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Thermodynamic analysis of a small scale combined cycle for energy generation from carbon neutral biomass

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Abstract

The aim of this paper is to investigate the thermodynamic performance of a novel small-scale power plant that employs a combined cycle for the energy generation from carbon-neutral biomass, such as pruning residues. The combined cycle is composed of an externally fired Joule Brayton cycle followed by a bottoming steam cycle. The topping cycle has the unique particularity of being composed of a cost-effective turbocharger taken from the automotive industry, in place of a more expensive commercial micro-turbine. The turbocharger can be either directly connected to the electric generator (after a few modifications) or coupled (without modifications) with a power turbine moving the generator. The use of solid biomass in the proposed plant is allowed by the presence of an external combustor and a gas-to-gas heat exchanger. The warm flue gases exhausted by the topping cycle are used in a bottoming cycle to produce steam, which can power a steam expander.
This paper thermodynamically assesses the novel combined cycle in the configuration for the topping cycle that employs a turbocharger coupled with a power turbine capable of generating 30 kW of electrical power. Furthermore, the comparison between the performance obtained using the bottoming water Rankine cycle and a bottoming Organic Rankine Cycle is provided.

Keywords: combined cycle; biomass; ORC; Rankine cycle

1. Introduction

The 2015 United Nations Climate Change Conference has stated that all nations have to promote the exploitation of renewable sources for energy generation in order to reach the target of limiting global warming to less than 2 °C compared to pre-industrial levels [1]. This goal can be achieved by boosting the employment of technologies able to produce electrical and useful thermal energy from wind and solar energy (which could become more effective if it was supported with energy storage systems [2]), hydropower, geothermal energy as well as biomass. This paper is focused on the latter renewable energy source, by proposing a novel, cost-effective power plant capable of generating electrical energy from some forms of biomass that can be considered carbon-neutral.

The term carbon-neutral means that the transformation of biomass into useful energy does not alter the overall amount of carbon present in the environment. However, not every form of biomass is really carbon neutral, as discussed in [3]; among those typologies of biomass that can be considered carbon neutral for energy generation are pruning residues, provided that they are exploited on site to avoid additional CO_2 generation from feedstock transportation [4-9].

The research carried out up to now by the authors of this paper intends to develop a novel, small-scale power plant capable of generating energy from solid biomass directly on site, without the need of transforming it into syngas or other biofuels [10-11], thus proposing a valid alternative to “stand alone” Organic Rankine cycles (ORCs), which are, at the

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state of the art, the best technology for the direct transformation of solid biomass into electrical and thermal energy [12]. With regard to commercially available small scale ORC plants, the electrical efficiency is usually below 15% when the generated electrical power is lower than 100 kW [13]. The electrical efficiency usually increases with the increasing size of the plant; e.g., an ORC plant produced by an important manufacturer [14] is capable of generating 200 kWe with an electrical efficiency of about 16.5%.

The proposed power plant has the particularity of being composed of an externally fired Joule Brayton cycle serving as the topping part, plus a Rankine cycle serving as the bottoming part. The novel idea is to apply a combined cycle (gas turbine + Rankine cycle) to dirty fuels (such as solid biomass) and small-scale generation. Instead, as it is widely recognised, combined cycles are only suited for both clean fuels (to prevent the gas turbine blades from being damaged) and large-scale generation. The novel configuration of the combined cycle was assessed thermodynamically in [10], with the assumption of using water as the working fluid of the bottoming cycle and for a generated electrical power of about 100 kW. The thermodynamic optimization performed in [10] showed that a maximum electrical efficiency of about 0.25 can be achieved, which is a very good level of performance for such a low power plant size. The configuration proposed in [10] was also capable of producing useful thermal power, thus achieving a combined heat and power (CHP) arrangement with a maximum overall efficiency (electrical plus thermal efficiency) of about 0.7 [10].

This paper further investigates this novel typology of power plant by exploring its thermodynamic feasibility for lower useful electrical power (less than 50 kW) and in a full electrical power generation arrangement. Furthermore, the main aim of this paper is to compare the plant configuration employing the bottoming water Rankine cycle with a similar plant configuration employing an ORC as the bottoming cycle. The comparison between the two plant configurations is performed in terms of overall electrical efficiency (useful electrical power over input thermal power).

2. Methodology

2.1 Proposal of plant configurations

The architecture of the proposed small scale combined cycle is shown in Figure 1. Clean air is compressed by a centrifugal compressor and delivered into a gas to gas heat exchanger aimed at increasing the air enthalpy. At first the high temperature air expands through a turbine (T_1) connected to the centrifugal compressor and then is conveyed into a second turbine (T_2) connected to an electric generator. The first turbine has the only task of driving the compressor, whereas the second turbine serves as the power turbine producing the shaft work output. Both turbines are moved by clean air, therefore their mechanical integrity is not at risk although dirty fuels are burned in the combustor. The air exhausted from the power turbine is delivered into the external combustor, where the combustion between the air and biomass generates high temperature flue gases. These exit the combustor and enter a cyclone separator (needed to clean the flue gases); afterwards, the flue gases enter the high temperature heat exchanger (HTHE) and transfer their thermal power to the clean air. After leaving the heat exchanger, the flue gases are conveyed into the bottoming cycle of the plant. In the configuration presented here, only the compressor is responsible for moving the air flow in the circuit, without the need for an additional fan; as a result, the discharge pressure of the power turbine can be greater than the atmospheric pressure in case of non-negligible pressure losses.

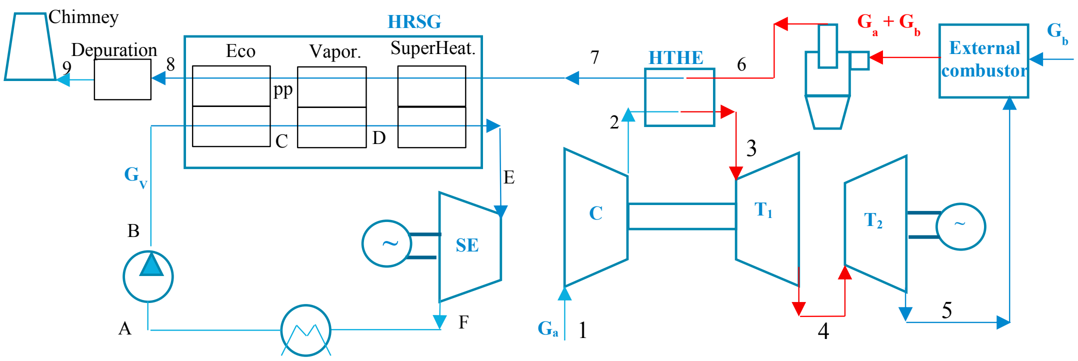

Fig.1. Scheme of the combined plant: topping part employing two turbines (right) and bottoming part employing a water Rankine cycle (left)

Summarizing, the topping cycle presents the following innovative aspects:

I. The compressor-turbine group does not need to be manufactured ex-novo, but a turbocharger and a power turbine for trucks can be used for this purpose.
II. The combustion chamber is external instead of being internal (as occurs in typical gas turbine plants).

III. The input thermal power is provided to the working fluid (clean air) by means of a gas to gas heat exchanger, which has to withstand the high temperatures required by a gas turbine while ensuring negligible pressure drops.

Item I), namely, the use of a turbocharger taken from the automotive industry, has the potential to reduce the capital costs of the plant, because this technology is mature and cheap. The plant analysed thermodynamically in [10] was based on a single-shaft configuration (i.e., one turbine and one compressor); instead, this paper analyses the two-shaft configuration (two turbines and one compressor), which can provide advantages at partial load.

Items II) and III) are necessary for the use of dirty fuels instead of clean ones. In order to burn solid biomass effectively, either a fluidized bed combustor or a moving grid burner can be used to accomplish this task. Several models of combustors are available on the market that can be used to perform the combustion of solid biomass with the high temperature air discharged from the turbine [10]. Several models are also characterized by excellent efficiency, as high as 90% [10].

With regard to the gas to gas heat exchanger, it is the most critical component, as several companies tried to design high temperature gas to gas heat exchangers, mostly with negative results due to high temperature cracking. However, the design of this component can take advantage of the fact that the pressures and the temperatures as well the gas flow rate of the proposed cycle are not as high as those of typical industrial gas turbines. The problematics concerned with the application of gas to gas heat exchangers were discussed thoroughly in previous papers [15-18], with two configurations being proposed by the same authors so far. The first one is called "the immersed particle heat exchanger (IPHE)" and is based on the concept of using an intermediate ceramic medium (namely small ceramic particles) to transfer heat from two fluids without mixing them. The second configuration employs a shell and tube heat exchanger with the particularity of being made of metallic materials capable of withstanding very high temperatures, such as a Nichel alloy or a stainless steel. The IPHE has a higher potential than the Nichel alloy heat exchanger, because the former does not employ metallic surfaces for the heat exchange and therefore can withstand higher temperatures [19]. Concerning the dimensions of the heat exchanger, it would not be an issue, because the heat exchanger would be very compact, with small size and reduced material content, considering that the flow rates of air and flue gas are very low in this type of power plant.

Figure 1 also shows a schematic representation of the bottoming cycle. The flue gases exiting the topping cycle are delivered into a heat recovery steam generator (HRSG) allowing the remaining enthalpy content to be transformed into useful energy. The architecture shown in Fig. 1 makes use of water as working fluid for the bottoming cycle. The use of water for such a low temperature level and gas flow rate was unfeasible in the past, because of the poor efficiency of steam expanders. However, recently there have been several advances in the design of steam turbines, which are more efficient than in the past even for very low levels of gross power. For example, the ''Green steam turbine'' is a commercially available model whose cost amounts to a few thousand Euros (by virtue of the single stage configuration) and which is capable of generating a maximum electrical power of 15 kW, with a maximum pressure of 10 bar and a maximum temperature of 225-250 °C [20]. Using the data provided by the manufacturer, and assuming an efficiency for the electric generator of about 95÷97%, the authors estimated an isentropic efficiency of approximately 50%, which represents a high level of performance despite the very low electrical power.

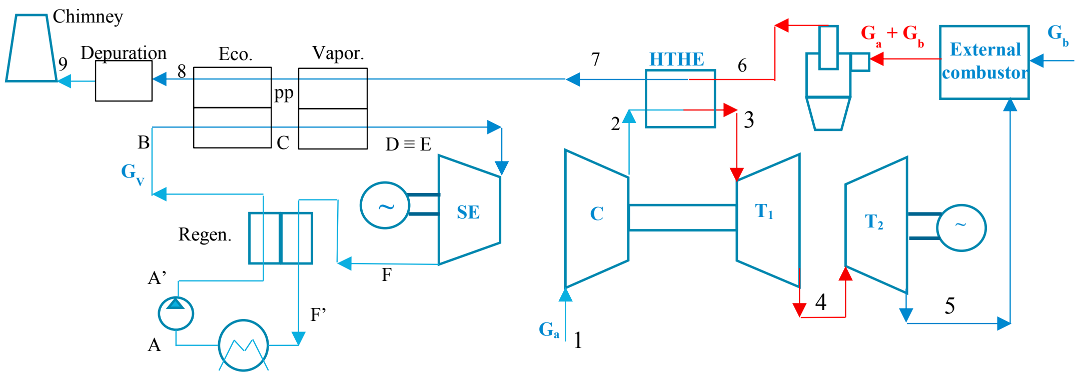

Fig. 2. Second scheme for the combined plant: topping part employing two turbines (right) and bottoming part employing an ORC (left)

Figure 2 reports a second configuration proposed for the bottoming cycle. In this case, the working fluid is based on an organic molecule, thus the bottoming part is to be defined as an Organic Rankine Cycle (ORC). The main difference with the previous architecture is that the presence of the super-heater is not mandatory in the scheme of Fig. 2, while a regenerator is employed to increase the efficiency of the cycle. The two configurations are compared using the thermodynamic model described in the following subsection. It should be noted that the bottoming cycle can be used either as a power plant for the only generation of electrical energy or as a plant for the generation of electrical and thermal energy (cogeneration). In the latter case, the condensation pressure must be raised in order to increase the condensation temperature, thus allowing high temperature heat recovery from the vapour condensation. Such a plant configuration was discussed in a previous paper [10], whereas the present analysis is only concerned with the generation of electrical power, so the condensation temperature will be kept very low to maximize the electrical power generated.

2.2 Thermodynamic model

The cycles of Figures 1 and 2 were modelled by means of a calculation code using the libraries "Fluid Prop" for the evaluation of the fluid enthalpies [21]. The flue-gas properties were assumed equal to those of air; the latter was modelled with the "gas mix model", which implements the ideal gas equation of state with a temperature dependent specific heat. The water was modelled by means of the IF97 model, which implements the thermodynamic and transport properties of water and steam according to the IAPWS -IF97 industrial standard [21]. The organic fluid was modelled with the "FreeStanMix" subroutines [21].

The following equations were employed for the simulation. The isentropic efficiency of the compressor and turbines ($\eta_{is,c}$, $\eta_{is,t1}$, $\eta_{is,t2}$) were calculated as follows (see Fig. 1 and Fig. 2 for explanation of symbols):

$$\eta_{is,c} = \frac{h_{2is}-h_1}{h_2-h_1}\ , \quad \eta_{is,t1} = \frac{h_3-h_4}{h_3-h_{4is}}\ , \quad \eta_{is,t2} = \frac{h_4-h_5}{h_4-h_{5is}}\ . \tag{1}$$

The equilibrium of the compressor-turbine group states that the power produced by the first turbine equals the power absorbed by the compressor:

$$\eta_{m,t1} G_a (h_3 - h_4) = \frac{G_a(h_2-h_1)}{\eta_{m,C}}\ , \tag{2}$$

where $\eta_{m,t1}$ and $\eta_{m,c}$ denote the mechanical efficiency of the first turbine and compressor, respectively. Furthermore, the pressure increase provided by the compressor, namely p_2-p_1, must be equal to the sum of the pressure jump across the two turbines, p_3-p_5, and the overall pressure drop along line 5-9 of the circuit, denoted by Δp_{loss} (assuming p_2=p_3) :

$$p_2\text{-}p_1 = p_3\text{-}p_5 + \Delta p_{loss}\ . \tag{3}$$

The power generated by the second turbine (power turbine) was evaluated as follows:

$$P_{u,TC} = \eta_{m,t2} G_a (h_4 - h_5)\ , \tag{4}$$

$P_{u,TC}$ and $\eta_{m,t2}$ denoting the useful power and the mechanical efficiency of the power turbine, respectively. The heat exchange within the gas-to-gas heat exchanger implies that the following equation must be satisfied:

$$(G_a + G_b)(h_6 - h_7)\eta_{th,HTHE} = G_a(h_3 - h_2)\ , \tag{5}$$

where $\eta_{th,HTHE}$ accounts for the thermal dispersions from the HTHE towards the environment. Instead, the heat exchange efficiency, ε_{HTHE}, is the ratio of the exchanged thermal power to the ideally exchanged thermal power, namely:

$$\varepsilon_{HTHE} = \frac{h_3-h_2}{h_6-h_2}\ . \tag{6}$$

The thermal balance of the external combustor was calculated as follows:

$$\eta_b G_b H_i + G_b h_b + G_a h_5 = (G_a+G_b)h_6\ , \tag{7}$$

η_b , G_b , H_i and h_b denoting the combustor efficiency, the mass flow rate of the fuel, the lower heating value of the fuel and the inlet enthalpy of the fuel, respectively. The term η_b accounts for both the imperfect combustion and the thermal dispersion towards the environment. The thermal dispersion from the cyclone separator was included in this term.

The overall power generated by the bottoming cycle was calculated as follows:

$$P_{u,BC} = \eta_{m,E}\eta_{is,E} G_V (h_E - h_{Fis}) - \frac{p_B - p_A}{\eta_{m,P}\eta_{y,P}\rho_l}\ , \tag{8}$$

$\eta_{m,E}$ and $\eta_{is,E}$ denoting the mechanical and isentropic efficiencies of the expander; $\eta_{m,P}$ and $\eta_{y,P}$ are the mechanical and hydraulic efficiencies of the pump. The liquid density is denoted by ρ_l. The heat power exchanged between the flue gases and the fluid of the bottoming cycle (either water or organic fluid) was calculated through the following equation:

$$(G_a + G_b)(h_7 - h_8)\eta_{th,HRSG} = G_V(h_E - h_B)\ , \tag{9}$$

where $\eta_{th,HRSG}$ takes into account the thermal dispersion towards the environment. Equation 9 can be employed to determine the mass flow rate of the vapour, G_V. It should be noted that the previous equations neglect the slight reduction in the mass flow rate of fuel stemming from the ash content; this assumption was made because the ash content is usually very low in flue gases generated by the combustion of pruning residues [22]. The enthalpy of point B must be determined using different procedures in the two cases. When water is employed (Fig.1), h_B can be assumed equal to the enthalpy of the saturated liquid exiting the condenser (h_A).

On the other hand, when the ORC is employed (Fig.2), the value of h_B can be calculated by using the following simplified equations:

$$h_B - h_{A'} = h_F - h_{F'} \,, \tag{10}$$

$$\varepsilon_{reg} = \frac{h_B - h_{A'}}{h_{Bid} - h_{A'}} \,, \tag{11}$$

ε_{reg} denoting the heat exchange efficiency of the regenerator, and h_{Bid} the ideal enthalpy of point B (namely, the liquid enthalpy calculated for $p=p_C$ and $T=T_F$). The minimum temperature difference between gas and vapour, denoted by ΔT_{pp}, can be calculated in both configurations using the following relations:

$$(G_a + G_b)(h_{pp} - h_8) = G_S(h_C - h_B) \tag{12}$$

$$T_{pp} = T_C + \Delta T_{pp} \tag{13}$$

where suffix pp stands for the flue gas conditions at the pinch point. Finally, the global (electrical) efficiency of the combined cycle was calculated as follows:

$$\eta_g = \frac{P_{u,TC} + P_{u,BC}}{G_b H_i} \tag{14}$$

3. Results

The aim of this paper is to investigate the global (electrical) efficiency of the combined cycle for a small-scale application, namely for an overall useful electrical power of less than 50 kW. Considering that the useful electrical power of the bottoming cycle is expected to be a half of that of the topping cycle, the latter was fixed equal to 30 kW. The calculation of the topping cycle was therefore performed for $P_{U,TC}$= 30 kW and by fixing some parameters according to the available technologies, as reported in Table 1. Specifically, the pressure at the outlet of the compressor was set to p_2 = 3.5 bar (which is a value achievable by commercially available turbochargers); the temperature of the flue gas entering the gas-to-gas heat exchanger was set equal to T_6=850 °C. This value represents the maximum temperature of the cycle and was fixed in order to preserve the integrity of the gas-to-heat exchanger, which, in this analysis, is supposed to be a tube-and-shell exchanger made of a high temperature material (Nichel alloy or stainless steel). The efficiency of the gas-to-heat exchanger was set to the value of ε_{HTHE}=0.8, in order to have an optimum compromise between the HTHE dimensions and the cycle efficiency (according to previous discussions [10]). The efficiency of the compressor and the two turbines were considered equal to $\eta_{is,c}$ = 0.75 and $\eta_{is,t1} = \eta_{is,t2}$= 0.8, respectively (which are very common values for commercially available turbochargers). The mechanical efficiencies of the compressor and the turbines were set to the value of 0.98. The thermal losses of the topping cycle were taking into account as follows: $\eta_{th,HTHE}$ = 0.95 and η_b = 0.9 (see equations 5 and 7 for the explanation of these terms). The lower heating value of the fuel was taken equal to H_i = 14000 kJ/kg, which corresponds to the lower heating value of pruning residues with 20% of humidity. With regard to the air temperature at the inlet of the first gas turbine (T_3), it depends on T_6 and ε_{HTHE} (efficiency of the HTHE). Given T_6 and ε_{HTHE}, it results that T_3=722.5° C, which should be a temperature level largely tolerable by turbochargers having their turbine blades made of Nichel alloys.

As far as the bottoming cycles are concerned, two calculations were performed, one for the water cycle (Fig.1) and the other one for the ORC (Fig.2). With regard to the former (see Fig. 1), the commercially available “Green steam turbine”, having an estimated isentropic efficiency equal to 0.5, is proposed to be used as the water steam expander. The temperature at the inlet of the steam turbine, T_E, was maintained constant and equal to 250 °C, which is the maximum temperature that the turbine can withstand according to the manufacturer specifications. The maximum water pressure was set equal to 10 bar, while the condensation pressure was considered equal to P_F=P_A=0.1 bar. These pressure levels are consistent with the maximum and minimum pressures required by the Green steam turbine.

With regard to the ORC (Fig. 2), toluene was selected as the working fluid because its thermodynamic properties are compatible with the temperature of the flue gases exiting the HTHE. The condensation pressure and the maximum pressure level of the vapour were set equal to the values employed for the water cycle, namely 0.1 bar and 10 bar, respectively. The efficiency of the regenerator was set equal to 0.6. The isentropic efficiency of the ORC expander, which is supposed to be a single stage axial turbine, was calculated by using the method proposed by Astolfi and Macchi [23]; according to this method, and using the operating conditions of the proposed plant (size parameter=0.037m and volume ratio=75.7), a maximum isentropic efficiency of approx. 0.53 can be achieved by properly designing the turbine. In the present analysis, a conservative value of $\eta_{is,E}$=0.5 was considered for the ORC turbine.

In both cases (water cycle and ORC), the thermal dispersions towards the environment were taken into account by fixing $\eta_{th,HRSG}$=0.95 (see equation 9); the temperature difference at pinch point was taken equal to ΔT_{pp}=10K. It should be noted that, in both cases (water cycle and ORC), the vaporization pressure was calculated by the code so as to respect the assigned value for ΔT_{pp}. Furthermore, in both cases, the temperature of the exhaust gas exiting the HRSG, T_8, was set equal to 150 °C to avoid the formation of acid rain caused by the sulphur content of biomass, typically 0.1–1%. The efficiencies of both the electric generators and the electric motors moving the pumps were not considered in this analysis.

With these assumptions, the code calculated the mass flow rates (G_a, G_b, G_V) and the unknown enthalpies in the circuit, the useful power of the bottoming cycle ($P_{u,BC}$), the input thermal power ($G_b H_i$) and the overall efficiency (η_g).

Table 1. Fixed parameters for the topping cycle

Fixed parameters	Value
$P_{u,TC}$	30 kW
p_2	3.5 bar
T_6	850 °C
$\eta_{is,c}$	0.75
$\eta_{is,t1} = \eta_{is,t2}$	0.8
$\eta_{m,c} = \eta_{m,t1} = \eta_{m,t2}$	0.98
ε_{HTHE}	0.8
T_3	722.5 °C
η_b	0.9
$\eta_{th,HTHE}$	0.95
H_i	14000 kJ/kg

Table 2. Fixed parameters for the bottoming cycles

Fixed parameters	Rankine cycle	ORC
Fluid	Water	Toluene
T_E	250 °C	-
p_A	0.1 bar	0.1 bar
T_A	45.83 °C	45.35 °C
$\eta_{th,HRSG}$	0.95	0.95
ε_{reg}	-	0.6
T_8	150 °C	150 °C
$\eta_{is,E}$	0.5	0.5
$\eta_{m,E} = \eta_{m,P}$	0.98	0.98
$\eta_{y,P}$	0.8	0.8
ΔT_{pp}	10 °C	10 °C

The calculation was performed by changing the pressure drop in line 5-9 (denoted by Δp_{loss}), so as to investigate the influence of Δp_{loss} on the plant performance. Figure 3 reports the overall efficiency of the combined cycle, as a function of the pressure drop. As expected, the increase in the pressure drop in line 5-9 causes a great decrease in the efficiency, evidencing that the components placed downstream of the power turbine need to be designed as effectively as possible as far as the associated pressure drops are concerned. However, in both cases, the overall efficiency is very high, showing that, even for large pressure drops in line 5-9, the combined cycle has a great potential, in spite of the small-scale application.

Figure 3 shows that, for a given pressure drop, the overall global efficiency of the combined cycle employing toluene is higher than that of the combined cycle employing water. However, the difference is slight, highlighting that, for such a low level of electrical power and by virtue of the employment of the Green steam turbine, the bottoming water cycle is very competitive.

Figure 4 reports the mass flow rates in the circuit, showing that the air mass flow was increased with the pressure drop in order to achieve the target power of 30 kW_e for the topping cycle. Nevertheless, the range calculated for the air mass flow rate results to be consistent with the mass flow rate values of typical turbochargers for trucks. As expected, because of the presence of the regenerator and the absence of the superheater in the ORC, the mass flow rate of toluene is remarkably greater than that of the water.

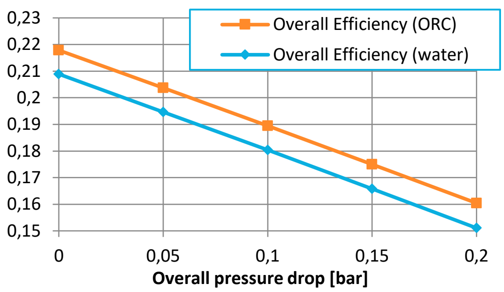

Fig. 3. Overall electr. efficiency (η_g) vs pressure drop (Δp_{loss})

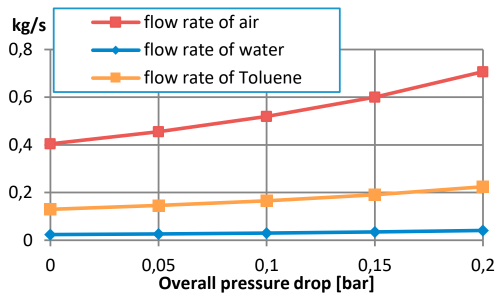

Fig. 4. Flow rates (G_a, G_V) vs pressure drop (Δp_{loss})

It should be noted that, in both cases, the increase in the air flow rate determines a consequential increase in the flow rate of both the water and toluene. As a result, the useful electrical power increases with the pressure drop, as shown in Figure 5 which reports the electrical power generated in the bottoming cycles and the overall input thermal power. The latter increases more than proportionally with the degree of pressure losses, because the overall efficiency has a corresponding decreasing trend, as shown in Fig. 3.

Figure 6 reports additional parameters, useful for the comparison, such as the vaporization pressure in the HRSG and the temperature of the flue gases entering the HRSG. It is noteworthy that, for a given pressure drop, the vaporization pressure of the water Rankine cycle is slightly lower than that of the ORC, both having a decreasing trend with the increasing pressure drop. This is because the vaporization pressure was changed to satisfy the imposed temperature difference at pinch point (10 °C) and the discharge temperature (150 °C). Finally, Fig. 7 provides the comparison between the thermodynamic diagrams of water (Fig. 7a) and toluene (Fig. 7b) for a selected case (Δp_{loss}=0.1 bar).

The difference in terms of efficiency between the bottoming ORC and the bottoming water cycle is not as high as expected, because the temperature of the flue gas exiting the topping cycle is sufficiently high to make the steam cycle competitive, in addition to the fact that the selected operating conditions do not allow a single stage ORC turbine to be designed effectively. However, the difference between the bottoming water cycle and bottoming ORC would increase if a more efficient ORC expander was employed, such as a multi stage axial turbine. In this regard, Table 3 shows that, for a selected case (Δp_{loss}=0.1 bar), the global plant efficiency would be remarkably improved with the adoption of a two stage ORC turbine, which can achieve efficiencies as high as 70% according to the method proposed in [23].

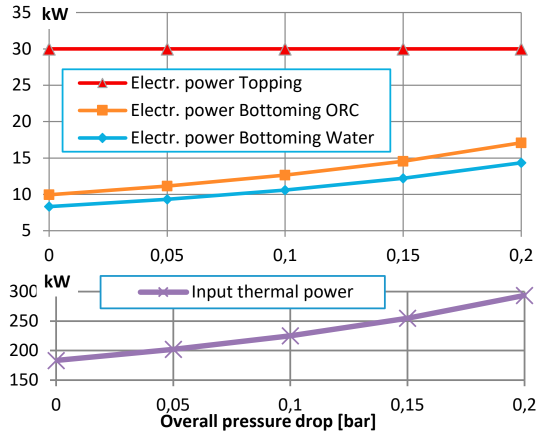

Fig. 5. Electrical power and input thermal power vs pressure drop (Δp_{loss})

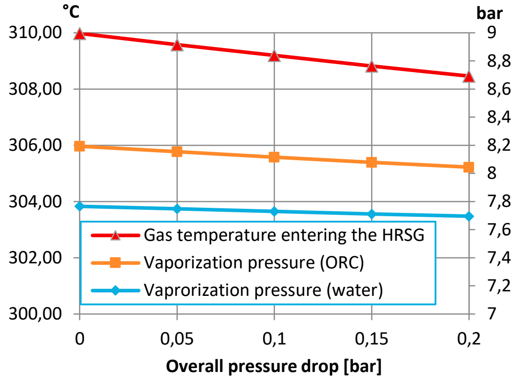

Fig. 6. Temperature of the gas entering the HRSG (T_7) and vaporization pressure (p_C), vs pressure drop (Δp_{loss})

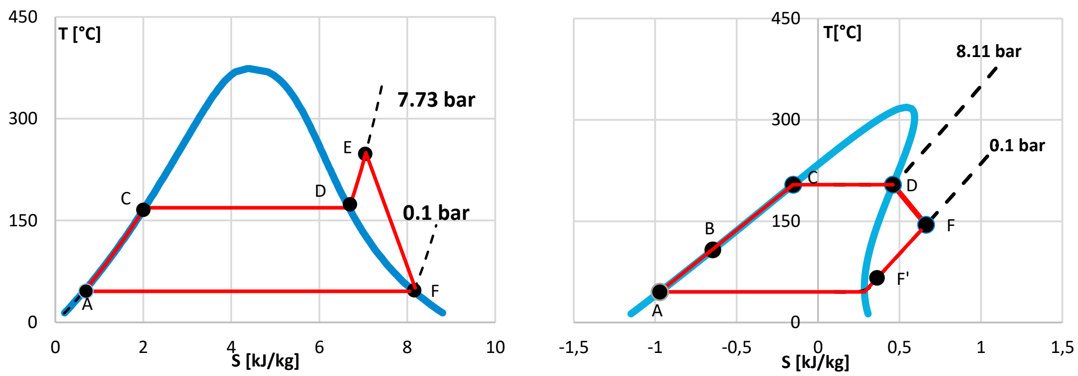

Fig.7. Thermodynamic diagrams (temperature-entropy): water (left) and toluene (right)

Table 3. Expander isentropic efficiency, electrical power and plant efficiency as a function of the type of expander (Δploss=0.1 bar)

Expander	Isentropic efficiency	Electrical power bottoming cycle	Plant efficiency (electrical)
Water steam turbine	50%	10.60 kW	18%
Single stage ORC turbine	50%	12.65 kW	19%
Two stage ORC turbine	70%	17.37 kW	21%

4. Conclusions

The aim of this paper is to investigate a novel, small scale, combined cycle for energy generation from biomass, by exploring its thermodynamic feasibility for very low electrical power generated (less than 50 kW) and in a full electrical power generation mode. The thermodynamic analysis regarded both the topping cycle and the bottoming one. Concerning the former, it was analysed with the hypothesis of being mainly composed of a cheap turbocharger from the automotive industry (instead of a more expensive commercial micro-turbine), an external combustor, a gas-to-gas heat exchanger and a power turbine moving an electric generator. Two configurations were analysed for the bottoming cycle, the first one being a water Rankine cycle (employing the highly efficient “green steam turbine”) and the second one being an Organic Rankine cycle (employing toluene because of the temperature levels).

The results have shown that the combined cycle, in spite of the economic solutions and the very low electrical power, can achieve very competitive levels of global efficiency (up to 22%). The comparison between the water Rankine cycle and the ORC, performed by employing a single stage turbine, has shown that the global efficiency achieved by the latter is slightly greater than that of the former. This results from the fact that toluene better adapts to the temperatures of the combined cycle. The difference was not as high as expected because of the high temperature of the flue gas exiting the topping cycle and because, for the selected operating conditions, the isentropic efficiency of a single stage axial ORC turbine is limited to about 50%. However, it was demonstrated that the use of a two-stage ORC turbine with an isentropic efficiency of 70% can allow a remarkable improvement in the plant performance. Forthcoming studies will investigate the comparison between the bottoming water cycle and the bottoming ORC in the case of a CHP arrangement, in which the use of an ORC is expected to be more profitable compared to the full electrical generation case study.

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