25th ABCM International Congress of Mechanical Engineering
October 20-25, 2019, Uberlândia, MG, Brazil

COB-2019-0658
STUDY AND DIMENSIONING OF A COIL TYPE HEAT EXCHANGER IN A VEHICULAR PNEUMATIC BRAKE SYSTEM

Carlos Henrique de Paula Junior, Student of Mechanical Engineering – UERJ; chpj_1993@hotmail.com
Luiz Carlos Cordeiro Junior, Professor UERJ; luiz.cordeiro@fat.uerj.br;
Universidade do Estado do Rio de Janeiro; Rod. Pres. Dutra Km 298 – Polo Industrial – Resende – RJ – CEP: 27.537-00

ABSTRACT. The present work aims a thermal analysis in a coil type heat exchanger, used in a vehicle with pneumatic brake system. In this analysis, the thermal mapping was constructed with the vehicle in the static and dynamic conditions, within the facilities of the Mechanics Laboratory / UERJ / Unit Resende. The analysis of calories by means of the global coefficient of heat transfer, natural and forced convection, as well as the external and internal patterns of linear and helical transfer (coil) were considered in these studies. We also considered analyzes by logarithmic mean method of temperature differences (MLDT) and the ε-Nut effectiveness method. The coil monitoring conditions were established with a continuous internal compressed air flow through the motor compressor, and the external air flow is simulated by a centrifugal fan with pre-defined speeds for analysis. For study purposes, atmospheric air was considered ideal, neglecting the possible thermal effects generated by the contaminants (water and oil) and or incrustations inside the heat exchanger piping. The result of this research served to better understand the design, the application of this heat exchanger, the influence of the geometric effects in its construction for thermal exchange.

Keywords: *heat exchanger, coil designer and vehicular pneumatic brake.*

1. INTRODUCTION

Within area of thermal engineering, heat exchangers are of great importance in the development of studies and research for process optimization, which drives the improvement of existing techniques, as well as the development of new technologies that culminate in ever increasing products and solutions. more efficient.

In the study of heat exchangers, there are several methods that have been used to analyze the performance of heat exchangers. Among them, the logarithmic mean temperature difference method (LMTD) can be highlighted in the literature. For use in the analysis of heat exchangers, when the inlet and outlet temperatures of both fluids are known, the mass flow rates, heat transfer coefficient and heat transfer area of the heat exchanger can be determined.(Incropera et al., 2008), furthermore, the ε-NUT Effectiveness method is recommended in situations where it is desired to determine heat transfer rate and hot and cold fluid outlet temperatures for fluid mass flow rates and inlet temperatures. The type of heat exchanger and its size are being specified. What is sought is to determine the heat transfer performance of a given heat exchanger or to determine if an available heat exchanger can be used. (Çengel and Ghajar, 2012).

The present work deals with a thermal analysis and design of a coil type heat exchanger, by dimensioning the linear length and the number of volutes necessary for the cooling of the compressed air of a vehicle brake system. It's important to point out that this model of heat exchanger and application is chosen by each assembler and can be used inside the chassis of the vehicle or in other cases mounted on the outside of this chassis. In this evaluation, it will be presented comparatively analytical model and data experimentally obtained in natural and forced convection, through four coil models built in different materials (steel and copper) and different geometries, all mounted internally to the chassis frame. For the purpose of heat exchanger analysis, control volumes were defined between the pipe segments in the horizontal and helical coil parts.

2. METODOLOGY

2.1 Experimental characteristics

The experiment was developed in a functional truck chassis whose heat exchanger mounting configurations can be easily adapted to the different geometries and materials of a coil heat exchanger, as shown in Figures 1 to 4, with their respective control volumes.

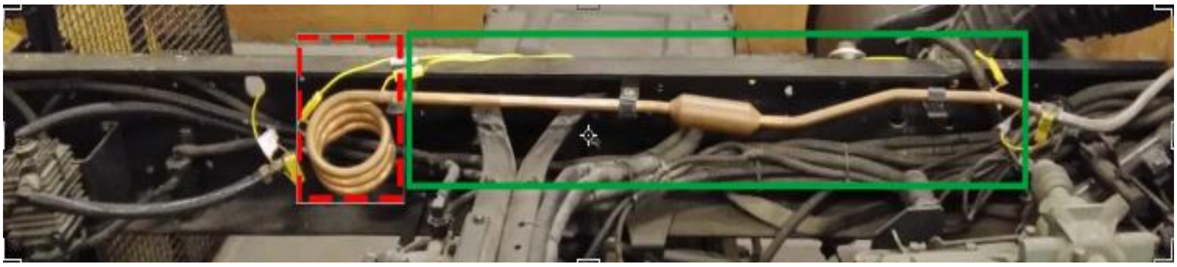

Figure 1. Representation of the copper heat exchanger with the linear control volumes (continuous line) and control volume of the 3-volute "coil" (dashed line). (Source: Prepared by the author).

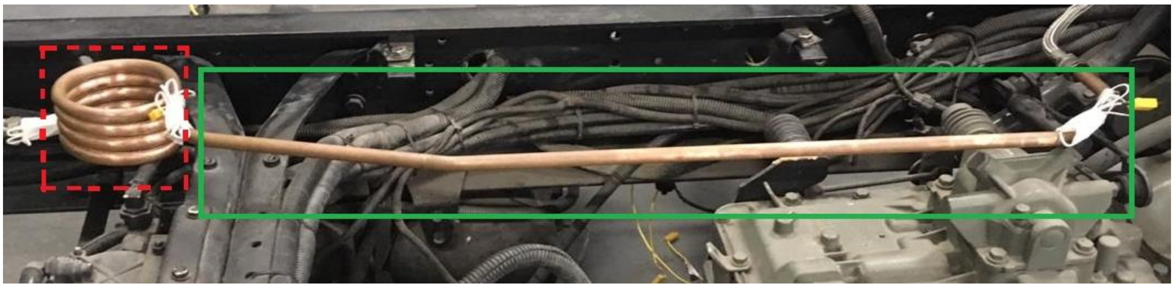

Figure 2. Representation of the copper heat exchanger with linear control volumes (continuous line) and 4-volute "coil" control volume. (Source: Prepared by the author).

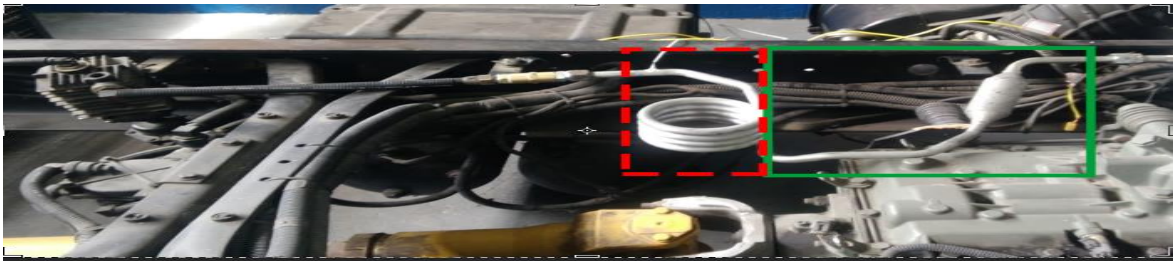

Figure 3. Representation of the steel heat exchanger with linear control volumes (continuous line) and 4-volute "coil" control volume. (Source: Prepared by the author).

Figure 4. Representation of the steel heat exchanger with the linear control volumes (continuous line) and the control volume of the coil (dashed line) with 5 volutes. (Source: Prepared by the author).

All heat exchangers have differences in material type, number of volutes, linear length and the spacing between volutes. To perform and analyze the experiments, was established that the engine always will work in a standard rotation, 1500 rpm, This decision was justified because it is a working range recommended by the engine manufacturer to the vehicle user, as well as careful maintenance of the filters, engine oil and use of additive S10 diesel oil.

For all control volume conditions imposed on the heat exchangers, air for both internal and external flow was considered to be a perfect gas. In this way, issues such as water and oil contamination of the air, and pipe fouling factors will not be considered, as influences of truck bodywork too. For forced convection analysis, wind flow was simulated through a centrifugal fan, see Fig. 5, controlled by a frequency inverter. It was adapted to meet the predefined velocities in the analysis of logarithmic mean temperature differences (LMTD) and ε-NUT methods.

Figure 5. Centrifugal Ventilator for performing simulated conditions under forced convection. (Source: Prepared by the author).

2.2 Data survey

For temperature measurement in both internal and external flows, K-type thermocouples were used for both steel and copper heat exchangers. As a standard, thermocouples were instrumented inside the pipes and close to the outer wall surface on both heat exchangers, as shown in Fig. 6.

Figure 6. Heat exchanger with the following instrumentation points: 1-Cold Inlet Fluid Temperature (Tf, e); 2-Inlet Hot Fluid Temperature (Tq, e); 3-Temperature of Hot Fluid Before Coil; 4-Outflow Hot Temperature (Tq, s) and 5-Outflow Cold Temperature (Tf, s).

All thermocouples were calibrated based on ice melting temperature, room temperature and water boiling temperature according to the Resende city of Rio de Janeiro state. From the results obtained in the fit curves by the least squares numerical method it was possible to extract a linear correction equation that correlates the ideal value for each thermocouple used in the experiment. All temperature data acquisition and monitoring were performed through a Lynx AqDados data acquisition board, which was configured according to the factory calibration report and parameterized according to the manufacturer's recommendation for temperature acquisition. Figure 7 below shows how the calibration curve of the thermocouples used was performed.

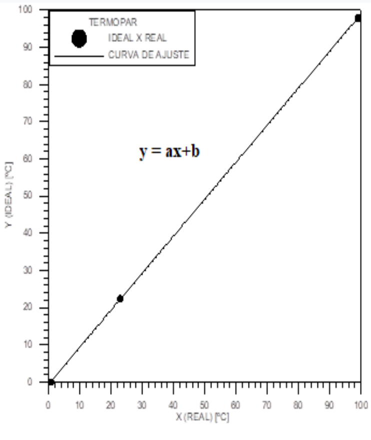

Figure 7. Least squares method applied to acquire thermocouple calibration curves.

Temperature monitoring in the heat exchangers under natural and forced convection was maintained until the system was established under steady state. Fig.8 shows the temperature curve as a function of time for natural convection and Fig.9 for forced convection on steel and copper heat exchangers in different geometries.

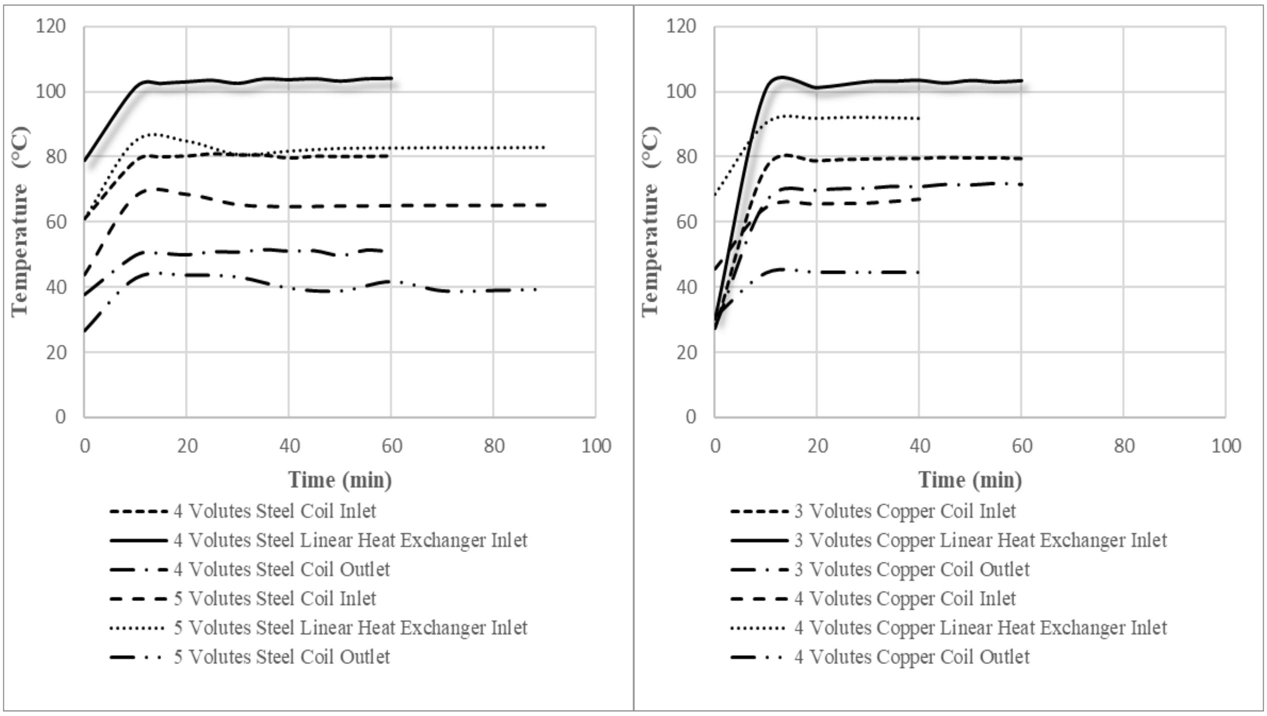

Figure 8. Characteristic curves of experimentally obtained data on heat exchanger temperatures, 4 and 5 volutes steel and 3 and 4 volutes copper, by natural convection.

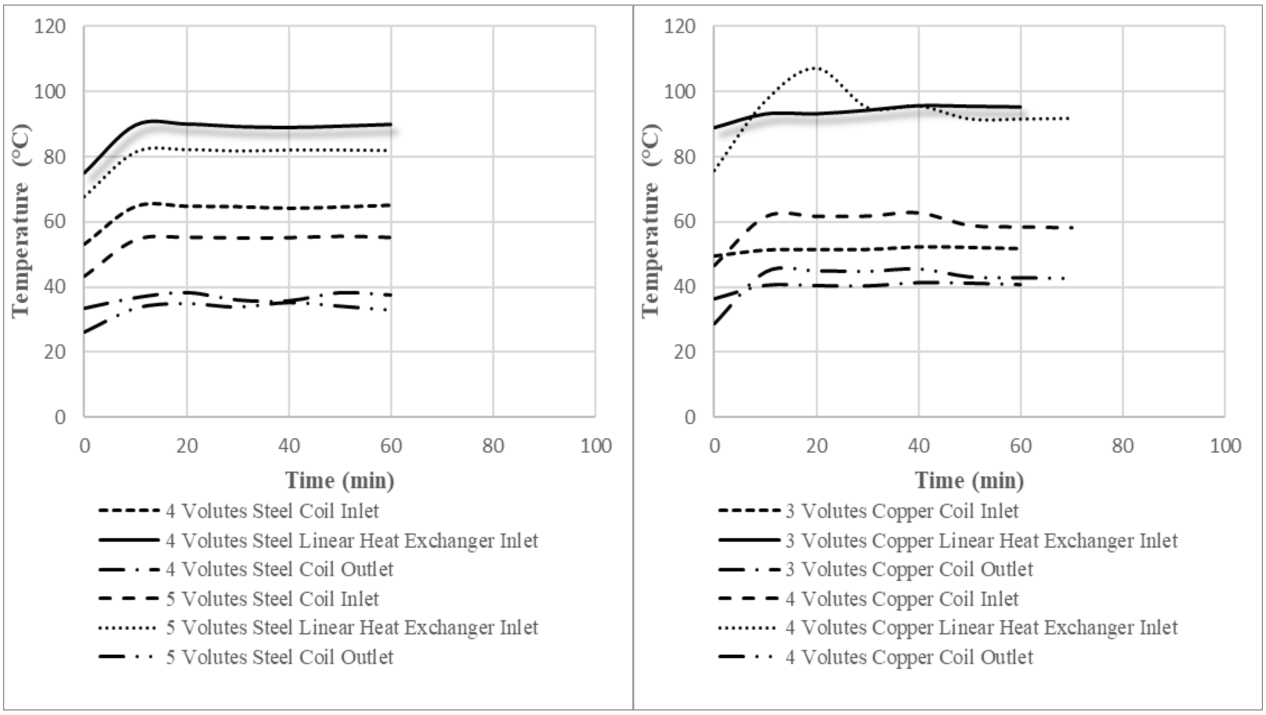

Figure 9. Characteristic curves of experimentally obtained data of the heat exchangers, 4 and 5 volutes steel and 3 and 4 volutes copper, by forced convection.

For external forced convection, wind flow velocities were verified at the inlet and outlet measurement points for both control volumes. To better work with the variations of velocities as a function of the geometric variations of the components, the average of the velocities in these points was established for the theoretical analyzes. The average velocities of external flow can be verified by Tab.1.

Average Outflow Speed (m/s)	
Copper heat exchanger: 3 volutes	5,03±0,01
Copper heat exchanger: 4 volutes	5,21±0,01
Steel heat exchanger: 4 volutes	5,52±0,01
Steel heat exchanger: 5 volutes	4,51±0,01

Table 1. Average external flow velocity with forced convection in the linear and coil control volumes for copper and steel.

Forced convection heat exchangers are typically classified according to flow configuration and construction type (Incropera et al., 2008). While mounting the experiment, it was more favorable to mount the heat exchangers on the vehicle chassis and the centrifugal fan position for a countercurrent arrangement. Thus, hot and cold fluids enter the system at opposite ends, flow in opposite directions and leave the system at opposite ends. This statement can be seen in Fig.10 for copper heat exchanger and in Fig.11 for steel heat exchanger, both for the first and second control volumes, associated with a counterflow flow heat exchanger.

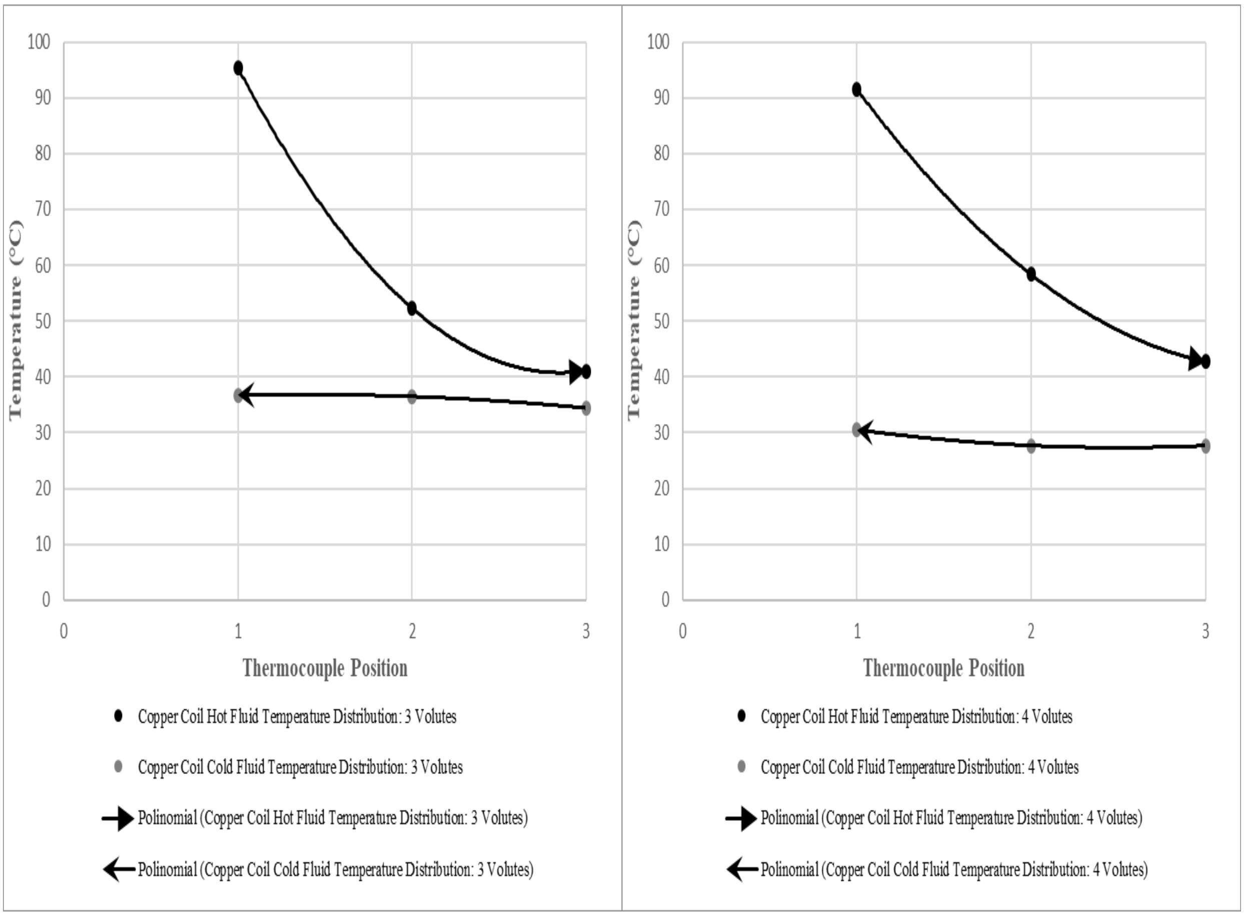

Figure.10. Temperature distribution on copper heat exchangers, with counterflow fluid in the first and second control volumes for the 3 and 4 volute coils, respectively.

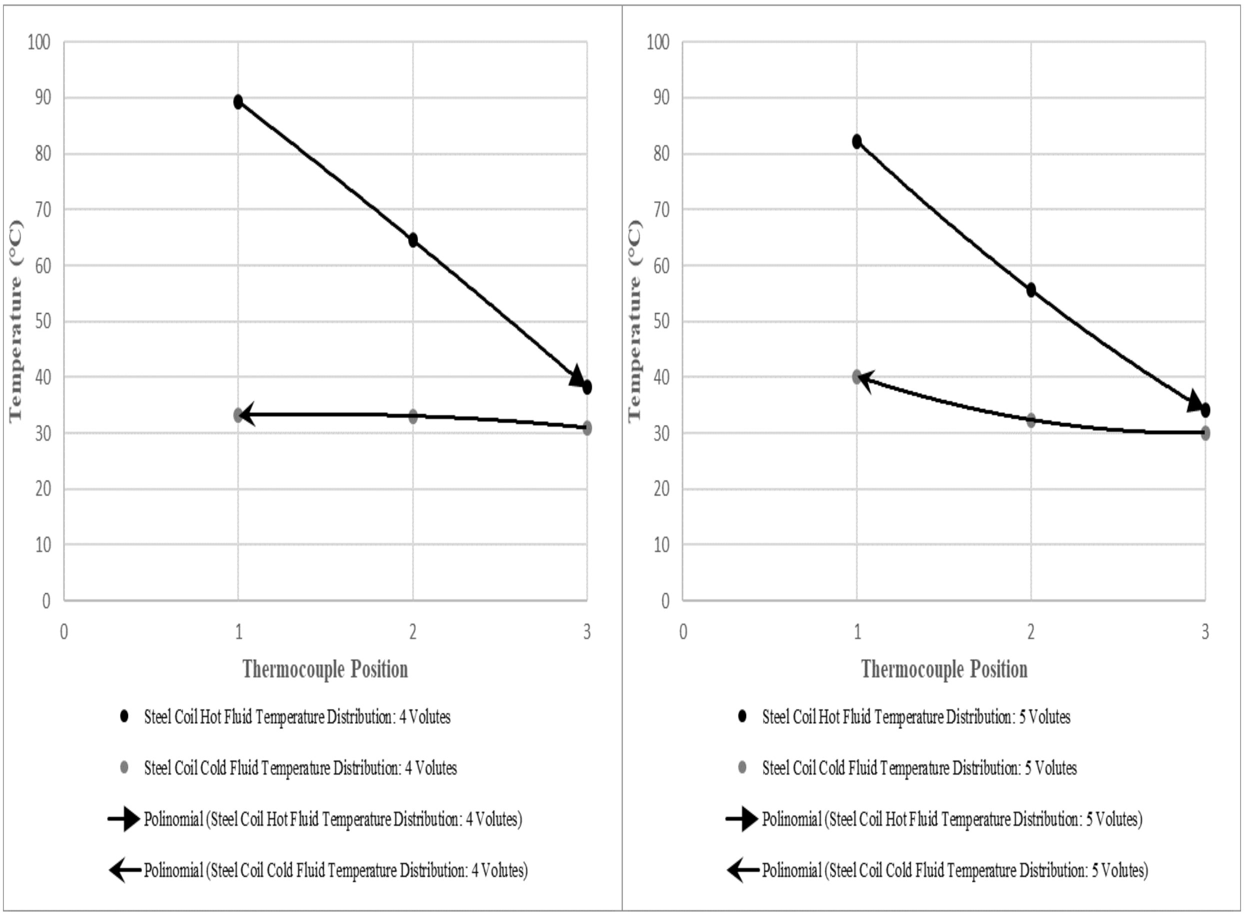

Figure.11. Temperature distribution in copper heat exchangers, with counterflow fluid in the first and second control volumes for the 4 and 5 volute coils respectively

.

Initially, the countercurrent flow configuration provides heat transfer between the warmer portions of the two fluids at one end, while heat transfer occurs between the coldest portions of the two fluids at the other end. It is important to note that the outlet temperature of the cold fluid may be higher than the outlet temperature of the hot fluid.

Subscribed indices 1 and 3 are used to designate opposite ends of the heat exchanger. This convention is used in all types of heat exchangers analyzed here. For countercurrent flow, subscript 1 has to be the temperature of the inlet hot fluid and outlet cold and subscript 3 is related to the temperatures of the outlet hot fluid and inlet cold (Incropera et al., 2008).

2.3 Formulations for calculation of heat exchange dimensioning

The study model for each heat exchanger was developed through 2 control volumes in each element. This is due to the different geometric shapes that constitute the heat exchanger, being a straight part and the other a part with volutes. Regarding the flow regime of air flow was considered permanent regime. For the development of the analysis, two analytical models were considered, one model for natural convection, which correlates the condition of low external air flow under high conditions of internal temperature and hot air flow, and another model for large amount of air. external air flow, where the LMTD and ε-Nut methods were considered.

Natural and mixed convection heat transfer is defined by the movement of the fluid by density difference in a heating process, resulting in thrust forces imposed on the fluid when its density decreases in the vicinity of the heat transfer surface, thereby resulting in natural convection currents and the process of thermal exchange (Holman, 1983).

To determine the variables of natural and mixed convection sections, the membrane temperature (T_f), corresponding between the ambient temperature ($T\infty$) and the temperature of the wall of the heat exchanger (Ts), according to Eq. (1).

$$\frac{T_\infty + T_S}{2} = T_f \qquad (1)$$

The characteristic of the flow of the fluid can be demonstrated by the value of the Rayleigh number (*Ra*), for laminar flow, $Ra < 10^9$, and turbulent flow, $Ra > 10^9$. Rayleigh is defined by the product of Grashof (*Gf*) and Prandtl (*Pr*). The values of the constants of "*C*" and "m" are found in the literature as a function of *Ra* and the surface geometry, so that the Nusselt (*Nu*) number can be determined, see Eq. (2), (Incropera et al., 2008). The heat transfer coefficient (*h*) is calculated between the Nusselt number, the heat transfer coefficient of the air (*k*) and the diameter of the heat exchanger (*D*), as presented by Eq. (3), (Holman, 1983).

$$C \cdot (Gf \cdot Pr)^{m} = Nu \tag{2}$$

$$\frac{Nu \cdot k}{D} = h \tag{3}$$

For the calculation of external forced convection for concentric counterflow tubes, the LMDT and ε-NUT methods are established. For the LMTD method the appropriate average temperature difference (ΔTm, l) is applied, where it relates the hot and cold inlet (*Tq,e;Tf,e*)and heat exchanger outlet temperatures,(*Tq,s;Tf,s*), as shown by Eq. 4, (Incropera and DeWitt, 2008) Through the average temperature difference, global heat transfer coefficient (*U*), external radius (*re*) and the rejected heat (*q*), it is possible to calculate the linear length (*L*) required for the exchanger. of heat through Eq.5.

$$\frac{(T_{q,s} - T_{f,e}) - (T_{q,e} - T_{f,s})}{\ln\left(\dfrac{T_{q,s} - T_{f,e}}{T_{q,e} - T_{f,s}}\right)} = \Delta Tml \tag{4}$$

$$\frac{q}{(U \cdot 2 \cdot \pi \cdot re \cdot \Delta T_{ml})} = L \tag{5}$$

The effectiveness (ε) is defined by as the ratio between the actual heat transfer rate of the heat exchanger (*q*) under study and the maximum heat transfer rate (q_{max}), which can be estimated from an infinite length pure countercurrent heat exchanger. This ensures the highest possible temperature in the fluid with the lowest thermal capacity. This relation is represented in Eq.6.

$$\frac{q}{q_{\max}} = \varepsilon \tag{6}$$

The calculation of the maximum rejected heat is the product of the minimum thermal capacity (C_{min}), which may be cold or hot fluid, and the difference in inlet or hot fluid temperatures as shown by Eq.7. Actual rejected heat is obtained from Eq.8, where the mass flow ($\dot{m}$), specific heat at constant fluid pressure (c_p), and the mixing temperature (T_m) of hot or cold fluid.

$$C_{\min} \cdot (T_{q,e} - T_{f,e}) = q_{máx} \tag{7}$$

$$\dot{m} \cdot c_p \cdot T_m = q \tag{8}$$

The effectiveness of a heat exchanger can be expressed as a function of Nut (number of thermal units), Eq.9. Nut can be described by the ratio of the global heat transfer coefficient (*U*), the heat exchange area (*A*), the minimum thermal capacity (*Cmin*) and the maximum thermal capacity (C_{max}), Eq10.

$$\frac{U \cdot A}{C_{\min}} = Nut \tag{9}$$

$$f\left(Nut, \frac{C_{mín}}{C_{máx}}\right) = \varepsilon \tag{10}$$

Based on the equations presented above, the specific formulation for the main types and arrangements of heat exchangers was developed, as can be found in, (Incropera et al., 2008) and (S.Kakaç, et al., 2012). For the present work, considering the external flows in parallel, counterflow and or crossed conditions, the analytical model was developed.

For forced convection within control volume 1 Eq. (3) must be respected, however the Nusselt value for straight tube (Nus) is related to the Reynolds (*Re*) and Prandtl (*Pr*) number, following Eq. (13) for laminar flow, *Re* <2000, and Eq. (14) for turbulent flow, *Re*> 2000, (Holman, 1983). The value of the constant "n" may be 0.4 for the external fluid heating condition if the internal wall temperature is higher than the fluid mixing temperature, or 0.3 for fluid cooling. Therefore, the inner wall temperature is lower than the fluid mixing temperature (Holman, 1983).

$$0,65 \cdot \mathrm{Re}^{0,65} \cdot \mathrm{Pr}^{n} = Nu_s \tag{13}$$

$$0,023 \cdot \mathrm{Re}^{0,8} \cdot \mathrm{Pr}^{n} = Nu_s \tag{14}$$

However, the internal convection for the second coil control volume can be determined by Schmidt's ratio, through Eq. (15), this Schimidt ratio considers Nusselt's number for the "coil" (*Nuc*), Nusselt number for straight tube (*Nus*), coil bending radius and inner radius of heat exchanger (*ri*). Reynolds must be respected between and the ratio 5 <*R / ri* <84. This correlation was developed using coil air and water under constant wall temperature boundary conditions. Properties were evaluated at mean fluid temperature (S.Kakaç et al., 2012).

$$1,0 + 3,4\left(\frac{ri}{R}\right) = \frac{Nu_c}{Nu_s} \tag{15}$$

Finally, the calculation of the number of volutes (*N*) can be obtained by Eq. (16), considering the heat exchange area of the heat exchanger (*A*) and the variables previously shown. If the value found is a noninteger number, simply the value of "*N*" is rounded to the next highest integer, and the spacing between the volutes must be 1.25 of the outer diameters of the coil, (Patil et al., 1982).

$$\frac{A}{\pi \cdot De \cdot \sqrt{(\pi \cdot 2 \cdot R)^2 + \left(1,25 \cdot De\right)}} = N \tag{16}$$

In equation 17, the total height of the coil can be calculated as a function of the number of volutes (*H*), the center-to-center distance of the pass between two volutes (*p*) and outside diameter (*De*), (Patil et al., 1982).

$$(N \cdot p) + De = H \tag{17}$$

3. RESULTS AND DISCUSSIONS

For the analysis result sought to find the ideal linear lengths for each heat exchanger in different analytical methods, according to equation 5, keeping the temperature variations and the heat exchanger geometries constant, the rejected heat and the coefficient. The overall heat transfer will be the important factors for the calculation of the heat exchanger total linear length design, consequently the determination of the estimated heat exchange area.

For the analysis of natural convection, it was sought to find, as a function of the control volumes in the straight part of the tube and the "coil", the total linear length of the heat exchangers. These analyzes are shown graphically by Fig.12.

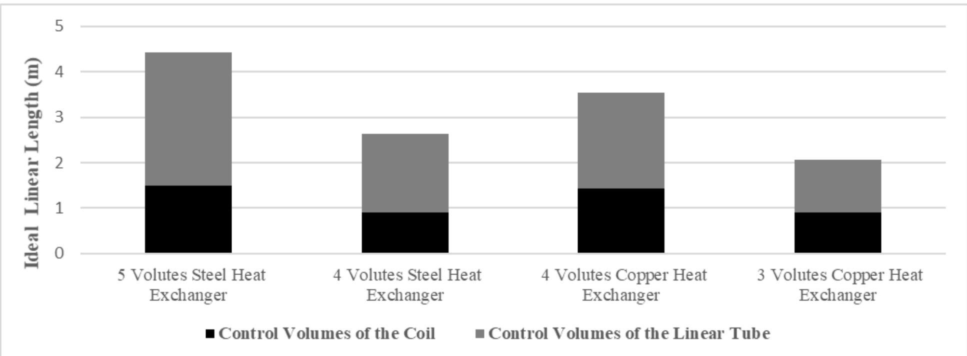

Figure 12. Histogram of natural convection analysis results, linear length for each control volume of 3 and 4-volute copper heat exchangers; and steel with 4 and 5 volutes. (Source: Prepared by the author).

The values found in external natural convection are equivalent for both control volumes. The heat rejection is dependent on the heat exchange area and the heat exchanger geometries, so it was observed from the analyzes performed that the linear pipes have a heat exchange area difference according to the heat rejected for each heat exchanger. heat. Thus, a positive average variation of 37% of the linear length was obtained for the 5-volute steel heat exchanger and for the 4-volute steel heat exchanger, the 4-volute copper heat exchanger varied 19% and with 3 volutes obtained 15%, these results were compared with their respective real linear dimensions. The analysis for linear and coil control volumes in external forced convection in constant counterflow mass flow can be applied to the analytical models of the logarithmic mean temperature difference (LMTD) and ε-Nut effectiveness methods. Thus, the linear length of the heat exchangers was generated as a function of the control volumes and type of materials. As shown graphically by Fig. 13.

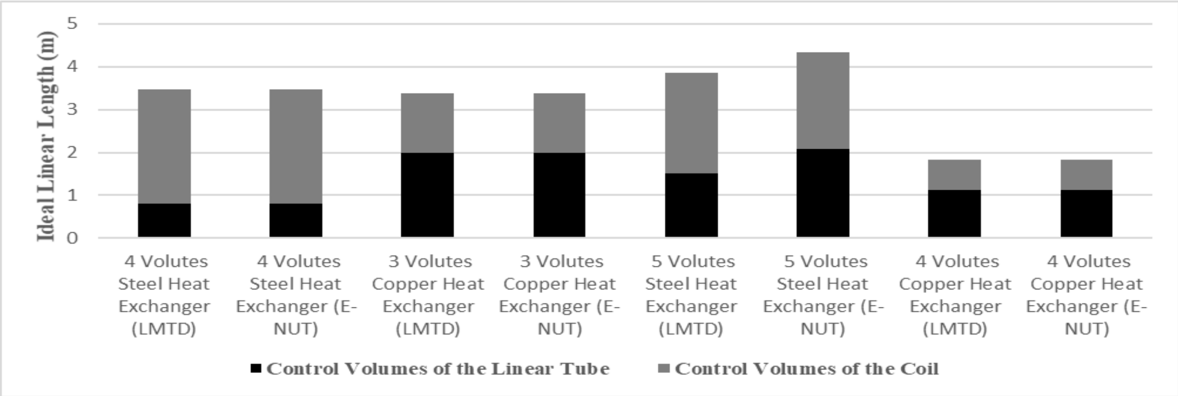

Figure 13. Histogram of forced convection analysis results for the E-Nut and DTML methods, linear length for each control volume of the 3-volute and 4-volute copper heat exchangers and 4- and 5-volute steel. (Source: Prepared by the author).

The results of the proposed methods were important to verify the design of the countercurrent flow heat exchanger. As shown earlier, the linear sizing difference between the two methods used and the actual dimensions obtained a positive variation of 27% for the 5-volute steel heat exchanger, 30% for the 4-volute steel heat exchanger, 38 % for 4-volute copper heat exchanger and 40% for 3-volute copper heat exchanger. The heat rejected on the heat exchanger under these conditions depends on the heat exchange area, heat exchanger geometries, and chassis mounting configuration on a commercial vehicle. Importantly, the greater the number of volutes and spacing between them, the better the heat exchange area for forced convection will benefit. This finding can be verified by the number of volutes as a function of the natural convection, forced convection LMTD and E-Nut methods, as shown in Fig.14. It was found that the number of positive average volutes of the three proposed methods varied by about 15. % for 5-volute steel heat exchanger, 25% for 4-volute steel heat exchanger, 7% for 4-volute copper heat exchanger and 9% for 3-volute copper heat exchanger, compared to the actual dimensions of the two coils, without considering the rounding to the upper integer of the number of volutes calculation.

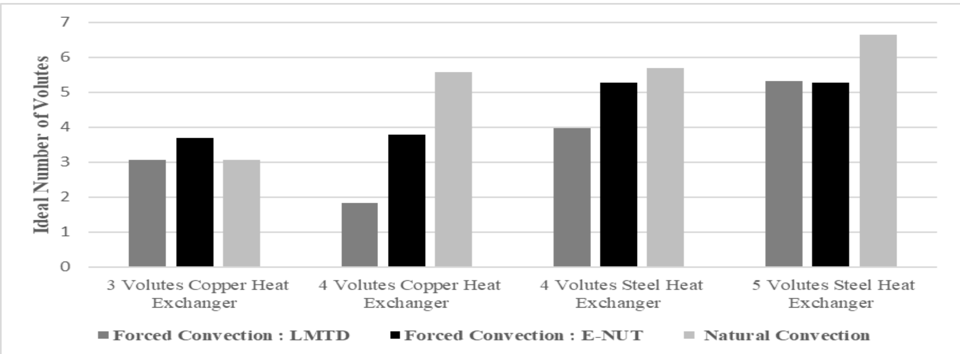

Figure 14. Histogram of the results of natural convection and forced convection analysis for the E-Nut and DTML methods, number of volutes of copper and steel heat exchangers, respectively. (Source: Prepared by the author).

4. CONCLUSION

For the present work, it can be concluded that the heat exchanger efficiency is influenced predominantly by the heat exchange area, tube manufacturing material, external and internal convections. Moreover, the results were due to air, considered in this study as ideal fluid, disregarding effects of contamination by impurities such as water (due to air humidity) and oil (contamination from the compressor) and or effects of scale and loss of impurities. load, which influence the theoretical results, due to the alteration of the considered parameters. In addition, the heat rejected by the compressor corresponds to a vehicle engine operating condition and was set to a speed of 1500 RPM not reaching maximum torque and engine power conditions. The results presented by the analytical proposal were satisfactory, as demonstrated by calculating the linear length and the number of volutes in comparison to the experimental model used in real application by the assembler.

Under the imposed conditions of forced and natural convection thermal rejection the model was analyzed by an arithmetic average of the three proposed methods for the definition of the linear length and the number of volutes, for the ideal total linear dimension resulted in an average variation of 31. % for 5-volute steel heat exchanger, 19% for 4-volute steel heat exchanger, 19% for 4-volute copper heat exchanger and 22% for 3-volute copper heat exchanger. Thus, the present study suggests that when sizing heat exchangers, it is necessary to analyze not only through a single method, but all possible conditions to better characterize the model. Regarding the test arrangement, it is important to note that the configuration of the external flow regime in parallel was not successful in the analyzes, as the heat exchanger mounting configurations did not allow the system mass flow to remain constant for the thermal exchange. In addition, vehicle body mounting conditions can cause heat exchanger encapsulation, thus causing unfavorable conditions for heat exchange and not favoring temperature variations, but with the proposed counter flow allowed a better flow analysis air mass and temperature profile, as referenced in several studies of this study. It is worthy that in the counterflow, configuration material savings and cost savings can be achieved for the manufacture of heat exchangers due to the need for a smaller heat exchange area to achieve the same thermal rejection results compared to a parallel mounting configuration.

For future work, the content of this study proved to be an efficient tool for concept optimization and development of a new model with differentiated geometries and working in more severe counterflow conditions, with the possibility of large external mass variation, simulating the vehicle in dynamic state. at different speeds and may even favor cost savings through the proposed optimization.

5. REFERENCE

Holman, J.P., 1983.” Transferência de Calor”, McGrawHill Ltda, São Paulo, Brasil.

Incropera, F.P. [et al.], De Witt, D. P., Bergman, T.L., Lavine, A.S., 2008. “Fundamentos de Transferência de Calor e de Massa”, Ed. LTC, 6ª edition., Rio de Janeiro, Brasil.

K. Patil, Ramachandra Shende, B.W. and Ghosh, Prasanta.,1982. “Designing A Helical-Coil Heat Exchanger”. Chemical Engineering (New York), Vol. 89, pp. 85–88.

S.Kakaç,Liu.H,APramuanjaroenkij,2012.”Heat Exchanges,Selection,Rating,and Thermal Design”,third Edition.

Yunus A. Çengel and Afshin I. Ghajar., “Transferência de Calor e Massa”. McGraw-Hill Ltda, São Paulo, Brasil.

6. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.