The design of an annular combustion chamber

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ABSTRACT

The design of an annular combustion chamber for a micro gas turbine engine is presented in this paper. The combustion chamber is designed for using biogas as fuel. It is designed based on the constant pressure, enthalpy addition process. The present methodology deals with the computation of the initial design parameters and arriving at optimized values. Then the dimensions of the combustor are calculated based on different empirical formulas. The air mass flow is then distributed across the zones of the combustor. The cooling requirement is met using cooling holes. The whole combustion chamber is modeled using Catia V5. The model is then analyzed using various parameters at various stages and levels to determine the optimized design. The aerodynamic flow characteristics are numerically simulated by means of the ANSYS CFX software. The air-fuel mixture, combustion-turbulence, the thermal and cooling analysis is carried out. The results are then presented in image outputs and graphs.

KEYWORDS: Aerodynamic design; annular combustion chamber; CFD (computational fluid dynamics) analysis.

NOMENCLATURE

LATIN
S-section area $[mm^2]$;
T-temperature [K];
$\dot{m}$-mass flow [kg/s];
v-velocity [m/s];
r-radius [mm];

GREEK
α-excess air;
ρ-density $\left[\frac{kg}{m^3}\right]$;
ϕ-hole diameter [mm];

1. INTRODUCTION

The scope of this article is to present a design methodology for a combustor which will be part of a micro gas turbine for a 350 kW cogeneration power plant which works after a Brayton recovery cycle. The compressor and the turbine are the single-stage centrifugal type. The combustion chamber is of annular type. The micro gas turbine is designed to function using biogas as fuel.
The combustion chamber, or combustor, of a gas turbine, is the device that receives the pressurized air from the compressor mixes it with fuel, and burns this mixture to release the heat energy through a combustion reaction. Gas turbines work with a high excess of air, usually out of the flammability limits, and so a flame tube, or liner, is used to improve the distribution of air through the reactor.

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Basically, the liner divides the combustion chamber into three zones: the primary zone, the secondary or intermediate zone and the dilution zone. In the primary zone, a recirculation zone is created to ensure the stability of the flame. The main function of the primary zone is to anchor the flame and provide sufficient time, temperature, and turbulence to achieve essentially complete combustion of the incoming fuel-air mixture. In the primary-zone a portion of the hot combustion gases are entrained and recirculated in order to provide continuous ignition to the incoming air and fuel. If the primary-zone temperature is higher than around 2000 K, dissociation reactions will result in the appearance of significant concentrations of nitrogen oxides. On the other hand, incomplete combustion will lead to the formation of carbon monoxide (CO) and unburned hydrocarbons (UHC) in the efflux gases. Should these gases pass directly to the dilution zone and be rapidly cooled by the addition of massive amounts of air, the gas composition would be "frozen," and CO, which is both a pollutant and a source of combustion inefficiency, would be discharged from the combustor unburned. Thus, in the secondary zone, the temperature is dropped to an intermediate level by the addition of small amounts of air encourages the burnout of soot and allows the combustion of CO and any other unburned hydrocarbons (UHC) to proceed to completion. The role of the dilution zone is to admit the air remaining after the combustion and wall-cooling requirements have been met and to provide an outlet stream with a temperature distribution that is acceptable to the turbine.
Combustion chamber design methodologies have been proposed by Lefebvre [1] and Melconian & Modak [2]. However, these methodologies have been developed for the design of aeronautical and large industrial gas turbine combustors. These methods have been successfully used to design combustion chambers [3, 5, 6] and there has even been developed software in which these methods have been implemented [4].
Mohammad and Jeng [7] have developed a computer code for annular combustor design. The design algorithm employs empirical and semi-empirical models which include diffuser section design, air distribution computations, combustor sizing, fuel nozzle design, axial swirler design, heat transfer calculations (with/without cooling, thermal barrier coating) and dilution holes design.
Conrado et al. [8] present a methodology for gas turbine combustor basic design. Criteria for selecting a suitable combustor configuration are examined followed by design calculations for the dimensions of the casing, the liner, the diffuser, and the swirler. Calculations of gas temperature in the various zones of the combustor and liner wall temperatures in the presence of film cooling are performed along with design calculations for the dimensions of the air admission holes. A computational program was developed based on the sequence of equations discussed in the paper.
In [9] is presented a combustion chamber design methodology using a software developed for this purpose. The software developed calculates the dimension for a straight through tubular combustion chamber that operates with gaseous fuel. This software allows visualizing the main aerodynamics parameters that are involved in the project of a simple combustor. The equations involved on the calculus of the program are based on the methodology developed by Lefebvre.

2. DESIGN METHODOLOGY

The first step in designing the combustor was the determination of the excess air and fuel mass flow starting from the input data presented in Table 1 and Table 2.

Table 1: Parameters of the gas turbine cycle

Parameter	Value	Units
Compression ratio	5:1	
Intake temperature of Turbine (T_3)	1173	K
Output temperature of Compressor (T_2)	750	K
Air Mass flow	2.6	kg/s
Efficiency of air compressor	80	%
Efficiency of turbine	85	%

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Table 2: Chemical composition of the fuel

	Volume percent (%)	Density (kg/Nm³)
CH_4	50	0,656
CO_2	45	1.842
N_2	3	1.165
O_2	0.8	1.331
NH_4	0.6	0.73
H_2	0.2	0.089
CO	0.2	1.14
H_2S	0.2	1.434

The density of the biogas, according to the chemical composition in Table 2 is 1.21 kg/Nm³.
The low calorific power of the biogas was determined using Eq. 1:

$$H_i = 12720 \cdot (CO) + 10800 \cdot (H_2) + 35910 \cdot (CH_4) + 23400 \cdot (H_2S), kJ/Nm^3 \tag{1}$$

where (CO), (H_2), (CH_4) and (H_2S) represent the volume percent for each component of the biogas [10]. Thus a low calorific power of 18048 kJ/Nm³ was obtained. Or taking into account the biogas density, a low calorific power of 14889 *kJ/kg*.
Eq. 2 was used for determining the theoretical quantity of oxygen necessary for complete combustion [10]:

$$O_{min} = 0{,}5 \cdot [(CO) + (H_2)] + \sum(m + \tfrac{n}{4}) \cdot (CmHn) + 1{,}5 \cdot (H2S) - (O2) \left[\frac{m_N^3}{m_N^3}\right] \tag{2}$$

where (CO), (H_2), (C_mH_n), (H_2S) and (O_2) represent the volumetric participations for each component of the biogas [10]. Using Eq. 2, for m=1 and n=4, the following value was obtained: $O_{min}=0.997$.
The theoretical quantity of air necessary for complete combustion was determined using Eq. 3:

$$L_{min} = \frac{O_{min}}{0{,}21} \tag{3}$$

obtaining the value: 4.74.

The excess of air was determined using Eq. 4:

$$\alpha = \frac{H_i - c_{pg} \cdot T_3}{c_{pg} \cdot T_3 \cdot L_{min} - c_{pa} \cdot T_2 \cdot L_{min}} \tag{4}$$

where c_{pa} represents the specific heat of air and c_{pg} represents the specific heat of exhaust gases [9]. For $T_2=750\ K$ and $T_3 = 1173\ K$ an air excess of 5.37 was obtained.
For an air mass flow of 2.6 kg/s and using Eq. 5:

$$m_c = \frac{m_a}{\alpha \cdot L_{min}} \left[\frac{kg}{s}\right] \tag{5}$$

a fuel mass flow of 0.092453 kg/s was obtained.

The next step in designing the combustor was to determine the excess of air and the temperature alongside the liner.
For this purpose the combustor was divided into three regions:

1. The fuel injectors region;
2. The primary zone;
3. The dilution zone;

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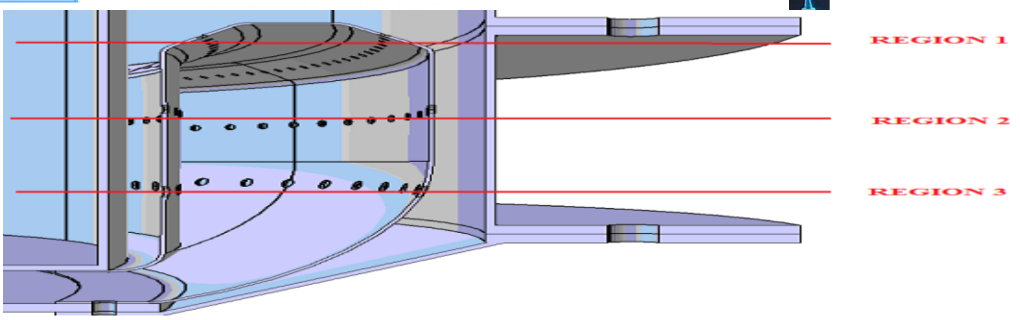

Figure 1: The three regions of the combustion chamber

Taking into consideration the information presented in the specialty literature [1], it was considered that 10% of the total air mass flow enters the fuel in the injectors region, 18% of the total air mass flow enters in the primary zone and the rest of 72% enters the dilution zone.

Table 3: Parameters resulted for the three regions of the combustion chamber

Regions	**1**	**2**	**3**
m_a(kg/s)	0.260	0.468	1.872
α (excess air)	0.593	1.661	5.932

The temperature along the combustion chamber was determined using Eq. 6:

$$T = \frac{efic \cdot H_i + \alpha \cdot L_{\min} \cdot T_2 \cdot H_i}{(1+\alpha \cdot L_{\min}) \cdot c_{p\_g}} \text{ [K]} \tag{6}$$

where *efic* represents the combustion efficiency. Its values were taken according to [11]. As expected the temperature maximum value is obtained in the primary zone.

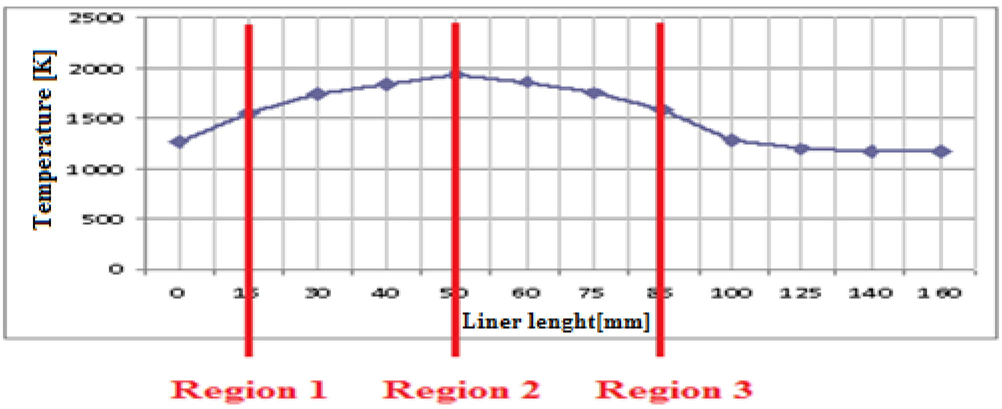

Figure 2: Temperature distribution

The final step in the design process was to determine the velocities outside and inside the fire tube and the liner holes diameters based on the jet penetration.

The pressure at the compressor's exit is 500000 Pa (p_2). Since the distance from the compressor exit to the combustor entrance is considerable due to the micro gas turbine constructive solution, it was considered that the air pressure at the combustor entrance is 475000 Pa.

The velocities outside and inside the fire tube have been calculated using Eq. 7:

$$v = \frac{\dot{m}}{\rho \cdot S} \left[\frac{m}{s}\right] \tag{7}$$

where S represents the section area.

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Table 4: Repartition of velocity exterior of the fire tube

Exterior air annular section			
r_{ext} (mm)	r_{int} (mm)	m_c (kg/s)	m_a (kg/s)
322	311	0.092453	2.6
S=21863.82 mm^2			
Interior air annular section			
r_{ext} (mm)	r_{int} (mm)	m_c (kg/s)	m_a (kg/s)
261	250	0.092453	2.6
S=17649.94 mm^2			
Region	**1**	**2**	**3**
Pressure loss (dp%)	1	2	3
Pressure (Pa)	467775	463050	458325
Air density (ρ_a)	2.1797	2.1577	2.1357
Mass flow (kg/s)	2.6	2.34	1.872
v(m/s)	30.1862	27.4448	22.1822

Table 5: Repartition of velocity interior of the fire tube

r_{ext} (mm)	r_{int} (mm)	m_c (kg/s)	m_a (kg/s)
311	261	0.092453	2.6
S=89804 mm^2			
Region	**1**	**2**	**3**
Pressure loss (dp%)	1.25	2.25	3.25
Pressure (Pa)	466593.75	461868.8	457143.8
Fuel density (ρ_g)	0.9359	0.8344	1.3338
Mass flow (kg/s)	0.3524	0.8204	2.6924
v(m/s)	4.1934	10.9490	22.4776

The jet penetration was calculated using Eq. 8 [11]:

$$H_{ji} = 3.1 \cdot \phi \cdot \left(0.3 + 0.415 \cdot \frac{V_a}{Vg}\right) \tag{8}$$

Table 6: Jet penetration

Region	1	2	3
Air Velocity(V_a)	30.18	27.44	22.18
Fuel Velocity(Vg)	4.19	10.94	22.47
Hole diameter ϕ (mm)	2.5	5	6.5
Jet penetration (H_{ji})	25.49	20.78	14.29

Based on the calculations presented above a first version of the combustor geometry, presented in Fig. 3, has resulted.

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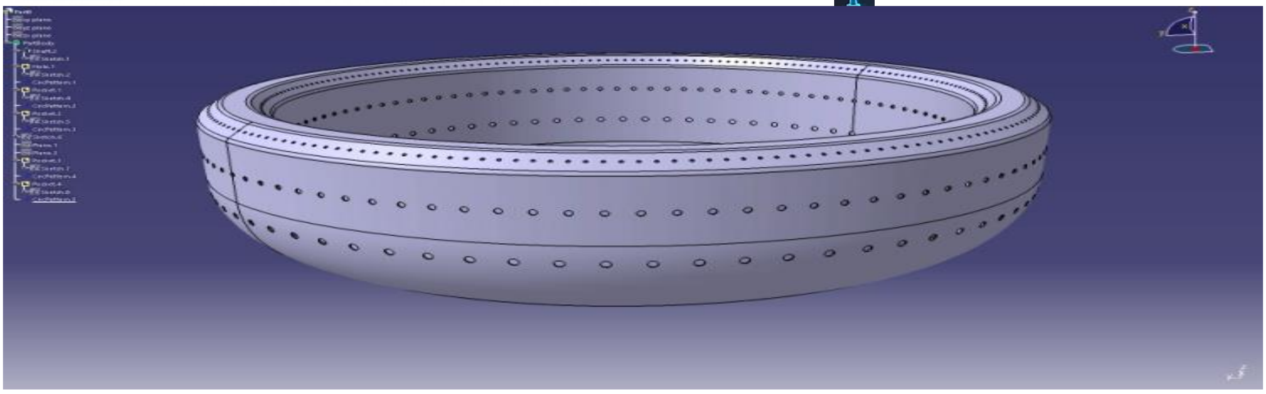

Figure 3: Fire tube

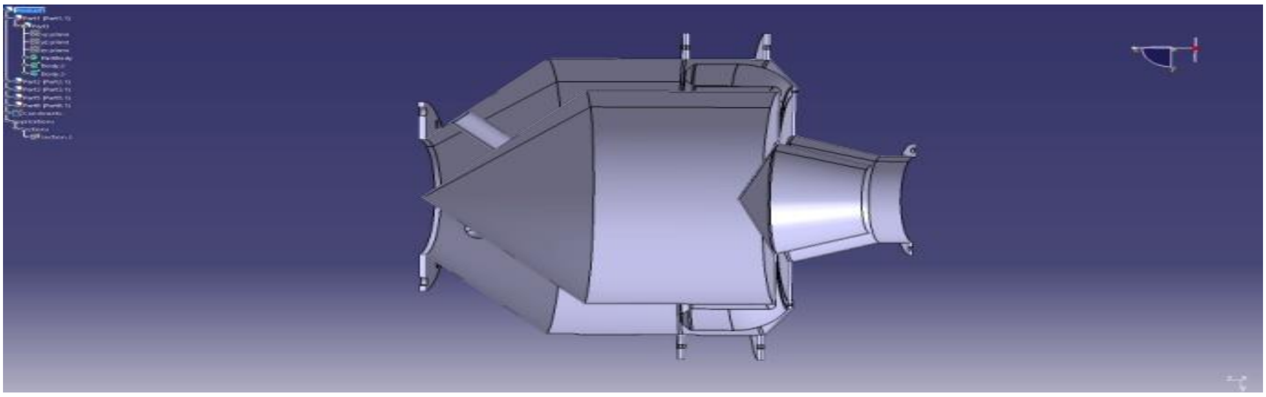

Figure 4: Combustion chamber assembly

3. Three-dimensional numerical simulation results

In order to obtain quicker results due computing limitation, the model was simplified into a 22.5-degree cut section for the combustor. The computational aerodynamic analysis is carried out to validate theoretical results and to obtain a detailed preview of the outcome design.
The numerical simulations were carried out using ANSYS CFX software. The RANS approach was used with an unstructured type mesh has been generated for the computational domain and with the Domain Motion Stationary. The total number of Tetrahedral Elements is 8231719.
The following boundary conditions were used. At air inlet, there were imposed the air mass flow and temperature, at fuel inlet there were imposed the fuel mass flow and temperature and at the outlet, the pressure was imposed.
The Eddy Dissipation combustion model was used, in combination with the K-epsilon turbulence model. This model was chosen because it allows accurate simulation of the heat release and the distribution of the main chemical species. Reaction mechanism used in the simulation was Methane-Air WD2 in two steps where mass fraction for CO_2 and CH_4. The other elements from the chemical composition of the fuel are neglected due to their the small volume percentage.
The velocity distribution presented in Fig. 5 shows high velocity in the central region. Jet penetration is very strong, thus the created turbulence will affect combustion process.

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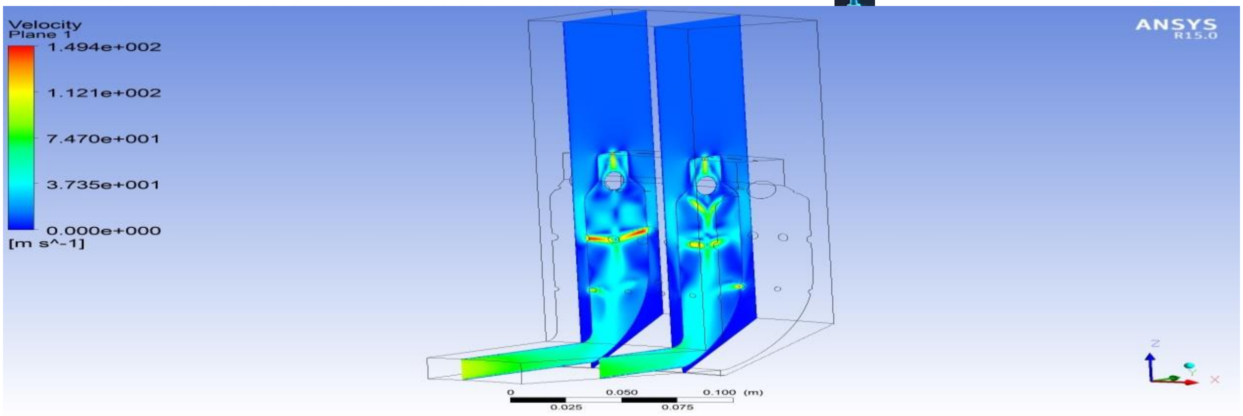

Figure 5: Velocity distribution

From Fig. 6 it can be observed that the flame temperature presents high values mainly near the combustor's walls. This is in good correlation with the velocity profile presented in Fig. 5. The high-velocity values from the central region of the fire tube make difficult the mixing of the air and fuel in the primary region.

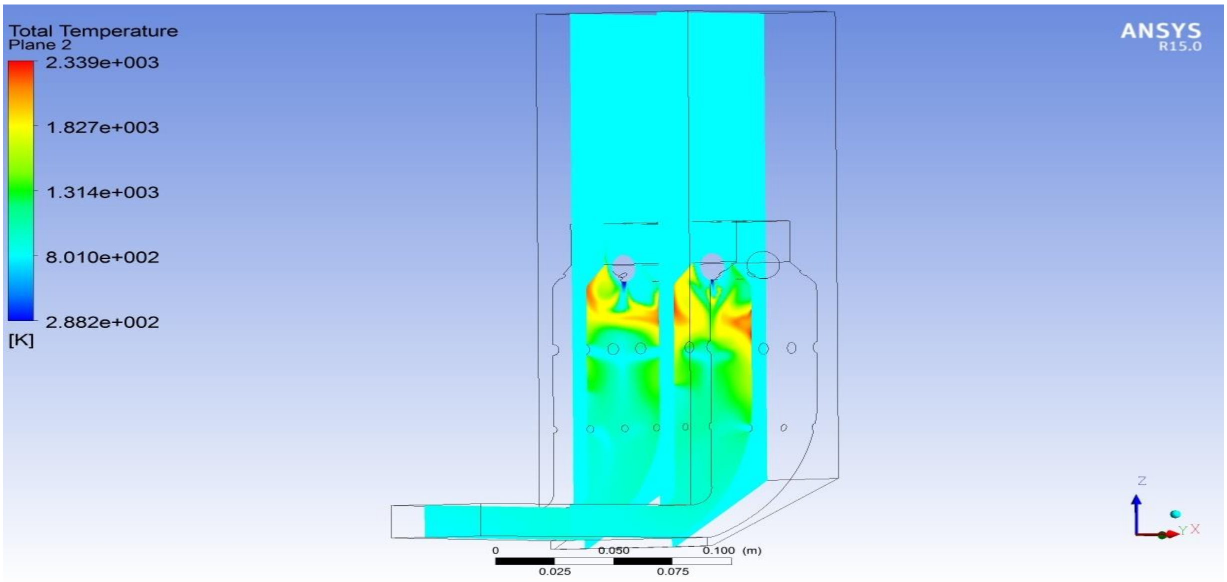

Figure 6: Total temperature distribution

In Fig. 7 are presented 4 temperature iso-surfaces inside the fire tube (red=2200 K, orange=2000 K, yellow=1800 K, green 1500 K). As it was observed before, the high flame temperatures are developing near the walls. This is not a good sign. It can lead to serious damage to the fire tube. It also can be seen that the flame has a very irregular structure.

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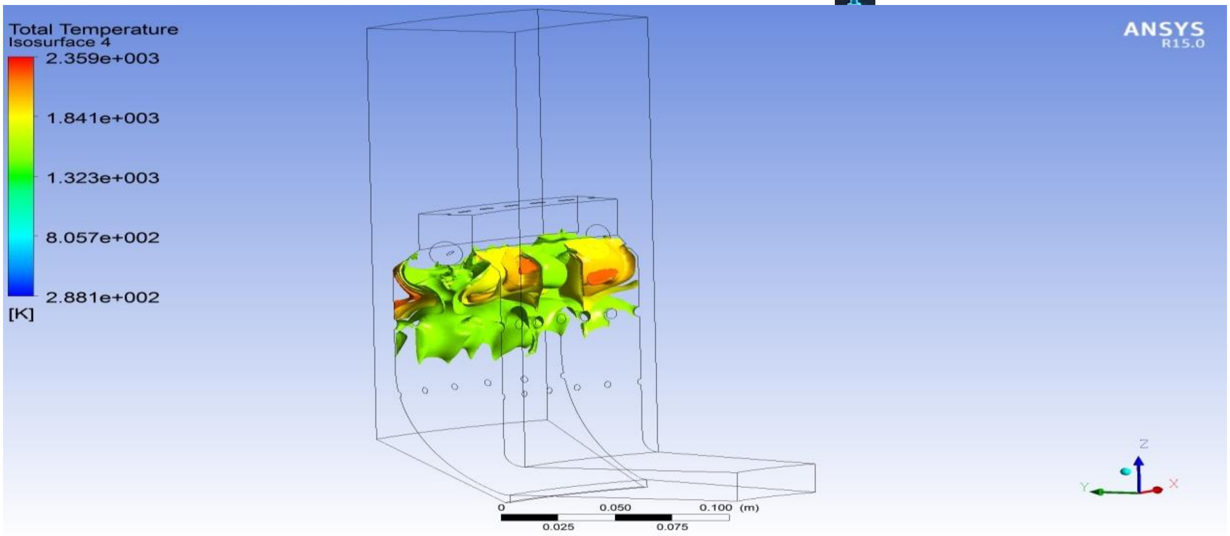

Figure 7: Total temperature Isosurface distribution

The average temperature at the exit of the combustor was 1150K. Even though this temperature value is very close to the one imposed in Table 1, based on the results obtained so far it was concluded that same changes have to be done to the combustor geometry.
A deflector was added to the original geometry in order to concentrate the flow in the central region of the fire tube and to prevent flame adhesion to the walls of the combustor. The improvement can be observed in Fig. 8 and Fig. 9. The flow velocity has diminished and the flame is concentrated in the center of the fire tube.

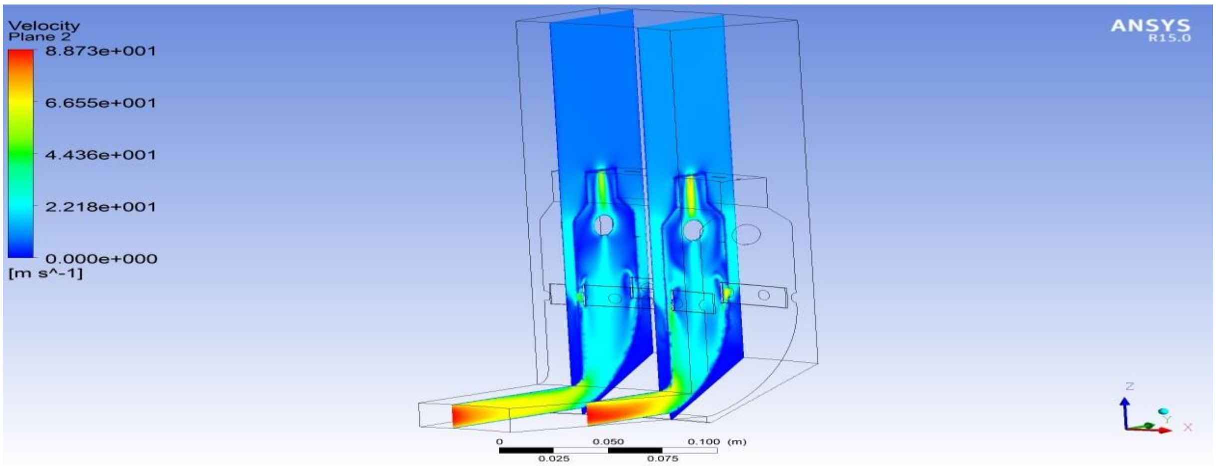

Figure 8: Velocity distribution

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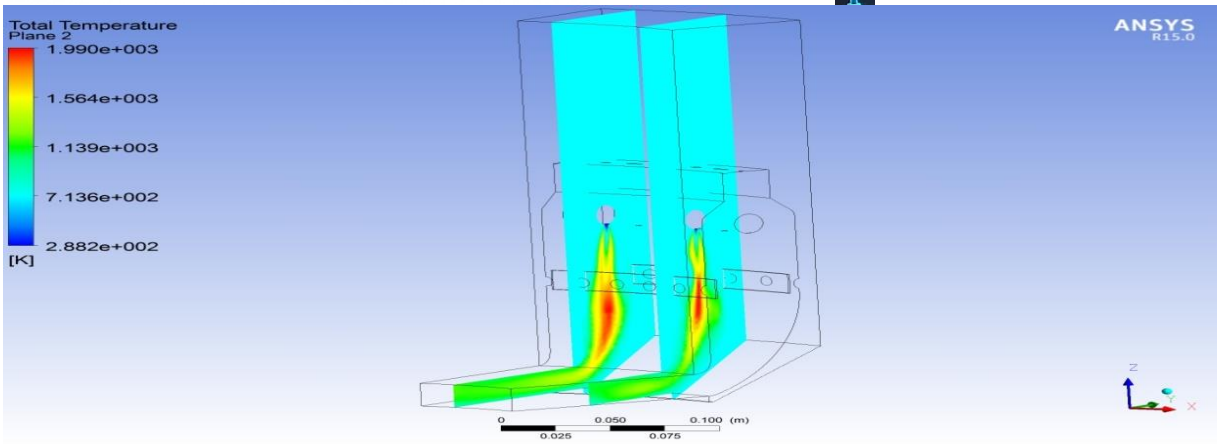

Figure 9: Total temperature distribution

In Fig. 10 the temperature distribution along the fire tube is presented. As it can be seen, the temperature rise up to 1400 K in the region of the injectors, afterward reaching a maximum of 2100 K in the primary zone. After that, the addition of air leads to a decrease of the flame temperature.

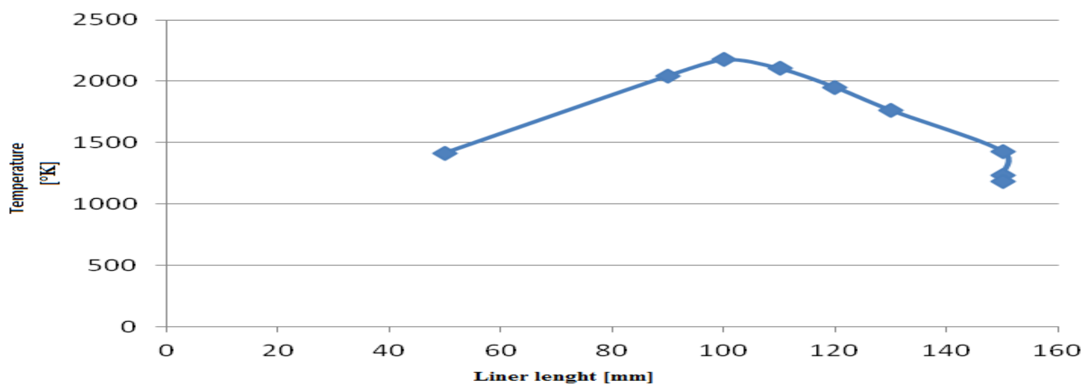

Figure 10: Total temperature distribution

Form Fig. 11 it can be seen that the flame has a uniform structure and the flame does not exceed the fire tube length. Thus it won't affect the turbine.

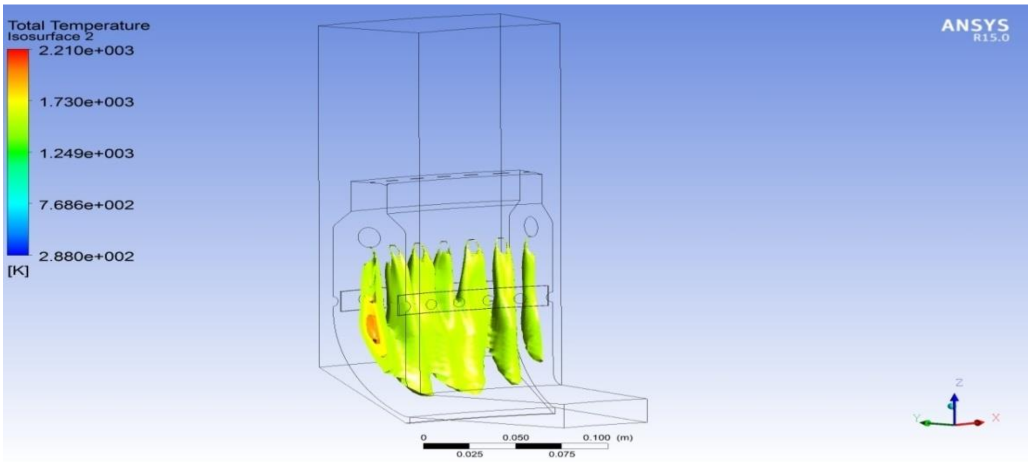

Figure 11: Total temperature Isosurface distribution

The average outlet temperature is 1180 K, being close to the required value.

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4.Conclusions

The complete annular combustor design using just the initial design parameters has been discussed in this paper. This is a design methodology which can be used for the preliminary design. The transparent and detailed approach is focused on reducing design time and complexity. This gives an overall advantage in total design time and prototype building. Using the methodology, a practical design is presented. The obtained values are used for modeling and further simplified for analysis. The analysis was also carried out with higher accuracy using the combustion-turbulence interaction model and the results show that the optimum gas exit temperature was obtained for the present design. The design was successfully calculated and modeled.

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