

Development and testing of a multi-fuel micro-CHP conversion kit

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1. Introduction

The economic, environmental and energy crisis determined a global situation in which urgent and effective actions are necessary in all areas of energy use. The global problems related to the energy and environmental impacts of the built environment, energy access, energy depletion and increasing urbanism are the issues on which many energy companies and governments are focusing their efforts. Focusing the attention on developing Countries, it is known that today about 20% of the world population still lives without access to electricity (IEA, 2014). In addition, power outages occur frequently, and also last for several hours.

All these conditions favored the penetration of off-grid power systems all over the world (Kaundinya, Balachandra, & Ravindranath, 2009), not only in developing Countries, but also in developed Countries, where they are in some cases preferable to the grid connection, because of their affordability and reliability.

In such critical context, prompt actions are required in order to make the current power systems more robust, safe and durable. It

means also increasing the reliability of the energy supply by reducing the dependence on imports from politically unstable regions and reducing the economic disruptions caused by the rising costs of energy, as mentioned also in (IEA, 2012). This is true in particular for the generation of electricity, because of the uncertainty of its market due to deregulation, environmental concerns and emissions trading mechanisms. The prices of fuel and electricity are characterized by a wide variability. Moreover, the same has gradually happened to the costs of the greenhouse gas emission allowances (Varympopiotis, Tolis, & Rentizelas, 2014), resulting in a even higher uncertainty in the energy market.

In this field, technological improvement, fuel switching and decentralized models can give significant contributions. In fact, due to the uncertainties, technical optimization may not necessarily guarantee the continuity and effectiveness of energy, whereas some aspects of flexibility in the programmable power-plants operation (i.e. strategic decisions, like the selection of an appropriate fuel) may significantly contribute to their financial performance and stability. It is therefore possible to state that, if technically feasible, the option of switching between multiple alternative fuels and operational modes assumes a core importance and may potentially lead to more promising energy investments in the future (Varympopiotis et al., 2014). It is in fact well known that the global energy system is drastically influenced by the oil crisis.

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Nomenclature

$P_{i,f}$	fuel power input (kW)
$P_{r,e}$	exhaust recoverable thermal power (kW) thermal
P_a	power available for other purposes (kW) power
P_v	required for the fuel vaporization (kW) rated actual
P_e	electrical power of the genset (kW) shaft power to
P_s	the generator (kW)
$FR_{m,a}$	air mass flow rate (kg/h)
$FR_{m,f}$	fuel rate consumption at the rated electrical output
	power (kg/h)
$T_{e,c}$	temperature of the exhaust at the cylinders outlet
	(K)
$T_{e,he}$	design temperature of the exhaust at the heat
	exchanger outlet (K)
ΔT	temperature difference of the exhaust across the
	heat exchanger (K)
SH_a	average exhausts specific heat before and after the
	heat exchanger (J/kg K)
S_R	stoichiometric air/fuel ratio by mass
LHV	fuel low heating value (MJ/kg)
HHV	fuel high heating value (MJ/kg)
H_v	vaporization enthalpy (kJ/kg)
$P_{i,f-STF}$	rated fuel power input when using the main tradi-
	tional fuel (kW)
$T_{e,c-STF}$	cylinder outlet exhaust temperature when using the
	main traditional fuel (K)
k	ratio of specific heat at constant pressure (Cp) to the
	specific heat at constant volume (Cv)
η_t	thermal efficiency
η_o	overall efficiency
η_e	electrical efficiency (on real electrical power)
D	engine displacement (L)
R_{PM}	revolution per minute

Fluctuations of the oil market influence the price of all the fossil fuels, and indirectly also that of other sources, like biofuels (bioethanol and biodiesel). In addition, the market of biofuels is also influenced by the food economy, due to the competitiveness between the use of certain agricultural products for energy or for food (Rathmann, Szklo, & Schaeffer, 2010). Nevertheless, the biomass and its derived biofuels can represent an important opportunity, in particular for undeveloped areas. In fact, the biomass is a versatile, renewable and persistent source, which has already been used extensively within Europe and other developed contexts for heating and/or power generation. Even greater benefits can be reached in the developing Countries, where the contributions of biomass to the national primary energy demands are much higher and where this source is generally exploited without any reliable conversion systems. Here the adoption of distributed, cost-effective, efficient, simple and easy to use biomass-fueled CHP systems could possess a very strong potential.

Such strategy is in concordance with the development of the decentralized generation, which focuses in particular on the transition from centralized to Distributed Generation (DG) energy systems. This new paradigm benefits also from the technological progresses toward the miniaturization of the energy conversion equipment, which determines the availability of a wide variety of small-scale forms of power. In this framework, the capability of a decentralized system to operate in order to generate power or to function in cogeneration, with multiple fuel sources, is the key aspect toward a flexible and adaptable system of energy supply.

Taking into account all the aforementioned considerations, this paper is devoted to the development of a kit for converting

a spark-ignition engine installed on small-size gensets in order to allow their functioning with multiple fuels, with particular reference to low-boiling-point fuels. Such kit allows to easily operate on the technological systems which have been on the market for a long time, reducing the application strain, since the kit is designed as an add-on component. Furthermore, depending on the local conditions of utilization, the use in cogeneration of the same engine is also possible, as explored in the following sections.

The strengths are that the system is technically simple, robust, reliable and versatile, since it can be fueled with different fuels, and it can be used both for grid-connected and off-grid applications. In these cases, it can be easily integrated in systems based on renewable sources when non-programmable sources (sun and wind) are not available.

The possibility to fuel a generator with different fuels, increasing its overall energy efficiency, makes its applications more stable and affordable in different conditions of the energy market. Other example of experimental fuel switching are available in the technical literature (Chen & Nishida, 2014; Maurya & Agarwal, 2014; Mustafa, Cenk, & Mustafa, 2014; Park, Youn, Lim, & Lee, 2012; Yamina, Sakhninib, Sakhriea, & Hamdana, 2013), but more applications are needed in order to better analyze and to promote this potential innovation.

In particular, this paper reports the first phase of technical development of the system and the results obtained testing the system in laboratory conditions. In order to test the system with available, common and compatible fuels, the test was carried out using a fossil fuel, LPG, and a biofuel, bioethanol. It must be noted that the latter fuel was considered as it is starting to become very interesting also in EU Countries, since it could have low environmental impact and its production cost is decreasing rapidly; although the price in Italy is still high and about 0.73–0.86 \$/L (Caputo, 2011), the production cost in new plants in Brazil is close to 0.20 \$/L (0.30 \$/L of gasoline equivalent) (ESSE, 2011) and the IEA foresees a reduction of one third in the cost of ethanol by 2030, even if a significant technological progress will be necessary to make this happen (Doumax, MarcPhili, & Sarasa, 2014).

2. Concept design of the multi-fuel micro-CHP conversion kit

As previously introduced, the research focused on the development of a conversion kit for small-size four-stroke, Otto-cycle reciprocating internal combustion engines equipped with a carburetor; this type of engines, widely used for small-size applications in the agricultural sector, in pumping systems and power units, is typically characterized by a simple design, high reliability and low maintenance, but offers however medium-low performances in terms of efficiency and environmental standard; in fact, in order to contain the costs and increase the simplicity of construction and maintenance, these engines typically do not integrate the same modern technology used on more advanced engines, such as those adopted in the automotive field, e.g. electronic fuel injection systems, EGR (Exhaust Gas Recirculator) or catalytic converter (Arias, 2005; Koederitz, 2003). The purpose of the research is therefore the development of a technique for the conversion of this type of engine, with specific reference to the ones installed on existing gensets, with the dual objective of:

- increase their flexibility of use, allowing them to be fueled with different alternative fuels, with a reduced environmental impact and a short chain;
- increase their overall generation efficiency, allowing the recovery of part of the thermal energy that would otherwise be lost during the engine running.

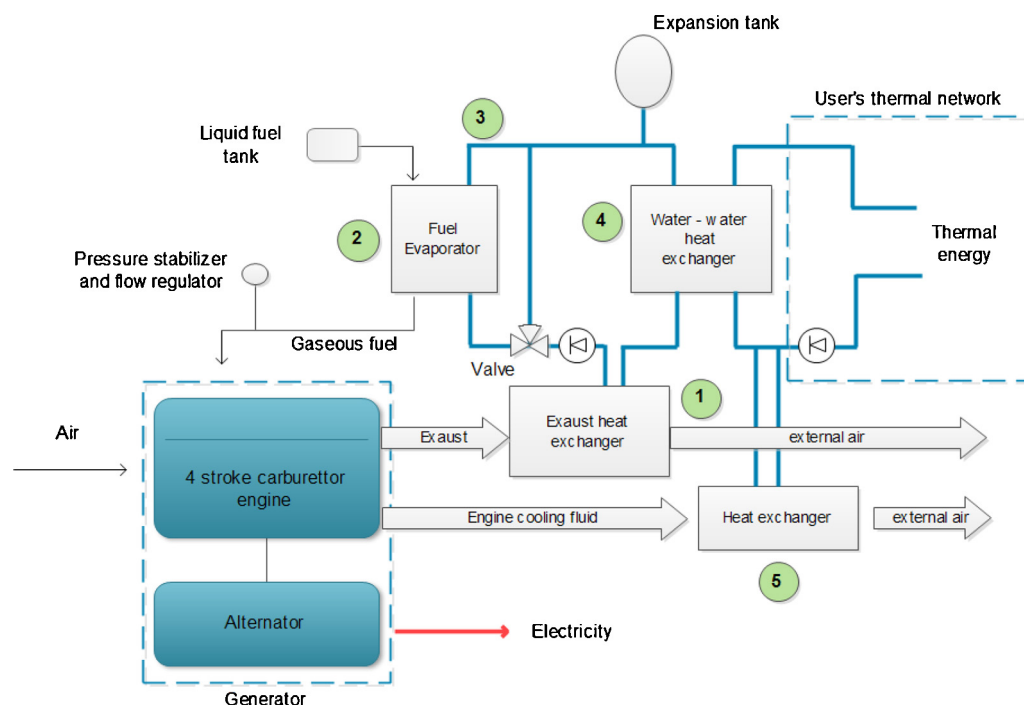


Fig. 1. Conceptual scheme of the system.

Conceptually, the conversion kit allows to use part of the thermal energy produced by the engine during the generation of electricity to bring different types of liquid fuels to the gaseous state, also allowing to recover part of the residual heat for other thermal uses.

In this sense, in the literature, several experimental studies dealing with the effects of biofuels usage in spark ignition engines on the engine performance, exhaust emissions and combustion characteristics have been carried out, with particular reference to alcoholic fuels (Celik, Ozdalyan, & Alkan, 2011; Eyidogan et al., 2011). In the concept design developed, the feeding with fuels at gaseous state can simplify the technical and economic aspects related to fuel dosage: although less accurate than a liquid-injection feeding, it is easier to implement; in addition, this solution drastically reduces the problems related to the deterioration of some of the components necessary for the injection of liquid-state fuel (gaskets and rubber elements, feeding pump, etc.), due to chemical-physical interactions, such as those caused by alcoholic substances (Agarwal, 2007).

In detail, the system aims to be an add-on easily applicable to engines of different sizes, since it does not require any transformation or relevant modification to the base genset and it is

specifically optimized for Otto-cycle reciprocating internal combustion engines.

3. System design

In order to actually make what is described in the concept design, the system architecture was defined, based on a uncomplicated conceptual scheme, with the objective of minimizing the use of electronic systems of management and control. In detail, the conversion kit is composed of the following main components:

1. *Exhaust heat exchanger*; this component aims to recover part of the heat contained in the exhaust gases of the engine; the device consists of a gas-water heat exchanger, which is installed on the exhaust pipe of the fumes of the engine and which allows to recover part of the residual heat and transfer it to a heat transfer fluid, even on air-cooled engines.
2. *Fuel evaporator*; this is a component which aims to use the thermal energy recovered from the exhaust heat exchanger, transferring it to the liquid feeding-fuel to allow its transition to the gaseous state. The evaporator must also be equipped with an electrical resistance, which is necessary to provide the

Table 1
Fuels' chemical and physical proprieties.

	Gasoline	Diesel	LPG	Ethanol	Methanol
Chemical structure	C ₄ to C ₁₂	C ₈ to C ₂₅	C ₃ H ₈ —C ₄ H ₁₀	CH ₃ CH ₂ OH	CH ₃ OH
Molecular mass (g/mol)	95–120	200	44	46.07	32.04
Density (g/cm ³ –20 °C)	0.72–0.76	0.85	0.52	0.79	0.792
Vaporization enthalpy at 1 atm and at standard boiling point (kJ/kg)	380–450	250–375	426	840–950	1100–1150
Octane number	95	15–25	111	116	113
Energy content					
LHV (MJ/kg)	43.44	44.4	46.1	27.1	19.7
HHV (MJ/kg)	46.53	47.3	50.15	29.84	22.88
Physical state	Liquid	Liquid	Press. liquid	Liquid	Liquid
Flash point (°C)	–43	74	–73 to –100	13	11
Boiling point at 1 atm (°C)	80–100	180–360	–42	78	65

thermal energy required for the evaporation of the fuel during the warming-up of the engine, that is until the temperature of the heat transfer fluid does not allow the fuel to have a steam pressure greater than atmospheric pressure.

3. *Hydraulic circuit*; a hydraulic circuit, equipped with a specific circulating pump and an expansion tank, conveys the thermal energy from the exhaust heat exchanger and from the engine cooling fluid heat exchanger to the evaporator and, subsequently, to a water-water heat exchanger. The circuit is equipped with a 3-way valve, for the adjustment of the flow rate to be sent to the evaporator.
4. *Water-to-water heat exchanger*; the heat in excess, that is not required for the vaporization of the fuel, is transferred through this heat exchanger to thermal appliances connected to the system.
5. *Engine cooling fluid heat exchanger*; a fraction of the heat contained in the engine cooling fluid (water or air) can be transferred through this heat exchanger to thermal appliances connected to the system.

Fig. 1 shows a conceptual scheme of the system described above.

As it can be seen from Fig. 1, the components required for the adjustment and control are only the gaseous fuel pressure regulator and stabilizer and the 3-way valve located on the hydraulic circuit, whose calibration is carried out according to the pressure of the vaporized fuel, to ensure a stable feeding.

The proposed configuration allows to feed any type of fuel to the engine; in particular, for the use of low-boiling-point fuels, the heat transfer fluid used in the hydraulic circuit will be water, whereas for fuels with a high boiling point it will be necessary to use diathermic oil.

It should be noted that, at this stage, the research focused on fuels with low-boiling-point temperature at atmospheric pressure, for the following key reasons:

- in order to contain the operating temperatures and then to use water as a heat transfer fluid;
- the use of fuels with a low octane number (diesel and biodiesel) require to change the compression ratio in order to optimize the operation of the engine (EPA, 2014).

In order to characterize the analyzed low-boiling-point fuels, their physical and chemical properties are summarized in Table 1 and compared to those of some traditional fuels (DOE, 2014).

The calculation of the performance of the conversion kit previously outlined and the subsequent prototyping are described below.

4. Materials and methods

In order to estimate the theoretical performance of the conversion kit described above, and subsequently to size the components of a prototype installation to be used for experimental tests, a simplified method for the estimation of the mass and energy balance was developed.

Such method, described below, is valid considering the application of the kit on an Otto-cycle engine, with a total power between approximately 10 and 30 kW.

4.1. Simplified heat and mass balances calculation methodology

The calculation procedure involves the execution of the energy and mass balance starting from known data, related to the engine and to the generator system used as a basis for the use of the kit. In this sense, certain data provided by the manufacturer are required, namely the power input through the traditional fuel for which the engine was designed, the electrical power generated and the energy efficiency. Depending on these values, it can be made the calculation of the heat flow related to the process of vaporization of the liquid fuel and the fraction of energy usable for other purposes, as shown in Fig. 2.

The first quantity that is to be calculated is represented by the thermal power recoverable through the heat exchanger from the exhaust gases (*Exhaust recoverable thermal power*), obtainable using the following formula.

$$P_{r,e} = \frac{(FR_{m,a} + FR_{m,f})}{3,600,000} \times \Delta T \times SH_a \quad (1)$$

where, $P_{r,e}$ is the exhaust recoverable thermal power, in kW, $FR_{m,a}$ is the air mass flow rate, in kg/h, calculated with (2), $FR_{m,f}$ is the fuel rate consumption at the rated electrical output power, in kg/h, calculated as indicated in (3), ΔT is the temperature difference of the exhaust across the heat exchanger, in K, calculated with (4), SH_a is the average exhaust specific heat before and after the heat exchanger, in J/kg K, assumed to be equal to that of the air and calculated according to Langen linear equation (Santoli, Lo Basso, & Caruso, 2011) with (6).

Air mass flow rate ($FR_{m,a}$)

$$FR_{m,a} = S_R \times FR_{m,f} \quad (2)$$

where, S_R is the stoichiometric air/fuel ratio by mass, given for each type of fuel.

$FR_{m,f}$ is the fuel rate consumption at the rated electrical output power, in kg/h, calculated as indicated in (3).

Fuel rate consumption at rated electrical output power ($FR_{m,f}$)

The fuel rate consumption at the rated electrical output power, in kg/h, is provided by the engine manufacturers for the main traditional fuel (e.g. LPG, gasoline) and can be calculated for alternative fuels proportionally to different LHV, by imposing the same fuel

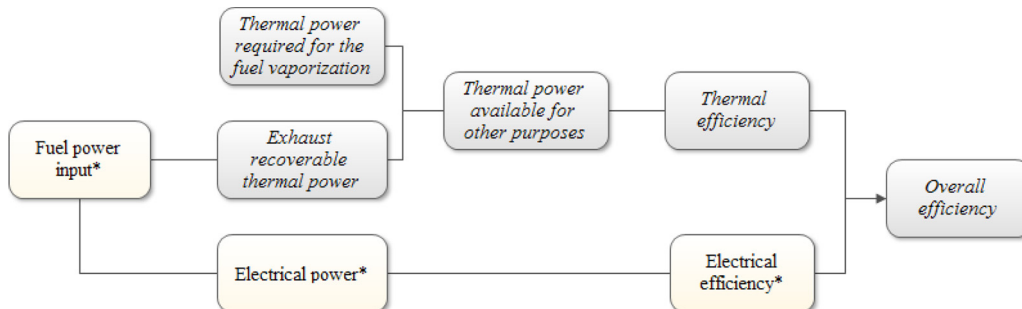


Fig. 2. Flow-chart of the main steps of the calculation method (*data provided by the manufacturer).

power input ($P_{i,f}$) related to the main traditional fuel, according to the following relation:

$$FR_{m,f} = P_{i,f} \times \frac{3.6}{LHV} \quad (3)$$

where, $P_{i,f}$ is the fuel power input, in kW, assumed to be constant for different fuels and equal to the nominal value related to the main traditional fuel for which the engine is designed, LHV is the fuel low heating value, in MJ/kg.

Exhaust temperature difference across the heat exchanger (ΔT)

$$\Delta T = T_{e,c} - T_{e,he} \quad (4)$$

where, $T_{e,c}$ is the temperature of the exhaust at the cylinders outlet, in K, calculated with (5), $T_{e,he}$ is the design temperature of the exhaust at the heat exchanger outlet, in K, assumed to be equal to 423 K in order to ensure the recovery of heat at a relatively high temperature, i.e. greater than 360 K, while enabling to keep contained the exchange surfaces of the heat recovery device. The temperature defined above is that required to evaporate all the low-boiling-point fuels identified (ethanol and methanol);

Temperature of the exhaust at the cylinders outlet ($T_{e,c}$)

The temperature of the combustion products at the outlet of the cylinders for the functioning with the main traditional fuel (e.g. LPG, gasoline) is provided by the engine manufacturer, while for the other fuels it is calculated assuming an adiabatic isentropic expansion, and therefore as a simple proportion between the power fed into the engine and the exhaust temperature, according to the following relation (Cengel, 2007).

$$T_{e,c} = \left(\frac{P_{i,f}}{P_{i,f-STF}} \right)^{(1-K)/K} \times T_{e,c-STF} \quad (5)$$

where, $P_{i,f}$ is the fuel power input, in kW, $P_{i,f-STF}$ is the rated fuel power input when using the main traditional fuel, in kW, $T_{e,c-STF}$ is the cylinder outlet exhaust temperature when using the main traditional fuel, provided by the engine manufacturer, k is the ratio of specific heats, that is the ratio of specific heat at constant pressure (C_p) to the specific heat at constant volume (C_v).

Average exhaust specific heat (SH_a)

$$SH_a = 953 + 0.15 \times \left(\frac{T_{e,c} + T_{e,he}}{2} \right) \quad (6)$$

where, $T_{e,c}$ is the temperature of the exhaust at the cylinders outlet, in K, calculated with (5), $T_{e,he}$ is the design temperature of the exhaust at the heat exchanger outlet, in K, assumed to be equal to 423 K.

Once the recoverable exhaust thermal power is known, this value can be compared with the thermal power required for the fuel vaporization, calculated according to (7):

$$P_v = \frac{FR_{m,f} \times H_v}{3600} \quad (7)$$

where, P_v is the power required for the fuel vaporization, in kW, $FR_{m,f}$ is the fuel rate consumption at the rated electrical output power, in kg/h, calculated as indicated in (3).

H_v is the vaporization enthalpy, in kJ/kg.

The recoverable exhaust thermal power, net of losses, must therefore be significantly higher than the power required for the fuel vaporization, in order to ensure the proper functioning of the system in all the conditions of use.

Subsequently, the thermal power available for other purposes can be calculated according to (8):

$$P_a = P_{r,e} - P_v \quad (8)$$

where, $P_{r,e}$ is the exhaust recoverable thermal power, in kW, P_v is the power required for the fuel vaporization, in kW.

Once the previously calculated quantities are known, it is possible to determine the energy efficiencies of the system, as indicated below.

Thermal efficiency (η_t)

$$\eta_t = \frac{P_a}{P_{i,f}} \quad (9)$$

where, P_a is the thermal power available for other purposes, in kW, $P_{i,f}$ is the fuel power input, in kW.

Overall efficiency (η_o)

$$\eta_o = \frac{P_a + P_e}{P_{i,f}} \quad (10)$$

where, P_a is the thermal power available for other purposes, in kW, P_e is the rated actual electrical power of the genset, in kW, $P_{i,f}$ is the fuel power input, in kW.

4.2. Prototype design and performance calculation

Once the calculation methodology has been set up, in order to properly assess the applicability of the technological solution to actual systems, a commercial generator was chosen as the applicative basis for prototyping; in detail, it is an Otto-cycle, V-twin, air-cooled, internal combustion engine, typically used for small-size electric generators, with a maximum shaft power of 18 kW at 3600 rpm. The engine, with a displacement of 725 cc, is already designed to be used with gaseous fuels, such as LPG or CNG. The electricity generation section consists of a generator with a rated power of 12.2 kVA, corresponding to 9.8 kW with a power factor of 0.8 at 3000 rpm and a nominal efficiency equal to 84%.

The maximum output power, resulting from the engine-generator coupling, is the following:

- Prime running power (PRP): 9 kW (11 kVA);
- Limited time running power (LTP): 9.8 kW (12.2 kVA).

The choice of this type of generator is related to the need to apply the experimental prototype of the conversion kit to a basic component which is technically elementary, and therefore characterized by high reliability and low costs, in accordance with the purposes of the research. The air cooling of the engine, however, does not allow an easy and convenient heat recovery from the cooling fluid of the engine itself, thus the prototyping of the kit is solely based on the possibility of heat recovery from the exhaust gases. Based on the hypothesis reported previously, in order to adequately size the various subcomponents of the prototype conversion kit and to estimate its global performances, the simplified calculation methodology described in Section 4.1 was applied, assuming at this stage the use of LPG, methanol and ethanol from renewable sources.

Table 2 summarizes the baseline data necessary for the calculation.

Table 2
Reference data.

		LPG	Ethanol	Methanol
Stoichiometric ratio by mass	S_R (kg _a /kg _f)	15.5:1	9:1	6.5:1
Vaporization enthalpy	H_v (kJ/kg)	426	842	1100
Temperature of the exhaust at the heat exchanger outlet ^b	$T_{e,he}$ (K)	423	423	423
Engine displacement ^a	D (L)		0.725	
Revolutions per minute ^a	R_{PM} (rpm)		3000	
Shaft power to generator ^a	P_s (kW)		13.00	
Real electrical power ^a	P_e (kW)		9	

^a Data provided by the manufacturer.

^b Design value.

Table 3
Mass and energy balance.

		LPG	Ethanol	Methanol
Fuel rate consumption	$FR_{m,f}$ (kg/h)	3.52 ^a	5.99	8.23
Fuel power input	$P_{i,f}$ (kW)	45.06	45.06	45.06
Cylinder outlet exhaust temperature	$T_{e,c}$ (K)	988.00 ^a	988.00	988.00
Exhaust temperature difference across heat exchanger	ΔT (K)	565.00	565.00	565.00
Exhaust gas mass flow rate	$FR_{m,a} + FR_{m,f}$ (kg/h)	58.06	60.46	58.47
Exhaust recoverable thermal power	$P_{r,e}$ (kW)	9.67	10.07	9.73
Thermal power required for fuel vaporization	P_v (kW)	0.00	1.4	2.52
Thermal power available for other purposes	P_a (kW)	9.67	8.67	7.22
Electrical efficiency (on real electrical power)	η_e (%)	19.97	19.97	19.97
Overall thermal and electrical efficiency	η_t (%)	41.4	39.2	36.0

^a Data provided by the manufacturer.

Table 3 summarizes instead the results obtained by the simplified calculation of the mass and energy balance.

The data reported above refer to the use of LPG, pure ethanol and methanol. As it can be seen, the conversion kit allows to recover from the exhaust gases the thermal power required for the vaporization of the alternative liquid fuels taken into consideration. In all three scenarios, a residual thermal power, to be used for other purposes, is also available, transforming to all intents and purposes the proposed generator in a micro-CHP system; in the specific prototypical case, the application of the kit allows to recover, in nominal operating conditions and with all the considered fuels, slightly less than 10 kW of thermal power, that would otherwise be lost through the exhaust.

The theoretical overall efficiency of such a system ranges therefore from a minimum value equal to 36%, using methanol, to a maximum of 41.4% when using LPG; these values thus confirm that the proposed kit allows to significantly increase the total efficiency of the generation system used as a basis of application, which through the simple generation of electricity involved values of approximately 20%.

It should be noted that the system can be fueled not only with pure ethanol, but also with 95° alcohol or containing progressively higher percentages of water; this is one of the strengths of the system designed. The feeding of the gaseous fuel, in fact, allows the possibility to use alcohol with significant percentages of water (up to 10–12%) without affecting the overall functionality of the engine. Specifically, since significant percentages of water are tolerable, the fuel can be obtained simply by thermal distillation, avoiding the mixing with hydrocarbons, which is instead necessary to obtain an anhydrous alcohol.

Once all the parameters of the mass and energy balance had been determined, the sizing of the main components of the prototype developed was carried out, as follows:

- *Fuel evaporator*: considering the thermal power required for the evaporation of the different types of fuel, the heat exchanging surface of the evaporator was assumed to be equal to a nominal

value of 7.5 kW. The fuel evaporator, manufactured from a classic tank-in-tank heat exchanger, consisting of two coaxial cylinders, has a volume of 4 L and contains the heat transfer fluid, while the inner cylinder, containing the fuel, has a total capacity of about 1.6 L, 0.9 of which is occupied by the liquid.

- *Exhaust heat exchanger*: the nominal thermal power of the heat exchanger was set to approximately 25 kW. This value is higher than the maximum power recoverable from the exhaust of the engine chosen for prototyping; the choice of this size arises from the ease of adapting to the purpose at hand components already present on the market (e.g. heat exchangers for the instantaneous production of DHW).
- *Hydraulic circuit*: it is made of a copper pipe with a diameter of 14 mm. The heat transfer fluid is circulated by an electric pump with a constant flow rate of 1 L/s. As already mentioned, the control of the thermal power fed to the fuel evaporator occurs through a 3-way valve, whose adjustment in the prototyping phase is performed manually. On this circuit is also installed a membrane expansion tank with a capacity of 5 L, which is sufficient to compensate for the thermal expansion of the heating fluid up to 95 °C, corresponding to the maximum design temperature of the system. On the circuit there is also an overpressure relief valve, set to 3 bar for safety reasons. The entire circuit is connected to a water–water exchanger with a nominal power equal to 25 kW.
- *Vaporized fuel circuit*: it is made of copper pipes with a diameter of 8 mm, it is equipped with a holding valve for the control of the steam flow rate and a manual shut-off valve, to block the inflow of fuel. There is also a pressure stabilizer, made of a membrane expansion tank, with a 0.3 bar preload and a capacity of 1 L.

It should be noted that no specific details are provided on the construction characteristics of the various components, since they are patenting pending.

5. Testing

The prototyping of the conversion kit and its application to the above described engine allowed to carry out experimental tests aimed at measuring the actual performance of the system designed. In this sense, several standard testing procedures were prepared, or are in a developing phase in many Countries, to provide methods for determining the performance of MCHP (micro combined heat and power) systems (Angrisani, Marrasso, Roselli, & Sasso, 2014). In any case, considering the preliminary and prototyping phase of the work, the testing activity was aimed just at validating the theoretical calculations and at ensuring the proper functioning of the conversion kit. In the forthcoming part of the research activity, a complete performance test, in accordance with the prEN 50465 (EN, 2014) or the prUNI E0204A073 (UNI, 2014) methodology, will be carried out.

At this stage, in order to test and measure the performance of the system, a first experimental campaign was carried out with LPG and bioethanol with a residual water content of 5%, using the following measurement devices:

- three-phase frequency, voltage and current meter;
- temperature sensors, placed on the hydraulic circuit, before and after the exhaust heat exchanger;
- exhaust gas temperature sensors, placed upstream and downstream of the exhaust heat exchanger;
- flow meters placed on the hydraulic circuits;
- precision scale for weighing the fuel used.

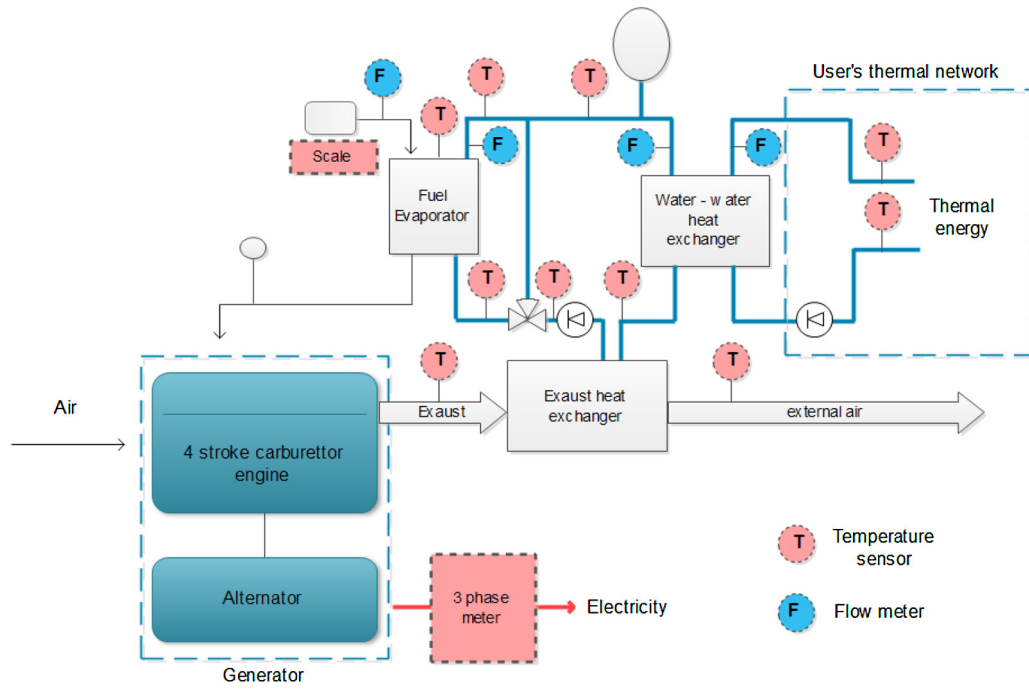


Fig. 3. Measurement devices positioning.

The positioning of the described measurement devices is shown in Fig. 3.

The experimental measurements were performed in an indoor environment, with an air temperature of 20 °C, at an altitude of 120 m above sea level, operating with the following procedure:

- 5 min engine warming-up, at zero load;
- detection of the quantity of fuel in the tank by means of weighing;
- application of the electrical load for 10 min and acquisition of the operating parameters;
- detection of the quantity of fuel in tank by means of weighing.

In order to assess the energy performance in terms of actual electrical load, the generator was connected with a three-phase load with a power factor of 0.8 and a variable electrical power consumption, according to the test requirements; a first test was performed with the rated load, equal to 9 kW, with the sole purpose of verifying the correct operation of the system. A second performance test was instead carried out with an electrical load of 4.5 kW, therefore corresponding to 50% of the maximum electric power deliverable by the system; this configuration corresponds in fact to a cautionary operating condition, since the thermal power generated by the engine, and thus the fraction available for fuel evaporation, is reduced compared to the functioning at full load; in addition, in such partial load condition the engine performance decreases, because of the losses in the air-intake duct. On this point, it must also be noted that the power generators are typically required to operate with load factors between 30 and 70% of the rated load, in order to cope with the inrush powers required by some electrical equipment. Load factors minor than 30% can significantly affect efficiency and reliability (Iverson, 2007).

The results obtained are summarized in Table 4.

It should be noted that the uncertainty on the calculated quantities is equal to:

- $\pm 2\%$ on electrical and thermal powers;
- $\pm 7\%$ on efficiencies.

As it can be seen from the experimental data, the measured electrical efficiency at the rated load, equal to 16.8% with both fuels used, is lower by about 15% than the theoretical values calculated in Section 4.2 of this paper. Such deviation is mainly due to the difference between the fuel consumption rate declared by the manufacturer for the operation with LPG and the data measured experimentally. It can be observed that the consumption of LPG at the rated load shown in Table 4 is consistent with that recorded with the same generator before the application of the conversion kit developed, confirming the fact that the changes do not adversely affect the rated performance of the engine.

The values obtained from the partial load test show, however, slightly lower performances than the theoretical ones, which are mainly related to a reduction in efficiency in partial load conditions, typical of Otto-cycle engines.

In any case, it can also be noted that, in all the test conditions analyzed, the application of the kit allows to obtain a quantity of thermal power sufficient for the evaporation of the liquid fuel, recovering at the same time a non-negligible amount of thermal power for other uses.

In the testing phase described, a specific monitoring of the environmental parameters related to pollutant emissions was not carried out, since such characterization is planned for the next phase of the work.

6. Discussion and conclusions

The aim of this paper is to assess the technical feasibility and the energy performance of a prototype multi-fuel micro-CHP conversion kit.

The kit enables to transform the existing single-fuel generators (or, more generally, the Otto-cycle internal combustion engines equipped with a carburetor) in multi-fuel ones, thanks to a simple, robust and cost-effective technical solution. The application of the kit also allows to perform a recovery of the residual heat produced by the engine used as the application basis, allowing therefore the CHP operation. The aim of the research was therefore to demonstrate, theoretically and experimentally, the possibility

Table 4
Test and performance parameters.

		Performance test 1		Performance test 2	
		LPG	Ethanol 95%	LPG	Ethanol 95%
Low heating value	LHV (kJ/kg)	46	25.74	46	25.74
Vaporization enthalpy	H_v (kJ/kg)	426	842	426	842
Fuel rate consumption ^a	$FR_{m,f}$ (kg/h)	4.7	8.2	3.46	5.82
Temperature of the exhaust at cylinders outlet ^a	$T_{e,c}$ (K)	990	964	990	929
Temperature of the exhaust at heat exchanger outlet ^a	$T_{e,he}$ (K)	422	420	419	421
Exhaust temperature difference across heat exchanger ^a	ΔT (K)	568	544	571	508
Average air specific heat	SH_a (J/kg K)	1060	1059	1060	1056
Fuel power input ^a	$P_{i,f}$ (kW)	52.4	52.2	34	34.3
Exhaust recoverable thermal power ^a	$P_{r,e}$ (kW)	11.3	12.2	7.3	8.1
Thermal power required for the fuel vaporization ^a	P_v (kW)	0	1.7	0	1.1
Real electrical power ^a	P_e (kW)	9	9	4.5	4.5
Thermal power available for other purposes ^a	P_a (kW)	11.3	10.5	7.3	7
Electrical efficiency (on real electrical power) ^a	η_e (%)	16.8	16.8	13.2	13.1
Overall thermal and electrical efficiency ^a	η_t (%)	38.3	36.9	34.7	33.6

^a Average value recorded during the testing period.

of converting in a simple, cheap and effective way, small-size generation systems which are typically used for off-grid applications, with particular reference to developing Countries, increasing their flexibility and efficiency of use.

The results obtained through the prototyping phase showed that it is possible to effectively apply the conversion kit to a commercial generator set; experimental tests allowed instead to check the proper functioning of the system over a limited period of time and to observe that the performances are broadly consistent with the calculated results. The kit developed ensured in fact the smooth functioning of the power generator with ethanol at 95°, allowing to increase the overall efficiency, thanks to the possibility of recovering a fraction of the thermal energy produced. In this regard, it is important to underline that the performances obtained are not comparable with those related to systems specifically designed as micro-CHP systems, but they should be compared with the generation efficiency of the component used as an application basis, before its conversion; the application of the kit allowed in fact to double the overall generation efficiency (first law efficiency), in all evaluated cases, compared to the original values before the conversion, reaching values close to 40% in rated load conditions.

The subsequent phases of the research will cover the complete characterization of the prototype developed, also from the environmental and economic points of view, by carrying out performance tests with different boundary conditions and over longer periods, also in order to assess the strain of the components. The application of the kit to a generator equipped with a liquid-cooled engine is also planned, in order to analyze the possibility of a further increase of the overall system efficiency: the additional recoverable energy can be quantified in a further 30–35% of the fuel power input.

In conclusion, it is possible to assess that, on the basis of the results obtained, the applicative potential of the kit are significant in all the contexts in which common, small size, Otto-cycle generation systems are present.

Additional tests will be carried out in the forthcoming part of the research in order to better frame the potential application of the kit.

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