HEAT TRANSFER ENHANCEMENT OF A PLATE FINNED ELLIPTICAL TUBE USING VORTEX GENERATORS: AN EXPERIMENTAL STUDY.

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SUMMARY. *Vortex generators have been extensively used in the recent years for different fin-tube compact heat exchanger geometries. The heat transfer enhancement effects and the relatively low increments in the flow losses were verified for the circular tube geometry. However, the use of vortex generators in elliptical tubes, which have better aerodynamic performance, was not carried out yet. In this work, the influence of the position and the angle of attack of the vortex generators and the Reynolds number on the heat transfer enhancement of a finned elliptical tube with eccentricity 0.5 was analyzed. The naphthalene sublimation technique was used to determine the average Nusselt number. The changes in the flow patterns on the fin surface were observed using a mass transfer technique based on color sensitive evaporation. An instrumented open circuit wind tunnel was used for the experimental tests.*
KEYWORDS. *Vortex generators, elliptical tube, compact heat exchangers, naphthalene sublimation.*

1. INTRODUCTION.

Heat transfer enhancement techniques have been applied to industrial compact heat exchangers over the years because they yield the reduction of production and operation costs. Vortex generators (VG) with different geometries have been used on the gas side of fin-tube heat exchangers to enhance the heat transfer. The generated vortices system produces a local increase of velocity, thinning the boundary layer and mixing the fluids with different temperatures. These mentioned mechanisms are responsible for the increase of the local and average heat transfer coefficients, as noted by different authors in the last years (Yanagihara & González, 1996). Flow losses reduction, especially for high Reynolds numbers, have been observed in some cases. The complex interactions between the vortex system, the boundary layer and the mean flow in a heat exchanger passage are very difficult to predict. Therefore, it is necessary to study separately each possible application with its particular geometry. For a fixed heat exchangers geometry, the influence of each vortex generator parameters, e.g. position, angle of attack, aspect ratio and type of generator must be investigated.

Early in the sixties, Bauer apud Kays and London (1984) compared the performances of staggered banks of oval and circular finned tubes with approximately similar pitches. The heat transfer coefficient was 15 % higher and the pressure drop was 25 % lower for the oval tubes. A smaller wake area and a lower form drag were found to be responsible for the advantages in the performance. Ximenes (1981) carried out a study of heat exchangers with finned elliptical tubes by means of naphthalene sublimation. Heat exchanger models with transversal and longitudinal adimensional pitches of 2.50 and 2.20, respectively, were tested. The average and local Nusselt numbers for the models with one and two rows were obtained experimentally. The Reynolds number was varied between 200 and 1400. Additionally, three values for the tubes eccentricity, 0,5; 0,65 and 1, were studied. In contrast with the Bauer's work, substantial differences were not found in the heat transfer parameters between circular and elliptical geometry of tubes, independently of the value of the eccentricity. The explanation for the above conclusion was found in a balance between a smaller wake region and a less significant horse vortex system, when the eccentricity is diminished from 1 (circular tubes) to 0.65. Although the pressure drop was not measured in this work, it constitutes an important initial reference to studies of heat exchangers with finned elliptical tubes.

Fiebig et al. (1994) used liquid crystal thermography to obtain friction and heat transfer data for heat exchangers with three rows of staggered flat and round tubes, for smooth fin and fin with vortex generators. The Reynolds number, using the height channel as characteristic dimension, was varied between 600 and 3000. Comparing the results for tround and flat tubes, they concluded that the influence of vortex generators was determinant in heat exchangers elements using flat tubes. The Nusselt number ratio (enhanced surface over smooth surface) was several times higher for flat tubes as compared to round tubes configuration. The poor thermal performance of the heat exchangers with inline arrangement was increased by the use of vortex generators because of the larger traveling path of the vortices. The pressure drop increased a lot by the use of vortex generators in flat tube configuration.

Rocha et al.(1997) developed a numerical two dimensional model to compare the thermal performance of elliptical and round tubes. The results of this work shown a fin efficiency gain of the elliptical tubes of up to 18 %, as compared to the circular tubes. In addition, the higher efficiency values was observed in tubes with eccentricity of 0,5.

Bordalo & Saboya (1999) carried out an experimental study of the pressure drop for elliptical tubes using the same eccentricity of Ximenes (1981). The aerodynamic advantage of elliptical tubes, as compared to round tubes, was evident for Reynolds numbers, based in the height channel, exceeding 1000. This behavior occurred because the viscous drag is larger than the form drag at low velocities.

In the last years some papers have dealt with heat exchangers with non circular tube geometry and vortex generators (Cheng et all. (1998a) and (1998b)). Further parametric studies with vortex generators applied on different heat exchangers models are necessary. In the authors knowledge, vortex generators and elliptical tubes have not been combined yet. Only oval tubes, with low eccentricity, and flat tubes had been studied.

The present work consists of an experimental investigation about the influence of combining a plate-finned elliptical tube with vortex generators mounted on the fin surface. The objective is to search for the best position to place the vortex generators on the fin surface. The experimental technique used for the determination of the average Nusselt number was the naphthalene sublimation technique because it is reliable, easy to use and with high accuracy. The local visualization was implemented spreading wetted chalk on the fin surface, as a color sensitive evaporation material, in order to get quantitative information about the local mass transfer coefficient. The knowledge of the better location is very useful for future investigations on more complex heat exchanger models formed by one, two or higher number of tube rows. Additionally, it was studied the influence of Reynolds number and angle of attack on the average Nusselt number.

2. EXPERIMENTAL APPARATUS AND PROCEDURE.

The experiments were conducted in an instrumented open circuit wind tunnel (Fig.1). It consists of a contraction at the inlet, a test section, a diffuser, a centrifugal fan and a discharge tube. A flow straightener and a grid are placed before the test section to guarantee uniformity of the velocity profile at the entrance of the tested model. The test section made of acrylic, with a transversal section of 0,26 *m* of width and 0,090 *m* of height, , was build in such a way to permit a rapid and easy access to the model. The wind tunnel should be open circuit when naphthalene sublimation technique is used, because it avoids the inner air contamination with vapors of naphthalene. A liquid thermometer with resolution of 0.1 ^oC was placed in the rear part of tunnel to measure the temperature of the air flowing through the model. The flow rate was measured by means of a vortex flowmeter with 1% of uncertainty. The atmospheric pressure was obtained with a barometer. A hygrometer furnished the air relative humidity. Finally, an electronic balance of high resolution (10^{-4} g) was used to weight the test specimen before and after each experimental run.

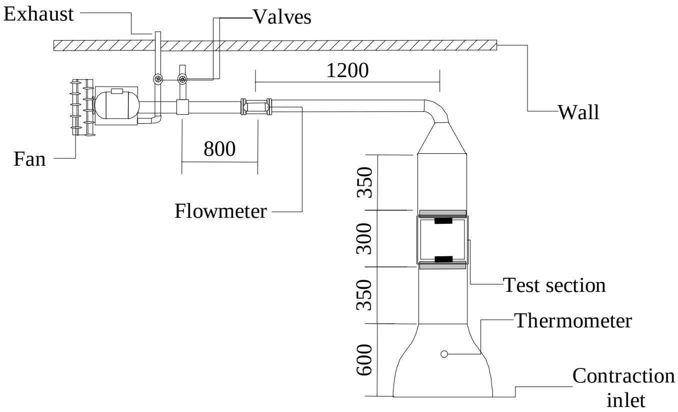

Figure. 1 Experimental open circuit wind tunnel

The finned elliptical tube model was constructed using polyurethane foam for the tubes and acrylic for the fins. Aluminum plate was the material used to manufacture the vortex generators. In the figure 2, it is presented a picture of the smooth model formed by four fins and a tube located in the centerline of the fins.

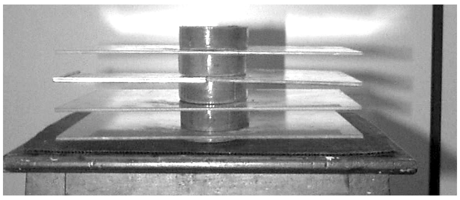

Figure 2. The smooth finned elliptical model.

Figure 2 show one acrylic fin substituted by an aluminum fin, but maintaining the same geometrical proportions. This special fin served as a mold to receive melted naphthalene. The surface of naphthalene that emulates the fin surface was obtained by casting and using a clean glass plate as cover for the mold. Additional details for the casting process can be found in the bibliographic reference (González, 1996).

The dimensions of the model were taken from real heat exchangers used in air conditioning equipments. It was adopted an adequate scale (1:10) to allow the vortex generators (VG) to be placed on the fin, as shown in table1 and figure 3. In this figure A_g is the area of the vortex generator, β the angle of attack, s the distance between the tips of a pair of vortex generators, H and b are the height and the base length of the vortex generators, respectively. All adimensional parameters were obtained with reference to the smaller diameter of the ellipse ($D_2=6.35$ cm). The eccentricity used was 0.5 because it was found to be the most efficient by Rocha et al. (1997).

Table 1. Dimension of model and vortex generators used in this work.

S_L/D_2	S_L/D_2	E/D_2 *and* H/D_2	$\Lambda=2H/b$ *(Aspect ratio)*
4	*3*	*0.28*	*1*

The experimental procedure consists of assembling the model with the special fin, covered by naphthalene, immediately after weighting the fin. The chronometer is started when the weight is taken and a series of determined procedures is followed until the model is placed in the tunnel. Then it is removed and its weight is taken again. This procedure permits to measure the mass of naphthalene lost by natural sublimation during the manipulation of the model as function of the time elapsed, for later mass correction.

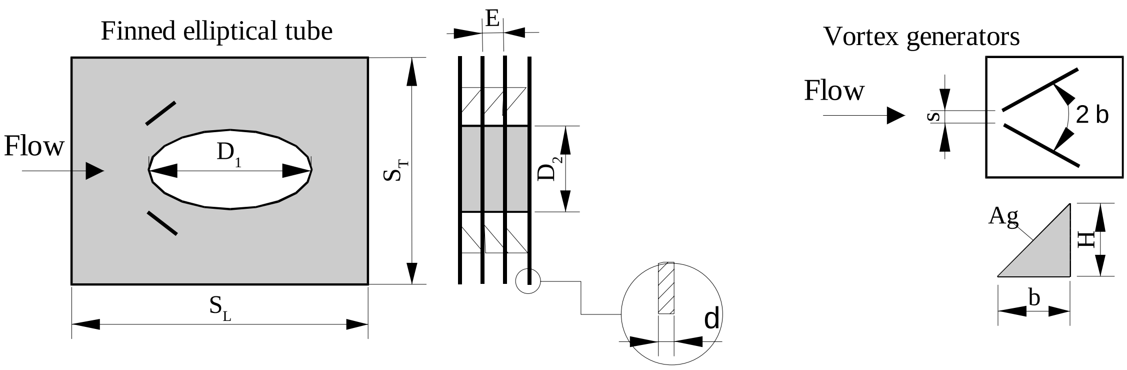

Figure 3. Parameters of heat exchanger model and vortex generators.

Finally, the fan is turn on and a warm-up period begun. After the tunnel warm-up, the model is placed into the test section and the chronometer is started. The air temperature was taken each two minutes and the room conditions were also monitored. After 35 or 40 minutes *($\Delta\tau$)* the fan is turned off and the model is removed, the special fin is dismounted and weighted. If the room temperature changed more than 0.7 ^{0}C during the test run, the test is interrupted and the naphthalene mold is rejected.

In this work it was used an aspect ratio of unitary value for one pair of vortex generators symmetrically placed in both of sides of the tube. A delta winglet pair of vortex generator type was chosen and an angle of attack of 45^o was initially used.

3. DATA REDUCTION PROCEDURE

The average Nusselt number is determined as follows. Each set of measurement is processed beginning with the calculation of the mass transferred during the test *(Δm)*, obtained by the difference between the initial and the final weight of the specimen (m_i e m_f) and considering also the mass lost by natural sublimation Δm_{ns} :

$$\Delta m = m_f - m_i - \Delta m_{ns} \tag{1}$$

The averaged mass transfer coefficient h_m was determined by :

$$h_m = \frac{\Delta m / \Delta\tau}{(\Delta\rho_{log})A_f} \tag{2}$$

where A_f is the fin area covered with naphthalene. In this equation the mean logarithmic vapor density $\Delta\rho_{log}$ is calculated by the equation 3:

$$\Delta\rho_{log} = \frac{(\rho_{vw} - \rho_{v\infty in}) - (\rho_{vw} - \rho_{v\infty out})}{ln\left(\dfrac{\rho_{vw} - \rho_{v\infty in}}{\rho_{vw} - \rho_{v\infty out}}\right)} \tag{3}$$

where the density of vapor at the fin surface level *(ρ_{vw})* is calculate using the ideal gas law at the surface temperature. The vapor pressure of naphthalene is obtained by means of Ambrose's correlation (1975). The vapor density of the mainstream *($\rho_{v\infty out}$)* at the exit of the channel was calculated using the equation 4. The vapor density of the mainstream at the entrance of the model *($\rho_{v\infty int}$)* is considered null because the wind tunnel is open circuit.

$$\bar{\rho}_{v\infty out} = \frac{\Delta m/(\Delta\tau)}{Q} \tag{4}$$

The denominator of equation 4 is the volumetric air flow rate in the channel, between two consecutive fins. Therefore, the mass transfer Stanton number could be determine by the following relation:

$$St_m = \frac{h_m}{u} \tag{5}$$

where u is the average velocity in the minimum free flow area of the channel, function of the flow rate through the tunnel.

From the analogy between the heat and mass transfer we could write:

$$St_h = St_m \left(\frac{Pr}{Sc}\right)^{2/3} \tag{6}$$

where St_h is the heat transfer Stanton number and Sc the Schmitd number, obtained from Cho's correlation (1992) as function of the temperature T_w

$$Sc = 2.28\left(\frac{T_w}{298.16}\right)^{-0.1526} \tag{7}$$

The Reynolds number was calculated using the Kays & London (1984) definition of the hydraulic diameter The Nusselt number was calculated by the equation 8.

$$Nu = St_h \; Re\, Pr \tag{8}$$

Finally the heat transfer enhancement E is evaluated using a enhancement ratio between the averaged Nusselt number for the smooth *(Nu_0)* and the augmented *(Nu)* surface.

$$E = \frac{Nu}{Nu_0} \tag{9}$$

4. RESULTS AND DISCUSSION.

The experimental technique and the procedure used in this work was certified by comparison with experimental results available in the literature. Figure 4 presents the experimental results by Ximenes (1981) and those calculated in the present work for the same transversal pitch. The values were plot using the Sherwood number *(Sh)* to avoid the possible divergence introduced by the exponent of the heat and mass transfer analogy and the Schmidt number. It can be said that the present results are in good agreement with the previous experiments.

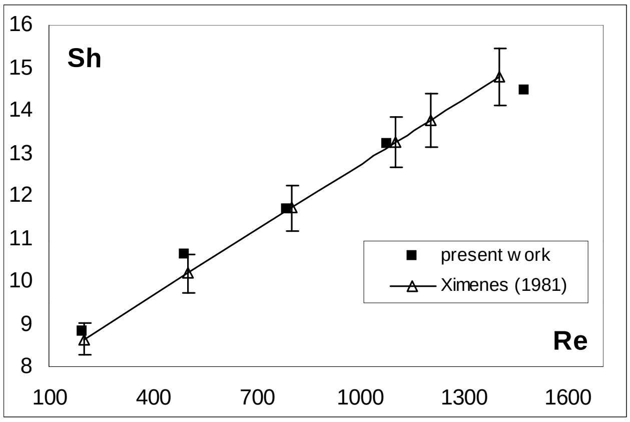

Figure 4. Sherwood number vs. Reynolds number for Ximenes (1981) and the present work.

Figure 6 presents the heat transfer enhancement results for the finned elliptical tube obtained using vortex generators. The generators position was varied to sweep all the fin surface around the tube. In the abscissas axis, they are plotted numbers from *0* up to *16*, each one of them corresponding to a different vortex generators position identified with the same number at figure 5

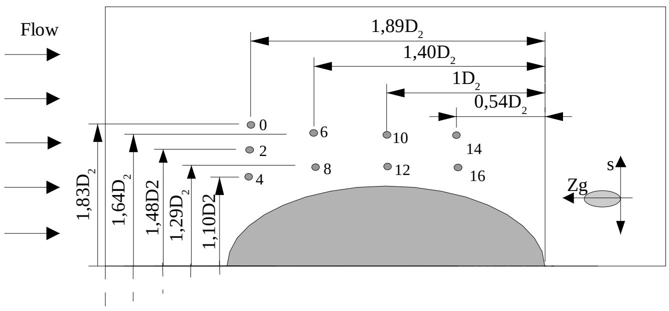

Figure 5. Vortex generators positions tested.

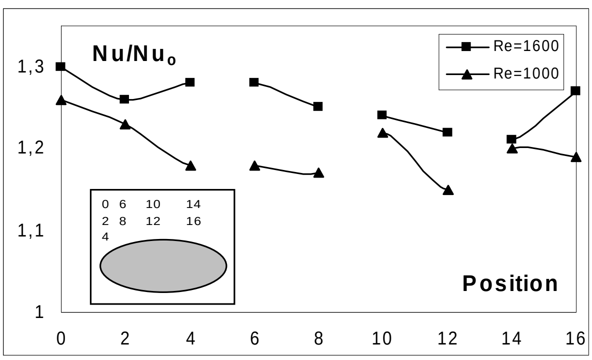

Figure 6. Nu/Nu_0 as function of the position of vortex generators for two Reynolds number.

The preliminary results plotted in figure 6 show approximately the same behavior for the two Reynolds numbers tested in this work. Generally speaking, the heat transfer enhancement diminishes when the vortex generators is placed farther from the fin entrance. This result is straightforward because the region of vortex actuation is increased gradually when the position is dislocated toward the entrance region of the duct. Therefore, the area with a higher local velocity is larger when compared to the non disturbed region.

Additionally, it can be noted in the same graphic another interesting behavior which is the increasing of heat transfer when the location is far from the tube surface. The interaction between the vortices generated by wings and the horseshoe vortices generated in front of the tube should be the reason for this fact. In this situation both vortical systems act in the same region (clearer areas in figure 7 are indicative of higher mass transfer coefficient). When the vortex generators are placed farther from the tube surface, their influence is exerted over a surface area where, in smooth condition, the heat transfer coefficient is lower. In other words, the potential for the heat transfer enhancement is lower when the vortex generator position is closer to the tube surface.

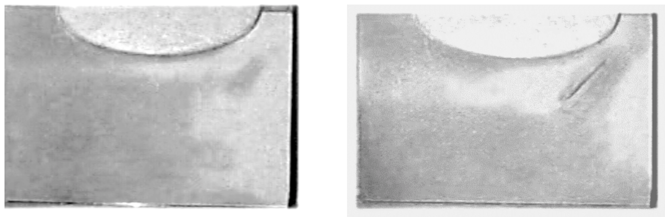

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Figure 7. Pictures showing a common actuation area for both of vortex systems.

a) Horseshoe vortex system (smooth fin). b) Both vortical systems (VG at position 4).

An explanation is necessary for the last positions (14 and 16) because their location is near the wake region where there is no influence of the horseshoe vortices and the heat transfer coefficient is low. The vortex generators in this region deviate the flow to the wake region, producing an increment of the heat transfer there. Some authors (Yanagihara & Gonzalez (1996)) reported that placing the vortex generators in the wake region of the tube improves the thermal performance. Finned elliptical tube with eccentricity of 0,5 presents also a recirculation region behind the tube, explaining the above results.

Considering these previous results, another group of experiments were conducted. The area of study was refined and the vortex generators were placed only around the position where the highest heat transfer enhancement was reached at the preliminary experiments. The new locations intent to refine the study and they are defined in the following figure.

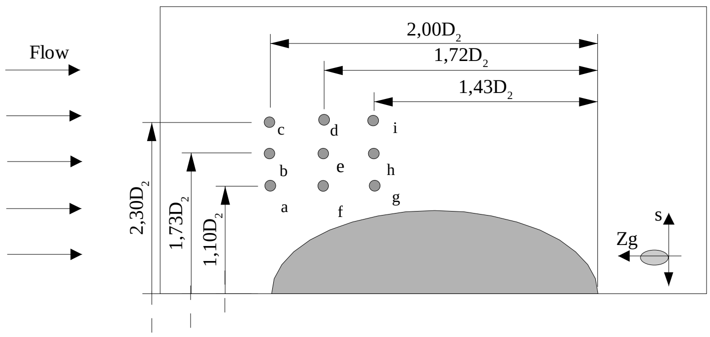

Figure 8. New locations for vortex generators around the position where the highest heat transfer enhancement was reached.

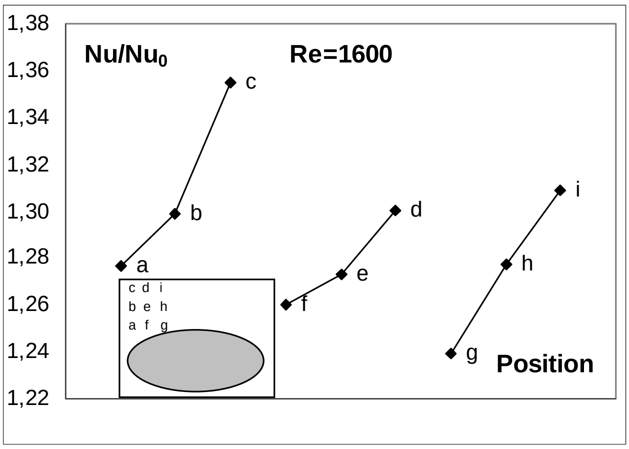

Figure 9. Nu/Nu$_0$ as function of the new locations of vortex generators.

In the figure 9, the tests for Reynolds numbers Re = 1600 are presented. In this figure, it can be observed the same tendencies already noted in the analysis of the figure 6: the intensification is increased when the VG location is farther from the tube surface and when it is dislocated toward the entrance region of the fin. The points located at larger distances from the entrance edge present the lower values for heat transfer enhancement. However, these values are higher than those for points located at the end of the fin in the preliminary experiments (Fig. 5 and 6) The authors did not seek the locations for the maximum heat transfer enhancement farther from the tubes because it has not practical importance in compact heat exchangers as the vortex generator need to be placed between tubes.

The influence of the Reynolds number was analyzed for the point where a highest heat transfer enhancement was found. This plot is shown in the figure 10 where the heat transfer enhancement is increased when the Reynolds number is augmented. The ratio Nu/Nu$_0$ did not vary linearly with the Reynolds number, probably because of the short length of the model and the increasing relative importance of the horseshoe vortices system, for higher velocities.

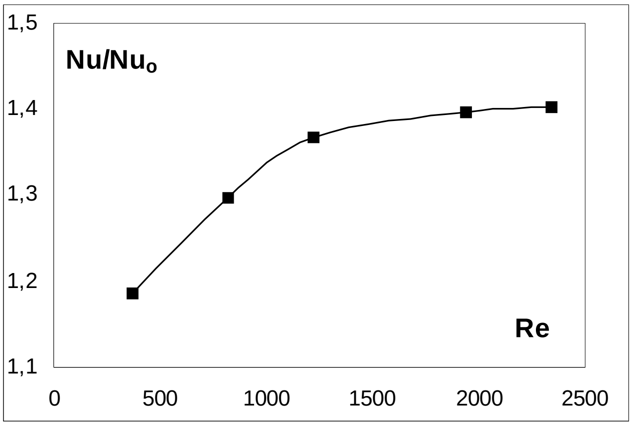

Figure 10. Influence of the Reynolds number in the heat transfer enhancement.

The angle of attack β formed between the vortex generator and the flow direction was varied and five different values were tested for the position "c". The results (figure 11) shown coincidence with other studies found in the literature (Gonzalez, 1996). For higher angles of attack the heat transfer increases, reaching a maximum near 55^o. When the angle of attack is too much increased, it occurs a flow deviation more than a stronger vortex formation.. The shots in the figure 12 show how an angle of attack of 60^0 produces a main vortex (visualized by a clear area situated behind the winglet) weaker than the same vortex generators with $\beta=45^0$.

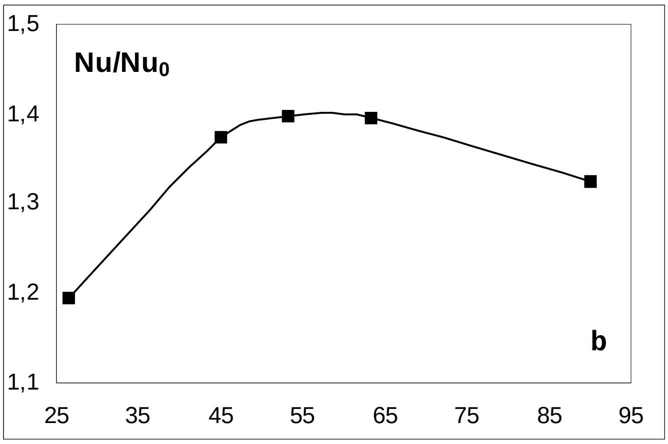

Figure 11. Influence of angle of attack on the heat transfer enhancement, for Re=1600.

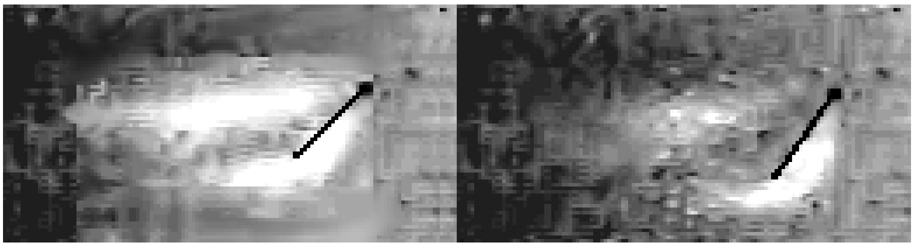

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Figure 12. Mass transfer visualization for two values of the angle of attack.
a) $\beta=45^0$ b) $\beta=60^0$.(González, J. J. (1999).)

5. CONCLUSION.

Vortex generators enhances the heat transfer when applied to a model of compact heat exchanger with finned elliptical tubes.

In the plate-finned elliptical tube, the average mass transfer coefficient was found to be higher when the location of the vortex generator was closer to the front leading edge of fin and farther from the tube surface.

The influence on heat transfer of the position of vortex generators on the fin surface was investigated and the best result was obtained for the point "c", located at $s/D_2= 2.30$ and $Zg/D_2=2$, with reference to the tube (figure 7). The heat transfer enhancement rate obtained at this location should not be considered a maximum because it was not detected a diminution beyond it. However, amplification of the range studied has no practical significance because of tube arrangement constraints.

The increasing of the Reynolds number always produces higher heat transfer enhancement for the range of Reynolds number tested.

Increasing the angle of attack β improves the heat transfer behavior but there is a maximum value for this parameter.

This paper demonstrate the possibility to improve the thermal performance of a plate-finned elliptical tube by the use of vortex generators. It indicates the need of future investigations where more complex models could be studied.

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