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EXPERIMENTAL EVALUATION OF HEAT PERFORMANCE ENHANCEMENT OF LOUVERED SERPENTINE FIN AND DIMPLED FLAT TUBE HEAT EXCHANGERS

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Abstract. *The use of concavities or dimples in heat exchangers inside tubes surface has become a common passive technique to enhance the heat transfer by increasing the surface area and internal fluid turbulence, disturbing the boundary layer. This study presents the measurement of two types of compact heat exchanger, serpentine louvered fins, being one produced with dimpled tubes and the other with smooth tubes, both with same core dimension, number of tubes, tube rows and fin pitch. The heat transfer measured for both compact heat exchangers had been evaluated thought a matrix formed by sixteen combinations of four water flow rates and four air flow rates, which is compared in order to quantify the gain of heat transfer due the dimples. The heat transfer and friction factor for the internal and external side are calculated thought the correlations found in the literature. The study concludes that the dimpled tube version had increased the overall heat transfer coefficient up to 29% at the combinations of the lowest Reynolds numbers in the water side and highest Reynolds numbers in the air side, and 1% at the combination of the highest Reynolds numbers in the water side and lowest Reynolds numbers in the air side.*

Keywords: *Heat transfer enhancement; Dimpled tubes; Compact heat exchangers; Smooth tubes;*

1. INTRODUCTION

The heat performance enhancement of heat exchangers used for automotive application is more than ever necessary since that added to the classic reasons such as less space in the vehicles engine compartment because of design and weight reduction there are also new environmental regulations such as Brazilian Proconve L6, US EPA-5 and European Euro 6 which demand a CO_2 and No_x emission reduction for diesel engines. One of the techniques applied to achieve such emission reductions is the utilization of Exhaust Gas Recirculation (EGR) process which requires an EGR cooler which utilizes the engine jacket water flow rate to cooling down the exhaust gas, adding heat to be rejected by the radiator. The use of concavities or dimples in tubes has become a common passive technique to enhance the heat transfer by increasing the surface area and internal fluid turbulence, disturbing the boundary layer. Such passive techniques to increase the heat transfer rate are advantageous with other techniques because the manufacturing processes are simple (Gupta,*et al*., 2012), adding no extra costs in raw material or labor. It will be examined the effect of dimpled tubes over a heat performance of radiators manufactured in cooper brass technology, louvered serpentine fin with two flat tubes rows (dimpled and smooth). Heat transfer enhancement in dimpled tubes has been studied by Chen, *et al.,* (2001) and a generalized heat transfer correlation for louver fin geometry has been provided by Chang and Wang (1997). A thermo-hydraulic characterization of a louvered fin and flat tube heat exchanger has been presented by Cuevas, *et al.,* (2011). The influence over the radiators heat performance will be compared in basis of ideal fan power consumptions and frictional power consumption per unit of internal tube surface area.

2. EXPERIMENTAL SETUP AND TEST RESULTS

2.1 Bench test and test procedure

The tests were conducted in an automotive calorimeter, at steady state condition (Fig. 1) through a matrix formed by the combination of four coolant flow rates and four air flow rates, as showed in the Table 1.

Table 1. Heat performance test points

Measurement	Unit	Measurement point targets
Water Flow	m^3/min	9×10^{-2}; 1.15×10^{-1}; 1.5×10^{-1} and 2.0×10^{-1}
Air Flow	kg/s	0.72; 1.34; 1.96 and 3.2
Water inlet temperature	ºC	90

The coolant used in this study is water and the external flow (fins) is air, being the water inlet temperature controlled by an electronic module which controls the electric current to the heater which heats up the water before it has been pumped through the radiator. The water flow rate is measured by a flow meter which sends the measurements to the PLC (programmable logic controller). Static pressure is measured by a pressure gage and water pressure drop is measured by a differential pressure gage. Temperature in both sides (air and water) is measured by thermo resistances device (RTD). The air flow rate is measured by venturi pipes constructed following the European standard NF EN ISO 5167-1, (1995), in carbon steel, in which is installed one differential pressure gage, one absolute pressure gage, one thermo resistance RTD and one electronic hygrometer. The measurements performed by this instrumentation are introduced in the Eq. (1) presented by Delmée (1983). The measurements are done when the steady state conditions have been reached. This moment is defined by SPC, or statistical process control, which is a method for measuring, understanding and controlling variation in a manufacturing process, as defined by AIAG (1997). Since that the steady state conditions could be defined as how much a certain value (or process) varies inside the control limits in the time domain, it is possible to trace a parallel between steady state conditions and a stable manufacturing process, thus allowing the use of SPC to determine when steady state conditions are in place and the measurement could be performed. During the test, the water flow rate, air flow rate, inlet temperature and energetic balance (between water and air heat transfers) are measured continuously and its measures can be used to evaluate the process capability (Cp) and the capability index (Cpk). The Cp is the process capability index, which is a measure of the ability of a process to produce consistent results, representing the process variation to the product specification, so that could be found by the ratio between the permissible spread and the actual spread of a process. The Cpk is an index (a simple number) which measures how close a process is running to its specification limits, relative to the natural variability of the process, meaning the accuracy of the process. The larger the index, the less likely it is that any value will be outside the specifications. Specifically for this study, the Cpk is 1.33 and the Cp is 2. When reached the Cpk and Cp target values, the test condition is considered at steady state and it triggers a data acquisition and test rig keeps the measuring point under steady state. At the end of the acquisition time, an average value is reported and the program starts the fluids flow adjustments for the next measurement point.

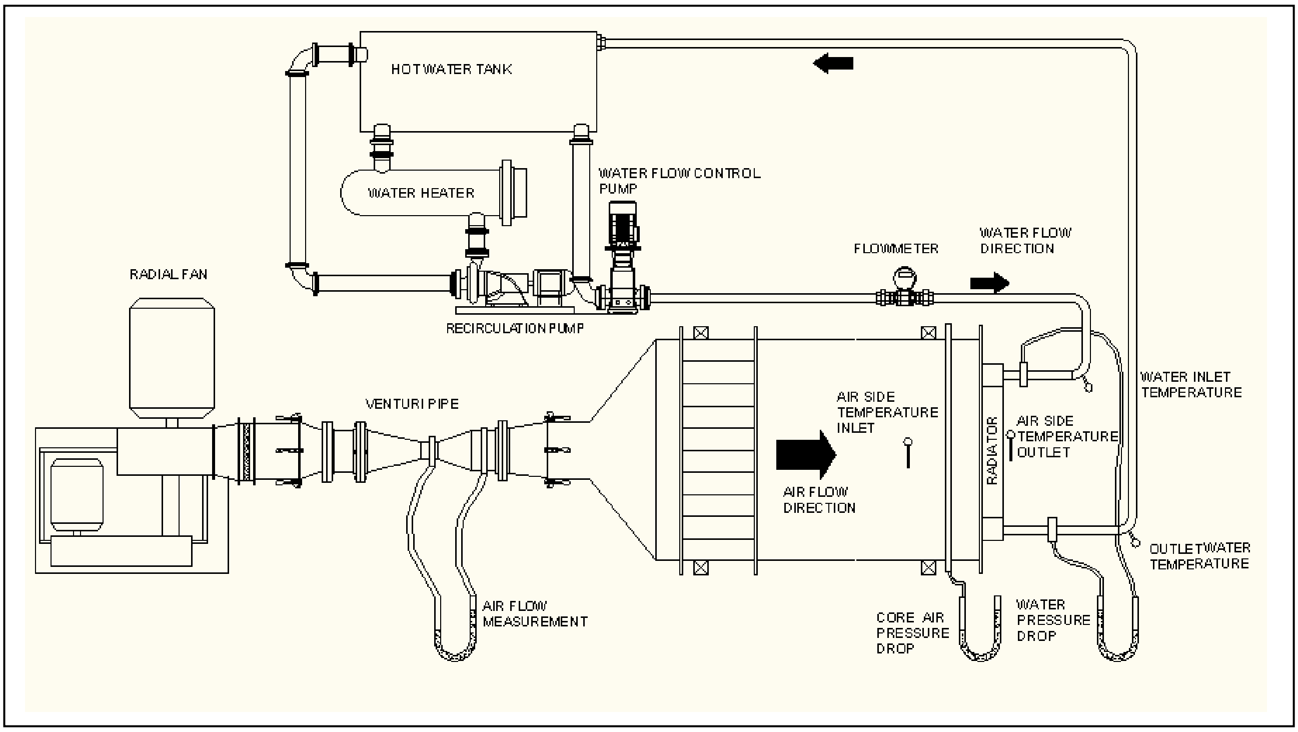

Figure 1. Diagram of test rig used to measure the hydraulic, aerodynamics and thermals characteristics.

The samples utilized in this present study have been applied in cooling modules for diesel engine applications. It is a cooper brass technology, where cooper has been used for fins manufacturing and brass for the tubes. There is one sample produced with flat tubes without dimples (smooth) and another one with dimpled tubes as presented in the Fig. 3(a) and Fig.(3b). Figure 2(a) is the samples sketch used in this study; it is also utilized here to indicate the meaning for some symbols used to represent some dimensions in the samples. Figure 2(b) shows the sample under testing in the test rig. Samples principals' geometrics characteristics are presented at Tab. 2.

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Fig. 2(a)

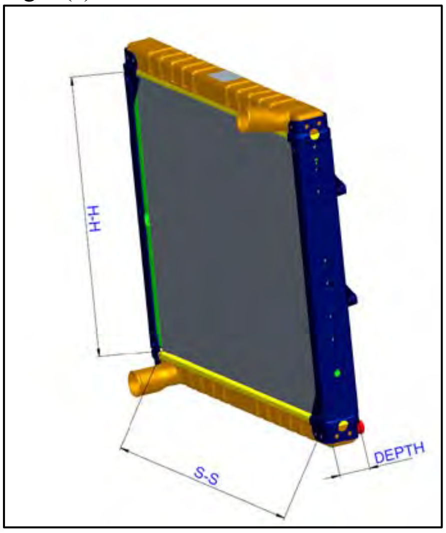

Fig. 2(b)

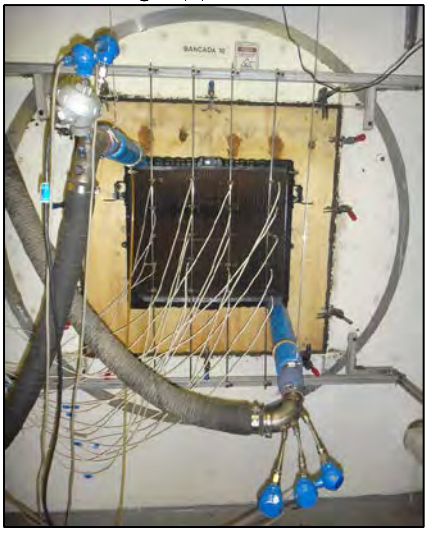

Figure 2(a). The heat exchanger basic dimensions indication. Figure 2(b). Heat exchanger under test

Fig. 3(a)

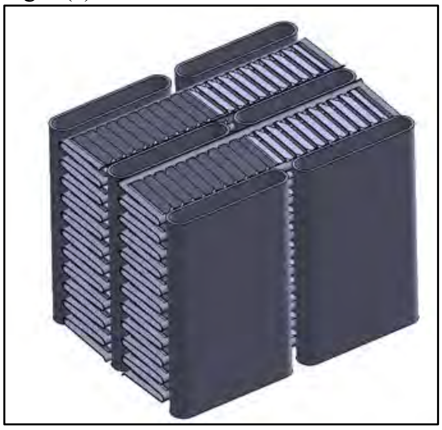

Fig. 3(b)

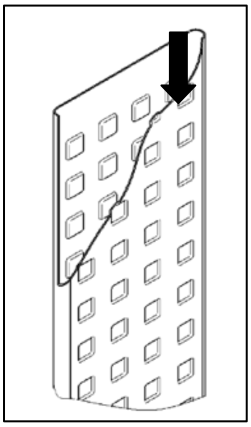

Figure 3(a). Core matrix configuration showing the louvered serpentine fin in a two tube rows configuration.
Figure 3(b). Shows a dimpled tube configuration.

Table 2. Geometrics characteristics for the samples tested for in this study.

Dimension	Symbol	Unit	Smooth tube sample (#1)	Dimpled Tube sample (#2)
Header to Header	H-H	m	0.500	0.500
Side to Side	S-S	m	0.602	0.602
Core Depth	D-D	m	0.042	0.042
Nº of Tube Rows		-	2	2
Nº of Tubes		-	112	112
Tube Hydraulic Diameter	Dh_t	m	$2.99x10^{-3}$	$2.99x10^{-3}$
Type of Tube		-	Smooth	Dimpled
Nº of Fins			57	57
Fin Pitch	Fp	m	1.69×10^{-3}	1.69×10^{-3}
Fin hydraulic diameter	Dh_f	m	2.83×10^{-3}	2.83×10^{-3}
Radiator Weight		kg	13.4	13.4
Radiator Water Volume		m^3	$3.8x\ 10^{-3}$	$3.8x\ 10^{-3}$

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2.2 Uncertantly analysis

The uncertainty sources could came from the transducers itself and from the data acquisition system.

There are not data manually recorded, thus this factor are not going to be accounted in this study.

For the transducers in which the calibrations have been performed including the data acquisition system, the declared uncertainty will be considered the relative uncertainty result and for those which the data acquisition system has been calibrated in separate; the uncertainties will be added per Eq. (1).

Table 3. Transducers characteristics.

Type of measurement	Symbol	Transducer uncertainly	Data Acquisition uncertainty	Data Acquisition and Transducer Uncertainty (Relative Uncertainty)
Water Pressure drop	Δ_{pi}	-	-	±0.4%
Venturi Pressure Drop	Δ_{pv}	-	-	±0.33%
Air fin pressure drop	Δ_{po}	-	-	±0.42%
		-	-	±0.35%
Venturi Absolute Pressure	P_v	-	-	±2.8%
Temperature	T	0.16ºC	0.15ºC	±0.22ºC
Volumetric water flow	q_v	2.3%	0.1%	±0.5%

Thus, uncertainty budget for transducer and data acquisition system is:

$$u_b = \sqrt{\left(u_1^2 + u_2^2\right)} \tag{1}$$

Where:

u_b is the uncertainty budget

u_1 is the transduce uncertainty

u_2 is the data acquisition uncertainty

The transducers calibration certificates provide the single uncertainty expression for each measurement, but it does not take in account how it will combine with the other measurements and the equations where those values will be used.

A method of estimating uncertainty in experimental results has been presented (Kline and McClintock, 1953).

Thus uncertainty propagation can be estimate by the Eq. (2).

$$\left(\frac{Ur}{r}\right)^2 = \left(\frac{x_1}{r}\frac{\partial r}{\partial X_1}\right)^2\left(\frac{Ux_1}{x_1}\right)^2 + \left(\frac{x_2}{r}\frac{\partial r}{\partial X_2}\right)^2\left(\frac{Ux_2}{x_2}\right)^2 + \ldots + \left(\frac{x_j}{r}\frac{\partial r}{\partial X_j}\right)^2\left(\frac{Ux_j}{x_j}\right)^2 \tag{2}$$

Where:

$\left(\frac{Ur}{r}\right)$ Is the relative uncertainty result, and $\left(\frac{Ux_i}{x_i}\right)$ is the relative uncertainty for each variable

The uncertainty has been estimated for the heat transfer in function of tube hydraulic diameter Reynolds number (Re_{dh}) for the water side evaluation and louver pith Reynolds number (Re_{LP}) for the air side evaluation.

Figure 4 presents the results for the water side and air side at the measurements points and refers to a reference baseline considered with zero uncertainty. The uncertainty is higher at low Re_{LP} and high Re_{dh} since that at this test conditions the water temperature differential between inlet and outlet radiator is very low, increasing the influence of temperature uncertainty at low heat transfer measurements, as long as the air flow increases (higher Re_{LP}) and water flow decreases (lower Re_{dh}) the uncertainties tends to converge to smaller values.

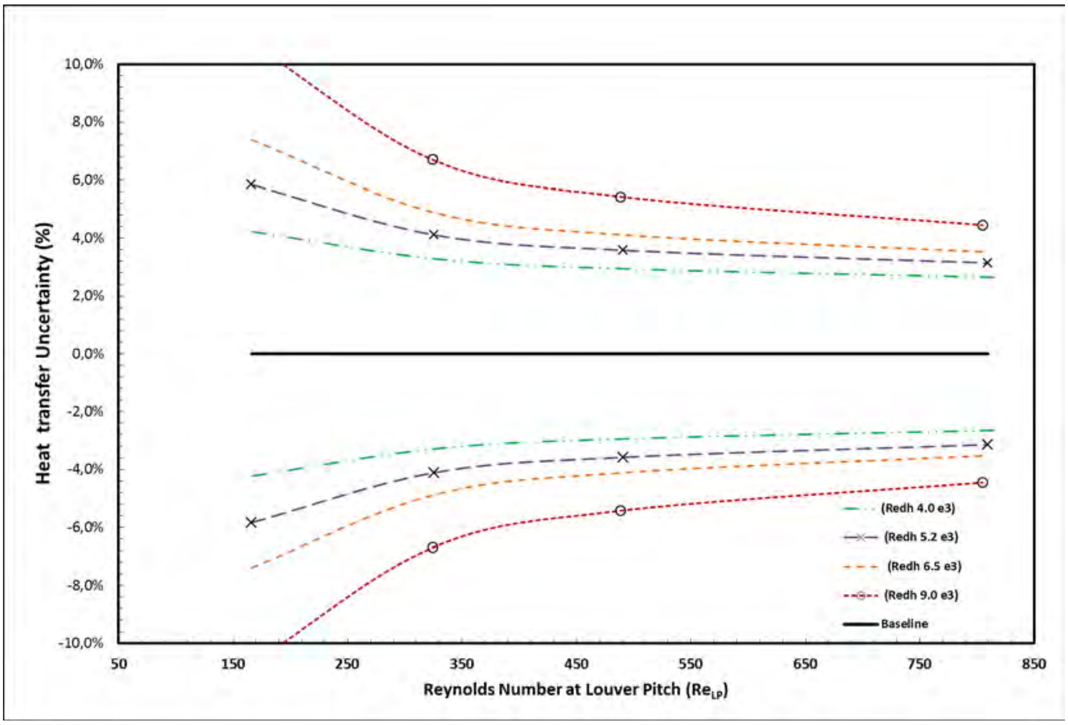

Figure 4. Uncertainty for the heat transfer results.

2.3 Heat transfer coefficients correlations and results

The air side volumetric air flow is determined by the Eq. (3) presented in the ISO 5167-1 (2003).

$$m_o = (K \;\; C \;\; E \;\; \beta^2 \; D^2 \; \varepsilon \sqrt{\rho_v \;\; \Delta_p}) \tag{3}$$

Where E = Approximation velocity factor

$$E = \frac{1}{\sqrt{(1-\beta^2)}} \quad \text{and} \quad \beta = \frac{d}{D} \qquad (4) \text{ and } (5)$$

C = Discharge coefficient $f(D, R_D, \beta)$

Thus, the relation by $C \, E \, \beta^2$ represents the flow coefficients for concentric measuring systems with reduction in the flow pressure and its value can be found in tables based in the Reynolds number R_D.

d = Venturi throat diameter.
D = Pipe line diameter or venturi major diameter
K = Venturi project characteristics grouped constant.
ε = Isentropic expansion factor
Δ_p= Pressure differential in the venture
ρ_v = Air density in the venturi pipe.

The radiator heat exchange for the water side and air side is given by the Eq. (6) and (7), (Incropera, 1996):

:

$$\dot{Q}_i = \dot{m}_i . c_{pi} . (t_{1i} - t_{2i}) \quad \text{and} \quad \dot{Q}_o = \dot{m}_o . c_{po} . (t_{1o} - t_{2o}) \qquad (6) \text{ and } (7)$$

Where, respectively we'll have:

$\dot{Q}_i$; $\dot{Q}_o$ Water side and air side heat flow (W)

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$\dot{m}_i$; $\dot{m}_o$ Water side and air side mass flow (kg s^{-1})
c_{pi} ; c_{po} Specific heat at average temperatures, J kg^{-1}K^{-1}
t_{1i} ; t_{2i} Water side inlet and outlet temperature (°C)
t_{1o} ; t_{2o} Air side inlet and outlet temperature (°C)

Since that at steady state conditions the heat supplied by the water flow rate must be the same as the exchanged thought the air flow rate($\dot{Q}_i = \dot{Q}_o$), the relationship between (6) and (7) must results the closest as 1 as possible.

For this study, the maximum allowed divergence between the measurements of each side was ±7%, and it has been found by the Eq. 8 for the energy balance.

$$BE = \left(1 - \frac{\dot{Q}_i}{\dot{Q}_o}\right).100 \leq \pm 7\% \tag{8}$$

Figure 5 shows how the energy balance for all measurements for this study, behaved through the adopted measurement range.

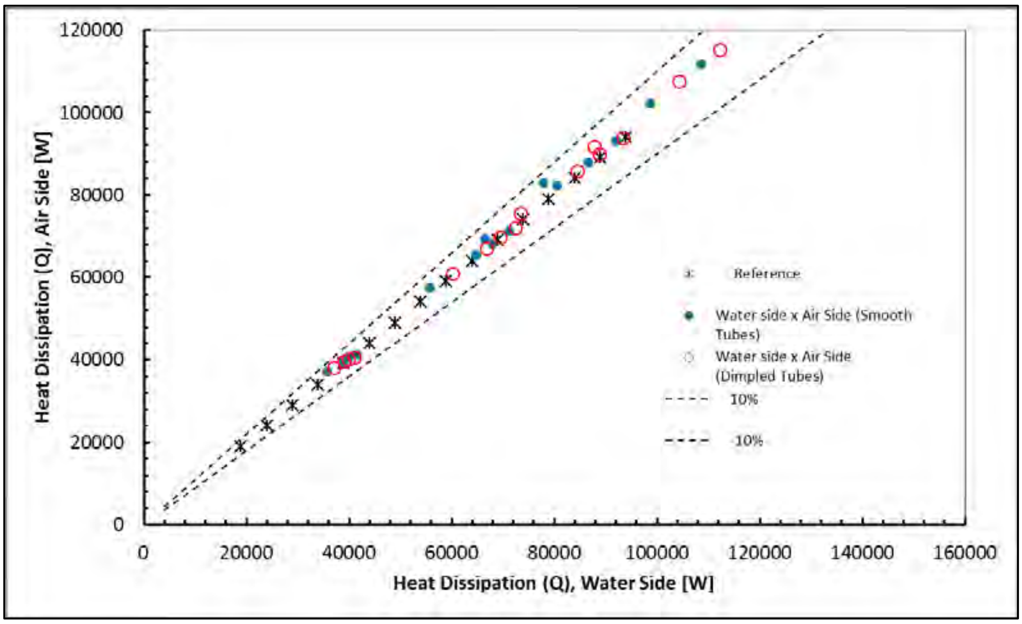

Figure 5. Air side heat transfer versus water side radiator heat dissipation

The number of transfer units, *NTU*, is a dimensionless parameter widely used for heat exchanger analysis, from what the overall heat transfer coefficient, *U*, can be determined. Thus, for the comparison between dimpled and smooth tubes in this study, the overall heat transfer will be used, coupled with the ideal fan power consumption and friction power expended per unit of surface area in order to evaluate the heat performance. The *NTU* can be found by the interaction between the Eq. (9) and Eq. (10) used for effectiveness, (Incropera, 1996).

$$\varepsilon = 1 - \exp\left[\left(\frac{1}{C_r}\right)(NTU)^{0.22}\left\{\exp\left[-C_r\,(NTU)^{0.78}\right] - 1\right\}\right] \tag{9}$$

$$\varepsilon = \frac{C_o\left(T_{2,o} - T_{1,o}\right)}{C_{\min}\left(T_{1,i} - T_{1,o}\right)} \tag{10}$$

Where:
The air side capacity flow is determined by the Eq. (11).

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$$C_o = \dot{m}_o \, cp_o \tag{11}$$

The water side capacity flow is determined by the Eq. (12).

$$C_i = \dot{m}_i \, cp_i \tag{12}$$

The C_{min} is equal to C_i or C_o whichever is smaller and C_{max} whichever is bigger.
The capacity flow ratio C_r is given by the Eq. (13).

$$C_r = \frac{C_{\min}}{C_{\max}} \tag{13}$$

$\dot{m}_i$ is the water mass flow rate, kg s^{-1}
$\dot{m}_o$ is the air mass flow rate, kg s^{-1}

The overall heat transfer coefficient is given by the Eq. (14).

$$NTU = \frac{AU}{C_{\min}} \tag{14}$$

Where A, is the tube internal area for the overall heat transfer coefficient based in the water side, U_i, and external area for the overall heat transfer coefficient based in the air side, U_o. The friction power expended per unit of surface area E (W m^{-2}), can be readily evaluated as a function of the Reynolds number, the friction factor and the specific fluids proprieties from Eq. (15) presented by Kays and London, (1998).

$$E = \frac{1}{2} \frac{\mu^3}{{\rho_o}^2} \left(\frac{1}{4 r_h} \right)^3 \zeta \, {\mathrm{Re}_{Dh}}^3 \tag{15}$$

The water side friction dimensionless factor ζ is given by the Eq. (16).

$$\zeta = \frac{2 \, \Delta p_o \, D_h}{L \, \rho_o \, v^2} \tag{16}$$

And:
L is the tube length, m
ρ_o is the water average density, kg m^{-3}
v is the water velocity in the tube m s^{-1}
Δp_o'is the water pressure drop, Pa
D_h is the tube hydraulic diameter, m
μ is the viscosity, N s m^{-2}
r_h is the hydraulic radius, m

Figure 6 presents the U_i in function of friction power expended per unit of surface area, E, at four different air flows indicated per Reynolds number at louver pitch and Fig.9 presents U_o in function of power expended for an ideal fan. As the friction power decreases due the decrease of the water flow rate, the heat transfer coefficients enhancement in the dimpled tubes are higher. At higher Reynolds louver pitch numbers and higher friction power, E, the water side overall heat transfer coefficient of both radiators tends to converge. If compared at same E, the U_i for the dimpled tubes radiator increases up to 29% over the smooth tubes version at the lowest E values. When it is compared at the highest E values, the difference between the two radiators versions reduces to 1%, as showed in the Fig.7. Since that there is no significant difference between the air pressure drop measurements for both radiators versions, as it can be seem at the Fig. 8, it is possible to evaluate the overall heat transfer coefficient based in the air side at same ideal fan power, allowing to consider the comparison of both radiators versions also under the point of view of fan power consumption. The ideal fan power consumption is estimated by the Eq. (17), Thome, (2010).

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$$P_i = \left(\frac{\dot{m}_o}{\rho_o} \right) \Delta p_o \tag{17}$$

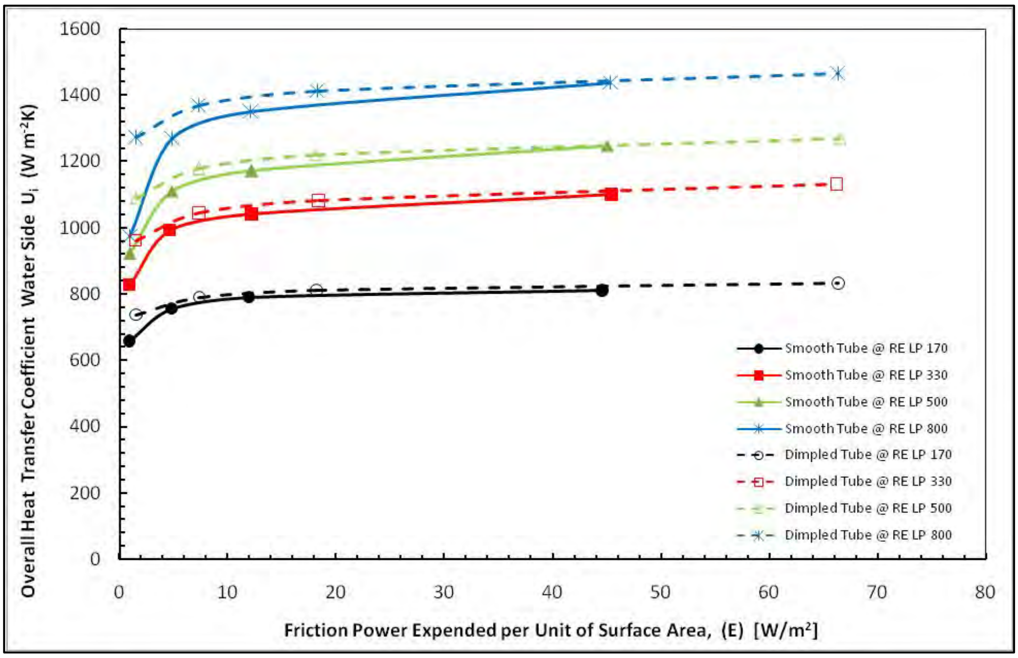

Figure 6. Overall heat transfer coefficient (water side) versus friction power, *E*, for both radiators versions

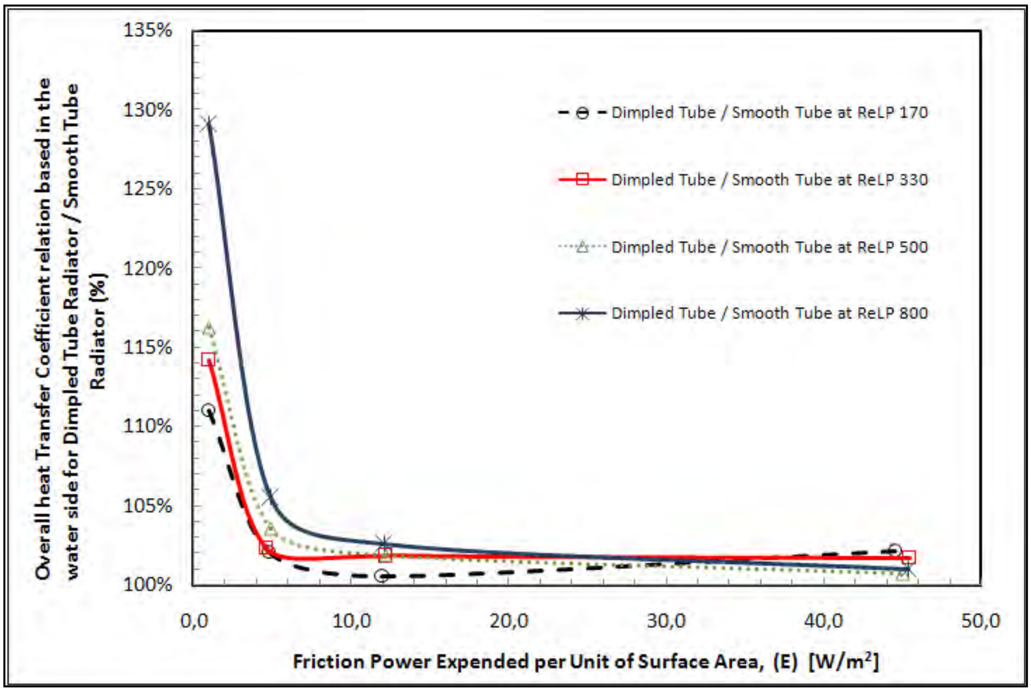

Figure 7. Internal overall heat transfer coefficient relation for dimpled and smooth tube at same friction power unit surface, *E*

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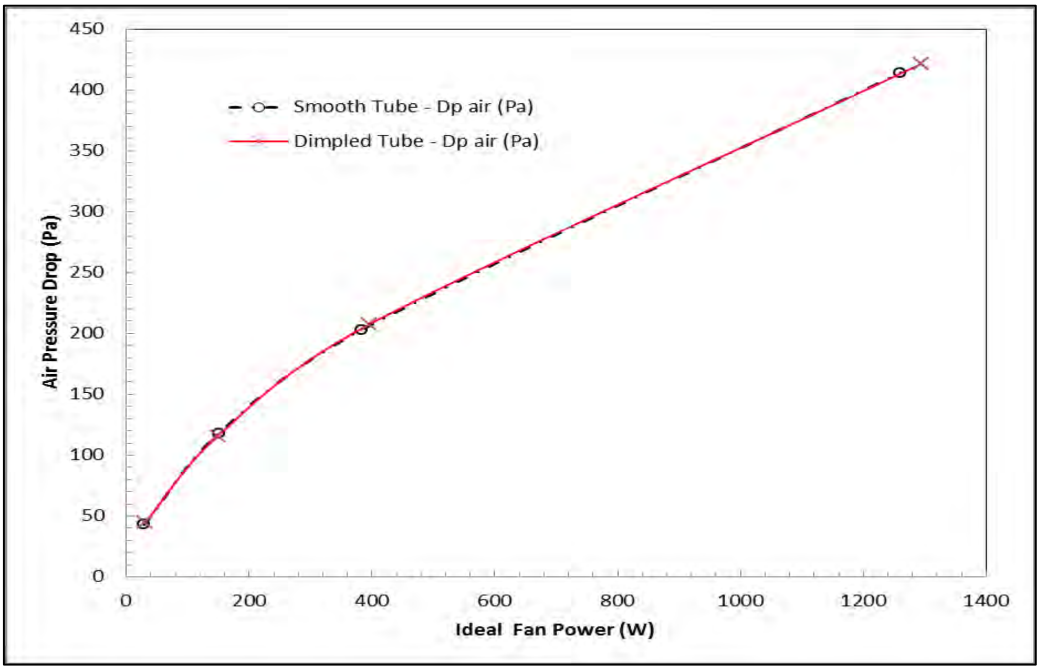

Figure 8. Air pressure drop for the smooth and dimpled tube radiator versions versus ideal fan power consumption

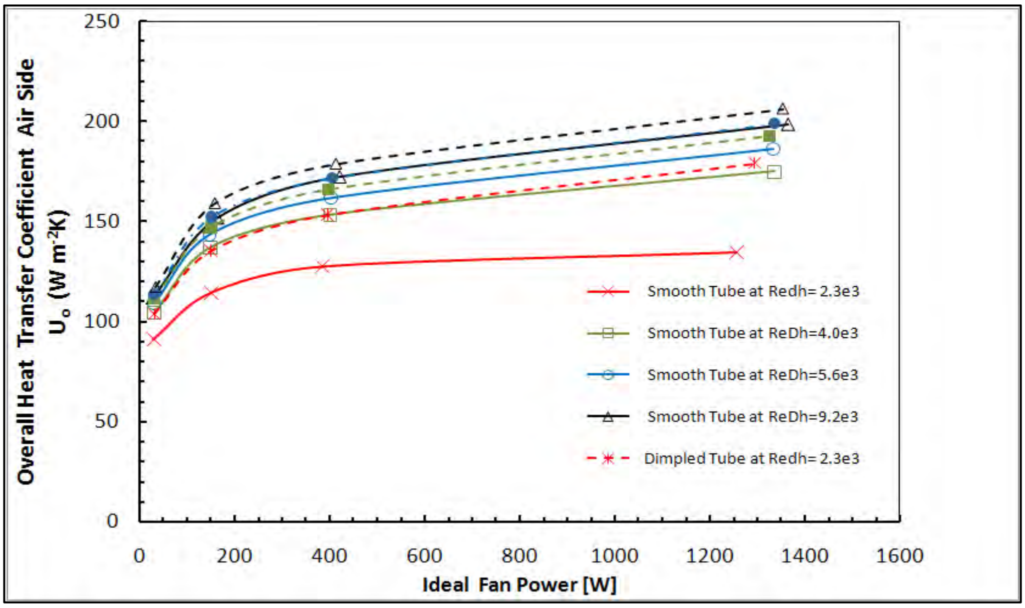

Figure 9. Overall heat transfer coefficient based in the air side for the smooth and dimpled tube radiator versions versus ideal fan power consumption.

3. CONCLUSION

The comparison between the dimpled and smooth tube radiators versions, presented in the Fig.7 had showed that when the analysis is performed at the highest Re_{Lp} and lowest friction power, E, which can be correlated to the lowest Re_{dh}, the overall heat transfer enhancement is up to 29%. As the friction power E increases from 1 to 10W m^{-2}, the overall heat transfer enhancement reduces. Above E = 10W m^{-2} until the maximum value measured of 45W m^{-2}, the dimples enhancement is about constant, where the overall heat transfer enhancement is about 1%.

Thus the dimples have a strong effect over the overall heat transfer coefficient in the lowers E and Re_{dh}, having its influence reduced as the E and Re_{dh} increases gradually. Since that both radiators have the same fan power consumption the overall heat transfer enhancement obtained does not need any further consideration.

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5. REFERENCES

AIAG, 1997. *Fundamentos de controle estatístico de processo CEP-Manual de Referência*. IQA, 1st edition.

Chan, Y.J. and Wang, C.C., 1997. "A generalized heat transfer correlation for louver fin geometry", *Int.J.Heat Mass Transfer,* Vol. 40, Nº 3, p. 533-564.

Chen, J., Müller-Steinhagen, H. and Duffy, G.G., 2001. "Heat transfer enhancement in dimpled tubes", *Applied Thermal Engineering* Vol.21, p.535-547.

Cuevas C., Makaire D., Dardenne L., Ngendakumana,P., 2011. "Thermo-hydraulic characterization of a louvered fin and flat tube heat exchanger", *Experimental Thermal and Fluid Science* Vol.35, p.154-164.

Delmée, G.J.,1983. *Manual de Medição de Vazão*. Edgard Blücher Ltd, 2nd edition.

Gupta, A. and Uniyal 2M., 2012. "Review of Heat Transfer Augmentation Through Different Passive Intensifier Methods", *IOSR Journal of Mechanical and Civil Engineering (IOSRJMCE)* ISSN:2278-1684 Vol. I, Issue 4, p.14-21.

Incropera, F.P. and DeWitt, D.P., 1996. *Fundamentals of Heat and Mass Transfer*, John Wiley & Sons, 4th edition.

ISO 5167-1, 2003. "Measurment of Fluid Flow by Means of Pressure Differential Devices Inserted in Circular Cross-Section Conduit Running Full- Part 1: General Principles and Requeriments".

Kays W.M. and London, A.L., 1998. *Compact Heat Exchangers*, 3rd Edition.

Kline, S.J. and McClintock,F.A., 1953."Describing Uncertainties in Single Sample Experiments". *Mechanical Engineering*, Jan. 1953.

Norme Européenne, NF EN ISO 5167-1, 1995. *Measure de debit des fluids au moyend'appareilsdéprimogènes*, *Partie 1: Diaphargmes, tuyères et tubes de Venturiinsérésdans des conduires en charge de section circulaire*.

Thome, J.R., 2010. *Engineering Data Book III*, Wolverine Tube Inc.

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