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HEAT TRANSFER AUGMENTATION AND FLOW LOSSES IN COMPACT HEAT EXCHANGERS BY USING VORTEX GENERATORS.

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Summary. *This work presents an experimental study of the influence of Delta Winglet vortex generators on the heat transfer enhancement and flow losses in fin-tube heat exchangers with two rows of tubes. The naphthalene sublimation technique was used to determine the heat transfer coefficient based on the heat-mass transfer analogy, while the flow losses were determined by means of a drag force measurement system. The performance of Delta Winglet mounted on the surface of the fins was investigated. The study was conducted for different transversal and longitudinal tube spacing. The results indicate that a reasonable heat transfer enhancement, with small increment of the destroyed exergy, can be achieved.*

Keywords: *Heat transfer enhancement, Vortex generator, Compact heat exchanger.*

1. INTRODUCTION

Compact fin-tube heat exchangers have wide industrial applications due to their high heat transfer capacity per unit volume. Many techniques to improve the performance of this kind of heat exchanger have been developed (such as corrugated fins, louvered fins, wavy fins, etc.) with the aim of reducing its dimension and the construction and operation costs.

The utilization of vortex generators to this type of heat exchangers has been considered promising by some authors in the recent years. The most important works about this theme are collected in the review papers of Jacobi & Shah (1995) and Torii & Yanagihara (1997).

Yanagihara & Sabanai (1996), Yanagihara & Bayón (1996) and Yanagihara & Rodrigues Jr. (1996) have conducted experimental and numerical studies about the influence of the vortex generators parameters on heat transfer and flow losses for fin-tube geometry, however these results can only be applied to one tube row heat exchangers.

Bayón & Yanagihara (1998a) investigated the influence of parameters such as position, attack angle and aspect ratio of Delta Winglet Pair vortex generators on the global heat transfer for a model of compact fin-tube heat exchanger with two rows of tubes in staggered arrangement. The best results of heat transfer enhancement were obtained for vortex generators position, attack angle and aspect ratio very similar to the values of these parameters carried out in the study of the vortex generators application in compact heat exchangers with only one row of tubes. Values of global heat transfer enhancement between 12,8% and 21,5% were achieved as function of the parameters set of the vortex generators. The heat transfer enhancement obtained in heat exchanger with two rows of tubes was ever higher (4-6% superior) that for heat exchanger with one row of tubes and the same vortex generators parameters.

Bayón & Yanagihara (1998 b) developed an experimental study with the objective to complement the results of heat transfer enhancement with flow losses introduced by vortex generators. This study was performed for the same heat exchanger model, the same Reynolds number values and the same vortex generators parameters that were encountered as better in Bayón & Yanagihara (1998a). The results shown that for the best position of the vortex generators a heat transfer enhancement of about 12,8% can be obtained, while the flow losses achieved small increments (less than 6%).

The present work pursues the objective to generalize the results achieved in previous studies through an experimental investigation about the influence of Delta Winglet Pair vortex generators on models of fin-tube compact heat exchangers with two rows of tubes in staggered arrangement. The influence of vortex generators in both rows of tubes with different values of transversal and longitudinal tube spacing on the global heat transfer and flow losses was studied. The results of heat transfer enhancement and of flow losses were analyzed by comparison with the smooth fin without vortex generators. The evaluation of the results was made applying the Second Law of Thermodynamics by using the exergy analysis. The heat transfer experiments were carried out using the naphthalene sublimation technique. The flow losses investigation was performed by means of a drag measurement apparatus.

2. EXPERIMENTAL MODELS

The range of geometrical parameters of the experimental models were chosen considering the typical dimensions of heat exchangers used in air-conditioning systems. The models were made with appropriate scale (5:1) to permit the work with the naphthalene plates. This allowed to evaluate The influence of vortex generators on heat transfer and flow losses was evaluated by comparing the performance of the heat exchanger with and without vortex generators. The models and the geometrical dimensions are presented in Fig. 1 and in Table 1.

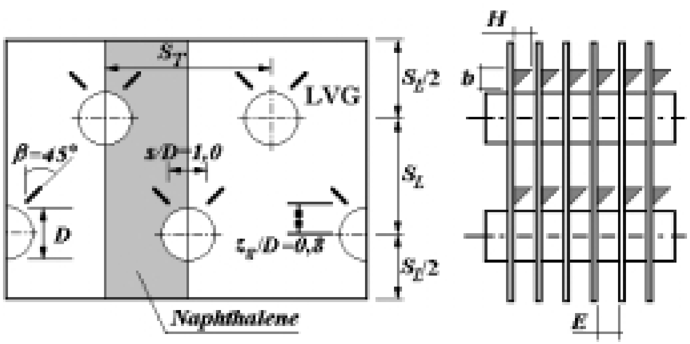

Figure 1. Experimental configuration.

The models were made by displaced fin-units, so that the fins could be mounted and removed. The test fins for the mass transfer measurements were located in the center of the model. These fins contained the cast naphthalene and were geometrically identical to the other fins. The test fins were placed forming the top and the bottom of the channel in order to consider the mass transfer in both surfaces. The other fins simply provided the proper flow conditions. The vortex generators were mounted on the surface of each fin symmetrically to the tubes.

Table 1. Dimensionless parameters of the geometry fin-tube.

Parameter	Value
S_L/D	1,5 –2,5
S_T/D	1,5-2,75
E/D	0,2

3. DATA REDUCTION.

Naphthalene sublimation. The global mass transfer coefficient h_m is given by:

$$h_m = \frac{\Delta m / \Delta\tau}{(\rho_{vw} - \rho_{v\infty})A_f} \qquad (1)$$

where Δm is the global sublimated mass, $\Delta\tau$ is the run time and $\rho_{v\infty}$ is the concentration of naphthalene vapor on the free mainstream. The concentration $\rho_{v\infty}$ was calculated applying the mass conservation equation to the control volume that involves the air inside the channel and the fin surface (naphthalene plate) from the entry to the exit, resulting:

$$\rho_{v\infty} = \rho_{v\infty\, e} + A_f(\Delta m / \Delta\tau \dot{V}) \qquad (2)$$

where $\dot{V}$ is the volumetric air flow over the naphthalene plate area and $\rho_{v\infty\, e}$ is the naphthalene vapor concentration present on the air stream at the naphthalene plate leading edge. The vapor concentration ρ_{vw} (and the saturated vapor pressure) of naphthalene at the surface is constant. This wall boundary condition is equivalent to an isothermal boundary condition in the heat transfer process. The naphthalene vapor density ρ_{vw} was determined from the ideal gas law using the vapor pressure (calculated by the correlation of Ambrose *et al.*, 1975) and the surface temperature.

The mass transfer Stanton number, St_m, was determined as:

$$St_m = \frac{h_m}{u} \qquad (3)$$

where u is the average velocity in the channel.

The heat transfer Stanton number, St_h, is given by the heat-mass transfer analogy:

$$St_h = St_m \left(\frac{Pr}{Sc} \right)^{2/3} \qquad (4)$$

The Schmitd number Sc was calculated using the correlation of Cho *et al*. (1989). The results were expressed in terms of Nusselt number through the expression:

$$Nu = St_h .Re.Pr \qquad (5)$$

Applying the methodology for uncertainty analysis, the uncertainty in *Re* were estimated to be 3% and uncertainty in St_h were estimated to be 4,5%, for 95% of confidence level.

The heat transfer enhancement factor was defined as the ratio between the Nusselt number of the fin surface with vortex generators (*Nu*) and the Nusselt number of the smooth fin (Nu_0).

Flow Losses. The additional pressure drop produced by the presence of the delta winglets inside the geometry of the heat exchanger is very small (less than 5 Pa for the conditions of the present work). Therefore, it was very difficult to use any type of pressure gauge for the measurement of this parameter. In order to overcome this problem, a drag measurement method was applied. This method is based on the measurement of the drag force on the heat exchanger model. The drag force is equal to the sum of the pressure force and the variation of the momentum in the main flow direction on the control volume that involve the heat exchanger model, thus:

$$\frac{D}{A} = \Delta p - (\beta_{out} - \beta_{in})\rho U^2 \tag{6}$$

In the Eq.(6), Δp is the pressure difference, A is the cross section area, ρ is the fluid density, U is the average velocity of the fluid on the section and β is the momentum coefficient for the inlet and outlet sections given by:

$$\beta = \frac{\int_0^A \rho u^2 dA}{\rho U^2 A} \tag{7}$$

where u is the local velocity.

When the momentum variation of the flow in the control volume is smaller than the drag force, it can be considered that the drag is equal to the pressure drop (Eq. 6). This drag measurement method for the determination of the pressure losses was previously used by Bayón & Yanagihara (1998b) with satisfactory results.

4. EXPERIMENTAL APPARATUS AND PROCEDURE.

The tests for the mass transfer measurements were conducted in an open circuit wind tunnel, shown schematically in Fig. 2.

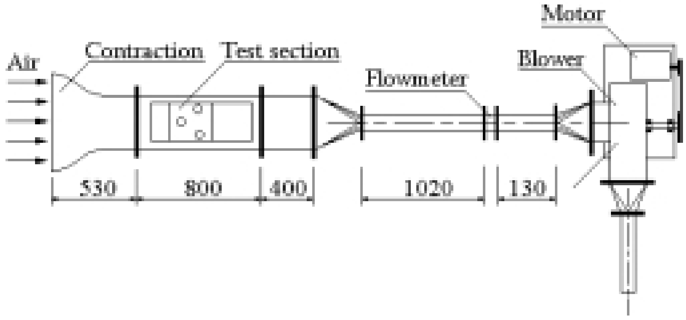

Figure 2. Experimental installation.

The tunnel is composed of an inlet, a test section, a blower and a discharge tube. The inlet is composed of a nozzle with a contraction ratio of 8:1. The test section is formed by a rectangular duct with 260mm × 90mm of cross section and 800 mm of length. It provided visual access during experiments and easy access to the model for quick installation and removal of specimens. The blower was located downstream of the test section to avoid heating of the free-stream flow. The discharge tube vented the air from the wind tunnel to the outside atmosphere, so that naphthalene did not contaminate the laboratory environment. The airflow rate was measured by a vortex flowmeter mounted in the tube between the test section and the blower and was controlled by varying the rotation frequency of the blower.

For the drag measurements, the same wind tunnel was used, mounted in vertical position. The system used for the drag measurement is shown in the Fig. 3. The heat exchanger model was hanged inside the test section by means of two rods connected to the extreme of a bar. The other extreme of the bar was rested on plate of a weight scale. A counterweight was placed in the bar to adjust the initial value in the balance reading. Therefore, when the air flows, the drag on the heat exchanger model reduces the balance reading. The measurement sensibility was adjusted varying the ratio between the lever arms (a/b).

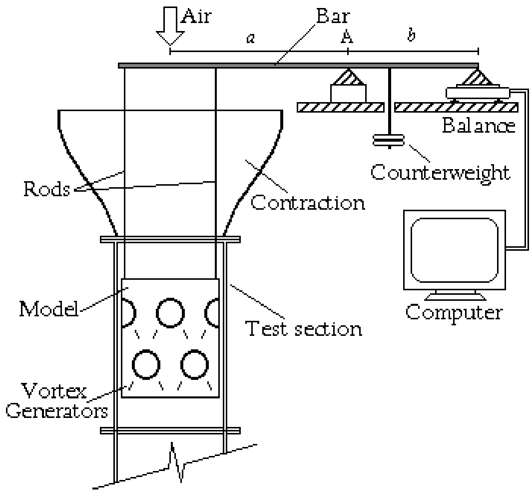

Figure 3. Apparatus for drag measurement in vertical test section.

The scale has a range of 0-2 N with resolution and accuracy of 10^{-5} N. The scale was connected to a computer to take the values of drag force each 5 seconds. The aim of this procedure was to reduce the uncertainty in the measurement produced by the oscillations of the drag force during the tests. The amplitude of the oscillations was smaller than 10^{-4} N. Considering 30 readings for each value of the drag, the standard deviation of the measurements was smaller than 10^{-4} N. The measured drag varied between 3.10^{-2} N and 7.10^{-2} N depending on Reynolds number. The uncertainty in the measurements was estimated to be 1%.

5. RESULTS AND DISCUSSION.

Heat transfer results. For the tests, the vortex generators parameters set obtained by Bayón & Yanagihara (1998b) were chosen. The parameter values used in tests were: vortex generators position z_g/D=0.4; s/D=1.0, attack angle β=45° and aspect ratio Λ=(2H/b)=2 (see Fig. 1).

The heat transfer tests were conducted for Reynolds numbers Re=300 and Re=1000 for each geometry. The heat transfer enhancement results as function of the longitudinal spacing are shown in Figures 4a (*Re*-300) and 4b (*Re*=1000). Each plotted curve corresponds to a transversal spacing.

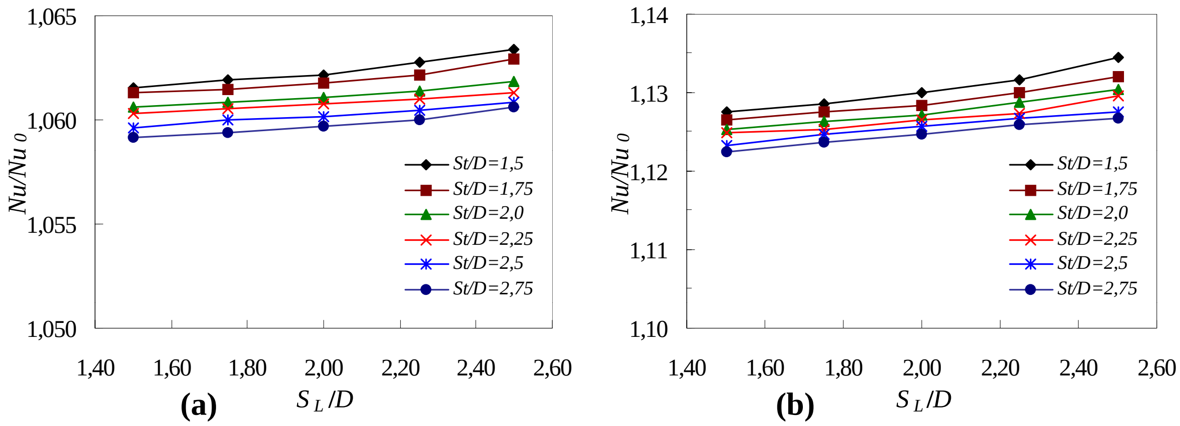

Figure 4. Influence of the vortex generator on the heat transfer .for different transversal and longitudinal spacing between tubes. **a)** *Re*=300, **b)** *Re*=1000.

The curves showed that the values of heat transfer enhancement varied between 5,9–6,3 % for *Re*=300 and between 12,2–13,5% for *Re*=1000 as a function of the heat exchanger geometry. The results indicate that for any transversal spacing, the achieved heat transfer enhancement showed tendency to grow when the longitudinal spacing increases. This behavior was observed for both of *Re*=300 and *Re*=1000 but it was significant for higher Reynolds number. This behaviour can be explained by the increase of the interaction area of the vortex on the fin. With increment of the longitudinal spacing, the vortex generated in the first row has a larger influence area, before reaching the second tube row. The same phenomenon happens to the vortex generated in the second row, covering a larger area.

The variation of the heat transfer enhancement with the transversal spacing is opposite to that observed for the longitudinal spacing. When the transversal spacing was increased, the heat transfer enhancement was diminished. The order of variation of heat transfer enhancement was 0,2–0,3% for *Re*=300 and 0,6-0,8% for *Re*=1000 when the transversal spacing was varied between 1,5-2,5*D*. In this case, the increase on the heat transfer enhancement produced by the vortex generators (when the transversal spacing diminishes) is due to the relative increment of vortex interaction area with respect to the total area of the fin. It is possible because the transversal spacing reduction increases the volume occupied by the tubes and reduces the fin surface but do not affect significantly the region where the vortex generators produce the heat transfer augmentation effect, i.e., on the recirculation region downstream of the tubes.

Flow losses results. Initially, the values of the friction factor *f* for the Reynolds number *Re*=300 and *Re*=1000, for the different heat exchanger geometries without and with vortex generators, were determined. The values of friction factor with vortex generators were obtained for the delta winglets parameters set used in the tests of heat transfer. The flow losses expressed in terms of f/f_0 are shown in figures 5a and 5b, where *f* is the friction factor for heat exchanger with vortex generators and f_0 the friction factor for the heat exchanger with smooth fin. For the heat exchanger geometries studied it was observed that the friction factor ratio (f/f_0) decreases when the Reynolds number increases from *Re*=300 to *Re*=1000. The ratio (f/f_0) varies between 1,14-1,21 for *Re*=300 and between 1,06-1,11 for *Re*=1000 as function of the longitudinal and transversal spacing between tubes.

The reduction of the flow losses, when vortex generators are placed in the fin-tube configuration, was observed initially by Fiebig *et al.* (1990) but for a simple geometry (only one finned tube) and for Reynolds numbers on the range of 2000-5000. The explanation for this behavior is the delay of the separation of the boundary layer on the tubes induced by the presence of the vortex generator. The delayed separation produces a form drag reduction on the tubes, while the vortex generators augment form drag and friction. Then, when the Reynolds number is increased, the reduction of the form drag on the tubes is higher than the increment of the form drag and friction introduced by the vortex generators and consequently the factor f diminishes respect to f_0.

It can be observed that the friction factor enhancement f/f_0 increases when the longitudinal spacing between tubes augments. This effect was observed for both of Reynolds number being more significant for Re=300 ($\Delta f/f_0 \approx$ 2,8%) than for Re=1000 ($\Delta f/f_0 \approx$ 1,3%).

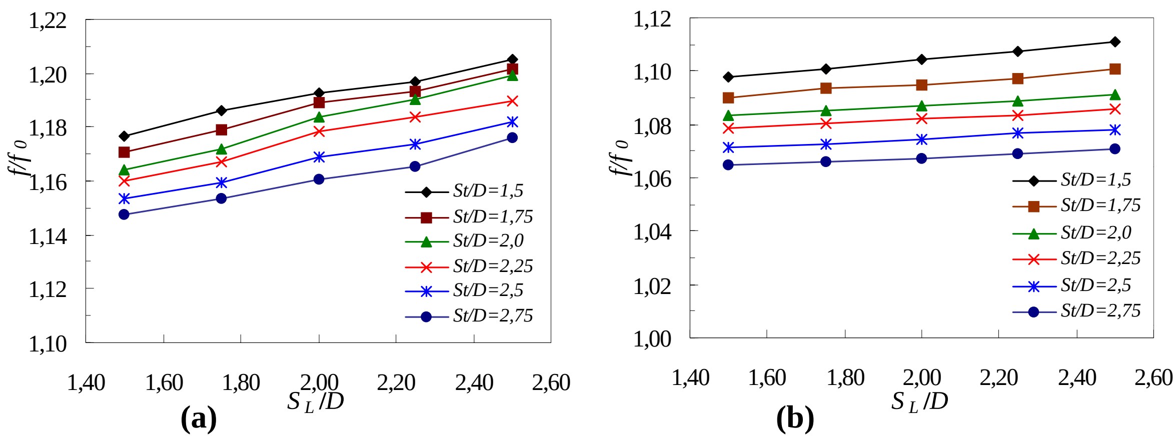

Figure 5. Friction factor enhancement in heat exchanger with two rows of tubes and vortex generators vs. the longitudinal spacing. **a)** Re=300, **b)** Re=1000.

The increase of the factor f/f_0 with the longitudinal spacing can be explained by the larger interaction area between the vortex and the fin. The extended fin, i.e., the heat exchanger with large longitudinal spacing permits the vortex to travel a larger length downstream both rows of tubes augmenting the friction in these regions. This effect is slightly smaller for Re=1000 than for Re=300 because for Re=1000 the free length is not enough to permit the vortices to dissipate before reaching the second row of tubes or the trailing edge of the fin.

The friction factor enhancement f/f_0 also varied with the changes of the transversal spacing. In this case, it was observed a diminution of f/f_0 when the transversal spacing was increased. The reduction of f/f_0 reached approximately 2,9% for Re=300 and 3,4% for Re=1000 when the transversal spacing was varied from S_T/D=1,5 to 2,75. This behaviour due to the fact that the area where the vortex act practically do not change while the fin area increases with the transversal spacing increment. Consequently, the friction losses produced by the vortex generators have smaller relative importance than the total losses, while f diminishes approaching to f_0 of the geometry without vortex generators.

Heat transfer enhancement and flow losses. Global analysis. For the evaluation of any heat transfer augmentation technique is necessary to consider, at the same time, the effect on the heat transfer and on flow losses. One criterion for this evaluation is to determine the value of the relation: (Nu/Nu_0 / f/f_0). This relation indicates which is the predominant effect from the point of view of the 1st Law of Thermodynamics. The values (Nu/Nu_0 / f/f_0) for heat exchangers with two rows of tubes in staggered arrangement with vortex generators are shown in the Figures 6a and 6b for different transversal and longitudinal spacing.

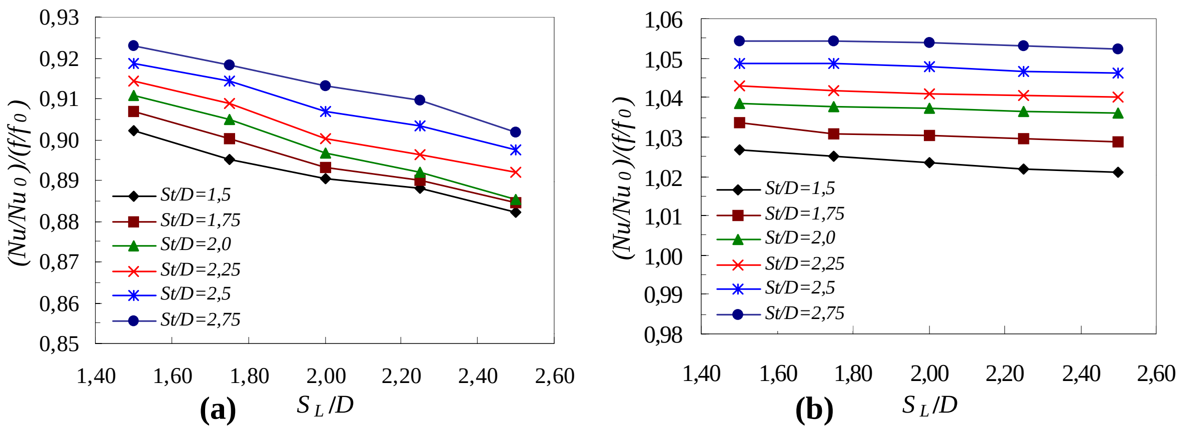

Figure 6. (Nu/Nu_0 / f/f_0) versus longitudinal spacing. **a)** *Re*=300, **b)** *Re*=1000.

The values of (Nu/Nu_0 / f/f_0) obtained for Reynolds number *Re*=300 were smaller than 1 for all tested geometries of heat exchanger (Fig. 6a), while for Reynolds number *Re*=1000 the factor (Nu/Nu_0 / f/f_0) was ever higher than 1 (Fig. 6b). This indicates that the enhancement effect of the vortex generators is superior to the effect on the flow losses for the Reynolds number *Re*=1000.

(Nu/Nu_0 / f/f_0) presents minimum variation with the increment of longitudinal spacing because the heat transfer enhancement Nu/Nu_0 and the flow losses f/f_0 increase practically in the same way with the increase in the longitudinal spacing. However, the factor (Nu/Nu_0 / f/f_0) augments when the transversal spacing increases because Nu/Nu_0 diminishes (0,2-0,3%) less than the flow losses f/f_0 (2,9%) with the increase of the transversal spacing. From the point of view of the 1st Law of Thermodynamics the vortex generators are more interesting as enhancement technique when the Reynolds number is large.

Second Law Analysis. The heat transfer enhancement evaluation criteria based on the 2nd Law of Thermodynamics compares the exergy destroyed by the enhanced surface with the exergy destroyed by the smooth surface.

For the determination of exergy values for both surfaces it was necessary to consider in a determined application. Here the surfaces were taken as part of a evaporator of domestic air-conditioning system. The ambient temperature was chosen as 25°C and the fin temperature was assumed as 10°C. For this conditions the relation *Ns* between the destroyed exergy of the surface with vortex generators, Ex_{GVL}, and the destroyed exergy of the smooth surface, Ex_0, was calculated. The results of *Ns* as function of the longitudinal spacing between tubes are shown in the Figures 7a and 7b for the values of Reynolds number *Re*=300 and *Re*=1000.

The obtained *Ns* values varied between 1,009-1,014 for *Re*=300 and between 1,04-1,07 for *Re*=1000 for the different geometries of tested heat exchangers. The change of the destroyed exergy *Ns* with respect to spacing between tubes was not significant for *Re*=300. For Reynolds number *Re*=1000 the change in the destroyed exergy resulted higher than for *Re*=300 when the transversal and longitudinal spacing were varied, being 1,6% approximately. The results indicate that the presence of the vortex generators for *Re*=300 practically do not destroys additional exergy with respect to the smooth fin. However, for Re=1000 the destroyed exergy reaches more significant values.

The Figures 8a and 8b confirm this affirmation. Here it is represented the relation between the heat transfer enhancement and the dimensionless destroyed exergy versus the longitudinal spacing. Comparing both cases (*Re*=300 and *Re*=1000) it is observed that the difference, in absolute values, of the relation $(Nu/Nu_0)/Ns$ for the same spacing is between 1,0% and 2,4%. Considering that $(Nu/Nu_0)/Ns$ is a *gain-penalty* relation of the effect introduced by the vortex generators inside heat exchanger geometry. It is possible to conclude that from the point of view of the 2nd Law of the Thermodynamics the effect of the vortex generators is positive (between 5-7,5%) and very similar for both values of Reynolds numbers. According to the 1st Law the vortex generators present positive effects only when the Reynolds number is large, but this result should not be taken absolutely because as it was demonstrated through the 2nd Law analysis the relation *heat transfer enhancement-destroyed exergy* of the vortex generators is almost similar for both Reynolds numbers used in the present work.

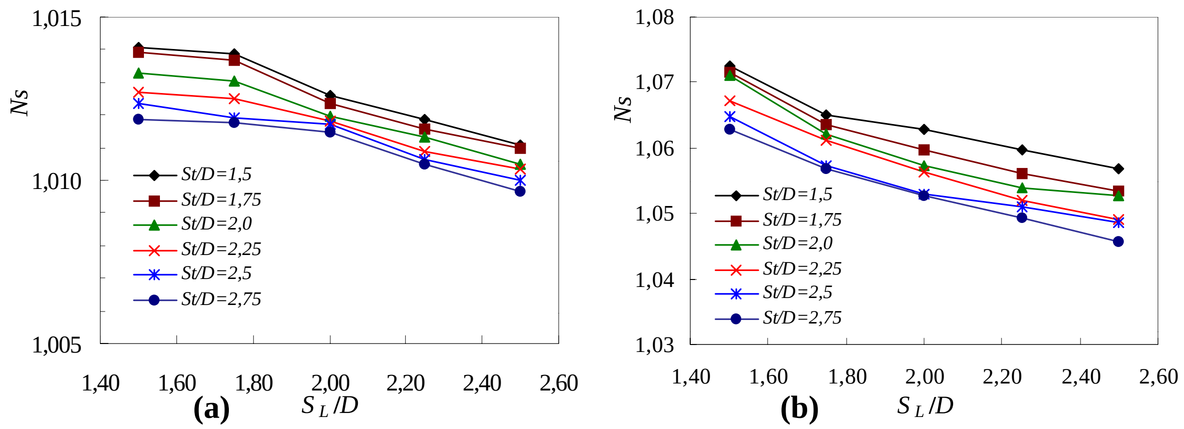

Figure 7. Destroyed exergy for the DWP with respect to the smooth fin in heat exchangers with two rows of tubes vs. the longitudinal spacing. **a)** *Re*=300, **b)** *Re*=1000.

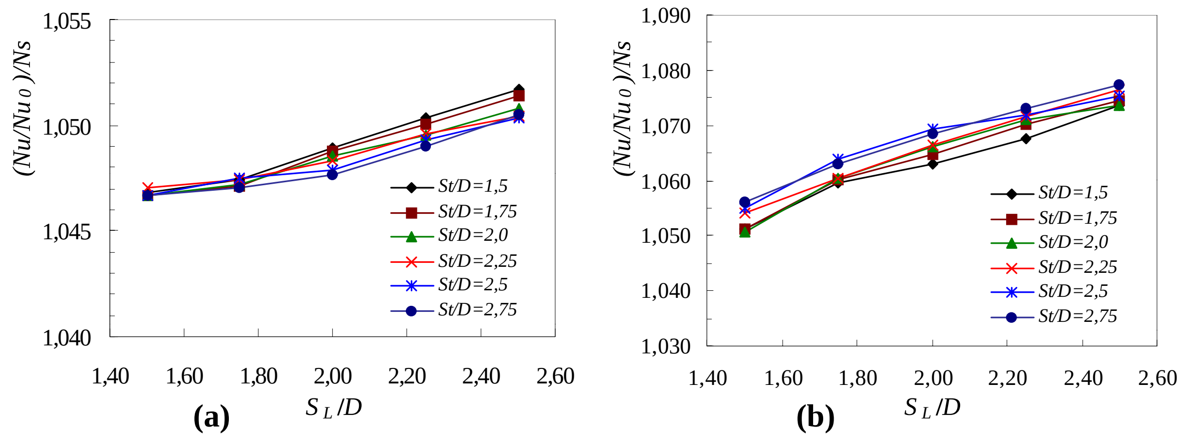

Figure 8. Relation heat transfer enhancement-dimensionless destroyed exergy vs. longitudinal spacing. **a)** *Re*=300, **b)** *Re*=1000.

CONCLUSIONS

The results showed that the Delta Winglet Pair vortex generators enhance the heat transfer on the airside of fin-tube heat exchangers with two rows of tubes. Heat transfer enhancement of up to 6,4% for *Re*=300 and 13,5% for *Re*=1000 were obtained with position z_g/D=0,4, s/D=1,0, aspect ratio of Λ=2 and attack angle β=45°. The heat transfer enhancement showed a slight growth with the increase of the longitudinal spacing between tubes (0,2-0,7% as function of Reynolds number). The behavior of the heat transfer enhancement with the transversal spacing was opposite to that observed for the longitudinal spacing. This indicates that the large influence of vortex generators on the heat transfer is obtained for small transversal spacing and large longitudinal spacing.

The results indicate that for *Re*=300 the use of vortex generators in heat exchangers with two rows of tubes in staggered arrangement produces an enhancement effect on the heat transfer 0,88-0,93 times smaller than the augmentation effect on the flow losses, meanwhile for *Re*=1000 the behavior is different, being the heat transfer enhancement 1,02-1,06 times larger than the flow losses augmentation.

However, the analysis through of the 2nd Law of Thermodynamics demonstrated that for all studied heat exchangers geometries and Reynolds numbers the heat transfer enhancement was 1,05-1,075 times superior than the augmentation of the exergy destroyed.

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