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Research Paper

Theoretical Analysis Versus Experimental Results Of The Flat Tube Compact Heat Exchanger (Automotive Radiator)

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ABSTRACT: *A compact heat exchanger analysis of the types used frequently in the automotive and aeronautical industries is performed. The compact finned-tube heat exchanger with flat fins was analyzed in greater depth since it is widely used in automotive radiators, the present study's primary object. The theory was developed to determine the heat transfer rate and pressure drop for this specific compact heat exchanger by applying the effectiveness (ε-NUT). The analyzed radiator is popularly known as a water cooler and is used in medium trucks with load capacities between 15 and 20 tons. The truck type has no turbo compressor, which implies the absence of an air cooler, or "intercooler. " The air that enters the fins makes it at room temperature. Theoretical versus experimental comparisons were made. Results allow us to conclude that the simulations performed are consistent when the input hypotheses defined as valid for the experiments are assumed. Stopping criteria for iterative processes, determining heat transfer rate and pressure drop, using the experimental data of a single available experiment, and need to be refined, for better characterization and robustness of theoretical procedure. There is a lack of empirical expressions in open literature. Large corporations, builders of compact heat exchangers, do not provide information obtained in their laboratories, since the data are strategic for the survival.*

KEYWORDS: *Compact heat exchanger, Automotive radiators, Theory of effectiveness (ε-NUT).*

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I. INTRODUCTION

The objective of the article is to the analysis of compact heat exchangers, with the ultimate goal of comparing theoretical and experimental results for a flat finned tube heat exchanger, presented by Ribeiro, L.N. (2007), with an emphasis on heat transfer rate and pressure drop. Also, apply the theory of effectiveness associated with compact heat exchangers.

The demand for ever more technologically advanced automotive vehicles leads to more efficient, light, and compact engines. The thermal control of these vehicles' engines is obtained through automotive radiators Kakaç and Liu (2002).

Research involving automobile radiators has been developed over the years. Companies in the automotive sector invest high resources in all types of techniques that can optimize energy performance. In light of this, new experimental devices are built to improve automotive radiators by incorporating new technologies, such as heat pipes and siphon terms Pabón, N. Y. L. (2014).

Compact flat plate heat exchangers, due to their compactness, low weight, and high effectiveness, have been used in automotive and aerospace applications. The thermal performance of flat plate type exchangers is determined, among other variables, by the fin geometry. Pressure drop and heat transfer in a flat plate type heat exchanger are characterized in terms of the dimensionless coefficients of Colburn, J, and Friction, f, as a function of the number of air Reynolds different types of heat exchange surfaces. Analytical, direct determination of dimensionless parameters such as heat transfer rate and pressure drop are incredibly complex. The difficulty is that the heat transfer coefficient and the friction factor are strongly dependent on geometric parameters. These parameters are the fin height, fin spacing, fin thickness. The heat exchanger needs to be characterized separately, according to Alur, S. (2012), and Mazumdar, S. (2007).

The text by Kays and London (1984) provides an excellent introduction to the analysis of compact plate-type heat exchangers. It contains data on the heat transfer coefficient and friction factor for various geometries. The authors provided J and f data for countless and different types of compact heat exchanger configurations. The results have been presented for more than three decades and continue to be applied. Due to the experimental difficulty obtaining this type of information in academic research laboratories, there is no information in the current open literature with such scope.

Relatively recent academic work Alur, S. (2012) presents experimental data for a countercurrent type heat exchanger, where highly pressurized air flows through a heat exchanger channel. The results are expressed in terms of the heat exchanger's effectiveness and the pressure drop as a function of the air's mass flow.

An experimental project was presented in a master's dissertation by Pabón, N. Y. L. (2014), to obtain the necessary conditions for an automotive radiator's thermal characterization. In this project, a wind tunnel and a hydraulic circuit thermally coupled to the automotive radiator were built. No experimental results were presented that could corroborate the data of the theoretical model developed in this work.

Nogueira, E., Aroucha, L.C.A., Pereira, F. L. (2019) implemented a solution to expand the application range of the Colburn Factor and the Friction Coefficient related to the compact flat tube heat exchangers. The theory was used in comparison with experimental results. Then, an expansion of the application range was proposed for f, the Friction Coefficient, and J, the Colburn Factor.

Vehicle manufacturing companies have perfected procedures and techniques to perform radiators' thermal evaluation under real operating conditions. However, large corporations throughout the world, which use compact heat exchangers, have no interest in presenting results obtained in their laboratories. The data are strategic for survival.

Figure 1, below, shows experimental results for a flat finned tube heat exchanger used in automotive radiators Özisik, M. N. (1990; page 487).

Even for simpler geometries of compact heat exchangers, there are many variables to characterize them, as previously mentioned properly. The variables found in the literature are a) the hydraulic diameter; b) the space between the ends; c) the height between fins; d) the thickness of the ends; e) the total volume of the heat exchanger; f) the minimum airflow area; g) the frontal area of the airflow.

The speed, pressure, and temperature fields have been determined using J and f as a function of Reynolds numbers associated with air, as already established, through Figure 1, below. These dimensionless factors are commonly characterized by Reynolds geometry and number in the form: J or f = F (Rea, geometry) = A Rea F (dimensionless geometric parameters).

In situations where there is the geometric similarity, kinematic, and dynamic similarity, the data obtained by Kays and London (1984; page 273) are used in practice.

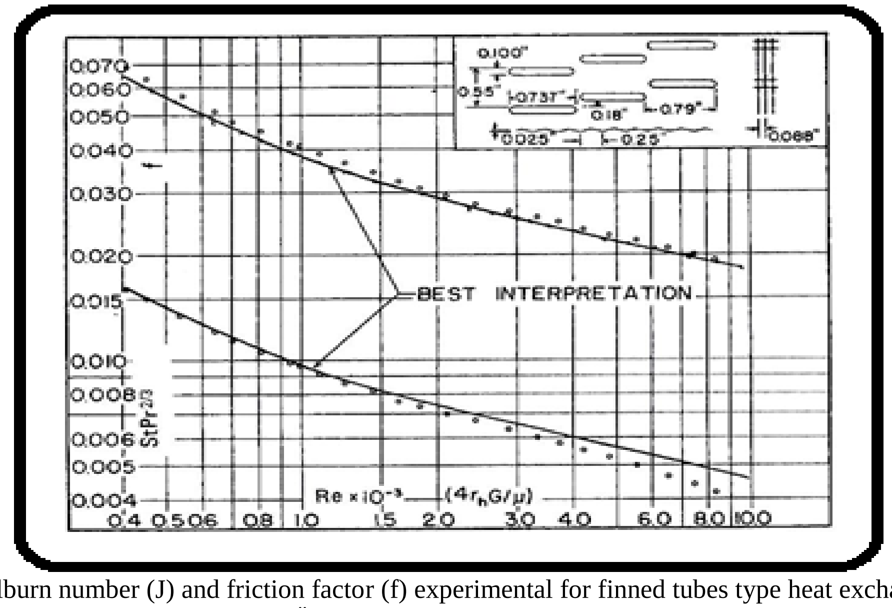

Figure 1 - Colburn number (J) and friction factor (f) experimental for finned tubes type heat exchanger. Source: Özisik, M. N. (1990; p. 487)

Interpolations were performed to compute the data available in Figure 1, in this work, and the following equations were used:

$$J = 0.02619862019 - 3.626028274^{-5}Re_a + 3.16047951^{-8}{Re_a}^2 - 1.568380435^{-11}{Re_a}^3 + 4.633774581^{-15}{Re_a}^4 - 8.056435353^{-19}{Re_a}^5 + 7.225312269^{-23}{Re_a}^6 - 8.564813179^{-28}{Re_a}^7 - 4.452384011^{-31}{Re_a}^8 + 3.807970933^{-35}{Re_a}^9 - 1.015364192d^{-39}{Re_a}^{10} \quad (1)$$

and

$$f = 1.199866203\,\mathrm{Re_a}^{(-0.4747697361)} \tag{2}$$

Using the expressions above, it became possible to determine the heat transfer rate and the head loss in flat finned tube heat exchangers.

The model developed, using the procedures specified below, used data from the master's dissertation on the Automotive Radiator of Ribeiro, L.N. (2007), according to data in the sections below. The radiator used in the experiment is popularly known as the water radiator. It is used in medium trucks, with load capacities between 15 and 20 tons. The truck does not have a turbocharger, which implies the absence of an air radiator, or "intercooler." And the air that enters through the ends does so at room temperature.

II. METHODOLOGY

The model developed, using the procedures specified below, used data from the master's dissertation on the Automotive Radiator of Ribeiro, L.N. (2007), according to data in the sections below.

2.1 Characterization of Automotive Radiator

The heat exchanger used for comparison between models is represented by Figures 2 - 3 and Tables 1 - 2, of physical dimensions and air properties:

Figure 2 - Water-to-air heat exchanger used to cool engines in cars and trucks. Source Özisik, M. N. (1990; p. 450)

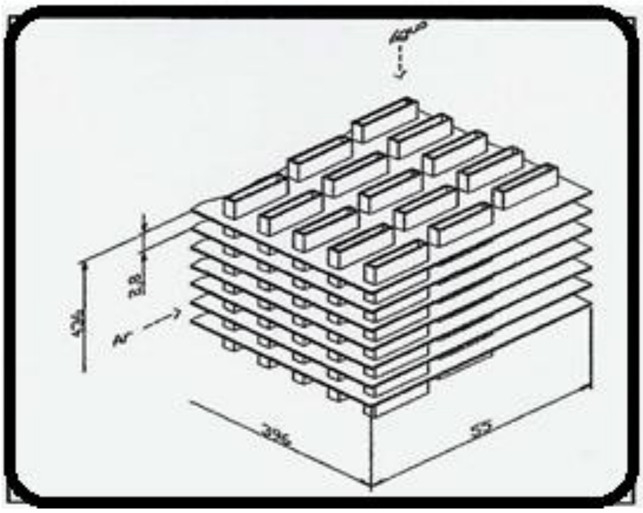

Figure 3 - Geometry of the finned flat tube heat exchanger. Source: [Master Thesis Ribeiro, L.N. (2007; Behr Brasil Ltda.)]

Table 1 - Dimensions of the finned flat tube heat exchanger

Features	*Nomenclature*	*Value*
Heat Exchanger Width	B	396 mm
Heat Exchanger Height	H	436 mm
Heat Exchanger Thickness	L	55 mm
Dimensions of tubes	a x b x c	13,3 x 2,6 x 448 mm
Wet perimeter of each tube	Pe	31,8 mm
Hydraulic diameter	D_H	4,36 mm
Number of pipe rows	Nf	3
Tubes per row	Tf	43
Total number of tubes	Qt	129
Cross-tube spacing	ST	8,8 mm
Distance between tubes	SL	18,3 mm
Thickness of fins	e	0,05 mm
Spacing between fins	E	2,8 mm
Number of fins	Qa	155

Table 2 - Thermo-physical properties of air

Property	*Value*
cp	1,008 kJ/(kg.K)
k	28,816 W/(m.K)
ν	19,31 x 10^{-6} m^2/s
ρ	1,048 kg/m^3
Pr	0,702

The relevant experimental data for the comparisons made, heat transfer rate and pressure loss, were obtained by Ribeiro, L.N. (2007) in the wind tunnel of Behr Brasil Ltda and are represented through Figure 4.
Similarly, to the dimensionless quantities J and f, interpolations provide the approximate equations for the quantities:

$$QK_{exp} = 74.96904025 + 53.18082817m_a - 3.148106367{m_a}^2 + 0.131605291{m_a}^3 - 0.002308469675{m_a}^4 \quad (3)$$

and

$$\Delta P_{Exp} = 5.833333333 - 2.91045066dm_a + 2.769230769{m_a}^2 - 0.2812742813{m_a}^3 + 0.008741258741{m_a}^4 \quad (4)$$

at where

QK_{exp} is the ratio between the experimental heat transfer rate and the average logarithmic temperature difference; ΔP_{Exp} is the experimental pressure drop and m_a is the mass flow rate of air.
With the information defined above, a theoretical, iterative procedure can be established to determine the radiator's heat transfer rate and the pressure drop on the airside (which makes it possible to define, a priori, the power of the fan at used).

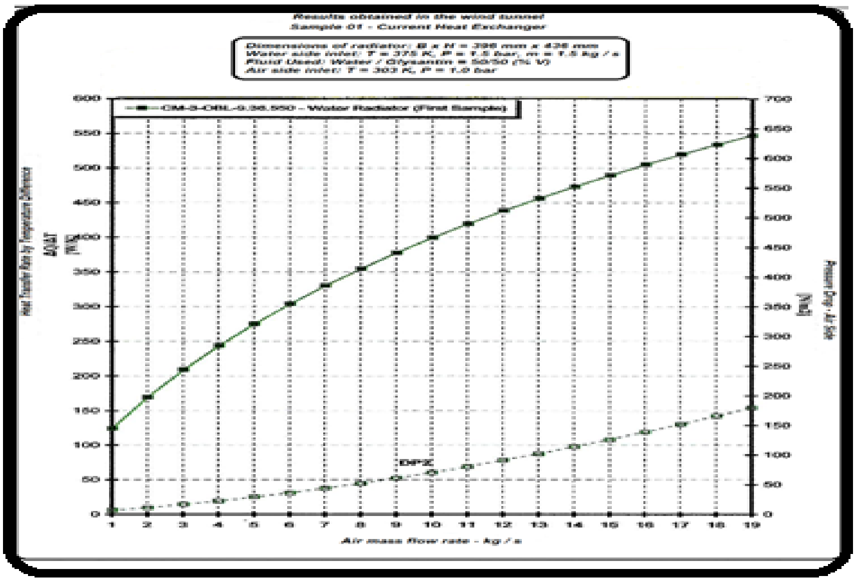

Figure 4 - Heat transfer rate and pressure drop for the Automotive Radiator. Source: Ribeiro, L.N. (2007; p. 72)

2.2 Theoretical Determination of the Heat Transfer Rate

The theoretical determination of the heat transfer rate depends on the overall heat transfer coefficient. The heat transfer coefficient depends on the heat transfer coefficients, ha, hw, air, and waterside. Initially, it is necessary to determine the physical properties according to the average temperatures of the fluids. However, in theory, the outlet temperatures are unknown a priori, and the average temperatures should be estimated initially. With the outlet temperatures initially stipulated, defined physical properties, and the geometric quantities of the exchanger provided, we have,

$$G_a = \frac{m_a}{A_{min}} = \frac{m_a}{\sigma_a A_{fr}} \tag{5}$$

$$Re_a = \frac{G_a D_{ha}}{\mu_a} \tag{6}$$

$$J = \frac{h_a}{G_a c_{pa}} Pr_a^{\ 2/3} \tag{7}$$

The Prandtl number for air, Pr_a is obtained for air as an ideal gas, published by Çengel and Boles (2013, pg. 934):

$$Pr_a = 1.005351636d0 + 0.01292094145Tmed_a + 2.524174317^{-5}Tmed_a^{\ 2} - 5.074647769^{-8}Tmed_a^{\ 3} + 1.564763295^{-8}Tmed_a^{\ 4} \tag{8}$$

$$h_a = J \frac{G_a c_{pa}}{Pr_a^{\ 2/3}} \tag{9}$$

for the water:

$$\mu_{máx} = \frac{m_w}{\rho_w A_{mín}} = \frac{m_w}{\rho_w \sigma_w A_{fr}} \tag{10}$$

$$Re_w = \frac{\mu_{máx}\ D_{hw}}{v_w} \tag{11}$$

Considering the water flow regime in the tube as fully developed, we have, for turbulent flow, approximately:

$$\mathrm{Nu} = 0,023\ \mathrm{Re}_w^{\ 0,8} Pr_w^{\ 0,4} \tag{12}$$

If the flow regime in the water is to be laminar, an interpolation of the data from the master's Dissertation of Nogueira, E. (1988; page 130), in the region of thermal input is used:

$$Nu = 1.409019812d0Z_w^{(-0.3511653489)} \quad \text{para } 10^{-5} \leq Z_w < 10^{-3} \tag{12.1}$$

$$Nu = 1.519296981d0Z_w^{(-0.3395483303d0)} \quad \text{para } 10^{-3} \leq Z_w < 10^{-2} \tag{12.2}$$

$$Nu = 10.8655 - 570.4671787Z_w + 28981.67578Z_w^2 - 950933.9838Z_w^3 + 20237498.47Z_w^4 - 276705269.6Z_w^5 + 2340349265Z_w^6 - 1.112482493^{10}Z_w^7 + 2.269345238^{10}Z_w^8;\ 10^{-2} \le Z_w 10^{-1} \qquad (12.3)$$

$$Nu = 5.261d0 - 19.930190482Z_w + 139.4921627Z_w^2 - 605.9954034Z_w^3 + 1716.100694Z_w^4 - 3217.96875Z_w^5 + 3954.86111Z_w^6 - 3056.051587Z_w^7 + 1344.246031Z_w^8 - 256.2830687Z_w^9;\ 10^{-1} \le Z_w = 10^0 \qquad (12.4)$$

Soon,

$$h_w = Nu\frac{k_w}{D_{hw}} \qquad (13)$$

The overall heat transfer coefficient is obtained through the heat exchange area of the air. It is necessary to determine the fin's efficiency to perform the calculations, as there is temperature variation between the base of the fin and its end. Soon,

$$\eta = \frac{tgh(mL)}{mL} \qquad (14)$$

at where,

$$mL = \sqrt{2h_a/k_a t} \qquad (15)$$

The fin efficiency, weighted by the area, is determined by:

$$\eta' = \beta\eta + 1 - \beta \qquad (16)$$

at where,

$$\beta = \frac{\acute{a}rea\ da\ aleta}{\acute{a}rea\ total} \qquad (17)$$

soon,

$$\frac{1}{U_a} = \frac{1}{\eta' h_a} + \frac{1.0}{A_{med}K_{aleta}} + \frac{1}{(A_w/A_a)h_w} \qquad (18)$$

at where

$$A_{med} = \frac{A_a + A_w}{2.0} \qquad (19)$$

and

$$\frac{A_w}{A_a} = \frac{\acute{a}rea\ de\ transfer\hat{e}ncia\ de\ calor\ do\ lado\ da\ \acute{a}gua}{\acute{a}rea\ de\ transfer\hat{e}ncia\ de\ calor\ do\ lado\ do\ ar} \qquad (20)$$

By the theory of effectiveness (ε-NUT), we have:

$$N = NTU = \frac{A_a U_a}{C_{min}} \qquad (21)$$

The thermal capacities of air and water are calculated by:

$$Ca = m_a * Cp_a \qquad (22)$$

and

$$Cw = m_w * Cp_w \qquad (23)$$

at where

C_{min} *is the lowest value between the thermal capacities of water and air*

Finally,

$$Q = \varepsilon C_{min}(T_{h,af} - T_{c,af}) \qquad (24)$$

$$\Delta T_{Ln} = \frac{Q}{U_a A_{total}} \qquad (25)$$

$$QK_{teo} = \frac{Q}{\Delta T_{Ln}} \qquad (26)$$

at where

$$\varepsilon = 1 - exp\left[\left(\frac{c_{min}}{c_{max}}\right)^{-1}(NTU)^{0,22}\left\{exp\left[-\frac{c_{min}}{c_{max}}(NTU)^{0,78}\right] - 1\right\}\right] \qquad (27)$$

according to Kakaç, S. (1991, p. 35).

QK_{teo} is the theoretical value for the ratio between the heat transfer rate in the air and the logarithmic mean temperature difference (DTML).
With the heat transfer rate determined, as a first approximation, the air and water outlet temperatures can be calculated using the energy balance equations:
With the heat transfer rate determined, as a first approximation, the air and water outlet temperatures can be calculated using the energy balance equations:

$$Q = \varepsilon C_{min}\left(T_{h,af} - T_{c,af}\right) \quad (28)$$

and

$$Q = m_a c_{pa}\left(T_{a,af} - T_{a,ef}\right) \quad (29)$$

The average outlet temperatures, air, and water can then be determined and compared with the temperatures initially
stipulated:

$$T_{m,ar} = \frac{T_{a,af} + T_{a,ef}}{2} \quad (30)$$

and

$$T_{m,água} = \frac{T_{a,wf} + T_{a,wf}}{2} \quad (31)$$

With the average temperatures finally calculated, it is possible to compare the heat transfer rate values. If they are outside an acceptable value, one can restart the term properties' calculations until a satisfactory convergence. The problem is obtained when compared with experimental values, or empirical expressions are lower than a precision.

2.3 Theoretical Determination of Pressure Drop in Air

In calculating the head loss in a finned heat exchanger, the main losses are related to the friction factor (f), and the head loss on the airside can be determined by:

$$\Delta P = \left[\frac{\left(G_a{}^2\right)}{2\rho_{ai}}\right]\left[\left(1.0 + \sigma_a{}^2\right)\left(\frac{\rho_{ai}}{\rho_{ao}} - 1.0\right) + \frac{4.0 f L_{aleta}\rho_{ai}}{Dh_a \rho_{med}}\right] \quad (32)$$

at where

$$\frac{1.0}{\rho_{med}} = \left(\frac{1.0}{\rho_{ai}} + \frac{1.0}{\rho_{ao}}\right) \quad (33)$$

The friction factor, f, is determined by Equation (2), obtained through the experimental values of Kays and London (1984), and the specific mass of air at the outlet of the heat exchanger, ρ_{ao}, can be estimated, initially, as a function of the air outlet temperature, using the equation:

$$\rho_{ao} = 1.28123142 - 0.004142716793 \mathrm{x} Ts_a + 1.921703199 x Ts_a{}^2 - 1.340288713 \mathrm{x} Ts_a{}^3 + 3.583356643 \mathrm{x} Ts_a{}^4 \quad (34)$$

The pressure loss determination procedure is also iterative and as complex as the one defined for determining the heat transfer rate. A very refined approximation to the specific mass of air at the heat exchanger outlet is required because the outlet temperature, a priori, is unknown.
The value obtained for the air outlet's specific mass, through the polynomial interpolation, Equation (34), allows the determination of approximate value for the pressure drop but does not initially represent the correct value admitted as a solution. The stopping criterion is also defined by comparing the result obtained for the experimental pressure drop within an error admitted as satisfactory.

III. RESULTS

The calculation methodology for determining the quantities considered in the analysis was presented above. Details related to the iterative procedures will be better defined throughout the discussions.
Table 3 presents a summary of the theoretical results obtained, which will be the object of analysis and discussion ahead.

Table 3 - Theoretical results obtained for the heat transfer rate.

m_a kg/s	Ts_a - °C	Ts_w - °C	$Tmed_a$ - °C	$Tmed_w$ - °C	$QK_{Teorico}$ W/K	$QK_{Experimental}$ W/K	Erro %
1.0	49.90	88.0	37.44	94.00	136.05	125.13	8.73
2.0	47.17	76.80	36.09	88.40	152.84	169.75	-9.96
3.0	42.82	67.31	33.91	83.65	188.60	209.55	-10.0
4.0	37.94	59.54	31.47	79.77	220.64	245.15	-10.0
5.0	33.14	53.29	29.07	76.64	249.52	277.18	-9.98
6.0	28.82	48.28	26.82	74.14	74.14	306.16	-10.0
7.0	24.55	44.26	24.78	72.13	72.13	332.58	-9.98

Figure 5 presents the interpolation results to determine the Colburn coefficient, J, for compact heat exchangers of the finned tubes type. The Colburn factor is critical for determining the heat transfer coefficient on the airside. The data used for interpolation were taken from Kays and London (1984; page 273).

Figure 6 presents the interpolation results for the average Nusselt number on the thermal input region's waterside. The determination of the Nusselt number is essential to determine the heat transfer coefficient on the waterside. The interpolation data was taken from the master's thesis of Nogueira, E. (1993; page 130).

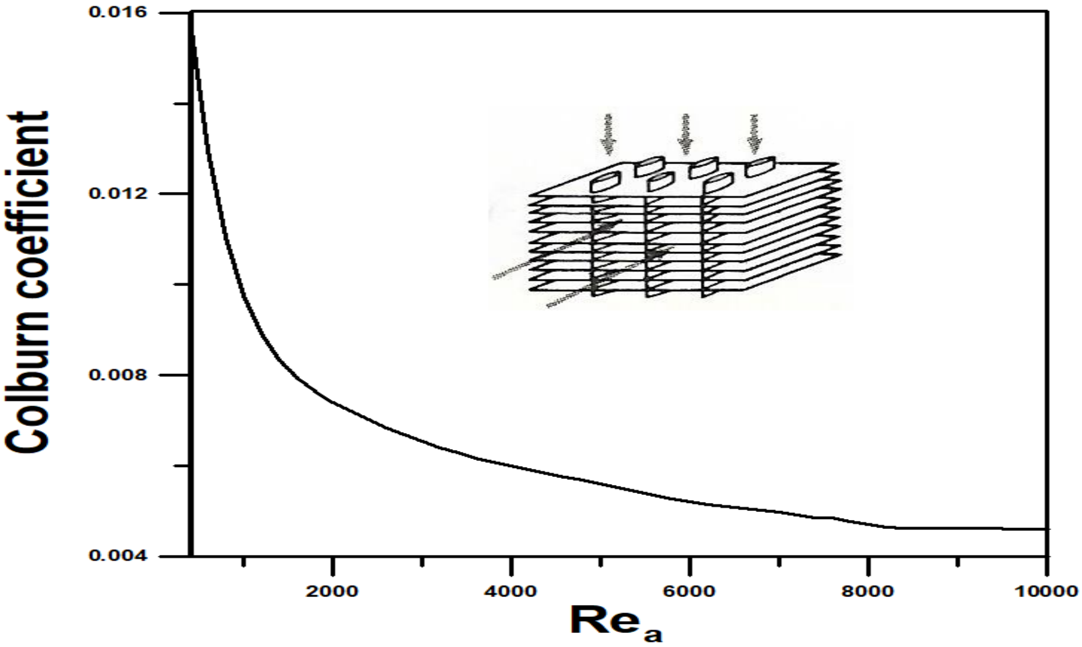

Figure 5 - Colburn's Coefficient, Equation 1, for compact finned tubes type heat exchangers.

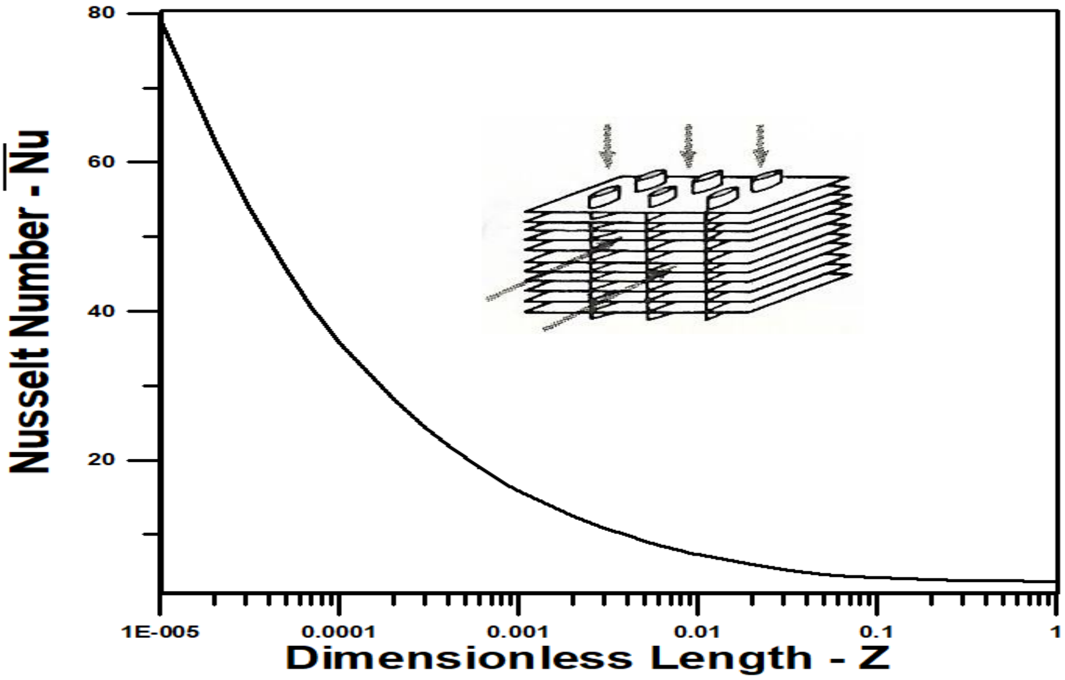

Figure 6 - Average Nusselt number, Equation 12, in the thermal inlet region for the duct of the circular, straight section in the laminar regime.

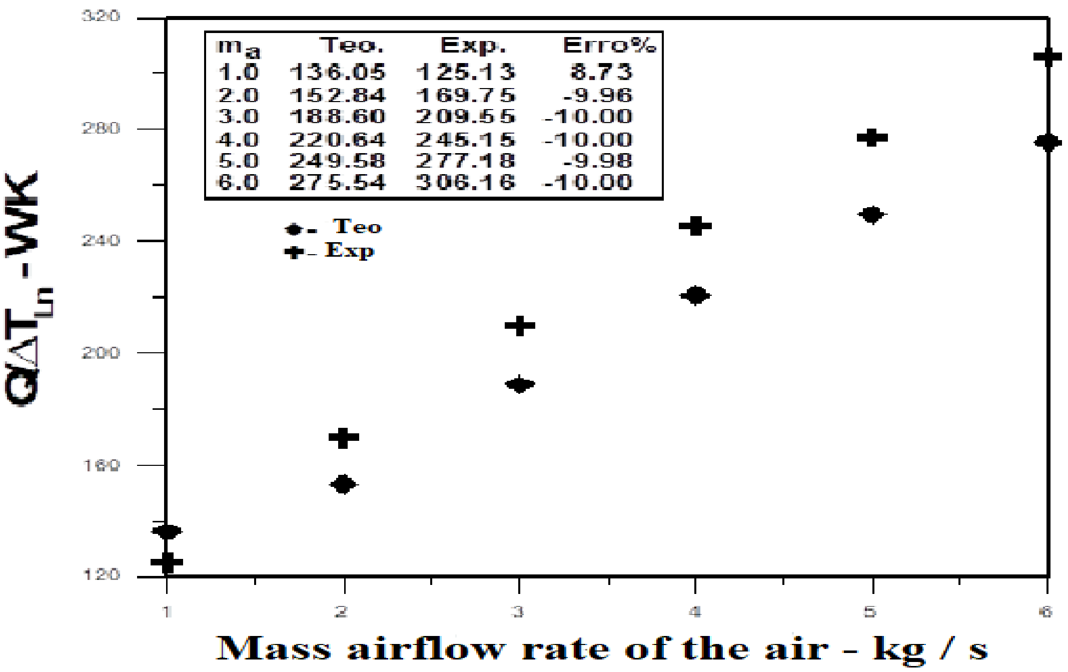

Figure 7 - Theoretical and experimental comparison of the heat transfer rate ratio to the mean logarithmic temperature on the airside.

Figure 7 presents theoretical and experimental results for determining the heat transfer rate in the analyzed heat exchanger (Automotive Radiator), presented in the master's dissertation by Ribeiro, L.N. (2007). All information related to the heat exchanger's geometry and physical properties' relevant parameters were taken from the dissertation. Some of them are shown in Tables 1 - 2 and Figure 3.

Some critical information for the characterization of the experiment are a) water inlet temperature in the tube equal to 100 ºC; b) Maximum water outlet temperature equal to 95 ºC; c) Average air inlet temperature admitted to being 25 ºC; d) Fixed water flow equal to 1.0 kg / s in all cases.

There is no analysis of errors in the master's dissertation, which were used for comparison, making it difficult to discuss further the theoretical versus experimental comparison obtained. However, within the range of airflow analyzed, in kg / s, a maximum error of about 10% can be observed in Figure 7 and Table 3. Considering that experimental errors in this type of experiment are high, it can be admitted that the comparison is in excellent agreement. The relevant fact is that the leaving water temperature is an input parameter for the simulation and must be defined a priori, to start the iterative procedure. It was used to reference the fact that for the lowest flow, the water leaving temperature was initially admitted as the maximum, which is 95 ºC.

In the absence of an empirical equation in the open literature for the heat transfer rate, the stopping criterion used was an error of less than 10% for the analyzed heat exchanger. When compared with the data provided by Ribeiro, L.N. (2007), since experimental errors of this order, or higher, are common in experiments that measure this type of physical quantity.

The author of the dissertation uses a wind tunnel from the automotive radiator's factory of Behr Brasil Ltda to obtain experimental data, and the analysis covers flow from 1.0 kg s to 19.0 kg/s for the air.

Figure 7 shows that the comparisons made have progressed until the flow of 6.0 kg / s since the iterative procedure is more costly for higher flows. Because parameters J and F do not apply for Reynolds number above 10,000. There are also other theoretical reasons, depending on the simulation performed, for the comparison to be made within the analyzed flow range (from 1.0 kg/s to 6.0 kg/s). In fact, according to the results presented in Figure 8, it can be observed that the temperature of the water outlet reaches a value close to the temperature of the air inlet, which is hypothetically equal to 25 ºC. For flow rates above 6.0 kg/s, see Table 3, the outlet temperature is below this value.

Leaving water temperatures well above 25 ºC, for mass flow equal to 6.0 kg/s, leads to theoretical results that are very far from the experimental reality. Above this flow range, it is difficult to justify the data obtained theoretically compared with the experimental data, unless the initial hypothesis of 25 ºC is changed as the air inlet temperature.

In a refrigerated environment, and thermally controlled, by hypothesis, a consistent theoretical analysis could be performed for higher flow rates and air inlet temperatures below 25 ºC.

Figure 9 shows an interpolation of the data presented by Kays and London (1984) for the friction factor in a compact finned tube heat exchanger. The friction factor presented is a fundamental factor in determining the heat exchanger's head loss on the airside. The data used for interpolation were taken from Kays and London (1984; page 273).

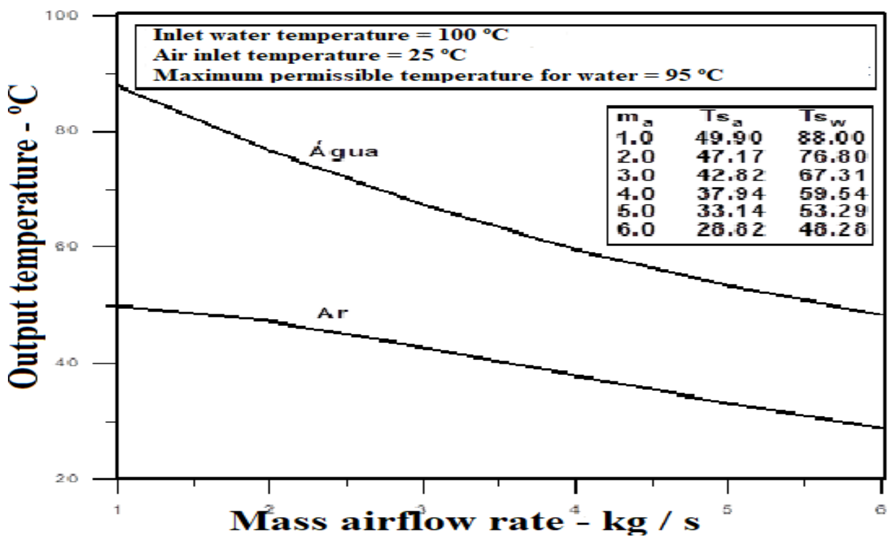

Figure 8 - Data for the water and air outlet temperature as a function of the air mass flow.

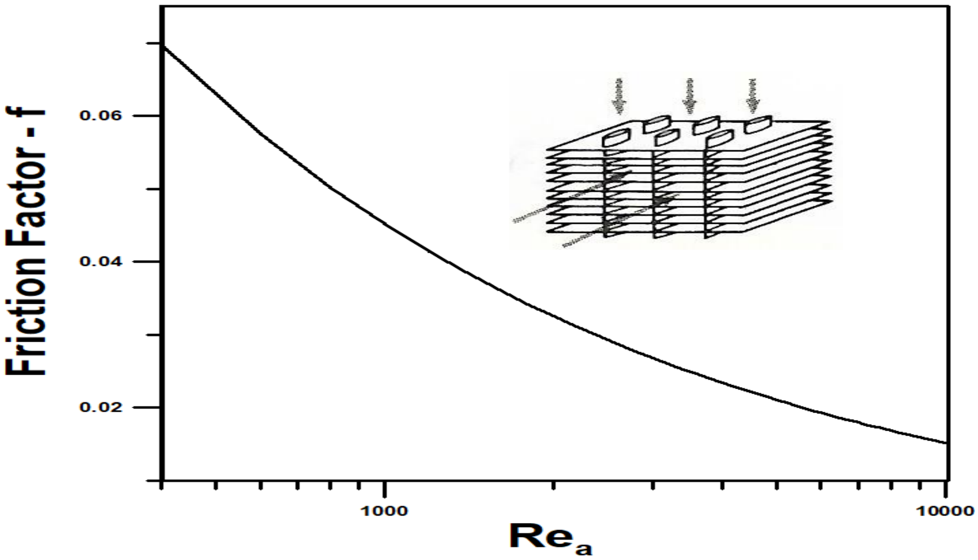

Figure 9 - Coefficient of Friction, Equation 2, for a compact heat exchanger of flat finned tubes versus Reynolds number for air.

The theoretical versus experimental comparison for airside head loss is shown in Figure 10. The comparison made is in excellent agreement with the experimental data when using the methodology described above in this work.

In this work's specific case, the result admitted as correct is that which results in comparison with the experimental data below 10% of error.

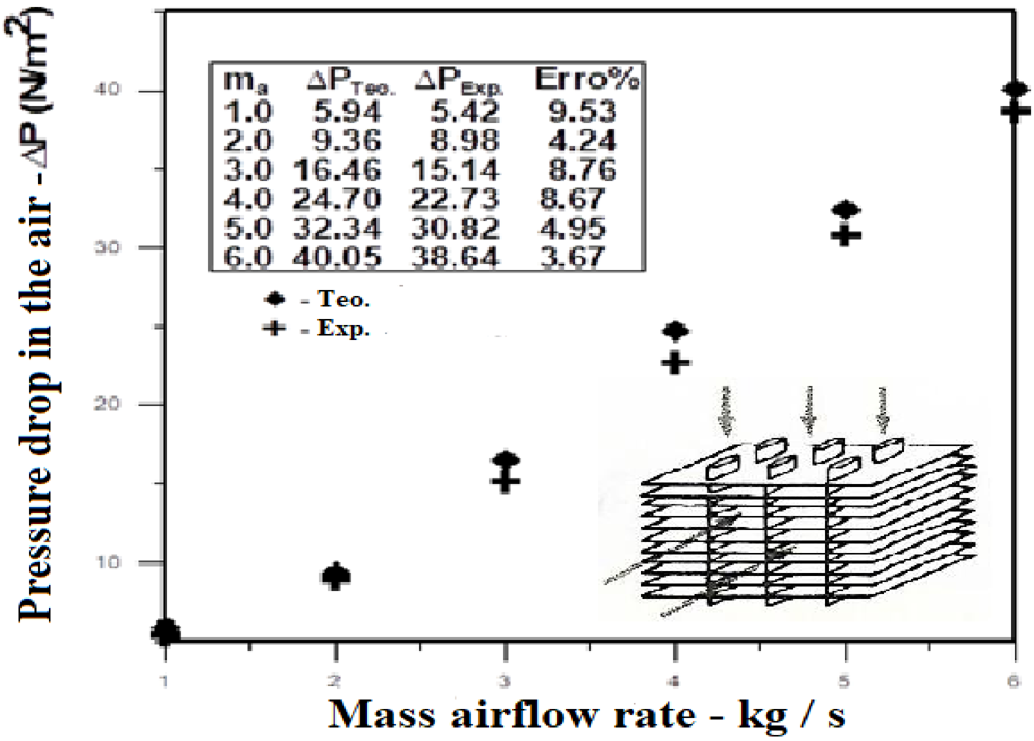

Figure 10 - Theoretical and experimental comparison for pressure loss, on the airside, in the flat finned tube heat exchanger.

IV. CONCLUSION

The analysis carried out makes it possible to create a solution methodology for flat finned tube heat exchangers. The methodology used makes it possible to determine the heat transfer rate and the pressure drop on the airside of a compact finned tube type heat exchanger (Automotive Radiator).

Concerning the experimental data of the master's thesis by Ribeiro, L.N. (2007), the comparisons made allow us to conclude that the model developed is consistent and well defined.

It is necessary to define a definitive stopping criterion for the model, both to the heat transfer rate and pressure drop. However, there is a lack of empirical equations in the open literature that are sufficiently general and represent a wide range of experimental results for the type of compact exchanger analyzed.

Regarding the aspect highlighted in the paragraph above, it is essential to highlight the Laboratory of Thermal Sciences of the Universidade Federal de Santa Catarina's effort. In a relatively recent period, they presented a master's thesis related to constructing a wind tunnel, with the specific objective of experimental characterization of automotive radiators [Pabón, NYL (2014)]. This high-cost enterprise underscores the importance of studies and analyzes carried out in this work.

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