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A hybrid numerical-experimental analysis of the effects of delta winglet vortex generators in plate-finned heat exchangers with forced convection

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Abstract. *The present work performs a numerical-experimental study, addressing the effects of delta winglet vortex generators on the heat transfer rate by forced convection in a plate finned heat exchanger. Thus, a hybrid approach is presented to obtain the thermal performance of the heat exchanger. The methodology used aims to determine the average convection heat transfer coefficient and the overall efficiency of the finned surface. For this, a combination of numerical simulations with experimental data from wind tunnel was applied in the finned arrangement with delta winglet turbulence promoters, the results were validated with empirical correlations for the Nusselt number. A study of the parameters of the turbulence promoters was carried out, analyzing the vertical dimension, H, and the height in relation to the heater's longitudinal axis, Hr. The main results obtained show that the parameters with* $H/D = 1$ *and* $Hr/D = 1$ *have the greatest heat exchange in the system, that is, when the vertical dimension of the promoter was greater and when the heater and the hang glider prosecutor were not aligned.*

Keywords: *Forced convection 1, Plate finned heat exchanger 2, Numerical-experimental analysis 3, Delta winglet 4, Vortex generator 5*

1. INTRODUCTION

There are several techniques used to increase the heat transfer rate in thermal devices, improving the heat exchange between the heaters and the fluid, resulting in a higher efficiency. Enhancing the heat exchange surface with fins is one of the most used methods González *et al.* (2019), however, vortex generators have shown great potential to increase, significantly, the heat transfer rate Ponjet *et al.* (2014). According to Ponjet *et al.* (2014) vortex generators are solid objects that obstruct the flow, causing fluid separation and leading to an increment in the heat transfer rate. One of the most common vortex generators in the literature has a delta winglet shape, which, despite enhancing the convective coefficient, produces a substantial growth in pressure drop Gajusingh *et al.* (2010).

In the case of finned surfaces, it can be seen in González *et al.* (2019) a hybrid methodology is proposed to determine the heat transfer coefficient in the fin and its efficiency, which was validated by comparing empirical correlations of the Nusselt number proposed in Kaminski and Gross (2000) and ESCOA (1979). This technique is applied to a tube heat exchanger with plate fins with in-line arrangement and the tests were carried out for fins of different materials in a speed range. In this study, the boundary conditions were obtained experimentally and used as parameters for numerical simulation.

On the other hand, Zdanski *et al.* (2015) carried out an experimental study addressing the effects of delta winglet turbulence promoters. The research consisted of varying some parameters of the promoters in order to evaluate their effects in convection heat transfer. The results indicated an improvement in the thermal exchange of the system, generating an increase of up to 30% in the Nusselt Number. In addition, with the experimental results obtained from the Nusselt Number, a new empirical correlation was developed. This growth happens due the flow displacement and, consequently, vortices that create circulations, which allow an increase in heat exchange Gajusingh *et al.* (2010).

The present study aims to unite the use of fins and turbulence promoters in order to enhance the heat exchange, based on the experimental numerical method proposed by González *et al.* (2019) and taking into account the best results of Zdanski *et al.* (2015) with new parameters analyzed.

2. EXPERIMENTAL PROCEDURE

The scheme of the experimental procedure is shown in Fig. 1, where there are 4 stainless steel heaters with three copper fins and the 2 lines of turbulence promoters in the test section (25cmx25cm) of the wind tunnel. This tunnel operates by suction with an aspect ratio of 1: 6 and with a speed range of 4.0 – 15.0 m/s. The intensity of the flow turbulence entering the empty test section is less than 1%.

The flow velocity and its temperature at the entrance are measured with a digital anemometer placed inside the test section. The surface temperatures of the heaters were read in a data logger by thermocouples of type "k" that were welded in the heaters, it was considered in the equation that the welded union does not have contact resistance.

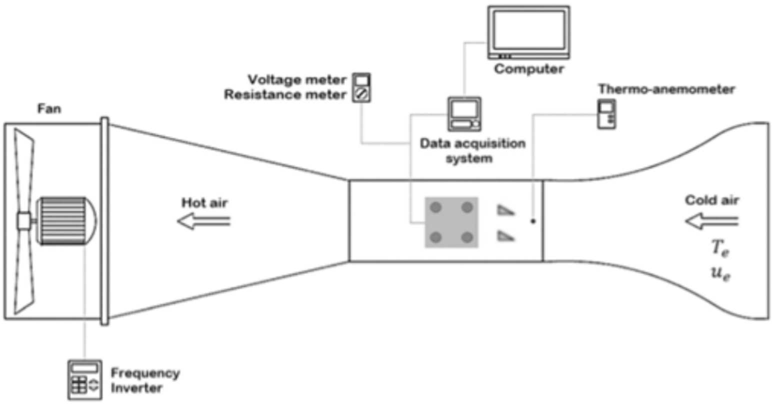

Figure 1. Schematic diagram of the heat exchanger in the test section

The resistance (R) and voltage (V) values were measured directly, using a digital multimeter. The calculation of the total heat transferred is obtained by applying Ohm's law given by Eq. 1, where $\phi = 0.98$, Ali (2009).

$$Q_m = \cos(\phi) \sum_{i=1}^{N} \frac{V_i^2}{R_i} \tag{1}$$

The average surface temperature of the heaters was obtained through Eq. 2, where the temperatures (T_i) was obtained with a total of 12 type k thermocouples, 3 in each heater, equally spaced as shown in Fig 2. The surface area of the resistance $As = 4\pi DL$ was measured indirectly using a digital caliper, which also obtained the other measurements from the fins and the promoters.

$$\overline{T}_s = \frac{\sum_{i=1}^{12} T_i A_i}{\sum_{i=1}^{12} A_i} \tag{2}$$

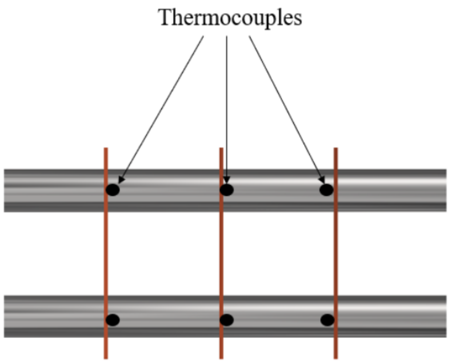

Figure 2. Position Thermocouples

The assembled apparatus with the fins follows the experimental reference of González *et al.* (2019), as shown in Fig 2, however, as shown in Table 1, it can be seen that a new speed range was verified in order to enhance the efficiency of the methods employed that work better in higher Reynolds values such as González *et al.* (2019) and Zdanski *et al.* (2015) have already concluded.

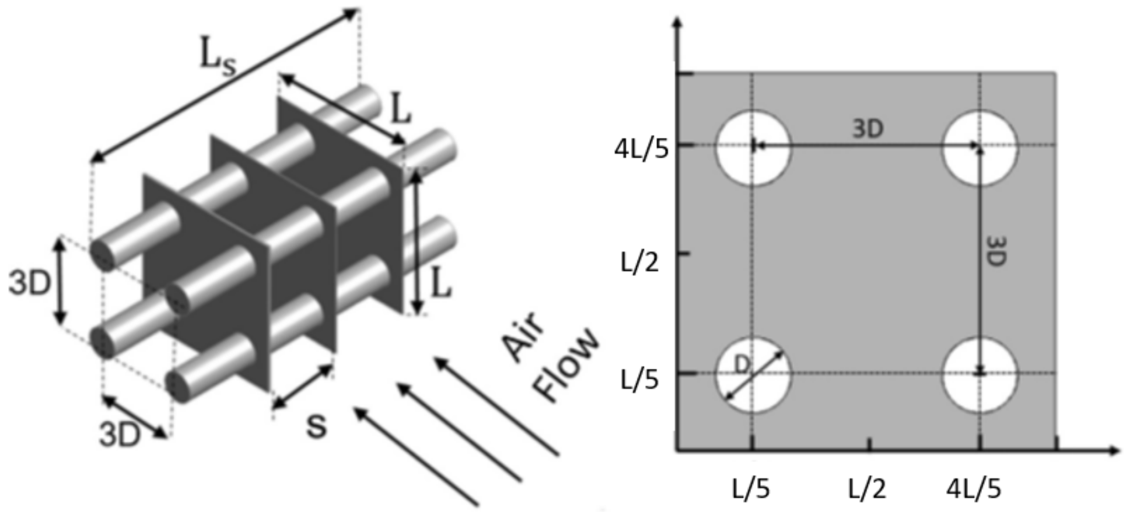

Figure 3. Computational domain and mesh for an individual fin

Table 1. Main parameters of the experiment.

Parameter	Symbol	Value	Unit
Airflow velocity	U_e	11.8; 10.1; 8.5 e 6.9	m/s
Fin spacing	S	42.5	mm
Fin length	L	80.0	mm
Tube diameter	D	16.00	mm
Tube lenght	Ls	170.00	mm
Fin thickness	δ_f	1.60	mm
Thermal conductivity	K_f	Copper: 401	W/mK

The turbulence promoters were printed in 3D with PLA material following the best parameters found in Zdanski *et al.* (2015), corresponding to case 1(Table 2). In order to expand the analysis already done, 2 new parameters were analyzed, namely the height of the promoter (H) and the relative height between the promoter and resistance (Hr), these parameters can be seen in Figure 4.

Figure 5, on the other hand, shows the isometric and side view of the set of the present work and Table 2 shows the cases evaluated for the 4 speeds presented in Table 1

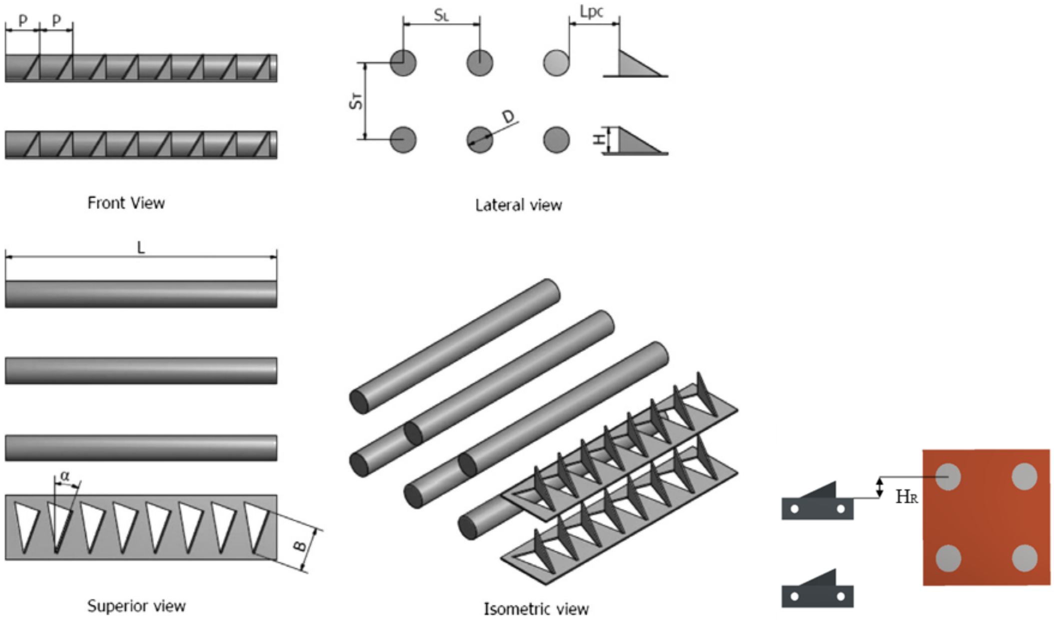

Figure 4. Illustration of the front, side, top and isometric views of the delta winglet promoters

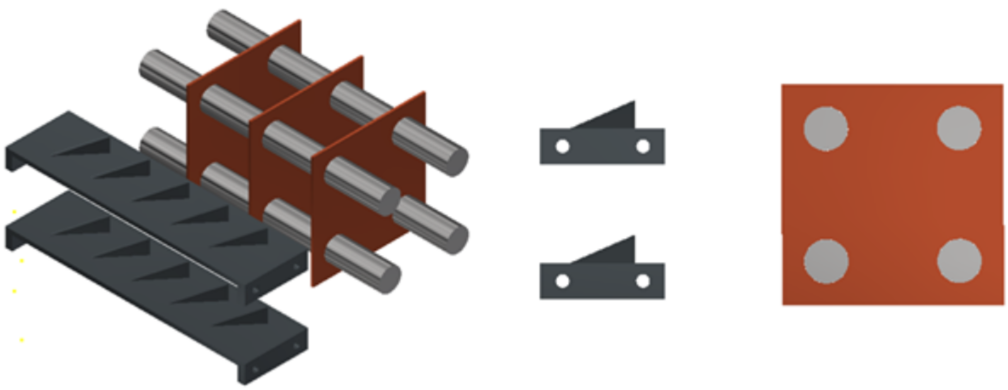

Figure 5. illustration of the front, side, top and isometric views of the delta winglet promoters

Table 2. Geometric parameter relationship of the arrangement.

Case	P/D	L_{pc} /D	α(rad)	B/D	H/D	St/D	Sl/D	Hr/D
1	1.75	1.94	0.72	1.72	1	3	3	0
2	1.75	1.94	0.72	1.75	1	3	3	1
3	1.75	1.94	0.72	1.75	0.625	3	3	0
4	1.72	1.94	0.72	1.75	0.625	3	3	1

2.1 Experimental Numeric Hybrid Method

The iterative calculation of the average heat transfer coefficient, h_0, and the overall efficiency of the finned surface, η_0, can be explained by the following iterative procedure:

i. Dimensional measurements: The cross-sectional area A_{SR} and the total heat transfer area of the finned surface, A_{tot}, are determined from linear dimension measurements so that

$$A_{tot} = NA_f + A_s \tag{3}$$

Where N is the number of fins, A_s, is the primary surface area (total surface area of heat in contact with air), and A_f is the finned surface area.

ii. Wind tunnel measurements: inlet speed measurement u_{in}, inlet and outlet temperatures (T_{in} e T_{out}) denergy input data, V_i, R_i, cowith the average surface temperature of the heaters, $\overline{T}_s$, so we can determine the film temperature as the usual setting, $T_{film} = (\overline{T}_s + T_{in})/2$.

iii. The evaluation of the total heat transfer rate, Q_m, using equation 1, and the outlet air temperature, T_{out}, is determined using the energy conservation principle in conjunction with the heat transfer rate (a similar procedure was used by Wang *et al.* (2017) and González *et al.* (2019)) as

$$T_{out} = \frac{Q_m}{\dot{m}c_a} + T_e \tag{4}$$

Where, $\dot{m} = \rho u_{in} A_{sr}$

iv. Calculation of the general heat transfer coefficient, U: the general heat transfer coefficient is obtained by Newton's law of cooling

$$U = \overline{h_o}\overline{\eta_o} = \frac{Q_m}{A_{tot}\Delta T_{LMTD}} \tag{5}$$

Where, ΔT_{LMTD} is the logarithmic difference of mean temperature

$$\Delta T_{LMTD} = \frac{(T_s - T_{in}) - (T_s - T_{out})}{ln(\frac{\overline{T}_s - \overline{T}_e}{\overline{T}_s - \overline{T}_{out}})} \tag{6}$$

v. Iterative calculation of the average convective heat transfer coefficient, h_0. The average convective heat transfer coefficient is determined by the following iterative procedure:

1. Define the initial estimate of the average convective heat transfer coefficient, $\overline{h}_0$. WHILE $\phi_h > TOL_h$ DO Calculate the heat transferred over the surface of all cylindrical heaters, $Q_s = \overline{h}_0 A_s \Delta_{LMTD}$;
2. Determine numerically the heat transferred by individual fins, Q_f, using the average experimental temperature of the fin, $\overline{T}_s$, and the average convective heat coefficient, h_0;
3. Calculate the total heat exchanged by the fins and surface of the heaters, $Q_{tot} = NQ_f + Q_s$;
4. The maximum ideal heat exchanged, $Q_{max} = \overline{h}_0 A_{tot} \Delta T_{LMTD}$;
5. Determine the overall efficiency of the finned surface, $\overline{\eta}_0 = \frac{Q_{tot}}{Q_{max}}$;
6. Evaluate the new heat transfer coefficient of $h_0 = \frac{U}{\overline{\eta}_0}$;
7. Calculate the convergence index, $\phi_h = |\overline{h}_0^{new} - \overline{h}_0^{old}|$
END WHILE
8. Final output $\overline{h}_0, \overline{\eta}_0$ and Q_f.

2.2 Numerical Evaluation

The fin geometry is represented in Figure 6. The mathematical model is derived from the energy conservation equation for a problem with two-dimensional heat conduction with constant temperature through the thickness, δ_f, as

$$\frac{\partial(k_f\frac{\partial T}{\partial x})}{\partial x}+\frac{\partial(k_f\frac{\partial T}{\partial y})}{\partial y}=\frac{2\overline{h}_0}{\delta_f}(\overline{T}_s-\overline{T}_e) \qquad (7)$$

For the numerical analysis, the commercial software Ansys Fluent was used. The simplifications adopted for the solution of the numerical model are: steady state, uniform fin convection, incompressible fluid, the average heat coefficient, $\overline{h}_0$, is assumed for both lateral sides of the fin with adiabatic fin tips. The average surface temperature of the heaters, $\overline{T}_s$, , is assumed to be the base temperature of the fin. The thermal contact resistance between heaters and fins is reduced by mounting the fin / heater configured with interference and applying thermal rate (contact resistances have been neglected).

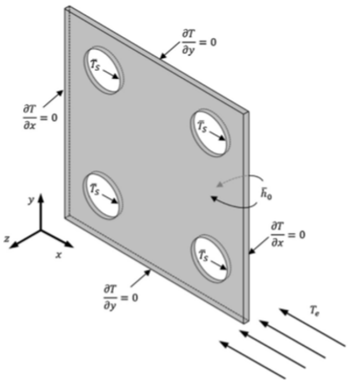

Figure 6. Boundary Conditions

3. RESULTS AND DISCUSSIONS

In this section, the results of the comparison between the arrangements with the turbulence promoters will be presented, whose analysis will be based on evaluating the influences of the geometric parameters H and Hr on the heat exchange of the arrangement. The present study was divided into two subsections: (i) Numerical validation and (ii) Analysis of thermal behavior with the introduction of delta winglet promoters.

3.1 Validation of numerical simulation

The hybrid methodology that determines the convection coefficient and the experimental apparatus was validated by González *et al.* (2019)), therefore, an analysis of the results obtained numerically was performed using the commercial software Fluent, Ansys®. The numerical process was validated by evaluating the number of iterations and elements in relation to the average convective coefficient, $\overline{h}_0$, as shown in Figure 7. This analysis was performed using experimental data from the heat exchanger without using the turbulence promoter at a speed of 12.7 m / s, where seven iterations were evaluated for each mesh size.

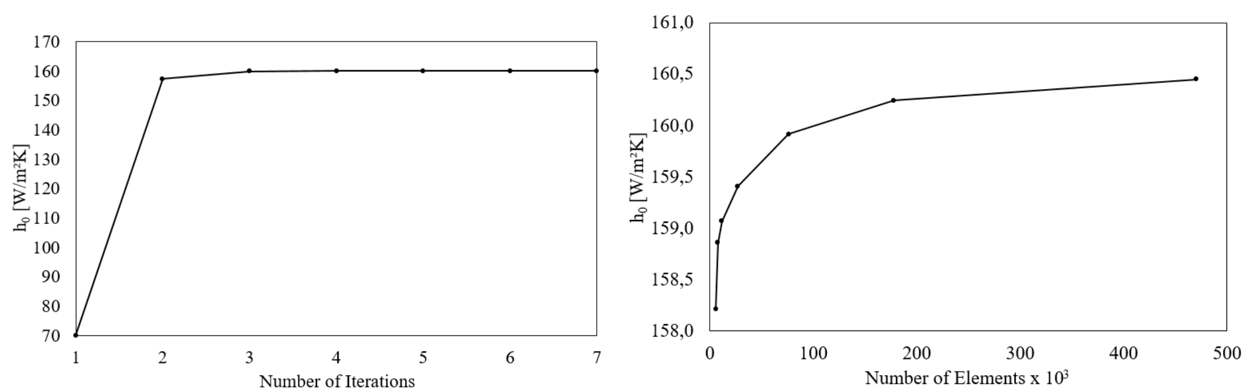

Figure 7. Number of iterations and elements in relation to the average convective coefficient

It can be seen in Figure 7 that from the third iteration the average convective coefficient, $\overline{h}_0$, has a minimal variation (0,05%), in addition, the graph in Fig. 7 shows us that the mesh with 76x10^3 elements is sufficient to the simulations considering that the relative error is less than 0,5% according to Fig.8, in comparison with the more refined mesh.

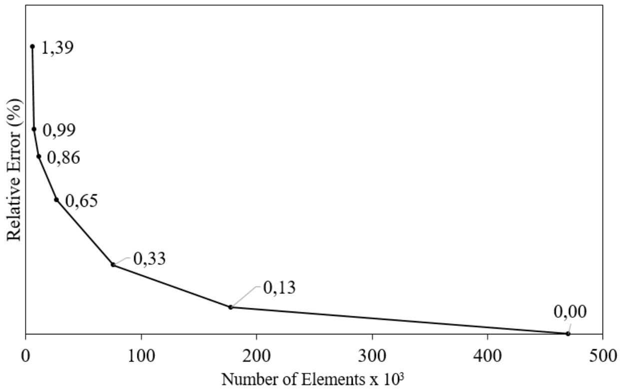

Figure 8. Relative error depending on the number of elements

3.2 Analysis of the thermal behavior of the system

In this section, the main results obtained through the analysis of the observed parameters of vertical dimension (H) of the turbulence promoter and their relative height (Hr) in relation to the longitudinal axis of the heaters are presented.

According to the graphs in Fig. 9, it is observed that the higher the value of H, the greater the number of Nusselt, this behavior occurs for cases with or without the variation of the relative height (Hr), it is observed that for the case with relative height, an increase in the number of Nusselt around 16%, was obtained, for the case without the relative height, the thermal gain in the number of Nusselt was around 19%. Thus, when analyzing the influence of Hr in the arrangement, it can be seen that the addition of this was important in increasing the heat exchange with the different values of H. It is concluded from the results that with the increase in height (H), there was a growth in the turbulent intensity that affected the heat exchange.

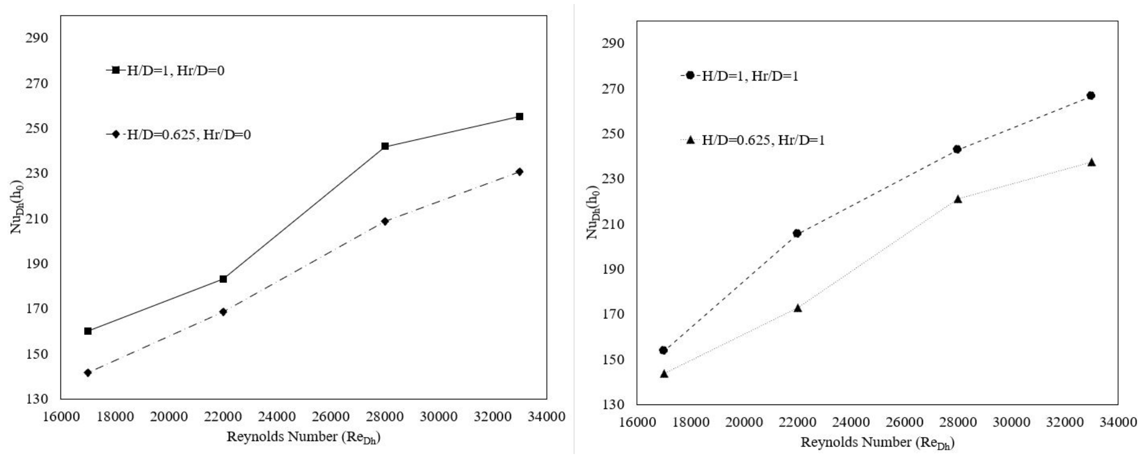

Figure 9. Graph of Nusselt Number as a function of Reynolds Number determined by hydraulic diameter with and without relative height variation

Figure 10 compares the arrangements with and without the relative height (Hr), in cases where $H/D = 1$ and $H/D = 0.625$, thus, it is noted that Hr increases a better heat exchange for the different values. However, for both cases, it was observed that for lower Reynolds values in the studied range, the results were not conclusive due to the fact that the parameter Hr did not induce a greater variation between the Nusselt numbers. In order to compare the best arrangement, Figure 11 presents a graph with all the studied variations and their respective Nusselt numbers are compared, thus, it is observed that both parameters positively influenced the Nusselt Number, with a greater emphasis for the value of H. The best evaluated case was the one with $H/D = 1$ and $Hr/D = 1$ and the one that contributed less to the gain was the $H/D = 0.625$ and $Hr/D = 0$, which presented a difference of up to 21% in Nusselt, thus showing the relevance of the analyzed parameters. Unlike the variation of parameter H seen previously, in which there was an increase in turbulent intensity, the variation of parameter Hr translated the zone of greater turbulent intensity, aligning this zone with the heater.

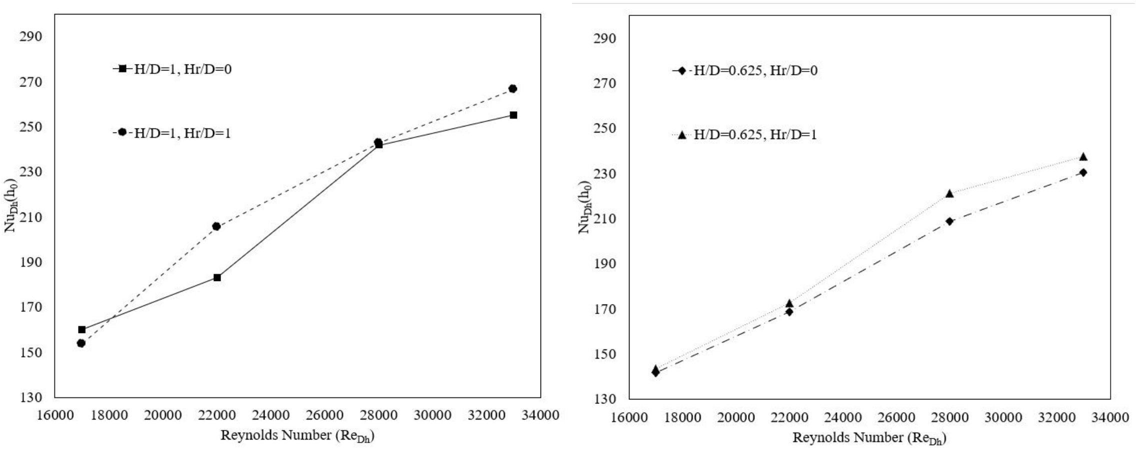

Figure 10. Graph of the number of Nusselt as a function of the Reynolds Number for the purchase of cases with and without the increment of the Hr parameter

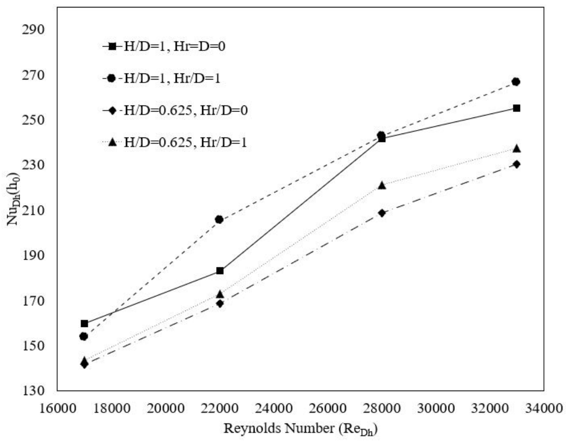

Figure 11. Graph of Nusselt Number by Reynolds Number of the analyzed arrangements

4. CONCLUSION

It was verified in the present study that, geometric parameters of delta winglet vortex generators, directly influence the heat exchange of the arrangement proposed in the work. It was observed that for a larger dimension of height H, the greater was the Nusselt Number, with an increase of up to 16% in relation to the lower height. In addition, the relative height (Hr) in the heat exchange effect was evaluated, the results show that with the addition of a relative height, there is an improvement in the heat exchange of the system, as it positions the zone of greater turbulent intensity aligning it to the heater.

The study points out that both parameters positively influence the Nusselt Number, detaching when the vertical dimension of the winglet is greater and when the heater and the winglet are not aligned with the arrangement ($H/D = 1$ e $Hr/D = 1$), which presents the greatest heat exchange. Thus, the parameters with the highest height of the vortex generator (H) and greater variation in relative height (Hr) have a greater heat exchange in the system. Finally, the authors of this work suggest that, for future work, a numerical investigation be made of the topology of the fluid generated by the promoters that affect the heat exchanger, in order to better understand the behavior of the fluid and its influence on the In this way, new geometric parameters can be investigated in order to improve the heat exchange in the system.

5. ACKNOWLEDGEMENTS

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