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Solar/biomass hybrid cycles with thermal storage and bottoming ORC: System integration and economic analysis

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Abstract

This paper focuses on the thermodynamic modelling and thermo-economic assessment of a novel arrangement of a combined cycle composed of an externally fired gas turbine (EFGT) and a bottoming organic Rankine cycle (ORC). The main novelty is that the heat of the exhaust gas exiting from the gas turbine is recovered in a thermal energy storage from which heat is extracted to feed a bottoming ORC. The thermal storage can receive heat also from parabolic-trough concentrators (PTCs) with molten salts as heat-transfer fluid (HTF). The presence of the thermal storage between topping and bottoming cycle facilitates a flexible operation of the system, and in particular allows to compensate solar energy input fluctuations, increase capacity factor, increase the dispatchability of the renewable energy generated and potentially operate in load following mode. A thermal energy storage (TES) with two molten salt tanks (one cold and one hot) is chosen since it is able to operate in the temperature range useful to recover heat from the exhaust gas of the EFGT and supply heat to the ORC. The heat of the gas turbine exhaust gas that cannot be recovered in the TES can be delivered to thermal users for cogeneration.
The selected bottoming ORC is a superheated recuperative cycle suitable to recover heat in the temperature range of the TES with good cycle efficiency. On the basis of the results of the thermodynamic simulations, upfront and operational costs assessments and subsidized energy framework (feed-in tariffs for renewable electricity), the global energy conversion efficiency and investment profitability are estimated.

Keywords: *CHP, cogeneration, biomass, gate cycle, concentrating solar, ORC, combined cycle, bottoming cycle*

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1. Introduction

The European Commission is introducing new and ambitious targets for the penetration of renewable energy (27% of internal energy consumption), energy efficiency (reduction of 25% of energy consumption) and the reduction of greenhouse gas emissions (40% relative to 2005 levels) by 2030. These targets can be pursued by distributed heat and power generation, where renewable energy sources integrated with suitable energy storage systems can provide efficiently heat and electric power close to the end users. Concentrating solar power (CSP) and biomass-fired combined heat and power (CHP) plants can contribute towards all of these goals. CSP technologies generate electricity by concentrating the incident solar radiation onto a small area, where a heat transfer fluid (HTF) is heated. This thermal energy is then transferred by the HTF to a power generating system to drive a thermodynamic energy-conversion cycle. The integration of thermal energy storage (TES) can make CSP dispatchable and facilitate the overall energy conversion process. Nevertheless, solar energy is inherently intermittent such that even with TES the capacity factor of solar power plants is limited and often needs to be integrated by fossil boilers. Biomass can be an interesting alternative to fossil fuels to compensate the lack of solar energy: however, the thermal inertia of the furnace makes this technology well suited for base load operation but not for fluctuating operation to meet variable requests of heat and electricity from end users. TES can compensate the input and output energy fluctuations and overcome the individual drawbacks of solar and biomass as primary energy resources and allows such plants to achieve either base load or flexible operation [1][2].

The performance of a variety of system configurations of such hybrid plants under a variable solar input has been investigated in literature [3][4]. Some solar-biomass hybrid configurations are based on parabolic-trough collectors (PTCs), backup boilers and Rankine cycles [5][6], on the substitution of steam bleed regeneration with water preheating by solar energy [7] or on Fresnel collectors [8] to achieve higher temperatures. Some applications consider the use of solar towers or solar dishes and compressed air as HTF [9]. None of the previous research has addressed the integration of parabolic-trough CSP and molten salt TES with biomass combustion in externally fired gas turbines (EFGT). The use of biomass has been widely investigated in the literature as it provides added socio-economic and environmental benefits [10]; in small-to-medium scale CHP plants this includes dual-fuelling of biomass and natural gas in externally/internally fired gas turbines [11][12][13]. The influence of part load efficiencies on optimal EFGT operation was investigated in [14], while the improved energy performance and profitability of employing a bottoming ORC has been investigated in different energy-demand segments [15][16]. The literature on ORC systems and working fluid selection is also extensive[17][18][19]. In particular, a combined cycle with a 1.3 MW biomass EFGT topping cycle and 0.7 MW bottoming ORC plant was proposed in [20].

In the present paper, which goes beyond the work proposed in Ref.[20][21], the heat and power generation system is composed by independent "power blocks", which are the generation sections (gas turbine and ORC), the thermal energy sources (biomass furnace and CSP plant), the TES and the thermal end users. The TES can compensate the solar input fluctuations and needs to be optimized to minimize exergy losses when heat is recovered from the topping cycle and from CSP to be transferred to the bottoming ORC. The technologies adopted for the TES and the ORC to meet these goals are described in the next paragraph.

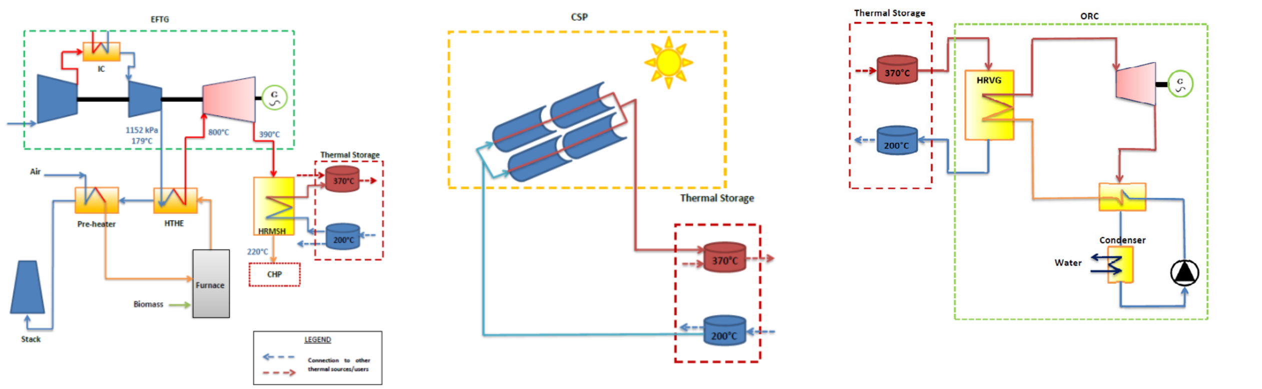
2. Technology description and thermodynamic analysis

The main power blocks that compose the power plant are depicted in Fig 1. A detailed thermodynamic analysis of the EFGT is described in Ref. [20], while a similar EFGT-ORC combined cycle coupled to a CSP section is proposed in [21]. However, in the last configuration, the solar input is used to feed the topping gas turbine in combination to biomass fuel. The overall cycle pressure ratio of the EFGT is 12 and the TIT is 800 °C, which allows a low cost for the heat exchanger material (steel). Combustion air in the biomass furnace is taken from the ambient for a more flexible regulation, since the circuit of the working air flowing into the turbine and the circuit of the combustion air flowing into the furnace are decoupled. The rated LHV input produced by the biomass combustion is 9050 kW, the net electric power output is 1388 kW while the available heat flow at the turbine exit is equal to 4093 kWt at 390°C. Therefore, the temperature of the Hot Tank of the TES has been accommodated to 370°C. The available heat of the air exiting the turbine is firstly recovered in the heat exchanger HRMSH (Heat Recovery

Molten Salts Heater) where molten salts coming from the Cold Tank are heated up to 370°C and conveyed to the Hot Tank. The Cold Tank temperature, as explained in the following, has been assumed of 200°C. Therefore, at rated operating conditions, considering a unitary value of heat capacity ratio between air flow and molten salts flow, and assuming ΔT of 20° at both the hot and the cold ends of the HRMSH, the thermal flow that can be recovered in the HRMSH is 1890 kW from the biomass EFGT. Under such conditions air has still a temperature of 220°C and its sensible heat can be further recovered for cogeneration.

The required area of the solar field is evaluated assuming a standard direct normal irradiance (DNI) of 800 W/m^2. The solar-collector section is based on ENEA technology of PTCs [22] [23][24][25] as from figure 2(b). Although this technology allows for temperature up to ~550 °C, in this work a lower temperature of about 370°C is considered in order to meet the temperature of the Hot Tank of the thermal storage. The CSP consists of a line with six collectors connected in series. This length, about 600 meters, is necessary to allow the Heat Transfer Fluid (HTF) to increase its temperature of 170°C (from 200°C to 370°C) under normal operating conditions of flow and irradiation. The solar collector field is sized to supply 900 kWt that is the 33% of the total rated thermal input to the ORC plant.

The TES section is a two-tank molten-salt system, where the temperature difference between the two tanks is moderately higher than conventional systems that use oil as HTF (170 °C instead of 100 °C) and therefore allows for a lower volume of the two tanks[26]. A mixture of molten salts (lithium, sodium and potassium nitrates) is chosen for both the HTF and the TES medium. These salts freeze at about 120°C and are liquid at temperatures higher than 200 °C [26]. For this reason, a temperature of 200°C is assumed for the cold tank. The molten salts flow in the solar field during normal operation but also at night, recycled from the cold tank. The system's heat losses are generally limited and the fluid will cool only a few degrees. In the event of a lack of heating from the sun, the temperature can be restored using some heaters inside the two tanks.



(a) (b) (c)

Fig 1. Plant layout of the power blocks that compose the plant. (a) EFGT; (b) CSP (c) ORC.

The technical specifications of the solar field under two different scenarios of the TES capacity are reported in Table 1, where the solar multiple represents the ratio of the solar-field thermal energy output to the total thermal energy demand (at design point conditions) from the bottoming ORC cycle. The required ground area is estimated assuming a distance between collector lines of 2.5 times the PTC aperture size In the first considered scenario one collector line was adopted and the amount of energy stored in the TES allows 6 hours of further production (SM 1.96). In the second case, two lines have been adopted and the amount of energy available in the TES allows 18 hours of further production (SM 3.9). The TES capacity is sized to account only for the fluctuations of the thermal input coming from the solar section. The EFGT input heat to the TES feeds directly the ORC, with the assumed baseload operation, hence it does not imply a TES sizing.

In this paper, the Hottel model is adopted for evaluating the average monthly reduction coefficient of the direct normal irradiance DNI, (kWh/m^2 month). The site of Priolo Gargallo (Siracusa, Italy, Latitude 37°08'04", Longitude 15°03'00", 30 m a.s.l., solar collector positioning N-S) has been selected, resulting in a DNI of 2,256 kWh/m^{2}yr and an effective radiance of 1,760 kWh/m^{2}yr. Adopting the methodology proposed in Ref.[27], the useful solar thermal power output is 3,978 and 7,956 MWh/yr for the two assumed CSP sizes (Cases B and C in Table 3, respectively).

Table 1. Design characteristics of the solar field and the thermal storage

Solar field characteristics		
Case study	**B**	**C**
Intercepting area (m^2)	3,228	6,457
Required ground area (m^2)	8,071	16,142
Thermal power output(MW)	1.808	3.616
Solar thermal power available for TES (MW)	0.887	2.6956
Design TES capacity (MWh)	5.178	16.02
Design TES discharge hours	5.48	16.96

The bottoming ORC recovers heat from molten salts flowing from the Hot Tank to the Cold Tank of the TES, with the adoption of a Heat Recovery Vapour Generator (HRVG) (Figure 2c). Since the heat is available at high temperature (from 370 to 200 °C) a recuperative configuration is chosen for the cycle. In particular, the cycle contains a pump (6-1) that supplies the fluid to the recuperator (1-2). The recuperator pre-heats the working fluid using the thermal energy from the turbine outlet. The HRVG produces the evaporation of the organic fluid up to the requested condition of the turbine inlet (2-3), by recovering the heat from the molten salts. Then, the vapour flows in the turbine (3-4) connected to the electric generator. At the exit of the turbine, the organic fluid goes to the hot side of the recuperator (4-5) where it is cooled before entering the condenser. Finally, the condenser closes the cycle (5-6). Considering the operating temperature range of the molten salts, toluene is a suitable working fluid for the ORC cycle since it shows a relatively high critical temperature. Subcritical cycles are firstly examined, considering both saturated and superheated cycles. The T-s diagrams of a saturated cycle and a superheated one, having the same evaporation pressure, are shown in Figures 3(a) and 3(b), respectively. In both figures, the lines representing molten salts flowing in the HRVG and cooling water flowing in the condenser are indicated. The comparison of the two cycles in Figure 2, shows that the saturated cycle is characterized by higher temperature difference between hot and cold side along the HRVG and by lower heat recovered in the recuperator. As a consequence, the heat exchange surfaces of both the HRVG and the recuperator are much lower for the saturated with respect to the superheated cycle. The ORC cycle is sized assuming Toluene working fluid with components efficiencies as reported in [25], condenser temperature of 40°C, ΔT_{min} in the RHE of 25°C and ΔTmin in the HRVG of 20°C. Under such hypotheses, the efficiency of the ORC cycle increases with the evaporation pressure; however, over 10 bar, the increase is low. The cycle efficiency also increases with superheating. Therefore, the plant performance have been evaluated considering an evaporation pressure of 10 bar and a superheated vapor temperature of 350°C. The total amount of the thermal input to the ORC cycle has been estimated assuming that, at rated operating conditions, 70% of the thermal input comes from the EFGT and 30% from the solar field. In particular, it is supposed that the heat flow of 1,890 kWt recovered at rated power by molten salts in the HRMSH is entirely transferred to the organic fluid in the HRVG. The thermal input to the ORC cycle is then integrated by the solar contribution of 900 kWt. Definitely, the total input to the ORC is 2790 kW and the electric power is 800 kWe.

3. Annual Energy Production

The annual energy output is estimated considering the two sizes of the TES (cases B and C) and compared to a plant without solar field (100% biomass fuel) as already examined in [20] (case A). The EFGT is supposed to be operated at baseload for all the time. The ORC plant, instead, is operated at baseload in case A while, in the cases B and C, the ORC is operated at full load when receiving heat from the EFGT and either CSP or TES. Instead the ORC is operated at 70% of full load when the stored thermal energy is over and it receives heat only from the EFGT. Part load efficiency reduction is neglected in this preliminary analysis.

Table 2. Description of the three case studies considered in the present work

Case study	**A**	**B**	**C**
Biomass furnace (kW_t)	9,050	9,050	9,050
Biomass input (t/yr)	25,694	25,694	25,694
Topping EFGT net electric power (kW)	1,388	1,388	1,388
Bottoming ORC net electric power (kW)	700	800	800
Electric efficiency gas turbine	15.3%	15.3%	15.3%
Electric efficiency of the ORC[(1)],	21.5%	29%	29%
Solar share (solar/total energy input yearly basis)	0	6.9%	13.3%
Net electric generation (MWh/yr)	16,786	16,710	17,223
Equivalent operating hours (hr/yr)	8,039	7,568	7,805

(1). Ratio of electric power output and thermal power transmitted in the HRVG.
Case A: 100% biomass input; Cases B and C: CSP with different TES capacity

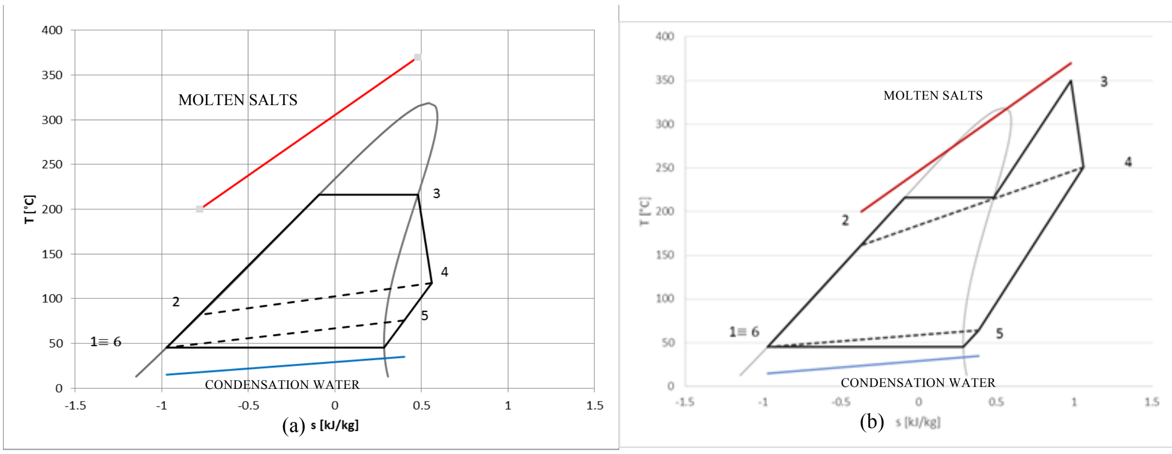


Figure 2. T-s chart for a saturated cycle (a) and a superheated one (b), fed by molten salts flowing from the Hot tank to the cold one.

The rated electric power of the combined cycle in case A (only biomass fuel, Lower Heating Value: LHV=9050 kJ/kg) is 2,088 kWe (with bottoming ORC of 700 kW) while in case B and C (biomass + solar input as from table 3) the combined cycle net power output is 2,188 kWe (bottoming ORC of 800 kW). The electric auxiliary consumption is 6%, and the thermal power output for CHP is of 963 kWt at 104 °C and 2106 kWt at 220 °C respectively for the case A and cases B and C. The modelling results report a net electric efficiency (electricity/input biomass energy at nominal solar energy input) of 23% for the 100% biomass. The energy generated is reported in Table 3. In all cases, the biomass energy input and the power output from the EFGT are the same.

4. Thermo-economic assumptions

A profitability assessment of the hybrid CSP-biomass combined EFGT-ORC CHP plant is proposed in this section. For each case study, a sensitivity analysis to the heat demand intensities and the biomass purchase price are considered. A basic strategy is assumed here of electricity fed into the grid, given that renewable CHP plants are eligible for feed-in tariffs in the Italian energy market.

The financial appraisal of the investment is carried out assuming the following hypotheses: (i) 20 years of operating life and feed-in tariff duration for renewable electricity; no 're-powering' throughout the 20 years; zero decommissioning costs, straight line depreciation of capital costs over 20 years; (ii) maintenance costs, fuel supply costs, electricity and heat selling prices held constant (in real 2017 values); (iii) cost of capital (net of inflation) equal to 5%, corporation tax neglected, no capital investments subsidies. Electricity is sold to the grid at the feed-in electricity price available in the Italian energy market[28], which is 180 and 296 Eur/MWh respectively for biomass electricity and CSP electricity[28]. The electricity generation is calculated at 8,040 operating hours per year. The further revenues from sales of cogenerated heat at high temperature (1890 kWt at 220°C) are here not considered, however they represent a significant increase of revenue in case of on site heat demand availability. The turnkey costs are estimated by means of interviews and data collection from manufacturers of the selected technologies, as described in [20]. For the CSP section, PTCs and TES costs were derived from NREL cost figures[29], according to the lessons learnt from ENEA/Enel Archimede project[30]. In particular, unitary PTC costs of 250 Eur/m^2 and TES costs of 20 kEur/MWh are assumed. The Capex cost are assumed respectively 4,700 – 5,984 and 7,031 kEur for cases A, B and C, with specific investment costs respectively of 2.26, 2.51 and 2.95 kEur/kWe. The annual O&M costs are assumed 3.5% of the turn key cost, biomass cost is 50 Eur/t and the ash discharge are accounted for assuming unitary cost of 70 Eur/t ash. Personnel costs are 268 kEur/yr [20].

5. Thermo-economic analysis results

Figure 3 reports on the energy performance (global electric efficiency and solar share) and Levelized Cost of Energy (LCE) at different biomass supply costs (in the case of electricity-only production) for the proposed case studies. The global electric efficiency is evaluated as the ratio between the annual electric energy production and the annual LHV energy input from biomass combustion. A comparison with the hybrid solar/biomass system configuration proposed in [21] where the solar input from the same typology of PTCs and TES is provided to the topping gas turbine at 550 °C reducing the biomass consumption, is also shown in Figure 3. The global electricity efficiency is the ratio of electricity annual sales and biomass energy input, while the solar share is the percentage of solar energy input on a yearly basis. The Net Present Value (NPV) and Internal Rate of Return (IRR) as a function of the biomass supply cost, for electricity-only scenario, are reported in Figure 4.

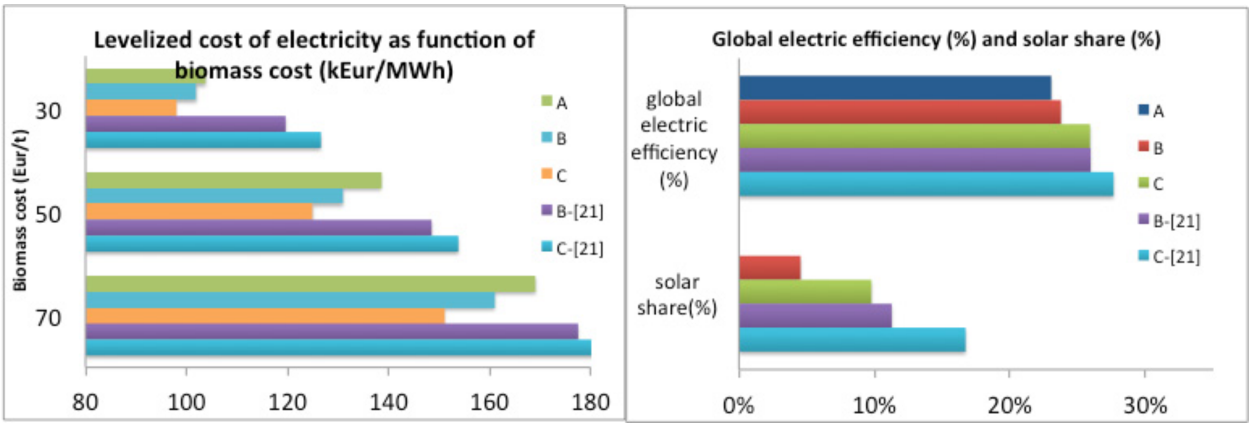


Figure 3. LCE as a function of the biomass purchase price (left) and energy balances as resulting from thermodynamic modelling (right) for Cases A, B and C and Cases B and C of ref [21]

The proposed hybridization of the biomass EFGT with CSP (Case B and C) presents comparable global electric efficiency (in comparison to only biomass - case A), while the LCE increases. In fact, the solar input increases the electricity generated via the bottoming ORC at fixed biomass supply cost but also increases the investment costs. Moreover, the trade-off between higher revenues from solar-based electricity and increased investment costs for the CSP and TES sections increases the NPV and IRR when the solar hybridization is considered, and this is more evident at low biomass supply costs. Moreover, increasing the size of CSP and TES (from case B to C) is beneficial for global energy efficiency balances, as expected, but also for investment profitability, due to the relatively low cost of the molten salt storage (in the proposed temperature range), in comparison to the increased revenues from solar electricity feed-in tariffs. These considerations are different from what reported in [21], where a different solar-biomass hybridization system was proposed. In that case, despite the higher solar share and electric efficiency, the LCE results higher and NPV, IRR are lower than in this configuration.

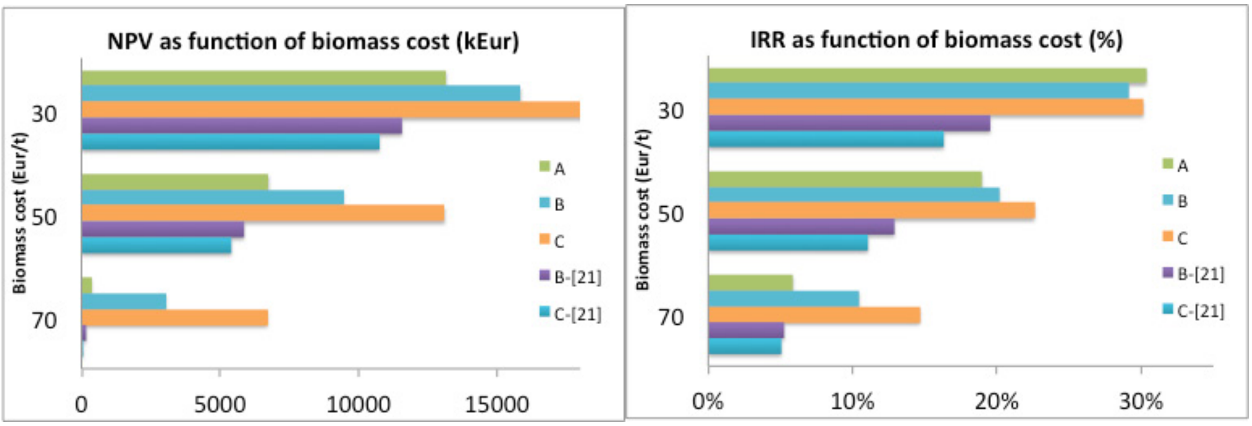


Figure 4. NPV (left) and IRR (right) for Cases A to C and Cases B and C of ref [21], as a function of biomass supply cost for electricity- only scenario

6. Conclusions

A thermodynamic and economic analysis has been performed on a hybrid (solar-biomass) combined cycle composed of an externally fired gas-turbine (EFGT) and a bottoming organic Rankine cycle (ORC) integrated by a linear parabolic trough collector field with molten salts as the heat transfer fluid. In order to improve the flexibility of the plant, a thermal storage recovers excess heat from the gas turbine and the solar field and transfers it to the ORC cycle and thermal end users, when requested. The thermal input of the gas turbine is about 9 MW, with a power output of 1.3 MW, while the bottoming organic Rankine cycle has electric output of 700 or 800 kW with or without the solar hybridization configuration. The thermodynamic modelling has been performed assuming two CSP sizes, and the energy performance results report higher global conversion efficiencies when using CSP integration and the thermo-economic analysis reports a higher NPV of the investment when integrating solar energy, due to the increased electric generation and higher value of solar-based electricity.. A comparison with a previously proposed solar/biomass hybridization with higher temperature (550°C) of CSP working fluid and direct solar energy input to the topping gas turbine demonstrates the higher profitability of this system configuration. Another advantage of this configuration, not been highlighted in this economic analysis, is the availability of high grade heat for cogeneration from the bottoming ORC, that could make the difference when a proper heat demand is available.

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