Available online at www.sciencedirect.com

ScienceDirect

Energy Procedia 126 (201709) 659–666

Energy Procedia

www.elsevier.com/locate/procedia

72nd Conference of the Italian Thermal Machines Engineering Association, ATI2017, 6-8 September 2017, Lecce, Italy

Experimental prototype development and performance analysis of a small-scale combined cycle for energy generation from biomass

R. Amirante[a], P. De Palma[a], E. Distaso[a], M. La Scala[b], P. Tamburrano[a]*

[a]Department of Mechanics, Mathematics and Management, Polytechnic University of Bari, Via Orabona 4, 70100 Bari, Italy
[b] Department of Electrical and Information Engineering, Polytechnic University of Bari, Via Orabona 4, 70125 Bari, Italy

Abstract

This paper presents a research activity aimed at exploiting combined cycles (gas turbine plants coupled with steam cycles) for small-scale energy generation from carbon-neutral biomass. Such a goal has never been achieved before, since combined cycles are generally suited only for large-scale applications and for clean fuels. In order to adapt combined cycles to small-scale energy generation using dirty fuels, the implementation of cost-effective and commercially available components is studied, such as the use of a turbocharger and a power turbine taken from the automotive industry. The ongoing realization of the first prototype of small-scale combined cycle is presented in this paper, providing a detailed description of both the plant architecture and the main components chosen. In addition, a commercially available tool (Cycle Tempo) is used to demonstrate the high feasibility and potential of the plant in terms of efficiency. To that end, different plant configurations are studied and the effects of losses on the plant performance are investigated in detail.

Peer-review under responsibility of the scientific committee of the 72nd Conference of the Italian Thermal Machines Engineering Association

Keywords: combined cycle; biomass; CHP; Cycle Tempo

1. Introduction

This paper presents an ongoing experimental and numerical activity aimed at constructing and testing the first prototype of small-scale combined cycle for the energy generation from solid biomass, such as pruning residues.

* Corresponding author. Tel. 0805963470.
E-mail address: paolo.tamburrano@poliba.it

1876-6102
Peer-review under responsibility of the scientific committee of the 72nd Conference of the Italian Thermal Machines Engineering Association
10.1016/j.egypro.2017.08.294

Unlike other forms of biomass [1], pruning residues can be considered carbon-neutral, both because pruning is a necessary practice for maintaining the health of trees, and because, when they are burned, the residues release the same amount of carbon as that absorbed from the atmosphere during their life. Unfortunately, pruning residues are often burned on fields without being valorized for energetic purposes [2]. As a demonstration of their unexploited potential, it was recently estimated that the quantity of oil tree pruning residues produced in a small area surrounded Bari Airport (Italy) would be able to satisfy most of the electrical and thermal demand of the airport [2].

An effective strategy in terms of sustainability is to burn pruning residues for energy generation on site (where the biomass is produced), thus avoiding the production of CO2 from feedstock transportation. To that end, small-scale power plants are needed to effectively transform the solid biomass input power into either electrical power or combined heat and power (CHP). However, the direct use of biomass needs further developments in relation to the stat of the art. Over the past years, the industrial and university research has been focused on the development and improvement in ORC power plants, which are, at the state of the art, the most widespread and profitable technology applicable to burning solid biomass for small-scale cogeneration [3]. The main problem regarding ORC power plants is the need for organic fluids, which usually have a global warming potential greater than zero and can be toxic and inflammable. In addition, commercially available units are mainly manufactured for electrical power greater than 100 kW_e [4]; for lower values of electrical power, few models are available, with an electrical efficiency that is usually well below 15% [5]. For these reasons, rather than focusing on the improvement in the ORC technology, the authors of this paper have been concentrating their efforts on a different strategy, which aims at developing a small-scale combined cycle based on a gas turbine and a water steam cycle, as a concrete alternative to stand alone ORC plants [6, 7]. A parallel research activity carried out by the same authors is focused on the realization of high temperature heat exchangers (HTHEs) to be applied to such combined cycles [8-10].

The presented combined cycle can be used either for the generation of electrical energy or for CHP generation. The latter configuration was studied in a previous paper, showing that an electrical efficiency of about 0.22 and an overall efficiency of about 0.65 can be obtained [6]. These values are related to the use of a shell and tube heat exchanger made of a high temperature alloy; higher values, namely an electrical efficiency of about 0.25 and an overall efficiency of about 0.7, can potentially be reached by using the novel immersed particle heat exchanger (IPHE), which is currently being developed [10]. This performance prediction was obtained for a size of the plant of about 100 kW_e, neglecting the pressure losses caused by the components placed downstream of the gas turbine. The latter assumption was made in order to determine the maximum potential of the combined cycle.

This paper, in addition to presenting the ongoing realization of the experimental plant, numerically assesses the plant performance in the case of non-negligible pressure drops downstream of the gas turbine (e.g., due to the use of a fluidized bed combustor). Furthermore, the effects of the HTHE efficiency upon the performance of the plant is investigated in detail. The analysis is performed for a size of the plant of about 50 kW_e and in the configuration employing two turbines in series for the topping cycle.

2. Architecture of the combined cycle and thermodynamic model

A scheme of the proposed power plant is reported in Fig. 1. The plant is composed of a topping cycle and a bottoming one. The former is an externally fired Joule Brayton cycle and the latter is a steam cycle. Both cycles have the particularity of being realized using commercially available components, such as the gas generator and the power turbine as well as the micro-steam turbine. Air at ambient conditions is compressed by the compressor (C) and is delivered to the high temperature heat exchanger (HTHE), where it undergoes a temperature increase necessary to run the turbine (T). After exiting the first turbine, the air enters the power turbine (PT) transforming the air enthalpy into mechanical work, which in turn is transformed into electrical energy by the electric generator connected with the power turbine. The warm air, rather than being exhausted into the environment, is delivered to an external combustor as oxidation stream converting the chemical energy possessed by the solid biomass into thermal energy.

Since the flue gases exiting the HTHE still have a high enthalpy content, they are used in a bottoming water-steam cycle generating additional electrical energy. The core of the bottoming cycle is the heat recovery steam generator (HRSG), which is composed of an economizer, a vaporizer and a super-heater. The HRSG allows superheated steam to be generated; the steam enters a small steam turbine (ST), which is connected to a second electric generator. After exiting the turbine, the steam is condensed and delivered again to the HRSG in order to close the cycle. This configuration can be used either for the production of electrical energy or for the combined generation of heat and power. The production of useful thermal power is achieved in the condenser: the hot water, generated by the steam condensation, can be used for thermal purposes at a temperature that depends on the condensation pressure chosen.

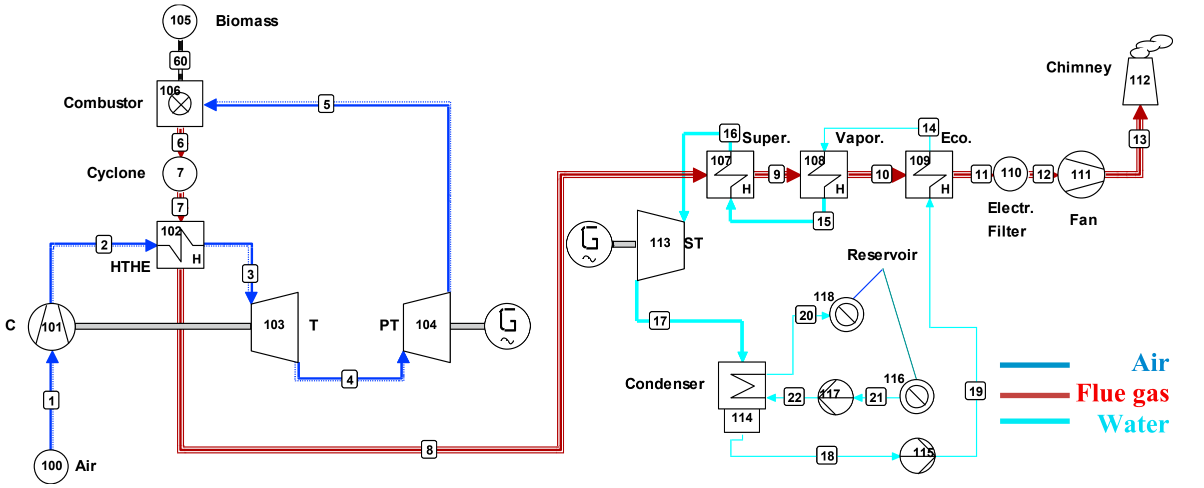

Fig.1. Scheme of the combined cycle (obtained with Cycle Tempo)

A fan is positioned at the end of the circuit of the flue gases, in order to compensate for the pressure losses in line 5-12. An alternative scheme consists in making the power turbine work with a discharge relative pressure greater than zero, without the need for an additional fan; as a result, in this alternative configuration, only the compressor is responsible for moving the air flow in the circuit.

The thermodynamic performance of the plant was investigated using the software Cycle Tempo [11]. Both the air and the flue gases were modelled using the GasMix model, which implements the ideal gas equation of state $pv=R_uT/M$ (p being the pressure, v the specific volume, R_u the universal gas constant, T the temperature and M the molecular weight of the gas) with a temperature dependent specific heat of the form: $c_p(T)=c_1+c_2T+c_3T^2+c_4T^3+c_5T^4$ (with the five constants depending on the gas composition). The water (both liquid and steam) of the bottoming cycle was modelled by means of the IF97model, which implements the thermodynamic and transport properties of water and steam according to the IAPWS -IF97 industrial standard.

3. Experimental plant

The combined plant is currently under construction at LabZERO, which is a multidisciplinary laboratory located at Politecnico di Bari. Fig. 2 shows a view of the plant, where it is possible to notice its main components. On the right of this figure is the reservoir for the solid biomass, from which the biomass is lifted by a screw conveyor and delivered to the combustor, where the combustion between the air discharged by the power turbine and the biomass takes place. A fluidized bed combustor was implemented, as it allows woody biomass to be burned efficiently without the necessity for expensive fuel preparation (e.g., pulverising).

Fig. 2 also shows the HTHE, having a U-shape. The HTHE is the most critical component of the plant, as it must be able to withstand the high temperature of the flue gases while ensuring low pressure drops. The technical difficulty in the realization of this component, due to high temperature cracking, has hindered the diffusion of externally fired gas turbines; as a matter of fact, only internal combustion gas turbines are employed for the generation of electrical energy. However, the topping cycle of the proposed plant has lower pressures and temperatures than typical industrial gas turbines. In fact, the maximum pressure of the air will be about 3.5-4 bar, while the temperature of the flue gases exiting the combustor will be maintained lower than 900 °C. In these conditions, the use of a metallic material for high temperatures, such as a Nichel alloy or stainless steel, can guarantee the thermo-mechanic resistance of the heat exchanger. In this case, the HTHE is made of stainless steel and has the typical architecture of a shell and tube heat exchanger, with the hot flue gases flowing inside the internal tubes and the air flowing outside the tubes but inside the shell. The tubes are bent in the shape of a U, in order to reduce the overall height of the heat exchanger.

Fig. 2. Prototype of combined cycle

An alternative to this HTHE architecture is currently being developed that is based on the use of ceramic particles capable of transferring heat between the flue gases and the air flowing in two columns separately. This heat exchanger, named the immersed particle heat exchanger, has the potential to reach higher temperatures and hence higher efficiency for the topping cycle [10].

Fig. 3 shows the gas-generator and power turbine that were purchased from a manufacturer of turbo-compounds for trucks. The power turbine is usually connected to the crank shaft by means of a gear train and has the function to provide a surplus power to the engine, which can be as high as 10% of the overall power generated by an engine. The application of commercially available turbo-compound components to a power plant for energy generation would not require substantial modification to the whole group (turbocharger and power turbine), except for the fact that, in the proposed scheme, the power turbine is connected to an electrical generator, rather than being connected to the crank shaft.

As far as the steam turbine of the bottoming cycle is concerned, the choice was between volumetric expanders and micro-steam turbines. The main advantage offered by the utilization of volumetric expanders (among which the most effective seem to be screw expanders) is that they can operate with wet steam, thus making the superheating unnecessary. Another peculiar characteristic is the low rotational speed, which can permit the direct connection with an electric generator without the need for an expensive inverter [6]. The main drawback is the low isentropic efficiency; in fact, it can be estimated, using available data, that the isentropic efficiency is very low for volumetric expanders. With regard to steam turbines, several advances have been recently achieved in the design of small- and micro-steam turbines. As an example, the “green steam turbine” is a commercially available model, whose cost amounts to a few thousand euros, capable of providing up to 15 kW_e with maximum pressure of 10 bar and maximum temperature of 225-250 °C. By virtue of its high level of performance, this model was selected as the steam expander for the bottoming cycle. Figure 3 shows the steam turbine with all the necessary components of the steam cycle, namely vacuum pump, vacuum vessel, condenser, feed pump, piping. It is also possible to note, at the top of the turbine, six nozzles which allow the output power to be modulated by adjusting the steam flow rate entering the turbine.

As occurs for heavy-duty steam turbines, superheating is mandatory because such small steam turbines can operate only with dry steam, and the amount of droplets at the turbine outlet must be kept as low as possible. As a result, the HRSG cannot be a simple saturated steam generator, but requires the super-heater, with the main drawback of slightly increasing the costs of the plant because of the presence of this additional component. HRSGs employed for large-scale plants are usually water-tube boilers, as these allow reaching higher pressures than fire-tube boilers. In the proposed plant, the thermomechanical resistance in the HRSG is not an issue since the maximum operating pressure is very low compared to large scale units. As a result, the selected unit is a fire-tube boiler for the generation of saturated steam followed by a shell and tube heat exchanger serving as the super-heater.

Fig. 3. Turbocharger (left), power turbine (center), steam turbine (right)

With regard to the abatement of emissions, the low temperature of the flue gases (850-900 °C) avoids the formation of thermal NOx; the sulphur dioxide content will be abated by injecting CaCo3 into the combustor (the reaction is active in the temperature range 850-900 °C). A cyclone separator and an electrostatic filter (positioned as shown in Fig. 1) will be responsible for the abatement of particulate matter.

The experimental plant will allow retrieving the thermodynamic cycles, measuring the pressure drops and evaluating the electrical efficiency under different operating conditions. Thermocouples and pressure transducers are going to be positioned at the inlet and outlet sections of the main components; the output electrical power will be measured by the power inverters, whereas the fuel consumption will be estimated by measuring the mass flow rate of the air and the flue gases through the use of flow meters to be inserted in different positions of the plant.

4. Results

The aim of the present calculations is to investigate the potential of the proposed plant by fixing some parameters according to the available technologies. The pressure at the exit of the compressor, isentropic efficiency of the compressor and isentropic efficiency of the gas turbines were set equal to 3.5 bar, 0.75 and 0.8, respectively; this is a very common combination according to the available maps of commercial turbochargers. The value of the mass flow rate of air was set equal to 0.45 kg/s in order to achieve an overall electrical power lower of about 50 kW_e. As mentioned before, the chosen typology for the combustor is a fluidized bed, whose operating temperature will be between 850 and 900 °C. In the present calculations, a value of 870 °C was fixed for the flue gas temperature at the exit of the combustor. The temperature at the inlet of the steam turbine was set to the value of 250 °C, which is the maximum temperature that the steam turbine can withstand. The isentropic efficiency and mechanical efficiency of the steam turbine were set to 0.5 and 0.98, respectively. The temperature of the exhaust gas exiting the HRSG was set to 150 °C to avoid the corrosion of the ducts caused by the sulphur content in the flue gases. The vaporization pressure was set equal to 10 bar (equal to the maximum operating pressure of the steam turbine); the minimum value allowed for the pinch point temperature difference was set to 10 °C. The overall pressure drop in line 5-12 was set to 0.1 bar, which is the expected pressure drop mainly caused by the use of a fluidized bed combustor.

The air composition was considered that of standard air with 60% of relative humidity, whereas the flue gas composition was automatically calculated by Cycle Tempo. The biomass composition was set equal to the typical composition of oil tree pruning residues, with 20% of humidity and a lower heating value of 14000 kJ/kg [2]. The efficiencies of the electric generators and the electric motors were not considered in this analysis.

Fig. 4 shows the results obtained for a HTHE efficiency (defined as the actual air enthalpy increase over the ideal air enthalpy increase) of 0.8 and for a steam condensation pressure of 0.1 bar. Hereafter, this case study is referred to as case study 1 for ease of comprehension; the corresponding performance parameters are shown in Table 1.

The electrical efficiency of the plant (defined as the total useful electrical power over the input thermal power) is 18% for case study 1, with a net overall electrical power of about 42 kW_e. It is noteworthy that the fan placed at the end of the path of the flue gases absorbs a great part of the generated electrical power. In fact, in the ideal conditions of null pressure losses in line 5-12 (and hence null power absorbed by the fan), the electrical efficiency would be raised to a very high level of about 22%. This suggests that alternative solutions to the fluidized bed combustor can be studied to be employed in the present plant, in order to reduce the pressure losses and hence increase the efficiency.

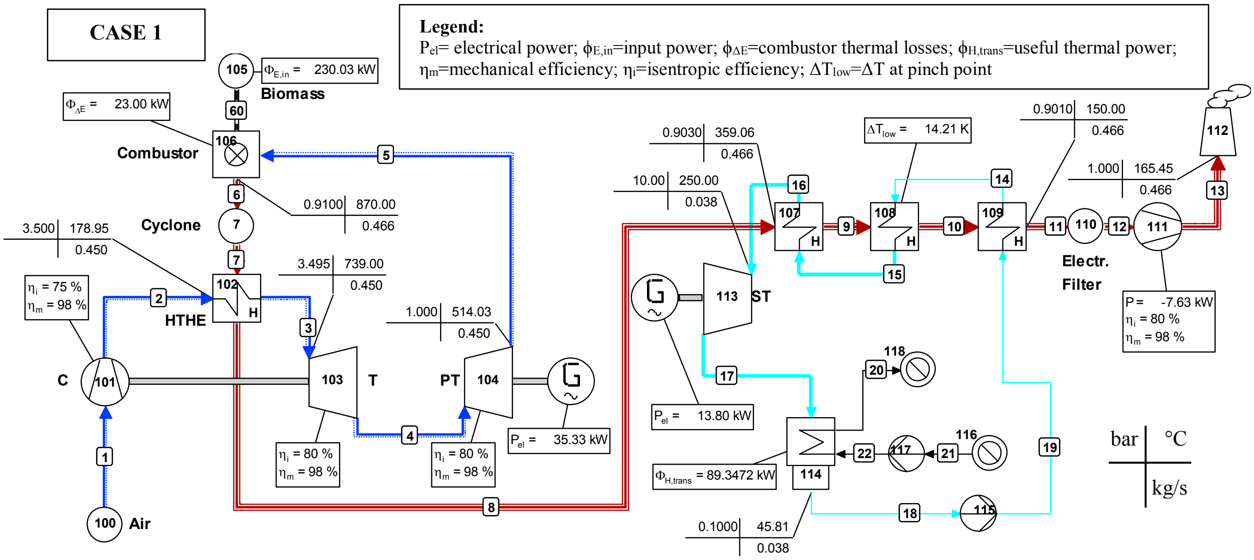

Fig. 4. Results for case study 1 (HTHE efficiency=0.8; steam condensation pressure=0.1 bar; overall pressure drop=0.1 bar)

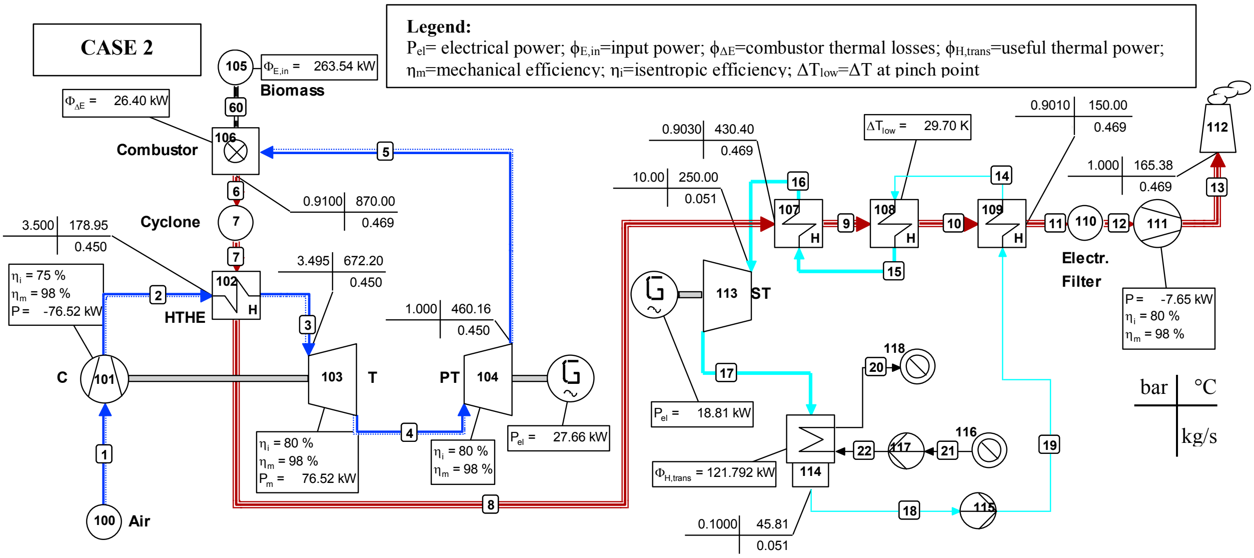

Fig. 5. Results for case study 2 (HTHE efficiency=0.7; steam condensation pressure=0.1 bar; overall pressure drop=0.1 bar)

Fig. 5 reports the calculation performed with the same assumptions as those of case study 1, except for the HTHE efficiency, which in this case was set to 0.7. This case study is referred to as case study 2. Although the HTHE efficiency was reduced by only 12.5%, it is shown that the results have changed remarkably. In fact, a lower value of the HTHE efficiency causes a lower value of the air temperature at the inlet of the gas turbine, which in turn causes a reduction in the electrical power of the topping cycle. In contrast, the temperature of the flue gases at the exit of the HTHE results to be increased, thus allowing a greater quantity of thermal energy to be transferred to the bottoming cycle, which therefore can generate more electrical power than the previous scheme. As a result, the HTHE efficiency can be considered as an “energy divider”, as its value determines how the input thermal power is divided between the topping cycle and the bottoming one. However, it should be noted that the temperature reduction at the inlet of the gas turbine also determines a reduction in the efficiency of the topping cycle; consequently, the overall electrical efficiency of the combined cycle has been lowered from 18 % to about 15 % (see Table 1), thus evidencing that the reduction in the HTHE efficiency has a negative effect on the electrical efficiency of the combined cycle. The use of a HTHE efficiency of 0.7 can become more profitable if a CHP arrangement is considered, in which the hot water generated in the condenser (condensation temperature=45.81°C) can be used for sanitary needs or for space heating with radiant panels. As shown in Table 1, both in case study 1 and in case study 2, the CHP arrangement allows a great increase in the first law efficiency (namely, thermal efficiency + electrical efficiency) of the combined cycle, by virtue of the generation of a large quantity of useful thermal power (89 kW and 122 kW, respectively). However, also in such a CHP arrangement, the use of a HTHE efficiency of 0.8 must be preferred to the use of a HTHE efficiency of 0.7; in fact, the comparison in terms of the second law efficiency (calculated, according to the symbols of Fig. 4 and Fig. 5, as: η_{II}=[P_{el} +(1-T_{amb}/T_{cond})$\phi_{H,trans}$]/$\phi_{E,in}$, where T_{amb} is the ambient temperature assumed equal to 288 K and T_{cond} is the condensation temperature) shows that the second law efficiency of case study 1 (approx. equal to 0.22) is remarkable, in spite of the small-scale application, and is greater than that of case study 2 (approx. equal to 0.19).

The plant performance is finally investigated for a steam condensation pressure of 0.2 bar (condensation temperature = 60 °C), allowing hot water to be generated in the condenser at a temperature higher than that of the previous case studies and more compatible with traditional space heating systems. Higher condensation pressures were not considered in this analysis, because the manufacturer of the green steam turbine suggests that the condensation pressure should not exceed 0.2 bar for a long time in order to extend the life of the bearings of the steam turbine, although the realization of a customized solution for the bearings should not be a particular issue in order to allow for long operating conditions with higher condensation pressures and, hence, higher condensation temperatures.

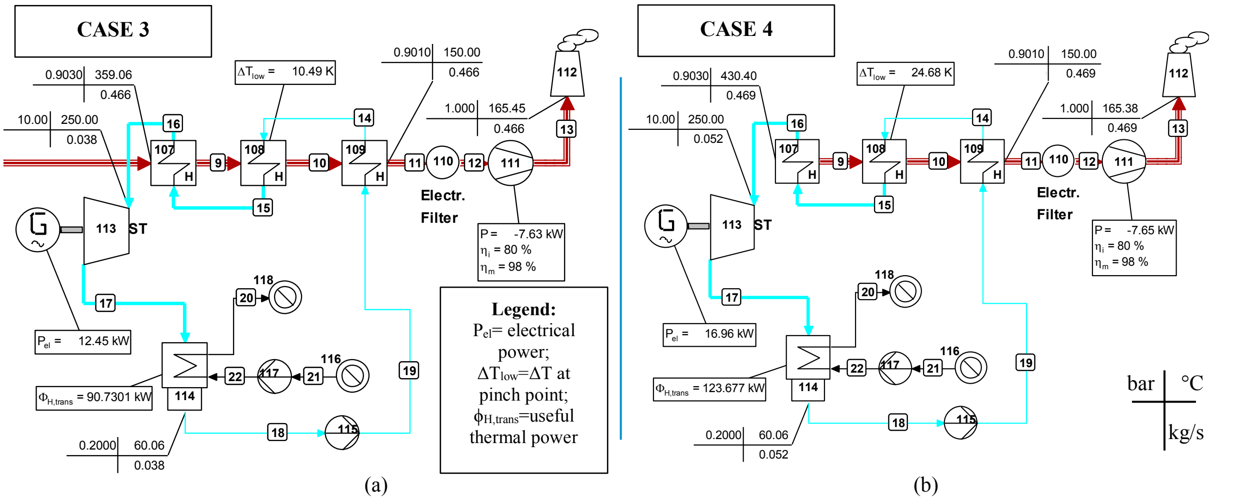

Fig. 6. (a): bottoming cycle of case study 3 (HTHE efficiency=0.8; steam condensation pressure=0.2 bar; overall pressure drop=0.1 bar); (b): bottoming cycle of case study 4 (HTHE efficiency=0.7; steam condensation pressure=0.2 bar; overall pressure drop=0.1 bar)

Fig. 6a shows the bottoming cycle achieved for a HTHE efficiency of 0.8 and a condensation pressure of 0.2 bar (the topping cycle is the same as that reported in Fig. 4), while Fig. 6b shows the bottoming cycle for a HTHE efficiency of 0.7 and a condensation pressure of 0.2 bar (the topping cycle is the same as that reported in Fig. 5). These two configurations are denoted by case study 3 and case study 4, respectively; the corresponding performance parameters are reported in Table 1, for ease of comparison. Table 1 and Figures 4-6 reveal that, for a given HTHE efficiency, the increase in the condensation pressure entails a decrease in the electrical power and electrical efficiency (because of the lower enthalpy drop in the steam turbine), but an increase in the generated useful thermal power. In addition, the comparison between case study 3 and case study 4 shows that, also for a condensation pressure of 0.2 bar, the higher the HTHE efficiency, the higher the second law efficiency (being approx. 0.23 for a HTHE efficiency of 0.8, and being about 0.2 for a HTHE efficiency of 0.7).

It results that the use of a HTHE with an efficiency of 0.8 must be preferred to enhance the plant performance, regardless of the value of the condensation pressure, which can be changed according to the thermal needs. As a final consideration, it should be noted that the electrical power of the bottoming cycles of Fig. 5 and Fig. 6b is greater than the maximum power generated by the green steam turbine (15 kW); in these cases, either two turbines in series could be used or the temperature of the flue gases could be slightly increased to adjust the steam flow rate.

Table 1. Performance parameters for the four configurations analyzed (overall pressure drop equal to 0.1 bar for all the case studies)

	Case study 1: HTHE eff.=0.8; cond. pressure=0.1 bar	**Case study 2:** HTHE eff.=0.7; cond. pressure=0.1 bar	**Case study 3:** HTHE eff.=0.8; cond. pressure=0.2 bar	**Case study 4:** HTHE eff.=0.7; cond. pressure=0.2 bar
Electrical efficiency	0.1802	0.1470	0.1743	0.1400
First law efficiency	0.5686	0.6092	0.5686	0.6092
Second law efficiency	0.2178	0.1918	0.2278	0.2036

5. Conclusions

This paper presents the architecture and the main components of the first prototype of small-sale combined cycle for energy generation from solid biomass, which is under construction. A commercially available software package is employed to investigate the performance of the power plant under different operating conditions. Both the thermal losses in the combustor and the pressure losses in the components downstream of the gas turbine are taken into account. The results show that a high electrical efficiency of about 18% can be achieved for a heat exchanger efficiency of 0.8, although a high value for the pressure losses was considered (0.1 bar). However, it was underlined that the reduction in the pressure losses (e.g., by adopting different solutions for the combustor) can be instrumental in further enhancing the plant performance, with theoretical values being estimated greater than 20% for negligible pressure losses. The effect of the reduction in the heat exchange efficiency upon the plant performance was also investigated by considering a HTHE efficiency of 0.7. It was shown that such a reduction penalizes the electrical efficiency but allows more useful thermal power to be generated in the CHP arrangement; however, the calculation of the second law efficiency demonstrated that the employment of a HTHE with an efficiency of 0.8 must be preferred to maximize the plant performance in all the case studies considered.

Acknowledgments

This research is co-funded by “Fondo di Sviluppo e Coesione 2007-2013 – APQ Ricerca Regione Puglia, Programma regionale a sostegno della specializzazione intelligente e della sostenibilità sociale ed ambientale - FutureInResearch”. This work is also supported by Progetto ZERO (Zero Emission Research Option).

References

[1] Zanchi, G., Pena, N., & Bird, N. (2012). Is woody bioenergy carbon neutral? A comparative assessment of emissions from consumption of woody bioenergy and fossil fuel. GCB Bioenergy, 4(6), 761-772.

[2] Amirante, R., Clodoveo, M. L., Distaso, E., Ruggiero, F., & Tamburrano, P. (2016). A tri-generation plant fuelled with olive tree pruning residues in Apulia: An energetic and economic analysis. Renewable Energy, 89, 411-421.

[3] Dong L, Liu H, Riffat S. Development of small-scale and micro-scale biomass fuelled CHP systems – a literature review. Appl Therm Eng 2009;29(11):2119–26.

[4] Tocci, L., Pal, T., Pesmazoglou, I., & Franchetti, B. (2017). Small Scale Organic Rankine Cycle (ORC): A Techno-Economic Review. Energies, 10(4), 413.

[5] http://www.zuccatoenergia.it/. Accessed on 1/05/2017

[6] Amirante, R., & Tamburrano, P. (2015). Novel, cost-effective configurations of combined power plants for small-scale cogeneration from biomass: Feasibility study and performance optimization. Energy Conversion and Management, 97, 111-120.

[7] Amirante, R., Coratella, C., Distaso, E., & Tamburrano, P. (2015). A small size combined system for the production of energy from renewable sources and unconventional fuels. Energy Procedia (2015) pp. 240-248. DOI: 10.1016/j.egypro.2015.12.090.

[8] Catalano, L. A., De Bellis, F., Amirante, R., & Rignanese, M. (2011). An immersed particle heat exchanger for externally fired and heat recovery gas turbines. Journal of Engineering for Gas Turbines and Power, 133(3) doi:10.1115/1.4002157.

[9] Amirante, R., & Tamburrano, P. (2014). High temperature gas to gas heat exchanger based on a solid intermediate medium. Advances in Mechanical Engineering, Volume 2014, 20 pages

[10] Amirante, R., Tamburrano, P. & Disatso E. (2017). Novel, cost-effective configurations of combined power plants for small-scale cogeneration from biomass: design of the Immersed Particle Heat Exchanger. Energy Conversion and Management, 148, 876-894.

[11] Delft University of Technology, Cycle-Tempo, version 5.1, software