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Geometry optimization of a commercial annular RQL combustor of a micro gas turbine for use with natural gas and vegetal oils

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Abstract

A new annular RQL combustion chamber of an 80 kWel Elliott TA80R micro gas turbine was designed and validated by means of CFD simulations of natural gas combustion on modified geometries to overcome known failures at low running hours (around 2500 hrs) caused by overheating. This work provides the results of the design optimization on some geometrical parameters for fuel injection, air-fuel mixing and mixture combustion. Moreover, the new design considered simplified manufacturability and flow optimization to reduce emission while maintaining similar temperatures and efficiencies. The new combustor can easily be built with affordable overall gross costs € guaranteeing similar TIT with respect to the original geometry and with a considerable reduction of NOx emission.

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Keywords: CFD, combution chamber, micro gas turbine

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1. Introduction

The Rich Quick Lean (RQL) combustor technology decreases pollutant emissions of gas turbines, mostly NOx emissions, in many fuels characterized by complex composition and low LHV, including wood gas, biogas and also fossil fuels such as natural gas (NG) and propane [1-3]. RQL combustors define the equivalent ratio in three different

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zones: gas moves from the rich fuel condition zone to the quick mix zone and finally to the lean zone. Combustion in the rich fuel condition zone improves the stability producing and maintaining high concentration of hydrogen and hydrocarbon radical species and minimizing NOx production due to low temperatures. In the quick mix zone effluents with a high concentration of carbon monoxide, hydrogen and partially oxidized hydrocarbon with a further supply of air are completely oxidized. Finally, in the lean burn zone temperature decreases (up to 1100-1200 K for microgas turbine, mGT) with a dilution air injection to avoid turbine blades damaging and reduce NOx and CO concentration [1-4]. The equivalent ratio of the rich fuel zone is generally between 1.2 and 1.6 while in the lean fuel zone between 0.5 and 0.7 [5].

A critical issue of RQL combustors is the complexity of the system requiring a careful definition of the values of equivalence ratios in rich and lean zone to operate with low emissions. A second critical factor of the RQL combustor is the turbulent jet in the crossflow. In Douglas et al. [3] the RQL technology was integrated with a Trapped Vortex Combustion system (TVC) to improve low emissions of different fuels. In Cozzi et al. [6] the coaxial swirling air is experimentally analyzed to evaluate the influence of the fuel injection on the pollutant emissions while in Jermakian et al. [7] the effects of high pressure and temperature on jet mixing and emissions was investigated. RQL tests by GE and Multi-Annular Swirl Burner tests by Siemens-Westinghouse are presented for application to combined cycles; main operational parameters for the GT may be obtained via artificial intelligence [8-10]. In other works the CFD approach was used to evaluate the performance of a RQL combustor to improve the operational lifetime changing its design [11-14].

This work is focused on a RQL annular combustion chamber (CC) of an 80 kWel Elliot T80 gas microturbine working on wood pyrolysis gas at the Integrated Pyrolysis Regenerated Plant (IPRP) designed and built at the University of Perugia, Italy [15-17]. In previous works the authors numerically characterized and experimentally verified the original RQL combustor of the Elliot T80. The CFD model was utilized to optimize the design of the original RQL combustor and build a new geometry for low LHV gases derived from gasification, pyrolysis and anaerobic digestion and liquid biofuels; the results were compared to natural gas and diesel fuel [18-24].

This work presents the results of a design optimization on some geometrical parameters for fuel injection, air-fuel mixing and mixture combustion, for a new design [25] that considers simplified manufacturability and flow optimization to reduce emission while maintaining similar temperatures and efficiencies, also when working with vegetal oils.

Nomenclature

CC	Combustion Chamber
CFD	Computational Fluid Dynamics
EBU	Eddy Break Up
mGT	micro Gas Turbine
RANS	Reynolds Averaged Navier Stokes
RQL	Rich Quick Lean
TIT	turbine inlet temperature

2. The Micro Gas Turbine and the Combustion Chamber

The mGT, ELLIOTT TA 80 R, is composed by a radial turbo-compressor system and a recuperative heat exchanger. At full load, the electrical power of the machine is 80 kW, efficiency 28% and the nominal value of TIT is 1010 °C.

The mGT is equipped with an annular RQL (Rich, Quick mix, Lean) combustion chamber. Figure 1a shows a scheme of this combustor with the location of the three zones. Dam walls separate the rich and the lean zones where the secondary air holes and the area of quick mixing are placed; here quenching air is injected into the liner through the inner and outer dam walls, around the whole circumference to achieve quick-mix. The fuel is injected into the combustion rich zone through 12 injectors placed inside 12 mixing tubes.

Figure 1b shows a detail of the injection zone. The axis of the injector is tilted compared to that of the mixing tube to allow the fuel to interfere with the inner wall of the tube and to mix with air before entering the combustion rich zone.

This arrangement is known to produce local temperature peaks on the internal wall of the mixing tubes that rapidly brings to rupture by high temperature corrosion of the mixing tube as shown by Yakuwa et al. [26] after only 2400 hrs. The overheating of the mixing tube above the metal creep limit was well described numerically by the authors in previous works [21-22, 25]as shown in figure 2.

(a) (b)

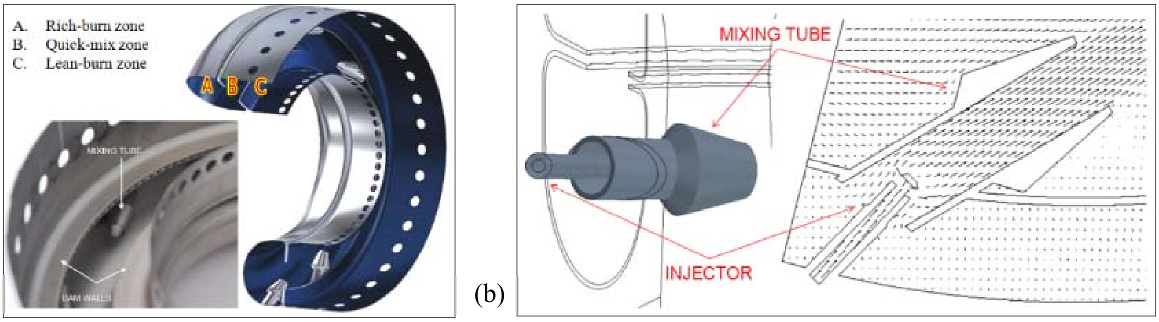

Fig. 1. (a) Annular RQL CC of Elliott TA 80 R; (b) Injector and Mixing Tube.

Fig. 2. High temperature corrosion of mixing tube [26] and its CFD explanation by the authors [21, 25]

CC substitution after as few as 2500 hrs is a critical issue for the economics of a natural gas fired mGT as a consequence of the high cost of the original spare part, given the complex geometry and costly material.

This work reports the CFD optimization analysis of the fuel injection and the of the air – fuel in a new RQL combustor concept with an easier manufacturing geometry (low cost), with respect to the original geometry, for natural gas and vegetal oil as fuel. The new concept also improves the liner lifetime, by reducing wall temperatures, while showing comparable performance in terms of emissions and TIT, with respect to the original one.

3. Methodology

3.1. Design of a new Combustion Chamber

The authors developed a new CC design to improve the combustion process and overcome the issues of the original CC in terms of manufacturability, extended lifetime and reduced emissions [25].

The length of the liner was increased by 100 mm to increase the time available for the combustion process and to decrease unwanted species due to incomplete combustion (CO, HC, intermediate products). The Secondary Air Holes are moved from the thickness of the dam wall to the external liner wall to simplify the manufacturing. The Dilution Air Holes are realized in 3 lines along the external and internal lateral walls to have small steps of equivalence ratio decreasing, to avoid the freezing of oxidation reactions of intermediate species. Figure 3 LEFT, shows the modified and original CC.

The mixing tubes were removed and changed with a thick wall provided with 12 holes where fuel and primary air are injected. This modification eliminates the complexity of manufacturing inherent to the mixing tube and eliminates the failures connected to this component.

This work presents previously unreleased analysis on the optimization of geometrical characteristics of the primary zone were to optimize fuel injection and the air – fuel mixing processing. A sensitivity analyses on the geometry of mixing hole was performed, varying the length L and the inclination α of the hole, indicated in Figure 3 RIGHT where a detail of the fuel injection zone of the new CC is shown. Three values of length L: 12.5, 20.3 and 26 mm were combined with three values of inclination α: 39, 45 and 51 degrees, obtaining 9 different geometrical configurations of the injection-mixing zone. Two different fuels are considered: natural gas and vegetal oil, this latter assumed as linoleic acid (which is the most present fatty acid in vegetal oils).

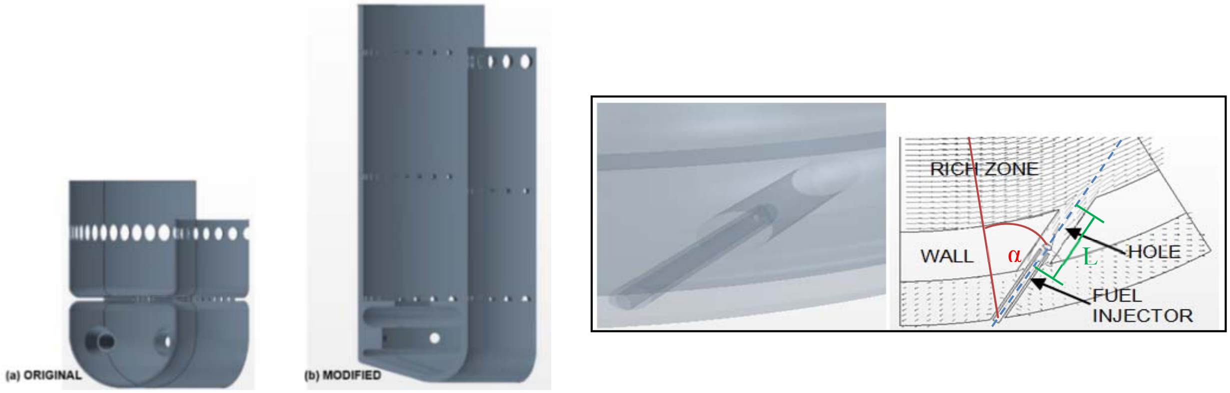

Fig. 3. LEFT 30 deg sectors of (a) original CC; (b) modified CC – RIGHT new CC: fuel injection zone [25]

3.2. CFD simulation method

Numerical simulations were carried out, using CFD software Star CCM+, in order to investigate how the geometrical variations of the primary zone influence the processes of injection, mixing and combustion.

A 3D steady RANS approach with a realizable K-ε model Two-Layer Formulation[28-29]was used respectively for turbulence and segregated fluid enthalpy model to solve the total energy equation, with the chemical thermal enthalpy as the solved variable. Temperature is then calculated from enthalpy according to the equation of state. This model is recommended for any simulation involving combustion and it is required with the non-adiabatic PPDF equilibrium combustion models, used for the cases of vegetable oil combustion. No radiation model was used based on CFD software advise [30] and on model validation with experimental data carried out in previous works [19-20].

Nine different computational domains were built, corresponding to nine different geometrical configurations. Each computational domain consists of two regions, a fluid one and a solid one. For the numerical analysis, a 30 degree sector of the annular CC, which contains one injector, is used, due to the periodical cylindrical symmetry of the volume. In all volumes, a grid mesh was generated by means of polyhedral cells with thin mesh for the solid region. The grid was generated customizing cell dimensions in order to have smaller cells (10% of the base size mesh equal to 2 mm) in critical areas, such as the injection zone and the mixing zone. Tests with different grid configurations were carried out in order to obtain grid independent solution.

A reduced combustion mechanism developed and validated by the authors [17-20] was adopted for the kinetic of chemical reaction. It involves a mixture of 7 gaseous compounds (CH_4, CO, CO_2, H_2, H_2O, O_2, N_2) and nine reactions. This scheme is based on 2 reduced mechanisms found in the Literature: the 2-step Westbrook and Dryer [31, 32] for CH_4 and CO oxidation and the 4-step Jones and Lindstedt mechanism [32] for the dissociation of CH_4 to H_2, for H_2 oxidation and reaction between CO and H_2O. A non-adiabatic PPDF approach was used to model Thermal and Prompt NOx formation. To simulate the combustion of NG in the CC, operating conditions resulting from the experimental analysis were set as boundary conditions.

For vegetal oil, assumed as linoleic acid ($C_{18}H_{32}O_2$), a Lagrangian Multiphase model is considered; it simulates the transport of the dispersed phase, consisting of liquid fuel droplets, in the gaseous continuous phase. The atomization of injected liquid phase occurs with the Reitz-Diwakar breakup model and the fuel vaporization was modeled with a Quasi-Steady Droplet Evaporation. The combustion was modeled using a Non-adiabatic Equilibrium

PPDF model. PPDF model is usable for non-premixed combustion calculations in turbulent reacting flow and it is suitable for multi-phase reacting flows, as the Lagrangian model case.

4. Results and Discussion

4.1 Natural gas simulation

Figure 4 LEFT shows the temperature distribution in the injection section, where the mixing hole is placed, for the different geometrical configurations. Observing the temperature of the jets from the nozzle, the cases with better results are those with inclination value equal to 45 degrees because in these cases (fig 4 LEFT d, e, f), the jet reaches the maximum value of temperature, close to 1900 K, due to a better air fuel mixing. In case f, with the maximum length of the mixing hole, a separation of the flame seems to begin inside the mixing hole, while in the cases with less inclination (fig 4 b, c) a separation of the flame can be observed in the internal wall of the hole. This phenomenon can lead to high temperature values in this part of the liner with the risk of corrosion of the part. For the cases with maximum value of inclination of the mixing hole (fig 4 LEFT g, h, i), the resulting swirl of the flow curves the flame leading to higher pressure drops in the primary zone.

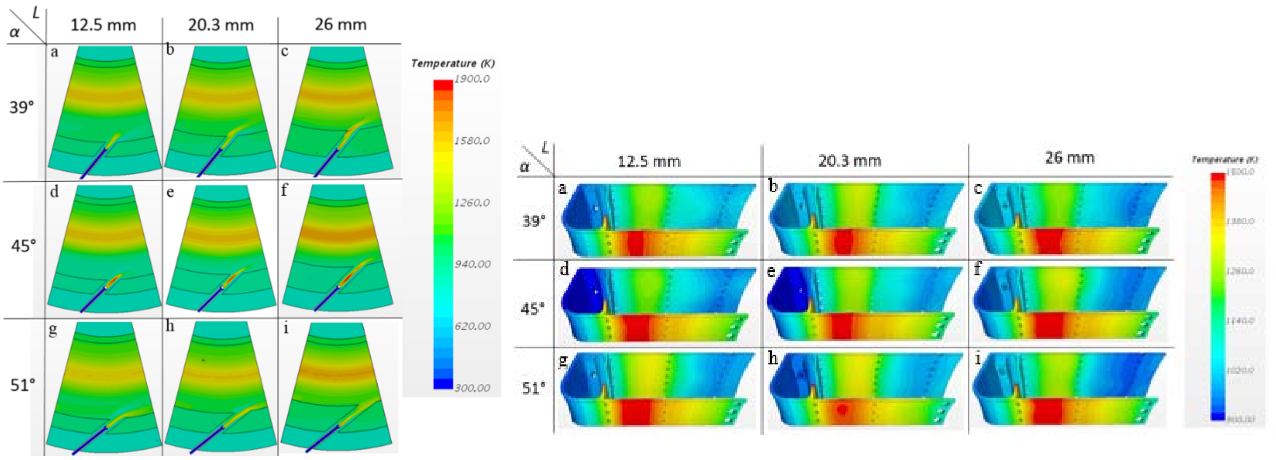

Fig. 4. LEFT: Temperature distribution in the injection-mixing zone; RIGHT: Temperature profile resulting from simulation in the new combustor for the different cases.

The effects of the different value of length L and inclination α on wall temperature are shown in figure 4 RIGHT. As expected from the observation of the jets temperature, higher temperatures are reached in case b and c, where the flames are placed in the edge of the mixing hole. However, the temperature value is slightly above 1000 K which is not harmful to the resistance of the material. The lowest values of the combustor temperature in the primary zone results for the cases d and e, with values close to 900 K. In the other cases the temperature value is between 900 and 1000 K. For all cases, the maximum value of temperature, close to 1500 K is placed in the internal part of the throat and in the internal side of the dilution zone. In this area, corresponding to the quick mix zone, the secondary air gets into the combustor and air fuel mixture are near stoichiometric condition. A secondary flame is generated with temperatures above 2000 K. The heat generated in the quick zone causes the higher temperature values of the liner.

Finally, Table \ shows the values of TIT and NOx concentration of exhaust gas exiting the combustor. The resulting values guarantee in all cases the performance of the gas turbine with expected efficiency and low emission values.

Table 1. TIT and NOx emissions resulting in CC outlet.

	a	b	c	d	e	f	g	h	i
TIT (K)	1160	1129	1129	1189	1131	1119	1112	1129	1140
Pressure (bar)	3.64	3.64	3.64	3.64	3.64	3.63	3.64	3.64	3.64
NOx (ppm@15%O2)	16	15	10	11	13	17	11	10	12

4.2 Vegetable oil simulation

Figure 5 LEFT shows the temperature distribution in the injection section for the different geometrical configurations in case of vegetable oil combustion. The most evident result for all cases is the formation of the flame inside the primary zone but outside the mixing hole. Inside the mixing hole, the fuel droplet evaporation occurs due to the high air temperature (about 700 K) and the fuel vapors are mixed with primary air. In the case of liquid fuel, the length and inclination of the mixing hole does not affect the shape of the flame, which in all cases develops at the outlet of the hole and propagates on the side of the mixing wall, where the higher temperatures are observed in the primary zone. These temperatures assume higher values than those seen for natural gas, reaching near 2500 K and the solid part of the liner (Figure 5 RIGHT), in the primary zone has also higher temperatures, with higher values for cases e and i with temperature close to 1300 K. The maximum value of temperature is always placed in the internal part of the throat and in the internal side, but for vegetal oil the values are lower than natural gas, close to 1400 K.

Table 2 shows the values of TIT and NOx concentration of exhaust gas exiting the combustor for vegetal oil combustion. TIT values result slightly lower than natural gas case but enough to guarantee in all cases the performance of the gas turbine efficiency. NOx values are very close to those of the natural gas case.

Table 2. Vegetable oil case: TIT, pressure and NOx emissions resulting in CC outlet

	a	b	c	d	e	f	g	h	i
TIT (K)	1104	1089	1107	1085	1106	1094	1086	1069	1084
Pressure (bar)	3.17	3.18	3.18	3.18	3.17	3.17	3.17	3.17	3.17
NOx (ppm@15%O2)	16	18	19	18	18	17	16	17	15

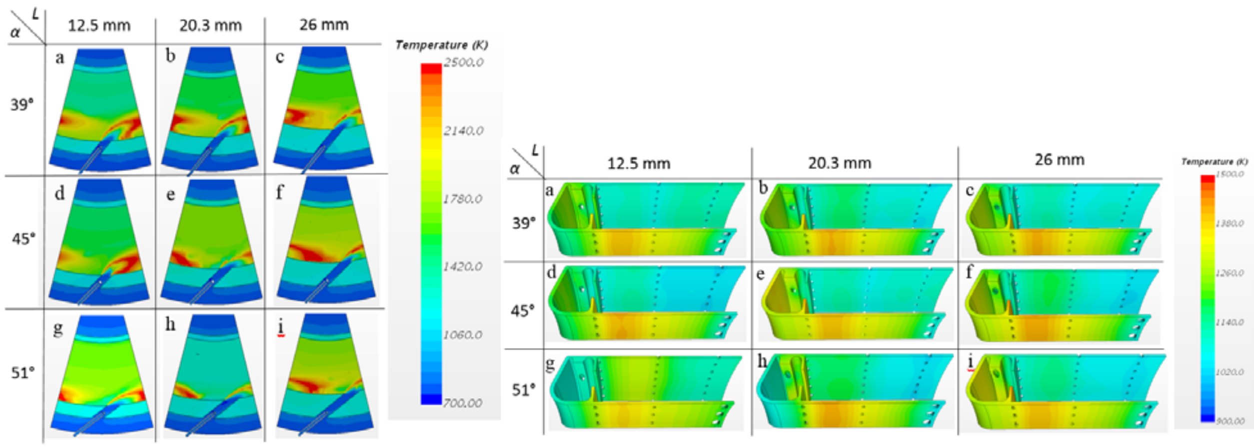

Fig. 5. Vegetable oil case: LEFT: Temperature distribution in the injection-mixing zone; RIGHT: Temperature profile resulting from simulation in the new combustor for the different cases.

4.3 Thermodynamic Analysis

Table 3 shows the values of power and efficiency of the turbine cycle calculated for both natural gas (NG) and vegetable oil (VO) cases by means thermodynamic analysis based on the values of TIT and pressure resulted from CFD analysis (table 1 – 2) and values of flow rates of air and fuels imposed as boundary conditions in the simulations.

Table 3. Thermodynamic analysis: Power and efficiency.

		a	b	c	d	e	f	g	h	i
Power (kW)	NG	77.2	70.1	80.2	83.8	70.6	67.7	66.5	70.2	72.8
	VO	41.8	38.8	42.6	38.8	41.8	39.3	37.5	34.2	37.2
Efficiency (%)	NG	23.1	20.9	24	25.1	21.1	20.3	19.9	21	21.8
	VO	12.5	12.6	12.8	11.3	12.5	11.8	11.2	10.2	11.1

For NG combustion the best turbine performances result for case d (L = 12.5 mm and α = 45°).

The corresponding power is 83.8 kW and the efficiency is 25.1%. Also case a (lower length and lower inclination) and c (higher length and lower inclination) present good results in terms of power and efficiency, close to the values of machine datasheet.

About the comparison between the fuels NG and VO, observing the net mechanical power and efficiency average values related to the fuel used, NG obtains performances higher than 40% with respect to VO. This is confirmed by a TIT that is 40 - 100 °C higher for NG and a CC outlet pressure 12-15% higher also for NG.

This difference is due to the physical nature of the fuel, since the oil is injected at a temperature close to the ambient conditions, and therefore for its heating and evaporation, an amount of heat is required and this is subtracted from the combustion process, thereby lowering the efficiency of the CC. It should also be considered that the injection zone is a cold point which also has negative effects on the distribution of pressures for the VO cases. For vegetable oil combustion the best turbine performances result for cases c and a, in agreement to the natural gas case, and for case e (mean length and inclination). For both fuels, the increase of inclination of mixing hole leads to lower values of resulting power and efficiency.

The combustor was eventually designed and built according to geometry e) and to verify the manufacturability and related costs, was mounted on an existing natural gas engine and it is has continuously run without failures for over 27.000 hours. Total cost is close to € 3.000 (against an estimated cost of 20,000 euro for the original combustor) and requires ca 10 hours of production time. The new liner was realized soldering 3 parts: main part comprehends the Rich zone and the throat, is realized by casting the material and then cutting the inner and the out profiles. The 12 holes are realized drilling the part. Two sheets are cut in rectangular shape and drilled, then are bended to get two cylinders and welded to the main part [26].

5. Conclusions

This work is focused on a RQL annular combustion chamber (CC) of an 80 kWel Elliot T80 gas microturbine and presents the results of a design optimization consisting in substituting critical mixing tubes with mixing holes and varying the length L and inclination α of the hole. Three values of L, namely 12.5, 20.3 and 26 mm were combined with three values of α, namely 39, 45 and 51 degrees, obtaining 9 different geometrical configurations of the injection-mixing zone. They are labeled a, b, c for L respectively 12.5, 20.3 and 26 mm and α = 39°, d, e, f, for 45° and g, h, i for 52°. Two different fuels were considered: natural gas and vegetal oil, this latter assumed as oleic acid (which is the most present fatty acid in vegetal oils). For Natural Gas cases d, e, f, show that the jet reaches the maximum value of temperature, close to 1900 K, due to a better air fuel mixing while in the cases with less inclination (b, c) a separation of the flame can be observed in the internal wall of the hole. Higher values of inclination induce a higher swirl of the flow which can lead to higher pressure drops. The effects of the different value of length L and inclination α on wall temperature show that higher temperatures are reached in case b and c, where the flames are placed in the edge of the mixing hole. The lowest values of the combustor temperature in the primary zone results for the cases d and e, with values close to 900 K. For vegetable oil the most evident result for all cases is the formation of the flame inside the primary zone but outside the mixing hole. In the case of liquid fuel, the length and inclination of the mixing hole does not affect the shape of the flame, which in all cases develops at the outlet of the hole and propagates on the side of the mixing wall, where the higher temperatures are observed in the primary zone. These temperatures assume higher values than those seen for natural gas, reaching near 2500 K. For this reason, the solid part of the liner, for the case of liquid combustion, in primary zone has temperature higher than the case of gas combustion natural gas.

When observing net mechanical power and efficiency related to the fuel used, NG results in higher than 40% performances with respect to the use of vegetable oil are obtained.

The combustor was eventually designed and built according to geometry e). The total production cost is close to € 3.000 and required roughly 10 hours, against an estimated cost of € 20,000 euro for the original combustor. The combustor runs without failures for over 27.000 hours.

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