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BIOMASS UTILIZATION IN DUAL COMBUSTION GAS TURBINES FOR DISTRIBUTED POWER GENERATION IN MEDITERRANEAN COUNTRIES

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ABSTRACT

In Mediterranean regions, such as Puglia in Italy, the supply chain constraints (i.e. local biomass availability, logistics of supply, storage and seasonality issues) limit the optimal size of a biomass fired power plant in a range of 5-15 MWe. In this scenario, innovative Dual Combustion Externally Fired Gas Turbine (DCGT) Power Plants cofired by natural gas and biomass are examined. For this purpose, biomass external firing is explored under two alternatives: direct combustion of solid biomass and atmospheric fixed bed biomass gasification with air.

The proposed cycles are analyzed considering both the Net Overall Electric Efficiency and the Marginal Efficiency of biomass energy conversion, defined for the cofiring of biomass and natural gas. Since natural gas represents a quite expensive fossil fuel resource, a Marginal Efficiency higher than zero indicates the convenience to burn natural gas in this typology of power plant rather than in traditional Combined Cycle with higher efficiency. The energy analysis has been carried out by varying pressure ratio, turbine inlet temperature, heat exchanger efficiency and considering the further option of steam injection.

The results of the thermodynamic assessment highlight that the gasification should be preferred to the direct combustion of biomass because of the higher marginal efficiency, although the net overall electric efficiencies of the two plants are almost the same (31%).

KEYWORDS

Biomass, gasification, externally fired, gas turbine

NOMENCLATURE

H	=	Enthalpy
$\dot{m}$	=	mass flow rate
p	=	Pressure
P	=	electric power
Q	=	loss coefficient of electric power transmission
T	=	Temperature

$\dot{v}$	=	volumetric flow rate
β	=	pressure ratio
η	=	efficiency

Subscripts and upscripts

cge	=	cold gas efficiency
comp	=	compressor
elect	=	Electric
el.gen.	=	electric generator
isent	=	isentropic
marg.	=	marginal
ng	=	natural gas
polyt.	=	polytropic
ref.		reference
therm.	=	Thermal
turb	=	Turbine

Acronyms

BC	=	Biomass Combustion
BIG	=	Biomass Integrated Gasifier
CC	=	Combined Cycle
DCGT	=	Dual Combustion Gas Turbine
EFGT	=	External Fired Gas Turbine
GDP	=	Gross Domestic Product
LHV	=	Lower Heating Value
STIG	=	Steam Injected Gas Turbine
SP	=	Size Parameter
TIT	=	Turbine Inlet Temperature

INTRODUCTION

The high cost of energy in Italy causes considerable economic losses and a slowdown in GDP (Gross Domestic Product) growth. It has been shown that the GDP grows proportionally to the amount of energy produced: a Country that wants to grow has to produce the necessary energy to enable the proliferation and development of industries. Not only the cost of energy depends on the availability of raw materials such as coal, natural gas, fuel oil but also the energy efficiency of thermal power plants. Italy already has a good average energy efficiency of the generation mix but it is not a producer of fossil fuels. The only solution to face with security of supply, sustainability and competitiveness issues related to the Italian energy system is to foster the development of alternative and renewable energy sources.

Among these, biomass to energy systems can play a relevant role to contribute to the national energy strategy goals. However, there is still a strong delay in the field of biomass rather than the other alternative sources. For this reason, it is natural to focus the studies towards innovative solutions. In fact, biomass is present in nature in great quantities and different qualities but with considerable supply costs and low energy conversion efficiency. To increase the competitiveness of the biomass market we need to optimize the logistics of biomass supply and propose high efficiency conversion technologies.

In this paper, innovative biomass conversion technologies are proposed and their efficiency and economic feasibility are assessed. The software Gatecycle 6.62 edited by General Electric is used in order to simulate the thermodynamic cycles. The study was applied to the case of Apulia Region, where the domestic biomass potentials indicate a reasonable power plant size range of 5-15 MWe [1].

SYSTEM DESCRIPTION

The field of biomass energy conversion is very wide not only because of the heterogeneity of the biomass but also for the type of energy conversion processes that can be implemented. In this work, we refer to thermo-chemical processes aimed at producing electricity and heat. With the crisis of coal occurred a few years ago, biomass was considered a good alternative for the external combustion of the classic coal-fired steam power plants. So the technology of biomass burned in the steam power plant was widely introduced: despite the very low efficiency of energy conversion (14-20% for biomass versus 30-40% for coal) due to the low calorific value of biomass and lower combustion temperatures, this technology allows low investment costs because it uses a technology already industrialized, with a combustion chamber of poor quality while the turbine and the condenser represent the most expensive elements in the system. In conclusion, it aims at a high return of investment (ROI) through a simple and relatively cheap but inefficient technology.

Trying to optimize the performance and efficiency of energy systems, recent studies involved biomass gasification and combined cycle [2][3]. Biomass integrated gasification combined cycle (BIGCC) technology is at an advanced stage of study and is characterized by high electric efficiency and high installation costs. The gasification of biomass, as for the combustion, is the natural evolution of coal gasification. When the cost of natural gas became excessive, coal gasification became an alternative in the internal combustion gas turbine. This solution was then given up when the cost of natural gas decreased and also for some technological problems concerning the gasifier and the de-rating of the turbine. In view the best reactivity of biomass with air and good cleanliness properties of the syngas, gasification of biomass seems quite promising. In conclusion, it aims at the rapid return of the investment by a complex and costly but very efficient technology.

The proposed work investigates the *Dual Combustion Gas Turbine (DCGT)* scenario. This combines the classic internal combustion turbine (fuelled by natural gas) with an external combustion of biomass or syngas produced by biomass gasification. Regardless of the choice of burning biomass or syngas from biomass gasification, the product of combustion is carried in a gas-air heat exchanger that pre-heats air exiting the compressor. The absence of direct mixing is widely justified by a series of past experiences regarding the problem of de-rating and chemical aggression of exhaust gas. The dual combustion turbine recalls the classic regenerative turbine and the co-firing technology.

The details of the two proposed scenarios are discussed in the following.

Scheme A. (Figure 1) Biomass is directly burnt in an atmospheric biomass furnace and exhaust gas goes to the regenerative heat-exchanger to warm up the compressed air. Combustion with natural gas in the internal combustion chamber allows for high temperature at the turbine inlet. Inter-refrigeration is adopted in order to reduce power absorbed for compression and to recover more thermal power from exhaust gas.

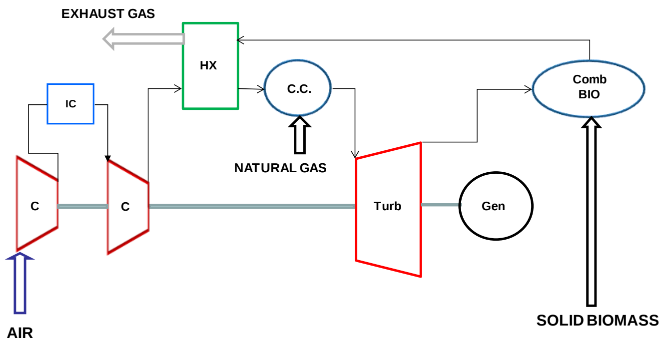

Figure 1 – Scheme A: Dual Combustion Gas Turbine with direct Biomass Combustion (BC-DCGT)

Scheme B. Figure 2 Biomass is gasified in a fixed bed updraft gasifier at atmospheric pressure (simple and cheap); a syngas atmospheric burner follows the gasifier. Apart from the biomass combustion, the two schemes are very close as the hot gas has the same path in the Joule-Brayton cycle. The gasification system has important disadvantages because of the presence of the cleaning system (cyclones, water scrubbers). The gasifier introduces a large efficiency loss and the contaminants (ash, tar, alkali metals, particulate, etc) in the syngas must be removed, otherwise they will cause blockage of injectors, valves, filters and, above all, the degradation of the turbine blades if used into internal combustion systems.

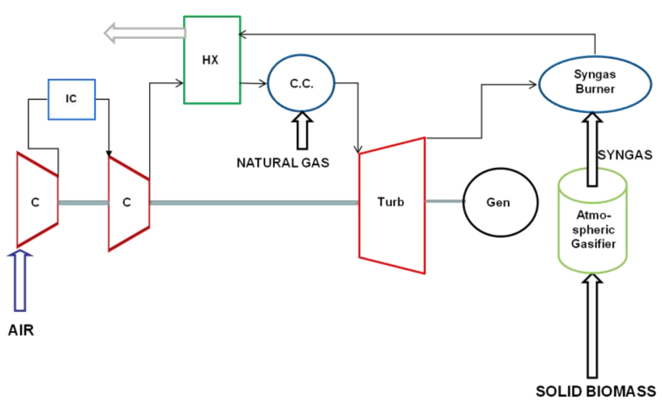

Figure 2 Scheme B: Dual Combustion Gas Turbine with Biomass Integrated Gasifier (BIG-DCGT)

The internal combustion of syngas is not proposed here in order to prevent the turbine from de-rating. The lower LHV of the fuel needs a specific design of the engine: indeed, the turbine is designed to work with natural gas. In fact, if the LHV is low, the mass flow is very high and this leads the compressor and the turbine to work at lower overall efficiency. In addition to this, there is also a serious risk for the compressor to surge. This is due to the mass balance and the mechanical connection of turbine and compressor. The system must use bleed valves and variable inlet guide vanes in the compressor or a new design of the nozzle guide vanes of the turbine has to be undertaken [7]. These drawbacks of the Direct Combustion prevent the growth of the classic GT and encourage the external Combustion (innovative EFGT technology). Here, syngas or biomass is burnt directly: the air exiting the compressor is heated by flue gas and enters directly in the turbine instead of the aggressive hot gas. The chemical hot corrosion is completely avoided as well as the risk of surging. In addition to these advantages, what draws more the attention is the separation of the combustion chamber where any kind of dirty fuels may be burnt without risks of chemical aggression to the turbine blades.

Tough less corrosive effects are present, two important drawbacks have to be considered:

1) The maximum cycle temperature is limited by the presence of the heat-exchanger; this temperature is about 1000 °C [2] and this causes a decrease of the efficiency in comparison to the classic internal combustion configuration.
2) The heat-exchanger is an expensive component and, being the weakest part of the plant, needs to be very efficient and sophisticated (ceramics) with evident growth of the specific costs.

So, the DCGT configuration seems to couple at best the positive aspects of internal and external combustion in the examined range of medium-low size (5-15 MWe) [1][1]. The difference between these technologies concerns mainly the combustion process. Direct combustion of biomass produces an aggressive gas with dramatic consequences on the heat-exchanger: that is why the temperature in the heat exchanger must be controlled under the melting point of biomass ashes (1073 K) [2][2]. In the more expensive case of gasified gas, the hot gas is cleaner and the temperature of the warmed air reaches 1000°C [2] that is200°C higher than in the scheme with with direct combustion. The outlet biomass combustor temperature has a crucial influence on the energy analysis and in scheme B the limit is the resistance of materials of heat exchanger instead of corrosive problems. In case B, the increased power of the plant, which is also more environmental-friendly, allows overcoming the problems of poor quality of syngas and cost of gasifier, even if there is the cost of biomass supplying that is to be considered.

BIOMASS SCENARIO IN APULIA

Apulia Region is chosen as a reference model among Mediterranean Countries because its geographical position strikes the balance between Northern Countries (Montenegro) and Southern ones (like Tunisia). That is not the same for the morphology of the territory; indeed, Apulia guarantees wide plains which facilitate biomass harvesting and transport, unlike other countries, such as Montenegro and Greece.

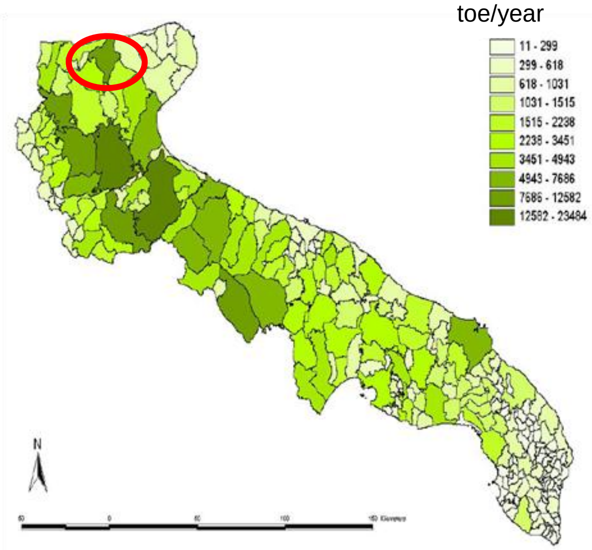

Figure 3 Map of Absolute Availability of Agricultural Residues in Apulia [1]

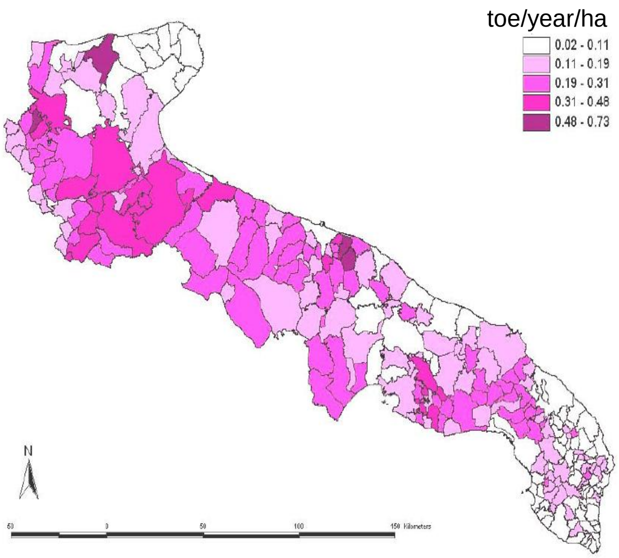

Figure 4 Map of Relative Density of agricultural residues in Apulia [1]

Apulia allows great potentials even in comparison with other Mediterranean areas. Indeed, plains cover 53,3% of the region while low hills are in 45,2% and mountains only 1,5%; 29% of the land is bound to unproductive surfaces whilst the remaining 71% is the whole arable surface.

As usual, biomasses count numerous and different kinds of residues as agricultural, forest, industrial, urban residues. In this work, agricultural and forest biomasses are considered because industrial and urban wastes require particular treatments against pollutant emissions.

Table 1. Amount of Agricultural Residues in Apulia (source [1])

Region surface	Kind of Residues	Potential absolute availability [t/year dry]	Potential availability percentage [%]	Annual relative availability [t/(year* km^2)]	
				Potential	Net
APULIA (19,358 km^2)	Herbaceous	467,181	48.3	24.1	8.7
	Woody	500,763	51.7	25.9	15.1
	Total	967,974	100	50.0	23.8

Agricultural biomass in this case consists on herbaceous and woody residues such as straw and pruning residues.

A local study has been carried out for agricultural residues in Apulia as they represent the largest biomass source: this is due to the presence of a wide plain in the north part of Apulia where herbaceous residues abound (73,5%) while woody ones are more present in the rest of the region.

The most important data in order to evaluate the local potentiality are those concerning the Net Relative Availability of Biomass which corresponds to a net availability/surface ratio and represents an Energy Density.

The two maps of Apulia Biomass Availability underline the difference between Absolute and Relative Availability (density): smaller zones can be characterized by low Absolute Availability (Figure 1) but, in some case, by high Relative Availability (Figure 2) and, on the contrary, there are some areas that present great Absolute Availability while the Relative Availability is poor because of the great dispersion.

The net values in Table 1 show that a part of the potential amount of biomass is destined to other uses like combustion in field, cattle feed, litters, firewood.

The analysis of maps and table shows that Region Apulia presents good quantities of biomass, mainly in the northern part of it. Considering, e.g., the small area highlighted in Figure 3 with the circle, it can be observed that it is one of the richest in terms of biomass availability in the region; its average diameter is approximately 13 km and the estimated net availability ranges from 0.48 and 0.73 tep/(year*km^2). Assuming that the power plant will be operated for 7500 h/year, the Installed Power of the plants will approximately range within 5-15 MWe, which is coherent with the choice of small-size Gas Turbines.

The forest residues are scarce in Apulia territory because of the lack of forest and the presence of numerous preserved areas.

The maximum forest residues availability is allocated where agricultural residues are absent and, moreover, the regional potential is lower than other countries' ones such as Serbia and Montenegro but is better than African Countries. Fires and tricks of hill slopes are serious issues and decrease the reliability of this kind of biomass.

ENERGY ANALYSIS

Thermodynamic Simulation.

This analysis has been carried out by fixing the isentropic efficiency of each turbomachinery, as usual in literature studies.

Due to the combined use of two different fuels, with different LHV properties, in order to analyze the plant energy performances the following efficiency definitions are given

- Thermodynamic Efficiency

$$\eta_{Therm} = \frac{P_{shaft}}{\dot{m}_{ng} \cdot LHV_{ng} + \dot{m}_{syngas} \cdot LHV_{syngas}} \qquad (1)$$

Takes into account the thermodynamic efficiency of the cycle as ratio between the shaft power produced by the turbine and the ideal energy input given by the combustion of natural gas and syngas,

- Electric Efficiency

$$\eta_{Elect} = \frac{P_{el}}{\dot{m}_{ng} \cdot LHV_{ng} + \dot{m}_{syngas} \cdot LHV_{syngas}} \qquad (2)$$

differs from the thermodynamic efficiency as it takes into account the electric losses in the electric generator, as

$$P_{el} = \eta_{el.gen.} P_{shaft}; \qquad (3)$$

- Total Electric Efficiency

$$\eta_{Total} = \frac{P_{el}}{\dot{m}_{ng} \cdot LHV_{ng} + \dot{m}_{biomass} \cdot LHV_{biomass}} = $$

$$= \frac{P_{el}}{\dot{m}_{ng} \cdot LHV_{ng} + \frac{\dot{m}_{syngas} \cdot LHV_{syngas}}{\eta_{cge}}} \qquad (4)$$

is instead referred to the LHV of the raw biomass and, therefore, takes into account the cold-gas-efficiency η_{cge} of the gasifier, if present.

Finally, in order to identify a parameter able to take into account the potential of efficient conversion of the natural gas LHV in electric energy, the Marginal Efficiency [4] is defined as

$$\eta_{Marg} = \frac{P_{el} - \eta_{el,ref} \cdot \dot{m}_{ng} \cdot LHV_{ng} \cdot q}{\dot{m}_{biomass} \cdot LHV_{biomass}} \qquad (5)$$

where $\eta_{el,ref}$ is the electric conversion efficiency of natural gas in conventional combined power plants fuelled by natural gas, and the coefficient q (<1) takes into account the electric transmission losses that could be avoided in the on-site power production of the electric power (assuming a local use of the produced electric power). The product $\eta_{el,ref} \cdot \dot{m}_{ng} \cdot LHV_{ng} \cdot q$ is, therefore, the electric power that could be produced elsewhere is a conventional power plant fuelled by the same amount of natural gas and connected to the electric grid. Consequently, the difference appearing at numerator of Eq.(5) represents the surplus of electric power produced by the energy conversion of biomass.

The marginal efficiency occurs whenever two different fuels are burnt and it has energy, economic and environmental importance because it describes how much natural gas is saved. For example, as the Natural Gas is often more expensive than Biomass in Italy, a higher marginal efficiency might be preferred to the thermal efficiency from an economic point of view. In terms of CO2 production, saving natural gas by using biomass allows to reduce the greenhouse gas emissions, that is more and more relevant in the modern international Energy Policy.

In the thermodynamic analysis of the different plant configurations, the specific work produced by the plant, W is evaluated as the ratio of electric power P_{el}, to the air mass flow rate $\dot{m}_{air}$ entering the compressor unit. In order to optimize the cycle, such analysis is carried out by varying the overall pressure ratio β and the turbine inlet temperature, TIT.

Table 2 summarizes all the parameters and data assumed for the thermal analysis.

Table 2. Assumptions for Energy Analysis [3].

		BC-DCGT-STIG	BIG-DCGT-STIG
Natural Gas	LHV [kJ/kg]	44140	44140
	p [kPa]	2000	2000
Syngas	T [°C]	-	325
	p [kPa]	-	150
	LHV [kJ/kg]	-	5000
Solid Biomass	T [°C]	150	-
	LHV [kJ/kg]	15000	15000
Air Compr.	$\dot{m}_{air}$ [kg/s]	11	11
	η_{isent}	0.85	0.85
Pump	η_{isent}	0.85	0.85
Intercooler	ε	0.70	0.70
	$\Delta p_{Hot/Cold}$ [%]	2	2
Recuperative Heat Exch.	ε	0.80	0.90
	$\Delta p_{Hot/Cold}$ [%]	2	2
Heat Recov. Steam Gen.	$\Delta T_{Sub-cool}$ [°C]	6	6
	ΔT_{PP} [°C]	24	24
Gas Turbine Combustor	$\eta_{comb.}$	0.93	0.93
	Δp [%]	3	3
Biomass Combustor	Δp [%]	10	10
	COT [°C]	800	1000
Turbine	η_{isent}	0.88	0.88
Gasifier	η_{cge}	-	0.78
Electric Generator	η_{MEC}	0.97	0.97
	η_{EL}	0.98	0.98

Dual Combustion Gas Turbine with Biomass Combustor (BC-DCGT).

In order to increase thermal efficiency and specific work, in the examined schemes, steam is injected in the main combustion chamber (STIG cycle) as shown in Figure 5 where a screenshot of the scheme developed within the Gate Cycle (R) software used for simulation.

In the simulation, turboexpander blades are assumed to be cooled by air bled from the compressor. The amount of the extracted air necessary to maintain a blade temperature of 1070 K, is evaluated through specific macros developed in Gate Cycle. The pressure ratio of each of the two compressors is assumed equal to the square root of the overall pressure ratio in order to minimize the compressor work. Furthermore, the maximum steam flow in combustion chamber is assumed equal to 20% of the inlet air [3].

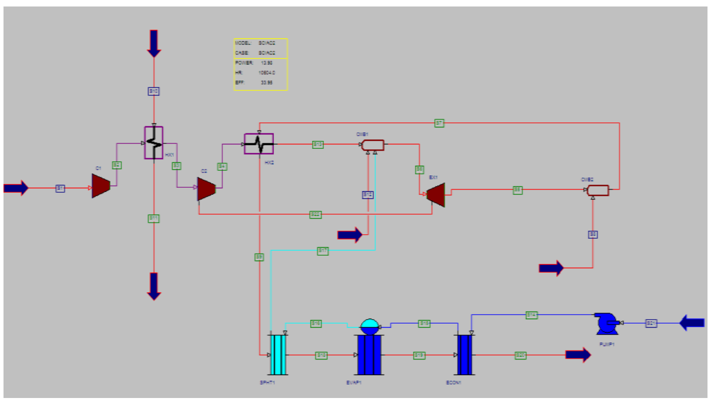

Figure 5-Dual Combustion Gas Turbine power plant (STIG) with Combustor of Solid Biomass (BC-DCGT-STIG). Screenshot by GATECYCLE.

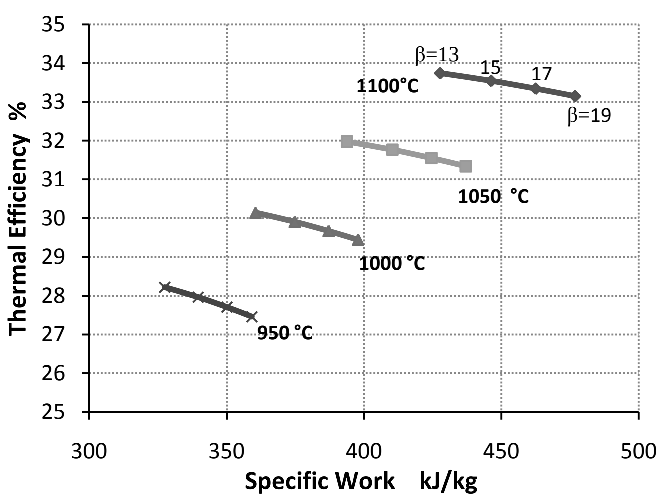

Figure 6 Curves of Thermal Efficiency at different TIT and β in BC-DCGT-STIG.

Curves in Figures 6 describe the influence of β and TIT on thermal efficiency. It appears that with the BC-DCGT-STIG cycle, it is possible to overcome 31% with a TIT of only 1050°C. A reduction of 50°C of the TIT implies a reduction of the efficiency of 2% (at the same value of pressure ratio) while the influence of β is much weaker.

The results of the marginal efficiency are given in Figure 7, where it is assumed the electric reference efficiency $\eta_{el,ref}$ =0.52 that is, averagely, the overall efficiency of the gas turbine combined cycle power plants actually operating, while it is assumed q=0.97.

It appears that, for all the considered values of pressure ratio, the marginal efficiency decreases with increasing TIT (a decrease of about 5% for each 50°C of increase in TIT). The reason of such apparently surprising result, can be cleared if one considers that, at constant pressure ratio, the flow rate of natural gas feeding the first (internal) burner increases with increasing TIT; moreover, as a consequence of the higher TIT, also the turbine exit temperature is increased. It descends that, the higher is the TIT, the lower is the rate of biomass feeding the second (external) burner, since the temperature of the gas entering the burner (equal to the turbine exit temperature) increases, while the temperature of the gas exiting the burner needs to remain constant and equal to the maximum allowed temperature at the recuperative heat exchanger inlet. Finally, the effects of the TIT on the marginal efficiency can be explained considering the definition given in Eq.(5). It can be considered that, the higher is TIT, the higher is the electric power output, P_{el}, but, also, the higher is the mass flow rate of natural gas, $\dot{m}_{ng}$, and, the higher is the product $(\eta_{el,ref} \cdot \dot{m}_{ng} \cdot LHV_{ng} \cdot q)$ that lowers the numerator. Since the electric efficiency assumed as reference value is $\eta_{el,ref} = 0.52$, that is the typical efficiency of a combined cycle gas turbine, the increase of TIT determines an increase of P_{el} lower than the increase of the product $\left(\eta_{el,ref} \cdot \dot{m}_{ng} \cdot LHV_{ng} \cdot q\right)$: this is the reason that causes the decrease of the marginal efficiency, notwithstanding the decrease of the thermal input of the biomass that appears at denominator of Eq.(5).

Figure 7 shows also the effects of the pressure ratio on the marginal efficiency: it appears that, at the same value of TIT, unlike the thermal efficiency, the higher is the pressure ratio the higher is the marginal efficiency; this can be clarified considering that, with the same TIT, the turbine exit temperature decreases with increasing pressure ratio; consequently, the biomass rate increases and finally, also the marginal efficiency increases.

In order to complete the survey on the direct combustion scheme, the performances of the cycle with steam injected(BC-DCGT-STIG) are compared in Figure 8 to the performance of the cycle without steam injection (BC-DCGT). It appears that the basic scheme without steam injection has a rather lower electric efficiency, mainly due to the higher exhaust heat losses at the exit of the heat-exchanger. The electric efficiency increases of 3-4% if steam injection is adopted in comparison to the basic plant; the marginal efficiency remains approximately unchanged while the electric power (not shown) grows of 60%.

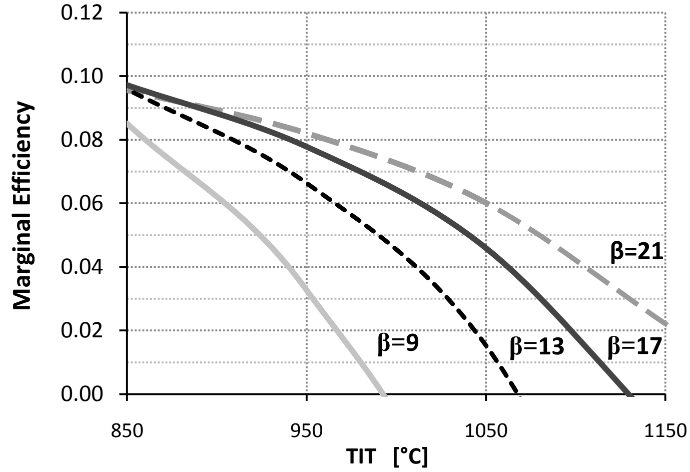

Figure 7. Marginal Efficiency in BC-DCGT-STIG.

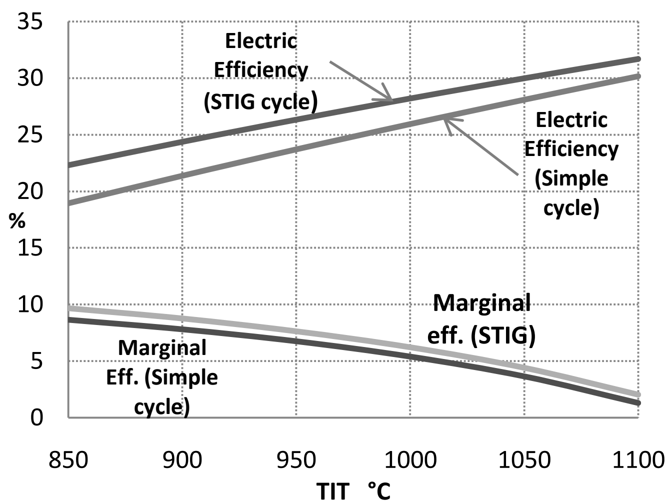

Figure 8. Comparison between STIG cycle and Simple cycle in BC-DCGT technology for Net Overall Electric Efficiency and Marginal Efficiency, as a function of Turbine Inlet Temperature.

The low values of the marginal efficiency are mainly due to the relatively high values assumed for $\eta_{el,ref}$ and q. It is reasonable that lower values could be assumed for small plants in order to favor the distributed generation from renewable sources. In analogy to the values adopted for evaluation of the Primary Energy Saving in combined Heat and Power generation, $\eta_{el,ref}$ can be lowered to 0.4 for plant with power output in the range between 1 to 10 MW.

The advantages given by the STIG scheme is that it allows for reducing the influence of the efficiency of the regenerative heat-exchanger. In fact, its influence in the basic cycle is, among any analyzed parameters, very important in order to improve the performances of the plant. The effectiveness of the regenerative heat exchanger has a large influence on the costs of the plant as this component is very complex.

On the other hand, the Heat Recovery Steam Generation (HRSG) allows a partial recovery of exhaust thermal losses and, therefore, the effectiveness of heat-exchanger can be lower without affecting the overall energy efficiency. The heat-exchanger becomes energetically less important with consequently improvement of simplicity and economy. The simulation highlights that, varying the regenerator effectiveness from 0.8 and 0.90, the thermal efficiency increases of 0.5% (with STIG) instead of 2% (without STIG).

Energy results for DCGT with Biomass Gasifier Integrated.

In the present work it is supposed that an atmospheric fixed bed updraft gasifier, characterized by a relatively simple, cost-effective and reliable technology, is directly connected to the "externally fired" burner. In comparison to the internal combustion of syngas, the external combustion scheme avoids that some work is consumed for compressing hot syngas coming from the gasifier. It is assumed a typical chemical composition of syngas with LHV=5000 kJ/kg and cold-gas-efficiency of the gasifier equal to 0.78. Also in this case the introduction of the STIG scheme (Figure 10) improved efficiencies and power output while reducing the influence of the regenerative heat exchanger efficiency on the performance.

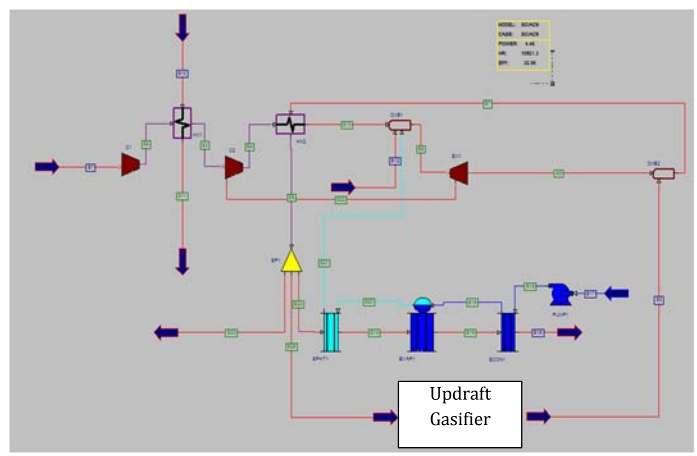

Figure 9. Dual Combustion Gas Turbine power plant (STIG) with Biomass Gasifier (BIG-DCGT-STIG). Screenshot by GATECYCLE.

In the proposed scheme, part of exhaust gas exiting the recuperator is carried to the gasifier in order to use its heat for increasing the mean gasification temperature and its cold-gas-efficiency. Furthermore, this scheme allows for reducing the exhaust thermal losses of the plant.

The graphs of Thermal Efficiency vs. Specific Work, evaluated for different values of TIT and β (Figure 10), show higher values of efficiency reachable by this configuration, in comparison to the scheme with direct biomass combustion (BC-DCGT-STIG). The comparison between the results given in Figures 6 and 10, shows that, at the same TIT, the thermal efficiencies of the two schemes have about the same values but the use of syngas in the atmospheric burner, allowing for a 200°C higher temperature at the burner exit, rises the maximum thermal efficiency from 33% shown in Figure 6 to about 37% reached in Figure 10.

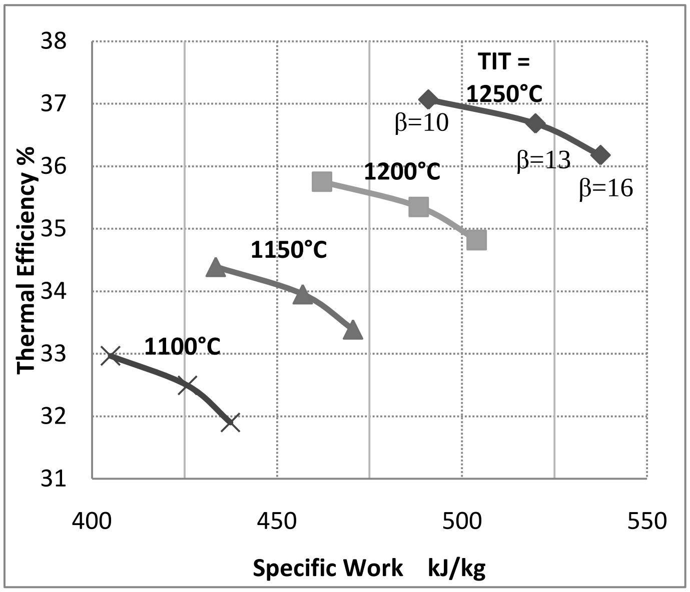

Figure 10. Curves of Thermal efficiency – Specific work in the BIG-DCGT-STIG scheme.

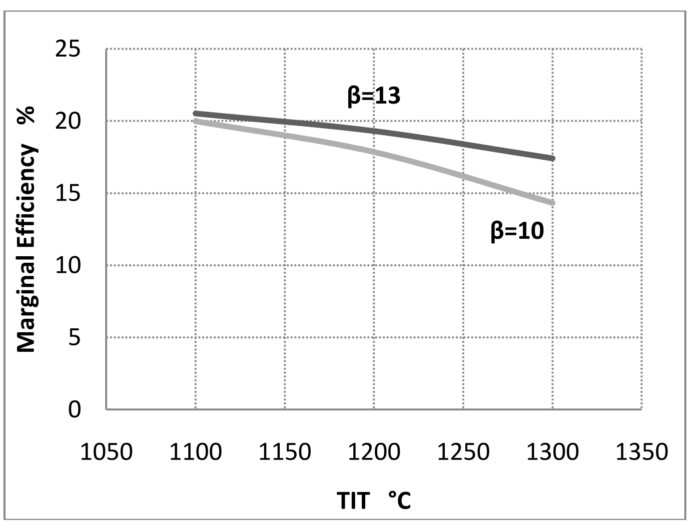

Figure 11. Curves of Marginal Efficiency in BIG-DCGT-STIG.

The marginal efficiency of the cycle with biomass gasifier is shown in Figure 11; in comparison to the results given in Fig.7, it appears that in this case the marginal efficiency is largely positive even at high values TIT. This positive feature is due to the higher temperature allowed at the syngas burner exit in comparison to the maximum exhaust gas temperature allowed with direct biomass combustion. Such increase of temperature favours the thermal power production originated from biomass instead of the natural gas.

CHOICE OF TIT IN THE PROPOSED SCENARIOS.

Due to the different influence of TIT on the two efficiencies, more than one optimization strategy is possible. On one hand, it could be recommended to increase the TIT in order to increase the net overall electric efficiency and the power output; the highest limit of TIT is the blade material resistance, even with an effective blade cooling system; in this work, the limit of 1200°C was assumed as a reasonable limit for small scale turbines. On the other hand, it could be suggested to decrease the TIT in order to increase the marginal efficiency, decreasing the consumption of natural gas, whether the biomass availability is sufficient. A moderate strategy might be considered in order to couple the economic and the energy/environmental feasibilities: the best solution might be characterized by the maximum thermal efficiency and a non-negative marginal efficiency, which is the so called “restricted optimization”. The corresponding value of TIT and further results are reported in Table 3. It appears that the Thermal Efficiency of BIG-DCGT-STIG is some percent better than BC-DCGT-STIG because of the greater TIT, and they at least can reach 33% thanks to the positive characteristics of STIG cycle.

Table 3. Results of the Restricted Optimization

	BC-DCGT-STIG	BIG-DCGT-STIG
β_{opt}	17	10
TIT_{opt} [°C]	1100	1200
$P_{steam,opt}$ [kPa]	2143	1639
$T_{steam,opt}$ [°C]	450	500
$\eta_{Thermal}$ [%]	34.48	37.64
$\eta_{Electric}$ [%]	32.78	35.78
$\eta_{Total\text{-}Electric}$ [%]	32.78	30.63

Considering the Total Efficiency, the Gas Turbine with Combustion system (30-34%) turns out to be better than the Gas Turbine with the Gasification system (28-32%). The two technologies have Total Efficiencies between 30% and 34%. The updraft gasifier offers good performances for the Dual Combustion Gas Turbine. The cold-gas-efficiency (0.78) does not reduce the electric efficiency of the whole system as in the Dual Combustion Gas Turbine there is a parallel between the Internal Firing and External Firing and the gasifier penalizes only the external firing contribution.

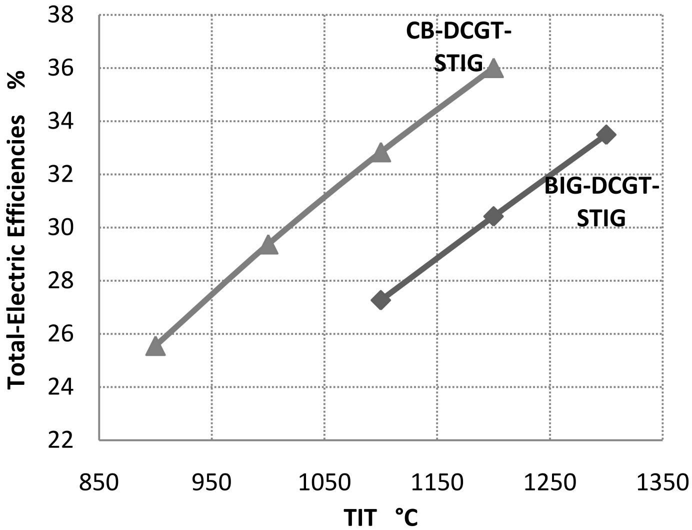

Figure 12. Comparisons of Total-Electric Efficiencies

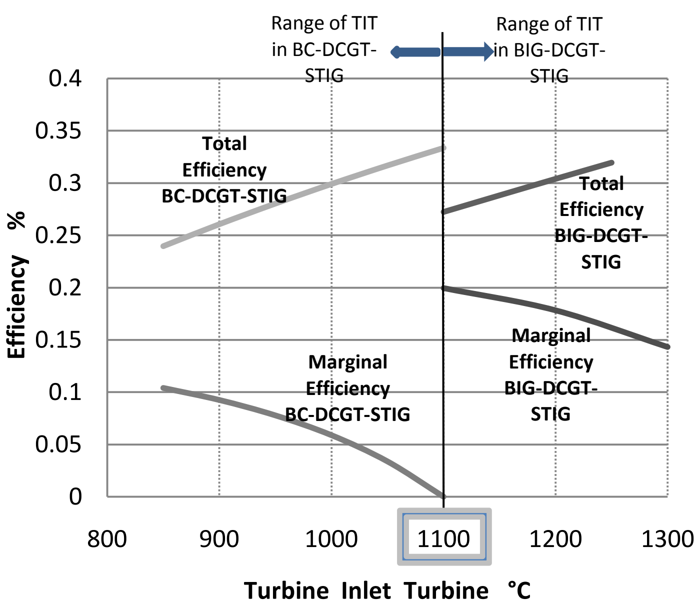

Figure 13. Curves of Marginal and Total-Electric Efficiencies in BC-DCGT-STIG and BIG-DCGT-STIG for the "Restricted Optimization".

Figure 12 seems to suggest that, considering the Dual Combustion Gas Turbine STIG technologies, it is not thermodynamically effective to use a gasifier and to reach greater TIT. It would be better to burn solid biomass with smaller TIT but without the problematic gasifier. Hovewer Figure 13 shows the range of TIT that are defined by the positivity of the marginal efficiency. As the allowable TIT in case of Direct Combustion of biomass is 1100°C (β=17 to maintain positive the marginal efficiency), the Total Efficiency is equal to the one in presence of the Gasification system at 1200°C (β=10) but in this last configuration a greater marginal efficiency positively affects the economic analysis since more natural gas is saved.

The optimization of the total efficiency would prefer the Direct Combustion configuration while the restricted optimization of total and marginal efficiencies promotes the Gasification system. Furthermore, this highlights the rationality of this kind of optimization that should be repeated whenever two different fuels are present.

CONCLUSIONS

In the present work the energy analysis of gas turbine cycle with dual combustion of natural gas and biomass was carried out. These cycles appear suitable for the production of electricity in small plants for the short supply chain in the production and use of biomass.

It has been shown that in dual combustion plants, internal combustion of natural gas increases the overall thermal efficiency and specific work; the thermodynamic analysis has highlighted the result that, the higher is the ratio between natural gas and biomass, the lower is the marginal electrical power produced by biomass with respect to what would be produced in a conventional plant fuelled by natural gas only. This result, paradoxically, thus drives to limit TIT.

Energy analysis has also highlighted the strong influence of the maximum allowed temperature of the heat recuperator; mainly, in the case of direct combustion.

In conclusion, this paper proposed a methodology and a new indicator in order to evaluate environmental matters during the standard energy optimization and design of dual combustion systems.

Although the direct combustion system can achieve higher electric efficiencies, the gasification one saves greater amount of natural gas and, consequently, is more environmentally-friendly. In particular, this last is able to reach almost the same overall efficiency (31%) of the direct combustion (33%) assuring much higher marginal efficiency (+17% compared with the direct combustion) with much stronger positive impact on the greenhouse gas emissions. This underlying aspect deserves to be reminded as it will have huge impact on the future energy policy and economics. These results are mostly subjected to the assumptions on the maximum temperature allowable for the heat exchanger, different for biomass direct combustion and syngas combustion.

All these results must be taken into account in the business plan to identify the economic feasibility the power plant, depending on the availability and costs of biomass, costs of transport, as well as economic incentives for power production from biomass within a short supply chain scheme.

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