Influence of Multi-Line of Fuel Injection (LPG) in Secondary Zone of Tubular Combustion Chamber of Micro Gas turbine

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Abstract

Experimental investigation of high performance and low emission colorless combustion in a micro gas turbine tubular combustor with multi-line of fuel injection located in secondary zone carried out at the atmospheric conditions, colorless oxidation reaction is characterized by dispersed homogenies flame and high uniformity of average temperature inside the combustion chamber. System performance, emission gases, the flame capturing photo, measurements of temperature, inlet air mass flow rate and gas fuel LPG flow rate for variable of fuel lines mass fraction F_2/F_1 are recorded at low emission colorless combustion condition. for 3.2 g/s of fuel flow rate with 6 holes and 1mm main injector holes diameter and 8 holes with 0.8mm diameter for secondary line of fuel injection, concluded that maximal air mass flow rate, with choked fuel flow in the main injector for each cases promotes the formation of colorless distributed pal blue flame combustion, with great effects for the equivalence ratio on the average temperature to limiting cooled flame quenching generation, desirable pattern factor < 0.28, increasing in combustion efficiency up to 99.68%, and decreasing in specific fuel consumption with rising of power generation.

Key words: Gas Turbine, Tubular Combustor, Colorless combustion, Distributed fuel injection, LPG Fuel.

Nomenclature

A	Area [m2].
D	Diameter [m].
F_1	Fuel Mass Flow Rate in main Injection Line.
F_2	Fuel Mass Flow Rate in Secondary Injection Line.
L	*Length [m]*.
$\dot{m}$	Flow Rate [kg/s].
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.

Ref Reference section.

RZ Recirculation Zone.

PZ Primary Zone.

SZ Secondary Zone.

DZ Dilution Zone.

diff Diffuser.

Lin Liner.

sw Swirler.

1. Introduction:

The improvement of low polluting and efficient combustion systems in gas turbine combustor is the main goal of the combustion equipment manufacturers'. For the NOx, CO and unburned hydrocarbon (UHC) emissions case, most control strategies in combustion processes are based on three variables: residence time, temperature and oxygen availability. These strategies focus on reducing peak temperature, keeping the residence time, and low oxygen concentration at high temperature zones. However, two techniques are generally used in order to achieve these objectives: a- changing the burning process through appropriate burners, that operate in the formation mechanism of the pollutants and b- treatment of exhaust gases, acting on the mechanism of destruction of pollutants.

Control and minimization of emissions of NOx and other pollutants are the flameless combustion (smoldering visible). This technique is a new type of combustion that changes the characteristics of the reaction zone and consequently the formation of combustion products by process modifications. 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, Cohen [1]. This work present the experimental results for the effects of the multi-zone of oxidation reaction on the tubular gas turbine combustor and capable of achieving colorless low emission combustion. Colorless low emission oxidation reaction is characterized by reacting of distributed fuel with a high oxidizer temperature with high levels of turbulence producing a highly dispersed reaction zone. 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 [2]. 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. [3], studied numerically the difference between the combustion models in CFD mode as numerical modelling and validation of the combustion in a gas turbine and identified the characteristic of each models concluded, The Eddy Dissipation Model (EDM) is especially suitable for the chemical reactions, which go through in a short time, and the fluid flow is turbulent and non-premixed. In EDM the regress of elementary reaction k, is determined by the smallest of the reactants limiter and products limiter The PDF Flamelet Model (PFM) can be used, if the flame is turbulent, the Damköhler number is much greater than 1 and non-premixed. Chemical reactions not influence the mixture fraction, because it deals with elements rather than molecules, and elements are not affected by chemistry. Ansys Fluent Non-Premixed model can be used for fast, turbulent reactions, in the case of chemical equilibrium, or laminar flamelet structure. With these certain assumptions, the Thermochemistry can be reduced to a single parameter. [4],studied numerically the effects of reference area of 600 kW gas turbine tubular combustor on the burning, using shear stress transfer model for turbulence flow field simulation by ANSYS CFX and PDF flamelet as combustion model with mixture of combination gaseous fuel and 4.288 kg/s air mass flow rate concluded as increasing in reference area caused reduction in burning rate with enhanced combustion process also the reduction of air velocity at the swirle region has develop the dispersal of flame along the combustor. [5], examined experimentally flameless oxidation to reduce thermal NO-formation they concluded that there are two approaches which are concerned with the reduction of NOx emission. In this regard, the first approach deals with NOx abatement strategies regarding control of NOx formation via thermal mechanisms, which avoid hot spot zones within the chamber. Other attempts to eliminate NOx after formation indicates the methods used in NOx abatement strategies. However most NOx technique is 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 into three main categories : injection of diluent, exhaust gases clean-up and NOx formation prevention. These methods are crucial in the reduction of NOx formation by reducing the combustion chamber temperature (thermal NOx prevention), improve mixing or exhaust gases clean-up in which Nox is reduced after formation. [6] studied experimentally, the effects of mixture preparing by varying the location of Methane fuel injection point with respect to air injection point, for high thermal intensity swirl distribution combustion designed for ultra-low emission of NOx and CO Results showed for high heat release about 27MW/m3-atm and equivalence ratio 0.6 with 600 k preheated air, NOx decrease from 21 ppm for coaxial fuel injection case to 10ppm for non-premixed case and CO emission very low due to preheated air effects for all case of fuel injection. [7],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 Increase in NO emissions outlined that there is an interaction between both injections jets leading to an unequal distribution in the flame region. [8] studied experimentally high thermal intensity 3.8 MW/m^3-atm small scale 13.4 inch combustor for furnaces application with equivalence ratio 0.9 and preheated air to 930 K related the air injection to the turbulent time scale to be0.104 ms, result showed moving the reaction zone toward the air injection as increasing the air flow rate with constant fuel flow ate and the lower NOx about 10ppm and CO about 12ppm. [9], 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 were 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 multi-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 mathcad15. Developed by Saywers [2], Lefebvre [9]. 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 (Aref), corresponding to the combustor in study. It is adopted in designing the reference area that meets both criteria above. Defined Aref, obtained the following calculations performed by the tool, the main ones being:

- Diameter of the liner (DLin).

- Longitudinal lengths of the primary zone (LPz), secondary zone (LSz) and dilution zone (LDz).

- Dimensions of the diffuser (LDiff) and swirler diameter (DSw).

- Diameter of primary zone holes (DPh), secondary zone hols (DSh) and dilution zone holes (DDh).

For the aerodynamic criterion if the combustor is dimensioned for a certain pressure loss, it will be large enough to accommodate the chemical reaction, J. Saywers [2]. 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, H. Lefever [9] and S. Cohen [1]. A satisfactory mixed air-fuel inside the flame tube, and a relatively steady flow throughout the chamber, are aimed at the design of combustor, leading consequently to shorter combustors and lower pressure losses.

Table 1, Pressure losses terms of aircraft and industrial engine combustor, Saywers [2].

Combustor type	$\frac{\Delta p_{3-4}}{p_3}$	$\frac{\Delta p_{3-4}}{q_{ref}}$	$\frac{m_3 \cdot \sqrt{T_3}}{p_3 \cdot A_{ref}}$
Multi-can	5.3	40	3e-3
Annular	6	20	4.5e-3
Can-Annular	5.4	30	3.5e-3

By the aerodynamic criterion, preliminary casing and flame tube diameters are estimated using equations 1, 2 and 3, Saywers[2].

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

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

$$\frac{2}{3} D_L \leq L_{Pz} \leq \frac{2}{3} D_L \tag{3}$$

$$L_s = \frac{1}{2} D_L \tag{4}$$

The dilution zone length ratio as a function of pattern factor for different value of pressure losses factor $\frac{\Delta P_{3-4}}{q_{ref}}$ =30, 50.

$$LR_{Dz,30} = 2.96 - 9.86Q + 13.3Q^2 \quad (5)$$

$$LR_{Dz,50} = 2.718 - 12.64Q + 28.51Q^2 \quad (6)$$

$LR_{Dz,40}$ Calculated by interpolation method.

$$L_{Dz} = LR_{Dz,40} \cdot D_L \quad (7)$$

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 to 7 for the combustor inlet boundary conditions in table 2 to get the final preliminary design results in table 3.

Table: 2, 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: 3, 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 3 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).

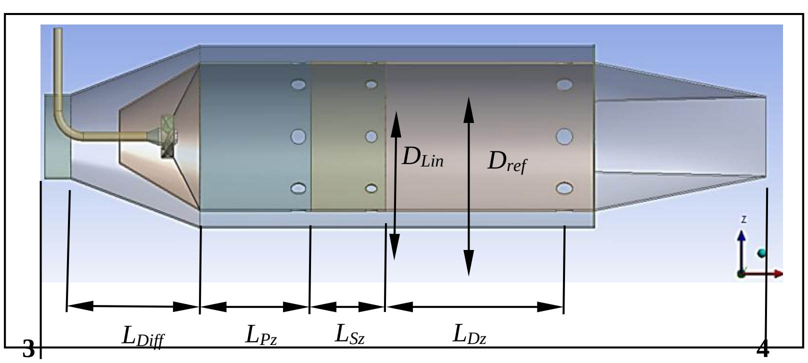

Figure: 1, Final combustor geometry for the preliminary design results.

4. Experimental 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.

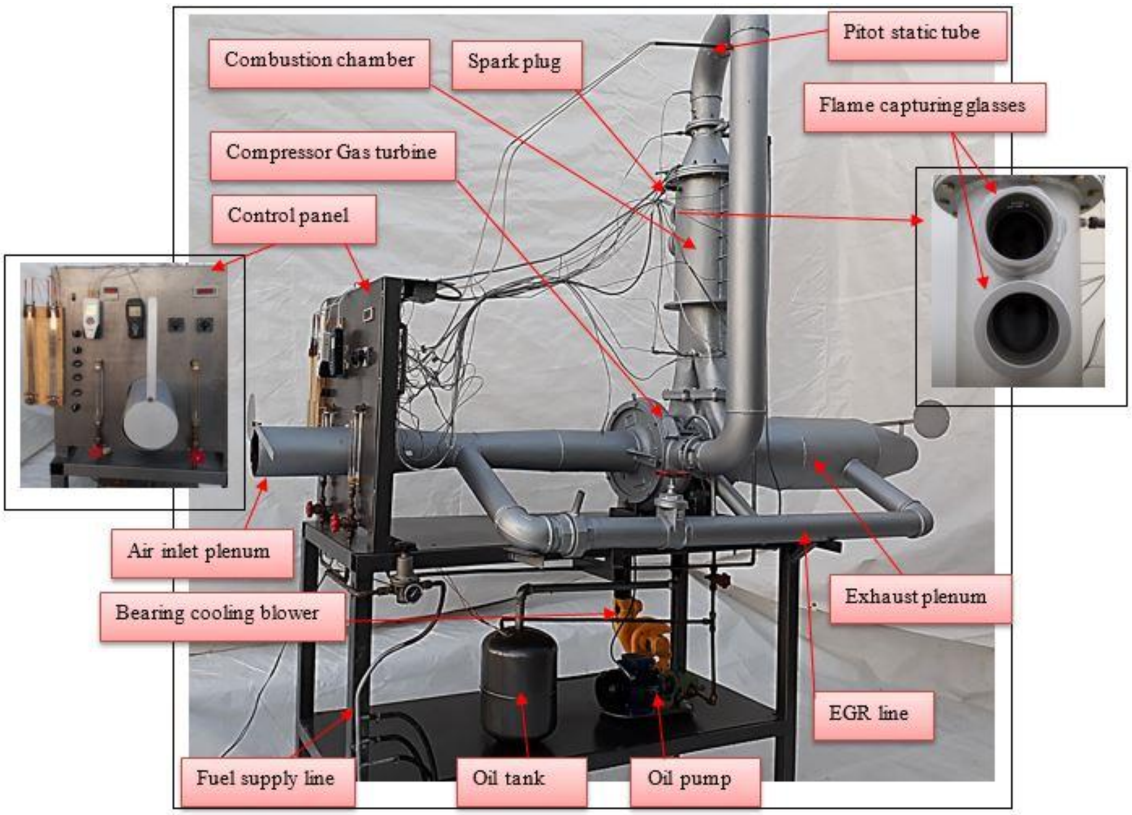

Figure 2. Experimental Test Rig.

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 about 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.

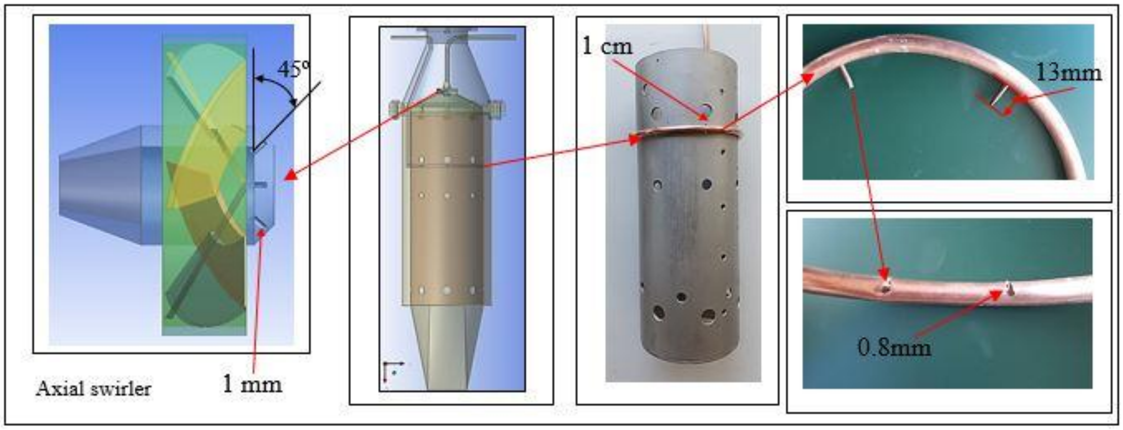

Figure: 3, One row of S.L. F. I. in secondary zone below primary holes about 1cm

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 [10]:

$$\eta_c = 1 - (UHC_{EI} + 0.211\ CO_{EI})10^{-3} \quad (8)$$

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:

$$UHC_{EI} = 0.0288UHC \quad (9)$$

$$CO_{EI} = 0.0503CO \quad (10)$$

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, Lefebver [11] as follows:

$$Pattern\ factor = \frac{T_{max} - T_4}{T_4 - T_3} \quad (11)$$

8. Results:

For 6 holes and 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, for 3.2 g/s of total fuel flow rate, results shows long pal blue continues flame (flameless considerations) as shown in Figure 4 produce low addle speed approximately 5500 RPM, complete combustion, without soot generation 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. For loaded turbine figures: 5, 6 and 7 shows the variation of temperature distribution profile along the center line of 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 and becomes more uniformity as decreasing in compressor pressure from 100 cm H2O to partial loaded turbine 60 cm H_2O

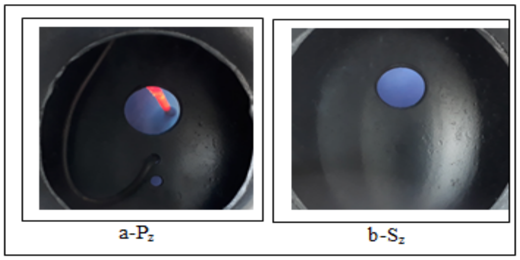

Figure 4 a, b Photographs of flame in primary zone and secondary zone respectively

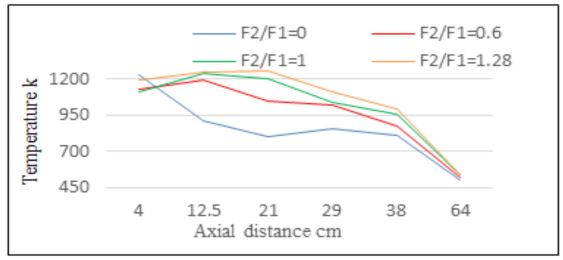

Figure 5, temperature profile along the center line of the combustor for 100 cm compressor outlet pressure.

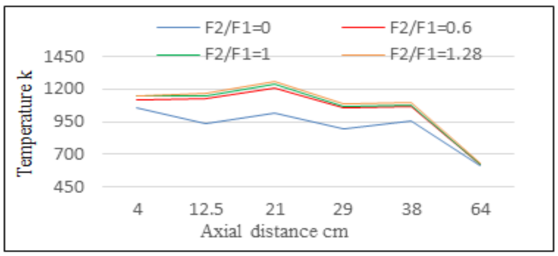

Figure 6, temperature profile along the center line of the combustor for 80 cm compressor outlet pressure.

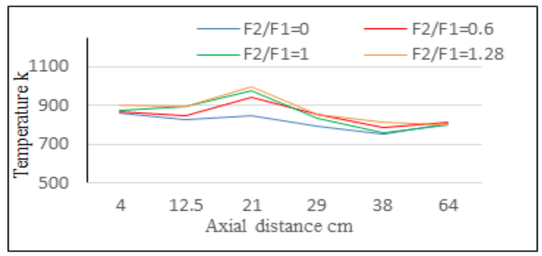

Figure 7, temperature profile along the center line of the combustor for 60 cm compressor outlet pressure.

for combustor performance figures: 8, 9 and 10 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 each running case of compressors pressure while the overall gradient in pattern factor decrease with the decreasing in compressor pressure due to the elongation of reaction zone by the secondary line of fuel injection (S. L. F. I.) and decreasing in effectiveness of penetration cooling effects of all zones holes caused more temperature homogeneity inside the combustion chamber, approximately constant average outlet temperature for all running cases.

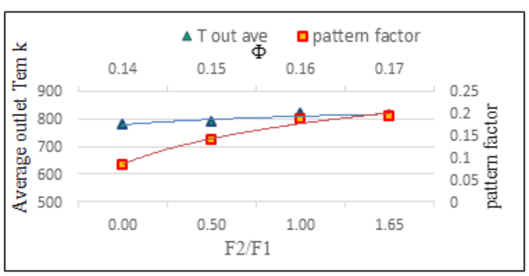

Figure 8, pattern factor and average outlet temperature for loaded turbine 100 cm compressor outlet pressure

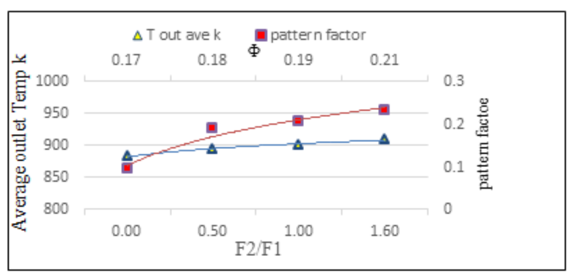

Figure 9, pattern factor and average outlet temperature for loaded turbine 80 cm compressor outlet pressure.

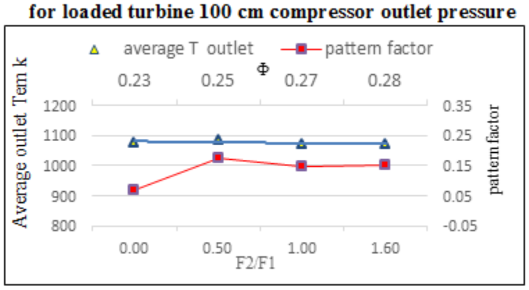

Figure 10, pattern factor and average outlet temperature for loaded turbine 60 cm compressor outlet pressure

Figures: 11, 12 and 13 shows the variation of combustion efficiency with fuel mass flow rate ratio and equivalence ratio the results shows decreasing in combustion efficiency for 100 cm H_2O head of compressor outlet pressure due to increasing in UHC and CO emissions figure 14 because of the high effectiveness of cooling holes the cooled mixture caused cooled quenching incomplete combustion for the distributed fuel while this effect decrease as decreasing in pressure 80 cm H_2O head to partial loaded turbine 60 cm H_2O head because of increasing in equivalence ratio caused continues flame caused increasing in efficiency and decreasing in emission gases CO and UHC as shown in figures 15 and 16.

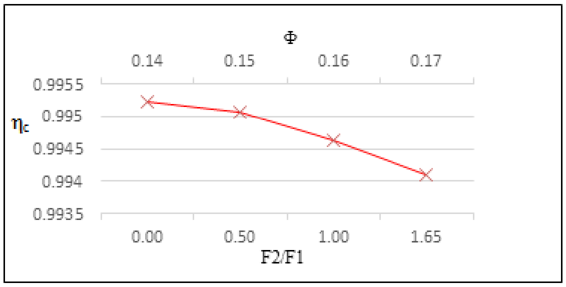

Figure 11, combustion efficiency for loaded turbine 100 cm H_2O compressor outlet pressure.

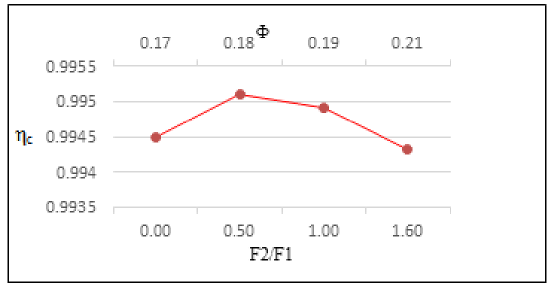

Figure 12, combustion efficiency for loaded turbine 80 cm H_2O compressor outlet pressure.

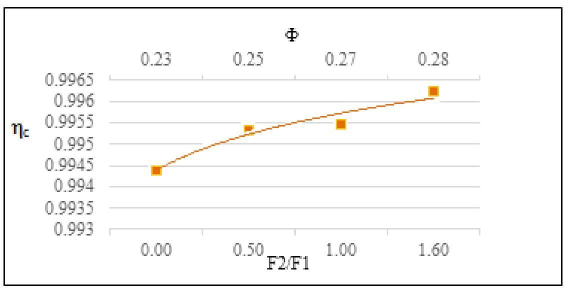

Figure 13, combustion efficiency for loaded turbine 60 cm H_2O compressor outlet pressure.

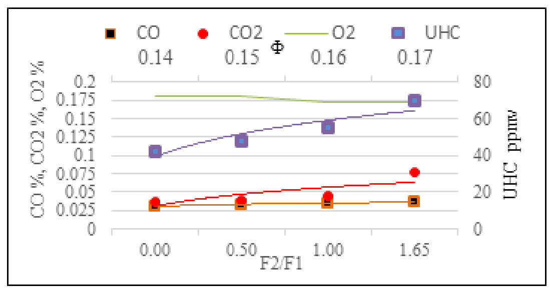

Figure 14, emission gases for loaded turbine 100 cm H_2O compressor outlet pressure.

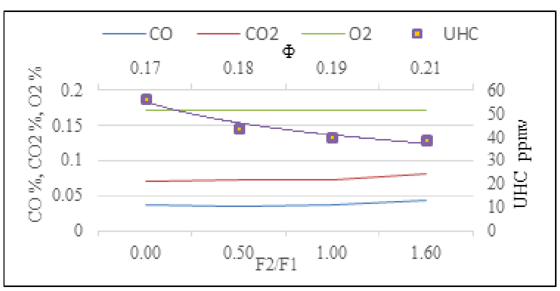

Figure 15, emission gases for loaded turbine 80 cm H_2O compressor outlet pressure.

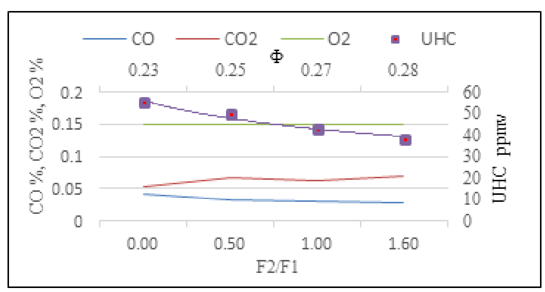

Figure 16, emission gases for loaded turbine 60 cm H2O compressor outlet pressure.

Figure 17, 18 and 19 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.

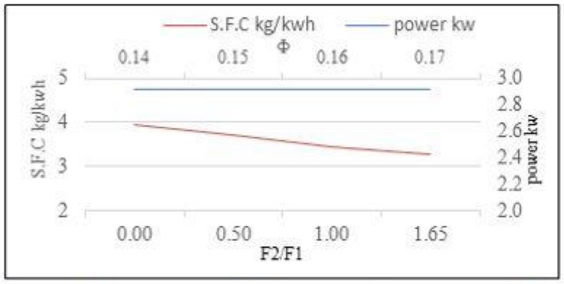

Figure 17, power and S.F.C loaded turbine 100 cm H_2O compressor outlet pressure.

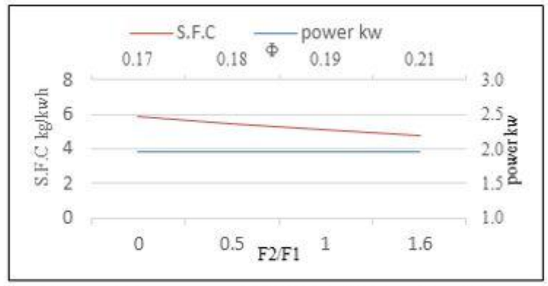

Figure 18, power and S.F.C loaded turbine 80 cm H_2O compressor outlet pressure.

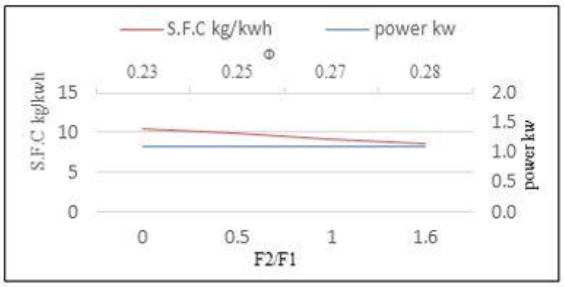

Figure 19, power and S.F.C loaded turbine 60 cm H_2O compressor outlet pressure.

9. Conclusions:

In this paper a colorless flame with high-performance combustion carried out by experimental investigation using optimization of volumetric reaction phenomenon by using multi-line of fuel injection method to burned gaseous fuel (LPG). It is found that to achieve colorless flame combustion is heavily dependent on combustor design, the method of fuel injection. The conclusions of this study are based on the objectives.

1. The colorless flame oxidation reaction mode occurred for distributed lean reaction (distributed fuel injection) and high air flow rates.
2. Equivalence ratio has an important role in distributed fuel combustion to forming continues distributed flame.
3. Cooled air and low average combustor temperature have a great effect on the distributed fuel combustion by causing cooled quenching and rise the emission gases generation.
4. The colorless flame combustion achieved extremely low emissions and high performance with the distributed flame, limit range of equivalence ration and high average temperature.
5. High speed of fuel mass flow rates (choking range) with respect to air mass flow rate helped in promoting good mixing and strong reaction resulting in a high temperature field.

10. References

[1] S. h.Cohen, G.F.C. Rogers, *Gas Turbine Theory*, Fourth ed. Longman Group Limited, Essex, England, 1996.

[2] J. W. Sawyer, "Gas Turbine Engineering Handbook,Theory and Design," *Turbomach. Int. Publ. USA, pp. 5-1 to 5-62.*, vol. 1, p. 350, 1985.

[3] G. B. Hornyák, Anita, "Numerical Modelling and Validation of the Combustion in a Gas Turbine," 2012.

[4] F. Luis Goular Dias, M. Antonio Rosa do Nascimento, and L. de Oliveira Rodrigues, "Reference Area Investigation in a Gas Turbine Combustion Chamber Using CFD," *J. Mech. Eng. Autom.*, vol. 4, no. 2, pp. 73–82, 2014.

[5] J. A. Wunning and J. G. Wunning, "FLAMELESS OXIDATION TO REDUCE THERMAL NO-FORMATION," *Prog. Energy ComlnaL Sci.*, vol. 23, pp. 81–94, 1997.

[6] A. E. E. Khalil, A. K. Gupta, K. M. Bryden, and S. C. Lee, "Mixture Preparation Effects on Distributed Combustion for Gas Turbine Applications," *J. Energy Resour. Technol.*, vol. 134, no. 3, p. 32201, 2012.

[7] A. E. E. Khalil and A. K. Gupta, "Dual Injection Distributed Combustion for Gas Turbine Application," *J. Energy Resour. Technol.*, vol. 136, no. 1, p. 11601, 2013.

[8] A. S. Veríssimo, A. M. A. Rocha, and M. Costa, "Operational, combustion, and emission characteristics of a small-scale combustor," *Energy and Fuels*, vol. 25, no. 6, pp. 2469–2480, 2011.

[9] E. Oliveira, J. R. Barbosa, and W. P. Martignoni, "Influence of Air Intake Holes's Position for Primary and Secondary Zones on the Pattern Factor for Gas Turbine with Annular Combustors Designed for Ethanol," in *International Conference Data Mining, Civil and Mechanical Engineering (ICDMCME'2014)*.

[10] W. S. Y. Hung, "Carbon Monoxide Emissions From Gas Turbines as Influenced by Ambient Temperature and Turbine Load," *J. Eng. Gas Turbines Power*, vol. 115, no. 3, pp. 1–10, 1993.

[11] H. Lefebvre and D. R. Ballal, *Gas Turbine Combustion alternative fuel and Emissions*, THIRD EDIT. Printed in the United States of America on acid-free paper, 1999.

تأثير حقن وقود متعدد الخطوط (غاز بترولي مسال) في المنطقه الثانويه لحارق أنبوبي لتوربين غازي صغير

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الخلاصة

دراسه تجريبيه لاحتراق بأداء عالي وانبعاثات قليله بدون لهب لحارق أنبوبي لتوربين غازي صغير بحقن وقود متعدد المنافذ في المنطقه الثانويه للحارق تم تطبيقه ضمن الضروف الجويه. تفاعل الأكسده بدون لهب يتميز بشعله متجانسة وموزعه ومستوى عالي من التماثل لمعدل درجات الحراره داخل غرفة الأحتراق. أداء المنضومه، ألأنبعاثات الغازيه، صور للشعله، قياس كل من درجات الحراره، معدل جريان الهواء الداخل و معدل جريان وقود غازي بترولي مسال لنسب مختلفه للتدفق الكتلي في خطوط حقن الوقود دونت لحالة أحتراق واطيئ الأنبعاثات بدون لهب، لمعدل جريان وقود 3.2 غم\ثا مع6 منافذ بقطر 1 ملم لحاقن الوقود الرئيسي و 8 منافذ بقطر 0.8 ملم لحاقن الوقود الثانوي بينت الدراسه لأعلى معدل تدفق للهواء مع جريان وقود في حالة الأختناق في الحاقن الرئيسي لكل معدل جريان وقود يعطي أحتراق بدون لهب بلون أزرق شاحب موزع بتجانس مع تأثير كبير لنسبة تكافؤ خلط الوقود على معدل درجات الحراره والتي بدورها تحدد امكانية تولد اخماد اللهب البارد ومعامل لنمط درجات الحراره مقبول تصميميا > 0.2 مع زياده في كفائة الأحتراق تصل الى 99.68% وأنخفاض في معدل استهلاك الوقود النوعي مع زياده في القدره المتولده.

الكلمات المفتاحية: – توربين غازي، حارق انبوبي، أحتراق بدون لون، انبعاث واطئ، وقود غازي بترولي مسال.