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Numerical-experimental evaluation of the performance of fined systems: Study of new geometric arrangements for piniform fins

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Abstract. *A hybrid numerical-experimental approach is adopted to obtain the thermal performance of a piniform finned heat exchanger. The methodology used aims to determine the average convective heat transfer coefficient and overall finned surface efficiency. For this, a combination of numerical simulations with wind tunnel experimental data was applied in the finned arrangement. Two arrangements were analyzed, one with fins arranged transversely (In line) to the heaters and the other where the fins have an inclination (Zig-zag) in order to create turbulence and increase the heat exchange transfer. The main results indicate that the arrangement "Zig-zag" achieved a better performance than the "In line" arrangement for a Reynolds number higher than* $6.0\ 10^3$*, for Reynolds lower than* $5.5\ 10^3$*, it's not possible to compare the result between the two arrangement, due to the results being in the experimental uncertainty.*

Keywords: *Heat transfer, Piniform Fins, Forced Convection, Alternative Geometric Arrangement, Numerical Simulation*

1. INTRODUCTION

Heat exchange devices are widely used in the most varied applications. One of the most important components are the heat exchangers. This thermal device is present at refrigerators, air condition systems, cars, computers, among others. Due to the great need for thermal exchange in equipment, studies have been development for configuration improvement and heat transfer optimization, with purpose to size reduction and improvement efficiency of the exchangers. Understanding the geometric parameters of heat exchangers are essential for improvement development. Kim and Kim (2005) developed a study about finned tubes heat exchangers, studying the fins spacing, number of rows and tubes alignment. The study showed that the heat transfer coefficient was 10% higher at the alternating tube matriz in comparison with the aligned tube matriz.

Some authors have been development numerical methodology, experimental and hybrids, in this study the authors use a hybrid methodology for study the influence of geometric arrangements on heat exchangers. A common challenge at researches of this nature is the determination of the boundary conditions that govern the problem's physics at numerical solutions. It's common to perform a wind tunnel test to obtain these parameters, Zdanski *et al.* (2014) developed a numerical-experimental study for evaluate factors that influence the heat transfer at tapered tubular matriz. The parameters were obtained through tests realized at a suction wind tunnel and numerical simulation using commercial software Ansys CFX®. The researches checked that the taper of the tubular matriz influenced the Reynolds number and the system heat transfer. In the same way, Bender *et al.* (2016) and Bender *et al.* (2018) studied a new trapezoidal geometric arrangement of tubes immersed at a turbulent flow through a numerical-experimental methodology. The results obtained show a better performance with the trapezoidal geometric for the Nusselt number, however a great increase at the pressure drop was checked.

Chen *et al.* (2013) investigated the heat transfer in the plates fin for some fin spacing, an inverse method with experimental data of temperature was used to determinate the heat transfer coefficient accuracy. In this study the researches used numerical simulation with commercial software for obtain heat exchange and fluid flow characteristic. The inverse method with the experimental study proved useful for new heat exchanger studies. On the same way, González *et al.* (2019) developed a hybrid methodology to determinate the average heat transfer coefficient and the efficiency of the exchanger, the boundary conditions were obtained experimentally and used as input at the numerical simulation using commercial software. Singh *et al.* (2017) analyzed alternative profiles for the fins, comparing the default rectangular geometry with two other alternatives profiles, which were named synodal profile and polynomial profile. The alternatives profiles resulted in a increase of heat transfer per fin weight when compared to the default rectangular profile.

The present study aims to evaluate the thermal performance of two geometric arrangement of piniform fins. The research was based on the methodology developed by González *et al.* (2019) for determination of the average heat transfer coefficient of the exchanger and the fin efficiency through wind tunnel tests and numerical simulation using commercial software Ansys CFX®.

2. EXPERIMENTAL SETUP AND PROCEDURE

In this section describes the materials used in the experiment and the methodology adopted to determine the main thermal parameters in the evaluation of the geometric arrangements of the heat exchanger.

2.1 Experimental Procedure

The schematic diagram of the installation used in the tests is presented in Fig. 1, where there is an open circuit wind tunnel of a suction type (range of velocities 4.0–15.0 m/s) with 1:6 contraction ratio and test section of 250 mm × 250 mm. The experiment bench is the same used in the work of Zdanski *et al.* (2014), Bender *et al.* (2016), Bender *et al.* (2018) and González *et al.* (2019). The turbulence intensity of the incoming flow in the empty test section is less than 1% and the velocity control of the flow in the wind tunnel is through the frequency inverter that controls the electric motor of the fan.

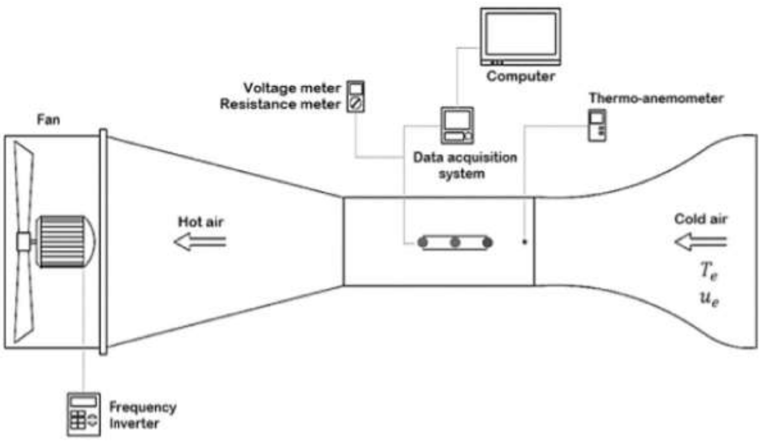


Figure 1: Schematic diagram of the heat exchanger in the test section

The experimental setup used in the present work is presented is shown in Fig. 2. The heat exchanger consists of a tubular matriz aligned with three heaters and piniform fins were welded to these heaters to increase the heat exchange area for two different arrangements. In the Figure 2a, it is possible to observe the mounting of arrangement 1 "In line", while the Fig. 2b it is observed the arrangement 2 "Zig-zag". The main geometric parameters of the arrangements are shown in Table 1.

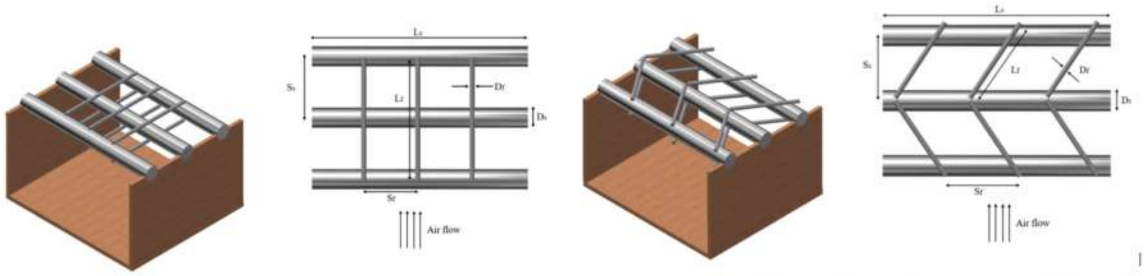
In stage of the built of the experimental apparatus, the fins were welded on the heaters and the addition material used in this joint is the same material as the other components of the system, therefore, it was considered that the welded joint has no contact resistance. To support the apparatus in the test section of the wind tunnel, a support was built, the material used in the manufacture is composed of fiberglass to reduce the heat transfer between heaters and support. The resistance (R) and voltage (V) values were measured directly, using a digital multimeter. The measured resistances are: 281, 282 and 282 Ω. The calculation that determines the total heat transferred is obtained by applying Ohm's law given by Eq. 1, where $\phi = 0.98$.

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

In addition, to obtain the average surface temperature ($\overline{T}_s$) Eq. 2 was used, where the temperatures (T_i) are the temperatures measured by type k thermocouples, so three resistors (R) were used, being the surface areas: heaters ($A_s = \pi D_s L_s$) and fins ($A_f = \pi D_f L_f$).

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

After preparing the experimental apparatus, measurements were started in a wind tunnel. The thermocouples welded in the heaters were connected to the data acquisition system (Field Data Logger) to obtain the temperatures (T_i). At the tunnel entrance, a hot wire anemometer was inserted to obtain the flow velocity and temperature. After the measurements of the described parameters, the boundary conditions measured experimentally could be used in the numerical methodology.



A) Experimental arrangement "In line"

B)Experimental arrangement "Zig-zag"

Figure 2: Organization of experimental arrangements: A) In line and B) Zigzag

Table 1: Geometric parameters of the Arrangements 1 e 2

Parameters	In line	Zig-zag
N_f (Número de aletas)	6	12
N_s (Número de aquecedores)	3	3
L_f [mm]	96	66,85
L_s [mm]	170	170
D_f [mm]	3,8	3,8
D_s [mm]	16	16
S_f [mm]	42,5	42,5
S_s [mm]	48	48
A_s [mm^2]/N_s	8550	8550
A_f [mm^2]/N_f	575	800
A_t [mm^2]	32550	35250

2.2 Experimental Numeric Hybrid Methodoly

For the numerical analyzes, the commercial program Ansys CFX®was used. The simplifications adopted for the solution of the numerical model are: permanent regime, uniform convection in the fins, all fins dissipate the same amount of heat, so it will be necessary to simulate only one fin. In addition, contact resistances were neglected as previously mentioned, incompressible fluid and fluid properties were obtained at the film temperature.

The methodology used in this work to determine the average heat transfer coefficient and the overall efficiency of the heat exchanger was proposed by González *et al.* (2019). The method consists of:

i. Dimensional measurements: The surface area of the heaters, A_s, fins surface area, A_f, and the total heat transfer area of the surface, A_{tot}, are determined from measurements of linear dimensions so that

$$A_{tot} = N_f A_f + N_s A_s \tag{3}$$

Where N_f is the number of fins, N_is the number of heaters, 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: Obtaining the main parameters for determining the global heat transfer coefficient, where the variables involved are: inlet velocity u_{in}, inlet temperature (T_{in}), mains voltage used during the experiment (V_i), electrical resistance of heaters (R_i) and the average surface temperature of the heaters, $\overline{T}_s$

iii. The evaluation of the total heat transfer rate, Q_m, using equation 1, $Q_m = \cos(\phi) \sum_{i=1}^{N_s} \frac{V^2}{R_i}$

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

$$U = \overline{h_o}\overline{\eta_o} = \frac{Q_m}{A_{tot}(\overline{T}_s - T_{in})} \tag{4}$$

Where $\overline{h_0}$ is the average heat transfer coefficient and $\overline{\eta_0}$ is the overall surface efficiency.

v. Iterative calculation of the average convective heat transfer coefficient, $\overline{h_0}$. 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 $\Psi_h > TOL_h$ DO
   Calculate the heat transferred over the surface of all cylindrical heaters, $Q_s = \overline{h}_0 A_s(\overline{T_s} - T_{in})$;
2. Heat transferred by individual fins, Q_f, using the average surface temperature of each heater (T1, T2, T3), and the average convective heat coefficient for convective heat, $\overline{h_0}$. For this stage of the calculation, computer simulation is used, where a three-dimensional model was adopted in the simulations, thus, two regions were considered in the fin to simulate the case as shown in Fig. 3. The heat dissipated in the fin is calculated according to Eq. 5, being composed of the heat dissipated in region 1 with fin region 2

$$Q_f = Q1 + Q2 \tag{5}$$

3. Calculate the total heat exchanged by the fins and surface of the heaters, $Q_{tot} = N_f Q_f + N_s Q_s$;
4. The maximum ideal heat exchanged, $Q_{max} = \overline{h}_0 A_{tot}(\overline{T_s} - T_{in})$;
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, $\Psi_h = |\overline{h}_0^{new} - \overline{h}_0^{old}|$
   END WHILE
8. Final output $\overline{h}_0, \overline{\eta}_0$ and Q_f.

To exemplify the iterative calculation described in the step-by-step of item (iv), a flowchart was created which can be seen in Fig. 4.

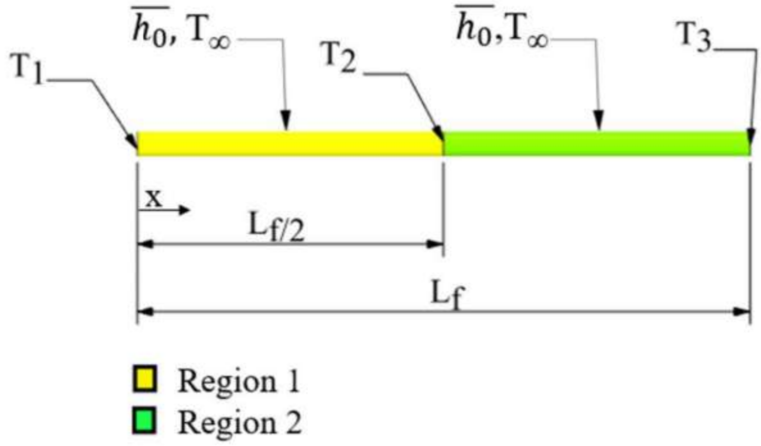


Figure 3: Fin heat exchange region

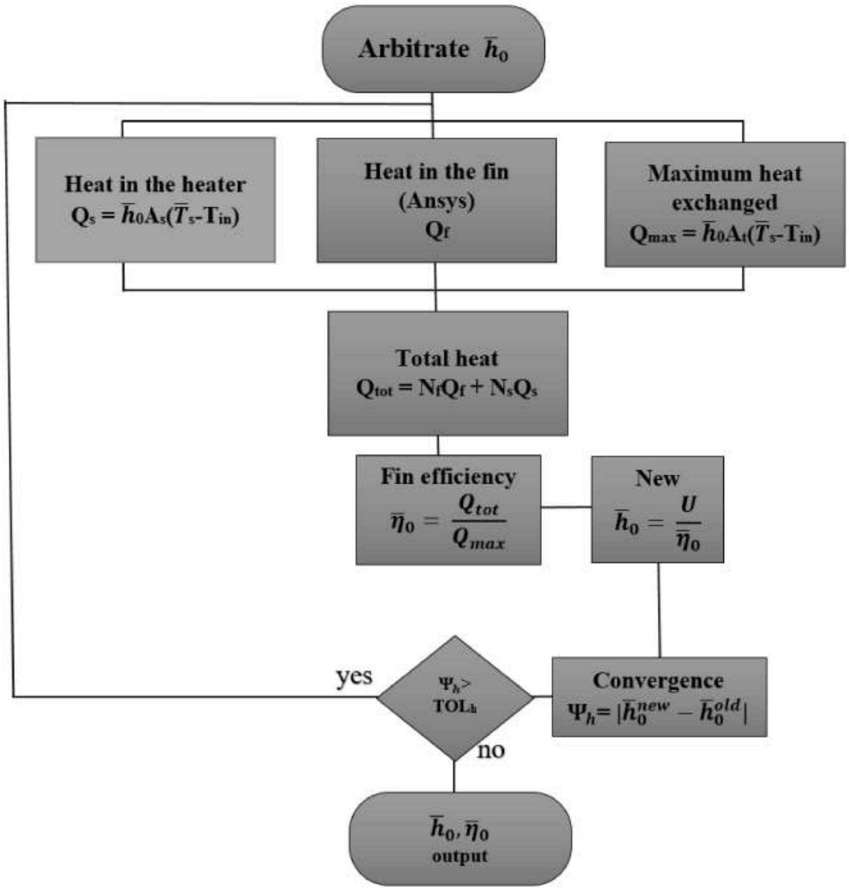


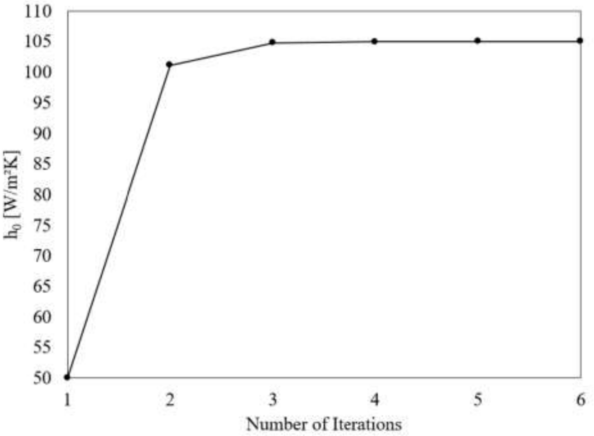
Figure 4: Iteration routine for calculating the average convective coefficient (h_0)

3. RESULTS AND DISCUSSIONS

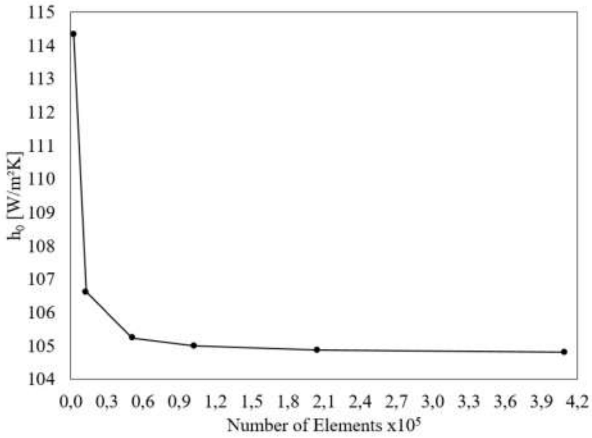
In this section, the results of the comparison between the two geometric arrangements of the piniform fins will be presented, whose analyzis will be based on the numerical-experimental evaluation describe in section 2. Thus, the present study was divided into two (i) Validation of the numerical model, and (ii) Analyzis of the thermal behavior of the listed surfaces, for both the proposed arrangements.

3.1 Validation of the numerical simulation

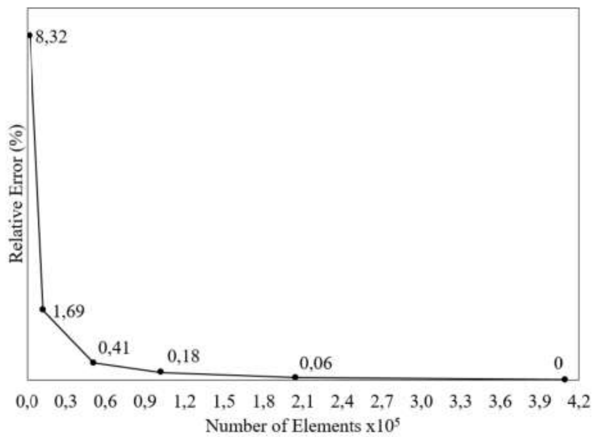
To validate the iterative process, the following were analyzed: convergence of the method and the variation of the calculation of the coefficient of heat exchange with the iterations, the results can be seen in Fig. 5a and 5b. The Figure 5a shows a rapid convergence for the calculation of the heat exchange coefficient, the convergence criterion (Ψ_h) adopted is 0,01% in relation to the convective coefficient calculated in the previous iteration, according to the algorithm in section 2.2. In Figure 5b, the mesh dependency of the adopted model is observed, thus, it is verified that from 6x10^4 number of elements, the result of the heat exchange coefficient remains constant. Figure 5c, it is shown that the relative error for the mesh adopted was approximately 0,18%. The topology of the mesh in the fin is shown in Fig. 6, where you can see that the mesh adopted was of the structured type.



(a) Convergence analysis



(b) Mesh refining analysis



(c) Relative error of the elements

Figure 5: Convergence and mesh analysis

Figure 6: Mesh topology

In addition to the previous analyzes, a check of the global energy balance was realized, to see that energy supplied to the system in the experiment (Q_m), is equals the energy obtained numerically, separating in heat transfer rate by the fins (Q_s) and the heaters (Q_f). The results are compared in Table 2, it turns out that the values of Q_m and Q_f were practically the same in six simulated cases, with the global energy balance being respected in the adopted procedure. The Figure 7 represents a ratio of heat transfer rate through the fin (Q_f) to the total heat transfer rate (Q_m)as a function of the Reynolds number for the two proposed arrangements. It was noted in Figure 7 that with the increase in the number of Reynolds, there was a reduction in the heat transfer rate dissipated by the fin in comparison compared to heat dissipated by the heaters, this occurs for both arrangements.It is observed with the increase of Reynolds, there is a greater distance between the behavior of the "Zig-zag" arrangement and the "In line" (the reason between the heat dissipated in the fin and the total heat dissipated in the heat exchanger differs approximately 9% between the two arrangements for the largest Reynolds number tested). In addition to energy conservation being seen as a complementary solution to a physical problem, the residual used in the commercial software Ansys CFX®were considered convergent with RMS 10^{-6}.

Table 2: Comparison between heats calculated numerically and theoretically

Flow speed range	Parameter	In line	Zig-zag
9,0 m/s - 9,3 m/s	$N_f q_f$ [W]	57,81	51,78
	$N_s q_s$ [W]	443,20	443,10
	q_m [W]	504,00	494,88
	q_t [W]	504,00	494,880
7,5 m/s - 7,7 m/s	$N_f q_f$ [W]	60,80	59,12
	$N_s q_s$ [W]	443,19	435,75
	q_m [W]	504,00	494,88
	q_t [W]	503,99	494,87
5,6 m/s - 6,1 m/s	$N_f q_f$ [W]	65,90	64,88
	$N_s q_s$ [W]	438,09	439,13
	q_m [W]	504,00	504,00
	q_t [W]	503,98	504,01

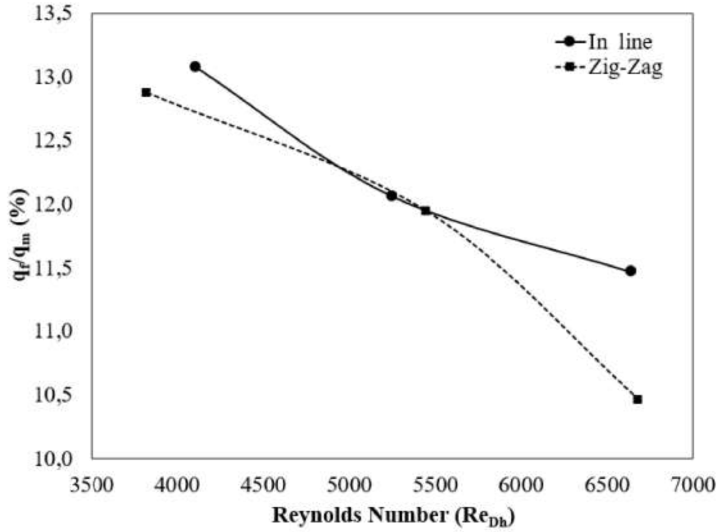
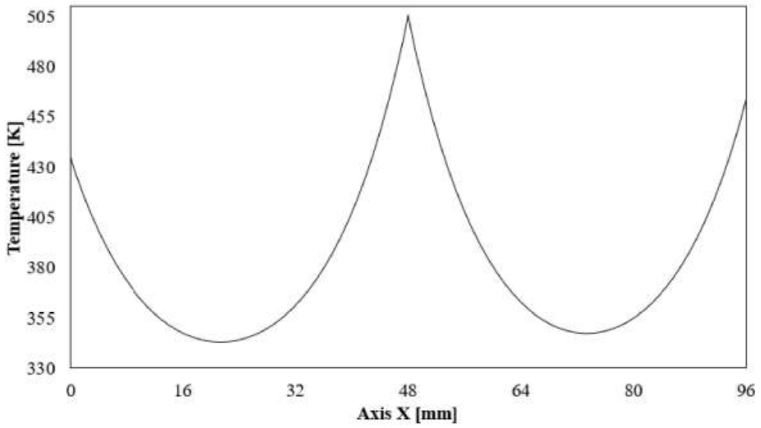
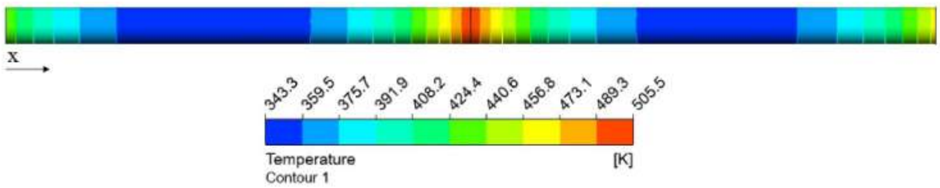


Figure 7: Ratio of heat transfer through the fin

Through the simulations performed, it was possible to generate a temperature profile along the fin as shown in Figure 8. Figure 8a represents the temperature curve of the center line of the fin in relation to the axial coordinate, the two points of inflection shown in graph represent the points where there is no heat flow in the axial direction of the fins, there is an inflection point in each simulation region (regions 1 and 2 shown in Figure 3). The distribution of the temperature can be seen in Fig. 8b, it turns out that the greatest heat dissipation occurs in the central region of the fin, because this region is in contact with the higher temperature heater. Lower temperatures are found in regions further away from the heaters, in these regions there is less heat dissipation from the fins.



(a) Temperature profile graph



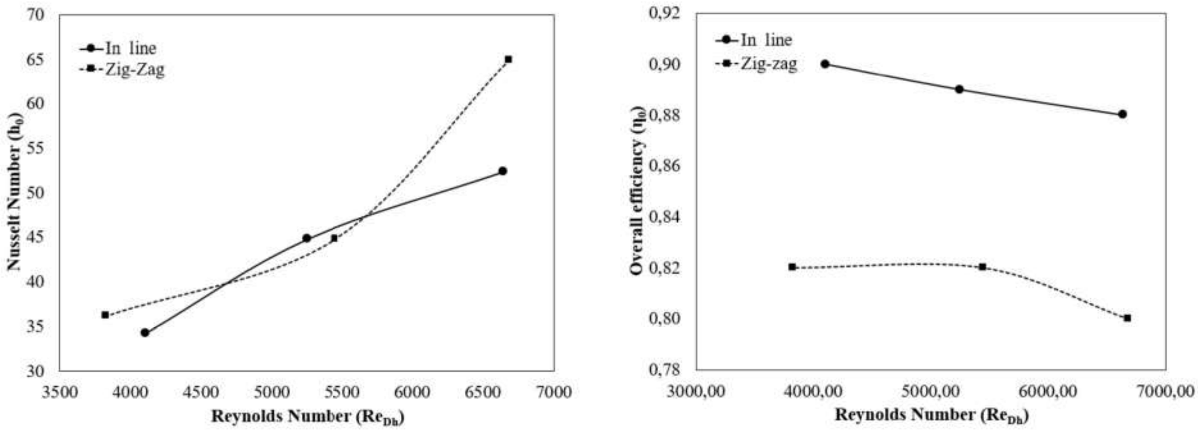
(b) Temperature distribution in the fin

Figure 8: Fin temperature profile

3.2 Analysis and thermal behavior of the fins

This section presents the results of the thermal behavior of piniform fin arrangements with their respective Reynolds numbers. Fig. 9a shows the average Nusselt number (Nu_{Dh}) calculated by the convective coefficient obtained by the hybrid methodology as a function of the Reynolds number, the uncertainty of the Nusselt number in the experiment is the around 4% according to Zdanski *et al.* (2015) , it is observed that in the Reynolds range less than 6000, there was neither improves or worsens, due to the difference between the two models being within the uncertainty range of the experiment, for Reynolds values greater than 6000, the Zig-zag arrangement has a better thermal performance (for the largest Reynolds number tested there is an increase of about 20% of the global Nusselt among the analyzed arrangements. In Figure 7b are presented the results of the overall efficiency of the heat exchanger in function of the Reynolds number. Analyzing the heat transfer rate of the "Zig-Zag" Arrangement, would lead us to believe that the efficiency would be superior to that of the "In-line" Arrangement, due to its better thermal behavior for Reynolds higher than 6000, however, this does not occur, given the fact of the finned area in the "Zig-zag" Arrangement to be larger than the "In line", this implies a reduction in the overall efficiency because with the increase in the area surface there is also a reduction in global temperature that is directly proportional to the calculation of the efficiency thermal.

In the Figure 10, are presented the values of the Nusselt number based on the global heat transfer coefficient. Is noted from the graph shown in Figure 10 that in the Reynolds range of 5000 the “In-line” arrangement has a better performance, this result meets the justification that the “Zig-zag” arrangement has a larger area finned when compared to the “In-line” arrangement. For the largest Reynolds number tested, it is observed that despite the “Zig-zag” arrangement has a lower overall surface efficiency due to the increased heat exchange area, its thermal performance was superior to the “In-line” arrangement. The graph can be obtained as a result of multiplying between the parameters of Fig. 9a and 9b according to Eq. 5.



(a) Average Nusselt number as function of Reynolds number (b) Overall fin surface efficiency as function of Reynolds number

Figure 9: Average Nusselt number and Overall fin surface efficiency

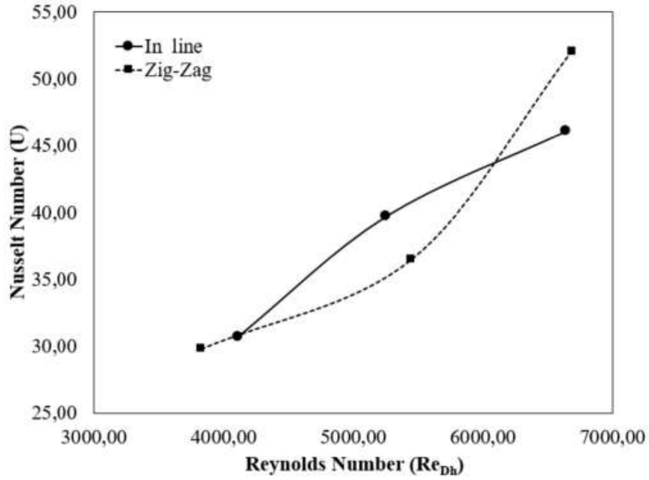


Figure 10: Average Nusselt number of the global heat transfer coefficient

4. CONCLUDING REMARKS

In the present study, the proposed hybrid evaluation for the two different geometric arrangement, obtained expected results, a small increase at the heat transfer coefficient for the Zig-zag arrangement for Reynolds number higher than 6000 when compared to the In line arrangement. For Reynolds number lower than 5500, it was found that the two arrangements when compared didn't showed improvement neither worsening in thermal performance, because the results are in the experimental uncertainty region. The hypothesis for the improvement at the thermal performance for the Zig-zag arrangement for large Reynolds number, can be explained due to the fins arrangement provide a higher turbulence intensity to the flow, turbulent flow has a greater mixture of layers and consequently a greater heat transfer coefficient. The validation for the method used to calculate the average heat transfer coefficient was concluded with the interaction's evaluation and the evaluation of the method stability.

Analyzing only the fins region, Figure 7 show that the ratio between the heat dissipated in the fins and heaters decreases with the increases on the flow speed. In the numerical simulation it was observed a higher heat dissipation at the central region of the fin, the Figure 8 present the fin temperature profile with two inflection points, fins regions away from heater present lower temperatures, due to this, arrangement with longer fins has more regions with lower heat transfer resulting in a worse thermal performance of the heat exchanger.

The authors of this work checked that new geometric arrangements con result in a thermal performance improvement, even though the obtained results are close it was observed that for higher Reynolds numbers the study show a growing distance on the results. The authors propose that new studies should be carried out with the proposed arrangement for higher Reynolds values, increasing the number of fins on the heat exchanger, thus increasing the heat transfer on the fins region. However, increasing the number of fins will decrease the global surface efficiency and could decrease the thermal performance of the exchanger. In addition, new studies with CFD simulation may be interesting to study the flow topology and better understand the phenomenon present at the thermal system.

5. ACKNOWLEDGEMENTS

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