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THE EFFECTS OF BURNING ALCOHOL FUELS IN FUEL FLEXIBLE ANNULAR GAS TURBINE COMBUSTORS ON THE OVERALL OPERATING CONDITIONS

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Abstract. *Renewable fuels, emissions and fuel flexibility are being subject of research these days, with high priority given to these topics. Much research is being carried out in this field, aiming at the design and development of gas turbine combustors to successfully burn conventional and renewable fuels. In particular, in this work the overall operating conditions and performance characteristics of an annular gas turbine combustor using ethanol and methanol fuels were calculated numerically using in-house developed computer codes, based on direct fuel injection, diffusion flames, stability loops, reaction mechanisms and combustor loading parameter. The main characteristics of the annular combustor, the flammability curves for ethanol and methanol fuels, the temperature distribution and combustion efficiency along the length of the combustor, the stability loops for the combustor loading parameter and some combustion products at combustor exhaust are predicted and analyzed. The work indicates that air mass flow rate, fuel mass flow rate, pressure, combustion volume and type of used fuel have a paramount effect on the stability of the combustion process and combustion advancement in the combustor zones. It is also shown that the equivalence ratio in the primary zone has important influence on the stability performance of the gas turbine combustion chamber.*

Keywords: *gas turbine, annular combustor, alternative fuels, loading parameter.*

1. INTRODUCTION

Gas turbines find enormous applications in aeronautical, automotive, naval, electric power generation and in oil industry. Emissions and the alternative fuels are being a major concern in recent years, so that high priority has been given to these topics (Rollbuhler, 1989). In the aeronautical industry, reducing emissions is being deeply considered. Thus, the use of mixtures of aeronautical fuels and their possible substitutes, the bio and synthetic fuels is being explored. Fossil fuels reserves are finite, non-renewable in nature and are increasingly difficult to be exploited. Moreover, problems in the international oil market are becoming increasingly common and their excessive utilization has caused serious environmental damages (Dark *et al.*, 2001).

Given these facts, programs focused on the research of alternative energy sources have been strongly encouraged in the United States, Canada, European Union, Brazil and other countries. In recent years, aiming at making aeronautical processes environmentally more friendly, researches of potential fuels for use in gas turbines have been promoted ((Minteer, 2006), (Wariman *et al.*, 2019)).

A gas turbine combustor is the device responsible for the increase of the incoming airflow temperature through the addition of fuel and its subsequent combustion. The combustor should meet several requirements, among them the capacity to easily starting the combustion and operating steadily on a wide range of conditions. Much research is being carried out in this field, aiming at the design and development of gas turbine combustors to successfully burning several types of fuels (Barbosa and Pilides, 2000).

This manuscript contributes to the development of gas turbine combustors design technology using alternative fuels. This research is focused on the development of a methodology for analyzing the reacting flow in these combustors. As part of the work performed, numerical simulations for medium and high calorific value liquid fuels were carried out. Fuels such as ethanol and methanol were used. From scientific literature one can find much information related to topics such as flammability limits, fuel flexible combustor design, performance of combustion and emissions in combustion chambers. But, these subjects are analyzed separately. It is necessary to emphasize that in this work, we show all these topics integrated into a global analysis using a computational tool, especially developed for this purpose.

2. METHODOLOGY

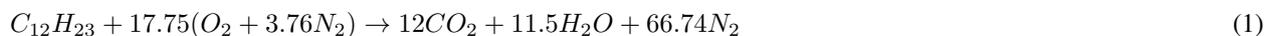
2.1 Alcohol fuels used as alternative fuels

A great advantage of any thermal engine using a continuous combustion process is its capacity to burn with success a wide variety of fuels. Usually, the gas turbine combustors are developed for a specific single fuel (for instance, kerosene or synthetic fuels). The various types of liquid fuels that are available for gas turbines depend on various factors such as costs, availability, storage, handling, pollution and combustor type. In this study, for the numerical simulation, ethanol and methanol fuels are proposed as alternative fuels instead of kerosene. Table 1 shows some of their main characteristics ((Minteer, 2006), (Lois *et al.*, 2002)).

Table 1. Typical properties for kerosene, ethanol and methanol fuels.

Property	Kerosene	Ethanol	Methanol	Unit
Formula	C ₁₂ H ₂₃	C ₂ H ₅ OH	CH ₃ OH	-
Enthalpy of vaporization	35.4	38.56	37.40	kJ/mol
Boiling point	540	351.2	337.7	K
Melting point	233	159	175.4	K
Flash point	326	286	285	K
Net Heating Value	43.2	27.2	22.7	MJ/kg
Critical temperature	683	514	513	K
Autoignition temperature	483	695	658	K
Critical pressure	3.45	6.3	8.07	MPa
Molecular weight	167.3	46.07	32.04	g/mol
Refractive index	3.64	1.36	1.33	-
Stoichiometric Air/Fuel	14.72	8.95	6.45	-
Specific gravity (15 °C)	0.808	0.794	0.787	-
Energy density	34.9	21.6	15.6	MJ/l

In the combustor the fuel is transformed from liquid to vapor, mixed with air and, then, burned with formation of combustion products (Edwards, 2002). Often the kerosene fuel is defined as C₁₂H₂₃ and the stoichiometry (molar basis) for complete combustion is calculated using the Eq. (1).



In Brazil, the anhydrous alcohol or simply bioethanol is a renewable fuel produced from sugar cane. Their properties as fuel are regulated by Resolution 36/2005 issued by the Brazilian National Petroleum Agency (ANP). This type of fuel can also be produced from waste wood and from agricultural residues (Dark *et al.*, 2001). There are physical and chemical properties of ethanol that can be beneficially explored in gas turbine combustors, while other technical difficulties must be overcome (Hamelinck and Faaij, 2006). The ethanol stoichiometric calculations in a molar basis, for complete combustion, are carried out using Eq. (2).

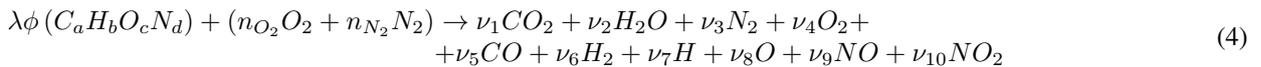


Methanol is an alcohol close relative of ethanol, excellent motor fuel and fuel blending component. It may be used as fuel in diesel engines, gas turbines and fuel cells. It is a renewable fuel with many uses. In the 19th century it was produced from biomass; presently it can be made from a series of materials and energy sources (Odgers and Kretschmer, 1986). When this fuel is produced from biomass, the sulfur content is 0.1% to 0.2%; up to 2% or higher when made from coal. Gas turbines are particularly sensible to sulfur hot corrosion and to metals in the fuel or oxidant agent. This is an injurious potential problem since methanol easily mixes with water, eventually contaminated with sodium salts and other contaminants (Paul, 1978). The stoichiometric equation for the complete combustion of methanol is given by Eq. (3).



Using a methodology based on the Gibbs free energy minimization, each fuel flammability curve is built to delimit the combustion chamber operational conditions. The fuel flammability curve is based on the adiabatic flame temperature as function of primary zone equivalence ratio. The adiabatic flame temperature can be calculated for each fuel/air ratio at given operating conditions, such as air mass flow, fuel mass flow, temperature and pressure at combustor inlet. Then a typical stability curve can be built, where the limiting fuel/air ratio can be determined against the flame limit temperature for lean and rich mixtures (Bohorquez *et al.*, 2012).

Several thermochemical assumptions were made for the calculation of the temperature at the combustor and the equivalence ratio at the primary zone. These considerations were made based on the ones adopted by McBride and Gordon (McBride *et al.*, 1993), which had been used in gas turbine combustors by Bohorquez (Bohorquez *et al.*, 2012). The chemical equilibrium condition was established using the Gibbs free energy minimization. Temperature and pressure were used as thermodynamic variables. Fuel reaction with air is modeled by Eq. (4), which includes the equivalence ratio. For this study the most significant combustion products in gas turbines were considered (Shakariyants, 2008).



where, λ is the dimensionless molar air-fuel ratio; ϕ is the dimensionless equivalence ratio; C is the monatomic carbon; CO is the carbon monoxide; CO_2 is the carbon dioxide; H is the monatomic hydrogen; H_2 is the hydrogen, H_2O is the water (gaseous state); n is the number of moles, in mol; N is the monatomic nitrogen; N_2 is the nitrogen; NO is the nitric oxide; NO_2 is the nitrogen dioxide; O is the monatomic oxygen; O_2 is the oxygen; and ν is the molar concentration, in mol.

In this research the equilibrium composition is obtained as function of temperature and pressure. The condition for equilibrium can be established in terms of the minimization of the Gibbs free energy or Helmholtz free energy or the maximization of entropy. The Gibbs free energy is most easily minimized if the variables are temperature and pressure (McBride *et al.*, 1993). The Helmholtz free energy is most easily minimized using as variables the temperature and volume (Bohorquez *et al.*, 2012).

Under several operating conditions, a set of solutions was obtained. Subsequently, based on energy conservation and adiabatic combustion, Eq. (5) was obtained. With equivalence ratio variation from lean to rich mixture, the flammability curve of each fuel was obtained. By inverse interpolation of Lagrange polynomials, for given temperature the equivalence ratio in the combustor primary zone was obtained (Bohorquez *et al.*, 2012).

$$\sum_r n_r (*hf_r + *\Delta hs_r) - \sum_p n_p (*hf_p + *\Delta hs_p) = 0 \quad (5)$$

where, $*hf$ is the enthalpy of formation, in kJ/kmol; $*\Delta hs$ is the sensible enthalpy, in kJ/kmol; p is products; and r is reactants.

It follows the calculation of the total number of moles of products, followed by the comparison of the enthalpies of the products with the enthalpies of the reactants. Afterwards, it is checked if the result of the difference between the enthalpies of the reactants and products is less than the tolerance. Then, obtain the flammability curve for each fuel using several values of the equivalence ratio as previously indicated. From the flammability curve, the equivalence ratio in the primary zone can be calculated inverse interpolation (Bohorquez *et al.*, 2012).

2.2 Stability performance

The gas turbine combustor parameters that are most important for the overall engine performance are the pressure loss through the combustor, combustion stability and combustion efficiency (Schobeiri, 2018). The pressure loss performance of a combustor is normally included in the diagram of the combustor loading and combustor intensity, which shows the corrected air mass flow rate through the combustion chamber for several fuel-air ratios. The stability loops and combustion efficiency of the gas turbine combustor can be shown as a plot of different fuel loading values for various fuel-air ratios (Mattingly, 2006).

One of the essential requirements of a fuel flex gas turbine combustion chamber is that combustion must be stable under a wide range of fuel-air ratios. This is a measure of the maximum air velocity that the combustion system can tolerate before flame extinction happens (Lefebvre and Ballal, 2010). For the flame stabilization, it is necessary to calculate its stability performance to determine a series of predetermined levels of air mass flow, fuel mass flow, air temperature and pressure, where lean blowout and rich extinction occur (Odgers and Carrier, 1973).

The stability loop must be calculated, an adequate approximation to the extinction limits may be obtained by using the Odgers-Carrier equations, solving the equations for the combustion efficiency and flammability limits for a given equivalence ratio (Odgers and Carrier, 1973). With the combustion efficiency calculated at any stable operating condition, it is possible to use these available data to find the flammability limits using the Eq. (6) to (9). There are two flammability loops for a fuel-air mixture. The weak fuel limit up to which the flame can propagate is denominated as the lower

flammability limit. The rich limit is denominated as the upper flammability limit ((Mattingly, 2006), (Odgers and Carrier, 1973)). Figure 1 shows a general diagram for the principal features of an annular gas turbine combustor. In that diagram are illustrated the basic patterns of recirculating and mixing flow in the different combustor zones (Mattingly, 2006).

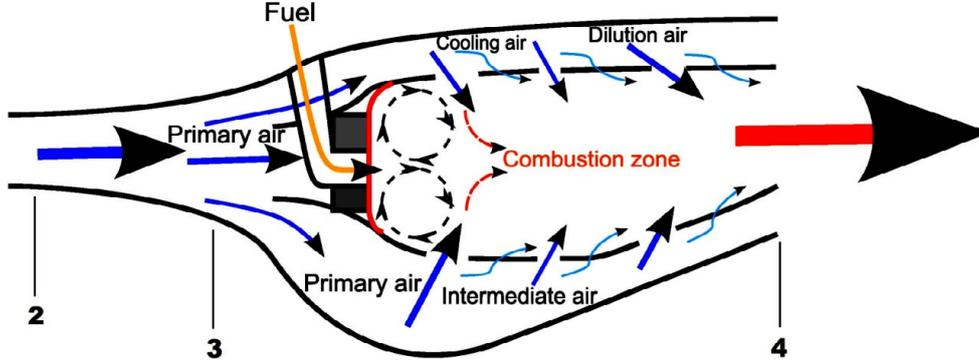


Figure 1. Flow patterns and air distribution in an annular combustor

$$\eta_c = \frac{(\dot{m}_a + \dot{m}_f) h_4 - \dot{m}_a h_3}{\dot{m}_f (LHV)} \quad (6)$$

For lean mixtures:

$$0 = 0.911 \log \left[\left(\frac{\dot{m}_f}{V_{pz} p_3^{2/\phi_{pz}}} \right) F \right] - \log \left[\log \left(\frac{1}{\eta_c} \right) \right] + 8\phi_{pz} - 1.097 + D_{weak}^* \quad (7)$$

For rich mixtures:

$$0 = 0.911 \log \left[\left(\frac{\dot{m}_f}{V_{pz} p_3^{2/\phi_{pz}}} \right) F \right] - \log \left[\log \left(\frac{1}{\eta_c} \right) \right] + \frac{8}{\phi_{pz}} - 1.097 + D_{rich}^* \quad (8)$$

$$F = \left[\left(10^{-3.054\phi_{pz}^{-1.205}} \right) \left(T_3^{1.2327\phi_{pz}^{-1.205}} \right) \right]^{-1} \quad (9)$$

where, D_{rich}^* is the dimensionless empirical constant for rich mixtures; D_{weak}^* is the dimensionless empirical constant for lean mixtures; F is the dimensionless temperature factor; h_3 is the specific enthalpy of gas entering the combustor, in kJ/kg; h_4 is the specific enthalpy of gas leaving the combustor, in kJ/kg; LHV is the lower heating value, in MJ/kg; \dot{m}_a is the air mass flow, in kg/s; \dot{m}_f is the fuel mass flow, in kg/s; p_3 is the static pressure entering the combustor, in Pa; T_3 is the temperature of gas entering the combustor, in K; V_{pz} is the primary zone volume, in m^3 ; η_c is the dimensionless combustor efficiency; and ϕ_{pz} is the dimensionless primary-zone equivalence ratio.

2.3 Temperature distribution at combustor zones

Modern combustion turbines operate at severe conditions of pressure and temperature to reduce their specific fuel consumption. Since combustor performance needs to be maximized, air available for wall cooling must be minimized in order to use most of the air leaving the compressor for the combustion, according to (Odgers and Kretschmer, 1986). Additionally, the size of the combustor is continuously decreased in volume, what requires precise calculation of the flame temperature in the combustor zones ((Motsamai, 2008), (Gherman *et al.*, 2019)).

The original methodology, developed by Kretschmer and Odgers (1986) and improved by Lefebvre and Ballal (2010), using a series of empirical formulae to calculate the flame temperature in the combustor main zones, is used in this work. For this purpose the combustion chamber is divided into four main zones: recirculation zone, primary zone, secondary zone and dilution zone. For each combustor zone, one-dimensional (1D) heat transfer analysis is made and it is assumed that the flame temperature varies linearly from the value at inlet to value at outlet ((Sawyer, 1985), (Odgers and Kretschmer, 1986), (Lefebvre and Ballal, 2010), (Topal and Turan, 2019)).

The increase on the compressor outlet temperature has a large effect on the combustor wall temperatures. It contributes to a higher temperature for the air available for cooling from the compressor. Gas turbine combustors use cooling air to reach long useful lifetime of the liner and participation of cooling air in the reaction is considered to be negligible. The liner air admission holes are needed to provide sufficient penetration of the jets into the gas turbine combustor to promote

fuel and air to flammable proportions, and rapid mixing with the combustion products and to improve the coefficient of discharge and jet positional stability ((Walsh and Fletcher, 2004), (Mattingly *et al.*, 2002)).

The annulus air flow in the neighborhood of the liner hole is generally influenced by the lower pressure inside the liner and considerable deflection of the streamlines occurs in their immediate vicinity. This produces important levels of turbulence in the gas turbine combustor. Turbulence affects the convection and radiation heat flux from the gas section as well as the convection and radiation heat flux to the annulus section. For the analysis, hot gases are flowing inside the liner and air is flowing between the liner wall and the casing. The liner wall is heated by convection and radiation from the combustion gases and it is convection cooled by the annulus airflow and radiation cooled to the outer casing. Thermal losses due to heat conduction along the liner wall are neglected. The heat balance is shown in Fig. 2 ((Lefebvre and Ballal, 2010), (Topal and Turan, 2019), (Ogders and Carrier, 1973), (Han *et al.*, 2013)).

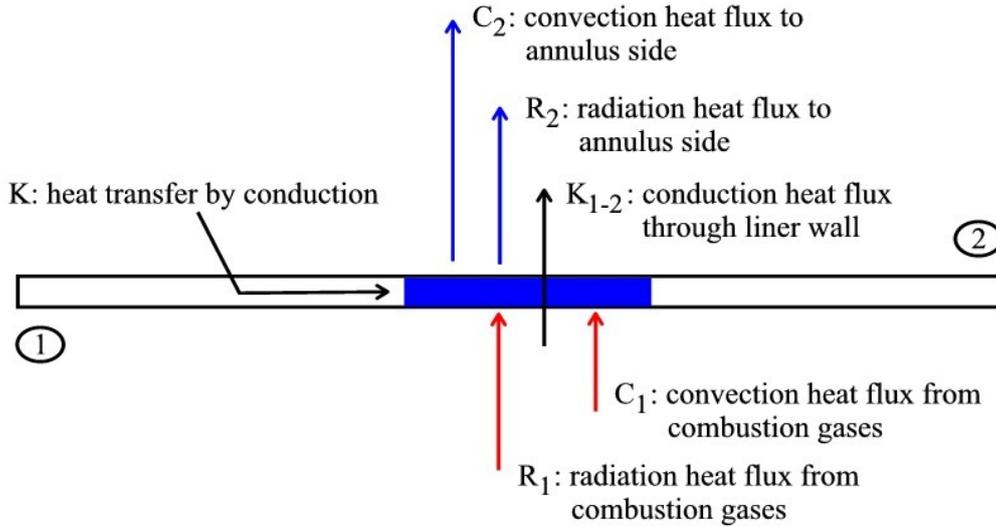


Figure 2. Heat transfer model on the liner wall of the gas turbine combustor (adapted from Lefebvre and Ballal, 2010).

Under steady-state operation conditions, thermal energy balance between hot gases side and cooling side is shown by Eq. (10) ((Lefebvre and Ballal, 2010), (Topal and Turan, 2019)).

$$\begin{cases} R_1 + C_1 = K_{wall\ 1-2} \\ R_2 + C_2 = K_{wall\ 1-2} \end{cases} \quad (10)$$

The mechanism to solve the Eq. (10) above is shown as follows ((Lefebvre and Ballal, 2010), (Topal and Turan, 2019)):

$$Nu = 0.032 \left(\frac{k}{D} \right) (Re)^{0.8} (Pr)^{0.33} \quad (11)$$

$$R_1 = 0.5\sigma (1 + \varepsilon_{wall}) \varepsilon_{gas} T_{gas}^{1.5} (T_{gas}^{2.5} - T_{wall\ 1}^{2.5}) \quad (12)$$

$$C_1 = 0.02 \left(\frac{k_{gas}}{D_L^{0.2}} \right) \left(\frac{\dot{m}_{gas}}{A_L \mu_{gas}} \right)^{0.8} (T_{gas} - T_{wall\ 1}) \quad (13)$$

$$C_2 = 0.02 \left(\frac{k_a}{D_{an}^{0.2}} \right) \left(\frac{\dot{m}_{an}}{A_{an} \mu_a} \right)^{0.8} (T_{wall\ 2} - T_3) \quad (14)$$

$$R_2 = \sigma \left[\frac{\varepsilon_{wall} \varepsilon_c}{\varepsilon_c + \varepsilon_{wall} (1 - \varepsilon_c) (A_{wall}/A_c)} \right] (T_{wall\ 2}^4 - T_3^4) \quad (15)$$

$$K_{wall\ 1-2} = \left(\frac{k_{wall}}{e_{wall}} \right) (T_{wall\ 1} - T_{wall\ 2}) \quad (16)$$

$$\varepsilon_{gas} = 1 - \exp \left[-290PL(ql_b)^{0.5} T_{gas}^{-1.5} \right] \quad (17)$$

$$L = 336/H^2 \quad (18)$$

$$T_{gas} = T_3 + \Delta T_{comb} \quad (19)$$

where, A_{an} is the area of the annulus air space, in m^2 ; A_c is the surface area of the casing, in m^2 ; A_L is the surface area of the liner, in m^2 ; A_{wall} is the area of the liner wall, in m^2 ; C_1 is the convection heat flux from gas, in W/m^2 ; C_2 is the convection heat flux to annulus air, in W/m^2 ; D is the diameter, in m; D_{an} is the diameter of the annulus air space, in m; D_L is the diameter of the liner, in m; e_{wall} is the liner wall thickness, in m; H is the fuel hydrogen content (by mass) in percent; k is the thermal conductivity, in $W/(m \cdot K)$; k_a is the air thermal conductivity, in $W/(m \cdot K)$; k_{gas} is the combustion gases thermal conductivity, in $W/(m \cdot K)$; k_{wall} is the liner wall thermal conductivity, in $W/(m \cdot K)$; $K_{wall1-2}$ is the conduction heat flux through liner wall, in W/m^2 ; L is the dimensionless luminosity factor; l_b is the beam length, in m; \dot{m}_{an} is the annulus air mass flow rate, in kg/s; \dot{m}_{gas} is the combustion gases mass flow rate, kg/s; Nu is the dimensionless Nusselt number; P is the gas pressure, in kPa; Pr is the dimensionless Prandtl number; q is the dimensionless fuel/air ratio by mass; R_1 is the radiation heat flux from gas, in W/m^2 ; R_2 is the radiation heat flux to casing, in W/m^2 ; Re is the dimensionless Reynolds number; T_3 is the combustor entry temperature, in K; T_{gas} is the gas temperature, in K; T_{wall1} is the liner wall temperature at hot gases side, in K; T_{wall2} is the liner wall temperature at annulus side, in K; ΔT_{comb} is the temperature rise due to combustion, in K; ϵ_c is the dimensionless casing emissivity; ϵ_{gas} is the dimensionless gas emissivity at temperature T_{gas} ; ϵ_{wall} is the dimensionless emissivity at liner-wall temperature; μ_a is the dynamic viscosity of air, in $kg/(m \cdot s)$; μ_{gas} is the dynamic viscosity of gas, in $kg/(m \cdot s)$; σ is the Stefan-Boltzmann constant: $5.67 \times 10^{-8} W/(m^2 K^4)$.

3. RESULTS

The combustor performance parameters on the gas turbine operational envelope, for a jet engine designed to burn ethanol and liquid methanol, were calculated using the methodology detailed above. The combustor inlet temperature varied from 295 K to 479 K; the combustor inlet pressure from 98 kPa to 359 kPa; and, the equivalence ratio from 0.1 to 2.0. Additionally, special attention was given to flammability limits and stability loops. The combustor operational characteristics are shown in Tab. 2. The stability curves for ethanol and liquid methanol are shown in Fig. 3 and 4.

Table 2. Combustor operational characteristics.

Parameters	100% Pn	85% Pn	75% Pn	50% Pn	30% Pn	25% Pn	5% Pn
Air mass flow rate (kg/s)	0.30485	0.28256	0.26988	0.22776	0.17724	0.17063	0.05850
Fuel mass flow rate (kg/s)	0.00455	0.00409	0.00390	0.00312	0.00238	0.00228	0.00130
Equivalence ratio (ϕ)	0.23284	0.22609	0.22543	0.21369	0.20939	0.2080	0.34667
Combustor entry temperature (K)	478.55	461.63	449.80	399.30	336.93	323.87	295.10
Combustor entry pressure (kPa)	358.70	327.30	307.10	220.21	138.30	124.70	97.65
Combustor outlet temperature (K)	1068.97	1018.46	987.68	880.83	759.53	739.16	798.87
Combustor outlet pressure (kPa)	326.00	298.50	280.00	196.50	118.00	104.00	93.30

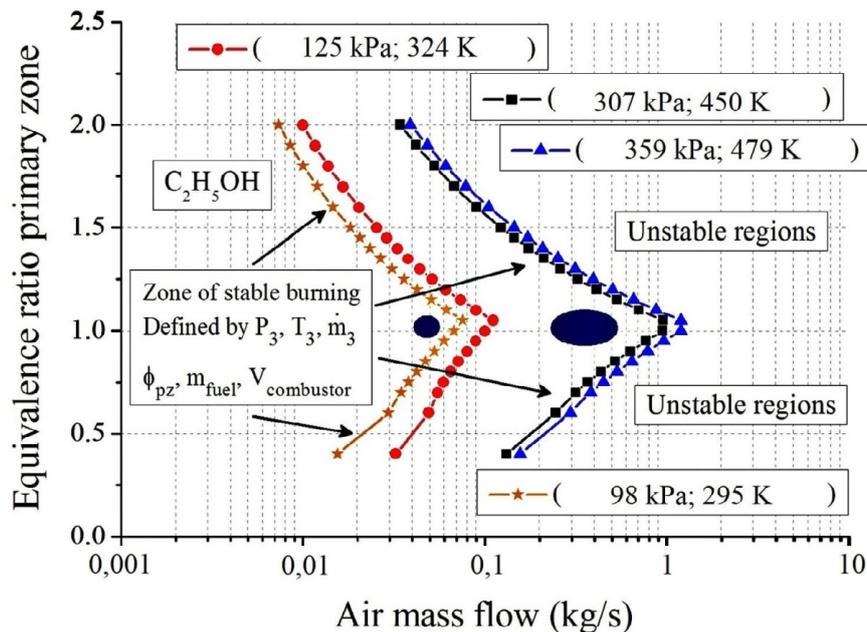


Figure 3. Stability loops for ethanol fuel.

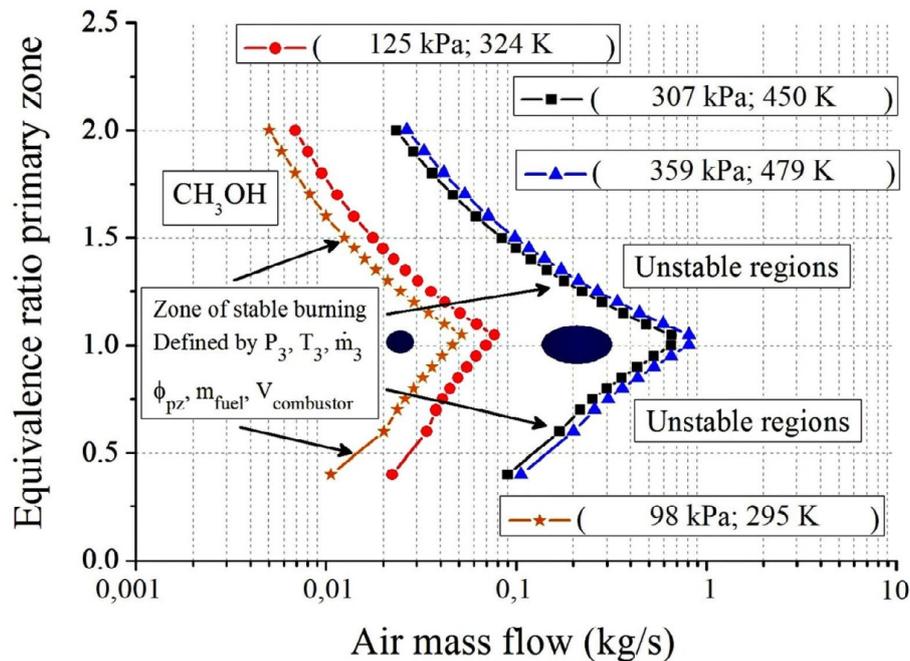


Figure 4. Stability loops for methanol fuel.

The terms stability loops and flammability zones are related parameters. Physically, these characteristics allow a correct interaction of the combustor with other components of engine (compressor and turbine). The values of the stability loops are affected by gas turbine combustor inlet boundary conditions (temperature, pressure and overall fuel-air). These factors will determine the values of the equivalence ratio primary zone within the flammability zone for all operational loads. Additionally, the values of the stability loops are affected by the combustor volume.

The combustor inlet temperature and pressure have significant influence on the combustion stability, as reflected by the stability curves. From Fig. 3 and 4, when the operating temperature is 295 K and the pressure is 98 kPa, it is seen that the stability zone is narrower. For the same operating conditions the air mass flow is less than one obtained for the operating temperature of 479 K and pressure of 359 kPa.

Figures 3 and 4 show that the pressure in the combustor is an important factor that affects the zone of stable burning, since the stability limits are reduced when pressure is reduced, causing reduction of the combustion speed and greater recirculation in the combustor primary zone. This is of great importance, because for any gas turbine combustion chamber there is a limit in equivalence ratio for both lean and rich mixtures, beyond which the flame will be unstable and flame out would occur.

Analysis of the stable burning zone for the two fuels studied, namely ethanol and liquid methanol, shows that when equivalence ratio of the air-fuel mixture at primary zone is lean, the air mass flow rate increases until the equivalence ratio reaches the stoichiometric ratio. On the other hand, if the air-fuel mixture is rich, the air mass flow rate decreases until the equivalence ratio is approximately 1 to 1.1. From the results shown before, it can be seen how the major differences take place in the air distribution for the primary, secondary and dilution zones, which would require an appropriate thermochemical development and reliable flame stability during the combustor operation.

The fuel flexible gas turbine combustors have severe operating conditions, especially due to the high compressor pressure ratios. Therefore, this will produce higher temperatures and pressures at the combustor entrance. As the inlet pressure in the combustor is increased, the reaction rate formation of products becomes less significant and the major problem is related to providing suitable conditions so that the combustor can withstand the combined effects of high load deformation and high temperature metals.

It is highlight that the type of fuel has a marked effect on the stability and combustion advancement in the combustor zones. This can be seen when the primary zone is analyzed under a steady state operating condition. At full power, the pressure is 358.7 kPa and the temperature 478.55 K at the combustor inlet. Then, the equivalence ratio in the primary zone is 1.05 for kerosene, 1.06 for ethanol and 1.03 for methanol. The primary zone airflow for the same operating conditions is 28% of the total airflow for kerosene, 28% of the total airflow for ethanol and 28% of the total airflow for methanol. This means that the combustor can reach similar flame stability conditions, whereas the combustion efficiency will require richer fuel/air mixtures of ethanol or weaker fuel/air mixtures of methanol are burnt instead of kerosene.

The approach used in this study allows the calculation of the temperature distribution in all the combustor zones. These results are summarized in Fig. 5 and 6. From the results shown in Fig. 5 and 6 it can be seen how the increase of pressure and temperature at combustor inlet has a double effect on the temperature of the combustor walls: increases

the heat transfer rate by radiation to the combustor walls, resulting in the increase of the cooling air temperature and the increase of cooling air temperature will produce a greater consumption of this flow to keep the temperatures of the combustor walls within the allowable range of temperature.

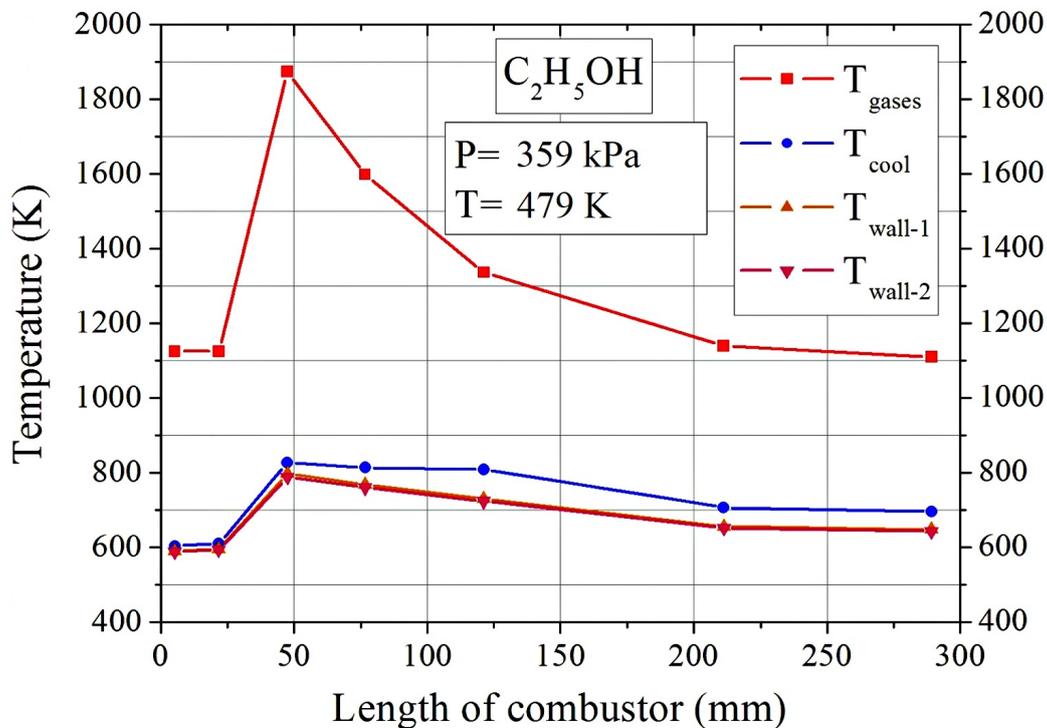


Figure 5. Temperature distribution for combustor burning ethanol.

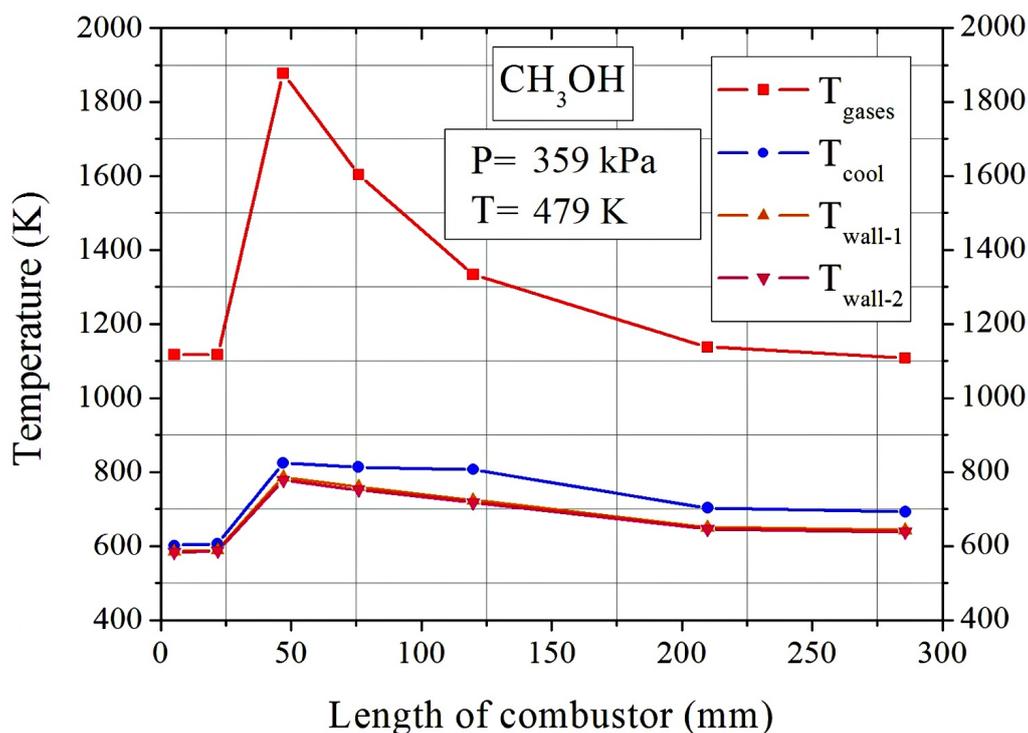


Figure 6. Temperature distribution for combustor burning methanol.

4. FINAL REMARKS

The main objective of this study is to develop a methodology to obtain the performance predictions of a fuel flexible gas turbine combustion chamber and systematically apply it as a computational tool. The methodology used was implemented in a FORTRAN code. A gas turbine combustor was analyzed and the results of the simulations indicate the functionality and versatility of the model. The use of successive substitution combined simultaneously with LU factorization and the Newton-Raphson procedure allowed the calculation more quickly, avoiding the usual problems with singularities.

Although the results show the outcomes for only two fuels (ethanol and liquid methanol), the program is able to run any type of gas turbine fuel, requiring only the fuel chemical composition and the thermodynamic coefficients. Main parameters considered for fuels implementation are: heating value of the fuel, fuel mass flow rate, air mass flow rate and air distribution in the combustor.

Despite the differences between thermochemical characteristics of kerosene with liquid methanol and ethanol, the optimal distribution of the airflow through the holes in the primary and secondary zones shows no great change with respect to that obtained for kerosene fuel.

5. ACKNOWLEDGEMENTS

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