

Analysis on using biomass lean syngas in micro gas turbines

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Abstract. The paper presents an analysis on small systems for converting biomass/wastes into power using Micro Gas Turbines (MGT) fed with gaseous bio-fuels produced by air-gasification. The MGT is designed for burning various fossil liquid and gas fuels, having catalogue data related to natural gas use. Fuel switch changes their performances. The present work is focused on adapting the MGT for burning alternative low quality gas fuel produced by biomass air gasification. The heating values of these gas fuels are 3 to 5 times lower than the methane ones, leading to different air demand for the stoichiometric burning. Validated numerical computation procedures were used to model the MGT thermodynamic process. Our purpose was to analyze the influence of fuel change on thermodynamic cycle performances.

1. Introduction

Biomass and organic residues can be used as renewable energy source for the production of gaseous derived fuels. The gasification process and technologies involve a decrease in process energy efficiency compared to combustion because of the endothermic characteristic that limits this efficiency to about 0.6 – 0.7 (cold gas efficiency) significantly below the 0.85 – 0.98 specific to combustion processes [1]. Biomass source size and time availability limit the power generation to small scale applications. Moreover, the thermodynamic cycles with good energy conversion efficiency in the low power range (up to 15-20 MWel) are Brayton and Otto-Diesel [2, 3, 4, 5]. The lower energy efficiency of the gasifying process is balanced by the superior energy efficiency of the gas turbines and reciprocating spark ignition engines in the scale range of the applications. The society evolution involves the increase of food industry as well as of its residues to be disposed. Nevertheless the waste streams have flows significant lower than fossil fuels ones. Consequently, waste to energy applications demand small power units in the range of 40 kW - 500 kW [1]. In this range the steam turbines efficiency is usually limited to 0.12 – 0.18 (with power output starting from 150–200 kW) while the gas turbines and engines generally maintain their conversion efficiency of about 0.3 – 0.4 (with power output starting from 20–200 kW) [3, 6, 7, 8].

The gas turbines directory performances are reached by using standard gas fuel – the methane. The shift from natural gas to lean gases, that are usually produced through gasification processes, conduct to significant changes in turbines behavior that are subject to current investigations. This research area is developing and the results provide information both to equipment manufacturers and system engineers for a better assessment of biomass to energy conversion solutions using gasification process.

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A MGT unit, using natural gas, up to 500kW_{el} that can be used also in cogeneration was analyzed for the case of burning alternative lean syngas from air gasification of organic waste. The syngas composition (specific energy content) was varied according to different feedstock (wood biomass and organic residues from food industry), and the thermo-chemical process (atmospheric pressure air gasification – the most common type) [9, 10, 11]. Consequently a range of syngas type was used in our computation from low to higher quality gas fuel using as reference the methane.

The paper presents the effect of syngas physical-chemical properties on Micro GT performances.

2. Materials and methods.

2.1. Gas fuel properties

The syngas fuel used in our research comes from the air gasification of wood biomass and organic residues from meat processing industry. Two different feedstock were considered as variation range for the produced syngas quality. The syngas produced by wood gasification has a higher specific energy content related to feedstock properties (high combustible fraction, average humidity and low inert content). The food industry waste consists in meat and bones residues (low specific energy content due to water content –45%, and inert fraction – 30%) [9]. Consequently, the syngas energy properties will be inferior to the first one. Feedstock and syngas detailed properties are presented in previous works of the authors [9]. The common bibliographic sources report the syngas composition on dry basis that is a theoretical situation only. The syngas humidity used in the analysis was chosen based on condensing the H₂O vapors in surface heat exchangers cooled with river water at average temperature in temperate climate. By cooling the syngas nearly to 31.4°C, the water vapors molar participation in the syngas, with a relative humidity near to 100%, will be about 4.6%.

The range of the syngas components are: CO (17.9÷33.5%); H₂ (11.1÷20.8%), N₂ (33.3÷57.9%), CO₂ (4.4÷7%), C_nH_m (0.5÷1.%). Syngas molar masses (22÷26 kg/kmol) are higher compared to methane. The volumetric Higher Heating Value of these gases (HHV varies between 4.3 MJ/Nm³ symbol D and 8 MJ/Nm³ symbol A) are about 5 to 15 times lower compared to CH₄ [9].

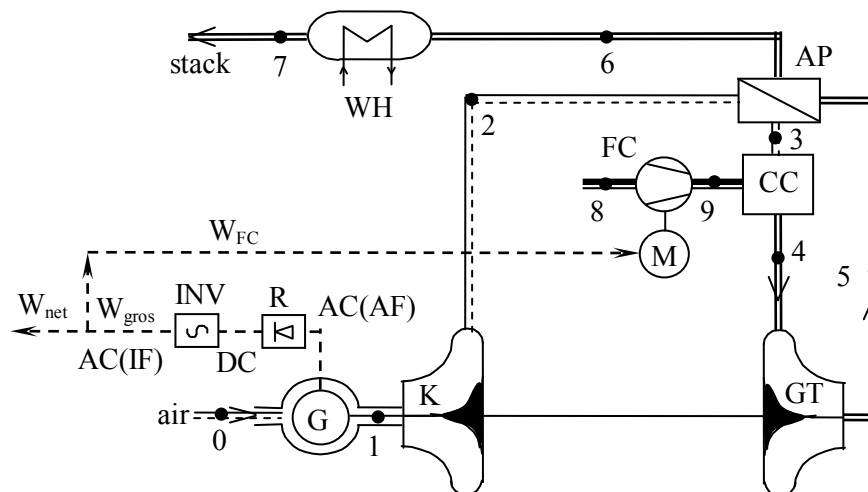


Figure 1. Typical micro GT design in single shaft arrangement.

Note: G-generator; K-air compressor; GT-gas turbine; M-engine; FC-fuel compressor; CC-combustion chamber; AP-air pre-heater; WH-water heater; AC-alternative current; R-rectifier; DC-direct current; INV-inverter; W-power.

Almost all MGT employs an internal recovery Air Preheater (AP). This consists into a

regenerative gas-air surface heat exchanger. The compressed air is pre-heated by recycling the thermal potential of turbines exhaust gases.

The syngas obtained by air gasification contents a reduced fraction of combustible gases. Consequently the process performances will be affected if this new gas fuel will be used and some components of the MGT will need redesigning [12].

2.2. Problem formulation

The two non-dimensional factors used to analyze the MGT thermodynamic process are: **1)** the compression ratio ($\Pi = p_{\max}/p_{\min}$), and **2)** the ratio of extreme absolute temperatures ($\theta = T_{\max}/T_{\min}$). When Π is fixed and θ increases, the electrical efficiency (η_{el}) and the specific work relative to flue gas mass flow rate (W_{sp} - kJ/kg), are increasing too. Consequently becomes interesting to raise $T_{\max}$, with minimum intervention on the MGT components.

It is generally known [13] that η_{el} and W_{sp} can be maximized at a specific value of Π . For classic Brayton cycle, without AP, at a given θ value:

$$\Pi (W_{sp \max}) < \Pi (\eta_{el \max}) \quad (1)$$

The influence of the AP on MGT process leads to [14, 15]:

$$\Pi (\eta_{el \max}) < \Pi (W_{sp \max}) \quad (2)$$

These characteristics ensure good electrical efficiency for low Π . The MGT are equipped with compressor for the fuel. Therefore the unit power and its efficiency given in directories consider the compressor energy consumption and the values listed are net.

The engineering computations necessary to analyze the MGT cycle use lots of variables and nonlinear equations. The analytical solving becomes unusable. Therefore, validated procedures [14,15] were used in the numerical computation customized to cycle.

The main parameters (temperatures of compressed air and exhaust gases, excess air at Combustion Chamber, α_{CC}) and indicators were computed: AP efficiencies (ε), η_{el} , W_{sp} , depending on: **1)** Π ($\Pi \in [2 \text{ to } 5]$, in geometric progression with $10^{1/50}$ ratio); **2)** θ ($\theta \in [3.9 \text{ to } 4.9]$, corresponding to $T_{\max} \in [1095 \text{ to } 1441] \text{ K}$, or $t_{\max} \in [822 \text{ to } 1168]^\circ\text{C}$); **3)** different fuels compositions. As reference fuel we used methane “**M**”, while the other 4 fuel gas properties (**A**, **B**, **C**, and **D**) are presented in previous work of the authors [9], and the range of syngas components and HHV is presented above.

The following assumptions were used:

- ◆ ISO parameters were used for the compressor air inlet ($T_{\text{atm}} = T_{\text{ref}} = 288.15 \text{ K}$, relative humidity $\varphi = 60\%$, and $p_{\text{atm}} = 101.324 \text{ kPa}$).
- ◆ Fuel gas pressures at FC inlet are considered equals to the atmospheric one, while the temperatures at FC inlet are different: $T_{8 \text{ methane}} = 283.15 \text{ K}$, and $T_{8 \text{ BS}} = 304.55 \text{ K}$. $T_{8 \text{ BS}}$ is the temperature of the syngas after cooling and water vapors removal (the condensing temperature of the water vapors at 4.6 kPa).
- ◆ The AF, CC, AP, and WH pressure losses are between 1.5 to 3 % from the specific inlet pressures.
- ◆ The energy losses in CC, AP, and WH were computed based on $\eta_{CC} = \eta_{AP} = \eta_{WH} = 0.99$.
- ◆ The heat losses in the compressors and turbines were neglected. The isentropic efficiencies of this equipment are: 0.81 for air compressor, 0.77 for fuel gas compressors, respectively 0.82 for turbine.
- ◆ The pressure ratios of fuel gas compressors are superior to the air compressor ratios, according to equipment manufacturer specifications.
- ◆ Electric gross efficiency, $\eta_{el \text{ g}}$, was calculated based on the AC power at inverter terminals. It includes the losses in rotating machines, Rectifier, and Inverter. We considered that $\eta_{\text{mec K\&T}} \eta_{\text{gen}} \eta_{\text{rectifier}} \eta_{\text{inverter}} = 0.915$.
- ◆ The electricity net efficiency is calculated taking into consideration the fuel gas compressor power consumption that is 1.12 times higher than the real work transmitted to the fuel gas into compression process.

- ♦ The mass flow rates were computed based on combustion and energy balance equations of CC.
- ♦ The compositions of fuel, air, and flue gases are mixtures of real gases (CO, H₂, CO₂, O₂, N₂, and others) considering their quota and thermodynamic properties.

Additionally we introduced a condition for AP, by fixing its exergetic efficiency:

$$\eta_{\text{ex AP}} = \{m_{\text{flue gas}} [h_5 - h_6 - T_{\text{ref}} (s_5 - s_6)]\} / \{m_{\text{air}} [h_3 - h_2 - T_{\text{ref}} (s_3 - s_2)]\} = 0.9 \quad (3)$$

m – mass flow rate [kg/s]; h – specific enthalpy [kJ/kg]; s – specific entropy [kJ/kg/K] in the points related to the schema.

The exergy degradation into AP, given mainly by differences of temperature has a limiting effect on the Number of Transfer Units (NTU). The ratio is chosen assuming that AP is designed according to the turbo-machineries, limiting the AP size and cost.

3. Results and their analyses

By using the computations results we built diagrams of parameter's variations and completed analyses on their evolutions.

Compressed fuel temperatures, t_9 and the compressed air ones, t_2 , depends mainly on Π , increasing with this ones. Assuming that compressors have the same inlet's temperatures ($t_1 = t_8$), pressures ($p_1 = p_8$), compression ratios ($p_2 = p_9$), and isentropic efficiencies, the outlet K and FC temperatures should be different due to the properties of gases ($t_2 \geq t_{9 \text{ BS}} > t_{9 \text{ Methane}}$).

For the same Π and θ , the flue gas temperatures at the gas turbine output (t_5) depend only on the molar participation of the components in the flue gases. Our results show that there are no significant differences between the studied cases. The variations of this parameter with Π and θ are: t_5 decreases with Π increase, at constant θ , and diminish with θ , at constant Π .

As previous mentioned, the peculiarity of Micro GT design is given by the AP. It recover a quota of heat from flue gas to the air, increasing the air temperature at CC inlet (reducing the fuel consumption) and decreasing the heat losses associated to flue gas flow. Consequently, all other parameters in the schema depend on AP performances.

The AP energy balance equation is:

$$\eta_{\text{en AP}} = [m_{\text{fg}} (h_5 - h_6)] / [m_{\text{air}} (h_3 - h_2)] = [m_{\text{fg}} c_{p \text{ av fg}} (t_5 - t_6)] / [m_{\text{air}} c_{p \text{ av air}} (t_3 - t_2)] \quad (4.1)$$

c_p – specific heat of working gases at constant pressure kJ/kg/K.

It results the following equation between the temperature differences:

$$(t_3 - t_2) = (t_5 - t_6) \eta_{\text{en AP}} (m_{\text{fg}} / m_{\text{air}}) (c_{p \text{ av fg}} / c_{p \text{ av air}}) \quad (4.2)$$

$\eta_{\text{en AP}}$ – energy efficiency of the air preheater.

Knowing t_2 and t_5 , the temperatures t_3 and t_6 could be theoretically computed from a system with two equations the imposed exergetic efficiency (3), and the AP energy balance equation (4.2). In reality there is a strong dependence between these parameters, processes evolutions and substance properties. Consequently all the parameters must be established by iterative computation. The iterations stop when the imposed exergetic efficiency of the AP is achieved. Thus we also computed the ratio $m_{\text{fuel}} / m_{\text{flue gas}}$ depending on Π , θ , and fuel type. Note that the lean syngas fuel mass flow rate is considerably bigger than the methane one. When the HHV decreases, the fuel mass flow rate increases. Therefore, per 1 kg air, the flue gas flow rate increase substantially from methane to syngas A, and moderate from syngas A (high quality) to D (lean). The wide range of the results conducted to the use of an average bio-syngas for representing in Figure 2 the variation of the ratio $m_{\text{fuel}} / m_{\text{flue gas}}$ for methane and the average bio-syngas (BS).

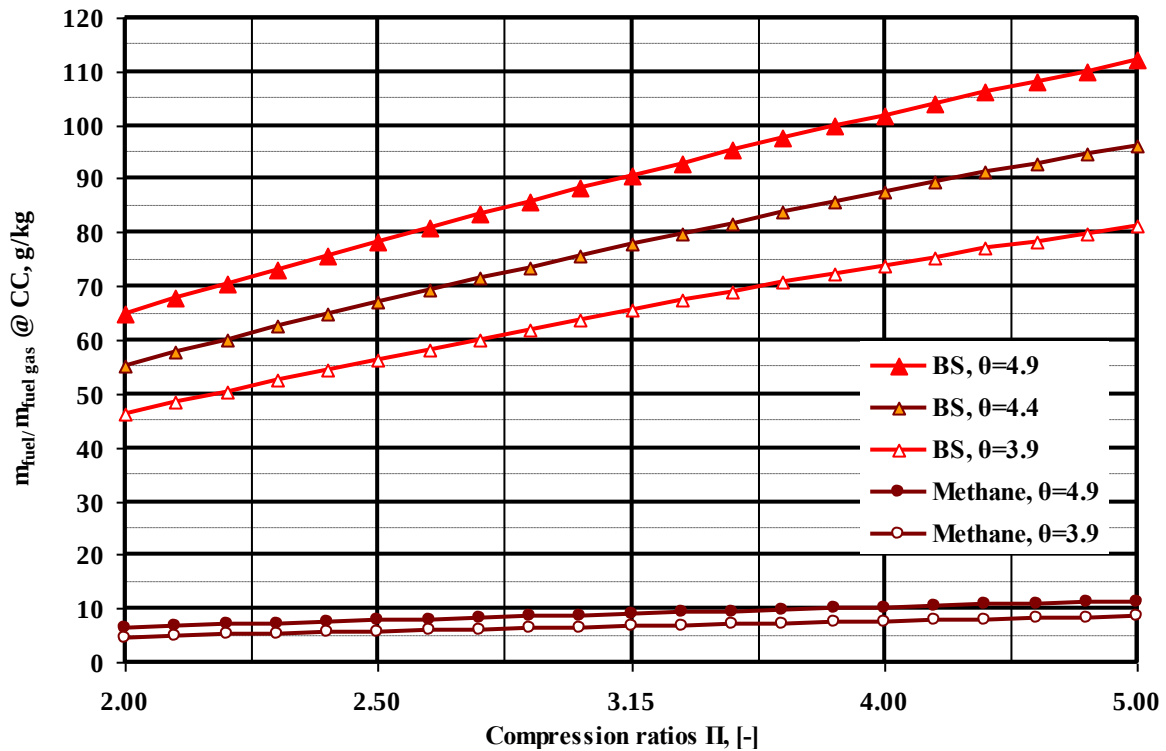


Figure 2. $m_{\text{fuel}}/m_{\text{flue gas}}$ for methane, and the average for all BS $m_{\text{fuel}}/m_{\text{flue gas}}$ values, in g/kg, vs. Π , for $\theta_{\text{ratios}}=3.9$ to 4.9

Rising Π , at constant θ , increase t_2 , which pushup t_6 ($t_6 > t_2$) (Figure 3). For given Π and θ , rising flue gas mass flow rate, decrease the temperature difference on the flue gas flow part and t_6 increase because t_5 has insignificant variation with the fuel type.

As mentioned before, rising Π , at constant θ , decrease t_5 . It pushes down t_3 ($t_5 > t_3$). As a result the temperature differences on both AP flows, ($t_3 - t_2$) and ($t_5 - t_6$), diminish. Consequently t_3 decrease with Π and increase with θ . Because the flue gas mass flow rate per 1 kg air is higher for syngas than for methane, the value of t_3 is also higher for syngas preheating compared to methane (Figure 4).

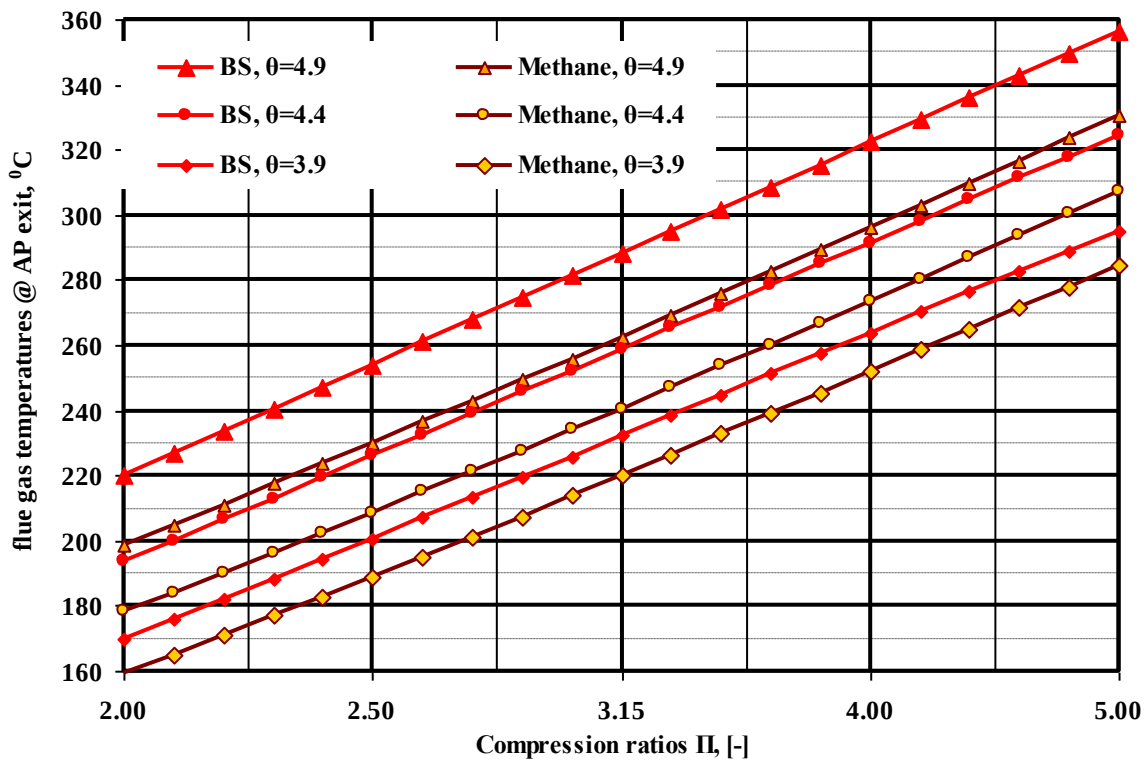


Figure 3. Variation of t_6 for CH_4 and the average t_6 values for all BS, vs. Π , for $\theta \in 3.9 \div 4.9$

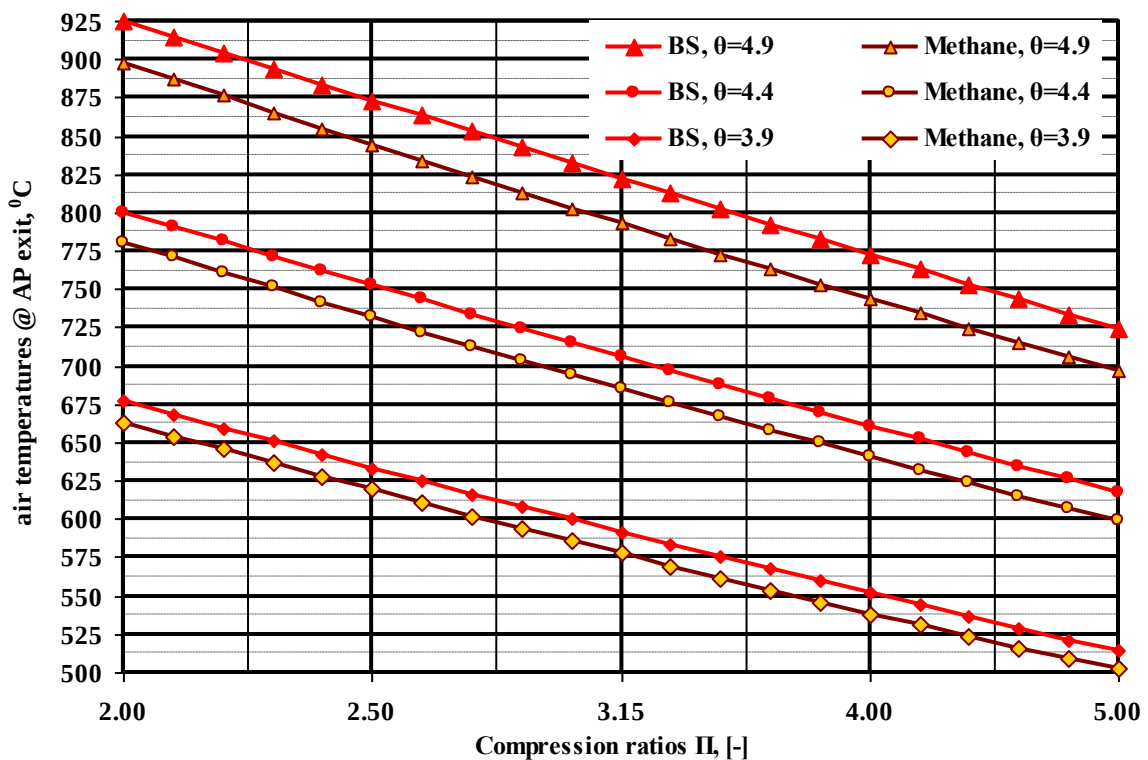


Figure 4. Variation of t_3 for CH_4 and the average t_3 values for all BS, vs. Π , for $\theta \in 3.9 \div 4.9$

The average AP heating effectiveness $\varepsilon_{\text{AP}} = (t_3 - t_2) / (t_5 - t_2)$ diminishes with Π , being higher for lean syngas and minimal for CH_4 (Figure 5). For methane the values are in range with available data [14].

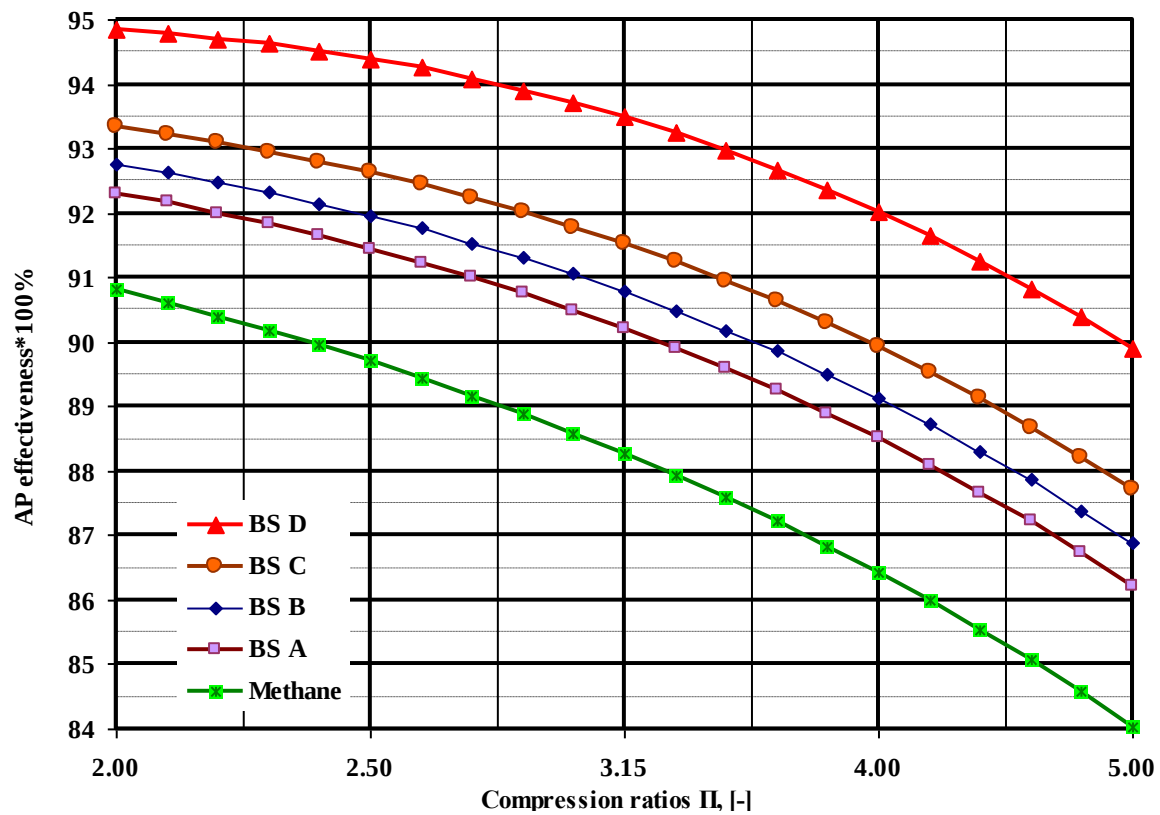


Figure 5. Average AP effectiveness, ε_{AP} , at $\theta_{ratio}=3.9$ to 4.9 , for different fuels, vs. Π

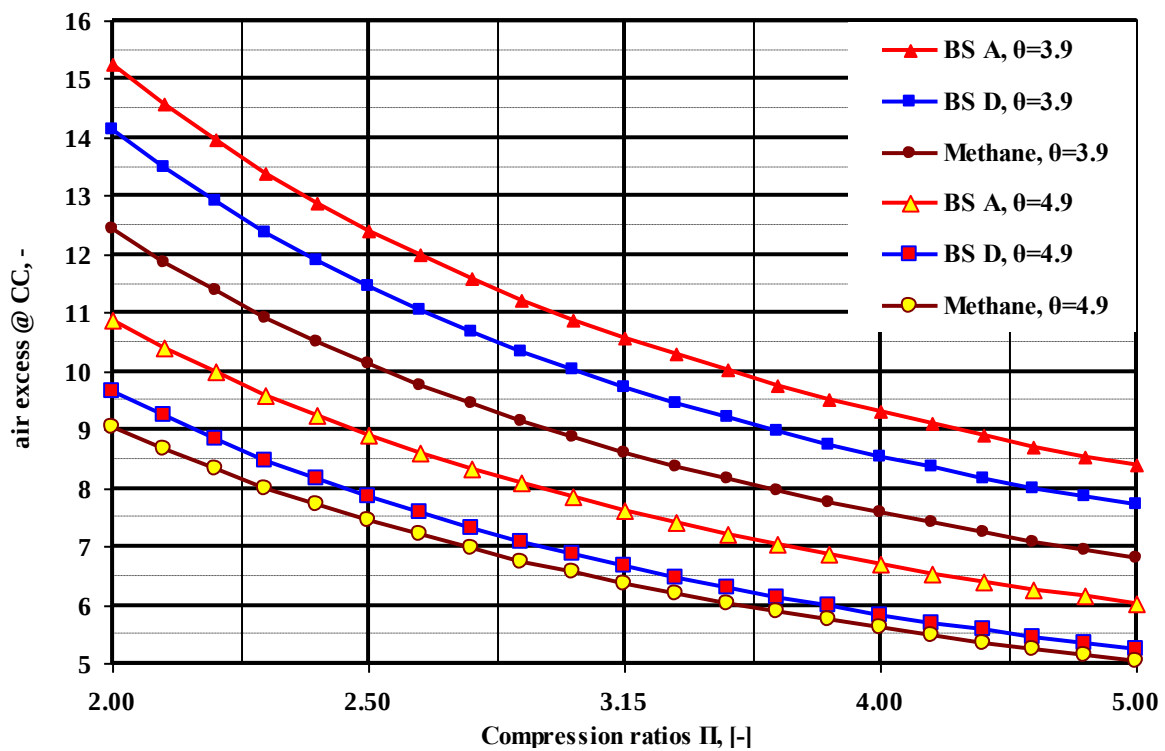


Figure 6. Variation of α_{CC} for CH_4 and different BS vs. Π , for $\theta=3.9$ to 4.9

Figure 6 shows the air excess coefficient in combustion chamber (α_{CC}) variation depending on Π , θ , and fuel type. It reveals the differences between syngases (BS D and BS A) and methane. For the same Π and θ values α increase when the fuel gas heat values are higher. For all analyzed fuels the evolution of air excess is similar to the conventional GT: for a given θ it diminishes with Π growth, while for a given Π it diminishes with θ growth. The differences are given by the numerical values of α_{CC} , higher than for classical GT without AP, because at micro GT the air temperature at CC input is higher.

Figure 7 shows the variation of ratio between $P_{th AP}$ (the thermal flow rate recovered by AP) and $P_{fuel@CC}$ (the heat input by burning fuel into CC) for an average syngas and methane. For all θ values these ratios decrease when Π increase. The values obtained when burning methane are higher compared to syngas use. If compression ratio increases, the difference between the temperatures at turbine and compressor exhaust decreases and the possibilities for internal heat recovery decrease.

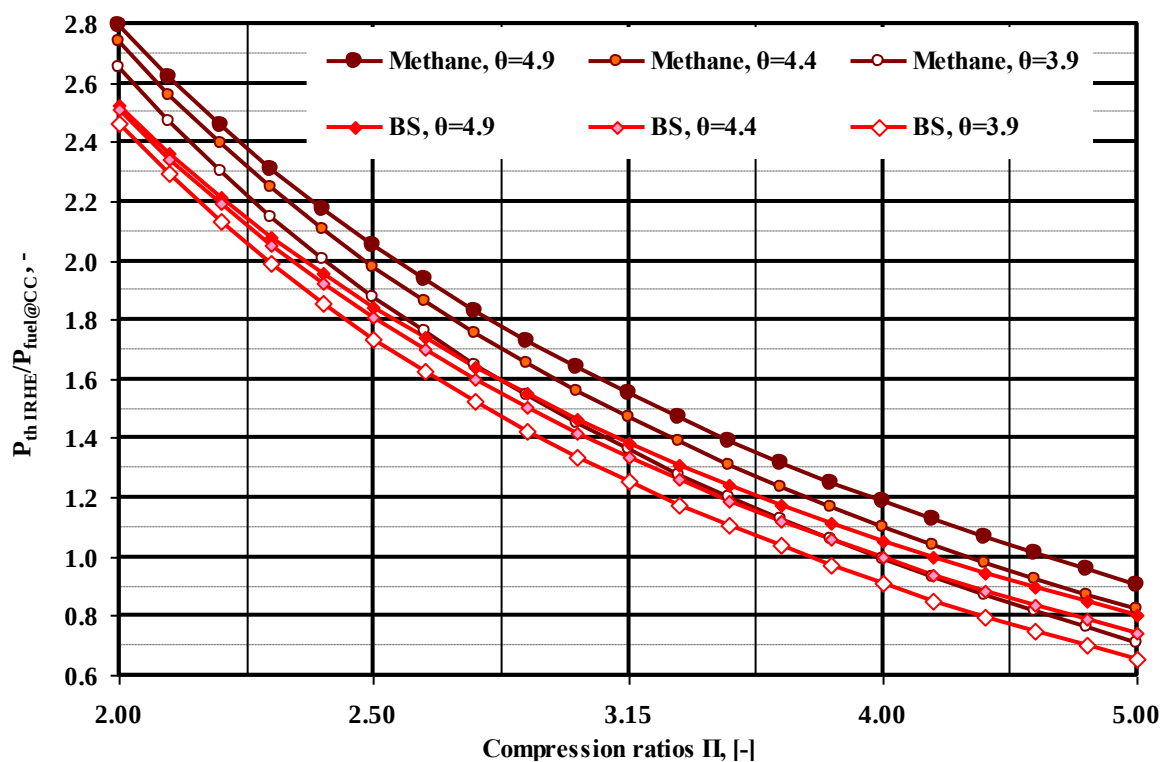


Figure 7. Variation of $P_{th AP}/P_{fuel@CC}$ for methane and the average values of $P_{AP}/P_{fuel@CC}$ for BS, vs. Π , at $\theta=3.9$ to 4.9

The generated power per mass gas flow, P_g/M_{fg} , increases with θ . Nevertheless this parameter does not reach a maximum value for Π range. P_g/M_{fg} for syngas is higher than for methane, but the fuel switch has a low effect.

For the end user it is important the effect of fuel change on the net power flow, respectively the net electrical efficiency. To establish the equipment performances were computed and represented in following diagrams the evolutions of: gross electric efficiency $\eta_{el gross}$ (Figure 8); fuel compressor relative power consumption $P_{el fuel compressor}/P_{el gross}$ (Figure 9); net electrical energy per flue gas unit $W_{el net}/m_{flue gas}$ (Figure 10) and net electric efficiency $\eta_{el net}$ (Figure 11).

The electric gross efficiency, $\eta_{el gross}$, has its maximum value in the Π range, for the used θ ratios (Figure 8). We noticed that fuel calorific value has a low effect on this parameter. When θ varies in the considered range the electric gross efficiency increases with about 5 percentage points.

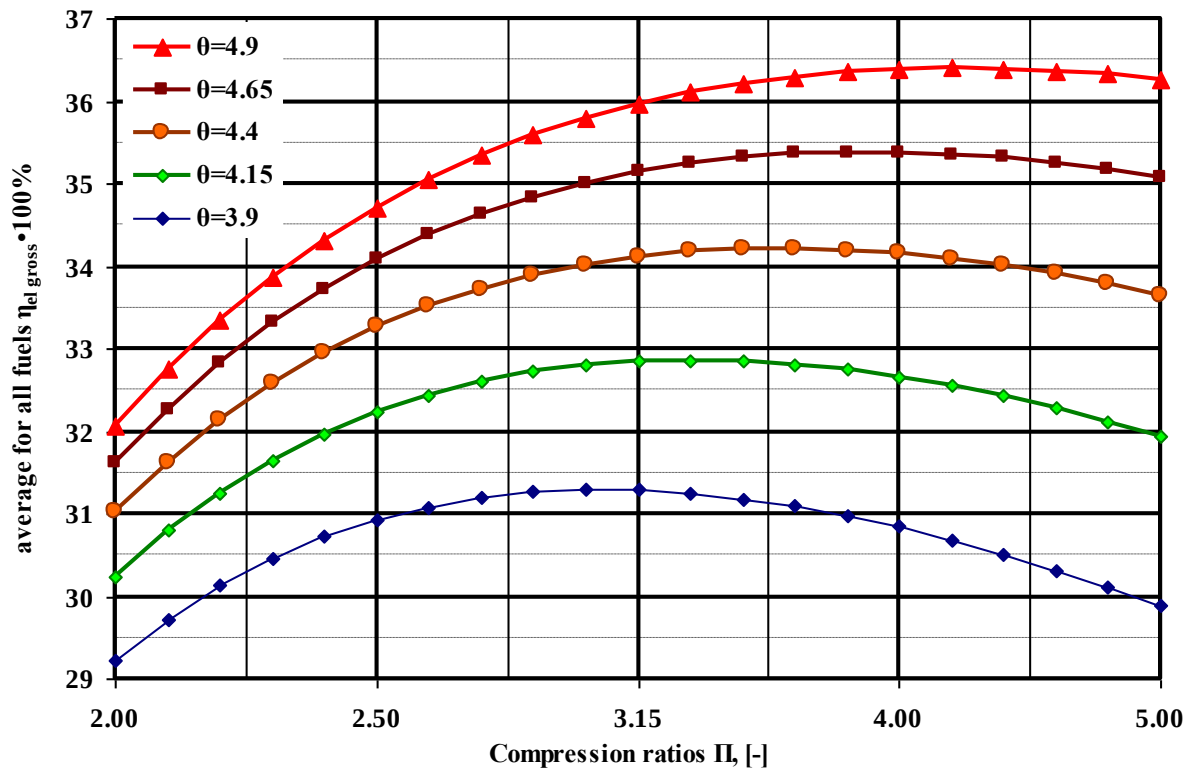


Figure 8. Average for all fuels $\eta_{el\ gross}$ vs. Π , for $\theta_{ratios}=3.9$ to 4.9

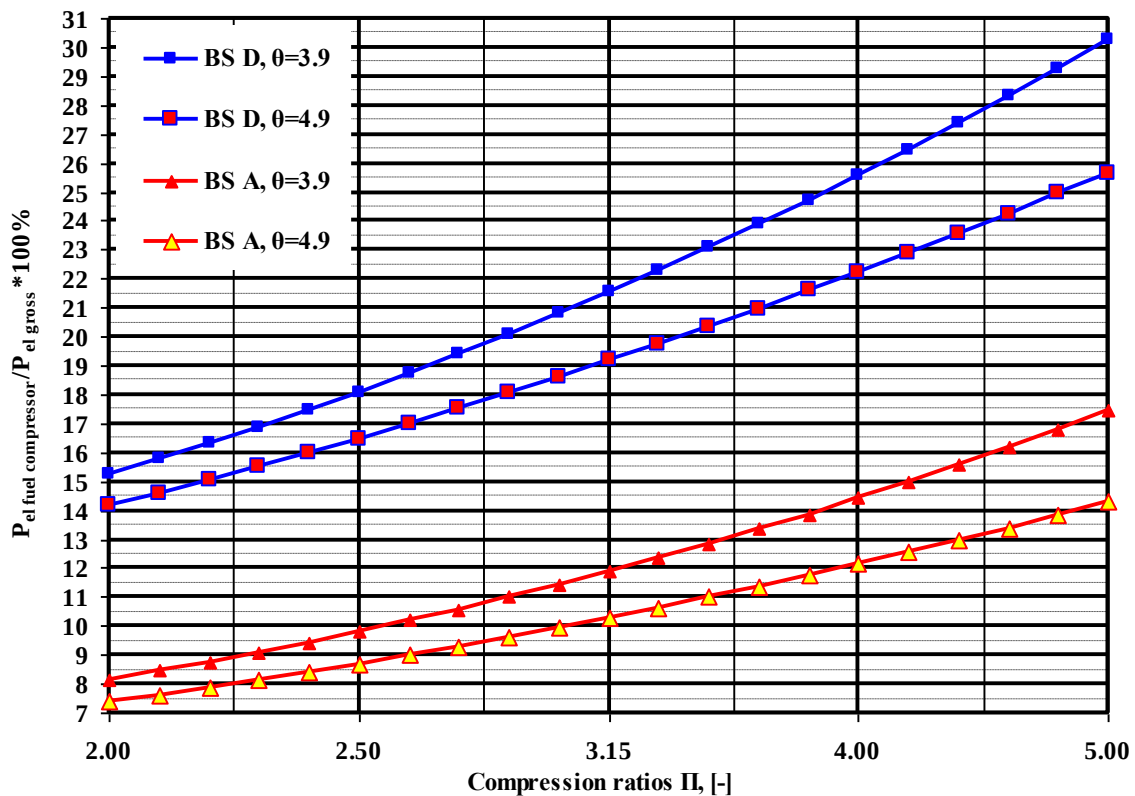


Figure 9. Variation of $P_{el\ fuel\ compressor} / P_{el\ gross}$ vs. Π , for BS "A" and "D", at $\theta = 3.9$ to 4.9

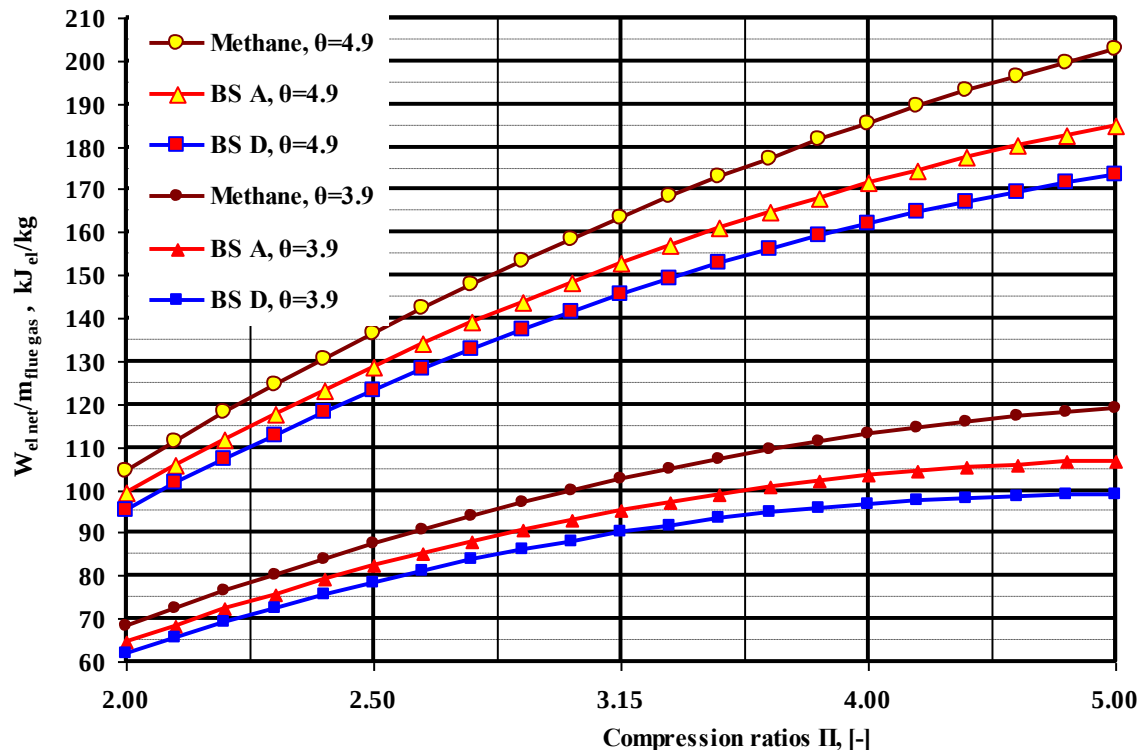


Figure 10. Variation of $W_{el\ net}/m_{flue\ gas}$ ratio, vs. Π , for methane, BS "A", and BS "D", at $\theta=3.9$ to 4.9

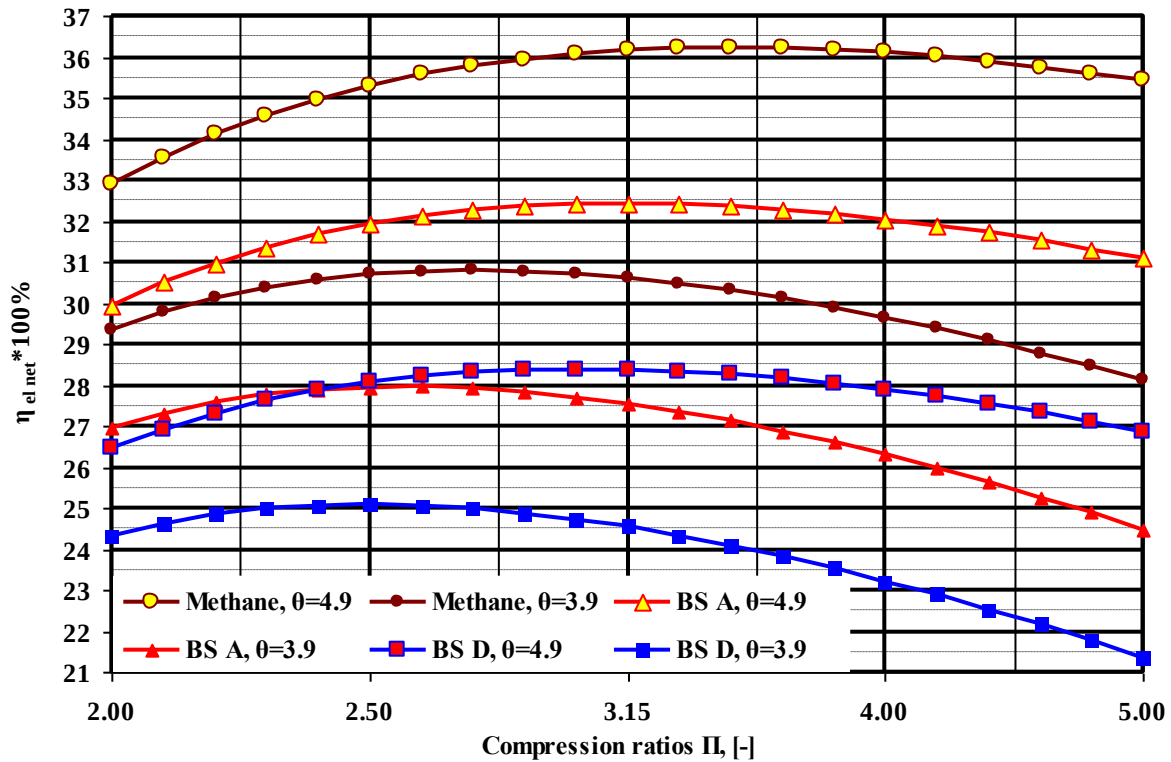


Figure 11. $\eta_{el\ net}$ vs. Π , for methane, BS "A", and BS "D" at different $\theta =3.9$ to 4.9

When using standard gas fuel (methane) the values of fuel compressor power consumption relative to gross electric power ($P_{el\ fuel\ compressor}/P_{el\ gross}$) are in the range of 1.4 (for $\theta=4.9$; $\Pi=2$) to 3.45 (for

$\theta=3.9$; $\Pi=5$). The power consumption varies directly with Π , and increases when fuel quality decreases. Figure 9 presents this parameter variation for syngas use highlighting the major influence of syngas quality on fuel compressor power consumption. For the BS A the gas compressor consumes starting from 7.4% to 8% (for $\Pi=2$) up to 14.2% to 17.4% (for $\Pi=5$) from the generated power. For the BS D these quota increase at 14% to 15.2% (for $\Pi=2$) up to 25.7% to 30.3% (for $\Pi=5$). That's why changing the fuel leads to a contrary effect on $W_{el\ net}/m_{flue\ gas}$ compared to $W_{el\ gross}/m_{flue\ gas}$. The $W_{el\ net}/m_{flue\ gas}$ is bigger when using CH_4 than syngas (Figure 10).

The higher flue gas compressor consumption when using syngas than for methane use has two effects (see Figure 11): 1) for the same heat flow rate input the net power and consequently the electrical net efficiencies decreases; 2) for the same θ , the values of Π that maximize $\eta_{el\ net}$ are smaller than the values of Π that maximize $\eta_{el\ gross}$.

4. Conclusions

The micro gas turbines can be modified to burn syngas obtained from air gasification of biomass and waste keeping the main equipment (gas turbine and air compressor). The paper focused on the thermodynamic aspects of the process. There are, of course, necessities some changes for adapting the Micro GT to syngas use, to the burner of the combustion chamber and the fuel compressor. The results of the study revealed an important influence of fuels composition on micro GT performances. At lower HHV of gas fuel and constant heat flow rate in the combustion chamber, the gas flow rate increases to keep the temperature of flue gases constant. For the analyzed case ($\Pi=4$ and $\theta=4.5$) the thermal power from fuel combustion increases in the range of 1.1% - 3.4 %. Even if the gross electric power increases with 0.9 to 2.9 %, the net electric output will decrease with about 0.15% - 0.35 % compared to methane use. The net energy efficiency will decrease with 0.9 to 3.75 % compared to the reference value (directory specifications of the unit).

The paper results can be used in design engineering for waste to derived fuels conversion chains using gasification conversion and power generation using micro gas turbines.

Acknowledgements

This work was supported by a grant of the Romanian National Authority for Scientific Research, CNDI – UEFISCDI, project number PN-II-PT-PCCA-2011-3.2- 1687 (62/2012).

References

- [1] Mărculescu C 2012 Comparative analysis on waste to energy conversion chains using thermal-chemical processes *Energy Procedia* **18** 604–11
- [2] Gupta K K, Rehman A and Sarviya R M 2010 Bio-fuels for the gas turbine: A review *Renew. Sustain. Energy Rev.* **14** 2946-55
- [3] Cameretti M C and Tuccillo R 2015 Combustion features of a bio-fuelled micro-gas turbine *Appl. Therm. Eng.* **89** 280-90
- [4] Arroyo J, Moreno F, Munoz M and Monne C 2013 Efficiency and emissions of a spark ignition engine fuelled with synthetic gases obtained from catalytic decomposition of biogas *Int. J. Hydrogen Energy* **38** 3784–92
- [5] Arroyo J, Moreno F, Munoz M, Monne C, *et al.* 2014 Combustion behavior of a spark ignition engine fueled with synthetic gases derived from biogas *Fuel* **117** 50-58
- [6] Delattin F, Lorenzo G D, Rizzo S, Bram S and Ruyck De J 2010 Combustion of syngas in a pressurized microturbine-like combustor: Experimental results *Appl. Eng.* **87** 1441-52
- [7] Barsali S, Marco De A, Giglioli R, Ludovici G and Possenti A 2015 Dynamic modelling of biomass power plant using micro gas turbine *Renew. Energy* **80** 806-18
- [8] Cavarzere A, Morini M, Pinelli M, Spina P R, Vaccari A and Venturini M 2014 Experimental Analysis of a Micro Gas Turbine Fuelled with Vegetable Oils from Energy Crops *Energy Procedia* **45** 91-100
- [9] Mărculescu C, Cenuşă V and Alexe F 2016 Analysis of biomass and waste gasification lean

- syngases combustion for power generation using spark ignition engines *Waste Manag.* **47** 133-40
- [10] Cascarosa E and Gasco L 2012 Meat and bone meal and coal co-gasification: Environmental advantages *Resources, Conservation and Recycling* **59** 32-37
- [11] Loha C, Chatterjee P K and Chattopadhyay H 2011 Performance of fluidized bed steam gasification of biomass – Modeling and experiment *Energy Convers. Manage.* **52** 1583–88
- [12] Neilson C 1998 LM 2500 gas turbine modifications for biomass fuel operation *Biomass and Bioenergy* **15** 269-73
- [13] Romier A 2004 Small gas turbine technology *Appl. Therm. Eng.* **24** 1709-23
- [14] Cenușă V and Alexe F 2007 Comparative Analysis on Performances of Micro Gas Turbines Burning Biogas vs. Natural Gas *WSEAS Trans. Environ. Dev.* **3** 72-80
- [15] Cenușă V, Benelmir R and Merabtine A 2012 Reference Calculations for Gas Turbine Installations *International Journal of Energy, Environment and Economics* **20** 247-65