Comparative Analysis on Performances of Micro Gas Turbines Burning Biogas vs. Natural Gas

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Abstract: - The Micro Gas Turbines could burn a wide scale of clean liquids and gaseous fuel, but their reference data are usually given for burning natural gas. Fuel's switch changes their performances. The paper analyzes the case of Micro GT designed for natural gas and adapted for burning biogas produced in anaerobic digesters. This one has Low Heat Values and air requests for stoechiometric burning, lower than methane. For modeling the processes from Micro GT components and cycle, we employ numerical computation, using procedures achieve and validate in our department. Our purpose is analyzing the results of gaseous fuel switch on Micro GT performances. We put into evidence that: **1)** burning biogas doesn't affect significantly the performances of Micro GT, but **2)** that choice request necessary actions for adapting the burner and the gas fuel compressor. The obtained results are in concordance with recently references.

Key-Words: - Micro-Turbines, Thermodynamic Analysis, Biogas, Computation and Modeling.

1 Introduction

Using Gas Turbines (GT) in distributed power-only generation systems, for electric outputs less than 500 kW, became possible due to new thermodynamic, constructive, and technological solutions. Micro Gas Turbines, using supersonic high-speed radial flow turbo-machineries (centrifugal compressors and centripetal turbines) are a result of improvement in small gas turbines and turbochargers technology. The height rotation speeds, variable on load, impose special solutions for obtaining AC power at network frequency, 50 or 60 Hz.

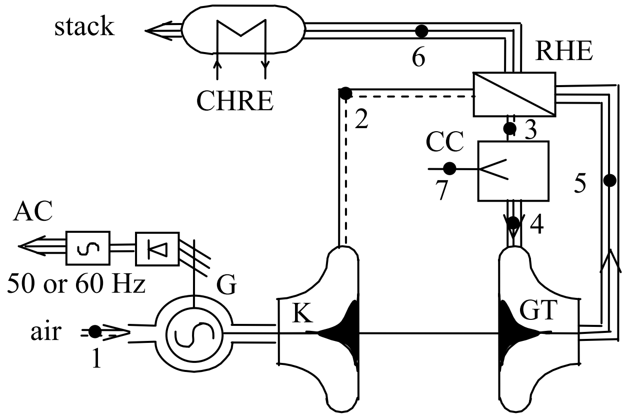

Fig.1 Single shaft micro gas turbine

Single shaft Micro GT (Fig.1), having rotation speeds over 60,000 rpm, variable with load, drives directly synchronous generators witch produce audio AC (over 1,000 Hz). This one is transformed into DC and converted to network frequency.

For double shaft Micro GT (Fig.2), the expansion is divided on two turbines: **a)** the expansion one, driving the compressor at high and variable speed, and **b)** the power turbine, coupled by a special gear with a network frequency synchronous generator.

Nearly all Micro GT employs a Regenerative Heat Exchanger (RHE), which preheats the compressed air with the heat from turbine's exhaust gases. The remaining heat of exhaust gases could be recuperated in a Cogeneration Heat Recovery Exchanger (CHRE). The RHE reduce the flue gas temperature at CHRE input and limit the recuperation at low temperature heat consumers.

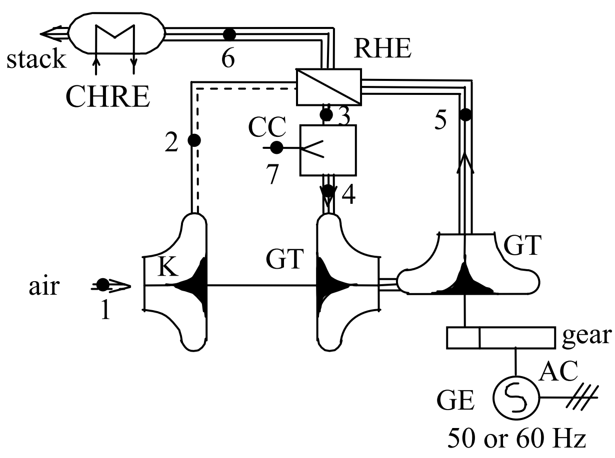

Fig.2 Double shaft micro gas turbine

For distributed generation in locations next to biogas digesters it is possible to use the recuperated heat to maintain the fermentation temperature. In this case the main result of the Micro GT process is the electricity. That's why this paper will analyze only the conversion of heat, developed by fuel burning, into work and power.

Like the medium and high power GT, Micro GT could burn a wide scale of gaseous fuels (natural gas, residual gases from industrial processes, biogas and others), but their base design is for natural gas. The biogas from organic wastes anaerobic controlled fermentation contents only 56 to 80 % combustible gases [5]. Burning it in GT will modify the performances comparing to the reference ones and may need redesigning some parts [6].

2 Problem Formulation

The main no dimensional parameters employed for gas turbines cycle thermodynamically analyses, are: the compression ratio ($\varepsilon=p_{max}/p_{min}$), and the ratio of extreme absolutes temperatures ($\theta=T_{max}/T_{min}$). It is generally known and accepted [7, 8] that for the classic Brayton cycle, without RHE:

- For a given compression ratio, when θ augment, the electrical efficiency, η_{el}, and the specific work per flue gas mass unit, W_{sp}, in kJ/kg, are growing too. It creates the interest for rising T_{max}, without using expensive materials or loosing availability. For single shaft Micro GT with steel rotor, the maximal temperatures are about 1250 K (980°C). For double shaft ones, with HP turbine ceramic rotors, it could be raised up to 1425 K (1150°C).
- For a given θ value, both, electrical efficiency and specific work, evolves on ε following curves with maximum values respecting the relation:

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

Increasing θ augment both $\varepsilon(W_{sp\,max})$ and $\varepsilon(\eta_{el\,max})$. Generally the GT producers choose, for the Brayton cycle without RHE, ε values in the interval $\varepsilon(W_{sp\,max})$ to $\varepsilon(\eta_{el\,max})$. For the Micro GT, due to the RHE effect, the relation between $\varepsilon(W_{sp\,max})$ and $\varepsilon(\eta_{el\,max})$ became [2, 4]:

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

That allows obtaining enough good electrical efficiency for lower compression ratio. On the other hand, for ε values witch can be reach by Micro GT, it can't obtain the maximum point of W_{sp}.

About electrical efficiency, it is important to note that Micro TG on the market incorporate into the delivered package the fuel gas compressor. That's why into directories the reference power and the efficiency are the net ones.

In this paper we are doing technical analyses in two steps. The big number of variables and the nonlinear mathematical relation made almost impossible finding analytical solutions. That lead as to choose the numerical way, for different set of data, using procedures achieves and validates in our department [1 to 3], and customized to cycle.

In the first step repetitive calculations are done for establishing the main parameters (temperatures, air excess at Combustion Chamber, α_{CC}, temperature differences) and indicators ($RHE_{efficiency}$, η_{el}, W_{sp}), depending on: **1)** $\boldsymbol{\varepsilon}$ ($\varepsilon\in[2\cdot\sqrt{2}$ to $4\cdot\sqrt{2}]$, in geometric progression with relative amount equal to $\sqrt[16]{2}$); **2)** $\boldsymbol{\theta}$ ($\theta\in[4$ to $5]$, witch correspond to $T_{max}\in[1153$ to $1441]$ K, or $t_{max}\in[880$ to $1170]$ °C) and **3)** gaseous fuels elementary compositions.

Relating to the third point, in this paper we consider three biogas elementary compositions: **A** - poor biogas, **B** - average biogas, and **C** - rich biogas. The reference fuel is considered the methane "**M**".

Table 1 shows the compositions, LHV, and air requirement for stoechiometric burning.

Table 1. Gaseous fuels data.

Data	Unit	Gaseous fuels			
		Biogas			Methane
		A	B	C	M
H_2	%, mol	1.26	1.53	1.8	0
CH_4	%, mol	54.32	65.96	77.6	100
CO	%, mol	0.14	0.17	0.2	0
H_2S	%, mol	0.28	0.34	0.4	0
N_2	%, mol	3.52	2.56	1.6	0
O_2	%, mol	0.22	0.16	0.1	0
CO_2	%, mol	40.26	29.28	18.3	0
μ_{fuel}	kg/kmol	27.65	24.43	21.21	16.04
LHV	kJ/kg	15,900	21,852	29,610	49,896
	kJ/m^3_N	19,615	23,818	28,021	35,707
m_{air}/m_{fuel}	kg/kg	5.465	7.516	10.191	17.199
V_{air}/V_{fuel}	m^3_N/m^3_N	5.238	6.365	7.493	9.563

The calculated mass LHV of biogas, in kJ/kg, represent 32 to 59.5 %, while the volumetric LHV, in kJ/m^3_N, is 55 to 78.5 % from the CH_4 ones. The humid air requirements (at p_{air}=101,3 kPa, $t_{air}=15^0$C, and φ_{air}=60%) for complete burning a mass unit (kg_{air}/kg_{fuel}) or a volume unit (m^3_N/m^3_N) of biogas, related to the methane one, represents almost the same percentages like the LHV ones.

In the second step, we suppose a Micro GT having average values of ε and θ (ε=4 and θ=4.5) which produces 100 kW AC power flow at generator terminals, on burning methane. We determine its reference flows. Considering that the flue gases rate flow is constant, we recalculate its performance indicators for other gaseous fuels.

In the both previously mentioned analysis steps we are using the following general hypotheses:

- Atmospheric air parameters are the ISO ones.
- The pressure losses in AF, CC, RHE, and CHRE are 3 % from the amount pressure.
- For taking into account the energy losses in CC and RHE we consider $\eta_{CC}=\eta_{RHE}=0.98$.
- We neglected the heat transfer during the compression and the expansion. The isentropic efficiencies of turbo-machineries are: 0.8 for compressor, respectively 0.82 for turbine.
- Electricity generation efficiency, $\eta_{el\ g}$, is determined considering AC power generated flow at inverter terminals (for single shaft), respectively alternator output (for double shaft). It takes into consideration the conversion losses in rotating machines ($\eta_{gen}\cdot\eta_{mec}$. = 0.92·0.95).
- Electricity net efficiency takes into consideration the power consumption of fuel gas compressor.
- The mass flow rates into compressor and turbine resulted from burning equations and mass / heat balance of CC.
- The air, fuel, and flue gases were considered like mixtures of real basis gases (CO_2, O_2, N_2, and others) taking into consideration their participation and thermodynamic properties.

At the above mentioned hypotheses we added a supplementary condition for RHE, by fixing the ratio between the AC power generated flow, P_g, kW, and the RHE conductance, Cd, kW/K. This datum is a dimensional one; its value is proportional with the logarithmical average temperature difference ($\delta t_{log\ av}$) and with the ratio between electrical power (P_g) and regenerative heat flow ($P_{th\ rec}$):

$P_g/Cd=\delta t_{log\ av}*(P_g/P_{HRE})$ (3)

The exergetic lack into RHE characterized by temperature differences, limits the NTU (Number of Transfer Unit) and thermal efficiency. The imposed ratio supposes that RHE design is correlated with the turbo-machineries, and limit the RHE size and cost. We supposed Micro GT with $P_{el}/Cd=16$.

The flow rates use the following nomenclature:

- $M_{f\ g}$ - flue gases mass flow rate, kg/h;
- M_{air} - air mass flow rate, kg/h;
- M_{fuel} - fuel mass flow rate, kg/h;
- $P_{th\ CC}$ - thermal flow obtained by burning, kW;
- $P_i=P_{i\ T}-P_{i\ k}$ - internal net work, as difference between gas turbine and air compressor work;
- P_g - AC power generated flow, kW;
- ΔP_{CC} - Combustion Chamber losses, kW;
- $P_{th\ 1\ c}$ - thermal flow from CC to the cycle, kW;
- ΔP_{RHE} - RHE thermal losses, kW;
- $P_{th\ rec}$ - recycled heat flow (into RHE), kW;
- $P_{th\ flue\ gas}$ - exhaust gas available heat flow, kW;
- ($P_{g\ biogas}$ - $P_{g\ methane}$)/ $P_{g\ methane}$ - relative increase of AC electricity flow comparing to methane;
- $P_{el\ k\ fuel}$ - AC power for fuel compressor, kW;
- P_{net} - net AC electricity flow, kW.

3 First analysis step results

Using the first analysis's step results, and restricting the field of ε values, we build Fig. 3 to 25.

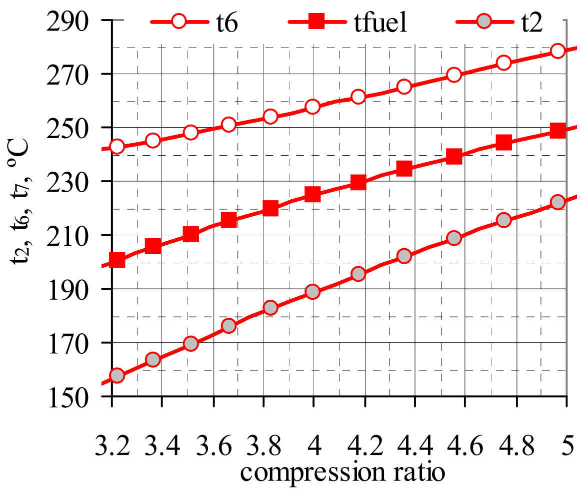

Fig.3 t_2, $t_{fuel}=t_7$, and t_6 (biogas B, θ=4.5), vs. ε

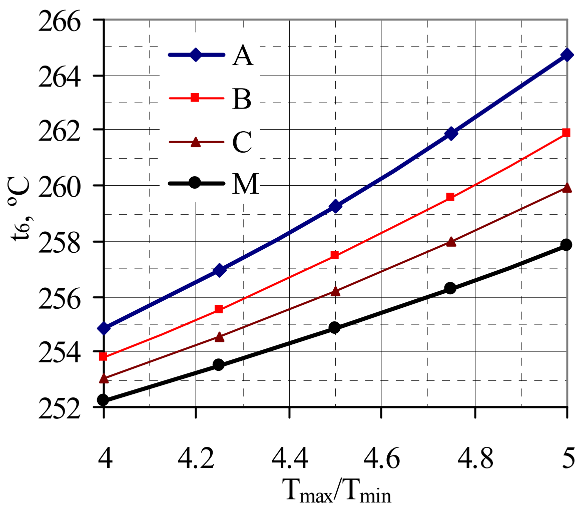

Fig.4 t_6 vs. θ, for ε=4, all fuels

Analysis of parameter's evolution shows that:

- Compressed air and compressed fuel temperatures, t_2 and t_7, are depending mainly on ε, increasing with this one (Fig.3). The flue gas exhaust temperature, t_6, increases with ε and θ

(Fig.4), but is lower than for conventional GT.

- The air excess α_{CC} decreases with ε and θ increase (Fig.5) and LHV reduce (Fig.6), remaining similar with the methane burning one.

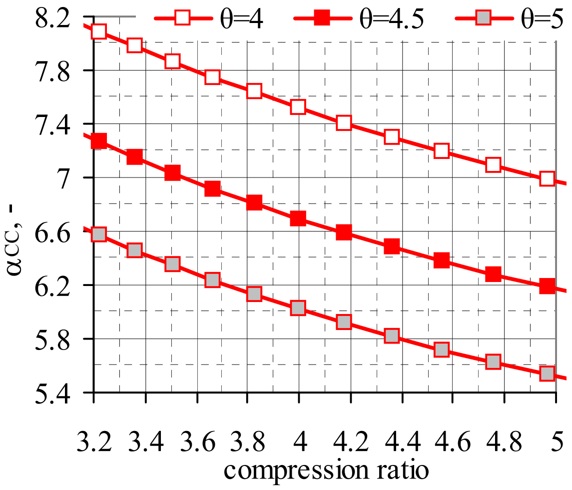

Fig.5 α_{CC} vs. ε, for biogas B, and θ=4 to 5

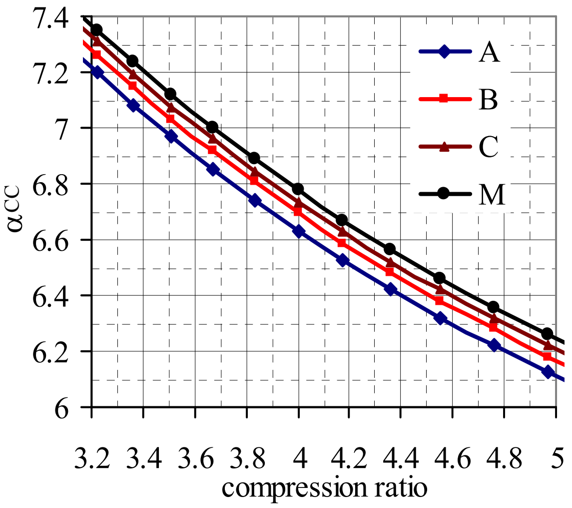

Fig.6 α_{CC} vs. ε, for θ=4.5, all fuels

- Due to the low expansion ratio, the hot flue gas temperature at turbine output, t_5, is upper than at conventional GT. It increases with θ augment and decrease with ε grow (Fig.7). That allows higher air temperature at CC inlet, t_3 increasing with θ augment and decreasing with ε grow, too (Fig.8). The higher t_3 values correspond to the lower fuel LHV (Fig.9).
- The θ values and fuel's LHV have low effect on logarithmic average temperature difference at RHE, $\delta t_{log\ av}$ (Fig.10). The main impact is given by ε, $\delta t_{log\ av}$ decreasing with ε augment.
- The RHE efficiency increases with ε and θ growth (Fig.11). For fuel switch, the RHE $_{efficiency}$ rises, when fuel's LHV diminish (Fig. 12).

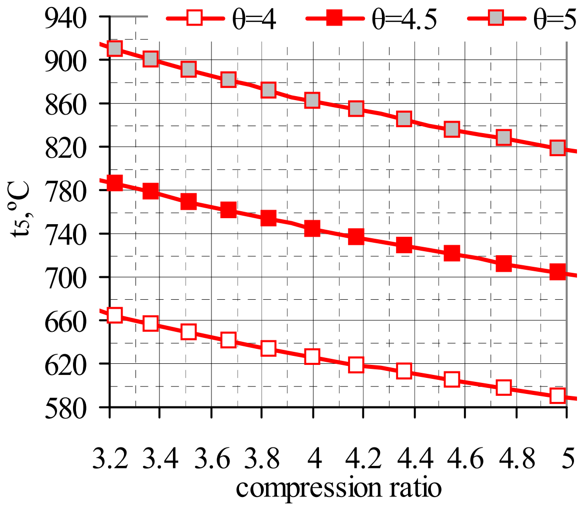

Fig.7 t_5 vs. ε, for biogas B, and θ=4 to 5

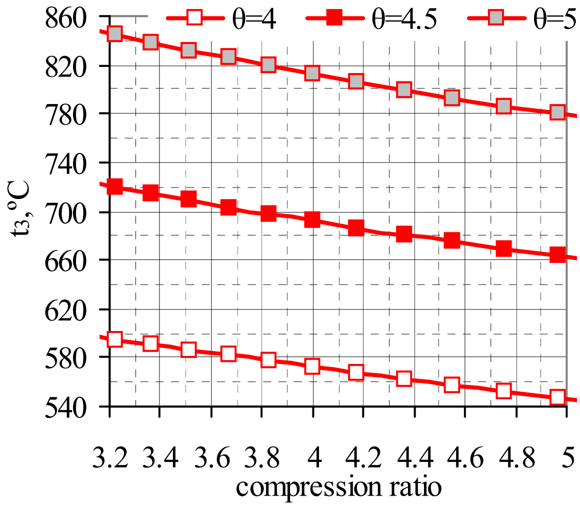

Fig.8 t_3 vs. ε, for biogas B, and θ=4 to 5

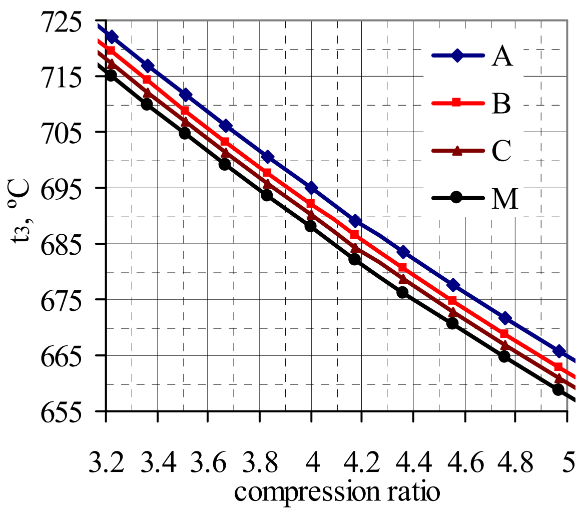

Fig.9 t_3 vs. ε, for θ=4.5, all fuels

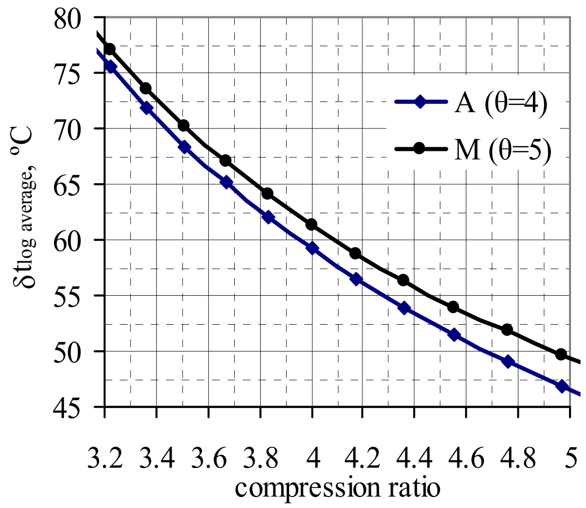

Fig.10 $\delta t_{log\ av}$ vs. ε, for biogas A (θ=4), and M (θ=5)

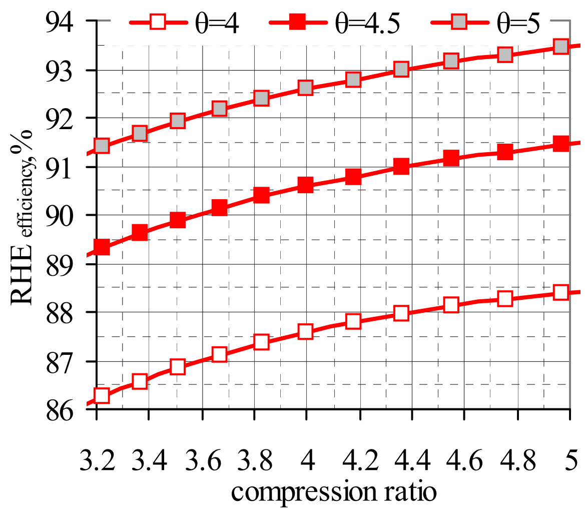

Fig.11 $RHE_{efficiency}$ vs. ε, for biogas B, and θ=4 to 5

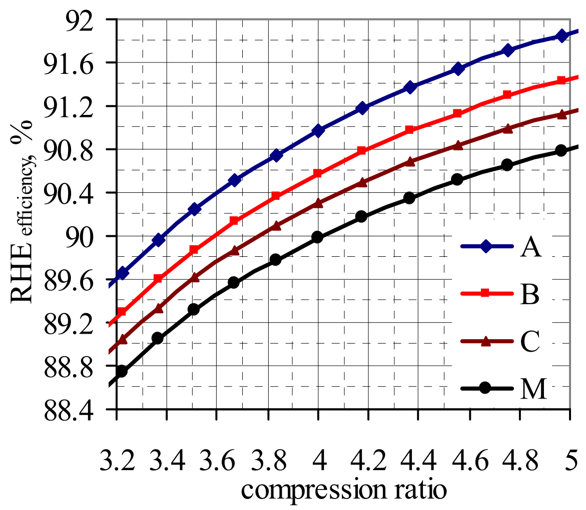

Fig.12 $RHE_{efficiency}$ vs. ε, for θ=4.5, all fuels

- The ratio between the recycling thermal flow rate at RHE, P_{RHE}, and the thermal flow rate given by fuel burning, P_{CC}, for average values of ε and θ (ε≅4, and θ≅4.5), is: $P_{RHE}/P_{CC}>1$ (Fig.13). The fuel switch has a low effect on P_{RHE}/P_{CC} ratio, major effects being given by ε (P_{RHE}/P_{CC} decreases with ε diminish), and θ (P_{RHE}/P_{CC} increases at once with this one).
- The air compressor takes around 50 % from the turbine's work. The quota increases with ε growth and diminishes with fuel LHV shrink, mainly because of reducing the air flow rate.
- The specific generated power per flue gas unit, $P_g/M_{f\,g}$, increases with θ augment, without reaching the maximum in the analyzed ε frame (Fig.14). The fuel switch has a low effect. $P_g/M_{f\,g}$ for biogas is higher than for methane (Fig.15).

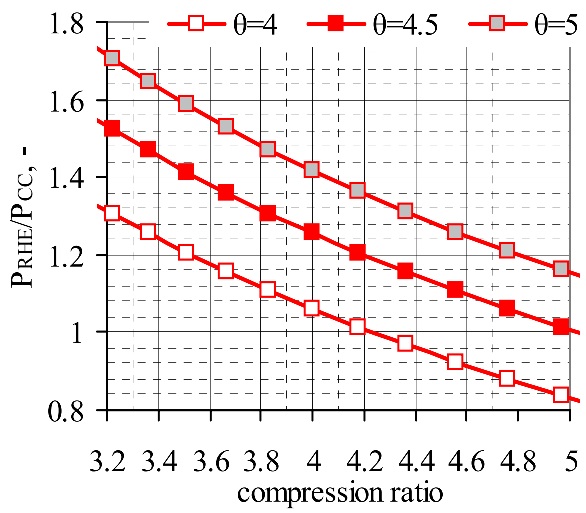

Fig.13 P_{RHE}/P_{CC} vs. ε, for biogas B, and θ=4 to 5

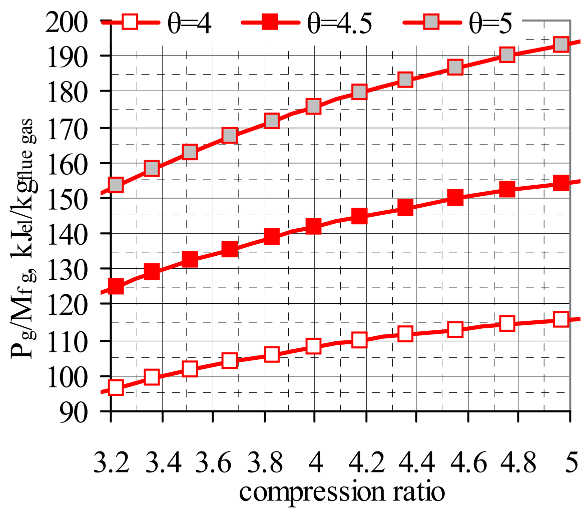

Fig.14 $P_g/M_{f\,g}$ vs. ε for biogas B and θ=4 to 5

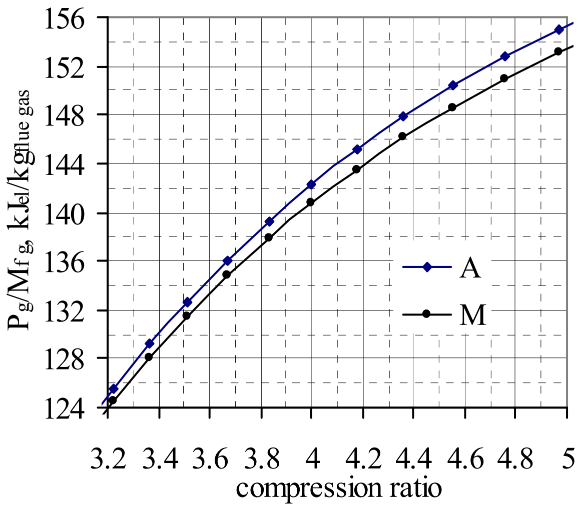

Fig.15 $P_g/M_{f\,g}$ vs. ε, for M and biogas A, θ=4.5

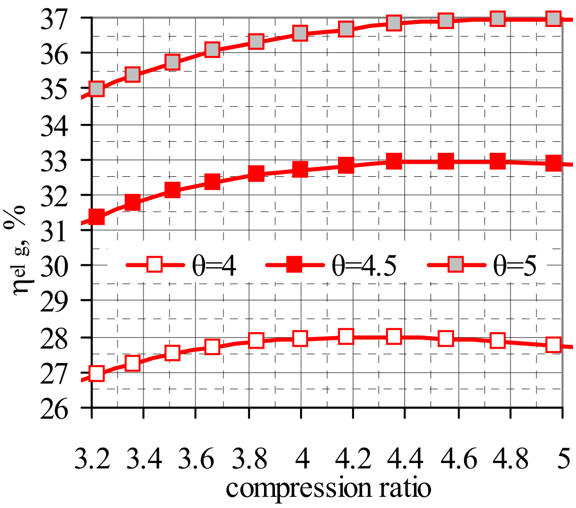

Fig.16 $\eta_{el\,g}$ vs. ε, for biogas B, and θ=4 to 5

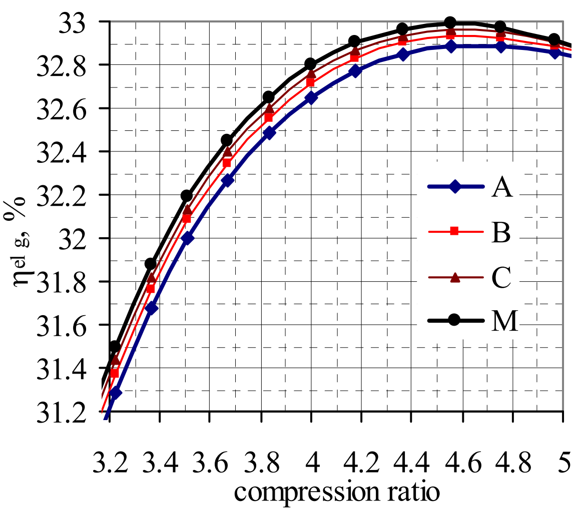

Fig.17 $\eta_{el\,g}$ vs. ε for θ=4.5, all fuels

- The electricity generation efficiency, $\eta_{el\,g}$, reaches the maximum in the analyzed ε frame, for all the considered θ ratios (Fig.16). The effect of fuel's LHV is low (Fig.17).
- The ratio between the fuel flow and air flow rates, M_{fuel}/M_{air}, increases with about 300 % for the A biogas, comparing to methane (Fig.18). For the same fuel and ε, it rises with θ (Fig.19).

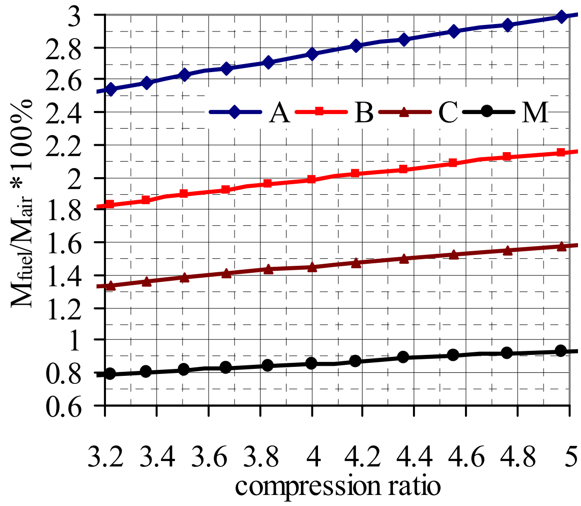

Fig.18 M_{fuel}/M_{air} vs. ε for θ=4.5, all fuels

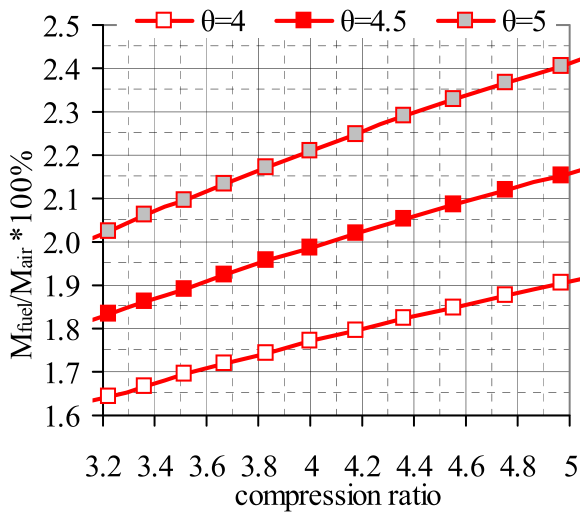

Fig.19 M_{fuel}/M_{air} vs. ε for biogas B, and θ=4 to 5

- The gas compressor takes in 3.4 to 7.8 % from AC power generated flow. The quota is proportional with ε, and increases when fuel LHV shrinks (Fig.20). Because of that the fuel switch has an opposite effect on $P_{net}/M_{f\,g}$, comparing with $P_g/M_{f\,g}$, the specific net power per flue gas unit, $P_{net}/M_{f\,g}$, being bigger for methane than for biogas (Fig.21).

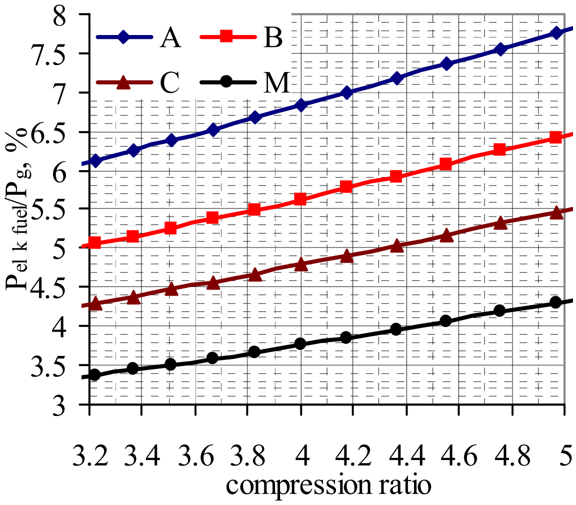

Fig.20 $P_{el\ k\ fuel}/P_g$ vs. ε for θ=4.5, all fuels

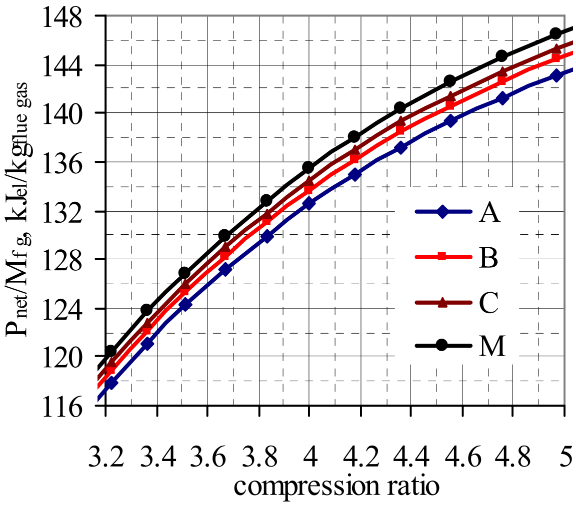

Fig.21 $P_{net}/M_{f\ g}$ vs. ε for θ=4.5, all fuels

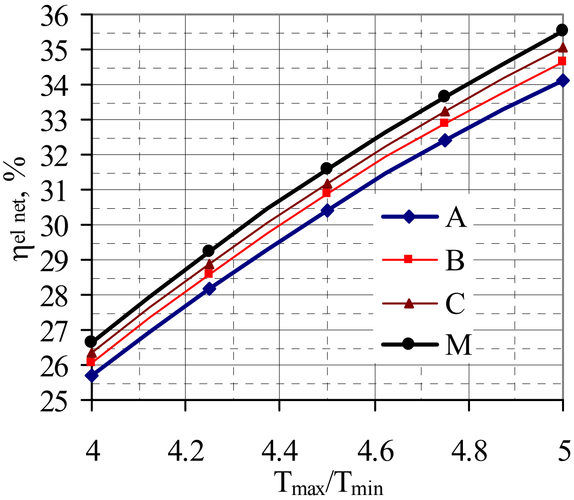

Fig.22 $\eta_{el\ net}$ vs. θ for ε=4, all gases

- Rising θ from 4 to 5 will allow electrical net efficiency, $\eta_{el\ net}$, growth from 26 to over 35 % (Fig.22); the higher values are for methane.
- For θ=4.5, electrical net efficiency have optimal values in the area of ε≅4.5. Decreasing LHV reduces $\eta_{el\ net}$ (Fig.23). For the same fuel, the differences between $\eta_{el\ net}$ and $\eta_{el\ g}$, $\Delta\eta_{net\ vs.\ g}$, because of the fuel gas compressor consumption, are 3.5 to 4.5 % for methane up to 6.5 to 8.5 % for biogas A (Fig.24).

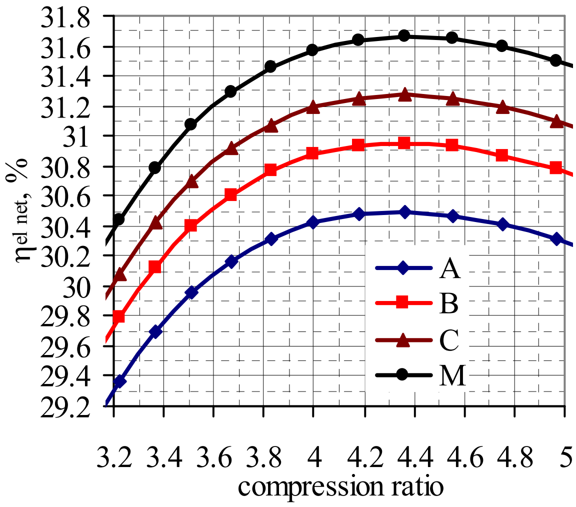

Fig.23 ηel net vs. ε for θ=4.5.

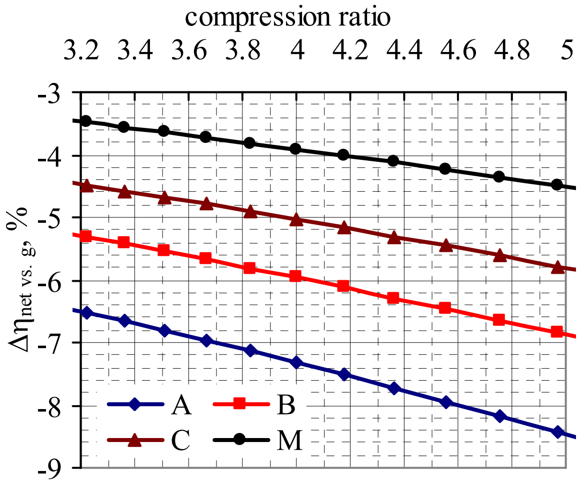

Fig.24 Δη net vs. g, vs. ε for θ=4.5, all gases

- The relative diminish of electrical net efficiency, comparing to methane $(\eta_{net\ meth}-\eta_{net\ biogas})/\eta_{net\ meth}$, is about 1.2 % for rich biogas (C), 2.2 % for average biogas (B), respectively 3.65 % for poor biogas (A) - see Fig. 25.

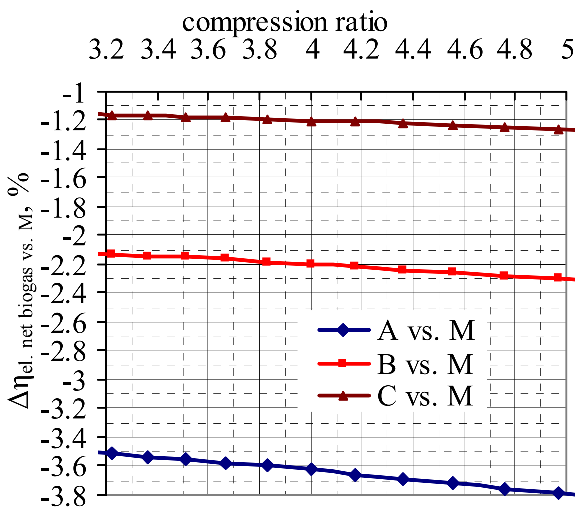

Fig.25 $\Delta\eta_{el.\ net}$ biogas vs. M, for θ=4.5

4 Second analysis step results

The obtained results in the second analysis step are given in Table 2. The mass and energy flows are given in absolute values, and evaluated comparing with methane burning situation data.

Table 2. Mass and energy flows for Micro Turbines

Data	Unit	A	B	C	M
$M_{f\,g}$	kg/h	2,535.46			
M_{air}	kg/h	2,467.39	2,486.05	2,499.06	2,513.89
	%	98.150	98.893	99.410	100
M_{fuel}	kg/h	68.0758	49.4094	36.4024	21.5677
	%	315.638	229.090	168.782	100
$P_{th\ CC}$	kW	315.234	310.982	308.086	304.853
	%	103.405	102.011	101.061	100
P_{int}	kW	117.766	116.408	115.474	114.416
P_g	kW	102.928	101.741	100.924	**100**
ΔP_{CC}	kW	6.305	6.220	6.162	6.097
	%	103.405	102.011	101.061	100
$P_{th\ 1\ c}$	kW	308.929	304.763	301.925	298.756
	%	103.405	102.011	101.061	100
ΔP_{RHE}	kW	7.856	7.809	7.777	7.740
	%	101.498	100.899	100.479	100
$P_{th\ rec}$	kW	384.929	382.654	381.062	379.246
	%	101.498	100.899	100.479	100
$P_{th\ flue\ gas}$	kW	183.307	180.545	178.674	176.599
	%	103.798	102.234	101.175	100
$P_{k\ fuel}$	kW	7.024	5.728	4.837	3.764
	%	186.610	152.182	128.502	100
$P_{el\ net}$	kW	95.904	96.013	96.087	96.236
	%	99.655	99.768	99.845	100

The examination of table's data shows that, when changing the fuel from methane to poor biogas, for the same flue gas mass flow rate, the fuel mass flow rate increases over 3 times. However, the air mass flow rate and the main Micro GT energetic flows don't change significant. The fuel burning heat flow increases with approximately 3.4 %. Simultaneously the internal net work and the AC power generated flow are growing with 2.9 %. The main reason of internal net work increase is rising of fuel flow rate and the decrease of air flow rate at the LHV diminishes. This requires less air compressor work.

The recycled into RHE heat flow rate is, in this case, a little bigger than thermal flow generated by burning fuel, and increases slower then this one. The exhaust gas available heat flow has a slow growth, approximately 1.5 %.

The 1.5 times augment of average fuel molar mass, lead, even the fuel mass flow rate raise more then 3 times, to the increase only 2 times of AC power for fuel compressor. This supplementary consumption is bigger then the gain of AC power generated flow. As a result the net AC electricity flow decreases.

5 Conclusion

The Micro Gas Turbines designed for use of methane could be adapted for bio-gas, using the same main turbo-machineries (air compressor and flue gas turbine). When the fuel gas LHV is decreasing, the fuel gas flow rate increases, in order to maintain the same temperature at CC output. Corresponding, in the typical analyzed situation (ε=4, θ=4.5) the thermal flow rate from CC to the cycle augments with 1.1 to 3.4 %.

Even if the generator (or converter) output will have growths from 0.9 to 2.9 %, the net output will be lower with 0.15 to 0.35 % comparing with the circumstances when using methane as fuel. The electrical net efficiencies will diminish with 0.9 to 3.75 % relatively to the reference one.

The main necessaries changes for adapting to bio-gas Micro Gas Turbines designed for methane burning are related to the: **a)** CC burners, and **b)** gas fuel compressors.

These ones should be redesigned for mass flow rates up to 3.15 times bigger and volume flow rates up to 1.9 times bigger than the methane ones, depending on gaseous fuel composition and LHV.

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