International Journal For Research in
Applied Science and Engineering Technology

INTERNATIONAL JOURNAL FOR RESEARCH

IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY

Volume: 7 **Issue:** II **Month of publication:** February

DOI: http://doi.org/10.22214/ijraset.2019.2126

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 6.887
Volume 7 Issue II, Feb 2019- Available at www.ijraset.com

Compact Heat Exchanger in Automotive Radiators: Theoretical Versus Experimental Analysis of Coefficient of Friction and Colburn Factor

Élcio Nogueira[1], André Luís do Carmo Aroucha[2], Fernando Lamim Pereira[3],
[1]*Assistent Professor – FAT/UERJ*
[2]*Student in Production Engineering with emphasis in Mechanics – FAT/UERJ*
[3]*Student in Production Engineering with emphasis in Mechanics – FAT/UERJ*

Abstract: A theoretical-experimental analysis of a compact heat exchanger, type finned flat tube, used in automotive radiators is performed. It is a first approximation solution for the expansion of the application range of the Colburn Factor and Friction Coefficient in relation to the data available in the literature related to compact heat exchangers. The theory of effectiveness (ε-NUT) is used in comparison with experimental results obtained in the laboratories of automotive radiator factory of Behr in Lorena – SP, Brazil. The experimental data were extract from Master Dissertation [Ribeiro, 2017]. From the comparison between theoretical and experimental results, it has been proposed, through successive approximations, an expansion of the application range for the Coefficient of Friction and Colburn Factor, in relation to the results presented by Kays and London (1984).

Keywords: Compact heat exchanger; Automotive radiator; ε-NUT theory; Friction coefficient; Colburn factor.

I. INTRODUCTION

The requirement of increasingly technologically advanced automotive vehicles leads to the search for more efficient, lightweight and compact engines. The thermal control of the motors of these vehicles is obtained through automotive radiators, a class of compact heat exchangers Kakaç and Liu (2002). Research involving compact heat exchangers of all types, mainly car radiators, has been developed over the years and automotive companies invest high resources in all sorts of techniques that can optimize energy performance. As a result, new experimental apparatuses are built with the aim of improving automotive radiators through the incorporation of new technologies such as heat pipes and thermosyphons Pabón, N. Y. L. (2014).

Heat exchanger of low-weight and high-efficiency, flat plate type, has been used in automotive and aerospace applications. The heat transfer and the pressure drop in a compact heat exchanger are characterized in terms of the non-dimensional coefficients of Colburn, J, and Friction, f, as a function of the Reynolds number of the air for different types of heat exchange surfaces. Direct analytical determination of dimensionless parameters associated with heat transfer rate and pressure drop are extremely complex. The difficulty in this case is associated with the fact that the heat transfer coefficient and the friction factor are strongly dependent on geometric parameters such as fin height, fin spacing, fin thickness, and each type of heat exchanger needs to be characterized separately Alur, S. (2012), Mazumdar, S. (2007). The text by Kays and London (1984) provides an excellent introduction for the analysis of compact flat plate type heat exchangers, and contains heat transfer coefficient and friction factor data for various geometries. In fact, the authors provide J and f data for many different types of compact heat exchanger configurations. Although the results have been presented for more than three decades, they continue to be applied, since there is no data in the current open literature with such a wide range, due to the experimental difficulty in obtaining this type of information in academic research laboratories. Vehicle manufacturers have improved procedures and techniques to perform the thermal evaluation of radiators under real operating conditions. However, large corporations throughout the world, using compact heat exchangers, are not interested in presenting results obtained in their laboratories, since the data are strategic for survival.

Recently, Aroucha, A. L. C. and Pereira, F. L. (2019) presented a final monograph of the undergraduate course in Production Engineering with emphasis in Mechanics on compact heat exchangers of the types used in the automotive and aeronautical industries. The compact heat exchanger, type finned flat tube, was analyzed in greater depth, since it is widely used in automotive radiators. The analyzed radiator is popularly known as water radiator and is used in medium trucks with load capacities between 15 and 20 tons. The truck, by hypothesis, does not have turbocharger, which implies in the absence of air radiator, or "intercooler", and the air that enters the fins does it at room temperature. A theoretical model was developed to determine the rate of heat transfer and pressure drop through the application of the theory of effectiveness (ε-NUT). Theoretical versus experimental comparisons were carried out and the results obtained allow to conclude on the consistency of the simulations, assuming the assumptions and inputs, input temperatures and flow rates, defined as valid for the experiments. However, because vital information to close the theoretical model has been suppressed from the available experimental data, since it is strategic for the survival of the company producing the radiator, comparisons were made only for a range of air flows. In addition, one of the limitations found by the authors to make comparisons in elevated airflows in the radiator is that the range of Reynolds number values for the determination of the coefficient of friction and Colburn factor, as presented by Kays and London (1984), has a limit of 10^4.

Patel, S. and Deshnoskh, S. P. (2017) state that the demand for increased compactness and high performance of automotive radiators is increasing over the years, due to fuel economy and greater comfort for automotive users. Therefore, they say, it becomes important to optimize the process of cooling air through the radiator of the vehicle, without overloading the aerodynamic drag. Introduce an analytical procedure for design and dimensioning for heat exchangers and provide a starting point for future research on automotive cooling systems. They conclude that the key to an optimized design of a radiator lies in the following aspects: compactness, low pressure drops, low cost, durability, heat transfer and fluid flow.

Datil, V. R. et. al (2017) state that more powerful and smaller vehicles create problems in the dissipation of the heat exchanged in automotive radiators. They discuss the various different procedures recently used to optimize the thermal performance of ever smaller radiators. Among them, we have: 1 - Design of fins; 2 - Changes in the radiator core; 3 - Type of tube used; 4 - Changes in fin material; 5 - Use of different types of refrigerant fluids. They claim that nanofluid is a potential candidate for improved performance of automotive radiators, since the heat transfer coefficient of nanofluid is higher than that of water or water-ethylene glycol mixture. They conclude that reducing the size of the radiator reduces drag, increases fuel economy and reduces vehicle weight. In addition, they claim that the efficiency of the radiator increases with the insertion of heat pipes into the core.

Bharathi Mahanti, V. D. M., Ravi Kumar. (2017) present an experimental study using nanofluids (CuO-Water) as refrigerant for an automotive radiator. They conclude, mainly, that nanofluids increase the thermal performance of an automotive radiator.

Nagar, U. T. and Trivedi, B. M. (2017) state that the main aspect for the cooling of air and oil in an automotive vehicle is in the design of the radiator. They claim that nanofluids are being used for effective cooling, but that the main focus lies in defining the geometry of the radiator core, since circular radiator section requires less material than others. In addition, they conclude, the use of aluminum tubes decreases the total weight and increases the heat transfer rate of the radiator.

Lin, W. (2014) argue that graphite is a novel highly-conductive porous material for high power equipment cooling application, but aluminum and copper are still preferred for thermal management. However, when weight is a decisive factor, the introduction of material with lower density, high thermal conductivity and large contact area, graphite foam is the appropriate material.

According to Henrikson, L (2015) there is a great demand for the increased performance of automotive refrigeration systems. They state that there are a number of solutions to solve this type of demand, and for heavy vehicles the most reasonable way is to install heat exchangers positioned in different locations of the vehicle due to space constraints. These types of installations can already be seen nowadays, and where this occurs the air flow is not, as is often the case, necessarily perpendicular to the core of the heat exchanger. In this sense, performance evaluations are presented for standard compact heat exchangers, used in automotive vehicles, in non-perpendicular flow of air in relation to the core. The results were correlated with experiments to try to detect deviations and similarities. Saidi, M.L. et. al (2006) present an experimental study to determine the performance of radiators used in passenger vehicles. The effectiveness method (ε-NUT) was used to determine the Nusselt number, heat transfer coefficient and global coefficient of heat exchange. The trial error method was used to determine the quantities of interest, from the available experimental data. Thus, the Colburn factor and the coefficient of friction were estimated as results of the procedure. The utility of the presented method is justified because it provides empirical data that can be used in projects of compact heat exchangers.

II. OBJECTIVES

To perform an analysis on compact heat exchangers, with the ultimate objective of comparing theoretical and experimental results for finned flat tube type heat exchanger (Automotive Radiator). To establish comparisons between theoretical results obtained by the theory of effectiveness (ε-NUT) with experimental results of the literature, presented by Ribeiro, L.N. (2007) with emphasis on the rate of heat transfer and pressure drop. To obtain an expansion of the application range for the Coefficient of Friction and Colburn Factor, in relation to the results presented by Kays and London (1984).

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 6.887
Volume 7 Issue II, Feb 2019- Available at www.ijraset.com

III. METHODOLOGY

We propose an extension, as a first approximation, for the data presented by Kays and London (1984, Figure 01), regarding Colburn factor and coefficient of friction.

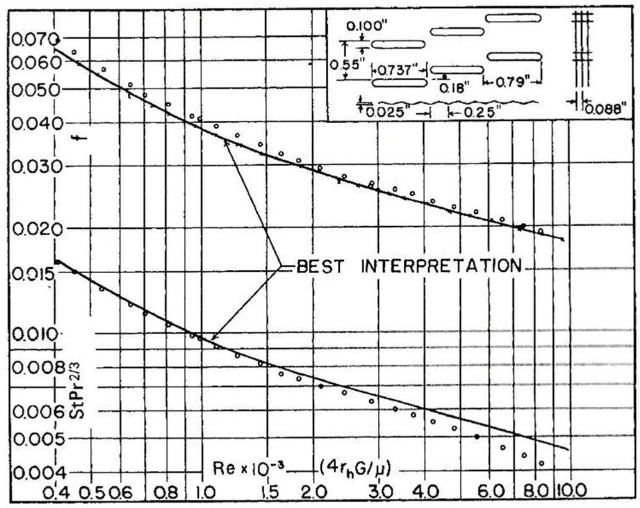


Figure 01 – Heat Transfer and Flow-Friction in Finned Flat Tubes

Kays and London (1984, page 273)

In situations where there is geometric, kinematic and dynamic similarity, the data of Figure 01 are used in applications where finned flat tube heat exchanger is used.

In order to compute the data available in Figure 01, in this work, interpolations were performed within the available Reynolds number range, the following equations were used:

$$J = 0.02619862019 - 3.626028274^{-5}Re_a + 3.16047951^{-8}Re_a^{\,2} - 1.568380435^{-11}Re_a^{\,3} + 4.633774581^{-15}Re_a^{\,4} - 8.056435353^{-19}Re_a^{\,5} + 7.225312269^{-23}Re_a^{\,6} - 8.564813179^{-28}Re_a^{\,7} - 4.452384011^{-31}Re_a^{\,8} + 3.807970933^{-35}Re_a^{\,9} - 1.015364192d^{-39}Re_a^{\,10} \qquad (01)$$

and

$$f = 1.199866203Re_a^{\,(-0.4747697361)} \qquad (02)$$

The model developed, through the procedures specified below, was used the experimental data of the dissertation on Radiator Automotive of Ribeiro, L.N. (2007), according to data from the sections below.

A. Radiator Characterization

The heat exchanger used for comparison between theoretical and experimental models is characterized and represented by Figures 02 - 03 and Tables 01 - 02, of physical dimensions and air properties:

Figure 02 - Compact water-air type heat exchanger used for cooling engines in cars and trucks

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 6.887
Volume 7 Issue II, Feb 2019- Available at www.ijraset.com

Özisik, M. N. (1990; page 450)

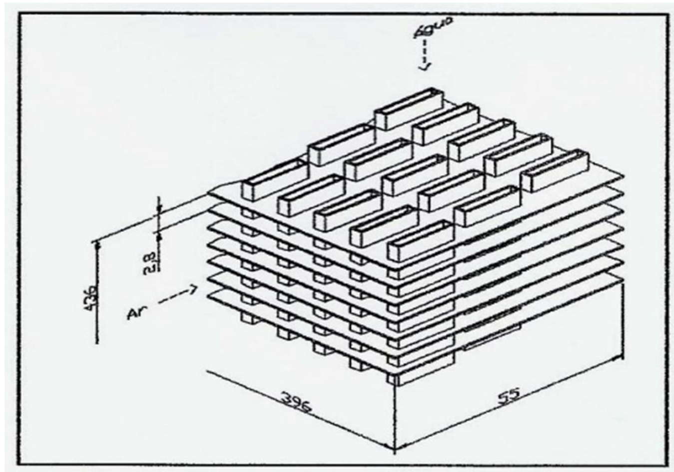


Figure 03 - Geometry of the heat exchanger of finned flat tubes.

Dissertation of Ribeiro, L.N. (2007; Behr Brasil Ltda.)

Table 01 - Dimensions of finned flat tubes 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

Dissertation of Ribeiro, L.N. (2007; Behr Brasil Ltda.)

Table 02 - 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

Dissertation of Ribeiro, L.N. (2007; Behr Brasil Ltda.)

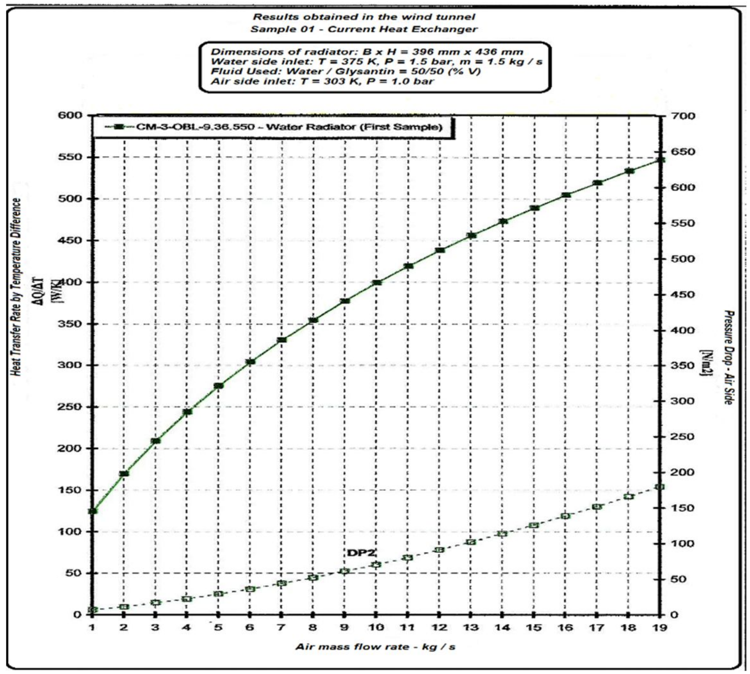


Figure 04 - Experimental value for heat transfer rate and pressure drop

Dissertation of Ribeiro, L.N. (2007, page 72)

The experimental data relevant to the comparisons, heat transfer rate and pressure drop were obtained by Ribeiro, L.N. (2007) in the wind tunnel of Behr Brasil Ltda and are represented by Figure 4.

Equivalent to that done for the dimensionless coefficients J and f, we performed interpolations of the above data and the approximated equations obtained are given by the equations:

$$QK_{exp} = 74.96904025 + 53.18082817m_a - 3.148106367m_a^2 + 0.131605291m_a^3 - 0.002308469675m_a^4 \quad (03)$$

and

$$\Delta P_{\text{Exp}} = 5.833333333 - 2.91045066dm_a + 2.769230769m_a^2 - 0.2812742813m_a^3 + 0.008741258741m_a^4 \quad (04)$$

QK_{exp} is the ratio between the experimental heat transfer rate and the Mean Logarithmic Temperature Difference – MLTD.

ΔP_{Exp} is the experimental loss of charge and m_a is the mass flow of air.

Theoretical analysis

B. Determination of Heat Transfer Rate

With the information defined above, it is possible to establish a theoretical, iterative procedure for the determination of the heat transfer rate of the radiator and the air-side pressure drop (which allows to define, a priori, the fan power at be used).

The theoretical determination of the heat transfer rate depends on the overall heat transfer coefficient, which in turn depends on the heat transfer coefficients, h_a and h_w, on the air side and the water side, respectively. To begin the calculations, it becomes necessary to determine the physical properties in function of the average temperatures of the fluids. However, the exit temperatures, in theory, are unknown a priori, and the average temperatures should be initially estimated.

With the initially stipulated output temperatures, defined physical properties, and the geometric quantities of the exchanger supplied, we have,

For the air:

$$G_a = \frac{m_a}{A_{min}} = \frac{m_a}{\sigma_a A_{fr}} \quad (05)$$

$$Re_a = \frac{G_a D_{ha}}{\mu_a} \quad (06)$$

$$J = \frac{h_a}{G_a c_{pa}} Pr_a^{2/3} \quad (07)$$

The Prandtl number for air, Pr_a, is obtained by interpolating the data, valid for air as the ideal gas, published by Çengel and Boles (2013, page 934):

$$Pr_a = 1.005351636d0 + 0.01292094145Tmed_a + 2.524174317^{-5}Tmed_a^2 - 5.074647769^{-8}Tmed_a^3 + 1.564763295^{-8}Tmed_a^4 \quad (08)$$

then,

$$h_a = J\frac{G_a c_{pa}}{Pr_a^{2/3}} \quad (09)$$

For water:

$$u_{máx} = \frac{m_w}{\rho_w A_{mín}} = \frac{m_w}{\rho_w \sigma_w A_{fr}} \quad (10)$$

$$Re_w = \frac{u_{máx}\, D_{hw}}{v_w} \quad (11)$$

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

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

If the flow regime in the water is laminar, it is used to interpolate the data of the Master Thesis of Nogueira, E. (1988, page 130), for the thermal input region under development:

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

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

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

$$Nu = 5.261d0 - 19.93019048Z_w + 139.4921627Z_w^2 - 605.9954034Z_w^3 + 1716.100694Z_w^4 - 3217.96875Z_w^5 + 3954.86111Z_w^6 - 3056.051587Z_w^7 + 1344.246031Z_w^8 - 256.2830687Z_w^9;\ 10^{-1} \le Z_w = 10^0 \quad (12.4)$$

Then, we have:

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

The overall heat transfer coefficient is obtained in relation to the air exchange area and, in order to perform the calculations, it is necessary to determine the efficiency of the fin since there is variation of temperature between the entrance of the plate of the exchanger (base of the fin) and its outlet:

$$\eta = \frac{tg\mathrm{h}(mL)}{mL} \quad (14)$$

where

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

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

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

where

$$\beta = \frac{fin\ area}{total\ area} \quad (17)$$

Then, we have:

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

where

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

and

$$\frac{A_w}{A_a} = \frac{water\ side\ heat\ transfer\ area}{air\ side\ heat\ transfer\ area} \quad (20)$$

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

$$N = NTU = \frac{A_a U_a}{C_{mín}} \tag{21}$$

The thermal capacities of air and water are calculated by:

$$Ca = m_a * Cp_a \tag{22}$$

and

$$Cw = m_w * Cp_w \tag{23}$$

C_{min} is the lowest value between the thermal capacities of water and air. Finally,

$$Q = \varepsilon C_{mín}\left(T_{h,af} - T_{c,af}\right) \tag{24}$$

$$\Delta T_{Ln} = \frac{Q}{U_a A_{total}} \tag{25}$$

and

$$QK_{teo} = \frac{Q}{\Delta T_{Ln}} \tag{26}$$

where

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

according to Kakaç, S. (1991, page 35).

QK_{teo} is the theoretical value for the ratio between the heat transfer rate in the air and the mean logarithmic temperature difference – MLTD.

With the heat transfer rate determined, as the first approximation, one can calculate the air and water exit temperatures, through the energy balance equations:

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

and

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

The average outlet, air and water temperatures can then be determined and compared to the initially set temperatures:
The mean air and water temperatures can then be determined and compared to the initially defined temperatures:

$$T_{m,a} = \frac{T_{a,af} + T_{a,ef}}{2} \tag{30}$$

and

$$T_{m,w} = \frac{T_{a,wf} + T_{a,wf}}{2} \tag{31}$$

With the average temperatures finally calculated, the values obtained for the heat transfer rate were compared and, if they are outside of an admissible value, when compared with experimental values or empirical expressions, the calculations for thermophysical properties can be re-started, until a satisfactory convergence is obtained for the problem.

C. Determination Of Air Pressure Drop

In the calculation of pressure drop in a finned heat exchanger the main losses are related to the friction factor (f), and the air-side pressure drop can be determined by [Kakaç, S. (2002)]:

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

where

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

The friction factor, f, is determined by equation 02, 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 outlet temperature of the air, through the equation:

$$\rho_{ao} = 1.28123142 - 0.004142716793xT_{oa} + 1.921703199xTo_a{}^2 - 1.340288713xT_{oa}{}^3 + 3.583356643xT_{oa}{}^4 \tag{34}$$

The pressure drop determination procedure is also iterative and as complex as that defined for determining the heat transfer rate, since a very refined approximation to the value of the specific mass of the air at the heat exchanger outlet is required.

The value obtained for the specific mass of the air outlet, through the polynomial interpolation, equation 34, allows the determination of an approximate value for the pressure drop, but does not initially represent the correct value accepted as solution. The stopping criterion, also here, is defined by comparing the result obtained for the experimental pressure drop, within an acceptable error.

IV. RESULTS AND DISCUSSION

The theoretical analysis presented in the previous section represents the procedure developed and applied by Aroucha, A.L.C. and Pereira, F.L. (2019). It was possible to obtain theoretical results comparable to the experimental data of Ribeiro, L.N. (2007), within a percentage error range of less than 10%, for relatively low air mass flow rates.

The authors found, however, difficulties in obtaining physically consistent theoretical results for relatively high mass flows, above 10 kg/s for air.

The inconsistencies found for relatively high mass flows into the air were justified by the lack of experimental information, since the author of the experiments does not present analysis of errors or additional information to those presented through Figures 02, 03 and 04 and Tables 01 and 02, which could contribute to the closure of the theoretical model. Such a lack of information must be credited to the fact that it is a company building radiators and experimental data may be relevant to its survival.

Assuming that the data provided for the heat transfer rate and pressure drop are correct, with relatively low experimental errors, as expected from data from a company that builds radiators, it is possible to advance in the analysis of the theoretical model and propose a more refined, and that allows satisfactory results for high flow rates. This is, therefore, intended with the results and discussions presented below.

Finally, the results presented are generalizations obtained by Aroucha, A.L.C. and Pereira, F.L. (2019), with a slightly different procedure to that recommended by them, and that allows to broaden the range of flows applicable by the theoretical model.

It should be noted that the application of the generalized theoretical model, as discussed below, allows to broaden, as the first approximation, the range of application of the Colburn Factor and the Friction Coefficient, in relation to the data presented by Kays and London (Figure 01).

As previously pointed out, the Colburn factor is fundamental for determining the coefficient of heat transfer on the air side. The data used for interpolation were taken from Kays and London (1984, page 273) for Reynolds number between 400 and 10^4.

Some important basic characteristics for the characterization of the experiment carried out by Ribeiro, L.N. (2007; Behr Brasil Ltda.) are: a) Water inlet temperature in the tube is equal to 100 ºC; b) Maximum outlet temperature of the water in the pipe equal to 95 ºC; c) Average air temperature is allowed to be 25 ° C; d) Water mass flow fixed and equal to 1.5 kg / s in all cases.

Aroucha, A.L.C. and Pereira, F.L. (2019) adopt, for the beginning of the iterative procedure of approximation and theoretical determination of results, the initial temperatures of exit for water and air. The limitations in this case are the maximum water outlet temperatures, equal to 95 ° C, and the air outlet temperature, equal to 25 ° C.

Assuming the maximum water outlet temperature for the lowest air mass flow rate, equal to 1.0 kg / s, Aroucha, A.L.C. and Pereira, F.L. (2019) find that the minimum percentage error admitted between the experimental and theoretical values should be greater than 10%. Otherwise, the iterative procedure does not converge.

From the result described above, they assume a minimum percentage error, for all air mass flow rates, equal to 10% and obtain convergence for the model.

However, when applying the procedure for these conditions, the water outlet temperatures, for mass air flows above 6.0 kg / s, are much lower than 25º, which leads to a physical inconsistency for the conditions allowed for the experiments. From these results, they conclude: the theoretical model developed does not work for high air mass flow rates.

However, as we will see from the results presented below, one of the limitations to the operation of the presented model, in every air mass flow rate analyzed experimentally, lies in the fact that the Reynolds number is greater than 10^4 for mass flow rates of more than 7,0 kg / s. For values of the Reynolds number greater than 10^4, as expected, the Colburn Factor and Friction Coefficient, as obtained by Kays and London (1984), do not apply. Of course, in these cases, the equations 01 and 02 present incompatible results with experiments and the errors are unpredictable.

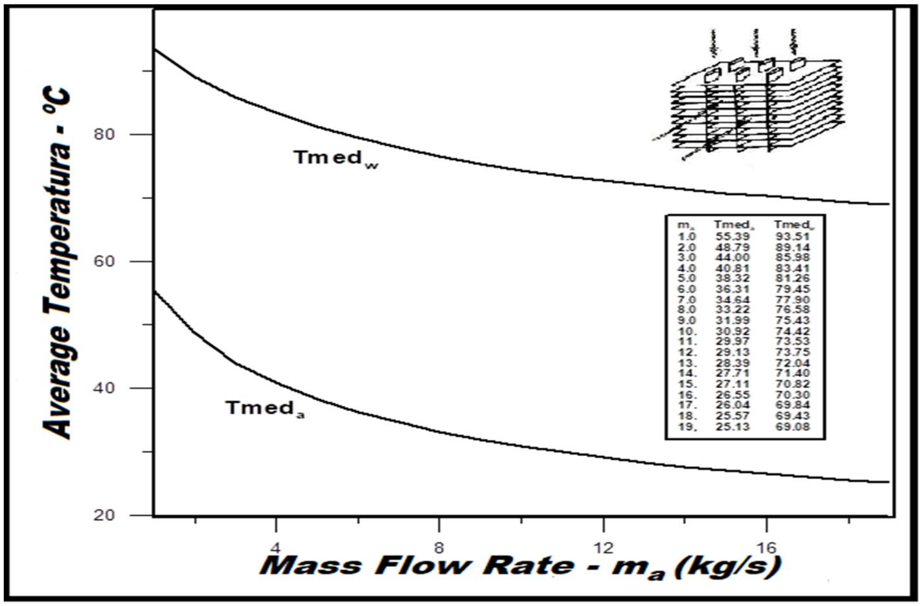


Figure 05 - Average temperatures for air and water

The results presented above, Figure 05, demonstrate that the procedure adopted in this work allows to advance to high flows and obtain results compatible with what is expected physically. Note that the air outlet temperature is slightly above 25 ° for the maximum mass flow of air analyzed experimentally, equal to 19.0 kg / s.

Equivalently, the results for the air and water outlet temperatures, Figure 06 below, are physically consistent for the entire mass flow rate of air analyzed.

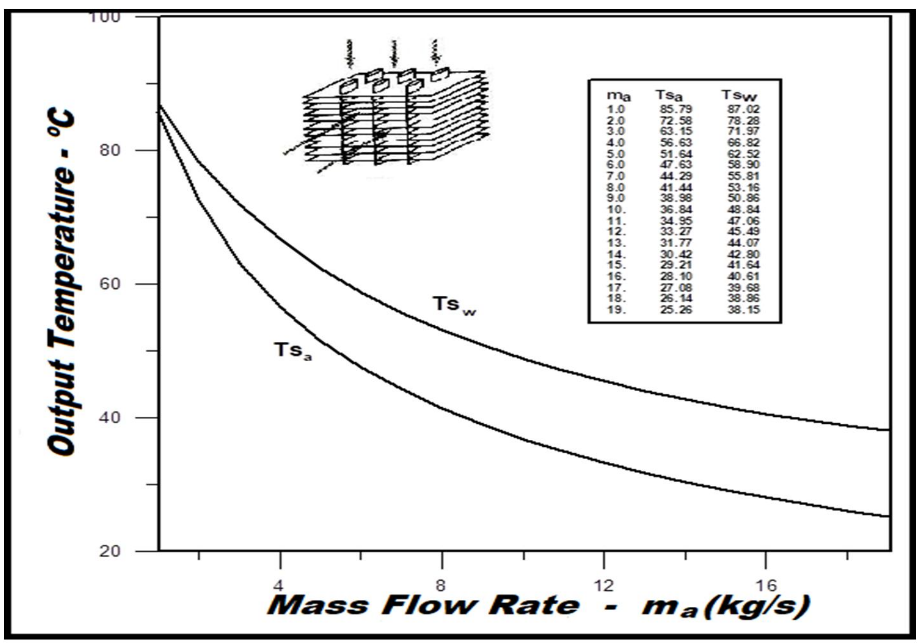


Figure 06 - Outlet temperatures for air and water

The results presented through Figure 07 present values for the experimental heat transfer rates and heat transfer rate for the theoretical model. When analyzing the experimental errors presented in the figure it can be seen that in practically the whole range the percentage errors are below 0.1%. In fact, these errors were admitted a priori, and are the key to the present procedure, which admits very low experimental error for the data obtained, once, as already pointed out, were obtained in a wind tunnel of a radiator factory. In this sense, assuming the experimental values as correct, an additional procedure to that already discussed for the theoretical model allows a balance between experimental and theoretical results.

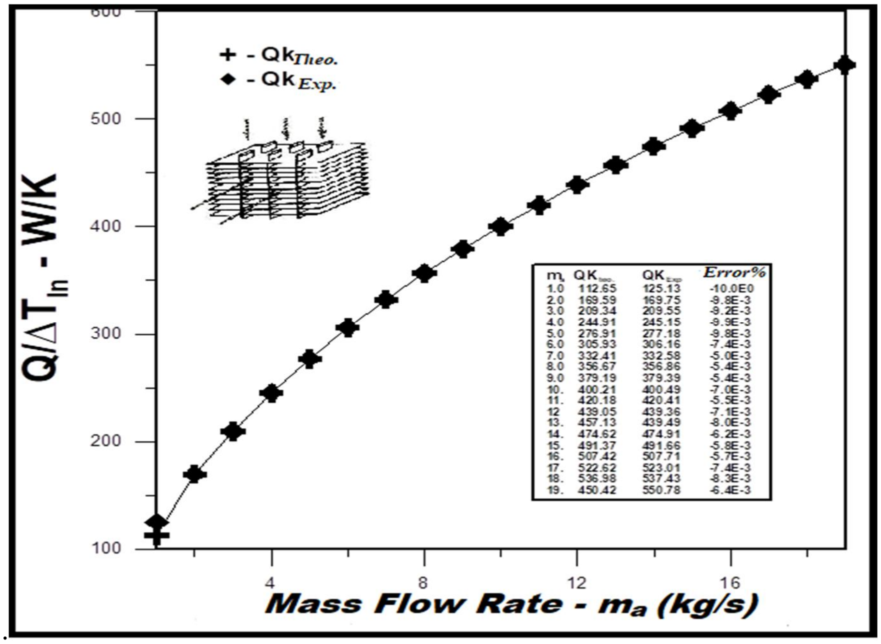


Figure 07 - Theoretical versus experimental heat transfer rate

The solution key, in this case, is in an iterative procedure for the Colburn Factor, for Reynolds numbers greater than 10^4. The value obtained, iteratively, for the Colburn Factor makes it possible to advance the theoretical model to high mass flow rates air.

It should be noted, however, that for air mass flow rate of 1.0 kg / s the minimum error allowed to close the model is 10%. In this case, for errors lower than this value the water outlet temperature is higher than that admitted in the experiment. We assume, in this situation, that the experimental error is high, since the flow is relatively low and the measurements performed have lower resolutions.

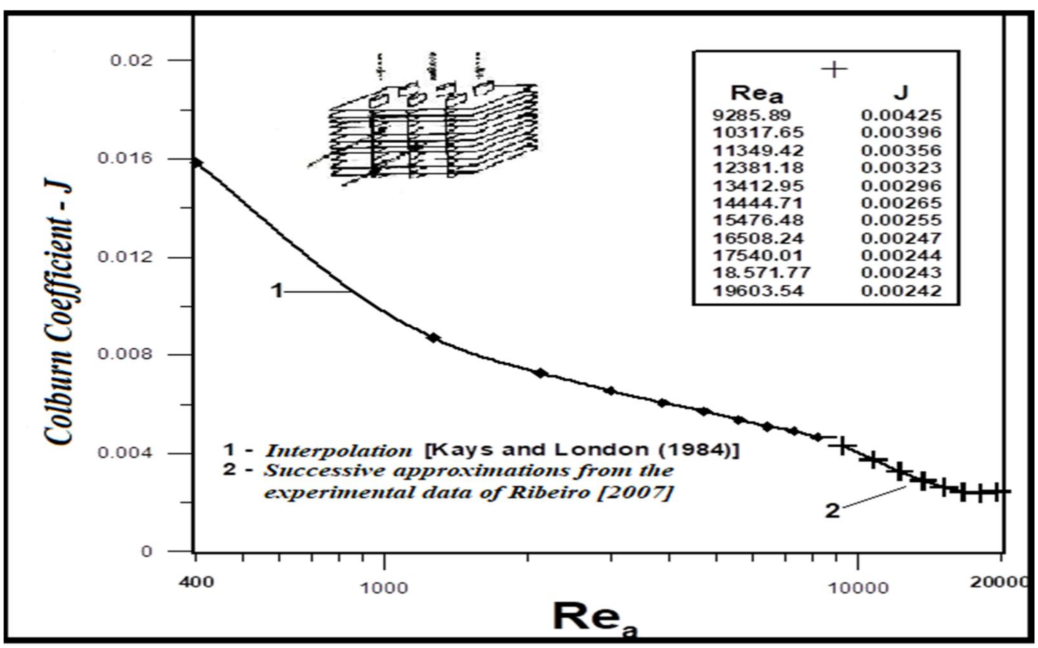


Figure 08 - Colburn Factor versus Reynolds number

Figure 8 shows results for Colburn Factor in all air mass flow rates analyzed experimentally by Ribeiro, L.N. (2007; Behr Brasil Ltda.). For Reynolds numbers greater than 10^4, we recommend, through the theoretical procedure, the method of successive approximations to obtain the values of **J**. The results obtained in this case are tabulated and indicated by index 2.

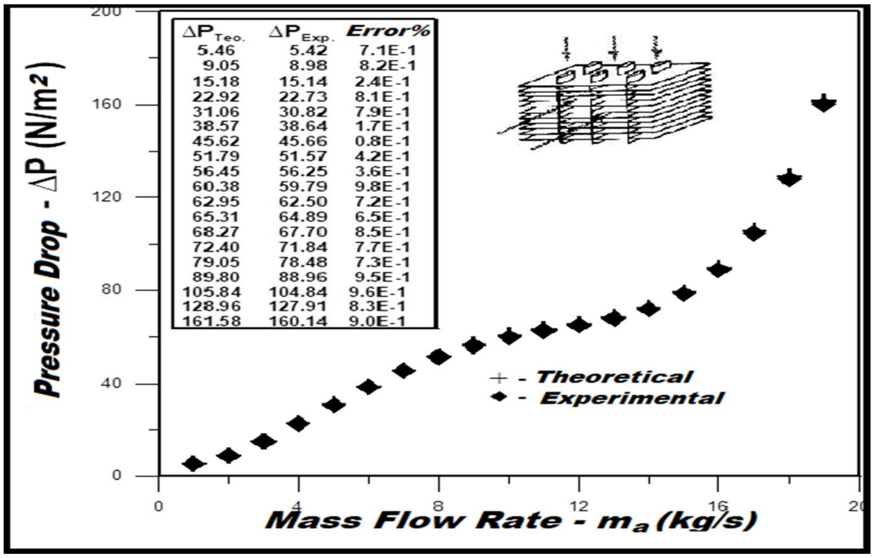


Figure 09 – Pressure drop versus mass flow rate of the air

As previously pointed out, the Friction Coefficient is fundamental for determining the pressure drop on the air side. The data used for interpolation were taken from Kays and London (1984, page 273) for Reynolds number between 400 and 10^4.

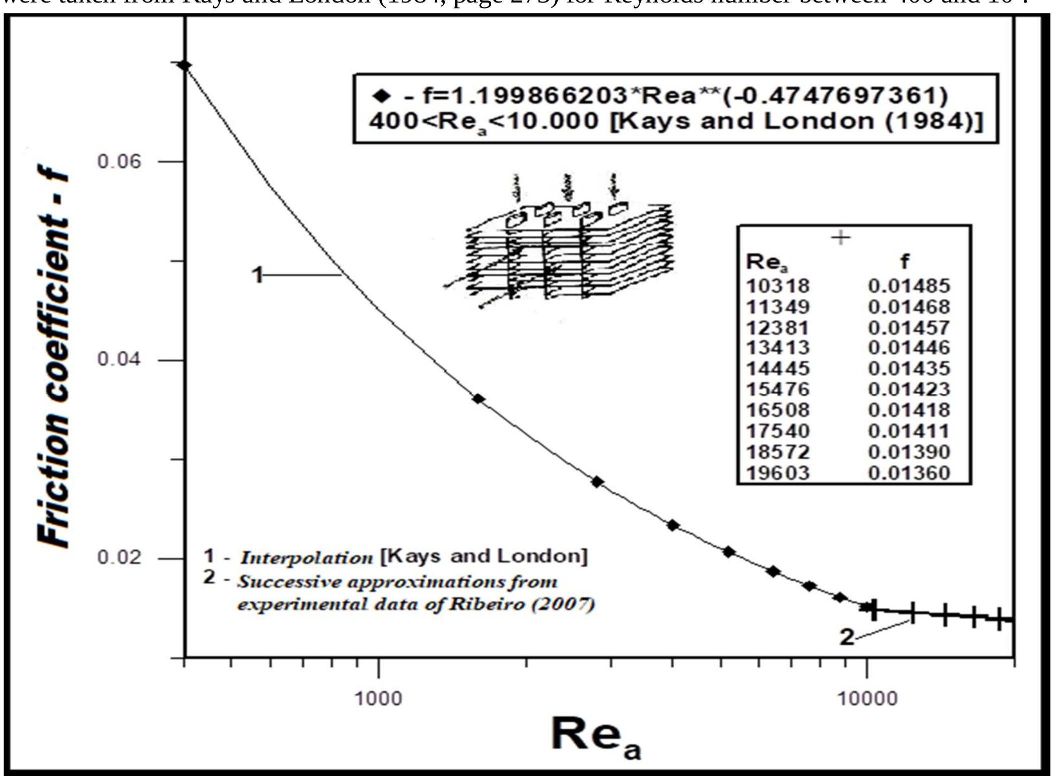


Figure 10 – Friction Coefficient versus Reynolds number

Figure 9 presents the results obtained for pressure drop as a function of the Reynolds number, in every air mass flow rate analyzed experimentally. It was assumed, equivalent to the heat transfer rate, that the experimental data are correct, and that experimental errors are negligible. The errors between the theoretical and experimental results are all below 0.1%.

Figure 10 shows results for Friction Coefficient in all air mass flow rates analyzed experimentally by Ribeiro, L.N. (2007; Behr Brasil Ltda.). For Reynolds numbers greater than 10^4, we recommend, through the theoretical procedure, the method of successive approximations to obtain the values of **f**. The results obtained in this case are tabulated and indicated by index 2.

V. CONCLUSION

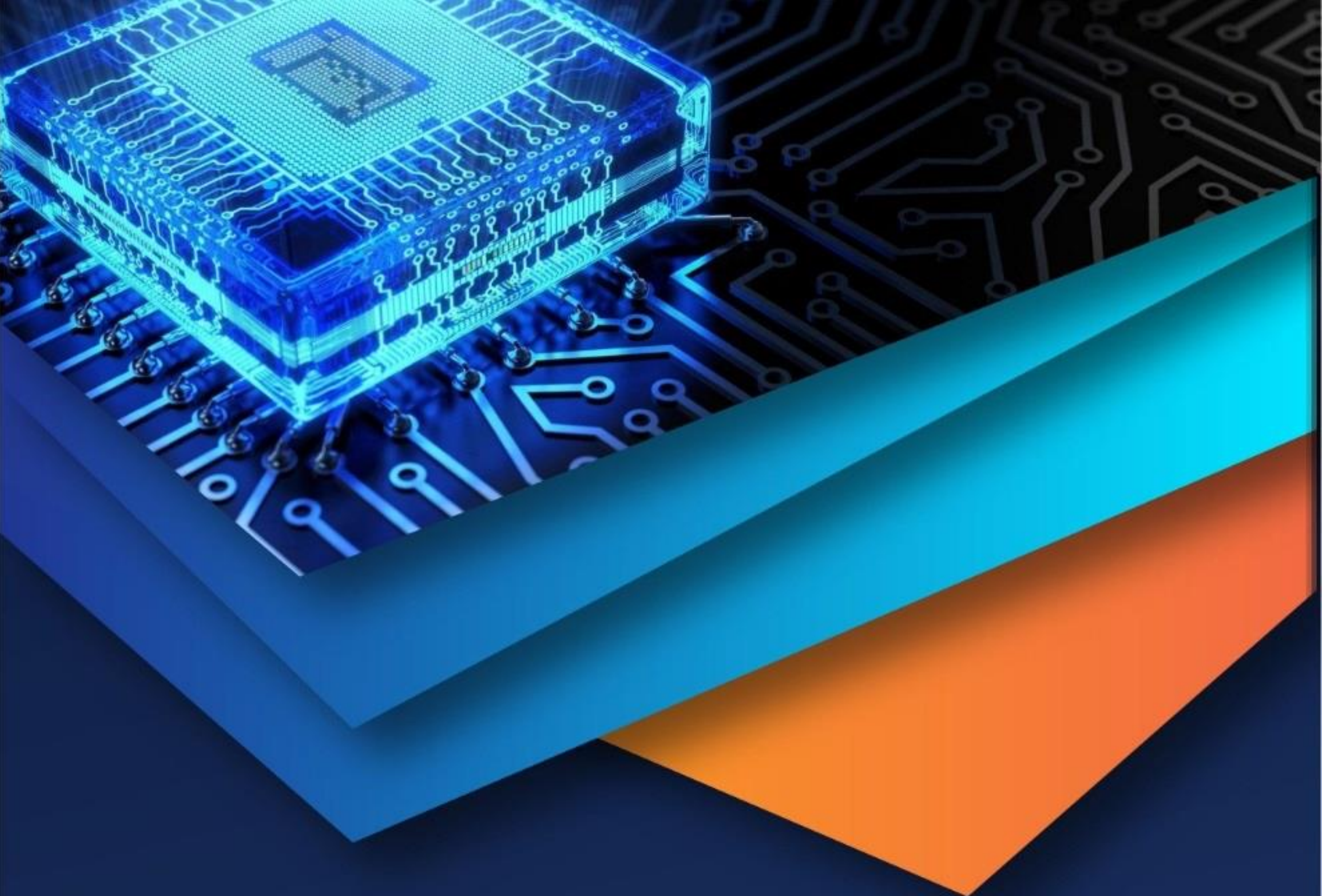
Experimental results obtained for finned flat-tube type heat exchanger, and results calculated using a theoretical model based on the theory of effectiveness (ε-NUT), were used to determine an expansion of the application range of the Colburn Factor and Friction Coefficient, in relation to the Reynolds number range reported in the literature on compact heat exchangers.

The theoretical method presented was used in a wide range of mass flows rate to the air and it was possible, after iterative procedure to determine the air and water exit temperatures, and the application of the method of successive approximations applied to the Colburn Factor and Coefficient of Friction, the determination of the heat transfer rate and the pressure drop on the air side with percentage errors below 0.1%.

It is, therefore, a proposal for a solution, in the first approximation, for the expansion of the application range of the Colburn Factor and Friction Coefficient. For a better characterization of the proposed expansion it becomes essential to compare with new reliable experimental results from the literature on compact heat exchangers.

REFERENCES

[1] Alur, S. “Experimental Studies on Plate Fin Heat Exchangers”. A Thesis Submitted for Award of Degree of Doctor of Philosophy. Mechanical Engineering Department, National Institute of Technology, Rourkela, 2012.
[2] Aroucha, A.L.C. and Pereira, F.L. "Study and theoretical and experimental analysis of a compact heat exchanger type finned tube (Radiator Automotive)". Graduation Final Project by the Course of Production Engineering with emphasis in Mechanics, Faculty of Technology of the State University of Rio de Janeiro - FAT / UERJ. Advisor: Prof. Dr. Élcio Nogueira, Resende, Brazil, 2019.
[3] Bharathi Mahanti, V. D. M., Ravi Kumar, V. “STUDY ON THE BEHAVIOUR OF CUO NANO PARTICLES IN RADIATOR HEAT EXCHANGER FOR AUTO MOBILES” International Journal of Advances in Engineering & Technology. ©IJAET ISSN: 22311963, 344 Vol. 10, Issue 3, pp. 344-353, June, 2017.
[4] Çengel, Y. A., Boles, M. A. "Thermodynamics". AMGH Editora Ltda, Porto Alegre, RS, 2013.
[5] Henrikson, L. “Performance of Compact Heat Exchanger in Non-Perpendicular Cooling Airflows” Thesis for the Degree of Doctor of Philosophy, Department of Applied Mechanics, Chalmers University of Technology Gothenburg, Sweden, 2015.
[6] Kakaç, S. “Boilers, Evaporators, and Condensers”. John Wiley & Sons, INC. New York, 1991.
[7] Kakaç, S., Liu, H. “Heat Exchangers: selection, rating, and thermal design”. 2nd ed., CRC Press Book Company, 2002.
[8] Kays, W. M., London, A. L. “Compact Heat Exchangers”. McGraw-Hill, New York, 1984.
[9] Lin, W. “Heat Exchangers for Heavy Vehicles”. Academic thesis, by due permission of the Faculty of Engineering at Lund University, 2014.
[10] Mazumdar, S., Singh, R. “CFD Studies on Flow Through Compact Heat Exchangers”. A Thesis Submitted in Partial Fulfillment of Requirements for Degree of Bachelor of Technology, under Guidance of Prof. Kumar Sarangi, Mechanical Engineering Department, National Institute of Technology, Rourkela, 2007.
[11] Nagar, U. T. and Trivedi, B. M. “PERFORMANCE ANALYSIS AND DESIGN OF AUTOMOBILE RADIATOR”. International Journal of Advance Engineering and Research Development Volume 4, Issue 11, November -2017.
[12] Nogueira, E. "Laminar Flow and Heat Transfer in Non-Stratified Immiscible Fluids". Thesis presented to the Postgraduate Division of the Technological Institute of Aeronautics – ITA, as part of the requirements for obtaining the Master's Degree in Mechanical Engineering, in the Aeronautics, Propulsion and Energy area of the Aeronautical and Mechanical Engineering Course. Advisor: Prof. Dr. Renato Machado Cotta, São José dos Campos, Brazil, 1988.
[13] Özisik, M. N. "HEAT TRANSFER - A Basic Approch". Publisher Guanabara Koogan S.A., Rio de Janeiro, 1990.
[14] Pabón, N. Y. L. "Design and Fabrication of a Wind Tunnel and Thermal Characterization of an Automotive Radiator". Dissertation submitted to the Post-Graduation Program in Mechanical Engineering of the Federal University of Santa Catarina to obtain the Master's Degree in Mechanical Engineering. Advisor: Profa. Dra. Márcia B. H. Mantelli. Florianópolis, Brazil, 2014.
[15] Patil, V. R. et. al “Review of Problems of Heat Transfer in Car Radiator and Suggested Solutions”. International Journal of Scientific Development and Research (IJSDR) www.ijsdr.org, IJSDR | Volume 2, Issue 1 (2017).
[16] Ribeiro, L. N. "Optimization of Flat Fin Compact Heat Exchanger through Limit Layer Analysis". Dissertation defended with the Department of Mechanical Engineering of Taubaté - UNITAU, as part of the requirements to obtain the Master's Degree by the Graduate Program in Mechanical Engineering. Advisor: Prof. Dr. Sebastião Cardoso, Taubaté, Brazil, 2007.
[17] Saidi, M.L. et. Al. “Experimental Study of Thermal Performance and Pressure Drop in Compact Heat Exchanger Installed in Automotive”. ASME Internal Combustion Engine Division, Spring Technical Conference May 8-10, 2006, Aachen, Germany, 2006.
[18] Stone, K. M. “Review of Literature on Heat Transfer Enhancement in Compact Heat Exchangers”. Air Conditioning and Refrigeration Center, University of Illinois, Urbana, 1996.



10.22214/IJRASET

INDEX COPERNICUS 45.98

IMPACT FACTOR: 7.129

TOGETHER WE REACH THE GOAL

IMPACT FACTOR: 7.429

INTERNATIONAL JOURNAL FOR RESEARCH

IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY