A helical radial fins heat exchanger proposal for cooling air in a pneumatic system of a heavy vehicle

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Abstract. *The present work has as theme the proposal of a heat exchanger for cooling air in a pneumatic system of a heavy vehicle. The boundary conditions of the problem were provided by physical test performed by a company that produces heavy vehicles. The heat exchanger proposed for this application has a different of the most used format in this industry; it has helical radial fins and is exposed to a transverse external flow to the internal flow. For the development was chosen a logarithmic method of temperature differences (LMTD) as the inlet and outlet temperatures of the fluid were known. With this, a mathematical model in Microsoft Excel software was developed taking into account all design parameters and calculation. To validate this mathematical model, it was compared to an experiment conducted in the fluids mechanics laboratory of Universidade Positivo and with Computational Analysis of Fluid Dynamics (CFD). Once the results provided by the three methods were analysed, it was proposed nine solutions that would meet the requirements of the studied application. It was also suggested the best proposal based on the criteria of less use of raw materials and higher efficiency.*

Keywords: *air cooling, CFD, Heat Exchanger, Helical radial fins*

1. INTRODUCTION

Companies must be competitive in the current market. Therefore, the development of its products must optimize the raw materials and manufacturing processes costs.

According to Black (1998), "Manufacturing processes are developed to add value to the materials as efficiently as possible. Advances in manufacturing technologies usually bring productivity gains". Thus, the methods and tools used in development projects can influence the efficiency of production processes.

Any waste of time or of raw materials directly affects the final cost of production and, hence, the selling price, causing the company to gain or lose market competitiveness. Using the example of Black (1998), normally the cost of manufacturing is 40% of the selling price of a product. Black (1998) concludes that "as the selling price is determined by the market, maintaining the profit depends on reducing manufacturing costs".

In an industrial application, where the product offered by the company will be a heavy vehicle type truck or bus, this quest for optimization in costs is no different, and this scenario will be the basis for the development of this work. The air cooling system used in these vehicles reduce the air exit temperature from the compressor to a temperature acceptable in the air dryer and all the pneumatic system of the vehicle.

A set of solutions can be considered for this system, for example, tubular coils and finned aluminum tubes axially extruded or other materials.

The estimated cost of a helical exchanger with radial fins compared to the other options mentioned above is lower, allowing the reduction in cost of the solution and maintaining efficiency. However, these heat exchangers are not currently used for this solution because there is no history of their behavior when used in an air cooling system applied in a pneumatic system of a heavy vehicle. This would imply a new process validation to this development, requiring an initial investment that must be paid over time and enable a cost reduction of the final solution.

Therefore, this research aims to propose an analysis on a helical heat exchanger as a possible solution to reduce the size of a heat exchanger, therefore, reducing the used material and its costs.

A helical fin heat exchanger was designed and implemented. Experiments measured the convection coefficient and it was performed a CFD analysis for the flow between these fins, compared to circular fins.

2. MATERIALS AND METHODS

The knowledge of the properties of materials used in the manufacture of products is important, as each material has a behavior that differentiates the manufacturing process to which it will be submitted. To make a choice, the designer must take into account the properties of the material before and after processing, because in the meantime, major changes occur and may affect the project. Some properties such as mechanical strength, density, thermal conductivity and / or electrical, plasticity among others, are important, as the cost and availability (Van Vlack, 1970).

There are three mechanisms on the heat transfer: conduction, convection and radiation. This research does not aims to study radiation phenomena.

2.1 Conduction

The ability of a material to conduct heat is called thermal conductivity (Çengel, 2009). The Table 1 exposes some material's thermal conductivities. The helical fins were composed of steel and the inside tube of copper.

Table 1. Materials and its thermal conductivities

Material	Thermal Condutivity (W/mK)
Steel	50.20
Copper	385.00

2.2 Convection

This heat exchange mechanism happens mainly between liquid and gas. Its function is to transfer heat within a fluid, through their own movements. Convection occurs as a result of differences between the densities of air. When the heat is conducted from the relatively hot surface for the overlying air, this air then becomes warmer than the surrounding air (Kreith, 1998).

The equivalent to the thermal conductivity is the convection coefficient. To evaluate the convection coefficient it is used a similarity approach. It is defined the dimensionless Nusselt number as the Eq. (1) (Incropera, 2008).

$$Nu \equiv \frac{hD}{k_f} \tag{1}$$

This equation does not define the convection coefficient, but the Nusselt number can be related to other dimensionless numbers using correlations as the Dittus-Boelter equation exposed in Eq. (2) (Incropera, 2008).

$$Nu = C \cdot Re^m \mathrm{Pr}^{1/3} \tag{2}$$

Re is the Reynolds number and Pr is the Prandtls number. C and m are determined using tables. Therefore, the convection coefficient can be determined using these equations.

The flow outside a heat exchanger is an example of convection phenomena.

2.3 Heat Exchangers

A heat exchanger is an equipment designed to exchange heat between fluids, (Incropera, 2008). The heat exchanger design involves determining the amount of heat that can be extracted from a fluid using the Eq. (3).

$$\dot{q} = UA\Delta T \tag{3}$$

Where $\dot{q}$ is the heat transfer rate, U is the global heat transfer coefficient, A is the area and ΔT is a temperature difference. However, it is necessary to define which temperature difference use, since there are two fluids and four temperatures involved. The Logarithmic Means Difference Temperatures (LMDT) method calculates this ΔT using the Eq. (4).

$$\Delta T = \frac{\Delta Ta - \Delta Tb}{l\,n\left(\frac{\Delta Ta}{\Delta Tb}\right)} \tag{4}$$

Where a and b are the heat exchanger borders.

2.4 Fins

Fin is a solid element which transfers energy by conduction within its borders and convection to the environment. The fins are used to increase the heat transfer rate between a solid body and an adjacent fluidThey are used to increase the heat transfer rate between a solid body and an adjacent fluid, (Ozisik, 1990).

The effectiveness of a fin is defined as the ratio between the heat transfer rate and fin heat rate if it does not exist. The effectiveness can be further understood as the ratio between thermal resistances. The shape of the effectiveness of a fin is shown in Eq. (5) and Eq. (6), Franco (2007).

$$\eta_a = \frac{tan\,h\; m\; L_c}{m\; L_c} \tag{5}$$

$$\boldsymbol{m} = \left(\frac{4h}{kw}\right)^{1/2} \tag{6}$$

Where η is the fin efficiency and m is the fin coefficient. Efficiency characterizes the performance of only a single fin. For the characterization of performance of a set of fins and the base to which they are attached and must calculate n_0 overall effectiveness shown in Equation (7) below Franco (2007).

$$\boldsymbol{n_0 = 1 - \frac{NA_a}{A_t}\,(1 - n_a)} \tag{7}$$

Where N is the number of fins, A_a is the surface area of each fin (m^2) A_t is the total surface area (m²) and η_0 is the effectiveness of a single fin, Franco (2007).

2.5 CFD Computer Fluid Dynamic

The CFD method is a part of Computational Mechanics, which is included in the Computer Aided Engineering simulation area. Using numeric and algorithmic methods via the CFD computational tool, it is possible to analyze the behavior of the fluids and the heat transfer along the flow (Maliska, 2004).

Using CFD programs analysis it is possible to can create virtual prototypes, and virtually predict their performance. The CFD analysis can be used finite volume method to define the volume control via geometry subdivision. The subdivision process geometry defines the boundaries of the volume control (Ferziger and Peric, 2002)

Because it can be applied to any kind of mesh, the CFD becomes the most advantageous method for analyzing complex geometries. This research used the finite elements approach with the STAR CCM+ software, with an educational license.

3. HEAT EXCHANGER DESIGN

3.1 Design Parameters

For an adequate heat exchanger design, it is necessary that some input parameters are known, namely:

a) mass flow rate of the compressor: ṁ = 0.0137 kg / s
b) Air speed the ventilation outlet: V = 25 m / s
c) Air temperature at the exchanger inlet: Te = 121 ° C
d) Air temperature at the exchanger outlet: Ts = 78 ° C (DT = 43 ° C)
e) Ambient temperature: T∞ = 24 ° C

These parameters are exposed in figure 1.

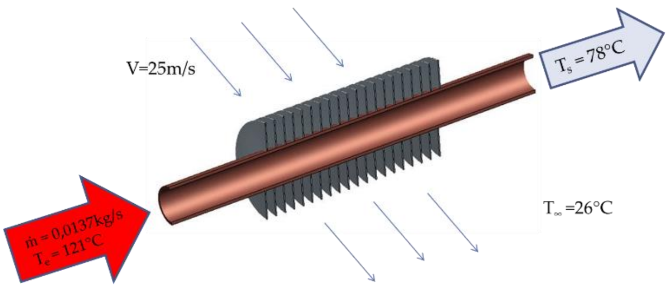


Figure 1. Heat exchanger project parameters

These parameters were obtained through experiment carried out by a company in the automotive sector of heavy vehicles at Curitiba metropolitan region. It was considered the worst operating conditions of the vehicle, for example, the maximum mass flow of the compressor, ie, there is less time for the heat exchange occurs. Another example of the air speed in the fan outlet, considers that the vehicle is stopped, that is, this is the lower speed of air supply fan, consequently, less heat exchange.

3.2 Design Dimensions

To achieve the design parameters, the proposed heat exchanger must be sized according to its geometry, illustrated in figure 2, and within the limits provided by the manufacturer manufacturing.

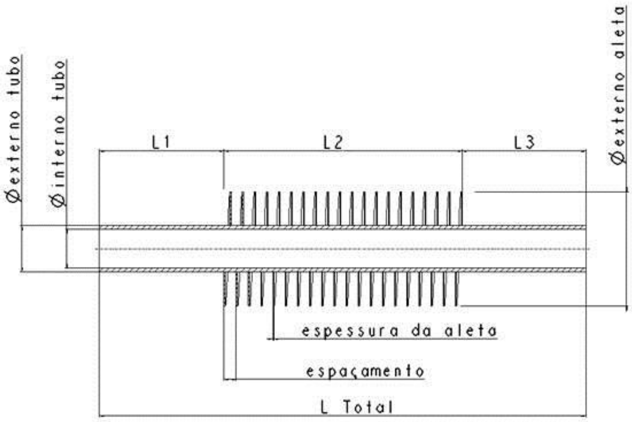


Figure 2. Heat exchanger dimension specifications

The geometry of the helical fins is distributed radially to the tube, having thickness, spaced apart, height, length finned among other parameters which will be explained in the following. L1 and L3 were both defined as 100mm, summing 200mm. L2, which is the heat exchanger with fins, measured 760mm. The total length should not exceed 960mm to fit the heavy vehicle. The internal diameter was 16mm, and the external diameter was 19,05mm.

The maximal fin diameter was 47mm due to space restrictions, so, the fin height was 28mm. The fin thickness was the least that could be produced, 0,5mm. The space between fins could be changed and will be evaluated in future. The fin numbers is the division of L2 and the space between fins.

The total heat exchange in L2 was determined using the previous equations.

3.3 Experiments

A physical heat exchanger was produced to measure its parameters. The design of the heat exchanger with finned tubes, was developed using design data supplied by an automaker company of heavy vehicles. The information provided, was obtained from of a truck that used the heat exchanger to cool the air that powers a pneumatic system, collected in tests. In order to compare the efficiency of a fin tube with tube without fins, an experiment was set in a heat exchanger system. Tests were performed in the wind tunnel Fluid Laboratory of the University Positivo, as shown in figure 3, to measure the inlet and outlet temperature difference of the fluid (air) passing through the inside heat exchanger.

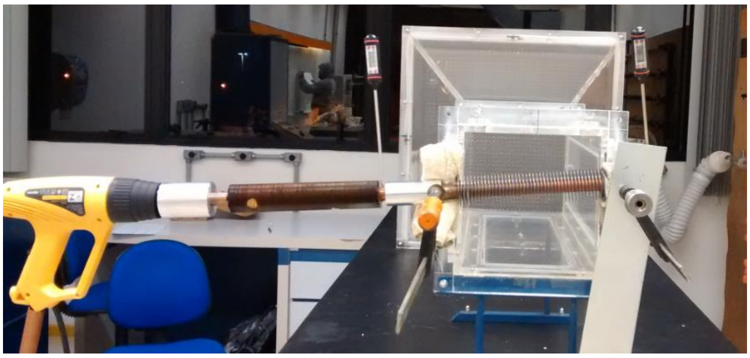


Figure 3. Experiment with heat exchanger using wind tunnel.

The hot fluid passing through the tube, was generated using a heat gun to 300C, and output volumetric flow $4,67 \times 10^{-3}$ m^3/s, as specified by the manufacturer.

For air temperature measurements, in and out of the heat exchanger, it was used two thermometers. The heat exchanger was set in the wind tunnel exit and the heat gun was attached to the heat exchanger inlet. The thermometers were attached at the heat exchanger in the inlet before the wind tunnel and the other at the exit, after the wind tunnel, as shown in figure 3. The thermometers were located outside the wind tunnel from the air flow.

The procedure used was varied wind tunnel air velocity over the heat exchanger in order to compare the heat flow. For this, the velocity was varied from 0m/s to 19,6m/s.

Due to the thermocouple installation in the experiment, the conduction effects at the walls of the heat exchanger tube was discarded. This was done to ensure better adhesion of the instrument and allow its measurement with the lowest possible temperature variation.

The first experiment was conducted with wind tunnel off just passing hot air through the tube that was the thermal exchange with the ambient air. The temperature was measured every 30 seconds until stabilization of it. After the first test wind tunnel was turned on and six measurements were carried out with different velocities of the air at the tunnel exit. For all measurements were conducted two tests, always with a five minute time between them, the recording being made at the moment when the temperature had stabilized.

The recording of air temperature measurements was possible before and after passing through the exchanger. With these measured temperatures, it was possible to calculate the efficiency of each heat exchanger.

4. RESULTS

For better understanding, this chapter is divided into three topics: experimental results, the CFD results and the mathematical model results.

4.1 Experiment results

As described in development, it conducted a test fluid mechanics laboratory at Universidade Positivo, in order to assist in the proposed design in this work. Thus, it was observed that the data acquisition, the temperature of the fluid measured with thermocouples oscillated. To justify the data, it was necessary to wait for a steady time which ranged between 5 and 15 minutes, as the graphs in Figures 4 and 5.

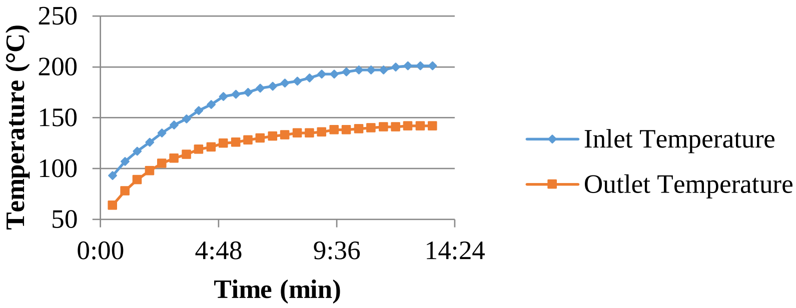


Figure 4. Steady time for thermometer measurements for V=0.

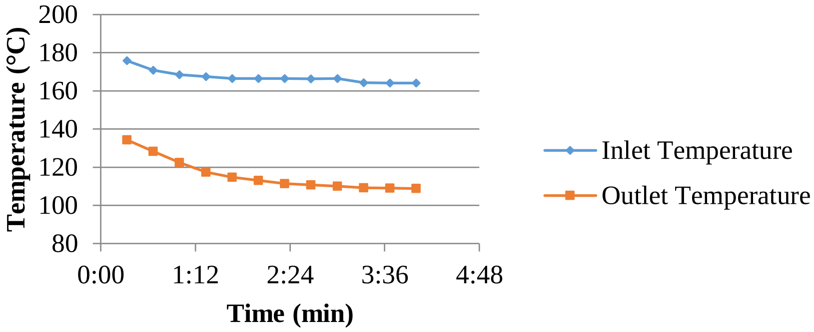


Figure 5. Steady time for thermometer measurements for V=3.2m/s.

Analyzing both figures, it can be seen that the tube without fins and without forced convection generated by wind tunnel, it took nearly three times longer to stabilize than the finned tube with a forced convection at a speed of 3.2m / s. This time was considered the stabilization temperature for the acquisition of all the subsequent data. Recalling that the two tests with and without fins for the seven-speed wind tunnel were made in duplicate, to make the data more reliable. All measurement data can be recorded in Table 2.

Table 2. Recorded data for experiment with fins

Room temperature: 15.4 °C						
air velocity (m/s)	Inlet temp. (°C)		Outlet temp. (°C)		Difference.	
	1° experiment	2° experiment	1° experiment	2° experiment	1° experiment	2° experiment
0.0	201.3	202.1	142.3	139.9	59.0	62.2
3.2	201.1	201.4	110.2	111.9	90.9	89.5
6.3	198.5	200.6	102.2	103.8	96.3	96.8
10.0	193.7	193.5	97.3	98.0	96.4	95.5
13.0	187.1	189.8	93.8	94.6	93.3	95.2
15.8	184.2	183.2	90.9	92.2	93.3	91.0
19.6	177.5	177.0	88.5	89.5	89.0	87.5

Analyzing the table, it is observed that the inlet temperature in the two tests did not remain constant, ranging 201.3 ° C to 177.5 ° C, both data for the first measurement. This was due to the fact the heat source, in the case the thermal puffer not be able to keep the inlet temperature in the pipe constant, and the transfer of heat generated due to convection outside the inlet temperature of the pipe was decreased.

Knowing the experimental data, it was used the Dittus-Boelter correlation to calculate the convection coefficient as the figure 6. The parameters adopted to this correlation were Nu=33.35, k_f=3,89·10^{-2}W/mK, C=0.023, m=0.8 and n=0.33, as INCROPERA et al. (2008).

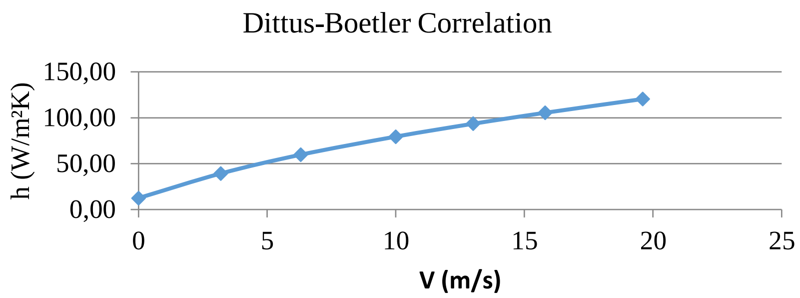


Figure 6. Calculated external convection coefficient with Dittus-Boelter correlation

The next step was to determine the temperature variation in the heat exchanger, so that it makes necessary the knowledge of the output temperature can now be calculated. Figure 7 shows a comparison between the calculated (using LMTD) and the measured ΔT.

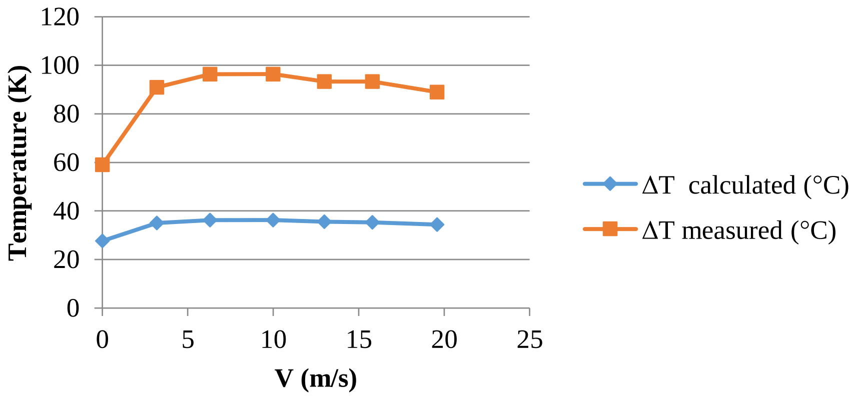
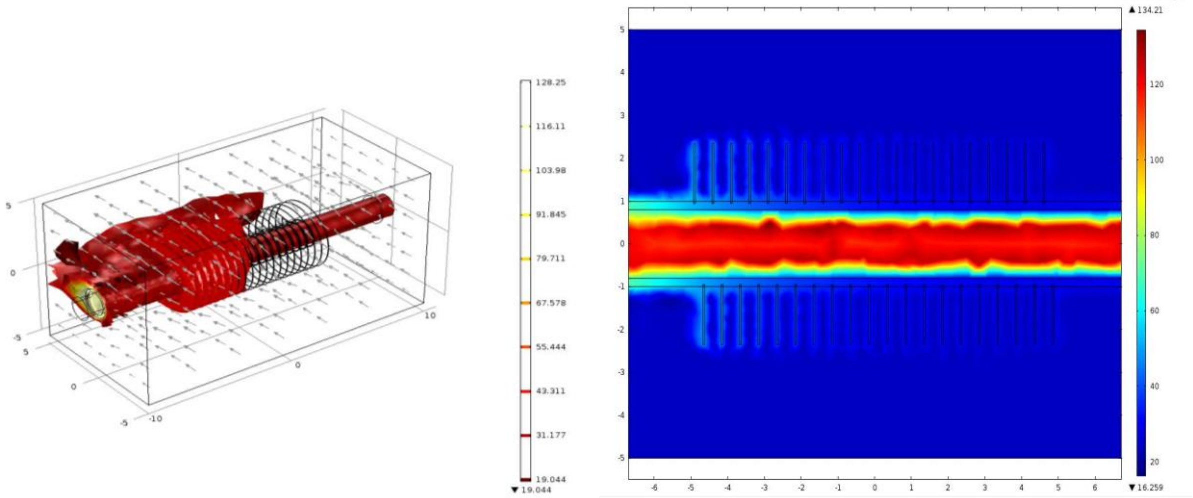


Figure 7. Comparison between the variation of temperature and calculated the change measure temperature

Analyzing the figure 7, it can be observed a difference between the calculated temperature and the measure. This difference is explained by several factors: in the experiment in spite of the driving effect have been neglected, it influences the thermocouple, increasing the temperature measured. Another factor is that the thermocouple has physical dimension and influences the flow, causing a small distortion in the measurement. Already in the analytical method MLDT despises among other items, the aerodynamic fins. If the fins were circular or helical and have the same area, for this method, there would be no temperature differences. This fact is not reflected as real. Finally, one should not consider either of the two measures as erroneous. Both observe different aspects of the same phenomenon, and one should analyze the trend of the two, which in this case is similar.

4.2 CFD results

In order to validate the mathematical model with another method, a CFD analysis was performed. However, due to restrictions on computer processing, it was not possible to simulate the same conditions of the experiment, rather than fin length of 200 mm, it was possible to simulate a finned pipe with finned 100mm length. This limitation has occurred in meshing, whereas the lower fabric element should be less than 0.5 mm, the thickness of the fin. This type of element consumed a considerable amount of memory. The Figures 8 (a) and (b) illustrate some results obtained by this computational analysis. figure 8 (a) shows the results of isotherms exposed to the flow and figure 8 (b) presents the temperature in a sagittal section along the exchanger.



(a) (b)

Figure 8. CFD Results (a) Flow isothermals and (b) Sagital temperature

Even using only fin length 100mm, the results obtained are representative, as they show the same trends of experimental curves of temperature and CFD difference. Figure 30 illustrates these trends, where "T_e" and "t_s" are respectively the input and output temperatures used in CFD, which are the same as those used in the experiment. On the horizontal axis, we see the influence that the air velocity in the wind tunnel carries on the temperature difference, being directly proportional quantities. The simulation was run with the same parameters as the experimental data.

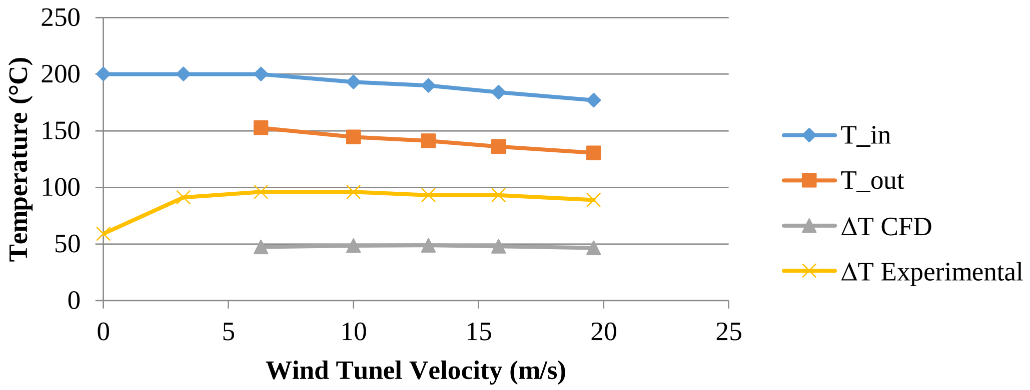


Figure 9. Temperature comparison for CFD and experiment

Applying the project boundary conditions and their restrictions, nine possible solutions have been found. More solutions could be found, but to facilitate the analysis of results only some values were arbitrary, for example, the fin length can assume any value less than 760mm, however, only four values were analyzed: 760mm, 710mm, 660mm and 610mm . Still, from these results one can see the trend of the graphs and say that the shorter fin to reach the temperature variation in the heat exchanger DT = 43 ° C is 610 mm. Below this level, the design parameters are not met.

In order to differentiate the nine possible solutions was adopted a mass criterion, trying to find what would be the lightest heat exchanger found consequently possess less use of raw materials in their manufacture.

5. CONCLUSION

With this study was possible to achieve the objective of this research, which was to design a heat exchanger with helical radial fins for an application in a pneumatic system of a heavy vehicle. The minimum mass criterion was adopted and a solution was reached. The helical fin had an adequate behavior.

6. ACKNOWLEDGEMENTS

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