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MODELLING AND SIMULATION OF A SHELL AND TUBE HEAT EXCHANGER USED FOR REDUCING FLUE GAS TEMPERATURE FOR MICROALGAE MITIGATION

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Abstract. *Microalgae culture has been used to obtain biomass to produced biofuels and other bio-products, such as antioxidants and food. Since microalgae are photosynthetic beings, the culture has also recently been used for mitigation of industrial flues gases due to its higher composition of carbon dioxide that are important for the microalgae metabolism. However, industrial flue gases cannot be injected directly into the culture because of its high temperature. Reducing the temperature of flue gases is then necessary to avoid microalgae injury and, consequently, biomass loss. To reduce the temperature, a shell and tube heat exchangers are commonly used because of its cost benefit and its ease of manufacturing. This work brings a modeling and simulation of a shell and tube heat exchanger of a single pass that is used to reduce the temperature of flue gases from an electric generator of 50 kVA power. The modeling and simulation were performed in Fortran® code using the finite volume method where the results obtained by simulation where compared to the real data after the heat exchanger installation. The simulation results and the real data obtained for the refrigerating fluid had a variation of only 0.84% while the simulation results and the real data for the final temperature of the flue gases had a variation of 1.67%.*

Key-Words: *Microalgae, Shell and Tube Heat Exchanger and Flue Gas.*

1. INTRODUCTION

The microalgae are one of the oldest living organisms on nature that can be found individually, in groups (or chains) and are the base of the alimentary chain in the seas and rivers. They are commonly known as “plankton” (Satyanarayana, Mariano and Vargas, 2011). Their importance is based on the participation of global balance of photosynthesis, since, only at sea, more than 90% of photosynthesis is done by microalgae’s organisms (Lourenço, 2006).

The commercial use of microalgae biomass is well diversified in many fields; however, their use depends on the culture medium employed so that the amount of biomass obtained is sufficient to generate products of commercial value. In the case of microalgae, their obtainment differs from vegetal and animal extractivism because microalgae are obtained collecting these beings from the nature and “domesticating” them in culture mediums to make them usable (Lourenço, 2006).

Microalgae cultures are artificially grown in open or closed system, under batch semi-continuous or continuous system. On all these cultures medium, microalgae can grow using inorganic carbon, either autotrophically or with addition organic carbon, in a heterotrophic or mixotrophic manner (Morais, 2011). The advantage of using open systems is related to their low manufacturing and maintenance costs (Borowitzka, 1999) when compared to closed-culture media, which are more complex and expensive (Suali and Sarbatly, 2012).

Since the microalgae culture medium needs several chemical compounds, especially nitrogenous, phosphorus and carbon, their culture medium can be used as a bioremediation of exhaust gas from industrial activities involving combustion process, like thermoelectric power plants, for example. Industrial flue gases contain a range of gases such as Carbon Dioxide (CO_2), Nitrogen Oxides (NO_x) and Sulphur Dioxide (SO_2) that can be solubilized in the medium and be used by the microalgae metabolism (Huang *et al.,* 2016).

Microalgae are very sensitive organisms to temperature variation, carbon source, pH, and light incidence (Rosa, 2011). Since exhausted gases have high temperatures, they must be cooled before being injected in the culture medium. This cooling can be done using heat exchangers designed to achieve this goal. The objective of this work is to design a shell-tube heat exchanger in order to reduce the temperature of exhausted gases coming from an electric generator of 50 kVA. The design was done using a Finite Element Method (FEM), simulated in Fortran® code. The heat exchanger was manufactured and the temperatures were measured to compare the simulation with real data. The Self-Sustainable Energy Research and Development Center (NPDEAS) at Federal University of Paraná maintain photobioreactors (PBR) of 10 m^3 volume with microalgae specie *Scenedesmus obliquus*.

This study aims to help NPDEAS to utilized gases from their electric generator for growing microalgae and, consequently, reducing atmospheric emissions that can be harmful for the environment. As the aim of the study is to reduce the flue gases temperature from an internal combustion engine to a level supported by the microalgae culture, the technique known as Volume Elements Model method is applied.

2. VOLUME ELEMENTS MODEL (VEM)

The Volume Elements Model (VEM) proposed by Vargas (2001) is based on the systems engineering optimization of the physical magnitude of study. The method follows a reduced-order precision model in a reasonable time, compared to the high-order models that take a long time for the simulation (Dilay *et al*., 2015). Vargas's model (2001) follows the systems engineering process shown in Fig. 1, where the modeler chooses the hypotheses through physical laws, followed by the choice of differential equations applied to each volume element that divides the system, identifying unknown variables and known parameters. In this case, the equations are simulated in a computational program (in Fortran® language) validating the solution by experimental methods and adjusting the order of precision to optimize it (Vargas and Araki, 2017). The advantage of this method is that with a few adjustments in the modeling process (mathematical systems), the obtained values can meet the experimental data.

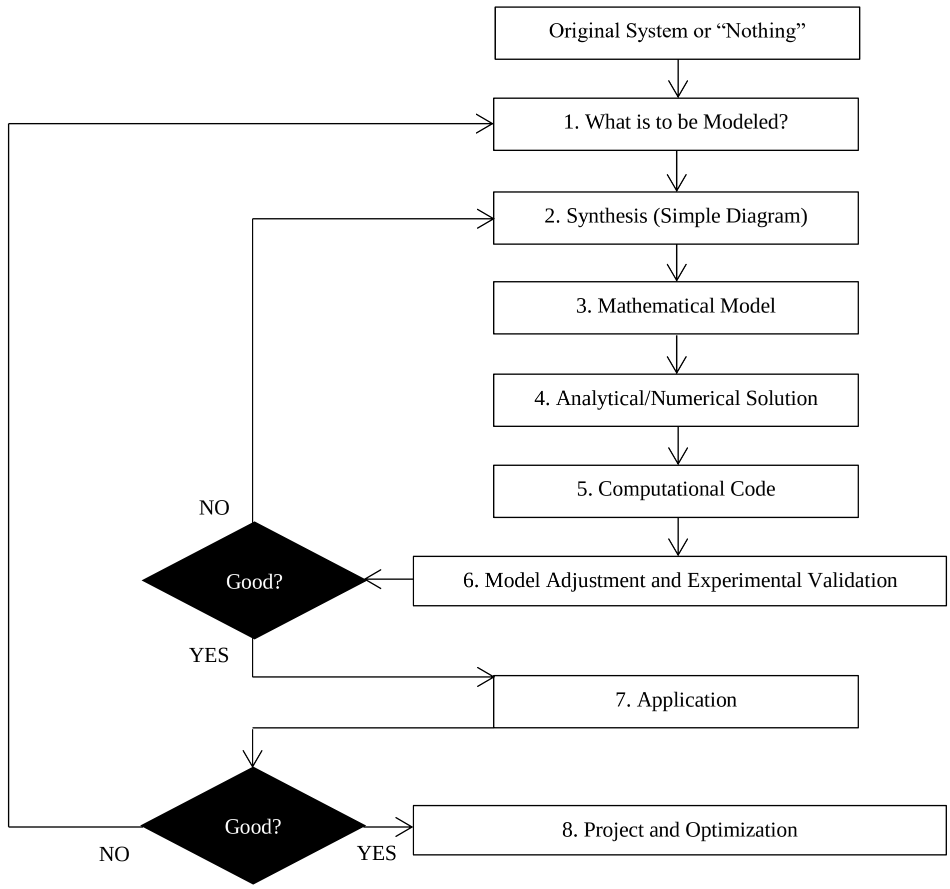

Figure 1. Schematic Representation of the Volume Element Model. Source: (Vargas, 2001).

3. EXPERIMENTAL PROCEDURE

The NPDEAS, located at UFPR, growths microalgae biomass in compact photobioreactors of $10m^3$ volume (Fig. 2) exposed to natural environmental conditions. The cultivation period takes an average of 20 days, which is enough for the microalgae to consume the medium nutrients. Each FBR contains 742 transparent tubes made of PVC, allowing the photosynthetic process of microalgae cells (Vargas, 2013).

Figure 2. Compact Photobioreactor of 10 m³. Source: The Authors (2017).

The microalgae are unicellular organisms that require an external source of carbon for their photosynthetic system, which usually can be supplied by the CO_2 present in the atmospheric air (Huang *et al.*, 2016). However, the biomass productivity is lower than cultivation systems that apply higher concentrations of carbon dioxide. Therefore, an alternative for obtaining elevated biomass productivity is to add higher loads of carbon dioxide coming from industrial applications, like the flue gas emissions from electric generators or from thermoelectric power plants, for example. Theses power plants are responsible for almost 22% of the total carbon dioxide released to atmosphere (Siegenthaler *et al*, 2005).

Since some industrial gases contains necessary elements for the microalgae metabolism, such as NOx and CO_2, microalgae cultures can serve as an alternative process for mitigation of flue gas emissions (Huang *et al*, 2016).The NPDEAS analyzes the real possibility of using exhausted gases from an electric generator (genset) to enhance microalgae biomass productivity. The genset flue gases are cooled down before being injected in the purification column, where the gases are dissolved in the culture. After passing thought the purification column, the microalgae culture is pumped to the photobioreactors. The schematic diagram of process is presented in Fig. 3.

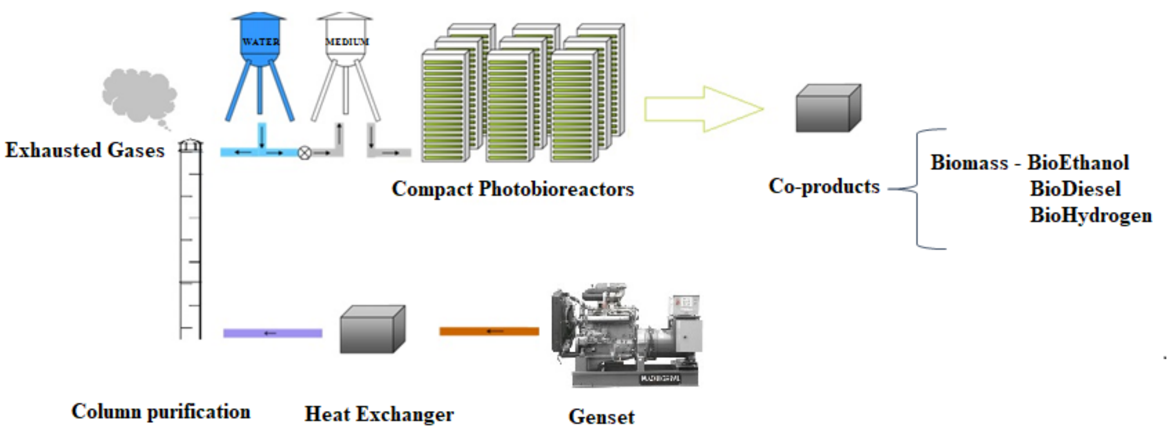

Figure 3. Schematic Representation of an Electric Generator Flue Gas Treatment. Source: The Authors (2017).

The electric generator (Fig. 4) is manufactured by Maquigeral, model 12W and it has a nominal power of 55 kWA/44 kWe with annual set limit of use of 200 hours. Since microalgae organisms are extremely sensitive to higher temperatures, the exhausted gases must be cooled to the maximum temperature of 40 ºC before feeding the microalgae cultures. Therefore, a shell and tube heat exchanger of single pass is modelled and constructed for this purpose and is the object of this study. The hot gases pass by the internal tube and the refrigerating fluid passes by the annular space between the shell and the internal tube. Fig. 5 brings the design of this heat exchanger that must be able to reduce the temperature of gases from 500 K to 300 K.

Figure 4. Electric Generator of 12MW. Source: The Authors (2017).

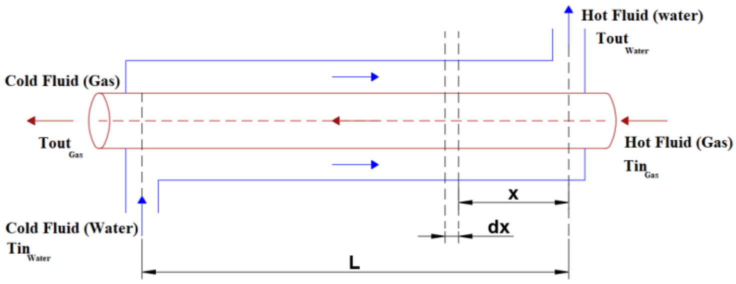

Figure 5. Single Pass Shell and Tube Heat Exchanger Scheme. Source: The Authors (2017).

4. MATHEMATICAL MODEL

The mathematical equations are based in a mass and energy conservation principles (Moran and Shapiro, 2014) with the following assumptions: (i) there isn't work produced in the system boundary; (ii) incompressible fluid; (iii) transient regime flow; (iv) the gas and water mass flow rate are constant and (v) fully developed flow. With these conditions, the 1st Thermodynamics Law can be write as:

$$\frac{dE}{dt} = \dot{Q} - \dot{W} + \dot{m}_{in} \cdot h_{in} - \dot{m}_{out} \cdot h_{out} \tag{1}$$

where dE/dt is the variation of the system total energy in relation to time, $\dot{Q}$ is the heat transfer rate on system boundary, $\dot{W}$ is the work rate realized through the system boundary, $\dot{m}_{in}$and $\dot{m}_{out}$ are, respectively, the mass flow rate that enters and leaves the system and h_{in} and h_{out} represents, respectively, the specific enthalpy at the system inlet and outlet.

Considering that all energy of the system (dE/dt) is present in this system only in the internal energy form (dU/dt), that is, by neglecting the kinetic (dK/dt) and potential $(dPot/dt)$ energies, the Eq. (1) can be rewritten as the following Equation:

$$\frac{dU}{dt} = \dot{Q} + \dot{m}_{in} \cdot h_{in} - \dot{m}_{out} \cdot h_{out} \tag{2}$$

Besides it, admitting that the internal energy U can be obtained through $U = m \cdot u$, according to Shah and Sekulic (2003), the dU/dt term can be write as:

$$\frac{dU}{dt} = \frac{d(m \cdot u)}{dt} = m\frac{du}{dt} + u\frac{dm}{dt} \tag{3}$$

Considering that there is no mass variation within the control volume, the expression dm/dt of Eq. (3) is considered zero. Besides it, taking into account the specific heat at constant volume definition, $c_v = du/dT$ [Van Wylen *et al.*, (1995)], and reorganizing the Eq. (2) and (3):

$$m \cdot c_v \cdot \frac{dT}{dt} = \dot{Q} + \dot{m}_{in} \cdot h_{in} - \dot{m}_{out} \cdot h_{out} \tag{4}$$

where dT/dt is the control volume temperature variation along the time. Now, taking into account the specific heat at constant pressure definition, $c_p = dh/dT$ [Van Wylen *et al.*, (1995)], and substituting into the Eq. (4):

$$m \cdot c_v \cdot \frac{dT}{dt} = \dot{Q} + \dot{m}_{in} \cdot c_p \cdot T_{in} - \dot{m}_{out} \cdot c_p \cdot T_{out} \tag{5}$$

where T_{in} and T_{out} are, respectively, the inlet and outlet fluid temperature. It is also noted, through the mass balance of the system, that the mass flow entering the system is equal to the mass flow leaving the system, making the mass flow rate constant throughout the system ($\dot{m}_{in} = \dot{m}_{out} = \dot{m}_{system}$). With this:

$$m \cdot c_v \cdot \frac{dT}{dt} = \dot{Q} + \dot{m}_{system} \cdot c_p \cdot (T_{in} - T_{out}) \tag{6}$$

The heat transfer rate, $\dot{Q}$, can be calculated through the heat transfer theory for circular section concentric tubes, as following:

$$\dot{Q} = U_g \cdot A \cdot \Delta T \tag{7}$$

where U_g is the heat transfer global coefficient and A is the lateral area of the inner circular section of the heat exchanger through which the flue gases pass, calculated, respectively, by Eq. (8) and (9).

$$U_g = \left(\frac{A_o}{A_i \cdot h_i} + \frac{A_o \cdot \ln(r_o/r_i)}{2 \cdot \pi \cdot k \cdot L} + \frac{1}{h_o}\right)^{-1} \tag{8}$$

$$A = 2 \cdot \pi \cdot r_i \cdot L \tag{9}$$

where A_o and A_i are, respectively, the lateral area of the external and internal tube; h_o and h_i are, respectively, the external (obtained by cold fluid) and internal (obtained by hot fluid) convection coefficient; r_o and r_i are, respectively, the heat exchanger external and internal tube ratio; k is the thermal conduction coefficient and L is the heat exchanger longitudinal length.

The external (obtained by cold fluid) and internal (obtained by hot fluid) convection coefficient can be calculated by the Dittus and Boelter (1930) correlation, valid for turbulent flow and Re > 10000.

$$Nu = \frac{h^* \cdot D_h}{k} = 0.023 \cdot Re^{0.8} \cdot Pr^n \tag{10}$$

where Nu is the Nusselt number; Pr ir the Prandtl number; n is a numerical coefficient defined 0.3 for cooling and 0.4 for heating; Re is the Reynolds number [Eq. (11)] and D_h is the hydraulic diameter [Eq. (12)].

$$Re = \frac{V \cdot D_h}{\vartheta} \tag{11}$$

$$D_{h,out} = \frac{4 \cdot A_t}{P_{eri}} \tag{12}$$

where V is the flow velocity; ϑ is the kinematic viscosity; A_t is the cross-sectional area of the tubular ring and P_{eri} is the ring perimeter. The hydraulic diameter for the inner tube of the heat exchanger is given by the tube diameter itself, while the hydraulic diameter of the tubular ring is calculated by Eq. (12).

The equations that govern the physical problem are simulated in a Fortran® language program, whose initial differential equations are solved by the Euler method. The parameters known and inserted in the simulation are shown in Table 1.

Table 1. Physical and Geometrical Parameters utilized in the Computational Simulation.

Physical or Geometrical Parameters	**Numerical Values**	**Units**
Water Density	998	$kg \cdot m^{-3}$
Gas Density	1.11	$kg \cdot m^{-3}$
Internal Diameter of Tube	0.11	m
External Diameter of Tube	0.15	m
Water Velocity	1.00	$m \cdot s^{-1}$
Gas Velocity	40.0	$m \cdot s^{-1}$
Specific Heat at Pressure Volume (Water)	4.182	$kJ \cdot kg^{-1} \cdot K^{-1}$
Specific Heat at Constant Volume (Gas)	0.721	$kJ \cdot kg^{-1} \cdot K^{-1}$
Specific Heat at Pressure Volume (Gas)	1.008	$kJ \cdot kg^{-1} \cdot K^{-1}$

Longitudinal Length of Tube	13.0	m
Gas Inlet Temperature	373	K
Water Inlet Temperature	298	K
Thermal Conductivity Coefficient (Material: Steel)	50.2	$W \cdot m^{-1} \cdot K^{-1}$
Number of Volume Elements	5.00	–
Conductivity Coefficient of Water	$640 \cdot 10^{-3}$	$W \cdot m^{-1} \cdot K^{-1}$
Conductivity Coefficient of Gas	$37 \cdot 10^{-3}$	$W \cdot m^{-1} \cdot K^{-1}$
Kinematic Viscosity of Water	$0.66 \cdot 10^{-6}$	$m^2 \cdot s$
Kinematic Viscosity of Gas	$32 \cdot 10^{-6}$	$m^2 \cdot s$
Prandtl Number for Water	3.770	–
Prandtl Number for Gas	0.686	–

Considering that the heat exchanger has two fluids that changes thermal energies in the heat form, the process must be analyzed for each individual fluid (water and gas). Eq. (6) is evaluated in the Control Volume 2 (CV_2) for the flue gases that are being cooled and in Control Volume 1 (CV_1) for the water being heated. The hot fluid, shown in "red" in Fig. 6, has a thermal exchange area given by Eq. (9) with water (a cold fluid shown in "blue" at the same Fig. 6). The equations to be simulated in each control volume are presented in the finite volume scheme of Fig. 7 and the notation "i" represents the control volume element being analyzed.

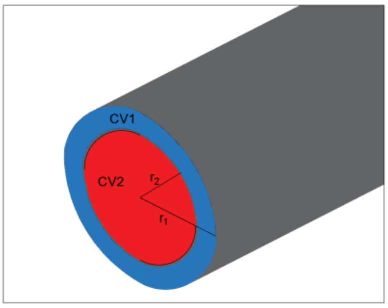

Figure 6. Tubes of the Heat Exchanger.
Source: The Authors (2017).

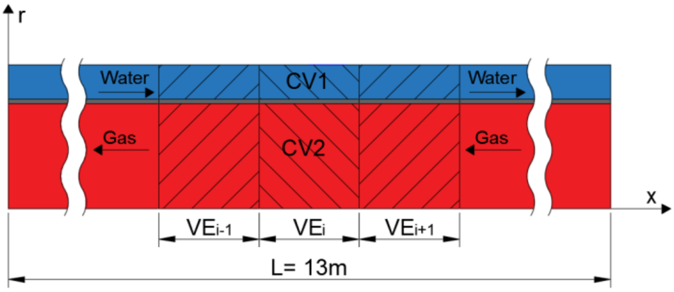

Figure 7. Volume Elements of the Heat Exchanger.
Source: The Authors (2017).

Defining the following variables $\Phi_w = \dot{m}_w \cdot c_{p,w}$, $\Phi_g = \dot{m}_g \cdot c_{p,g}$, $\xi_w = m_w \cdot c_{p,w}$ and $\xi_g = m_g \cdot c_{p,g}$, the Eq. (13) and (14) represents, respectively, the equations implemented in the computational program for VE_{i-1} formulated for the Control Volume VC_1 and VC_2.

$$\frac{dT_1}{dt} = \frac{\Phi_w \cdot \left(T_{in,w} - T_{i-1}\right) + U_g \cdot A \cdot (T_2 - T_1)}{\xi_w} \tag{13}$$

$$\frac{dT_2}{dt} = \frac{\Phi_g \cdot (T_i - T_{i-1}) - U_g \cdot A \cdot (T_2 - T_1)}{\xi_g} \tag{14}$$

The volume element VE_i formulated for the Control Volume CV_1 and CV_2 are given by Eq. (15) and (16), respectively. And, besides it, the Eq. (17) and (18) are applied to the volume element VE_{i+1} for, respectively, the Control Volume CV_1 and CV_2.

$$\frac{dT_1}{dt} = \frac{\Phi_w \cdot (T_{i-1} - T_i) + U_g \cdot A \cdot (T_2 - T_1)}{\xi_w} \tag{15}$$

$$\frac{dT_2}{dt} = \frac{\Phi_g \cdot (T_{i+1} - T_i) - U_g \cdot A \cdot (T_2 - T_1)}{\xi_g} \tag{16}$$

$$\frac{dT_1}{dt} = \frac{\Phi_w \cdot (T_i - T_{i+1}) + U_g \cdot A \cdot (T_2 - T_1)}{\xi_w} \tag{17}$$

$$\frac{dT_2}{dt} = \frac{\Phi_g \cdot \left(T_{in,g} - T_i\right) - U_g \cdot A \cdot (T_2 - T_1)}{\xi_g} \tag{18}$$

5. RESULTS AND DISCUSSIONS

The equations obtained by modelling were simulated in a Fortran® code and the results compared to experimental data. For the experimental data, a thermistor was used, in which the resistance was measured by a multimeter (BK® 2703A) and analyzed by Steinhart & Hart (1968) correlation shown in following Equation:

$$\frac{1}{T} = \frac{1}{\alpha} + \frac{1}{\beta}\ln(R) + C[\ln(R)]^3 \tag{19}$$

where R is the resistance value taken in Ω, α is the calibrated value for thermistor (α = 803.6), β represents the calibrated value for thermistor (β = 3666.4), T is the temperature in Kelvin and C represents the correction coefficient. The correction value C in Eq. (19) can be neglected due to its small value.

The exhausted gas and water temperatures exiting the heat exchanger were measured in triplicate. The Fig. 8 shows the variation of exiting water temperature versus time (T x t), where the values obtained by the simulation are within the acceptable error range. The experimental data and those obtained by a computational simulation had a difference of 0.84%.

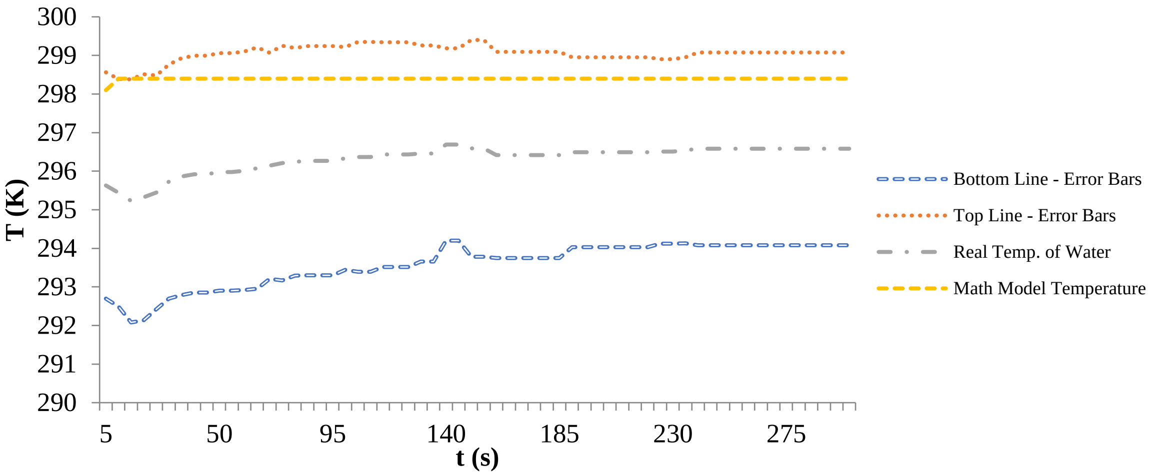

Figure 8 – Model Validation with Experimental Data for Water Fluid. Source: The Authors (2017).

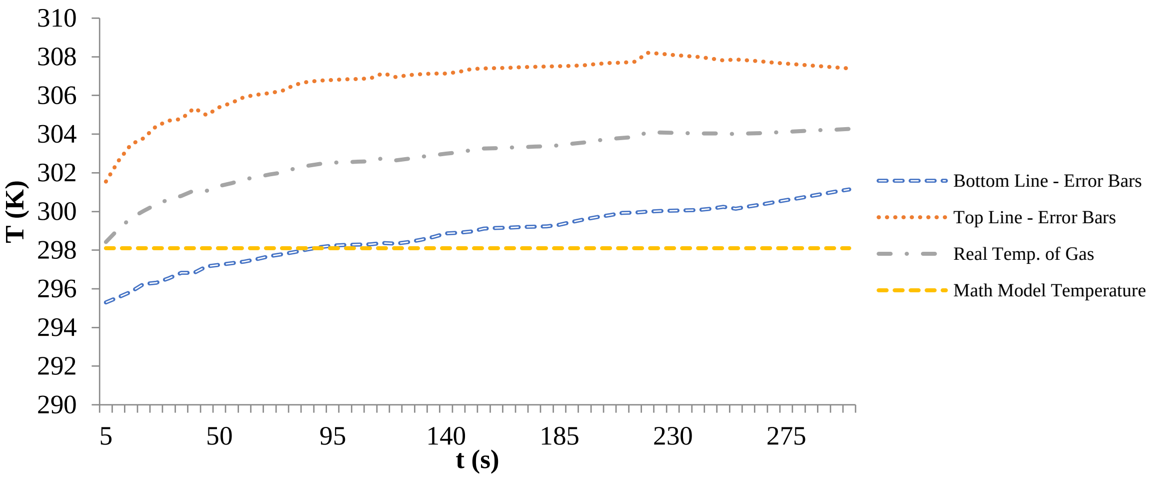

Figure 9 – Model Validation with Experimental Data for Gas Fluid. Source: The Authors (2017).

Exhausted gas-related temperature values taken experimentally and by simulation are shown in Fig. 9. It can be noted that the simulation values are out of the error range bars. The values obtained by computational simulation were 298 K while experimental data gave us an average value of 303 K for the existing gas temperatures. Even though this occurs, the real data and the experimental data differs only 1.67 %, which doesn't address the math model as a bad one.

For the simulation of the model equations, initially five-volume elements were used. The temperature distribution for each element of volume is shown in Fig. 10. By analyzing the graph, it can be noted that the temperature of exhausted gases is reduced to the minimum temperature right in the initial volume elements, which means the heat exchanger could have a shorter length than was previously designed. Increasing the number of elements of volume to 10 (Fig. 11) and 20

(Fig. 12), the simulation presents similarly behaviors when used a smaller number of volume elements; however, the time required for the simulation gets longer when the number of elements of volume used in the simulation is increased.

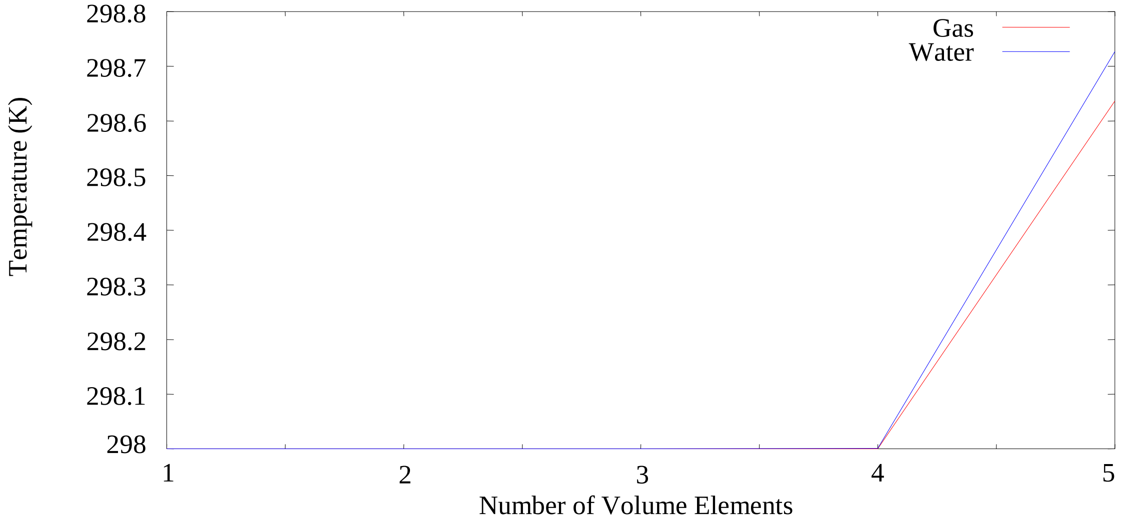

Figure 10 – Fluid Temperature in the 5 Volume Elements. Source: The Authors (2017).

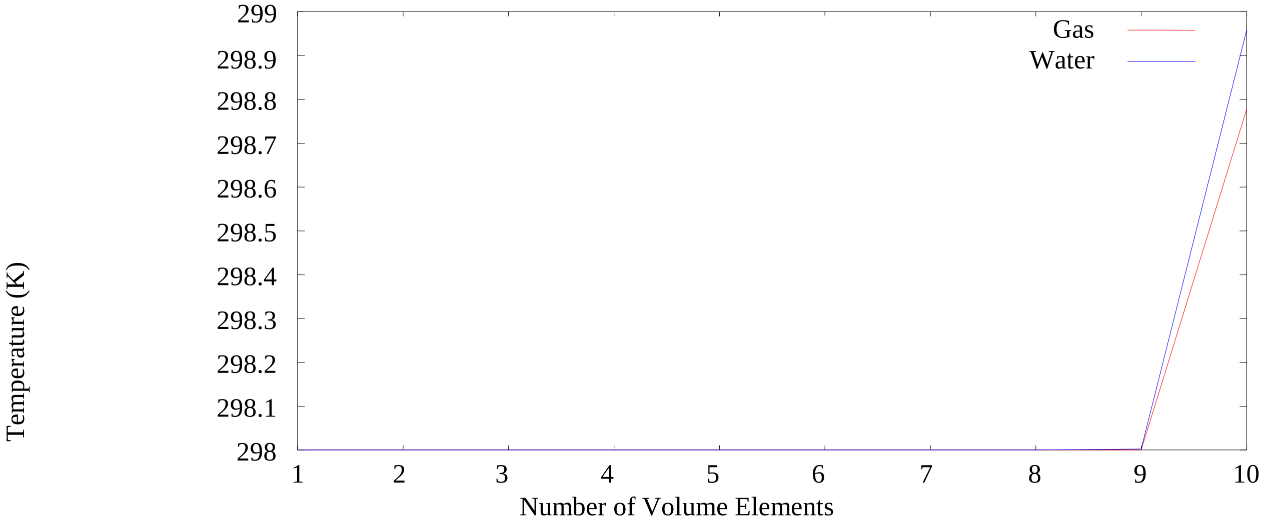

Figure 11 – Fluid Temperature in the 10 Volume Elements. Source: The Authors (2017).

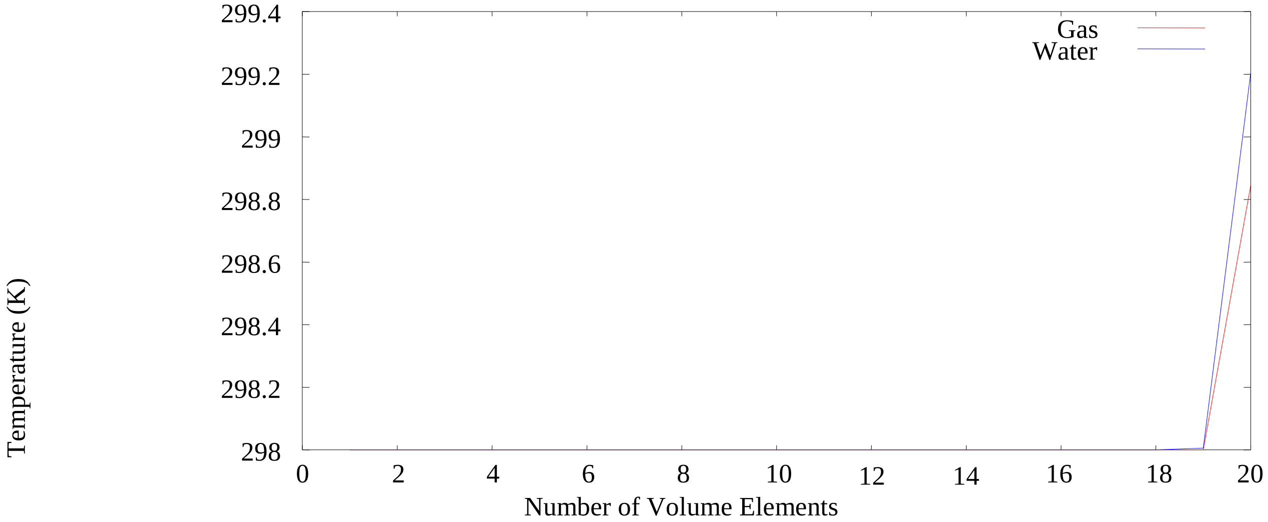

Figure 12 – Fluid Temperature in the 20 Volume Elements. Source: The Authors (2017).

6. CONCLUSIONS

Over the last decade, lab researches involving the use of microalgae cultures for flue gases treatment have growth significantly. However, scaling up for field applications, the use of exhausted gases derived from industrial activities must be cooled before feeding the microalgae cultures to avoid cells losses. Modelling and simulating a shell and tube heat exchanger of single pass for refrigerating exhausted gases from a diesel genset using the method of elements of volume (MEV) was presented in this work. The experimental results and the data obtained from the model have shown approximated values. For the exhausted gases temperature, the difference between experimental values and those obtained by modelling were 1.67%, while the difference for the water temperature exiting the heat exchanger was 0.84%. The modeling and simulation have also shown that the longitudinal length of the heat exchanger could be 50% shorter, which would implicate in cost savings during the manufacturing. The time required for refrigerating the flue gases from 100ºC until the room temperature of 30 ºC were approximately 200 seconds.

7. ACKNOWLEDGEMENTS

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