

DESIGN OF A COMBUSTOR FOR A BIOGAS DRIVEN MICRO-TURBINE AND MODELING OF THE COMBUSTION PROCESS

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Abstract: Micro-turbines are employed for small scale power generation and have advantages over other power generating machines in that they are compact, have few rotating parts and low vibrations. Micro-turbines driven by biogas have additional advantage of utilizing a renewable fuel source which results in cleaner environment. The dimensions of the casing, inner and outer liners, as well as primary and secondary holes were estimated based on empirical relations and standard equations for a 10kW micro turbine specification. The model of the designed combustor was modeled using Solid works and the simulation of the combustion was carried out using ANSYS 16.0 CFX software. The temperature variation, density variation, flow velocity, methane mass fraction and pressure loss in the combustor were therefore evaluated. The combustor had a maximum temperature of 1800K at the combustion zone and dropped to 900K at the exit. The density decreases drastically from 1.85 kg/m³ at the inlet to 0.7 kg/m³ at the combustion zone. The methane mass fraction increased from 0 at the inlet to 0.0025 at the combustion zone and dropped back to 0 at the exit. It was observed that the micro turbine combustor which uses biogas as fuel tends to require higher volume for optimal performance when compared with micro-turbines of same power ratings that are fueled with natural gas.

Keywords: Biogas, combustor, Micro-turbine, model, methane

1. INTRODUCTION

Micro turbine combustor is one of the major components that need to be painstakingly designed in a turbine engine for optimal performance. Combustor design needs to meet some requirements such as low emission, minimum pressure loss, complete combustion of fuel, uniform exit temperature among others. Many combustors tend to have better efficiency when they are run on specific fuel in which they are designed for. Some biogas users directly channel this fuel to their generating sets which could damage the cylinder and piston because of its corrosive nature if not treated before use. Loss of efficiency could also be recorded if turbine engine combustors designed to operate using natural gas are run on other types of fuel. Because of the increased awareness that fossil fuels are non-renewable and therefore are expected to get depleted in coming years, renewable sources like biogas become paramount and special combustors are needed to burn them. This paper therefore aims to design biogas combustor and investigates its performance using a computational fluid dynamics software package. In the first part, the definitions of combustor flow parameters and combustor dimensions parameters are presented. The equations used to determine the dimensions are equally presented and explained which are basically based on the empirical relations available in literature. Thereafter, the 3-D model of the combustor is presented, flow simulation is performed and results are presented.

2. MATERIALS AND METHODOLOGY

The design methodology used is similar to the one described by Melconian and Modak, [1] and Conrado [2]. The inlet flow conditions into the annular combustor are got from the outlet conditions of the centrifugal compressor designed to operate on a pressure ratio of 2.8, 0.14 kg/s air flow rate for a 10 kW micro gas turbine. The simple Brayton cycle is given in fig. 2.0

Cycle of the micro turbine is shown in Fig 2.0 below:

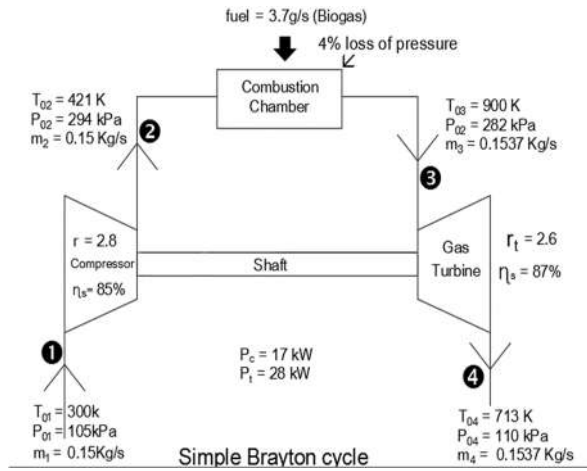


Fig. 2.0 Brayton Cycle for a 10kW micro gas turbine.

2.1. Preliminary Design Procedure

The design procedure is presented in figure 2.1. Though the major steps have been followed, the swirler and diffuser designs are not the focus of attention in this paper.

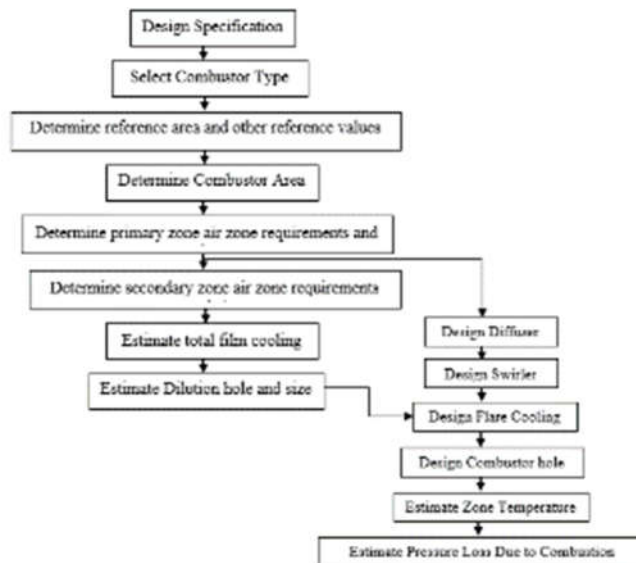


Fig. 2.1 Major steps for combustor design

2.1.1. Parameters at Combustor Inlet

As pointed out earlier, the parameters at combustor inlet are got from the centrifugal compressor design for a 10kW micro gas turbine. These parameters are presented in Table 1 below:

Table 1: *Inlet flow parameters and fuel data*

| No. | Parameter | Value | Units |
|-----|------------------------------------|--------|-------|
| 1 | Inlet air temperature, T_4 | 421 | K |
| 2 | Inlet fuel temperature, T_{fuel} | 300 | K |
| 3 | Inlet air pressure, P_4 | 270 | kPa |
| 4 | Air mass flow rate, \dot{m}_4 | 0.14 | kg/s |
| 5 | Fuel mass flow rate, \dot{m}_f | 0.0037 | kg/s |
| 6 | Air fuel ratio | 38 | |
| 7 | Heating value of the fuel | 19000 | kJ/kg |

Biogas chemical composition varies according to the biomass used during preparation. The composition used here is based on the one presented by Gokalp and Lekas [3].

$$\left. \begin{array}{l} CH_4 = 61\% \\ CO_2 = 34\% \\ N_2 = 5\% \end{array} \right\} \text{percent by volume}$$

Three major zones of an annular combustor and air apportioning into sections of the combustor are shown in figure 2.2. Figure 2.3 shows its cross section with major dimensions.

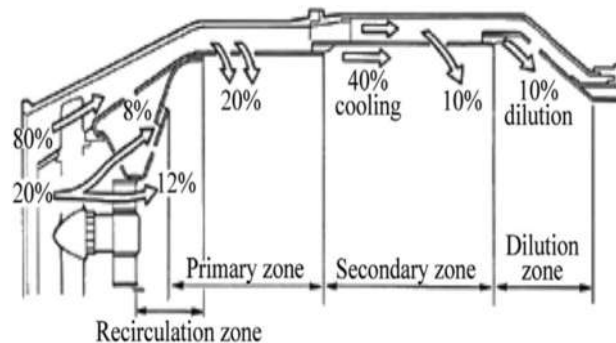


Fig. 2.2 Air flow apportioning in annular combustor [4]

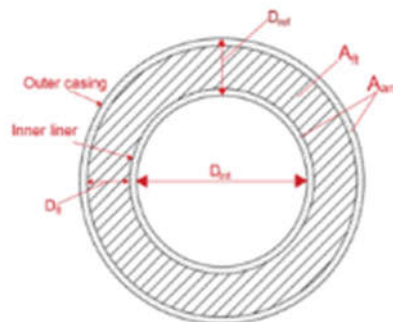


Fig. 2.3 Cross section of combustor showing dimensions

2.1.2 Estimation of Combustor Dimensions

2.1.3 Reference Area, A_{ref}

A_{ref} is the area of space bounded by the outer casing and the inner casing as shown in figure 2.3 by D_{ref} (reference length). The A_{ref} is given by equation 1.

$$A_{ref} = \left[\frac{R}{2} \left(\frac{\dot{m}_4 \sqrt{T_4}}{P_4} \right)^2 \left(\frac{\Delta P_{4-5}/q_{ref}}{\Delta P_{4-5}/P_4} \right) \right]^{0.5} \quad (1)$$

R is the gas constant = 287 J/kg.K

$\Delta P_{4-5}/P_4$ in equation (1) represents the ratio of pressure drop across the combustor to the inlet pressure and is about 0.06 for annular combustors [5] and also (Sharma et al, 2015).

$\Delta P_{4-5}/q_{ref}$ is the ratio of pressure drop to the reference dynamic pressure. Its value used by (Sharma et al, 2015) for annular combustor is 53. A_{ref} can therefore be calculated from equation (1).

$$A_{ref} = 0.00365 \text{ m}^2$$

Also, [2] presented equation for reference area of annular combustor in terms of their internal and reference diameters given by equation (2)

$$A_{ref} = \left[\pi \frac{(2D_{ref} + D_{int})^2}{4} - \frac{\pi(D_{int})^2}{4} \right] \quad (2)$$

D_{int} is chosen to accommodate aerodynamic considerations in all operating conditions.

$$D_{int} = 96 \text{ mm}$$

D_{ref} (reference length) is calculated from equation (2) having known A_{ref} from equation (1).

$$D_{ref} = 11 \text{ mm}$$

Diameter of outer casing, D_c from the geometry in figure 2.3 is given by equation (3)

$$D_c = 2D_{ref} + D_{int} \quad (3)$$

2.1.5 Combustor Sectional Area

The combustor sectional area is determined from equation (4)

$$A_{ft} = 0.7 A_{ref} \quad (4)$$

Where A_{ft} is the combustor sectional area

$$A_{ft} = 0.00256 \text{ m}^2$$

2.1.6 Annulus Area

Annulus area is given by equation (5). It represent the difference between A_{ref} and A_{ft}

$$A_{an} = A_{ref} - A_{ft} \quad (5)$$

$$A_{an} = 0.00109 \text{ m}^2$$

2.1.7 Combustor Zone Length

Primary Zone: The length of the primary zone can be calculated from equation (6)

$$L_{PZ} = \frac{3}{4} D_{ft} \quad (\text{Melconian and Modak, 1985}) \quad (6)$$

$$L_{PZ} = 8.25 \text{ mm}$$

Secondary and Dilution Zones: The combined zones length is given by equation (7)

$$L_{SZ} + L_{DZ} = 0.5D_{ft} + 1.5D_{ft} \quad (7)$$

$$L_{SZ} + L_{DZ} = 22 \text{ mm}$$

$$\text{Total combustor length, } L_{TC} = L_{PZ} + L_{SZ} + L_{DZ} = 30.25 \text{ mm}$$

Equations (1) to (7) describe the estimation of basic geometric parameters often used in annular combustor design such as the combustor area and length which together represents the combustor volume. Estimation of total holes' area and number of holes required at the primary and other zones are described next.

2.1.8 Combustor Holes' Diameter Determination

The six steps iteration method described by (Conrado et al, 2004) is used in estimating the required number of holes and diameter needed. The primary holes are determined as follows:

1. Determination of bleed ratio, B given by

$$B = \frac{\text{mass flow rate of air into the holes, } \dot{m}_h}{\text{mass flow rate of air in the annulus, } \dot{m}_{an}} \quad (8)$$

40% of mass flow in the annulus is made to enter the primary zone; $B = 0.4$

$$\dot{m}_4 = \dot{m}_{an}, \quad \dot{m}_h = 0.4\dot{m}_4 = 0.056 \text{ kg/s}$$

2. Hole discharge coefficient $C_{dh} = 0.5$ (assumed as the guess)
3. Total hole area is determined to be

$$A_{ph} = \sqrt{\frac{\frac{R}{2} \dot{m}_h^2 T_4}{\frac{\Delta P_h}{P_4} P_4^2 C_{dh}^2}} \quad (9)$$

$\Delta P_h/P_4$ represents the pressure loss factor due to the presence of the holes and is approximately 0.06 (Conrado et al, 2004).

$R = 287 \text{ J/kg.K}$ while T_4 and P_4 are given in table (1)

4. Determination of orifice area ratio, $\alpha_o = \frac{A_{ph}}{A_{an}} =$ and bleed/orifice area ratio $\mu = \frac{B}{\alpha_o}$
5. In equation (10) below, K is the pressure loss factor and δ is the loss due to momentum. $\delta = 0.85$ for this design.

$$K = 1 + \delta^2 [2\mu^2 + \{4\mu^4 + (\mu^2/\delta^2)(4B - B^2)\}^{0.5}] \quad (10)$$

6. The value of the discharge coefficient is tested using equation (11) below

$$C_{dh} = \frac{(K-1)}{\delta [4K^2 - K(2-B)^2]^{0.5}} \quad (11)$$

Equation (11) is used to recover C_{dh} and compare its value with the initial guess of 0.5. If close enough, then A_{ph} given by equation (9) represents the actual total primary hole area. The hole diameter is gotten from equation below having decided the number of primary holes, N_{ph} .

$$D_{ph} = \sqrt{\frac{4A_h}{\pi N_{ph}}} \quad (\text{from area of a circle}) \quad (12)$$

We determined the area of primary hole using the method described above as

$$A_{ph} = 0.000359 \text{ m}^2$$

Diameters of primary holes, $D_{ph} = 3\text{mm}, 2\text{mm}$

Total number of primary holes, $N_{ph} = 41(3\text{mm}), 20(2\text{mm})$

Notice that two different diameters are used, this is done in order that flow recirculation is increased for continuous burning of fuel. The total area of the holes is still conserved even with two different diameters.

The same method is used for secondary and dilution zones lumped together by making $B = 0.6$ in step 1 and follow the same procedure to get the hole area required.

Area of Secondary and Dilution holes put together is

$$A_{sdh} = 0.0006 \text{ m}^2$$

Diameter of secondary and dilution holes are

$$D_{sdh} = 3\text{mm}, 4\text{mm}, 5\text{mm}$$

Total number of dilution holes (inner and outer liners)

$$N_{sdh} = 18(5\text{mm}), 16(4\text{mm}), 15(5\text{mm})$$

After the estimation of combustor geometric parameters, the amount of fuel required to raise the temperature at inlet of the combustor to 900K is estimated from chemical reactions of the species in the combustion chamber.

Amount of fuel required is as follow

Inlet temperature to the combustor, $T_4 = 421\text{K}$ and outlet temperature is $T_5 = 900\text{K}$

Enthalpy increase of air,

$$\Delta H = \dot{m} C_{p_{avg}} (T_5 - T_4) = 0.14 \times 1.063 \times (900 - 421)$$

$$\Delta H = 71 \text{ kJ/s}$$

$$\text{Mass flow rate of fuel required, } \dot{m}_f = \frac{\Delta H}{\hat{h}_c} = \frac{71 \text{ kJ/s}}{19000 \text{ kJ/kg}}$$

Where h_c is the heating value of biogas composition used.

$$\dot{m}_f = 0.0037 \text{ kg/s as presented in table 1.}$$

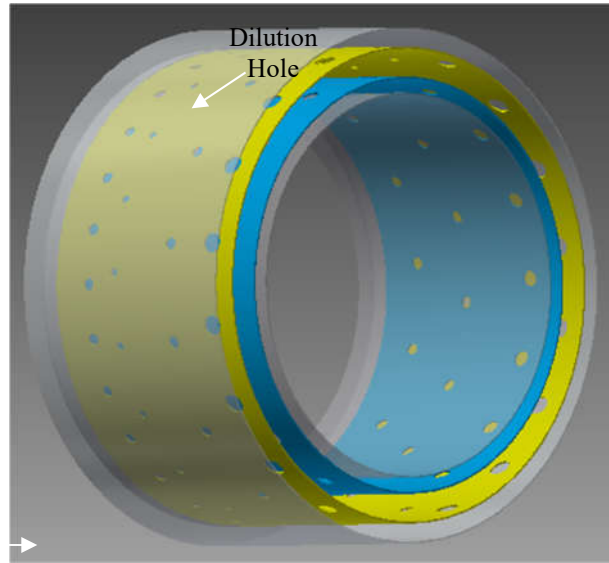


Fig 2.4: Annular combustor model

2.2 COMBUSTOR FLOW SIMULATION

ANSYS CFX 16.0 is used in carrying out simulation of combustion reaction in the combustor. Five steps are taken on ANSYS before the results of simulation could be achieved.

- i. Modeling of fluid domain
- ii. Discretization of fluid domain in meshing environment.

Table 2: Mesh statistics and relevant information for combustor simulation

| N o | MESH INFORMATION | VALUES |
|--------|---------------------|--------------|
| 1 | Elements | 2434787 |
| 2 | Nodes | 473261 |
| 3 | Smoothing | Low |
| 4 | Solver preference | CFX |
| 5 | Mesh arrangement | Unstructured |

- iii. Problem Set up

This involves setting of boundary conditions, specifying air/fuel flow rates, type of reaction model and turbulence flow model among others. Some basic settings on ANSYS are presented in table 3.

Table 3: Combustor simulation requirements and response using CFX

| Requirements | Response | Requirements | Response |
|------------------|------------------------|--------------|------------------|
| Turbulence model | Shear stress transport | Fuel inlet | 0.0037kg/s, 300K |
| Combustion model | Eddy dissipation | Air inlet | 0.14kg/s, 421K |

| | | | |
|--------------------|---|-----------------------|--------------------|
| Reaction list | Methane Air WD1 | Flow direction | Normal to boundary |
| Components model | CH ₄ , CO ₂ , H ₂ O, N ₂ , O ₂ | Static pressure inlet | 270kPa |
| Maximum flame temp | 1800K | Domain motion | Stationary |
| Reference pressure | 1 atm | Maximum iterations | 1000 |

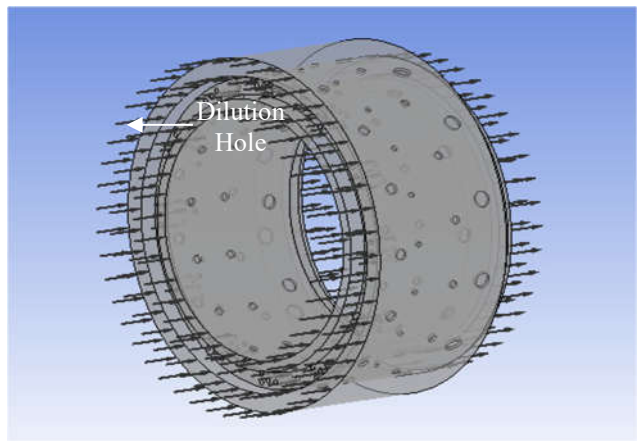


Fig 2.5: Problem set up for simulation of annular combustor.

3. RESULTS AND DISCUSSION

The result of the fluid flow simulation for the annular combustor is presented in Fig 3.1 through 3.5. The Global temperature distribution in the combustor is shown in figure 3.1. Streamlines of cool air can be seen entering the primary and dilution holes. The red portions show region of high temperature during combustion before cooling down to the yellow portion. Maximum temperature is observed to be 2128 K while the minimum temperature is 300K.

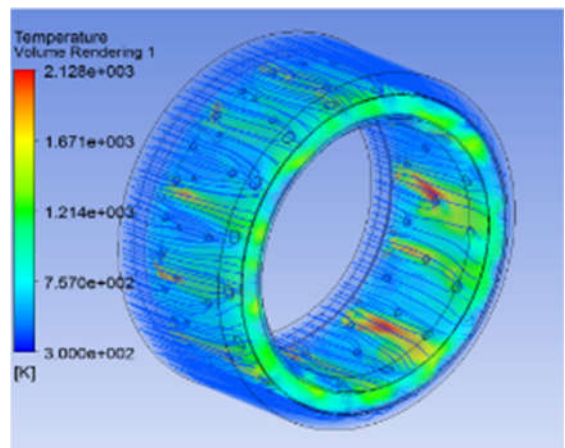


Fig 3.1 Global temperature distribution in the combustor

The local temperature contour at the combustor outlet is shown in fig 3.2. The maximum temperature is 1430K which is much lower than the global maximum temperature of 2128K.

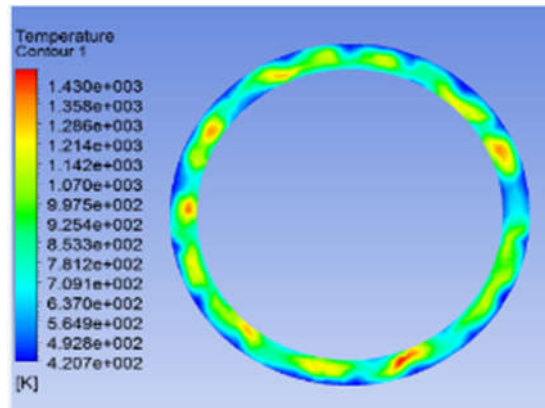


Fig 3.2 Local temperature contour at the combustor outlet

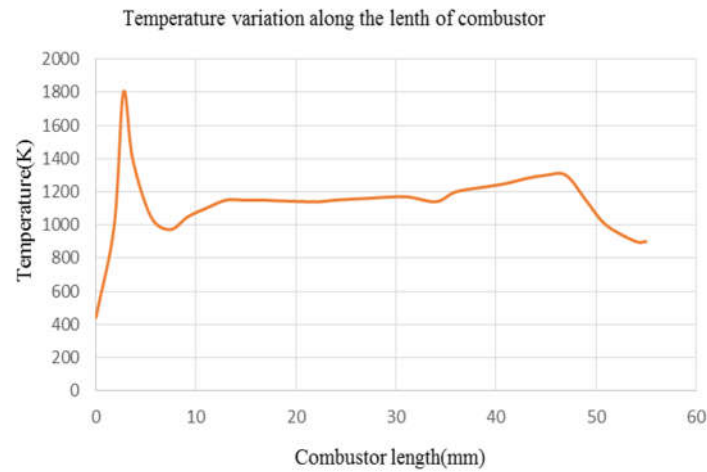


Fig 3.3 Temperature variation along the combustor length

Fig 3.3 presents the temperature variation along the length of the combustor. The lowest temperature of 443 K is at the inlet when unburnt gas and air are introduced into the combustor to 1800K when the gas is burning to 900 K when combustion has been completed.

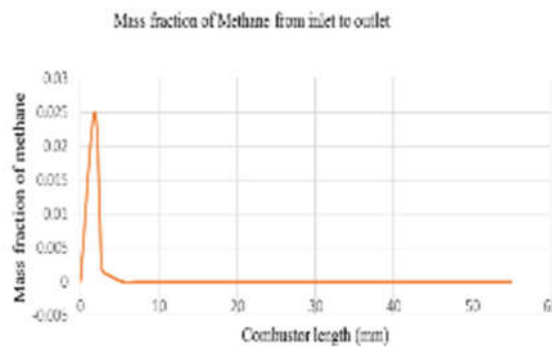


Fig 3.4 Mass fraction of CH_4 variation along the length of combustor

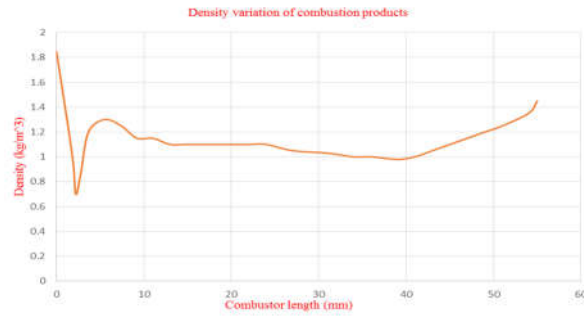


Fig 3.5 Density variation of combustion products along the length of combustor

The density in fig. 3.5 reduces from 1.85 kg/m^3 when the gas is introduced and combustion is initiated to 0.7 kg/m^3 when combustion takes place to 1.45 kg/m^3 when the gases exit the combustor.

4. CONCLUSION

The design of a combustor model of a biogas micro gas turbine has been discussed using the procedure described in this paper. The geometry was optimized by simulation of fluid flow in the combustor. This methodology requires less design time and shows that biogas combustor tends to require larger volume than conventional fuels.

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