THERMAL AND FRICTION CHARACTERIZATION OF COMPACT HEAT EXCHANGER WITH ONE AND TWO ROWS OF ELLIPTICAL TUBES.

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SUMMARY. *Compact heat exchangers are used in many technical applications. This work presents an experimental characterization of plate fin compact heat exchangers with one and two rows of elliptical tubes in the staggered arrangement. The average Nusselt number and friction factor for several transversal and longitudinal pitches were obtained. The naphthalene sublimation technique and the mass transfer analogy were applied to determine the Nusselt number. The friction factor was calculated by flow losses measurements using a weighted method by means of a balance. Experiments were carried out in an open circuit wind tunnel in the laminar flow regime. The Reynolds number based on the hydraulic diameter were varied between 200 and 1500. Correlations for the Colburn and Friction factors were obtained.*

KEYWORDS. *Elliptical tubes, compact heat exchangers, naphthalene sublimation.*

1. INTRODUCTION

Compact heat exchangers are used extensively in the automotive industry and refrigeration applications. Minimal space requirement, heavy duty heat service and the lowest pressure drop possible are common exigencies for these equipments. Many applications involving gas-to-gas or liquid-to-gas heat transfer, with low values of heat transfer coefficients at the gas side, require the determination of the overall thermal capacity with high accuracy. In general, correlations for heat exchangers are very useful for designers but usually these are restricted to few geometries and arrangements.

Correlations for plate fin compact heat exchanger with staggered round tubes were developed by McQuiston (1978) and Gray & Webb (1986) apud Webb (1994), the latter being more accurate. These correlations are valid for the following ranges: $2.55 \geq S_T/d_0 \geq 1.97$ and $2.58 \geq S_L/d_0 \geq 1.7$, $500 < \mathrm{Re} < 24\,700$.

Recently Abu et al (1998) carried out and experimental study where *28* heat exchangers samples with plain and corrugated fins were tested. This paper presents correlations for the prediction of Colburn factor *j* and the *f* factor with higher accuracy than earlier correlations, being valid for a larger range of the geometrical parameters.

The present literature lacks information about correlations for heat exchangers using non circular tubes. The elliptical tube geometry is one of the most studied non circular arrangement. However the information is not complete because of the large number of parameters affecting the performance of the heat exchanger using this geometry.

Bauer (1964) apud. Gray & Webb (1994) and Ximenes (1981) were the first to study this kind of geometry, comparing its performance to the round one. Bordalo & Saboya (1999) focused their work in the hydraulic performance of heat exchangers with one and two rows of tubes using a plain finned elliptical tube with the same pitches than those tested by Ximenes (1981).

The small number of correlations for non circular arrangement and their narrow scope of validity are the most important limitations found in the bibliography at present. The increasing number of papers dealing with tubes with non circular geometry are an indicative of the importance of this subject. The improvement of thermohydraulic performance of compact heat exchanger with finned tube is necessary because of the economical and environmental constraints. One way to reach this objective is to use tubes with better hydrodynamic performance and an smaller wake region. In this case, there is a new constraint which is the lower limit of internal pressure withstood by the tube when his cross section is farther from the circular one.

The present work is an experimental study of the thermal and friction performance of scaled models of plate fin heat exchanger with one and two rows of elliptical tubes. The analogy between the heat and mass transfer by means of the naphthalene sublimation technique was used for the average Nusselt number determination. The friction factors was evaluated using a weighted method by means of a balance.

The objective of this work is to obtain the average Nusselt number *(Nu)* and the friction factor *(f)* as function of the Reynolds number in the laminar regime for 36 heat exchanger models where the transversal and longitudinal pitches were systematically varied.

The importance of this study lays in the lack of information about heat exchangers with finned elliptical tube. Additionally, the thermal heat transfer characterization of this kind of heat exchanger will be used in future investigations as basis to calculate the effect of heat transfer enhancement technique applied on the surface of the fin.

2. MODELS

The geometrical parameters were chosen considering the dimensions of a real heat exchanger used in domestic air conditioning systems and scaling it to larger dimensions *(1:10)*. The flow regime was maintained laminar for all models and Reynolds numbers tested in this work.

In the table 1 it is presented the parameters of the models, the range of study and the step increment used for the variations.

Table 1. Parameter of the models, range of study and step increment used.

Dimensionless Parameter	*Range of study*	*Increment*
Transversal Pitch S_T/D_2	*1.75 – 3.00*	*0.25*
Longitudinal Pitch S_L/D_2	*2.25 – 3.25*	*0.25*
Reynolds number Re	*200 – 1500*	≈ 300
Fin pitch E/D_2	*0.26*	*Constant*
Fin thickness δ/D_2	*0.025*	*Constant*

The dimensionless parameter were obtained using the smaller diameter of the ellipse D_2 in agreement with table 1 and the Kays & London (1984) definition for the hydraulic diameter of the channel, as shown in the equation 1.

$$D_h = \frac{4A_{min}L_{ch}}{A_{ht}} \tag{1}$$

The variation of the transversal pitch required in this paper was made by changing the diameter of the tubes. Six different values to this magnitude were used. Combining these six values with the five longitudinal pitches tested results in thirty different models with two rows. Additionally, for one tube row, the longitudinal pitch was maintained constant and the six transversal pitches were tested too. The model's dimensions are resumed in the tables 2 to 8. D_1 is the larger diameter of ellipse and L_{ch} is the heat exchanger length.

Table 2. One row models. Fundamentals dimensions.

S_T/D_2	1,75	2	2,25	2,5	2,75	3.00
$S_L/D2$	2,75	2,75	2,75	2,75	2,75	2,75
D_2(cm)	7,14	6,25	5,56	5,00	4,55	4,17
D_1(cm)	14,29	12,50	11,11	10,00	9,09	8,33
L_{ch}(cm)	19,64	17,19	15,28	13,75	12,50	11,46
D_h(mm)	19.78	19.59	18.92	18.06	17.17	16.30

Table 3. Two row. Fundamental dimensions.(S_T/D_2=1.75 & 2.00)

S_T/D_2	1.75	1.75	1.75	1.75	1.75	2.00	2.00	2.00	2.00	2.00
S_L/D_2	2.25	2.50	2.75	3.00	3.25	2.25	2.50	2.75	3.00	3.25
D_2(cm)	7.14	7.14	7.14	7.14	7.14	6,25	6,25	6,25	6,25	6,25
D_1(cm)	14.29	14.29	14.29	14.29	14.29	12,5	12,5	12,5	12,5	12,5
D_h(mm)	18.07	17.83	17.63	17.48	17.34	18.14	17.93	17.76	17.62	17.51
L_{ch}(cm)	32.14	35.71	39.29	42.86	46.43	28.13	31.25	34.38	37.50	40.63

Table 4. Two row. Fundamental dimensions. (S_T/D_2=2.25 & 2.50)

S_T/D_2	2.25	2.25	2.25	2.25	2.25	2.50	2.50	2.50	2.50	2.50
S_L/D_2	2.25	2.50	2.75	3.00	3.25	2.25	2.50	2.75	3.00	3.25
D_2(cm)	7.14	7.14	7.14	7.14	7.14	7.14	7.14	7.14	7.14	7.14
D_1(cm)	14.29	14.29	14.29	14.29	14.29	14.29	14.29	14.29	14.29	14.29
D_h(mm)	17.68	17.50	17.36	17.24	17.14	17.01	16.86	16.74	16.63	16.55
L_{ch}(cm)	25.00	27.28	30.56	33.33	36.11	22.50	25.00	27.50	30.00	32.50

Table 5. Two row. Fundamental dimensions. (S_T/D_2=2.75 & 3.00)

S_T/D_2	2.75	2.75	2.75	2.75	2.75	3.00	3.00	3.00	3.00	3.00
S_L/D_2	2.25	2.50	2.75	3.00	3.25	2.25	2.50	2.75	3.00	3.25
D_2(cm)	7.14	7.14	7.14	7.14	7.14	7.14	7.14	7.14	7.14	7.14
D_1(cm)	14.29	14.29	14.29	14.29	14.29	14.29	14.29	14.29	14.29	14.29
D_h(mm)	16.27	16.14	16.03	15.94	15.87	15.52	15.40	15.31	15.23	15.17
L_{ch}(cm)	20.45	22.73	25.00	27.27	29.55	18.75	20.83	22.92	25.00	27.08

The materials selected to build the models were acrylic for the fins and polyurethane foam for the tubes. The picture in the figure 1 presents a photo of some models characterized in this work.

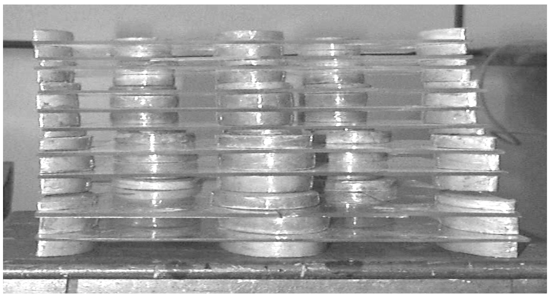


Figure 1. Picture of some models built during this work.

3. EXPERIMENTAL APPARATUS AND PROCEDURE.

The experiments were conducted in an instrumented open circuit wind tunnel consisting 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 consecutively located before the test section to guarantee uniformity in 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 0.045 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 a resolution of 0.1 ^{o}C was placed in the rear part of tunnel to measure the temperature of the air flowing trough the model. The flow rate was measured by means of a vortex flowmeter with 1% of uncertainty. The atmospheric conditions were obtained with a mercury barometer and a hygrometer. Finally, an electronic balance of high resolution (0.1 mg) was used to weigh the test specimen before and after each experimental run. A chronometer was used to measure the time elapsed for the naphthalene sublimation.

The pressure measurement was carried out by developing a novel weighting method. It consists of a tube with a pressure tap at one end and the other end submersed in a beaker with oil. The beaker is placed in a balance, care being taken to avoid the influence of the tube weight on the balance. The general arrangement of the system, including part of the tunnel with two pressure plenum chambers, are shown in the Fig. 2. The pressure difference, due to the flow losses, produces the elevation of the liquid inside the tube, making lower the scale reading. The relation between this reading and the pressure difference is called the manometer constant and can be obtained manipulating the equations of hydrostatic.

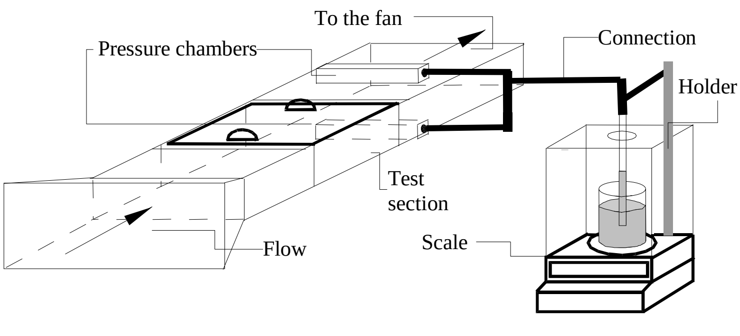


Figure 2. Experimental apparatus for pressure drop measurements.

4. 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} :

$$\Delta m = m_f - m_i - \Delta m_{ns} \tag{2}$$

The averaged mass transfer coefficient h_m was determined by :

$$h_m = \frac{\Delta m / \Delta\tau}{(\Delta\rho_{log})A_f} \tag{3}$$

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 4:

$$\Delta\rho_{log} = \frac{(\rho_{vw} - \rho_{v\infty in}) - (\rho_{vw} - \rho_{v\infty out})}{ln\left(\dfrac{\rho_{vw} - \rho_{v\infty in}}{\rho_{vw} - \rho_{v\infty out}}\right)} \tag{4}$$

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 the Ambrose's correlation (1970). The vapor density of the mainstream *($\rho_{v\infty out}$)* at the exit of the channel was calculated using the equation 5. The vapor density of the mainstream at the entrance of the model *($\rho_{v\infty int}$)* is considered null the wind tunnel is open circuit

$$\overline{\rho}_{v\infty out} = \frac{\Delta m/(\Delta\tau)}{Q} \tag{5}$$

The denominator of equation 5 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:

$$St_m = \frac{h_m}{u} \tag{6}$$

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

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

$$St_h = St_m \left(\frac{Pr}{Sc}\right)^{2/3} \tag{7}$$

where St_h is the heat transfer Stanton number and Sc is the Schmitd number, obtained from Cho's correlation (1989) as function of the temperature T_w

$$Sc = 2.28\left(\frac{T_w}{298.16}\right)^{-0.1526} \tag{8}$$

The Nusselt number could be calculated by the equation 9.

$$Nu = St_h\ Re\, Pr \tag{9}$$

The friction factor was calculated by the equation 10, where ΔP is the pressure drop produced by the model, ρ is the air density and L_{ch} the channel length. An area correction was used to normalize the friction factor.

$$f = \frac{\Delta P D_h}{2\rho L_{ch} u^2}\frac{A_f - A_{ff}}{A_f} \tag{10}$$

where A_{ff} is the frontal fin area of model and A_f is the frontal area of model.

The pressure drop through the model is obtained by the following manometer equation.

$$\Delta P = \frac{\Delta m_p\left(\frac{F_1}{F_2}+1\right)g}{\left(F_{ext}+F_1\right)} \tag{11}$$

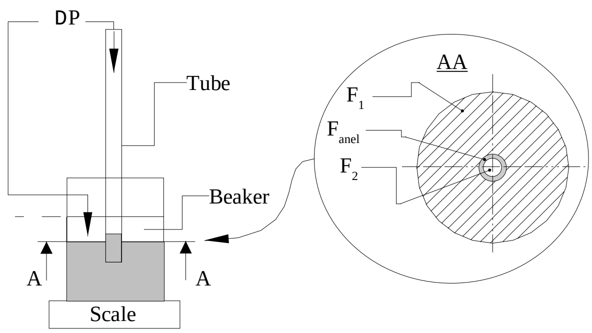


Figure 3. Beaker, tube, scale and areas definition.

where Δm_p is the scale reading, g is the acceleration of the gravity and the rest are areas defined in the Fig. 3.

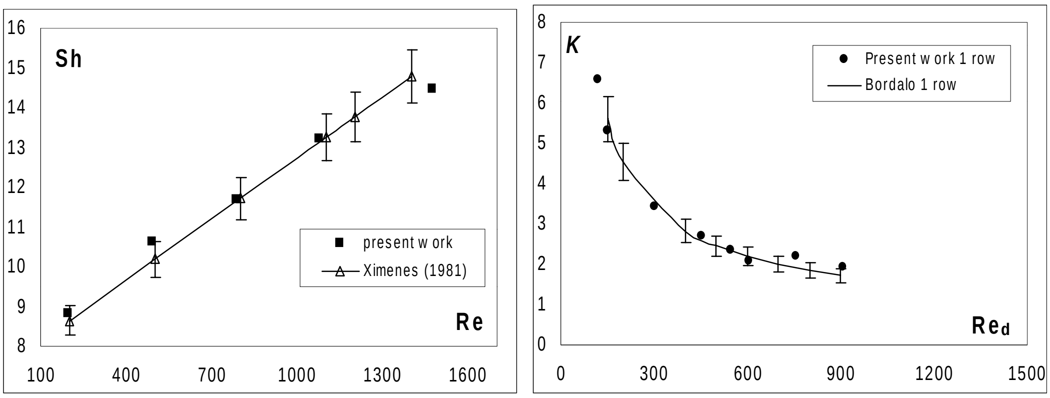
The uncertainty analysis (ANSI/ASME PTC 19.1-1985) was carried out the results indicated an uncertainty of 6 % for the Nusselt number and 10% for the friction factor, both uncertainties with 95% of confidence level.

5. RESULTS AND DISCUSSION.

The validation of the experimental technique was carried out by comparison with global heat transfer experimental results obtained by Ximenes (1981) and pressure drop results by Bordalo & Saboya (1999).

Figure 4a presents the experimental data reported by Ximenes (1981) and those calculated in the present work for one row of finned elliptical tubes with $S_T/D_2=2.5$. The values were plotted 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.

For the pressure drop, figure 4b presents the loss coefficient calculated by Bordalo & Saboya (1999) for one row of elliptical tubes and the same parameter used in the present work (same longitudinal and transversal pitches). The agreement is good, with the differences averaging *5.42%*. This value is lower than the experimental uncertainty. The Reynolds number (Re_δ) is based on the channel height and the front velocity. In general, the present results are in good agreement with previous works, although some deviations can be noted at lower Reynolds number, in which the uncertainties is higher for both referenced works.



a b

Figure 4. a)Sherwood number vs. Reynolds numbers for Ximenes (1981) and the present work. b) Loss coefficient versus Reynolds numbers for one row of finned elliptical tube. Certification.

Figure 5 presents the experimental results for models with one row, grouped in the table 2. The figure presents a log-log plot of the Colburn and friction factor for only three different S_T/D_2 values; the rest is not presented to avoid visual confusion in the graphics.

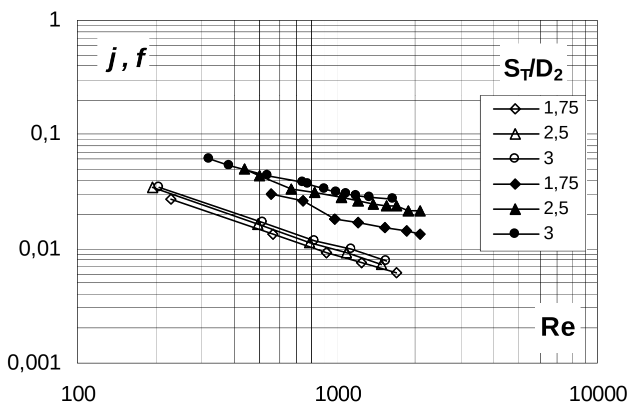


Figure 5. Colburn and Friction factors for one rows of finned elliptical tube with $S_L/D_2=2.75$.

The results of models having two rows of finned elliptical tubes is presented in the Fig. 6, 7 and 8 for S_L/D_2 values of *3.25, 2,75* and *2.25* and to the same values of S_T/D_2 already presented for one row models.

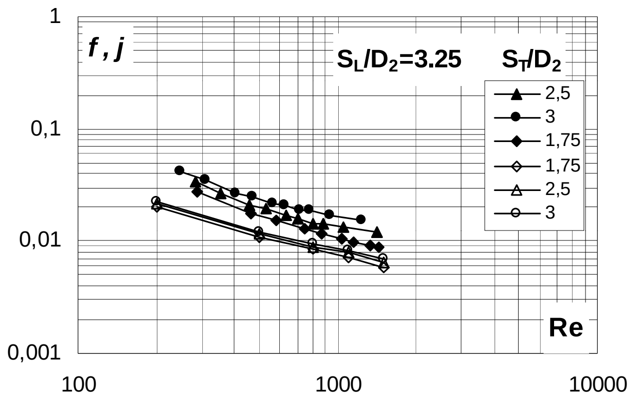


Figure 6. Colburn factor and Friction factor for two rows of finned elliptical tubes with $S_L/D_2=3.25$.

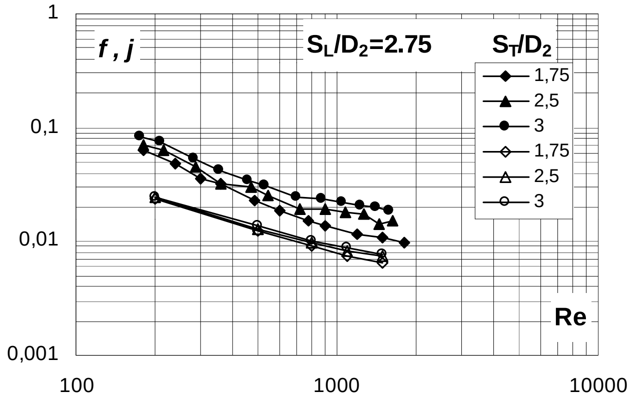


Figure 7. Colburn factor and Friction factor for two rows of finned elliptical tubes with $S_L/D_2=2.75$.

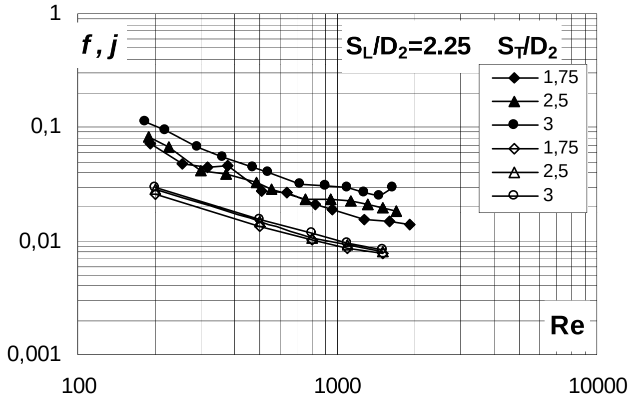


Figure 8. Colburn factor and Friction factor for two rows of finned elliptical tubes with $S_L/D_2=2.25$.

The observation of Fig. 5 to Fig. 8 indicates the influence of the transversal and longitudinal pitches and the number of tube rows on the thermal performance of heat exchanger models with finned elliptical tubes. The increase of the dimensionless transversal pitch produces an increment of the Colburn factor. This behavior happens for higher values of the Nusselt number. This is possible only if the heat transfer coefficient grows, because the hydraulic diameter decreases as the transversal pitch is increased. This phenomena could explain the small sensitivity to heat transfer when S_T/D_2 is augmented. Although the tendency was similar for models with one or two rows, it was observed a larger sensitivity in the first case. The increment in the average heat transfer coefficient is possible when S_T/D_2 is higher, because of the consequent smaller length of the models and the gradual reduction of wake region when the tube diameter is reduced. This is more significant for one-row models, considering the higher value of the mean heat transfer coefficient over the channel. It is also necessary to consider the higher velocities present in the channel when D_h is lower, for a certain Reynolds number.

Diminishing the longitudinal pitch has a positive impact on the heat transfer behavior, producing an increment of the Nusselt number by reducing the total fin length. In this case the hydraulic diameter grow when the length is reduced. At the same time, the heat transfer coefficient increases and both have a large influence on the variation of the Nusselt number with S_L/D_2.

The friction factor has the same behavior of the Colburn factor as far as the pitch variation is concerned. When the transversal pitch is augmented the friction factor f increases because the relation D_h/L_{troc} also increases and its effect is predominant over the friction factor, despite the fact that the pressure drop diminishes with S_T/D_2. Analyzing the influence of S_L/D_2, it is observed a reduction of the friction factor when the longitudinal pitch is increased. In this case, the diminishing of the relation D_h/L_{troc} is determinant despite the higher pressure drop caused by larger fins.

Finally the influence of the row number could be analyzed by comparison of the Fig 5 and 7 corresponding to one and two rows models having the same transversal and longitudinal pitches. It was noted an approximation between their behaviors for the highest Reynolds numbers. On the other hand, the models with one row of tubes have a better thermal performance when the velocities are low (note a more negative slope of j vs Re plotted in Fig. 5 when compared to Fig 7). The growing importance of the vortical system and a minor contribution of the boundary layer in the second row of models having two rows, is the reason for the improvement of the thermal performance when the velocity grows. The absence of the vortical system in one row model affects its performance at higher velocities.

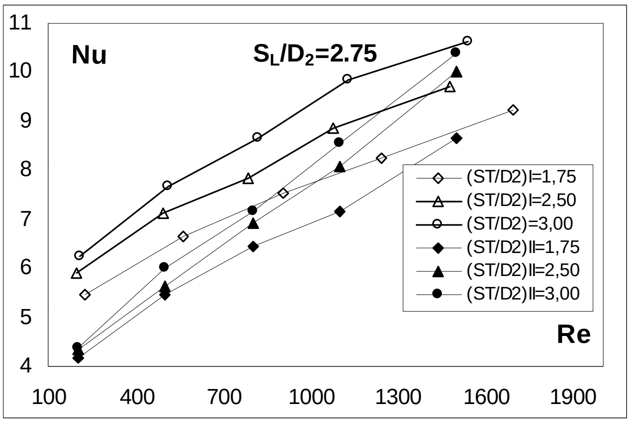


Figure 9. Experimental Nusselt number as a function of Reynolds number for one and two rows models.

Using approximately six hundred experimental values obtained in this work, correlations for friction factor and Colburn factor were determined. The correlations were obtained using the Mathcad software and are presented below:

$$j_{1row} = 1.327105\, Re^{-0.755}\left(S_T/D_2\right)^{0.333} \qquad f_{1row} = 0.66631\, Re^{-0.564}\left(S_T/D_2\right)^{0.606} \tag{12}$$

$$j_{2row} = 0.915761\, Re^{-0.61}\left(S_T/D_2\right)^{0.218}\left(S_L/D_2\right)^{-0.616} \quad f_{2row} = 4.4193\, Re^{-0.661}\left(S_T/D_2\right)^{0.736}\left(S_L/D_2\right)^{-1.563} \tag{13}$$

The statistical parameters for these correlations are presented in the table 6, where R^2 is the correlation coefficient according to Wadsworth (1989).

Table 6. Statistical result of correlations equations

Parameter correlated	R^2
$j_{1\ row}$	0.98
$j_{2\ row}$	0.90
$f_{1\ row}$	0.92
$f_{2\ row}$	0.80

6. CONCLUSIONS.

The thermal and friction characterization of thirty six models (six with one row) of compact heat exchanger with finned elliptical tubes was carried out. The effects resulting from variations of the transversal and longitudinal pitches in a wide range were analyzed.

The Nusselt number was found to increase when the transversal pitch is augmented or the longitudinal pitch is diminished and the causes of this behavior were analyzed. The horseshoe vortex system generated in front of the tubes and the developing boundary layer were the mechanisms responsible for the heat transfer enhancement. A novel technique for pressure measurement was developed and certified during this work. A group of simple empirical correlations were obtained considering the influence of the Reynolds numbers and the longitudinal and transversal pitches. These correlations for compact heat exchangers with one and two rows of elliptical tubes, considering smooth fin and isothermal condition, were never presented before.

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