COMPARISON OF PERFORMANCE AND APPLICATION OF HEAT EXCHANGERS OF ELLIPTICAL FINNED PIPES, GLOBALLY OPTIMIZED, USED IN CLIMATIZATION SYSTEMS

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Abstract. *This paper has as objective to prove experimentally the relevance and technical superiority of using pipes with elliptical profile in heat exchangers, mainly with regard to the ones of circular profile. Based in preliminary results of simulations and experimental analyzes in laboratory, a heat exchanger prototype was confectioned with geometric characteristics optimized in previous works. It was then used as an evaporator in a domestic Split type air conditioned system; so its performance could be compared in different work situations, as the system was using the conventional evaporator. To set the parameters of the comparisons, both evaporators had occupied the same cold unit. They had exactly the same external format, were composed by the same number of pipes, followed the same cooling fluid circulation way and represented the only component which was replaced in all the experimental apparatus. To minimize the technological difference between the fabrication process employed in both situations (handmade prototype in one case and baseline in the other) a third heat exchanger was made. It had the same geometrical characteristics, but employing the same fabrication characteristics. The previous analysis (numerical simulations and laboratory assays) show that the maximum thermal efficiency and the minimum external fluid pressure loss happen when the optimized pipes disposal and fin density are applied. The manufacture of the elliptical archetype was made with the plastic conformation of commercially available circular copper pipes, using devices developed with this purpose. It was used aluminum sheets with thickness of 0,3mm cut in the fin format, with the optimized geometric conditions. These conditions consist in elliptical pipes external profile (eccentricity), in the distance between these pipes rows and in the fin density. All these variables had been previously parameterized and represented for dimensionless numbers. The analysis of the heat transference of both assemblies was made through the verification of the efficiency of a residential Split HVAC system using each one of the evaporators. The main comparative element was the system COP (Coefficient of Performance). The experiments happened by the system operation, installed as indicated by the manufacturer, with the cold unit located inside an acclimatized chamber. It was possible to change the temperature inside the chamber, and the air speed that passed through it as well. The interpretation of the thermodynamic air conditions before and after the evaporator, based on the mass outflow and on the electric energy consumption of the compressor, had complemented the system efficiency analyzes. All the tests results extrapolations comparing the three heat exchangers had shown significant profits of energy efficiency by the order of 10 - 15%, for the bigger air flow regime in question. Considering the same occupied space by the heat exchanger (external dimensions), that would be better distributed for the elliptical tubes, it could make possible the installation of more tubes in the same available area. It is possible to preview still better results for optimized projects using these optimized geometric characteristics.*

Keywords: *air conditioner, heat exchanger, refrigeration capacity, coefficient of performance, elliptical tubes*

1. INTRODUCTION

The industrial processes optimization for maximum exploitation of the available energy (exergy) has been a very worded scientific research line in recent times.

This scientific work has the objective of comparing the performance of heat exchangers in a conventional project with the globally optimized elliptical pipes heat exchangers project, with similar dimensional characteristics. It is a natural sequence of other works carried by the Thermal Sciences Research Group from the Mechanical Engineering Department of the Universidade Federal do Paraná (UFPR). Some essays had been carried through inside this context, aiming at the thermal transference optimization in not finned circular and elliptical pipes heat exchangers, in two dimensions and laminar flow (Matos, Vargas, et al., 2001), experimental finned pipes (Matos, Vargas, et al., 2004A), numerical and experimental laminar flow of finned pipes (Matos, Vargas, et al., 2004B), and experimental in turbulent flow (Mainardes, Matos and Vargas, 2007). In addition, the minimization of the pressure loss in arrangements of circular and elliptical pipes was studied the (Mainardes, 2007), where had been concluded that the excellent architecture of these arrangements are approximately the same one for the two objectives, i.e., maximum heat transference and minimum pressure loss.

In a special way, the scientific works presented by Matos (2003) and Mainardes (2007) can be considered theoretical base for the present work. In relation to these studies, this essay tries to complement them through an experimental / practical implementation of the results previously obtained. Its focus is to get the comparison of the heat transference process of an archetype evaporator confectioned based on the optimums parameters (geometric and dimensional) obtained in both previews analyses and an commercially used (conventional) equipment operating in a Split type residential HVAC system.

In order to have a complete demonstration through a realistic obtained improvement performance quantification with the optimized configuration in comparison with conventional HVAC system project, tests on the experimental air conditioned unit constructed for the experimental validation had been carried. The tests used the conventional evaporator and the globally optimized archetype, with the objective of comparing the system Coefficient of Performance endowed with each one of the heat exchangers.

Obviously the handmade methods of production results in technically poorer final products than those products proceeding from large scale production process, due to the experience and improved manufacture methods obtained throughout years. To minimize the differences between the heat exchanger to be analyzed and the baseline, one third exchanger was built. It had the same dimensions and operational/geometric characteristics of the baseline, but used handmade technology. A comparison between these three exchanger gives more interesting data for the desired analyzes.

The elliptical heat exchanger was made using the globally optimized geometric characteristics from the previous scientific works. It consists of commercially available set copper pipes, duly conformed in the elliptical format, jointed in its extremities in order to form two circuits for fluid flow, just as the original exchanger (baseline). With the objective of simulate the fins, 0,3mm thickness aluminum plates were cut in the desired format, enabling that these plates to support all the pipes in the adjusted position. That allows us to make sure that there is contact between them and the pipes and guarantee the same external dimension (occupied volume) in both similar to baseline (Fig. 1). Unions manufactured exclusively for this experiment had been welded in the pipes extremities, by tin brazing process, getting sure the optimized distance between the pipe rows.

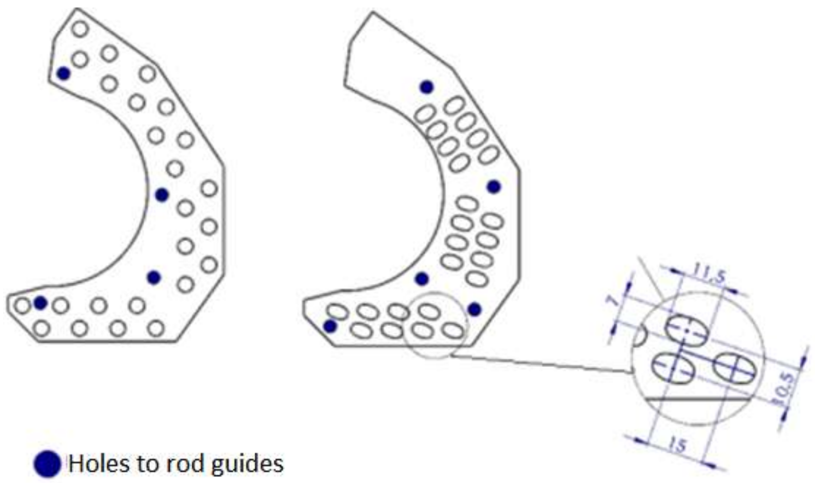

Figure 1. Representation of both exchangers - baseline `left and elliptical to the right

The analyzed equipment is a climatization split set, only cold homeuse type. All the equipment is manufactured by Springer®, of the Maxiflex™ series, with 12,000 btu/h of nominal cooling capacity. Composed by a evaporator unit, model 42MCC012515LS and a condensator unit, model 38MCC012515MS. This set was projected to operate with refrigerant fluid R22, and 220V tension. According to manufacturer datasheet, its average consumption is of 1270 w, and its COP is of 2,77.

The evaporator used in the analyzed equipment is a crossed draining tubular finned type coil, with tubing in copper material, of diameter 6,35mm [1/4"] and about 25 fins per inch, its fins are in aluminum thickness 0,1 mm, conformed in order to involve the copper pipes, being joined by none welding process.

As it can be inquired in Fig. 2 the equipment in question consist in 20 pipes with 600 mm length, posicioned transversally to the air flow, jointed in its extremity in order to form two independent cooling fluid circulation circuits, each one of them represented with distinct colors in this figure.

It is a radial evaporator format, minimizing the occupied space by the set. It uses siroco type fan.

Figure 2. Baseline evaporator

2. BIBLIOGRAPHICAL REVISION

Recently, Matos (2003) searched the heat exchangers optimization in circular and elliptical finned pipes in laminar flow, showing that heat exchanger with finned elliptical pipes presents greater global taxes of heat transference than the finned circular pipes heat exchanging. This work was complemented by Mainardes (2007); that carried through experimental analysis and numerical simulation, but operating under the turbulent flow confirming the results already gotten by Matos (2003); beyond adding optimized resulted with regard to the pressure loss on the part of the external fluid.

Finned elliptical pipes heat exchangers had been studied in the past, showing a relative increase in the heat transference in the elliptical pipes arrangements in comparison with the circular pipes arrangements. A relative reduction in the pressure loss up to 30% was also observed. Rocha et al. (1997) had developed experimentally an hybrid mathematical model for finned circular and elliptical pipes arrangements based in the energy conservation and the heat transference coefficients obtained through the naphthalene subliming technique and by heat and mass transference analogy. Saboya and Sparrow (1976); and Saboya and Saboya, (2001) had numerically obtained the temperature distribution and the fin efficiency for 1 and 2 rows of elliptical pipes in finned heat exchangers. The fin efficiency results had been compared with the Rosnan et al (1984) results for finned circular pipes heat exchangers, where a relative profit in the fin efficiency up to 18% for heat transference was observed with the elliptical pipes arrangement.

Bordalo and Saboya, (1999) more recently had presented measurements for the pressure loss comparing circular and elliptical finned heat exchanger, with 1, 2 and 3 rows of pipes. Reductions of up to 30% of the coefficient of pressure loss (coefficient of pressure fall per unit of row due only the pipe presence) had been observed, for the elliptical configuration. The comparison was carried through between circular and elliptical pipes arrangements with the same blockage area for free flow, and $200 \le \mathrm{Re}_\delta \le 2000$ that has covered the band of air speed of interest for applications in climatization (Matos, 2003).

Matos et al. (2001), carry through heat transference analysis in non finned circular and elliptical pipes heat exchangers, in two dimensions. The method of finite elements was used to discretize the governing equations of the air flow and the heat transference, and a bi-dimensional, iso-paramethric, quadrilateral element with linear interpolation functions, was implemented and added to FEAP code. The numerical results for the equilateral triangular arrangement configuration, obtained with the new element were validated by direct comparison with the previously published experimental results for circular pipes heat exchangers. Results of numerical geometry optimization show a relative profit in the heat transference up to 13% in the optimum elliptical arrangement, in comparison with the optimum circular arrangement. The previously observed heat transference profit combined with the reduction in pressure loss up to 30% shows that elliptical pipes arrangements has potential for a considerably better performance than conventional circular arrangements.

Other studies also come being carried through under the objective of others heat exchangers conceptions use, as can be observed in the Qi (2009) and Pettersen (1998) scientific works, which focus on the use of exchangers based in micro channels to get greater energy income.

The use of elliptical tubing in heat exchangers is an interesting subject studied in recent years, the possibility of use of bigger pipes density, in a same occupied total volume, and the already observed optimization between external fluid pressure loss and the heat transference show a great advantage of this technology comparing to the traditional one.

The previous works take to the main geometric characteristics arrangement optimization, based on the Matos (2003) and Mainardes (2007) results, the optimum architecture are given as dimensionless properties that can be presented be this form:

$$(S/2b,\ e,\ \phi_f)ot \cong (0{,}5;\ 0{,}6;\ 0{,}094^{1}).$$

[1] Equivalent to 2,89mm – for a sheet thickness of 0,3mm e ϕ_f = 0,094; e.i., about to 8 fins per inch.

where:

S/2b	= gap between rows of pipes - relation between the gap between pipes rows ("S)") and the secondary ellipse axle ("b")	[ ]
e	= pipe eccentricity, - relation between secondary and main ellipse axles	[ ]
$\emptyset_f$	= dimensionless fin density, defined by the equation (1) -	[ ]

for:

$$\emptyset_f = \frac{t_f}{t_f + \delta} \tag{1}$$

where:

t_f	= fins thickness	[m]
δ	= distance between fins	[m]

3. METHODOLOGY

In order to determinate the system cooling capacity looking for evaluate its performance, the experimental procedure begins measuring the cold air flow velocity[2] "V" into the evaporator using a thermo-anemometer model AVT55 (manufactured by Alnor Products). Then its thermodynamics properties were calculated in the psychometric state. The dry bulb temperature and relative humidity were measured using thermistors (PTC type) and the humidity sensors transducer (model HU1015NA) were installed in the cooling box (evaporator)air inlet and outlet. To facilitate the reading and to obtain homogeneous values of the variables, inflating ducts were fabricated as shown in Fig. 3, enabling the transition from the evaporator rectangular area (80 x 600 cm) to a square area (220 x 220 mm), adequate to the data acquisition.

Figure 3. Instrumented evaporator Air duct

The cooling capacity was then given by the Eq. 2:

$$\dot{Q}_e = \frac{\dot{V}}{v} * (h_{ins} - h_{ret}) \tag{2}$$

where:

$\dot{V}$	= volumetric air flow	m^3/s
v	= especific air volume	m^3/kg
h_{ins}	= inlet air enthalpy – f(T_{ins}, $Ø_{ins}$)	[kJ/kg]
h_{ret}	= outlet air enthalpy - f(T_{ret}, $Ø_{ret}$)	[kJ/kg]

After the calculations presented above, was determined the system efficiency based on the 1st Thermodynamics Law. The system coefficient of performance (COP) can be represented by Eq 3:

$$COP = \frac{\dot{Q}_e}{P_{cp}} \tag{3}$$

where:

COP	= Coeficiente of Performance	[ ]
$\dot{Q}_e$	= Cooling capacity	[kJ/s] ou [kW]
P_{cp}	= Consumed power by de compressor	[kJ/s] ou [kW]

The power consumed by the compressor (energy parcel used to transform useful work - refrigerant compression) was obtained by the difference of the total electric power used to operate the equipment (and therefore the compressor)

[2] The air flow velocity was post corrected by the log-Tchebycheff rule (ISO Standard 3966)

in steady state. The power necessary to operate all other electrical components (blowers and electronic circuits) and all this values were obtained by a power meter ESB (model SAGA 2300 / 1388 RS 485), and better described by Eq 4:

$$P_{cp} = P_{el_{Total}} - P_{el_{Empty}} \tag{4}$$

onde:

P_{cp}	= Consumed power by compressor	[kJ/s] ou [kW]
$P_{el_{Total}}$	= Total consumed power by the system	[kW]
$P_{el_{Empty}}$	= Consumed power only by the accessories	[kW]

Both electrical powers consumed were obtained by averaging six readings direct from the digital power meter at intervals of 30 seconds, when the operation was in steady state.

3.1 Thermodynamic air properties

All the psychometric calculations were performed based on standard atmospheric pressure of Curitiba as ASHRAE (2001) - 90.88 kPa>. It was used a spreadsheet (Microsoft Excel ™), which had added a package of formulas and equations (Add -in) called Psyched iGet ™, supplied by kWEngineering. Using these methods, all calculations were performed numerically, including the uncertainties analysis.

4. EXPERIMENTAL APPARATUS

4.1 Prototype Exchangers

Composed by copper tubes properly conformed, jointed and arranged, two heat exchangers were manufactured exclusively for these tests, taking into account the previously optimized features. The first step of the fabrication was the elliptical conformation, performed by a specially developed device, shown in Fig. 4.

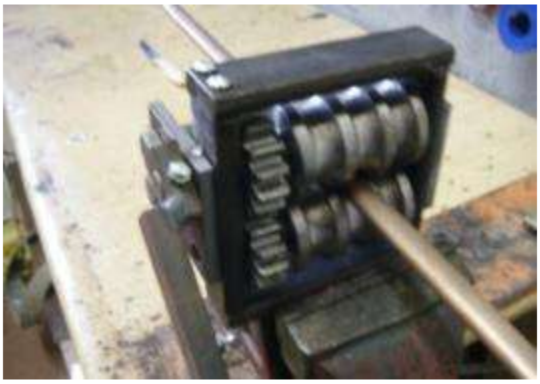

Figure 4. Elliptical conformation device

After that, the connections shown in Fig. 5 were made (as ones with the required dimensions are not available for purchase), using circular copper tubes with the radial conformation desired for each position in the exchanger.

Figure 5. Connections to be welded in pipe extremities

In the next step, the fins were installed and pipes extremities welded, as represented in Figs. 6 and 7, using laser cutting technology (fins manufacture) and brazing welding in the ends;

Figure 6 . Fins installed in prototype exchanger

Figure 7 . Welding process

After performed tightness assays in the exchanger, it was installed inside a test chamber, represented in Fig 8, composed by an inner chamber where the temperature is controlled by an electronic system. All other measuring instruments are available, in this case 12 thermistors were used, and all others instruments cited in tha methodology, which them had acquired its values by electronic system, based on the block diagram, using LabView ® software.

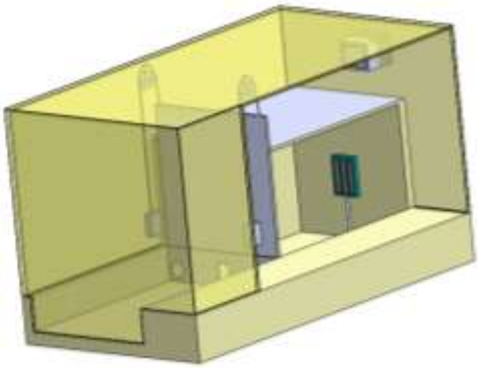

Figure 8. Testing Chamber of LMH / UFPr

4.2 Experimental Apparatus

As shown in Fig 9, the HVAC systems were installed one set once a time - condensator and evaporator unit (with the specific evaporator for the assay). All evaporators were placed in the same original evaporator unit, where the evaporation air condition was varied by increasing the chamber temperature. By this way the inlet air parameters repeatability were made sure (same pressure loss due to auxiliary accessories, and same available power).

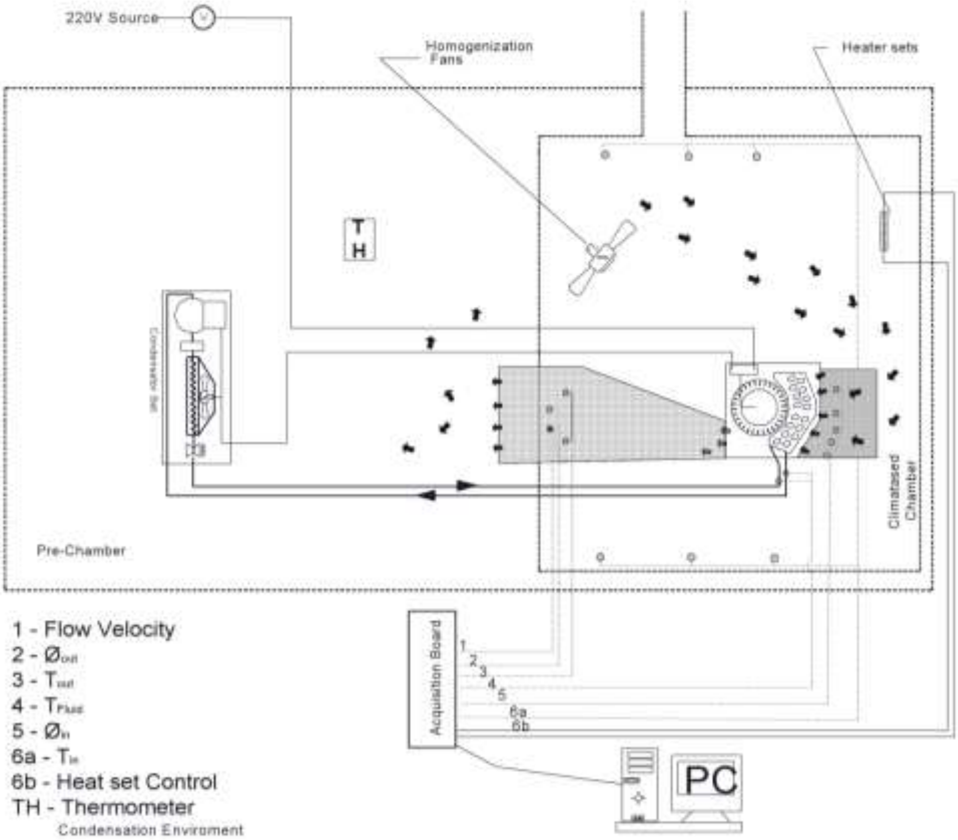

Figure 9. Schematic experimental apparatus

Besides, the condensator unit was placed in an environment (pre-chamber) where the temperature kept almost constant (at 25 ± 5°C). In Fig. 10 is represented the entire Split system installation in the acclimatized chamber and pre-chamber available in the laboratory.

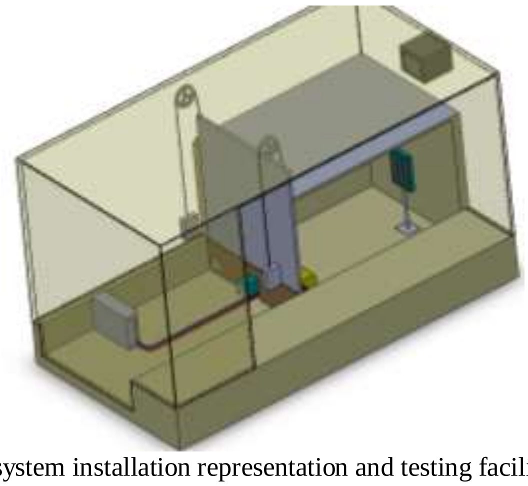

Figure 10. Split system installation representation and testing facilities for physical

Aiming to work with the equipment commonly found values, the following values for the variables were considered:

• Evaporation Air **-** entering the evaporator: Dry bulb temperature ranged from 20 to 30 and 40°C (operating within the manufacturer set limits)

• Air flow – the fan speed values used were pre-established by the manufacturer and designated as High and Medium speed, being informed by the manufacturer maximum original flow of 550m³/h. Which one was measured for each assay to consider the effects of pressure loss due each analyzed evaporator, as well as the set volumetric efficiency variation due to changes in air density. The low speed option available in the equipment can be operated only in exhaust function (when there is no cooling air blowing)

During each gas load, all procedures recommended by the manufacturing were employed (BRAZIL ‘s SPRINGER, 2008). Evacuation in lines and evaporating unit were held and the amount of refrigerant fluid (R22) suitable to result in an overheating of 6 ± 1°C was added, with the operating system on the most adverse conditions, i.e. in the lower chamber temperature (20°C) and air flow velocity devised in automatic. During this procedure manifold and a digital thermometer were used, being conducted under a qualified technician supervision / assistance.

When the gas charge was done, the system operation began on the desired air intake condition and exhaust fan speed. We waited for the stabilization of all parameters, i.e., steady state operation. It was followed by the data acquisition. The experiment was repeated for the next fan rotation value, and then repeated for the other evaporator air inlet temperatures. After the six experiments performed, the evaporator was changed and a new gas charge was performed to repeat the battery readings.

Each battery had considered six test measurements, according to NBR 15627-2, held for one hour in 10 minute intervals for each of the six measurements. A complete set of readings of all instruments was carried out at intervals of 30 seconds, in a total of 5 minutes, totaling 66 points of data acquisition.

5. RESULTS AND CONCLUSIONS

Performed according to the methodology described in item 3, the assay had its characteristics amount digitally measured and acquired for then by processed, resulting in specific tables and graphs. They show only the values for the desired analysis, as well as their applicable uncertainty.

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5.1 Power Consumption

Based on equation (4), the electric power consumed by the compressor was calculated assuming a power request in:

- Condensator set [3]: 88Watts, in both regimens
- Evaporator set [4]: 20 Watts, to medium speed fan
  31 Watts, to high speed fan

In a first analysis we can compare the values of consumed power by two principal sets (elliptic prototype and baseline), these results are present in Fig 11.

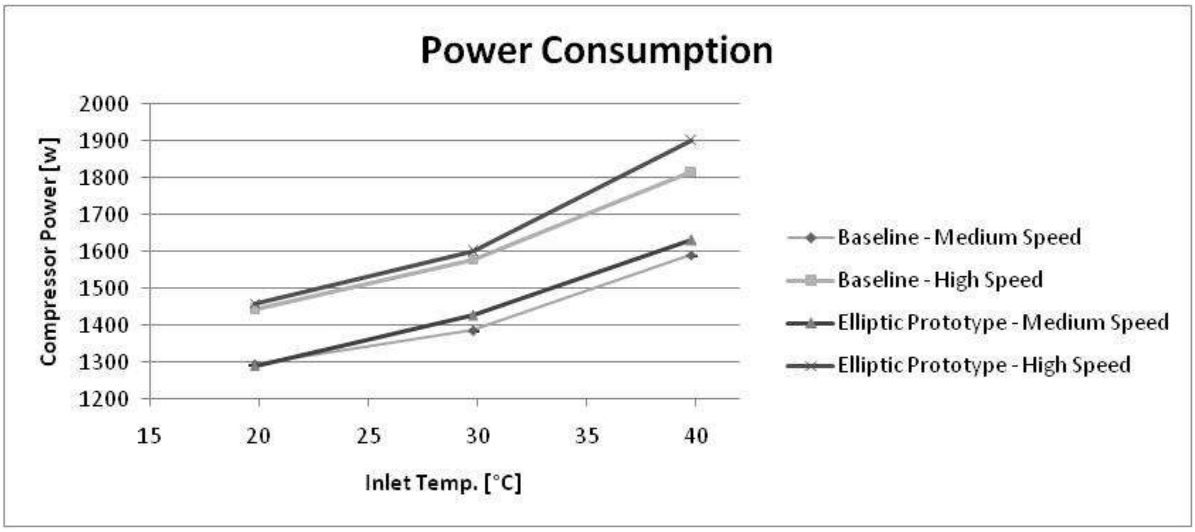

Figure 11. Power consumption comparison

As expected, there is a significant increase in compressor power consumption when we increase the chamber temperature testing. It happens because a greater pressure loss, caused by the fluid overheats resulting in phase transformation still along the heat changer tubing.

It was observed, in general, greater power consumption by the system that uses elliptic exchanger (an increase of about 10 to 15%). This fact can be explained by inherent internal condition of the manufacturing process of the whole prototype, which may have resulted in refrigerant higher pressure loss conditions. Some points which might motivate this behavior are the lower R/D ratio welded extremities and possible welding residues existing inside the pipes, not ruling out the possibility of small deformations resulting from the connections axial conformation, that could lead to choke the fluid way.

[3] . Unable to be measured for confirmation due to the electric drive occur simultaneously with the compressor.
[4] Energy measurement system read, while equipment is in Fan mode (without refrigerator cycle).

5.2 Coefficient of Performance.

As explained before, the way thought to decrease our technological procedures engaged in the prototype manufacture sets in comparison to the baseline set, manufactured in series, was to obtain a correction factor between both sets through the comparison between the two circular evaporators, reaching the results presented in Table 1.

COP								
Fan Speed	Temp.	Baseline	Elliptic Prot.	Tubular Prot.	Prototype Efficiency	Elliptic Corrected	Percentual increase
Medium	20	1,87 ± 0,431	1,53 ± 0,500	1,59 ± 0,432	85%	1,80	-4%
	30	2,81 ± 0,511	2,50 ± 0,681	2,29 ± 0,688	82%	3,06	9%
	40	4,20 ± 1,184	3,82 ± 1,022	3,42 ± 0,973	81%	4,69	12%
High	20	2,68 ± 0,800	2,42 ± 0,837	2,38 ± 0,813	89%	2,72	2%
	30	4,00 ± 1,112	3,82 ± 1,127	3,46 ± 1,045	86%	4,42	10%
	40	5,98 ± 1,401	5,73 ± 1,393	5,18 ± 1,583	87%	6,62	11%

Table 1 - Coefficient of performance

The same data is presented in Fig 12, in which the superiority of the original tubular assembly is more evident if we compare with the elliptical prototype. But when we make a comparison between the original baseline and the corrected tubular elliptical (correction factor included), we can see considerable COP increasing when the chamber temperatures are 30 and 40°C. It is better visualized in Fig 13, where the percentage COP increase is represented in function of the chamber temperature, for both analyzed fan speeds. It is possible see that the elliptical exchanger is more interesting for bigger temperatures and fan speeds, what confirm Matos(2003) and Minardes(2007) results, that show bigger heat transference for bigger Reynolds numbers.

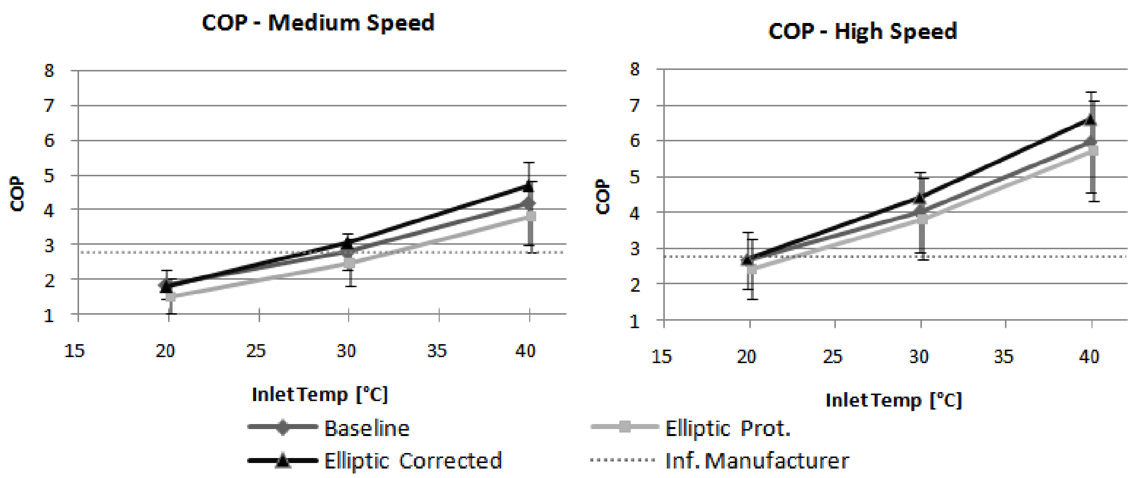

Figure 12 – COP walking for all exchanger

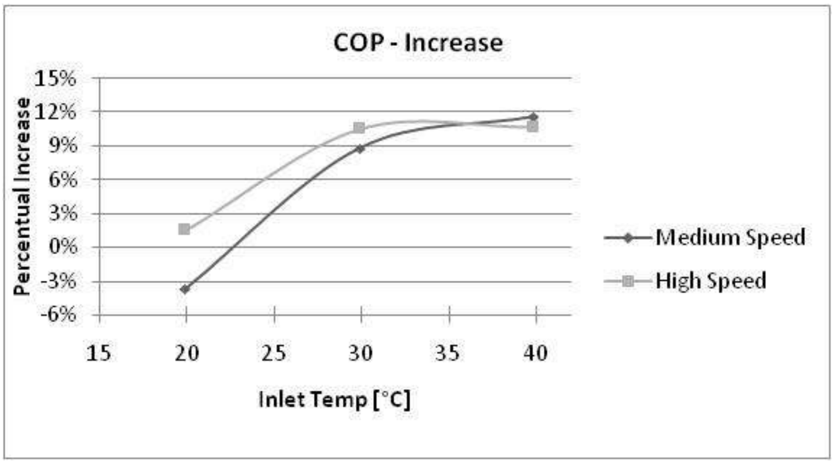

Figure 13 – COP increase, comparing baseline em corrected heat exchanger

Another interesting comparison is the one shown in Fig 14figure 14, where we can see different air's speed for each analyzed case, in the baseline and elliptical prototype. It is possible to confirm another time the results obtained before, showing that the optimum geometry besides provide larger heat transference else results in smaller pressure loss, evidenced by the air flow increase with the same electrical power consumption by its fan.

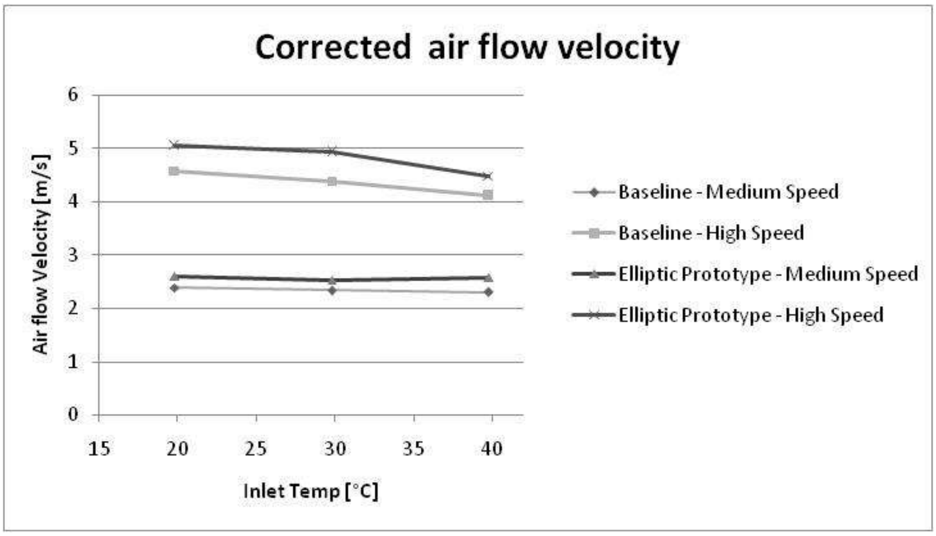

Figure 14 – Air speed comparison

It is not difficult to prevent by qualitative analyze that this optimized geometric characteristics, if used with industrial serial manufacture process, might result in more efficient products.

Another advantage of this technology is the possibility to use more tubes in the same area (smaller distance between tube's rows) or still use the same number of tubes, but in a smaller equipment keeping the heat exchange characteristics, and enabling the desired final product characteristic optimization.

To finalize this scientific work, a suggestion for future related essays is the evaluation of these two alternatives:

- *Possibilities of using more tubes in the same occupied volume, positioning all them a more adequate disposal*
- *To analyze the possibility of getting smaller products, with less material employed, and smaller occupied volume, but with the same heat exchange capacity.*

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