INFLUENCE OF HEAT DEMAND ON TECHNO-ECONOMIC PERFORMANCE OF A BIOMASS/NATURAL GAS MICRO GAS TURBINE AND BOTTOMING ORC FOR COGENERATION

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ABSTRACT

This paper presents a thermo-economic analysis of small scale Combined Cycle power plant composed by a Micro Gas Turbine (MGT) and a bottoming Organic Rankine Cycle (ORC) for cogenerative application. For the topping cycle three different configurations are examined: 1) a simple recuperative micro gas turbine fuelled by natural gas, 2) an externally fired gas turbine (EFGT) with direct combustion of biomass, and 3) a dual fuel EFGT cycle, fuelled both by biomass and natural gas.
For the bottoming cycle, a saturated recuperative Rankine cycle is examined under two different condensation temperatures, in order to vary the ratio of heat to power generated. The research assesses the global energy efficiency and profitability of the different schemes, as a function of the thermal energy demand intensity, represented by the annual equivalent heat demand hours.

Keywords: CHP, microturbine, EFGT, ORC, biomass, dual fuel

1. INTRODUCTION

Small scale CHP (Combined Heat and Power) generation can contribute to a number of energy and social policy aims, such as the reduction of greenhouse gas (GHG) emissions and fossil fuel depletion, reduction of energy costs, increased decentralization of energy supply and improved energy security.
The use of biomass as an integrative energy source in natural gas fired plants has been widely addressed in literature [EC, 2009a,b; Franco and Giannini, 2005]. In small size range (50 kWe- 1 MWe), one interesting option is the use of dual fuel biomass/natural Micro Gas Turbines (MGT), in particular by means of Externally Fired Gas Turbines (EFGT) [Camporeale *et al.,* 2013]. A bottoming ORC could be coupled to the MGT in order to increase the electric efficiency of the system. The ORC are much more suited than conventional steam plants for small and micro generation from a few dozen to some hundreds kWe, because of the low enthalpy drop that allows a turbine expansion through few stages. There is a large literature on ORC cycles and in particular on the fluid selection for waste heat recovery applications (Chen *et al.,*2010; Invernizzi *et al.,* 2011; David *et al.,* 2011). In the present work, different firing schemes are examined for the MGT: only biomass (B), only natural gas (NG) and dual fuel (B and NG) are considered as energy input and compared, as proposed in *Pantaleo et al, 2014* and Camporeale *et al.*2014. The investment profitability is appreciated on the basis of thermo-economic methodologies [Pantaleo *et al*., 2013, Al-Sulaiman *et al*., 2013, Galanti *et al*. ,2013, Ferreira *et al*., 2012, Pantaleo *et al*. 2014], with specific costs assessment proposed in [Pantaleo *et al.* 2013] and in light of the Italian policy measures for renewable energy and high efficiency CHP [Ministry Decrees, 2011, 2012].

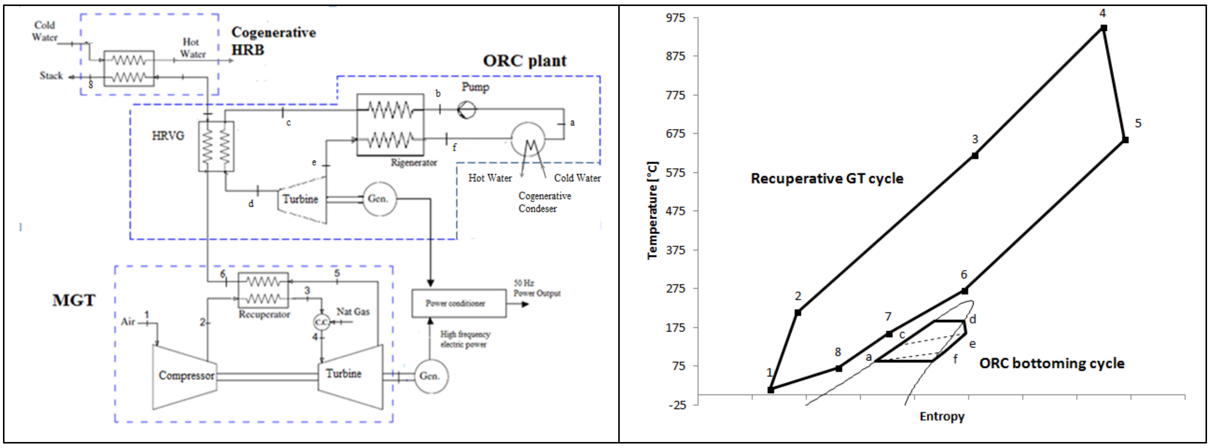

Figure 1 - MGT-ORC combined cycle for CHP generation, and related T-s diagram.

2. TECHNOLOGY OVERVIEW

2.1 Micro gas turbines fuelled by natural gas

Among distributed generation technologies, micro gas turbines (MGT) are expected to have steady growth in future energy systems, mainly for CHP applications. MGTs are typically single-shaft engines (Fig. 1), where compressor, turbine and electric generator have a common shaft rotating at high speed (typically between 60,000 and 90,000 rpm [Rosa do Nascimento *et al.*, 2013]). The high-frequency current from the generator is converted to grid frequency by an inverter, which enables variable-speed operation [Hamilton, 2003]. The turbine inlet temperatures (TIT) are typically in the range 800–1,000 °C, since no blade cooling systems are adopted, while the pressure ratio is low (3.5-5). A surface regenerative heat exchanger (recuperator) is used to increase the electric efficiency that can reach values of 30%. The turbine outlet temperature TOT is about 500-650°C, hence the material costs for the recuperator can be kept at a reasonable level. The total energy efficiency of a CHP-MGT is in the range 70–80%, and is influenced by the temperature of heat demand.

2.2 Externally fired micro gas turbines fuelled by biomass

Since several years, the use of biomass in MGT is considered a profitable option [Obernberger, 1998] for renewable energy production, however one of the key technical issues is the fuel quality. The EFGT cycle presents the advantages of gas turbines (low operational costs, high lifetime and reliability, relatively high energy efficiency even at small size) and the capability of using low quality biofuel. In the conventional scheme of an EFGT, the biomass is fed to an external furnace together with hot air coming from the turbine exhaust, and the turbine is fed by hot compressed air, heated in a high temperature heat exchanger (HTHE) to the required TIT by the hot gas of the biomass combustion. In this work, the following scheme has been selected: external combustion of biomass in a furnace, and heating of the MGT cycle working fluid (air) by means of a surface heat exchanger.

2.3 Natural gas/biomass dual fuelling

The external combustion of biomass combined with direct combustion of natural gas (often referred as "dual-fuel" or "cofiring") is a promising, cost-effective and reliable small scale generation system, that offers plant flexibility, high conversion efficiency and possibility to use commercially available components. In conventional EFGT cycle the turbine inlet temperature is limited by material of the HTHE to 850–900 °C [Riccio *et al.* 2009; Yan and Eidensten, 2000; Ferreira and Pilidis, 2001; Knoef, 1998; Soltani *et al.*, 2013; Evans and Zaradic, 1996; Savola *et al.,* 2005; Cocco *et al.,* 2006; Riccio *et al.*, 2000, Camporeale *et al.,*2014]. For this reason, a dual fuel configuration (biomass-natural gas) is investigated. With the internal combustion the TIT can reach a maximum TIT of about 950° compatible with the metal of the turbine blades, and consequently a higher efficiency and power output of the conversion process, with respect to only biomass input.

Table 1. Technical parameters for the topping MGT (AE Turbec T100, 2015)

Case study	Unit	100% NG	50-50% NG-B	100% B
Net electric power output (ISO)	kW	100.1	89.6	77.5
Turbine Inlet Temperature	°C	950	950	900
Energy input	kW	332.9	373.4	404.0
Net electric-efficiency. ISO	%	30.1	24.0	19.2
Gas temperature at turbine exit	°C	652.7	656.4	609.0
Exhaust gas temper. (recuperator exit.)	°C	270	272	262
Air mass flow rate	kg/s	0.7833	0.7833	0.7833

2.4 MGT+ORC combined cycles

The use of combined cycle schemes can increase the electric efficiency on respect to either two plants that compose the combined cycle. In this work, we consider a combined cycle composed by a MGT as topping cycle and an ORC as bottoming cycle, which converts part of the heat from the exhaust gas in useful work (Fig.1). In this scheme, the exhaust gas exiting the gas turbine is conveyed to a heat recovery heat exchanger that heats the organic fluid from liquid to saturated vapor or superheated vapor, depending on the chosen ORC cycle. It is composed by an economizer, an evaporator and, if present, a superheater. In analogy to gas-steam combined cycles, it is called Heat Recovery Vapor Generator (HRVG). In the HRVG, the organic fluid is brought to the thermodynamic condition requested for the admission in the turbine, while, on the other side, the gas exiting the HRVG has still a temperature suitable for cogeneration. Such heat is recovered in a cogenerative HRB where the gas can be cooled to a temperature that depends on the fluid temperature requested by the industrial or residential users. In the adopted configuration, a minimum gas temperature of 80°C is adopted. This limit can be applied either to MGT fuelling natural gas or EFGT fuelling biomass. In fact, in the adopted configuration of the biomass furnace, the products of biomass combustion do not contaminate the gas that flows across the turbine.

The bottoming cycle is an ORC in a recuperative configuration (Fig. 1). We considered the use of "dry fluids" that are characterized by a dry expansion in the turbine, avoiding drop generation that can damage turbine blades [Chen *et al.,* 2010]. In particular, the cycle contains a pump (a-b) that supplies the fluid to the recuperator (b-c). The recuperator pre-heats the working fluid using the thermal energy from the turbine outlet. The evaporator produces the evaporation of the organic fluid up to the requested condition, by recovering the heat from the topping cycle (c-d). Thus, the vapor flows in the turbine (d-e) connected to a high-speed electric generator. At the exit of the turbine, the organic fluid goes to the hot side of the recuperator (e-f) where it is cooled to a temperature a little higher than the condensation temperature. Finally, the condenser closes the ORC cycle (f-a). Two options are investigated for the condensation temperature:

- Option ORC1: low condenser temperature (40°C), Figure 2.(a); in this case the electric efficiency of the bottoming cycle is higher, but heat rejected by the ORC cannot be used for cogeneration;
- Option ORC2: high condenser temperature (100°C), Figure 2(b); in this case, the ORC output is lower, but the heat rejected at the condenser is useful for residential heating.

In both options, useful heat can be also recovered from the gas at the exit of the HRVG.

For the ORC cycle, considering that the exhaust gas temperature at the gas turbine outlet (recuperator exit) is 270°C, siloxanes and toluene are examined as suitable working fluids. The choices of thermodynamic parameters of the ORC cycle are related to the temperature of the heat to be recovered. In particular, turbine inlet temperature and evaporation pressure are the most influential properties. In relation to the choice of the evaporation pressure, p_{ev}, and of the maximum ORC cycle temperature, it is possible to examine saturated, superheated and supercritical cycles. A detailed discussion of the optimization of the cycle parameters can be found in [He *et al.* 2009; Kusterer *et al.* 2010]. Here some results can be summarized. The higher is the evaporation pressure in supercritical and subcritical cycles, the higher is the cycle efficiency. However, when the critical temperature is close to the heat source temperature, lower heat recovery efficiency is found. Furthermore, these cycles are characterized by high inlet / outlet turbine volumetric ratio. Therefore, the advantages of ORC cycles with very high-pressure levels are limited. For the above reasons, a simple recuperative ORC cycle is examined with a hexa-methyl-disiloxane "dry fluid".

Table 2. Technical parameters for the bottoming ORC cycles (topping cycle MGT fuelling NG)

Description	Unit	ORC 1	ORC 2
Evaporation pressure	bar	8	8
Turbine inlet temperature	°C	192	192
Condenser temperature	°C	40	90
Available Thermal Flow from exhaust gas, $\dot{Q}_{av}$	kW	199.7	199.7
HRVG Heat flow, $\dot{Q}_{rec}$	kW	105.0	95.2
Heat recovery ratio, χ	%	52.6	47.8
Electric power output, $P_{el.ORC}$	kW	20.0	11.4
Cycle Efficiency, η_L	%	19.9	11.9
Overall efficiency as recuperative cycle, η_{rec}	%	10.0	5.7
Heat rejected from condenser	kW	83.44	79.4

3. PERFORMANCE ANALYSIS

The biomass/natural gas energy input ratio has been varied and the following schemes of the topping cycle have been examined: 100% of natural gas (NG), dual fuel scheme with 50% of energy input from biomass and 50% from natural gas (DF) and, finally, 100% of energy input from biomass (B).
Thermodynamic simulations have been carried out by means of Gate-Cycle® software for the MGT and Cycle-Tempo® for the ORC section. Gate Cycle is a commercial software for thermal power plants including gas turbines, steam and combined cycles plants [https://getotalplant.com/GateCycle, 2015]. The Cycle Tempo software is used for thermodynamic cycles and in particular organic and other non conventional fluids [http://www.asimptote.nl/software/cycle-tempo/, 2015]. Both the codes have been tested and validated by the authors. In the following, the methodologies for: (i) energy balances and efficiency analysis, (ii) primary energy savings and profitability of investment are reported.

3.1 First Law analysis and Definitions

All calculations are performed for ISO standard conditions (15 °C, 1.013 bar and 60% relative humidity). Based on the cycle thermodynamic analysis, the following equations are used to evaluate the net electric power output (P_e), the total thermal power input ($\dot{E}_{in,tot}$), electric efficiency (η_e), thermal power supplied to hot water for cogeneration ($\dot{Q}_{th}$), thermal efficiency (η_{th}), total ("first law") efficiency for CHP generation (η_{CHP}).
The overall electric efficiency of the combined cycle $\eta_{e,cc}$ is reported in (1), where $P_{e,CC}$ is the electric power output of the combined cycle as reported in (2).

$$\eta_{e,cc} = \frac{P_{e,CC}}{\dot{E}_{in,tot}} \qquad (1) \qquad\qquad P_{e,CC} = P_{e,GT} + P_{e,ORC} \qquad (2)$$

The energy input is due to the combustion of NG and biomass according to (3).

$$\dot{E}_{in,tot} = \dot{E}_{in,NG} + \dot{E}_{in,biom} = \dot{m}_{NG} LHV_{NG} + \dot{m}_{biom} LHV_{biom} \qquad (3)$$

The energy performance of the bottoming cycle is evaluated from (i) the "internal thermal efficiency" of the bottoming cycle η_L of eqn (4), being $P_{e,ORC}$ the power output of the bottoming cycle and $\dot{Q}_L$ the heat input to the bottoming cycle (heat recovered in the HRVG), and (ii) the "heat recovery ratio" or "HRVG efficiency χ, defined as the ratio between heat recovered $\dot{Q}_L$ and heat rejected from the topping cycle $\dot{Q}_{HR}$.

$$\eta_L = \frac{P_{e,ORC}}{\dot{Q}_L} \qquad (4) \qquad\qquad \chi = \frac{\dot{Q}_L}{\dot{Q}_{HR}} \qquad (5)$$

The heat rejected is here the sensible heat of the gas exiting the gas turbine, which is the theoretically available if the gas were cooled to the ambient temperature. Due to this reason, $\dot{Q}_{HR}$ is also called "heat available" and indicated as $\dot{Q}_{av}$. The heat exchanged $\dot{Q}_L$ in the HRVG, is evaluated in Eq.(6) from gas the temperature drop $(T_6 - T_7)$, the mass flow rate $\dot{m}_g$ exhausted by the gas turbine and the average specific heat $c_{p,g}$ of the exhaust gas. The available heat is given by Eq. (7).

$$\dot{Q}_L = \dot{m}_g\ c_{p,g}(T_6 - T_7) \qquad (6) \qquad\qquad \dot{Q}_{av} = \dot{m}_g\ c_{p,g}(T_6 - T_{\mathrm{amb}}) \qquad (7)$$

The overall efficiency of the bottoming cycle $\eta_{rec}$is defined as the ratio between $P_{e,ORC}$ and the heat rejected from the topping cycle $\dot{Q}_{av}$. From the above definitions,

$$\eta_{rec} = \frac{P_{e,ORC}}{\dot{Q}_{av}} = \chi \cdot \eta_L\ . \tag{8}$$

The maximum power output for the combined cycle is obtained when the product of η_L and χ is maximum. The exhaust gas exiting the evaporator has still enthalpy level suitable for cogeneration. The heat that can be recovered in the cogenerative HRB can be evaluated from the temperature drop $(T_7 - T_8)$ of the exhaust gas. In this work, we assume a minimum temperature of 50°C for the return water from residential users. Therefore, assuming $\Delta T_{min} = 20°C$, the minimum exhaust gas temperature $T_8 =\ 70$°C and the heat rate recovered can be evaluated from

$$\dot{Q}'_{th} = \dot{m}_g\ c_{p,g}(T_7 - T_8). \tag{9}$$

Useful heat for residential users can be recovered also from the condenser of the ORC plant, if the condensation temperature is sufficiently high. Assuming $\Delta T_{min} = 20°C$, a condensation temperature of 90°C can be adopted. In this case, the heat flux rejected from the ORC plant is fully available for cogeneration,

$$\dot{Q}''_{th} = \dot{m}_v\ (h_f - h_a), \tag{10}$$

where $\dot{m}_v$ is mass flow rate of organic fluid and $(h_f - h_a)$ is the enthalpy drop of the organic fluid across the condenser. The total thermal heat recovered for cogeneration is the sum of the heat recovered from exhaust gas and recovered from the heat rejected from the condenser

$$\dot{Q}_{th} = \dot{Q}'_{th} + \dot{Q}''_{th} \tag{11}$$

In the other case of condensation temperature lower than the temperature requested for cogeneration, the heat flux rejected by the ORC condenser is wasted to the environment. In this last case $\dot{Q}''_{th} = 0$. The instantaneous values of thermal efficiency, η_{th}, and first law (total heat and power) efficiency, η_{CHP}, of the plant, can be evaluated as

$$\eta_{th} = \frac{\dot{Q}_{th}}{\dot{E}_{in,tot}} \qquad (12) \qquad\qquad \eta_{CHP} = \eta_{e,CC} + \eta_{th} = \frac{P_{e,CC} + \dot{Q}_{th}}{\dot{E}_{in,tot}} \qquad (13)$$

The energy performance of the plant has to be evaluated over the annual production of electric and useful thermal energy output, considering the annual fuel energy consumption. In this work, energy and economic evaluation are carried out under the hypothesis of baseload operation

$$E_{e,CC} = \int_0^{t_{an.}} P_{e,CC}\mathrm{d}t, \qquad E_{th} = \int_0^{t_{an.}} \dot{Q}_{th}\ \mathrm{d}t, \qquad E_{in,tot} = \int_0^{t_{an.}} \dot{E}_{in,tot}\ \mathrm{d}t. \tag{14}$$

The annual averaged first law (total heat and power) efficiency is then

$$\bar{\eta}_{CHP} = \frac{E_{e,CC} + E_{th}}{E_{in,tot}}. \tag{15}$$

3.2 IFGT and EFGT gas turbine models

The natural gas fired IFGT simulation (case 1) is based on the MGT Turbec T100 having a maximum power output of 100 kWe when fuelling natural gas (AE-T100 data sheet, 2015). A recuperator is used to raise the net electric-efficiency from 16% of the simple cycle gas turbine to 30% of the recuperative Joule-Brayton cycle. The design hypotheses for these sections of the plant are described in [Pantaleo *et al.,* 2013]. The main performance data are in Table 1. In the case of dual combustion of biomass and natural gas, an EFGT scheme is adopted. The biomass feeds the external furnace while combustion air is pre-heated in a dedicated heat exchanger, which recovers heat from exhaust combustion gas. Details of the EFGT scheme can be found in Pantaleo *et al.* (2013) and Camporeale *et al.*(2014). It should be noted that the combustion air is conveyed in the furnace by a fan independently from the gas flowing in the turbine. In this case, no dirty gas flows at the exhaust of the turbine, hence facilitating cogeneration and increasing the flexibility of input fuel, which could be particularly important in case of seasonal biomass availability. The results of the energy analysis reported in Table 3 (case A) show that option of 100% biomass fired EFGT is affected by a de-rated electric power on respect to option of 100% NG fired IFGT. The main causes are related to: (i) lower TIT (reduced to 900°C from the original 950°C of the turbine fuelled by NG,) and (ii) power absorbed by the fan of the furnace and the other auxiliaries.

Table 3. Energy performance of the different examined case studies.
Fuel type: NG: natural gas; DF: dual fuel natural gas/biomass; B: biomass

Description	Unit	MGT cogeneration			MGT + ORC 1 no cogeneration			MGT+ORC 1 cogeneration			MGT- ORC 2 cogeneration		
Fuel type		**NG**	**DF**	**B**	**NG**	**DF**	**B**	**NG**	**DF**	**B**	**NG**	**DF**	**B**
Electric power output	kW	101.9	93.7	85.0	121.77	113.8	105.0	119.0	111.3	102.4	110.4	102.5	93.7
Thermal input	kW	332.2	348.6	366.4	332.2	348.6	366.4	332.2	348.6	366.4	332.2	348.6	366.4
Cogenerated Heat flux	kW	165.0	163.2	160.8				57.2	56.5	55.6	148.6	143.2	137.9
Electric efficiency	%	30.7	27.0	23.2	36.6	32.8	28.6	35.8	31.9	28.0	33.2	29.4	25.6
Thermal efficiency	%	49.7	46.8	43.9				17.2	16.2	15.2	44.7	41.1	37.6
CHP efficiency	%	80.3	73.7	67.1				53.0	48.1	43.1	77.9	70.5	63.2

3.3 Performance of ORC bottoming cycle

The topping cycle supplies exhausted gas (case NG, case DF) or air (case B) to the bottoming cycle at 270°C (262 °C in the case B, that is, 100% biomass), hence the suitable working fluid can be chosen among siloxanes. In particular, hexamethyldisiloxane –MM- with a critical temperature of 245.6°C has been chosen. From a previous analysis, where saturated, superheated and supercritical cycles were compared, the saturated cycles are considered the best choice. The evaporation pressure of 8 bar is selected in order to have a good electric power output with acceptable turbine volume ratio. The isentropic and mechanical efficiencies of the pump are set to 0.75 and 0.86, respectively, while turbine isentropic and mechanical ones are set to 0.8 and 0.96. A pressure drop of 2 kPa is assumed in all the heat exchangers.

Let us consider first the ORC cycle (indicated by “ORC-1”) with a condensation temperature of 40°C. The following discussion is referred to calculations carried out considering a MGT fuelling NG as topping cycle. From Table 2, it appears that about 53% of the available heat is recovered in the HRVG. The efficiency η_L of the bottoming cycle is about 20% that is relatively high if one considers that the maximum cycle temperature is 192°C. The overall efficiency η_{rec} of the bottoming cycle in such configuration is about 10 %. In this case, heat rejected from condenser is not useful for cogeneration but heat can be recovered from exhaust gas exiting the HRSG.

Then, let us consider the ORC cycle (indicated by “ORC-2”) with a condensation temperature of 90°C. From Table 2, it appears that about 48% of the available heat is recovered in the HRVG (5% less than ORC-1), while the efficiency η_L of the bottoming cycle is 11% that is about the half of the previous case due to the higher condensation temperature. The overall efficiency η_{rec} of the bottoming cycle in such configuration becomes about 6 %. In this case, heat flux rejected from condenser, about 80 kW, is useful for cogeneration as well as heat from exhaust gas of the HRVG.

About identical results are obtained with a dual-fuel MGT, since the temperature of the gas exiting the topping cycle is about the same (272°C) while some difference are obtained with MGT fuelling only biomass due to the lower temperature at the recuperator exit (262°C). A complete description of the combined cycle MGT+ORC under the different configurations is given in Table 3, where the performance of the gas turbine alone is also reported. The performance is examined for the MGT under cogeneration configuration while the combined cycle is examined under either only electric power production or cogeneration.
It appears that the combined cycle allows for a significant increase of the power output in all the cases with condensation temperature $T_c = 40°C$, with an increase of the overall electric efficiency of about 5-6 %. However, there is a relevant reduction of the thermal efficiency η_{th} that drops from much more than 40% of the MGT alone to about 16% for the combined cycle: this is due to the utilization of the exhaust gas heat for the bottoming cycle. In the cases of combined cycle with ORC condensation temperature $T_c = 90°C$, the electric efficiency is 3% higher than the simple MGT, but there is a low decrease of thermal efficiency with respect to the MGT, thanks to the utilization of the heat flux rejected from condenser. These results will be examined in the next section in order to identify the relative profitability and attractiveness of the different plant configurations.

4. Thermo economic methodology

The aim of the methodology is to provide a preliminary assessment of the global energy efficiency and profitability of the selected plant configurations as a function of the heat demand for cogeneration. For this purpose, the scenario of a baseload plant operation is assumed, with all electricity fed into the grid and cogenerated heat delivered to the load (50-70°C). The intensity of heat demand is taken into account by means of the equivalent hours of thermal energy consumption per year h_T, neglecting transients and part load efficiency. Further CHP operating strategies, including part load operation, can be found in previous works [Pantaleo *et al.* 2013, Camporeale *et al*, 2014, 2015]. The plant is operated at rated power for the maximum number of hours compatible with maintenance (7,500 hr/year). The revenues from electricity are in the form of fixed guaranteed remuneration (for the natural gas-based fraction) and feed-in tariff [Ministry Decree, 2012] (for the biomass-based fraction), including white certificates for the quota of natural gas based high efficiency cogenerated energy [Ministry Decree, 2011]. In the case of only biomass fuel, this operating strategy is the most profitable one, since the feed-in price for biomass electricity is higher than the cost of electricity supply. The reason is that, according to Italian RES subsidy mechanism [Ministry Decree, 2012], only renewable electricity fed into the grid is eligible for feed-in tariff, while on-site electricity is not subsidized. However, in case of natural gas based electric generation, matching on site electric demand before feeding the excess to the grid could be more profitable, as resulting from [Pantaleo et al, 2014a]. The calculation of incentives available for high efficiency cogeneration (HEC) is based on the assessment of the primary energy saving PES of the CHP plant, according to the procedures described in [Pantaleo et al, 2013a, 2014b].

4.1 Financial appraisal

The financial appraisal of the investment is carried out assuming the following hypotheses: (i) 20 years of operating life; no 're-powering' throughout the 20 years; zero decommissioning costs; (ii) maintenance costs, fuel supply costs, electricity and heat selling prices held constant (in real 2015 values); (iii) duration of feed-in tariff for biomass electricity of 20 years and duration of HEC incentive for gas cogeneration of 10 years (as stated by the 'white certificates' mechanism, which also includes a multiplicative coefficient of 1.4 [Ministry Decree, 2011]); (iv) capital assets depreciated using a straight line depreciation over 20 years; (v) cost of capital (net of inflation) equal to 8%, corporation tax neglected, capital costs and income do not benefit from any support.

4.2 Costs and revenues assessment

The turnkey capital cost, including storage and civil work costs of the biomass section, is assumed from [Pantaleo et al, 2013]. The cost of gas turbine recuperator is calculated assuming the maximum metal temperature of 1,000 °C and the use of Ni-Cr 40-20 alloy. The costs of ORC turbines vary in a wide range depending on the technology. In this work, after personal interviews to ORC turbine manufacturers, a cost of 100 and 60 kEur is assumed for the 20 and 12 kW bottoming ORC turbine.

Table 4. Fuel consumption of the MGT plant with different configurations, CAPEX and OPEX of the investment, assuming baseload operation mode. NG: natural gas fuel; DF: dual fuel 50% natural gas and 50% biomass (referred to energy input); B: biomass

Fuel	Configuration	Fuel consumption		CAPEX (kEur)	OPEX (kEur/y)
		NG (Nm3/y)	Biom (t/y)		
NG	MGT+CHP	225,575	0	188	99.4
	MGT+ORC 1 without CHP (T_C=40°C)			278	101.2
	MGT+ORC 1 + CHP (T_C=40°C)			288	100.9
	MGT+ORC 2 + CHP (T_C=90°C)			240	100.2
DF	MGT+CHP	118,352	312	289	83.2
	MGT+ORC 1 without CHP (T_C=40°C)			379	85.3
	MGT+ORC 1 + CHP (T_C=40°C)			389	85.1
	MGT+ORC 2 + CHP (T_C=90°C)			349	84.2
B	MGT+CHP	0	657	470	65.0
	MGT+ORC 1 without CHP (T_C=40°C)			560	67.4
	MGT+ORC 1 + CHP (T_C=40°C)			570	67.1
	MGT+ORC 2 + CHP (T_C=90°C)			530	66.1

Table 5. Electricity and heat selling prices; the feed-in tariff for biomass electricity includes remuneration for electricity fed into the grid and the subsidy; the white certificate is issued for primary energy saved through natural gas fuelled high efficiency CHP

Parameter	Eur/MWh
Natural gas electricity price PE_{NG}	150
Bio-electricity feed-in price PE_b	287
White certificate price	8.14
Thermal energy price PT_t	40

The fuel cost is assumed of 80 Eur/t and 40 cEur/Nm3 respectively for biomass and NG. These figures are obtained from market prices for wood chips in the Italian market and data from Italian Energy Authority for natural gas price for use in CHP plants (in this case, low tax rates apply in comparison to heating use) [Pantaleo *et al.,* 2013]. The O&M costs depend on the technology generation, the type of service and the capacity factor. In this work, the estimated O&M costs are 12 Eur/MWh for natural gas based electricity and 16 Eur/MWh for biomass based electricity, on the basis of global service contracts proposed by manufacturers of these small scale CHP plants to final customers. Biomass ash discharge costs are 70 Eur/t. In the case of biomass electricity generation, a plant self -consumption of 5% of generated electricity is assumed, which takes in account the energy consumption for biomass handling and furnace operation. The reference thermal and electrical efficiencies, to calculate the primary energy savings, are respectively 90% and 45.15% [Ministry Decree, 2011]. Table 4 summarizes the fuel consumption of the MGT with different fuels and the CAPEX and OPEX of the investments. The heat and electricity selling prices and the incentives for biomass electricity and HEC are reported in Table 5. The natural gas electricity price and the thermal energy price are estimated assuming the scenario of on site generation to match the heat and power demand of a block of residential buildings in the Italian energy market. In particular, the avoided electricity and thermal energy costs for residential end users are considered, assuming statistical data of electricity and natural gas costs from Italian Authority of Energy (AEEG, 2015) without taxes. In the case of thermal energy, the selling price is defined assuming a reference scenario of natural gas based heating system (average thermal energy cost of 80 Eur/MWh) and a discount of 50% to incentivize customer to connect to the heating network of the CHP plant.

5. RESULTS AND DISCUSSION

5.1 Energy performance assessment

The first law cogenerative efficiency, $\bar{\eta}_{CHP}$, vs the equivalent hours of thermal load is reported in Figure 3. The plant configurations under investigation are: only MGT with cogeneration (MGT-cog), MGT+ bottoming ORC and low T_C without cogeneration (MGT+ORC1), MGT + bottoming ORC with cogeneration and low T_C (MGT+ORC1-cog) and MGT + bottoming ORC with cogeneration and high T_C (MGT+ORC2-cog). These configurations are examined considering different input fuel mixes (natural gas, dual fuel and biomass). The following main conclusions can be drawn:

1. Input fuel: the use of natural gas presents the best energy performances, because of the higher conversion efficiency of the thermodynamic cycle on respect to the case of biomass;
2. Thermodynamic cycle: the use of bottoming ORC and low condensation temperature does not increase the global efficiency of the system (in particular at high heat demand levels) since, despite the higher electric efficiency of cases MGT+ORC1 with and without cogeneration, the cogenerated heat is lower (case MGT+ORC1-cog) or equal to zero (MGT+ORC1) on respect to case MGT-cog or MGT+ORC2-cog. At high energy demand levels (equivalent heat demand hours respectively above 4,000 – 3,500 and 3,000 for natural gas, dual fuel or biomass feed configurations) the use of only MGT presents higher global energy efficiency than the bottoming ORC2. However, the operating strategy to switch off the bottoming ORC during high heat demand periods, so increasing the cogenerated heat, could improve the energy performance (Pantaleo et al, 2015).

6.2 Profitability assessment

The internal rate of return (IRR) and net present value (NPV) of the investments are reported in Fig 4 and 5. The following considerations can be drown:

1. Fuel mix: the dual fuel option presents the highest IRR, as a result of the trade-off between energy efficiency decrease and investment cost/electricity revenues increase when increasing biomass use; on the contrary, the option of only biomass input fuel presents the highest NPV, as a result of the highest investment cost of this technology, balanced by the highest revenues from electricity sales (mainly because of the high feed-in tariff available for biomass electricity);
2. System configuration: the bottoming ORC reduces the IRR of the investment in the case of natural gas fuel on respect to only MTG, even at high heat demand levels (where the bottoming ORC2 presents on the contrary the highest energy performance); this is not valid for the NPV and in the case of biomass fuel, where, at high energy demand levels, the case MGT+ORC2-cog presents higher NPV than case MGT-cog. On the contrary, at low heat demand levels and with dual fuel or only biomass options, the bottoming ORC cycle presents higher IRR than the only MGT option; this is due to the different electricity price for natural gas and biomass; in particular, and the higher ORC investment cost and reduced income from heat sale (less o no cogenerated heat) does not balance the increased revenues from electricity sale when using natural gas fuel.

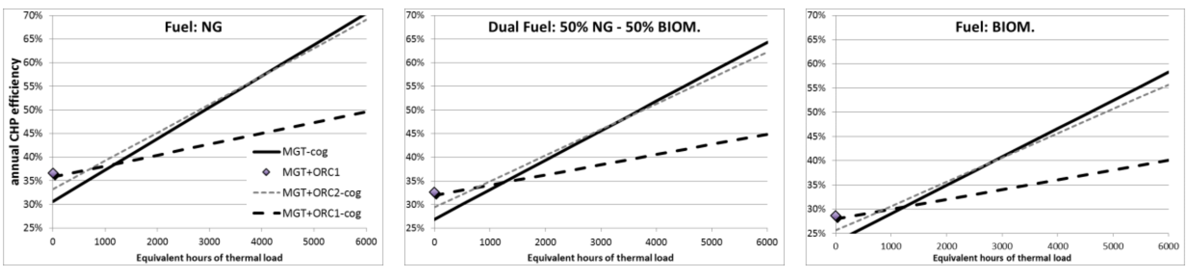

Figure 3. Annual CHP efficiency ($\bar{\eta}_{CHP}$) as a function of thermal energy demand.

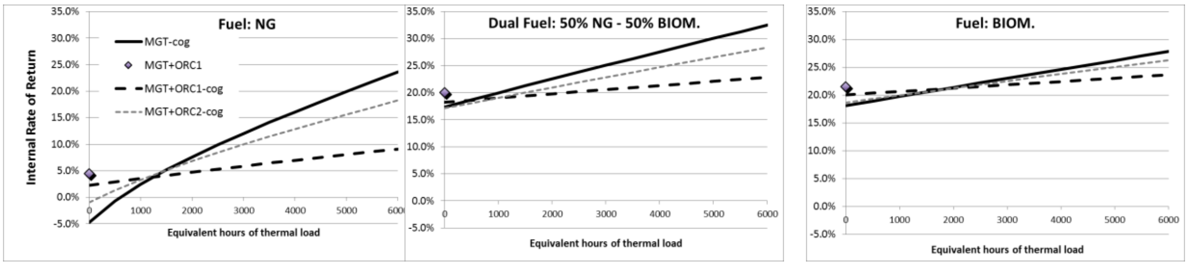

Figure 4. Internal Rate of Return (%) of the investments as a function of thermal energy demand equivalent hours for the case studies only natural gas (left), dual fuel (middle) and only biomass (right)

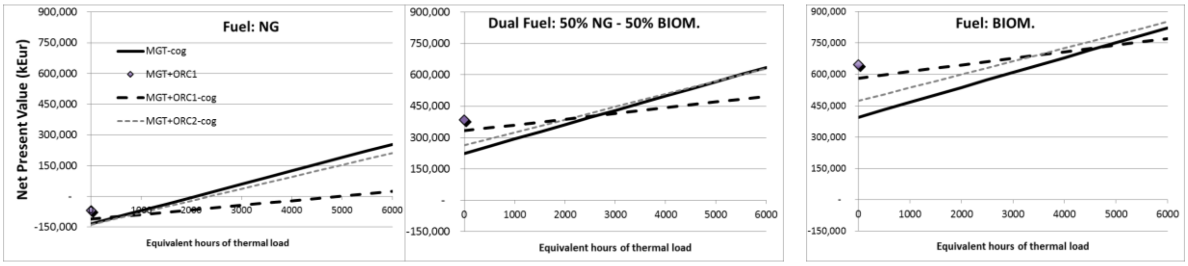

Figure 5. NPV (kEur) of the investments as a function of thermal energy demand equivalent hours for the case studies only natural gas (left), dual fuel (middle) and only biomass (right)

7. CONCLUSIONS

In this paper, the results of a thermo-economic assessment of micro gas turbine (100 kWe) with bottoming ORC fed by natural gas and biomass is presented. The systems configurations of MGT with cogeneration, MGT+bottoming ORC without cogeneration, and MGT with bottoming ORC and cogeneration at different condensing temperature are modelled, considering the three input fuel options of only natural gas, dual fuelling of natural gas and biomass, and only biomass. The results are used to inform a techno-economic methodology to estimate economic indices and energy performances in different scenarios. The influence of fuel characteristics on: (i) technical parameters, (ii) conversion efficiencies, (iii) capex and opex, (iv) profitability of investments are investigated.

The following main conclusions can be drawn:

(i) the global conversion efficiency ranged respectively between 30-70% and 23-60% for the natural gas and biomass fired case studies and the MGT with cogeneration, as a function of the thermal energy demand operating hours, with lower performances in case of biomass energy input. The combined cycle of the MGT with a bottoming ORC and low condensing temperature (case ORC 1) increases electric conversion efficiency to 36-28% and reduces the thermal efficiency to 17-15% respectively for the natural gas and biomass fired case studies. In the case of high condensing temperature (ORC 2), the electric and thermal conversion efficiencies are respectively of 33-25% and 45-38% for the natural gas and biomass fired cycles.

(ii) the ORC bottoming cycle increases the NPV when using biomass fuel, while the optimal IRR is obtained with dual fuel configurations; CHP investment profitability, and the incremental profitability of bottoming ORC on respect to MGT cycle, are highly influenced by thermal energy demand.

The scenario of 100% NG has the highest conversion efficiency and primary energy saving; however, the 100% biomass option has the highest NPV. This is due subsidies available by feed-in tariffs for electricity produced from biomass.

In conclusion, the profitability of adding a bottoming ORC is dependent on the trade-off between increased upfront costs, reduced heat available for cogeneration, and increased electricity generation. According to the simulation results, the bottoming ORC with low condensing temperature (ORC1) is profitable when using biomass fuel and with low heat demand levels, while the bottoming ORC with high condensing temperature (ORC2), despite presenting lower electric conversion efficiency, is more profitable than the only MGT cycle, at high energy demand levels. Further researches will be focused on algorithms to optimize the CHP operation with the option to switch on and off the bottoming ORC on the basis of the energy demand and other techno-economic parameters.

REFERENCES

AEEG (Italian Authority of Electricity and Gas), statistical data, 2015 http://www.autorita.energia.it/it/dati/elenco_dati.htm, (accessed may 2015)

AE-T100 Micro Turbine, Natural Gas, DATA SHEET, available from (accessed on Jan 4th, 2015) http://www.ansaldoenergia.com/easyUp/file/ae-t100_micro_turbine_natural_gas_sheet_englis.pdf

Al-Sulaiman FA, Dincer I, Hamdullahpur F. Thermoeconomic optimization of three trigeneration systems using organic Rankine cycles: Part I – Formulations. Energy Conversion and Management, 2013;69:199–208.

C. He, C. Liu, H. Gao, H. Xie, Y. Li, S. Wu, J. Xu. "The optimal evaporation temperature and working fluids for subcritical organic Rankine cycle". ELSEVIER, 2012.

C. M. Invernizzi, P Iora, and R. Sandrini. "Biomass combined cycles based on externally fired gas turbines and organic Rankine expanders". Journal of Power and Energy. DOI: 10.1177/0957650911415188

Camporeale S, Ciliberti P, Torresi M, Fortunato B, Pantaleo A (2015), Externally fired micro gas turbine and ORC bottoming cycle: optimal biomass/natural gas CHP configuration for residential energy demand, Proceedings of ASME Turbo Expo 2015, Montreal, Canada, June 15-19 2015

Camporeale S., Turi F., Torresi M., Fortunato B., Pantaleo A, Pellerano A. (2014) Part load performances and operating strategies of a natural gas-biomass dual fuelled microturbine for CHP operation, Proc. of ASME Turbo Expo 2014: Düsseldorf, Germany, June 16-20, 2014 GT2014- 27109, and ASME Journal of Engineering for Gas Turbines and Power

Cheap, I.R., Fee, W., 1986, Conference Attendance of a Shoestring Budget, *Int. J. Cost-Effective Research*, vol. 7, no. 4: p. 78-93.

Cocco D, Deiana P, Cau G. Performance evaluation of small size externally fired gas turbine (EFGT) power plants integrated with direct biomass dryers. Energy 2006;31(10-11):1459–1471.

Decision 406/2009/EC of the European Parliament and of the Council of 23 April 2009

Directive 2009/28/EC of the European Parliament and of the Council of 23 April 2009

Duminil, M., 1995, Principes de la production du froid, Chapter 2, *In*: Côme, D., Ulrich. R., *La chaîne du froid – Le froid au service de l'homme*, Hermann, Paris: p. 33-144.

Evans RL, Zaradic AM. Optimization of a wood-waste-fuelled indirectly fired gas turbine cogeneration plant. Bioresource Technology 1996;57: 117–26.

Ferreira ACM, Nunes ML, Teixeira SFCF, Leão CP, Silva ÂM, Teixeira JCF, Martins LSB. An economic perspective on the optimisation of a small-scale cogeneration system for the Portuguese scenario. Energy, 2012;45(1):436–444.

Ferreira SB, Pilidis P. Comparison of externally fired and internal combustion gas turbines using biomass fuel. ASME J. Energy Resour. Technol. 123 (2001) 291–296.

Forbes-Pearson, S., 1996, Uses of Hydrocarbon Refrigerants, *Proc. Aarhus Meet.*, IIF/IIR: p. 439-446. Janna, S.W., 1986, *Engineering Heat Transfer*, PWS Publishers, Boston, 769 p.

Fortunato B, Camporeale S.M., Torresi M., A Gas-Steam Combined Cycle Powered by Syngas Derived from Biomass, Procedia Computer Science, (2013) Volume 19, 736-745

Franco A, Giannini N. Perspectives for the use of biomass as fuel in combined cycle power plants. International Journal of Thermal Sciences, 2005; 44(2):163–177.

G. David, F. Michel, L. Sanchez. "Waste heat recovery projects using Organic Rankine Cycle technology – Examples of biogas engines and steel mills applications." Geneve 2011- World Engineer's Convention

Galanti L, Massardo AF. Thermoeconomic Analysis of Micro Gas Turbine Design in the Range 25-500 kWe. Proceedings of ASME Turbo Expo 2010 (pp. GT 2010–22351; pp1–11).

H. Chen, D.Yogi Goswami, E.K. Stefanakos. "A review of thermodynamic cycles and working fluids for the conversion of low-grade heat". ELSEVIER, 2010

Hamilton SL. The Handbook of Microturbine Generators. Tulsa: PennWell Corporation; 2003.

http://www.asimptote.nl/software/cycle-tempo/(accessed Jan. 3rd, 2015)

http://www.autorita.energia.it/it/dati/condec.htm accessed oct 2013

https://getotalplant.com/GateCycle/docs/GateCycle/index.html (accessed Jan. 3rd, 2015)

K. Kusterer, R. Braun, D. Bohn. "Organic Rankine cycle working fluid selection and performance analysis for combined application with a 2 MW class industrial gas turbine". Proceedings of ASME Turbo Expo 2014: - GT2014 June 16 – 20, 2014, Düsseldorf, Germany

Kautz M, Hansen U. The externally-fired gas-turbine (EFGT-Cycle) for decentralized use of biomass. Applied Energy. 2007;84(7-8):795–805.

Knoef H. The indirectly fired gas turbine for rural electricity production from biomass, Project Brochure and Reports; 1998. Contract FAIR-CT95–0291.

M. A. Rosa do Nascimento, L. de Oliveira Rodrigues, E. Cruz dos Santos, E. E. Batista Gomes, F. L. Goulart Dias, E. I. Gutiérrez Velásques, R. A. Miranda Carrillo. " Micro Gas Turbine Engine: a Review". Progress in Gas Turbine Performance", edited by Ernesto Benini, Published: June 19, 2013, DOI: 10.5772/54444

Ministry Decree 5-09-2011 on incentives for High Efficiency Cogeneration in Italy (in italian)

Ministry Decree 6-07-2012 on supporting mechanism for renewable electricity in Italy (in italian)

Obernberger I. Decentralized biomass combustion: state of the art and future development. Biomass and Bioenergy 1998;14(1):33–57.

P. Arvay, M. R. Muller, V. Ramdeen, Economic Implementation of the Organic Rankine Cycle in Industry, ACEEE Summer Study on Energy Efficiency in Industry, 2011, Editor: ACEEE, 529 14th Street, N.W., Suite 600, Washington, D.C. 20045, ISBN 0-918249-61-0

Pantaleo A, Camporeale S, Shah N. Thermo-economic assessment of externally fired micro gas turbine fired by natural gas and biomass: applications in Italy, Energy Conversion and Management, (2013a);75:202-213

Pantaleo A, Shah N, Keirstead J, Bioenergy and other renewables in urban energy systems (2013b) in Urban Energy Systems – an integrated approach, by J Keirstead and N Shah, Routledge, NY

Pantaleo A, Candelise C, Bauen A, Shah N, ESCO business models for biomass heating and CHP: case studies in Italy, Renewable and Sustainable Energy Reviews (2014a) 30, 237-253

Pantaleo A., Camporeale S., Shah N., Natural gas – biomass dual fuelled microturbines: comparison of operating strategies in the Italian residential sector Applied thermal engineering, (2014b)

Pantaleo A, Ciliberti P, Camporeale S, Shah N (2015) Thermo-economic assessment of small scale biomass CHP: steam turbines vs ORC in different energy demand segments, Proceedings of 7th International Conference on Applied Energy – ICAE, Abu Dhabi, March 28-31 2015

R. Tidball, J. Bluestein, N. Rodriguez, and S. Knoke (2010) Cost and Performance Assumptions for Modeling Electricity Generation Technologies, Report NREL/SR-6A20-48595, November 2010, available from http://www.nrel.gov/docs/fy11osti/48595.pdf (accessed Jan 4th, 2015)

Riccio G, Chiaramonti D. Design and simulation of a small polygeneration plant cofiring biomass and natural gas in a dual combustion micro gas turbine Biomass and Bioenergy 2009; 33(11).

Riccio G, Martelli F, Maltagliati S. Study of an external fired gas turbine power plant fed by solid fuel. In: Proceeding of ASME turbo expo 2000. ASME editor. Paper 0015-GT-2000

Savola T, Tveit T-M, Laukkanen T. Biofuel indirectly fired microturbine state of the art. TKK, Laboratory of Energy Engineering and Environmental Protection, Espoo, http://eny.hut.fi/research/process_integration/bioifgt_Jan.pdf ; 2005.

Soltani S, Mahmoudi SMS, Yari M, Rosen MA. Thermodynamic analyses of an externally fired gas turbine combined cycle integrated with a biomass gasification plant. Energy Conversion and Management, 2013; 70:107–115.

Yan J, Eidensten L. Status and perspective of externally fired gas turbines, J. Propulsion and Power 2000; 16(4).

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NOMENCLATURE

$\dot{E}$	energy flux	(kW)
h	enthalpy	(kJ/kg)
$\dot{m}$	mass flow	(kg/s)
p	pressure	(bar) or (kPa)
P	electric power	(kW)
$\dot{Q}$	heat flux	(kW)
T	temperature	(K) or (°C)
W	work	(kJ/kg)
η	efficiency	
χ	heat recovery factor	

subscripts

av	available
biom	biomass
cc	combined cycle
g	exhaust gas
e	electric
in	input
gen	Electric generator
L	Low (bottoming cycle)
th	thermal
rec	Recuperative cycle

acronyms

CHP	Combined Heat and Power generation
HRB	Heat Recovery Boiler
MGT	Micro Gas Turbine
ORC	Organic Rankine Cycle
LHV	Lower Heating Value
EFGT	Externally Fired Gas Turbine
IFGT	Internally Fired Gas Turbine
MM	hexa-methyl-disiloxane

Fuel supply : NG = natural gas ; DF = dual fuel (50% natural gas and 50% biomass input energy) ; B= biomass