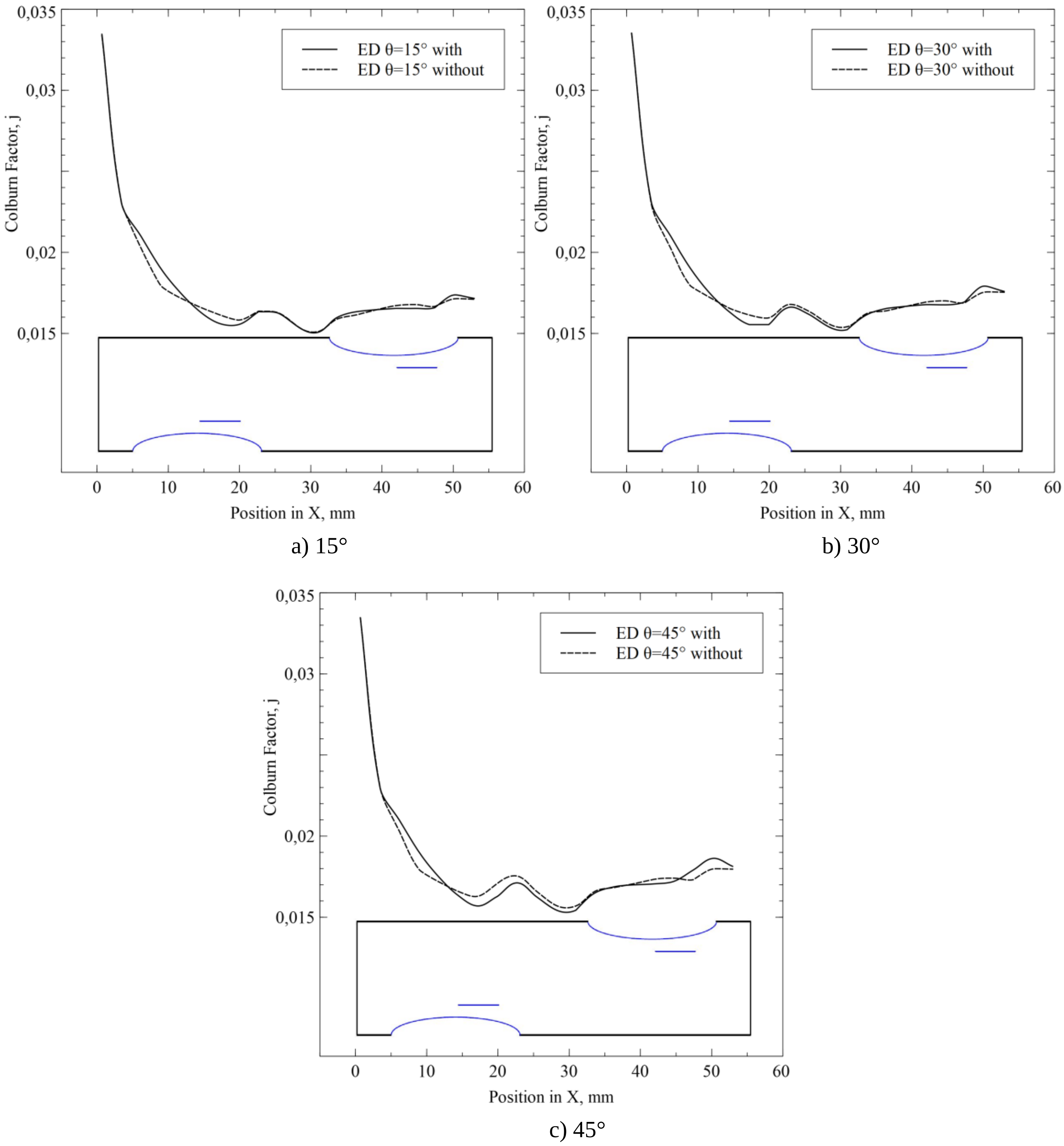
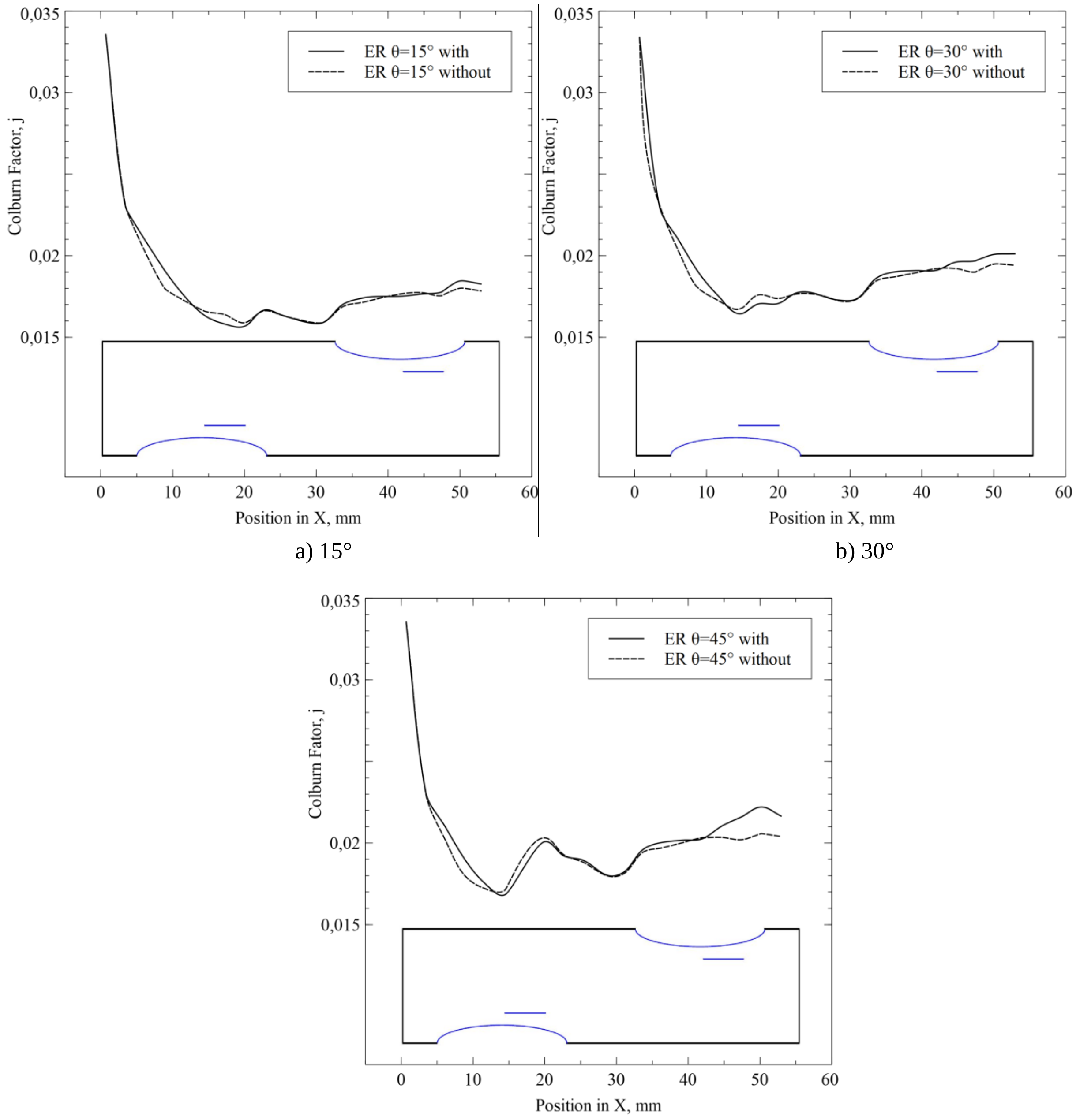


Figure 7. Colburn factor profile for the ED case: a) 15° attack angle, b) 30° attack angle, and c) 45° attack angle.

For the ED case, as shown in Fig. 7, using elliptical tubes and a delta-winglet vortex generator, a negligible drop is observed at the end of the main domain, which is experienced for all attack angles. Therefore, it can be concluded that the second delta-winglet vortex generator, considering elliptical tubes, does not yield significant gains in terms of heat transfer.

Finally, we evaluate the ER configuration, as depicted in Fig. 8, with a) representing a 15° attack angle, b) 30° attack angle, and c) 45° attack angle.



a) 15° b) 30°

c) 45°

Figure 8. Colburn factor profile for the ER case: a) 15° attack angle, b) 30° attack angle, and c) 45° attack angle.

In the case evaluated in Fig. 8, we observe a similar behavior to the case of elliptical tubes and a delta-winglet vortex generator (ED). However, upon analyzing the profile shown in Figure 51c, a significant change is observed only for the 45° attack angle case, while it is negligible for the 15° and 30° attack angles.

5. RESULTS AND DISCUSSIONS

Through computational modeling of a heat exchanger with wavy fins and vortex generators using elliptical and circular tubes, it was possible to compare and understand the dynamic behavior of the flow. The analysis of vortex generator insertion allows us to conclude that:

- For the CD configuration with a 15° attack angle, the effects of the second vortex generator can be neglected.
- In the CR configuration, for all analyzed attack angles, the second vortex generator should be retained.
- For configurations with elliptical tubes, there is no need for the second vortex generator for all evaluated angles in the ED case. However, in the ER case, the second generator should be considered when used with a 45° attack angle.
- The ability to eliminate the unnecessary use of vortex generators in certain configurations saves resources and reduces manufacturing costs.

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7. RESPONSIBILITY NOTICE

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