Experimental Investigation of Preheated Air Effects with Multi-line of Fuel Injection (LPG) on a Tubular Combustor of Micro Gas turbine

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Abstract

This work presents the experimental investigation carried out for preheated air with multi-line of fuel injection in a tubular combustor of micro gas turbine to produce high performance, Low emission and distributed flameless combustion, flameless combustion characterized by dispersed and homogeneous flame and temperature uniformity under the conditions of preheated air and adequate amount of recirculation of products gases. System performance, emissions gases and temperature measured at variable inlet air mass flow rate and gas fuel LPG flow rate for variable flow of secondary fuel radial inward 90º injection in secondary zone F_2 with respect main fuel injector flow rate F_1. concluded that maximal preheated air mass flow rate, with distributed fuel injection with high speed of fuel flow in the main and secondary injectors for each cases promotes the formation of distributed dark pal blue flame combustion, for 3.2 g/s of fuel flow rate with 1mm main injector holes diameter and 8 radial inward 90º holes with 1cm depth of injection ports concluded, lower carbon monoxide CO emissions and decreasing in unburned hydro carbon UHC emissions from 60 to 20 ppmv with increasing in power generation from 2.7 to 5.2 kW and decreasing in specific fuel consumption (S.F.C) from 4 to 2.8 kg/kWh.

Key words: Gas Turbine, Tubular Combustor, Preheated air, Low Emission, LPG Fuel.

Nomenclature

A	Area [m^2].
D	Diameter [m].
F_1	Fuel Mass Flow Rate in main Injection Line.
F_2	Fuel Mass Flow Rate in Secondary Injection Line.
F_t	Total Fuel Injected into Combustor.
L	Length [m].
$\dot{m}$	Flow Rate [kg/s].
M.L.F.I	main line of fuel injection.
P	Total Pressure [Pa].
P_r	Pressure Ratio ($P3/P1$).
q_{Ref}	Dynamic pressure along the combustor.
S.L.F.I	secondary line of fuel injection.

T Total Temperature [K].
V Velocity [m/s].

Greek Symbols

Φ Equivalence Ratio
θ Angle (Diffuser or Snout or Dome) [°]
βsw Swirler Blade Stagger Angle (Flat Blade) [°]

Subscripts

1 At Compressor Inlet
3 At Combustor Inlet
4 At Combustor Outlet
f Fuel
Ref Reference section
RZ Recirculation Zone
PZ Primary Zone
SZ Secondary Zone
DZ Dilution Zone
diff Diffuser
Lin Liner
sw Swirler

1. Introduction:

Increasingly strict emissions protocols, driven by increasing attentiveness and apprehension of environmental dilapidation, have enthused gas turbine manufacturers to explore emissions decline plans. The interaction between the distributed fuel jet and the hot combustor gases during high temperature air combustion (HiTAC) determine combustion stability, combustion efficiency and pollutant emission. This transference in technology has been supplemented by a downward trend in levels of emission gases. We are concerned about CO, CO2 and UHC emissions because its presence in the atmosphere ultimately results in global warming by green house and photochemical smog.

Flameless/Mild combustion has been gained significant importance due to its ability to suppress thermal NO formation and improve thermal efficient of combustion systems. Flameless combustion has been primarily identified with gaseous fuels and extensive work has been reported by [1], and [2]. Despite being a technique recently discovered and little known, nowadays the flameless combustion has gained attention in scientific circles because of the great advantages that offers in its use compared to conventional combustion.

Combustion chambers are designed to mix and ignite the air and fuel mixture, and then mix in more added air to complete the combustion, [3]. This work present the experimental results for the combined effects of the preheated air and multi-zone of oxidation reaction on the tubular combustor of micro gas turbine and capable of achieving colorless low emission combustion. The highly dispersed reaction zone eliminates hot spots, the turbulence levels of the colorless oxidation reaction is so high that if operated in the current diffusion flame combustion technology, **Saywers** [4]. Swirl flows has been adopted to obtain internal recirculation rates in colorless combustion mode. The preference of LPG is chosen from this work for micro gas turbines have a wider scope of fuel. [5], studied numerically and experimentally the flameless combustion in high intensity swirl combustor with liquid fuel, for combustor with variable higher power intensities i.e. 10.2, 16.3 and 21.1 MW/m3, with fuel injection pressure of 9 bar and mass flow rates of 1.72, 3.27, 5.21 and 6.78 kg/h respectively, were used to provide 21.5, 40.8, 65.1 and 84.7 kW thermal inputs respectively Concluded that the an increasing in fuel flow rate led to incomplete combustion and accumulation of unburned fuel in the combustor. Decreasing in CO and UHC emissions and increasing of NO_x emissions with increasing of equivalence ration due to increasing of the average temperature.

[6], examined experimentally flameless oxidation to reduce thermal NO-formation they concluded that there are two approaches are concerned with the reduction of NO_x emission. In this regard, the first approach deals with NO_x abatement strategies regarding control of NO_x formation via thermal mechanisms which avoid hot spot zones within the chamber. Another attempts to eliminate NO_x after formation indicates the methods used in NO_x abatement strategies. However most NO_x technique are aimed at lowering the peak temperature and maintaining the residence time along with lower concentration of oxygen via dilution in high

temperature zone, these strategies can be classified in to three main categories : injection of diluent, exhaust gases clean-up and NOx formation prevention. These methods are crucial in the reduction of NO_x formation by reducing the combustion chamber temperature (thermal NO_x prevention), improve mixing or exhaust gases clean-up in which Nox is reduced after formation. [7]**,**studied experimentally and numerically the effects of fuel mixture on the flameless combustion behavior for the fuel mixture variety using methane, ethylene, and propane for 7kW reverse flow furnaces with premixed of methane air mixture, results showed that the dilution of fuel with CO_2 or N_2 to fuel mass fraction of 0.15 caused reduction in NO_x emissions less than 1ppm and invisible flame this reduction due to the dilution effects on the ignition delay which allows the fuel to mixed with more flue gases which mitigate the NO_x formation, while with 100% propane the NOx emission is 25 ppm and with 100% ethylene about 42 ppm. [8],studied experimentally, the effects of the duel fuel injection on the flameless oxidation reaction by using double tangential inward fuel/air premixed injection point of Methane for local heat load 3.9-6.25 kW and equivalence ratio 0.6 with preheated air to 600k Results showed ultra-low with single injection of air and fuel as 5 ppm NO with 10 ppm of CO, while for the same equivalence ratio with dual injection showed higher emissions than with single injection for NO emission increased by about 20%, with minimal change in CO emissions. The increasing in NO emissions outlined that there is an interaction between both injections jets leading to an unequal distribution in the flame region. [9]**,** investigated experimentally the effects of reverse flow configuration on the flame stability characteristics of a MILD combustion in 23 inch length scale burner by using LPG parallel fuel and air injection port, preheated air to 723 K with low thermal intensity 0.3 MW/m^3-atm, results showed a great role of the preheated air temperature caused decreasing in NO_x emissions from 14ppm at air preheat temperature of 723K to about 12ppm for air preheat temperature of 300K. Amount of heat extracted had significant effect of NO_x emissions as the emission level decreased to 7ppm from 14 ppm by increasing heat extraction from 25% to 42%. Peculiar trend was observed with increase in equivalence ratio where increase in equivalence ratio from 0.8 to 0.9 resulted in decrease in NO_x emission to 8ppm from 14 ppm. [10], studied numerically the effects of the Position of injected air holes in primary and secondary zones on the exit temperature profile (pattern factor) on can combustor of Gas Turbine for Ethanol fuel, using the FLUENT package with SST adopted model for turbulent flow and Non-Premixed Combustion PDF flamlet model foe combustion processes, were varied air injection holes positions of the primary and secondary zone and fixed the location of the dilution zone holes. For this purpose each zone primary and secondary lengths are divided into four sub length: 25%, 50%, 75% and 100% of their original length dimensions. Concluded that the positioning of the Primary zone holes has a great influence on the pattern factor than the positioning of Secondary zone holes, pattern factor shows high sensitive to rows of primary holes displacements. For the Primary zone holes, a reasonable explanation is based on the fact that there is a strong revers flow zone in the first three quarters of the Primary zone length. This condition is observed when positioning the Primary zone holes row in downstream these 0.75 of the primary zone length, in other words, thereby reducing the demand on the downstream zones in obtaining lower pattern factor.

Developed design methodologies would be useful for researchers for preliminary design assessments of a gas turbine combustor. In this study, step by step preliminary design methodologies of tubular combustor have been used.

The effect of the preheated air with the recirculated flue gases with radial inward 90º secondary line of fuel injection in the secondary zone has also been studied experimentally, the overall contribution to knowledge of this study is development of combustor fuel injection methodologies with different variants. The other contribution to knowledge is related to novel combustors with a capability to produce low emissions.

2. Combustor Preliminary design calculation:

For the preliminary design of the combustor, a computational tool for Gas Turbine Combustor Design (GTCD) was used and implemented in Mathcad 15. Developed by Saywers [4] and Lefebvre [12]. With this tool, it is possible to get the preliminary design of tubular combustor. The GTCD enables the design of combustors fueled by LPG fuel, provided that changed the thermochemical parameters of temperature increase as a function of equivalence ratio for fuel adopted. Such a design methodology, in which GTCD is based, considers for the design of combustors, two criteria that must be met in all conditions of the operating envelope of the combustor: aerodynamic and thermochemical. Obtained for both criteria the reference area of the casing cross section (A_{ref}), corresponding to the combustor in study. It is adopted in designing the reference area that meets both criteria above. Defined A_{ref}, obtained the following calculations performed by the tool, the main ones being:

- Diameter of the liner (D_{Lin}).

- Longitudinal lengths of the primary zone (L_{Pz}), secondary zone (L_{Sz}) and dilution zone (L_{Dz}).

- Dimensions of the diffuser (L_{Diff}) and swirler diameter (D_{Sw}).

- Diameter of primary zone holes (D_{Ph}), secondary zone hols (D_{Sh}) and dilution zone holes (D_{Dh}).

For the aerodynamic criterion if the combustor is dimensioned for a certain pressure loss, it will be large enough to accommodate the chemical reaction, Saywers [4]. The mixing process of fuel and air is extremely important. A good mix in the primary zone is essential for high burning rate and to minimize UHC and soot formation, Lefever [12] and Cohen [3]. A satisfactory mixture air-fuel inside the flame tube, and a relatively steady flow throughout the chamber, are aimed in the design of combustor, leading consequently to shorter combustors and lower pressure losses.

By the aerodynamic criterion, preliminary casing and flame tube diameters are estimated using equations (1) and (2).

$$A_{ref} = \left[\frac{R_{air}}{2} \cdot \left[\frac{m_3 \cdot \sqrt{T_3}}{p_3}\right]^2 \cdot \left[\frac{\frac{\Delta p_{3-4}}{q_{ref}}}{\frac{\Delta p_{3-4}}{p_3}}\right]\right]^{0.5} \tag{1}$$

$$\frac{A_{lin}}{A_{ref}} = 0.65 \tag{2}$$

The pressure drop across the swirler is calculated as J. W. Sawyers [4] and H. Lefebvre [12]:

$$\frac{\Delta p_{sw}}{q_{ref}} = \frac{\Delta p_{3-4}}{q_{ref}} - \frac{\Delta p_s}{q_{ref}} - \frac{\Delta p_{diff}}{q_{ref}} \tag{3}$$

$$\frac{\Delta p_{diff}}{q_{ref}} = \frac{0.01 P_{t3}}{q_{ref}} \tag{4}$$

$$\frac{\Delta p_s}{q_{ref}} = 0.25 \frac{{A_{ref}}^2}{A_o} \tag{5}$$

$$A_{sw} = \left(\frac{{A_{ref}}^2}{\left(\frac{\Delta p_{sw}}{K_{sw}} \cdot \left(\frac{m_a}{m_{sw}}\right)^2 + \left(\frac{A_{ref}}{A_{lin}}\right)^2\right) \cdot \left(\cos\left(\beta_{sw}\frac{\pi}{180}\right)\right)^2}\right)^{0.5} \tag{6}$$

Where K_{sw} is 1.3 for flat vanes and 1.15 for curved vanes

m_{sw} is the swirler mass flow rate

A_{sw} is the swirler frontal area, calculated as (Lefebvre, 1999):

$$A_{sw} = \left(\frac{\pi}{4}\right) \cdot \left({D_{sw}}^2 - {D_{hub}}^2\right) - 0.5 n_v t_v (D_{sw} - D_{hub}) \tag{7}$$

$$D_{hub} = 0.1 \cdot D_{lin} + 6.35 \cdot 10^{-3} \tag{8}$$

$$D_{sw} = 0.225 \cdot D_{lin} + 0.01905 \tag{9}$$

Where n_v is the number of vanes

$t_v$is the vane thickness

Where diffuse angle defined as:

$$\varphi_{diff} = tan^{-1}\left[\left(\frac{\Delta p_{diff} \cdot A_3{}^2 \cdot p_{t3}{}^2}{502.4 \cdot \left(1-\frac{A_3}{A_o}\right)^2 \cdot m_3{}^2 \cdot T_3}\right)^{\frac{1}{1.22}}\right] \quad (10)$$

Diffuser length defined as:

$$L_{diff} = \frac{(R_o - R_3)}{tan(\varphi)} \quad (11)$$

The aerodynamic phenomena play a vital role in the design and performance of the gas turbine combustion. As already mentioned, generally, if the aerodynamic design is satisfactory and the fuel injection system is suitable for the combustor, so do not expect operational problems. Using Mathcad package to programming the above equations 1 to11 for the combustor inlet boundary conditions in table 1 to get the final preliminary design results in table 2.

Table: 1, inlet boundary condition

Variable	*Value*	*Unite*
$\dot{m}_3$	0.6	Kg/s
$\dot{m}_f$	0.0032	Kg/s
P_3	1.5E5	pa
Pr	1.4	-
T_3	600	K
V_3	50	m/s

Table: 2, preliminary design results.

Variable	*Value*	*Unite*
A_{ref}	0.0346	m^2
A_{lin}	0.0224	m^2
A_{sw}	8.329e-4	m^2
D_{ref}	0.21	m
D_{lin}	0.17	m
D_{sw}	0.048	m
$D_{sw,hub}$	0.025	m
L_{pz}	0.1269	m
L_{sz}	0.0846	m
L_{dz}	0.225	m
L_{Diff}	0.038	m
L_{Dom}	0.0223	m
D_{ph}	0.022	m
D_{sh}	0.014	m
D_{Dh}	0.032	m
θ_{Dom}	69.86	^{o}
θ_{Diff}	26.3	^{o}

3. Combustor Geometry:

In this section, the final dimensions results of preliminary design in table 2 have drawn as geometry through employing AutoCAD 2016. The mainstream of the main injector line. However, the geometry that utilized in the experimental study as shown in Figure (1).

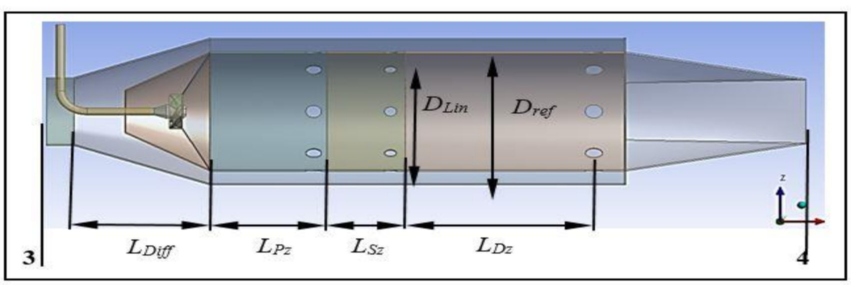

Figure: 1, final combustor geometry for the preliminary design results.

4. Experimantal Setup:

The design and manufacturing of the test rig, consisting of micro gas turbine close loop system, was carried out at the Department of Mechanical Engineering, University of Technology, Exhaust gases turbocharger BBC type, exhaust plenum, exhaust gases recirculation (EGR) line, intake projection venture's tube, oil lubrication system, ignition system, bearing cooling system, pitot-tube and other measurement device, the detail of the overall test rig setup is presented in figures: 2.

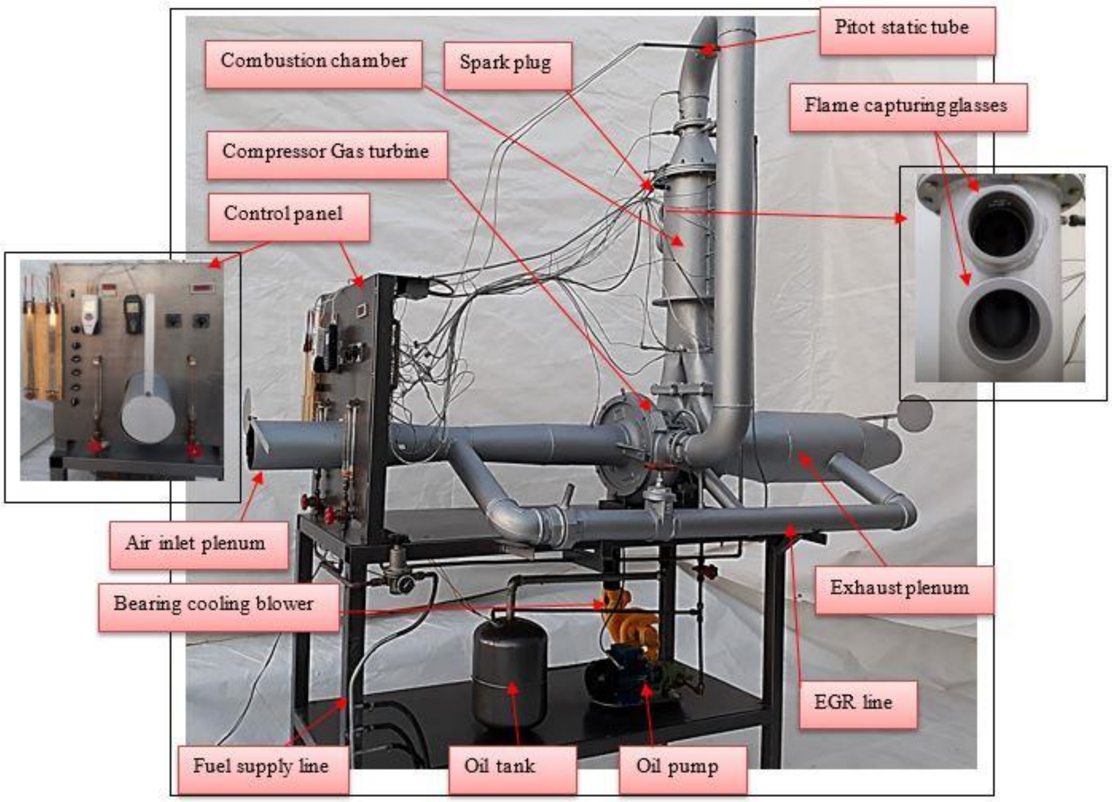

Figure 2. Details of engine parts.

5. Radial inward 90º fuel injection in secondary zone:

Related with the experimental steady-state test method and verification of the combustion with conventional main injector case for injection holes size (1mm) with swirler of 6 vanes, 60º degrees with the axial direction for each vane, [8] as shown in Figure: 3, And using distribution fuel injection method by radial inward 90º of eight port with 0.8mm hole diameter of fuel injection port in secondary zone below the primary holes in abot 1 cm, as shown in figure 4, In order to measure the centerline temperature along the combustor, outlet temperature profile, emissions gases and capturing flame by camera and combustion efficiency calculation. Both tests were achieved for the same primary, secondary and dilution holes geometry of inline arrangements, number of holes, and size, constant fuel pressure about 2 bar and the range of injected fuel approximately about 0-3.2 g/s of LPG gases fuel.

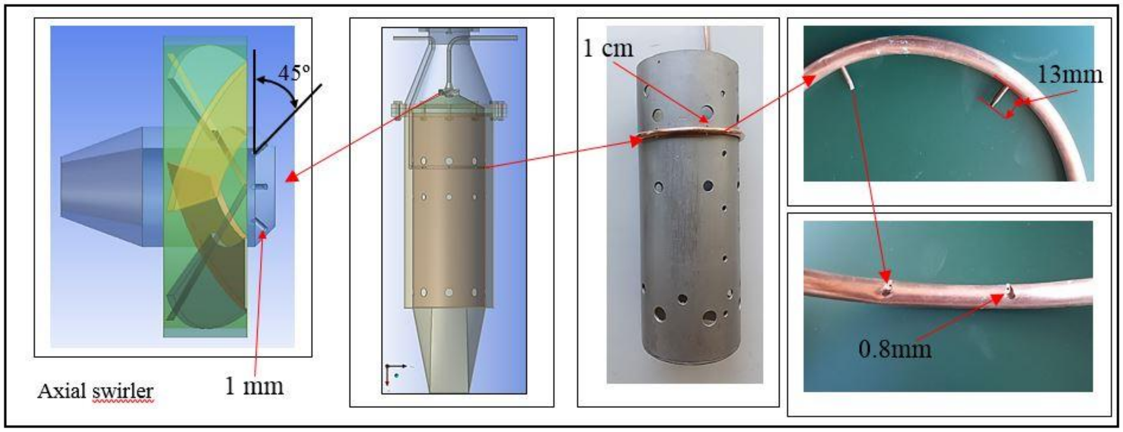

Figure: 3, one row of S.L. F. I. in secondary zone below primary holes about 1cm

6. Combustion efficiency calculation:

The combustion efficiency determined through a measurement of UHC and CO for LPG gas fuel, the imperial relationship of the combustion efficiency, UHC and CO, emissions is as follows, Hung [11]:

$$\eta_c = 1 - (UHC_{EI} + 0.211\ CO_{EI})10^{-3} \quad (12)$$

Where η_c= combustion efficiency

UHC_{EI} =emissions index of UHC, g/kg fuel

CO_{EI} = emission index of CO, g/kg fuel

The relation between emission index and emission expressed for UHC and CO at ISO condition are follow as:

$$UHC_{EI} = 0.0288UHC \quad (13)$$

$$CO_{EI} = 0.0503CO \quad (14)$$

Where UHC= emission of UHC in ppmv

CO= emission of CO in ppmv

7. Pattern factor calculation: temperature profile at the exit section of the combustor defined as pattern factor represented the temperature homogeneity at the combustor outlet, as follows, Lefebver [12]:

$$Pattern\ factor = \frac{T_{max} - T_4}{T_4 - T_3} \quad (15)$$

8. Results:

For 6 holes with 1mm injection hole diameter of main injector and 8 holes with 0.8mm diameter of secondary line of fuel injection located in secondary zone where the fuel injector located below the primary holes in 1 cm as shown in figure 3 with EGR effects results shows long pal dark blue continues flame (flameless considerations) as shown in figure 4 produce low addle speed approximately 5500 rpm (complete combustion, complete heat release) for this case and ultimate maximum speed about 15000 rpm, easy to start and accelerated the engine by both the primary and secondary fuel line.

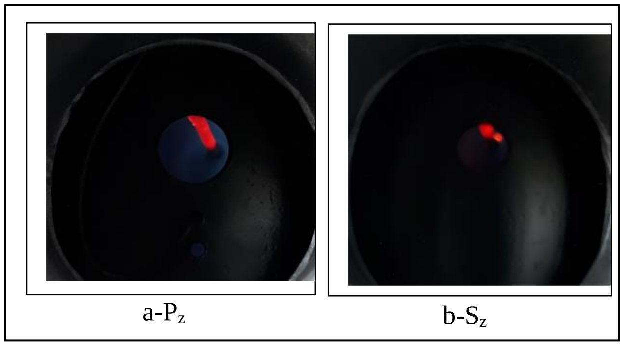

Figure 4 a and b Photographs of flame in primary zone and secondary zone respectively

For loaded turbine and with EGR effects figures 5 and 6 shows the variation of temperature distribution profile along the combustor for different fuel mass flow rate ratio and equivalence ratio, the outcome reveals that there is increasing in temperature in the secondary zone due to the effects of the secondary line of fuel injection at the expense of the primary zone and then decreasing in temperature along the combustor while the overall temperature gradient decrease as decreasing in compressor pressure 95 to 80 cm H2O due to the decreasing in the primary and secondary holes effects head also the overall combustor temperature of the combustor rise about 80 to 100 C when EGR ratio increase from20% to 25% .

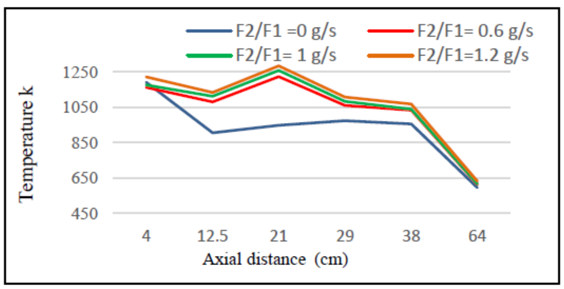

Figure 5, temperature profile along the combustor for loaded turbine 95 cm H2O compressor outlet pressure with EGR ratio

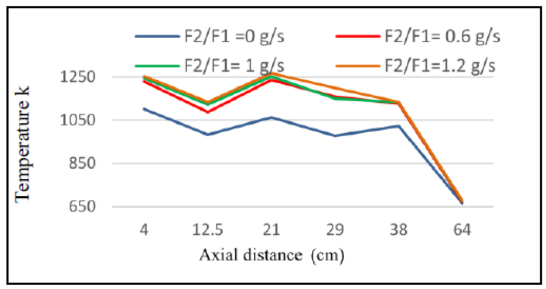

Figure 6, temperature profile along the center line of the combustor for 60 cm compressor outlet pressure.

for combustor performance figure 7, shows the variation of pattern factor and average outlet temperature with the fuel mass flow rate ratio and equivalence ratio the outcome reveals desirable increasing in pattern factor for case of running without EGR effects while the figure 8 reveals the decreasing in pattern factor with the activation of EGR line with ratio about 20% to 25% due to the elongation of reaction zone which give more temperature uniformity.

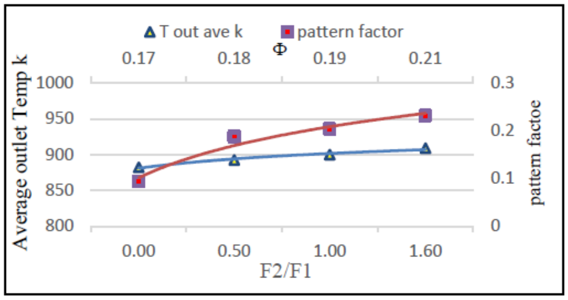

Figure 7, pattern factor and average outlet temperature without EGR effects.

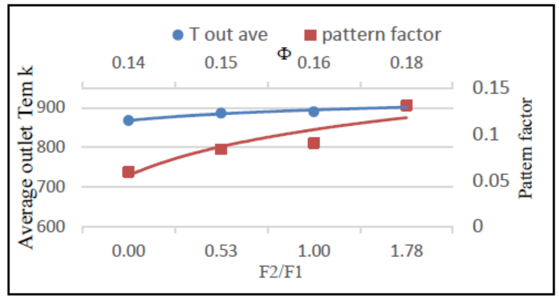

Figure 8, pattern factor and average outlet temperature with EGR ratio 20%.

Figures 9 and 10 shows the variation of combustion efficiency by different three correlations with fuel mass flow rate ratio and equivalence ratio the results shows increasing in combustion efficiency for each cases of 95 and 80 cm H2O head of compressor outlet pressure due to decreasing in UHC and CO emissions figures 11 and 12 because of increasing in combustor overall temperature and increasing of inlet air temperature the low effectiveness of cooling holes with the hot air caused complete combustion for the distributed fuel while this effect increase as decreasing in pressure 80 cm H2O because of increasing in equivalence ratio caused continues flame caused more increasing in efficiency and more decreasing in emission gases CO and UHC.

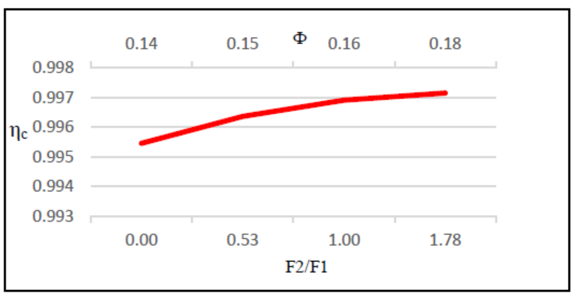

Figure 9, combustion efficiency for loaded turbine 95 cm H_2O compressor outlet pressure EGR ratio 20%.

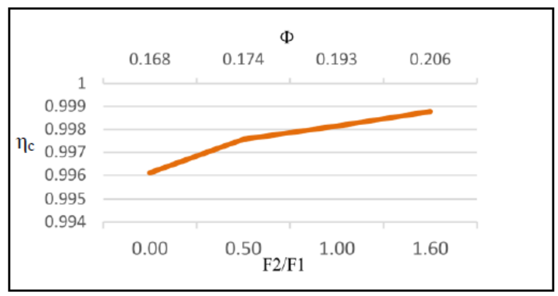

Figure 10, pattern factor and average outlet temperature for loaded turbine 80 cm compressor outlet pressure EGR ratio 25%..

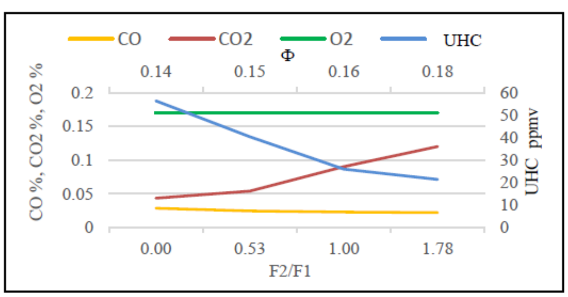

Figure 11, emission gases for loaded turbine 95 cm compressor outlet pressure EGR ratio 20%.

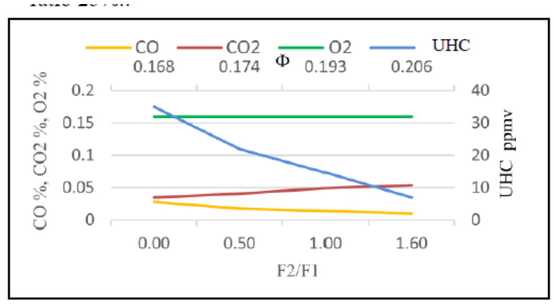

Figure 12, emission gases for loaded turbine 80 cm compressor outlet pressure EGR ratio 25 %.

Figure 13 and 14 shows variation of power and S.F.C results shows for constant outlet power for each case of compressor pressure decreasing in S.F.C as increasing in secondary line of fuel injection activation for each case due to increasing in fuel energy release (complete combustion) and increasing in S.F.C as decreasing in pressure and increasing in EGR ratio from 20% to 25%.

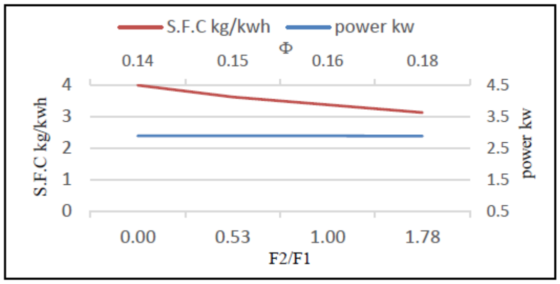

Figure 13, power and S.F.C loaded turbine 95 cm compressor outlet pressure EGR ratio 20%.

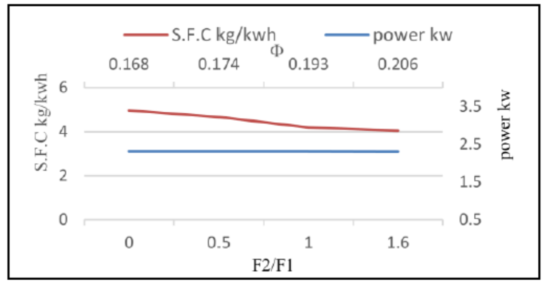

Figure 14, power and S.F.C loaded turbine 80 cm compressor outlet pressure EGR rati 25 %.

For variable compressor pressure for each fuel mass flow rate ratio enhancement effects results shows,

for partial loaded turbine and with EGR effects figure 15 shows the variation of temperature distribution profile along the center line of the combustor for different fuel mass flow rate ratio, equivalence ratio and pressure, the outcome reveals that there is slow decreasing in temperature in the secondary zone due to the effects of the secondary line of fuel injection at the expense of the primary zone and then decreasing in temperature along the combustor after dilution effects. for combustor performance figure 16 shows the variation of pattern factor and average outlet temperature with the fuel mass flow rate ratio, equivalence ratio and the pressure, the outcome reveals desirable increasing in pattern factor for each running case, approximately constant average outlet temperature.

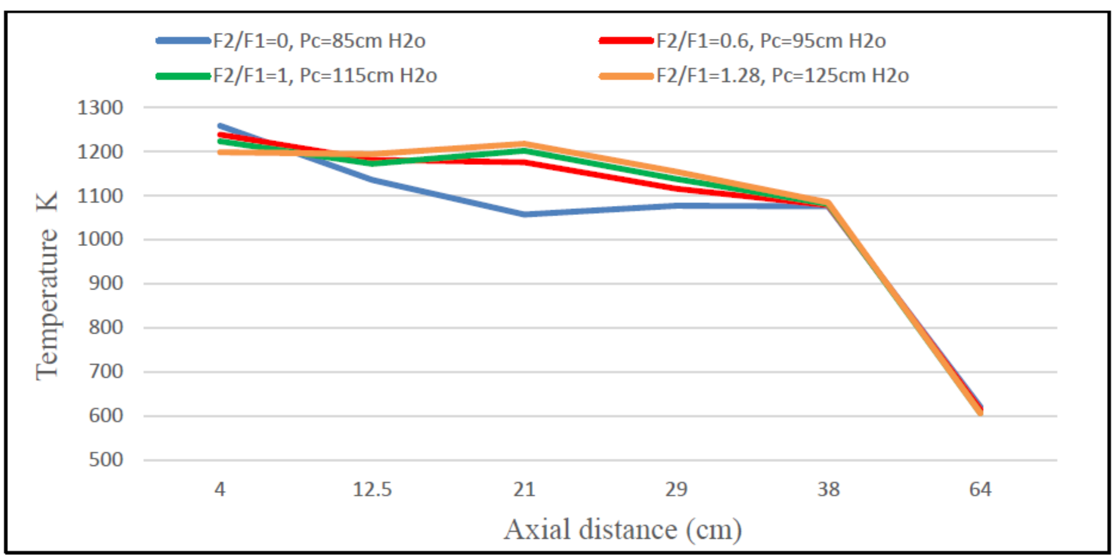

Figure: 15 temperature profile along the centerline of the with EGR effects loaded turbine with difference compressor outlet pressure.

Figure 17, shows the variation of combustion efficiency with fuel mass flow rate ratio, equivalence ratio and the pressure, results shows increasing in combustion efficiency for each cases of running due to decreasing in UHC and CO emissions figure 18 because of increasing in combustor overall temperature. Figure 19 shows variation of power and S.F.C results shows increasing in outlet power and decreasing in S.F.C with the increasing of pressure and increasing if S.F.I activation due to increasing in fuel energy release (complete combustion).

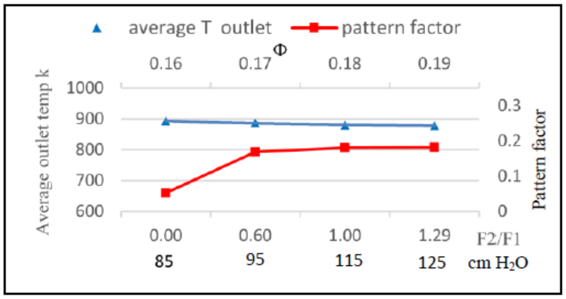

Figure 16, Pattern factor and average outlet temperature loaded turbine with difference compressor outlet pressure.

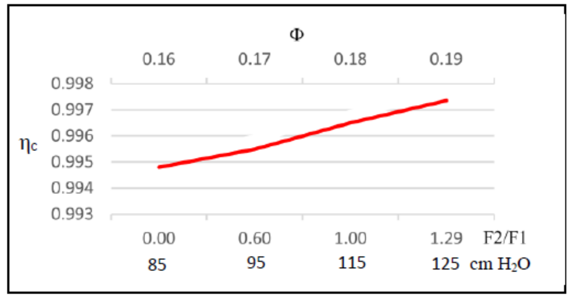

Figure 17, Combustion efficiency for loaded turbine with difference compressor outlet pressure.

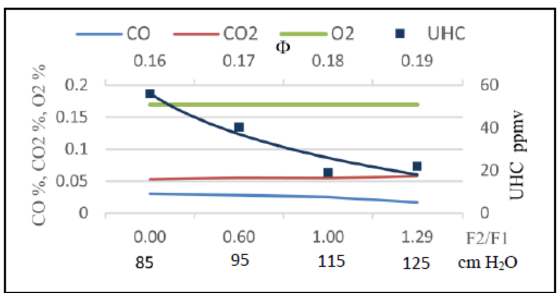

Figure 18, Emission gases with equivalence ratio for loaded turbine and with EGR effects.

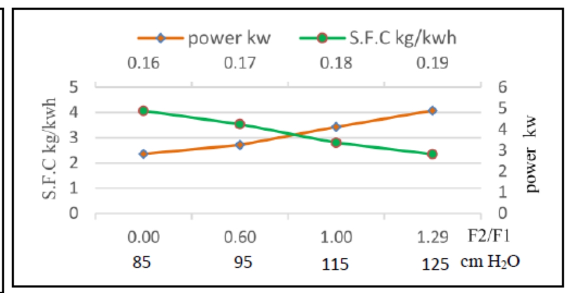

Figure 19, Power and S.F.C for loaded turbine with difference compressor outlet pressure

9. Conclusions:

A preheated air by exhaust gases recirculation to produce colorless flame with high-performance combustion carried out by experimental investigation on micro gas turbine with multi-port of fuel injection in secondary zone to burned gaseous fuel (LPG). It is found that to achieve high performance combustion is heavily dependent on combustor design, the method of fuel injection and oxidizer temperature. Following are the conclusions of this study based on the objectives.

1. The way the fuel and air are injected into the combustion chamber is of primary importance for the distributions of furnace temperature, oxygen, and fuel that thus affects emission gases and combustion efficiency.

2. The difficulty in designing a flameless prototype arises from the fact that there are no standard design tools. In spite of the innovative concept, its design and implementation involves the traditional issues of a gas turbine combustor.

3. It is not necessary to diffuse the compressed air to very low inlet velocities because high air inlet velocities can be used to enhance the internal recirculation.

4. The homogeneous distributed flame combustion mode happened only at fuel lean conditions and high-preheated airflow rates.
5. The flameless combustion achieved when the preheated air overpass the limit sill which abut 340K promote extremely low, CO and UHC when the distributed flame was formed (distribution of hotspot).
6. High fuel mass flow rates for each main injector and secondary injector with respect to high preheated air mass flow rate helped in promoting good mixing helped in forming the distributed flame, thus reducing the possibility to combustion instability.

CONFLICT OF INTERESTS.

- There are no conflicts of interest.

10. References

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بحث تجريبي لتأثير هواء مسبق التسخين مع حقن وقود متعدد الخطوط (غاز بترولي مسال) لحارق أنبوبي لتوربين غازي صغير

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الخلاصة

يقدم هذا العمل بحث تجريبي تم تطبيقه لهواء مسبق التسخين مع حقن وقود متعدد الخطوط لحارق أنبوبي لتوربين غازي صغير لأنتاج أحتراق بأداء عالي وواطي الأنبعاثات وبدون لهب، الأحتراق بدون لهب يتميز بشعله متجانسة وموزعه ودرجة حرارة متماثله لحالة هواء مسخن وكميه كافيه من أعادة التدوير للغازات الناتجه. أداء المنضومه ومعدل ألأنبعاثات الغازيه ودرجات الحراره تم قياسها عند تغير معدل تدفق الهواء ومعدل تدفق الوقود الغازي البترولي المسال في الخط الثانوي F_2لمنفذ حقن شعاعي الى الداخل بزاوية 90^o نسبة الى معدل الجريان في الحاقن الرئيسي F_1. بينت النتائج لأعلى معدل تدفق للهواء مسبق التسخين بحقن وقود موزع بسرعه جريان عاليه في الحاقن الرئيسي والثانوي لكل حاله تعطي شعله موزعه متجانسه بلون أزرق شاحب مظلم ، لجريان وقود بمعدل 3.2 غم\ثا مع قطر 1 ملم للحاقن الرئيسي و 8 حواقن بحقن شعاعي الى الداخل بزاوه 90^o وبعمق 1cm أستنتج معدل انبعاثات واطيئ لكل من اول اوكسيد الكربون CO وأنخفاض في الهيدرو كربونات الغير محترقه UHC من 60 الى 20 جزء بالمليون كنسبه حجميه مع زياده في القدره المنتجه من 2.7 الى 5.2 كيلو واط وانخفاض في معدل الأستهلاك النوعي للوقود من 4 الى 2.8 كغم\كيلو واط .ساعه .

الكلمات الداله: توربين غازي، حارق انبوبي، هواء مسبق التسخين، انبعاث واطيئ، وقود غاز بترولي مسال.