THE EFFECT OF USING BIOFUELS ON GAS TURBINE PERFORMANCE

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Abstract. *Power gas turbines are increasingly used in a lot of applications ranging from small to medium and heavy-duty engines. Although they are fed by conventional fuels, when powered by biofuels they become "greener". The utilization of biofuels in gas turbines is being investigated during the last decades. In the present work, a set of liquid and gaseous biofuels is considered as candidate substitutes of the original conventional fuel (diesel oil and natural gas) in gas turbines. Data from a commercial single-shaft power gas turbine are used to set up a virtual test case. Aiming to investigate the effect of the use of biofuel to the performance of the gas turbine, a simple model based on thermodynamic relations is set up and used to simulate the steady state gas turbine operation at design and off-design loads.*

Περίληψη. *Οι αεριοστρόβιλοι ισχύος χρησιμοποιούνται όλο και περισσότερο σε πλήθος εφαρμογών σε όλο το εύρος ισχύος τους, από μικρής ή μέσης ισχύος ως και βαρέος τύπου μηχανές. Αν και οι αεριοστρόβιλοι τροφοδοτούνται με συμβατικά καύσιμα, όταν χρησιμοποιούν βιοκαύσιμα θεωρούνται πιο «πράσινοι». Η χρησιμοποίηση βιοκαυσίμων σε αεριοστροβίλους αποτελεί αντικείμενο έρευνας τις τελευταίες δεκαετίες. Στην παρούσα εργασία, ένα σύνολο από υγρά και αέρια βιοκαύσιμα θεωρούνται ως υποψήφιοι αντικαταστάτες των συμβατικών καυσίμων (πετρελαίου και φυσικού αερίου) σε αεριοστροβίλους. Με βάση δεδομένα εμπορικού αεριοστροβίλου καθορίζεται μία εικονική περίπτωση μελέτης. Με σκοπό να εξεταστεί η επίδραση των βιοκαυσίμων στις επιδόσεις του αεριοστροβίλου, χρησιμοποιείται ένα απλό μοντέλο που στηρίζεται σε απλές θερμοδυναμικές σχέσεις και προσομοιώνεται η μόνιμη λειτουργία του στο ονομαστικό σημείο λειτουργίας και σε φορτία εκτός σημείου λειτουργίας.*

INTRODUCTION

Power gas turbines are increasingly used nowadays in a lot of land applications ranging from small to medium and heavy-duty engines. Although gas turbines are fed by conventional fuels, when powered by biofuels, they are considered to be “greener”. However, “many critics express concerns about the scope of the expansion of certain biofuels because of the economic and environmental costs associated with the refining process and the potential removal of vast areas of arable land from food production” [1]. The utilization of biofuels in gas turbines is being investigated during the last decadeσ both from a scientific [2], [3], as well as from an industrial point of view [4]. Apart from issues having to do with the type of combustion and burner modifications that are required in order to burn such fuels effectively and efficiently, the performance of the gas turbine is affected by the physical and chemical properties of the biofuel and mainly by its lower heating value.

In the present work, some liquid and gaseous biofuels, known from the literature, are considered as candidate substitutes of the original usual conventional fuel (diesel oil and natural gas) in gas turbines. Information concerning a commercial, heavy duty, single-shaft power gas turbine applications is used to define a virtual gas turbine test case with realistic data. In order to model its design load operation, a simulation tool based on simple thermodynamic relations (zero-order model) was programmed. By a trial-and-error method, the required data were evaluated so that the predicted performance approximately matches that provided by the manufacturer data sheet. Then, the above tool was extended by means of an iterative method in order to predict the design load operation when another fuel of a different lower heating value is considered. Thus, the design load operation is simulated and compared for various biofuels. Aiming to also investigate the effect of using biofuels to the part-load performance of the gas turbine, the simulation tool is further extended by means of a second iterative procedure in order to match the given partial loads for each biofuel. The results, concerning the design and off-design operation of the gas turbine under consideration for various biofuels, are presented and compared each other. Conclusions are drawn and future directions for the continuation of this research are proposed.

THE SINGLE-SHAFT POWER GAS TURBINE

Gas Turbine (GT) is a heat engine, i.e. an engine that transforms heat energy into mechanical work, based on the Joule-Brayton ideal thermodynamic cycle. Gas turbines are compact engines that are used either as jet engines for aircraft propulsion or as power engines in land

and marine applications. As power engines, they are utilized in a wide power range. We are concerned herein with power GT in its simplest configuration, namely the single-shaft one, the model of which is depicted in Fig.1(left). In its steady state operation, air is sucked by the compressor (C) that raises its pressure, then it is burned with fuel in the burner (B) and finally the high-pressure hot gas expands in the turbine (T) to produce work; a part of it moves the compressor, while the rest is the net work of the plant. Referring to the real rather than the ideal cycle of a real gas turbine, an inlet (points 1-2 in Fig.1(left)) and an exit duct (points 5-6) are also used, the compression in the compressor and the expansion in the turbine are not isentropic and the burner has not an ideal performance.

Various methodologies and tools of different levels of sophistication are used in order to simulate and study the operation of a GT. In the present work, a zero-order thermodynamic model has been programmed and implemented for the simulation of the steady state operation of the gas turbine at its design point. The model applies simple thermodynamic relations (under appropriate assumptions) that relate the working medium state before and after each component of the system. The data required for such a tool for the simulation of the performance of a GT are: air mass flow rate at the inlet ($\dot{m}_a$), environmental conditions (pressure p_o and temperature T_o), pressure loss coefficients for the inlet duct (K_{in}), exit duct (K_{ex}) and burner (K_b), pressure ratio (r_C) and isentropic efficiency ($\eta_{is,C}$) for the compressor, combustion chamber efficiency (η_{cc}), turbine inlet temperature (T_4) and isentropic efficiency ($\eta_{is,T}$), mechanical efficiency for the shaft (η_m), properties (isentropic exponent, specific heat capacity under constant pressure) of air (γ_a, cp_a) and gas (γ_g, cp_g), as well as Lower Heating Value (LHV) of fuel (q_f). (These data can be found in summary in a next section in Table 2).

Referring to Fig.1(left), in order to quantify the performance of the GT, calculation of the state of the working fluid (pressure and temperature) is first required at each characteristic point of the plant (1)-(6). To this end, the steps presented in the following Simulation Procedure 1 (SP1) are carried out.

SP1 (given T_{t4}, $\dot{m}_a$):

1. $T_1 = T_o$, $\quad p_1 = p_o$, $\quad T_2 = T_1$, $\quad p_{t2} = \left(1 - K_{in}\right) p_{t1}$
2. $T_3 = T_2 \left\{1 + \dfrac{1}{\eta_{is,C}}\left(r_C^{\varepsilon_a} - 1\right)\right\}$, $\quad \varepsilon_a = \dfrac{\gamma_a - 1}{\gamma_a}$, $\quad p_3 = r_C\, p_2$
3. $\dot{W}_C = \dot{m}_a c_{pa}\left(T_3 - T_2\right)$ (power provided to C), $\quad w_C = \dfrac{\dot{W}_C}{\dot{m}_a} = c_{pa}\left(T_3 - T_2\right)$ (specific work)

4. $f=\dfrac{\dot{m}_f}{\dot{m}_a}=\dfrac{c_{pg}T_4-c_{pa}T_3}{\eta_b q_f-c_{pg}T_4}$, $\quad p_4=\left(1-K_b\right)p_3$, $\quad p_{t5}=\dfrac{p_{t6}}{\left(1-K_{ex}\right)}$

5. $r_T=\dfrac{p_4}{p_5}$ (turbine pressure ratio), $T_5=T_4\left\{1-\eta_{is,T}\left(1-\left(\dfrac{1}{r_T}\right)^{\varepsilon_g}\right)\right\}$, $\varepsilon_g=\dfrac{\gamma_g-1}{\gamma_g}$, $T_6=T_5$

Then, the performance of the GT is quantified by evaluating the following quantities:

Net power of the GT: $\quad \dot{W}=\dot{W}_T/\eta_m-\dot{W}_C$

Specific net work of the GT: $\quad w=w_T/\eta_m-w_C$

Thermal efficiency of the GT: $\quad \eta_{th}=\dfrac{w_\kappa}{fq_f}$

Specific fuel consumption of the GT: $\quad sfc=\dfrac{1}{\eta_\theta q_f}$

In the above formulas, the following quantities have also been used:

Power produced by the T: $\quad \dot{W}_T=\left(\dot{m}_a+\dot{m}_f\right)c_{pg}\left(T_{t4}-T_{t5}\right)=\dot{m}_a\left(1+f\right)c_{pg}\left(T_{t4}-T_{t5}\right)$

Specific work of T (per unit air mass entering the engine): $w_T=\dfrac{\dot{W}_T}{\dot{m}_a}=\left(1+f\right)c_{pg}\left(T_4-T_5\right)$

Specific heat provided to the working medium in B: $\quad q=(1+f)c_{pg}T_4-c_{pa}T_3=\eta_b fq_f$

It is well known that two parameters play important role for the operation of a GT, namely the pressure ratio r_C (the higher it is the greater the thermal efficiency of the cycle is) and the maximum cycle temperature T_4 (the higher it is the greater the work produced by the cycle is). Due to the fact that the inlet temperature corresponds to environmental temperature, the maximum temperature, i.e. the so-called Turbine Inlet Temperature (TIT) is usually expressed by the ratio $t=T_4/T_1$. By using the above procedure SP1, parametric studies concerning the operation at design point of the gas turbine can be performed. A usual approach is to vary the pressure ratio (r_C) of the compressor for standard value of the maximum to minimum cycle temperature ratio t (i.e. $t=T_4/T_1$). A typical result of such a study (in a form that can be found in many textbooks, e.g. [5], [6]) is the bi-parametric performance diagram presented in Fig.1(right). This diagram depicts the performance of the gas turbine in terms of specific work-thermal efficiency (w-η_{th}) curves for the values of t that are shown in the figure, while the variation of r_C was limited in the range 4-20 with step 2 (thus, 8 values were used to generate each of the t=constant performance curves). For each couple (r_{C1}, t_1), the point on the t_1-curve that corresponds to the value r_{C1} has as coordinates the specific work and the corresponding efficiency at design load operation of a gas turbine with those characteristics.

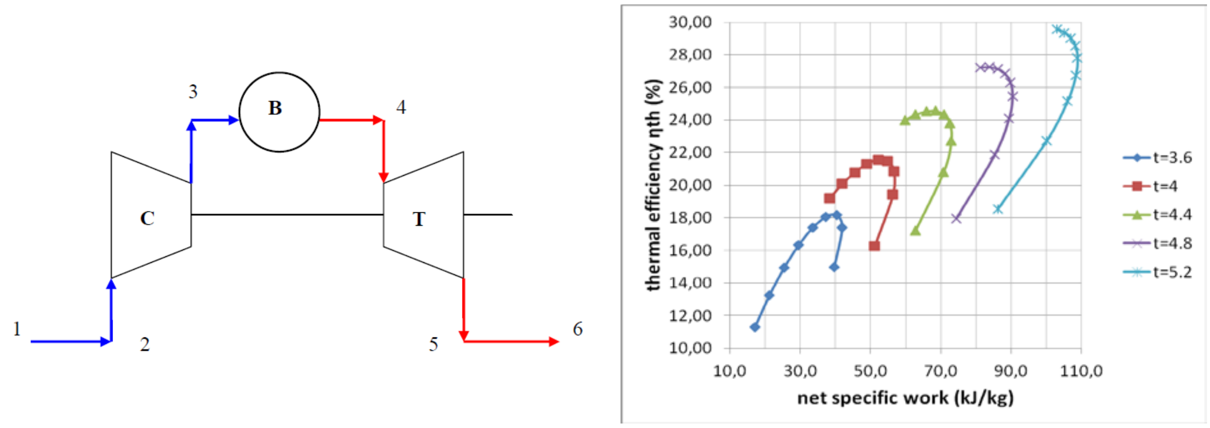

Fig.1 *Single-shaft gas turbine (left). Typical bi-parametric performance map (right).*

BIOFUELS AND GAS TURBINES

Biofuels are organic products that are derived from biomass, and they are classified in renewable sources of energy. Their use leads to reduced greenhouse gas emissions having a positive contribution to the environmental impact of their combustion. Both liquid and gaseous biofuels are referred in the literature. Their main properties, playing an important role for their combustion, are the Lower Heating Value (LHV), viscosity, density, molar weight and C/H ratio. Consideration of biofuels as alternative fuels for gas turbines and discussion of their properties can be found in [7]. Various aspects of the use of biofuels in gas turbines can also be found in the review papers [8] and [9].

Three liquid and three gaseous biofuels from those mentioned in the above literature are considered in the present work. The liquid ones are biodiesel, bioethanol and biomethanol, while the gaseous ones are syngas, biogas and hydrogen. Biodiesel is produced by vegetable oils or animal fat, bioethanol is produced by alcoholic fermentation of various vegetable products containing sugar, while biomethanol is produced by syngas. Syngas is produced by means of thermal gasification of biomass, biogas is a mixture of different gases that are produced in the course of anaerobic fermentation of organic material and hydrogen is produced by gasification of biomass or electrolysis. Two conventional fuels are also been considered for comparison purposes, one liquid (diesel fuel) and one gaseous (natural gas). Table 1 presents these biofuels along with their state and LHV (found in the literature). According to it, all biofuels have a lower LHV compared to the two conventional fuels except hydrogen that has about three times greater LHV than them.

In the present study, no further topics are discussed for the biofuels under consideration, like for example chemical aspects for their production, their availability concerning the secure of energy, the energy or the possible environmental impact required for their production, their economics, issues concerning health impact (for example toxicity), flammability (being important also for their storage), etc. In addition, possible modifications required in conventional gas turbine combustion chambers are not considered or discussed herein. Instead, the focus of the present study is on the effect of the utilization of biofuels in the performance of the GT.

Fuel	kind	state	LHV (kJ/kg)
Diesel	conventional	liquid	44800
Biodiesel	biofuel	liquid	40000
Bioethanol	biofuel	liquid	25000
Biomethanol	biofuel	liquid	20000
Natural gas	conventional	gaseous	42800
Biogas	biofuel	gaseous	22500
Syngas	biofuel	gaseous	19000
Hydrogen	biofuel	gaseous	120000

Table 1. *Lower Heating Value (LHV) of conventional fuels and biofuels (liquid and gaseous) that are considered in the calculations of the present study.*

SIMULATION OF GAS TURBINE AT DESIGN LOAD OPERATION

Case study description

In order to study the effect of the use of different biofuels to feed a gas turbine, a virtual case study was set up. In order to have realistic data, a commercial, heavy-duty, power gas turbine was selected and its data-sheet was accessed in the internet [10]. Based on the information provided for this engine, a trial-and-error procedure was manually performed in order to find appropriate values for the data required by the simulation tool SP1, such that the predicted performance of the engine approximately matches that of the manufacturer datasheet. The set of data for this engine together with the performance indices estimated by SP1 for design load operation are summarized in Table 1.

Data	Value	Performance index	Value
r_c	18	$\dot{W}$ (kW)	25064
$\eta_{ic,c}$ (%)	83	η_{th} (%)	0,337
$\eta_{ic,t}$ (%)	90	sfc (kg_f/kWh)	0,249
T_4 (K)	1523	T_6	834,1
η_b(%)	98	$\dot{m}_g$ (kg/s)	70,5
q_f (kJ/kg)	42800		
K_{in} (%)	1,5		
K_b (%)	4		
K_{ex} (%)	2		
η_m (%)	100		
$\dot{m}_a$ (kg/s)	68,76		

Table 2. *Evaluation of data for SP1 in order to approximately match design load performance of a commercial gas turbine.*

Simulation of design load operation for different fuels

In order to simulate this particular engine when utilizing different fuels, only the fuel LHV was considered to change. No other modifications concerning fuel or combustion gas properties were taken into account. Thus, to simulate the performance of the GT when using a different fuel, the LHV of the new fuel can be substituted in SP1 and new results are obtained. If doing so, higher fuel consumption is estimated for fuels with lower LHV than the conventional one. This is a consequence of the fact that in the methodology used in SP1, the maximum temperature after the combustion chamber (TIT) is prescribed, i.e. in order to retain the same TIT using a fuel with lower LHV, more fuel is required. However, in this case, the power produced by the gas turbine is higher than its nominal one due to the higher exhaust mass flow rate. This of course is not correct, since the capability of a particular engine to produce power is standard as prescribed by the designer (and in case of choked turbine, the maximum exhaust mass flow rate cannot be overcome).

The implementation of such an (inappropriate) approach would lead to results like the ones depicted in Fig. 2, where the bi-parametric map of the engine has shifted to the right, i.e. each w-η_{th} curve has moved towards greater specific work values for the same pressure ratio and maximum cycle temperature that is definitely not correct. Instead, the manual trial-and-error procedure that was described in a previous section has to be repeated for each fuel. In order to avoid such a hard approach and to be able to simulate correctly the engine performance (since for the same engine design, the same maximum power has to be produced independent what the fuel is), a second simulation procedure (SP2) was set up. According to this, the nominal engine power and the exhaust gas mass flow rate were considered as given data and new values of f, m_a, T_5 corresponding to the new LHV were calculated. Simulation Procedure (SP2) is described below.

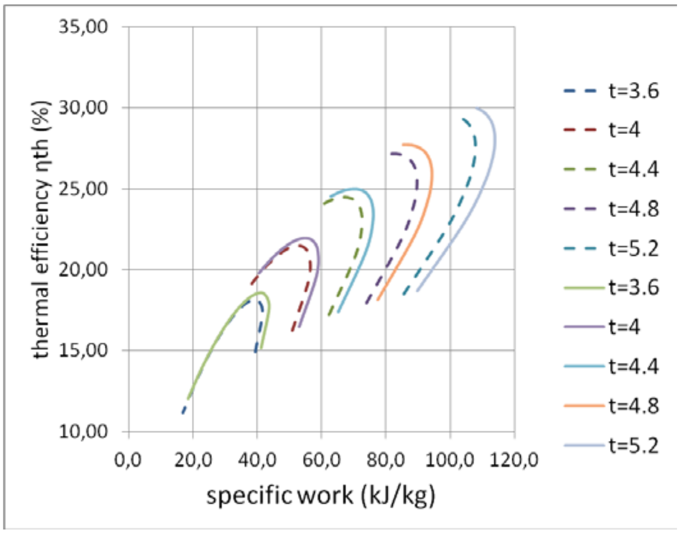

Fig. 2. *Generation of bi-parametric maps for use of natural gas (dashed lines) and biogas (continuous lines) in the same engine, as an example of curve shifting, corresponding to wrong prediction of different work capability as a consequence of inappropriate calculations.*

SP2 (given $\dot{W}_\kappa$, $\dot{m}_g$):

Steps 1-5 as in SP1. After assigning initial values to T_4 and f, an iterative procedure starts:

6. $T_5 = T_4\left\{1-\eta_{is,T}\left(1-\left(\frac{1}{r_T}\right)^{\varepsilon_g}\right)\right\}$, $\quad \varepsilon_g = \frac{\gamma_g - 1}{\gamma_g}$ $\quad \dot{m}_a = \frac{\dot{m}_g}{1+f}$

7. $w_\kappa = \frac{\dot{W}}{\dot{m}_a}$, $\quad w_T = \eta_m\left(w_C + w\right)$, $\quad T_4 = T_5 + \frac{w_T}{c_{pg}\left(1+f\right)}$

8. $f = \frac{c_{pg}T_4 - c_{pa}T_3}{\eta_b q_f - c_{pg}T_4}$. If $\left|\frac{f-\tilde{f}}{f}\right| > 10^{-5}$ go to (6), otherwise stop and set $T_6 = T_5$.

Results for design load operation

By means of the procedure SP2, the required data for the engine under consideration, as well as some performance indices were calculated for each different biofuel at the design load. Fig. 3 presents graphically the corresponding results in barchart form. In particular, Fig. 3(a) presents graphically the biofuels in increasing LHV order. Figures 3(b), 3(c) and 3(d) present the fuel-air ratio, the TIT and the thermal efficiency of the GT at its design load corresponding to each of these fuels. As the LHV increases, the fuel-air ratio decreases, i.e. less fuel is required for the production of the same power. The corresponding TIT increases as a consequence of the greater LHV and the thermal efficiency also increases. Except hydrogen that has by far the greatest LHV, for all other biofuels that have lower LHV than the conventional fuels, significant increase in fuel consumption is predicted, the TIT is decreased and thermal efficiency decreases.

Fig. 4 presents similar information in the form of a curve. Thus, Figures 4(a), 4(b), 4(c) and 4(d) depict the curve of TIT, fuel-air ratio, thermal efficiency and specific fuel consumption against fuel LHV. As one can notice TIT and thermal efficiency increase, while fuel-air ratio and specific fuel consumption decrease with the increase of the fuel LHV. However, two parts can be identified in these curves. The first of them

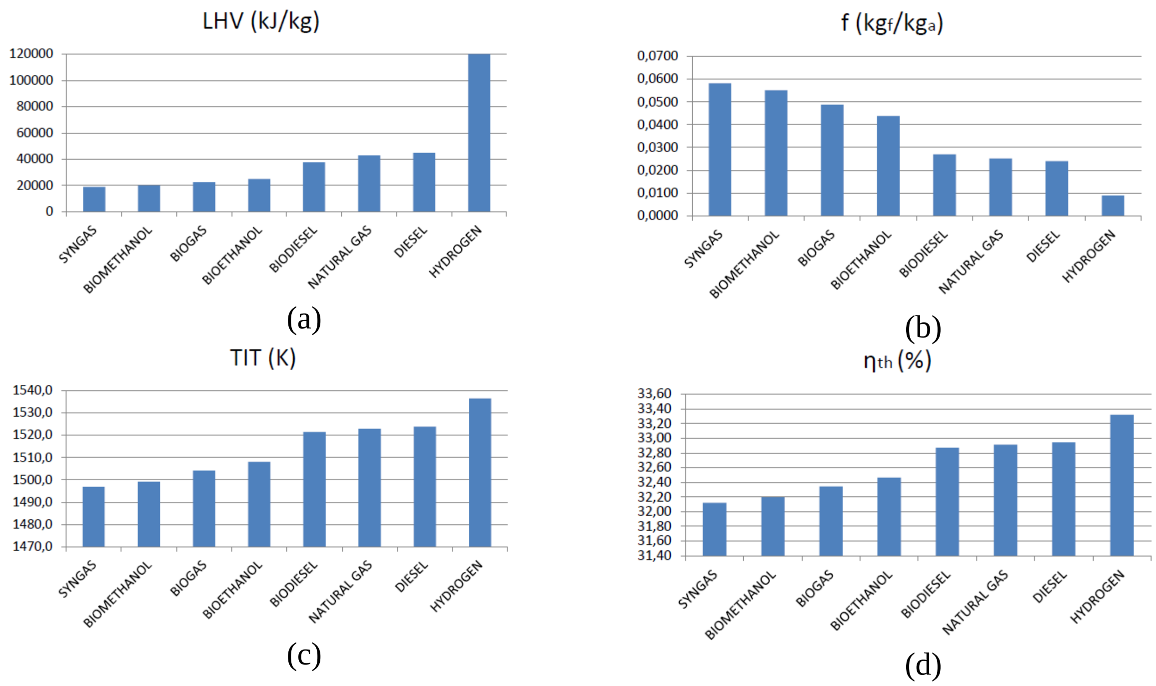

Fig. 3. *(a) Fuels considered in increasing LHV order. (b) Required fuel-air ratio for each fuel. (c) Achieved TIT for each fuel. (d) Thermal efficiency for each fuel.*

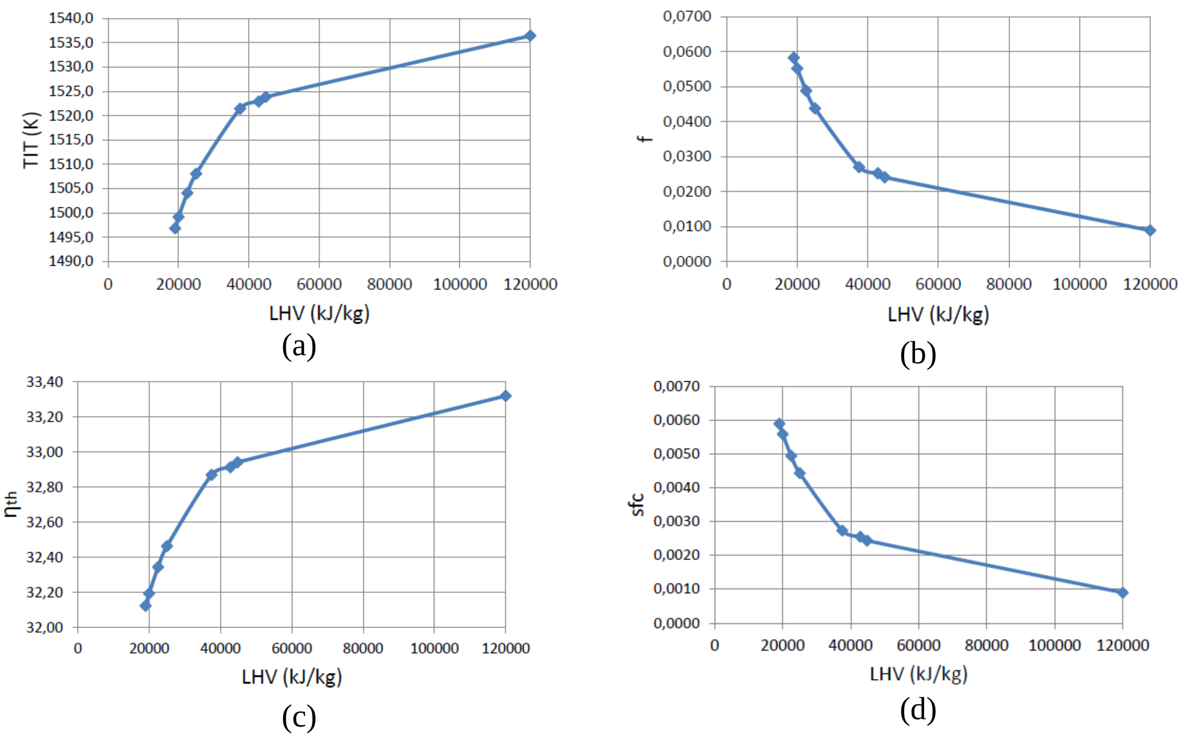

Fig. 4. *(a) TIT, (b) fuel-air ratio f, (c) thermal efficiency and (d) special fuel consumption vs fuel LHV.*

SIMULATION OF GAS TURBINE AT PART-LOAD OPERATION

Methodology for part-load GT operation

In order to investigate the effect of using different biofuels to the gas turbine also in part-load operation, a model to simulate such an operation had to be set up and programmed. Whenever a power turbine operates in partial loads, a control strategy is required to adapt its operation to part-load conditions. According to [11], where such strategies are reviewed, in the case of heavy duty gas turbines, the methods that are mainly used are the reduction of the air mass flow rate entering the compressor, realized by throttling an inlet guide vane (IGV), the reduction of the TIT, realized through the control of fuel mass flow rate entering the combustion chamber or combination of these two methods. According to the literature [12], the latter method is necessary in order to cover a wide range of part-load conditions with high efficiency operation.

In the present work, the control of TIT by varying the fuel mass flow rate was implemented as described in [12]. In this case, the mass flow rate entering the engine was kept constant (constant engine rotation speed rpm and constant isentropic compressor efficiency were assumed), while the compressor pressure ratio was considered to be appropriately reduced.

The fact that the rotation speed remains constant is compatible for example with the scenario that the gas turbine is connected with a generator for electric power generation. In order to implement the TIT control method, a third Simulation Procedure (SP3) was set up. Having solved for the design load for each fuel by means of SP2, SP3 solves for part-load operation, when the partial (off-design) load $\dot{W}_{off}$ is given. The procedure is described below.

SP3 (given $\dot{W}_{off}$):

Steps 1-8 as in SP2. Then, an iterative procedure starts:

9. $w_{C,off} = w_C = \dot{W}_C / \dot{m}_a$, $\quad w_{off} = \dot{W}_{off} / \dot{m}_a$

10. $T_{4,off} = T_4$, $\quad T_{5,off} = T_5$

11. $\tilde{T}_{4,off} = T_{4,off}$

12. $w_{T,off} = \eta_m (w_{off} + w_{C,off})$, $\quad T_{4,off} = T_{5,off} + \dfrac{w_{T,off}}{(1+f_{off})c_{pg}}$

13. $f_{off} = \dfrac{c_{pg}T_{4,off} - c_{pa}T_{3,off}}{\eta_b q_f - c_{pg}T_{4,off}}$, $\quad p_{4,off} = \dfrac{(1+f_{off})\sqrt{T_{4,off}}}{(1+f)\sqrt{T_4}/p_4}$

14. $p_{3,off} = \dfrac{p_{4,off}}{(1-K_b)}$, $\quad r_{C,off} = \dfrac{p_{3,off}}{p_2}$

15. $T_{3,off} = T_2\left\{1+\dfrac{1}{\eta_{is,C}}\left(r_{C,off}^{\varepsilon_a}-1\right)\right\}$, $\quad \varepsilon_a = \dfrac{\gamma_a - 1}{\gamma_a}$

16. $r_{T,off} = (1-K_{in})(1-K_b)(1-K_{ex})r_{C,off}$

17. $P_r = \dfrac{r_{T,off}}{r_T}$, $\quad \eta_{is,T,off} = \eta_{is,T}(0.6849 + 0.6866P_r - 0.3714P_r^2)$

18. $T_{5,off} = T_{4,off}\left\{1-\eta_{is,T,off}\left[1-\left(\dfrac{1}{r_{T,off}}\right)^{\left(\frac{\gamma-1}{\gamma}\right)_g}\right]\right\}$

19. If $\left|\dfrac{T_{4,off}-\tilde{T}_{4,off}}{T_{4,off}}\right| > 10^{-4}$ go to (11), otherwise stop.

Results for part-load operation

By means of the SP3, the part-load operation was simulated for ten different conditions for each fuel. In particular, the the partial to the design load ratio, i.e $a = \dot{W}_{off} / \dot{W}$ was used to describle part-load conditions. Thus, simulations were performed for the values of α=0.1-1.0 with step 0.1 (i.e. for the 10%, 20%, …, 100% of the design load).

Fig. 5 presents the TIT for various part-load conditions for liquid, as well as for gaseous fuels. For each fuel, the TIT increases as the load increases and attains its maximum value at the design load. Generally, different TIT values are achieved by the various fuels for a fixed load, but no significant differences are noticed due to the different fuels.

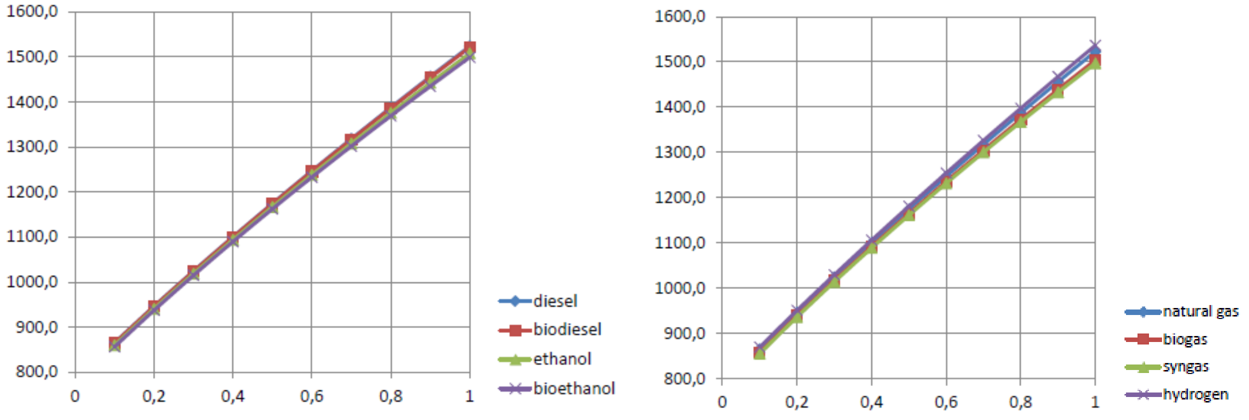

Fig. 5. *TIT vs % of design load for liquid fuels (left) and gaseous ones (right).*

Similarly, Fig. 6 presents the thermal efficiency for various part-load conditions for liquid, as well as for gaseous fuels. For each fuel, thermal efficiency increases as the load increases and attains its maximum value at the design load. Again, no significant differences in thermal efficiency are noticed for a fixed load due to different fuel.

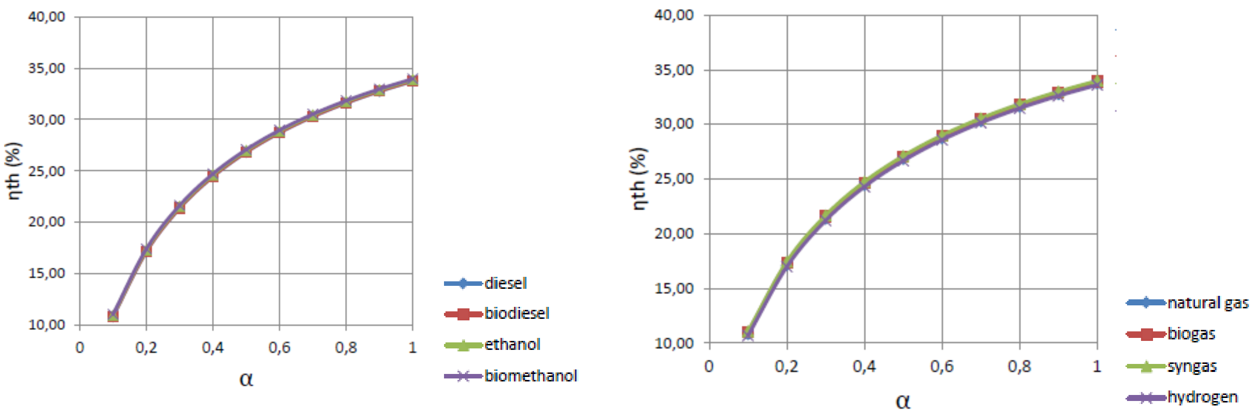

Fig. 6. *Thermal efficiency vs % of design load for liquid fuels (left) and gaseous ones (right).*

Fig. 7 presents the fuel consumption in part-load operation for the various fuels considered, in terms of mass flow-rate. For a fixed fuel, the consumption increases linearly with the increase in load. The lower the LHV, the greater the slope of this line is. For a fixed load, the lower the LHV, the greater the fuel consumption is.

Fig. 8 presents the special fuel consumption sfc in part-load operation for the various fuels. For a fixed fuel, the consumption is minimum at design load, then exhibits a slow almost linear increase with the decrease in load up to the 40% of its design value and finally increases abruptly for small partial loads, i.e. at small loads significantly more fuel is required to produce a unit of energy. The lower the LHV, the greater the slope of the linear part is. For a fixed load, the lower the LHV, the greater the sfc is.

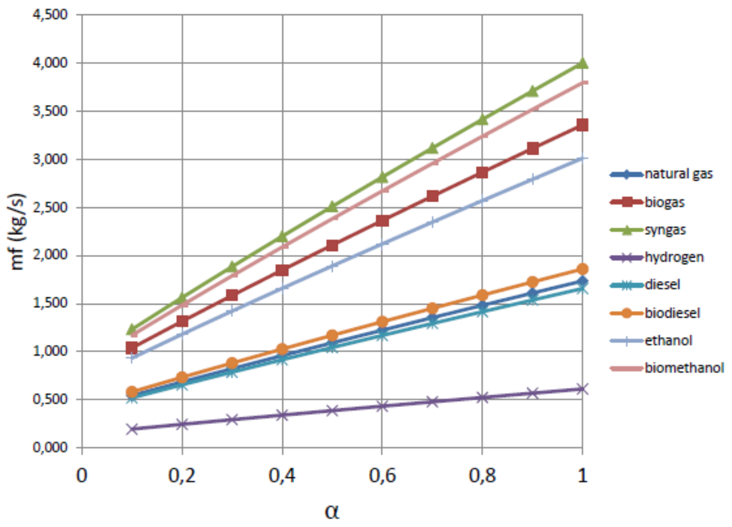

Fig. 7. *Fuel consumption in part-load operation for the various fuels considered.*

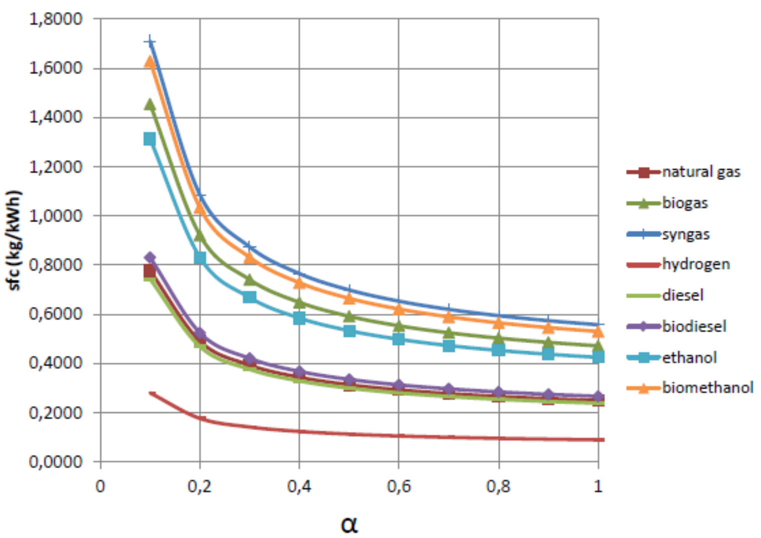

Fig. 8. *Special fuel consumption in part-load operation for the various fuels considered.*

CONCLUSIONS

In the present work, the effect of the utilization of biofuel in a heavy-duty gas turbine at design and off-design loads was studied by a simple model under appropriate assumptions. According to the results, the LHV of biofuels, being generally lower compared to that of conventional gas turbine fuels (natural gas, diesel oil), affects the performance of the engine. The lower the heating value of the fuel is, the lower the max cycle temperature achieved in the combustion chamber and the higher the required fuel consumption for the same design load. Thermal efficiency of the engine remains almost constant for various fuels. Similar remarks were found for part-load operation. In this case turbine inlet temperature was found to increase linearly with load for all fuels. Fuel consumption also increased linearly with load; the lower the heating value of the fuel, the higher the rate of increase.

Some directions for the continuation and gradual extension of the present work could be to implement a more sophisticated simulation tool (that will consider blade cooling and also simulate two-shaft gas turbines), to carry out similar studies for medium, small, micro gas turbines, to quantify the effect of using of biofuels on the environmental impact (e.g. in terms of CO_2 emissions), to consider other factors like toxicity selecting biofuels as alternative fuels for gas turbines. Furthermore, to search for Greece-oriented biofuelled GT applications and corresponding cost-effective techniques for sufficient biofuel production that ensure availability of the fuel and secure of energy (since lower LHV of biofuels requires significantly more amount of biofuel), to investigate the potential and viability of utilizing biofuels in maritime applications and finally to put the whole study in a Life Cycle Analysis (LCA) perspective.

ABBREVIATIONS

GT = Gas Turbine

LHV = Lower Heating Value

TIT = Turbine Inlet Temperature

SP = Simulation Procedure

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