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DESIGN OF A SHELL AND TUBE HEAT EXCHANGER FOR A DIFFUSION ABSORPTION REFRIGERATION DRIVEN BY A DIESEL ENGINE

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Abstract. *Due to the growing global demand for energy and concern for the environment, several studies on clean and renewable energy sources are being developed. Nowadays, a large amount of scientific research seeks to reuse waste exhaust gases in various production processes. One of the means of reusing these exhaust gases is as a heat source for absorption refrigeration cycles. This article proposes to project and build a heat exchanger to use the exhaust gases from a diesel engine as a heat source for a diffusion absorption refrigerator.*

Keywords: *shell and tube heat exchanger, Absorption refrigeration cycle, waste exhaust gases, engine, refrigeration.*

1. INTRODUCTION

In the last decade, a high quantity of research focuses on the absorption refrigeration cycles because of their capacity to work with a heat source. This capability makes systems based on this cycle a good option for reusing heat waste in different types of industries. However, to recover the heat wasted in different production areas, it is necessary to build a heat transfer exchanger. The most common heat exchanger used is the shell and tube type due to its capability to transfer heat even in a small space with the help of baffles and tubes, which increases the heat exchanger area.

For Mohammadi and Mcgowan (2019), the increase in the refrigeration and air conditioning market calls special attention to studies involving refrigerants that consider the environmental impact on the ozone layer and climate change. This fact was evidenced by Jiang et al. (2019) that state that refrigeration cycles that use ammonia/water as a working fluid have become the focus of several kinds of research. This means that they are considered less harmful to the environment than CFCs (chlorine-fluorine-carbon) and HFCs (hydrofluorocarbon) (most common gases used as refrigerants).

Furthermore, on-board refrigeration in cars can account for up to 40% of the total vehicle fuel consumption and on-board air conditioning that uses vapour compression refrigeration systems can increase fuel consumption by 20% (Venkataraman et al, 2020).

In this scenario, refrigeration systems that use alternative energy sources, such as absorption refrigeration systems, play a prominent role in thermodynamic studies involving cogeneration, especially when considering the recovery of lost energy as heat in several industrial processes as well as the use of refrigerants as an alternative to CFCs and HFCs. As related by Yuan et al. (2018) among the possible heat recovery techniques, the absorption refrigeration cycle shows potential as it can convert the exhaust thermal energy into refrigeration output and meet the on-board refrigeration requirement.

Aly et al. (2017) investigated the thermal performance of a DARC (diffusion absorption refrigeration cycle) using as an energy source the exhaust gases of a diesel 1.8L 4-cylinder engine with direct injection. The authors adapted the electric refrigerator of the DARC type to receive a heat exchanger of the shell type and tubes to heat the generator of the refrigerator. The experimental tests that were carried out used the ammonia/water pair as a refrigerant/absorbent pair and hydrogen as an inert gas. When the temperature was above 200 ºC, the bubble pump and the cycle started. Two manual valves controlled the exhaust gases to maintain temperatures between 210ºC and 231ºC in the cycle generator. The engine rotation was maintained at 1750 rpm to avoid the effects of overheating on the generator. All tests started 5 minutes after the engine started, to ensure steady-state speed of the engine. They investigated the effects on different engine quantities such as loads, exhaust gas temperature, and exhaust mass flow. The results showed that the heat recovery rate was 10%

bigger when all the exhaust gas pass through the generator. However, when the exhaust gas temperature was above 220ºC, there was a decrease in the cooling effect by up to 40%.

Ribeiro (1984), made a comparison between the methods of Kern, Tinker, and Bell, to calculate thermo-hydraulic shell and tube heat exchangers. According to the author, Kern's method does not consider the type of tubes arrangement, cuts other than 25%, leaks, and deviation. Tinker's method is the most elaborated, allowing to quantify and identify the effect of various currents in which the flow next to the shell, having interaction between then, that does not occur in another's methods. The Bell-Delaware method considers the factors of correction independent of each other, showing a better grouping of the points for heat exchange as well as pressure drop. For Reynolds above 500, the results of Bell's method were more effective with more approximated results than the others.

Manzela et al (2010) proposed the use of a single full pressure absorption refrigeration cycle to replace the compression refrigeration cycle, traditionally used in air conditioning in motor vehicles. For this, an helical heat exchanger was used. The results showed the possibility to use the waste exhaust gases to run a refrigeration system in an Otto engine since by using a mass flow rate controller. Then, Rego et al (2014) developed a methodology for controlling the heat supplied by exhaust gases from an Otto cycle internal combustion engine for application in refrigeration of refrigerated chambers. In their works, Rego et al (2014) suggest the built of a heat exchanger to improve the heat exchange of the waste energy from exhaust gases to make work a diffusion absorption refrigeration system in different loads of the engine.

Thus, a shell and tube heat exchanger was designed as proposed by Rego et al (2014), it was used the same model used by Aly et al. (2017) a shell and tube heat exchanger. Therefore, the model for the heat exchanger project followed a shell and tube heat exchangers methodology created by Kakaç et al. (2002). This paper aims to design a shell and tube heat exchanger to make a Diffusion Absorption Refrigeration Cycle works with exhaust gases from a Diesel engine.

2. METHODOLOGY

The engine was an MWM D229-4 model stationary diesel engine, Figure 1, which is in the Engine Laboratory of the Pontifical Catholic University of Minas Gerais and is part of a generator set. This means that the engine is coupled to a generator that converts electrical-mechanical energy into electrical energy. The composition of the generator set allows the introduction of a load in full operation, with the application of loads being the factor that alters the temperature of the exhaust gases. The engine is part of a generator-engine operation at a constant working speed of 1800 ± 20 rpm (Moreira, 2018)

The experimental DAR used is a commercial refrigerator Norcold N305-3R with a power of 180 W when it is turn on with a 120V alternating current. This refrigeration uses ammonia and water as the working fluid. The heat exchanger project considered a mass fraction of ammonia in the water of 0.1 kg/kg, as suggested by Zohan et. al. (2005), as a cold source. The engine used to provide the mass flow from the hot source is an MWM D229-4, with 4 strokes, direct injection, 3.92 liters, and a compression ratio of 17:1. Figure 1 shows the refrigerator and the engine.

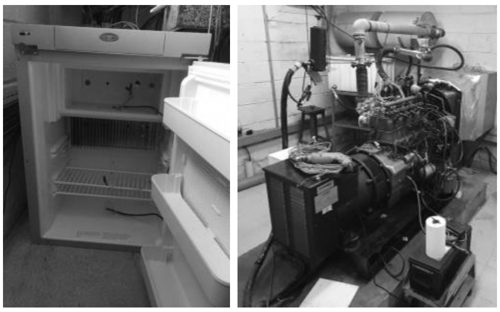


Figure 1. Refrigerator and Diesel engine.
Available from: Elaborated by the authors.

All the calculations and experiments were performed for working temperatures with the engine running at a load of 27.5 kW and 1800 rpm. The temperatures for the calculations were considerate the experimental temperatures. This temperatures were 567±6ºC for the exhaust gases, 25±2ºC for ambient and 180±4ºC for the outlet generator (when the refrigeration cycle is working on normal mode, without the introduction of the heat exchanger, with an electrical resistance to run its cycle). The experimental setup used thermocouple type k to measure the temperatures.

It is important to note that the design of a heat exchanger, Figure 2, must analyze the thermal performance, considering the coefficient of the heat exchanger and the pressure drop (Kakaç et al., 2002). If the pressure drop found in the heat exchanger design is above a certain limit, the heat exchanger parameters must be changed, seeking a reduction in the pressure drop. On the other hand, if the pressure drop found is less than expected, the parameters must be revised to reduce the costs of the heat exchanger project (Shah et al., 1988).

Thus, this study considered a mixture of ammonia and water with a mass fraction concentration of 0.1, as cold fluid flowing in the tubes of the heat exchanger and exhaust gases from the engine as hot fluid flowing in the shell of the heat exchanger Generally, designs of heat exchanger that uses exhaust gases as working fluid utilize the exhaust gases in the tubes, due to their facilities to clean and a higher control in their temperature of the outlet when compared witch the mass flow in the shell (Oliveira et al., 2017). However, due to the characteristics of the system, the exhaust gases were chosen to flow in the shell.

An analytical approach was performed to find the drop pressure lost at the shell. For this, the method of Bell-Delaware (Kakaç et al., 2002) was used. This method is one of the most used methods in the design of a shell and tube heat exchanger (Ozden; Tari, 2010). This happens because, unlike another popular methods, the Kern method is considers several factors that occur in a shell-and-tube heat exchanger, such as flow recirculation and flow in contact between baffles and tubes (Oliveira et al., 2017).

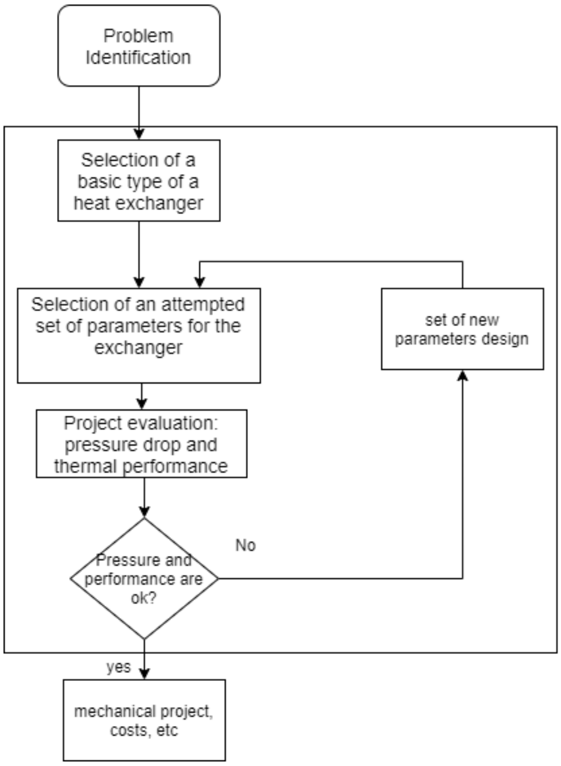


Figure 2. The general design of a heat exchanger.
Available from: Shah, Subbarao e Mashelkar (1988).

The experimental data show that the driven power required to start the DARC execution was 175 ± 5W. This value is close to the 180W power reported by the producing company. The mass flow of the cold fluid was found through the experimental power and the known inlet and outlet experimental temperatures, when the refrigerator is working with a thermal resistor, as in equation (1).

$$\dot{Q} = \dot{m}_f c_{p,f}(T_{f,s} - T_{f,e}) \tag{1}$$

where $\dot{Q}$ is the heat exchanger rate (W), $\dot{m}_f$ is the mass flow rate of the cold fluid (kg/s), water and ammonia mixture, $c_{p,q}$ is the specific heat at a constant pressure of the cold fluid and $T_{q,e}$ e $T_{q,s}$ are the inlet and outlet temperatures from the cold fluid respectively.

The temperature of the exhaust gas outlet was found by equation (2) because the heat received by the cold fluid must be the same as the heat supplied to the hot fluid. Thus, the temperature of the inlet of the exhaust gases, experimental of 300 ± 3ºC (573.15 ± 1K), the heat transfer rate, the specific heat at constant pressure (iteratively calculated), and the mass flow of the reservoir were used to find.

$$\dot{Q} = \dot{m}_q c_{p,q}(T_{q,e} - T_{q,s}) \tag{2}$$

where $\dot{m}_q$ is the mass flow rate of the hot fluid (kg/s), exhaust gases, $c_{p,q}$ is the specific heat at constant pressure of the hot fluid (J/kg-K) e $T_{q,e}$ e $T_{q,s}$ are the temperatures of the inlet and outlet of the hot fluid (K).

For this Project, the Logarithmic Means Temperature Method was used (Equation (3)). As it is possible to predict only one passage through the heat exchanger, the correction factor used was equal to 1.

$$\Delta T_{lm_{cf}} = \Delta T_m = \frac{\left((T_{inexa} - T_{outNH4OH})\right) - (T_{outexa} - T_{inNH4OH})}{\ln\left(\frac{T_{inexa} - T_{outNH4OH}}{T_{outexa} - T_{inNH4OH}}\right)} \quad (3)$$

where $\Delta T_{lm_{cf}}$ is logarithmic means temperature (K or ºC), ΔT_m is the correction factor of the logarithmic means (K ou ºC),T_{inexa} is the inlet temperature of the hot fluid (K ou ºC), $T_{outNH4OH}$ is the outlet temperature from the cold fluid (K ou ºC), T_{outexa} is the outlet temperature from the hot fluid (K ou ºC) e $T_{inNH4OH}$ is the outlet temperature from the cold fluid (K ou ºC).

The diameter of the shell can be found by Equation (4) (KAKAÇ et al., 2002).

$$D_s = 0.637 \cdot \left(\frac{CL}{CTP}\right)^{\frac{1}{2}} \cdot \left(A_0 \cdot PR^2 \cdot \frac{D_{etube}}{L}\right)^{\frac{1}{2}} \quad (4)$$

where Ds is the diameter of the shell (m), CL is the constant of the triangular tube configuration (-) – for thus 30º with a value of a 0,87 -, CTP is the constant of counting tubes (-) (0,93 for only one pass through the heat exchanger), A_0 is the heat exchange effectiveness area (m²), D_{etube} is the external diameter of the tubes (m) e L is the length of the shell (m).

The number of tubes of the shell and tube heat exchanger can be calculated through the equation (5).

$$N_t = 0.785 \cdot \left(\frac{CTP}{CL}\right) \cdot \frac{(D_s)^2}{(PR)^2 \cdot D_{etube}^2} \quad (5)$$

where Nt is the number of tubes (-), PR is the relation of passes of the tubes is the ratio of the passes of the tube given by the ratio of the distance between the centers of the tube and the inner diameter of the tube, generally between 1,25 and 1,33.

The number of baffles is given by Equation (6).

$$N_b = \frac{L}{\text{Bafflespace}} - 1 \quad (6)$$

where Nb is the number of baffles (-), and Bafflespace is the spacing between the baffles.

The heat exchanger ratio from the heat exchanger is given by the product of the convective heat exchange coefficient, the heat exchange area, and the temperature difference between the fluid and the body with which it exchanges heat, as in Equation (7).

$$Q = h \cdot A \cdot (T - T_\infty) \; ou \; Q = h \cdot A \cdot (T_\infty - T) \quad (71)$$

where h is the convection heat exchange ratio (W/m²·K), A is the heat exchanger area (m²), T is the body temperature (K) and T_∞ is the fluid temperature (K).

The overall heat transfer coefficient considering the deposition factor of the exhaust gases can be seen in equation (8) (BERGMAN et al., 2011).

$$\frac{1}{U_f} = \frac{1}{h_0} + \frac{\frac{D_{etube}}{2}}{\frac{D_{itube}}{2}} \cdot \left(\frac{1}{h_i}\right) + R_{ft} + \left(\frac{D_{etubo}}{2}\right) \cdot \left(\frac{ln\left(\frac{\frac{D_{etube}}{2}}{\frac{D_{itube}}{2}}\right)}{k_{material}}\right) \quad (8)$$

where U_f é is the overall dirty heat transfer coefficient (W/m²·K), h_0 is the convective coefficient external to the tube (W/(m²·K)), Detubo is the external diameter of the tube(m), D_{itube} is the internal diameter of the tube (m), h_i is the internal convection coefficient (W/(m²·K)), R_{ft} is the exhaust gas deposition factor in the tube ((m²·K)/W) e $k_{material}$ is the thermal conductivity of the material of the tube (W/(m·K)).

The Global Heat Transfer Coefficient, when fluid deposition in the tubes is not considered, can be seen in Equation (9).

$$\frac{1}{U_c} = \frac{1}{h_0} + \frac{\frac{D_{etubo}}{2}}{\frac{D_{itubo}}{2}} \cdot \left(\frac{1}{h_i}\right) + \left(\frac{D_{etubo}}{2}\right) \cdot \left(\frac{ln\left(\frac{\frac{D_{etubo}}{2}}{\frac{D_{itubo}}{2}}\right)}{K_{material}}\right) \tag{9}$$

where U_c is the Overall Heat Transfer Coefficient (W/m²·K).

The thermal and dynamic boundary layers of the shell were considered to be fully developed. According to Bejan (2013), for flows with constant heat flow in circular ducts, the Nusselt number assumes a value of 4,364. This value was considered for the preliminary calculations of the project. After performing the preliminary calculations, the Nusselt number for the shell was calculated using the McAdams correlation, equation (10), shown by Kakaç and Liu (2002).

$$\mathrm{Nu_{casco}} = 0.36 \cdot \left(\mathrm{D_e} \cdot \frac{\mathrm{G_s}}{\mu_{\mathrm{exa}}}\right)^{0{,}55} \cdot \left(\mathrm{c_{p_{exa}}} \cdot \frac{\mu_{\mathrm{exa}}}{\mathrm{k_{exa}}}\right)^{\frac{1}{3}} \cdot \left(\frac{\mu_{\mathrm{exa}}}{\mu_{\mathrm{exawall}}}\right)^{0.14} \tag{10}$$

where De is the commercial diameter of the shell, G_s is the mass velocity of the shell (kg /(m²·s)), $\mu_{exa}$is the viscosity of the exhaust gases (Pa·s), $c_{p_{exa}}$ é is the specific heat at a constant pressure of the exhaust gases (J/kg-K), k_{exa} is the thermal conductivity of the exhaust gases(W/m-K).

The Reynolds number has been calculated and the flow is considered to be laminar. (Bergman et al., 2011) show a relationship between the dimensionless number of Nusselt (Nu) for the flow with combined inlet length, where the temperature and speed profiles develop simultaneously, given by Eq. (11)

$$\overline{Nu_D} = h_i \cdot \frac{2r_i}{k_f} = 1.86 \cdot \left(\frac{Re_D Pr_f}{\frac{L}{d_i}}\right)^{\frac{1}{3}} \left(\frac{\mu_f}{\mu_s}\right)^{0.14} \tag{11}$$

where $Nusselt_{out}$ is the Nusselt number for the tubes (-), Re_{tubo} is the Reynolds number for the tubes (-), Pr_{NH4OH} is the Prandtl number for the tubes, a mixture of water and ammonia (-), μ_{NH4OH} is the dynamic viscosity of the average cold fluid temperature (Pa·s) and μ_{NH4OHw} is the dynamic viscosity of the average wall temperature (Pa·s).

The deposition factors, both for the tube and for the shell, were removed from a specific table to return these values present in the EES software.

After calculating the heat exchange coefficients, it is necessary to know the heat exchange areas, both with the clean heat exchanger, equation (12), and with it dirty, equation (13).

$$\mathrm{A_f} = \frac{\mathrm{Q}}{\mathrm{U_f} \cdot \Delta \mathrm{T_m}} \tag{12}$$

$$\mathrm{A_c} = \frac{\mathrm{Q}}{\mathrm{U_c} \cdot \Delta \mathrm{T_m}} \tag{13}$$

The Bell-Delaware method was empirically formulated to calculate the head loss in single-phase heat exchangers. This method adds factors to the calculation of heat transfer and heat loss on the side of the shell to represent flow distortions caused by deflectors and tubes, such as leaks between deflectors and fluid leakage (Kakaç; et al., 2002; Serna; Jiménez, 2005). According to Milcheva et al. (2017), in the design of shell and tube heat exchangers installed in heat recovery systems, several aspects must be taken into account, such as the working fluid phase, the material selected for construction, the type of construction and the optimization of geometric parameters. Figure 3 shows the flows considered by the Bell Delaware method, which are:

- Current A: represents the leaks through the gaps between pipes and baffles;
- Current B: represents the main flow, through the tubular beam;
- Current C: represents the flow in the spacing between the outer limit diameter of the tubular beam and the inner diameter of the shell;
- Current E: represents the flow through the gaps between the baffles and the inner diameter of the shell;
- Current F: represents the flow through the openings in the tube bundle, formed by the absence of tubes, which only occurs in heat exchangers with more than one fluid pass in the tubes.

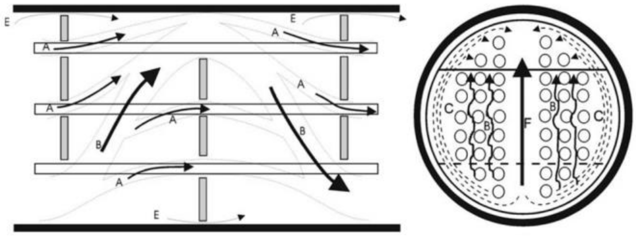


Figure 3. Mass flow in the shell of a heat exchanger.
Available from: Serna and Jiménez, 2005.

The convection heat transfer coefficient in the shell ($h_{e,B}$) is calculated by Equation (14), it is determined by a correction of the ideal convection heat transfer coefficient (h_{ideal}).

$$h_{e,B} = (J_C J_L J_B J_S J_R) h_{ideal} \tag{14}$$

Where: Jc is the correction factor due to the configuration effects on the baffles, taking into account the cut in the window and the spacing between the pieces, J_L is the correction factor due to the leakage effects on the baffles, J_B is the correction factor due to the by-pass effects on the tube bundle, J_S is the correction factor due to the effects of different deflectors spacing at the inlet or outlet of the heat exchanger and J_R is the correction factor due to the adverse temperature gradient in the laminar flow. Where h_{ideal} (W/m²-K) is given by Equation (15).

$$h_{ideal} = j_i \cdot cp_f \cdot \left(\frac{\dot{m}_f}{A_s}\right) \cdot \left(\frac{k_f}{cp_f}\right)^{\frac{2}{3}} \cdot \left(\frac{\mu_f}{\mu_s}\right)^{0.14} \tag{15}$$

Where ji is the correlation of Colburn for a group of tubes.

To determine the pressure drop in the shell, the Bell-Delaware method considers the total pressure drop (ΔPc, total) as the sum of the loss for cross-flow (ΔPc), the loss in the deflector window (ΔP_W) and the loss in the sections of input and output of the exchanger (ΔPs). These losses are corrected through empirical factors. The total pressure drop is calculated by equation (16).

$$\Delta P_{c,total} = \left[(N_c - 1)\, \Delta P_{b,ideal}\, R_b + N_c\, \Delta P_{w,i}\, \right] R_L + \; 2\Delta P_{b,ideal} \left(1 + \frac{N_{cw}}{N_a}\right) R_b R_s \tag{16}$$

Where $\Delta P_{b,ideal}$ is the pressure drop in a group of tubes between two baffles (Pa), given by equation (17), R_b is the correction factor for flow deviations, R_l is the correction factor for due to leaks from baffles, Rs is the correction factor for the inlet and outlet with different spacing between the baffles, Nc is the number of tubes crossed in a section of the baffles and N_{cw} s the effective number of cross-flow lines in a window. The correction factors for calculating the heat exchange and head loss coefficients were obtained from Kakaç et al. (2002) and Serth (2007).

$$\Delta P_{b,ideal} = 4 f_i \frac{{G_s}^2}{2\rho_f} \left(\frac{\mu_f}{\mu_s}\right)^{0.14} \tag{17}$$

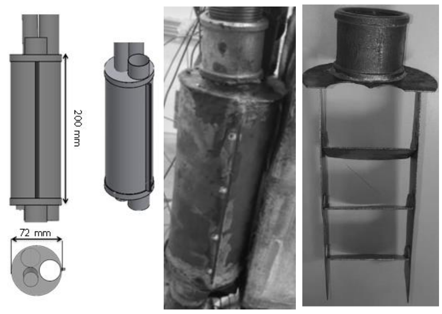
2.1 Results

The main numerical results for the designed heat exchanger are available in Table 9. For the designed heat exchanger, it was found that the diameter of the pipe must be 0.032 m, but for mounting reasons, the constructed diameter was 0.092 m. After adjusting the shell diameter construction values, the number of tubes and the number of baffles was found to be 1 and 3, respectively, with the shell and tube heat exchanger being built to these specifications. Besides, the weak solution concentration values found were 0.14 in the molar base, mass flow rate of the 5×10^{-4} kg / s mixture, and total pressure drop in the shell of 395 Pa.

Table 1. Numerical results for the shell and tube heat exchanger

Main Results – Shell and tube	Value
Shell Diameter	0.03264 m
Construction Diameter	0.092m
Number of tubes	1.001
Number of baffles	2.623 (3 built-in)
Molar fraction of ammonia into the water	0.1414 (-)
Mixture mass flow rate	$5x10^{-4}$ kg/s
Exhaust temperature estimated	567.6 K
Pressure drop in the shell	395 Pa

The shell and tube heat exchanger built can be seen in Figure 3.



a) Designed b) Shell c)Baffles

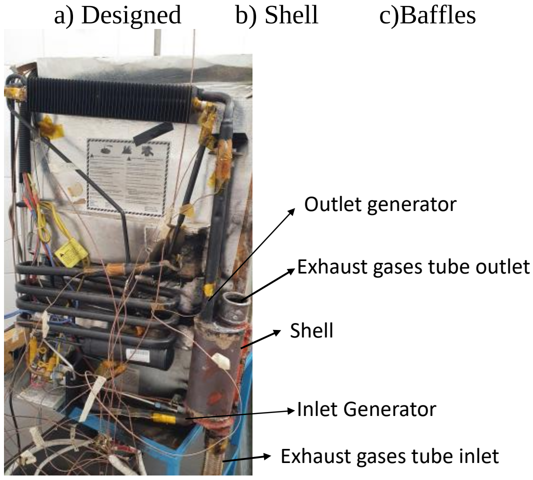


Figure 3. Shell and tube heat exchanger

After the construction of the heat exchanger, its functioning was evaluated through experimental tests with the exhaust gases of the Diesel engine. The heat exchanger was able to activate the refrigerator, proving itself capable of supplying the necessary energy for the functioning of the used refrigerator, as can be seen in the results of Figure 4. The results show that experimentally it is not possible to find a difference between the inlet and outlet temperatures of the shell and tube heat exchanger (inlet temperature of 241 ± 4 ºC and outlet temperature of 239 ± 4ºC). Based on these data, it is noted that the highest mass flow received at least 30% of the mass flow of the exhaust gases in the built-in heat exchanger. The lowest temperature at the evaporator inlet was -7.0 ± 2.0ºC after 4 hours of testing and -7.5 ± 2.0ºC after 4h 30 minutes since the beginning of the test. The maximum temperature in the evaporator was 188ºC.

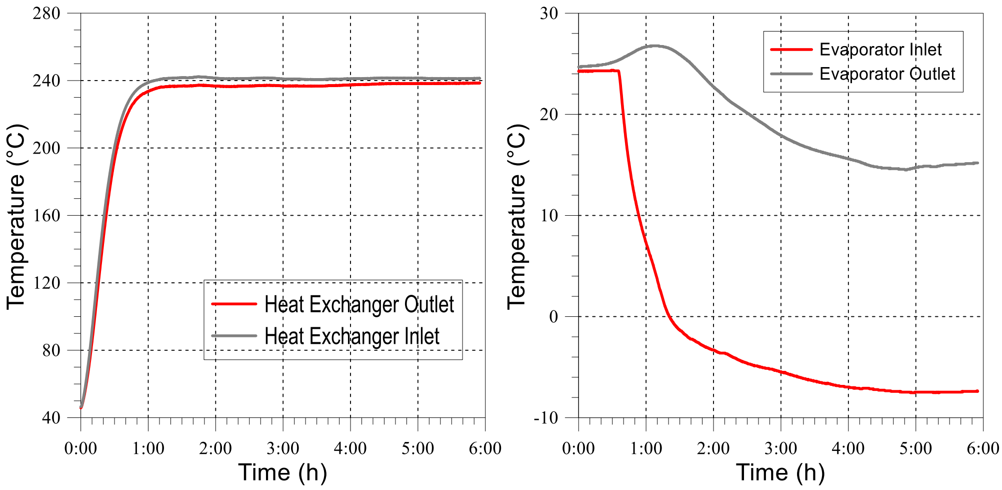


Figure 4. Experimental results.

2.2 Conclusion

A shell and tube heat exchanger was built to meet the demand for heat exchange between the cooler and the exhaust gases of a diesel engine for the correct functioning of the Refrigeration Absorption Diffusion Cycle of a refrigerator. The construction of the heat exchanger was based on the experimental temperatures of the original refrigerator and the model was designed using heat transfer equations. The project showed the need for the installation of 3 baffles and only one tube to perform the exchange. The heat exchanges found in the heat exchanger designed for the tests with the motor-generator group were able to maintain temperatures close to 180 ºC at the generator outlet, that temperatures was the reference temperature to makes a diffusion absorption refrigeration cycle works, as referenced by Manzela et al. (2010). The heat exchanger can provide energy for lower loads for which it was designed, suggesting that it can be classified more appropriately for other loads. This means the shell and tube heat exchanger is a good choice when it is used to auxiliary the heat recovery of waste exhaust gases from engines to make a diffusion absorption refrigeration cycle works. The shell and tube heat exchanger is a good alternative to the helical heat exchanger used by Manzela et al. (2010) and Rêgo et al. (2014) due its lower pressure drop inside the tubes and simple shape then the helical heat exchangers inside the tubes.

3. ACKNOWLEDGEMENTS

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