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# Design and numerical optimization on annular type combustion chamber for small gas turbine application with CNG as fuel: A Review

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**A B S T R A C T:** The turbines having capacity between 5kW-500kW are known as micro turbine and it can be utilize in the field of distributed power generation, as uninterrupted power supply source instead of DG set, used as power source in satellite instead of battery system and also in robot to operate various mechanism due to its less weight, higher energy density of hydrocarbon compare to battery, less reciprocating part and less emission compare to diesel engine but while designing various challenges are there such as short residence time due to its small volume, high heat loss due to high surface to volume ratio and some additional constraints like material availability, cycle limitations, bearing selection and fabrication technology.



**Keywords:** Micro combustor, numerical optimization, design, CNG

## 1. Introduction

Micro gas turbines have experienced a growing interest during the last decade. Their large energy density makes them attractive for portable power units as well as for propulsion of small airplanes (UAV). They are also of interest for distributed power generation in applications where heat and power generation can be combined.

Major advantages of micro gas turbine (MGT) are adaptability to distributed power generation and combined heating and power applications. Other benefits include lower weight per unit power, lesser number of moving parts, lower maintenance and lower in cost. This technology used for mobile power plant. Combustor for Micro Gas Turbine should be compact in size, inexpensive; mode of fabrication is easy and robust in construction.

The basic components of a micro gas turbine are the compressor, combustion chamber, turbine, generator and regenerator. Micro gas turbines are very small gas turbines that usually have an internal heat recovery heat exchanger (called a regenerator) to improve thermal efficiency of the plant. The heart of the micro turbine is the compressor-turbine package, which is most commonly mounted on a single shaft along with the electric generator. Moderate- to large-sized micro gas turbines use multistage axial flow compressors and turbines, in which the gas flows parallel to the axis of the shaft and then is compressed and expanded in multiple stages. Most current micro gas turbines are based on single-stage radial flow compressors and either single-or double-

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stage turbines single shaft along with the electric generator. The single shaft is supported by two (or more) high-speed bearings. The combustion chamber uses in micro gas turbine are mostly tubular, lean –premixed combustor, can combustor and rich burn combustor.

**1.1 Combustion Challenges in Micro Gas Turbine**

The limited flow residence time within the combustor that comes with the small size of the device, because many fluid mechanical processes such as injection and mixing may scale with size of the device, and the chemical reaction time is independent in size. It is associated with the increased surface area to volume ratio which is such that heat loss to walls of the combustor can be significantly relative to the heat released in combustion process. Studies on the mechanism of spark ignition and its special requirements in regards to fuel distribution should lead to more efficient utilization of available energy. New methods of fuel injection must be devised which can facilitate injection and eliminate exhaust smoke without impairing normal combustion. There is an urgent need for efficient methods of flame cooling to cope with the increasing heat transfer rates arising from the trend towards higher combustion process and temperature.

## 2. Literature survey

R R SHAH and D B Kulshrestha [1] has carried out numerical and analytical analysis of micro combustor for gas turbine for different reactant temperature and equivalence ratio using statistical analysis method o'conaire with zeldovich NO and CFD studies with ANSYS CFX and found that flame temperature increases for equivalence ratio 0.1 to 1.2 up to stoichiometric ratio and also found that for low equivalence ratio 0.132 heat loss is low and also better flame stabilization due to low velocity in reactive zone.

C P. mark and Selwyn [2] has carried out design and analysis of annular type combustion chamber based on constant pressure enthalpy process with initial design using benchmarking of industrial standard and arrived to optimized value. The combustion chamber dimensions are calculated based on various imperial value and its modelling using siemens NX 8 and flow simulation using ansys 14.5 and found that after optimization in preliminary design there is improvement in pattern factor, reduction in SFC, shorter length of combustor in its class and enthalpy addition with 96% efficiency and higher exit temperature.

Srinivasa Sharma Gangaraju, M.V.S. Murali Krishna et al [3] has carried out and design and analysis of gas turbine combustion chamber for Producer Gas as Working Fuel and state that the designed combustion chamber should assist the distribution of primary air required for ignition near the burner, and quenching of Flame by secondary air distribution with flame tube cooling. In conventional combustion chamber design, the primary air is always less than the secondary air, whereas for producer gas, as A/F ratio is ranging between 0.8 –1.2 (due to varying gas composition) the primary air and secondary air ratio is almost 0.93. As energy density of producer gas with Air is more than methane, and other conventional liquid fuels, the generation of NOX is predicted at higher temperatures. Hence estimation of adiabatic Flame temperatures, exhaust temperature of gases leaving the combustion chamber, limiting it to turbine exit temperature is a parameter of design match. The design of complete combustion chamber involves swirler design, burner design, and location of igniter. In the present work, the flame tube is designed with distribution of primary and secondary air.

Jinsong Hua, MengWu, et al [4] has carried out numerical simulation of combustion of hydrogen air mixture in micro combustor using CFD and evaluate the performance of the combustor under different conditions of various fuel/air ratios, mass flow rates and different heat loss conditions. through such systematic numerical analysis, an optimised operating condition for the micro-combustor is suggested which may be used as design guidelines and recommendations for the future development of micro-combustion systems. To achieve a lower wall temperature, the fuel lean burning combustion is suggested to obtain a lower flame temperature. The simulation results under different equivalence ratios illustrate that the flame can be blown out/quenched for small equivalence ratio and upstream burning may happen under large equivalence ratio conditions.

A moderated fuel/air flow rate is required for the stable power generation in the micro-combustor. When fuel/air mass flow rate is too low, the performance of the combustor is limited by poor thermal efficiency and flame quenching in the recirculation jacket. However, at high mass flow rates of fuel/air, the flame may be blown out and the chemical efficiency drops off rapidly, which also leads to low combustion efficiency.

Li Zhang, Junchen Zhu et al [5] has carried out numerical investigation of combustion characteristics of methane in micro combustor with hollow hemispherical bluff body which shows that Blow-off limit of the micro-combustor with a hollow hemisphere bluff body is 2.5 times than the micro-combustor without a bluff body. methane conversion rate of the micro-combustor with a hollow hemisphere bluff body sharply increases at the location of bluff body, which is not affected by Φ and inlet velocity. methane conversion rate is mainly affected by and inlet velocity, which increases firstly and then decreases with the increase of velocity. The maximum methane conversion rate is 97.8% when Φ is 1.0 and inlet velocity is 0.02 m/s. methane conversion rate sharply increases when the velocity increases from 0.08 m/s to 0.02 m/s and then the growth trend of methane conversion rate gradually decreases. Methane conversion rate increases with the increase of Φ when Φ is below 1.0, which reaches the maximum value at 1.0 when inlet velocity of mixture gas keeps constant state. Exist of hollow hemisphere bluff body forms recirculation zone. Recirculation zone is formed back the bluff body, which can effectively prolong the residence time of the mixture gas and increase the methane conversion rate. Recirculation zone is mainly affected by inlet velocity, which is increasing with the increase of inlet velocity.

K. V. Chaudhari, D. B. Kulshreshtha, S.A. Channiwala [6] has carried out design of annular type combustor chamber using the initial conditions as obtained from the compressor outlet and parameters are used from brayton cycle analysis which provides simplified design philosophy for the design of annular type combustor and simulation using CFD with k-ω which shows that temperature profile in the annular type combustion chamber is not uniform at exit of the combustor but dilution is achieved better so not uniform distribution of air takes place near all fuel injectors in primary zone which suggest primary holes are taken twice the number of injectors so uniform air distribution near each injectors and uniform temperature distribution at exit of chamber was achieved.

Jai Ganesh Chetiyar R, Hemanathan et al [7] evolve a systematic design procedure for combustion chamber and design combustion chamber for the 20-kW gas turbine engine. An experimental setup was specifically developed to measure the centreline temperature, liner wall temperature and annulus wall temperature using in total 24 numbers of K-Type thermocouples at different air/fuel ratios ranging from fuel rich ratio of 22.7396 to fuel lean ratio of 152.4. It is observed that the centreline temperature is in the vicinity of 1200°C to 1300°C and liner wall temperature is in the vicinity of 250°C to 350°C. Pressure drop along the length of the combustion chamber is at different air/fuel ratios of 22.7396, 26.72, 122.106 and 152.4, respectively. the achievement of near adiabatic flame temperature in the centreline of combustion chamber and liner wall temperature in vicinity of 300°C may be treated as the major outcome of this work which clearly substantiates the of the design guidelines proposed in the present work. The pressure drops along the length of the combustion chamber being not more than 10% of the delivery pressure clearly advocates the aerodynamic design superiority.

Dr. Digvijay B. Kulshreshtha,Dr. S. A. Channiwala, Saurabh B. Dikshit [8] has carried out CFD approach using CFX 11 to analyse the flow patterns within the combustion liner and through different air admission holes, namely, primary zone, intermediate zone, dilution zone and wall cooling, and from these the temperature distribution in the liner and at walls as well as the temperature quality at the exit of the combustion chamber is obtained for tubular type combustion chamber designed for gas turbine engine. The design optimization is then carried out using the numerical results. Distribution of air, flow recirculation, jet penetration and mixing are achieved in all the zones of the combustion chamber. The maximum centerline temperature recorded by CFD simulation is in the vicinity of 2200K while for Experimental Investigations is around 1600K while exit gas temperatures are in good agreement This may be due to the reason that adiabatic wall is considered in

simulation, which leads to no losses, and single step chemistry is considered, with no dissociation. Therefore, the temperature levels are higher compared to Experimental Results. Thus, CFD can be used to understand the combustion phenomena and can be applied as a design validation tool for tubular type gas turbine combustion chamber.
The temperature levels near the wall region of the dilution zone suggest some lacuna in the design of this zone. The distortion of dilution zone witnessed during the experimental investigations as shown in Figure 2.7, not only validates the CFD results but advocates for redesign of this zone.

D. G. Norton, D. G. Vlachos [9] has investigated Combustion characteristics and flame stability at the microscale: a CFD study of premixed methane/air mixtures, the combustion characteristics in micro-burners were studied using an elliptic two-dimensional computational fluid dynamics model. Transverse gradients in temperature and reaction rate were observed for many conditions, despite the small scale of the burner. Thick walls and/or larger thermal conductivities tend to make the burner more isothermal-like. The wall thermal conductivity plays a vital role in flame stability. Two mechanisms of losing flame stability have been observed. Low wall thermal conductivity limits the upstream heat transfer through the wall, which limits the preheating of the feed, inhibiting the onset of combustion, and causes blowout. Low wall thermal conductivity also causes hot spots of high temperatures within the wall, which can lead to mechanical failure. High thermal conductivity walls are essentially isothermal and have lower temperatures. However, they offer a larger hot area for external heat transfer and become susceptible to spatially global-like extinction. An optimum wall thermal conductivity of approximately 3–5 W/m/K, typical of ceramics, allows the largest external heat transfer coefficient for the conditions studied.

Ms. Urvashi. T. Dhanre, Ms. Sonam V. Sontakke [10] designed various combustor components such as diffuser, swirler, combustor liner, combustor casing and different zones are systematically designed and c-programming is provided for quick access and change of the design parameters on combustor performance and found that maximum adiabatic flame temperature achieve at equivalence ratio 0.9 and for strong recirculation swirl number greater than 0.6 and optimum length for liner is achieved with diffusion angle between 8-10˙.the numbers of holes are maximum on primary zone as there is main combustion process takes place, next on dilution zone for hot gases temperature reduction and last is intermediate zone as less air required for burning of unburned hydrocarbon.

Stephan Kruse, Bruno Kerschgens et al [10] investigate the effect of pressure and mixing on stability of MILD combustion studied for gas turbines on NOx and CO emission and found that high pressure increases NOx emissions and destabilizes MILD combustion mode while Enhanced mixing stabilizes MILD combustion and lowers NOx emission. Mixing is key parameter to control and stabilize MILD combustion. Measurements were performed in a reverse flow combustor with partially premixed air fuel inlet stream at atmospheric and elevated pressures of 2.5 bar and 5.0 bar.
At ambient pressure, it is found that conditions for MILD combustion are improved by decreasing Damköhler number, which is accomplished in the present study by smaller inlet
nozzle diameters resulting in higher nozzle exit velocities and at the same time higher burnt gas recirculation into the fresh gas stream, which lowers the chemical time scales. This significantly decreases NOx emissions with almost unchanged CO emissions, revealing the mixing rate as a particularly important combustion control parameter.
For the elevated pressure investigations, the residence times were increased. Furthermore, the elevated pressure changes the NOx formation pathways, while the ratio of recirculated to injected mass remains unchanged. The results show that the range in equivalence ratio with jointly low NOx and CO emissions becomes smaller compared to ambient conditions and it changes to lower equivalence ratios. This poses a challenge for MILD combustion applications in gas turbines.
Finally, recirculated burnt gas mixing rate was increased also at higher pressure by using smaller air nozzles. With higher mixing rates, where inlet velocities were matched across the different pressures, both NOx and CO

emissions can be decreased even below the values at atmospheric conditions. This again confirms mixing is a key control parameter in achieving MILD combustion conditions for gas turbines.

Ibrahim I. Enagi, K.A. Al-attab et al [11] carried-out design and development of MGT combustion chamber was performed, using CAD SOLID-WORKS and ANSYS-FLUENT 16.1 CFD simulation software. The main geometry manipulated variables were: chamber height, flame tube diameter (chamber diameter will change accordingly) and hole zones on the flame tube. Zone arrangement on the flame tube included holes diameters, number of holes in one row, number of rows per zone and distance between zones which will be referred to as dead zones since it does not include any holes and combustion models were used to produce an optimum design configuration.
Chambers with low height below 600mm did not show a good performance in terms of flame stability and CO emissions. Therefore, three heights of 600, 700 and 800 mm were further compared with 25, 75 and 100 mm flame tube diameters. Results shown here in this section are for LPG PDF simulation. However, diesel fuel has shown similar trends with the different geometries, but with slightly higher temperature and CO emissions. In order to determine the optimum geometry, the main manipulated parameters were the average CO emissions and temperature at the chamber outlet. Figure 2.12. (a) and (b)show average CO emissions and temperature, respectively, for the three heights with LPG fuel and 70% excess
air. It can be noticed that most of the geometries generated outlet temperature slightly above1200 °C, hence, temperature was not a considerable variable for the comparison between geometries. On the other hand, flame tube diameter showed a considerable effect on CO emissions. Although, the flame tube diameter had no effect on combustion setup in terms of air-fuel (AF) ratio, it affected the velocity and turbulence profiles inside the tube. Therefore, the smaller 50 mm diameter flame tube with higher flame turbulence and better air-fuel mixing inside the tube resulted in a slight enhancement in combustion reducing CO emissions in the range of 100–150 ppm compared to 200–300 ppm for larger flame tubes.

H.L. Cao, J.L. Xu [12] conducted experiment on thermal performance of a micro-combustor for micro-gas turbine system and found that as per Figure 2.13 at each hydrogen flow rate, the wall temperature of the micro-combustion chamber gradually increases with increasing excess air ratio from 0.445 to 0.903 and then begins to decrease with continually increasing excess air ratio from 0.903 to 3.823. The same phenomenon can also be observed in other conditions. Therefore, the wall temperature reaches the maximum value at the excess air ratio of about 0.9. when the excess air ratio is up to about 0.9, the heat loss will reach its maximum value. Because the wall temperature of the micro-combustor reaches the maximum value at an excess air ratio of about 0.9, the natural convection heat loss and radiant heat loss to the environment are bound to increase to the peak value correspondingly. With the increase of the excess air ratio, the heat loss begins to decrease due to the reduction of the wall temperature of the combustor.
The temperature of the exhaust gas of the micro-combustor will also increase further with continuing increase of thermal power. The highest temperature of the exhaust gas reaches 1142 K when the thermal power is increased to 156.3 W, and the excess air ratio is about 0.9. However, if the thermal power is increased further, the temperature of the exhaust gas of the micro-combustor will also increase further, even exceeding the serviceable range of the material used to fabricate the micro-turbine blades and finally leading to failure of the material.
Syed Alay Hashim [13] has carried out design and fabrication of an annular combustion chamber for the micro gas turbine engine applications using LPG as fuel. In 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. 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.

J. Guidez, P. Roux et al [14] assessing the feasibility of combustion in miniaturized combustors (volume less than 1 cm3) is a key point for the development of micro gas turbines. They investigate the combustion chamber operating with a hydrogen and air mixture and found that stable combustion was obtained with an output power between 100 and 1200 W, for air mass flow rate from 0.1 to 0.5 g/s, and equivalence ratio between 0.3 to 0.7 and Combustion efficiency is about 0.8 for equivalence ratios greater than 0.4.

Liqiao Jiang, Daiqing Zhao et al [15] developed micro combustion chamber with porous chamber wall and found that temperature of porous chamber wall found 200K lower than that of a solid wall micro combustor this is due to fact that heat exchanging manner was different between preheated gas and chamber wall in two kinds of micro combustors. In the solid-wall micro combustor with heat recovery channel, the heat transfer was convection between channel wall and premixed gas, and the premixed gas was continuously preheated along the heat recovery channel. But in the porous-wall micro combustor, the premixed gas flowed in a short path in the gas distribution channel, and the heat transfer took place in the porous wall, which did not contact with the flame directly, so the temperature of the premixed gas and porous chamber wall was lower than that of the solid-wall micro combustor, which finally decreased the temperature of the combustor's outside wall.

Yang ChunHsiang, Lee ChengChia et al [16] investigates the effects of using fuels with low heating values on the performance of an annular micro gas turbine (MGT) experimentally and numerically. The MGT used in this study is MW-54, whose original fuel is liquid (Jet A1). Its fuel supply system is re-designed to use biogas fuel with low heating value (LHV). The purpose is to reduce the size of a biogas distributed power supply system and to enhance its popularization. This study assesses the practicability of using fuels with LHVs by using various mixing ratios of methane (CH4) and carbon dioxide (CO2). Prior to experiments, the corresponding simulations, aided by the commercial code CFD-ACE+, were carried out to investigate the cooling effect in a perforated combustion chamber and combustion behavior in an annular MGT when LHV gas was used. The main purposes are to confirm that there are no hot spots occurring in the liners and the exhaust temperatures of combustor are lower than 700°C when MGT is operated under different conditions. In experiments, fuel pressure and mass flow rate, turbine rotational speed, generator power output, and temperature distribution were measured to analyses MGT performance. Experimental results indicate that the presented MGT system operates successfully under each tested condition when the minimum heating value of the simulated fuel is approximately 50% of pure methane. The power output is around 170 W at 85000 r/min as 90% CH4 with 10% CO2 is used and 70 W at 60000 r/min as 70% CH4 with 30% CO2 is used. When a critical limit of 60% CH4 is used, the power output is extremely low. Furthermore, the best theoretical Brayton cycle efficiency for such MGT is calculated as 23% according to the experimental data while LHV fuel is used. Finally, the numerical results and experiment results reveal that MGT performance can be improved further and the possible solutions for performance improvement are suggested for the future studies.

## Conclusion

From literature survey, it can be concluded that Good amount of literature is available for design of gas turbine combustion chamber. However, this is available in a discrete manner and there is a need to streamline design procedure. Flame temperature increases for equivalence ratio 0.1 to 1.2 (up to stoichiometric ratio) and also found that for low equivalence ratio 0.132 heat loss is low also better flame stabilization due to low velocity in reactive zone. For uniform air distribution near each injectors and uniform temperature distribution at exit primary holes are taken twice the number of injectors.

Pressure loss should be less than 8% and Pattern factor lies in between 0.025-0.3 for good design of combustion chamber. Swirl number greater than 0.6 to achieve strong recirculation, diffusor angle should be 8-

10 for optimum length of liner and equivalence ratio should be maintained 0.9 to obtain maximum adiabatic flame temperature.

To reduce heat losses and to maintain the uniform exit temperature the percentage of airflow inside the combustion chamber at each zone namely primary, secondary and dilution zones are designed for 25%, 30% and 45% of total airflow inside combustion chamber respectively. When the excess air ratio is up to about 0.9, the heat loss will reach its maximum value. Because the wall temperature of the micro-combustor reaches the maximum value at an excess air ratio of about 0.9, the natural convection heat loss and radiant heat loss to the environment are bound to increase to the peak value correspondingly. With the increase of the excess air ratio, the heat loss begins to decrease due to the reduction of the wall temperature of the combustor.

Flame tube diameter affects the flame turbulence and air-fuel mixing inside the tube and due to that also affect combustion small size flame tube reducing CO emissions in the range of 100–150 ppm compared to 200–300 ppm for larger flame tubes. It is feasible to use annular type combustion chamber of steel and silicon carbide for low heating value gaseous fuel as no hot spot found during experiment work and also exit temperature is also uniform.

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