Predicting the Performances of Biogas Fuelled Distributed Power-Only Generation Micro Gas Turbines, with Internal Heat Recovery

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Abstract: - The Micro Gas Turbines could burn a wide scale of clean liquids and gaseous fuel, but their reference performances are usually given for burning natural gas. Fuel's switch influences their performances. The paper analyzes the case of 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, the authors employ numerical computation, using procedures achieve and validate in our department. Their purpose is to analyze the influence of the gaseous fuel composition on the performances of Micro GT. It 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: - Biogas, Micro-Turbines, Thermodynamic Analysis, 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.

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, **b)** the power turbine, coupled by a special gear with a network frequency synchronous generator.

The Micro GT particular thermodynamic feature is using an Internal Heat Recovery Exchanger (IHRE), which preheats the compressed air with the heat extracted from the turbine exhaust gases. The remaining heat of exhaust gases could be recovered for CHP purposes, using an External Heat Recovery Exchanger (EHRE). The IHRE reduce the flue gas temperature at EHRE and limit the recovery at low temperature heat consumers.

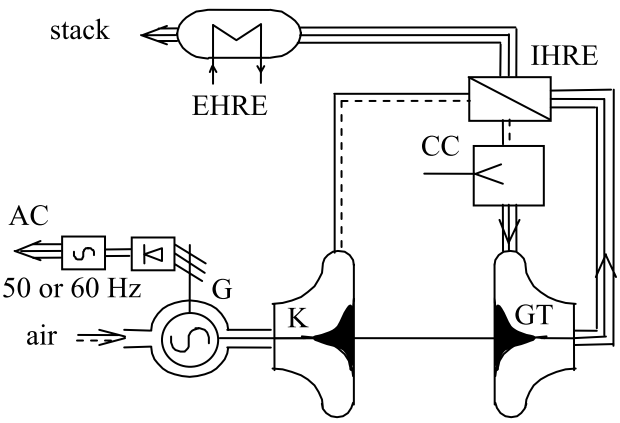

Fig.1 Single shaft micro gas turbine

For distributed generation in locations next to biogas digesters it is possible to use the recovered 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 burning the fuel, into work and power.

Like the medium and high power GT, Micro GT could burn a wide scale of clean liquids and gaseous fuel (natural gas, including low Sulfur content one, residual gases from industrial processes, biogas and others). Fuel switch modifies the performances of GT comparing to the reference ones. That should require a redesign of some parts. This paper determines the influences of biogas burning on performances and design of Micro GT manufactured for natural gas.

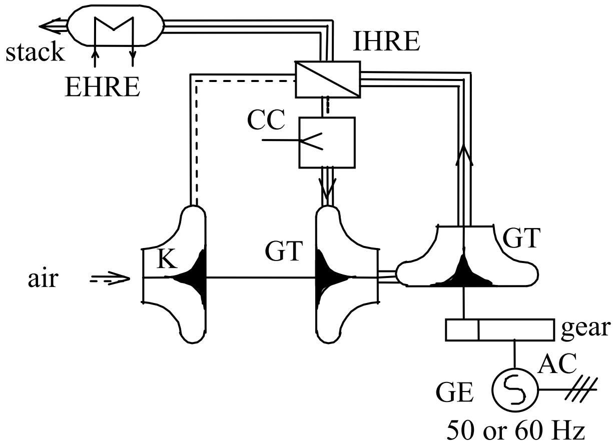

Fig.2 Double shaft micro gas turbine

2 Problem Formulation

The biogas from anaerobic controlled fermentation of organic wastes contents 56 to 80 % burnable gases [1]. 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 the requirement of air for stoechiometric burning.

Table 1. Data of gaseous fuels.

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 %, and the volumetric LHV, in kJ/m^3_N, 55 to 78.5 % from the CH_4 ones. The required humid air (ISO conditions: 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 volume unit (m^3_N/m^3_N) of biogas, related to the methane ones, represents almost the same percentages like the LHV ones.

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 [2, 3] that for the classic Brayton cycle, (without IHRE):

- 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 IHRE, ε values in the interval $\varepsilon(W_{sp\,max})$ to $\varepsilon(\eta_{el\,max})$. For the Micro GT, due to the IHRE influence, the relation between $\varepsilon(W_{sp\,max})$ and $\varepsilon(\eta_{el\,max})$ became [4 to 6]:

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

That allows obtaining enough good electric 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.

3 Problem Solution

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 [4, 7], and adapted to the cycle peculiarities.

In this paper we are doing two steps of analyses. 1) In the first step we suppose a Micro GT designed for burning methane and determine its reference flows. By changing the fuel data we recalculate the cycle, for the same flue gases flow rate, putting into evidence the influence of gaseous fuel elementary composition. 2) In the second step we are doing repetitive calculations in order to determine the variation of main indicators, like η_{el}, W_{sp}, and others, depending on ε, θ, and gaseous fuels elementary compositions. In the both steps we are using the following hypotheses:

- Atmospheric air parameters are the ISO ones.
- The pressure losses in AF, CC, IHRE, and EHRE are 3 % from the amount pressure.
- For taking into account the energy losses in CC and IHRE we consider $\eta_{CC}=\eta_{IHRE}=0.98$.
- 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 this general hypotheses we added a supplementary condition for IHRE, by fixing the ratio between the AC power generated flow, P_g, kW, and the IHRE conductance, Cd, kW/K. This one is proportional with logarithmical average temperature difference ($\Delta t_{med\ log}$) and with ratio between electrical power (P_g) and recycled heat flow ($P_{th\ rec}$):

$$\frac{P_g}{Cd} = \Delta t_{med\ \log} \cdot \frac{P_g}{P_{th\ rec}} \tag{3}$$

The temperature difference characterizes exergetic degradations into IHRE, limiting the NTU (Number of Transfer Unit) and thermal efficiency. The imposed ratio supposes that IHRE design is correlated with the turbo-machineries, and limit the IHRE size and cost.

In the first step we supposed a Micro GT with $P_{el}/Cd=16$, ε=4 and θ=4.5, which produces 100 kW AC power generated flow on burning methane. Considering that the flue gases rate flow is constant, we recalculated its performance indicators for other fuels. The obtained results are given in Table 2, with 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_{IHRE} – IHRE thermal losses, kW;
- $P_{th\ rec}$ – recycled heat flow (into IHRE), 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;
- $P_{k\ fuel}$ – AC power for fuel compressor, kW;
- P_{net} - net AC electricity flow, kW.

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
M_{fuel}	kg/h	68.0758	49.4094	36.4024	21.5677
$P_{th\ CC}$	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_{IHRE}	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 analysis of table 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 IHRE 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.

In the second step we enlarged the analysis area on: **a)** $\varepsilon \in [2\cdot\sqrt{2}$ to $4\cdot\sqrt{2}]$, in geometric progression with relative amount equal to $\sqrt[16]{2}$, **b)** $\theta \in [4$ to $5]$, witch correspond to $T_{max} \in [1153$ to $1441]$ K, respectively $t_{max} \in [880$ to $1170]$ °C. With the obtained results, restricting the field for ε values, we build Fig. 3 to 7. We added following supplementary nomenclatures:

- $(P_{g\ biogas} - P_{g\ meth})/P_{g\ meth}$ - relative increase of AC electricity generated flow, comparing to methane;
- α_{CC} air excess coefficient at combustion chamber;
- $\eta_{el\ g}$ - electricity generation efficiency;
- $\eta_{el\ net}$ - electricity net efficiency;
- $\Delta\eta_{g\ vs\ net} = (\eta_{el\ g} - \eta_{el\ net})/\eta_{el\ g}$ - relative variation of electrical efficiencies.

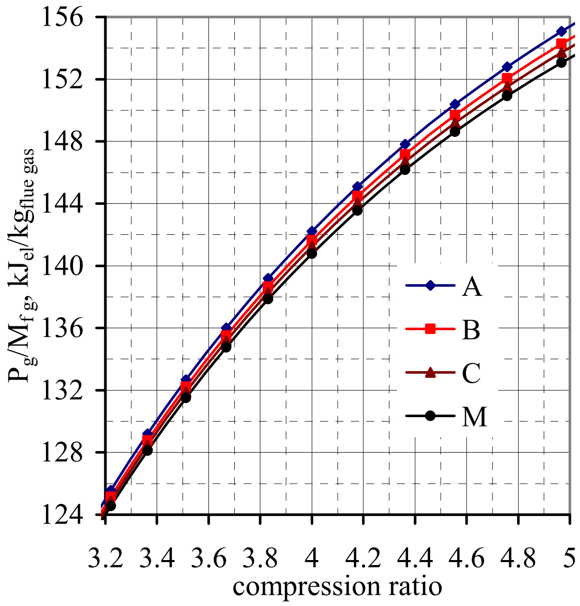

Fig.3 $P_g/M_{f\,g}$ vs. ε for θ=4.5.

The Fig.3 to 14 put into evidence that the identified trends on the peculiar case from Table 2 are maintaining on the entire analysis field, but the relative percentages of gains or losses changes with ε and θ values.

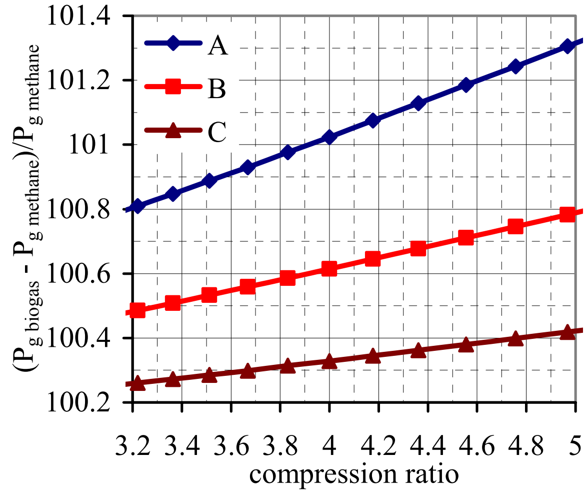

Fig.4 $(P_{g\ biogas} - P_{g\ methane})/P_{g\ methane}$ vs. ε for θ=4.5.

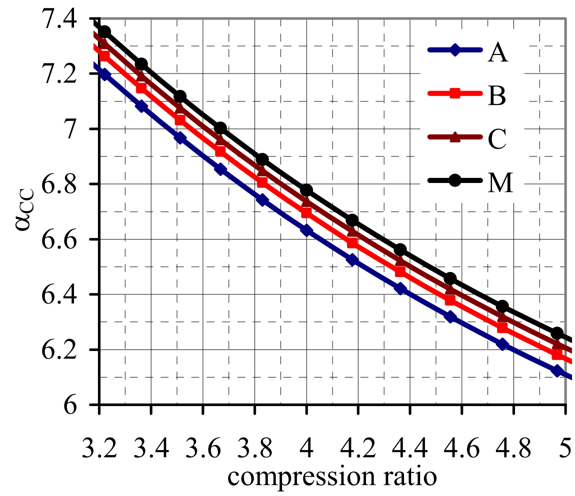

Fig.5 α_{CC} vs. ε for θ=4.5

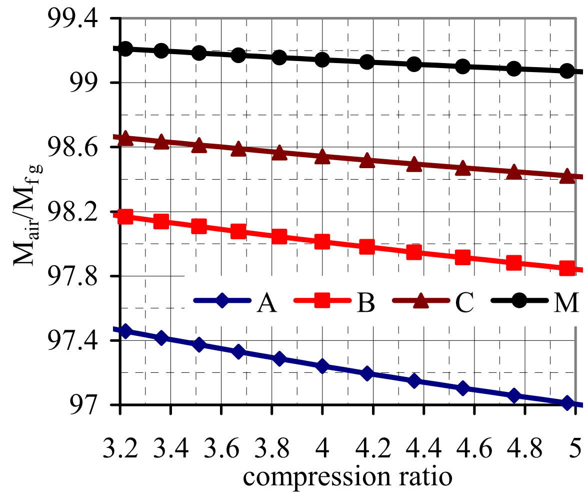

Fig.6 $M_{air}/M_{f\,g}$ vs. ε for θ=4.5.

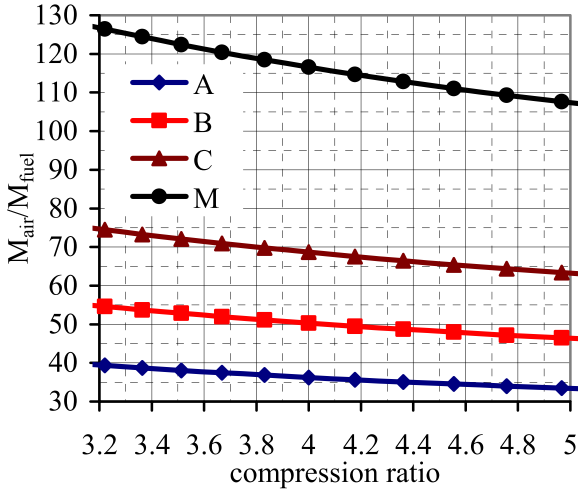

Fig.7 M_{air}/M_{fuel} vs. ε for θ=4.5.

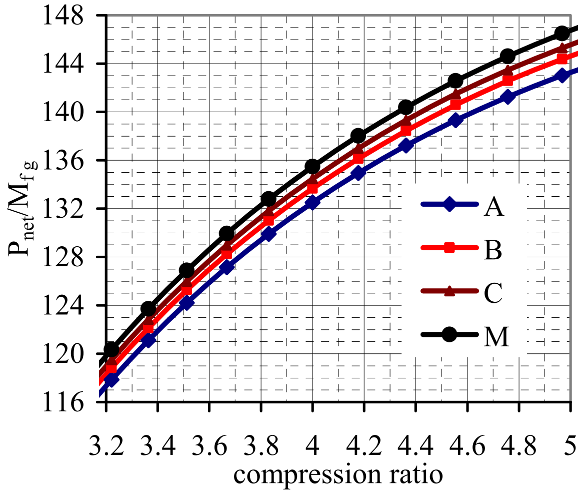

Fig.10 $P_{net}/M_{f\,g}$ vs. ε for θ=4.5.

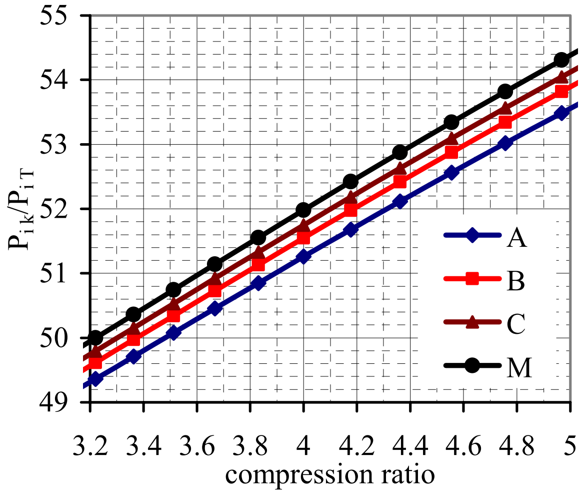

Fig.8 $P_{i\,k}/P_{i\,T}$ vs. ε for θ=4.5.

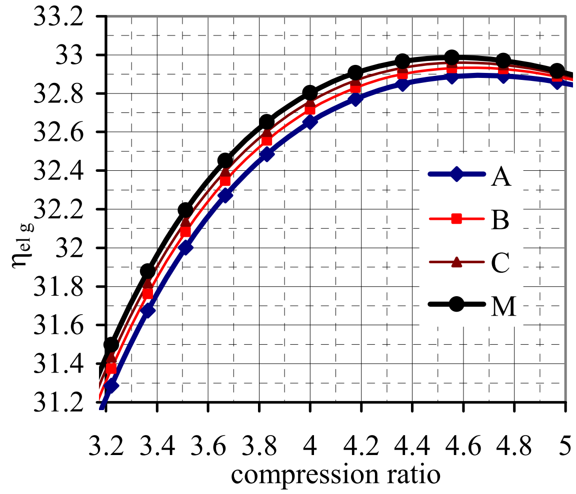

Fig.11 $\eta_{el\,g}$ vs. ε for θ=4.5.

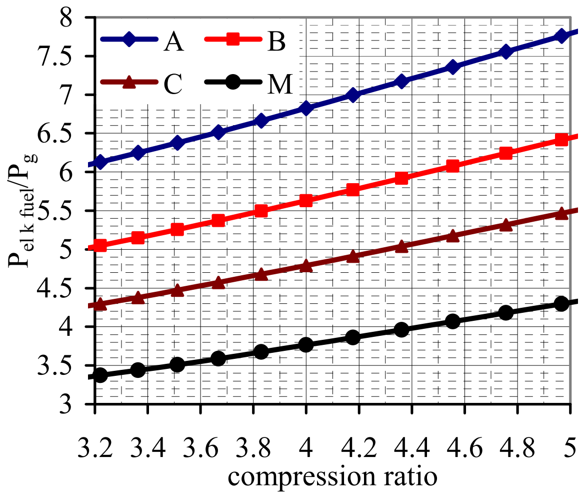

Fig.9 $P_{el\,k\,fuel}/P_g$ vs. ε for θ=4.5.

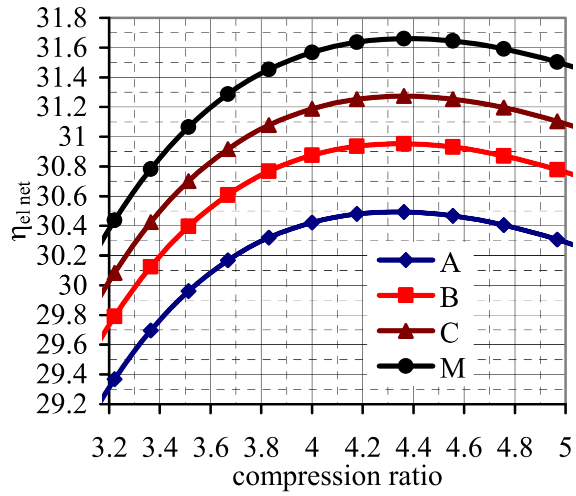

Fig.12 $\eta_{el\,net}$ vs. ε for θ=4.5.

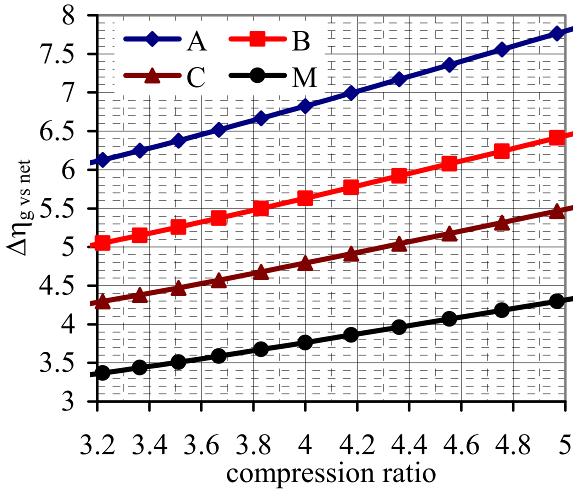

Fig.13 $\Delta\eta_{g\ vs.\ net}$ vs. ε for θ=4.5.

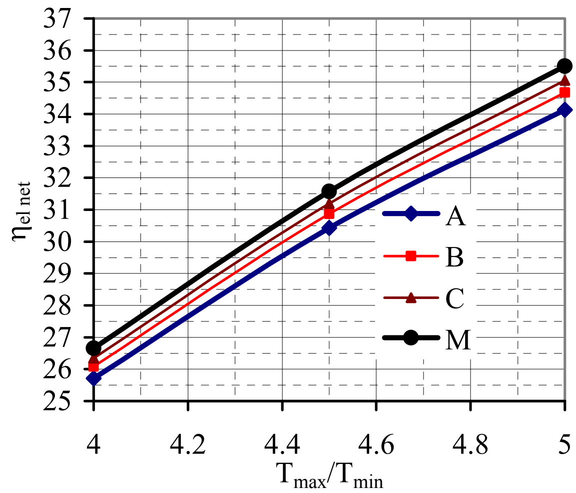

Fig.14 $\eta_{el\ net}$ vs. θ for ε=4

Analysis of main indicators evolution shows that:

- The augmentation of AC power generated flow is 0.3 to 2.3 %. It increases with ε rising and on LHV decreasing (Fig.4).
- The air excess coefficient at CC decreases with ε and LHV diminish, but remains almost similar with the methane burning one (Fig 5).
- The air compressor takes between 49.5 and 53.4 % from the mechanical work of the turbine. The quota increases with ε growing and diminishes with fuel LHV shrink (Fig.8).
- 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.9).
- The electrical efficiencies have optimal values for $\varepsilon \cong 4.5$ (Fig.11 and 12).
- The decrease of LHV reduces $\eta_{el\ net}$ (Fig.12). The differences between $\eta_{el\ g}$ and $\eta_{el\ net}$ are given, mainly, by the gas fuel power consumption (Fig.13).
- Increasing θ from 4 to 5 will allow efficiency growth from $\eta_{el\ net} \cong 26\%$ to over 35%.

4 Conclusion

The Micro Gas Turbines designed for use of methane could be adapted for bio-gas, using the same turbo-machineries. Even if the generator (or converter) output will have growths from 0.3 to 2.3 %, the net output will be with 0.7 to 2.5 % lower 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 are related to: 1) the burners of the CC and 2) the gas fuel compressors. These ones should be redesigned for mass flow rates up to 3 times bigger and volume flow rates up to 2 times bigger than the methane ones, depending on gaseous fuel composition and LHV.

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