International Conference on Applied Energy 2019
Aug 12-15, 2019, Västerås, Sweden
Paper ID: 1059

AN EXTERNALLY FIRED MICRO COMBINED-CYCLE, WITH LARGELY ADJUSTABLE STEAM TURBINE, IN A CHP SYSTEM

by
S. Barbarelli*, E. Berardi, M. Amelio, N. M. Scornaienchi
Department of Mechanical, Energy and Management Engineering
University of Calabria
I87036 Rende - Italy
* Corresponding Author

ABSTRACT

This work stems from the idea of demonstrating the feasibility of using micro combined plants for the simultaneous generation of electrical and thermal energy, by external combustion of low quality fuels. In particular, this study is aimed at acquiring the knowledge necessary for the development of a small power plant consisting of a micro gas turbine and an innovative steam microturbine. The topping cycle is carried out with a micro gas turbine coupled with a heat exchanger and an external combustor, in which biomass is burned. For the bottoming cycle an impulse turbine has been adopted, characterized by a variable section nozzle, capable of expanding widely variable steam flow rates, with small efficiency losses.

Keywords: renewables; biomass; combined heat and power; combined cycle; externally fired gas turbine; steam micro turbine.

1. INTRODUCTION

To reduce dependence on fossil fuels, the generation of electricity from renewable sources has increased over the years; however, due to the asynchronism between demand and production, the development of technologies based on non-programmable sources must be accompanied by the development of adequate storage systems or supported by more continuous energy sources, such as biomass [1]. Besides, progress in the renewable energy sector has led to the introduction of the concepts of distributed generation and Smart Grids [2].

It is in this context that the idea of a biomass small-scale externally fired combined cycle plant develops, capable of generating up to around 100 kW of electricity, with the possibility of cogeneration. This system is compatible with self-consumption, greatly simplifying the problems related to the control of electrical networks [3].

The combined system of the type presented here has several advantages: in addition to the low capital costs, high reliability and low maintenance, there is particular interest for the flexibility of the fuel. In fact, this cycle is characterized by externally fired, so it can also be fed with low quality fuels, without having to resort to a complex gasification section [4].

Thanks to external combustion, the turbine blades will not be crossed by flue gases but only by clean air, avoiding the problem of damaging the blades. The nominal size of the plant presented here was chosen for ease of integration in distributed generation and for the possibility of local supply of fuel: a small plant like this can be built directly in the biomass production area [5].

The energy recovery of waste heat further increases the potential of the plant, making it self-sufficient even from a thermal point of view. This makes the whole process even more sustainable, as well as economically advantageous. The burnt gases at the steam generator exit and the heat deriving from the condensation of the exhausted steam are used to supply a thermal user: the nature of the heat carrier adopted and the supply temperature are chosen based on the requests of the user to be satisfied.

In this perspective, this plant fits well in the context of distributed generation, with the dual advantage of minimizing the losses related to the transporting electricity along the distribution network and recovering the thermal energy available for discharge through cogeneration.

Selection and peer-review under responsibility of the scientific committee of the 11th Int. Conf. on Applied Energy (ICAE2019).

The proposed system can therefore be exploited to meet the needs of electricity and heat in hotels, tourist villages, small factories, shopping centers, food industries, hospitals, offices, industrial laundries, schools, greenhouses and in all those users characterized by a contemporary demand for the two useful effects. The economic and environmental advantages are obvious: fuel consumption is lower (compared to the separate generation of the two effects) and therefore the emissions of CO_2 and other pollutants are significantly reduced.

The topping cycle is the classic Joule-Brayton cycle performed by a commercial gas turbine (MGT); the bottoming cycle, on the other hand, is made by a specially designed innovative steam micro-turbine, which has the advantage of being able to expand variable steam flow rates without excessive losses.

2. PLANT LAYOUT

Fig. 1 shows the proposed layout for the small combined plant analyzed. MGT is commercially available; the steam turbine, instead, was designed in the Department of Mechanical, Energy and Management Engineering of the University of Calabria.

The exhaust gases, at the exchanger exit, are still at high temperature; therefore, their energy content can be exploited in heat recovery steam generator (HRSG), to produce steam to feed the bottoming cycle. The steam turbine powers the second generator. The expansion is stopped at 1 bar to simplify the plant, removing the deaerator, and to carry out cogeneration, recovering the thermal energy coming from the condensation of the steam. Finally, to operate in a closed circuit, the bottom cycle has a water supply pump, to increase the pressure again up to the initial conditions.

A further fraction of thermal energy can be recovered from the burnt gases coming out of the recovery boiler, by means of a gas-liquid exchanger. These gases are not cooled below 150 ° C, to avoid condensation of the sulfur compound derived from biomass combustion and prevent corrosion. The hot water produced can be used for heating or sanitary use.

For the bottom cycle, the use of the Rankine steam cycle is preferred to the ORC cycle, since it requires lower investment costs; furthermore, unlike organic fluids, water is easy to find, cheap and non-toxic. The various components that make up the plant will be described below.

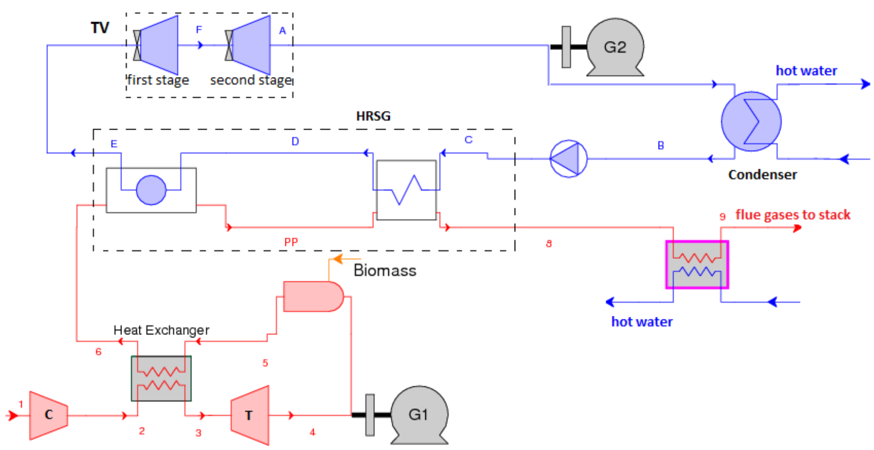

Fig 1 Plant Layout

Since this power plant is externally fired, combustion does not involve the air that expands in the turbine, but takes place in an external combustor. In a heat exchanger, the air flow coming from the compressor outlet is heated by the combustion products, to increase the enthalpy content. Thus, hot air expands into the turbine. Then it enters as combustion air in the combustor, which can thus exploit the heat still present in the working fluid leaving the turbine. The turbine drives the compressor and simultaneously drives the first electric generator.

2.1 Micro gas turbine (MGT)

The externally fired microturbine used in the topping cycle is the AE-T100E from Ansaldo Energia [6]. It allows to obtain 75 kW of electric power, with a maximum inlet temperature of 830 °C and a nominal air flow G_a of 0.79 kg / s.

2.2 Heat exchanger

To transfer to the clean air the heat of the gases deriving from the combustion of biomass, a gas-to gas heat exchanger is required; this air/burnt gases heat exchanger is the critical element of the plant in question, due to the high operating temperatures and fouling of the flue gases side. This component has represented in the past a limit for the development of plants externally fired with low quality fuels, due to the trade-off between high temperatures and high efficiency.

2.3 Combustor

To ensure the complete combustion of the biomass with high efficiency and reduced polluting emissions, a fluid bed combustor could be used. These combustors allow to burn the biomass without particular interventions [7], like the atomization of the fuel or the homogenization of the dimensions.

2.4 Heat recovery steam generator (HRSG)

In the layout examined, the HRSG is a simple saturated steam generator, without superheater: anyway the output of the turbine has a title higher than 95%, a satisfactory value to avoid erosion problems. A simple fire-tube boiler was adopted, since the maximum operating pressure is low and the thermo-mechanical resistance of the boiler is not compromised.

2.5 Steam turbine

The steam expander of the bottoming cycle is a small turbine, entirely designed in the Department of Mechanical, Energy and Management Engineering (DIMEG) of the University of Calabria. It is a turbine with an innovative design, which allows the flow to be regulated over a wide range, without significant losses in efficiency [8] [9] [10]. (Fig. 2)

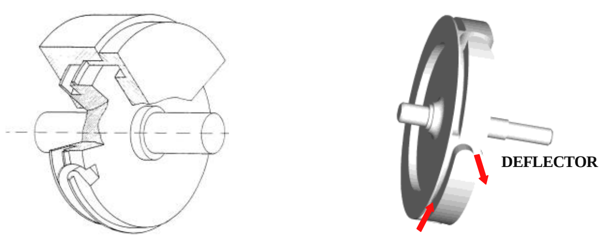

Fig 2 Cross section of the single stage rotor

The steam expander conceived at DIMEG is a tangential flow turbine with a tangential supply nozzle and a rotating channel, on the periphery of which three deflector ducts are dug. The absence of the blades simplifies the structure and guarantees a better resistance to fatigue, since there is no possibility of erosion caused by condensate drops. Unlike traditional small turbines, it is able to operate with low rotational speeds. The considerable advantage of this turbine is its ability to operate with small flow rates, with small variations in efficiency as the power supplied varies over a wide range.

To regulate the flow, this innovative turbine is equipped with a tangential nozzle with variable throat area: by means of a movable spear, the opening of the nozzle can be varied continuously from zero to area equal to the cross section of the rotating channel, without that the efficiency decreases significantly. The opening fraction is expressed by the opening coefficient Kp defined as:

$$K_p = \frac{nozzle\ \ area}{channel\ \ area}$$

As shown in Fig. 3, the fluid, with tangential admission, accelerates in the nozzle; flows into the rotating channel and axially discharges into the stator casing through the curved channel of the deflectors.

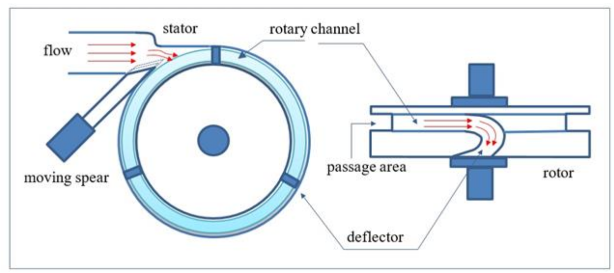

Fig 3 Single stage turbine prototype scheme

It has been observed that as the number of deflectors increases, the efficiency increases: going from 1 to 3 deflectors the improvements are consistent, while passing from 3 to 7 this advantage becomes less significant [11].

The efficiency of the turbine at nominal conditions is 50%; an efficiency greater than 47% is guaranteed if the flow decreases to around 63% of the nominal flow (Fig.4).

The prototype tested in the laboratory allows to dispose of up to 44 gr/s of steam. Since in the nominal conditions the plant produces a flow rate of 75.6 gr/s, it was necessary to design ad hoc a machine similar to the one tested. From experimental tests on the existing prototype, it was observed that the highest efficiency is obtained for an expansion ratio of 4. For this reason, to increase the performance of the bottoming cycle, a two-stage plant solution was chosen, with admission pressure equal respectively to 16 bar and 4 bar.

Figure 5 shows the prototype of a two-stage turbine designed ad hoc for the plant under study. The axial output of the first stage is connected by a tube to the tangential inlet of the second stage.

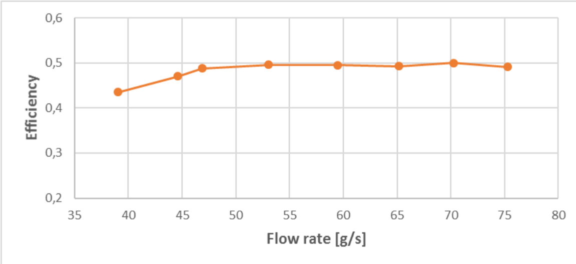

Fig 4 Efficiency curve

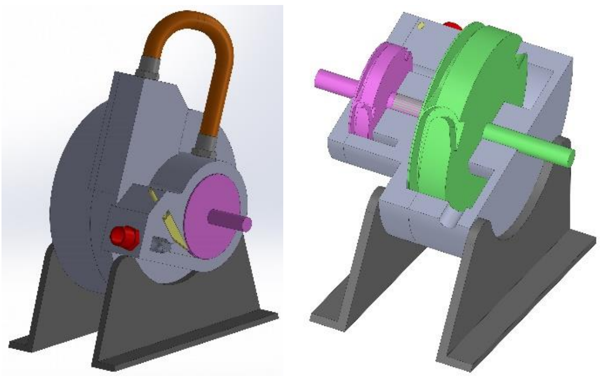
Fig 5 Two-stage turbine

2.6 *Construction criteria of the two-stage steam turbine*

In this work, the authors designed the turbine prototype in the two-stage version, based on the experience gained in this field and considering the steam flow generated in the HRSG and the chosen pressure levels.

By choosing a number of rotations N equal to 12000 rpm, the diameter of each stage can be determined as:

$$D_{rot} = \frac{60 \cdot u}{\pi \cdot N}$$

where u is the peripheral speed, determined starting from the spouting speed c_0:

$$u = K \cdot c_0$$

$$c_0 = \sqrt{2 \cdot \Delta h_{is}}$$

The prototype is able to work in the best conditions with a low velocity ratio, unlike traditional axial turbines; this allows operation at low rotational speeds. To design a machine similar to the tested prototype, it was chosen K = 0.1 for the first stage and K = 0.2 for the second stage.

The throat area of the nozzle was determined as:

$$A_{TH} = \frac{G_v}{\sqrt{\frac{k}{R} \cdot \frac{P_{in}}{T_{in}} \cdot \left(\frac{2}{k+1}\right)^{\frac{k+1}{2(k-1)}}}}$$

where

$$k = 1.035 + 0.1\,x$$

x is the steam quality.

To minimize secondary flow losses, the ratio between the height h and the width b of the rotating channel was chosen equal to 2 [12].

The Table 1 shows the geometric parameters of the two stages of the turbine.

Parameter	Symbol	First stage	Second stage
Nominal rotational speed [rpm]	N	12000	
Feeding pressure [bar]	Pin	16	4
Temperature at the turbine inlet [°C]	Tin	201.4	143.6
Isentropic enthalpy drop [kJ/kg]	Δh_{is}	251.8	228.9
Velocity ratio u/c_0	k	0.1	0.2
Nozzle throat area [mm^2]	A_{TH}	34.86	130.68
Rotating channel width [mm]	b	5.9	11.43
Deflector height [mm]	h	11.81	22.86
Rotor diameter [mm]	D_{rot}	113	215

Table 1 Sizing of the steam turbine

3. *THERMODYNAMIC MODELING*

To estimate the performance of the combined cycle, the energy balances relating to some components of the plant were exploited. The input thermal power is $G_b \cdot H_i$. By indicating the fuel mass flow rate with G_b, the combustor energy balance is given by:

$$\eta_c \cdot G_b \cdot H_i + G_b \cdot h_b + G_a \cdot h_4 = G_g \cdot h_5$$

where η_c is the combustor's efficiency, H_i and h_b respectively indicate the lower calorific value of the biomass and its initial enthalpy, h_4 is the enthalpy of the combustion air at the turbine outlet and h_5 is the enthalpy of the burnt gases at the combustor outlet. G_g is the flow rate exiting from the combustor, less than the sum of G_a and G_b due to the small ash content generated by biomass combustion [13].

The energy balance for the heat exchanger is expressed by:

$$G_g \cdot (h_5 - h_6) = G_a \cdot (h_3 - h_2)$$

where h_3 and h_2 are the enthalpies of the air entering and exiting the exchanger, while h_6 is the enthalpy of the burnt gases that leave the exchanger.

The electric power generated in the gas cycle is given by the difference between the power supplied by the turbine and the power absorbed by the compressor:

$$P_{el,TOP} = G_a \cdot (h_3 - h_4) \cdot \eta_{m,T} - \frac{G_a \cdot (h_2 - h_1)}{\eta_{m,C}}$$

In the recovery boiler, the heat exchange between the flue gases and the water/steam is regulated by the following equations:

$$G_g \cdot (h_6 - h_8) = G_V \cdot (h_E - h_C)$$

$$G_g \cdot (h_{PP} - h_8) = G_V \cdot (h_D - h_C)$$

$$T_{pp} = T_D + \Delta T_{pp}$$

G_V is the mass flow rate of steam produced in the boiler; T_{pp} and h_{PP} respectively represent the temperature and enthalpy of the flue gases at the evaporator outlet; ΔT_{pp} is the temperature difference at the pinch point; h_C is the enthalpy of the water entering the economizer; the subscript D indicates the water at the evaporator's entrance; h_E represents the enthalpy of the steam at the entrance of the steam turbine and h_8 is the enthalpy of the flue gases at the outlet of the boiler. The power supplied by the bottoming cycle can be evaluated as:

$$P_{el,BOT} = G_v \cdot \left(\Delta h_{is,1} \cdot \eta_1 + \Delta h_{is,2} \cdot \eta_2\right)$$

The power absorbed by the pump is negligible. The thermal power recovered respectively from the condensation of the steam and in the final exchanger, has been calculated as:

$$P_{th,cond} = G_V \cdot (h_A - h_B)$$
$$P_{th,scamb} = G_g \cdot (h_8 - h_9)$$

where h_A is the enthalpy of the steam at the exit of the turbine's second stage, h_B is the enthalpy of the water exiting the condenser and h_9 is the enthalpy of the flue gases sent to the chimney.

For the analysis of the plant in nominal conditions the assumptions are reported in Table 2.

Symbol	Value	
G_a	0.79	kg/s
p_1	1.013	bar
T_1	15	°C
β	4.7	
$TIT=T_3$	830	°C
T_5	950	°C
η_c	0.997	
Hi	18757.6	kJ/kg
p_E	16	bar
p_F	4	bar
P_A	1	bar
ΔT_{pp}	10	°C
T_9	150	°C
$\eta_{m,T} = \eta_{m,C}$	0.98	
$\eta_{is,TV}$	0.5	

Table 2 Assumed parameters

4. *RESULTS*

To estimate the performance of the proposed plant, simulations were made using the Thermoflex code. The electrical efficiency and thermal efficiency have been calculated as:

$$\eta_{el} = \frac{P_{el,TOP} + P_{el,BOT}}{G_b H_i} = \frac{P_{el,tot}}{G_b H_i}$$

$$\eta_{th} = \frac{P_{th,cond} + P_{th,exch}}{G_b H_i} = \frac{P_{th}}{G_b H_i}$$

The first law efficiency, given by the sum of η_{el} and η_{th}, is not sufficient to compare the performance of the plants in cogeneration configuration; with the second law efficiency, which can be evaluated as:

$$\eta_{II} = \frac{P_{el,tot} + P_{th,tot} \cdot \left(1 - \frac{T_1}{T_{CHP}}\right)}{G_b H_i}$$

it is possible to compare the performance of the real process with those of the ideal process, considering the T_{CHP} temperature at which the heat is made available. T_1 represents the ambient temperature.

Another important parameter is primary energy savings, assessed as:

$$PES = 1 - \frac{1}{\frac{\eta_{el}}{\eta_{el,ref}} + \frac{\eta_{th}}{\eta_{th,ref}}}$$

EU Regulation 2015/2402 [14] suggests $\eta_{el,ref} = 0.3$ and $\eta_{th,ref} = 0.8$; according to the European Directive 2004/8 / EC, a new cogeneration plant with a production capacity of less than 1 MWel is highly efficient if PES> 0. The system under examination allows a considerable energy saving, since PES = 23.32%.

The table 3 summarizes the results obtained.

Symbol	Value	
G_b	0.0218	kg/s
G_v	0.0756	kg/s
$P_{th,in}$	409.2	kW
$P_{el,TOP}$	75	kW
$P_{el,BOT}$	16.5	kW
$P_{el,tot}$	91.5	kW
$P_{th,cond}$	162.6	kW
$P_{th,exch}$	20.6	kW
$P_{th,tot}$	183.2	kW
η_{el}	22.35	[%]
η_{th}	44.76	[%]
η_I	67.11	[%]
η_{II}	31.61	[%]
PES	23.32	[%]

Table 3 Main results

5. *CONCLUSIONS*

The purpose of this work was to demonstrate the feasibility of a combined micro plant, consisting of an MGT and an innovative design steam microturbine. The plant is in CHP configuration and produces around 91 kW of electricity from biomass combustion. At the same time 183.2 kW of heat are generated and delivered by a hot water flow rate.

The system is innovative for the idea of creating a small-sized plant (<100 kW) with a combined gas-steam cycle, flexible in the power supply, being external combustion. As far as the authors know, there is only one other example of the same type made at the Polytechnic of Bari [7]. The objective of the present study was the development of a system suitable for distributed generation, with the use of a local renewable primary source, such as biomass. The efficiency achieved is lower than that of microturbines fueled by natural gas but, thanks to the combination with the steam cycle, it is still acceptable, given the energy source used and the consequent external combustion.

This system can also be used in areas not reached by electrification and is not affected by the volatility of fuel prices. It does not require excessive maintenance, it is based on established and economical technologies and has a low environmental impact. The proposed plant differs from the one studied in Bari for the dimensions (90 kW versus about 50 kW) and for the type of steam turbine. The system presented here is designed with a single-shaft, two-stage steam turbine with atmospheric pressure condensation to prevent the addition of the deaerator but, above all, to favor the thermal output that will be available at a temperature of 90/100 °C instead of 60 °C, as in the other known case. This also facilitates the feeding of an absorption refrigerator group, making it possible trigenerative applications.

From the simulations carried out using the Thermoflex software, it emerged that the proposed layout can be competitive in the distributed generation of small powers and to reduce dependence on fossil sources: in fact, in nominal conditions, the plant guarantees an electrical efficiency of 22.35 %, a thermal efficiency of 44.76%, a second law efficiency of 31.61% with primary energy savings of 23.32%.

REFERENCE

[1] Amirante R., Cassone E., Distaso E., Tamburrano P. Overview on recent developments in energy storage: Mechanical, electrochemical and hydrogen technologies. Energy Convers. Manage 2017;132.

[2] Zupone G., Amelio M., Barbarelli S., Florio G., Scornaienchi N.M., Cutrupi A. Levelized Cost of Energy: A First Evaluation for a Self Balancing Kinetic Turbine. Energy Procedia Volume 75, 2015, Pages 283-293

[3] Nastasi B. Hydrogen Policy, Market and R&D Projects. In: Calise F et al., editors. Hydrogen Production from Solar Energy, Amsterdam: Elsevier; 2019, edited by F.Calise et al. (Elsevier, Amsterdam, 2019).

[4] G. Cau, D. Cocco, F. Serra, Energy Convers. Manage. 56 (2012) 121–129.

[5] Nastasi B, Lo Basso G, Astiaso Garcia D, Cumo F, de Santoli L. Power-to-gas leverage effect on power-to-heat application for urban renewable thermal energy systems, Int J of Hydrogen Energy 2018;43(52):23076-23090.

[6] www.ansaldoenergia.com/business-lines/new-units/microturbines

[7] Amirante R., De Palma P., Distaso E., Tamburrano P. Thermodynamic analysis of small-scale externally fired gas turbines and combined cycles using turbo-compound components for energy generation from solid biomass. Energy Conversion and Management 2018.

[8] Barbarelli S., Florio G., Scornaienchi N.M. Developing of a Small Power Turbine Recovering Energy from low Enthalpy Steams or Waste Gases: Design, Building and Experimental Measurements. Thermal Science and Engineering Progress, Volume 6, June 2018, Pages 346-354

[9] Barbarelli S., Florio G., Scornaienchi N.M. (2005). Performance analysis of a low-power tangential flow turbine with rotary channel, Journal of Energy Resources Technology, Transactions of the ASME, Vol. 127, No. 4, pp. 272-279. DOI: 10.1115/1.1944008

[10] Barbarelli S., Florio G., Scornaienchi N.M. Theoretical and experimental analysis of a new compressible flow small power turbine prototype. International Journal of Heat And Technology. Vol. 35, Special Issue 1, September 2017. DOI: 0.18280/ijht.35Sp01XX

[11] Scornaienchi N.M., Florio G., Belli M. A Small Power Turbine Characterized by a Tangential Flux Within a Rotating Channel in ESDA 94 Proceedings of Second Biennal European Conference on Engineering System & Analysis, ASME Vol. 64-8.2, pp. 511-518, 1994, London.

[12] J. H. Horlock. “Losses and Efficiencies in Axial Flux Turbines” Int. J. Mech. Sci., 2, 1960.

[13] Vamvuka D, Bandelis G. Evaluation of wood residues from Crete as alternative fuels. Int J Energy Environ 2010;1(4):667–74.

[14] Commission Delegated Regulation (EU) 2015/2402 of 12 October 2015 reviewing harmonised efficiency reference values for separate production of electricity and heat in application of Directive 2012/27/EU of the European Parliament and of the Council and rep.

