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Perspectives for the use of biomass as fuel in combined cycle power plants

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Abstract

The paper analyzes the perspectives of biomass and biomass derived fuels utilization for energetic use. After a brief review about the current technologies for biomass conversion to energy and biomass based power plants, an exergy loss based economic analysis of biomass utilization is proposed. This analysis shows the opportunity of using biomass in plants with a thermodynamic efficiency higher than a minimum value. Thus the attention is focused on the use of thermal energy from biomass as integrative source together with natural gas in combined cycle power plants, considering methods for upgrading biomass energy conversion to power. The paper provides a thermodynamic analysis of combined plants using biomass to obtain exhaust gas aftertreatment with atmospheric postcombustion (reheat). Two different technical solutions are proposed. A general optimization of the two solutions shows the possibility of obtaining plant efficiency up to 60% in perspective and of 57% by using currently available gas turbine models.

Keywords: Biomass; Biomass conversion; Power generation; Combined cycle; Postcombustion; Thermodynamic optimization; Exergy analysis

1. Introduction

Energy policy is promoting many researches both for the enhancement of utilization of renewable energy and low enthalpy fuels for power generation and for finding the most effective ways of using them. The use of renewable energy technologies would reduce the current global environmental problems as well as the national energy insecurity of a lot of countries related to the use of fossil fuels.

Among the low enthalpy fuels for power generation, biomass (like paper, cardboard, agricultural and forestal residues, straw, wood wastes, sawdust, manufacturing scraps and cotton stalk), biomass derived fuel and the non-recyclable part of municipal solid waste (MSW) are the most popular currently. Biomass is generally a "fuel" with heating value (HV) in the range 8–25 $MJ\cdot kg^{-1}$, value quite low with respect to the 25–30 $MJ\cdot kg^{-1}$ of coal, 40–45 $MJ\cdot kg^{-1}$ of oil and 50–55 $MJ\cdot kg^{-1}$ of natural gas. Like other renewable energy sources, biomass is gaining increasing acceptance worldwide, making possible the perspective of reducing both fossil fuel depletion and greenhouse gas (mainly CO_2) and NO_X emissions due to fossil fuel utilization.

The conversion of biomass and wastes into energy encompasses a wide range of different conversion options, end-use applications and infrastructure requirements [1]. Although the environmental and other benefits of using biomass to displace fossil fuels are well known, it seems that they cannot compete effectively in the current market without tax credits, subsidies and other artificial measures.

A drawback to the diffusion of biomass use for thermoelectric generation is the fact that biomass is actually converted into energy in plants with a low thermal efficiency. Overall efficiencies to power tend to be rather low at typically 15% for small plants up to 30% for larger plants. These reference values are very far from the typical efficiencies of the most efficient energy conversion plant: the natural gas combined cycle plant [2]. Moreover biomass combustion releases various different chemical pollutants so that the environmental effects of burning biomass are generally considered less harmful than those associated with coal, but

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Nomenclature

c_p	specific heat at constant pressure	$J \cdot kg^{-1} \cdot K^{-1}$
E	exergy flow rate	W
fg	gain function	€
I	exergy losses	W
I^*	dimensionless exergy losses	
k	specific cost	$€ \cdot MJ^{-1}$
K_{plant}	fixed cost of the plant, including components installation and maintainance	€
m	mass flow in the steam cycle	$kg \cdot s^{-1}$
M	mass flow in the gas cycle	$kg \cdot s^{-1}$
p_{el}	selling price of electricity	$€ \cdot MJ^{-1}$
P	pressure	bar
Q_H	thermal energy input at high grade	W
Q_L	thermal energy output to the cold reservoir	W
Q^*	input thermal energy in the bottoming cycle	W
s	specific entropy	$J \cdot kg^{-1} \cdot K^{-1}$
S [ ]	vector of the parameters	
T	temperature	K
T_a	environmental temperature	K
T_{bf}	biomass flame temperature	K
w	specific power	$MJ \cdot (kg\ air)^{-1}$
W	power	W
x	steam quality	
X [ ]	vector of the input variables	
Y [ ]	vector of the output variables	

Greek symbols

ε	air pre-heater effectiveness	
λ	pressure ratio	
η	efficiency	
μ	ratio between thermal power from gas and total thermal power	
η_{conv}	profitable conversion efficiency	
η_{is}	isentropic efficiency	
τ	temporal basis	s
ΔP	pressure drop	bar

Subscripts, acronyms and abbreviations

1	of the first step
2	of the second step
b	of the biomass (as fuel)
BC	of the bottoming cycle
Carnot	of the Carnot cycle
C	compressor
c.c.	combustion chamber
cond	condenser
ECO	economizer
EVA	evaporator
g	of the exhaust gas
gas	of the natural gas (as fuel)
GT	(of the) gas turbine
HRSG	heat recovery steam generator
HP	high pressure
I	of the exergy losses
l	of the liquid streams
LP	low pressure
in	of the inputs
o, out	of the output, at the outlet
RIG	of the air pre-heater
ST	(of the) steam turbine
TC	of the topping cycle
TIT	Turbine Inlet Temperature
tot	of the total system

more harmful than those associated with natural gas [3]. Despite these drawbacks, from an energetic point of view, the use of biomass presents a lot of favorable aspects.

In recent years various solutions for using biomass in combined cycle plant configurations have been proposed. Concepts such as combined heat and power (CHP) and gas turbine combined cycle (GTCC) schemes, where thermal power from biomass or waste fuels is used, can be easily conceived.

Despite the large interest, those technical solutions cannot yet be considered as assessed technologies and they require further investigations concerning thermodynamic analysis, design and optimization of the components and economic analysis [4,5].

The aim of the present paper is to define a new approach in order to consider the use of biomass and derived fuels as thermal sources and different methods for their efficient energetic use, focusing on their use in thermoelectric plants.

After a classification of biomass and a review of their main conversion technologies, the paper analyzes the perspectives of using biomass as fuel for thermoelectric generation proposing a new point of view. The integration of economic and thermodynamic analysis shows the convenience of using biomass in quite efficient plants.

A feasible solution, proposed and examined in the present paper, is the use of the thermal energy from biomass as integrative source in combined cycle plants. In particular biomass can be used for atmospheric postcombustion of the exhaust gas at the outlet of the gas turbine. As will be shown later, this solution enables to get a reduction of the fossil fuel consumption (natural gas), a reduction of CO_2 emission and an increase of the combined plant efficiency by the optimization of the heat recovery steam generator (HRSG) and of the bottoming cycle. The technological feasibility and the thermodynamic performances are examined with reference both to repowering and upgrading of existing GT plants and to optimized plant configurations.

2. Biomass and biomass conversion technology

Biomass fuels can be generally divided into three primary classes:

(1) wood and woody materials;
(2) herbaceous and other annual growth materials such as straws, grasses, leaves, herbaceous energy crops, short rotation woody crops;
(3) agricultural and forestal by-products and residues including shells, hulls, pits.

The majority of biomass sources are represented by wood and wood wastes and by agricultural and forestal wastes: in some countries forest residues represent more than 70% of the potential of dry biomass [6]. The organic material derived from animals is also referred to as biomass. A fourth category of biomass is represented by refuse-derived fuels (RDF) from municipal solid waste (MSW) and non-recyclable papers. The class of MSW is often excluded from the category of biomass, but the origin, with the exception of mixed plastics, is appropriate for inclusion as a biomass type [1,7]. Some papers like [8] discuss the properties of biomass, important for the design and development of combustion and other types of energy conversion systems. The main selection criteria for biomass species are: growth rate, ease of management, harvesting and other intrinsic properties, such as moisture/ash/alkali content. About the intrinsical properties, it is important to underline that biomass is mainly made up of carbon. That makes it looked at with great attention among the alternative fuels: it can be either directly used as fuel in order to produce energy, or be turned into liquid or gaseous fuels. Different types of biomass with their heating values, are summarized in Table 1.

Biomass is a complex resource that can be processed in many ways leading to a variety of products [9]. Three processes are mainly used for their thermo-chemical conversion: *combustion*, *gasification* and *pyrolysis* [4,10]. The first two are the most efficient, requiring a less costly drying process and their products are more easily usable.

Most electricity generation from biomass is based on the Rankine (steam turbine) cycle. In this case biomass can be directly burnt in small-scale boilers for heating purposes, or in larger boilers for the generation of electricity or combined heat and power (CHP).

Gasification converts biomass to a low or medium calorific value gaseous fuel which can be used to generate heat and electricity by direct firing in engines, turbines and boilers after a suitable clean up.

Biomass pyrolysis produces a liquid fuel which can be transported and stored, and allows for de-coupling of the fuel production and of energy generation stages. This fuel can be used to generate heat and electricity by combustion in boilers, engines and turbines. Different products than liquid fuels can be obtained from pyrolysis, such as charcoal and biogas [11].

3. Use of biomass for power generation

Three routes are mainly available for power generation from biomass. These are:

- Biomass combustion-turbine cycle;
- Gas turbine based solutions;
- Hybrid solutions.

Today's capacity of converting energy from biomass is often based on mature, direct-combustion boiler/steam turbine technology. This is not a good solution especially in view of the disadvantage of needing large-scale collection and transportation over large areas.

The average size of existing biopower plants is 10–20 MW (the largest approach 75 MW). Many older wood-burning steam power plants produce steam at temperatures below 400 °C. Those plants operate at an electric efficiency of 10–15% when backpressure turbines are used to provide both power and process steam, or of 15–20% with condensing turbines (no process steam supply). In more modern plants, especially those of size over 50 MW, steam temperatures up to 480 °C have been used and electric efficiency around 25% can be reached [4,5,10].

Gas turbine-based solutions appear to be more interesting in terms of efficiency, size, cost, and fuel flexibility. Two radically different options can be found in the literature [3], but real applications are not yet developed.

Table 1
Higher Heating Value (HHV) and chemical analysis of different biomass fuels

Biomass and composition	Rice hulls	Rice straw	Sugar cane bagasse	Switchgrass	Wood wheat straw	Paper	MSW	Lignite
C [%]	38.8	38.2	48.6	46.7	47.5–52.5	48	39.7	61
H [%]	4.7	5.2	5.9	5.8	4.2–5.9	6.6	5.8	4
O_2 [%]	35.5	36.3	42.8	37.4	37.9–41.5	36.9	27.25	18.5
N_2 [%]	0.5	0.9	0.16	0.8	0.27–0.65	0.14	0.80	1
S [%]	0.05	0.2	0.04	0.2	0.03–0.12	0.07	0.35	1.8
Cl [%]	0.12	0.6	0.03	0.2	0.01–0.13			0.05
ash [%]	20.3	18.7	2.44	8.9	2.5–8.2	8.3	26.1	13.7
HHV [$MJ \cdot kg^{-1}$]	15.84	15.09	18.99	18.06	15.9–20.5	20.78	15.54	23.35

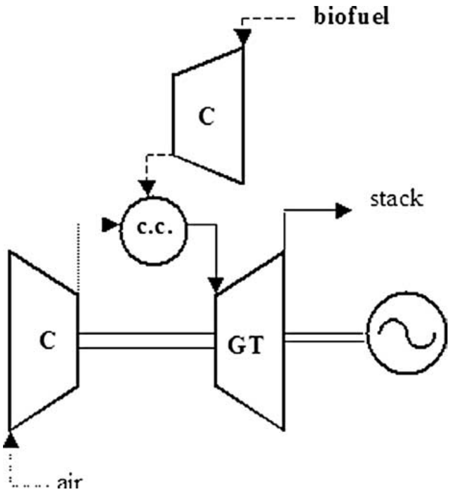

Fig. 1. Schematic of the BIG/GT plant.

The first option, shown in Fig. 1, is based on a gasifier that produces biofuel to be used in the combustion chamber of a gas turbine (BIG/GT), or in a combined cycle plant (BIG/CC). Integrated gasification and combined cycle gas turbine plants (BIG/CC) offer high efficiencies at a relatively small scale (30–50 MW); they are expected to achieve efficiencies of about 40% for wood based biomass and atmospheric gasification. But BIG/CC is currently at the pre-commercial demonstration stage. Pilot projects are operating or being developed in the UK, Sweden, the US and Brazil [12]. Among the demonstration units that are under consideration, the largest one (for 32 MW electrical) is likely to be built in Bahia, Brazil [13].

In the second option compressed air is heated indirectly in a heat exchanger and the biomass or the derived biofuel are directly burnt in an external combustor. Fig. 2 provides a scheme of such a plant known as externally fired gas turbine combined cycle (EFGT/CC): the hot gases pass at first through a heat exchanger, where the compressed air is heated to the Turbine Inlet Temperature (TIT), and then through a HRSG, where superheated steam is produced. The topping cycle is basically a conventional gas turbine cycle and the combustion chamber, is replaced by a heat exchanger. Although there are some experiences in operating ceramic heat exchangers at temperatures as high as 1500 K, in the usual cases the TIT is constrained at 1050–1100 K, if stainless steel heat exchangers are used, or in the range of 1100–1300 K by using advanced metallic alloys heat exchangers. Resorting to conventional stainless steel heat exchangers, the gas turbine cycle efficiency cannot be higher than 30% and EFGT/CC plant, with difficulties could reach an efficiency higher than 40%.

Different solutions proposed in the literature suggest a combination of a gas turbine plant with a biomass burner or a waste incinerator in repowering configurations. In this case, the steam from the incineration plant is passed to the HRSG, where it is superheated and then directed to the steam turbine [14].

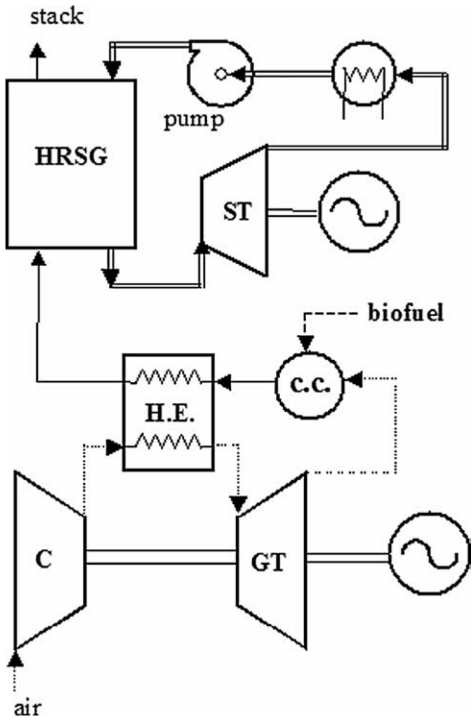

Fig. 2. Schematic of the EFGT/CC plant.

Table 2
Performance of typical plant configurations using biomass as fuel

Plant	Efficiency	Power	Features
Steam power plants (backpressure turbines)	10–20%	20–25 MW	wood-burning
Steam power plants (condensing turbines)	22–28%	5–50 MW	
Externally-fired gas turbines	25–30%	5–25 MW	simple cycle
	→ 40%	10–30 MW	combined cycle
Biomass integrated gasification	21–25%	40–60 MW	simple cycle
Combined cycle	35–40%	90–100 MW	combined cycle

Table 2 summarizes the main categories of biomass plants providing reference values for size and efficiency. As clearly appears, the power of the plants with thermal input from biomass are rated in the range 10–100 MW. Some papers like [15] discuss also the possibility of constructing power plants, using biomass as fuel, of size higher than 100 MW. But this possibility is limited to country with great agricultural and forestal extension.

Really the problem of the energy sources availability is an important prerequisite for the design of each kind of energy conversion plant. This problem becomes really important in case of thermoelectric conversion from biomass due to the low energy density (approximately 1 MWel for 1000 ha of surface) and scattered dispersion. Thus an accurate preliminary analysis of the territorial potential energy of the products coming from higher-value activities in agricultural district, agroindustrial or wood-farming area, is necessary in order to protect from uncertainty in the operating life of the plant. So for energy conversion of biomass, the definition of the size of the plant seems to represent the real preliminary key element and for a lot of countries the technological increase of biomass conversion seems to be very difficult.

4. An exergy loss based economic analysis of biomass fuelled power plants

There are often social, political and economic constraints to biomass, that do not encourage its use in substitution of fossil fuels. In several studies, as [10], attempts were made to measure the economic benefits of biomass facilities and in particular their effect on the job market. The monetary evaluation of the job is surely the main problem, since it is highly dependent on the economic conditions of the site, the political environment, the social value of jobs and the approach followed. So energy prices do not generally reflect the environmental benefits of biomass like of the other renewable sources.

Biomass is a fuel abundant in many countries, especially in developing ones, where it can be considered a relatively cheap fuel. So, on an economical point of view, it could appear convenient to convert biomass or biomass derived fuels in old and inefficient steam cycles as usually done in practice. But it is not always possible to consider biomass as a low cost fuel, even when it is available as a waste or by-product of a higher-value activity because it is not in ideal form for fuel use and for energy conversion, the start up of a sequence of operations with their own cost is necessary. These consecutive operations can be summarized in: transport, storage, handling, drying process, pre-treatment. At the end of all treatments the cost of biomass and of the derived fuel is raised so that, especially in industrial countries, it often becomes comparable with the cost of fossil fuels, even if the users do not generally pay directly a great part of the above-mentioned cost. Therefore, in order to evaluate more correctly the use of biomass or biomass derived fuels, it is necessary to define and consider all the costs. Due to the low energy density and scattered dispersion, the cost of collection and transportation is usually the main part of the biomass price. The energetic conversion of biomass is generally competitive if the necessity of collection, handling and transportation is reduced. This condition is verified when a large amount of biomass is available at a short distance from the plant (e.g., rice mill in China, sugar cane bagasse in Brazil, soybean-based crop in India or forestal residues in Canada).

Actually, in the European market, if the charges are related to distance below 50 km, the cost of biomass range from 0.5 to 4 $€\cdot\mathrm{GJ}^{-1}$ compared with about 5 $€\cdot\mathrm{GJ}^{-1}$ of natural gas. In the meantime, to make biomass a competitive source and to reassess the problem of environmental impact it is necessary to improve its conversion efficiency.

As for the fossil fuels, the thermal conversion of biomass implies a loss. So the available chemical energy of biomass is degraded to thermal energy at low temperature and this degradation causes an environmental damage too. The most popular currently economic neoclassic theory does not take into account the value of this degradation. A way to integrate the 2nd law of Thermodynamics in the economic analysis of an energy conversion system is to consider not only the cost of input "fuel" and of the components, but also to take into account the economic value of the energy degradation, i.e., of the exergy losses. This particular method of analysis, already tested by the authors in the design of the HRSG and gas fired combined cycle plants [16], can be extended also to plants that convert energy from biomass and renewable sources. The convenience of a conversion plant, that uses biomass as input fuel to obtain a well defined output power W_o, corresponds to satisfy the condition:

$$fg = \left[p_{\mathrm{el}} W_o - \left(k_I I + k_b \frac{W_o}{\eta}\right)\right]\tau - \sum K_{\mathrm{plant}} > 0 \tag{1}$$

where p_{el} is the selling price of electric energy and $\sum K_{\mathrm{plant}}$ is the sum of all the costs related to the plant including installation, operation and maintenance. The terms inside the round brackets are the key elements for the application of the method and require to be correctly defined, mostly for what concern the cost of exergy loss, k_I, and the cost of biomass, k_b. This last is correctly defined if the whole biomass life cycle, before its use in the plant, is considered. The exergy loss of the plant, I, can be defined with reference to the ratio between the efficiency of the conversion plant, η, and the ideal Carnot efficiency, η_{Carnot}, of a cycle operating between the environment, at temperature T_a and a maximum temperature equal to the biomass flame temperature, T_{bf}, considered as an exergy source at high temperature:

$$I = \left(\frac{\eta_{\mathrm{Carnot}}}{\eta} - 1\right) W_o \tag{2}$$

where η_{Carnot} represents the Carnot efficiency, defined as

$$\eta_{\mathrm{Carnot}} = 1 - \frac{T_a}{T_{\mathrm{bf}}} \tag{3}$$

The higher the flame temperature, the higher the theoretical maximum efficiency. So the maximum efficiency is dependent on the flame temperature, which is related to the heating value and to the composition of biomass as well as to the air excess used for the combustion [17]. The gain function defined in Eq. (1) is positive if:

$$\left\{p_{\mathrm{el}} W_o - \left[k_I \left(\frac{\eta_{\mathrm{Carnot}}}{\eta} - 1\right) W_o + k_b \frac{W_o}{\eta}\right]\right\}\tau > \sum K_{\mathrm{plant}} \tag{4}$$

that can be also given in dimensionless form as:

$$1 - \left(\frac{\eta_{\mathrm{Carnot}}}{\eta} - 1\right)\frac{k_I}{p_{\mathrm{el}}} - \frac{1}{\eta}\frac{k_b}{p_{\mathrm{el}}} > \frac{\sum K_{\mathrm{plant}}}{p_{\mathrm{el}} W_o \tau} \tag{5}$$

A reasonable choice for the cost of exergy losses is to assume a value ranging between the real cost of biomass and the cost of a conventional fossil fuel, like natural gas or coal, that is replaced by biomass or biomass derived fuel in the plant. From this point of view the use of biomass can be

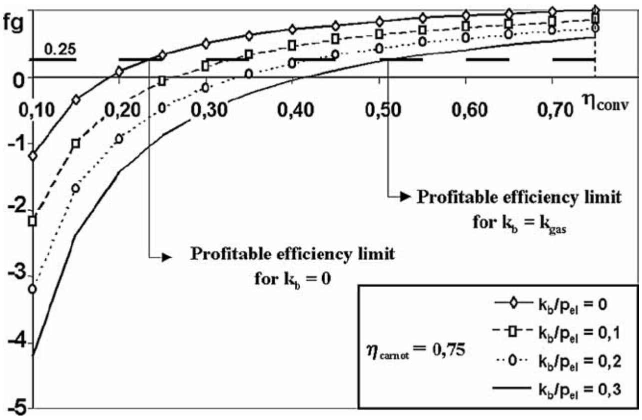

Fig. 3. Gain function as a function of the plant efficiency for different values of the ratio k_b/p_{el} ($k_I/p_{\mathrm{el}} = 0.3$).

considered as the saving of fossil fuels. The second term of Eq. (5) is the ratio between the time base cost of the plant (e.g., annual) and the revenue from the selling of energy. It depends on the cost of the components of the plant, including maintenance costs and working capital and on the economic life of the plant.

The analysis of Eq. (5) shows that a minimum value of the plant efficiency (η_{conv}) is necessary to satisfy the condition defined by Eq. (1), even if the biofuel is available at null cost ($k_b = 0$). If, for example, the ratio at the second member is equal to 0.25, meaning that in four years the cost of the plant is equal to the amount obtained by energy selling, the plant efficiency must be higher than 20% (Fig. 3). This limit rises up to 40%, when biofuel cost approaches the cost of natural gas, becoming about the 30% of the selling price of energy. Accepting this point of view, the previous analysis leads to reject a lot of "low efficient" plants, used today in the world, at the end of their economic life, to convert energy from biomass.

An interesting way to use biomass for energy conversion, seems to be the biomass integrated gasification combined cycle (BIG/CC), operating with syngas obtained from gasification of biomass in pressurized bed gasifier systems. Due to the intrinsic low temperature achieved after the combustion process, this kind of plants can approach an efficiency limit of the order of 45%, a little bit lower than in the IGCC plants using coal but they require high installation costs.

Other interesting developments concern the co-firing of coal and biomass. This option is encouraged by the similar chemical characteristic of the two fuels, but it is unlikely to achieve an efficiency level of 40%.

From the analysis previously exposed, it appears clear that a good option to convert energy from biomass to electric energy is to reconsider the use of biomass in combined plant configurations, also conceiving new more effective solution.

With the aim of increasing thermal efficiency of biomass power conversion up to 45%, resorting to currently available technology, the solution that appears more interesting concerns the joined use of natural gas and biomass or biomass derived fuel in combined cycle plants. Considering the point of view exposed by Eqs. (1)–(5), a possible development of the method, for the analysis of thermal plant where two different fuels (e.g., natural gas and biomass) are used together, is to perform the maximization of a gain function, defined taking into account both gas and biomass thermal power and exergy loss.

5. Use of biomass as integrative fuel in combined cycle plants for exhaust gas aftertreatment

As previously discussed, the solutions proposed in the literature are generally characterized by the use of biomass or biomass derived fuel instead of a fossil fuel. Due to different motivations, using biomass as fuel is not possible to reach high TIT and plant efficiency. Moreover biomass and derived fuels show some problems concerning reliability regarding fuel flexibility and availability. For these reasons it seems more convenient to propose the combined use of a fossil fuel and energy from biomass.

As previously discussed, a profitable way to use biomass for energy conversion is to introduce the derived thermal energy in a gas fired combined cycle plant.

An option considered in the literature for the development of this idea is the co-combustion (co-firing) of natural gas and fuel (syngas) derived from biomass gasification [18]. Syngas derived from biomass gasification is mixed with natural gas to be burnt in the combustion chamber.

This solution has interesting perspectives from a thermodynamic viewpoint because thermal power from biomass is introduced in the topping cycle at the higher thermal level, but its applications requires modification of the gas turbine, that has to be adapted to receive 4–5 times the volume flow of gas [19].

A different option proposed and discussed with major details in the present paper, is represented by the use of biomass as thermal source for atmospheric postcombustion after the discharge from GT. The idea is that, contrary to what happen for co-firing solutions, gas and biomass are used in sequence according to their quality: natural gas in the topping cycle and biomass or derived fuel in the bottoming one.

Two different solutions can be proposed:

(1) post-combustion with an additional firing of biomass before the HRSG or biomass cofiring in the HRSG: this solution corresponds to a Biomass Integrated Postcombustion (reheat) Combined Cycle (BIPCC) plant and is represented in Figs. 4 and 5;
(2) combustion of biomass or syngas derived from biomass to preheat air before entering the gas turbine: this solution represents a Biomass Integrated Fired Recuperated Combined Cycle (BIFRCC) plant and is schematically described in Figs. 6 and 7.

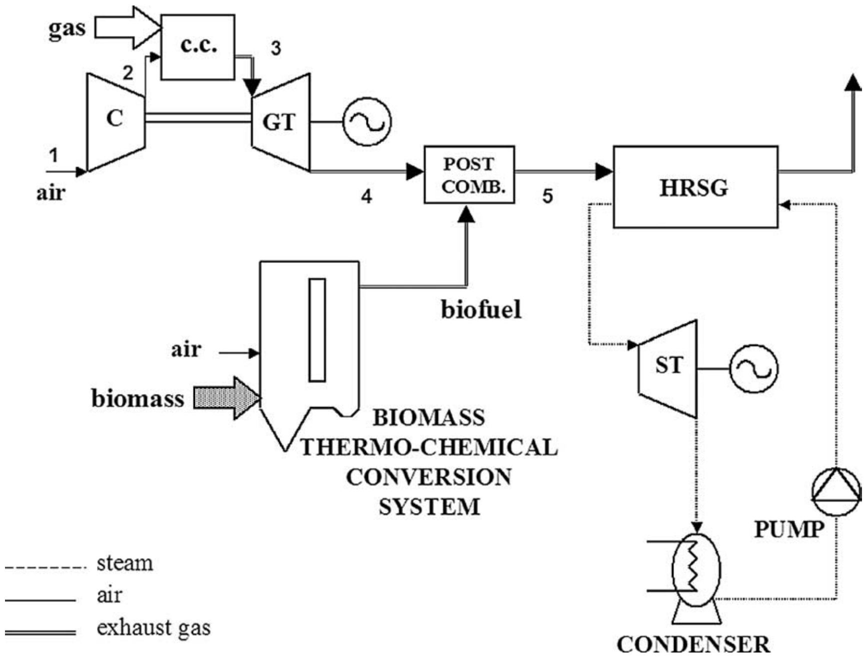

Fig. 4. Schematic of the BIPCC plant configuration.

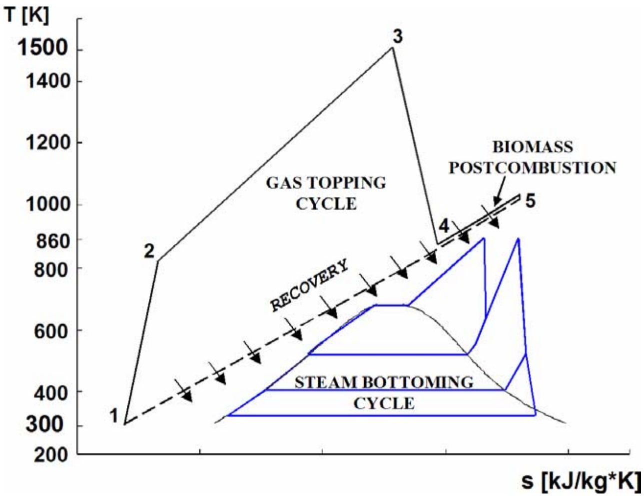

Fig. 5. T–s diagram of the BIPCC plant.

The increase of the inlet temperature of the gas entering the HRSG, by mixing biomass and exhaust from the GT, enables to obtain, as will be shown later, a higher amount of power from the steam cycle, resorting to the HRSG optimisation and to an increase of the steam cycle efficiency.

The solution BIPCC is apparently the less efficient one, since energy from biomass is used at low thermal level. But it appears interesting mainly for repowering of gas turbine based plants where a GT model with high efficiency (40%) and low discharge temperature (440–480 °C) is used.

In the plant configuration BIFRCC, described in Figs. 6 and 7, the combustion products, coming out from biomass firing unit, determines an increase of the GT exhaust gas temperature that can be extended towards the value of 1100 K. This temperature is sufficiently high to preheat air flow entering the GT combustion chamber, in a conventional gas-to-gas heat exchanger, before going through the HRSG. This solution allows the most efficient use of biomass giving an increase of the gas cycle performances and in the meantime an optimization of the HRSG and of the steam bottoming cycle.

5.1. Analysis of the proposed solutions

In a previous study the possibility of drawing up power plants with high thermodynamic efficiency has been shown, joining HRSG optimization [16], air pre-heating, reheat and reconsidering the selection of the gas turbine operating parameters. The gross efficiency could be increased, with respect to a basic combined cycle configuration, from 2%, by resorting to HRSG optimization only, up to 7%, joining reheat, air pre-heating and HRSG optimization [20].

The solutions proposed in this paper are derived from those ideas. In case of biofuel integration, it seems more rational to use different grade fuels in sequence, according to their thermal grade. For example in the solution BIFRCC, the low-grade fuel (biofuel) operates to pre-heat air at a temperature compatible with the maximum combustion temperature, while high-grade fuel (natural gas) can furnish power in the combustion chamber.

Moreover an additional reason support the idea of the convenience of introducing biomass reheat. It is generally accepted the idea that it is not convenient, on a thermodynamic point of view, to introduce thermal energy at low temperature grade.

So that if an analysis of a post-fired combined cycle power plant is carried out according to the scheme described in Fig. 8, through simple energy balances we reach the fol-

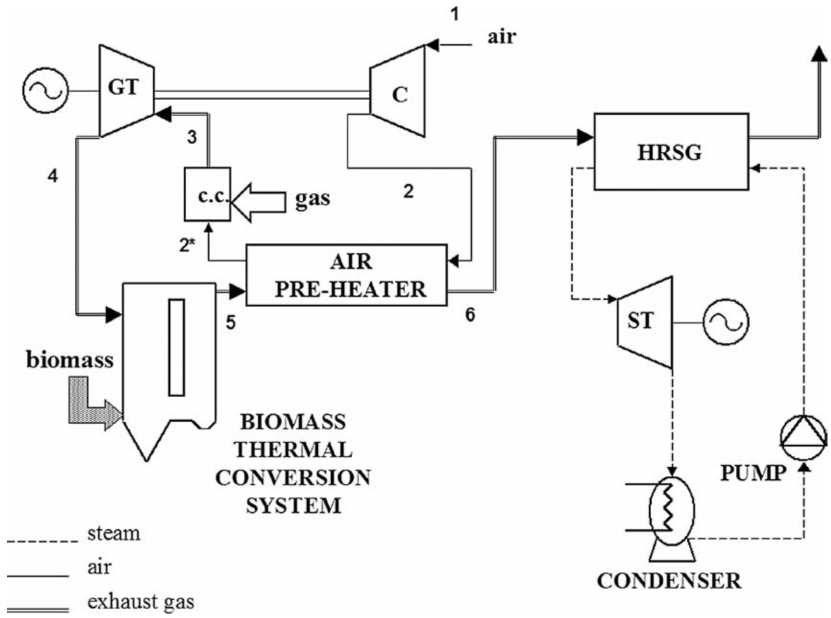

Fig. 6. Schematic of the BIFRCC plant configuration.

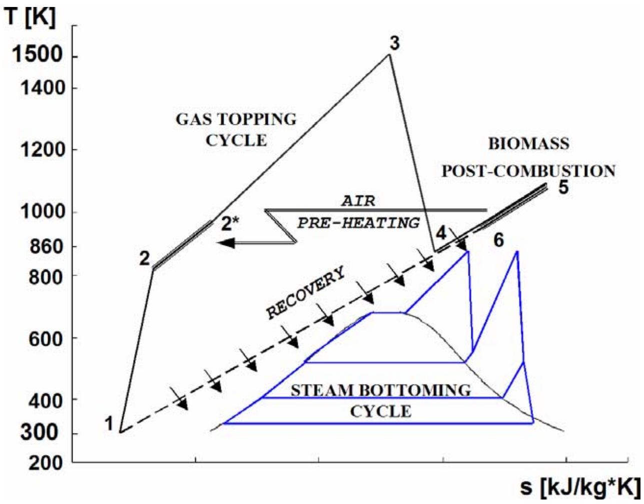

Fig. 7. T–s diagram of the BIFRCC plant.

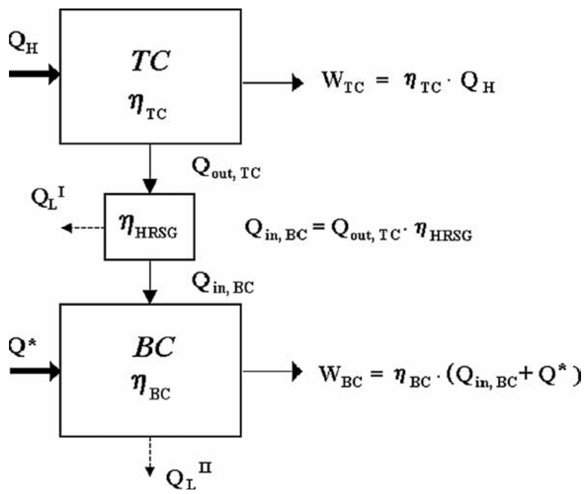

Fig. 8. Schematic of post-fired combined plant.

lowing mathematical expression for the efficiency of the plant:

$$\eta_{\mathrm{TOT}} = \frac{W_{o,\mathrm{tot}}}{Q_H + Q^*} = \frac{\eta_{\mathrm{GT}} + \eta_{\mathrm{ST}} \cdot [\eta_{\mathrm{HRSG}} \cdot (1 - \eta_{\mathrm{GT}}) + (\mu - 1)]}{\mu} \tag{6}$$

where

$$\mu = \frac{Q_H + Q^*}{Q_H}, \quad \mu \geqslant 1 \tag{7}$$

indicates the ratio between the total input thermal power and those from fossil fuel (Q_H).

If the efficiencies of gas ‘topping’ (η_{GT} i.e. η_{TC}) and steam ‘bottoming’ cycle (η_{ST} i.e. η_{BC}) are fixed, it is easy to demonstrate that the maximum value of the plant efficiency is obtained when $\mu = 1$, so when all the heat is introduced in the topping cycle. But, as shown in details in [16], the efficiency of the bottoming cycle is dependent on the inlet temperature of the gas to the HRSG according to the law shown in Fig. 9 and consequently it is dependent on the variable μ.

Fig. 9 shows that it is necessary to operate with an inlet gas temperature to the HRSG of the order of 550 °C to achieve a quite high efficiency of the bottoming cycle.

If the efficiency of the bottoming cycle improves with an introduction of thermal power, Q^*, before the HRSG, then the idea of arranging biomass postcombustion equipment has to be reconsidered. This explains the possible thermodynamic convenience of a combined cycle plant solution using reheat with low enthalpy fuels. Obviously, a real advantage is related to an optimization of the HRSG and of the bottoming cycle operating parameters.

6. Thermodynamic optimization of BIPCC and BIFRCC configuration

On a thermodynamic point of view, the problem of power plant design is to determine the parameters of an optimal configuration. With regards to combined cycle power plants with integration of thermal power from biomass, due to the complexity of the system, it is necessary to analyze the optimum design problem at different levels, decomposing the multistage decision problem as a sequence of sub-problems. Leading to a current engineering methodology of dynamic programming, the examined plant configurations can be converted into equivalent serial systems in such a way that the output of any component is the input of the subsequent component.

In the present paper the optimization procedure is related to the operating parameters of the plant (temperatures of gas and steam, pressures, mass flows), that are necessary for a successive more detailed optimization in terms of technological features and structural arrangement of the components.

The system under consideration is decomposed into two main sub-systems: the topping cycle, including the GT system, the biomass conversion equipment and the air preheater and the bottoming cycle, including the HRSG and the steam turbine (Fig. 10).

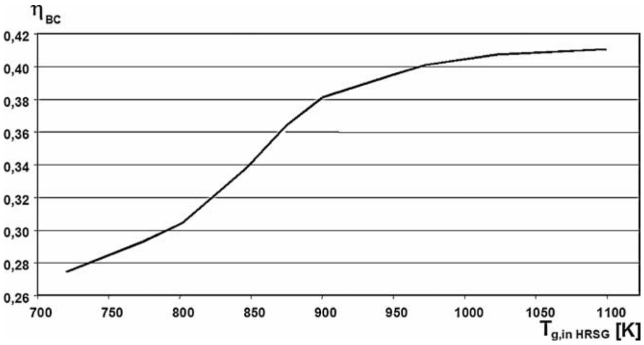

Fig. 9. Efficiency trend for a bottoming cycle as a function of the inlet temperature to the HRSG.

The link between the two subsystems is the inlet temperature of the exhaust gases to the HRSG: $T_{g,\text{in HRSG}}$. The decomposition into two different sub-problems is carried out in such a manner that the optimal solution of the original multi-variable problem can be obtained from the optimal solution of the two sequential sub-problems.

Applying the sub-optimization concept, the two components [HRSG + ST], are optimized at a first step. In a second step the two components can be considered together as a single element in order to carry out the optimization of the whole system.

6.1. First step—Bottoming cycle optimization

The method for determining the optimized HRSG operating parameters is based on exergy analysis. When the aim of heat recovery is to obtain power from the steam turbine, it is suitable to refer the thermodynamic optimization to the minimization of the exergy loss, which corresponds, in this particular case, to the maximization of the energetic efficiency of the cycle. In the first stage, without considering the detailed description of the sub-system, the exergy loss analysis can take into account the exergy loss between gas and steam in the HRSG and all the losses related to the steam evolution in the bottoming cycle defined as

$$I_{\text{BC}} = I_{\text{HRSG}} + I_{\text{ST}} + E_{\text{VAP}} \tag{8}$$

where I_{ST} is the exergy loss due to no isentropic expansion of the steam and E_{VAP} is the exergy of steam flow at the exit of the turbine that is not furtherly possible to recover and it is lost in the condenser and

$$I_{\text{HRSG}} = E_{g,\text{in}} + E_{l,\text{in}} - E_{l,\text{out}} \tag{9}$$

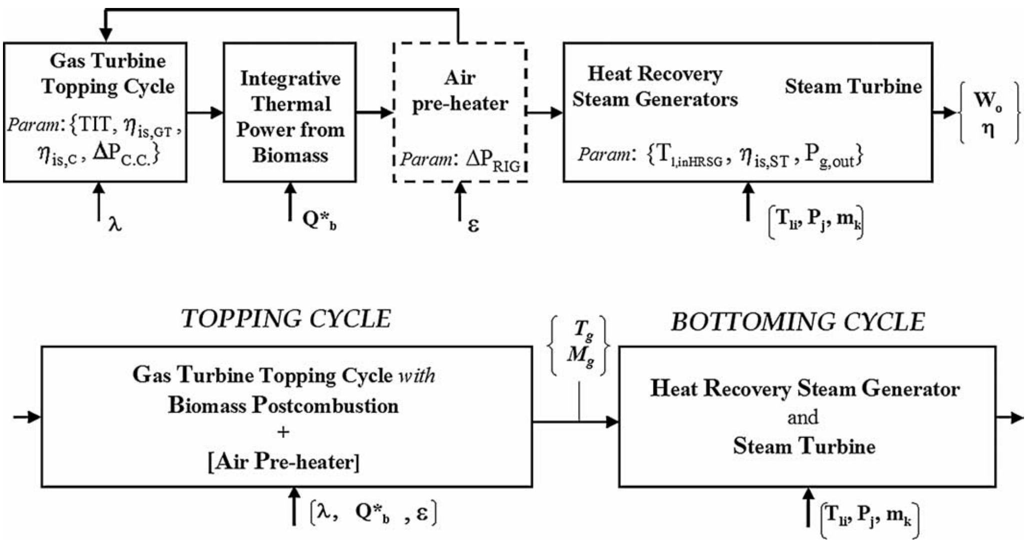

Fig. 10. Schematic of a plant configuration and equivalent serial system for the optimization.

Concerning the exergy loss due to non-isentropic expansion of the steam and the residual exergy E_{VAP} of the steam flow that is not furtherly possible to recover, Fig. 11 shows on a T–s diagram the specific irreversibility of a single pressure steam cycle. The colored areas identify the specific exergy losses:

$$\frac{I_{\text{ST}}}{m_l} = T_a \cdot (s_5 - s_4) \tag{10}$$

$$\frac{E_{\text{VAP}}}{m_l} = (T_{\text{cond}} - T_a) \cdot (s_5 - s_1) \tag{11}$$

The HRSG structure has been composed, using everywhere it is possible, heat exchanger sections with more water streams on the liquid side (as much water stream as the pressure levels are) that exchange heat from hot gas stream coming out from the gas turbine. The detailed description of this kind of HRSG arrangement is reported in [16].

So these consideration leads to conceive heat recovery structures in which, with the exception of evaporators and low pressure economizer, all the remaining units could be arranged with more water streams exchanging heat with the hot gas stream. Fig. 12 represents a possible scheme of a double pressure HRSG with reheat sections.

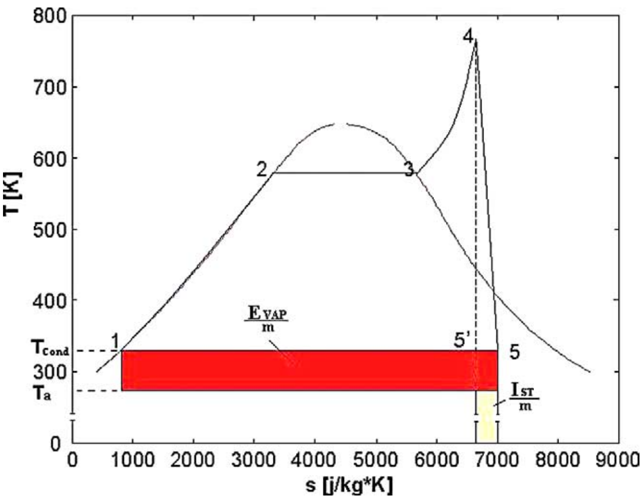

Fig. 11. Specific exergy destruction of a simple single-pressure steam cycle.

Supposing that the heat transfer is mainly convective, the HRSG is described by a sequence of elementary interconnected units; each unit is characterized by its operating parameters (temperatures, pressures, and mass flow rate of water streams).

Referring to a given mass flow rate of the exhaust gas [M_g], for the generic input temperature to the HRSG [$T_{g,\text{in HRSG}}$], the variables of the sub-system HRSG + ST must be selected in order to minimize the exergy losses, opportunely reduced in dimensionless form as:

$$I^*_{\text{BC}} = \frac{I_{\text{BC}}}{(M_g \cdot c_{p,g} \cdot T_a)} \tag{12}$$

The results of previous analysis show how the methodic use of multi-stream water sections and the possibility of reaching steam pressure and temperature close to the critical values, enable to get a substantial reduction of the exergy losses due to the heat transfer through a lower temperature difference. It seems to be generally convenient to resort to double or triple-pressure HRSG with reheat, that is the current technological trend in the combined plant technology. The detailed results in terms of exergy losses of several configurations are reported in [16]. In the plant configuration examined in this work only solutions with two and three pressure levels with reheaters have been considered.

6.2. Second step—Global optimization of the plant

The results of the first step optimization consist on a set of optimized bottoming plant configurations (HRSG + steam turbines) in terms of minimum of exergy loss. For each value of the gas temperature entering the HRSG, that is the interface value between topping and bottoming cycle, a configuration of the bottoming cycle that minimize the exergy losses is achievable.

Considering the global plant, the object of the thermodynamic optimization can be simply the maximization of the First Law efficiency defined as:

$$\eta = \frac{W_{o,\text{tot}}}{Q_{\text{in,tot}}} = \frac{W_{o,\text{GT}}}{Q_{H\,\text{gas}} + Q^*_b} + \frac{W_{o,\text{ST}}}{Q_{H\,\text{gas}} + Q^*_b} = \varphi_{\text{GT}} + \varphi_{\text{ST}} \tag{13}$$

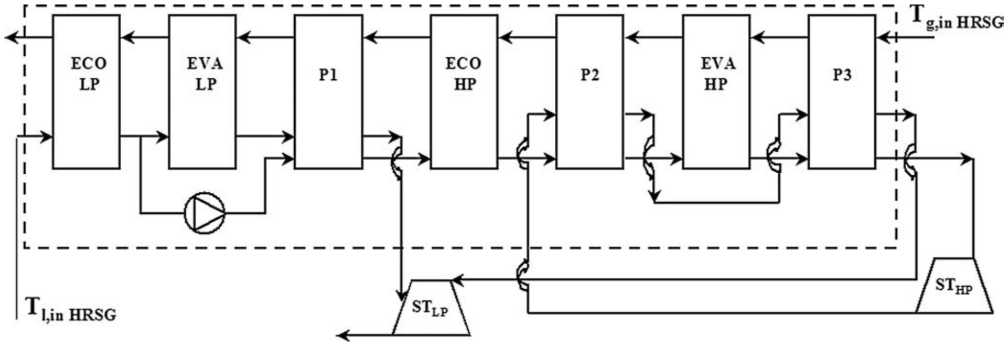

Fig. 12. Schematic of HRSG with two pressure levels with reheat.

In the expression of the efficiency defined by Eq. (13) the thermal energy input from biomass is directly considered, neglecting the biomass conversion process. An upper constraint on the maximum temperature of the gas after biomass postcombustion is defined. A further constraint is represented by the exhaust gas mass flow rate at the inlet of the HRSG, defined at the previous step $[M_g]$.

6.3. Mathematical features and implementation of the optimum design method

On a mathematical point of view each step of the problem is a non-linear, constrained optimization problem. The problem can be characterized, besides the vector of design variables $[X]$, by certain input parameters, $[S]$, and results in a vector of output variables $[Y]$.

The solution of a general constrained optimization problem consists on finding the vector $[X]$ which maximizes (minimize) the requested objective function subject to equality $g_i([X];[S]) = 0$ and inequality constraints $f_j([X];[S]) \leqslant 0$.

With reference to the plant configuration represented in Figs. 4–7, the aim of the considered double steps optimization problem is to find the vectors $[X_1]$, $[X_2]$, that optimize the objective function at each step, satisfying the relations between the input parameters and the design variables, stated by the constraints.

Particular inequality constrains are the maximum pressure of steam in the HRSG, the maximum temperature of the superheated steam the gas turbine inlet temperature and the temperature of the gas after the postcombustion of biomass or biomass derived fuel, the minimum pressure of the gas at the outlet of the HRSG.

Equality constraints can be considered the isentropic efficiency of compressor, steam turbine and gas turbine and the pressure losses in the heat exchangers.

$[X_1] = (T_{li}, P_j, m_k)$ is the design vector of the bottoming cycle, whose components are involved in the first step of the system optimization. The variables of the second step of the optimization process are the pressure ratio of the GT unit, the ratio between the thermal input from biomass and from natural gas and the air pre-heater effectiveness, so that $[X_2] = \{\lambda, Q_b^*/Q_{H\,\mathrm{gas}}, \varepsilon\}$. The last two variables are not completely independent but are linked to a particular value of the temperature of the gas entering the HRSG, thus $T_{g,\mathrm{in\ HRSG}} = T_{g,\mathrm{in\ HRSG}}([X_2])$, representing the interconnection element between the two steps of the optimum design process.

The two-steps optimization problem can be solved by using the Bellman's principle of optimality [21]: the general problem is decomposed in two separate problems, each one involving few decisions variables with respect to the general problem.

At the first step the HRSG and the steam cycle are optimized for a well defined value of the inlet temperature to the HRSG, by the minimization of the total bottoming cycle exergy loss. If the optimum of the objective function is defined as $\bar{I}_1$ and the optimized design vector as $\overline{X}_1$, it is:

$$\begin{aligned}\bar{I}_1\big(T_{g,\mathrm{in\ HRSG}};\big[\overline{X}_1\big]\big) &= \min_{[X_1]}\big\langle I_1\big(T_{g,\mathrm{in\ HRSG}};[X_1]\big)\big\rangle \\ &= \min_{[X_1]}\big\langle I_{\mathrm{BC}}\big(T_{g,\mathrm{in\ HRSG}};[X_1]\big)\big\rangle \qquad (14)\end{aligned}$$

The previously defined value of the exergy losses is a function of the value $T_{g,\mathrm{in\ HRSG}}$. By running the minimization of the exergy losses, using a penalty function method, [16], the optimal value of the objective function and the optimized design variables of the bottoming cycle are completely defined.

It is important to emphasize that, referring to the complete HRSG + ST sub-system, for a given value of the inlet temperature to the HRSG, the minimization of the bottoming cycle exergy losses is coincident with the maximization of the power output from the steam cycle so

$$\min\langle I_{\mathrm{BC}}\rangle \Longleftrightarrow \mathrm{Max}\langle W_{o,\mathrm{BC}}\rangle \qquad (15)$$

Then let consider the second part of the optimum design problem including in the analysis the topping cycle too. As stated above the maximization of the total efficiency of the plant is coincident with the maximization of the sum of the two terms φ_{TC} and φ_{BC}. The maximum efficiency of the plant $\bar{\eta}_{\mathrm{tot}}([S_3])$ is, obtained for the specified values of the input parameters $[S_3] = \{\mathrm{TIT}, \Delta P_{\mathrm{RIG}}, \Delta P_{\mathrm{c.c.}}, \eta_{\mathrm{is,GT}}, \eta_{\mathrm{is},C}, M_g\}$ and particular values of the design variables of the topping cycle $[X_2]$ joining the optimized bottoming cycle configuration related to the value of $T_{g,\mathrm{in\ HRSG}}$ obtained:

$$\begin{aligned}\bar{\eta}_{\mathrm{tot}}\big([S_3]\big) &= \mathrm{Max}_{[X_1,X_2]}\big\langle\eta_{\mathrm{tot}}\big(\big[\overline{X}_1\big];[X_2];[S_3]\big)\big\rangle \\ &= \mathrm{Max}_{[X_1,X_2]}\big\langle\varphi_{\mathrm{TC}}\big([X_2];[S_3]\big) \\ &\quad + \varphi_{\mathrm{BC}}\big(T_{g,\mathrm{in\ HRSG}}\big([X_2]\big);\big[\overline{X}_1\big];[S_3]\big)\big\rangle \qquad (16)\end{aligned}$$

The vector $\overline{X}_1$ has been selected at the first step of the optimisation process, with reference to the selected value of $T_{g,\mathrm{in\ HRSG}}$.

Since it is not possible to give in an explicit form neither the objective function, nor the constraints by the independent variables and due to the inherent non-linear nature of the problem, it is necessary a numerical solution by a non-gradient direct method.

Given a set of input parameters, by means of the connection with the subroutines of the fluids properties, a plant simulator determines the output of the topping cycle from which it is possible to obtain the input value, $T_{g,\mathrm{in\ HRSG}}$, for the bottom cycle optimization.

The main program is interconnected with the topping system simulator and with a bottom cycle optimized solution library, where the result obtained for different discrete value of $T_{g,\mathrm{in\ HRSG}}$ are collected. The main program receives the optimized bottoming cycle data together with the results coming from the topping cycle simulation and combines the two sets of results, returning the suitable data for the global system and the values of the objective function related to

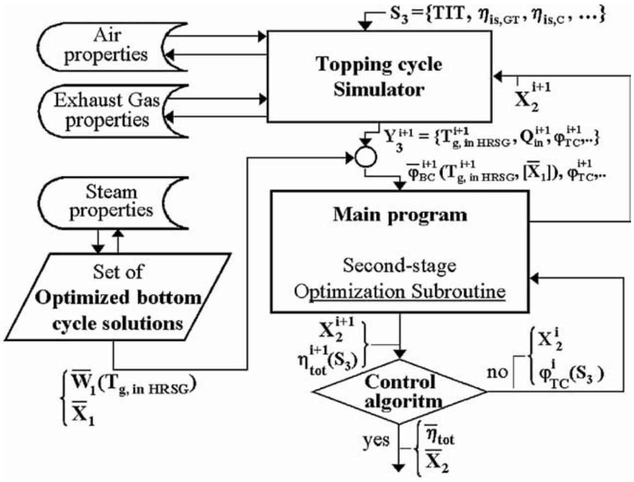

Fig. 13. Diagram of the optimization procedure.

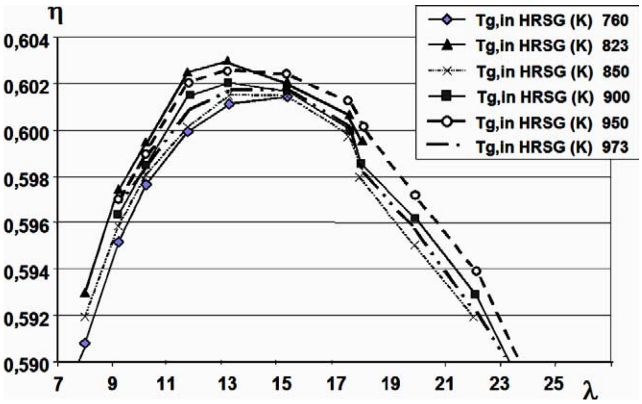

Fig. 14. Efficiency of the optimized BIFRCC plant as a function of the pressure ratio.

the current design variables. The comparison among the results obtained in term of efficiency for different values of the input variables λ, $Q_b^*/Q_{H,\mathrm{gas}}$ and of the interconnection variable $T_{g,\mathrm{in\ HRSG}}$ finally determines the results of the optimization.

The optimization method has been implemented according to the scheme reported in Fig. 13. Optimization is based on grid search method and simplex method; both of them can work even if the objective function is discontinuous and nondifferentiable in some points of the definition domain, as required by the problem under consideration [21,22].

7. Results of the optimization of BIFRCC and BIPCC plant configurations

After the detailed description of the method used to perform the optimization, in this section an application of the method is shown with reference to the two proposed configurations, BIFRCC and BIPCC respectively. The optimization is developed, at first, with reference to a general case, taking into account only some general parameters regarding the technological scenario. After that a study is carried out also with reference to two GT commercial models.

7.1. General case

For the second stage of the optimization, the following parameters, related to the technological availability of the components, have been assumed:

- Turbine inlet temperature (TIT): 1500 K;
- Maximum gas temperature after biomass based post-combustion: 1100 K;
- Isentropic efficiency of compressor ($\eta_{\mathrm{is},C}$): 0.87;
- Isentropic efficiency of gas turbine ($\eta_{\mathrm{is,GT}}$): 0.89;
- Pressure loss in the air pre-heater (ΔP_{RIG}): 5%;
- Pressure loss in the TG combustion chamber ($\Delta P_{\mathrm{c.c.}}$): 3%;
- Efficiency of the air pre-heater (η_{RIG}): 0.98;
- Limit temperature difference in the recuperator: 40 K.

The air mass flow rate is considered a fixed value (it is an additional parameter) meaning that the gas turbine size is well defined. The variables of the optimization are the pressure ratio, the thermal input from biomass and for BIFRCC configuration, the air pre-heater effectiveness too.

Concerning the bottoming cycle, we refer to subcritical steam cycle with the following constraint conditions:

- Maximum steam temperature ($T_{\mathrm{SH\text{-}RH}}$): 850 K;
- Minimum steam quality (x_{out}): 0.9;
- Isentropic efficiency of the steam turbine ($\eta_{\mathrm{is,ST}}$): 0.9;
- Maximum steam pressure (P_{MAX}): 200 bar;
- Inlet liquid temperature ($T_{l,\mathrm{in\ HRSG}}$): 308 K.

The optimization of the bottoming cycle has been carried out for a discrete set of different values of the inlet temperature of the gas to the HRSG, that represent the connection element between the two sub-systems, being fixed the gas mass flow rate (700 K $< T_{\mathrm{gin\ HRSG}} <$ 1100 K). So for each discrete value of $T_{g,\mathrm{in\ HRSG}}$, an optimized structure of the bottoming cycle in terms of minimum exergy losses is defined. Table 3 and Fig. 14 concisely provide the results of the optimization of BIFRCC power plant configuration. The analysis of these results shows that optimizing biomass integration and using optimized HRSG structure, it could be theoretically possible to obtain a gross efficiency of the plant of the order of 60% and the optimal solution can be obtained with medium pressure ratio ($\lambda = 11.7$–17.5). In the optimal solution the integrative thermal power from biomass contributes in the range of 25–32% with respect to the global power input.

The application of the aforesaid concepts does not require further enhancements in the gas turbine and gas-to-gas recuperator technology. The optimal air-preheater effectiveness values does not exceed 0.78, besides the limit of 1100 K imposed to the maximum temperature of the gas after post-combustion, obtained with thermal energy from biomass,

Table 3
Optimized parameters for the BIFRCC plant configuration

λ	$Q_{H,\mathrm{gas}}/Q_b^*$	$\varepsilon_{\mathrm{RIG}}$	Q_b^*/Q_{tot}	$T_{g,\mathrm{in\ HRSG}}$ [K]	$w_{o,\mathrm{ST}}$ [MJ·(kg air)$^{-1}$]	$w_{o,\mathrm{tot}}$ [MJ·(kg air)$^{-1}$]
9.2	3.96	0.620	0.202	823	0.203	0.592
11.7	2.82	0.686	0.262	823	0.203	0.600
13.2	2.43	0.731	0.292	823	0.203	0.597
17.5	2.20	0.472	0.313	950	0.294	0.680
20	1.93	0.520	0.341	950	0.294	0.672
22	1.76	0.563	0.362	950	0.294	0.664

Table 4
Optimized parameters for the BIPCC plant configuration

λ	$Q_{H,\mathrm{gas}}/Q_b^*$	$T_{g,\mathrm{in\ HRSG}}$ [K]	Q_b^*/Q_{tot}	w_o [MJ·(kg air)$^{-1}$]	η
13.2	10.23	950	0.089	0.688	0.599
17.4	5.45	950	0.155	0.680	0.595

makes possible the use of conventional stainless steel heat exchangers for the recuperator.

The results of the optimization of BIPCC power plant show that it is possible, in some special cases, to obtain a gross thermal efficiency of the plant approaching the level of 60%. The value is lower than the one that can be obtained with the BIFRCC configuration (Table 4), but with the advantage that BIPCC solution can be obtained only resorting to the use of optimized HRSG structure. However in this last case such efficiency solutions are achieved when the additional thermal input from biomass is of the order of 10% with respect to the power input from natural gas. In this case the thermal energy from biomass permits to increase the efficiency of the bottoming cycle. The interest of BIPCC configuration is mainly due to the possibility of a simple technological application.

7.2. Optimization of BIPCC and BIFRCC plants with reference to available gas turbine models

The possible application of biomass reheat has been analyzed considering existing commercial GT models, in order to show the possibility of upgrading plant performance by using optimized bottoming steam unit and atmospheric post-combustion.

Two different gas turbine models have been selected. The first one, GE LM6000PD, represents a highly efficient (40.6%) gas turbine: it is the smallest high performance commercial GT present in the market. This turbine seems to be not particularly suitable for use in combined cycle, due to the quite low discharge temperature (721 K). The second model, GE MS6001FA, has a lower pressure ratio but a quite high discharge temperature (870 K). It is a medium size GT model, produced by General Electric too. The compared performances of the two models are given in Table 5.

While the first model seems to be suitable for use in a BIPCC configuration, the second one seems more indicated for application in BIFRCC plants. In both the cases the optimization has been performed with the objective of maximizing the total efficiency. In the BIFRCC configuration, the temperature of the gas after reheat is always 1100 K, so that the thermal power input from biomass is fixed. In the case of BIPCC configuration, the temperature of the gas after reheat, coincident with inlet temperature to the HRSG ($T_{g,\mathrm{in\ HRSG}}$), can range between a minimum equal to $T_{g,\mathrm{out\ GT}}$ and a maximum of 1100 K varying the thermal power input from biomass. Figs. 15 and 16 compare the two different cases with reference to the inlet gas temperature to the HRSG.

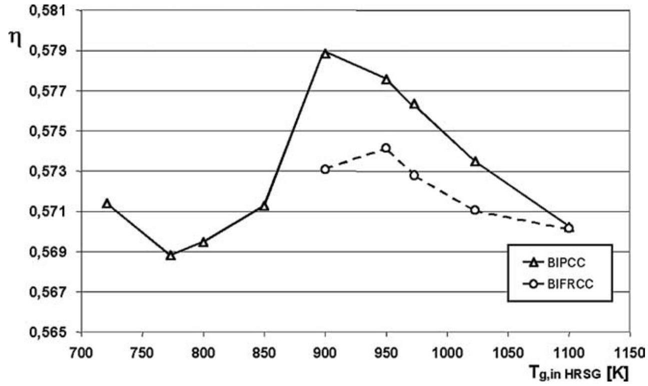

Fig. 15. Comparison between BIPCC and BIFRCC optimized plant configuration based on LM6000 PD GT.

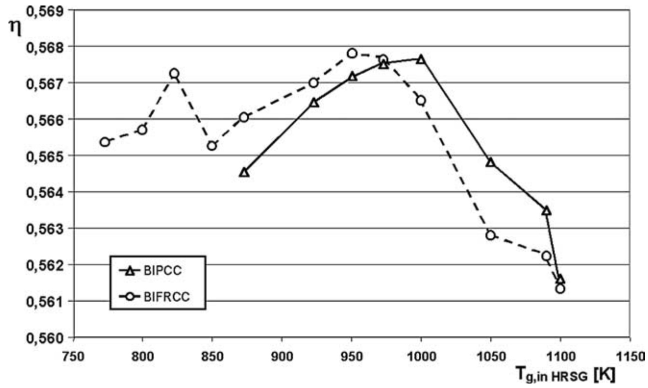

Fig. 16. Comparison between BIPCC and BIFRCC optimized plant configuration based on MS6001FA GT.

The concept of biomass postcombustion has been adapted also to the optimization of the two plants configurations based on the MS6001FA GT.

In this last case, the BIFRCC solution shows good results in a lot of cases, permitting to obtain a gross plant efficiency

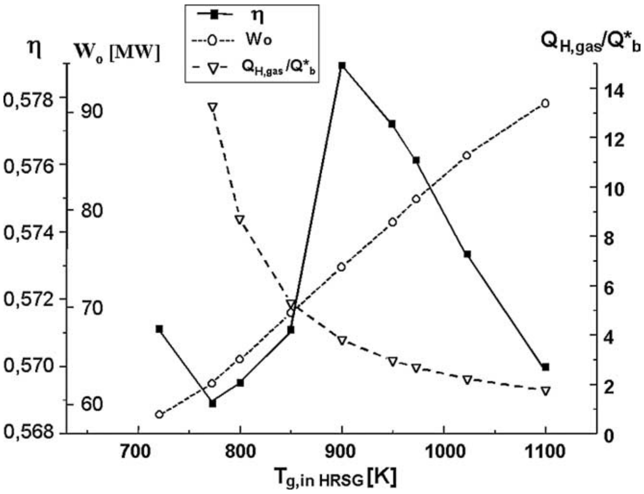

Fig. 17. Efficiency of the BIPCC plant as a function of reheat temperature for a plant based on LM6000PD GT.

Table 5
Characteristics of the commercial GT used to perform a constraint optimization of BIPCC and BIFRCC configuration (for both models $TIT = 1561$ K)

GT model	λ	M [kg·s^{-1}]	W [MW]	η [%]	$T_{g,\text{out GT}}$ [K]
LM6000PD	30	125	41.5	40.6	721
MS6001FA	14.9	195.8	70.1	34.6	870

of about 57%, with a gross output power of 120–130 MW: the thermal input from biomass is about the 23% of the total thermal input.

Fig. 17 provides more detailed results of the optimization for a BIPCC plant configuration based on the use of the LM6000PD gas turbine model. Together with the efficiency of the plant, output power and ratio between the thermal input from gas and from biomass, are reported as a function of the inlet temperature to the HRSG, corresponding to the exhaust gas temperature after reheat.

Using optimized HRSG structure for the exhaust gas recovery, it is possible to increase the output power of the plant up to 58% of efficiency, with about 27 MW of the thermal input coming from biomass (about 20% of the global thermal input). If a BIFRCC plant based on the same GT LM6000PD model is analyzed and optimized, the efficiency always results lower than that of the BIPCC configuration. Considering the results of this analysis, referred to two well defined gas turbine models, it seems clear that, in order to obtain a real advantage from gas to gas recuperation (BIFRCC plant configuration), it is necessary to consider also the GT operating parameters as design variables.

8. Conclusions

Biomass can be converted into useful forms of energy using a number of different processes and different plant configurations. The paper analyzes the perspective of improving the efficiency of biomass for power generation.

The main conclusion of the papers that should be emphasized are the following:

(1) The actual capacity of converting energy from biomass is often based on direct combustion boiler steam/turbine technology with efficiency lower than 20%. A thermoeconomic method of analysis based on the consideration of the cost of the exergy losses shows how the conversion efficiency of renewable sources as biomass, has a lower limit that, in a lot of cases, is considerably higher than the actual efficiency level.
(2) To promote the use of biomass conversion in a more profitable way, an interesting strategy seems to be the use of biomass joined with natural gas, in high efficient energy conversion system as the combined cycle power plants.
(3) The most promising solution in order to use a biomass derived fuel, in the combined cycle plants, is their use together with natural gas as integrative fuel to obtain a reheat of the exhaust gas and consequently and increase of the bottoming cycle efficiency. Two plant configurations, BIPCC and BIFRCC have been analyzed both with reference to possible optimized configurations and with reference to existing high efficiency gas turbine market models. The thermodynamic feasibility of approaching 60% of efficiency in combined cycle using natural gas and thermal energy from biomass as fuel has been shown.
(4) The large scale development of the solution proposed in the present paper has to be related to the development of small-size gas turbine, giving the possibility of building plants using approximately 10–20 MW of thermal input from biomass.

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