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# A Preliminary Energy Analysis of a Commercial CHP Fueled with $H_2NG$ Blends Chemically Supercharged by Renewable Hydrogen and Oxygen

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## Abstract

Currently, Power-to-Gas technologies are considered viable solutions to face the onset problems associated with renewable capacity firming. Indeed, carbon-free hydrogen production converting renewable electricity excess and its injection into natural gas pipelines is considered a short- to medium-term solution. In this way, the so-called $H_2NG$ blends can be fired within internal combustion engines and micro gas turbines operating in CHP mode, offering better environmental-energy performances in machines. As regards the distributed energy generation scenario, the local $H_2$ production by means of electrolysis for methane enrichment will be more cost-effective if the oxygen is fruitfully used instead of venting it out like a by-product, as usually occurs. This study focuses on the usefulness of using that oxygen to enrich the air-fuel mixture of an internal combustion engine for micro-CHP applications, once it has been fuelled with $H_2NG$ blends. Thus, the main aim of this paper is to provide a set of values for benchmarking, in which $H_2NG$ blends, ranging in 0%-15% vol., burn within an ICE in partial oxy-fuel conditions. In particular, a preliminary energy analysis was carried out based on experimental data, reporting the engine operating parameters, gains and losses in both electrical and heat recovery efficiency. The oxygen content in the air varies up to 22% vol. A Volkswagen Blue Tender CHP commercial version (19.8 $kW_{el.}$ of rated electrical power output) was considered as the reference machine and its energy characterization was reported when it operated under those unconventional conditions.

*Keywords:*Micro CHP; Hydrogen; $H_2NG$; ICE; Energy Efficiency; Oxy-Fuel.

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## 1. Introduction

In recent times the share of renewable energy sources in electricity generation has been rising worldwide and, consequently, the high penetration of intermittent energy sources leads to increasing numbers of hours in which generation exceeds demand. Moreover, the growing deployment of distributed generation contributes to increasing the onset of time-mismatch phenomena between production and demand. Hence, the most critical issues in effective energy management are those that regard the installation of expensive energy storage systems (electrical and thermal) resulting from the integration of traditional thermal cycles with non-programmable renewable sources.

Another viable option consists of merging different sectors such as electric grids, heating grids and transportation to overcome the current barriers through the new Smart Energy systems development. In this way, P2G, P2F and P2H, by integrating renewable hydrogen production, represent promising solutions to accomplish higher energy efficiency and environmental sustainability level [1,2].

In this framework, hydrogen could be used for both NG and biogas enrichment or for the synthesizing of liquid and gaseous eco-fuels. Those ones can be burned in the most common end-user devices as well [3-5]. Hence, assessing the contribution of feasible hydrogen integration in well-proven technologies is the first step towards promoting so-called electro-fuels and establishing a hydrogen energy demand on the market that will move towards hydrogen economy. For reasons such as these, several authors have investigated clean hydrogen production and its end uses [6-9]. Indeed, according to Bysveen [10], a good transition solution could be the use of hydrogen and hydrocarbon mixtures. Specifically, the use of $H_2$NG blends could constitute a realistic bridge solution to achieve these goals once the associated technical features are resolved. The latter include distributed hydrogen production, its cost effectiveness and its treatment, in order to meet end user needs; these constitute the imminent challenges that need to be overcome by adopting new policies and novel incentive strategies [11].

Local $H_2$ production by means of the electrolysis for NG enrichment will be more cost effective if the oxygen is fruitfully used, instead of venting it out like a by-product as usually occurs. Consequently, this paper focuses on the possibility to use the oxygen derived from the water-splitting reaction to chemically supercharge an internal combustion engine designed for micro-CHP applications and fueled, locally, with $H_2$NG blends. A few studies are available in the literature regarding oxy-fuel combustion and air enrichment by oxygen applied to ICEs. Van Blarigan et al. [12] investigated the potential $NO_x$ emission reduction in a spark-ignition engine fuelled with NG in a low nitrogen oxy-combustion environment while Bilger and Wu [13] have analyzed the potential of oxy-combustion application for increasing $CO_2$ capture efficiency related to end-pipe technologies. Similarly, Salt, Tree and Kim [14] have studied the effects of pure oxygen use on a diesel engine equipped with the EGR system so as to enrich the recirculated exhaust gas. However, the scope of that study was not to operate in full oxy-fuel conditions but in partial ones, limiting the nitrogen concentration above 50% vol. Then Killingsworth et al. [15] carried out an energy performance analysis on ICE using Argon rather than $CO_2$ or $H_2O$ as the main chemical component to mitigate flame temperature enhancement due to oxygen enrichment.

The present research project seeks to contribute an innovative approach in both scientific and industrial terms, since oxy-combustion applied to $H_2$NG mixtures has not yet been widely investigated. This preliminary study is expected to assist in developing a hydrogen-based economy and to provide more guidance for industrial applications such as CHP, which can be integrated within the forthcoming hybrid energy systems. As this study focuses on assessing engine performance, analyzing issues such as management strategies will be part of subsequent study.

**Nomenclature**

| | | | |
|---|---|---|---|
| CHP | Combined Heat and Power | A/F | Stoichiometric Air to Fuel ratio |
| EGR | Exhaust Gas Recirculation | $c_{p,I}$ | Specific Heat at constant pressure (kJ/molK) |
| $H_2$NG | Hydrogen enriched Natural Gas blend | $H$ | Specific enthalpy (kJ/mol) |
| ICE | Internal Combustion Engine | $\Delta H^0_f$ | Enthalpy of formation (kJ/mol) |
| P2F | Power to Fuel | $T_{ad}$ | Adiabatic flame temperature (K) |
| P2G | Power to Gas | | |
| P2H | Power to Heat | | |

## 2. Test rig description and methodology

The preliminary experimental campaign was carried out on a commercial version of a four-cylinder natural gas engine based on the Otto cycle. It is equipped with a 3-way catalyst and two Lambda probes, operating in stoichiometric combustion conditions and at fixed rotational speed. The engine's small electrical size (19.8 $kW_{el}$) was designed expressly for both dwellings and small industrial applications. In particular, the on-board control unit allows users to modulate the fuel flow through the carburetor slightly so as to keep the air to fuel ratio under control along with the shaft rotational speed. Indeed, that CHP engine is not able to run at partial load and it is usually connected to thermal storage in order to decouple the heating demand and the heat production derived from the CHP. The engine data sheet is summarized in Table 1.a.

Table 1.a CHP VolksWagen Eco Blue Tender technical specifications

| Parameters | Values |
|---|---|
| Engine Type | Natural Gas Engine water/Ethylene Glycol cooled |
| Rotational speed | 1,520 rpm |
| Compression ratio | 13.5:1 |
| Cylinders number | 4 |
| Displacement | 1,984 $cm^3$ |
| Type of Electric generator | cooled asynchronous generator |
| Active power | 19.7 kW generator |
| Voltage | 400 V, AC, three-phase |
| Frequency | 50 Hz |
| Generator efficiency | 0.94 |
| CHP electrical efficiency | 0.34 |
| CHP heat recovery efficiency | 0.58 |
| Max. Thermal power output | 36.1 kW |
| Heating water Flow rate | 2.9 $m^3/h$ |
| Supply temperature | 70-90°C |
| Return temperature | 60-70 °C |
| Maximum operating pressure | 6 bar |
| Required methane number | > 80 |
| Rated NG consumption | 5.84 $Nm^3/h$ |
| Fuels | Natural gas, H / L according to DVGW G 260, |

The water of the end-user hydraulic loop is pre-heated by routing it through the annular heat exchanger of the electric generator. Subsequently, a plate heat exchanger transfers the overall recovered heat from the engine jackets and exhaust gas to the water. The engine coolant consists of an ethylene glycol solution (i.e. VW G13) at 50% by mass which flows, firstly, through the cylinder jackets, and secondly, through a liquid-to-gas heat exchanger so as to increase the coolant temperature up to 95 °C (see Fig 1 part a)). Finally, a 3-way proportional valve completely bypasses the water flow rate since the electrical generator inlet temperature is lower than 70°C during the start-up transient. The required hydrogen for NG blending is produced by an alkaline electrolyser of rated capability equal to 1.2 $Nm^3/h$. Additionally, an automatic mixing device is integrated into the machine, in order to make several $H_2NG$ mixtures as needed. For the purposes of this research project, the oxygen venting line has been modified by installing a pressurized buffer, control valves, a pressure reducer and a flow meter for modulating oxygen additions to the air during the engine intake phase (see Fig.1 part b)).

Having done the above, the CHP engine was then tested in seven different fueling conditions deriving from the following combinations: i) pure NG as fuel and oxygen in air equal to 20.9% vol. (normal air composition), which was considered the base case study; ii) $H_2NG$ @ 10% vol. as fuel and standard air composition; iii) $H_2NG$ @ 15%

vol. as fuel and standard air composition; iv) $H_2NG$ @ 10% vol. as fuel and enriched air at 21.5% vol. oxygen; v) $H_2NG$ @ 10% vol. as fuel and enriched air at 22% vol. oxygen; vi) $H_2NG$ @ 15% vol. as fuel and enriched air at 21.5% vol. oxygen; vii) $H_2NG$ @ 15% vol. as fuel and enriched air at 22% vol. oxygen.

In all of these operating conditions, which were characterized by a constant electrical output, both NG and hydrogen consumptions were measured by two mass flow meters; thermal power output along with coolant temperatures, exhaust gas temperatures and hot water temperatures were recorded by the heat counter and PT 100 sensors. The outcomes were then time-averaged and compared in a systemic overview in order to show how hydrogen and oxygen affect engine energy performance. At the time of writing, the engine had reached more than 700 h running in unconventional ways and no failures had occurred. Specifically, CHP performance was recorded for over 100 h for each feeding set up with the sample time equal to 15 minutes.

Finally, it is important to point out that the stoichiometric air-to-fuel ratio was kept under control by manual carburetor tuning, measuring the oxygen residual volumetric fraction within the exhaust gas with an additional electrochemical probe. Even though the engine is equipped with Lambda probes and with an Electronic Control Unit (ECU) acting on the throttle body, such manual adjustment was required given that the control loop was not able to completely offset the fuel density reduction due to the hydrogen addition.

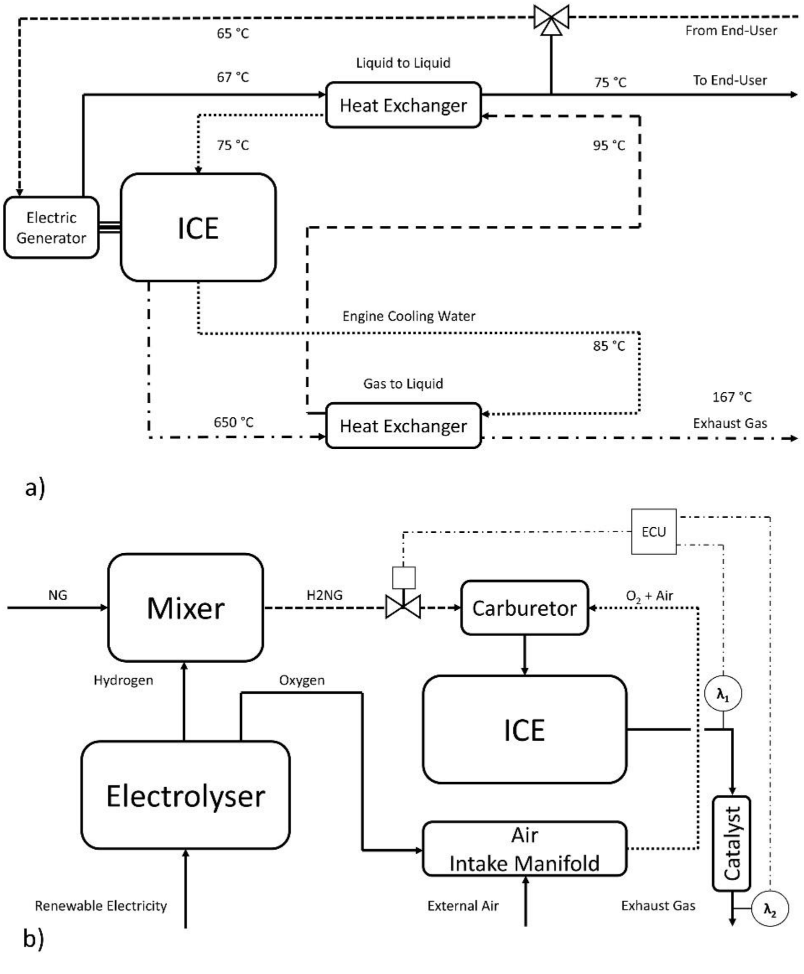

Fig. 1. (a) Micro CHP heat recovery architecture; (b) Hydrogen and oxygen mixing layout.

## 3. Oxygen enrichment effects on the combustion of $H_2$NG blends

It is well known that hydrogen is characterized by a high burning velocity, high gravimetric energy content, and low ignition energy. In the last decade, several research projects demonstrated that the addition of hydrogen to hydrocarbon fuels improves their combustion performance (e.g. combustion ignitability and stability) and, given that the blends are able to burn leaner, the pollutant emissions can be further reduced [16,17]. In regard to heat transfer issues, Wu L. et al. [18] found that radiative heat flux decreases when more hydrogen is added, while it is significantly enhanced with $O_2$ concentration. Moreover, the degree of reduction in radiative heat flux, as $H_2$ enrichment grows, is larger at higher $O_2$ fractions. Finally, the total heat flux rises with the addition of $H_2$, and much more with higher $O_2$ concentration, while the degree of increase in total heat flux, with the addition of $H_2$, was smaller at higher oxygen concentrations. This is mainly due to the fact that flame temperature is generally higher, so that the burning process is more complete when more $H_2$ is blended and enhanced convective heat transfer within the combustion chamber occurs. Although the radiative heat flux decreased with the $H_2$ addition, the combined effect of these two factors (radiative and convective) is that the total heat flux increases with the addition of $H_2$, owing to the dominating role of the convective heat flux. In accordance with Equation 1, the adiabatic flame temperature values, as a function of hydrogen percentage with changes in oxygen enriched air composition have been calculated and reported in Table 1.b.

$$H_{products} = \sum n_i \left[ \Delta H^0{}_{f,i} + \overline{c_{p,i}} \left( T_{ad} - 298 \right) \right] = H_{reactants} \qquad (1)$$

From the data it emerges that the adiabatic temperature gain is more sensitive to oxygen enrichment than to that of hydrogen. For this reason, the conservative oxygen value of 22% vol. was chosen as the maximum threshold during experimental tests in order to preserve the CHP mechanical components.

Table 1.b Adiabatic flame temperature vs hydrogen volumetric fraction in the blend with changes in oxygen enrichment.

| | Oxygen fraction in air [% vol.] | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $f_{H2}$ | 20.9 | 22 | 23 | 24 | 25 | 26 | 27 | 28 | 29 | 30 |
| 0% | 2094.805 | 2166.342 | 2229.043 | 2289.614 | 2348.141 | 2404.706 | 2459.387 | 2512.262 | 2563.404 | 2612.882 |
| 5% | 2097.753 | 2169.267 | 2231.938 | 2292.472 | 2350.955 | 2407.469 | 2462.094 | 2514.908 | 2565.984 | 2615.392 |
| 10% | 2100.929 | 2172.418 | 2235.057 | 2295.550 | 2353.984 | 2410.444 | 2465.008 | 2517.755 | 2568.759 | 2618.091 |
| 15% | 2104.362 | 2175.822 | 2238.425 | 2298.874 | 2357.256 | 2413.655 | 2468.153 | 2520.828 | 2571.754 | 2621.004 |
| 20% | 2108.082 | 2179.511 | 2242.075 | 2302.475 | 2360.799 | 2417.133 | 2471.558 | 2524.154 | 2574.995 | 2624.155 |
| 25% | 2112.129 | 2183.523 | 2246.043 | 2306.388 | 2364.649 | 2420.911 | 2475.257 | 2527.765 | 2578.514 | 2627.575 |
| 30% | 2116.548 | 2187.901 | 2250.372 | 2310.658 | 2368.848 | 2425.030 | 2479.288 | 2531.701 | 2582.348 | 2631.301 |

Table 2. Stoichiometric A/F (Air to Fuel ratio) by volume [$Nm^3_{Air}/Nm^3_{Fuel}$] vs. hydrogen volumetric fraction in the blend with changes in oxygen enrichment.

| | Oxygen fraction in air [% vol.] | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $f_{H2}$ | 20.9 | 22 | 23 | 24 | 25 | 26 | 27 | 28 | 29 | 30 |
| 0% | 9.569 | 9.091 | 8.696 | 8.333 | 8.000 | 7.692 | 7.407 | 7.143 | 6.897 | 6.667 |
| 5% | 9.211 | 8.750 | 8.370 | 8.021 | 7.700 | 7.404 | 7.130 | 6.875 | 6.638 | 6.417 |
| 10% | 8.852 | 8.409 | 8.043 | 7.708 | 7.400 | 7.115 | 6.852 | 6.607 | 6.379 | 6.167 |
| 15% | 8.493 | 8.068 | 7.717 | 7.396 | 7.100 | 6.827 | 6.574 | 6.339 | 6.121 | 5.917 |
| 20% | 8.134 | 7.727 | 7.391 | 7.083 | 6.800 | 6.538 | 6.296 | 6.071 | 5.862 | 5.667 |
| 25% | 7.775 | 7.386 | 7.065 | 6.771 | 6.500 | 6.250 | 6.019 | 5.804 | 5.603 | 5.417 |
| 30% | 7.416 | 7.045 | 6.739 | 6.458 | 6.200 | 5.962 | 5.741 | 5.536 | 5.345 | 5.167 |

Since the blends are burnt within ICE cylinders the stoichiometric air to fuel ratio plays a key role for determining the expansion starting point of the Otto cycle.

Table 2 summarizes the stoichiometric Air to Fuel ratio by volume for both different hydrogen additions and oxygen enrichments up to 30% vol. It is noteworthy that the larger the hydrogen fraction, the lower the required air volume is. Similarly, as the air becomes richer in oxygen, the A/F ratio tends to diminish. Additionally, if the A/F ratio is considered by mass, its values will increase with hydrogen additions while they will shrink at higher oxygen levels owing to the partial substitution of nitrogen atoms. Once the A/F is known, the specific mass of the exhaust gas per fuel unit by volume can be calculated. Given that ICEs operating in CHP mode are also able to recover heat from exhaust gas, the effects of both hydrogen and oxygen enrichment on the exhaust mass flow rate have been investigated to assess the potential energy losses or gains associated with the liquid to gas heat exchanger derating.

Table 3 outlines the calculation results related to the wet exhaust gas specific mass showing that, for each unit of fuel burnt, the mass flow rate decreases considerably compared to those with pure NG under common combustion conditions.

When considering a hydrogen fraction equal to 20% vol. and 20.9% vol. of oxygen, the reduction is 15.13% approximately, whilst in air-enriched conditions (e.g. $O_2$ = 22% vol.) the reduction changes to 19%.

Next, the same approach was used to evaluate all numerical variations of exhaust gas specific heat at constant pressure, which is another required parameter for the heat exchanger energy balance.

From data reported in Table 4, it can be noticed that the specific heat increases when more hydrogen is blended, as well as when the oxidizer is richer in oxygen than traditional air. That feature is due, on the one hand, to the higher water steam content deriving from the molecular hydrogen combustion. Indeed, the steam is characterized by a high specific heat. On the other hand, the higher number of oxygen atoms relative to those of nitrogen entails a lower nitrogen mass fraction within the exhaust gases. As a consequence, the increasing specific heat values partially offset the exhaust gas specific mass reduction caused by these unconventional combustion settings.

Table 3. Wet Exhaust gas specific mass [kg/$Nm^3_{fuel}$] vs. hydrogen volumetric fraction in the blend with changes in oxygen enrichment.

| | Oxygen fraction in air [% vol.] | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $f_{H2}$ | 20.9 | 22 | 23 | 24 | 25 | 26 | 27 | 28 | 29 | 30 |
| 0% | 13.025 | 12.427 | 11.934 | 11.481 | 11.065 | 10.680 | 10.324 | 9.994 | 9.686 | 9.399 |
| 5% | 12.532 | 11.957 | 11.482 | 11.046 | 10.645 | 10.275 | 9.933 | 9.615 | 9.318 | 9.042 |
| 10% | 12.039 | 11.486 | 11.030 | 10.611 | 10.226 | 9.870 | 9.541 | 9.235 | 8.951 | 8.685 |
| 15% | 11.546 | 11.016 | 10.578 | 10.176 | 9.806 | 9.465 | 9.149 | 8.856 | 8.583 | 8.328 |
| 20% | 11.053 | 10.545 | 10.126 | 9.741 | 9.387 | 9.060 | 8.758 | 8.477 | 8.215 | 7.971 |
| 25% | 10.561 | 10.075 | 9.674 | 9.306 | 8.968 | 8.655 | 8.366 | 8.098 | 7.848 | 7.614 |
| 30% | 10.068 | 9.604 | 9.222 | 8.871 | 8.548 | 8.250 | 7.975 | 7.718 | 7.480 | 7.257 |

Table 4.Exhaust gas Specific Heat at constant pressure @ 250 °C [kJ/kgK] vs. hydrogen volumetric fraction in the blend with changes in oxygen enrichment.

| | Oxygen fraction in air [% vol.] | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $f_{H2}$ | 20.9 | 22 | 23 | 24 | 25 | 26 | 27 | 28 | 29 | 30 |
| 0% | 1.1900 | 1.1968 | 1.2028 | 1.2089 | 1.2148 | 1.2208 | 1.2266 | 1.2325 | 1.2383 | 1.2440 |
| 5% | 1.1914 | 1.1982 | 1.2043 | 1.2104 | 1.2164 | 1.2224 | 1.2283 | 1.2342 | 1.2401 | 1.2459 |
| 10% | 1.1928 | 1.1997 | 1.2059 | 1.2120 | 1.2181 | 1.2241 | 1.2302 | 1.2361 | 1.2420 | 1.2479 |
| 15% | 1.1944 | 1.2013 | 1.2076 | 1.2138 | 1.2199 | 1.2261 | 1.2321 | 1.2382 | 1.2441 | 1.2501 |
| 20% | 1.1961 | 1.2031 | 1.2094 | 1.2157 | 1.2220 | 1.2282 | 1.2343 | 1.2404 | 1.2465 | 1.2525 |
| 25% | 1.1979 | 1.2050 | 1.2115 | 1.2178 | 1.2242 | 1.2304 | 1.2367 | 1.2428 | 1.2490 | 1.2551 |
| 30% | 1.2000 | 1.2072 | 1.2137 | 1.2202 | 1.2266 | 1.2329 | 1.2393 | 1.2455 | 1.2518 | 1.2579 |

## 4. Results and Discussion

The main aim of this study has been to investigate the $H_2NG$ fueling effects in partial oxy-fuel combustion set up on the energy performance related to a commercial micro CHP. In Fig.2 the preliminary experimental data have been summarized and once the fuel type along with the oxygen air enrichment were fixed, the engine behaviors related to the different operating conditions were compared. Specifically, electrical efficiency, heat recovery efficiency, both electrical and thermal power output, and blend consumptions were depicted in part a), part b), part c), and part d) of Fig. 2, respectively. In calculating them, the lower heating value of the $H_2NG$ mixtures was deduced from the literature [19]. Having analyzed the data, a beneficial effect on electrical efficiency emerged when the hydrogen fraction was equal to 10% vol. burning with traditional air, while at 15% vol. the engine needed higher mixture flow rate, penalizing that value. As regards heat recovery efficiency, it decreases as hydrogen is added to NG, in accordance with previous studies [19-21], owing to the liquid-to-gas heat exchanger off-design operation. Indeed, the exhaust mass flow rates were higher when hydrogen is added even though the specific mass is lower, as reported in Table 3. This is due to the fact that the engine requires a higher fuel flow rate [22], as shown in Fig. 2 part d), in order to offset the lower volumetric energy density of the inlet charge and to keep the stoichiometric combustion conditions under control.

In essence, less thermal power output can be recovered and higher heat losses to the stack occur. Furthermore, from the mechanical point of view, the electrical power output was the same in all tests. In contrast, when oxygen is added, more electrical power can be generated, as shown in Fig.2 part c). Moreover, in order to increase the hydrogen fraction in oxy fuel conditions, the electrical output goes down slightly. The higher fuel flow rates are mainly due to control system intervention which regulates the gas injection valve opening; this regulation is achieved via the feedback signal generated by the Lambda probes.

In addition, as regards the thermal power output and heat recovery efficiency, oxygen enrichment negatively affects their values since the exhaust gas specific mass is much lower compared to that of the NG levels.

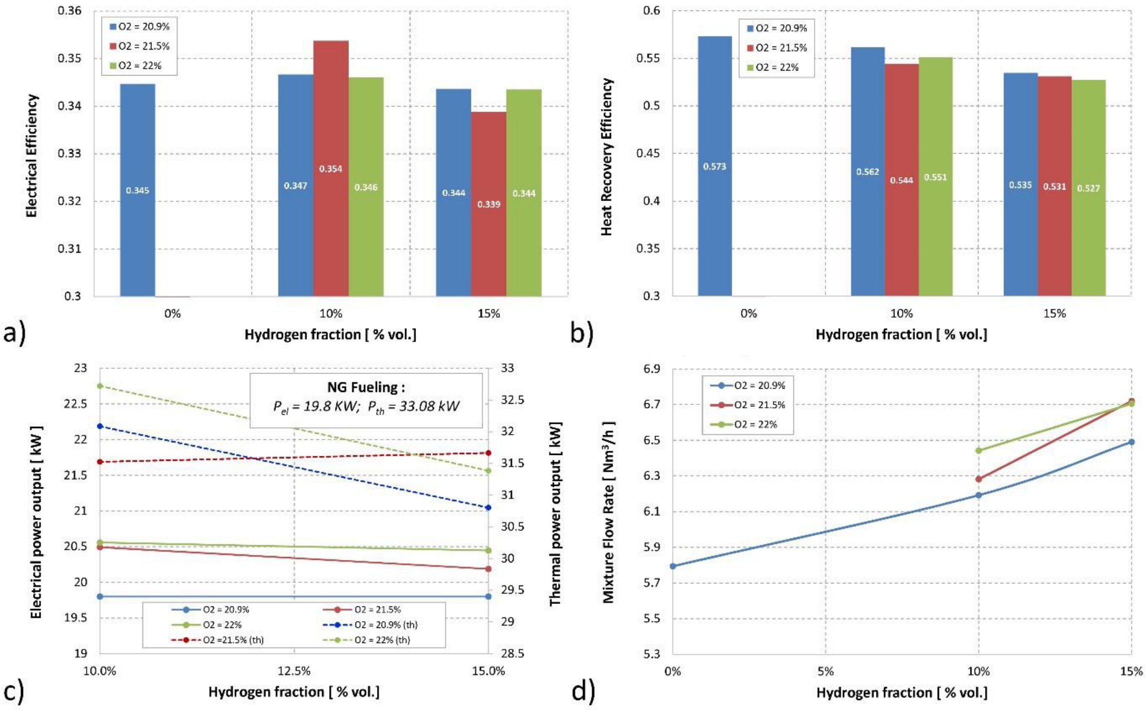

Fig. 2. (a) Micro CHP electrical efficiency vs. hydrogen fraction with changes in oxygen addition; (b) Micro CHP heat recovery efficiency vs. hydrogen fraction with changes in oxygen addition; (c) Micro CHP electrical and thermal power output vs. hydrogen fraction with changes in oxygen addition; (d) Micro CHP mixtures consumption vs. hydrogen fraction with changes in oxygen addition.

## 5. Conclusions

Based on the outcomes of this preliminary experimental campaign, the CHP performed the best, in terms of electrical efficiency, when it operated with $H_2$NG@ 10% vol. and oxygen enrichment equal to 21.5 % vol. In the other cases, the engine seems to be penalized by the default setup parameters implemented within the ECU. As a consequence, further studies using different engine tunings will be required in order to expand our research findings. Moreover, adequate energy management strategies, when such CHP solution is built in new smart energy systems, have to be defined accounting for environmental performance as well. Nonetheless, given that the engine is a commercial version, any external modification of safety and control parameters was not possible due to manufacturer settings.

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