SEASONAL ENERGY AND ENVIRONMENTAL CHARACTERIZATION OF A MICRO GAS TURBINE FUELED WITH H_2NG BLENDS

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Abstract

This paper deals with the seasonal energy and environmental characterization of a commercial micro gas turbine fuelled with hydrogen enriched natural gas blends, to implement a Power-to-Gas option. When the renewable electricity excess occurs in a hybrid system, that energy can be converted into hydrogen to increase micro-turbine environmental performance. The experimental approach consists of testing on field that device at rated and partial load over the hot and cold seasons. The energy and environmental performance, along with the method to properly allocate the pollutants emissions for such applications, have been presented when the hydrogen fraction ranges in 0% vol. - 10% vol, by means of a 2% vol. step. From the main findings it emerges that both heat recovery and electrical efficiency undergo a slight increase even if the machine is strongly affected by the environmental derating. However, the larger beneficial effects occurred over the summertime, since the hydrogen mixtures can partially offset the outdoor environmental conditions change. Indeed, beyond the hydrogen fraction of 6% vol. and beyond 15 kW of electric output, the enrichment increases the rotational speed providing a leaner combustion. In so doing, the CO emissions are equal to zero, while the NO_x are almost constant.

Keywords: Power to Gas; H2NG blends; renewable hydrogen; hydrogen use; CHP; MGT

Highlights:

- Experimental analysis on Micro Gas Turbine fuelled with hydrogen and Natural Gas
- Micro Gas Turbine seasonal energy and environmental characterization
- Electric and heat recovery efficiency evaluation in rated and partial loads
- Pollutants emissions assessment and carbon dioxide reduction
- The most effective method to allocate emissions hailing from Micro Gas Turbine

1-INTRODUCTION

Climate change is a worldwide recognized problem and its mitigation was identified as one of the greatest challenges. The way to achieve this target is the reduction of the greenhouse gases (GHG) emission by the energy system decarbonization using renewables [1].

The major issue of climate change has necessitated timely intervention by the European Union (EU). A roadmap has been drawn up to make the European economy more climate-friendly and energy-efficient [2]. The fundamental objective is to ensure that the energy consumed is safe, reliable, competitive, locally produced and sustainable [3].

There is a large variety of technologies which exploits renewable sources like wind energy, solar energy, the potential energy of water, the energy of waves and of tides, the geothermal heat, solid biomass, biofuels and biogas [4].

The growth has depended mainly from the rates of the solar photovoltaic and wind growth respectively for the 65% and the 20%. The production of hydroelectric power has remained more or less the same [5]. The potential of these two technologies is great and their importance is expected to grow in the next years, but their use introduced new problems [6].

Another way to store electricity is the use of hydrogen as an energy carrier through a process called Power-To-Gas. The electricity is converted into hydrogen by water electrolysis and can be stored in pressured tanks or in metal hydride tanks [7].

There are several pathways for the use of the hydrogen. One of the most affirmed is its use as energy carrier for transportation purposes. Initially, there were a great interest in on-board fuel cells, but nowadays their development is far from what was expected entailing a partial loss of interest in this technology [8]. However, lately there is a growing attraction on using hydrogen or hydrogen mixtures in spark ignition ICEs (Internal Combustion Engines). The use of pure hydrogen shows the great advantage of almost zero greenhouse emissions. Notwithstanding, it also presents a lot of drawbacks such as the need of a high-pressure tanks and a change in designing combustion chambers [9].

Constrisciani et al. [10] in 2005 coined the term hydro-methane in order to indicate a mixture composed by methane and hydrogen in which the hydrogen fraction could range between 5 - 30% by volume. Hydrogen, compared to methane, is characterized by a higher combustion speed than pure NG. Indeed, it is characterized by a flame speed burning equal to 2.6 - 3.2 m/s against 0.37 - 0.45 m/s of methane (about 7 times faster). The greater flame propagation speed allows a rapid mixtures combustion with less release of unburnt residues. [11].

The development of advanced combustion technologies for gaseous hydrogen blended hydrocarbon fuels in gas turbine applications is an area of much current interest [12].

Sarli et al. [13] also found linear correlation of laminar flame speed to the hydrogen concentration in the fuel blend via numerical simulations on hydrogen-methane air premixed combustion.

Imteyaz et al.[14] performed experimental investigations on the characteristics of hydrogen-enriched flames in a swirl-stabilized premixed combustor.

Dinesh et al. [15] also compare the difference between H_2 and CO enrichment syngas non-premixed swirling flame characteristics. It indicated that hydrogen enriched fuel burns to form a much thicker flame with a larger vortex breakdown bubble compared to H_2-lean but CO-rich syngas. Rajpara et al. [16] studied the emission characteristics of CH_4/H_2 blend fuel in a swirl can combustor and pointed out that hydrogen addition can increase flame temperature, decrease flame dimensions and thus reduces CO emissions with marginal increase in NO_x emissions.

In addition to the experimental and computational studies on the combustion speeds of methane-air mixtures [17], many research projects were also carried out on the combustion speeds of air-hydrogen mixtures [18,19]. Other research studies [20] were focused on thermo-economic assessment of a novel integrated CHP system incorporating solar energy-based technologies, biogas-steam reformer with methanol and hydrogen production. In addition to NG, other gaseous fuels characterized by a relatively low L HV (Lower Heating value), such as coal bed methane and biogas, have become the investigation object in all countries of the world [21-23]. Thus, some works, such as [24, 25], carried out the analysis on dual-fuel Hydrogen-diesel engine optimization for CHP systems, while Schefer et al. [26] studied experimentally and numerically the combustion of hydrogen-enriched methane under lean premixed combustion conditions.

Furthermore , result of simultaneous reduction of NO_x and emissions in the operation of a dual-fuel engine with hydrogen rates of over 70% when a 40% exhaust gas recirculation (EGR) rate was applied, while the diesel injection timing controlled the local equivalence ratio [27-29]. An impressive review [30] studied hydrogen as a fuel for compression-ignition (CI) engines. The authors conclude that hydrogen is a promising fuel but further extensive research is required to assess whether low-temperature combustion techniques can assist in NO_x reduction [31].

Other authors investigated on the hydrogen addition for triggering combustion in diesel engines. In that case, when the engine is operating at low-load, with an extremely high hydrogen energy share ratio, the reduction of carbon and NO_x emissions of over 90%, have been registered. Moreover, soot emission was dropped by 85% compared to the conventional diesel feeding [32]. S. Ouchikh et al. [33] studied the effect of natural gas enrichment with hydrogen on combustion characteristics of a dual fuel diesel engine. Bing Ge et al. [34] were focused on the effect of hydrogen enrichment to natural gas swirling flame at atmospheric pressure conditions using a radially-staged DLE (Dry Low Emission) burner. Zhang et al. and Wang et al. [35, 36] carried out the experimental study on cyclic variation in a spark ignited engine fueled with biogas and hydrogen blends; the result shows that at hydrogen fractions of 15% to 35% in the blends, the engine operating stability is increased and cyclic variations are decreased. Halter et al. [37] provided the characterization of the effects of hydrogen addition in premixed methane/air flames. The experiments and analyses show that a small amount of hydrogen addition in turbulent premixed methane–air flames introduces changes in both instantaneous and average flame characteristics. Other works [38-39] investigated, by numerical and experimental approaches, on the hydrogen potential effects, when spark ignition engines are fueled with different fractions of CNG, CO_2, H_2 blend were conducted, The results showed that with the increase of CO_2 fraction, the effect of vortex on the flame wrinkle is weakened, and the flame development period increases. Hu et al. [40] carried out experimental study on performance and emissions of engine fueled with low LHV gas-hydrogen mixtures, The results show that the influence of equivalence ratio on laminar speed burning of flame is quite complex. Generally, the burning speed of the rich mixture is greater than the lean mixture ones. With the increase of non-hydrocarbon fraction in low energy density gas, the flame laminar speed burning decreased; on the contrary, with the increase of hydrogen fraction, the flame laminar speed burning of hydrogen-blended gas increases; the larger the hydrogen fraction, the larger the flame propagation rate is, increasing also the flame instability . Chiesa et al. [41] investigated on the possibility to burn hydrogen in a large size, heavy-duty gas turbine designed to run on natural gas as a possible short-term measure to reduce greenhouse emissions of the power industry. The work of di Gaeta et al [42] deals with the development of a dynamic model of a commercial 100 kW Micro Gas Turbine (MGT) fueled with mixtures of natural gas or methane and alternative fuels (i.e. Hydrogen). Currently, the most installed MGTs can be fuelled with low concentrations of hydrogen mixed with natural gas [43]. Cappelletti et al. [44] described numerical redesign of 100 kW MGT combustor for 100% of hydrogen, concluding that generally the hydrogen fueled combustor produce more NO_x than a CH_4 fueling, but it operates in very lean condition thus with very low temperature field. On the contrary, Lee et al. [45] worked on Gas turbine combustion performance, testing hydrogen and carbon monoxide synthetic gas as fuel for MGTs. Then, de Santoli et al. [46] were focused on the potential use of H_2NG blends in the well-proven technologies, such as internal combustion engines, for evaluating strengths and weaknesses of that application. In other research project the main findings, hailing from the experimental campaign performed on a pre-commercial version of micro CHP (60 kW_{el}), were presented taking into account the requirements related to dwellings application [47]. Furthermore, Lo Basso et al. [48, 49] investigated on the hydrogen addiction effects on the electrical and heat recovery efficiency as well as on pollutant emissions of a condensing micro-CHP. Hydrogen gives the possibility to work even with very lean mixtures [50]. One of the most attractive techniques to compensate the demerits of limited lean-burn ability and slow burning velocity of natural gas is to mix the natural gas with a fuel that possesses wide flammability limit and fast burning velocity. Hydrogen (H_2) can be considered the best gaseous candidate for natural gas. The thermal efficiency can be increased and exhaust emissions reduced by the addition of hydrogen [51-52]. Lounici et al. [53] reported that addition of hydrogen to NG is a promising method for improving the dual fuel mode, especially at low engine loads. The results showed that THC and CO emissions are in general reduced by H_2 enrichment as a result of the improvement of gaseous fuel utilization. From that literature survey it emerges that the hydrogen addiction it was widely considered for fuelling ICEs which were dedicated to transportation applications. Therefore, only a few research projects dealing with MGTs in CHP mode and with their main components can be found in the recent literature. For that reason, the authors decided to provide benchmark values that could be interesting for scientists and energy managers. It is noteworthy how the MGT performance are strongly related to the local conditions of the installation site and the environment, where pressure and temperature values are of great importance. So, this work consists basically of evaluating both the environmental and energy performance of such devices, when they are fed by a mix of natural gas and hydrogen in the summer and winter operation.

From the environmental point of view, other authors investigated on pollutants emissions calculation related to energy systems able to produce multiple outputs. Furthermore, the most of the scientific studies, referring for instance to the CO_2 emissions assessment, consider only those due to electricity production, neglecting the evaluation and quantification of those related to the thermal energy [54, 55]. So that, for evaluating better the energy system emissions which has multiple products and inputs, as in the case of CHP, it would be appropriate to take into account also the different method for their allocations [56]. Hence the final scope of this paper is to discuss how the specific emissions should be corrected to meet the local environmental constraints. Since the specific emissions can be calculated as the ratio between the emitted pollutants mass flow rate and the energy output, their values change over different seasons. Indeed, the environmental derating influences, on one hand the air to fuel ratio within the combustion chamber, on the other hand the energy output of MGT. That entails higher specific emissions values when the energy output decreases over the summertime. All of those issues have been deeply discussed in Section 4.

2-Test rig description and methodologies

This paper consists of evaluating the environmental energy performance of a micro gas turbine fed with a mix of natural gas and hydrogen in summer and winter operation. The experimental campaign was carried out at the DIAEE (Department of Astronautically, Electrical and Energy Engineering of Sapienza University of Rome), when the MGT was operating to meet the research Centre energy needs. The MGT was manufactured by Capstone Corporation and it is able to produce rated power output and heat equal to 30 kW_{el} and 63 kW_{th}, respectively. The work consists of a series of tests in which the percentage of hydrogen in the fuel is gradually increased. Hydrogen is produced directly in the laboratory by the use of an alkaline electrolyser.

2.1 Device characteristics and test procedure definition

MGTs are small size power generation system, less than 50 kW_{el}, based on the Brayton cycle. The main differences between heavy duty gas turbine and MGT are that the second ones are based on recuperative cycle and they are equipped with radial turbo machinery. Indeed, if they were just a scuffle of the heavy duty, the net efficiency of the MGT would be very low and would make this technology less competitive than reciprocating internal combustion engine.

Since this technology has just reached industrial maturity, nowadays there are about a dozen of MGT's manufacturers, `whose products are based on a well-known technology using metallic materials, and with very low emissions compared to reciprocating Internal Combustion Engines (ICE).

The MGT tested in this project is the Capstone C30, in grid-connected version. Capstone Turbine Corporation ® is the world's leader in manufacturing innovative and clean solutions for micro gas turbines. It is the first manufacturer to use high efficiency air bearing, thus drastically lowering the required maintenance. The main technical characteristics of tested machine have been outlined in Table 1.

Table 1 – Capstone C30 datasheet

Parameters	Values
Engine type	Natural gas Micro Turbine
Rotational speed	96,000 rpm
Compression ratio	4:1
Electric generator type	Synchronous air-cooled generator
Gross Active power	30 kW
Width	0.76 m
Depth	1.5 m
Height	1.8 m
Weight	Grid Connect - 405 kg
Net Active Power	28 kW
Voltage	400 -480 V, AC, PPPN
Frequency	50-60 Hz (Grid Connected)
CHP electric efficiency (based on LHV)	0.26
CHP heat recovery efficiency (based on LHV)	0.58
Max. thermal output power	60 kW

Hot water flow	2.9 m³/h
Exhaust Temperature	275ºC (530ºF)
Exhaust Gas Flow	0.31 kg/s
Max. operating pressure	6 bar
Average fuel consumption	11 Nm3/h
Fuel	Natural gas, Liquid Fuels, Biogas, Associated gas, Propane Gas

During the electrical set-point operation the guiding parameter of the MGT is the electrical power, hence the available heat at the exhaust is a result of this functioning. It can be directly exploited by the HVAC systems or, if the thermal power required is lower than the produced one, it can be dissipated in a heat sink.

During the tests the turbine operated at rated and partial load conditions, since the device is integrated in the heating and cooling plant for the research centre. For that reason, the control logic has been setup in thermal tracking mode. Indeed, the MGT control unit acts on the rotational speed to keep under control the supply water temperature at 70 °C.

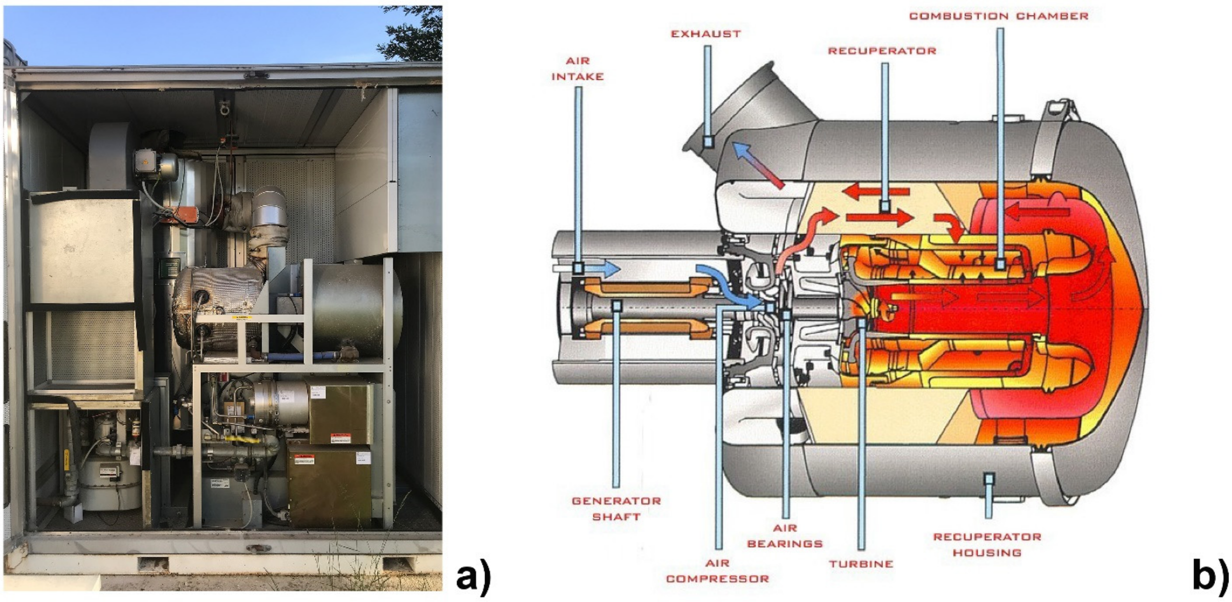


Fig.1 – Capstone C30 MGT overview: a) the installed MGT; b) cross section of combustion chamber and annular regenerator [54]

Figure 1 reports the package of the installed MGT and some technical details of combustion chamber and heat exchanger. Specifically, a cutaway view of Capstone C30 is shown in Figure1-b. The single stage Turbine and Compressor impellers are inertia welded to the shaft, which supports the electric generator and provides also its cooling. MGTs typically comprise an air compressor, combustor, turbine, heat regenerator and an alternator. In the commercial Capstone C30 package (see Figure 2) the process air in the gas turbine is firstly compressed by a radial compressor (from state 1to state 2) and further pre-heated by the annular regenerator (i.e. the gas to gas heat exchanger) using the hot turbine exhaust gas (from state 2 to state 2'). This form of heat recovery allows a significant increase of the electrical efficiency at typically low-pressure ratios of micro gas turbines. Indeed, that machine is characterised by a compression ratio equal to 4. The process air is then fed to the combustion chamber (state 3), mixed with the fuel and burned. Subsequently, the combustion by-products are depressurised via an expander (4), they are cooled by the heat regenerator (from the state 4 to the state 4') and finally they flows through the liquid to gas heat exchanger (from state 5 to state 6) to generate the hot water for the HVAC system.

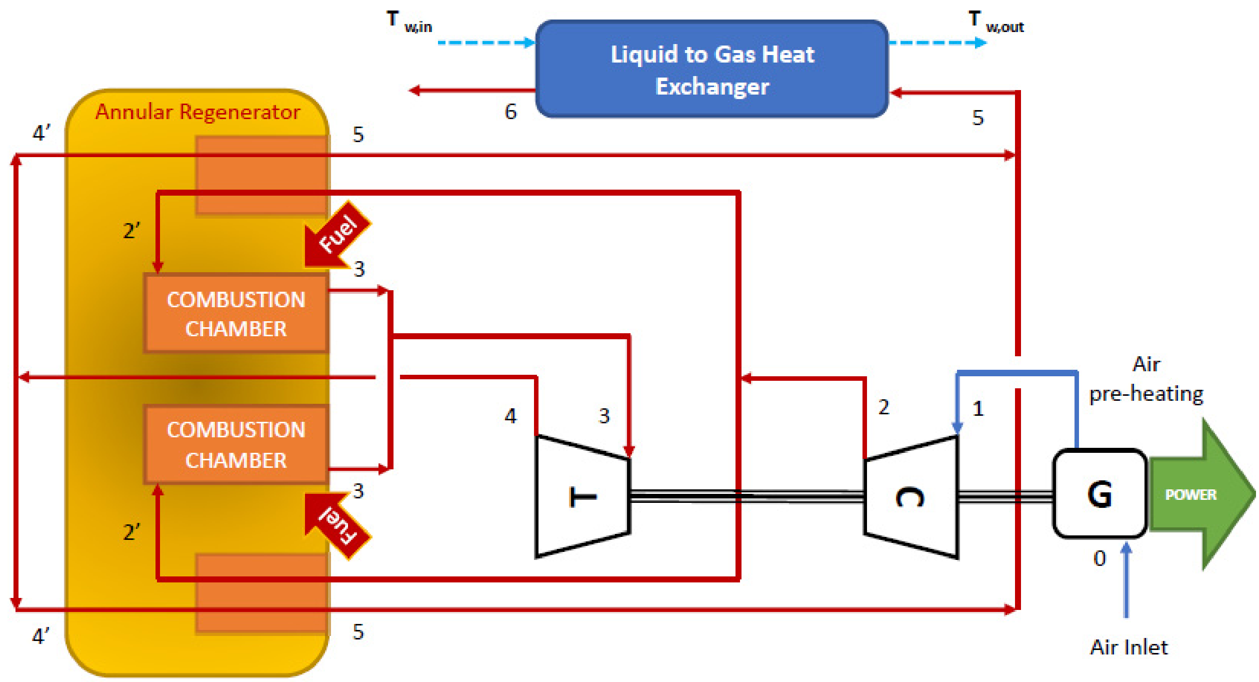


Fig 2. Capstone C30 process layout

The experimental campaign has been divided in two different phases: the first one consisted of collecting all the most meaningful parameters over the summer operation, starting from April to September, using hydrogen and NG mixtures in the range 0% vol. -10% vol. by adding 2% vol. of hydrogen in each steps; on the contrary, the second one has been carried out starting from the end of November up to March using only

H_2NG blends at 0% vol., 6% vol. and 10% vol. That choice was due to the fact that 6% vol. of H_2 is the threshold fraction beyond which something happens from the energy and environmental point of view.
Furthermore, the sample time for measurements has been fixed equal to 30 minutes, so that it has been possible to collect 16 readings per day, for a total number of 3,200 values over the whole test period.
In the end, environmental values have been collected as well by a portable gas analyser, in terms of O_2 concentration [% vd.], CO concentration [ppm vd.], CO_2 concentration [% vd.], NO concentration [mg/Nm^3 vd.] and NO_2 concentration [mg/Nm^3 vd.]. The exhaust gas analyser was manufactured by TESTO, and the model is 350-S. It is noteworthy that the device was not equipped with an NDIR (Non-Dispersive Infra- Red) probe for the carbon dioxide measurement, therefore that pollutant concentration has been calculated according to the Ostwald combustion equation:

$$\frac{[\mathrm{CO_2}]}{[\mathrm{CO_2}]_{\mathrm{th}}}+\frac{[\mathrm{CO}]}{[\mathrm{CO}]_{\mathrm{th}}}+\frac{[\mathrm{O_2}]}{\mathrm{m}}=1 \qquad (1)$$

Here, the terms in square brackets on fractions numerators represent the measured concentrations by the analyser probes, while those ones marked by the subscript "th" are the so-called undiluted or theoretical concentrations which are peculiar of any fuels. Then, *m* denotes the common oxygen concentration in the air.
Thus, Table 2 reports the main technical characteristics of each electrochemical probe of gas analyser.

Tab. 2 – Gas analyser probes technical specification.

Parameters	Range	Resolution	Accuracy
O_2	0-25vol.%	0.01vol.%	±0.2Vol.%
CO, H_2-comp.	0-10000ppm	1ppm	±10ppm (0-199ppm) ±5% of reading (200-2000ppm) ±10% of reading (rest of range)
NO	0-4000ppm	1ppm	±5ppm (0-99ppm) ±5% of reading (100-1999ppm) ±10% of reading (rest of range)
NO_2	0-00ppm	0.1ppm	±5ppm (0-99.9ppm) ±5% of reading (rest of range)

When H_2NG blends are used, the commercial gas analysers are not able to calculate directly the carbon dioxide concentration due to the fact that they do not have, in their software, the undiluted values for such mixtures. For that reason, some of the authors of this article provided in previous works those theoretical values. Table 3 outlines the theoretical concentrations associated to the H_2NG blends used over the experimental campaign.

Table 3 – CO and CO_2 theoretical concentration by volume within the dry exhaust gas for H_2NG mixtures

f_{H2}	0% vol.	2% vol	4% vol.	6%	8%	10% vol.
$[CO]_{th}$	14.976%	14.890%	14.801%	14.710%	14.616%	14.519%
$[CO_2]_{th}$	11.669 %	11.617%	11.563%	11.507%	11.449%	11.389%

2.2 Numerical approach to evaluate intermediate state values

In order to assess the MGT energy performance some parameters have been directly measured by instruments and probes while the other ones, especially those related to the intermediate states, have been calculated by a reverse engineering process. In detail, the electric power output, the rotational speed, the inlet temperature T_1, the combustion temperature T_3 have been registered by the onboard PLC (Programmable Logic Controller) unit. Thus, the final exhaust gas temperature T_6 has been measured by the gas analyser temperature probe and the hot water temperature as well as the water flow rate have been registered by a thermal counter. Finally, the NG and hydrogen flow rate have been measured by two different dedicated mass flow rate meters.

For a complete characterization of the thermodynamic cycle, T_2, T_2', T_4, T_4' and T_5 are necessary, which can not be experimentally collected. Those parameters have been computed by an analytical approach by hypothesizing some design values as reported in Table 2:

Table 4 – Technical assumptions for calculations: Source [41]

Compressor polytropic efficiency	Turbine polytropic efficiency	Combustor efficiency	$\Delta T_{4-4'}$ [°C]
0.85	0.9	0.99	15

In addition, to perform more realistic calculations the pneumatic efficiency has been introduced according to the Equation 2:

$$\eta_p = \frac{\beta_t}{\beta_c} \tag{2}$$

Where, β_t is the turbine expansion ratio, while β_c is the compressor's one. That efficiency has been considered variable with changes in the exhaust gas flow rate according to a quadratic trend ranging between the maximum and minimum values of 0.85 and 0.75.

It is important to highlight that corrected parameters maps related to the compressor and the expander were not available. Indeed, those ones are generally an exclusive property of manufactures, therefore it is not very easy to get those data to build an effective analytical model. For that reason, in this paper the authors decided to apply a semi-empirical approach so as to estimate some required parameters.

Having said this, on the basis of fundamental laws of thermodynamic, the outlet air temperature from the compression stage reads as:

$$T_2 = T_1 \beta_c^{\frac{R_{exh}}{c_{p1,2}\eta_{pol,c}}} \tag{3}$$

By the heat balance equation, the outlet air temperature from the annular regenerator can be evaluated as follows:

$$T_{2'} = T_2 + \frac{\dot{m}_{exh} \cdot c_{p4,5}(T_4 - T_5)}{\dot{m}_{air} \cdot c_{p2,2'}} \tag{4}$$

Given that the combustion temperature is a measured parameter, the energy balance equation related to the combustion chamber (Eq. 5), once it is solved by T_3, it has been used as a check value for the convergence of iterative procedure calculation on exhaust gas composition.

$$T_3 = T_4 + \frac{\alpha}{c_{p3,4}(1+\alpha)} \cdot \left[\frac{P_{el}}{\dot{m}_{air} \cdot \eta_{el}} - c_{p1,2}(T_2 - T_1)\right] \tag{5}$$

Here, α denotes the MGT actual Air to Fuel ratio which can be calculated by the Equation 6:

$$\alpha = \frac{\dot{m}_{air}}{\dot{m}_{fuel}} \tag{6}$$

The same iterative approach has been used for calculating the T_5 temperature. Indeed, the final exhaust gas temperature to the stack can be correlated to T_5 by the heat balance equation related to the gas to liquid heat exchanger, as reported in Equation 7.

$$T_6 = T_5 - \frac{\dot{m}_{H2O} \cdot c_{p,H2O}(T_{w,out} - T_{w,in})}{\dot{m}_{exh} \cdot c_{p5,6}} \tag{7}$$

In that way, it is possible to check the convergence to T_6 once the T_5 temperature has been preliminarily evaluated by the effectiveness equation associated to the annular regenerator. Therefore, assuming plausible values for such parameter the T_5 temperature can be calculated as follows:

$$T_5 = T'_4 - \varepsilon \cdot \frac{\dot{m}_{air} \cdot c_{p,air}}{\dot{m}_{exh} \cdot c_{p,exh}} (T'_4 - T_2) \quad (8)$$

It is important to point out that the fourth order polynomial equation for specific heat at constant pressure has been used for all calculations, as reported in Equation 9:

$$c_p = a + bT + cT^2 + dT^3 + eT^4 \quad (9)$$

Where *a*, *b*, *c*, *d*, and *e* are the peculiar coefficients of each gaseous components within the exhaust, which were assumed from literature and T is the average temperature. The calculation of the exhaust gas specific heat was done by making the weighted average of the chemical species properties.

Having said this, the MGT performance have been evaluated referring to the electric, heat recovery and First Law efficiencies, which can be calculated as follows:

$$\eta_{el} = \frac{P_{el}}{\dot{m}_{fuel} \cdot LHV} \quad (10)$$

$$\eta_{hr} = \frac{P_{th}}{\dot{m}_{fuel} \cdot LHV} \quad (11)$$

$$\eta_I = \eta_{el} + \eta_{th} \quad (12)$$

According to previous equations, the electric power output has been measured by a wattmeter, while the thermal power output has been registered by a thermal counter, as aforementioned, and the fuel flow rate by mass flow meters which are integrated within the electrolyser mixing section. In Table 5 the measurement equipment technical characteristics have been summarised.

Table 5 – Measurement instruments accuracy and scale.

Thermal counter (Honeywell EW5451A4100)	
Parameters	Values
Temperature range	5-130° C
Temperature difference range	3-120° K
Temperature drop resolution	ΔT 0.2 K
Temperature measurement error	max 1.5% at ΔQ 3 K (0.5+ ΔQ min/ΔQ)%
Flow rate	0.6 (m^3/h)
Flow rate relative error	± 1%
Watt meter	

Electrical power output range	0-30 kW
Relative error	± 1.5%
Bronkhhorst H_2 mass flow rate meter	
Measurement Range	0-1.5 Nm^3/h
Accuracy	± 0.5% RD plus ± 0.1%
Bronkhhorst NG mass flow rate meter	
Measurement Range	0-20 Nm^3/h
Accuracy	± 0.5% RD plus ± 0.1% FS
PT100 High temperature	
Temperature range	-60 + 600° C
Temperature measurement error	±0.3° C at 0° C
PT100 Low temperature	
Temperature range	-30 + 200° C
Temperature measurement error	±0.15° C at 0° C

In order to calculate the MGT thermodynamic cycle, the air flow rate has been deduced by means of Equation 13, along with the mass balance in Equation 14, once the actual fuel consumption has been registered.

$$(1-f_{H2}) \cdot CH_4 + f_{H2} \cdot H_2 + \lambda \cdot \left[2 \cdot (1-f_{H2}) + \frac{f_{H2}}{2}\right] \cdot (O_2 + 3.7846N_2)$$
$$\rightarrow (1-f_{H2}) \cdot CO_2 + (2-f_{H2}) \cdot H_2O + [2 \cdot (1-f_{H2}) + \frac{f_{H2}}{2}] \cdot [(\lambda-1) \cdot O_2 + \lambda \cdot 3.7846N_2] \quad (13)$$

$$\dot{m}_{exh} = \dot{m}_{air} + \dot{m}_{fuel} \quad (14)$$

In detail, the air flow rate calculation can be performed when the fuel flow rate q_{fuel}, the stoichiometric air to fuel ratio by volume $\alpha_{vol,stoic}$ and the relative equivalence ratio λ are known, according to the Equation 15 . This latter can be directly measured by the exhaust gas analyser probes, registering the oxygen residual content to the stack.

$$q_{air} = \lambda \cdot q_{fuel} \cdot \alpha_{vol,stoic} \quad (15)$$

3.Results and Discussion

The main aim of this study is to investigate the H_2NG fuelling effects on the energy and environmental performance of a commercial MGT for CHP applications. The experimental campaign has been carried out over one-year period dividing tests by season. In this section, the summer and winter data have been presented and compared each other in order to understand when and how the hydrogen enrichment is more effective for feeding the MGT.

It is well known that gas turbines are strongly penalised by the outdoor environmental temperature variations as well as the inlet air pressure value. Since the corrected rotational speed n_c is dependent on the air temperature, while the corrected mass flow rate m_c is dependent also on the inlet pressure (see Equation 16 and 17), when the air temperature increases, the former parameter tends to decrease, and the latter enhances.

$$n_c = \frac{n}{\sqrt{T_{inlet}}} \tag{16}$$

$$m_c = \frac{m \cdot \sqrt{T_{inlet}}}{p_{inlet}} \tag{17}$$

In the single shaft turbines, the corrected parameters variations imply the rotational speed reduction and the compression ratio decreasing, respectively due to the matching issues between compressor and expander. As a consequence, when gas turbines operate at constant rotational speed the mechanical power output largely lessens. On the contrary for gas turbines operating at variable rotational speed, to keep under control the power output the electronic control unit accelerates the shaft rounds per minute. For those reasons, all the data presented here derive from a filtering process so as to make values comparable. Indeed, a frequency analysis has been carried out to find the most frequent operating power of MGT. Thereafter, those values have been selected and sorted by the same inlet conditions to the compressor side. In such a way, the comparison between the NG feeding case and the H_2NG ones is reliable. It is important to highlight that, even if the MGT has been forced to run at rated power output in the summertime for the experimental purpose, only 22 kW of electrical output has been reached owing to the derating effect, whereas in the winter tests the maximum has been equal to 27.2 kW. That value is the useful electric output since the remaining 2.8 kW are used for driving fan and circulation pump which are installed within the Capstone C30 package.

The compressor inlet temperature values over the test period with the hydrogen enrichment have been reported in Figure 3 and Figure 4. By those data, it is possible to clarify the different boundary conditions during the hydrogen enrichment test as well as to distinguish partly the effects caused by the environmental derating and the hydrogen use on the MGT performance.

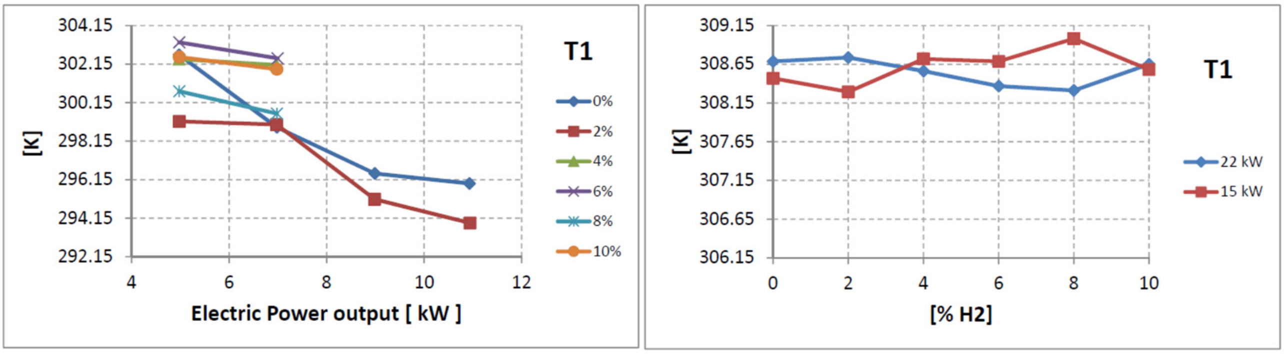


Fig.3 – Air inlet temperature: T_1 vs. Electric power output with changes in H_2 fraction at low partial load (left side); T_1 vs. H_2 fraction with change in electric power output (right side)

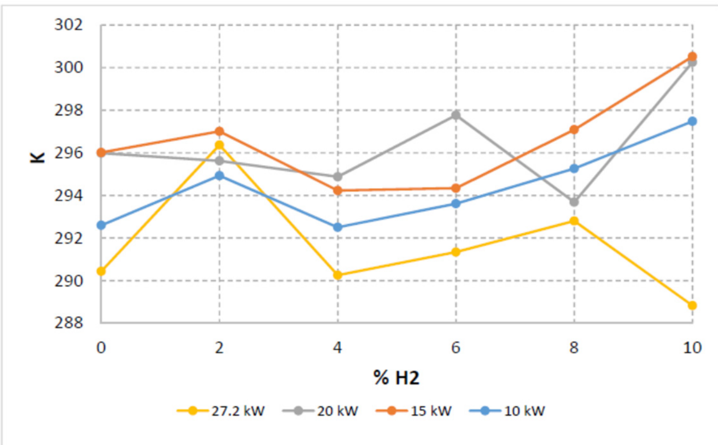


Fig.4 – Air inlet temperature vs. H_2 fraction with changes in electric power output over the winter experimental campaign.

The collected data confirm that as the inlet temperature increases the MGT' s electric power output decreases. When the hydrogen is added the same electric loads can be achieved also with different inlet air temperatures, showing that the enrichment is able to partially offset the outdoor temperature negative effect. It is then more evident when MGT runs at partial load; referring to Figure 4 it is possible to notice that 15 kW can be achieved anyhow with hydrogen fraction equal to 10% and T_1 equal to 300 K, approximately. Since the air inlet temperature variations strongly affect the rotational speed, it has been also investigated on the potential correlation between the hydrogen fraction and that parameter. Looking at Figure 5 and Figure 6 it emerges that hydrogen does not contribute significantly to accelerate the MGT shaft once it operates close to the maximum load conditions and with small hydrogen addition. Indeed, referring to the hot season data, the rotational speed enhancement in partial load conditions is caused by the higher outdoor temperature which has been registered during the hydrogen use. Notwithstanding, when the MGT was running at 22 kW and the hydrogen enrichment tests have been set in the range 4% vol. – 8% vol., in that case positive contributions have been occurred, although the inlet temperature values have been lower.

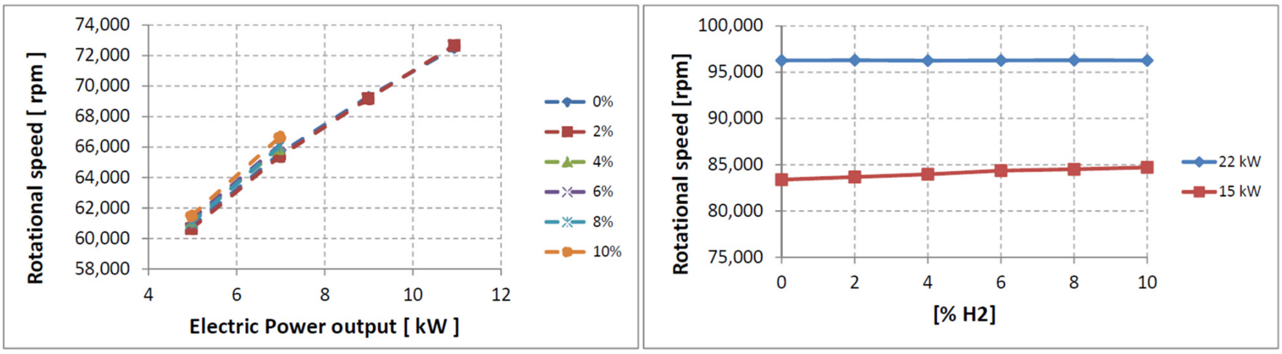


Fig.5 - Rotational speed vs. electric Power Output with varying H_2 fraction over the winter tests

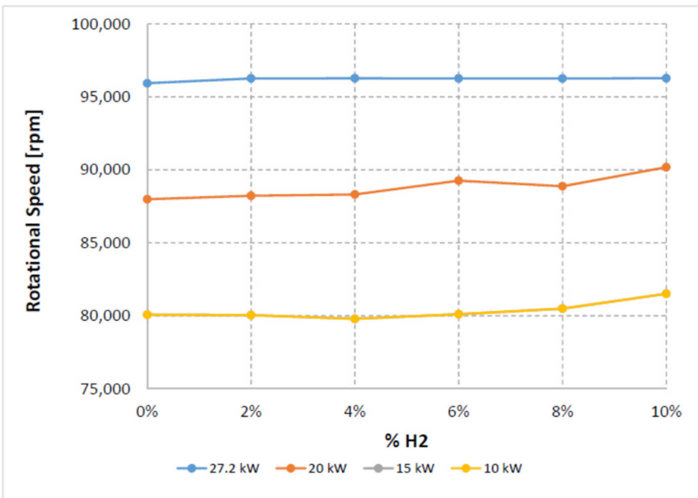


Fig.6– Rotational speed vs. H_2 fraction over the winter tests

A quite similar behaviour has been noticed in the winter operation. Indeed, according to the Figure 6 the rotational speed enhances slightly as the hydrogen fraction increases starting from 4% vol. up to 10% vol, especially when the MGT runs at 27.2 kW. That is due to the control unit attempt to compensate the lower fuel energy density within the combustion chamber, since the machine has to follow the required power set up to match the thermal tracking logic.

Furthermore, the changes in rotational speed entail variations in the air flow rate elaborated by the compressor as reported in Figure 7 and Figure 8. From those charts it emerges that as the rotational speed and hydrogen fraction increase the air flow rate grows accordingly, exception for MGT partial load conditions in the summertime test. Indeed, looking at Figure 8 at the same electric power output, when the hydrogen fraction in the blend increases the air flow rate tends to shrink. As a consequence, the air to fuel ratio values in those operating conditions are lower than the NG case, influencing the combustion temperature T_3. More generally, it can be state that as the MGT runs beyond 50% of rated load the air flow rates are higher, and the hydrogen enrichment contribute to make combustion leaner reducing the flame temperature.

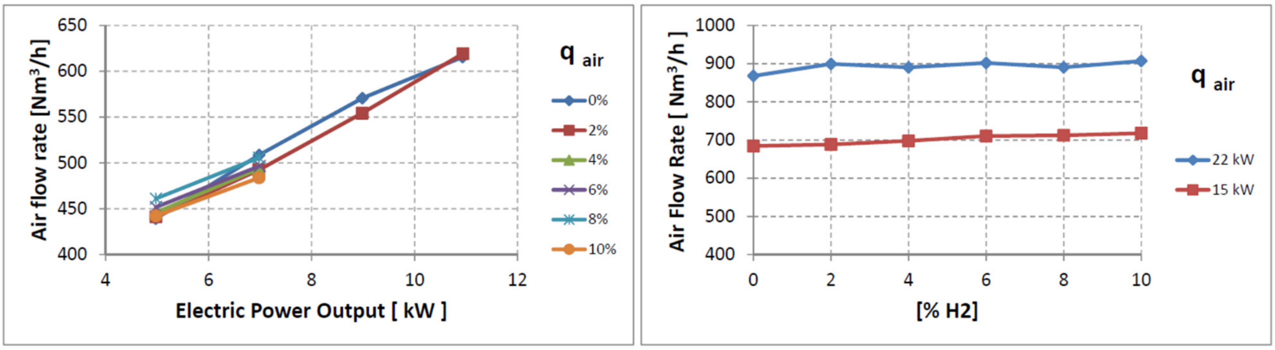


Fig.7 – MGT Air flow rate over the summer tests: air flow rate vs. electric power output with changes in H_2 fraction (left side); air flow rate vs. H_2 fraction with changes in electric power output

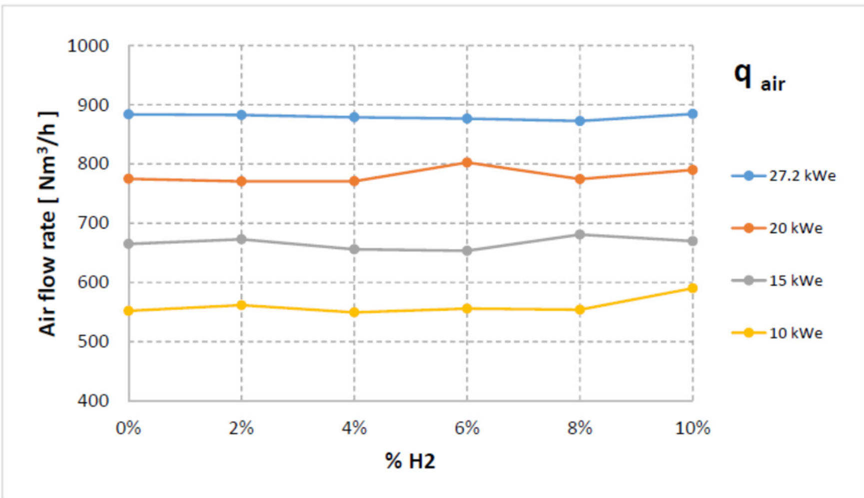


Fig.8 – MGT Air flow rate vs. H_2 fraction with changes in electric power output over the winter tests

It is well known that the hydrogen addiction to other fuels affects their flame temperature and for that reason the hydrogen use within conventional devices has not taken into account due to its own dangerousness. Yet, when it is used in cofiring applications the flame temperature slightly increases. The present experimental campaign on MGT shows how that effect has been mitigated by the air to fuel ratio changes. All of data related to both the hot and cold season tests (see Figure 9 and Figure 10) confirm that at fixed power output the combustion temperature is almost constant. Moreover, the T_3 values are only dependent on the electric load and consequently on the fuel flow rate which is controlled by the onboard electronic control unit. It is worth of noticing that the combustion temperatures at maximum loads, in the hot and cold seasons, are very close, since they range in 1073 K – 1081 K, showing that the MGT thermodynamic efficiency (i.e. Carnot efficiency), and consequently the electric efficiency, is weakly affected by hydrogen but strongly dependent on the outdoor environmental conditions. That issue can be also verified referring to the charts reported in Figure 13 and Figure 14

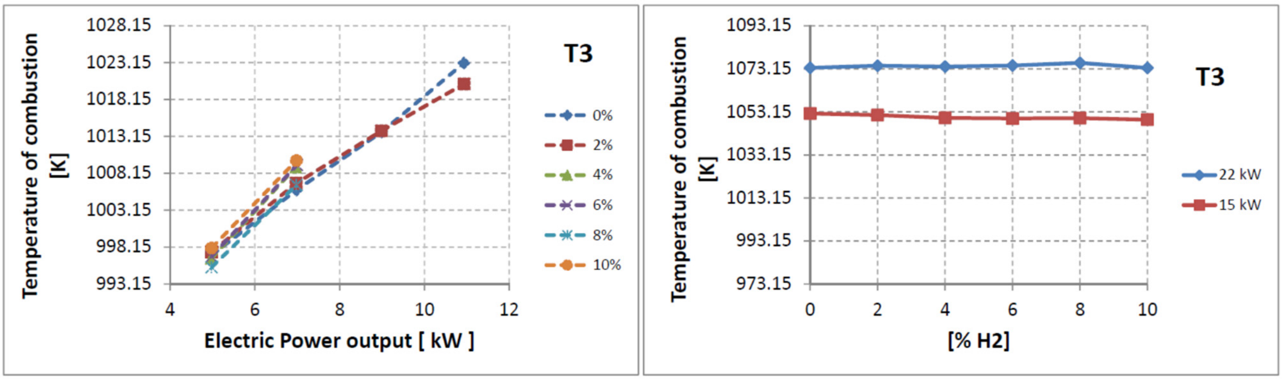


Fig.9 – Temperature of combustion in the summertime tests: T_3 vs. Electric power output with change in H_2 fraction (left side); T_3 vs. hydrogen fraction with changes in electric power output (right side)

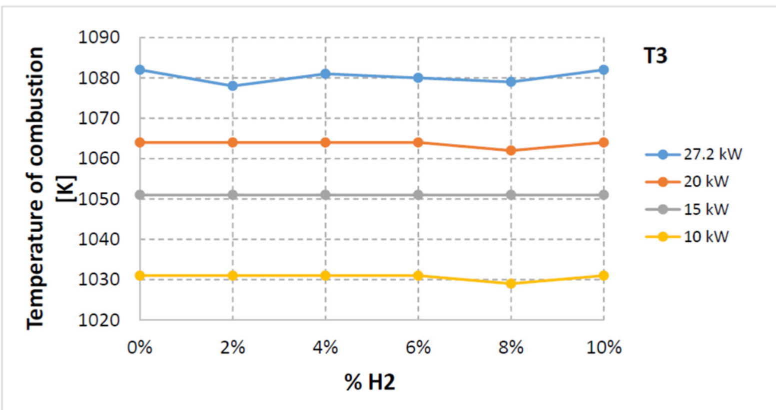


Fig.10 - T_3 vs. hydrogen fraction with changes in electric power output during the winter tests

Nevertheless, the small fluctuations of combustion temperature lead to limited gains in the energy conversion efficiency owing to a scant Carnot effect.

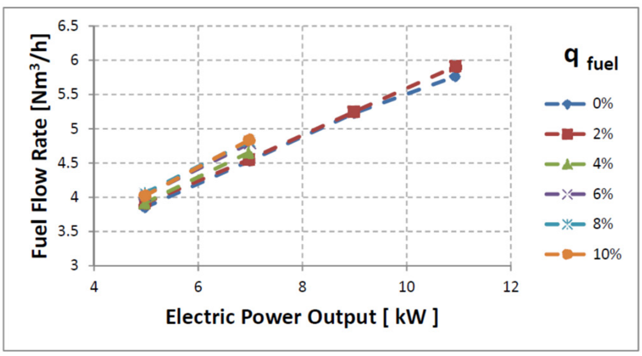
3.1 Micro gas turbine electric efficiency

In this section the MGT turbine electric efficiency values at different load conditions, with changes in hydrogen amount, have been presented. In order to evaluate that parameter, according to the Equation 9, the fuel flow rates have been registered and the mixtures LHV has been calculated by the weighted average by volume of NG and hydrogen energy densities. In Table 6 the reference Lower Heating Value of each burnt H_2NG mixture over the experimental campaign, have been reported. In the same table the theoretical relative flow rate variations have been presented as well. Basically, as the hydrogen fraction by volume increases, the mixtures volumetric LHV is lessened owing to the hydrogen lower normal density. That implies a larger flow rate to be burnt so as to release the same thermal energy produced by $1Nm^3$ of NG.

Table 6 – Lower Heating value of H_2NG blends and relative flow rate enhancement with varying hydrogen fraction up to 10% vol.

f_{H2} [% vol.]	0%	2%	4%	6%	8%	10%
LHV [MJ/Nm^3]	35.69	35.19	34.69	34.19	33.7	33.2
Relative flow rate enhancement	1	1.014	1.028	1.043	1.059	1.075

By the use of those theoretical values it is possible to verify in a first approximation how the hydrogen enrichment affects the MGT energy performance. In detail, comparing the theoretical relative flow rate enhancement to those ones related to the experimental data, thermodynamic effect on the Brayton-Joule can be assessed.



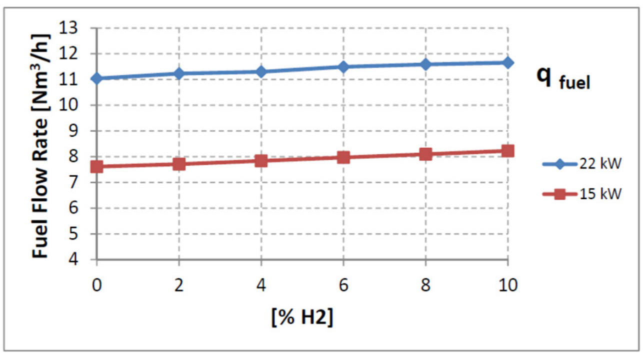


Fig.11 – Fuel flow rate during the summertime tests: q_{fuel} vs. Electric Power output with changes in H_2 fraction (left side); q_{fuel} vs. H_2 fraction with changes in electric power output (right side)

Indeed, referring to the charts in Figure 11, when the MGT generates 22 kW in the hot season and is fed by H_2NG@10% vol. the relative fuel flow rate enhancement is equal to 1.055, which is lower than the theoretical one reported in Table 6. Conversely, once the power output is 15 kW the relative fuel flow rate is 1.08 which is just a little bit higher than the theoretical value. Furthermore, it is interesting how in the power output range 5kW – 7 kW the associated relative parameters are lower than the theoretical ones, since they are equal to 1.044 and 1.063, respectively. For that reason, it is possible to conclude that the hydrogen addiction is able to improve the MGT performance especially at partial load conditions very close to the minimum one.

Using the same approach to analyse data related to the winter experimental campaign, when the H_2NG@10% vol. is used for feeding the MGT at the maximum power output (see Figure 12), the relative fuel flow rate is equal to 1.072, while at 15 kW and 10 kW the values are 1.107 and 1.103, respectively. As a consequence, the hydrogen enrichment does not lead to significant gain at rated power output, whereas it penalises the efficiency at partial loads.

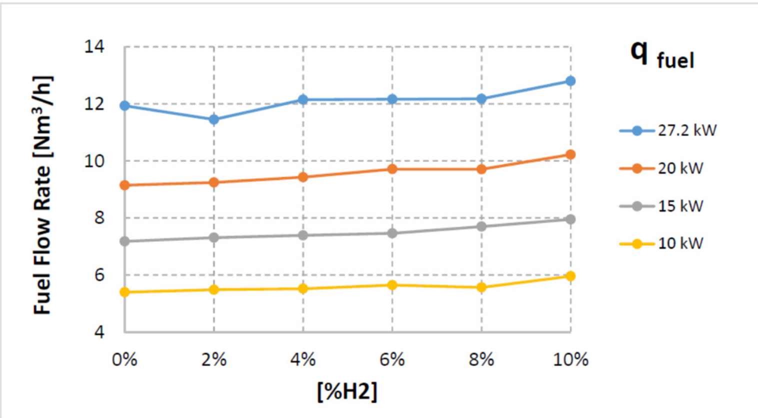


Fig.12 – Fuel flow rate during the winter tests: q_{fuel} vs. H_2 fraction with changes in Electric Power output

Thereafter, those findings can be summarized by the electric efficiency charts, as depicted in Figure 13 and Figure 14.

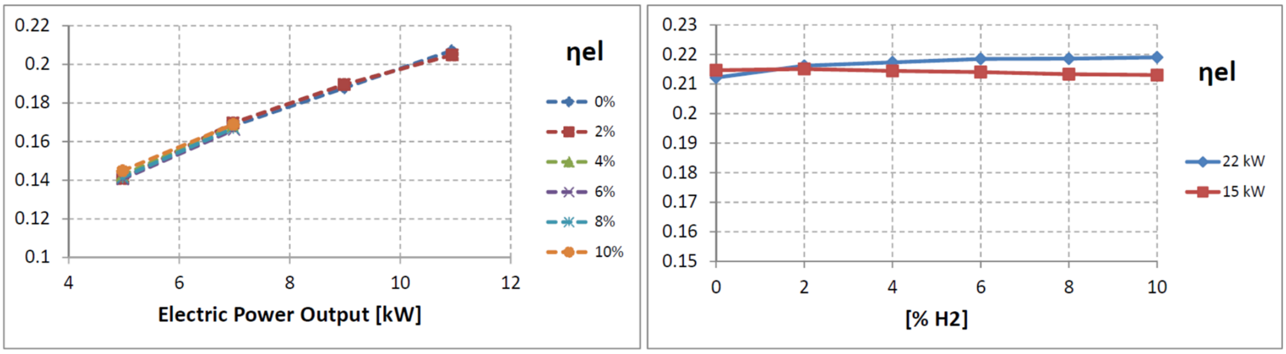


Fig.13– Electric efficiency during the summertime tests: η_{el} vs. Electric Power output with changes in H_2 fraction (left side); η_{el} vs. H_2 fraction with changes in electric power output (right side)

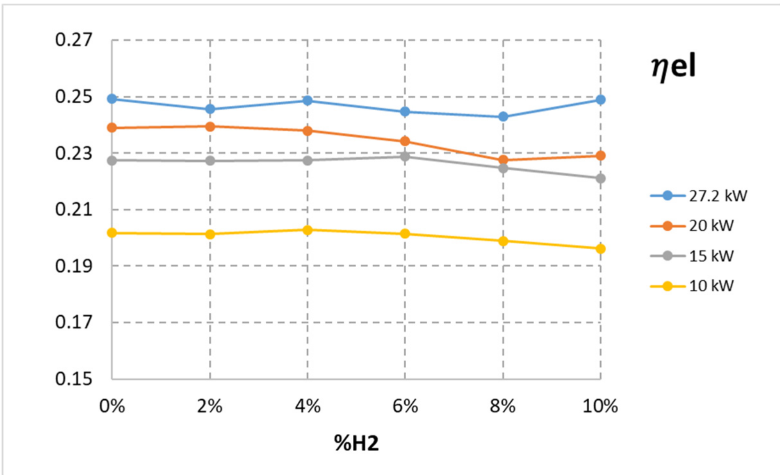


Fig.14– Electric efficiency during the winter tests: η_{el} vs. H_2 fraction with changes in Electric Power output

From data it emerges that, in the hot season the electric efficiency marginal gain, due to the hydrogen enrichment equal to 10% vol., is limited to 1 percentage point, starting from 0.21 up to 0.22 at maximum power. Moreover, when the MGT operates in the range 5kW – 7kW that enhancement does not exceed 0.5 percentage points, while at 50% of rated power output the electric efficiency slightly lessens as more hydrogen is added.

On the contrary, referring to the winter data, it can be stated that at maximum power output the electric efficiency values are substantially unaffected by the hydrogen addition and their fluctuations can be caused by the different outdoor environmental conditions during data collection. In addition, significant energy penalties occur at partial loads beyond the hydrogen threshold value equal to 6% vol.

In the end, it can be stated that the use of H_2NG blends favours the MGT mechanical performance, especially in the hot season for offsetting the environmental derating instead of utilizing it when climate conditions are colder.

3.2 MGT heat recovery efficiency

Since the Capstone C30 has been installed in the HVAC plant for CHP application, investigating on the hydrogen effect on thermal power output it has been necessary. So that, the main results and implications have presented and discussed here.

As reported in Section 2, the thermal power output in the selected operating conditions has been measured by means of a thermal counter downstream the MGT. Thus, according to the Equation 10 the heat recovery efficiency has been calculated having fixed the water flow rate within the hydraulic loop equal to 2.6 m^3/h and the set point temperature for supply water equal to 70 °C. Owing to the thermal-tracking mode adoption, the temperature difference of water changes as the MGT rotational speed increases, generating thereby more heat. Consequently, given that the circulation pump is not able to provide a variable flow and the exhaust gas mass flow rate increases as the turbine shaft accelerates, the liquid to gas heat exchanger effectiveness changes as well.

Having said this, the Figure 15 reports the thermal power output measured in the summertime tests. It shows how the hydrogen addiction slightly affects the recovered heat amount once the MGT runs underneath 11 kW. In all other cases, the thermal power output can be practically considered a constant value

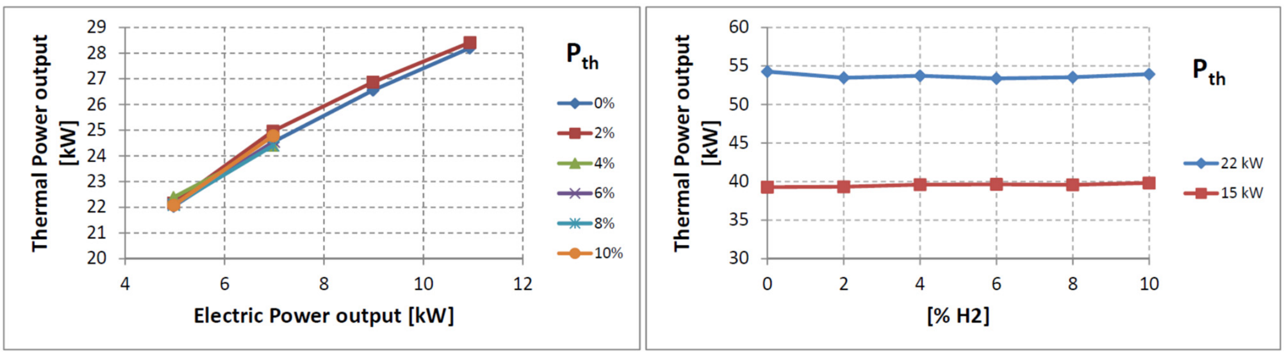


Fig.15– Thermal Power output during the summertime tests: P_{th} vs. Electric Power output with changes in in H_2 fraction (left side); P_{th} vs. H_2 fraction with changes in electric power output (right side)

Notwithstanding, due to the fact that the H_2NG blend consumptions are higher relative to the NG case but lower than the theoretical values, the heat recovery efficiency reaches the maximum of 0.642 at 5 kW with H_2NG@10% vol. Up to 11 kW the heat recovery efficiency trends as a function of hydrogen are quite undefined. However, it can be noticed that almost 1.5 percentage points of marginal gain has been accomplished at 22 kW once more hydrogen is added, while a modest lowering has been registered at 15 kW and H_2NG@8% vol.

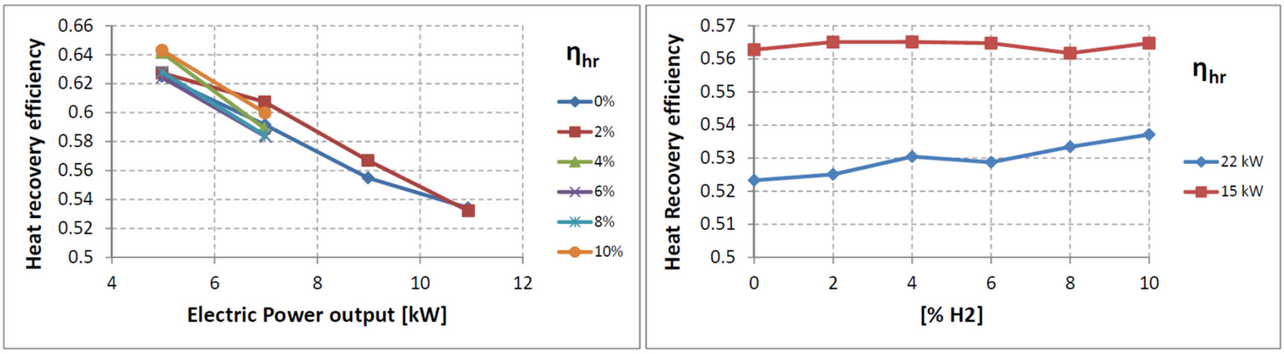


Fig.16– Heat Recovery efficiency during the summertime tests: η_{hr} vs. Electric Power output with changes in in H_2 fraction (left side); η_{hr} vs. H_2 fraction with changes in electric power output (right side)

Referring to the winter data, the recovered thermal power value depends on the MGT load conditions and it is almost constant up to the hydrogen fraction equal to 6% vol., as shown in the right side of Figure 17. Despite of that trend, the heat recovery efficiency tends to shrink on average of 3 percentage points less in all tested loads exception for 10 kW. In that case the highest efficiency gain caused by the hydrogen addiction is approximately 3 percentage points more.

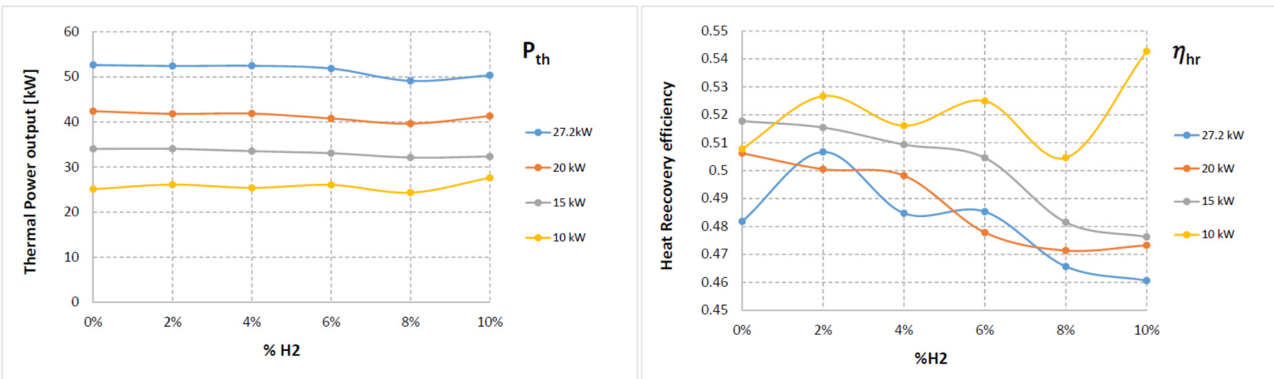


Fig.17– Thermal power output (left side) and Heat recovery efficiency (right side) during the winter tests: P_{th} and η_{hr} vs. H_2 fraction with changes in Electric Power output

Finally, by summing the electric and the heat recovery efficiency, the First Law efficiency can be evaluated.

From data analysis, it can be stated that the highest First Law efficiency (i.e. 0.788) can be attained at 5 kW and 15 kW along with a hydrogen fraction equal to 10% vol. in the hot season. Thus, starting from the minimum load, for growing MGT power output the First Law efficiency goes down as long as the CHP attains 15 kW. Anyway, at maximum power output in the hot season 10% vol. of hydrogen allows to get an efficiency very close to 0.76.

Conversely, in the winter period, the MGT at full load shows a First Law efficiency which does not exceed 0.685 for any hydrogen fraction while when it runs at 15 kW the highest achievable efficiency is limited to 0.715.

To sum up, in this section has been demonstrated that the H_2NG mixtures are more suitable from the energy point of view for feeding MGTs suffering the environmental derating over the hot season, even if the electric power generation is strongly reduced.

4-Environmental performance assessment

Any combustion process produces several pollutants emission depending on the fuel typology. Indeed, liquid and solid fuels are usually responsible of sulphur oxides, unburned hydrocarbons and particulate, while gaseous fuels such as NG and H_2NG mixtures show only small traces of those compounds, especially when they burn in MGTs. For those reasons, only NO_x, CO and CO_2 emissions have been detected in this work to characterise the commercial MGT environmental performance running over one-year period, in actual operating conditions. Additionally, the most effective method to properly allocate the specific pollutants emissions for such CHP systems has been presented and discussed.

Generally, the most common national regulations and technical standards on the air quality call for threshold limits based on pollutants concentration either by mass or by volume. For instance, those concentrations are typically referred to the dry exhaust gas and they can be also expressed by mg/Nm3 or ppm. Thus, in order to avoid counterfeit measurements, the pollutants concentrations are normalised by a reference oxygen content within the dry exhaust gas depending on the conversion system technology. In example, when boilers are considered the reference oxygen value is equal to 3% vol. while for ICEs (Internal Combustion Engines) and MGTs is equal to 5% vol. and 15% vol., respectively. Nevertheless, the concentration does not provide further information about the peculiar machines environmental performance accounting for the fuel use, the system size and the useful output typology.

Hence, to overcome those issues, it is possible to convert and allocate the pollutants concentration by using two different approaches: the first one is the so-called input-based method, which refers the pollutants emission to the unit of input energy; the second one is the output-based, where the emissions are typically associated to the useful outputs, taking also into account the energy conversion efficiency. According to the recent literature [11], the emitted pollutant mass per unit of input energy is indicted by λp (i.e. mg/kWh LHV), whereas for the mass per unit of output energy δ is frequently used. However, it is possible to alternatively apply both approaches by means of Equation 18, once the conversion efficiency related to the specified output has been fixed:

$$\delta = \frac{\lambda_p}{\eta} \tag{18}$$

It is important to point out that the second methodology is generally more effective, since it provides the direct comparison between the useful output and the environmental performance, showing also the role of conversion efficiency to lessen further the actual pollutants emission.

Having said this, when CHPs are considered for that analysis, the main issue is how allocating properly their specific emissions since they have two different outputs: electric energy and heat. For that purpose, several methods are available in literature, such as the Method of avoided electric production, the Method of the equivalent heat generator, the Method of the avoided heat generator and, the Method of Ecabert.

The two latter allocation methodology refer basically to the CHPs electric output, once thy are considered as prime movers. In order to account also for the heat generation, the electric specific emission can be reduced by a correction factor depending on the adopted methodology. So that, the corrected specific emission reads fundamentally as:

$$\delta_{corrected} = \delta_{el,CHP} - \delta_{avd} \tag{19}$$

As regards the Avoided heat generator method the correction factor can be calculated by:

$$\delta_{avd} = \frac{\lambda'_{th}}{\eta'_{th}} \cdot \frac{\eta_{hr}}{\eta_{el}} \tag{20}$$

Where λ'_{th} represents the specific emission related to the avoided heat generator, η'th is its thermal efficiency, η_{el} and η_{hr} are the CHP electric and heat recovery efficiencies, respectively. Conversely, for the Ecabert method the correction factor can be calculated by the equation (21):

$$\delta_{avd} = \delta_{CHP} \frac{\eta_{hr}}{\eta'_{th}} \tag{21}$$

Here, δ_{CHP} denotes the CHP specific emission value as a prime mover and it is not required to specify the environmental performance of the avoided heat generation.

It is worth of noticing that first method leads to reliable results especially for those CHPs which are ICE-based, while is not completely suitable for MGTs, since in some cases it provides negative values for the specific emissions. For that reason, in this section, the corrected specific emissions by the Ecabert method related to the Capstone C30 have been presented and discussed. Moreover, the reference value for the boiler efficiency has been assumed equal to 0.9, while for the CHP heat recovery efficiency the outcomes of this experimental campaign have been used for all calculations.

In order to better understand the MGT behaviour from the environmental point of view, in this subsection have been firstly reported the relative equivalence ratio and the normalized pollutants concentration according to the Italian standards. In detail, Figure 18 and Figure 4 depict the MGT combustion conditions as a function of electric load with changes in hydrogen fraction over the summertime and winter season, respectively.

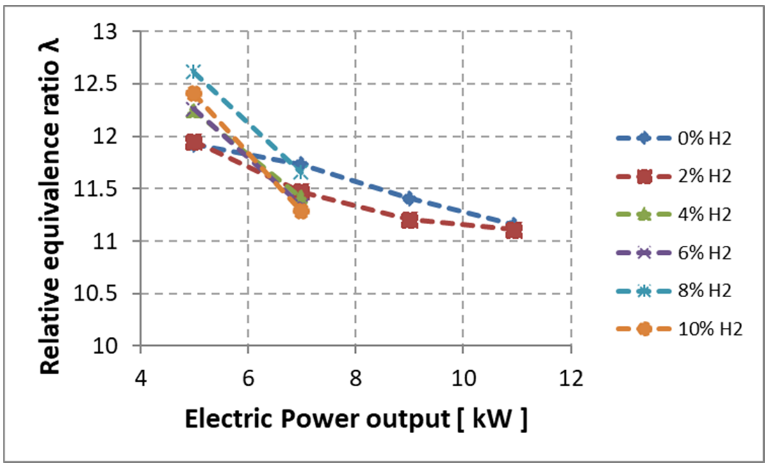


Fig.18– Relative equivalence ratio vs. electric power output with changes in H2 fraction (hot season)

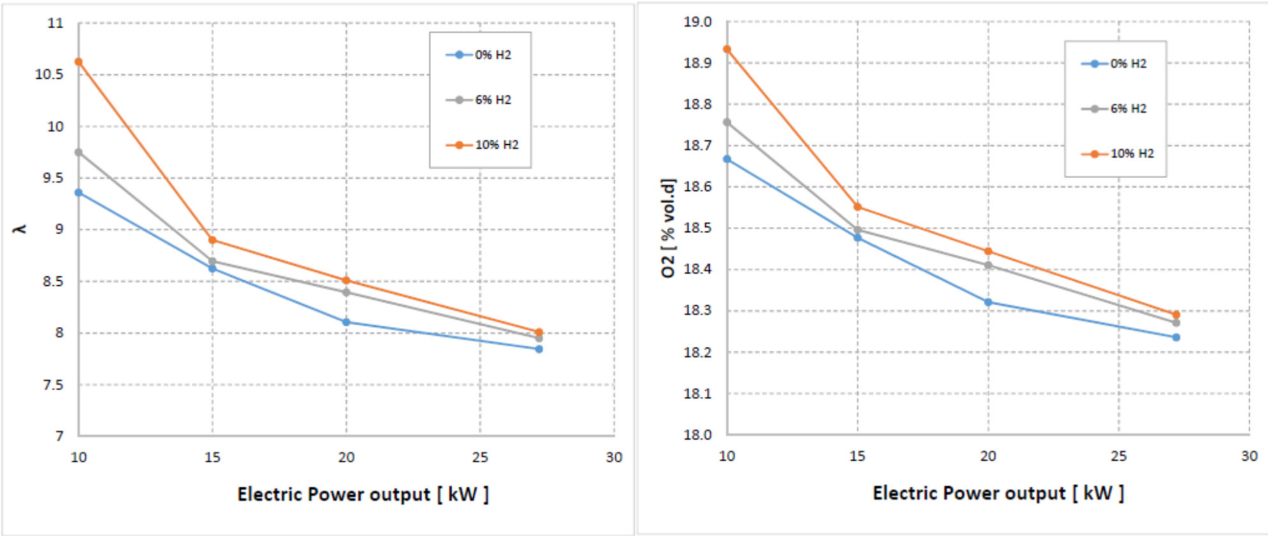


Fig.19– Relative equivalence ratio vs. electric power output with changes in H_2 fraction (left side); Oxygen concentration in exhaust gas vs. electric power output with changes in H_2 fraction (right side). Both charts refer to the cold season.

Looking at data it emerges how the lower the electric load, the higher the relative equivalence ratio is. In addition, when more hydrogen is added to the blend, the MGT combustion conditions tend to become leaner, exception for partial load values beneath 7 kW and hydrogen fraction equal to 10% vol. in the hot season. In addition, comparing data of Figure 18 with those reported in Figure B it can be noticed how in the cold season the combustion conditions are richer when MGT generates the same power output, Indeed, the relative equivalence ratio is very close to 9.5 at 10 kW over the winter campaign, whereas is higher than 11 at the same power output in the hot season. Therefore, it is clear how pollutants emissions, in terms of concentration and specific values, are strongly affected by the automatic combustion setup that the MGT electronic control unit imposes.

As regards the NO_x formation, Figure 20 and Figure 21 show what is the trend when the MGT is operating at different electric load and it is fuelled with H_2NG mixtures. Generally, it is possible to state that starting from the minimum load (i.e. 5 kW) the NO_x concentration tends to increase as the power output enhances up to 15 kW. In that load condition the NO_x concentration gets the maximum value in the hot season as much as in the cold one. Yet, when pure NG is used the worst performance (i.e. 182 mg/Nm3@15% O2) has been registered in the hot season, while in the winter this value is limited to 160 mg/Nm3@15% O_2. Thus, the hydrogen addition contributes to increase the NO_x formation as it was expected, owing to the higher flame temperature. Notwithstanding, the leaner combustion caused by the hydrogen use mitigates that drawback in all load conditions. Additionally, when MGT runs beyond 15 kW the NO_x concentrations are very low and they do not exceed 20 mg/Nm3@15% O_2 with the hydrogen as well.

The carbon monoxide concentration has a different trend as reported in Figure 20 and Figure 21. Basically, it shrinks almost linearly as the MGT load is higher and the hydrogen influences positively the concentration reduction. Looking at the figures it is possible to notice that CO is approximately equal to zero when the MGT runs at high loads, close to the rated one. Referring to the pure NG setup in the hot season, the CO concentration at 10 kW of power output is almost five times lower than the cold season one. Adding more hydrogen up to 10% vol. the CO concentration can be reduced of 27% at 15 kW of power output in the hot season. Conversely, during the winter test that reduction can accomplish 50% when the MGT operates at 10 kW. Therefore, all of data show that the hydrogen addition is very useful in partial load conditions, playing the role of a combustion stimulator. However, the carbon monoxide lowering implies the complete oxidation of carbon atoms to CO_2.

Nevertheless, the carbon dioxide molar fraction lessens with varying the hydrogen amount in the blend, but it tends to grow as the MGT operates close to the rated power output. Moreover, from charts it is possible to notice how the CO_2 concentrations are lower in the winter tests than in the summer ones and that reduction can achieve more than 55%.

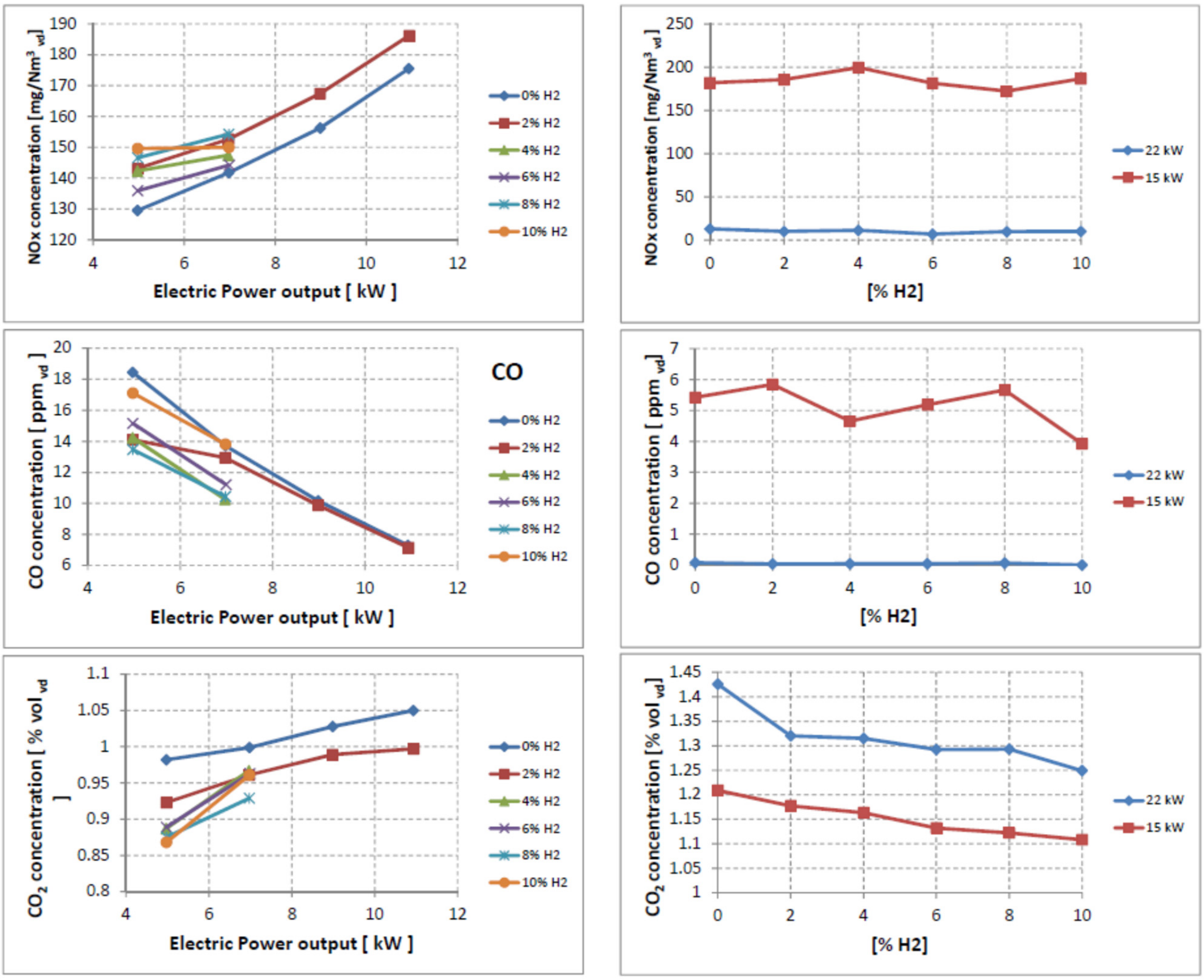


Fig.20– NO_x, CO and CO_2 concentrations vs. electric power output with changes in H_2 fraction (hot season)

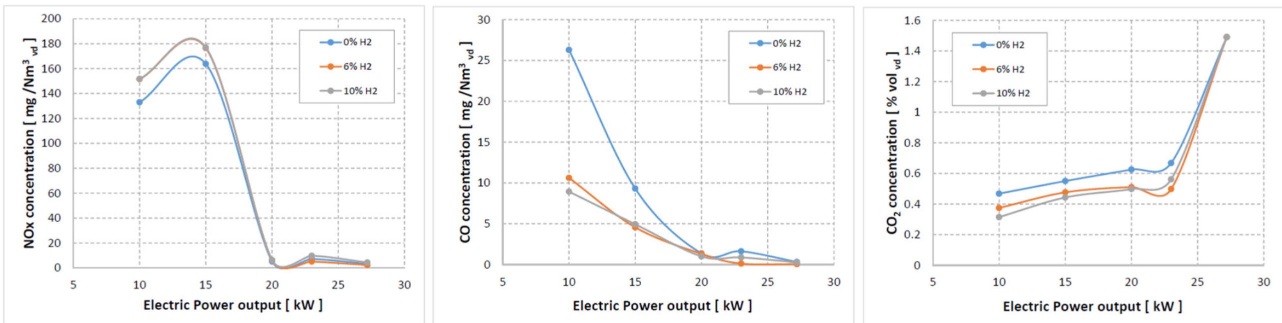


Fig.21– NO_x, CO and CO_2 concentrations vs. electric power output with changes in H_2 fraction (cold season)

That unexpected behaviour is likely due to the MGT environmental derating since the fuel consumption is higher in the hot season and consequently the electric efficiency is lower. Indeed, the relative equivalence ratio in the summer tests is generally higher than in the winter ones. So that, the CO_2 concentrations should have been more diluted since the combustion conditions are leaner. For those reasons, the specific emissions calculation by the Ecabert method has been applied in order to better understand those findings.

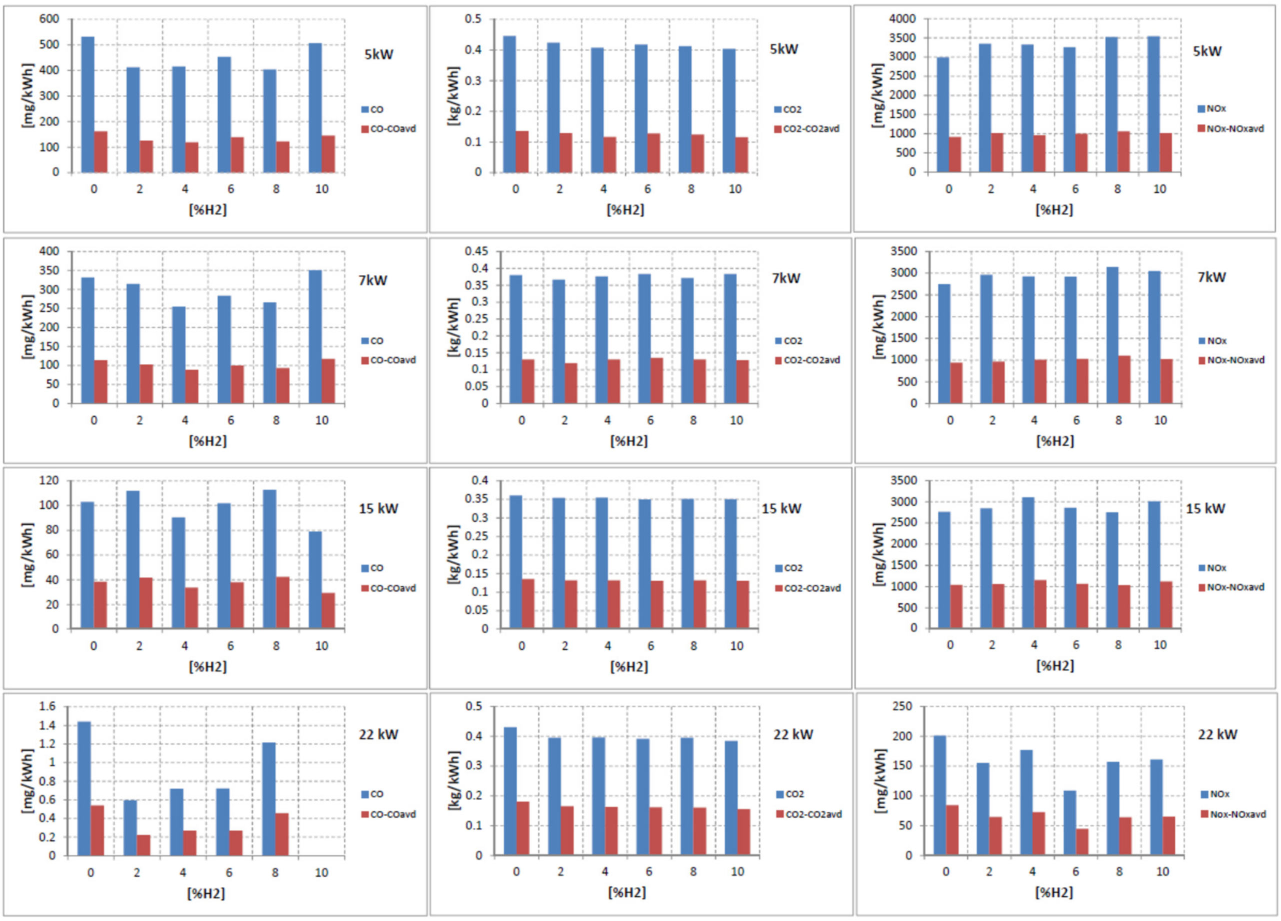


Fig.22- NO_x, CO and CO_2 specific emissions comparison between the traditional method and the Ecabert one. The specific emissions have been plotted vs. hydrogen fraction and sorted by the electrical power output in the hot season.

Figure 22 outlines in a systemic overview the CO, CO_2 and NO_x specific emissions referred to the electric power output only, and those related to the Ecabert method implementation hailing from the summer experimental campaign. It is important to highlight that the corrected specific emissions are affected by the CHP heat recovery efficiency value. Indeed, according to the Equation 14, the larger the heat recovery efficiency, the larger the correction factor is. Looking at charts related to the 22 kW setup it emerges that CO corrected specific emission can be reduced by 50.1% when hydrogen enrichment is equal to 6% vol.; the CO_2 values diminishes of 13.8% with hydrogen fraction equal to 10% vol., while the NO_x ones attain the maximum reduction of 46.7% using H_2NG@6% vol. Additionally, when the MGT operates at 15 kW the hydrogen effect on the NOx corrected emission consists of increasing that value of 10% compared to the pure NG feeding. However, it is possible to state that generally the hydrogen enrichment provides beneficial effects on MGT partial loads and, when the H2NG@10% vol. is used, the NOx corrected emissions enhancement is limited to 10% approximately.

On the contrary, referring to the same setup over the winter campaign those emissions rise up to 23% more, as reported in Figure 24. The same trend can be also registered at maximum power output, i.e. 27.2 kW, where the corrected specific emissions of nitrogen oxides increases of 23%, once the blended hydrogen is equal to 10% vol. (see Figure 23).

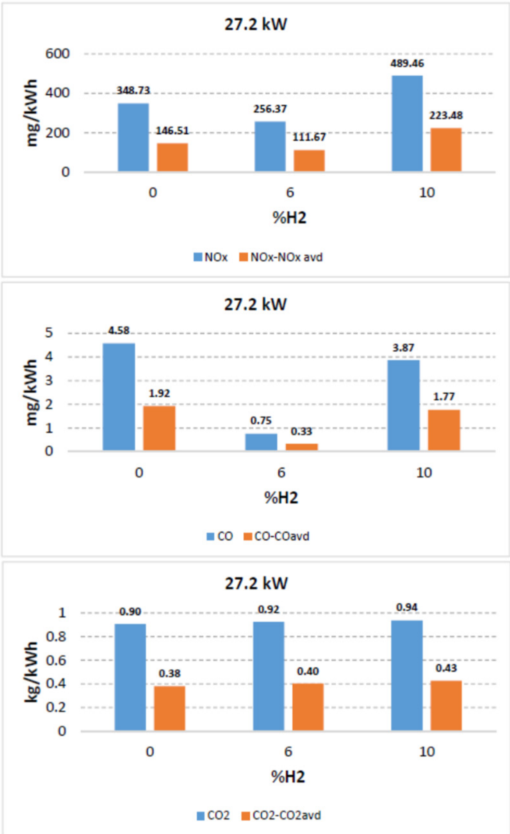


Fig.23- NO_x, CO and CO_2 specific emissions comparison between the traditional method and the Ecabert one. The specific emissions have been plotted vs. hydrogen fraction at 27.2 kW of electrical power output, in the cold season.

It is noteworthy how the CO_2 corrected emissions value corresponding to that setup, tends to slightly increase as the hydrogen fraction changes. This is due to the fact that the exhaust gas flow rate, hailing from the H_2NG burning, is higher than the NG one, although the carbon dioxide volumetric concentration goes down as the hydrogen is added. Moreover, owing to the heat recovery efficiency reduction, the final value of corrected specific emissions is not lessened as it was expected. That entails the carbon dioxide concentration variation is not able to offset the volumetric effect caused by the enhanced exhaust gas flow rate.

Finally, the NO_x, CO and CO_2 corrected emissions sorted by MGT electric power output related to the cold season have been depicted in Figure 24, Figure 25 and Figure 26, respectively.

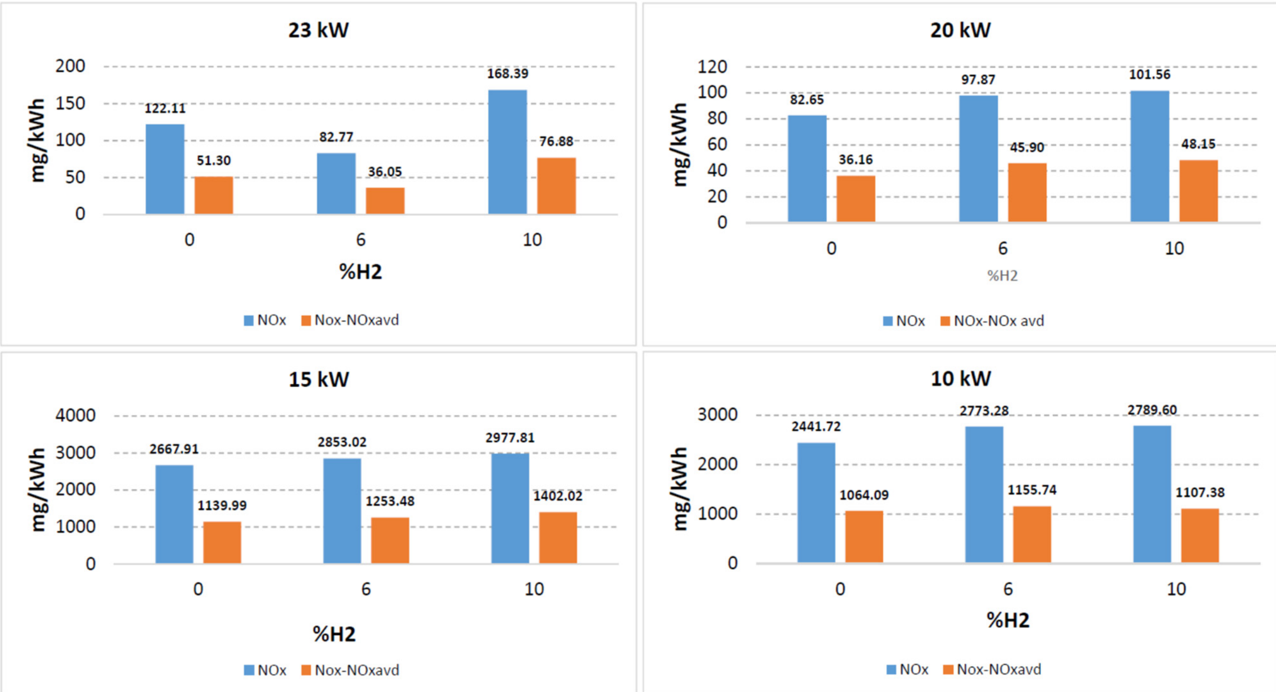


Fig.24 - NO_x specific emissions comparison between the traditional method and the Ecabert one. The specific emissions have been plotted vs. hydrogen fraction with changes in electrical power output, in the cold season.

Referring to the Figure 24, it is remarkable how at 15 kW of power output the NO_x specific emissions associated to the NG set up are higher than those related to the same operating conditions in the hot season. Indeed, in the first case the specific emissions are equal to 1139.99 mg/kWh, whereas they are 1035 mg/kWh in the second case. Furthermore, the hydrogen addiction up to 10% vol. leads to 1402 mg/kWh and 1120.8 mg/kWh. respectively.

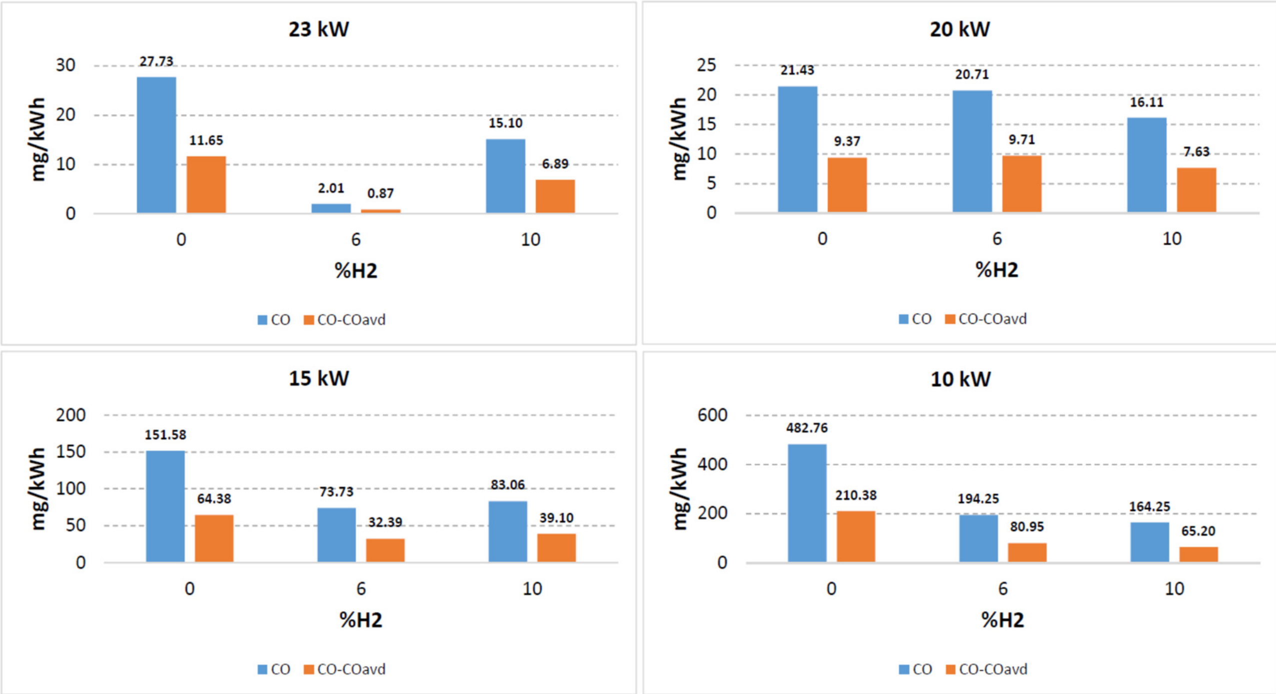


Fig.25- CO specific emissions comparison between the traditional method and the Ecabert one. The specific emissions have been plotted vs. hydrogen fraction with changes in electrical power output, in the cold season.

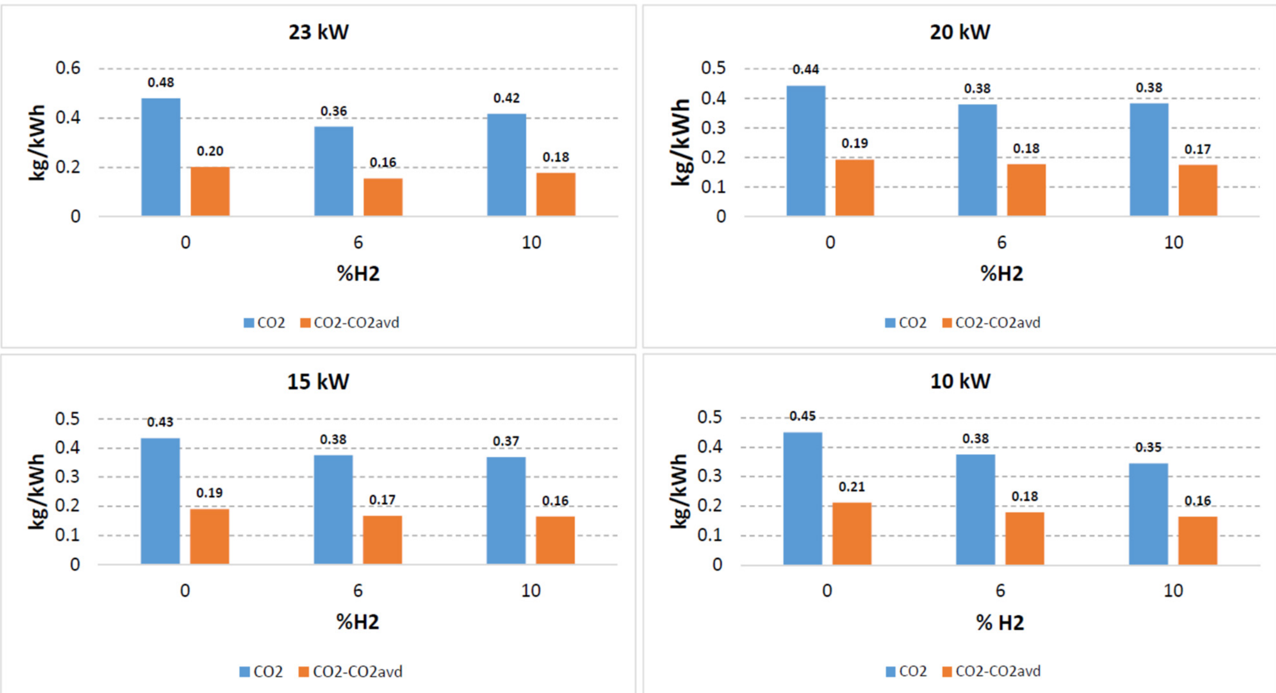


Fig.26– CO_2 specific emissions comparison between the traditional method and the Ecabert one. The specific emissions have been plotted vs. hydrogen fraction with changes in electrical power output, in the cold season.

As regards the CO and CO_2 corrected emissions the highest percentage reduction caused by the hydrogen enrichment can be registered at 15 kW of power output in the winter tests and are equal to 64.6% and 18.75%, respectively.

Having said this, it is possible to conclude that the hydrogen enriched natural gas blends are quite suitable for commercial MGTs to enhance their environmental performance at partial load and in the hot season.

5.Conclusions

This paper deals with the energy and environmental characterisation of a commercial MGT package for CHP applications when it is fuelled with H_2NG mixtures up to 10% vol. of hydrogen fraction. To do so, an experimental campaign, aiming at measuring the main operating parameters, has been carried out over one-year period. In addition, all of the performance indicators have been presented and discussed distinguishing the different seasonal behaviour of such well-proven device. The most significant findings of this research project can be summed up as follows:

- the MGT net electric power output is strongly penalised by the outdoor environmental conditions; indeed, during the summer tests the maximum achievable load has been equal to 22 kW, while in the winter it has been 27.2 kW.
- due to the environmental derating the MGT control unit increases the rotational speed to keep under control the power output. The hydrogen enrichment partially offsets the outdoor temperature negative effect especially in the summertime. On the contrary, at partial load, such as 15 kW, the hydrogen effect, when its fraction ranges in 4%vol. - 8% vol., is more visible since the rotational speed tends to increase although the compressor inlet temperatures are lower; during the cold season the MGT behaviour is quite similar, but the hydrogen fraction range affecting the cycle is extended up to 10% vol.;
- as the rotational speed increases, the air flow rate grows influencing the MGT air to fuel ratio and the combustion temperature. From data it emerges that the combustion temperature fluctuations from summer to winter are limited within the span 1073 K – 1081 K. That implies the weak influence of hydrogen enrichment, since the higher dilution contributes to cool down the flame temperature;
- as it was expected, the fuel flow rate enhances as the hydrogen fraction in blend is higher. Yet, in some operating conditions the relative enhancement is higher than the theoretical one. In detail,

referring to the hot season, when the power output ranges in 5kW – 7 kW those values for H_2NG@10% vol. parameters are equal to 1.044 and 1.063, respectively. Conversely, during the winter campaign, the hydrogen enrichment does not lead to significant gain at rated power output, whereas it penalises the efficiency at partial loads.

- Regarding the electric efficiency, the highest marginal gain, i.e. 1 percentage point more, has been registered in the hot season at maximum load (22 kW) and with H_2NG@10% vol. In partial load conditions, very close to the minimum power output, that gain does not exceed 0.5 percentage points. Thus, in the cold season, when MGT runs at rated power the hydrogen addiction is basically unaffecting. Nevertheless, at partial loads, beyond the threshold value of 6% vol. of hydrogen fraction, energy penalties have been registered.
- The heat recovery efficiency shows a marginal gain of 0.5 percentage point only at 22 kW in the hot season and using 10% vol. of hydrogen, while in all other cases it is penalised by hydrogen addiction. A quite similar trend has been registered in the cold season.
- NO_x emissions are not constant at each electrical power but drop down when the gas turbine reach higher electrical load. CO and CO_2 emissions have been significantly reduced with the addition of hydrogen unlike those of NO_x which have slightly increased.

In the end, it is possible to conclude that the hydrogen addiction for feeding MGTs leads to lower energy gains compared to ICE technologies. However, the new electro-fuel can be useful to partially mitigate the derating effects due to the outdoor conditions' changes. As a consequence, once a national policy for power to gas deployment is implemented, the renewable hydrogen production could be used effectively to feed also those devices. In such a way, the electricity peak production hailing from RES, which typically occurs in the summertime, it could be exploited efficiently, contributing to mitigate the renewable capacity firming issues as well. Further developments of this project consist of testing on field that MGT once it is fuelled with higher hydrogen enriched blends up to 25% vol., since that fraction has been considered by the authors the limit value for optimising the carbon avoidance cost.

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References

[1] European Commission, "2018-2020"Climate and Energy Package" https://ec.europa.eu

[2] Ministry of Economic Development and the Ministry of the Environment and Protection of the Territory of the Sea, "National Energy Strategy (SEN)", 10 November 2017

[3]European Commission, "2030 Climate and Energy Framework", available at https://ec.europa.eu/clima/policies/strategies/2030_en

[4] European power to gas, "Power to gas", available at http://www.europeanpowertogas.com/about/

[5] London research international, "Power-to-Gas Energy Storage and Transportation", February 2015

[6]European Commission, "2020 Climate and Energy Package" https://ec.europa.eu

[7]Dan Li, Roger A. Dougal, Eshwarprasad Thirunavukarasu, A. Ouroua. Variable speed operation of turbogenerators to improve partial load efficiency.

[8] Penzo, Marco. "Gas microturbines, characteristics and use as energy sources for residential use." 2012.

[9]Therkelsen P., Mauzey J., McDonell V. G., and Samuelsen S., 2006, “Evaluation of a Low Emission Gas Turbine Operated on Hydrogen,” ASME Turbo Expo 2006: Power for Land, Sea and Air, Asme, pp. 557

[10]Nicola Constrisciani, ENEA, "Hydromethane sustainability and market opportunities", April 2009.

[11]Michele Bianchi,Lisa Branchini,Francesco Melino."Power to gas: renewable energy for the gas network", in Info build energia, Int J Hydrogen Energy, 43 (2014), pp. 16652-16666.

[12] Ruscica G, Badami M, Portoraro A, Mura M, Di Pietra B of the Ministry of Economic Development and ENEA, "Electrical system research", Report RSE / 2009.

[13] V.D. Sarlia, A.D. BenedettobLaminar burning velocity of hydrogen–methane air premixed flames Int J Hydrogen Energy, 32 (2007), pp. 637-646.

[14] B.A. Imteyaz, M.A. Nemitallah, A.A. Abdelhafez, M.A. HabibCombustion behavior and stability map of hydrogen-enriched oxy-methane premixed flames in a model gas turbine combustor Int J Hydrogen Energy, 43 (2018), pp. 16652-16666.

[15] K.K.J.R. Dinesh, K.H. Luo, M.P. Kirkpatrick, W. Malalasekera,Burning syngas in a high swirl burner: effects of fuel composition Int J Hydrogen Energy, 38 (2013), pp. 9028-9042.

[16] P. Rajpara, R. Shah, J. Banerjee, Effect of hydrogen addition on combustion and emission characteristics of methane fuelled upward swirl can combustor Int J Hydrogen Energy, 43 (2018).

[17] U.S. Department of Energy, "Combined Heat and Power Technology Fact Sheet Series - Gas Turbines", 2016.

[18] L. Wei, P. GengA , review on natural gas/diesel dual fuel combustion, emissions and performance Fuel Process Technol, 142 (2016), pp. 264-278.

[19] Menter F. R., Kuntz M., and Langtry R. Ten years of industrial experience with the SST turbulence model Turbulence, heat and 2003.

[20] Pontecorvo Antonino, "Study of a gas microturbine for cogeneration and hybrid systems", PhD Thesis, University of Naples, Chapter 1.

[21] Penzo Marco, “Gas microturbines characteristics and use as energy sources for residential use” Volume 160, 1 October 2015, Pages 750-759.

[22] Elisa Valentini, “Energy-environmental performance of cogeneration systems advanced and analysis of application cases ", PhD Thesis, University of Bologna, 2014, pp 38-51

[23] W. Kreuter, H. Hofmann. «Electrolysis: the important energy transformers in a world of sustainable energy.» International Journey of Hydrogen Energy, 1998: 661-666.

[24] PavlosDimitriou, TakuTsujimura, YasumasaSuzuki ,Hydrogen-diesel dual-fuel engine optimization for CHP systems Energy Volume 160, 1 October 2018, Pages 740-752.

[25] Carlo Villante, Antonino Genovese. «Hydromethane: A bridge towards the hydrogen economy oran unsustainable promise» International Journey of Hydrogen Energy 37 (2012): 11541-11548.

[26] Schefer, R.W., Wicksall, D.M., Agrawal, A.K, Combustion of hydrogen-enriched methane in a lean premixed swirl-stabilized burner, (2012) Proceedings of the Combustion Institute29(1), pp. 843-851

[27] Xiao Sen, HouXin, ZhangZhun ,QingHu , Experimental and numerical study on engine fueled with different fractions of natural gas–carbon dioxide-hydrogen blends International Journal of Hydrogen Energy, Volume 44, Issue 11, 26 February 2019, Pages 5599-5606.

[28] Riccio G., Marini A., and Martelli F., 2009, "Numerical investigations of gas turbine combustion chamber Hydrogen fired" ISABE-2009-1112.

[29] Marini A., Cappelletti A., Riccio G., and Martelli F., 2010, "Cfd re-design of a gas turbine can-type combustion chamber hydrogen fired" ECCOMAS CFD 2010, June 14-17, Lisbon, Portugal.

[30] Hsin-Yi Shih, Chi-Rong Liu. "A computational study on the combustion of hydrogen / methane blended fuels for a micro gas turbine." International Journey of Hydrogen Energy 39, 2014: 15103-15115.

[31] Cadorin M., Pinelli M., Vaccari A., Calabria R., Chiariello F., Massoli P., and Bianchi E., 2012, "Analysis of a Micro Gas Turbine Fed by Natural Gas and Synthesis Gas: MGT Test Bench and Combustor CFD Analysis" Journal of Engineering for Gas Turbines and Power, 134(7), p. 071401.

[32] panel, A.H.Mosaffaa, Z.Ghaffarpourb, L.Garousi Farshic ,Thermoeconomic assessment of a novel integrated CHP system incorporating solar energy based biogas-steam reformer with methanol and hydrogen productionSolar Energy Volume 178, 15 January 2019, Pages 1-16.

[33] S.Ouchikha, M.S.Lounicia, L.Tarabetb, K.LoubarcM.Tazeroutc, Effect of natural gas enrichment with hydrogen on combustion characteristics of a dual fuel diesel engine International Journal of Hydrogen Energy, 142 (2016), pp. 264-278.

[34] BingGe,YongbinJi,ZilaiZhang,ShushengZang,YinshenTian,HaiYubMingminChen ,GuangyunJiao , DongfangZhang ,Experiment study on the combustion performance of hydrogen-enriched natural gas in a DLE burner, International Journal of Hydrogen Energy Volume 44, Issue 26, 21 May 2019, Pages 14023.

[35] Xu, J., Zhang, X., Zheng, S.-Z., Liu, J.-H., Experimental on combustion stability in an engine fueled with biogas and hydrogen blends, Neiranji Xuebao/Transactions of CSICE (Chinese Society for Internal Combustion Engines)29(3), pp. 236-240.

[36] Y. Wang, X. Zhang, C. Li, Experimental and modeling study of performance and emissions of SI engine fueled by natural gas–hydrogen mixtures Int J Hydrogen Energy, 7 (2010), pp. 2680-2683.

[37] Halter, F., Chauveau, C., Gökalp, Characterization of the effects of hydrogen addition in premixed methane/air flames, International Journal of Hydrogen Energy,(2008),32(13), pp. 2585-2592.

[38] Hsin-Yi Shih, Chi-Rong Liu. "A computational study on the combustion of hydrogen / methane blended fuels for a micro gas turbine." International Journey of Hydrogen Energy 39, 2014: 15103-15115.

[39] C. Altantzis, C.E. Frouzakis, A.G. Tomboulides Detailed numerical simulations of intrinsically unstable two-dimensional planar lean premixed hydrogen/air flames Proc Combust Inst, 1 (2011), pp. 1261-1268.

[40] Zhunqing Hu, Xin Zhang Study on laminar combustion characteristic of low calorific value gas blended with hydrogen in a constant volume combustion bomb,Int J Hydrogen Energy, 44 (2019), pp. 487-493.

[41] Chiesa, P., Lozza, G., Mazzocchi, Using hydrogen as gas turbine fuel, Journal of Engineering for Gas Turbines and Power,2005,127(1), pp. 73-80.

[42] di Gaeta, A., Reale, Massoli, P, A dynamic model of a 100 kW micro gas turbine fuelled with natural gas and hydrogen blends and its application in a hybrid energy grid, Energy ,2017 ,129, pp. 299-320.

[43] Zhunqing Hu, Xin Zhang, Experimental study on performance and emissions of engine fueled with lower heat value gas-hydrogen mixtures Int J Hydrogen Energy, 37 (2012), pp. 1080-1083.

[44] Cappelletti, A., Martelli, F., Bianchi, E., Trifoni, Numerical redesign of 100kw MGT combustor for 100% H2 fueling,2014, Energy Procedia, 45, pp. 1412-1421.

[45] Lee M. C., Seo S. Bin, Chung J. H., Kim S. M., Joo Y. J., and Ahn D. H., 2010, “Gas turbine combustion performance test of hydrogen and carbon monoxide synthetic gas,” Fuel, 89(7), pp. 1485-1491.

[46] Livio de Santoli, Gianluigi Lo Basso, Daniele Bruschi. "Preliminary Experimental Analysis of a CHP Hydro methane system." Journal of Energy and Power Engineering 7, 2013: 1681-1690.

[47] Gianluigi Lo Basso, Livio de Santoli, Angelo Albo, Benedetto Nastasi. “H2NG (hydrogen-natural gas mixtures) effects on energy performances of a condensing micro-CHP (combined heat and power) for residential applications: An expeditious assessment of water condensation and experimental analysis.” Energy 84 (Energy) 84 (2015): 397-418.

[48] Gianluigi Lo Basso, Benedetto Nastasi, Davide Astiaso Garcia, Fabrizio Cumo. «How to handle the Hydrogen enriched Natural Gas blends in combustion efficiency measurement procedure of conventional and condensing boilers.» Energy 123 (energy) 123 (2017): 615-636.

[49] Gianluigi Lo Basso, Livio de Santoli, Angelo Albo, Benedetto Nastasi. «H2NG (hydrogen-natural gas mixtures) effects on energy performances of a condensing micro-CHP (combined heat and power) for residential applications: An expeditious assessment of water condensation and experimental analysis.

[50] Gianluigi Lo Basso, Benedetto Nastasi, Davide Astiaso Garcia, Fabrizio Cumo. "How to handle the Hydrogen enriched Natural Gas blends in combustion efficiency measurement procedure of conventional and condensing boilers." Energy 123 (energy) 123 (2017): 615-636.

[51] Benedetto Nastasi, Gianluigi Lo Basso. «Hydrogen to link heat and electricity in the transition towards future Smart Energy Systems.» Energy 110, 2016: 5-22.

[52] A.R.Khan, M.R.Ravi, AnjanRay Experimental and chemical kinetic studies of the effect of H_2 enrichment on the laminar burning velocity and flame stability of various multicomponent natural gas blends International Journal of Hydrogen Energy ,Volume 44, Issue 2, 8 January 2019, Pages 1192-1212.

[53] Lounici, M.S., Boussadi, A., Loubar, K., Tazerout, Experimental investigation on NG dual fuel engine improvement by hydrogen enrichment, 2014, International Journal of Hydrogen Energy39(36), pp. 21297.

[54] Marco Antonia Rosa Do Nascimento, Lucilene De Oliveira Rodrigues ,Micro gas turbine engine , Eraldo Cruz dos Santos, Eli Eber Batista Gomes, A review, www.intechopen.com.