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Effect of the Regenerator Efficiency on the Performance of a Micro Gas Turbine Fed with Alternative Fuels

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Abstract

In this paper a validated in-house MATLAB© model was used to assess the behaviour of the Turbec T100 MGT when operated with alternative low lower heating value (LHV) fuels; moreover, the effect of the cycle humidification is assessed. In both the aforementioned cases, the flow rates through the turbomachines, their operating points, and the effectiveness of the recuperator might change and determine performance losses. In particular, the recuperator in a MGT is a crucial component that allows to achieve good thermodynamic performance, also in presence of low compression ratios, and its performance can strongly influence the final output of the machine. Therefore, the aim of the work is to evaluate the effect of the variation of the operating conditions on the performance of the recuperator and, therefore, of the whole MGT. The use of alternative fuels with low LHV and of steam injection shifts the operative points of the turbomachines without strongly affecting their isentropic efficiency; in general, compression ratio is reduced and the flow rate of the compressor is reduced. Therefore, attention must be paid for the compressor stall limit. The recuperator shows a slight variation of the temperature of the fluids, but a higher efficiency is recorded as the flow rate are typically reduced and a better heat recovery performance can be obtained.

Keywords: Recuperator; Biofuels; Micro Gas Turbine; Cogeneration; MATLAB®; GRI-MECH

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Nomenclature

CFD	Computational Fluid Dynamics
CHP	Combined Heat and Power
DG	Distributed Generation
HRSG	Heat Recovery Steam Generator
LHV	Low Heating Value [MJ/kg]
$\dot{m}$	Mass Flow Rate [kg/s]
$\dot{M}$	Corrected Mass Flow Rate [$m \cdot s \cdot K^{0.5}$]
MGT	Micro Gas Turbine
NG	Natural Gas
p	Pressure
S	Syngas
STIG	Steam Injection Gas Turbine
T	Temperature [K]
TIT	Turbine Inlet Temperature

Greek symbols

β	Pressure Ratio
ε	Effectiveness
ω	Rotational Speed [rpm]
Ω	Corrected Rotational Speed [$rpm \cdot K^{0.5}$]

Subscripts

c	Compressor
exh	Exhaust Gas
in	Inlet Section
out	Outlet Section
r	Recuperator
t	Turbine

1. Introduction

In the Distributed Generation (DG) energy context Micro Gas Turbines (MGTs) represent an important and viable solution as a system for intelligent energy networks, supplementing renewable sources and the traditional community-scale power production. MGTs are small-scale power generation systems based on a Brayton cycle: they consist of a compressor, a turbine, a regenerator, an alternator and an electric system, which manages the power supply to the electrical grid. MGTs design does not simply involve a scale reduction of a standard traditional gas turbine, but a complete reassessment of the engine's architecture [1]: they operate a regenerative cycle and use a radial compressor and a radial turbine, that are much cheaper than the axial ones and that are designed to operate at a high rotational speed.

Due to the single stage of the radial compressor, the pressure ratio is approximately 3-5. Because the small size of the turbo-compressor group, MGTs use not air-cooled blades and the turbine inlet temperature is restricted by the material performance, usually below 950°C for higher-temperature alloys [2].

Even if they are still an emerging technology with relatively high investment costs, MGTs offer many advantages if compared to other CHP units, like internal combustion engines: higher load regulation capability, reduced emissions (about one order of magnitude), lower weights and dimensions, easier installation, low noise and vibrations, significantly reduced maintenance, high reliability and fuel flexibility [3]. As for the electric efficiency, values are slightly lower but the overall cogeneration efficiency is typically higher; however, MGTs are still susceptible to improvement thanks to technological advancement in the fluid-dynamics of the machines, on the materials of some of the components and on some thermodynamic cycle optimization [4]. Moreover, several solutions can be adopted to enhance the electric performance and to increase the operational flexibility in terms of electric index, like humid air cycles and inlet air cooling techniques. On the environmental point of view, the adoption of alternative renewable fuels, like syngas from a gasification process, further reduces the footprint of this technology. Typically, these fuels have low lower heating values and some modifications in the operating conditions of the MGT are involved. Indeed, because of the higher amount of fuel required to run the machine, the flow rates through the turbomachines, their operating points, and the effectiveness of the recuperator might change and determine performance losses. In particular, the recuperator in a MGT is a crucial component that allows to achieve good thermodynamic performance, also in presence of low compression ratios. While the effect of cycle modifications and alternative fuel on the turbomachines and on the combustion process has been widely investigated in literature, the study of the effects on the recuperator, though important, has been only partially assessed.

The recuperator is a small heat exchanger that preheats the compressed air from the radial compressor before it enters the combustor by recovering heat from the exhaust gas; this allows to considerably increase the electric efficiency of the MGT up to approximately 30% [5]. However, when varying the operative conditions, the effectiveness of the recuperator can be affected, strongly influencing the final performance of the machine.

On the market there are several types of recuperators available for MGTs and, in these years, many numerical codes based on multi-objective methodologies have been developed to design micro gas turbine recuperators [2], [6]–[9]. Most of the times the results available in literature show only few steady-state operative points close to design conditions. When varying the operative conditions, the effectiveness of the recuperator can be affected, strongly influencing the final performance of the machine. Ferrari and others carried out an experimental test on a Turbec T100 machine focusing the attention on the recuperator working in transient conditions, start-up operations and load rejection tests, producing data useful for the recuperator design as well as and theoretical model validation [10]-[11]. The aim of the work is to evaluate the effect of the variation of the fuel feed and the use of humid cycles on the performance of the recuperator of a Turbec T100 MGT fed with two different fuels and at various load and rotational speeds. A validated simulation algorithm, written in Matlab, allows to evaluate the MGT's performance when fueled with Natural Gas (NG) or Syngas (S) both performing a standard regenerative cycle and a regenerative cycle with steam injection upstream the combustor (mixed air-stream cycles, STIG) [12]. In particular, a sensibility analysis will be carried out to assess the effect of the variation of the compounds and of the compressor and turbine flow rate on the performance of the recuperator and, therefore, of the whole plant.

2. Research and methods

The MGT modelled in this work is a Turbec T100 built from Ansaldo Energia group. It generates 100 kW of electrical power with an efficiency of 30% in ISO 2314 conditions. The emissions of both NOx and CO are less than 15 ppm (at full load and 15% O_2) [13]. The layout of the cycle is reported in Figure 1.

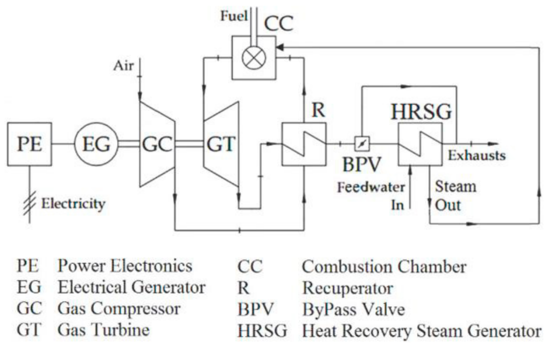

Figure 1 Layout of a Micro Gas Turbine

A Matlab code was developed by some of the authors of the present work [12], [14]–[16] consisting of 82 non-linear equations, describing the operation and the performance of the most important components of the MGT. The system of equations is solved using a non-linear solver based on Hessian matrixes. A set of equations describes the characteristic maps of the turbomachines, namely the compression ratio ($\beta_c = f_c(\dot{M}_c, \Omega_c)$; $\beta_t = f_t(\dot{M}_t, \Omega_t)$ and the isentropic efficiency of the turbine and of the compressor ($\eta_c = g_c(\dot{M}_c, \Omega_c)$; $\eta_t = g_t(\dot{M}_t, \Omega_t)$), as a function of the corrected mass flow rate and of the corrected speed ($\dot{M}_c = (\dot{m}_c\sqrt{T_{c_{in}}})/p_{c_{in}}$; $\Omega_c = \omega_c/\sqrt{T_{c_{in}}}$; $\dot{M}_t = (\dot{m}_{exh}\sqrt{TIT})/p_{t_{in}}$ and $\Omega_t = \omega_t/\sqrt{TIT}$). The maps were described thanks to experimental tests carried on using ambient air as working fluid. The equations are obtained from a regression curve of the experimental points that minimizes the R^2 index [17]. In the set of equations, the variation of the working fluid composition is taken into account with the ratio of the specific gas constant. The model also comprises mass flow rate balances and energy conservation balances of all the compounds flowing through the components of the machine. Specifically, the overall mass balance takes into account the mass flow rate of fresh air, the mass of fuel and, when adopted, the steam injection.

$$\dot{m}_{flue} = \dot{m}_{air} + \dot{m}_{fuel} + \dot{m}_{steam} \tag{1}$$

The energy balance considers the enthalpy content of all the compounds involved in the operation of the machine, starting from the humid air at the inlet to the compounds developed during the combustion. It is applied in all the components where a change in the composition, pressure and temperature take place. Combustion occurs in a reverse flow tubular combustor; downstream the turbine, the combustor products cross the recuperator to pre-heat the fresh air at the discharge of the compressor. After the recuperator, the thermal power content of the exhaust flue gas can be further recovered for cogeneration appliances. In case of steam injection, a Heat Recovery Steam Generator (HRSG) is used to produce the steam required for the humid cycle. In order to simulate the combustion process, the model comprises a routine developed in Matlab using Cantera library in order to evaluate temperature, emissions and the combustion kinetic of the MGT's combustor. This type of simulation has the advantage of being a low computational complexity algorithm, avoiding the long computational efforts required by a CFD model; a description of the combustor model is reported in [18]. It provides a sufficiently detailed description of the combustion process, giving information on the difference in operation between different fuels and it also takes into account the polluting compounds that are generated during the combustion. The behaviour of the combustor is simulated from a physical and thermodynamic (temperature and velocity) point of view, as well as from a kinetic point of view (molar fraction, reaction rate). In this work, the GRIM-Mech 3.0 library is used which contains the required species and reactions [18]. In particular, the effect on the mass flows, compression ratios, machine's efficiency and thermodynamic points are computed and used as initial compositions for the combustor model.

Also the regenerator was modelled using the data of the experimental analysis with NG feed. In particular, for a given rotational speed, the effectiveness of a heat exchanger can be assessed with the following equation:

$$\varepsilon_r = \sum_{i=0}^{3} A_i \cdot n^i \tag{2}$$

which is correlated to the ratio of the temperature differences:

$$\varepsilon_r = \frac{T_{air_{out}} - T_{air_{in}}}{T_{exh_{in}} - T_{air_{in}}} \tag{3}$$

where T_{air_in}, T_{air_out}, $T_{exahust\_in}$, are the temperatures of recuperator inlet air, outlet air and exhaust gas. The electronic control system of the MGT controls the fuel valve to keep the outlet temperature of the turbine at a fixed value of 645 °C. Also this operating constrain is taken into account in the model. The experimental data were used to define a relation linking the elaborated mass flow rate with the recuperator effectiveness. As aforementioned, the aim of the work is to assess the variation of the performance of the recuperator and of the whole MGT with varying fuel feed and operating conditions. Specifically, in this study a syngas produced through forestry wastes pyrolysis is considered. Chemical composition and LHV of the used S is reported in Table 1.

Table 1 NG and Forestry S composition and LHV

Molecule	Chemical formula	NG mole fraction	S mole fraction
Methane	CH_4	91.2%	21%
Ethane	C_2H_6	6.7%	-
Propane	C_3H_8	2.1%	-
Nitrogen	N_2	-	5%
Carbon Dioxide	CO_2	-	38%
Carbon Monoxide	CO	-	29%
Ethylene	C_2H_4	-	-
Hydrogen	H_2	-	7%
Total LHV [MJ/kg]		38.86	8.99

Besides the turbine and the compressor and the regenerator, also the diffuser downstream the turbine, the fuel compressor and most of the other auxiliaries and losses are modelled and taken into account thanks to experimental results [17]. The system of equations was run to assess the MGT performance when fueled with NG and S, both performing a standard regenerative cycle and a regenerative cycle with steam injection that can be produced by means of a dedicated HRSG, which exploits residual heat in the exhaust. Steam is injected upstream the regenerator with a pressure of 6 bar and with a superheating of 100 °C which resulted to be the best operative condition in terms of electric efficiency enhancement [16]. Obviously, the amount of steam that can be injected with the aforementioned conditions depends on the thermal power availability in the HRSG, namely the exhaust gas temperature and flow rate; therefore, at partial load, the maximum amount of steam produced is lower, while at full load up to 50 g/s can be produced, as will be shown in the following section.

3. Results and comments

The result reported in this section refer to the operation of the MGT with NG and S with and without steam injection upstream the combustor and with full and part-load electric output. Depending on the fuel feed and the standard or humid cycle operation, the operative point of the turbomachines change, affecting the mass flow rates, the gas temperature and the effectiveness of the recuperator. Table 2 reports the main operative conditions of the thermodynamic cycle and of the turbocharger for the nominal net power output of 100 kWe. With S feed, the fuel flow rate in the compressor is reduced much more sensibly than that of the turbine; as a consequence, the nominal power output is obtained at a lower rotational speed. Conversely the consumption of the fuel compressor increases because of the lower LHV of the fuel. A similar behaviour is obtained with the addition of steam: as steam increases the energy content of the flue gases, the desired electric power output is typically obtained at a lower rotational speed.

Table 2. Operative conditions of the MGT at 100 kWe

	Natural gas	Syngas	Natural gas with 20 g/s steam	Syngas with 20 g/s steam
rpm	70000	68000	68000	66600
Air flow rate [kg/s]	0.794	0.730	0.729	0.697
Flue gas flow rate [kg/s]	0.805	0.773	0.759	0.754
Turbine isentropic efficiency	84.8%;	85.4%	85.4%	85.7%
Turbine expansion ratio	4.1	3.8	3.8	3.7
Turbine Power [kW]	287.3	275.6	266.9	265.7
Compressor isentropic efficiency	78.6%	78.7%	78.8%	78.8%
Compressor compression ratio	4.5	4.2	4.2	4.1
Compressor Power [kW]	159.8	139.6	139.2	128.4
Fuel compressor consumption [kW]	3.1	10.6	3.0	10.9
Fuel flow rate [g/s]	7.0	41.4	6.8	42.7
TIT [K]	1217	1203	1205	1191

The introduction of an alternative fuel and steam injection modifies the operating conditions of the turbine and the compressor, but the operating points are still suitable in terms of isentropic efficiency; in particular, the turbine isentropic efficiency is increased, while attention must be paid in case of syngas use combined with steam injection as the compressor operates close to the stall limit. Moreover, the cycle compression ratio is reduced, as well as the TIT. These results are all consistent with the characteristic maps of the machines.

In order to assess the recuperator effectiveness with varying operating conditions, a set of simulation were carried out with rotational speed varying from 56000 rpm to 70000 rpm. Figure 2 reports the trend of the recuperator effectiveness and the exchanged thermal power as a function of the electric power output for the two considered fuels and without or with 20 g/s of steam injection.

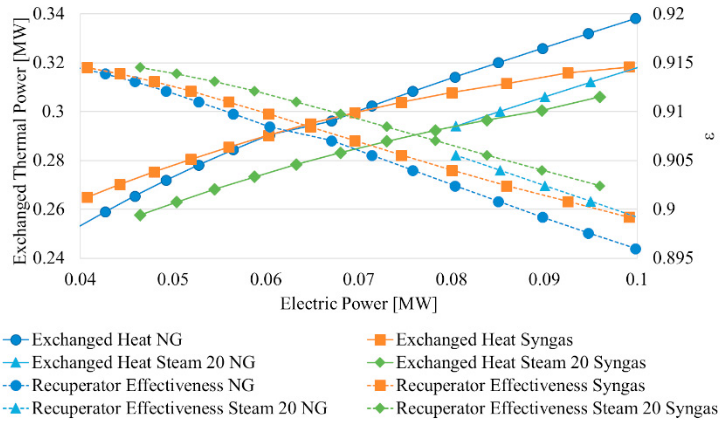

Figure 2 Exchanged heat and effectiveness of the recuperator as a function of the electric power output for NG and S with and without steam injection

The heat exchanged at the recuperator at full load with NG is the highest, while the efficiency is the lowest; conversely, with S the exchanged heat is slightly lower and the effectiveness is higher. With steam injection and NG feed, the heat exchanged in the recuperator is further reduced and the effectiveness is even higher. The combination of S and steam injection further increase this effect. This behaviour can be explained by analyzing the same trends as a function of the dry air flow rate and of the exhaust flow rate flowing in the recuperator (Figures 3a and 3b).

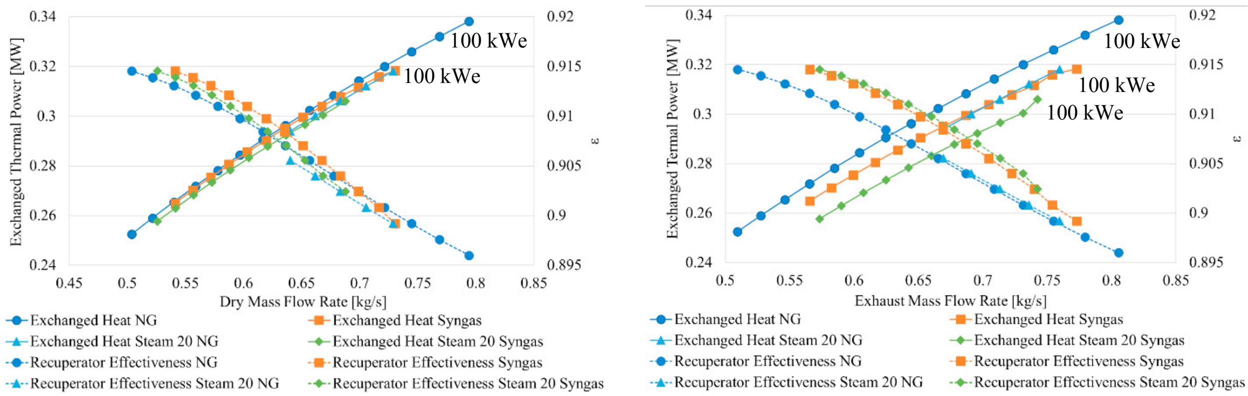

Figure 3 Exchanged heat and effectiveness of the recuperator as a function of the dry air (a) and exhaust mass flow rate (b) for NG and S with and without steam injection

When operating the MGT with S and, in particular, with steam injection, the flow rate of flue gases reduces but the reduction of the dry air flow rate is even higher, as previously reported. This means that the heat exchange surface of the recuperator is oversized and the flow velocity in the channels is lower, facilitating the heat exchange process. As a consequence, the recuperator effectiveness results to be higher and the exchanged thermal power is lower. Other parameters influencing, to a minor extent, the recuperator performance are the specific heat of the gases and their temperatures that, obviously, are affected by the presence of different compounds and in different shares.

In this respect, it is also worth to notice that the operation with 20 g/s of steam is not possible with NG feed when the power output gets lower than 80 kWe; this is due to the fact that the exhaust flow rate and heat downstream the recuperator is lower and, therefore, the HRSG cannot produce the required flow rate of steam at 6 bar and 100 °C superheating. When using syngas, instead, it is possible to produce 20 g/s of steam down to a power output of 55 kWe; indeed, the flow rate of dry air is reduced, therefore, the exhaust exits the recuperator with a higher energy content, terms of temperature and flow rate, that can be exploited by the HRSG to produce steam.

Figure 4 gives a better insight of the temperature figure in the two flows crossing the recuperator. As previously reported, besides the variation of the flow rates, also the compression ratio of the compressor tends to reduce whit S feed and with steam injection. This means that the inlet temperature of the fresh air in the recuperator is reduced while the exhaust at the exit of the turbine remain at 645 °C because of the fuel valve control strategy of the MGT. As regards the regenerated air conditions, it is possible to see that the outlet temperature is almost unaffected by the change of fuel and steam injection. Because of the lower amount of heat exchanged in the recuperator, also due to the lower flow rate of flue gases and, in particular, of the flue gas, the temperature of the exhaust at the exit of the regenerator results to be higher. As a consequence, there is also an advantage in terms of recoverable heat for cogeneration purposes or, as already stated, it is possible to exploit the residual heat content of the flue gases for the production of steam in case of application of a humid cycle.

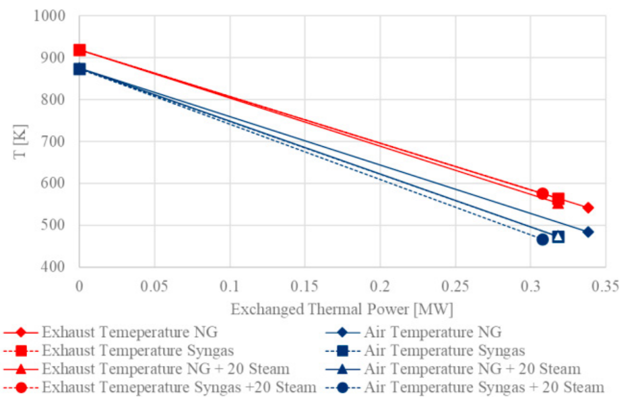

Figure 4 Recuperator inlet and outlet temperature and exchanged heat for NG and S with and without steam injection at the power of 100 kWe

Finally, Table 3 reports the electric efficiency of the MGT for the two tested fuels with and without steam injection. It is possible to conclude that steam injection gives advantages for NG feed. The use of S involves a higher fuel flow rate and fuel compressor consumption, which reduces the overall final electric efficiency.

Table 3 Electric Efficiency of the MGT fed with NG and S with and without Steam Injection

		Electric Efficiency	
Electric Power [kW]	60	80	100
NG	27.7%	28.8%	28.6%
S	23.2%	25.1%	26.7%
NG + Steam 20 g/s	-	29.1%	29.3%
S + Steam 20 g/s	22.8%	24.8%	26.1%

The introduction of steam with S has a further detrimental effect because the compression ratio and the TIT are reduced if compared to the standard operation of the cycle. As a consequence, the overall performance of the MGT is lower than the simple S case.

4. Conclusions

An analytical model developed in Matlab has been used to assess the behaviour of the turbomachines and the recuperator of a Turbec T100 MGT fed with natural gas or with a biomass-derived syngas, having a LHV equal to about 9 MJ/kg. The performance of the MGT was evaluated both with a traditional regenerated cycle and with steam injection upstream of the combustion chamber. The results show that, using syngas and steam injection, a variation of the operative points of the turbine and the compressor can be noted; however minor impacts on the overall performance are found; specific attention must be paid to the risk of stall of the compressor. As regards the recuperator, its effectiveness is enhanced and the exchanged heat is reduced; this is mainly due to the variation of the dry air flow rate, the gas temperature and the composition of the gases. As a consequence, the MGT proved its good flexibility in terms of fuel feed and cycle modifications. However, some considerations must be taken into account to grant the reliable operation of the system and the desired performance.

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