Design and Fabrication of an Annular Combustion Chamber for the Micro Gas Turbine Engine Applications

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Abstract-Combustion and its control are essential for the existence of an engine as we know. Investigators put their energies from many years on combustion and its stabilization. Temperature rise, mixing, liner cooling, permanence, fuel effects, temperature profile control and emissions control continue to brazen out the aerodynamic and mechanical designers with a embarrassment of engineering dilemmas and trade-offs. This paper illuminates on a small facet combustor and addresses the designing and modeling process of an annular combustion chamber for the application of micro gas turbine engines. The corporeal challenges of its design are to blaze the fuel in a small radial passageway of 41 mm aspect. For minimizing the machining complications of the combustor components, the combustor exit is through the holes in place of exit ring. Based on these challenges, some strategies for micro combustor are developed and explain here. The complete designing and modeling processes are carried out by using CATIA V5R19 design software.

Keywords: ***Flow turning (30^0 swirlers), working fluid LPG, Combustor exit design technique.***

I. INTRODUCTION

In a gas turbine engine a high rotational speed in an atmosphere of elevated temperature is really a challenge for the designers and also the selection of material is possible only based on the research on materials having strength at high temperature. Here the design is in Mini and Micro level, so the percentage of thermal expansion of high temperature exposed material should be according to the given dimensions of the engine. Some new materials for example Si_3N_4 and SiC have enough strength even at elevated temperature. The developments of gas turbine engine technologies enable dimensions in millimeters or centimeters. These assemblies are known as power MEMS or micro gas turbines and getting emergent interest in the last few years. The applications of these engines are mainly

suitable for small scale power generation or for the MAVs and UAVs propulsion. This mini and micro size gas turbine engines eradicate fully or partially the weight of the on board batteries [ref. (3-5)]. The combustion chamber, Heart of an engine is one of the most critical mechanisms to be designed for achieving the goal. The significance of the combustor design is a need for suitable operation over extensive range of air/fuel ratios. Compared to early days investigations improve the design and performance of the combustion chamber that exhibits nearly 100% combustion efficiency over their nominal working range, also improve the phenomena of pressure loss and substantial reduction in pollutant emissions and allow a linear life. This paper is focused on the design, fabrication and testing of an annular micro combustion chamber for the application of micro gas turbine engines. Even if the scale reduction of the flame tubes arise some difficulties for the other components too, these will not be addressed here [ref (6-9)]. The special feature of this combustor is that the fuel injection having two options, first is along with the flow and the other is tangential to the flow. The swirl vanes are design in such a manner that flow turns 30 degree from the previous path. This combustor has also provision for the flame tube cooling; one row of holes of both the inner and outer domes are made parallel to the axis of the combustor compared to the other rows of holes, which are at 30 degree angle with the axis of the combustion chamber. As I concise in the abstract that the exit of the combustor has number of holes in place of the slit for over come from the fabrication complications.

II. MAJOR CHALLENGES

The sight of the Micro combustor minimize mirror image of a conventional combustor, this is prime functional requirements. The combustion efficiency should be high enough to convert the chemical energy into thermal energy and kinetic energy with lower percentage of stagnation pressure loss, optimum flame stability and low NO_x emissions. The micro level design and development of combustor faces material challenges and also thermodynamic constraints. These challenges are shortly reviewed here in the next paragraphs.

II.A. CONSTRAINTS OF THE COMBUSTOR STAY TIME

The requirements of high power density by a combustor needs high mass flow rate per volume. Since the chemical reaction time do not scale with mass flow rate, the required high power density depends on whether combustion can be completed efficiently within a shorter combustor through flow time or not. These limitations have most significant technical challenging aspects of the design of a micro combustor. This is only possible if available stay time is greater than the needed reaction time.

II.B. TIME AVAILABLE RESTRICTIONS

The combustor stay time depends upon the flow rate trough the combustion chamber.

$$\tau_{res} = \frac{VP}{\dot{m} RT}$$

The above equation states that stay time can only be augmented by adopting a longer chamber or a lower volumetric flow rate. Reduction of combustor flow rate is possible by decreasing the mass flow rate or by increasing the static pressure. The operational speed depends upon the compressor/ rig and cannot be increased while the high mass flow rate per volume is desirable for the required high power density. The stay time is fully depends upon the required power density. The primary zone has strong recirculation design, which can augment stay time. The idea which gives artificially lengthens the path of the gases in the combustor, results longer stay or residence time without moderating the power density. The optimum ignition of incoming mixture and for high degree of flame stabilization, the idea to incorporate a recirculation zone is most universal.

II.C. TIME REQUIRED RESTRICTIONS

The complete combustion of the fuel is only possible by the rapid chemical reactions of the combustible mixture.

$$\tau_{reac} = \frac{[f]}{A*[f]^a*[o_2]^b*e^{\frac{-E_a}{RT}}}$$

The above equation shows that the reaction time is strongly influenced by the fuel choice. The lesser the energy required, the lower the reaction time. The other imperative parameters are air to fuel ratio and the combustor gas temperature; the stay time has noteworthy effects on preheating of the fuel to the combustor. The above equation also shows that this will effect on both E_a and T. However due to low Reynolds number in the combustor, the time needed to mix the fuel and the air are also critical as well as the injection conditions of the fuel. The stay time will be more for liquid fuel compared to gasses fuel, because liquid needs time to evaporate the droplets.

II.D. CONSEQUENCE OF HEAT TRANSFER

Generally the heat transfer or loss is neglected for a conventional gas turbine engine. The reduction of chamber dimensions due to increased surface to volume ratio, the heat loss effect is a fundamental factor here. The production of heat inside the combustion chamber is nearly proportional to the volume of the combustor and the loss of heat is almost proportional to the surface area, the surface to volume ratio leads to a significant increase in relative heat loss. Reference [11] gives the mathematical relationship with the scaling and the ratio of heat lost to heat generated.

$$\frac{Q_{los}}{Q_{gen}} \alpha \frac{1}{D_h^{1.2}}$$

The micro combustor dimensions are in millimeters, so that heat ratio will be greater than large scale combustor. The performance of the combustor due to large surface heat loss has seen in two different ways. First, the overall combustion efficiency is directly affected due to large thermal losses. Second, the large heat losses can increase kinetin reaction times and narrow flammability limits through lowering reaction temperatures.

III. COMBUSTOR PRINCIPLES FOR ANNULAR COMBUSTOR

According to physical constraints and scaling effects, the idea behind the designing of a micro combustor should be based upon the following concepts.

i. Relative to the engine size, combustor dimensions are increased slightly which increases combustion stay time or residence time.

ii. Premixing to decrease the time inside the combustion chamber.

iii. Burning should be lean to make the turbine inlet temperature tolerable.

As a large part of the combustion stay time in the current combustor is devoted to fuel air mixing, removal of this mixing from the combustor seems the only way to meet the residence time requirement. However, if the reactants are mixed upstream, the stability benefits of a near stoichiometric primary zone are lost. At low equivalence ratios, the stable burning is a must seen the limits on the turbine inlet temperature. This is possible by two different ways; the use of LPG or catalytic combustion of hydrocarbons. The latter solution will not be covered here as it seems to entail significant problems for micro scale applications (ref [11]). Compared to other hydrocarbons, gases LPG is ideal fuel for micro level design such as greater heating values, a rapid vaporization, a faster diffusion velocity, a short reaction time and a significantly higher flame speed. The most important thing is the broad flammability limits remove the requirements for a relatively rich primary burning zone which is necessary for hydrocarbon fuels. This allows an upstream mixing with the fuel air ratio required by the allowable turbine inlet temperature. This implies that LPG is certainly the first fuel to be investigated if successful; another gases fuel like propane could be studies later (ref [14]).

IV. DESIGN AND DEVELOPMENT

It is complex and also difficult to design a gas turbine engine. The difficulties are usually solved by reaching a reasonable compromise between the conflicting requirements. The gas turbine engine involves a broad range of technical disciplines including combustion chemistry, fluid dynamics, heat transfer, stress analysis and metallurgy. The major design parameters for the combustor designs are listed below.

a. Combustion chamber inlet conditions
b. Cooling air and the flow distribution
c. Exit pressure of the combustor
d. Pattern factor
e. Flame tube length
f. Exit area of the combustion chamber

Table 1: COMBUSTOR DESIGN PARAMETER

Inlet Temperature (T_{03})	330 K
Mass Flow rate of Fuel ($\dot{m}_f$)	2.97×10^{-2} Kg/s
Inlet Pressure(P_{03})	2.95 bar
Mass Flow rate of Air($\dot{m}_a$)	0.46 Kg/s
Fuel to Air Ratio	6.46×10^{-2}

IV.A. COMBUSTOR REFERENCE AREA

There are many constraints with which the combustor design is achieved. The dimension is decided on the size of the compressor and the turbine. The combustor has to accept the compressor exit conditions and the combustor exit conditions should cater to the required turbine inlet conditions for maximized turbine performance.

IV.B. AERODYNAMICS

We cannot leave out the aerodynamic contributions to the performance of a gas turbine engine. It is probably no great exaggeration to state that when good aerodynamic design is allied to a matching fuel-injection system, a trouble-free combustor requiring only nominal development is virtually assured. This means that aerodynamics leads to thermodynamically ideal constant pressure combustion.

IV.C. CHEMICAL REACTIONS

The chemical reactions evaluate the size of the combustion chamber to produce maximum efficiency. Combustion inefficiency represents a waste of fuel, which is clearly unacceptable in view of the world's dwindling oil supply and escalation of fuel costs. Another important consideration is that combustion inefficiency is manifested in the form of undesirable or harmful pollutant emissions, notably unburned hydrocarbons and carbon monoxide. These implications relate the combustion efficiency with the casing area of the combustor, which in turn relates it with the liner diameter and the performance of the combustion chamber. There are three types of controls as far as combustion is concerned.

a. Reaction Rate Controlled System
b. Mixing Rate Controlled System
c. Evaporation Rate Controlled System

Of the above-mentioned controlled system, the reaction rate controlled system is considered. The burning velocity model described by Green Hough and Lefebvre (ref. [1-2]) is used to determine the casing area. The above mentioned considerations of aerodynamic and chemical will give the reference areas for the combustion chamber. The reference area is the casing area for the combustion chamber.
The annulus casing area calculated from the above values is,

$$A_{ref} = 8.05 \times 10^{-2}\, m^2$$

V. AIRFLOW DISTRIBUTION

This section identifies and briefly describes the airflow distribution in, around, and through the combustor, resulting in the four basic airflow regions, i.e., primary air, intermediate air, dilution air and cooling air. Effective control of this air distribution is vital to attainment of the complete and stable combustion, correct burner exit temperature profile, and acceptable liner temperature for long life. Since the stoichiometric ratio is approximately 15:1, the essential feature is that the air should be introduced in stages. About 15-20% of air in the primary zone provided for rapid combustion. Around 30% of total air is then introduced in secondary zone to complete the combustion. Finally, in dilution zone the remaining air is mixed with the products of combustion to cool them down to the temperature required at the inlet of the turbine.

The percentage of airflow through inner casing and outer casing is calculated using the reference area and diameter values are,

In Inner Casing ≅ 28% of airflow
In Outer Casing ≅ 28% of airflow

V.A. EXIT TEMPERATURE PROFILES

Performance parameters are related to the temperature uniformity of the combustion gases as they enter the turbine. To ensure this the proper temperature profile has been established at the main burner exit. A simplified expression called the pattern factor or peak temperature factor may be calculated from these exit temperature data.

$$PF \equiv \frac{T_{t\,max} - T_{t\,av}}{T_{t\,av} - T_{t\,in}}$$

Using above expression the pattern factor (PF) is calculated as,

PF = 0.15

V.B. LENGTH OF FLAME TUBE

The cross sectional area of burner can be easily determined based on one dimensional gas dynamics, but the length requires scaling laws. The length of the main burner is primarily based on distance required for combustion to come to near completion. The length of the flame tube is a function of the pattern factor and the flame tube diameter. The length can be calculated using the following equation,

$$P.F = 1 - e^{\left(-0.05 x \frac{L_{linear}}{D_{linear}} x \frac{\Delta P_{3-4}}{q_{ref}}\right)^{-1}}$$

Using the equation the linear length is found to be 277 mm.

V.C. LINER AIR ADMISSION HOLES

The need of the liner holes is to provide enough air in the primary zone for complete combustion, to provide enough air to the intermediate zone for cooling the products of combustion and to provide a uniform temperature profile at the exit in the dilution zone and to cool the liner wall configuration. The diameter of the air admission holes depends on the maximum penetration required. The effective diameter of the holes will be calculated by the following equations:

$$\frac{Y_{max}}{d_j} = 1.15\left(\frac{\rho_j U_j^2}{\rho_g U_g^2}\right)^{0.5} \sin\psi$$

The number of holes can be calculated using one of the forms of continuity Equation (ref. [14])

$$nd_j^2 = \frac{15.25 m_j}{\left(\frac{P_3 \Delta P_L}{T_3}\right)^{0.5}}$$

The geometrical diameter can be found using the coefficient of discharge through the air admission holes.

$$d_h = \frac{d_j}{C_D^{0.5}}$$

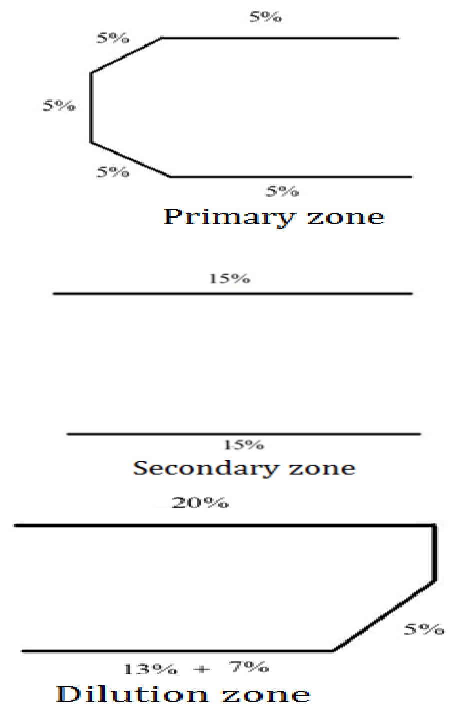

FIG.1.PERCENTAGE OF AIR ADMISSION IN PRIMARY, SECONDARY AND DILUTION ZONES

Using the above mentioned criteria, the number of holes and their diameter for one liner of the annular combustion chamber for different zones are as given below.

TABLE 2: GEOMETRICAL PARAMETERS OF AIR ADMISSION HOLES

Zone	Diameter	Percentage	No. of Holes
Primary	10 mm	5%	76
Secondary	8 mm	15%	88
Dilution	10mm	20%	76
	,,	5%	18
	,,	7%	27
	25mm	13%	8

V.D. SWIRL VANES

The swirlers are the part of the combustor that the primary air flows through as it enters the combustion zone. Their role is to generate turbulence in the flow to rapidly mix the air with fuel. Early combustors tended to use bluff body domes (rather than swirlers), which used a simple plate to create wake turbulence to mix the fuel and air. Most modern designs, however, are swirl stabilized (use swirlers). The swirlers establish a local low pressure zone that forces some of the combustion products to recirculate and creates the high turbulence.
However, the higher the turbulence, the higher the pressure loss will be for the combustor, so the swirlers must be carefully designed so as not to generate more turbulence than is needed to sufficiently mix the fuel and air. The swirl vanes are designed in such a manner that flow turns at 30^o from the previous path. There are totally 88 vanes present in the swirlers, each with length of 11 mm and width of 4 mm.

V.E. COMBUSTOR EXIT AREA

The core (air) mass flow rate and the compressor pressure ratio largely, determine the combustor exit area. The flow area takes the form of an annulus and it is denoted as A^*. We may calculate the flow area A^* by an one dimensional continuity equation,

$$A^* = \frac{\dot{m}_{\max}\sqrt{T_3}}{P_3}\sqrt{\frac{R}{\gamma}}\left(\frac{\gamma+1}{2}\right)^{\left[\frac{\gamma+1}{2(\gamma-1)}\right]}$$

Using the above equation the combustor exit is calculated as,

$$A^* = 1.615193131\times10^{-3}m^2$$

At the combustor exit, in place of slit it is replaced by a circular plate with holes in it. This arrangement is done mainly to reduce the difficulties in fabrication, in this circular plate 20 holes are made according to the combustor exit area and each hole are made with diameter of 10mm.

VI. FUEL AND COMBUSTION

In the early development of gas turbine engine, it was common belief that engine could use any fuel that would burn. This is true in theory, but not in practice. The turbojet engine is quit particular about the fuel used due to the high rate of the fuel flow, wide temperature and pressure variations.

Recently gaseous fuels are preferred most over liquid and solid fuels as it is easier to control emissions from gaseous fuels operated combustion devices as they do not contain any mineral impurities and are easier to for achieving higher efficiency. Liquefied petroleum gas (LPG) is used as fuel due to its greater heating value, a more rapid rate of vaporization, a faster diffusion velocity, a shorter reaction time, and a significantly higher flame speed. Interestingly, LPG can be stored as liquid in tank at around pressure of 0.8 MPa at normal atmospheric temperature, but it becomes gas when it is released to ambient pressure (0.1 MPa), because butane has a boiling point of -0.5^0 C, while propane has a B.P of -42.1^0 C at ambient pressure. The major constituents of LPG are propane (30 %), Iso-butane (15 %) and butane (55%).

Mass flow rate of air (m_a) = 0.46 kg/s

By mass fraction,

Total butane = 70% and Propane = 30%

$$\text{Moles of Butane} = \frac{70}{C_4H_{10}}$$

$$= \frac{70}{(12\times40+(1\times10)} = 1.21$$

$$\text{Moles of Propane} = \frac{30}{C_3H_8}$$

$$= \frac{30}{(12\times3)+(1\times8)} = 0.682$$

Therefore,

Total Moles =1.21+ 0.682 =1.8920

% of Butane by volume in LPG,

$$= \frac{1.21}{1.892}\times100$$

= 63.95%

% of Propane by volume in LPG,

$$= \frac{0.682}{1.892}\times100$$

= 36.05 %

VI.A.STOICHIOMETRIC AIR/FUEL RATIO

The mass balance of a chemical reaction, describing exactly how much oxidizer has to be supplied for complete combustion of certain amount of fuels is generally termed as stoichiometry. This stoichiometric ratio always needs not to be in stoichiometric proportions in combustion problems. On several occasions, hydrocarbons fuels are burnt in the presence of air. For hydrocarbon fuel it's represented by $C_XH_{Y,}$ the stoichiometric relation is given below,

$$C_xH_x + a(O_2 + 3.76N_2) \rightarrow xCO_2 + \left(\frac{y}{2}\right)H_2O + 3.76aN_2$$

Then the stoichiometric air/fuel ratio would be

$$\left(\frac{A}{F}\right)_{stio} = \left(\frac{\dot{m}_{air}}{\dot{m}_{fuel}}\right) = 4.76a\left(\frac{MW_{air}}{MW_{fuel}}\right)$$

By using the above expression, the stoichiometric ratio can be obtained very easily for most of the hydrocarbon fuels such as methane, butane, propane etc, when they react with air. For higher hydrocarbons such as propane and butane air/fuel ratio is around 15, it must be kept in mind that the stoichiometric air/fuel ratio for higher hydrocarbons (kerosene, diesel, gasoline, ATF) up to hexadecane is around 15.

Considering 1 mole of fuel,

$0.361\ C_3H_8 + 0.639\ C_4H_{10} + a\,O_2 +$

$3.76\ a\ N_2 \rightarrow b\ CO_2 + d\ H_2O + e\ N_2$

a= 5.96 b = 3.64 d = 4.64 e = 22.41

$$\left(\frac{a}{f}\right) stio = 15.5$$

Mass of air $\left(\dot{m}_a\right) = 0.4606 Kg/s$

At this mass flow rate of air,

$$\left(\dot{m}_f\right)_{stio} = \frac{0.4606}{15.5}$$

$$\left(\dot{m}_f\right)_{stio} = 0.0297 Kg/s$$

VI.B. IGNITION SYSTEM

Igniter used in this micro gas turbine combustor application is electrical spark igniter, which is similar to automotive spark plugs. The igniter needs to be in the combustion zone where the fuel and air are already mixed, but it needs to be far enough upstream so that it is not damaged by the combustion itself. The LPG gas has to be injected into the combustion chamber at a velocity; this velocity of injection of LPG gas can be calculated from the below expression,

$$V_{inj} = \sqrt{\frac{2 \times (\Delta P)}{\rho_{fluid}}}$$

The velocity of injection calculated from the expression is 24.3 m/s. The diameter of the fuel injector is calculated from the below expression,

$$\dot{m}_f = \rho_f A_f V_f C_D$$

From the above expression, value of diameter of the fuel injector is calculated as 0.6mm.

RESULTS AND DISCUSSION

We have assembled all the components of the combustion chamber at their respective positions and locations. Before testing we passed cold air to check the leakage in the system and at the same time the performance of the diffuser and rig setup are also checked.

In our first experiment, the fuel pressure level had been taken as 4 Bar and the air pressure as 3 Bar and the corresponding exit temperature is noted as 500 K. In the next experiments both the fuel and the air pressure are increased to 6 Bar and the corresponding temperature is noted a 695K.

These results show that our estimated calculations are achieved.

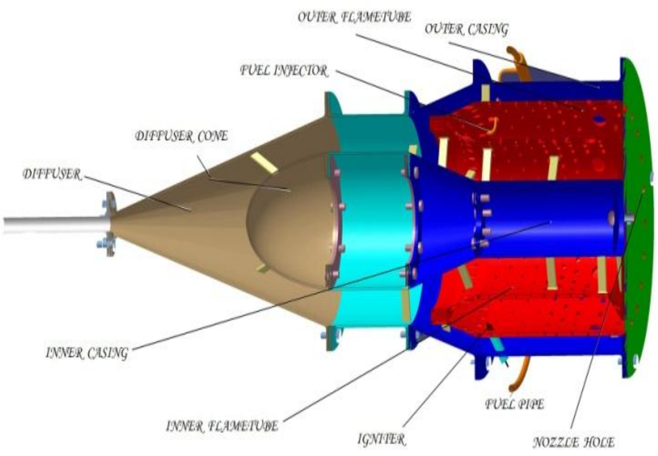

FIG.2.COMBUSTOR (BOTH CAD AND ACTUAL MODEL)

The design of CC is carried out on annular type combustion chamber using LPG as fuel leads to few of the conclusions. Reducing the size of a CC to a centimeter or even millimeter scale entails several new restrictions to the design space of such a device. The most important new limitations arise from the residence time constraint and the increased importance of heat losses.

To reduce these losses and to maintain the exit temperature quality the percentage of airflow inside the CC at each zone namely primary, secondary and dilution zones are designed for 25%, 30% and 45% of total airflow inside CC respectively. The swirl vanes are designed in such a manner that airflow inside CC deviates at an angle 30° from their previous path, this helps in producing required turbulence for a complete mixing of air and fuel for complete combustion. Thus the design of annular combustor for micro gas turbines are made simple and less complicated for fabrication. The material selection and flame stabilization, precautions have been taken .Design is economically done and assembly is made also less complicated

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