Experimental Characterization of a Small Cogeneration Plant at Full and Partial Loads

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ABSTRACT

This article presents the results of an experimental analysis performed on a small cogeneration plant equipped with a microturbine as a prime mover. The aim of the experimental campaign was to verify behavior and performances of the machine at both full and partial load. Attention was particularly paid to the energy saving indexes defined in the AEEG 42/02 Directive (Italian Authority for Electrical Energy and Gas) and in the more strict EU Directive 2004/8/EC. The results of the experimental campaign show that the cogeneration plant can easily exceed the minimum values of the energy saving index imposed by the current directives during operation at full load. On the contrary, very accurate design of the plant and prediction of the load profiles are required to get acceptable results if operation at partial load becomes necessary.

Key Words: Cogeneration, microturbine, experiment, energy saving index, savings verification, off-design operation, cogeneration regulation.

INTRODUCTION

In order to meet the national energy demand, power generation is traditionally obtained by a few large plants, typically processing fossil or nuclear fuels. However, a more recent approach, named "distributed generation" (DG), is based on a lot of small plants, spread on the country and using fossil fuels or renewable sources [1].

Both in the traditional and in the distributed power generation systems, significant improvement in the overall efficiency of fossil-fuel plants can be reached if power generation and heat recovery are joined together. The so-called cogeneration can be obtained using both a "topping" or a "bottoming" approach. In the former and more used, the first aim is power generation, but the wasted heat is recovered instead of being discharged into the cold sink. In small scale plants, characterized by nominal power between 50 kW and 1 MW, gas microturbines appear to be very promising as prime movers [2].

Large-scale cogeneration plants are usually installed on sites where power and thermal demands are constantly present, so that the plant can be operated at rated conditions. On the contrary, small-scale and micro-cogeneration plants (electrical power < 1 MW) are mostly installed in the tertiary and residential fields, where electrical and thermal loads are subjected to relevant variations, due to the discontinuous nature of the end user demand. This implies that such systems usually work in conditions which can be even very far from the nominal ones, resulting in a not cost-effective operating policy. Hence, plant optimization, in terms of both design and operation, is of the maximum importance to achieve relevant energetic and economic savings as well as pollution reduction [3].

In considering energy sources, it should be noticed that natural gas is today and will be more and more in the future the fuel of reference for the generation of both electrical and thermal power. Moreover, thermodynamic performances in the present-day use of natural gas are modest, particularly concerning heat production at low enthalpy which represents slightly less than 50% of gas consumption. Thus, it is evident that small- and micro-cogeneration could lead to a more rational use of fuel and consequently to a reduction of its global consumption.

The use of smaller gas turbines (<500 kW_{el}) may be considered an innovation; actually, this technology results really cost-effective only for a power generation greater than 5-10 MW_{el} [4]. However, microturbines are not conceived simply as the scaling down of bigger machines, but they present innovative features, such as the use of radial turbo-machinery—a cheaper solution—operating at very high rates (50000-120000 rpm) in regenerative cycles [5]. In fact, at present time, cogeneration is the essential condition to make advantageous the use of microturbines. References on this subject are wide-ranging and only a very little sample is listed in bibliography [6]-[10].

On the basis of these considerations in the present article an experimental characterization of a cogeneration plant based on a microturbine is presented in order to assess its performances from an energetic standpoint. Specifically, the aim of the experimental campaign was to characterize the response of the microturbine when it has to face imposed variations of both electrical and thermal load, with a particular focus on its behavior during operation far-from-the-nominal design or off-design conditions.

EXPERIMENTAL SETUP

A sketch of the cogeneration plant, located in Milan at the Italian research center RSE S.p.A. (formerly ERSE S.p.A.), and the nomenclature used in this article are shown in Figure 1. The plant is essentially composed of a microturbine (mT), a gas compressor, a dry cooler, a plate heat exchanger and a control panel. The microturbine Turbec T100 is able to produce up to 167 kW heat power, 105 kW electrical power at ISO conditions. Figure 1 also shows the main points where heat and flow variables will be measured for the experiment. Complete technical specifications may be found in [4].

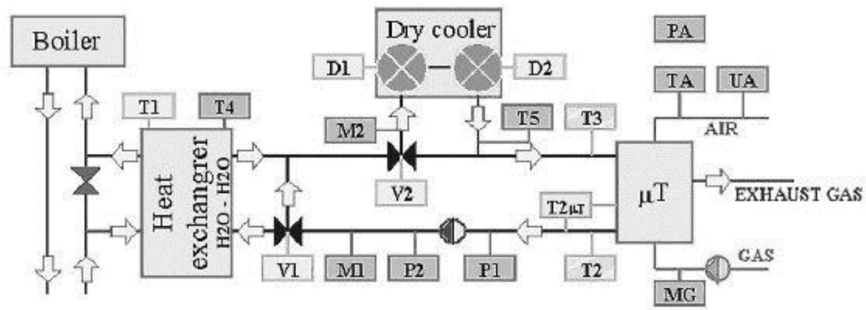
The hot water circuit consists of two separable rings:

1- The plate heat exchanger line, which connects the microturbine with the district heating of RSE S.p.A.;

2- The dry cooler line, which simulates a variable heat load. This circuit, used for the experiment, is the heat sink which dissipates all the thermal power produced by the microturbine unit. Its main component is an air-water heat exchanger. It's provided with two fans the operation of which can be regulated to obtain the desired heat dissipation.

A proportional, integrated, derivative control (PID) is implemented in the PLC that works up the information of the transducers and regulates the opening of the valves V1 and V2 and the operation of the fans D1 and D2. The PLC is connected to a PC through a LAN and the microturbine is linked to the same PC through a web server. A user friendly graphical interface was implemented by N. I. Labview®. The control system enables the definition of different operating conditions in order to simulate different heat loads and study the consequent response of the microturbine. In particular the following programs are available:

- Program B (base or security program): it is an automatic procedure which ensures that the dry cooler dissipates all the heat power generated;



Simbol	Element	Model	Function
D1	Dry cooler - Fan 1	Alfa Laval	Thermal load
D2	Dry cooler - Fan 2	Alfa Laval	Thermal load
V1	Three port valve 1	Siemens VFX 31	Admission to the heat exchanger
V2	Three port valve 2	Siemens VFX 31	Admission to the dry cooler
T1	Thermoresistance Pt100 2 wires	Siemens QAE22	Water outlet temperature of the heat exchanger water-water (side District heating circuit)
T2	Thermoresistance Pt100 4 wires	Electrotherm	Water outlet temperature of the exhaust gas heat exchanger of the microturbine

Figure 1. Plant setup and main points of measurement and nomenclasure.

$T2_{\mu T}$	Thermoresistance Pt100 2 wires	Plate sensor	Water outlet temperature of the exhaust gas heat exchanger of the microturbine (measured by the microturbine)
T3	Thermoresistance Pt100 4 wires	Electrotherm	Water inlet temperature of the exhaust gas heat exchanger of the microturbine
T4	Thermoresistance Pt100 4 wires	Electrotherm	Water outlet temperature of the heat exchanger water-water
T5	Thermoresistance Pt100 4 wires	Electrotherm	Water outlet temperature of the dry cooler
TA	Thermoresistance Pt100 2 wires	Siemens QAE22	Air temperature
P1	Pressure transducer	Nagano ADZ	Relative pressure before pump
P2	Pressure transducer	Nagano ADZ	Relative pressure after pump
PA	Pressure transducer	Nagano ADZ	Air pressure
UA	Humidity transducer	Electronik EE20-FT6	Air relative humidity
M1	Electromagnetic flow meter	Badger Meter	Outlet water flow of the microturbine
M2	Electromagnetic flow meter	Badger Meter	Inlet water flow of the dry cooler
MG	Mass flow meter	Fox Thermal Instr.	Inlet gas mass flow of the turbine

Figure 1. Plant setup and main points of measurement and nomenclasure.

- Program A1: the electrical power produced and the water temperature at the outlet of the microturbine (T_2) are set manually; all the heat can be dissipated. The valve V1 is completely opened towards the air dryer and both the fans D1 and D2 are set on. The PLC regulates the opening of the valve V2 in order to maintain the set-point value of temperature T_2;

- Program A2: the electrical power set-point, the heat load to be dissipated and the water temperature at the outlet of the microturbine are set manually, by acting on the opening of the valve V2 and on the value of $T_{2\mu T}$. Consequently, the microturbine is forced to regulate the electrical power to follow the required heat generation.

The experimental setup allows the real-time calculation of the following thermodynamic quantities:

combustion heating power, $Q_{fuel} = MG \cdot LHV_{fuel}$;

cogeneration heating power, $Q_{th} = M_1 c_p \left(T_2 - T_3\right)$;

heat load at the air dryer, $Q_{th} = M_2 c_p \left(T_2 - T_5\right)$;

electrical efficiency, $\eta_{el} = \dfrac{P_{el}}{Q_{th}}$;

thermal efficiency, $\eta_{th} = \dfrac{Q_{th}}{Q_{fuel}}$;

first law efficiency, $\eta_{tot,I} = \dfrac{P_{el} + Q_{th}}{Q_{fuel}}$;

thermal limit, $LT = \dfrac{Q_{th}}{P_{el} + Q_{th}}$

primary energy saving index, $PES = 1 - \left(\dfrac{\eta_{el}}{c \cdot \eta_{el,ref}} + \dfrac{\eta_{th}}{\eta_{th,ref}} \right)^{-1}$

where the reference values for the efficiencies and the grid loss correction terms were assumed according to both the Italian AEEG 42/02 Directive (indicated in the following as PES_1) and the more strict EU Directive 2004/8/EC (indicated in the following as PES_2).

EXPERIMENTAL PLAN

Test conditions were designed in the hypothesis to use the microturbine for residential or tertiary applications and considering that higher efficiency is attained during operation at nominal conditions. According to the available operating programs, the experimental plan was organized for both electrical and thermal tracking tests, as follows:

Electrical tracking tests, based on program A1. The electrical power generated (P_{el}) was varied in the range 50-110 kW, with 10 kW span. For each step, the set point of the temperature at the outlet of the microturbine (T_2) was varied in the range 60-80°C, with 5°C span; values lower than 60°C are unsuitable as thermal demand and were not considered.

Thermal tracking tests, based on program A2. The set-point temperature at the outlet of the microturbine $T_{2\mu T}$ was varied in the range 70-90°C, with 10°C span. Starting from an electrical power set-point, ranging between 80 and 105 kW, with 10 kW step, the heating power dissipated was varied from 0% to 100%, with 20% span. The valve V1 was always closed toward the district heating line, such that all the available thermal power was dissipated by the fans D1 and D2. Hence, the thermal load variation was provided by regulating the three-way valve V2 at the inlet of the dry cooler. The time needed to reach stationary conditions was about 15 minutes. It is worth noting that only the results for 0% and 100% thermal load will be presented because all the partial load conditions ranging between them follow the same transient as the 0% condition.

ANALYSIS OF RESULTS

Electrical Tracking Tests

These tests are suitable to obtain an energetic characterization of the plant. All the measured quantities are reported in diagrams as functions of the electric power P_{el}, parameterized by the heat exchanger water outlet temperature T_2. To improve readability, error bars are not reported and the maximum standard deviations of the presented data presented are here summarized: 0.63% for η_{el}, 3.01% for η_{th}, 5.96% for $\eta_{tot,I}$ and 5.69% for PES_1.

Figure 2 shows the electrical efficiency η_{el}. It is noticed that, in the range 90-110 kW, η_{el} is greater than 31%, a good value for microturbines technology. Reducing the power output P_{el}, the electrical efficiency decays almost linearly until about 60 kW, then it seems to fall down; at 50 kW η_{el} is equal to 24.5%. As expected, the electrical efficiency is independent of the water temperature.

Inspection of Figure 3 shows that the thermal efficiency η_{th} may reach a maximum in the range between 80 and 100 kW. This behavior might be an artifact produced by oscillations in the temperature of the exhaust gases entering the heat exchanger after the expansion. It is worth noting

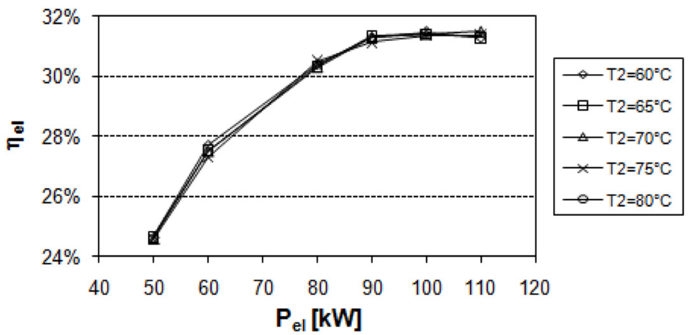


Figure 2. Electrical efficiency versus electrical power.

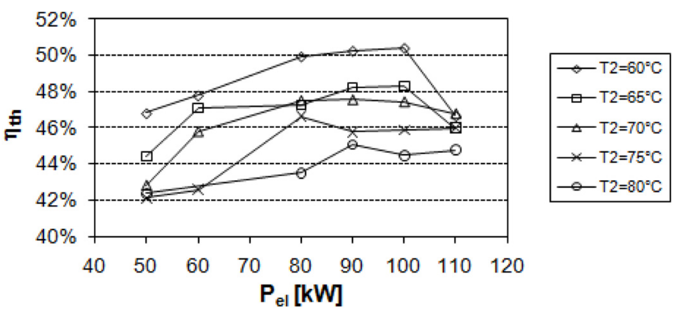


Figure 3. Thermal efficiency versus electrical power.

that thermal efficiency drops with increasing the water temperature. For example the difference between the values of η_{th} at water temperatures of 60 °C and 80 °C respectively is 5%. This may be explained in considering that the heat exchanger efficiency lowers/improves by reducing/increasing the larger temperature difference between water and exhaust gases.

The behavior of the total first law efficiency $\eta_{tot,I}$ depicted in Figure 4 is consistent with the above considerations. In particular the relevant decrease for partial electrical loads and the strong dependence on the water temperature should be noticed.

From Figure 5 it is seen that, with the microturbine operating at full electric load, about 30% of the primary energy can be saved. This value remains essentially constant by reducing the electrical power production until 80 kW; then it decreases at a rate that seems to be greater for increasing values of the water temperature. Moreover at about 60 kW a strong

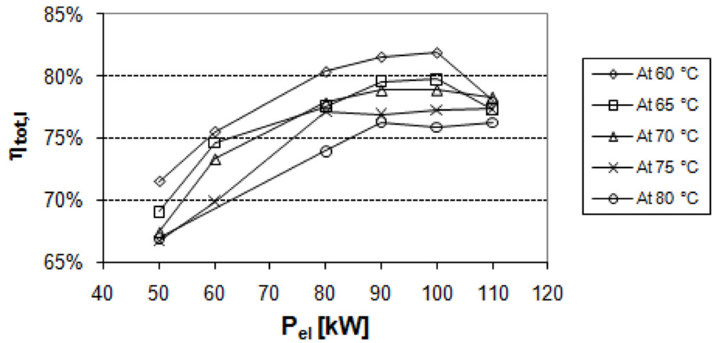


Figure 4. Total first law efficiency versus electrical power for various T_2

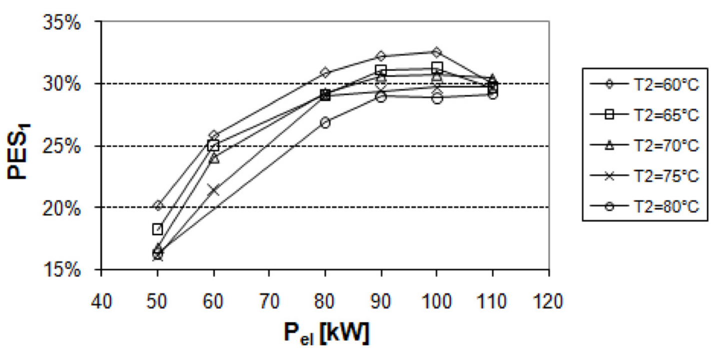


Figure 5. PES_1 versus electrical power.

diminution occurs and PES_1 reaches a minimum value between 16% and 20% depending on T_2, at 50 kW.

The behavior of PES_2 is qualitatively similar to that of PES_1. Quantitative considerations will be presented in the following sections.

Thermal TrackingTests

These tests allow to assess the ability of the microturbine to follow a variable thermal load in operating conditions even far from the nominal ones. According to the operating procedure described above (program A2), the results relative to full thermal load (valve V2 completely opened) and partial thermal load (valve V2 closed) are presented separately.

Tests at full thermal load

In Figure 6 the electrical power is reported versus the measured mass flow rate of the exhausted gases. At the nominal operating conditions (105 kW_{el}) the mass flow rate ranges between 0.74 and 0.78 kg/s, which agrees fairly well with the design data (0.76 kg/s). As expected, if the electrical power set-point is reduced, the mass flow is seen to decrease at a rate which increases by decreasing the power output. The average value of the mass flow rate is reported in Table 1.

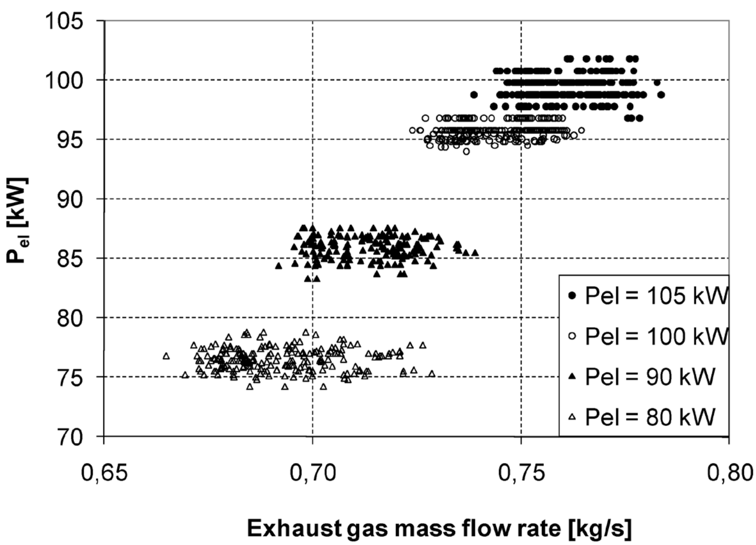


Figure 6. Electrical power versus exhaust gas mass flow rate.

Table 1. Average mass flow rate of the exhaust gases.

P_{el} [kW]	Average exhaust mass flow rate [kg/s]
80	0.700
90	0.719
100	0.742
105	0.757

Figure 7 shows the electrical efficiency (η_{el}) as function of thermal efficiency (η_{th}). On the same plot the relationship between η_{el} and η_{th} are reported for different values of the thermal limit (LT) and the energy saving index PES_1.

It can be seen that if the power output is reduced to 80% of the rated value, the thermal efficiency lowers only slightly, about 1.5%. Moreover, even if the power output is decreased, PES_1 is about 18% which is much greater than the lower limit (10%). On the other hand, the thermal limit is 0.64, that is, about twice the lower limit (0.33). Such values allow to define the system as cogeneration in compliance with the AEEG 42/02 regulation.

The AEEG 42/02 regulation defined the reference energy saving indexes for cogeneration plants up to December 31st, 2010. On such a date the new regulation 2004/8/CE will come into force and PES_2 will be used to define the lower limit for a plant to be considered as cogeneration: for small-scale plants (P_{el} <1 MW), $PES_2 > 0$.

Even though the new regulation is more strict than the previous one, PES_2 values correspond practically to the lower limit, such that the plant can be considered as cogeneration.

Tests at Partial Thermal Load

During operation at partial thermal loads a marked reduction of the exhaust gas flow rate is observed because thermal load is set to zero. The following figures refer to a test condition characterized by 100 kW electrical power set-point and 90°C output water temperature.

In Figure 8 the electrical power is shown as a function of the rotational speed during the whole transient: the behavior is almost linear.

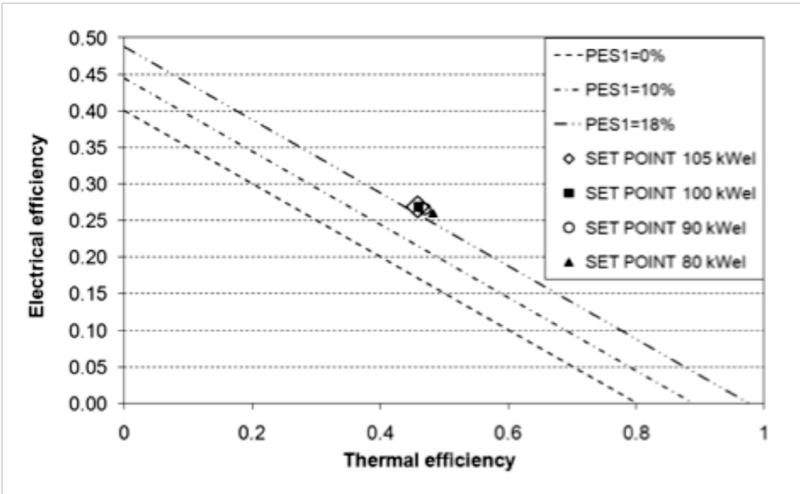


Figure 7. Electrical versus thermal efficiency for stationary operation at full thermal load

A similar result is observed for the exhaust gas mass flow rate, which is strictly linked to the rotational speed (Figure 9). The behavior agrees with the characteristic curves of the turbine.

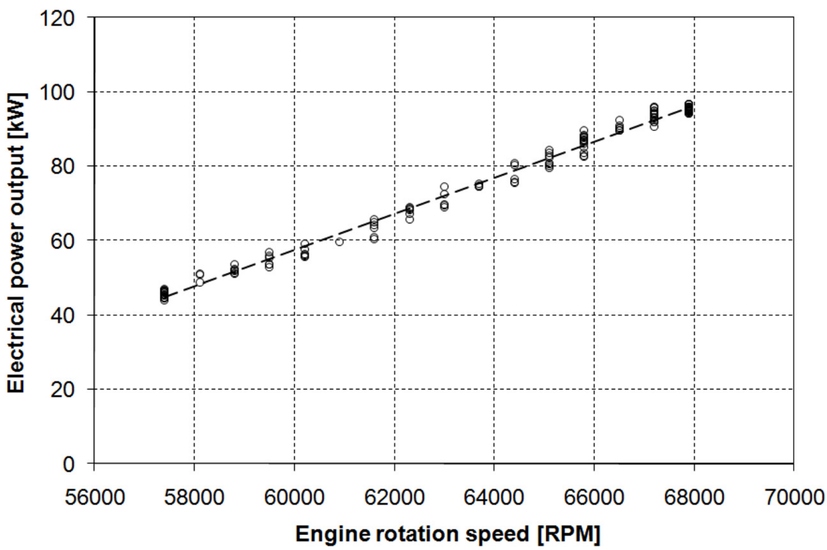


Figure 8. Electrical power output versus engine speed.

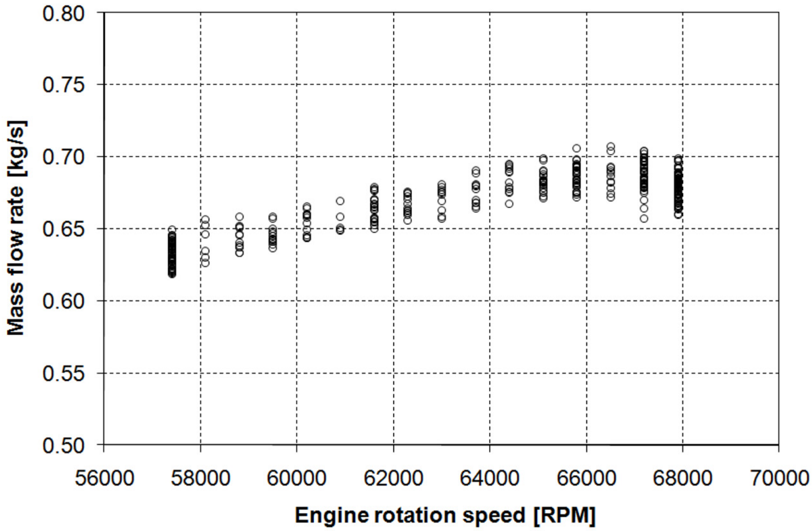


Figure 9. Exhaust mass flow rate versus engine rotation speed.

Concerning the exhausts temperature at the exit of the recuperator (Figure 10), it is seen to vary almost linearly with the electrical power output and a relevant decrease (up to 30°C) is observed during operation at partial loads.

In Figure 11 the behavior of the electrical efficiency η_{el} as a function of the electrical power output is reported. At full load η_{el} is about 28% and it decreases almost linearly till 24% at 50% partial load.

It is interesting to observe the variation of the energy saving indices during operation at partial load (Figure 12). In particular, PES_1 decreases from 18% to about 10% which is the lower limit according to the Italian regulation AEEG 42/02. It can be noticed a cloud of data corresponding to PES_1 = 18%, on the right of the PES_1 = 10% line. These data correspond to the beginning of the transient, when a fictitious increase of the thermal efficiency is observed due to the fact that the produced thermal power is not dissipated. Actually, at such conditions the system still operates at full load, because the control system has not yet started the regulation of the engine speed. On the other hand, the value of the thermal limit ranges from 0.60 to 0.66 and results always much greater than the prescribed lower limit. However, concerning the new 2004/8/CE regulation, the condition about the PES_2 lower limit is never satisfied, being the values always negative (up to –10%).

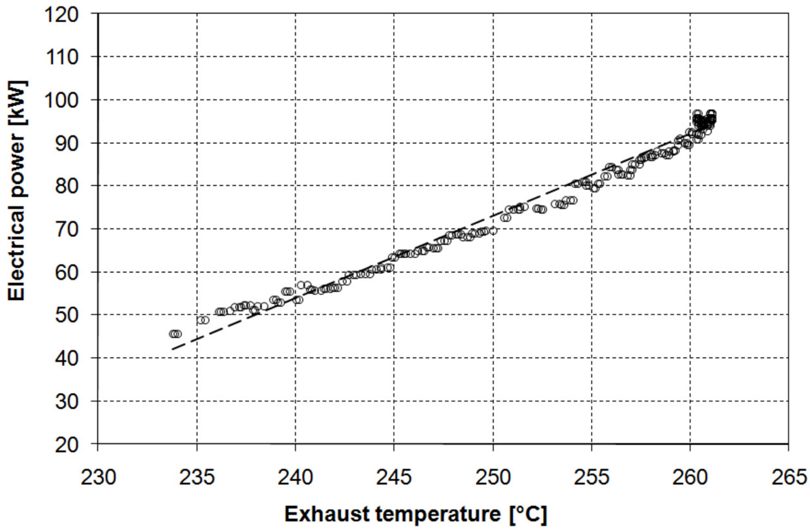


Figure 10. Electrical power output versus exhaust temperature.

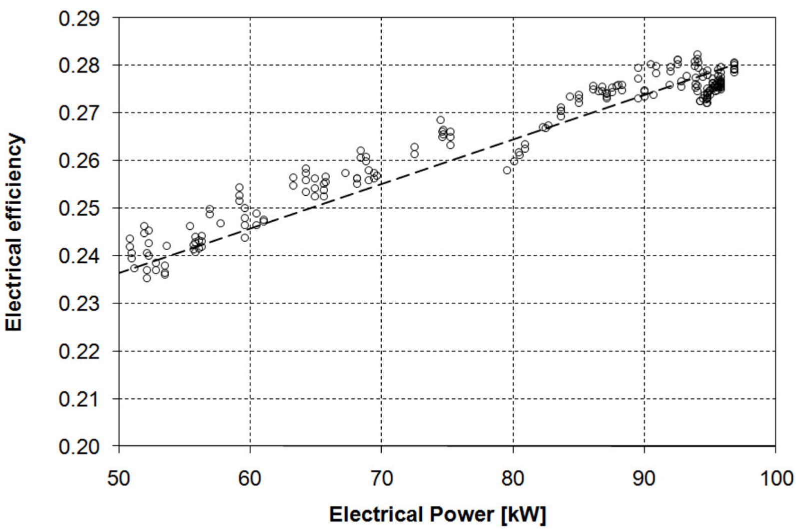


Figure 11. Electrical efficiency versus electrical power output.

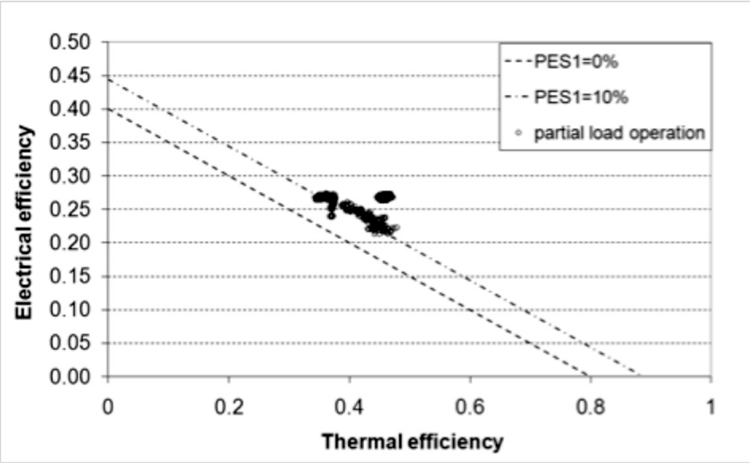


Figure 12. Electrical versus thermal efficiency for partial thermal load operation.

CONCLUSIONS

In this article the results of an experimental analysis on a small cogenerative plant based on a microturbine have been presented and discussed. In particular, attention was devoted to the operation at both partial electrical and thermal loads.

Tests at variable electrical load revealed that performances remain essentially constant in the range 80-110 kW. A moderate decrease is then observed until about 60 kW, while a further reduction of the electrical power implies a clear worsening.

Tests at variable heat loads clearly showed that if the power output is reduced from the nominal value towards the lower operating limit (50% partial load), the microturbine performances decay progressively. Such decrease is not dramatic for most of the parameters, but it has a heavy effect on the achievable energy saving.

Concluding, the results of the experimental campaign show that a cogenerative plant equipped with a microturbine as a prime mover, during operation at full load, can easily provide significant energy savings in comparison to the separate electricity and heat production. On the contrary, to get acceptable results in case of variable load operation, very accurate design of the plant and proper consideration of load profiles are required.

Acknowledgments

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